New pipe sizing charts are now available
for Refrigerants 134a, 404A, 407C and 507A.
See the 1999, "HCFC/HFC Refrigerant Piping Guide for Commercial Chillers."
Order Catalog No. 510-313.

Carrier®

system design manual

PART 3

PIPING DESIGN

CARRIER AIR CONDITIONING COMPANY • SYRACUSE, NEW YORK

PART 3 PIPING DESIGN

CONTENTS

SYSTEM DESIGN MANUAL

SUMMARY OF PART THREE

This part of the System Design Manual presents data and examples to guide the engineer in practical design and layout of normal air conditioning piping systems.

The text of this manual is offered as a general guide for the use of industry and of consulting engineers in designing systems. Judgment is required for application to specific installations, and Carrier is not responsible for any of the uses made of this text.

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Carrier Air Conditioning Company

CHAPTER 1. PIPING DESIGN—GENERAL

Piping characteristics that are common to normal air conditioning, heating and refrigeration systems are presented in this chapter. The areas discussed include piping material, service limitations, expansion, vibration, fittings, valves, and pressure losses. These areas are of prime consideration to the design engineer since they influence the piping life, maintenance cost and first cost.

The basic concepts of fluid flow and design information on the more specialized fields such as high temperature water or low temperature refrigeration systems are not included; this information is available in other authoritative sources.

GENERAL SYSTEM DESIGN

MATERIALS

The materials most commonly used in piping systems are the following:

1. Steel – black and galvanized
2. Wrought iron – black and galvanized
3. Copper – soft and hard

Table 1 illustrates the recommended materials for the various services. Minimum standards, as shown, should be maintained. *Table 2* contains the physical properties of steel pipe and *Table 3* lists the physical properties of copper tubing.

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TABLE 1—RECOMMENDED PIPE AND FITTING MATERIALS FOR VARIOUS SERVICES

SERVICE		PIPE	FITTINGS
REFRIGERANTS 12, 22, 500 and 502	Suction Line	Hard copper tubing, Type L*	Wrought copper, wrought brass or tinned cast brass
		Steel pipe, standard wall Lap welded or seamless	150 lb welding or threaded malleable iron
	Liquid Line	Hard copper tubing, Type L*	Wrought copper, wrought brass or tinned cast brass
		Steel pipe, standard wall Lap welded or seamless	300 lb welding or threaded malleable iron
	Hot Gas Line	Hard copper tubing, Type L*	Wrought copper, wrought brass or tinned cast brass
		Steel pipe, standard wall Lap welded or seamless	300 lb welding or threaded malleable iron
CHILLED WATER		Black or galvanized steel pipe†	Welding, galvanized; cast, malleable or black iron‡
		Hard copper tubing†	Cast brass, wrought copper or wrought brass
CONDENSER OR MAKE-UP WATER		Galvanized steel pipe†	Welding, galvanized; cast or malleable iron‡
		Hard copper tubing†	Cast brass, wrought copper or wrought brass
DRAIN OR CONDENSATE LINES		Galvanized steel pipe†	Galvanized drainage; cast or malleable iron‡
		Hard copper tubing†	Cast brass, wrought copper or wrought brass
STEAM OR CONDENSATE		Black steel pipe†	Welding or cast iron‡
		Hard copper tubing†	Cast brass, wrought copper or wrought brass
HOT WATER		Black steel pipe	Welding or cast iron‡
		Hard copper tubing†	Cast brass, wrought copper or wrought brass

*Except for sizes ¼″ and ⅜″ OD where wall thicknesses of 0.30 and 0.32 in. are required. Soft copper refrigeration tubing may be used for sizes 1⅜″ OD and smaller. Mechanical joints must not be used with soft copper tubing in sizes larger than ⅞″ OD.

†Normally standard wall steel pipe or Type M hard copper tubing is satisfactory for air conditioning applications. However, the piping material selected should be checked for the design temperature-pressure ratings.

‡Normally 125 lb cast iron and 150 lb malleable iron fittings are satisfactory for the usual air conditioning application. However, the fitting material selected should be checked for the design temperature-pressure ratings.

TABLE 2—PHYSICAL PROPERTIES OF STEEL PIPE

NOM. PIPE SIZE (in.)	SCHEDULE NO.†	OUTSIDE DIAM (in.)	INSIDE DIAM (in.)	WALL THICK-NESS (in.)	WEIGHT OF PIPE (lb/ft)	WT OF WATER IN PIPE* (lb/ft)	OUTSIDE SURFACE (sq ft/ft)	INSIDE SURFACE (sq ft/ft)	TRANS-VERSE AREA (sq in.)
⅛	40(S)	.405	.269	.068	.244	.0246	.106	.0705	.0568
	80(X)	.405	.215	.095	.314	.0157	.106	.0563	.0364
¼	40(S)	.540	.364	.088	.424	.0451	.141	.0955	.1041
	80(X)	.540	.302	.119	.535	.0310	.141	.0794	.0716
⅜	40(S)	.675	.493	.091	.567	.0827	.177	.1295	.1910
	80(X)	.675	.423	.126	.738	.0609	.177	.1106	.1405
½	40(S)	.840	.622	.109	.850	.1316	.220	.1637	.3040
	80(X)	.840	.546	.147	1.087	.1013	.220	.1433	.2340
¾	40(S)	1.050	.824	.113	1.130	.2301	.275	.2168	.5330
	80(X)	1.050	.742	.154	1.473	.1875	.275	.1948	.4330
1	40(S)	1.315	1.049	.133	1.678	.3740	.344	.2740	.8640
	80(X)	1.315	.957	.179	2.171	.3112	.344	.2520	.7190
1¼	40(S)	1.660	1.380	.140	2.272	.6471	.434	.3620	1.495
	80(X)	1.660	1.278	.191	2.996	.5553	.434	.3356	1.283
1½	40(S)	1.900	1.610	.145	2.717	.8820	.497	.4213	2.036
	80(X)	1.900	1.500	.200	3.631	.7648	.497	.3927	1.767
2	40(S)	2.375	2.067	.154	3.652	1.452	.622	.5401	3.355
	80(X)	2.375	1.939	.218	5.022	1.279	.622	.5074	2.953
2½	40(S)	2.875	2.469	.203	5.79	2.072	.753	.6462	4.788
	80(X)	2.875	2.323	.276	7.66	1.834	.753	.6095	4.238
3	40(S)	3.500	3.068	.216	7.57	3.20	.916	.802	7.393
	80(X)	3.500	2.900	.300	10.25	2.86	.916	.761	6.605
3½	40(S)	4.000	3.548	.226	9.11	4.28	1.047	.929	9.89
	80(X)	4.000	3.364	.318	12.51	3.85	1.047	.880	8.89
4	40(S)	4.500	4.026	.237	10.79	5.51	1.178	1.055	12.73
	80(X)	4.500	3.826	.337	14.98	4.98	1.178	1.002	11.50
5	40(S)	5.563	5.047	.258	14.62	8.66	1.456	1.321	20.01
	80(X)	5.563	4.813	.375	20.78	7.87	1.456	1.260	18.19
6	40(S)	6.625	6.065	.280	18.97	12.51	1.735	1.587	28.99
	80(X)	6.625	5.761	.432	28.57	11.29	1.735	1.510	26.07
8	40(S)	8.625	7.981	.322	28.55	21.6	2.26	2.090	50.0
	80(X)	8.625	7.625	.500	43.39	19.8	2.26	2.006	45.6
10	40(S)	10.750	10.020	.365	40.48	34.1	2.81	2.62	78.9
	60(X)	10.750	9.750	.500	54.70	32.4	2.81	2.55	74.7
	80	10.750	9.564	.593	64.33	31.1	2.81	2.50	71.8
12	30(S)	12.750	12.090	.330	43.80	49.6	3.34	3.17	115.0
	40	12.750	11.938	.406	53.53	48.5	3.34	3.13	111.9
	(X)	12.750	11.750	.500	65.40	46.9	3.34	3.08	108.0
	80	12.750	11.376	.687	88.51	44.0	3.34	2.98	101.6
14	30(S)	14.000	13.250	.375	54.60	59.8	3.67	3.46	138.0
	40	14.000	13.125	.438	63.37	58.5	3.67	3.44	135.3
	(X)	14.000	13.000	.500	72.10	55.8	3.67	3.40	133.0
	80	14.000	12.500	.750	106.31	51.2	3.67	3.27	122.7
16	30(S)	16.000	15.250	.375	62.40	79.1	4.18	3.99	183.0
	40(X)	16.000	15.000	.500	82.77	76.5	4.18	3.93	176.7
	80	16.000	14.314	.843	136.46	69.7	4.18	3.75	160.9
18	(S)	18.000	17.250	.375	70.60	100.8	4.71	4.52	234.0
	(X)	18.000	17.000	.500	93.50	98.3	4.71	4.45	227.0
	40	18.000	16.874	.562	104.75	97.2	4.71	4.42	224.0
	80	18.000	16.126	.937	170.75	88.5	4.71	4.22	204.2
20	20(S)	20.000	19.250	.375	78.60	126.7	5.24	5.04	291.0
	30(X)	20.000	19.000	.500	104.20	122.5	5.24	4.97	284.0
	40	20.000	18.814	.593	122.91	120.4	5.24	4.93	278.0
	80	20.000	17.938	1.031	208.87	109.4	5.24	4.70	252.7
24	20(S)	24.000	23.250	.375	94.60	184.6	6.28	6.08	426.0
	(X)	24.000	23.000	.500	125.50	179.0	6.28	6.03	415.0
	40	24.000	22.626	.687	171.17	174.2	6.28	5.92	402.1
	80	24.000	21.564	1.218	296.36	158.2	6.28	5.65	365.2

*To change "Wt of Water in Pipe (lb/ft)" to "Gallons of Water in Pipe (gal/ft)," divide values in table by 8.34.

†S is designation of standard wall pipe.
X is designation of extra strong wall pipe.

TABLE 3—PHYSICAL PROPERTIES OF COPPER TUBING

CLASSIFICATION	NOM. TUBE SIZE (in.)	OUTSIDE DIAM (in.)	STUBBS GAGE	WALL THICK-NESS (in.)	INSIDE DIAM (in.)	TRANS-VERSE AREA (sq in.)	MINIMUM TEST PRESSURE (psi)	WEIGHT OF TUBE (lb/ft)	WT OF WATER IN TUBE* (lb/ft)	OUTSIDE SURFACE (sq ft/ft)
HARD	¼	⅜	23	.025	.325	.083	1000	.106	.036	.098
	⅜	½	23	.025	.450	.159	1000	.144	.069	.131
	½	⅝	22	.028	.569	.254	890	.203	.110	.164
	¾	⅞	21	.032	.811	.516	710	.328	.224	.229
	1	1⅛	20	.035	1.055	.874	600	.464	.379	.295
	1¼	1⅜	19	.042	1.291	1.309	590	.681	.566	.360
Govt. Type "M"	1½	1⅝	18	.049	1.527	1.831	580	.94	.793	.425
250 Lb	2	2⅛	17	.058	2.009	3.17	520	1.46	1.372	.556
Working	2½	2⅝	16	.065	2.495	4.89	470	2.03	2.120	.687
Pressure	3	3⅛	15	.072	2.981	6.98	440	2.68	3.020	.818
	3½	3⅝	14	.083	3.459	9.40	430	3.58	4.060	.949
	4	4⅛	13	.095	3.935	12.16	430	4.66	5.262	1.08
	5	5⅛	12	.109	4.907	18.91	400	6.66	8.180	1.34
	6	6⅛		.122	5.881	27.16	375	8.91	11.750	1.60
	8	8⅛		.170	7.785	47.6	375	16.46	20.60	2.13
HARD	⅜	½	19	.035	.430	.146	1000	.198	.063	.131
	½	⅝		.040	.545	.233	1000	.284	.101	.164
	¾	⅞		.045	.785	.484	1000	.454	.209	.229
	1	1⅛		.050	1.025	.825	880	.653	.358	.295
	1¼	1⅜		.055	1.265	1.256	780	.882	.554	.360
Govt. Type "L"	1½	1⅝		.060	1.505	1.78	720	1.14	.770	.425
250 Lb	2	2⅛		.070	1.985	3.094	640	1.75	1.338	.556
Working	2½	2⅝		.080	2.465	4.77	580	2.48	2.070	.687
Pressure	3	3⅛		.090	2.945	6.812	550	3.33	2.975	.818
	3½	3⅝		.100	3.425	9.213	530	4.29	4.000	.949
	4	4⅛		.110	3.905	11.97	510	5.38	5.180	1.08
	5	5⅛		.125	4.875	18.67	460	7.61	8.090	1.34
	6	6⅛		.140	5.845	26.83	430	10.20	11.610	1.60
HARD	¼	⅜	21	.032	.311	.076	1000	.133	.033	.098
	⅜	½	18	.049	.402	.127	1000	.269	.055	.131
	½	⅝	18	.049	.527	.218	1000	.344	.094	.164
	¾	⅞	16	.065	.745	.436	1000	.641	.189	.229
	1	1⅛	16	.065	.995	.778	780	.839	.336	.295
	1¼	1⅜	16	.065	1.245	1.217	630	1.04	.526	.360
Govt. Type "K"	1½	1⅝	15	.072	1.481	1.722	580	1.36	.745	.425
400 Lb	2	2⅛	14	.083	1.959	3.014	510	2.06	1.300	.556
Working	2½	2⅝	13	.095	2.435	4.656	470	2.92	2.015	.687
Pressure	3	3⅛	12	.109	2.907	6.637	450	4.00	2.870	8.18
	3½	3⅝	11	.120	3.385	8.999	430	5.12	3.890	.949
	4	4⅛	10	.134	3.857	11.68	420	6.51	5.05	1.08
	5	5⅛		.160	4.805	18.13	400	9.67	7.80	1.34
	6	6⅛		.192	5.741	25.88	400	13.87	11.20	1.60
SOFT	¼	⅜	21	.032	.311	.076	1000	.133	.033	.098
	⅜	½	18	.049	.402	.127	1000	.269	.055	.131
	½	⅝	18	.049	.527	.218	1000	.344	.094	.164
	¾	⅞	16	.065	.745	.436	1000	.641	.189	.229
	1	1⅛	16	.065	.995	.778	780	.839	.336	.295
	1¼	1⅜	16	.065	1.245	1.217	630	1.04	.526	.360
Govt. Type "K"	1½	1⅝	15	.072	1.481	1.722	580	1.36	.745	.425
250 Lb	2	2⅛	14	.083	1.959	3.014	510	2.06	1.300	.556
Working	2½	2⅝	13	.095	2.435	4.656	470	2.92	2.015	.687
Pressure	3	3⅛	12	.109	2.907	6.637	450	4.00	2.870	.818
	3½	3⅝	11	.120	3.385	8.999	430	5.12	3.89	.949
	4	4⅛	10	.134	3.857	11.68	420	6.51	5.05	1.08
	5	5⅛		.160	4.805	18.13	400	9.67	7.80	1.34
	6	6⅛		.192	5.741	25.88	400	13.87	11.2	1.60

*To change "Wt of Water in Tube (lb/ft)" to "Gallons of Water in Tube (gal/ft)," divide values in table by 8.34.

SERVICE LIMITATIONS

The safe working pressure and temperature for steel pipe and copper tubing, including fittings, are limited by the USAS codes. Check those codes when there is doubt about the ability of pipe, tubing, or fittings to withstand pressures and temperatures in a given installation. In many instances cost can be reduced and over-design eliminated. For example, if the working pressure is to be 175 psi at 250 F, a 150 psi, class A, carbon steel flange can be safely used since it can withstand a pressure of 225 psi at 250 F. If the code is not checked, a 300 psi flange must be specified because the 175 psi working pressure exceeds the 150 psi rating of the 150 psi flange.

The safe working pressure and temperature for copper tubing is dependent on the strength of the fittings and tube, the composition of the solder used for making a joint, and on the temperature of the fluid conveyed. *Table 4* indicates recommended service limits for copper tubing.

EXPANSION OF PIPING

All pipe lines which are subject to changes in temperature expand and contract. Where temperature changes are anticipated, piping members capable of absorbing the resultant movement must be included in the design. *Table 5* gives the thermal linear expansion of copper tubing and steel pipe.

There are three methods commonly used to absorb pipe expansion and contraction:

1. *Expansion loops and offsets — Chart 1* shows the copper expansion loop dimensions required for expansion travels up to 8 inches. *Chart 2* shows the sizes of expansion loops made of steel pipe and welding ells for expansion travels up to 8 inches.

TABLE 5—THERMAL LINEAR EXPANSION OF COPPER TUBING AND STEEL PIPE

(Inches per 100 feet)

TEMP RANGE (F)	COPPER TUBING	STEEL PIPE
0	0	0
50	.56	.37
100	1.12	.76
150	1.69	1.15
200	2.27	1.55
250	2.85	1.96
300	3.45	2.38
350	4.05	2.81
400	4.65	3.25
450	5.27	3.70
500	5.89	4.15

NOTE: Above data are based on expansion from 0°F but are sufficiently accurate for all other temperature ranges.

Table 6 gives the minimum length from an elbow to expansion-type supports required for offsets in steel piping up to 5 inches. Expansion loop sizes may be reduced by cold-springing them into place. The pipe lines are cut short at about 50% of the expansion travel and the expansion loop is then sprung into place. Thus, the loops are subject to only one-half the stress when expanded or contracted.

2. *Expansion joints* — There are two types available, the slip type and the bellows type. The slip type expansion joint has several disadvantages: (a) It requires packing and lubrication, which dictates that it be placed in an accessible location; (b) Guides must be installed in the lines to prevent the pipes from bending and binding in the joint.

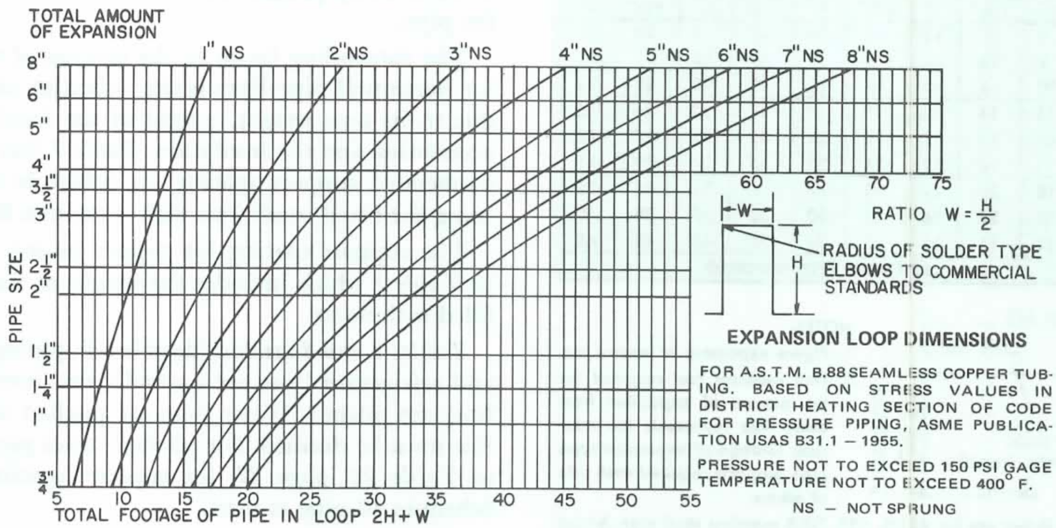
Bellows type expansion joints are very satisfactory for short travels, but must be guided or

TABLE 4—RECOMMENDED MAXIMUM SERVICE PRESSURE FOR VARIOUS SOLDER JOINTS

SOLDER USED IN JOINTS	SERVICE TEMP (F)	MAXIMUM SERVICE PRESSURE (PSI)			
		Water			Steam
		¼" to 1⅛" Incl.	1⅜" to 2⅛" Incl.	2⅝" to 4⅛" Incl.	All
50-50 Tin-Lead	100	200	175	150	—
	150	150	125	100	—
	200	100	90	75	—
	250	85	75	50	15
95-5 Tin-Antimony or 95-5 Tin-Lead	100	500	400	300	—
	150	400	350	275	—
	200	300	250	200	—
	250	200	175	150	15
Solders Melting At or Above 1100 F	350	270	190	155	120

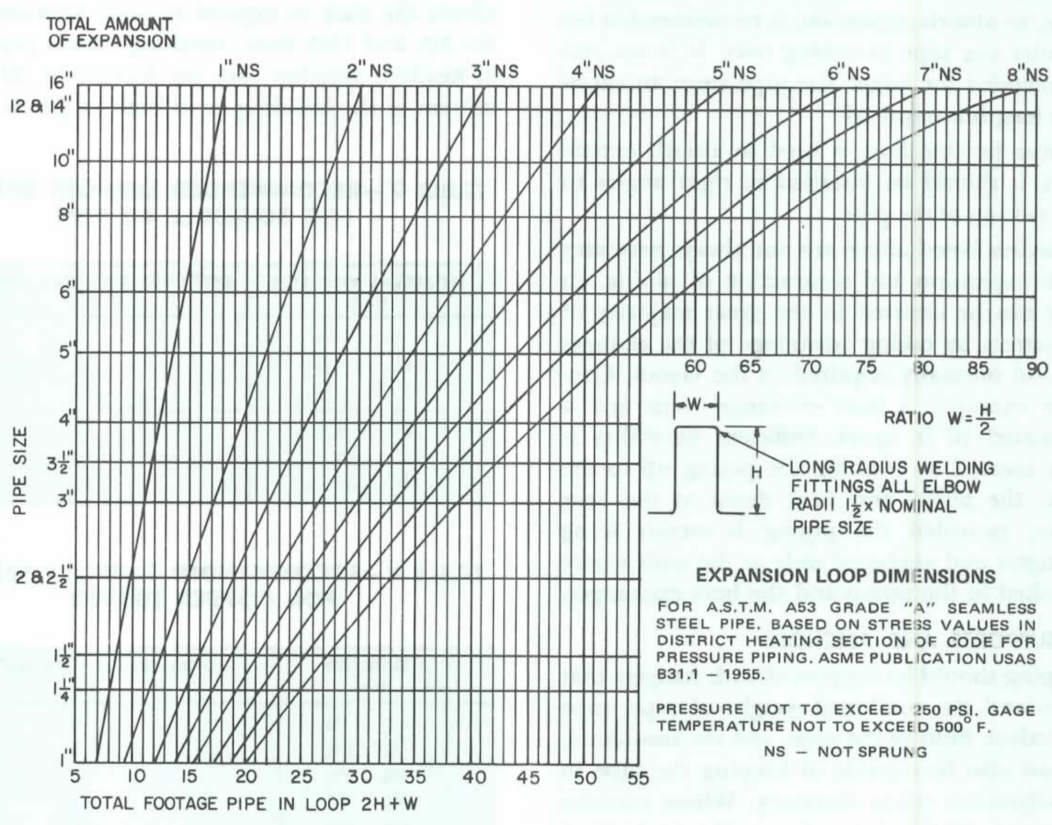
Extracted from *American Standard Wrought-Copper and Wrought-Bronze Solder-Joint Fittings*, (USAS B16.22), with the permission of the publisher, The American Society of Mechanical Engineers, 29 West 39th Street, New York 18, New York.

CHART 1—COPPER EXPANSION LOOPS



Data from Ric-Wil Co.

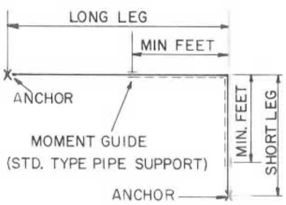
CHART 2—STEEL EXPANSION LOOPS



Data from Ric-Wil Co.

TABLE 6—STEEL EXPANSION OFFSETS

PIPE SIZE	EXPANSION OF LONGEST LEG								
	1″	1½″	2″	2½″	3″	3½″	4″	4½″	5″
2″	8	11	13	15	16	17	18	19	20
2½″	9	12	14	16	17	18	19	21	22
3″	10	13	15	17	18	19	20	22	23
4″	11	14	16	18	19	22	22	24	25
5″	12	15	17	19	21	23	25	27	28
6″	13	16	19	21	23	25	27	29	31
8″	18	20	22	25	27	29	31	33	35
10″	20	23	26	28	30	33	35	38	40
12″	22	26	29	32	34	37	40	43	45
	MINIMUM FEET REQUIRED								



NOTES:

1. **Figure expansion of longest leg.**
2. **Find minimum feet required for this amount of expansion from chart. This represents the minimum footage of expansion type pipe supports required *each* side of elbow.**

Dimensions shown are for ASTM, A53, GR.B seamless steel pipe, based on stress values in district heating section of code for pressure piping USAS B.31.1.

Data from Ric-Wil Co.

in some other way restrained to prevent collapse.

3. *Flexible metal and rubber hose* — Flexible hose, to absorb expansion, is recommended for smaller size pipe or tubing only. It is not recommended for larger size pipe since an excessive length is required.

   Where flexible hose is used to absorb expansion, it should be installed at right angles to the motion of the pipe.

The devices listed above are not always necessary to absorb expansion and contraction of piping. In fact they can be omitted in the great majority of piping systems by taking advantage of the changes in direction normally required in the layout. Consider, for example, a heat exchanger unit and a pump located 50 ft. apart. Sufficient flexibility is normally assured by running the piping up to the ceiling at the pump and back down at the heat exchanger, provided the piping is merely hung from hangers and anchored only at the ends where it is attached to the pump and the heat exchanger.

PIPING SUPPORTS AND ANCHORS

All piping should be supported with hangers that can withstand the combined weight of pipe, pipe fittings, valves, fluid in the pipe, and the insulation. They must also be capable of keeping the pipe in proper alignment when necessary. Where extreme expansion or contraction exists, roll-type hangers or saddles should be used. The pipe supports must have a smooth, flat bearing surface, free from burrs or other sharp projections which would wear or cut the pipe.

The controlling factor in the spacing of supports for horizontal pipe lines is the deflection of piping due to its own weight, weight of the fluid, piping accessories, and the insulation. *Table 7* lists the recommended support spacing for Schedule 40 pipe, using the listed conditions with water as a fluid.

The support spacing for copper tubing is given in *Table 8* which includes the weight of the tubing filled with water.

Tables 7 and 8 are for "dead level" piping. Water and refrigerant lines are normally run level; steam lines are pitched. Water lines are pitched when the line must be drained. For pitched steam pipes, refer to *Table 22, page 82,* for support spacing when Schedule 40 pipe is used.

Unless pipe lines are adequately and properly anchored, expansion may put excessive strain on fittings and equipment. Anchors are located according to job conditions. For instance, on a tall building, i.e. 20 stories, the risers could be anchored on the 5th floor and on the 15th floor with an expansion device located at the 10th floor. This arrangement allows the riser to expand in both directions from the 5th and 15th floor, resulting in less pipe travel at headers, whether they are located at the top or bottom of the building or in both locations.

TABLE 7—RECOMMENDED SUPPORT SPACING FOR SCHEDULE 40 PIPE

NOMINAL PIPE SIZE (in.)	DISTANCE BETWEEN SUPPORTS (ft)
¾ - 1¼	8
1½ - 2½	10
3 - 3½	12
4 - 6	14
8 - 12	16
14 - 24	20

TABLE 8—RECOMMENDED SUPPORT SPACING FOR COPPER TUBING

TUBE OD (in.)	DISTANCE BETWEEN SUPPORTS (ft)
⅝	6
⅞ - 1⅛	8
1⅜ - 2⅛	10
2⅝ - 5⅛	12
6⅛ - 8⅛	14

On smaller buildings, i.e. 5 stories, risers are anchored but once. Usually this is done near the header, allowing the riser to grow in one direction only, either up or down depending on the header location.

The main point to consider when applying pipe support anchors and expansion joints is that expansion takes place on a temperature change. The greater the temperature change, the greater the expansion. The supports, anchors and guides are applied to restrain the expansion in a desired direction so that trouble does not develop because of negligent design or installation. For example, if a take-off connection from risers or headers is located close to floors, beams or columns as shown in *Fig. 1,* a change in temperature may cause a break in the take-off with subsequent loss of fluid and flooding damage. In this figure trouble develops when the riser expands greater than dimension "X." Proper consideration of these items is a must when designing piping systems.

VIBRATION ISOLATION OF PIPING SYSTEMS

The undesirable effects caused by vibration of the piping are:

1. Physical damage to the piping, which results in the rupture of joints. For refrigerant piping, loss of refrigerant charge results.
2. Transmission of noise thru the piping itself or thru the building construction where piping comes into direct contact.

It is always difficult to anticipate trouble resulting from vibration of the piping system. For this reason, recommendations made toward minimizing the effects of vibration are divided into two categories:

1. *Design consideration* – These involve design precautions that can prevent trouble effectively.
2. *Remedies or repairs* – These are necessary where precautions are not taken initially or, in a minority of cases, where the precautions prove to be insufficient.

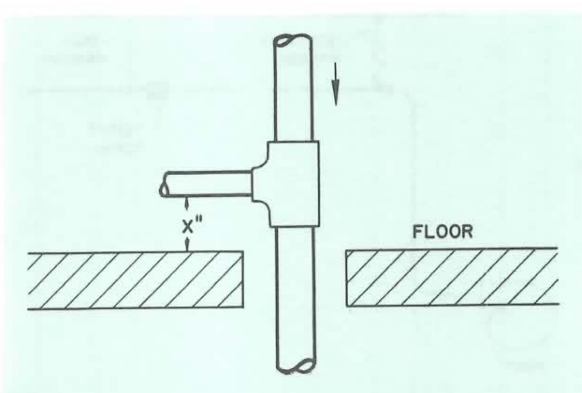


Fig. 1 – Take-off Too Close To Floor

Design Considerations for Vibration Isolation

1. In all piping systems vibration has an originating source. This source is usually a moving component such as a water pump or a compressor. When designing to eliminate vibration, the method of supporting these moving components is the prime consideration. For example:
   a. The weight of the mass supporting the components should be heavy enough to minimize the intensity of the vibrations transmitted to the piping and to the surrounding structure. The heavier the stabilizing mass, the smaller the intensity of the vibration.
   b. Vibration isolators can also be used to minimize the intensity of vibration.
   c. A combination of both methods may be required.
2. Piping must be laid out so that none of the lines are subject to the push-pull action resulting from vibration. Push-pull vibration is best dampened in torsion or bending action.
3. The piping must be supported securely at the proper places. The supports should have a relatively wide bearing surface to avoid a swivel action and to prevent cutting action on the pipe.

   The support closest to the source of vibration should be an isolation hanger and the succeeding hangers should have isolation sheaths as illustrated in *Fig. 2, page 8.* Non-isolated hangers (straps or rods attached directly to the pipe) should not be used on piping systems with machinery having moving parts.
4. The piping must not touch any part of the building when passing thru walls, floors, or furring. Sleeves which contain isolating material should be used wherever this is anticipated. Isolation hangers should be used to suspend the piping from walls and ceilings to prevent transmission of vibration to the building.

   Isolation hangers are also used where access to piping is difficult after installation.
5. Flexible hose is often of value in absorbing vibration on smaller sizes of pipe. To be effective, these flexible connectors are installed at right angles to the direction of the vibration.

1

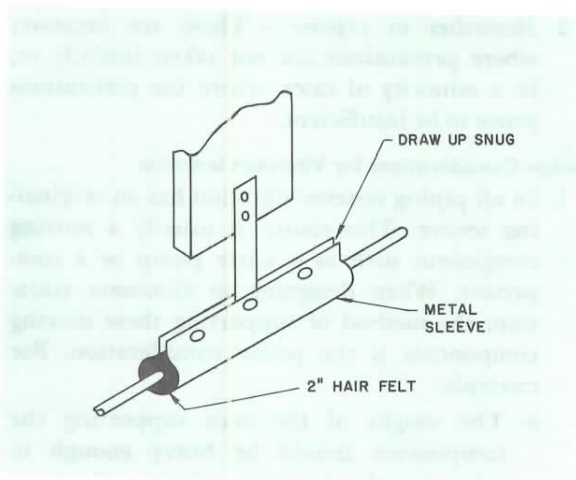


FIG. 2 — ISOLATED SHEATH PIPE HANGER

Where the vibration is not limited to one plane or direction, two flexible connectors are used and installed at right angles to each other. The flexible hose must not restrain the vibrating machine to which it is attached. At the opposite end of the hose or pair of hoses, a rigid but isolated anchor is secured to the connecting pipe to minimize vibration.

Generally, flexible hose is not recommended in systems subject to pressure conditions. Under pressure they become stiff and transmit vibration in the same manner as a straight length of pipe.

Flexible hose is not particularly efficient in absorbing vibration on larger sizes of pipe. Efficiency is impaired since the length-to-diameter ratio must be relatively large to obtain full flexibility from the hose. In practice the length which can be used is often limited, thus reducing its flexibility.

Remedies or Repairs for Vibration Isolation After Installation

1. Relocation of the piping supports by trial and error tends to dampen extraordinary pipe vibration. This relocation allows the piping to take up the vibration in bending and helps to correct any vibrations which cause mechanical resonance.
2. If relocation of the pipe supports does not eliminate the noise problem caused by vibration, there are several possible recommendations:
   a. The pipe may be isolated from the support by means of cork, hair felt, or pipe insulation as shown in *Fig. 2*.
   b. A weight may be added to the pipe before the first fixed support as illustrated in *Fig. 3*. This weight adds mass to the pipe, reducing vibration.
   c. Opposing isolation hangers may be added.

FITTINGS

Elbows are responsible for a large percentage of the pressure drop in the piping system. With equal velocities the magnitude of this pressure drop depends upon the sharpness of the turn. Long radius rather than short radius elbows are recommended wherever possible.

When laying out offsets, 45° ells are recommended over 90° ells wherever possible. See *Fig. 4*.

Tees should be installed to prevent "bullheading" as illustrated in *Fig. 5*. "Bullheading" causes turbulence which adds greatly to the pressure drop and may also introduce hammering in the line. If more than one tee is installed in the line, a straight length of 10 pipe diameters between tees is recommended. This is done to reduce unnecessary turbulence.

To facilitate erection and servicing, unions and flanges are included in the piping system. They are installed where equipment and piping accessories must be removed for servicing.

The various methods of joining fittings to the piping are described on *page 12*.

GENERAL PURPOSE VALVES

An important consideration in the design of the piping system is the selection of valves that give proper performance, long life and low maintenance.

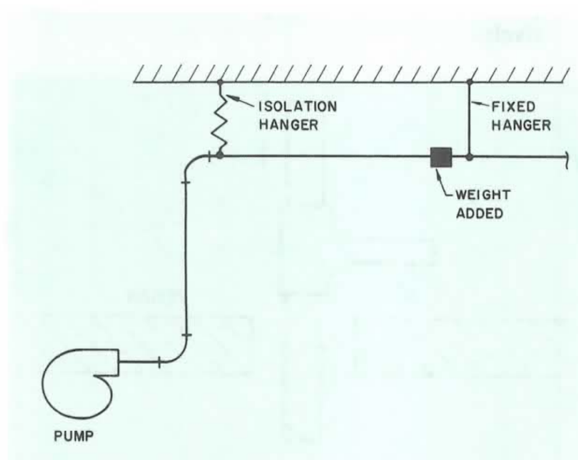


FIG. 3 — WEIGHT ADDED TO DAMPEN VIBRATION

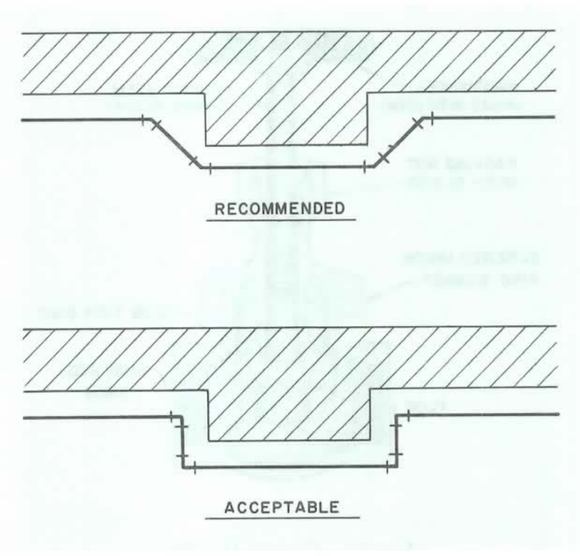


Fig. 4 – Offsets to Avoid Obstructions

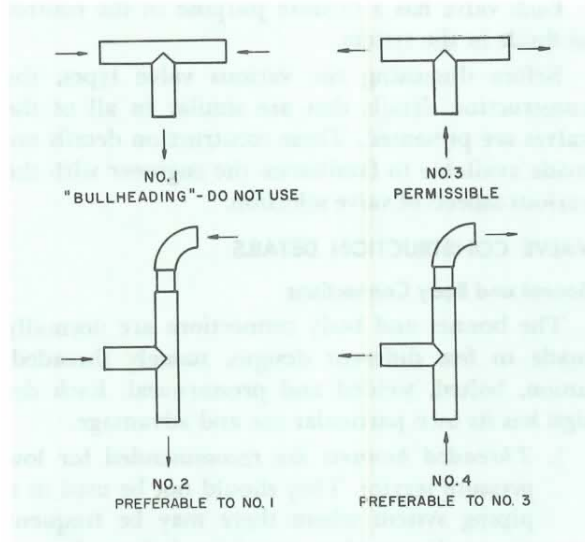


Fig. 5 – Tees

The design, construction and material of the valve determines whether or not it is suited for the particular application. *Table 9* is for quick reference in selecting a valve for a particular application. There are six basic valves which are commonly used in piping systems. These are gate, globe, check, angle, "Y" and plug cock.

TABLE 9—GENERAL PURPOSE VALVES

	WATER	STEAM	REFRIGERANT*
VALVE CONSTRUCTION			
A. Bonnet and body connections			
1. Threaded	Satisfactory	Satisfactory (Low Press)	Not Recommended
2. Union	Satisfactory	Satisfactory	Not Recommended
3. Bolted	Satisfactory	Satisfactory	Satisfactory
4. Welded	Satisfactory	Satisfactory (High Press)	Satisfactory
5. Pressure-seal	Satisfactory	Satisfactory	Satisfactory
B. Valve stem, operation			
1. Rising stem, outside screw	Satisfactory	Satisfactory	Satisfactory
2. Rising stem, inside screw	Satisfactory	Satisfactory	Satisfactory
3. Non-rising stem, inside screw	Satisfactory (non-corrosive brines)	Not Recommended	Not Recommended
4. Sliding stem	Satisfactory	Satisfactory	Not Recommended
C. Valve connections to pipe			
1. Screwed	Satisfactory	Satisfactory	Not Recommended
2. Welded	Satisfactory	Satisfactory	Recommended
3. Brazed	Satisfactory	Satisfactory (Low Temp)	Recommended
4. Soldered	Satisfactory	Satisfactory (Low Temp)	Satisfactory
5. Flared	Satisfactory	Satisfactory	Satisfactory
6. Flanged	Satisfactory (non-corrosive brines)	Satisfactory	Satisfactory
DISC CONSTRUCTION			
Gate Valve			
1. Solid wedge	Satisfactory	Satisfactory	Not Recommended
2. Split wedge	Satisfactory	Satisfactory	Not Recommended
3. Flexible wedge	Satisfactory	Recommended	Not Recommended
4. Double disc, parallel seat	Satisfactory	Not Recommended	Not Recommended
Globe, Angle, "Y" Valve			
1. Plug disc	Satisfactory	Satisfactory	Satisfactory
2. Conventional (Narrow-seat)	Satisfactory	Not Recommended	Satisfactory
3. Needle valve	Satisfactory	Satisfactory	Satisfactory
4. Composition disc	Satisfactory	Satisfactory (Low Press)	Satisfactory
Plug Cock Valve	Satisfactory	Satisfactory	Not Recommended

*For Refrigerants 12, 22, 500 and 502 only.

Each valve has a definite purpose in the control of fluids in the system.

Before discussing the various valve types, the construction details that are similar in all of the valves are presented. These construction details are made available to familiarize the engineer with the various aspects of valve selection.

VALVE CONSTRUCTION DETAILS

Bonnet and Body Connections

The bonnet and body connections are normally made in five different designs, namely threaded, union, bolted, welded and pressure-seal. Each design has its own particular use and advantage.

1. *Threaded bonnets* are recommended for low pressure service. They should not be used in a piping system where there may be frequent dismantling and reassembly of the valve, or where vibration, shock, and other severe conditions may strain and distort the valve body. Threaded bonnets are economical and very compact. *Fig. 6* illustrates a threaded or screwed-in bonnet and body connection in an angle valve.
2. *Union bonnet* and body construction is illustrated in *Fig. 7*. This type of bonnet is normally not made in sizes above 2 in. because it would require an extremely large wrench to dismantle. A union bonnet connection makes a sturdy, tight joint and is easily dismantled and reassembled.

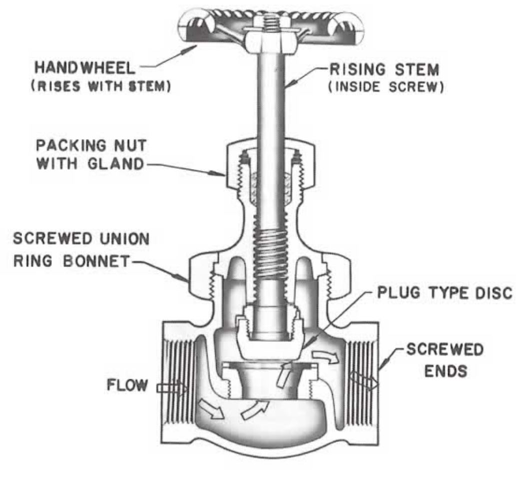


FIG. 7 — GLOBE VALVE

3. *Bolted bonnets* are used on practically all large size valves; they are also available for small sizes. This type of joint is readily taken apart or reassembled. The bolted bonnet is practical for high working pressure and is of rugged, sturdy construction. *Fig. 8* is a gate valve illustrating a typical bolted bonnet and body valve construction.
4. *Welded bonnets* are used on small size steel valves only, and then usually for high pressure, high temperature steam service *(Fig. 9)*. Welded bonnet construction is difficult to dismantle

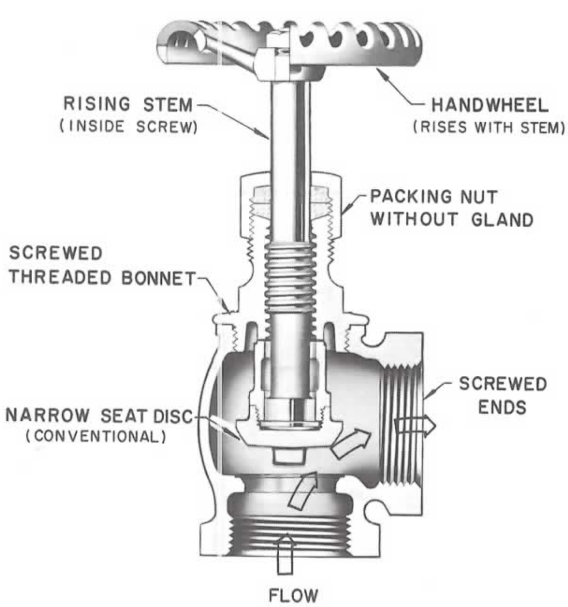


FIG. 6 — ANGLE VALVE

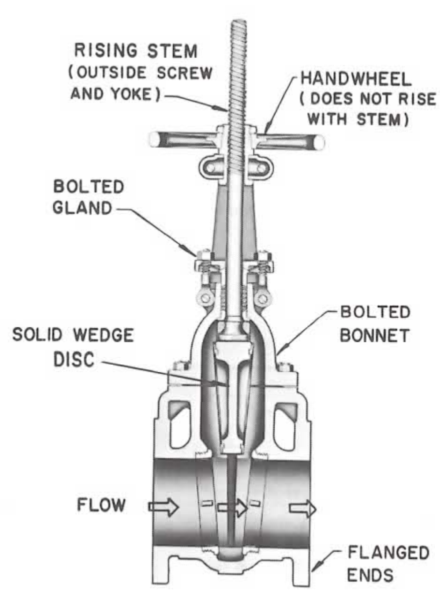


FIG. 8 — GATE VALVE (RISING STEM)

Figures 6-10, courtesy of Crane Co.

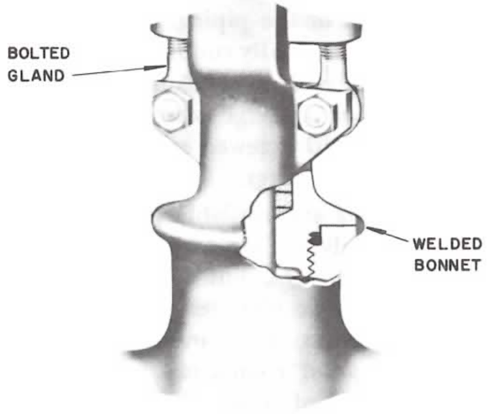


FIG. 9 – WELDED BONNET CONSTRUCTION

and reassemble. For this reason these valves are not available in larger sizes.

5. *Pressure-seal bonnets* are for high temperature steam. *Fig. 10* illustrates a pressure-seal bonnet and body construction used on a gate valve. Internal pressure keeps the bonnet joint tight. This type of bonnet construction simplifies "making" and "breaking" the bonnet joint in large high pressure valves.

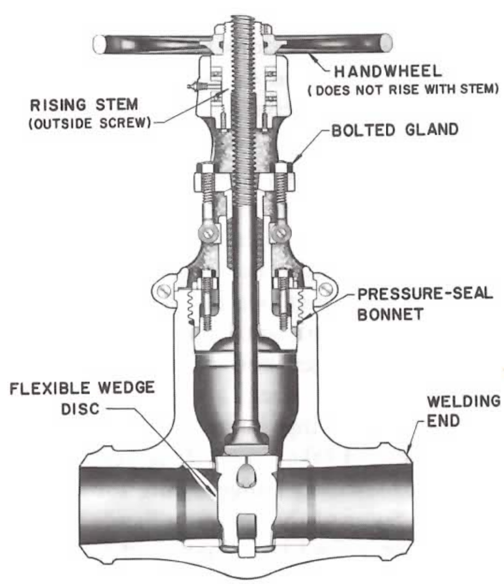


FIG. 10 – FLEXIBLE WEDGE DISC (GATE VALVE)

Valve Stem Operation

In most applications the type of stem operation does not affect fluid control. However, stem construction may be important where the need for indication of valve position is required or where head room is critical. There are four types of stem construction: rising stem with outside screw; rising stem with inside screw; non-rising stem with inside screw; and sliding stem (quick opening).

1. *Rising stem with outside screw* is shown in *Fig. 8.* The gate valve illustrated in this figure has the stem threads outside of the valve body in both the open and closed position. Stem threads are, therefore, not subject to corrosion, erosion, sediment, and galling from extreme temperature changes caused by elements in the line fluid flow. However, since the valve stem is outside the valve body, it is subject to damage when the valve is open. This type of stem is well suited to steam and high temperature, high pressure water service. A rising stem requires more headroom than a non-rising stem. The position of the stem indicates the position of the valve disc. The stem can be easily lubricated since it is outside the valve body.

2. *Rising stem with inside screw* is probably the most common type found in the smaller size valves. This type of stem is illustrated in an angle valve in *Fig. 6,* and in a globe valve in *Fig. 7.* The stem turns and rises on threads inside the valve body. The position of the stem also indicates position of the valve disc. The stem extends beyond the bonnet when the valve is open and, therefore, requires more headroom. In addition it is subject to damage.

3. *Non-rising stem with inside screw* is generally used on gate valves. It is undesirable for use with fluids that may corrode or erode the threads since the stem is in the path of flow. *Fig. 11* shows a gate valve that has a non-rising stem with the threads inside the valve body. The non-rising stem feature makes the valve ideally suited to applications where headroom is limited. Also, the stem cannot be easily damaged. The valve disc position is not indicated with this stem.

4. *Sliding stem (quick opening)* is useful where quick opening and closing is desirable. A lever and sliding stem is used which is suitable for both manual or power operation as illustrated in *Fig. 12.* The handwheel and stem threads are eliminated.

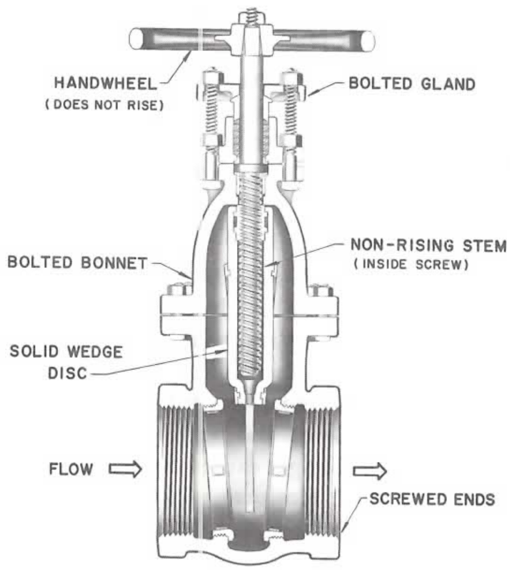


FIG. 11 — GATE VALVE (NON-RISING STEM)

Pipe Ends and Valve Connections

It is important to specify the proper end connection for valves and fittings. There are six standard methods of joints available. These are screwed, welded, brazed, soldered, flared, and flanged ends, and are described in the following:

1. *Screwed ends* are widely used and are suited for all pressures. To remove screwed end valves and fittings from the line, extra fittings (unions) are required to avoid dismantling a considerable portion of the piping. Screwed end connections are normally confined to smaller pipe sizes since it is more difficult to make up the screwed joint on large pipe sizes. *Fig. 7* is a globe valve with screwed ends that connect to pipe or other fittings.
2. *Welded ends* are available for steel pipe, fittings, and valves. They are used widely for all fitting connections, but for valves they are used mainly for high temperature and high pressure services. They are also used where a tight, leak-proof connection is required over a long period of time. The welded ends are available in two designs, butt weld or socket weld. Butt weld valves and fittings come in all sizes; socket weld ends are usually limited to the smaller size valves and fittings. *Fig. 10* illustrates a gate valve with ends suitable for welding.
3. *Brazed ends* are designed for brazing alloys. This type of joint is similar to the solder joint but can withstand higher temperature service because of the higher melting point of the brazing alloy. Brazing joints are used principally on brass valves and fittings.
4. *Soldered ends* for valve and fitting are restricted to copper pipe and also for low pressure service. The use of this type of joint for high temperature service is limited because of the low melting point of the solder.
5. *Flared end* connections for valves and fittings are commonly used on metal and plastic tubing. This type of connection is limited to pipe sizes up to 2 in. Flared connections have the advantage of being easily removed from the piping system at any time.
6. *Flanged ends* are higher in first cost than any of the other end connections. The installation cost is also greater because companion flanges, gaskets, nuts and bolts must be provided and installed. Flanged end connections, although made in small sizes, are generally used in larger size piping because they are easy to assemble and take down. It is very important to have matching flange facing for valves and fittings. Some of the common flange facings are plain face, raised face, male and female joint, tongue and groove joint, and a ring joint. Flange facings should never be mixed in making up a joint. *Fig. 8* illustrates a gate valve with a flanged end.

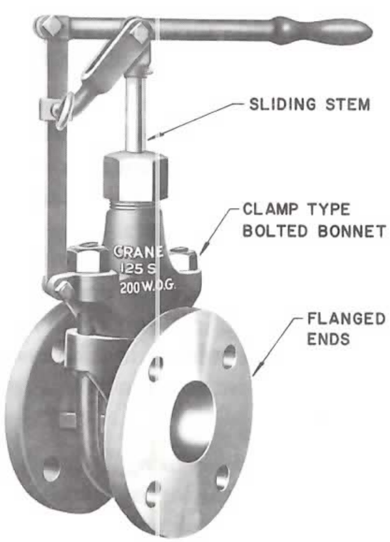


FIG. 12 — SLIDING STEM GATE VALVE

GATE VALVES

A gate valve is intended for use as a stop valve. It gives the best service when used in the fully open or closed position. *Figures 8 and 10 thru 14* are typical gate valves commonly used in piping practice.

An important feature of the gate valve is that there is less obstruction and turbulence within the valve and, therefore, a correspondingly lower pressure drop than other valves. With the valve wide open, the wedge or disc is lifted entirely out of the fluid stream, thus providing a straight flow area thru the valve.

Disc Construction

Gate valves should not be used for throttling flow except in an emergency. They are not designed for this type of service and consequently it is difficult to control flow with any degree of accuracy. Vibration and chattering of the disc occurs when the valve is used for throttling. This results in damage to the seating surface. Also, the lower edge of the disc may be subject to severe wire drawing effects. The wedge or disc in the gate valve is available in several forms: solid wedge, split wedge, flexible wedge, and double disc parallel seat. These are described in the following:

1. *Solid wedge disc* is the most common type. It has a strong, simple design and only one part. This type of disc is shown in *Figs. 8 and 11*. It can be installed in any position without danger of jamming or misalignment of parts. It is satisfactory for all types of service except where the possibility of extreme temperature changes exist. Under this condition it is subject to sticking.
2. *Split wedge disc* is designed to prevent sticking, but it is subject to undesirable vibration intensity. *Fig. 13* is a typical illustration of this type of disc.
3. *Flexible wedge disc* construction is illustrated in *Fig. 10*. This type of disc is primarily used for high temperature, high pressure applications and where extreme temperature changes are likely to occur. It is solid in the center portion and flexible around the outer edge. This design helps to eliminate sticking and permits the disc to open easily under all conditions.
4. *Double disc parallel seat (Fig. 14)* has an internal wedge between parallel discs. Wedge action damage at the seats is minimized and transferred to the internal wedge where reasonable wear does not prevent tight closure.

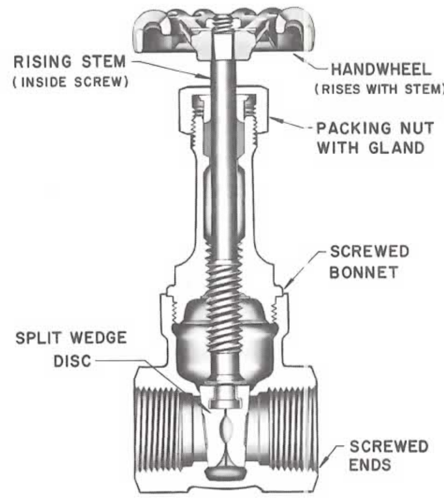


Fig. 13 – Split Wedge Disc (Gate Valve)

The parallel sliding motion of the discs tends to clean the seating surfaces and prevents foreign material from being wedged between disc and seat.

Since the discs are loosely supported except when wedged closed, this design is subject to vibration of the disc assembly parts when partially open.

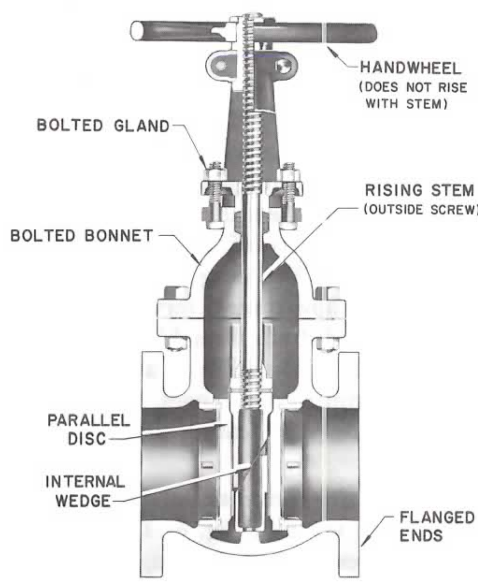


Fig. 14 – Double Disc Parallel Seat (Gate Valve)

Figures 11-14, courtesy of Crane Co.

When used in steam service, the closed valve may trap steam between the discs where it condenses and creates a vacuum. This may result in leakage at the valve seats.

GLOBE, ANGLE AND "Y" VALVES

These three valves are of the same basic design, use and construction. They are primarily intended for throttling service and give close regulation of flow. The method of valve seating reduces wire drawing and seat erosion which is prevalent in gate valves when used for throttling service.

The angle or "Y" valve pattern is recommended for full flow service since it has a substantially lower pressure drop at this condition than the globe valve. Another advantage of the angle valve is that it can be located to replace an elbow, thus eliminating one fitting.

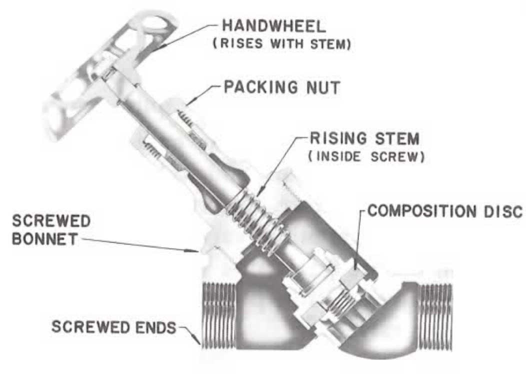
Fig. 7, page 10, is a typical illustration of a globe valve, and *Fig. 6, page 10,* shows an angle valve. The "Y" valve is illustrated in *Fig. 15.*

Globe, angle and "Y" valves can be opened or closed substantially faster than a gate valve because of the shorter lift of the disc. When valves are to be operated frequently or continuously, the globe valve provides the more convenient operation. The seating surfaces of the globe, angle or "Y" valve are less subject to wear and the discs and seats are more easily replaced than on the gate valve.

Disc Construction

There are several different disc and seating arrangements for globe, angle and "Y" valves, each of which has its own use and advantage. The different types are plug disc, narrow seat (or conventional disc), needle valve, and composition disc.

1. The *plug disc* has a wide bearing surface on a long, tapered disc and matching seat. This type of construction offers the greatest resistance to the cutting effects of dirt, scale and other foreign matter. The plug type disc is ideal for the toughest flow control service such as throttling, drip and drain lines, blow-off, and boiler feed lines. It is available in a wide variety of pressure temperature ranges. *Fig. 7, page 10,* shows a plug disc seating arrangement in a globe valve.
2. *Narrow seat* (or conventional disc) is illustrated in an angle valve in *Fig. 6.* This type of disc does not resist wire drawing or erosion in closely throttled high velocity flow. It is also subject to erosion from hard particles. The narrow seat disc design is not applicable for close throttling.



Courtesy of Jenkins Bros.

FIG. 15 — "Y" VALVE

3. *Needle valves,* sometimes referred to as expansion valves, are designed to give fine control of flow in small diameter piping. The disc is normally an integral part of the stem and has a sharp point which fits into the reduced area seat opening. *Fig. 16* is an angle valve with a needle type seating arrangement.
4. *Composition disc* is adaptable to many services by simply changing the material of the disc. It has the advantage of being able to seat tight with less power than the metal type discs. It is also less likely to be damaged by dirt or foreign material than the metal disc. A composition disc is suitable to all moderate pressure services but not for close regulating and throttling. *Fig. 15* shows a composition disc in a "Y" valve. This type of seating design is also illustrated in *Fig. 19* in a swing check valve and in *Fig. 20* in a lift check valve.

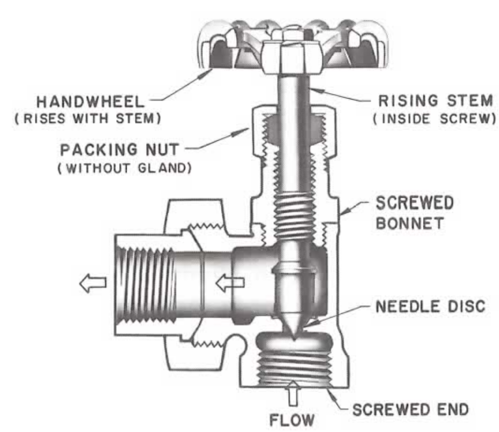


FIG. 16 — ANGLE VALVE WITH NEEDLE DISC

PLUG COCKS

Plug cocks are primarily used for balancing in a piping system not subject to frequent changes in flow. They are normally less expensive than globe type valves and the setting cannot be tampered with as easily as a globe valve.

Plug cocks have approximately the same line loss as a gate valve when in the fully open position. When partially closed for balancing, this line loss increases substantially. *Fig. 17* is a lubricated type plug valve.

REFRIGERANT VALVES

Refrigerant valves are back-seating globe valves of either the packed or diaphragm packless type. The packed valves are available with either a hand wheel or a wing type seal cap. The wing type seal cap is preferable since it provides the safety of an additional seal.

Where frequent operation of the valve is required, the diaphragm packless type is used. The diaphragm acts as a seal and is illustrated in the "Y" valve construction in *Fig. 18*. The refrigerant valve is available in sizes up to $1\frac{5}{8}$ in. OD. For larger sizes the seal cap type packed valve is used.

CHECK VALVES

There are two basic designs of check valves, the swing check and the lift check.

The swing check valve may be used in a horizontal or a vertical line for upward flow. A typical swing check valve is illustrated in *Fig. 19*. The flow thru the swing check is in a straight line and without restriction at the seat. Swing checks are generally used in combination with gate valves.

Courtesy of Walworth Co.

Fig. 17 – Plug Cock

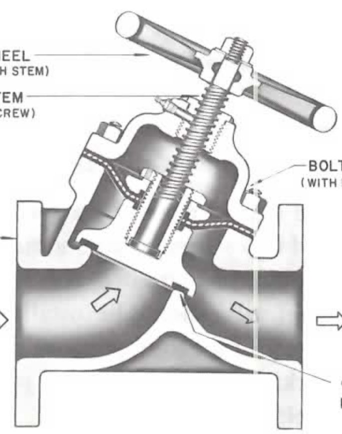


Fig. 18 – "Y" Valve (Diaphragm Type)

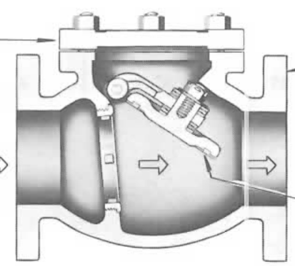


Fig. 19 – Swing Check Valve

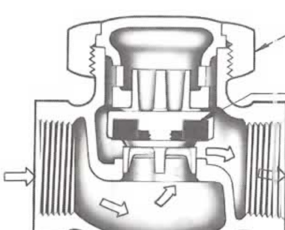


Fig. 20 – Lift Check Valve

The lift check operates in a manner similar to that of a globe valve and, like the globe valve, its flow is restricted as illustrated in *Fig. 20*. The disc is seated by backflow or by gravity when there is no flow, and is free to rise and fall, depending on the pressure under it. The lift check should only be installed in horizontal pipe lines and usually in combination with globe, angle and "Y" valves.

Figures 16, 18-20, courtesy of Crane Co.

1

SPECIAL SERVICE VALVES

There are several types of valves commonly used in different piping applications that do not necessarily fall into the classification of general purpose valves. Expansion, relief, and solenoid valves are some of the more common special purpose valves.

A relief valve is held closed by a spring or some other means and is designed to automatically relieve the line or container pressure in excess of its setting. In general a relief valve should be installed wherever there is any danger of the fluid pressure rising above the design working pressure of the pipe fittings or pressure vessels.

VALVE AND FITTING PRESSURE LOSSES

To properly design any type of piping system conveying a fluid, the losses thru the valves and fittings in the system must be realistically evaluated. Tables have been prepared for determining these losses in terms of equivalent length of pipe. These values are then used with the correct friction chart for the particular fluid flowing thru the system.

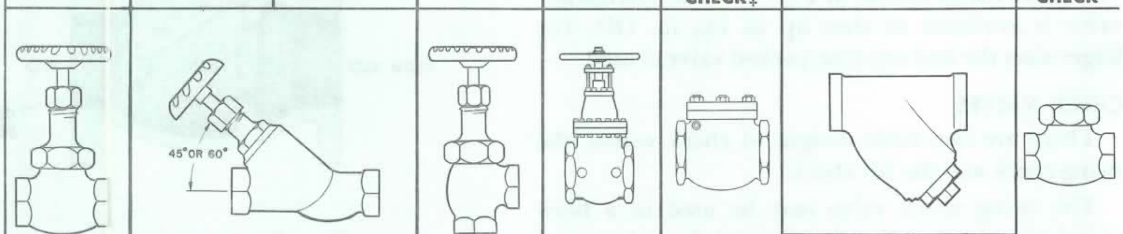
Table 10 gives valve losses with screwed, flanged, flared, welded, soldered, or brazed connections.

Table 11 gives fitting losses with screwed, flanged, flared, welded, soldered, or brazed connections.

Table 12 lists the losses for special types of fittings sometimes encountered in piping applications.

TABLE 10—VALVE LOSSES IN EQUIVALENT FEET OF PIPE*

Screwed, Welded, Flanged, and Flared Connections



NOMINAL PIPE OR TUBE SIZE (in.)	GLOBE†	60°-Y	45°-Y	ANGLE†	GATE††	SWING CHECK‡	Y-TYPE STRAINER‡‡ Flanged End	Y-TYPE STRAINER‡‡ Screwed End	LIFT CHECK
3/8	17	8	6	6	0.6	5	—	—	
1/2	18	9	7	7	0.7	6	—	3	
3/4	22	11	9	9	0.9	8	—	4	Globe & Vertical Lift Same as Globe Valve**
1	29	15	12	12	1.0	10	—	5	
1¼	38	20	15	15	1.5	14	—	9	
1½	43	24	18	18	1.8	16	—	10	
2	55	30	24	24	2.3	20	27	14	
2½	69	35	29	29	2.8	25	28	20	
3	84	43	35	35	3.2	30	42	40	
3½	100	50	41	41	4.0	35	48	—	
4	120	58	47	47	4.5	40	60	—	
5	140	71	58	58	6	50	80	—	
6	170	88	70	70	7	60	110	—	
8	220	115	85	85	9	80	150	—	
10	280	145	105	105	12	100	190	—	
12	320	165	130	130	13	120	250	—	Angle Lift Same as Angle Valve
14	360	185	155	155	15	135	—	—	
16	410	210	180	180	17	150	—	—	
18	460	240	200	200	19	165	—	—	
20	520	275	235	235	22	200	—	—	
24	610	320	265	265	25	240	—	—	

*Losses are for all valves in fully open position and strainers clean.

†These losses do not apply to valves with needle point type seats.

‡Losses also apply to the in-line, ball type check valve.

**For "Y" pattern globe lift check valve with seat approximately equal to the nominal pipe diameter, use values of 60° "Y" valve for loss.

††Regular and short pattern plug cock valves, when fully open, have same loss as gate valve. For valve losses of short pattern plug cocks above 6 ins. check manufacturer.

‡‡For .045 thru 3/16 in. perforations with screens 50% clogged, loss is doubled.

TABLE 11—FITTING LOSSES IN EQUIVALENT FEET OF PIPE

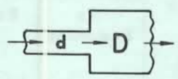
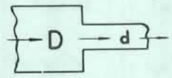
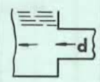
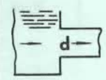
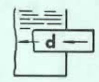
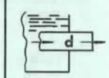
Screwed, Welded, Flanged, Flared, and Brazed Connections

NOMINAL PIPE OR TUBE SIZE (in.)	SMOOTH BEND ELBOWS						SMOOTH BEND TEES			
	90° Std*	90° Long Rad.†	90° Street*	45° Std*	45° Street*	180° Std*	Flow-Thru Branch	Straight-Thru Flow		
								No Reduction	Reduced ¼	Reduced ½
⅜	1.4	0.9	2.3	0.7	1.1	2.3	2.7	0.9	1.2	1.4
½	1.6	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.6
¾	2.0	1.4	3.2	0.9	1.6	3.2	4.0	1.4	1.9	2.0
1	2.6	1.7	4.1	1.3	2.1	4.1	5.0	1.7	2.3	2.6
1¼	3.3	2.3	5.6	1.7	3.0	5.6	7.0	2.3	3.1	3.3
1½	4.0	2.6	6.3	2.1	3.4	6.3	8.0	2.6	3.7	4.0
2	5.0	3.3	8.2	2.6	4.5	8.2	10	3.3	4.7	5.0
2½	6.0	4.1	10	3.2	5.2	10	12	4.1	5.6	6.0
3	7.5	5.0	12	4.0	6.4	12	15	5.0	7.0	7.5
3½	9.0	5.9	15	4.7	7.3	15	18	5.9	8.0	9.0
4	10	6.7	17	5.2	8.5	17	21	6.7	9.0	10
5	13	8.2	21	6.5	11	21	25	8.2	12	13
6	16	10	25	7.9	13	25	30	10	14	16
8	20	13	—	10	—	33	40	13	18	20
10	25	16	—	13	—	42	50	16	23	25
12	30	19	—	16	—	50	60	19	26	30
14	34	23	—	18	—	55	68	23	30	34
16	38	26	—	20	—	62	78	26	35	38
18	42	29	—	23	—	70	85	29	40	42
20	50	33	—	26	—	81	100	33	44	50
24	60	40	—	30	—	94	115	40	50	60

NOMINAL PIPE OR TUBE SIZE (in.)	MITRE ELBOWS			
	90° Ell	60° Ell	45° Ell	30° Ell
⅜	2.7	1.1	0.6	0.3
½	3.0	1.3	0.7	0.4
¾	4.0	1.6	0.9	0.5
1	5.0	2.1	1.0	0.7
1¼	7.0	3.0	1.5	0.9
1½	8.0	3.4	1.8	1.1
2	10	4.5	2.3	1.3
2½	12	5.2	2.8	1.7
3	15	6.4	3.2	2.0
3½	18	7.3	4.0	2.4
4	21	8.5	4.5	2.7
5	25	11	6.0	3.2
6	30	13	7.0	4.0
8	40	17	9.0	5.1
10	50	21	12	7.2
12	60	25	13	8.0
14	68	29	15	9.0
16	78	31	17	10
18	85	37	19	11
20	100	41	22	13
24	115	49	25	16

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

TABLE 12—SPECIAL FITTING LOSSES IN EQUIVALENT FEET OF PIPE

NOM. PIPE OR TUBE SIZE (in.)	SUDDEN ENLARGEMENT* d/D			SUDDEN CONTRACTION* d/D			SHARP EDGE*		PIPE PROJECTION*	
	¼	½	¾	¼	½	¾	Entrance	Exit	Entrance	Exit
										
⅜	1.4	0.8	0.3	0.7	0.5	0.3	1.5	.8	1.5	1.1
½	1.8	1.1	0.4	0.9	0.7	0.4	1.8	1.0	1.8	1.5
¾	2.5	1.5	0.5	1.2	1.0	0.5	2.8	1.4	2.8	2.2
1	3.2	2.0	0.7	1.6	1.2	0.7	3.7	1.8	3.7	2.7
1¼	4.7	3.0	1.0	2.3	1.8	1.0	5.3	2.6	5.3	4.2
1½	5.8	3.6	1.2	2.9	2.2	1.2	6.6	3.3	6.6	5.0
2	8.0	4.8	1.6	4.0	3.0	1.6	9.0	4.4	9.0	6.8
2½	10	6.1	2.0	5.0	3.8	2.0	12	5.6	12	8.7
3	13	8.0	2.6	6.5	4.9	2.6	14	7.2	14	11
3½	15	9.2	3.0	7.7	6.0	3.0	17	8.5	17	13
4	17	11	3.8	9.0	6.8	3.8	20	10	20	16
5	24	15	5.0	12	9.0	5.0	27	14	27	20
6	29	22	6.0	15	11	6.0	33	19	33	25
8	—	25	8.5	—	15	8.5	47	24	47	35
10	—	32	11	—	20	11	60	29	60	46
12	—	41	13	—	25	13	73	37	73	57
14	—	—	16	—	—	16	86	45	86	66
16	—	—	18	—	—	18	96	50	96	77
18	—	—	20	—	—	20	115	58	115	90
20	—	—	—	—	—	—	142	70	142	108
24	—	—	—	—	—	—	163	83	163	130

*Enter table for losses at smallest diameter "d."

CHAPTER 2. WATER PIPING

This chapter presents the principles and currently accepted design techniques for water piping systems used in air conditioning applications. It also includes the various piping arrangements for air conditioning equipment and the standard accessories found in most water piping systems.

The principles and techniques described are applicable to chilled water and hot water heating systems. General piping principles and techniques are described in *Chapter 1*.

WATER PIPING SYSTEMS

Once-Thru and Recirculating

The water piping systems discussed here are divided into once-thru and recirculating types. In a once-thru system water passes thru the equipment only once and is discharged. In a recirculating system water is not discharged, but flows in a repeating circuit from the heat exchanger to the refrigeration equipment and back to the heat exchanger.

Open and Closed

The recirculating system may be classified as open or closed. An open system is one in which the water flows into a reservoir open to the atmosphere; cooling towers, air washers and collecting tanks are examples of reservoirs open to the atmosphere. A closed system is one in which the flow of water is not exposed to the atmosphere at any point. This system usually contains an expansion tank that is open to the atmosphere but the water area exposed is insignificant.

Water Return Arrangements

The recirculating system is further classified according to water return arrangements. When two or more units of a closed system are piped together, one of the following piping arrangements may be used:

1. Reverse return piping.
2. Reverse return header with direct return risers.
3. Direct return piping.

If the units have the same or nearly the same pressure drop thru them, one of the reverse return methods of piping is recommended. However, if the units have different pressure drops or require balancing valves, then it is usually more economical to use a direct return.

Reverse return piping is recommended for most closed piping applications. It is often the most economical design on new construction. The length of the water circuit thru the supply and return piping is the same for all units. Since the water circuits are equal for each unit, the major advantage of a reverse return system is that it seldom requires balancing. *Fig. 21* is a schematic sketch of this system with units piped horizontally and vertically.

There are installations where it is both inconvenient and economically unsound to use a complete reverse return water piping system. This sometimes exists in a building where the first floor has previously been air conditioned. To avoid disturbing the first floor occupants, reverse return headers are located at the top of the building and direct return risers to the units are used. *Fig. 22* illustrates a reverse return header and direct return riser piping system.

In this system the flow rate is not equal for all units on a direct return riser. The difference in flow rate depends on the design pressure drop of the supply and return riser. This difference can be reduced to practical limits. The pressure drop across the riser includes the following: (1) the loss thru the supply and return runouts from the riser to the unit, (2) the loss thru the unit itself, and (3) the loss thru

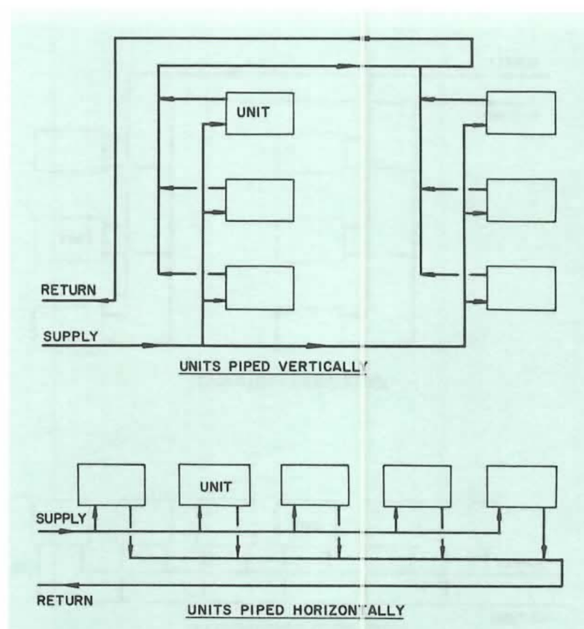


Fig. 21 – Reverse Return Piping

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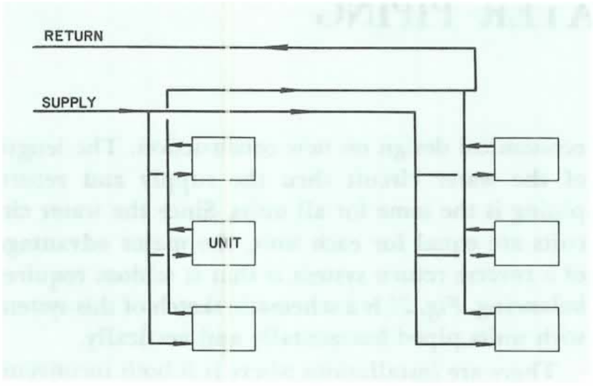

Fig. 22 — Reverse Return Headers with Direct Return Risers

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the fittings and valves. Excessive unbalance in the direct supply and return portion of the piping system may dictate the need for balancing valves or orifices.

To eliminate balancing valves, design the supply and return pressure drop equal to one-fourth the sum of the pressure drops of the preceding *Items 1, 2 and 3.*

Direct return piping is necessary for open piping systems and is recommended for some closed piping systems. A reverse return arrangement on an open system requires piping that is normally unnecessary, since the same atmospheric conditions exist at all open points of the system. A direct return is recommended for a closed recirculating system where all the units require balancing valves and have different pressure drops. Several fan-coil units piped together and requiring different water flow rates, capacities and pressure drops is an example of this type of system.

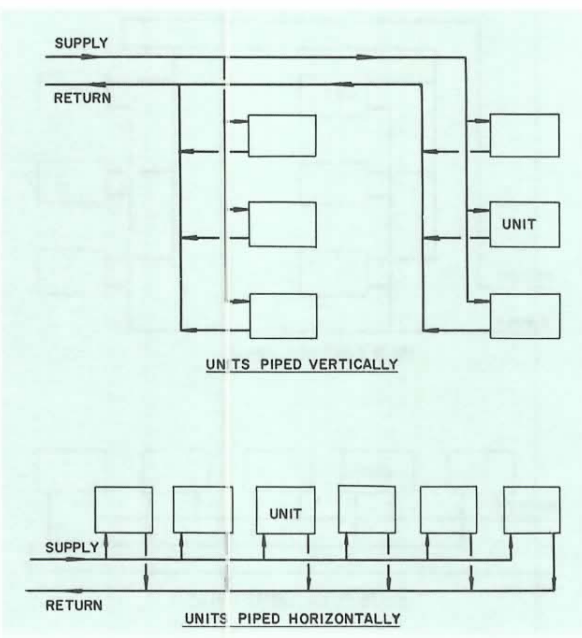

Fig. 23 — Direct Return Water Piping System

The direct return piping system is inherently unbalanced and requires balancing valves or orifices, and provisions to measure the pressure drop in order to meter the water flow. Although material costs are lower in this system than in the two reverse return systems, engineering cost and balancing time often offset this advantage. *Fig. 23* illustrates units piped vertically and horizontally to a direct return.

CODES AND REGULATIONS

All applicable codes and regulations should be checked to determine acceptable piping practice for the particular application. Sometimes these codes and regulations dictate piping design, limit the pressure, or qualify the selection of materials and equipment.

WATER CONDITIONING

Normally all water piping systems must have adequate treatment to protect the various components against corrosion, scale, slime and algae. Water treatment should always be under the supervision of a water conditioning specialist. Periodic inspection of the water is required to maintain suitable quality. *Part 5* of this manual contains a discussion of the various aspects of water conditioning including cause, effect and remedies for corrosion, scale, slime and algae.

WATER PIPING DESIGN

There is a friction loss in any pipe thru which water is flowing. This loss depends on the following factors:

1. Water velocity
2. Pipe diameter
3. Interior surface roughness
4. Pipe length

System pressure has no effect on the head loss of the equipment in the system. However, higher than normal system pressures may dictate the use of heavier pipe, fittings and valves along with specially designed equipment.

To properly design a water piping system, the engineer must evaluate not only the pipe friction loss but the loss thru valves, fittings and other equipment. In addition to these friction losses, the use of diversity in reducing the water quantity and pipe size is to be considered in designing the water piping system.

PIPE FRICTION LOSS

The pipe friction loss in a system depends on water velocity, pipe diameter, interior surface roughness and pipe length. Varying any one of these factors influences the total friction loss in the pipe.

Most air conditioning applications use either steel pipe or copper tubing in the piping system. The friction loss based on the Darcy-Weisbach formula is presented in *Charts 3 thru 5* in this chapter.

Charts 3 and 4 are for Schedule 40 pipe up to 24 in. in diameter. *Chart 3* shows the friction losses for closed recirculation piping systems. The friction losses in *Chart 4* are for open once-thru and for open recirculation piping systems.

Chart 5 shows friction losses for Types K, L and M copper tubing when used in either open or closed water systems.

These charts show water velocity, pipe or tube diameter, and water quantity, in addition to the friction rate per 100 ft of equivalent pipe length. Knowing any two of these factors, the other two can be easily determined from the chart. The effect of inside roughness of the pipe or tube is considered in all these values.

The water quantity is determined from the air conditioning load and the water velocity by predetermined recommendations. These two factors are used to establish pipe size and friction rate.

Charts 3 thru 5 are shaded to indicate velocities above 15 fps and friction rates above 10 ft per 100 ft of length. It is normally good practice not to exceed these values.

Water Velocity

The velocities recommended for water piping depend on two conditions:

1. The service for which the pipe is to be used.
2. The effects of erosion.

Table 13 lists recommended velocity ranges for different services. The design of the water piping system is limited by the maximum permissible flow velocity. The maximum values listed in *Table 13* are based on established permissible sound levels of moving water and entrained air, and on the effects of erosion.

Erosion in water piping systems is the impingement on the inside surface of tube or pipe of rapidly moving water containing air bubbles, sand or other solid matter. In some cases this may mean complete deterioration of the tube or pipe walls, particularly on the bottom surface and at the elbows.

Since erosion is a function of time, water velocity, and suspended materials in the water, the selection of a design water velocity is a matter of judgment. The maximum water velocities presented in *Table 14* are based on many years of experience and they insure the attainment of optimum equipment life under normal conditions.

TABLE 13—RECOMMENDED WATER VELOCITY

SERVICE	VELOCITY RANGE (fps)
Pump discharge	8 - 12
Pump suction	4 - 7
Drain line	4 - 7
Header	4 - 15
Riser	3 - 10
General service	5 - 10
City water	3 - 7

Friction Rate

The design of a water piping system is limited by the friction loss. Systems using city water must have the piping sized so as to provide the required flow rate at a pressure loss within the pressure available at the city main. This pressure or friction loss is to include all losses in the system, as condenser pressure drop, pipe and fitting losses, static head, and water meter drop. The total system pressure drop must be less than the city main pressure to have design water flow.

A recirculating system is sized to provide a reasonable balance between increased pumping horsepower due to high friction loss and increased piping first cost due to large pipe sizes. In large air conditioning applications this balance point is often taken as a maximum friction rate of 10 ft of water per 100 ft of equivalent pipe length.

In the average air conditioning application the installed cost of the water piping exceeds the cost of the water pumps and motors. The cost of increasing the pipe size of small pipe to reduce the friction rate is normally not too great, whereas the installed cost increases rapidly when the size of large pipe (approximately 4 in. and larger) is increased. Smaller pipes can be economically sized at lower friction rates (increasing the pipe size) than the larger pipes. In most applications economic considerations dictate that larger pipe be sized for higher flow rates

TABLE 14—MAXIMUM WATER VELOCITY TO MINIMIZE EROSION

NORMAL OPERATION hr/year	WATER VELOCITY (fps)
1500	15
2000	14
3000	13
4000	12
6000	10
8000	8

CHART 3—FRICTION LOSS FOR CLOSED PIPING SYSTEMS

Schedule 40 Pipe

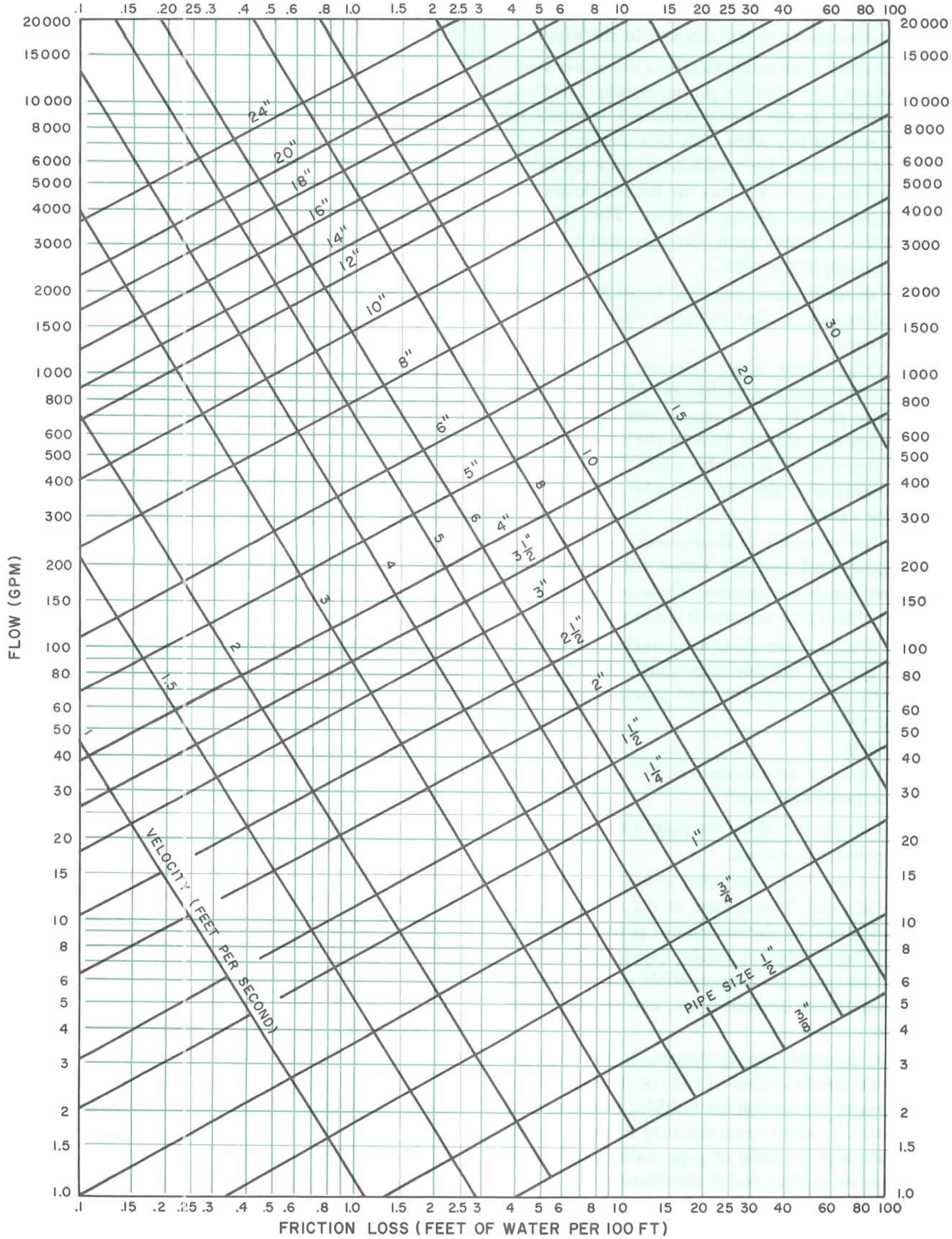

CHART 4—FRICTION LOSS FOR OPEN PIPING SYSTEMS

Schedule 40 Pipe

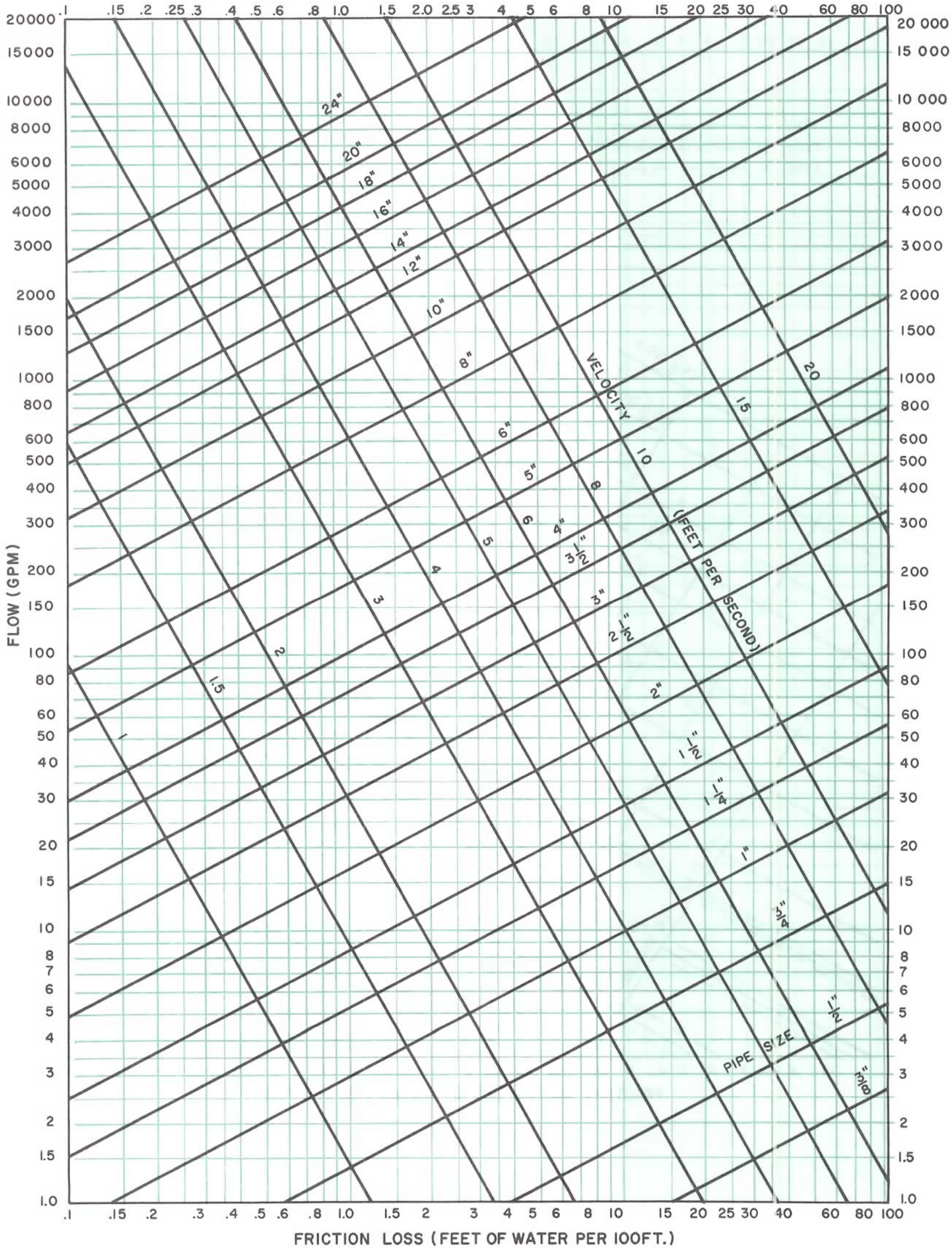

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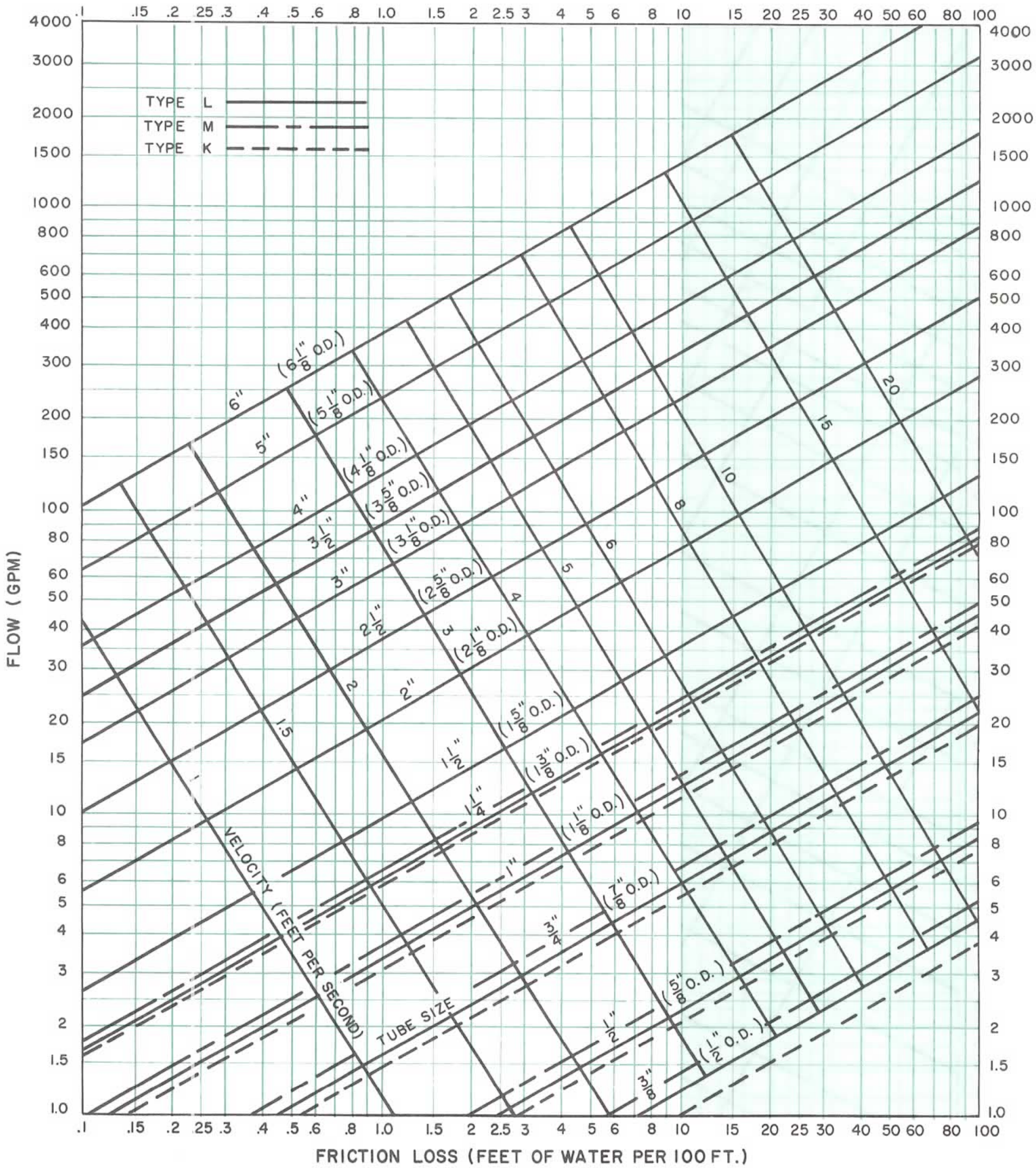
CHART 5—FRICTION LOSS FOR CLOSED AND OPEN PIPING SYSTEMS

Copper Tubing



and pressure drops than smaller pipe which is sized for lower pressure drops and flow rates.

Exceptions to this general guide often occur. For example, appearance or physical limitations may dictate the use of small pipes. This is often done for short runs where the total pressure drop is not greatly influenced.

Each system should be analyzed separately to determine the economic balance between first cost (pipe size, pump and motor) and operating cost (pressure drop, pump and motor).

Pipe Length

To determine the friction loss in a water piping system, the engineer must calculate the straight lengths of pipe and evaluate the additional equivalent lengths of pipe due to fittings, valves and other elements in the piping system. *Tables 10, 11 and 12* give the additional equivalent lengths of pipe for these various components. The straight length of pipe is measured to the centerline of all fittings and valves *(Fig. 24)*. The equivalent length of the components must be added to this straight length of pipe.

WATER PIPING DIVERSITY

When the air conditioning load is determined for each exposure of a building, it is assumed that the exposure is at peak load. Since the sun load is at a maximum on one exposure at a time, not all of the units on all the exposures require maximum water flow at the same time to handle the cooling load. Units on the same exposure normally require maximum flow at the same time; units on the adjoining or opposite exposures do not. Therefore, if the individual units are *automatically controlled* to vary the water quantity, the system water quantity actually required during normal operation is less than the total water quantity required for the peak design conditions for all the exposures. Good engineering design dictates that the water piping and the pump be sized for this reduced water quantity.

The principle of diversity allows the engineer to evaluate and calculate the reduced water quantity. In all water piping systems two conditions must be satisfied before diversity can be applied:

1. The water flow to the units must be automatically controlled to compensate for varying loads.
2. Diversity may only be applied to piping that supplies units on more than one exposure.

Figure 25 is a typical illustration of a header layout to which diversity may be applied. In this illustration the header piping supplies all four exposures. Assuming that the units supplied are automatically controlled, diversity is applied to the west, south and east exposures only. The last leg or exposure is never reduced in water quantity or pipe size since it requires full flow at some time during operation to meet design conditions.

Figure 26 illustrates another layout where diversity may be used to reduce pipe size and pump capacity. In this illustration diversity may be applied to the vertical supply and return headers and also to the supply and return branch headers at each floor. Diversity is not applied to pipe section 7-8 of both the supply and return vertical headers. In addition

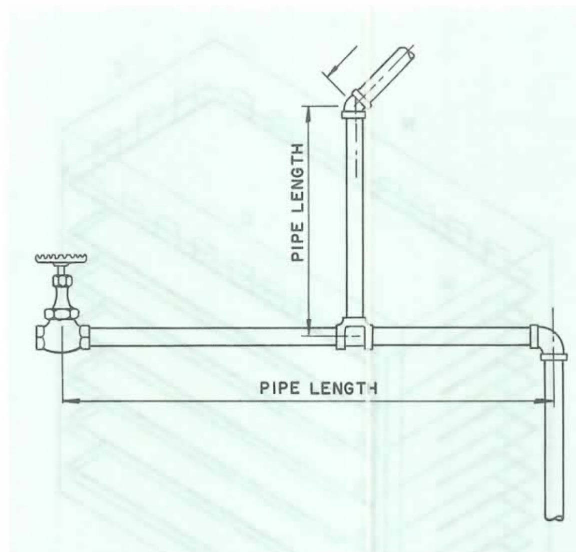


FIG. 24 — PIPE LENGTH MEASUREMENT

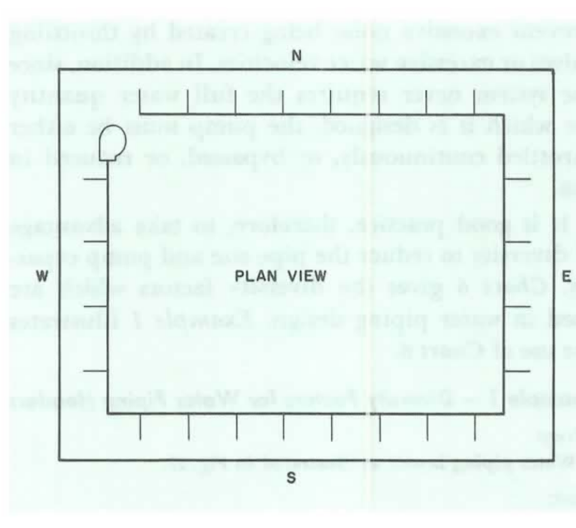


FIG. 25 — HEADER PIPING

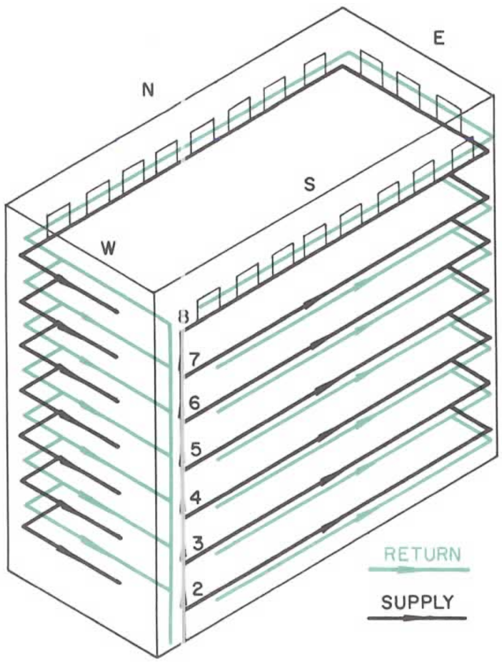


FIG. 26 — HORIZONTAL WATER PIPING LAYOUT

the south leg of the return piping and the west leg of the supply piping on each floor must be full size.

In any water piping system with automatically controlled units, the water requirements and pump head pressure varies. This is true whether or not diversity is applied. However the water requirements and pump head vary considerably more in a system in which diversity is not considered.

In a system in which diversity is not applied, greater emphasis is required for pump controls to prevent excessive noise being created by throttling valves or excessive water velocities. In addition, since the system never requires the full water quantity for which it is designed, the pump must be either throttled continuously, or bypassed, or reduced in size.

It is good practice, therefore, to take advantage of diversity to reduce the pipe size and pump capacity. *Chart 6* gives the diversity factors which are used in water piping design. *Example 1* illustrates the use of *Chart 6*.

Example 1 — Diversity Factors for Water Piping Headers

Given:

Water piping layout as illustrated in *Fig. 27*.

Find:

1. Diversity factor to be applied to the water quantity.
2. Water quantities in header sections.

Solution:

1. Pump A supplies north and west exposure but diversity can be applied to north exposure only. The total gpm in pump A circuit is 280 gpm and the accumulated gpm in the north exposure is 160 gpm. The ratio of accumulated gpm to the total water quantity in the circuit is:

$$\frac{160}{280} = .57$$

Enter *Chart 6* at the ratio .57 and read the diversity factor .785.

Pump B circuit has a ratio for the east exposure of:

$$\frac{120}{280} = .43$$

Entering *Chart 6* at the ratio of .43, the diversity factor is read as .725.

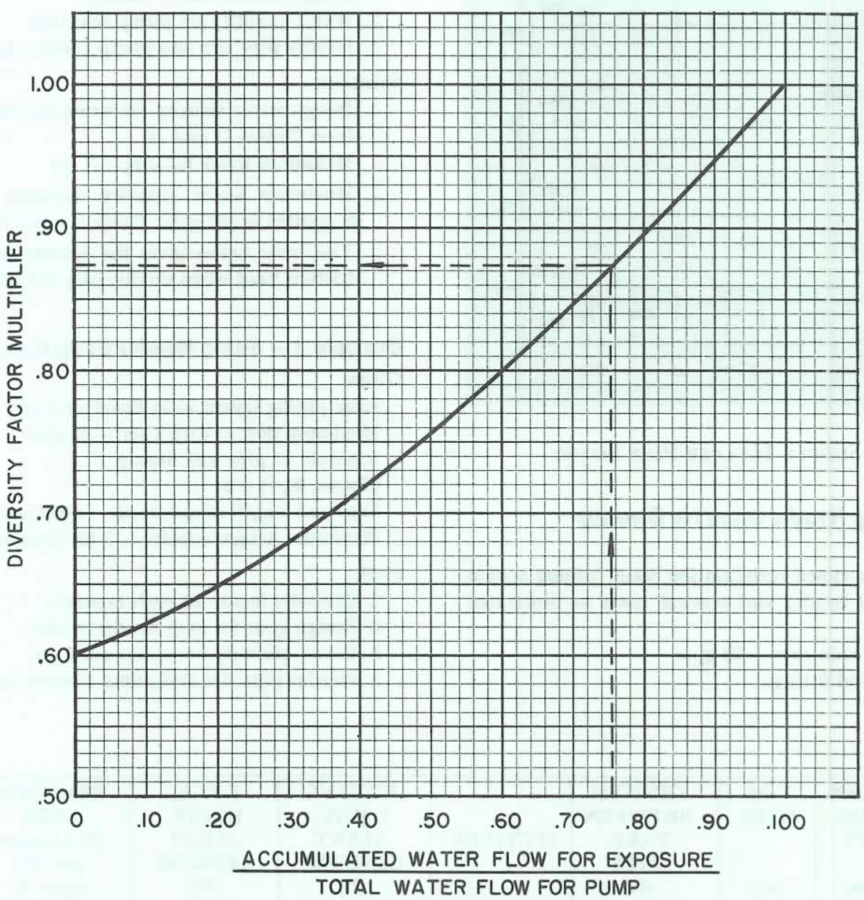
2. The following table illustrates how the diversity factors are applied to the maximum water quantities to obtain the design water quantities.

PUMP "A" CIRCUIT			
Section	Max Quantity (gpm)	Diversity Factor	Design Quantity (gpm)
A-R1	280	.785	220
R1-R2	260	.785	204
R2-R3	240	.785	188
R3-R4	220	.785	173
R4-R5	200	.785	157
R5-R6	180	.785	141
R6-R7	160	.785	126
R7-R8	140	.785	(110)120*
R8-R9	120	1.00	120
R9-R10	100	1.00	100
R10-R11	80	1.00	80
R11-R12	60	1.00	60
R12-R13	40	1.00	40
R13-R14	20	1.00	20

PUMP "B" CIRCUIT			
Section	Max Quantity (gpm)	Diversity Factor	Design Quantity (gpm)
B-R28	280	.725	203
R28-R27	260	.725	188
R27-R26	240	.725	174
R26-R25	220	.725	160
R25-R24	200	.725	(145) 160*
R24-R23	180	.725	(130) 160*
R23-R22	160	1.00	160
R22-R21	140	1.00	140
R21-R20	120	1.00	120
R20-R19	100	1.00	100
R19-R18	80	1.00	80
R18-R17	60	1.00	60
R17-R16	40	1.00	40
R16-R15	20	1.00	20

*When applying diversity, the design water quantity in the last section of the exposure is usually less than the water quantity in the first section on the adjoining exposure. When this occurs, the water quantity in the last section or last two sections is increased to equal the water quantity in the first section of the next exposure.

CHART 6—DIVERSITY FACTORS



In *Example 1* pump "A" is selected for 220 gpm and pump "B" is selected for 203 gpm. The pipe sizes in the north and east exposures are reduced using the design gpm, whereas the pipes in the south and west exposures are selected full size.

Example 2 and 3 illustrate the economics involved when applying diversity. *Example 2* shows a typical header layout with one pump serving all four exposures. The header is sized without diversity.

Example 3 is the same piping layout but diversity is used to size the header.

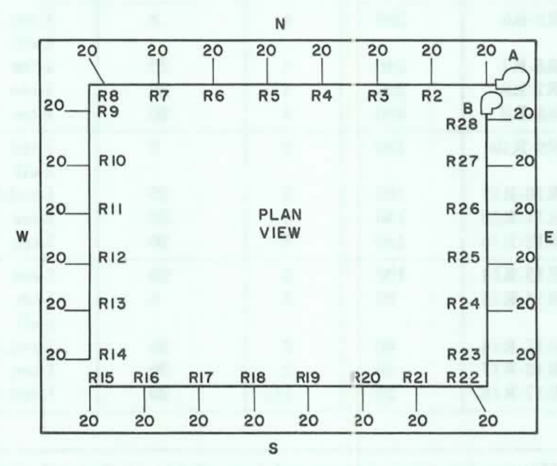


FIG. 27 — SUPPLY WATER HEADER

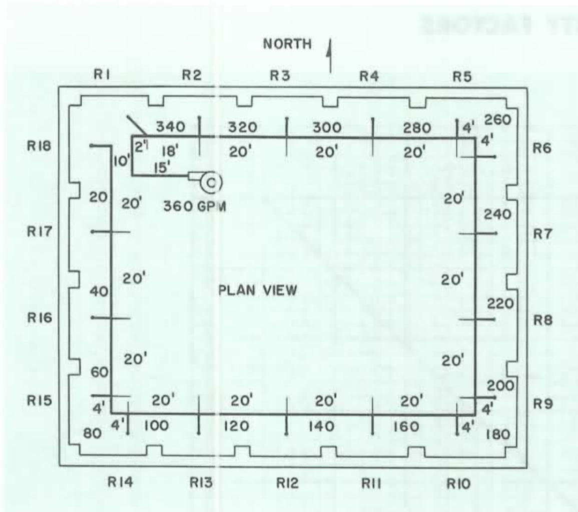


FIG. 28 — SUPPLY HEADER PIPE SIZING

Example 2 — Sizing Header Using No Diversity

Given:

A building with a closed recirculation water piping system using a horizontal header and vertical risers as illustrated in *Fig. 28*.

Maximum flow to each riser — 20 gpm

Schedule 40 pipe and fittings.

Elbows, R/D — 1

Expected length of operation — 6000 hours

Find:

1. Design header water velocity
2. Water quantity for pump selection
3. Header pipe size and pump friction head

Solution:

1. Design water velocity for sizing the headers is determined from *Tables 13 and 14*.
   Maximum water velocity — 7 fps
2. Maximum water quantity required when no diversity is applied is 360 gpm. Pump is selected for 360 gpm.
3. The table below gives the header pipe sizes and pump friction head when no diversity is applied.

Example 3 — Sizing Header Using Diversity

Given:

Same piping layout as in *Example 2* and *Fig. 28*.

Maximum flow to each riser — 20 gpm

Schedule 40 pipe and fittings

Elbows, R/D = 1

Expected length of operation — 6000 hours

Maximum design velocity — 7 fps *(Example 2)*

Find:

1. Diversity factor for each exposure
2. Design gpm for each header section
3. Water quantity for pump selection
4. Header pipe size and pump friction head

HEADER SECTION	WATER QUANTITY (gpm)	PIPE SIZE† (in.)	LENGTH BETWEEN TAKE-OFFS (ft)	FITTINGS	FITTING EQUIVALENT LENGTH* (ft)	TOTAL EQUIVALENT LENGTH (ft)	FRICTION LOSS† (ft of water per 100 equiv ft)	FRICTION HEAD (ft of water)
To R1	360	5	27	2-ells	26	53.0	2.3	1.22
R1-R2	340	5	18	1-tee	8.2	26.2	2.0	.53
R2-R3	320	5	20	1-tee	8.2	28.2	1.8	.51
R3-R4	300	5	20	1-tee	8.2	28.2	1.6	.45
R4-R5	280	4	20	1-red. tee	12.0	32.0	4.4	1.41
R5-R6	260	4	8	1-tee	6.7			
				1-ell	10.0	24.7	3.8	.94
R6-R7	240	4	20	1-tee	6.7	26.7	3.2	.85
R7-R8	220	4	20	1-tee	6.7	26.7	2.7	.72
R8-R9	200	4	20	1-tee	6.7	26.7	2.3	.62
R9-R10	180	4	8	1-tee	6.7			
				1-ell	6.0	20.7	2.1	.43
R10-R11	160	3	20	1-red. tee	9.0	29.0	5.5	1.59
R11-R12	140	3	20	1-tee	5.0	25.0	4.6	1.15
R12-R13	120	3	20	1-tee	5.0	25.0	3.2	.80
R13-R14	100	3	20	1-tee	5.0	25.0	2.5	.63
R14-R15	80	3	8	1-tee	5.0			
				1-ell	7.5	20.5	1.6	.33
R15-R16	60	2	20	1-red. tee	7.0	27.0	6.8	1.84
R16-R17	40	2	20	1-tee	3.3	23.3	3.2	.75
R17-R18	20	1¼	20	1-red. tee	5.0	25.0	6.5	1.62
							Pump friction head‡	16.39

*Fitting losses are determined from *Table 11*. For reducing tees enter *Table 11* at the larger diameter.

†Friction rate and pipe size are determined from *Chart 3* not exceeding the maximum design water velocity (7 fps).

‡Pump friction head does not include losses for valves, strainers, etc., which must be included in the actual design.

Solution:

1. *Chart 6* is used with the ratio of accumulated gpm in the exposure to the total pump gpm, in order to determine the diversity factors. The following table illustrates the method of determining diversity factors. (First exposure listed is always first exposure served by pump.)

EXPOSURE	EXPOSURE WATER QUANTITY (gpm)	ACCUM. GPM / TOTAL PUMP GPM	DIVERSITY FACTOR
North	100	100/360 = .28	.67
East	80	180/360 = .50	.76
South	100	280/360 = .78	.89
West	80	360/360 = 1.00	1.00

2. The diversity factor found in *Step 1* is applied to the maximum water quantity in each header section to establish the design gpm for sizing the header. The table at right gives the design water quantity for the various header sections.

3. The design water quantity required for pump selection when diversity is applied is 240 gpm.

4. The design water quantity found in *Step 2* is used in sizing the header pipe and in establishing the pump friction head. The table below illustrates the header pipe sizing:

HEADER SECTION	MAXIMUM WATER QUANTITY (gpm)	DIVERSITY FACTOR	DESIGN WATER QUANTITY (gpm)
To R1	360	.67	240
R1-R2	340	.67	227
R2-R3	320	.67	214
R3-R4	300	.67	201
R4-R5	280	.67	(187)197*
R5-R6	260	.76	197
R6-R7	240	.76	182
R7-R8	220	.76	167
R8-R9	200	.76	(152)160*
R9-R10	180	.89	160
R10-R11	160	.89	142
R11-R12	140	.89	124
R12-R13	120	.89	106
R13-R14	100	.89	90
R14-R15	80	1.00	80
R15-R16	60	1.00	60
R16-R17	40	1.00	40
R17-R18	20	1.00	20

*When applying diversity, the design water quantity in the last section of the exposure is usually less than the design water quantity in the first section of the adjoining exposure. When this occurs, the water quantity in the last section or last two sections is increased to equal the water quantity in the first section of the next exposure.

HEADER SECTION	DESIGN WATER QUANTITY (gpm)	PIPE SIZE† (in.)	LENGTH BETWEEN TAKE-OFFS (ft)	FITTINGS	FITTING EQUIVALENT LENGTH* (ft)	TOTAL EQUIVALENT LENGTH (ft)	FRICTION LOSS† (ft of water per 100 equiv ft)	FRICTION HEAD (ft of water)
To R1	240	4	27	2-ells	20.0	47.0	3.4	1.60
R1-R2	227	4	18	1-tee	6.7	24.7	3.0	.74
R2-R3	214	4	20	1-tee	6.7	26.7	2.7	.72
R3-R4	201	4	20	1-tee	6.7	26.7	2.3	.61
R4-R5	197	4	20	1-tee	6.7	26.7	2.3	.61
R5-R6	197	4	8	1-ell	10.0			
				1-tee	6.7	24.7	2.3	.57
R6-R7	182	4	20	1-tee	6.7	26.7	2.0	.53
R7-R8	167	4	20	1-tee	6.7	26.7	1.8	.48
R8-R9	160	3	20	1-red. tee	9.0	29.0	5.6	1.62
R9-R10	160	3	8	1-ell	7.5			
				1-tee	5.0	20.5	5.6	1.15
R10-R11	142	3	20	1-tee	5.0	25.0	4.5	1.12
R11-R12	124	3	20	1-tee	5.0	25.0	3.5	.87
R12-R13	106	3	20	1-tee	5.0	25.0	2.7	.68
R13-R14	90	3	20	1-tee	5.0	25.0	2.0	.50
R14-R15	80	3	8	1-ell	7.5			
				1-tee	5.0	20.5	1.6	.33
R15-R16	60	2	20	1-red. tee	7.0	27.0	6.8	1.84
R16-R17	40	2	20	1-tee	3.3	23.3	3.2	.75
R17-R18	20	1¼	20	1-red. tee	5.0	25.0	6.5	1.63
							Pump friction head‡	16.35

*Fitting losses are determined from *Table 11*. For reducing tee enter *Table 11* at the larger diameter.

†Friction rate and pipe size are determined from *Chart 3* with the water velocity not exceeding 7 fps.

‡Pump friction head does not include losses for valves, strainers, etc., which must be included in the actual design.

Examples 2 and 3 indicate that the following reductions in pipe and fitting size can be made when diversity is used:

1. 57 ft of 5 in. pipe replaced with 4 in. pipe.
2. 28 ft of 4 in. pipe replaced with 3 in. pipe.
3. 8 fittings reduced 1 size.

In addition the pump can be selected for 240 gpm instead of 360 gpm which is approximately a ⅓ reduction. Other areas where a reduction in size is possible are:

1. Pipe and fittings in the return piping header.
2. Valves, unions, couplings, strainers and other elements located in the supply and return headers.

2

PUMP SELECTION

Pumps are selected so that there is no sustained rise in pressure when the water flow is throttled. Systems having considerable throttling have the pump selected on the flat portion of the "head-versus-flow" curve.

Normally, new installed pipe has less than design friction and, therefore, the pump delivers greater gpm than design and requires more horsepower. For this reason a centrifugal pump is always selected for the calculated pump head without the addition of safety factors. If the pump is selected for the calculated head plus safety factors, the pump must handle a larger water quantity. When this occurs and provision is not made to throttle or bypass the excess water flow, the possibility of pump motor overload exists.

Again, if the pump is selected for maximum water quantity and diversity is not applied, the water flow must be throttled. This increases the pump head.

SYSTEM ACCESSORIES AND LAYOUT

This section discusses the function and selection of piping accessories and describes piping layout techniques for coils, condensers, coolers, air washers, cooling towers, pumps and expansion tanks.

ACCESSORIES

Expansion Tanks

An expansion tank is used to maintain system pressure by allowing the water to expand when the water temperature increases, and by providing a method of adding water to the system. It is normally required in a closed system but not in an open system; the reservoir in an open system acts as the expansion tank.

The open and closed expansion tanks are the two types used in water piping systems. Open expansion tanks are open to the atmosphere and are located on the suction side of the pump above the highest unit in the system. At this location the tank provides atmospheric pressure at or above the pump suction, thus preventing air leakage into the system. The static head on the pump due to the expansion tank must be greater than the friction drop of the water in the pipe from the expansion line connection to the pump suction. In *Fig. 29* the static head *AB* must be greater than the friction loss in line *AC*. Adding any accessories such as a strainer in line *AC* increases the friction drop in *AC* and results in raising the height of the expansion tank. To keep the height of the tank at a reasonable level, accessories should be placed at points 1 and 2 in *Fig. 29*. At these designations the friction loss in line AC is not affected.

The following procedure may be used to determine the capacity of an open expansion tank:

1. Calculate the volume of water in the piping, from *Tables 2 and 3, pages 2 and 3.*
2. Calculate the volume of water in the coils and heat exchangers.
3. Determine the percent increase in the volume of water due to operating at increased temperatures from *Table 15*.
4. Expansion tank capacity is equal to the percent increase from *Table 15* times the total volume of water in the system.

The closed expansion tank is used for small or residential hot water heating systems and for high temperature water systems. Closed expansion tanks are not open to the atmosphere and operate above

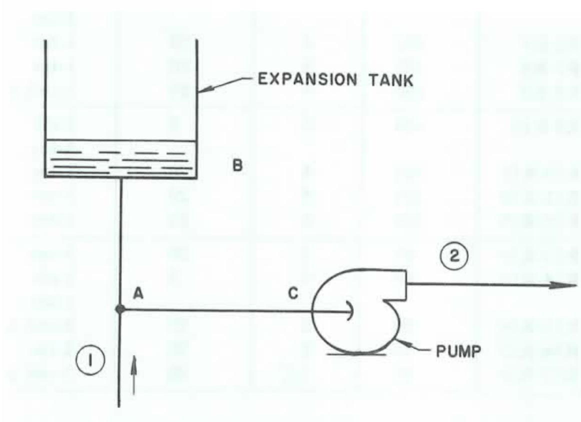


FIG. 29 – STRAINER LOCATION IN WATER PIPING SYSTEM

TABLE 15—EXPANSION OF WATER

(Above 40 F)

TEMP (F)	VOLUME INCREASE (%)	TEMP (F)	VOLUME INCREASE (%)
100	.6	275	6.8
125	1.2	300	8.3
150	1.8	325	9.8
175	2.8	350	11.5
200	3.5	375	13.0
225	4.5	400	15.0
250	5.6		

atmospheric pressure. Air vents must be installed in the system to vent the air. Closed expansion tanks are located on the pump suction side of the system to permit the pump suction to operate at or near constant pressure. Locating the expansion tank at the pump discharge is usually not satisfactory. All pressure changes caused by pump operation are subtracted from the original static pressure. If the pressure drop below the original static is great enough, the system pressure may drop to the boiling point, causing unstable water circulation and possible pump cavitation. If the system pressure drops blow atmospheric, air sucked in at the air vents can collect in pockets and stop water circulation.

The capacity of a closed expansion tank is larger than an open expansion tank operating under the same conditions. ASME has standardized the calculation of the capacity of closed expansion tanks. The capacity depends on whether the system is operating above or below 160 F water temperature.

Water temperatures below 160 F use the following formula to determine the tank capacity:

$$V_t = \frac{E \times V_s}{\dfrac{P_a}{P_f} - \dfrac{P_a}{P_o}}$$

where: V_t = minimum capacity of the tank (gallons).

E = percent increase in the volume of water in the system *(Table 15).*

V_s = total volume of water in the system (gallons).

P_a = pressure in the expansion tank when water first enters, usually atmospheric pressure (feet of water absolute).

P_f = initial fill or minimum pressure at the expansion tank (feet of water absolute).

P_o = maximum operating pressure at the expansion tank (feet of water absolute).

When the system water temperature is between 160 and 280 F, the following equation is used to determine the expansion tank capacity:

$$V_t = \frac{(0.00041\ t - 0.0466)\ V_s}{\dfrac{P_a}{P_f} - \dfrac{P_a}{P_o}}$$

where t = maximum average operating temp (F).

Strainers

The primary function of a strainer is to protect the equipment. Normally strainers are placed in the line at the inlet to pumps, control valves or other types of equipment that should be protected against damage. The strainer is selected for the design capacity of the system at the point where it is to be inserted in the line. Strainers for pump protection should be not less than 40 mesh and be made of bronze. On equipment other than pumps the manufacturer should be contacted to determine the degree of strainer protection necessary. For example, a control valve needs greater protection than a pump and, therefore, requires a finer mesh strainer.

Thermometers and Gages

Thermometers and gages are required in the system wherever the design engineer considers it important to know the water temperature or pressure. The following temperatures and pressures are usually considered essential:

1. Water temperature entering and leaving the cooler and condenser.
2. Pump suction and discharge pressure.
3. Spray water temperature and pressure entering the air washer.

Water thermometers are usually selected for an approximate range of 30 F to 200 F; they should be equipped with separable wells and located where they can be easily read.

Pressure gages are selected so that the normal reading of the gage is near the midpoint of the pressure scale.

Air Vents

Air venting is an important aspect in the design of any water system. The major portion of the air is normally vented thru the open expansion tank.

Air vents should be installed in the high points of any water system which cannot vent back to the open expansion tank. Systems using a closed expansion tank require vents at all high points. Runoff drains should be provided at each vent to carry possible water leakage to a suitable drain line.

PIPING LAYOUT

Each installation has its own problems regarding location of equipment, interference with structural members, water and drain locations, and provision

2

for service and replacement. The following guides are presented to familiarize the engineer with accepted piping practice:

1. Shut-off valves are installed in the entering and leaving piping to equipment. These are normally gate valves. This arrangement permits servicing or replacing the equipment without draining the entire system. Occasionally a globe valve is installed in the system to serve as one of the shut-off valves and in addition is used to balance water flow. Most often it is located at the pump discharge. In a close coupled system the shut-off valves may be omitted if the time and expense required to drain the system is not excessive. This is a matter of economics, the first cost of the valves versus the cost of new water treatment and time spent in draining the system.
2. Systems using screwed, welded or soldered joints require unions to permit removal of the equipment for servicing or replacement. If gate valves are used to isolate the equipment in the system, unions are placed between the equipment and each gate valve. Unions are also located before and after control valves, and in the branch of a three-way control valve. It is good practice to locate the control valve between the equipment and the gate valve used to shut off flow to the equipment. This permits removal of the control valve from the system without draining the system. By locating the control valve properly, it is possible to eliminate the unions required for removal of the equipment. If the system uses flanged valves and fittings, the need for unions is eliminated.
3. Strainers, thermometers and gages are normally located between the equipment and the gate valves used to shut off the water flow to the equipment.

The following piping diagrams are illustrated with screwed connections. However, flanged, welded or soldered connections may be used. These layouts have been simplified to show various principles involved in piping practice.

Water Coils

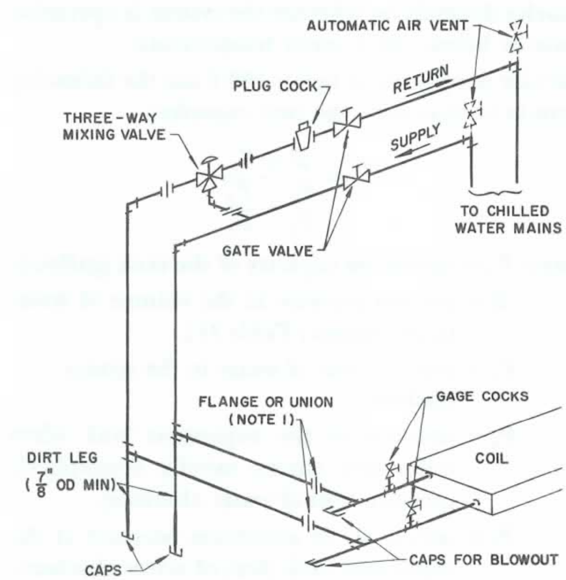
Figures 30 thru 36 illustrate typical piping layouts for chilled water coils in a closed piping system.

The coil layout illustrated in *Fig. 30* contains a three-way mixing valve. This valve, located at the cooling coil outlet, maintains a desired temperature by proportioning *automatically* the amount of water flowing thru the coil or thru the bypass. It is regulated by a temperature controller. Gage cocks are usually installed in both the supply and return lines to the coil. This permits pressure gages to be connected to determine pressure drop thru the coil. The plug cock is manually adjusted to set the pressure drop thru the coil.

Figure 31 illustrates an alternate method of piping a water coil. The plug cock shown is used to adjust manually the water flow for a set pressure drop thru the coil. The pressure drop is determined by connecting pressure gages to the gage cocks. In this piping layout control of the leaving air temperature from the coil is maintained within a required range since normally the entering water is controlled to a set temperature. Often an air bypass around the coil is used to maintain final air temperature.

Figure 32 illustrates a multiple coil arrangement. Piping connections for drain and vent lines for the coil are included and should be ½ in. nominal pipe size. The same principles covered in *Figs. 30 and 31* are applicable to multiple coil arrangements.

A globe valve may be substituted for the plug cock and gate valve combination in the return lines in *Figs. 30, 31 and 32.* In this arrangement the globe valve is used to balance the pressure drop thru the coil, and also to shut off the water when servicing is required. However, it has disadvantages that



NOTE: Flange or union is located so coil may be removed.

FIG. 30— CHILLED WATER PIPING FOR COILS (AUTOMATIC CONTROL)

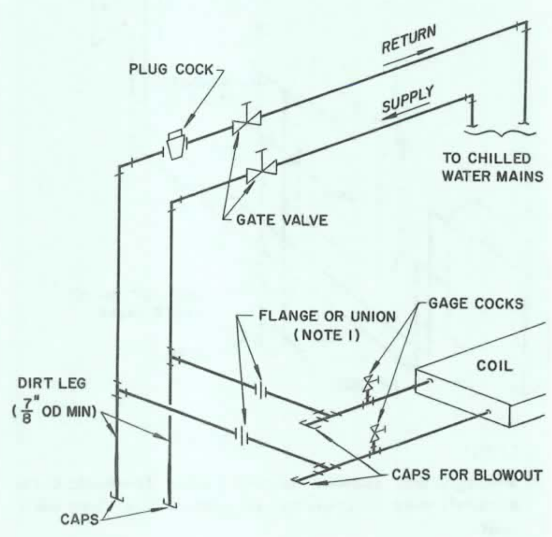
should be considered. First, the valve setting is not fixed and, therefore, can be changed accidentally; secondly, the valve must be reset whenever it is used to shut off the flow.

Capped tees are provided on coils as a means of blowing out the water when the system is drained for freeze protection.

The use of a gate valve with a hose bib should be considered in the dirt leg when floor drains are remote in relation to the unit. The bib makes it possible to connect a hose to the dirt leg for draining the coil. The dirt leg has a 7/8 in. nominal minimum diameter, should be approximately 18 in. long and be located at an accessible servicing point. A gate valve is preferred in the dirt leg because sediment passes thru it more freely than thru a globe valve.

Figure 33 illustrates multiple coil units piped with vertical risers. This is a common approach to air conditioning multi-room, multi-story buildings. The units are connected to common supply and return risers which pass thru the floors of the building. A dirt leg is required at the bottom of each riser as shown in *Fig. 33*.

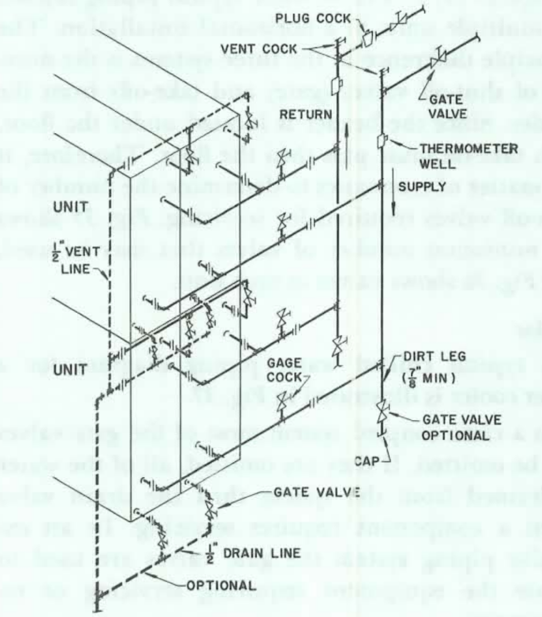
Gate valves are recommended as illustrated to permit servicing without disturbing the remainder of the system. On very small systems these valves may be omitted.



NOTES:

1. Flange or union is located so coil may be removed.
2. Plug cock is used for adjusting flow thru coil.

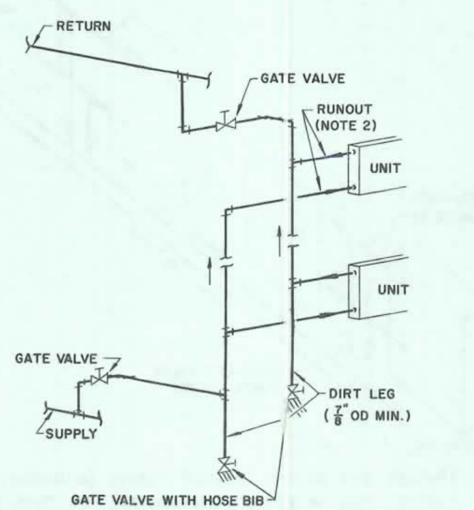
FIG. 31 – CHILLED WATER PIPING FOR COILS (MANUAL CONTROL)



NOTES:

1. Plug cock is used to adjust pressure drop thru coil.
2. All valves shown are gate valves.

FIG. 32 – CHILLED WATER PIPING FOR MULTIPLE COILS



NOTES:

1. Headers are pitched upward in the direction of water flow so that air can be vented thru the expansion tank.
2. Supply and return runouts to the coil should have flared connections if runouts are soft copper. Otherwise unions or flanges are installed to facilitate servicing units.

FIG. 33 – PIPING LAYOUT FOR VERTICAL MULTIPLE COILS

Figures 34, 35 and 36 show typical piping layouts for multiple units in a horizontal installation. The principle difference in the three systems is the number of shut-off valves (gate) and take-offs from the header. Since the header is located under the floor, each take-off must pass thru the floor. Therefore, it is a matter of economics to determine the number of shut-off valves required for servicing. *Fig. 35* shows the minimum number of valves that may be used, and *Fig. 36* shows valves at each unit.

Cooler

A typical chilled water piping diagram for a water cooler is illustrated in *Fig. 37*.

In a close coupled system most of the gate valves can be omitted. If they are omitted, all of the water is drained from the system thru the drain valve when a component requires servicing. In an extensive piping system the gate valves are used to isolate the equipment requiring servicing or replacement.

Figure 37 illustrates the recommended water piping and accessories associated with a cooler.

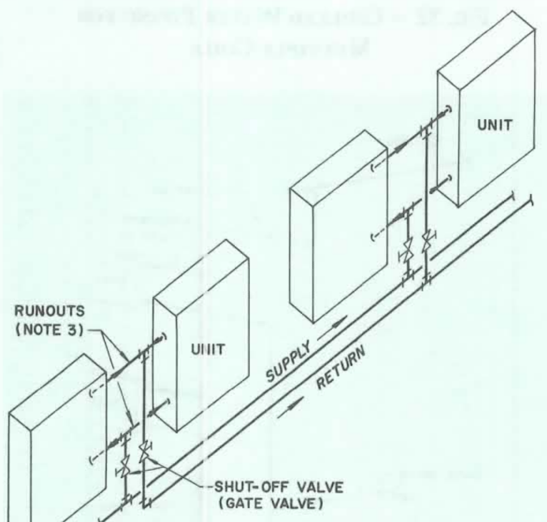
Condenser

Figure 38 shows a water-cooled condenser using city, well or river water. The return is run higher than the condenser so that the condenser is always full of water. Water flow thru the condenser is modulated by the control valve in the supply line.

Figure 39 is an illustration of an alternate drain arrangement for a condenser discharging waste water. Drain connections of all types must be checked for compliance with local codes. Codes usually require that a check valve be installed in the supply line when city water is used.

Figure 40 illustrates a condenser piped up with a cooling tower. If the cooling tower and condenser are close coupled, most of the gate valves can be eliminated. If the piping system is extensive, the gate valves as shown are recommended for isolating the equipment when servicing.

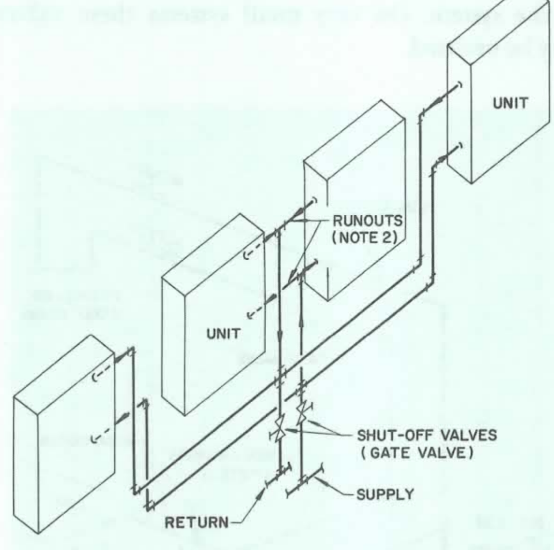
When more than one condenser is to be used in the same circuit, the flow thru the condensers must be equalized as closely as possible. This is complicated by the following:



NOTES:

1. Though not shown, control valves (automatic or manual) may be required to control flow thru each unit.
2. A shut-off valve may be installed in the supply and return branch headers when headers serve 3 to 5 units.
3. Supply and return runouts to the coil should have flared connections if the runouts are soft copper. Otherwise unions or flanges are installed to facilitate servicing units.

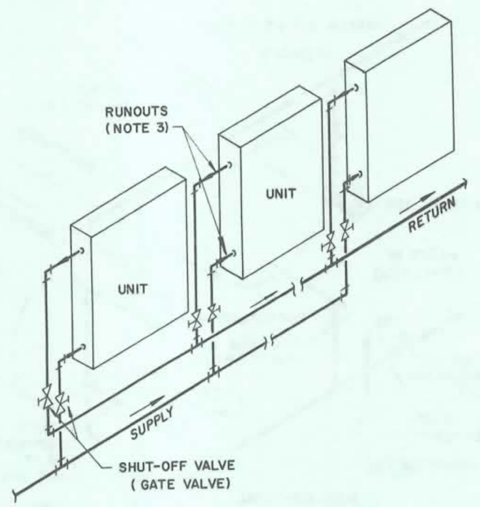
Fig. 34 — Piping Layout for Horizontal Multiple Coils (4 Units — 4 Shut-Off Valves)



NOTES:

1. Though not shown, control valves (automatic or manual) may be required to control flow thru each unit.
2. Supply and return runouts to the coil should have flared connections if the runouts are soft copper. Otherwise unions or flanges are installed to facilitate servicing units.

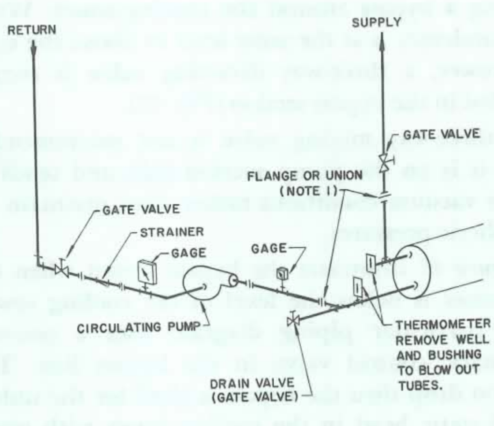
Fig. 35 — Piping Layout for Horizontal Multiple Coils (4 Units — 2 Shut-Off Valves)



NOTES:

1. Though not shown, control valves (automatic or manual) may be required to control flow thru each unit.
2. A shut-off valve may be installed in the supply and return branch headers when headers serve 3 to 5 units.
3. Supply and return runouts to the coil should have flared connections if the runouts are soft copper. Otherwise unions or flanges are installed to facilitate servicing units.

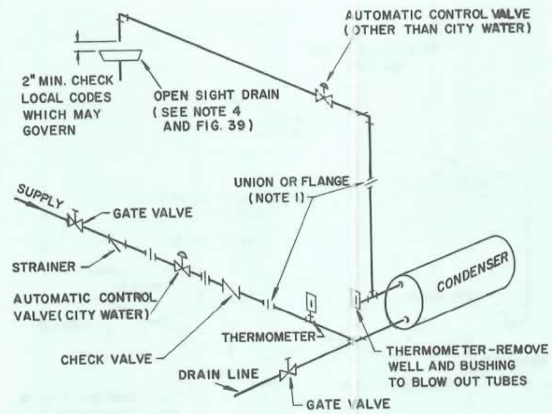
FIG. 36 — PIPING LAYOUT FOR HORIZONTAL MULTIPLE COILS (3 UNITS — 6 SHUT-OFF VALVES)



NOTES:

1. Flange or union is located to allow cooler head removal.
2. Gate valves shown may be eliminated in a close coupled system.

FIG. 37 — PIPING AT A WATER COOLER



NOTES:

1. Flange or union is located to allow condenser head removal.
2. With outlet at top, condenser will be flooded even though automatic control valve is in modulating position.
3. Check valve is required by most sanitary codes (city water).
4. Required for city water only.

FIG. 38 — CONDENSER PIPING FOR A ONCE-THRU SYSTEM

1. The pressure drops thru the condensers are not always equal.
2. Water entering the branch line and leaving the run of tees seldom divides equally.
3. Workmanship in the installation can affect the pressure drop.

To equalize the water flow thru each condenser, the pipe should be sized as follows:

1. Size the branches for a water flow of 6 fps minimum. The branch connections to each condenser should be identical.

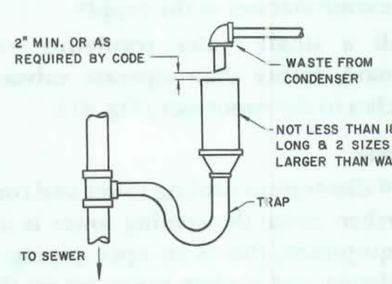
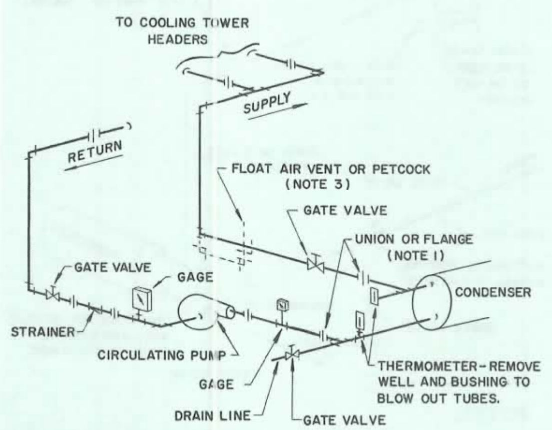


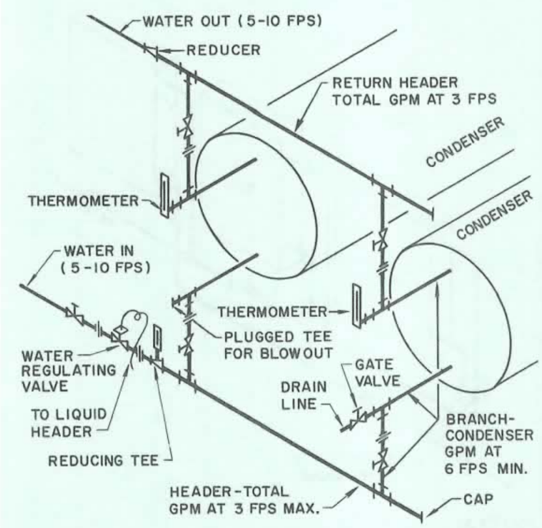
FIG. 39 — ALTERNATE DRAIN CONNECTION



NOTES:

1. Flange or union is located to allow condenser head removal.
2. Gate valves shown may be eliminated in a close coupled system (except drain valve).
3. When water enters bottom of condenser, air will vent naturally thru cooling tower sprays. If it is necessary to drop piping after leaving condenser, install air vent at high point of line before drop. See dotted line in figure.

FIG. 40 — CONDENSER PIPING FOR A COOLING TOWER



NOTES:

1. Thermometer wells are inserted in tees. Remove wells to blow out coils.
2. A single water regulating valve must be used as shown. If under capacity, install two valves in parallel and connect pressure tube in liquid header.
3. Water supply in or return out can be at any point in the headers.

FIG. 41 — MULTIPLE CONDENSER COOLING WATER PIPING (REFRIGERANT CONNECTIONS PARALLEL)

2. Size the header for the total required water quantity for all the condensers with a velocity of not more than 3 fps. The header is extended approximately 12 in. beyond the last branch to the condenser.
3. Size the water main supplying the header for a velocity of 5 to 10 fps with 7 fps a good average. The water main may enter the header at the end or at any point along the length of the header. Care should be used so that crosses do not result.
4. Size the return branches, header and main in the same manner as the supply.
5. Install a single water regulating valve in the main, rather than separate valves in the branches to the condenser *(Fig. 41)*.

Cooling Tower

Figure 40 illustrates a cooling tower and condenser piped together. Since the cooling tower is an open piece of equipment, this is an open piping system. If the condenser and cooling tower are on the same level, a small suction head for the pump exists. The strainer should be installed on the discharge side of the pump to keep the suction side of the pump as close to atmospheric as possible.

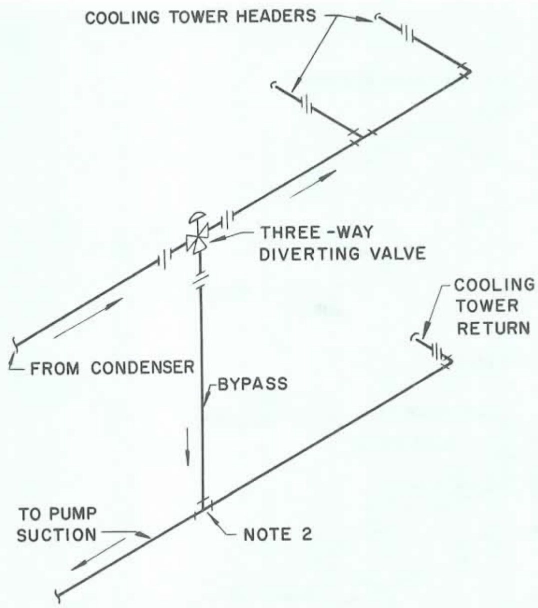
It is often desirable to maintain a constant water temperature to the condenser. This is done by installing a bypass around the cooling tower. When the condenser is at the same level or above the cooling tower, a three-way diverting valve is recommended in the bypass section *(Fig. 42)*.

A three-way mixing valve is not recommended since it is on the pump suction side, and tends to create vacuum conditions rather than maintain atmospheric pressures.

Figure 43 illustrates the bypass layout when the condenser is below the level of the cooling tower. This particular piping diagram uses a two-way automatic control valve in the bypass line. The friction drop thru the bypass is sized for the unbalanced static head in the cooling tower with maximum water flow thru the bypass.

If multiple cooling towers are to be connected, it is recommended that piping be designed such that the loss from the tower to the pump suction is approximately equal for each tower. *Fig. 44* illustrates typical layouts for multiple cooling towers.

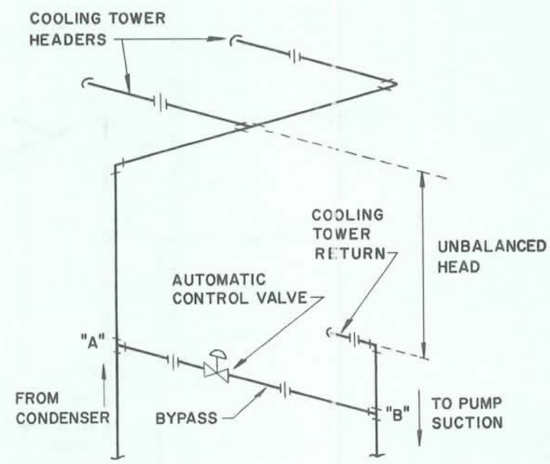
Equalizing lines are used to maintain the same water level in each tower.



NOTES:

1. A three-way diverting valve is used when the condenser is at the same level as or above the cooling tower. See *Fig. 43* for piping layout when condenser is below the cooling tower.
2. A three-way mixing valve is not recommended at this point as it imposes additional head at the pump suction.

Fig. 42 — Cooling Tower Piping for Constant Leaving Water Temperature (Condenser and Tower at Same Level)



NOTES:

1. **A two-way automatic control valve is used when the condenser is below the cooling tower. See *Fig. 42* for piping layout when the condenser is at the same level as or above the cooling tower.**
2. **The friction loss from "A" to "B" includes the loss thru that section of the pipe and the loss thru the two-way automatic control valve. This friction loss should be designed for the unbalanced head of the cooling tower.**
3. **Locate the automatic control valve close to the cooling tower to minimize pump motor overload and waterflow thru the tower when valve is in full open position.**

Fig. 43 — Cooling Tower Piping for Constant Leaving Water Temperature (Condenser Below Tower)

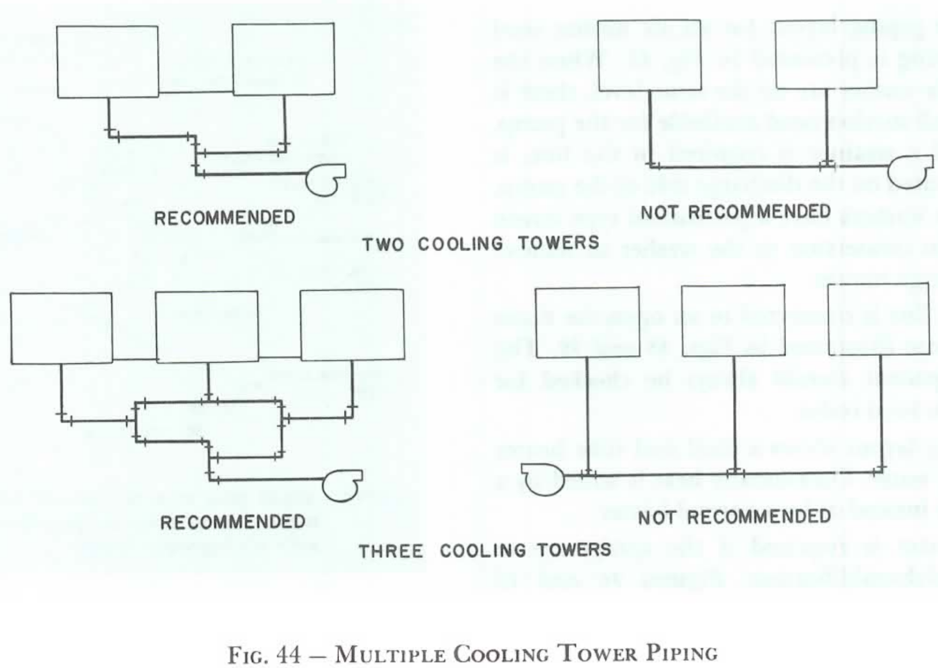
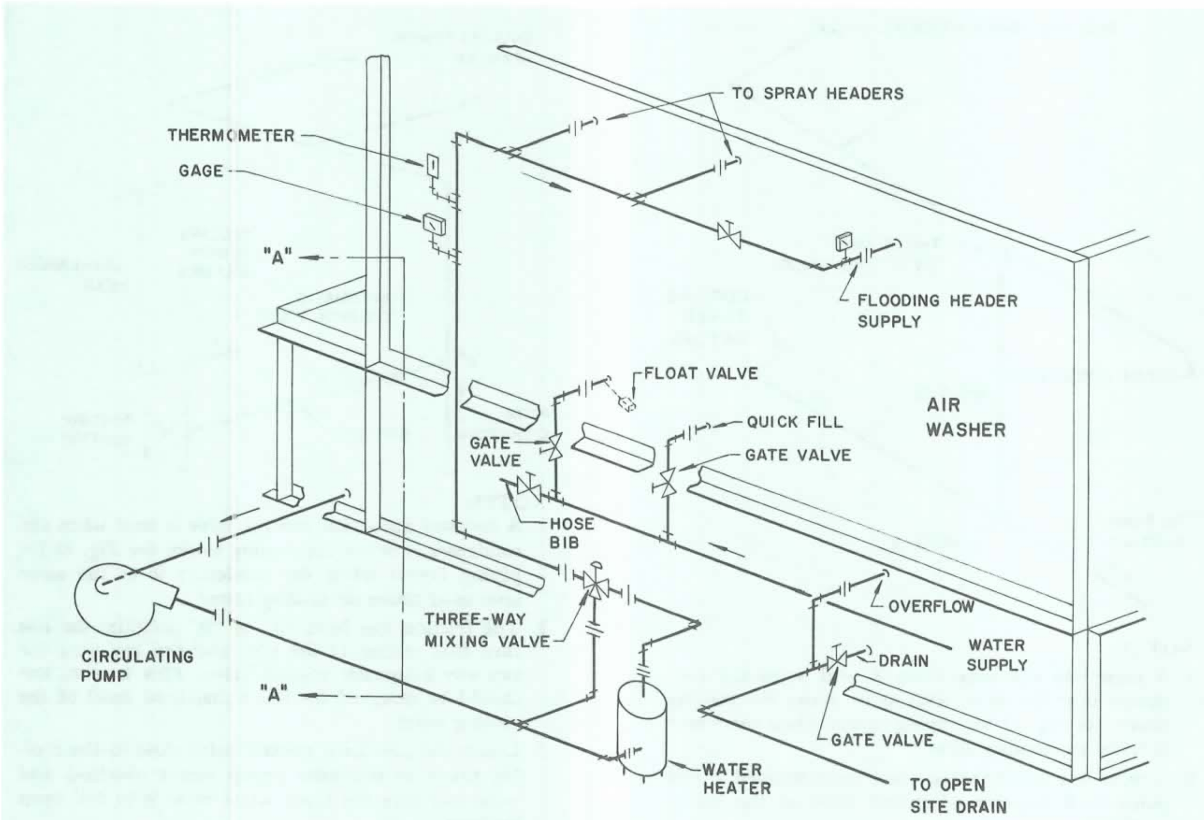


Fig. 44 — Multiple Cooling Tower Piping



NOTE: See *Figs. 46 and 47* for typical piping when an air washer is used for the dehumidifying system (section "A - A").

FIG. 45 — AIR WASHER PIPING

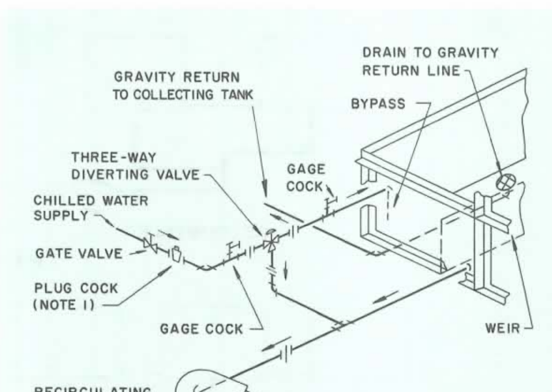
Air Washer

The water piping layout for an air washer used for humidifying is presented in *Fig. 45*. When the pump and air washer are on the same level, there is usually a small suction head available for the pump. Therefore, if a strainer is required in the line, it should be located on the discharge side of the pump. Normally air washers have a permanent type screen at the suction connection to the washer to remove large size foreign matter.

The drain line is connected to an open-site drain similar to those illustrated in *Figs. 38 and 39*. The drain arrangement should always be checked for compliance to local codes.

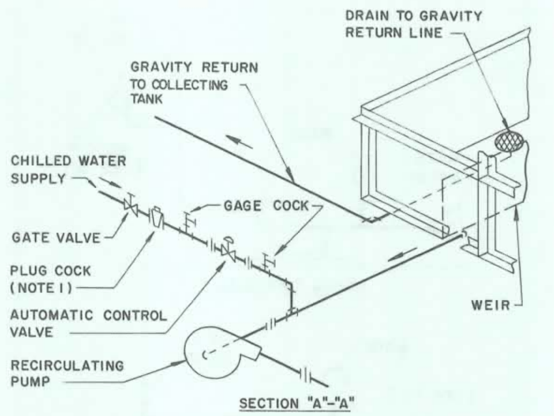
The piping layout shows a shell and tube heater for the spray water. Occasionally heat is added by a steam ejector instead of by a normal heater.

Chilled water is required if the sprays are to accomplish dehumidification. *Figures 46 and 47* illustrate two typical methods of connecting the chilled water supply. The plug cock in both dia-



NOTE: Adjust plug cock so that full flow thru automatic control valve is approximately 90% of recirculating water design.

FIG. 46 — AIR WASHER PIPING USING A THREE-WAY CONTROL VALVE



NOTE: Adjust plug cock so that full flow thru diverting valve is approximately 90% of recirculating water design.

Fig. 47 — Air Washer Piping Using a Two-Way Control Valve

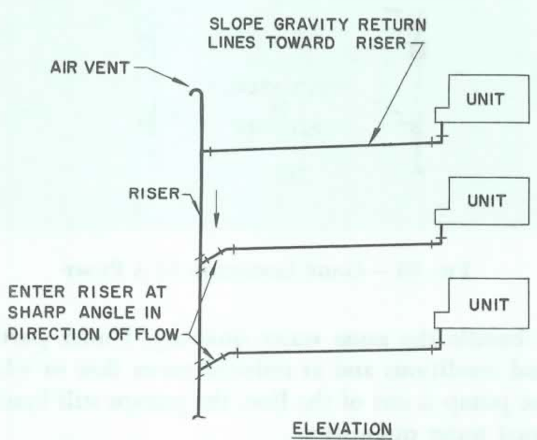


Fig. 48 — Air Washer Return Connections at Different Elevations

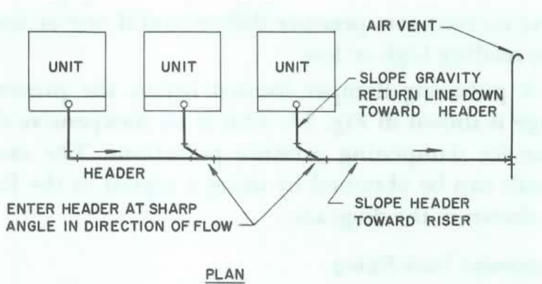


Fig. 49 — Air Washer Return Connections at the Same Level

grams is adjusted so that full flow thru the three-way diverting valve *(Fig. 46)* and thru the automatic control valve *(Fig. 47)* is approximately 90% of the recirculating water design quantity.

Figures 48 and 49 are schematic sketches of multiple air washers with gravity returns piped to the same header.

Sprayed Coil

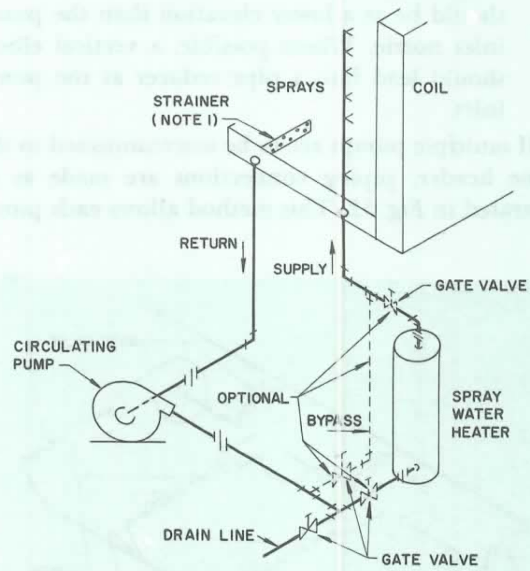
A typical layout for a sprayed coil piping system is shown in *Fig. 50*. The diagram shows a water heater which may be required for humidification. If a preheat coil is used, the water heater may be eliminated.

The drain line should be fitted with a gate valve rather than a globe valve since it is less likely to become clogged with sediment.

Pump Piping

The following items illustrated in *Fig. 51* should be kept in mind when designing piping for a pump:

1. Keep the suction pipe short and direct.
2. Increase the suction pipe size to at least one size larger than the pump inlet connection.
3. Keep the suction pipe free from air pockets.
4. Use an eccentric type reducer at the pump suction nozzle to prevent air pockets in the suction line.



NOTE: If no strainer is installed in this location, then a strainer is recommended on the pump discharge.

Fig. 50 — Spray Water Coil With Water Heater

2

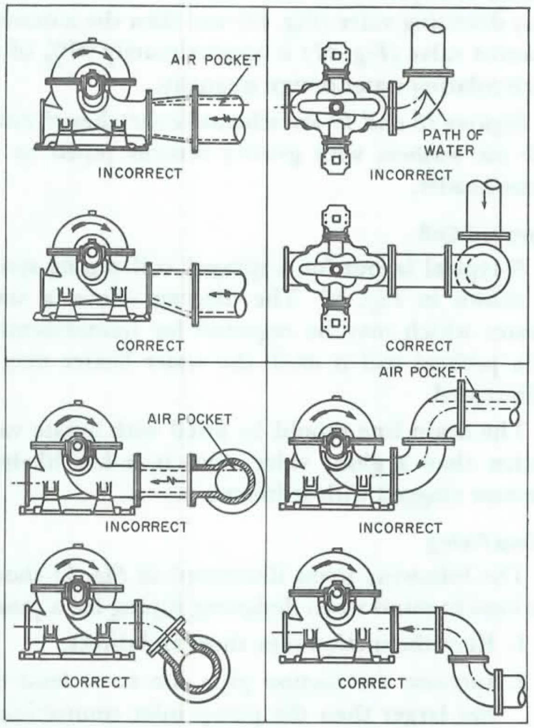


FIG. 51 — PUMP SUCTION CONNECTIONS

5. Never install a horizontal elbow at the pump inlet. Any horizontal elbow in the suction line should be at a lower elevation than the pump inlet nozzle. Where possible, a vertical elbow should lead into a pipe reducer at the pump inlet.

If multiple pumps are to be interconnected to the same header, piping connections are made as illustrated in *Fig. 52.* This method allows each pump

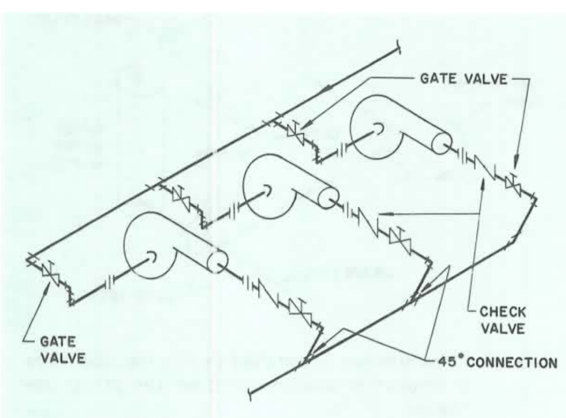


FIG. 52 — MULTIPLE PUMP PIPING

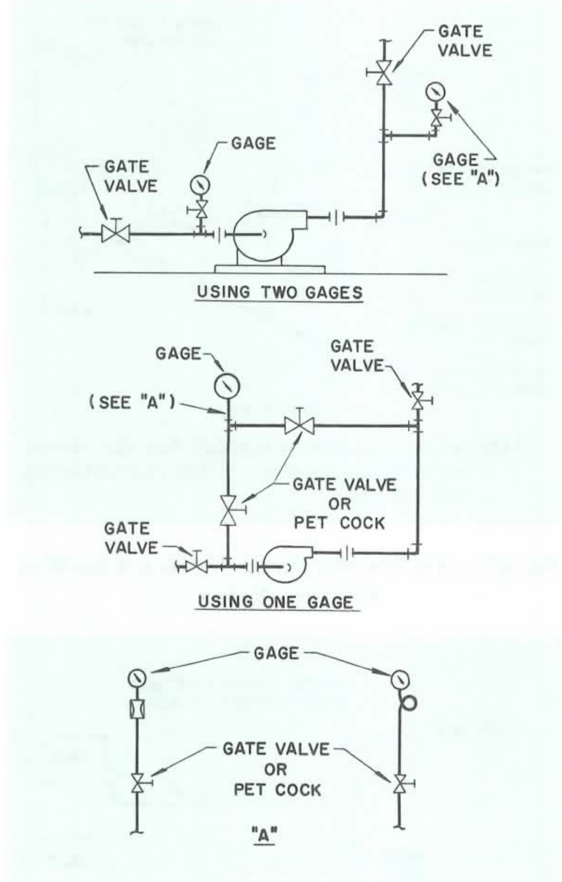


FIG. 53 — GAGE LOCATION AT A PUMP

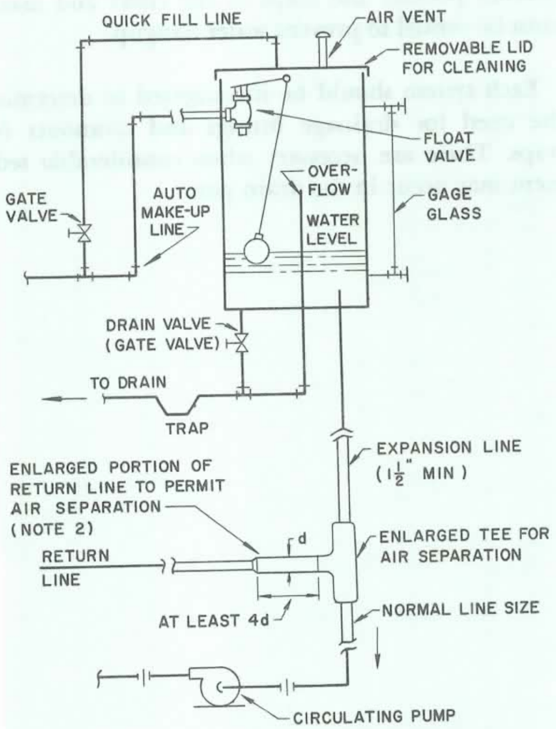
to handle the same water quantity. Under partial load conditions and at reduced water flow or when one pump is out of the line, the pumps still handle equal water quantities.

Figure 53 illustrates two methods of locating pressure gages at the pump; one method uses two gages and the other uses one. The use of one gage has the advantage of always giving the correct pressure differential across the pump. Two gages may give an incorrect pressure differential if one or both are reading high or low.

A pulsating damper located before the pressure gage is shown in *Fig. 53.* This is an inexpensive device for dampening pressure pulsations. The same result can be obtained by using a pigtail in the line as shown in the diagram.

Expansion Tank Piping

Figure 54 is a suggested piping layout for an open expansion tank. Piping is enlarged at the connection to the expansion tank. This permits air entrained or carried along with the water to separate



NOTES:

1. Do not put any valve strainer or trap in the expansion line.
2. Enlarged portion of return line and enlarged tee are each two standard pipe sizes larger than return line.

FIG. 54 – OPEN EXPANSION TANK PIPING

and be vented thru the tank. The expansion tank should be located at the pump suction side at the highest point in the system.

Valves, strainers and traps must be omitted from the expansion line since these may be accidentally turned off or become plugged.

Figure 55 illustrates the piping diagram for a closed tank.

Drain Line Piping

Moisture that forms on the cooling coils must be collected and carried off as waste. On factory fabricated fan-coil units a drain pan is used to collect this moisture. For built-up systems the floor or base of the system (before and after the cooling coil) is used to gather the moisture.

Since, under operating conditions, the drain water is subject to pressure conditions slightly above or slightly below atmospheric pressure, the line used to carry off this water must be trapped. This trap prevents conditioned air from entering the drain line when the drain water is under positive pressure as in a blow-thru fan-coil unit. When the system is under negative pressure as in a draw-thru unit, the trap prevents water from hanging up in the drain pan.

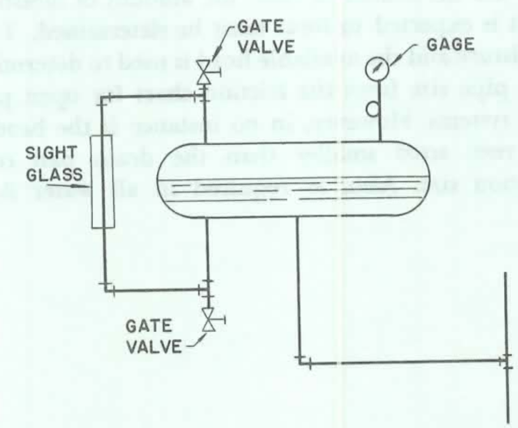


FIG. 55 – CLOSED EXPANSION TANK PIPING

Figure 56 illustrates the trapping of a drain line from the drain pan. The length of the water seal or trap depends on the magnitude of the positive or negative pressure on the drain water. For instance, a 2-inch negative fan pressure requires a 2-inch water seal.

Normally, under-the-window fan-coil units have the drip pan subject to atmospheric conditions only and the drain line from these units is not trapped.

The drain line runout for all systems is pitched to offset the line friction. For a single unit the runout is piped to an open site drain. Local codes and regulations must be checked to determine proper piping practice for an open site drain. The runout is run full size corresponding to the drain pan connection size.

Some applications have multiple units with the drain lines connected to a common header or riser.

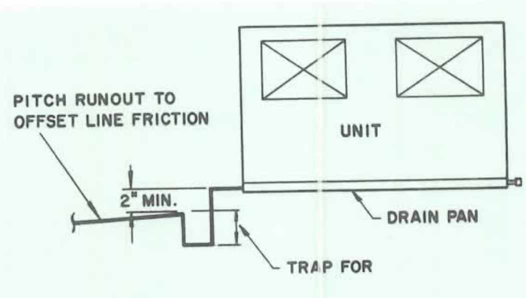


FIG. 56 – PIPING FOR DRAIN PANS

To size the header or riser, the amount of moisture that is expected to form must be determined. This moisture and the available head is used to determine the pipe size from the friction chart for open piping systems. However, in no instance is the header or riser sized smaller than the drain pan connection size. Also, as required in all water flow systems, pockets and traps in the risers and mains must be vented to prevent water hangup.

Each system should be investigated to determine the need for drainage fittings and cleanouts for traps. These are necessary when considerable sediment may occur in the drain pan.

CHAPTER 3. REFRIGERANT PIPING

GENERAL SYSTEM DESIGN

This chapter includes that practical material required for the design and layout of a refrigerant piping system at air conditioning temperature levels, using either Refrigerant 12, 22, 500 or 502.

APPLICATION CONSIDERATIONS

A refrigerant piping system requires the same general design considerations as any fluid flow system. However, there are additional factors that critically influence system design:

1. The system must be designed for minimum pressure drop since pressure losses decrease the thermal capacity and increase the power requirement in a refrigeration system.
2. The fluid being piped changes in state as it circulates.
3. Since lubricating oil is miscible with Refrigerants 12, 22, 500 and 502, some provision must be made to:
   a. Minimize the accumulation of liquid refrigerant in the compressor crankcase.
   b. Return oil to the compressor at the same rate at which it leaves.

Piping practices which accomplish these objectives are discussed in the following pages.

CODE REGULATIONS

System design should conform to all codes, laws and regulations applying at the site of an installation.

In addition the Safety Code for Mechanical Refrigeration (USAS-B9.1) and the Code for Refrigeration Piping (USAS-B31.5) are primarily drawn up as guides to safe practice and should also be adhered to. These two codes, as they apply to refrigeration, are almost identical, and are the basis of most municipal and state codes.

REFRIGERANT PIPING DESIGN

DESIGN PRINCIPLES

Objectives

Refrigerant piping systems must be designed to accomplish the following:

1. Insure proper feed to evaporators.
2. Provide practical line sizes without excessive pressure drop.
3. Protect compressors by –
   a. Preventing excessive lubricating oil from being trapped in the system.
   b. Minimizing the loss of lubricating oil from the compressor at all times.
   c. Preventing liquid refrigerant from entering the compressor during operation and shutdown.

Friction Loss and Oil Return

In sizing refrigerant lines it is necessary to consider the optimum size with respect to economics, friction loss and oil return. From a cost standpoint it is desirable to select the line size as small as possible. Care must be taken, however, to select a line size that does not cause excessive suction and discharge line pressure drop since this may result in loss of compressor capacity and excessive hp/ton. Too small a line size may also cause excessive liquid line pressure drop. This can result in flashing of liquid refrigerant which causes faulty expansion valve operation.

The effect of excessive suction and hot gas line pressure drop on compressor capacity and horsepower is illustrated in *Table 16*.

TABLE 16—COMPRESSOR CAPACITY VS LINE PRESSURE DROP

42 F Evaporator Temperature

SUCTION AND HOT GAS LINE PRESSURE DROP	COMPRESSOR	
	Capacity (%)	Hp/Ton (%)
No Line Loss	100	100
2F Suction Line Loss	95.7	103.5
2F Hot Gas Line Loss	98.4	103.5
4F Suction Line Loss	92.2	106.8
4F Hot Gas Line Loss	96.8	106.8

Pressure drop is kept to a minimum by optimum sizing of the lines with respect to economics, making sure that refrigerant line velocities are sufficient to entrain and carry oil along at all loading conditions.

For Refrigerants 12, 22, 500 and 502, consider the requirements for oil return up vertical risers.

Pressure drop in *liquid lines* is not as critical as in suction and discharge lines. However, the pressure drop should not be so excessive as to cause gas formation in the liquid line or insufficient liquid pressure at the liquid feed device. A system should normally be designed so that the pressure drop in the liquid line is not greater than one to two degrees change in saturation temperature. In terms of pressure drop, this corresponds to about 1.8 to 3.8 psi for R-12, 2.9 to 6 psi for R-22, 2.2 to 4.6 psi for R-500 and 3.1 to 6.3 psi for R-502.

Friction pressure drop in the liquid line includes accessories such as solenoid valve, strainer, drier and hand valves, as well as the actual pipe and fittings from the receiver outlet to the refrigerant feed device at the evaporator.

Pressure drop in the *suction line* means a loss in system capacity because it forces the compressor to operate at a lower suction pressure to maintain the desired evaporator temperature. Standard practice is to size the suction line for a pressure drop of approximately two degrees change in saturation temperature. In terms of pressure loss at 40 F suction temperature, this corresponds to about 1.8 psi for R-12, 2.9 psi for R-22, 2.2 psi for R-500, and 3.1 psi for R-502.

3

Where a reduction in pipe size is necessary to provide sufficient gas velocity to entrain oil upward in vertical risers at partial loads, a greater pressure drop is imposed at full load. To keep the total pressure drop within the desired limit, excessive riser loss can be offset by properly sizing the horizontal and "down" lines.

It is important to minimize the pressure loss in *hot gas lines* because these losses can increase the required compressor horsepower and decrease the compressor capacity. It is usual practice not to exceed a pressure drop corresponding to one to two degrees change in saturation temperature. This is equal to about 1.8 to 3.8 psi for R-12, 2.9 to 6 psi for R-22, 2.2 to 4.6 psi for R-500, and 3.1 to 6.3 psi for R-502.

REFRIGERANT PIPE SIZING

Charts 7 thru 21 are used to select the proper steel pipe and copper tubing size for the refrigeration lines. They are based on the Darcy-Weisbach formula:

$$h = f \times \frac{L}{D} \times \frac{V^2}{2g}$$

where h = loss of head in feet of fluid
f = friction factor
L = length of pipe in feet
D = diameter of pipe in feet
V = velocity in fps
g = acceleration of gravity = 32.17 ft/sec/sec

The friction factor depends on the roughness of pipe surface and the Reynolds number of the fluid. In this case the Reynolds number and the Moody chart are used to determine the friction factor.

Use of Pipe Sizing Charts

The following procedure for sizing refrigerant piping is recommended:

1. Measure the length (in feet) of straight pipe.
2. Add 50% to obtain a trial total equivalent length.
3. If other than a rated friction loss is desired, multiply the total equivalent length by the correction factor from the table following the appropriate pipe or tubing size chart.
4. If necessary, correct for suction and condensing temperatures.
5. Read pipe size from *Charts 7 thru 21* to determine size of fittings.
6. Find equivalent length (in feet) of fittings and hand valves from *Chapter 1* and add to the length of straight pipe *(Step 1)* to obtain the total equivalent length.
7. Correct as in *Steps 3 and 4* if necessary.
8. Check pipe size.

In some cases, particularly in liquid and suction lines, it may be necessary to find the *actual pressure drop*. To do this, use the procedure described in *Steps 9 thru 11:*

9. Convert the friction drop (F from *Step 3*) to psi, using refrigerant tables or the tables in *Part 4*.
10. Find the pressure drop thru automatic valves and accessories from manufacturers' catalogs. If these are given in equivalent feet, change to psi by multiplying by the ratio:

$$\frac{Step\ (9)}{Step\ (6)}$$

11. Add *Steps 9 and 10*.

In systems in which automatic valves and accessories may create a relatively high pressure drop, the line size can be increased to minimize their effect on the system.

Example 1 — Use of Pipe Sizing Charts

Given:
Refrigerant 12 system
Load — 46 tons
Equivalent length of piping — 65 ft
Saturated suction — 30 F
Condensing temperature — 100 F, of subcooling
Type L copper tubing

Find:
Suction line size for pressure drop corresponding to 2 F.
Actual pressure drop in terms of degrees F for size selected.

Solution:

See *Chart 7*.

1. Line sizes for 40 F saturated suction and 105 F condensing temperature are shown on *Chart 7*. Determine the correction factor for a 30 F suction temperature of 1.18 from table in notes following *Chart 9*.
2. Determine adjusted tons to be used in *Chart 7* by multiplying correction factor in *Step 1* by load in tons:
   $1.18 \times 46 = 55$ tons
3. Enter *Chart 7* and project upward from 55 tons, to the 0°F subcooling line of a 2⅝ in. OD pipe size, then to a 3⅛ in. OD pipe size. At 2⅝ in. OD, a 2 F drop is obtained with 34 ft of pipe; at 3⅛ in. OD a 2 F drop is obtained with 80 ft of pipe. Select a 3⅛ in. OD pipe to obtain less than a 2 F drop.
4. Use the following equation to determine actual pressure drop in terms of degrees F in the 3⅛ in. OD pipe with a 46 ton load:

Actual pressure drop

$$= \frac{\text{equivalent ft of pipe}}{\text{piping allowed for 2 F drop}} \times 2\text{ F}$$

$$= \frac{65}{80} \times 2 = 1.6\text{ F}$$

LIQUID LINE DESIGN

Refrigeration oil is sufficiently miscible with these refrigerants in the liquid phase to insure adequate mixing and oil return. Therefore low liquid velocities and traps in liquid lines do not pose oil return problems.

The amount of liquid line pressure drop which can be tolerated is dependent on the number of degrees subcooling of the liquid. Usually this amounts to 5 F to 15 F as the liquid leaves the condenser. Liquid lines should not be sized for more than a 2 F drop under normal circumstances. In addition, liquid lines passing thru extremely warm spaces should be insulated.

Friction Drop and Static Head

With an appreciable friction drop and/or a static head due to elevation of the liquid metering device above the condenser, it may be necessary to resort to some additional means of liquid subcooling to prevent flashing in the liquid line. Increasing the liquid line pipe size minimizes pipe friction and flashing due to friction drop.

In large systems where the cost is warranted, a liquid pump may be used to overcome static head.

An arrangement shown in *Fig. 57* illustrates a method which may be used to overcome the effect of excessive flash gas caused by a high static head in the system. This arrangement does not prevent the forming of flash gas, but does offset the effect it might have on the operation of the evaporator and valves.

Liquid Subcooling

Where liquid subcooling is required, it is usually accomplished by one or both of the following arrangements:

1. A liquid suction heat interchanger (heat dissipates internally to suction gas).
2. Liquid subcooling coils in evaporative condensers and air-cooled condensers (heat dissipates externally to atmosphere).

The amount of liquid subcooling required may be determined by use of a nomograph, *Chart 22* or by calculation. The following examples illustrate both methods.

Example 2 — Liquid Subcooling from Nomograph

Given:

Refrigerant 12 system
Condensing temperature — 100 F (131.6 psia)
Liquid line pressure drop (incl. liquid lift) — 29.9 psi

Find:

Amount of liquid subcooling in degrees F required to prevent flashing of liquid refrigerant.

Solution:

Use *Chart 22*.

1. Determine pressure at expansion valve:
   $131.6 - 29.9 = 101.7$ psia
2. Draw line from point *A* (100 F cond temp) to point *B* (101.7 psia at expansion valve).
3. Draw line from point *C* (intersection of *AB* with line *Z*) thru point *D* (0% flash gas) to point *E* (intersection of *CD* with liquid subcooling line).
4. Liquid subcooling at point *E* = 18 F. Liquid subcooling required to prevent liquid flashing = 18 F.

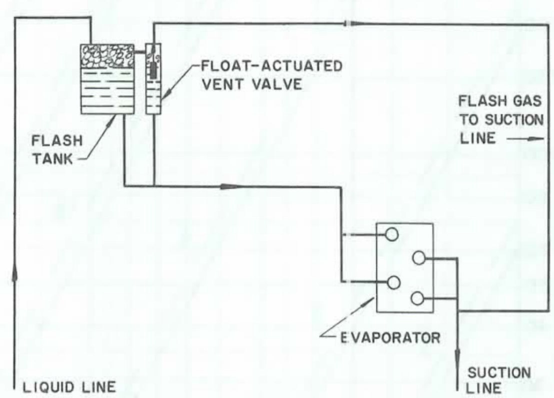
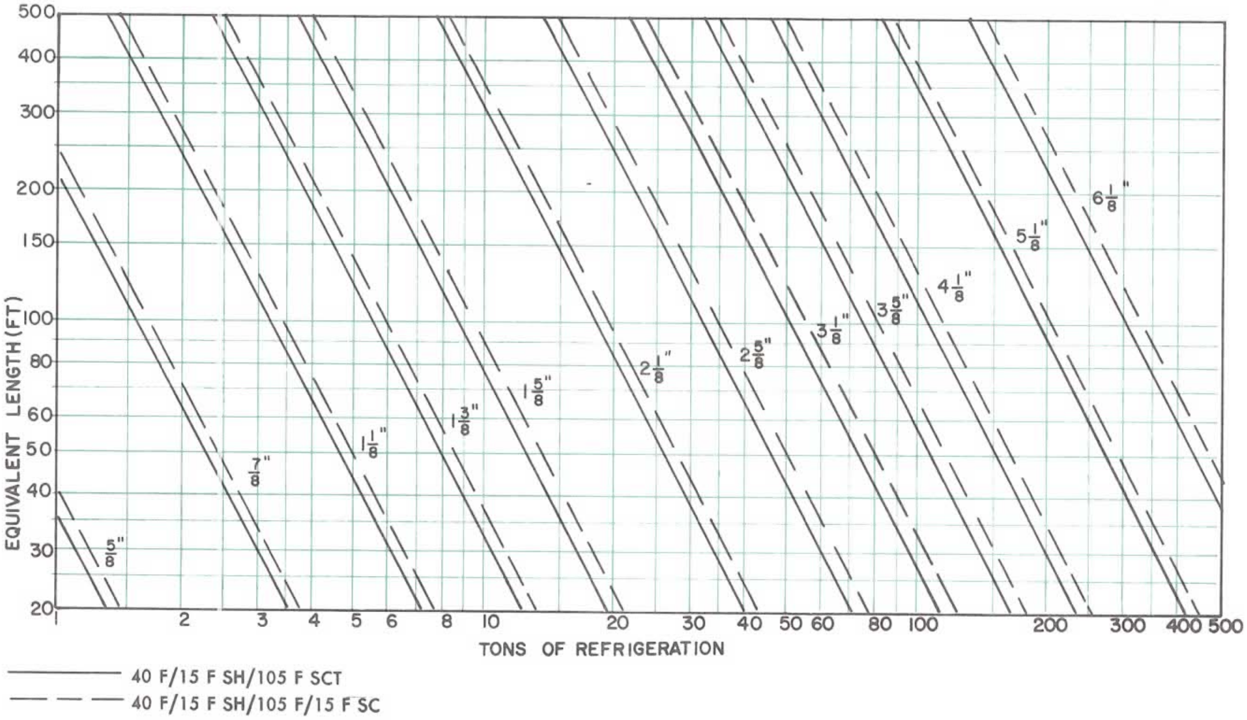


Fig. 57 — Method of Overcoming Ill Effects of System High Static Head

REFRIG. 12
40°/105°

CHART 7—SUCTION LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F



REFRIG. 12
40°/105°

CHART 8—HOT GAS LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

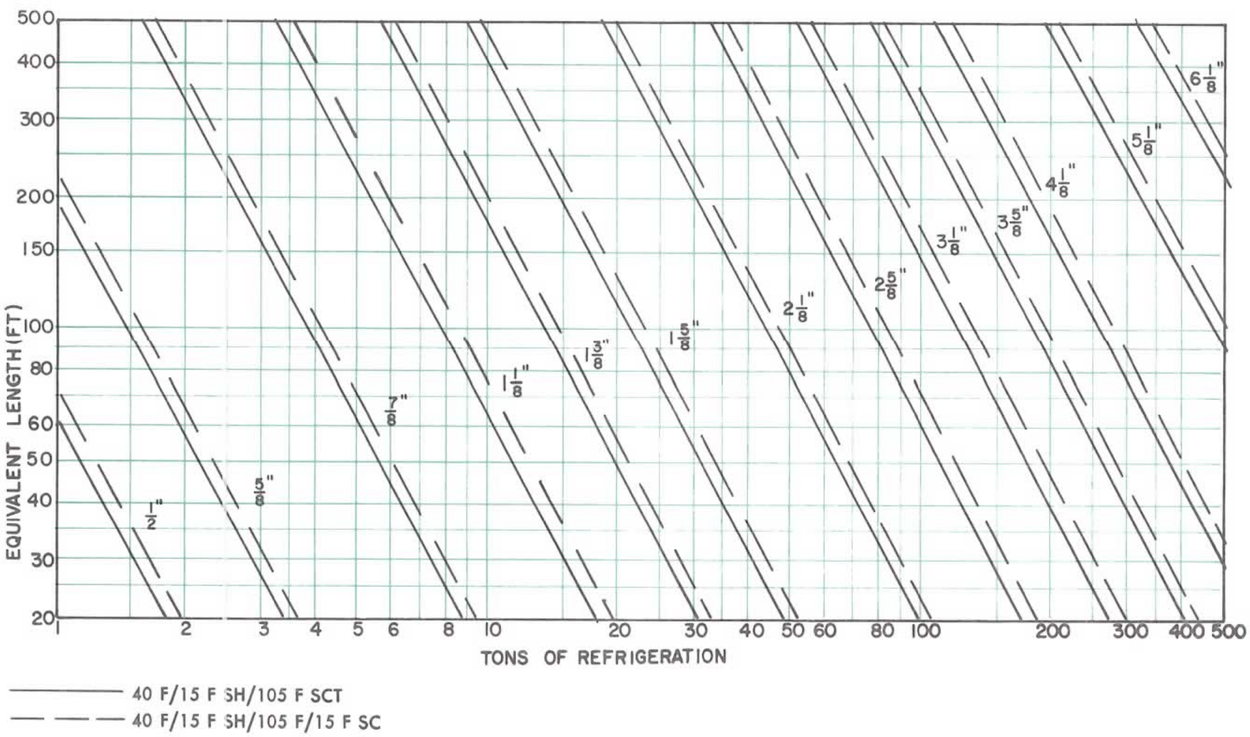
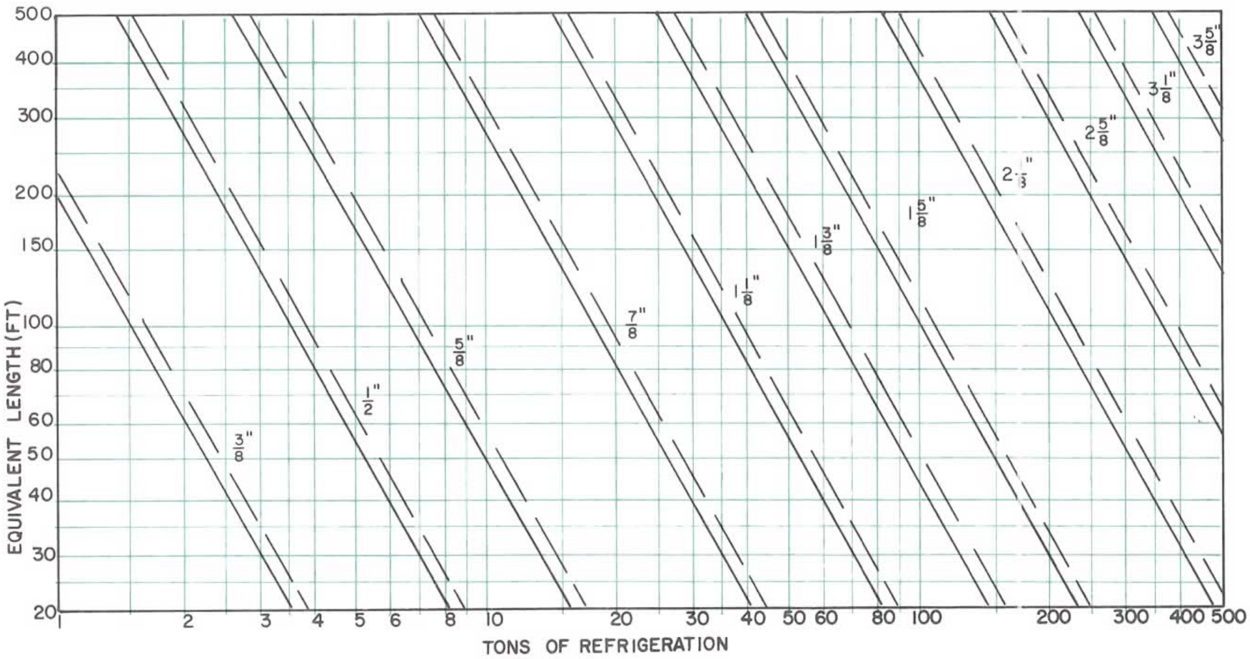


CHART 9—LIQUID LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

REFRIG. 12
40°/105°



——— 40 F/15 F SH/105 F SCT
— — — 40 F/15 F SH/105 F/15 F SC

Range of Chart 9: Saturated Suction Temperatures —40 F to 50 F; Condensing Temperatures 80 F to 160 F

Pressure drop is given in equivalent degrees because of the general acceptance of this method of sizing. The corresponding pressure drop in psi may be determined by referring to the saturated refrigerant tables.

To use Charts 7 and 8 **for conditions other than 40 F saturated suction, 105 F condensing,** multiply the refrigeration load in tons by the factor below and use the product in reading the chart (S = Suction, HG = Hot Gas).

COND TEMP (F)	SATURATED SUCTION TEMPERATURE (F)																			
	—40		—30		—20		—10		0		10		20		30		40		50	
	TONS MULTIPLYING FACTOR																			
	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG
80	5.07	1.50	3.92	1.46	3.08	1.43	2.45	1.39	1.96	1.36	1.59	1.33	1.31	1.30	1.08	1.28	0.89	1.25	0.75	1.22
90	—	—	4.11	1.34	3.22	1.31	2.55	1.27	2.05	1.24	1.66	1.21	1.36	1.19	1.12	1.16	0.93	1.14	0.78	1.11
100	—	—	4.33	1.23	3.39	1.20	2.69	1.17	2.15	1.14	1.74	1.11	1.43	1.09	1.18	1.06	0.98	1.04	0.82	1.02
110	—	—	—	—	3.58	1.10	2.83	1.08	2.27	1.05	1.84	1.02	1.50	1.00	1.24	0.98	1.02	0.96	0.85	0.93
120	—	—	—	—	3.80	1.03	3.00	1.00	2.40	0.97	1.94	0.95	1.58	0.93	1.30	0.90	1.08	0.88	0.90	0.86
130	—	—	—	—	—	—	3.19	0.93	2.55	0.91	2.06	0.88	1.68	0.86	1.37	0.84	1.14	0.82	0.95	0.80
140	—	—	—	—	—	—	3.42	0.88	2.73	0.85	2.20	0.83	1.79	0.81	1.46	0.78	1.21	0.76	1.00	0.75
150	—	—	—	—	—	—	—	—	2.29	0.81	2.35	0.78	1.91	0.76	1.56	0.74	1.29	0.72	1.07	0.70
160	—	—	—	—	—	—	—	—	3.17	0.77	2.54	0.74	2.06	0.72	1.69	0.70	1.39	0.68	1.15	0.66

NOTES:

1. To use suction hot gas and liquid line charts **for friction drop other than 2 F,** multiply equivalent length by factor below and use product in reading chart.

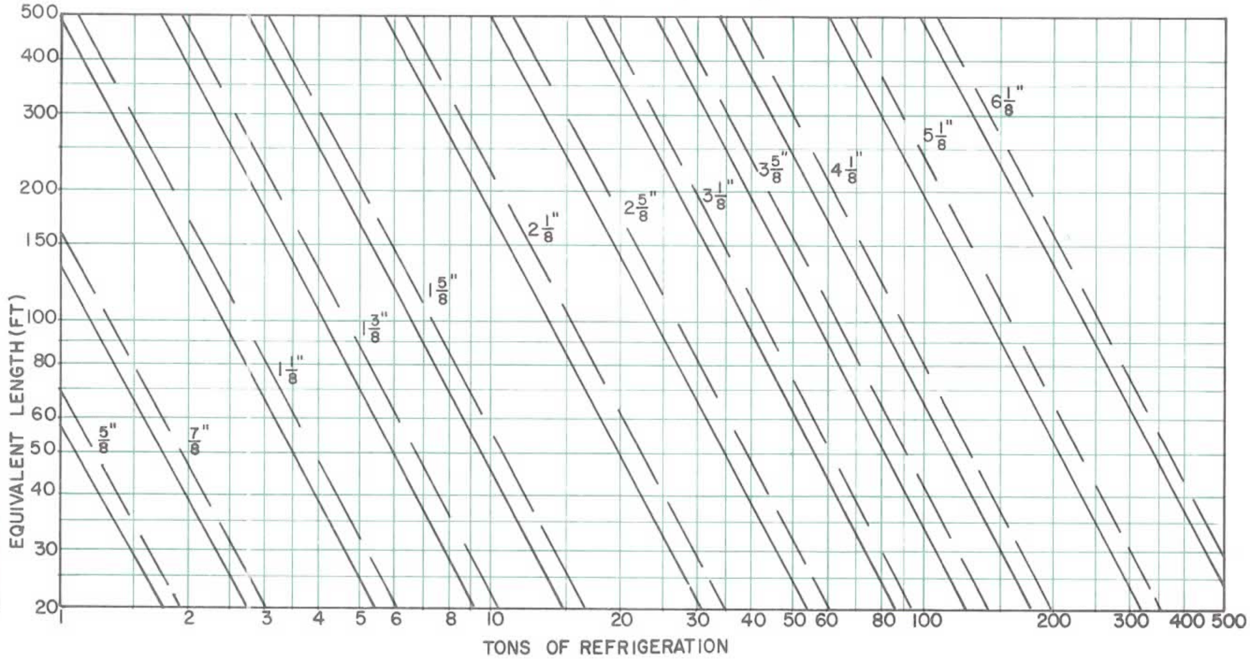
Friction Drop (F)	Liquid Line Hot Gas Line Suction Line	.5	1.0	1.5	2.0	2.5	3.0	4.0	5.0	6.0
Multiplier		4.0	2.0	1.3	1.0	0.8	0.7	0.5	0.4	0.3

2. Pipe sizes are OD and are for Type L copper tubing.

REFRIG. 502
0°/105°

CHART 10—SUCTION LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

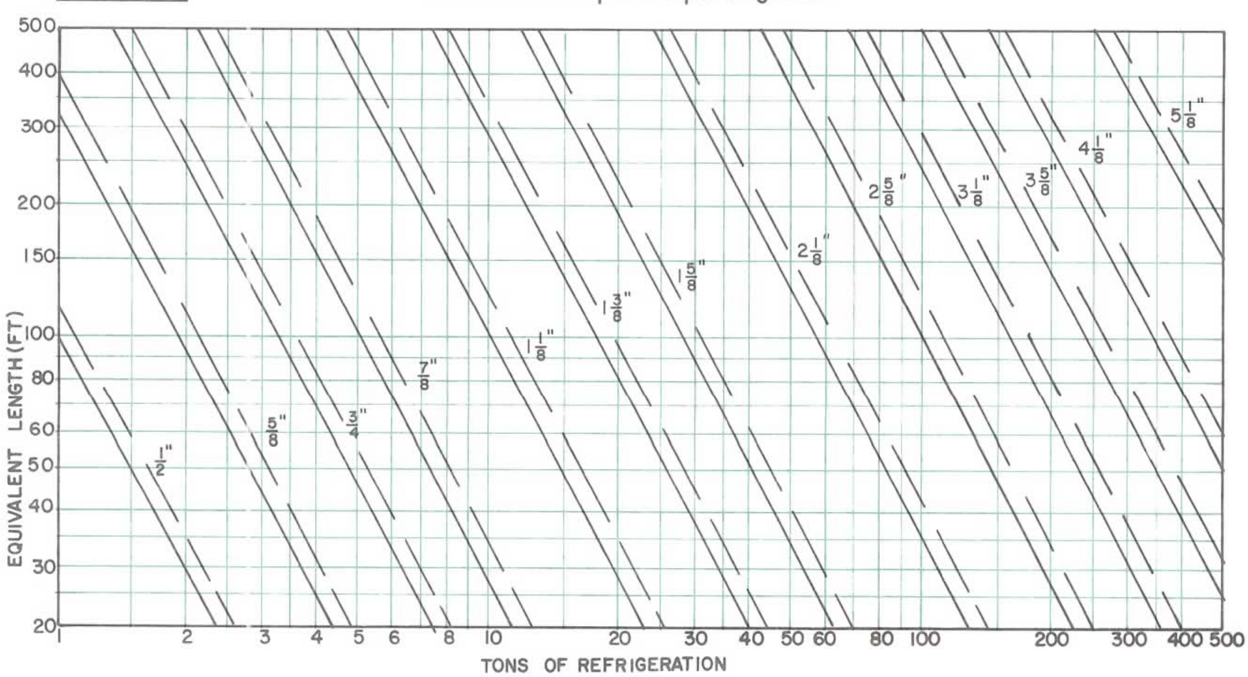


——— 0 F/15 F SH/105 F SCT
— — — 0 F/15 F SH/105 F/15 F SC

REFRIG. 502
0°/105°

CHART 11—HOT GAS LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

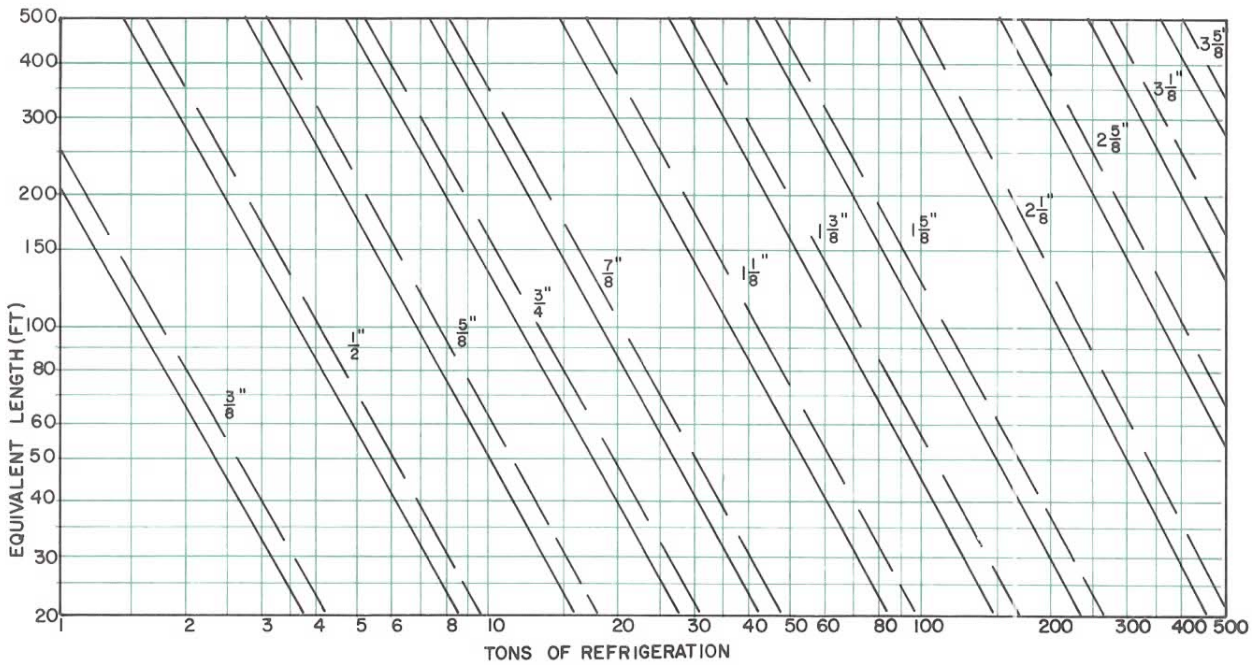


——— 0 F/15 F SH/105 F SCT
— — — 0 F/15 F SH/105 F/15 F SC

CHART 12—LIQUID LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

REFRIG. 502
40°/105°



——— 0 F/15 F SH/105 F SCT
— — — 0 F/15 F SH/105 F/15 F SC

Range of Chart 12: Saturated Suction Temperatures —60 F to +40 F
Condensing Temperatures 80 F to 160 F

3

Pressure drop is given in equivalent degrees because of the general acceptance of this method of sizing. The corresponding pressure drop in psi may be determined by referring to the saturated refrigerant tables.

To use Charts 10 and 11 **for conditions other than 0° F saturated suction, 105 F condensing,** multiply the refrigeration load in tons by the factor below and use the product in reading the chart (S = Suction, HG = Hot Gas).

COND TEMP (F)	SATURATED SUCTION TEMPERATURE (F)																			
	—50		—40		—30		—20		—10		0		10		20		30		40	
	TONS MULTIPLYING FACTOR																			
	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG
80	2.82	1.44	2.15	1.34	1.67	1.27	1.31	1.22	1.04	1.18	0.84	1.14	0.69	1.11	0.56	1.07	0.47	1.04	0.39	1.00
90	—	—	2.31	1.29	1.79	1.20	1.40	1.14	1.12	1.10	0.90	1.07	0.73	1.04	0.60	1.00	0.49	0.97	0.41	0.94
100	—	—	—	—	1.93	1.17	1.51	1.10	1.20	1.05	0.96	1.01	0.78	0.98	0.64	0.95	0.53	0.92	0.44	0.89
110	—	—	—	—	2.11	1.16	1.64	1.07	1.30	1.01	1.04	0.97	0.84	0.94	0.69	0.91	0.57	0.88	0.47	0.85
120	—	—	—	—	—	—	1.79	1.07	1.42	0.99	1.13	0.94	0.91	0.91	0.74	0.88	0.61	0.85	0.51	0.82
130	—	—	—	—	—	—	1.98	1.08	1.56	0.99	1.24	0.93	0.99	0.88	0.81	0.85	0.66	0.82	0.55	0.79
140	—	—	—	—	—	—	—	—	1.74	1.01	1.37	0.93	1.10	0.88	0.89	0.84	0.73	0.81	0.60	0.78
150	—	—	—	—	—	—	—	—	1.94	1.05	1.53	.096	1.22	0.88	0.98	0.83	0.80	0.80	0.66	0.77
160	—	—	—	—	—	—	—	—	—	—	1.72	1.00	1.36	0.91	1.10	0.84	0.89	0.80	0.73	0.77

NOTES:

1. To use suction hot gas and liquid line charts **for friction drop other than 2 F,** multiply equivalent length by factor below and use product in reading chart.

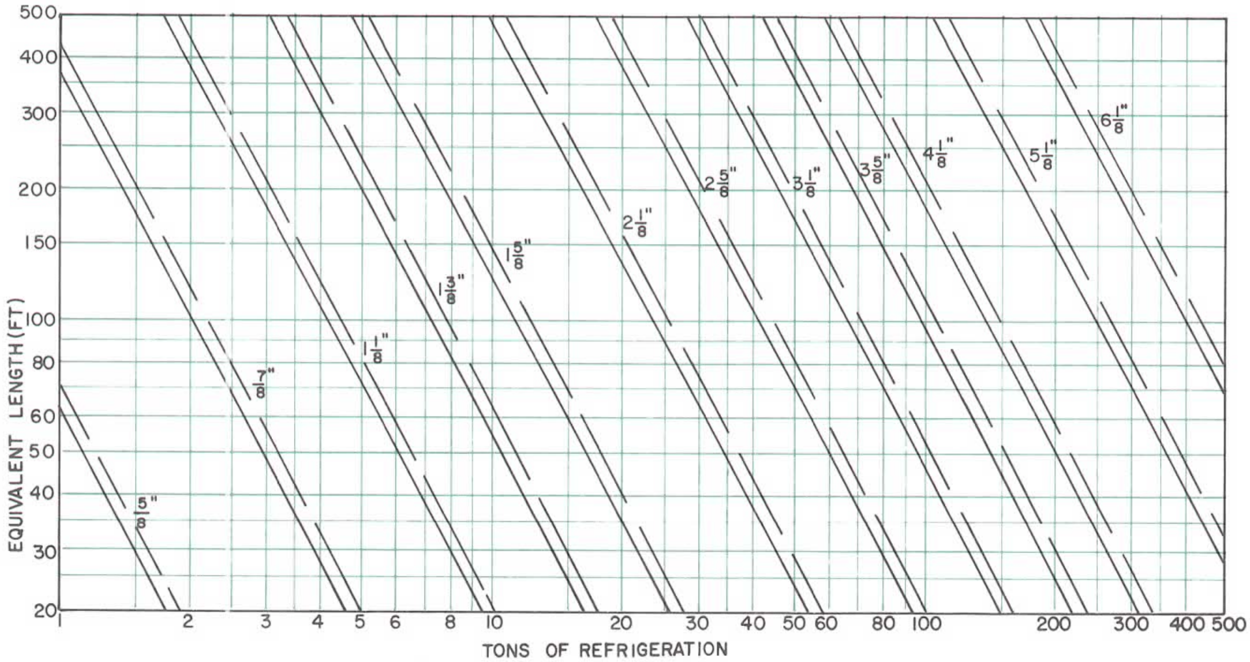
Friction Drop (F)	**Liquid Line Hot Gas Line Suction Line**	**.5**	**1.0**	**1.5**	**2.0**	**2.5**	**3.0**	**4.0**	**5.0**	**6.0**
Multiplier		4.0	2.0	1.3	1.0	0.8	0.7	0.5	0.4	0.3

2. Pipe sizes are OD and are for Type L copper tubing.

REFRIG. 500
40°/105°

CHART 13—SUCTION LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F



40 F/15 F SH/105 F SCT

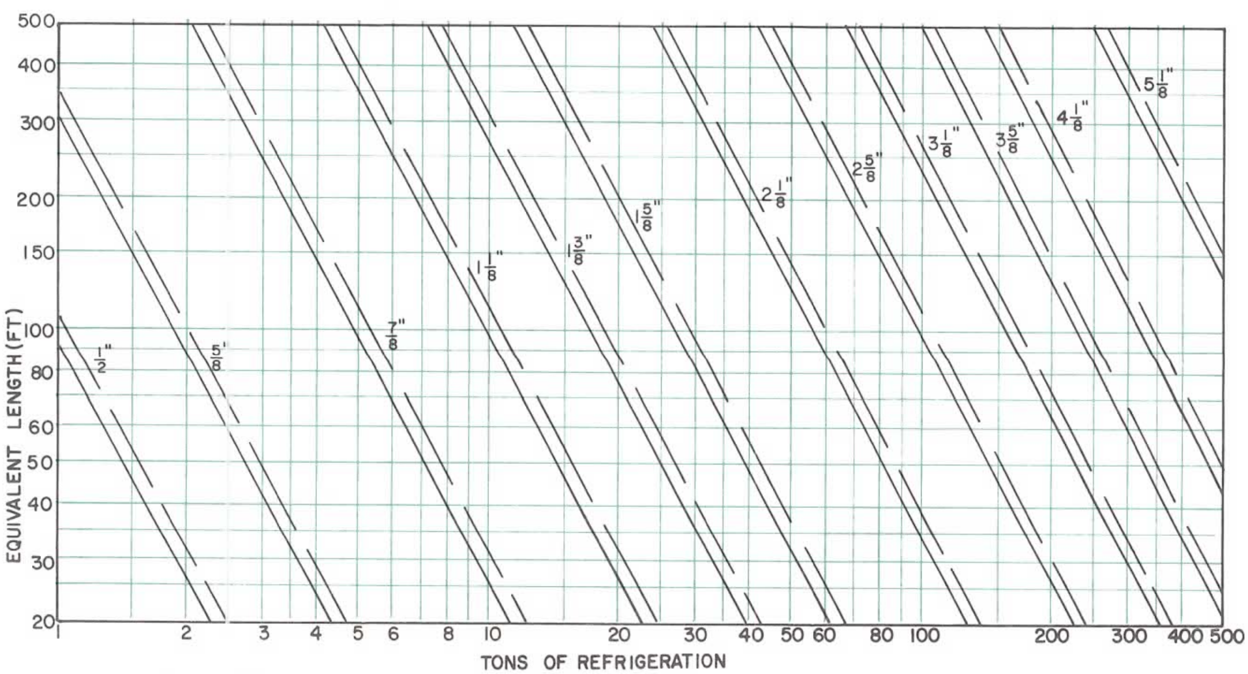
40 F/15 F SH/105 F/15 F SC

3

REFRIG. 500
40°/105°

CHART 14—HOT GAS LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F



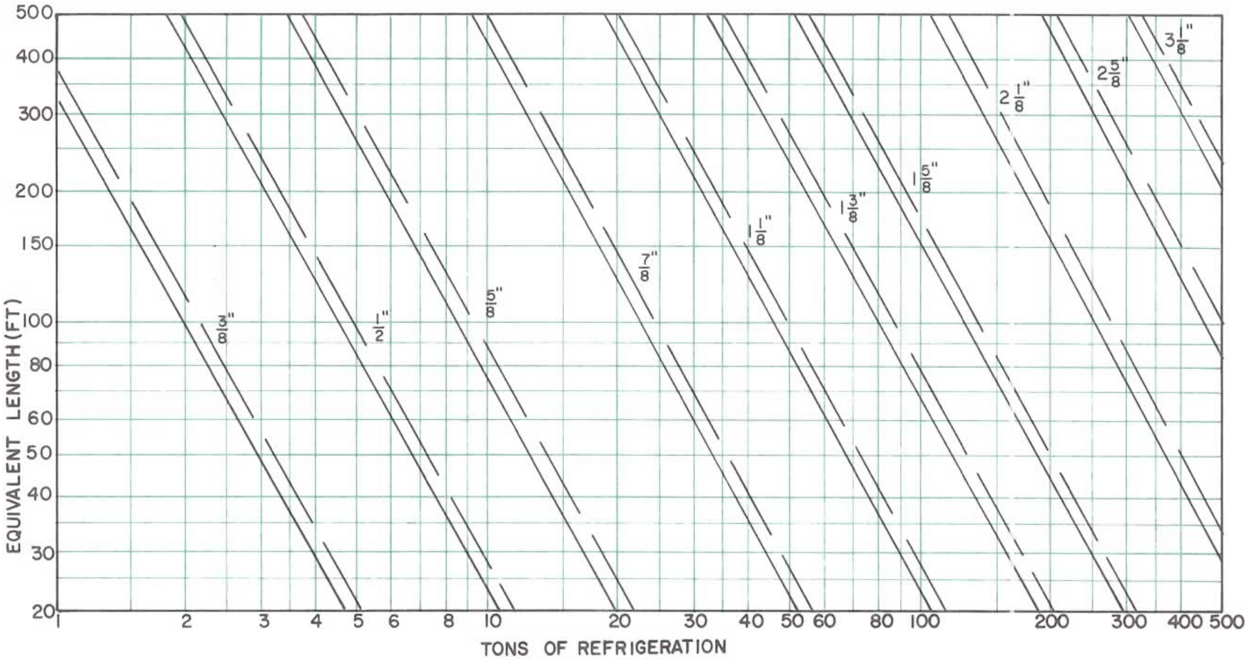
40 F/15 F SH/105 F SCT

40 F/15 F SH/105 F/15 F SC

CHART 15—LIQUID LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

REFRIG. 500
40°/105°



———— 40 F/15 F SH/105 F SCT
— — — 40 F/15 F SH/105 F/15 F SC

Range of Chart 15: Saturated Suction Temperatures —40 F to 50 F
Condensing Temperatures 80 F to 160 F

Pressure drop is given in equivalent degrees because of the general acceptance of this method of sizing. The corresponding pressure drop in psi may be determined by referring to the saturated refrigerant tables.

To use Charts 13 and 14 **for conditions other than 40 F saturated suction, 105 F condensing,** multiply refrigeration load in tons by factor below and use product in reading chart (S = Suction, HG = Hot Gas).

COND TEMP (F)	SATURATED SUCTION TEMPERATURE (F)																			
	—40		—30		—20		—10		0		10		20		30		40		50	
	TONS MULTIPLYING FACTOR																			
	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG
80	5.09	1.53	3.93	1.49	3.08	1.44	2.44	1.39	1.95	1.35	1.58	1.32	1.29	1.29	1.07	1.26	0.89	1.23	0.74	1.20
90	—	—	4.14	1.37	3.23	1.33	2.56	1.28	2.05	1.24	1.66	1.21	1.36	1.19	1.12	1.15	0.93	1.13	0.77	1.10
100	—	—	4.37	1.26	3.41	1.23	2.70	1.19	2.16	1.15	1.74	1.12	1.42	1.09	1.17	1.06	0.97	1.04	0.81	1.02
110	—	—	—	—	3.61	1.15	2.85	1.11	2.28	1.08	1.84	1.04	1.50	1.01	1.23	0.99	1.02	0.96	0.85	0.94
120	—	—	—	—	3.84	1.08	3.03	1.05	2.42	1.01	1.95	0.98	1.59	0.95	1.30	0.92	1.08	0.90	0.90	0.88
130	—	—	—	—	—	—	3.24	0.99	2.58	0.96	2.08	0.93	1.69	0.90	1.38	0.87	1.14	0.85	0.95	0.83
140	—	—	—	—	—	—	3.49	0.95	2.77	0.92	2.23	0.89	1.81	0.86	1.48	0.83	1.22	0.80	1.01	0.78
150	—	—	—	—	—	—	—	—	3.01	0.89	2.41	0.86	1.95	0.83	1.59	0.80	1.31	0.77	1.09	0.75
160	—	—	—	—	—	—	—	—	3.29	0.88	2.63	0.84	2.13	0.82	1.73	0.78	1.42	0.75	1.18	0.73

NOTES:

1. To use suction hot gas and liquid line charts **for friction drop other than 2 F,** multiply equivalent length by factor below and use product in reading chart.

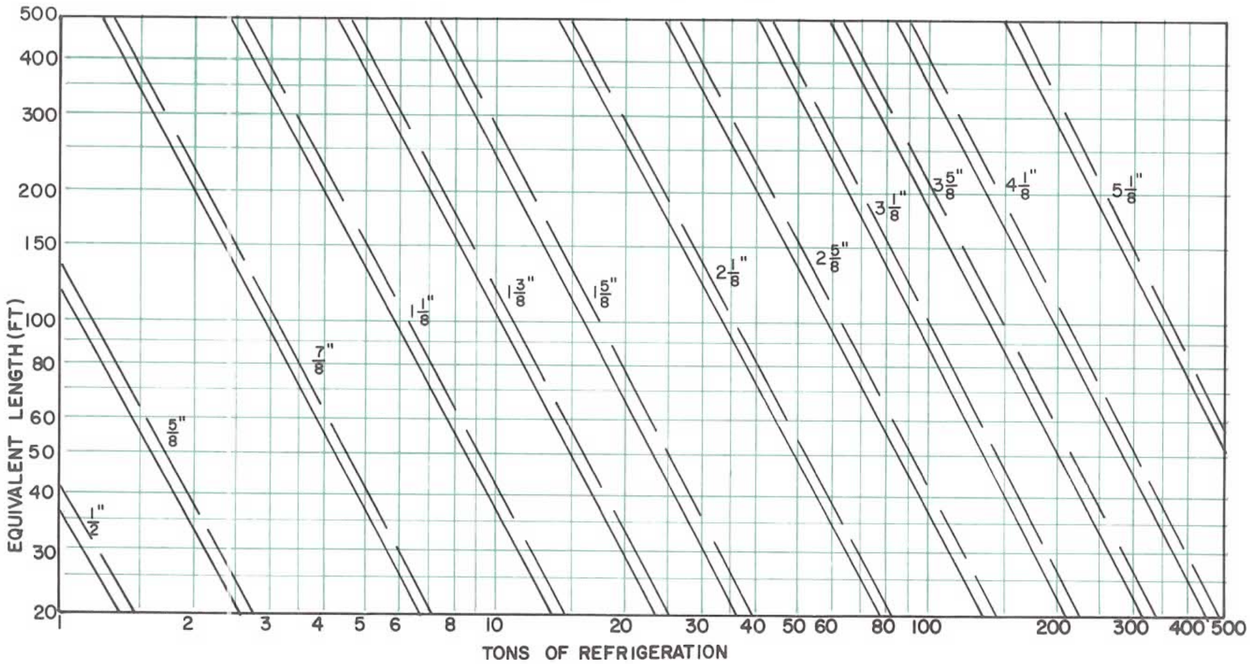
Friction Drop (F)	**Liquid Line Hot Gas Line Suction Line**	**.5**	**1.0**	**1.5**	**2.0**	**2.5**	**3.0**	**4.0**	**5.0**	**6.0**
Multiplier		4.0	2.0	1.3	1.0	0.8	0.7	0.5	0.4	0.3

2. Pipe sizes are OD and are for Type L copper tubing.

REFRIG. 22
40°/105°

CHART 16—SUCTION LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

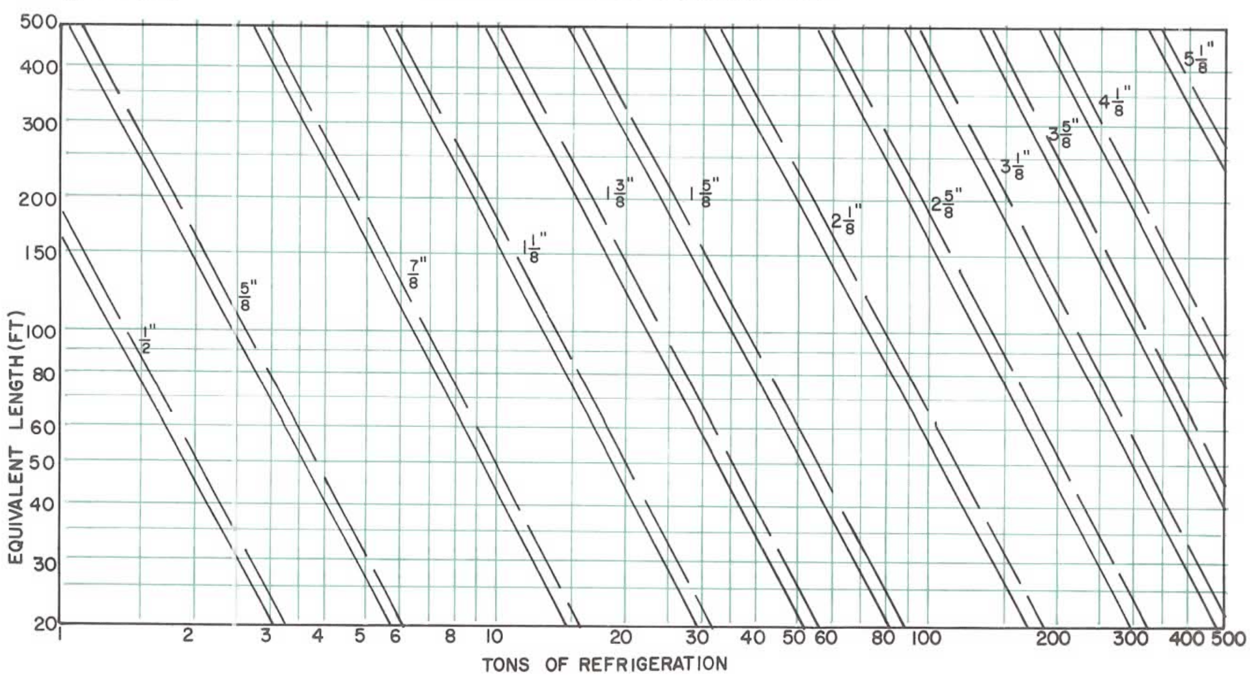


40 F/15 F SH/105 F SCT
40 F/15 F SH/105 F/15 F SC

REFRIG. 22
40°/105°

CHART 17—HOT GAS LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F

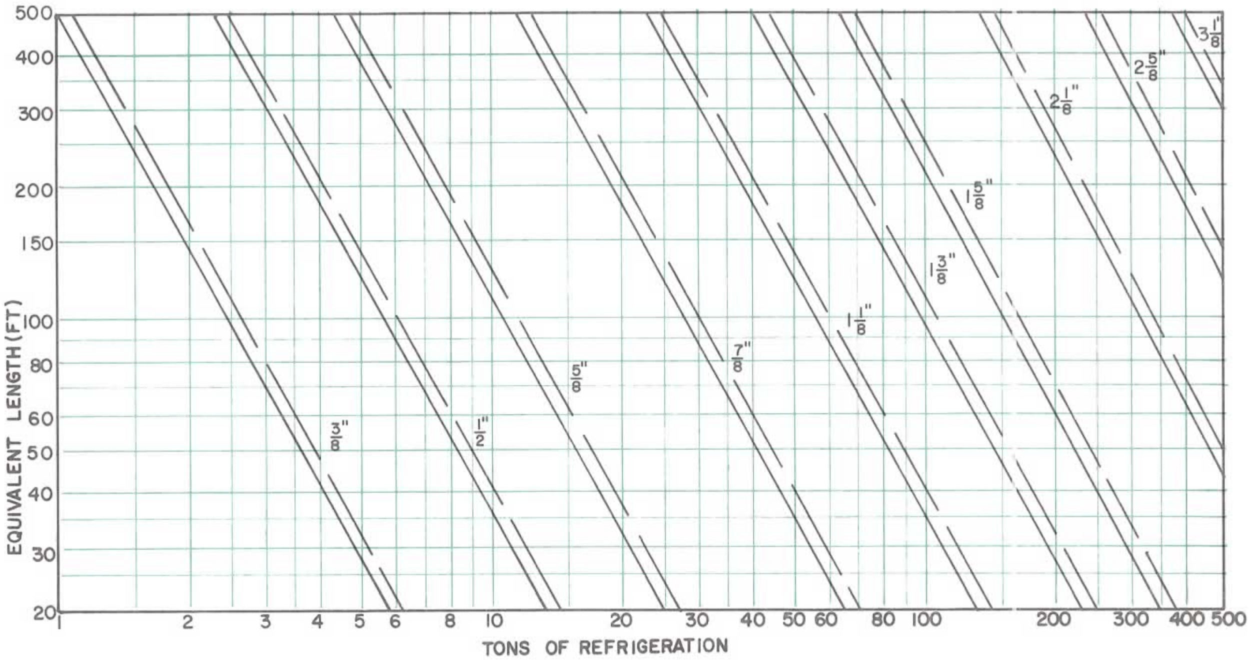


40 F/15 F SH/105 F SCT
40 F/15 F SH/105 F/15 F SC

REFRIG. 22
40°/105°

CHART 18—LIQUID LINES—COPPER TUBING

For Pressure Drop Corresponding to 2 F



———— 40 F/W15 F SH/105 F SCT
— — — 40 F/W15 F SH/105 F/W15 F SC

Range of Chart 18: Saturated Suction Temperatures —40 F to 50 F
Condensing Temperatures 80 F to 160 F

Pressure drop is given in equivalent degrees because of the general acceptance of this method of sizing. The corresponding pressure drop in psi may be determined by referring to the saturated refrigerant tables.

To use Charts 16 and 17 **for conditions other than 40 F saturated suction, 105 F condensing,** multiply the load in tons by the factor below and and use the product in reading the chart (S = Suction, HG = Hot Gas).

COND TEMP (F)	SATURATED SUCTION TEMPERATURE (F)																			
	—40		—30		—20		—10		0		10		20		30		40		50	
	TONS MULTIPLYING FACTOR																			
	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG
80	4.75	1.52	3.72	1.49	2.95	1.41	2.37	1.35	1.92	1.30	1.57	1.28	1.29	1.25	1.07	1.23	0.90	1.20	0.76	1.16
90	—	—	3.86	1.38	3.06	1.33	2.45	1.26	1.99	1.21	1.63	1.18	1.34	1.15	1.12	1.13	0.94	1.11	0.79	1.08
100	—	—	4.05	1.28	3.21	1.25	2.57	1.19	2.08	1.14	1.70	1.09	1.41	1.07	1.17	1.05	0.98	1.03	0.82	1.01
110	—	—	—	—	3.37	1.18	2.70	1.14	2.19	1.08	1.79	1.03	1.47	1.00	1.23	0.98	1.02	0.96	0.86	0.94
120	—	—	—	—	3.55	1.11	2.84	1.09	2.29	1.05	1.87	0.99	1.54	0.95	1.28	0.92	1.08	0.90	0.90	0.89
130	—	—	—	—	—	—	3.02	1.03	2.44	1.01	1.99	0.96	1.64	0.91	1.36	0.88	1.13	0.86	0.95	0.84
140	—	—	—	—	—	—	3.20	1.00	2.58	0.97	2.11	0.94	1.73	0.89	1.44	0.85	1.20	0.82	1.01	0.80
150	—	—	—	—	—	—	—	—	2.78	0.94	2.26	0.92	1.86	0.88	1.54	0.83	1.28	0.80	1.08	0.77
160	—	—	—	—	—	—	—	—	3.00	0.93	2.44	0.90	2.00	0.87	1.65	0.83	1.37	0.78	1.15	0.75

NOTES:

1. To use suction hot gas and liquid line charts **for friction drop other than 2 F,** multiply equivalent length by factor below and use product in reading chart.

Friction Drop (F)	Liquid Line Hot Gas Line Suction Line	.5	1.0	1.5	2.0	2.5	3.0	4.0	5.0	6.0
Multiplier		4.0	2.0	1.3	1.0	0.8	0.7	0.5	0.4	0.3

2. Pipe sizes are OD and are for Type L copper tubing.

CHART 19—SUCTION LINES—STEEL PIPE

For Pressure Drop Corresponding to 2 F

SCHEDULE 40

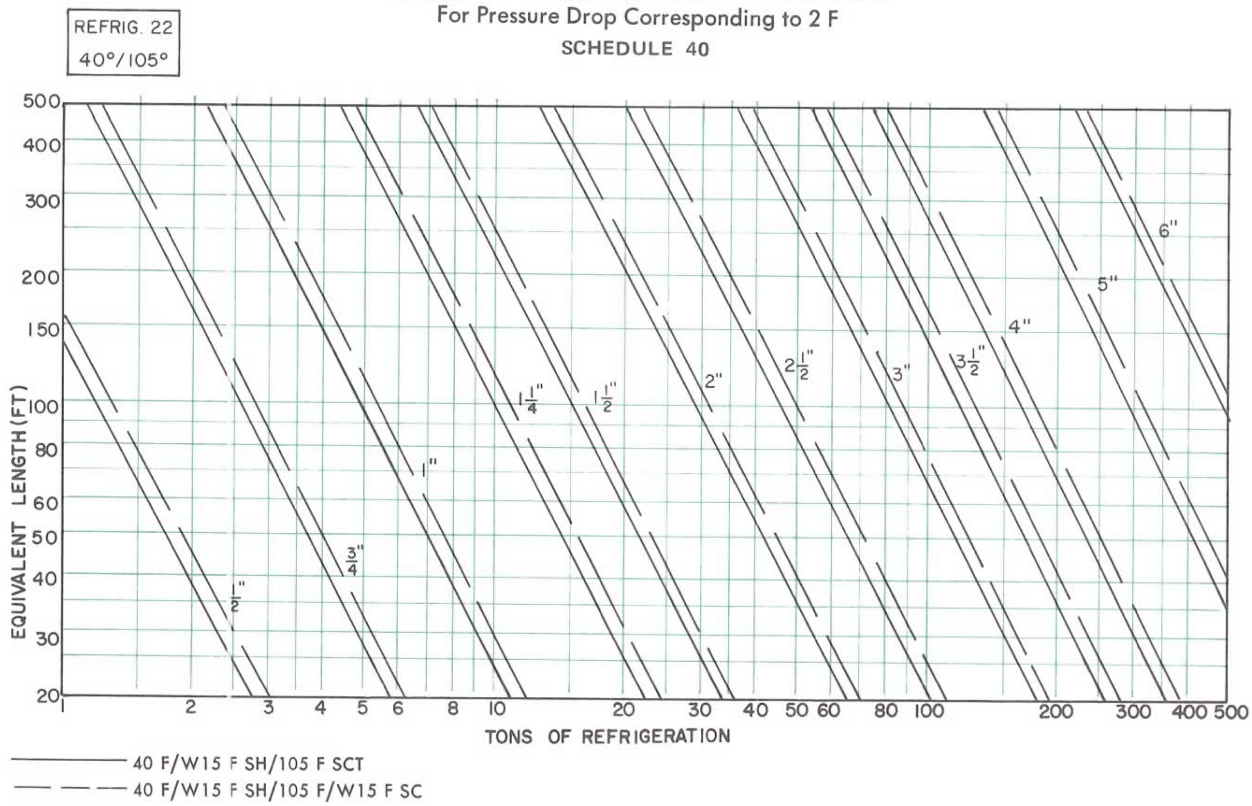
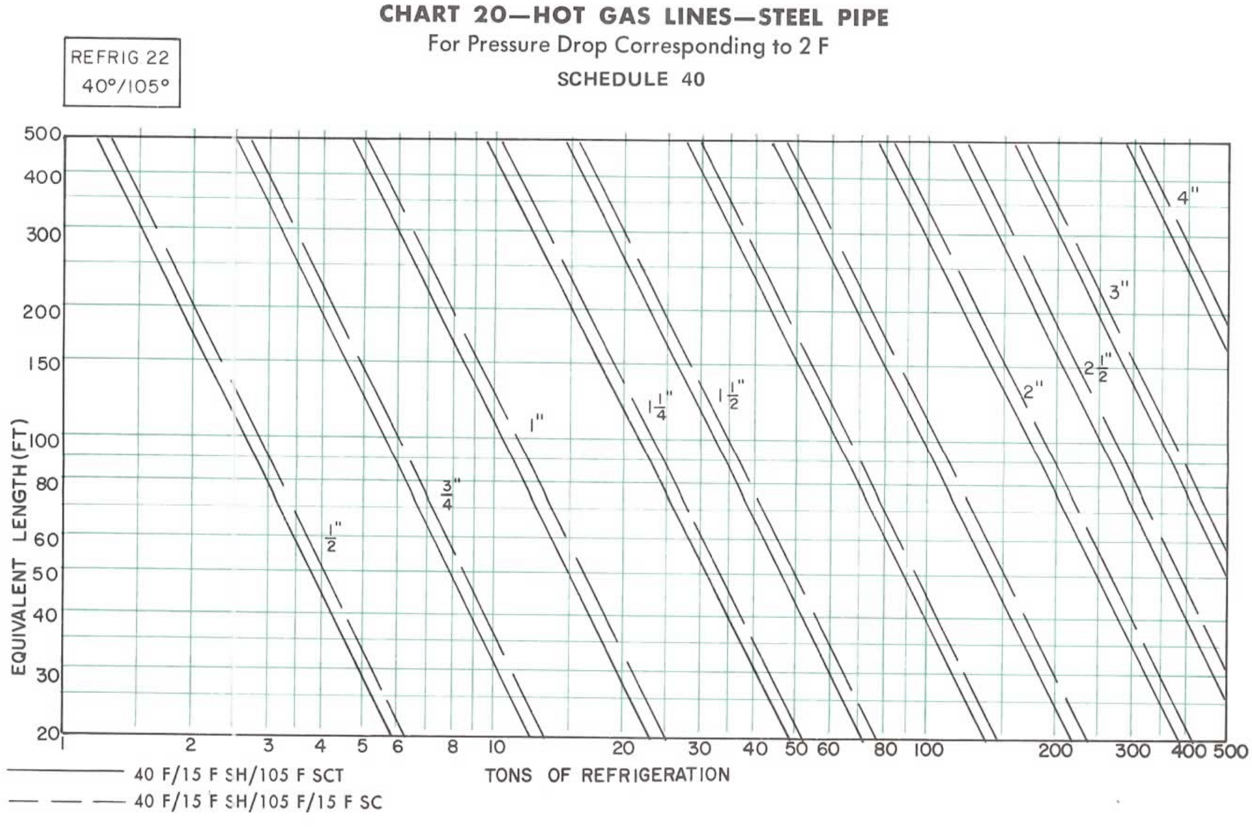


CHART 20—HOT GAS LINES—STEEL PIPE

For Pressure Drop Corresponding to 2 F

SCHEDULE 40

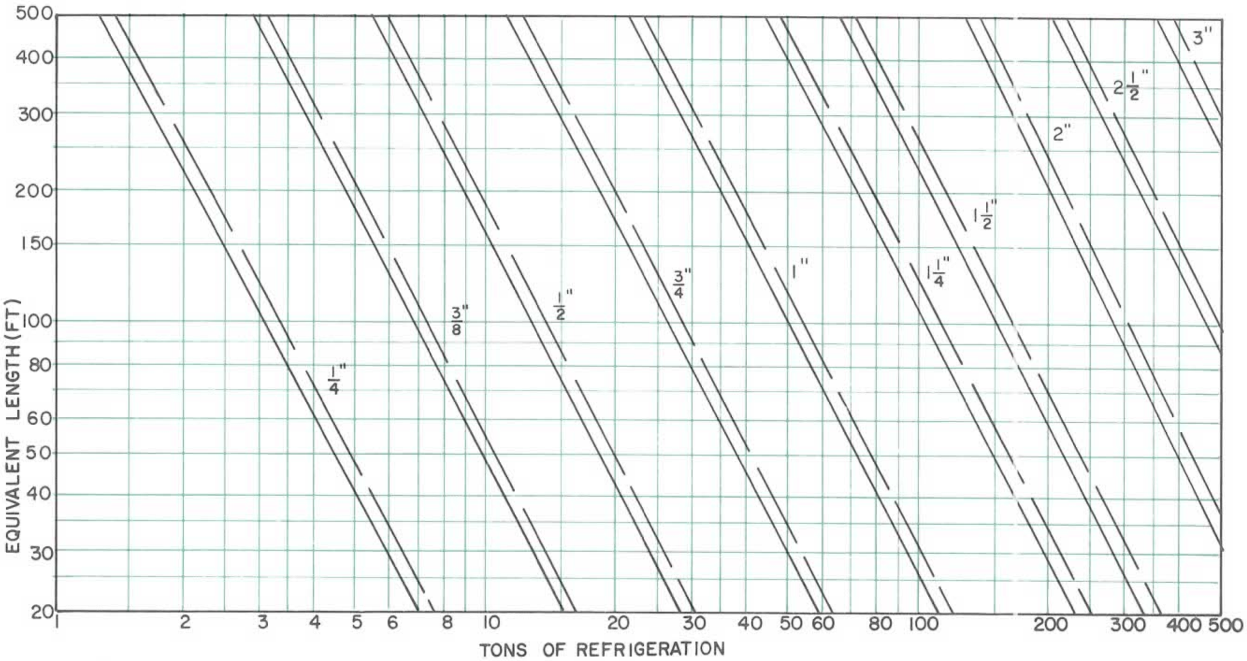


3

CHART 21—LIQUID LINES—STEEL PIPE

For Pressure Drop Corresponding to 2 F

REFRIG. 22
40°/105°



———— 40 F/15 F SH/105 F SCT
— — — 40 F/15 F SH/105 F/15 F SC

Range of Chart 21: Saturated Suction Temperatures —40 F to 50 F
Condensing Temperatures 80 F to 160 F

Pressure drop is given in equivalent degrees because of the general acceptance of this method of sizing. The corresponding pressure drop in psi may be determined by referring to the saturated refrigerant tables.

To use Charts 19 and 20 **for conditions other than 40 F saturated suction, 105 F condensing,** multiply the refrigeration load in tons by the factor below and use the product in reading the chart (S = Suction, HG = Hot Gas).

COND TEMP (F)	SATURATED SUCTION TEMPERATURE (F)																			
	—40		—30		—20		—10		0		10		20		30		40		50	
	TONS MULTIPLYING FACTOR																			
	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG	S	HG
80	4.41	1.45	3.48	1.42	2.78	1.37	2.25	1.33	1.84	1.29	1.52	1.27	1.26	1.25	1.06	1.22	0.89	1.20	0.76	1.17
90	—	—	3.64	1.32	2.90	1.28	2.35	1.24	1.92	1.20	1.58	1.17	1.31	1.15	1.10	1.13	0.93	1.11	0.79	1.08
100	—	—	3.81	1.22	3.04	1.20	2.46	1.16	2.00	1.12	1.65	1.09	1.37	1.07	1.15	1.05	0.97	1.03	0.82	1.01
110	—	—	—	—	3.20	1.13	2.58	1.10	2.10	1.06	1.73	1.02	1.44	1.00	1.20	0.98	1.01	0.96	0.86	0.95
120	—	—	—	—	3.37	1.06	2.72	1.04	2.22	1.01	1.82	0.97	1.51	0.94	1.26	0.92	1.07	0.90	0.91	0.89
130	—	—	—	—	—	—	2.88	0.99	2.34	0.97	1.93	0.93	1.60	0.90	1.33	0.87	1.12	0.85	0.95	0.84
140	—	—	—	—	—	—	3.07	0.95	2.49	0.93	2.05	0.90	1.70	0.87	1.42	0.84	1.19	0.81	1.01	0.80
150	—	—	—	—	—	—	—	—	2.67	0.89	2.19	0.88	1.81	0.85	1.51	0.81	1.27	0.79	1.08	0.77
160	—	—	—	—	—	—	—	—	2.89	0.88	2.36	0.85	1.95	0.83	1.63	0.80	1.37	0.77	1.16	0.74

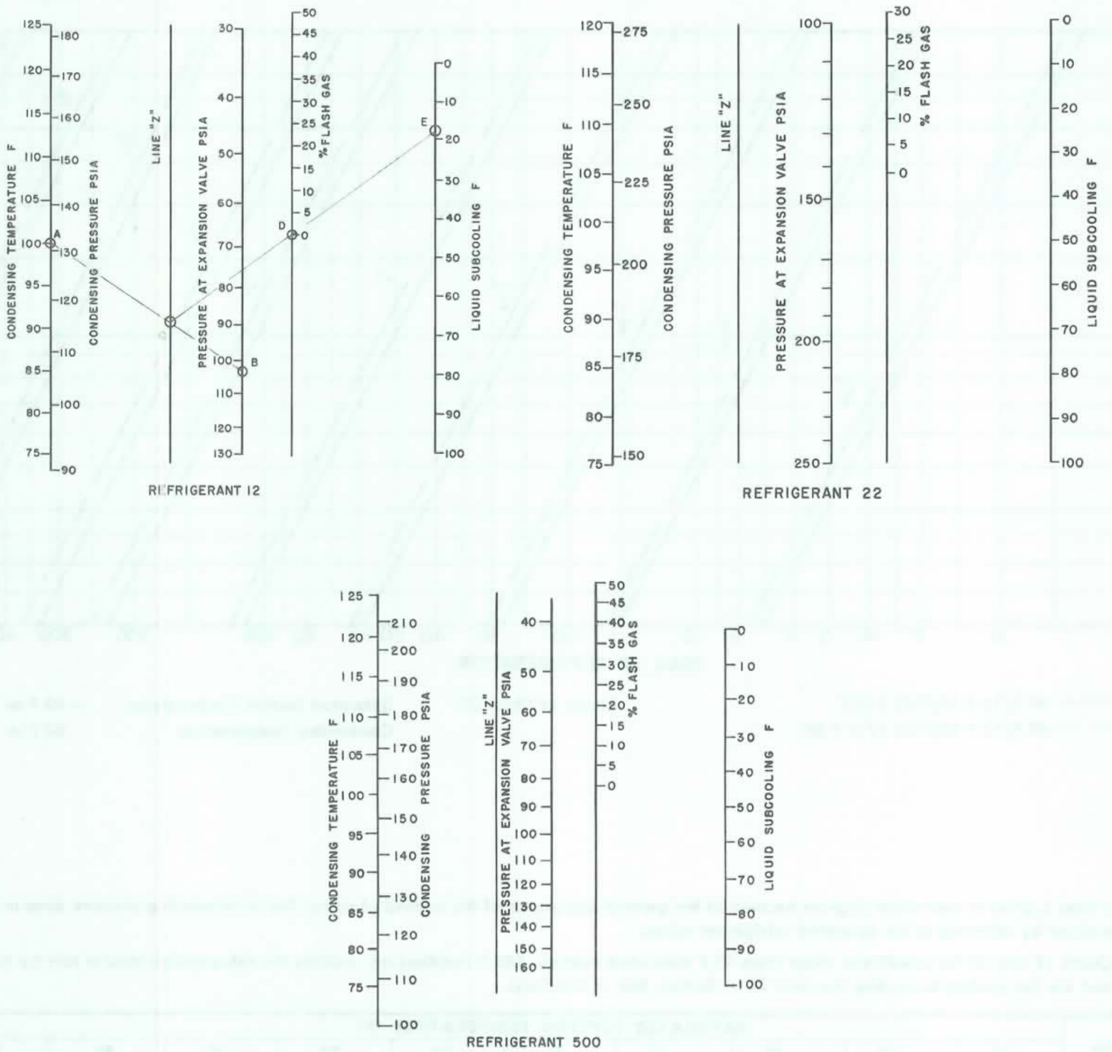
NOTES:

1. To use suction hot gas and liquid line charts **for friction drop other than 2 F,** multiply equivalent length by factor below and use product in reading chart.

Friction Drop (F)	Liquid Line Hot Gas Line Suction Line	.5	1.0	1.5	2.0	2.5	3.0	4.0	5.0	6.0
Multiplier		4.0	2.0	1.3	1.0	0.8	0.7	0.5	0.4	0.3

2. Pipe sizes are nominal and are for schedule 40 steel pipe.

CHART 22—SUBCOOLING TO COMPENSATE FOR LIQUID LINE PRESSURE DROP



Example 3 — Liquid Subcooling by Calculation

Given:

Refrigerant 12 system
Condensing temperature — 100 F
Liquid lift — 35 ft
Piping friction loss — 3 psi
Losses thru valves and accessories — 7.4 psi

Find:

Amount of liquid subcooling required to prevent flashing of liquid refrigerant.

Solution:

1. Pressure loss due to pipe friction = 3.0 psi
   Pressure loss due to valves and accessories = 7.4 psi
   Pressure loss due to 35 ft liquid lift = 35/1.8* = 19.5 psi
   Total pressure loss in liquid line = 29.9 psi

2. Condensing pressure at 100 F = 116.9 psig
   Pressure loss in liquid line = 29.9 psi
   Net pressure at liquid feed valve = 87 psig

3. Saturation temperature at 87 psig = 82 F
   (from refrigerant property tables)

4. Subcooling required
   = condensing temp − saturation temp at 87 psig
   = 100 − 82 = 18 F
   Liquid subcooling required to prevent liquid flashing = 18 F.

*At normal liquid temperatures the static pressure loss due to elevation at the top of a liquid lift is one psi for every 1.8 ft of Refrigerant 12, 2.0 ft of Refrigerant 22, and 2.1 ft of Refrigerant 500.

Sizing of Condenser to Receiver Lines (Condensate Lines)

Liquid line piping from a condenser to a receiver is run out horizontally (same size as the condenser outlet connection) to allow for drainage of the condenser. It is then dropped vertically a sufficient distance to allow a liquid head in the line to overcome line friction losses. Additional head is required for coil condensers where the receiver is vented to the inlet of the coil. This additional head is equivalent to the pressure drop across the condenser coil. The condensate line is then run horizontally to the receiver.

Table 17 shows recommended sizes of the condensate line between the bottom of the liquid leg and the receiver.

TABLE 17—CONDENSATE LINE SIZING

CONDENSER TO RECEIVER

(Based on Type L Copper Tubing)

CONDENSATE LINE SIZE (OD, In.)	REFRIGERATION, MAX. TONS Refrigerant 12	Refrigerant 22	Refrigerant 500	"X" MIN.*
½	1.2	1.4	1.2	
⅝	2.3	2.5	2.4	8″
⅞	6.4	7.7	6.8	
1⅛	13.3	15.9	14.0	
1⅜	22.5	26	23.6	15″
1⅝	34.6	41	36	
2⅛	69.0	83	72	
2⅝	119	143	125	18″
3⅛	184	220	194	
3⅝	261	312	274	

*This is the minimum elevation required between a condenser coil outlet and a receiver inlet for the total load when receiver is vented to coil outlet header (based on 10 ft of horizontal pipe, 1 valve and 2 elbows).

SUCTION LINE DESIGN

Suction lines are the most critical from a design standpoint. The suction line must be designed to return oil from the evaporator to the compressor under minimum load conditions.

Oil which leaves the compressor and readily passes thru the liquid supply lines to the evaporators is almost completely separated from the refrigerant vapor. In the evaporator a distillation process occurs and continues until an equilibrium point is reached. The result is a mixture of oil and refrigerant rich in liquid refrigerant. Therefore the mixture which is separated from the refrigerant vapor can be returned to the compressor only by entrainment with the returning gas.

Oil entrainment with the return gas in a horizontal line is readily accomplished with normal design velocities. Therefore horizontal lines can and should be run "dead" level.

Suction Risers

Most refrigeration piping systems contain a suction riser. Oil circulating in the system can be returned up the riser only by entrainment with the returning gas. Oil returning up a riser creeps up the inner surface of the pipe. Whether the oil moves up the inner surface is dependent upon the mass velocity of the gas at the wall surface. The larger the pipe diameter, the greater the velocity required at the center of the pipe to maintain a given velocity at the wall surface.

Table 18 shows the minimum tonnages required to insure oil return in upward flow suction risers.

Vertical risers should, therefore, be given special analysis and should be sized for velocities that assure oil return at minimum load. A riser selected on this basis may be smaller in diameter than its branch or than the suction main proper and, therefore, a relatively higher pressure drop may occur in the riser.

This penalty should be taken into account in finding the total suction line pressure drop. The horizontal lines should be sized to keep the total pressure drop within practical limits.

Because modern compressors have capacity reduction features, it is often difficult to maintain the gas velocities required to return oil upward in vertical suction risers. When the suction riser is sized to permit oil return at the minimum operating capacity of the system, the pressure drop in this portion of the line may be too great when operating at full load. If a correctly sized suction riser imposes too great a pressure drop at full load, a double suction riser should be used *(Fig. 58)*.

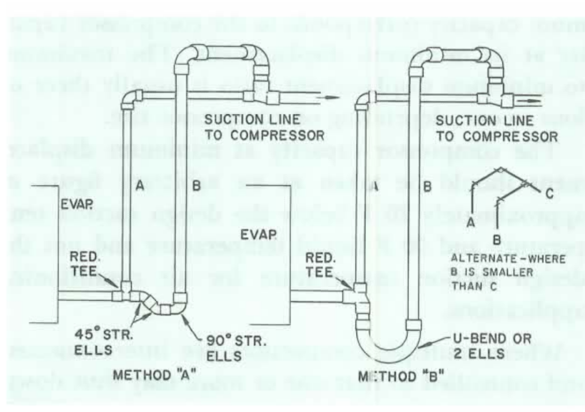


Fig. 58 — Double Suction Riser Construction

—TABLE 18—MINIMUM TONNAGE* FOR OIL ENTRAINMENT UP SUCTION RISERS

		COPPER TUBING—TYPE L											
Pipe OD		½	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝	4⅛
Area, Sq In.		.146	.233	.348	.484	.825	1.256	1.78	3.094	4.77	6.812	9.213	11.97
Ref.	**Sst.**												
R-12†	−40	.061	.110	.182	.27	.54	.91	1.4	2.79	4.78	7.49	10.9	15.1
	−20	.077	.138	.228	.34	.67	1.13	1.75	3.49	5.99	9.36	13.7	19.0
	0	.93	.167	.278	.42	.82	1.38	2.14	4.26	7.32	11.4	16.6	23.2
	20	.112	.201	.332	.50	.97	1.65	2.55	5.1	8.73	13.6	19.9	27.6
	40	.132	.238	.390	.59	1.15	1.94	3.0	6.0	10.3	16.1	23.4	32.6
R-22†	−40	.09	.16	.27	.41	.79	1.34	2.1	4.1	7.1	11.1	16.1	22.4
	−20	.11	.20	.33	.50	.96	1.60	2.5	5.0	8.7	13.5	19.6	27.4
	0	.13	.24	.39	.59	1.2	1.96	3.0	6.1	10.4	16.2	23.6	32.8
	20	.16	.28	.46	.70	1.4	2.30	3.5	7.1	12.1	18.9	27.6	38.1
	40	.18	.33	.54	.81	1.6	2.70	4.1	8.2	14.1	22.0	32.1	44.6
R-500†	−40	.068	.12	.20	.31	.60	1.0	1.6	3.1	5.4	8.4	12.2	16.9
	−20	.086	.16	.26	.39	.75	1.3	2.0	3.9	6.8	10.5	15.3	21.4
	0	.110	.19	.31	.47	.92	1.6	2.4	4.8	8.2	12.8	18.7	26.0
	20	.130	.23	.37	.56	1.1	1.9	2.9	5.7	9.9	15.3	22.4	31.2
	40	.150	.27	.44	.67	1.3	2.2	3.4	6.8	11.6	18.2	26.6	36.8
R-502‡	−60	.053	.10	.16	.24	.46	.78	1.2	2.4	4.1	6.4	9.4	13.0
	−40	.070	.12	.20	.30	.59	1.0	1.5	3.1	5.3	8.3	12.0	16.8
	−20	.084	.15	.25	.38	.74	1.3	1.9	3.8	6.6	10.3	15.0	20.9
	0	.104	.19	.31	.47	.91	1.5	2.4	4.7	8.1	12.7	18.4	25.7
	20	.120	.22	.37	.56	1.1	1.8	2.9	5.7	9.8	15.2	22.2	30.8
	40	.146	.26	.43	.65	1.30	2.2	3.3	6.7	11.4	17.8	26.0	36.1
Pipe OD		½	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝	4⅛

*Minimum tonnage values are based on the indicated saturation temperatures (SST) with 15 F of superheat and 90 F liquid temperature.

†For R-12, R-22 and R-500 reduce or increase table values 1% for 10 F less or more superheat.

‡For R-502 reduce or increase table values 2% for 10 F less or more superheat. For liquid temperatures other than 90 F, multiply the table values by the corresponding factor listed in the following table.

Liquid Temperatures F		50	60	70	80	90	100	110	120	130	140
Correction Factors	**R-12, R-22, R-500**	1.20	1.15	1.10	1.05	1.00	.95	.90	.85	.80	.75
	R-502	1.26	1.20	1.13	1.07	1.00	.94	.88	.82	.76	.70

Liquid temperature equals condensing temperature minus subcooling.

Double Suction Risers

There are applications in which single suction risers may be sized for oil return at minimum load without serious penalty at design load. Where single compressors with capacity control are used, minimum capacity corresponds to the compressor capacity at its minimum displacement. The maximum-to minimum displacement ratio is usually three or four to one, depending on compressor size.

The compressor capacity at minimum displacement should be taken at an arbitrary figure of approximately 20 F below the design suction temperature and 90 F liquid temperature and not the design suction temperature for air conditioning applications.

Where multiple compressors are interconnected and controlled so that one or more may shut down while another continues to operate, the ratio of maximum to minimum displacement becomes much greater. In this case a double suction riser may be necessary for good operating economy at design load. The sizing and operation of a double suction riser is described as follows:

1. In *Fig. 58* the minimum load riser indicated by *A* is sized so that it returns oil at the minimum possible load.
2. The second riser *B* which is usually larger than riser *A* is sized so that the parallel pressure drop thru both risers at full load is satisfactory, providing this assures oil return at full load.
3. A trap is introduced between the two risers as shown in *Fig. 58*. During partial load operation when the gas velocity is not sufficient to return oil through both risers, the trap gradually fills with oil until the second riser *B* is sealed off. When this occurs, the gas travels up riser *A* only and has enough velocity to carry oil along with it back into the horizontal suction main.

The fittings at the bottom of the riser must be close coupled so that the oil holding capacity of the trap is limited to a minimum. If this is not done, the trap can accumulate enough oil on partial load operation to seriously lower the compressor crankcase oil level. Also, larger slug-backs of oil to the compressor occur when the trap clears out on increased load operation. *Fig. 58* shows that the larger riser *B* forms an inverted loop and enters the horizontal suction line from the top. The purpose of this loop is to prevent oil drainage into this riser which may be "idle" during partial load operation.

Example 4 — Determination of Riser Size — (Single Riser)

Given:

Refrigerant 502 system
Refrigeration load — 20.0 tons
Condensing temperature — 105 F, 15 F subcooling
Suction temperature — 0° F, 15 F superheat
Minimum load — 4 tons
(One of four cylinders operating at — 20 F SST, 15 F superheat, 90 F liquid temperature.)
Type L copper tubing
Height of riser — 10 ft (See *Fig. 58A* for general arrangement)
Allowable pressure drop in suction line is 2 F

Find:

Size of riser
Suction line pressure drop

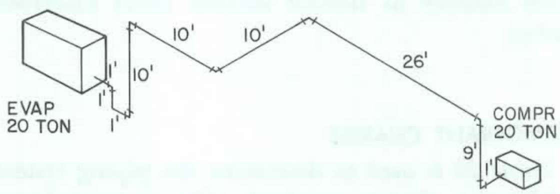


FIG. 58A — SUCTION LINE LAYOUT

Solution:

1. Estimated total equivalent length equals $1\frac{1}{2}$ times the actual lineal length. $1.5 \times 70 = 105$ ft. From *Chart 10,* a $2\frac{5}{8}$ in. line will carry 20 tons for 165 ft at a pressure drop of 2 F with 15 F subcooling.
2. The actual equivalent length of the suction line is determined in two parts, as follows:
   a. Riser section at evaporator — 10 ft plus one $2\frac{5}{8}$ in. ell at 4.1 ft. = 14.1 ft.
   b. Balance of suction line system — 60 ft plus seven $2\frac{5}{8}$ in. ells at 4.1 ft = 88.7 ft total equivalent length = 102.8 ft.
3. From *Table 18,* R-502, the minimum load for returning oil up a $2\frac{5}{8}$ in. riser at −20 F suction is $6.6 \times 1.00 =$ 6.6 tons. This is greater than the minimum load. Thus, the riser must be sized at $2\frac{1}{8}$ in. which will return oil down to $3.8 \times 1.00 = 3.8$ tons.
4. Determine if the $2\frac{1}{8}$ in. riser is adequate to carry full load at 0°F suction without exceeding the specified pressure drop. The equivalent length of the riser is: 10 ft + one $2\frac{1}{8}$ in. ell at 3.3 ft = 13.3 ft. *From Chart 10,* the $2\frac{1}{8}$ in. riser will carry 20 tons for 58 ft at a pressure drop of 2 F. The loss in the riser for 13.3 ft:

$$= \frac{13.3 \text{ ft}}{58.0 \text{ ft}} \times 2 \text{ F} = .46 \text{ F}$$

The loss in the balance of the suction line:

$$= \frac{88.7 \text{ ft}}{165 \text{ ft}} \times 2 \text{ F} = 1.07 \text{ F}$$

The total loss = .46 + 1.07 = 1.53 F which is within the 2 F specified.

Example 4A — Determination of Riser Size — (Double Riser)

Given:

Refrigerant 12 system
Refrigeration load — 98.5 tons
Condensing temperature — 105 F, no subcooling
Suction temperature — 40 F, 15 F superheat
Minimum load — 7.0 tons
(Two of 16 cylinders operating at 20 F SST, 15 F superheat, 105 F liquid temperature.)
Type L copper tubing
Height of riser — 10 ft (*See Fig. 58A* for general arrangement)
Allowable pressure drop in suction line is 2 F

Find:

Size of Riser
Suction line pressure drop

Solution:

1. Estimated total equivalent length equals $1\frac{1}{2}$ times the actual lineal length $1.5 \times 70 = 105$ ft. From *Chart 7,* a $4\frac{1}{8}$ in. line will carry 98.5 tons for 110 ft at a pressure drop of 2 F with no subcooling.
2. The actual equivalent length of the suction line equals 70 ft plus eight $4\frac{1}{8}$ in. ells at 6.7 ft = 123.6 ft. This is greater than 110 ft which gives a 2 F loss for the $4\frac{1}{8}$ in. size. Therefore, select a $5\frac{1}{8}$ in. line which will carry 98.5 tons for 340 ft at a pressure drop 2 F.
3. The actual equivalent length of the suction line is determined in two parts, as follows:
   a. Riser section at evaporator — 10 ft + one $5\frac{1}{8}$ in. ell at 8.2 ft = 18.2 ft.
   b. Balance of suction line system — 60 ft + seven $5\frac{1}{8}$ in. ells at 8.2 ft = 117.4 ft. Total equivalent length = 135.6 ft.
4. From *Table 18,* R-12, a $2\frac{1}{8}$ in. riser will return oil when carrying $5.1 \times .925 = 4.7$ tons at 20 F suction temperature.
5. Determine if the $2\frac{1}{8}$ in. riser is adequate to carry full load at 40 F suction without exceeding the specified pressure drop. The equivalent length of the riser is 10 ft + one $2\frac{1}{8}$ in. ell at 3.3 ft = 13.3 ft. The pressure drop in the balance of the suction line is:

$$\frac{117.4 \text{ ft}}{340 \text{ ft}} \times 2 \text{ F} = 69 \text{ F}$$

The pressure drop allowed for the riser is then 2 F − .69 F or 1.31 F. The chart equivalent length of a $2\frac{1}{8}$ in. line having a pressure drop or 1.31 F is:

$$\frac{2 \text{ F}}{1.31 \text{ F}} \times 13.3 \text{ ft} = 20 \text{ ft}$$

Chart 7 at 20 ft shows the $2\frac{1}{8}$ in. line is only capable of carrying 42 tons. Thus a second riser is necessary.

6. Fig. 58B illustrates a recommended arrangement of double suction risers. Riser A size would be the $2\frac{1}{8}$ in. selected in Step 4. Riser B must be sized by a "cut and try" method. As a first trial, size Riser B one size smaller than the selected suction line of $4\frac{1}{8}$ in. Equivalent lengths for Risers A and B are as follows:

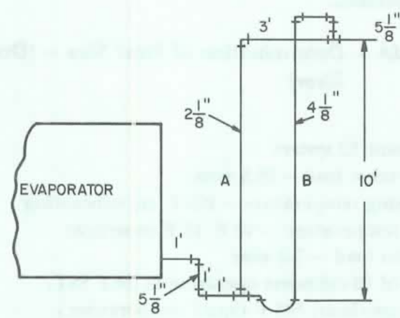


FIG. 58B—DOUBLE SUCTION RISER

Riser A	
Equivalent pipe lengths	
One $2\frac{1}{8}$ in. ell	3.3
One $2\frac{1}{8}$ in. branch tee	10.0
One $5\frac{1}{8}$ in. reducing tee	13.0
Actual pipe length	13.0
Total equivalent length	39.3 ft

Riser B	
Equivalent pipe lengths	
Three $4\frac{1}{8}$ in. ells	20.1
One $5\frac{1}{8}$ in. reducing tee	12.0
One $4\frac{1}{8}$ in. branch tee	21.0
One $4\frac{1}{8}$ in. "U"	17.0
Actual pipe length	13.0
Total equivalent length	83.1 ft

Pressure drop available for the riser is 1.31 F from Step 5. For Riser A, the chart equivalent length of a $2\frac{1}{8}$ in. line is:

$$\frac{2\text{ F}}{1.31\text{ F}} \times 39.3\text{ ft} = 60.0\text{ ft}$$

For Riser B, the chart equivalent length of a $4\frac{1}{8}$ in. line is:

$$\frac{2\text{ F}}{1.31\text{ F}} \times 83.1\text{ ft} = 127.0\text{ ft}$$

From *Chart 7*, the $2\frac{1}{8}$ in. line is capable of carrying 23 tons and the $4\frac{1}{8}$ in. line is capable of carrying 92 tons. The combined capacity is 115 tons, which is acceptable. Riser pressure drop is equal to:

$$\frac{98.5\text{ tons}}{115\text{ tons}} \times 1.31\text{ F} = 1.12\text{ F}$$

Total pressure drop = 1.12 F + .69 F = 1.81 F

DISCHARGE (HOT GAS) LINE DESIGN

The hot gas line should be sized for a practical pressure drop. The effect of pressure drop is shown in *Table 16, page 43.*

Discharge Risers

Even though a low loss is desired in the hot gas line, the line should be sized so that refrigerant gas velocities are able to carry along entrained oil. In the usual application this is not a problem. However, where multiple compressors are used with capacity control, hot gas risers must be sized to carry oil at minimum loading.

Table 19 shows the minimum tonnages required to insure oil return in upward flow discharge risers. Friction drop in the risers in degrees F per 100 ft equivalent length is also included.

Double Discharge Risers

Sometimes in installations of multiple compressors having capacity control a vertical hot gas line sized to entrain oil at minimum load has an excessive pressure drop at maximum load. In such a case a double gas riser may be used in the same manner as it is used in a suction line. *Fig. 59* shows the double riser principle applied to a hot gas line.

Sizing of double hot gas risers is made in the same manner as double suction risers described earlier.

REFRIGERANT CHARGE

Table 20 is used to determine the piping system refrigerant charge required. The system charge should be equal to the sum of the charges in the refrigerant lines, compressor, evaporator, condenser and receiver (minimum operating charge).

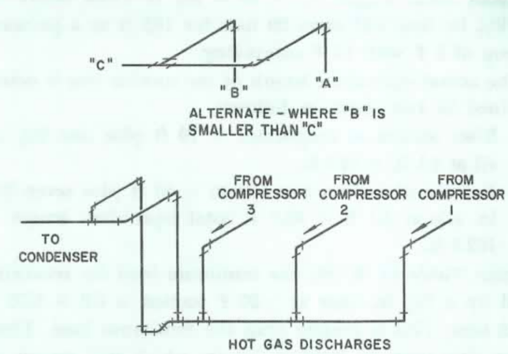


FIG. 59 — DOUBLE HOT GAS RISER

TABLE 19—MINIMUM TONNAGE FOR OIL ENTRAINMENT UP HOT GAS RISERS

COPPER TUBING—TYPE L

Pipe OD		½	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝	4⅛
Area Sq In.		.146	.233	.348	.484	.825	1.256	1.78	3.094	4.77	6.812	9.213	11.97
Ref.	Sct.												
R-12*	80	.17	.31	.51	.77	1.51	2.54	3.93	7.84	13.5	21.0	30.7	42.6
	90	.17	.31	.51	.77	1.51	2.54	3.92	7.84	13.5	21.0	30.7	42.6
	100	.17	.31	.51	.77	1.51	2.54	3.92	7.84	13.5	21.0	30.7	42.6
	110	.17	.31	.51	.77	1.50	2.53	3.90	7.81	13.4	20.9	30.5	42.6
	120	.17	.30	.50	.75	1.47	2.49	3.84	7.66	13.2	20.6	30.0	41.6
	130	.17	.30	.49	.72	1.45	2.44	3.77	7.54	12.9	20.3	29.4	40.8
	140	.16	.28	.47	.71	1.38	2.33	3.61	7.20	12.4	19.4	28.2	39.9
	150	.15	.28	.47	.71	1.38	2.33	3.60	7.18	12.3	19.3	28.1	38.9
	160	.15	.27	.45	.68	1.34	2.24	3.47	6.95	11.9	18.6	27.1	37.6
R-22*	80	.23	.42	.69	1.04	2.0	3.4	5.3	10.6	18.2	28.3	41.5	57.5
	90	.23	.42	.69	1.04	2.0	3.4	5.3	10.6	18.2	28.2	41.3	57.3
	100	.23	.42	.69	1.03	2.0	3.4	5.3	10.5	18.0	28.1	41.0	56.7
	110	.23	.41	.67	1.02	2.0	3.4	5.2	10.4	17.9	27.9	40.8	56.5
	120	.22	.40	.66	1.00	2.0	3.3	5.1	10.2	17.5	27.4	39.9	55.4
	130	.22	.39	.64	.98	1.9	3.2	5.0	10.0	17.2	26.8	39.0	54.0
	140	.21	.38	.63	.96	1.9	3.2	4.9	9.7	16.7	26.1	38.0	52.6
	150	.21	.37	.61	.93	1.8	3.1	4.7	9.4	16.2	25.2	36.8	51.0
	160	.20	.36	.59	.89	1.7	2.9	4.5	9.0	15.5	24.1	35.2	48.8
R-500*	80	.20	.36	.59	.89	1.73	2.92	4.51	9.0	15.5	24.2	35.4	49.0
	90	.20	.35	.58	.88	1.73	2.86	4.49	8.9	15.4	24.0	35.0	48.5
	100	.20	.35	.58	.88	1.73	2.86	4.47	8.8	15.3	23.8	34.9	48.2
	110	.20	.35	.57	.87	1.70	2.86	4.45	8.7	15.2	23.7	34.7	48.0
	120	.19	.34	.56	.86	1.66	2.82	4.44	8.7	15.0	23.3	34.1	47.3
	130	.19	.34	.56	.85	1.64	2.78	4.29	8.6	14.7	23.0	33.6	46.5
	140	.18	.33	.54	.83	1.61	2.71	4.20	8.4	14.4	22.5	32.8	45.5
R-502†	80	.18	.32	.53	.80	1.55	2.7	4.1	8.2	14.1	21.9	32.5	44.3
	90	.17	.31	.51	.77	1.49	2.52	3.92	7.8	13.4	20.9	30.5	42.3
	100	.165	.30	.50	.74	1.44	2.45	3.8	7.55	13.0	20.2	29.5	40.9
	110	.160	.29	.48	.72	1.41	2.38	3.71	7.35	12.7	19.7	28.7	39.8
	120	.154	.28	.46	.69	1.33	2.26	3.52	7.0	12.4	18.7	27.3	37.9
	130	.145	.26	.43	.65	1.27	2.14	3.34	6.62	11.4	17.8	25.9	35.9
	140	.135	.24	.40	.61	1.18	1.98	3.08	6.15	10.6	16.4	24.0	33.3
Pipe OD		½	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝	4⅛

SCT—Saturated Condensing Temperature.

*Minimum tonnages are based on a saturated suction temperature of +20 F with 15 F of superheat at the indicated saturated condensing temperatures with 15 F subcooling and actual discharge temperature based on 70% compressor efficiency.

For suction temperatures other than the 20 F, multiply the table values by the following factors:

Sat. suct temperature	−40	−20	0	+20	+40
Correction factor	.85	.90	.95	1.0	1.06

†Minimum tonnages are based on a saturated suction temperature of −20 F. All other conditions are the same as above.

For suction temperatures other than the −20 F, mutiply the table values by the following factors:

Sat. suct temperatures	−60	−40	−20	0	+20	+40
Correction factor	.87	.94	1.0	1.08	1.15	1.21

TABLE 20—FLUID WEIGHT OF REFRIGERANT IN PIPING

(Lb/10 ft of length)

PIPE SIZE*		SUCTION LINES 40 F SAT. SUCT TEMP				LIQUID LINES 100 F TEMP				HOT GAS LINES 100 F SAT. COND TEMP			
Copper OD In.	Steel Nom. In.	R-12	R-500	R-502	R-22	R-12	R-500	R-502	R-22	R-12	R-500	R-502	R-22
½	⅜	.013	.013	.022	.016	.80	.70	.75	.72	.032	.032	.044	.047
⅝	½	.021	.02	.036	.025	1.28	1.13	1.20	1.15	.051	.052	.071	.075
⅞	¾	.043	.042	.075	.051	2.65	2.33	2.48	2.40	.105	.11	.148	.16
1⅛	1	.073	.072	.128	.087	4.52	3.98	4.23	4.09	.18	.18	.252	.27
1⅜	1¼	.110	.11	.195	.13	6.87	6.06	6.44	6.22	.27	.28	.384	.40
1⅝	1½	.16	.15	.276	.19	9.74	8.56	9.12	8.81	.39	.39	.543	.57
2⅛	2	.27	.27	.481	.33	16.9	14.9	15.9	15.3	.67	.69	.945	.99
2⅝	2½	.42	.41	.74	.51	26.1	23.0	24.5	23.6	1.04	1.1	1.46	1.5
3⅛	3	.60	.59	1.06	.72	37.3	32.9	35.0	33.7	1.5	1.5	2.08	2.2
3⅝	3½	.81	.80	1.43	.98	50.5	44.3	47.3	45.6	2.0	2.0	2.81	3.0
4⅛	4	1.05	1.04	1.86	1.27	65.5	57.6	61.3	59.3	2.6	2.7	3.66	3.8
5⅛	5	1.64	1.62	2.90	1.98	102.0	90.0	95.6	93.0	4.1	4.1	5.70	6.0
6⅛	6	2.36	2.33	4.17	2.84	147.0	130.0	137.5	133.0	5.8	6.0	8.20	8.6
8⅛	8	4.11	4.06	7.40	4.96	257.0	227.0	244.0	232.0	10.2	10.4	14.50	15.0

To Correct for Temperatures Other Than Above, Multiply by the Following Factors:

REFRIGERANT	SUCT LINE—SAT. TEMP F					LIQUID LINE—SAT. TEMP F					HOT GAS LINE—SAT. TEMP F				
	50	30	10	−10	−30	40	60	80	100	120	80	90	100	110	120
12	1.18	.84	.59	.39	.26	1.09	1.06	1.03	1.00	.97	.75	.87	1.00	1.15	1.33
500	1.18	.84	.58	.39	.26	1.10	1.07	1.04	1.00	.96	.74	.87	1.00	1.15	1.33
502	1.17	.85	.59	.41	.27	1.12	1.09	1.05	1.00	.95	.73	.85	1.00	1.16	1.37
22	1.18	.84	.58	.39	.25	1.11	1.08	1.04	1.00	.98	.74	.86	1.00	1.16	1.35

*Refrigerants 12, 22, 500 and 502 weights are for OD sizes of Type L copper pipe.

3

REFRIGERANT PIPING LAYOUT

EVAPORATORS

Suction Line Loops

Evaporator suction lines should be laid out to accomplish the following objectives:

1. Prevent liquid refrigerant from draining into the compressor during shutdown.
2. Prevent oil in an active evaporator from draining into an idle evaporator.

This can be done by using piping loops in the lines connecting the evaporator, the compressor and the condenser. Standard arrangements of suction line loops based on standard piping practices are illustrated in *Fig. 60*.

Figure 60a shows the *compressor located below a single evaporator*. An inverted loop rising to the top of the evaporator should be made in the suction line to prevent liquid refrigerant from draining into the compressor during shutdown.

A *single evaporator below the compressor* is illustrated in *Fig. 60b*. The inverted loop in the suction line is unnecessary since the evaporator traps all liquid refrigerant.

Figure 60c shows *multiple evaporators on different floor levels with the compressor below*. Each individual suction line should be looped to the top of the evaporator before being connected into the suction main to prevent liquid from draining into the compressor during shutdown.

Figure 60d illustrates *multiple evaporators stacked on the same floor level* or may represent a two-circuit single coil operated from one liquid solenoid valve with the compressor located below the evaporator. In this arrangement it is possible to use one loop to serve the purpose.

Where coil banks on the same floor level have separate liquid solenoid valves feeding each coil, a separate suction riser is required from each coil, similar to the arrangements in *Fig. 60c and 60e* for best oil return performance. Where separate suction risers are not possible, use the arrangement shown in *Fig. 60f*.

Figure 60g shows *multiple evaporators located on the same level and the compressor located below the evaporators*. Each suction line is brought upward and looped into the top of the common suction line. The alternate arrangement shows individual suction lines out of each evaporator dropping down into a common suction header which then rises in a single loop to the top of the coils before going down to the compressor. An alternate arrangement is shown in *Fig. 60h* for cases where the *compressor is above the evaporator*.

When automatic compressor pumpdown control is used, evaporators are automatically kept free of liquid by the pumpdown operation and, therefore, evaporators located above the compressor can be free-draining to the compressor without protective loops.

The small trap shown in the suction lines immediately after the coil suction outlet is recommended to prevent erratic operation of the thermal expansion valve. The expansion valve bulb is located in the suction line between the coil and the trap. The trap drains the liquid from under the expansion valve bulb during compressor shutdown, thus preventing erratic operation of the valve when the compressor starts up again. A trap is required only when straight runs or risers are encountered in the suction line leaving the coil outlet. A trap is not required when the suction line from the coil outlet drops to the compressor or suction header immediately after the expansion valve bulb.

Suction lines should be designed so that oil from an active evaporator does not drain into an idle one. *Fig. 60e* shows *multiple evaporators on different floor levels and the compressor above the evaporators*. Each suction line is brought upward and looped into the top of the common suction line if its size is equal to the main. Otherwise it may be brought into the side of the main. The loop prevents oil from draining down into either coil that may be inactive.

Figure 60f shows *multiple evaporators stacked with the compressor above the evaporators*. Oil is prevented from draining into the lowest evaporator because the common suction line drops below the outlet of the lowest evaporator before entering the suction riser.

If evaporators must be located both above and below a common suction line, the lines are piped as illustrated in *Figs. 60a and 60b*, with (a) piping for the evaporator above the common suction line and (b) piping for the evaporator below the common suction line.

Multiple Circuit Coils

All but the smallest coils are arranged with multiple circuits. The length and number of circuits are determined by the type of application. Multiple circuit coils are supplied with refrigerant thru a distributor which regulates the refrigerant distribution evenly among the circuits. Direct expansion coils can be located in any position, provided proper

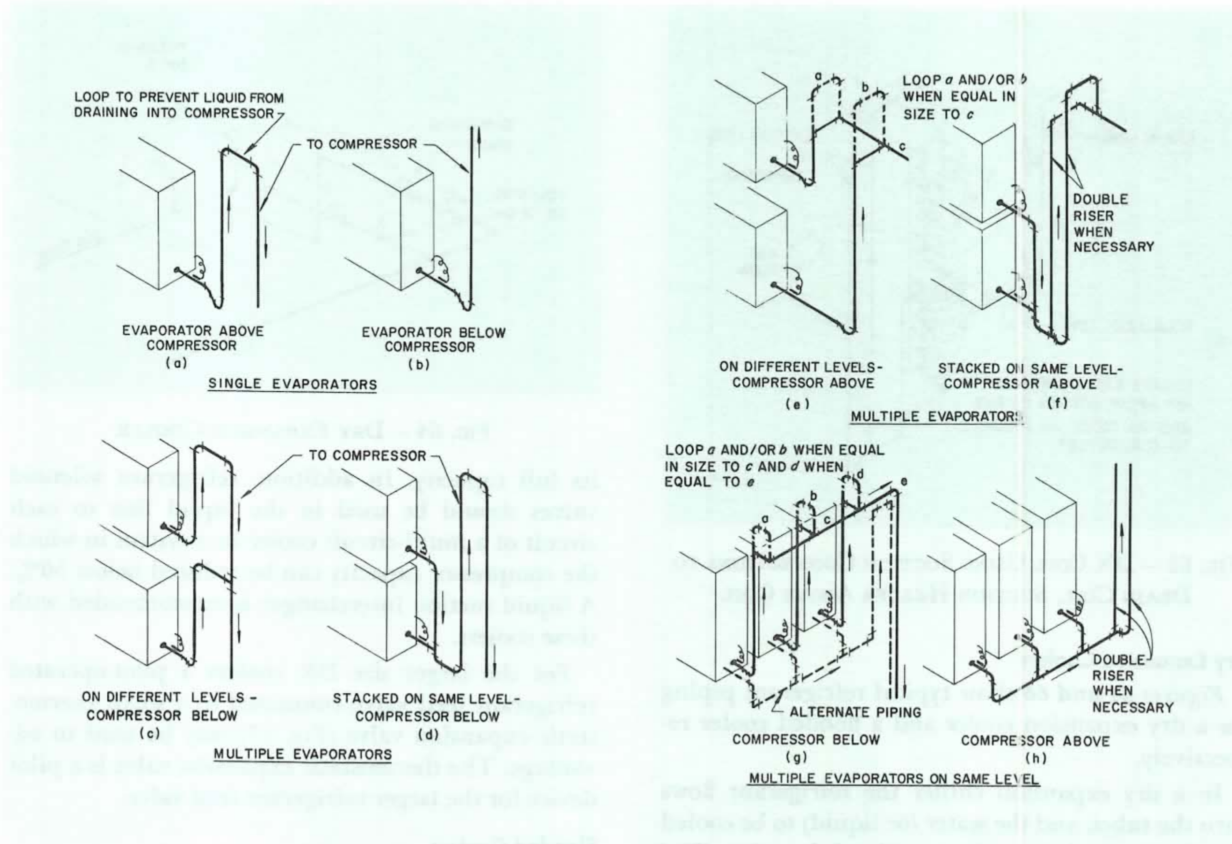


FIG. 60 — STANDARD ARRANGEMENTS OF SUCTION LINE LOOPS (ONE-CIRCUIT COILS SHOWN)

refrigerant distribution and continuous oil removal facilities are provided.

In general the suction line piping principles shown in *Fig. 60* should be employed to assure proper expansion valve operation, oil return and compressor protection.

Figures 61 and 62 show direct expansion air coil piping arrangements in which the suction connections drain the coil headers effectively. *Fig. 61* shows individual suction outlets joining into a common suction header below the coil level. *Fig. 62* illustrates an alternate method of bringing up each suction line and looping it into the common line. The expansion valve equalizing lines are connected into the suction header downstream of the expansion valve thermal bulb.

Figure 63 illustrates the use of a coil having connections at the top or in the middle of each coil header, and piped so that this connection does not drain the evaporator. In this case oil may become trapped in the coil. The figure shows oil drain lines from connections supplied for this purpose. The drain lines extend from the suction connection at the bottom end of each coil header to the common suction header below the coil level.

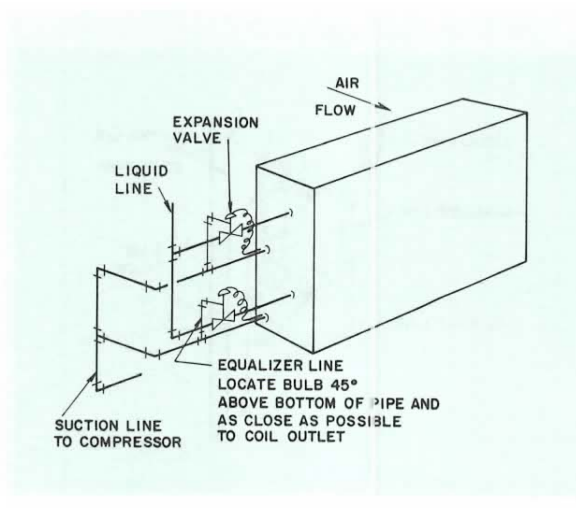


FIG. 61 — DX COIL USING SUCTION CONNECTIONS TO DRAIN COIL, SUCTION HEADER BELOW COIL

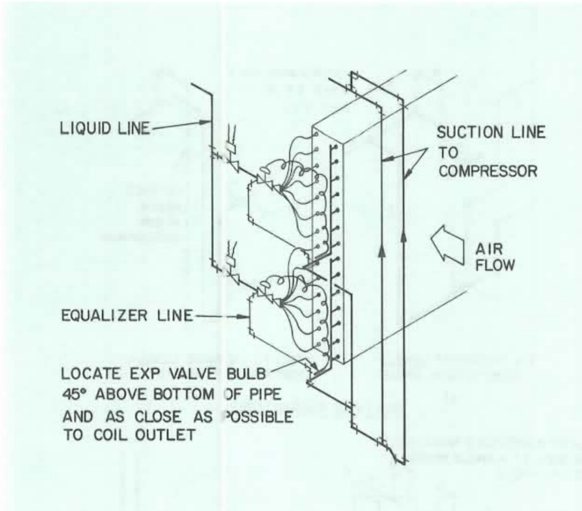


FIG. 62 — DX COIL USING SUCTION CONNECTIONS TO DRAIN COIL, SUCTION HEADER ABOVE COIL

Dry Expansion Coolers

Figures 64 and 66 show typical refrigerant piping for a dry expansion cooler and a flooded cooler respectively.

In a dry expansion chiller the refrigerant flows thru the tubes, and the water (or liquid) to be cooled flows transversely over the outside of the tubes. The water or liquid flow is guided by vertical baffles. Multi-circuit coolers should be used in systems in which the compressor capacity can be reduced below 50%. This is recommended since oil cannot be properly returned and good thermal valve control cannot be expected below this minimum loading per circuit.

It is also recommended that the minimum capacity of a single circuit should be not less than 50% of its full capacity. In addition, refrigerant solenoid valves should be used in the liquid line to each circuit of a multi-circuit cooler in a system in which the compressor capacity can be reduced below 50%. A liquid suction interchanger is recommended with these coolers.

For the larger size DX coolers a pilot-operated refrigerant feed valve connected to a small thermostatic expansion valve *(Fig. 65)* may be used to advantage. The thermostatic expansion valve is a pilot device for the larger refrigerant feed valve.

Flooded Coolers

In a flooded cooler the refrigerant surrounds the tubes in the shell, and water or liquid to be cooled flows thru the tubes in one or more passes, depending on the baffle arrangement.

Flooded coolers require a continuous liquid bleed line from some point below the liquid level in the cooler shell to the suction line. This continuous bleed of refrigerant liquid and oil assures the required return of oil to the compressor. It is usually

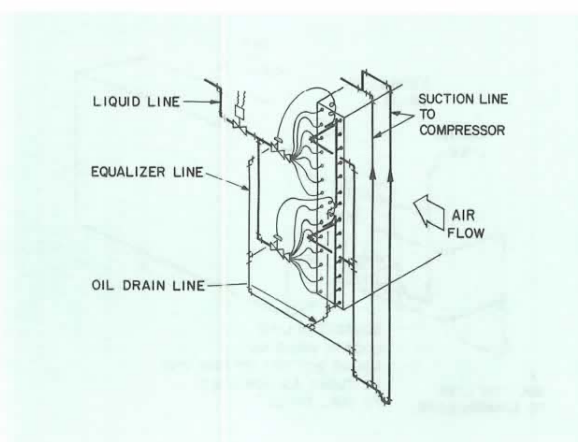


FIG. 63 — DX COIL USING OIL RETURN DRAIN CONNECTIONS TO DRAIN OIL

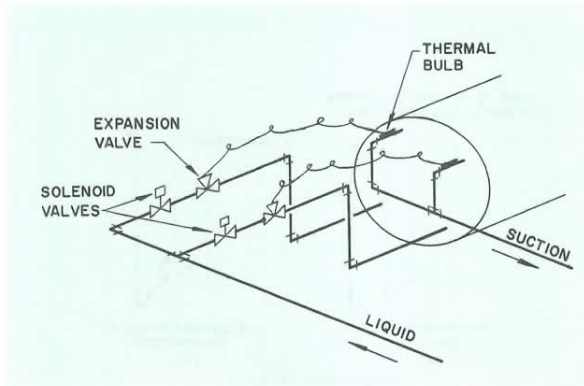


FIG. 64 — DRY EXPANSION COOLER

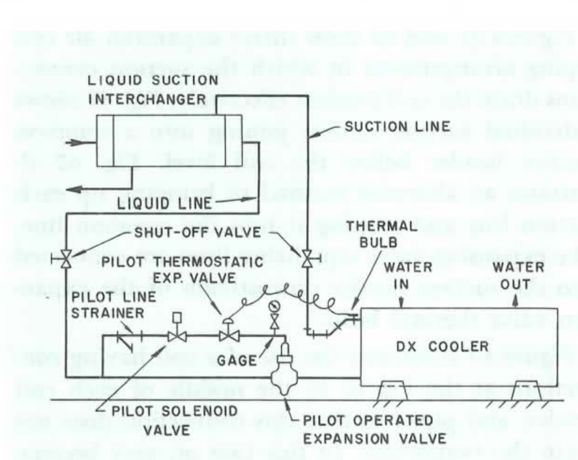


FIG. 65 — HOOKUP FOR LARGE DX

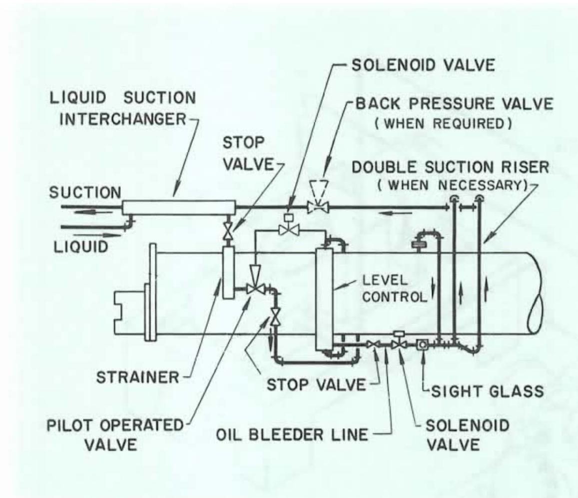


Fig. 66 — Flooded Cooler

drained into the suction line so that the oil can be returned to the cooler with the suction gas. This drain line should be equipped with a hand shut-off valve, a solenoid valve and a sight glass. The solenoid valve should be wired into the control circuit in such a manner that it closes when the compressor stops.

A liquid suction interchanger, installed close to the cooler, is required to evaporate any liquid refrigerant from the refrigerant oil mixture which is continuously bled into the suction line.

Since flooded coolers frequently operate at light loads, double suction risers are often necessary.

To avoid freeze-up the water supply to a flooded cooler should never be throttled and should never bypass the cooler.

COMPRESSORS

Suction Piping

Suction piping of parallel compressors should be designed so that all compressors run at the same suction pressure and so that oil is returned to the running compressors in equal proportions. To insure maintenance of proper oil levels, compressors of unequal sizes may be erected on foundations at different elevations so that the recommended crankcase operating oil level is maintained at each compressor.

All suction lines are brought into a common suction header which is run full size and level above the compressor suction inlets. Branch suction line take-offs to the compressors are from the side of the header and should be the same size as the header. No reduction is made in the branch suction lines to the compressors until the vertical drop is reached.

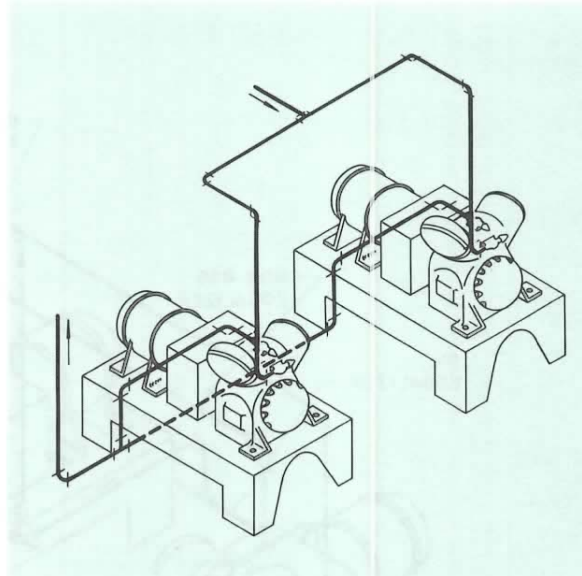


Fig. 67 — Layout of Suction and Hot Gas Lines for Multiple Compressor Operation

This allows the branch line to return oil proportionally to each of the operating compressors.

Figure 67 shows suction and hot gas header arrangements for two compressors operating in parallel.

Discharge Piping

The branch hot gas lines from the compressors are connected into a common header. This hot gas header is run at a level below that of the compressor discharge connections which, for convenience, is often at the floor. This is equivalent to the hot gas loop for the single compressor shown in *Fig. 68*.

The hot gas loop accomplishes two functions:

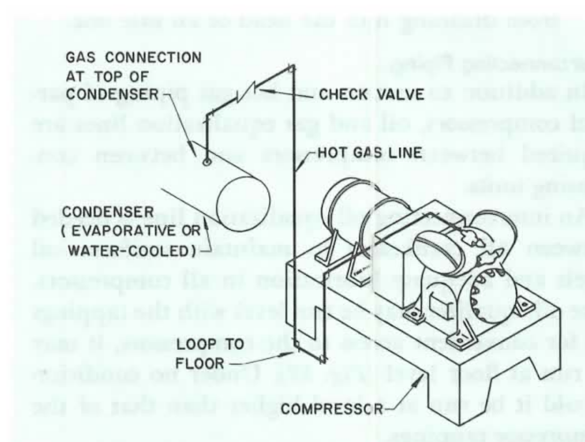


Fig. 68 — Hot Gas Loop

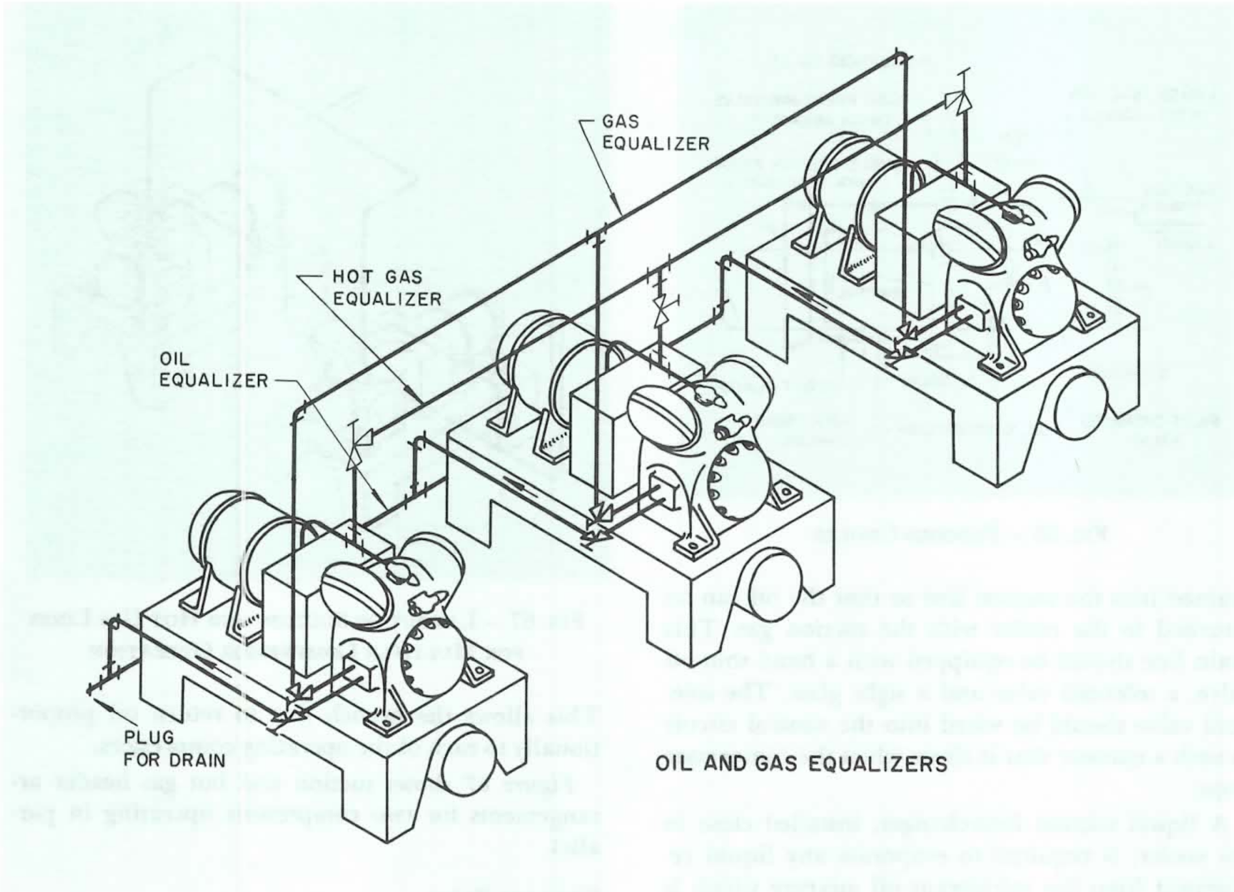


FIG. 69 – INTERCONNECTING PIPING FOR MULTIPLE CONDENSING UNITS

1. It prevents gas, which may condense in the hot gas line during the off cycle, from draining back into the heads of the compressors. This eliminates compressor damage on start-up.
2. It prevents oil, which leaves one compressor, from draining into the head of an idle one.

Interconnecting Piping

In addition to suction and hot gas piping of parallel compressors, oil and gas equalization lines are required between compressors and between condensing units.

An interconnecting oil equalization line is needed between all crankcases to maintain uniform oil levels and adequate lubrication in all compressors. The oil equalizer may be run level with the tappings or, for convenient access to the compressors, it may be run at floor level *(Fig. 69)*. Under no condition should it be run at a level higher than that of the compressor tappings.

Ordinarily, proper equalization takes place only if a gas equalizing line is installed above the compressor crankcase oil line. This line may be run level with the tappings, or may be raised to allow head room for convenient access. It should be piped level and supported as required to prevent traps from forming.

Shut-off valves should be installed in both lines so that any one machine may be isolated for repair without shutting down the entire system. Both lines should be the same size as the tappings on the largest compressor. Neither line should be run directly from one crankcase into another without forming a U-bend or hairpin to absorb vibration.

When multiple condensing units are interconnected as shown in *Fig. 69,* it is necessary to equalize the pressure in the condensers to prevent hot gas from blowing thru one of the condensers and into the liquid line. To do this a hot gas equalizer line is installed as shown. If the piping is looped as shown, vibration should not be a problem. The equalizer line between units must be the same size as the largest hot gas line.

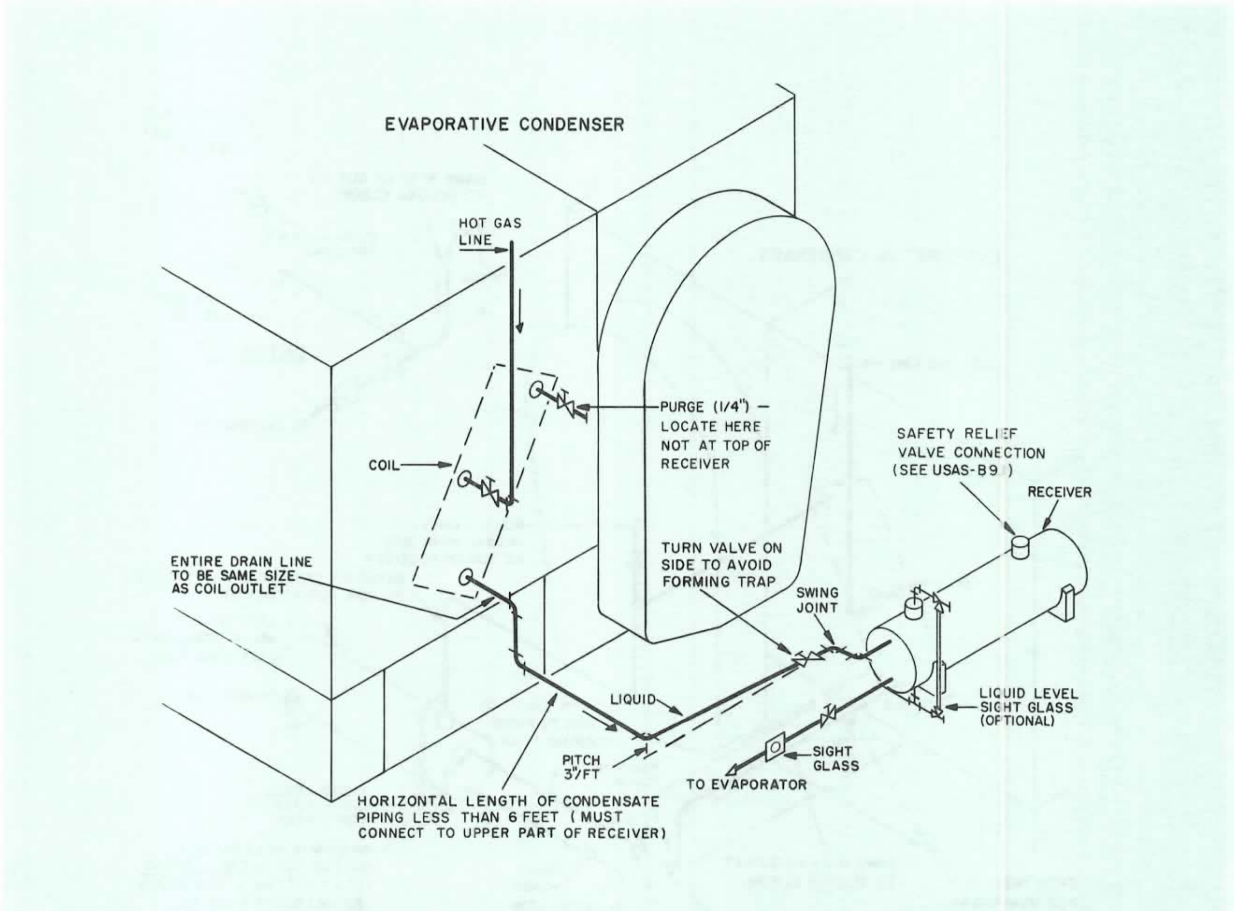


Fig. 70 — Hot Gas and Liquid Piping, Single Coil Unit Without Receiver Vent

CONDENSERS

Liquid receivers are often used in systems having evaporative or air-cooled condensers and also with water-cooled condensers where additional liquid storage capacity is required to pump down the system. However, in many water-cooled condenser systems the condenser serves also as a receiver if the total refrigerant in the system does not exceed its storage capacity.

When receivers are used, liquid piping from the condenser to the receiver is designed to allow free drainage of liquid from the condenser at all times. This is possible only if the pressure in the receiver is not allowed to rise to the point where a restriction in flow can occur.

Liquid flow from the condenser to the receiver can be restricted by any of the following conditions:

1. Gas binding of the receiver.
2. Excessive friction drop in the condensate line.
3. Incorrect condensate line design.

The following piping recommendations are made to overcome these difficulties.

Evaporative Condenser to Receiver Piping

Liquid receivers are used on evaporative condensers to accommodate fluctuations in refrigerant liquid level, to maintain a seal, and to provide storage facilities for pumpdown. An equalizing line from the receiver to the condenser is required to relieve gas pressure tending to develop in the receiver. Otherwise liquid hang-up in the condenser due to restricted drainage can occur. The receiver can be vented directly thru the condensate line to the condenser outlet, or by an external equalizer line to the condenser.

Figure 70 shows a single evaporative condenser and receiver vented back thru the condensate drain line to the condensing coil outlet. Such an arrangement is applicable to a close coupled system. A separate vent is not required. However, it is limited to a horizontal length of condensate line of less than

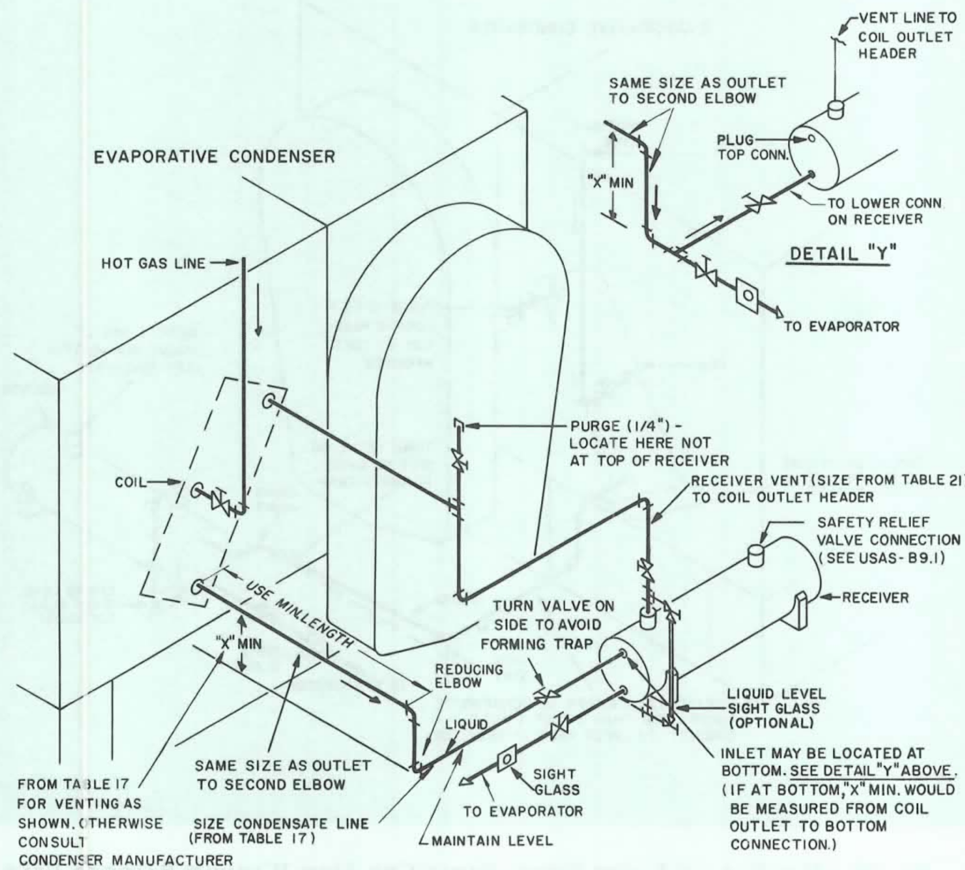


Fig. 71 – Hot Gas and Liquid Piping, Single Coil Unit With Receiver Vent

6 ft. The entire condensate line from the condenser to the receiver is the same size as the coil outlet. The line should be pitched as shown.

Figure 71 shows the refrigerant piping for a single unit with receiver vent. Note that the condensate line from the condenser is the full size of the outlet connection and is not reduced until the second elbow is reached. This arrangement prevents trapping of liquid in the condenser coil.

Table 21 lists recommended sizing of receiver vent lines.

There are some systems in current use without a receiver but it must be recognized that problems can occur which can be avoided if a receiver is used.

Such an arrangement is more critical with respect to refrigerant charge. An overcharged system can waste power and cause a loss of capacity if the overcharge backs up into the condenser. An undercharged system also wastes power and causes a loss of capacity because the evaporator is being fed partially with hot gas. Therefore, if the receiver is omitted, an accurate refrigerant charge must be maintained to assure normal operation.

TABLE 21—RECEIVER VENT LINE SIZING

Receiver to Condenser

VENT LINE SIZE BASED ON TYPE L COPPER TUBING (In. OD)	REFRIGERATION (Tons, Max.)
5/8	to - 40
7/8	40 - 80
1 1/8	80 - 120
1 3/8	Above 120

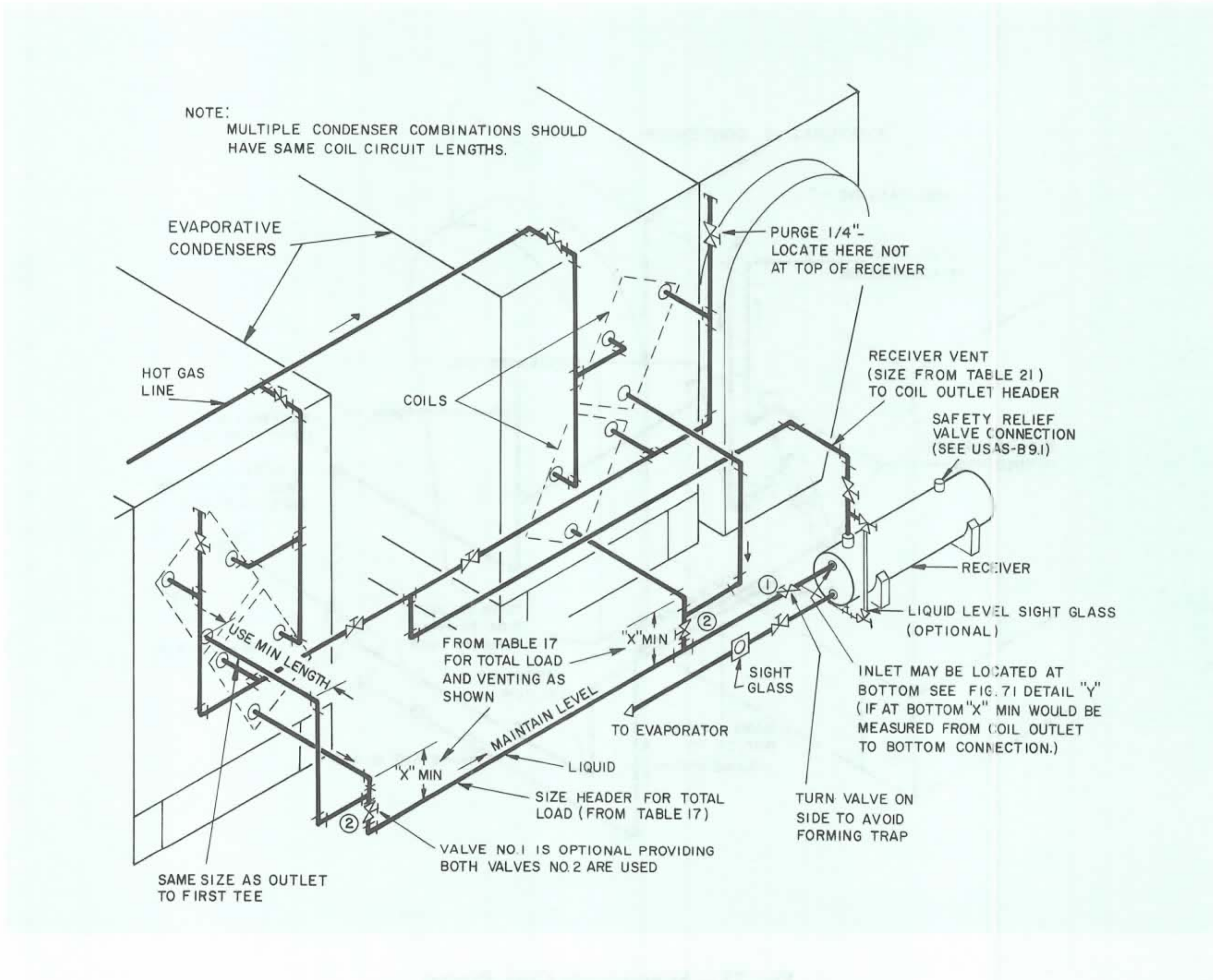


Fig. 72 — Hot Gas and Liquid Piping, Multiple Double Coil Unit

The advantage of such an arrangement is an economic one; equipment cost is lower since receiver and valves are eliminated and the system operating charge is lower if charged accurately.

Figure 72 shows a piping arrangement for multiple units. Note that there are individual hot gas and vent valves for each unit. These valves permit operation of one unit while the other is shut down. These are essential because otherwise the idle unit, at lower pressure, causes hot gas to blow thru the operating unit into the liquid line. Purge cocks are also shown, one for each unit.

The hot gas piping should be such that the pressure in each condenser is substantially the same. To accomplish this the branch connection from the hot gas header into each condenser should be the same size as the condenser coil connection.

Figure 73 shows a subcooling coil piping arrangement. The subcooling coil must be piped between the receiver and the evaporator for best liquid subcooling benefits.

Multiple Shell and Tube Condensers

When two or more shell and tube condensers are applied in parallel in a single system, they should be equalized on the hot gas side and arranged as shown in *Fig. 75*.

The elevation difference between the outlet of the condenser and the horizontal liquid header must be at least 12 in., preferably greater, to prevent gas blowing thru. The bottoms of all condensers should be at the same level to prevent backing liquid into the lowest condenser.

When water-cooled condensers are interconnected as shown, they should be fed from a common water regulating valve, if used.

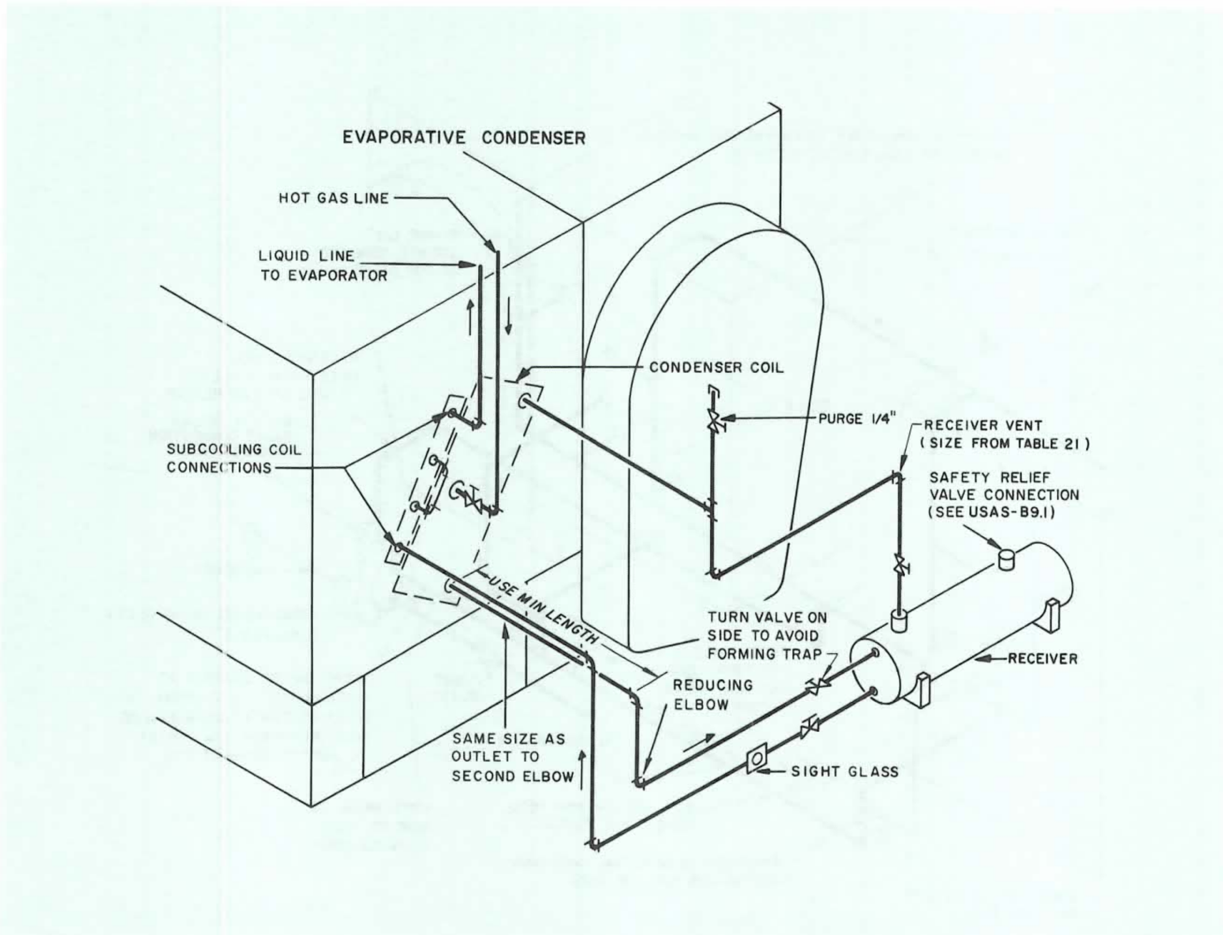


FIG. 73 — SUBCOOLING COIL PIPING

An inverted loop of at least 6 ft is recommended in the liquid line to prevent siphoning of the liquid into the evaporator (or evaporators) during shutdown. Where a liquid line solenoid valve or valves are used, the loop is unnecessary.

Figure 74 shows a similar loop for a single condenser with the evaporator below.

Vibration of Piping

Vibration transmitted thru or generated in refrigerant piping and the objectionable noise which results can be eliminated or greatly minimized by proper design and support of the piping.

The best way to prevent compressor vibration from being transmitted to the piping is to run the suction and discharge lines at least 6 pipe diameters in each of three directions before reaching the first point of support. In this manner the piping can absorb the vibration without being overstressed.

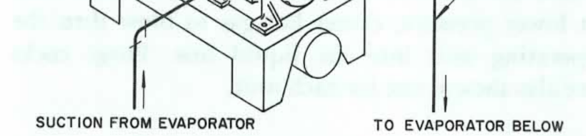


FIG. 74 — LIQUID LINE FROM CONDENSER OR RECEIVER TO EVAPORATOR LOCATED BELOW

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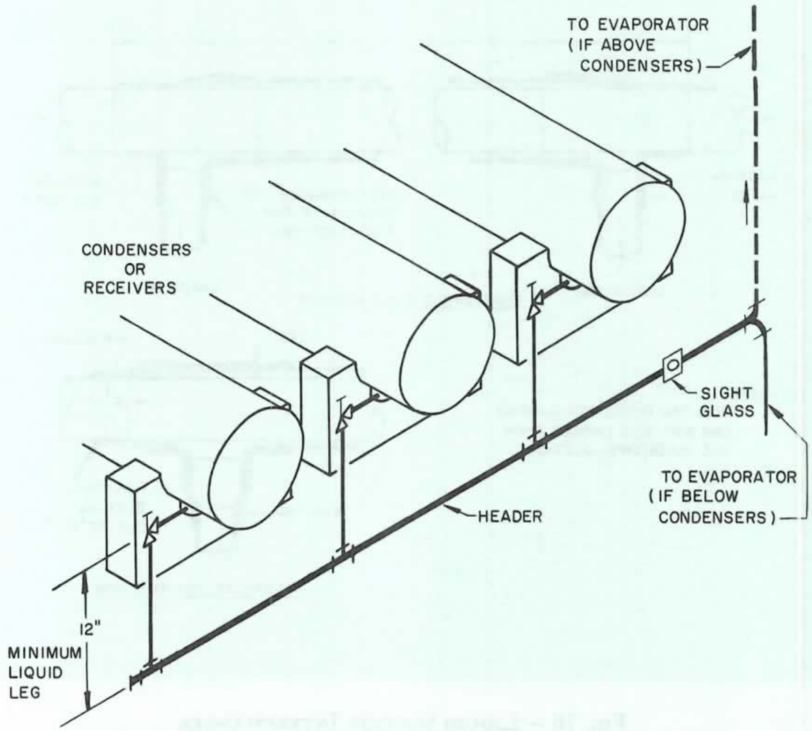


FIG. 75 – LIQUID PIPING TO INSURE CONDENSATE FLOW FROM INTERCONNECTED CONDENSERS

The hot gas loop from the compressor can be attached to the compressor base by means of a bracket if the base is isolated. If there is enough space in the horizontal run of the loop, two brackets are recommended to eliminate excessive rocking movement of the pipe. Brackets should be attached at the point of minimum movement of the compressor assembly. The riser following the loop is supported as close as possible to the compressor.

If the compressor is mounted on a resilient base, the pipe support should have a resilient isolator. The isolator is selected for four times the deflection in the spring support of the compressor base.

See *"Vibration Isolation of Piping Systems"* in *Chapter I* for further discussion of the subject.

REFRIGERANT PIPING ACCESSORIES

LIQUID LINE

Liquid Suction Interchangers

These are devices which subcool the liquid refrigerant and superheat the suction gas. The following describes four reasons for their use and the best location for each application:

1. To subcool the liquid refrigerant to compensate for excessive liquid line pressure drop. Location – near condenser. Liquid suction interchangers are not recommended for single stage applications using Refrigerant 22 because superheating of the suction gas must be limited to avoid compressor overheating. However, where they are used to prevent liquid slop-over to the compressor, superheating of the suction gas should be limited to 20 F above saturation temperature. A liquid suction interchanger so designed to limit the superheat of the suction gas should have a bypass so that operating adjustments may be made.
2. To act as an oil rectifier. Location – near evaporator.
3. To prevent liquid slop-over to the compressor. Location – near evaporator.
4. To increase the effciency of the Refrigerant 12 and 500 cycles. Location – near evaporator to avoid insulation of subcooled liquid line.

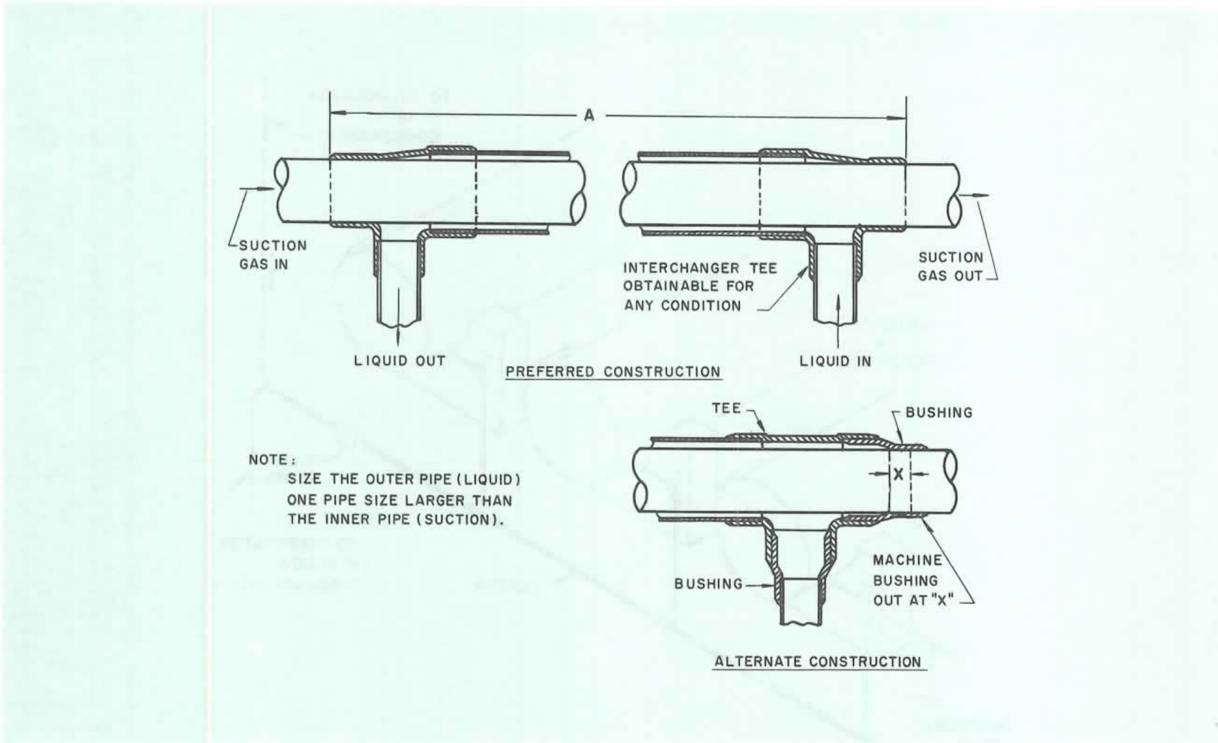


FIG. 76 — LIQUID SUCTION INTERCHANGER

Two common types of liquid suction interchangers are:

1. The *shell and coil* or the *shell and tube exchanger,* suitable for increasing cycle efficiency and for liquid subcooling. This type is usually installed so that the suction outlet drains the shell to prevent oil trapping.
2. The *tube-in-tube interchanger (Figs. 76 and 77),* a preferable type for controlling slop-over caused by erratic expansion valve feed or for "rectifying" lube oil from a refrigerant oil mixture bled from a flooded evaporator.

Excessive superheating of the suction gas must be avoided with heat exchangers since it causes excessive compressor discharge temperatures. Therefore, the amount of liquid subcooling permissible by a liquid suction interchanger is limited to the amount of suction gas superheating that does not cause compressor damage when the gas is compressed to the discharge pressure. Beyond this point additional subcooling should be obtained from other sources.

Charts 23 and 24 are used to determine the length (A) of a *concentric tube-in-tube interchanger (Fig. 76).* The amount of liquid subcooling available is

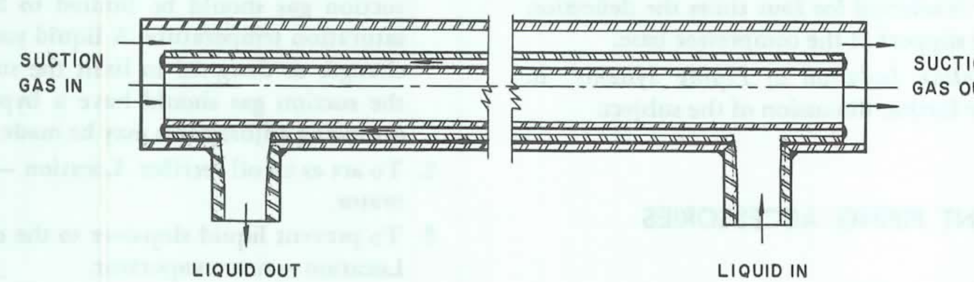
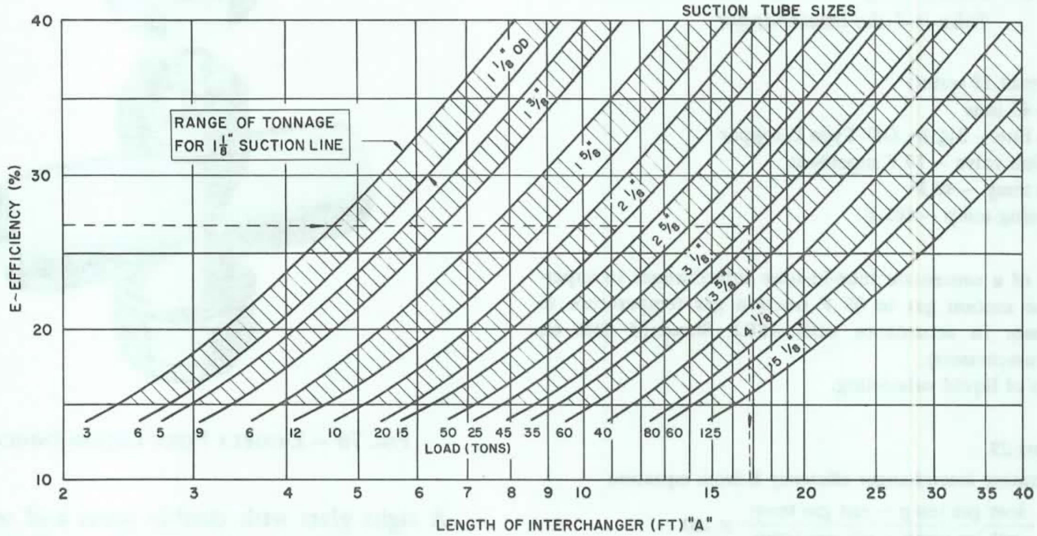


FIG. 77 — ECCENTRIC THREE-PIPE LIQUID SUCTION INTERCHANGER

CHART 23—EFFICIENCY CURVES, DOUBLE TUBE LIQUID SUCTION INTERCHANGER,

Refrigerants 12 and 500



NOTES: 1. E (exchanger efficiency) = $\frac{\text{Leav gas temp} - \text{Ent gas temp}}{\text{Ent liq temp} - \text{Ent gas temp}} \times 100$

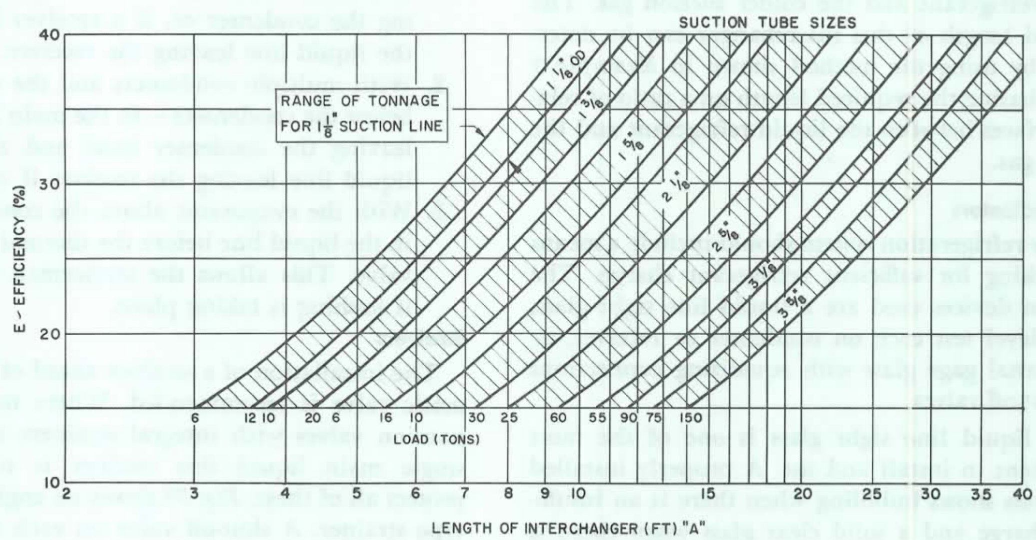
2. Liquid refrigerant sub-cooling multipliers — Refrigerant 12 = .653, Refrigerant 500 = .571

3. Type L tubing for Refrigerants 12 and 500. Liquid in annular space.

4. Liquid tube is next larger size.

CHART 24—EFFICIENCY CURVES, DOUBLE TUBE LIQUID SUCTION INTERCHANGER,

Refrigerant 22



NOTES: 1. E (exchanger efficiency) = $\frac{\text{Leav gas temp} - \text{Ent gas temp}}{\text{Ent liq temp} - \text{Ent gas temp}} \times 100$

2. Liquid refrigerant sub-cooling multiplier — Refrigerant 22 = .458

3. Type L tubing for Refrigerant 22. Liquid in annular space.

4. Liquid tube is next larger size.

calculated by using the ratio of the specific heats of the suction gas and of the liquid (subcooling multiplier). *Example 5* illustrates the use of these charts.

Example 5 — Determining the Length of a Concentric Tube-in-Tube Interchanger

Given:

Refrigerant 12 system
Load — 45 tons
Suction line — 3⅛ in. OD Type L copper
Expansion valve — 10 F superheat
Suction temp — 40 F
Condensing temp — 105 F

Find:

Length of a concentric tube-in-tube interchanger to superheat the suction gas to 65 F (suction gas temperature to compressor in accordance with ASRE Standard 23R on rating compressors).
Amount of liquid subcooling.

Solution:

See *Chart 23.*

1. Determine interchanger efficiency E from equation

$$E = \frac{\text{leav gas temp} - \text{ent gas temp}}{\text{ent liq temp} - \text{ent gas temp}} \times 100$$
$$= \frac{65 - 50}{105 - 50} \times 100 = \frac{15}{55} \times 100 = 27.2\%$$

2. With an efficiency of 27.2% a 3⅛ in. OD pipe size and a 45 ton load, enter *Chart 23* as indicated per dashed line to determine a length (A) of 17 ft.
3. For Refrigerant 12 the ratio of gas to liquid specific heat is .653. Therefore, subcooling of liquid refrigerant is .653 × 15 F (leav gas temp — ent gas temp) or 9.8 F.

An *eccentric three-pipe interchanger* is shown in *Fig. 77.* The inner pipe and the outer pipe offer two surfaces for the exchange of heat between the warm liquid refrigerant and the colder suction gas. The required length of this interchanger can be determined by using the method shown in *Example 5* and by basing the required length on a ratio of relative surfaces between the liquid refrigerant and the suction gas.

Liquid Indicators

Every refrigeration system should include a means of checking for sufficient refrigerant charge. The common devices used are a liquid line sight glass, liquid level test cock on condenser or receiver, or an external gage glass with equalizing connections and shut-off valves.

The liquid line sight glass is one of the most convenient to install and use. A properly installed sight glass shows bubbling when there is an insufficient charge and a solid clear glass when there is sufficient charge.

Sight glasses should be installed full size in the main liquid line and not in a bypass line that parallels the main line.

FIG. 78 — DOUBLE PORT LIQUID INDICATOR

A sight glass with double ports and seal caps is preferable. The double ports allow a light to be put behind one port so that the state of the refrigerant is easily seen. The seal caps serve as an added protection against leakage or breakage since they are removed only when checking the refrigerant. *Fig. 78* shows a double port liquid indicator with seal caps.

The installation of a double port or see-through sight glass is recommended in the following locations:

1. With a single condenser and the evaporator below the condenser — in the liquid line leaving the condenser or, if a receiver is used, in the liquid line leaving the receiver.
2. With multiple condensers and the evaporator below the condensers — in the main liquid line leaving the condenser bank and also in the liquid line leaving the receiver if used.
3. With the evaporator above the condenser(s) — in the liquid line before the thermal expansion valve. This allows the serviceman to observe if flashing is taking place.

Strainers

The installation of a strainer ahead of each automatic valve is recommended. Where multiple expansion valves with integral strainers are used, a single main liquid line strainer is sufficient to protect all of these. *Fig. 79* shows an angle cartridge type strainer. A shut-off valve on each side of the strainer is desirable and should be located as close to the strainer as possible.

On steel piping systems an adequate strainer should be installed in the suction line and a filter-

Figures 78-80, courtesy of Mueller Brass Co.

FIG. 79 — ANGLE CARTRIDGE TYPE STRAINER

FIG. 80 — ANGLE TYPE DRIER-STRAINER

drier in the liquid line to remove the scale and rust inherent in steel pipe.

Refrigerant Driers

A permanent refrigerant drier is recommended for most systems and is essential for all low temperature systems. It is also essential for all systems using hermetic compressors since the compressor motor winding is exposed to refrigerant gas. If the gas contains excessive moisture, the winding insulation may break down and cause the motor to burn out. A full-flow drier must be used for this type system.

Figure 80 shows an angle type cartridge drier. The drier should be mounted vertically in the liquid line. A three-valve bypass *(Fig. 81)* should be used to permit isolation of the drier for servicing and to allow partial refrigerant flow thru the drier.

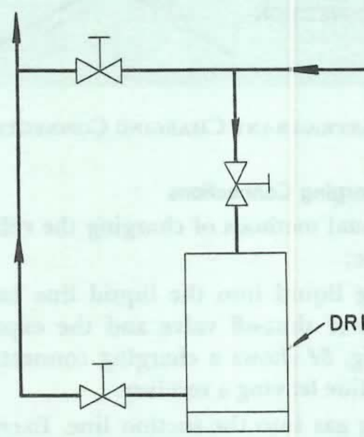


FIG. 81 — THREE-VALVE BYPASS FOR REFRIGERANT DRIER

Reliable *moisture indicators (Fig. 82)* for liquid refrigerant lines are available. These devices indicate the proper time to replace the drier cartridge.

FIG. 82 — COMBINATION MOISTURE AND LIQUID INDICATOR

Filter-Driers

Filter-driers *(Fig. 83)* are more commonly used than strainers and driers together. The drier material actually filters the liquid refrigerant.

FIG. 83 — FILTER-DRIER

Figures 82 and 83, courtesy of Sporlan Valve Co.

Solenoid Valves

Solenoid valves are commonly used in the following places:

1. In the liquid line of any system operating on single pump-out or pump-down compressor control.
2. In the liquid line of any single or multiple DX evaporator system.
3. In the oil bleeder lines from flooded evaporators to stop the flow of oil and refrigerant into the suction line when the system shuts down.

In many cases it is desirable to use solenoid valves with opening stems. The opening stem serves as a by-pass so that the system may continue to operate in case of solenoid coil failure.

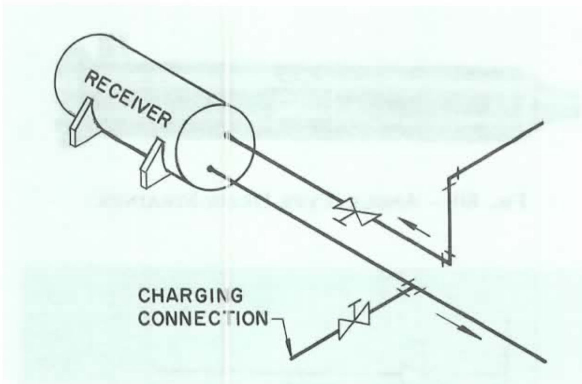


FIG. 84 — REFRIGERANT CHARGING CONNECTIONS

Refrigerant Charging Connections

The two usual methods of charging the refrigeration system are:

1. Charging liquid into the liquid line between the receiver shut-off valve and the expansion valve. *Fig. 84* shows a charging connection in a liquid line leaving a receiver.
2. Charging gas into the suction line. Except for very small systems this method is not practical because of the time required to evaporate the refrigerant from the drum and because of the danger of dumping raw liquid into the compressor.

Expansion Valves

Thermal expansion valves should be sized to avoid both the penalties of being undersized at full load and of being excessively oversized at partial load. The following items should be considered before sizing valves:

1. Refrigerant pressure drop thru the system must be properly evaluated to determine the correct pressure drop available across the valve.
2. Variations in condensing pressure during operation affect valve pressure and capacity. Condensing pressure should be controlled, therefore, to maintain required valve capacity.
3. Oversized thermal expansion valves do not control as well at full system capacity as properly sized valves and control gets progressively worse as the coil load decreases. Capacity reduction, available in most compressors, further increases this problem and necessitates closer selection of expansion valves to match realistic loads.

When sizing thermal expansion valves, make the selection on the basis of maximum load at the design operating pressure and at least 10 degrees superheat. Five degrees is the usual change in superheat between a full open and closed position. This is called the operating superheat. Thus a valve which operates at 10 degrees superheat at design load balances out at 5 to 6 degrees superheat at low load. A low superheat setting at design load, therefore, does not provide sufficient margins of safety at low loads because of the 5 degrees necessary for operating superheat.

The expansion valve bulb should be located immediately after the coil outlet on the suction line and 45° above the bottom of the pipe. With this arrangement the coil is the source of superheat for valve operation. The valve should be set so that overfeeding does not occur at times of partial load.

The effect of condensing temperature on the capacity of an expansion valve for two different systems is illustrated in *Example 6.*

Example 6 — Effect of Condensing Temperature on Expansion Valves

Given:

Two refrigeration systems using Refrigerant 500, one operating at 40 F suction and 90 F condensing, the other operating at 40 F suction and 130 F condensing.

	SYSTEM 1 40 F Suction 90 F Condens.	SYSTEM 2 40 F Suction 130 F Condens.
Condensing pressure	121.2 psig	218.2 psig
Liquid line drop	6.2	6.2
Pressure at thermal expansion valve inlet	115.0 psig	212.0 psig
Suction pressure	46.2 psig	46.2 psig
Suction line losses	2.8	2.8
Coil pressure drop	7.0	7.0
Distributor pressure drop	17.0	17.0
Pressure at thermal expansion valve outlet	73.0 psig	73.0 psig
Pressure drop available across valve	42.0 psi	139.0 psi

Assume that a valve of 27.5 ton capacity at 40 F suction and 60 psi differential is selected.

Find:

Capacity at the pressure drop available across the valve of systems 1 and 2.

Solution:

The capacities will vary approximately as the square root of the pressures:

$$\frac{\text{Capacity 1}}{\text{Capacity 2}} = \left(\frac{P_1}{P_2}\right)^{1/2}$$

For system 1 $\quad \frac{\text{Cap.}}{27.5} = \left(\frac{42}{60}\right)^{1/2} \quad$ Cap. = 23 tons

For system 2 $\quad \frac{\text{Cap.}}{27.5} = \left(\frac{139}{60}\right)^{1/2} \quad$ Cap. = 42 tons

Note that the expansion valve capacity is nearly double at the higher head pressure.

On certain low temperature applications and on high temperature applications where the design or partial load least temperature difference (L.T.D.) between the refrigerant and air or water is extremely small, it may become necessary to consider the use of the liquid suction interchanger as a source of superheat. This is done to increase the effective evaporator surface by allowing the liquid suction interchanger to supply the superheat function.

If only one liquid suction interchanger is used for the applications just mentioned, it should be an eccentric three-pipe interchanger as shown in *Fig. 77*. This arrangement permits the expansion valve bulb to sense the suction gas temperature from the outside surface of the interchanger. Otherwise two tube-in-tube interchangers should be used with the thermal expansion valve bulb located between the interchangers.

The preferred refrigerant flow in a coil circuit to obtain superheat is illustrated in *Fig. 85*.

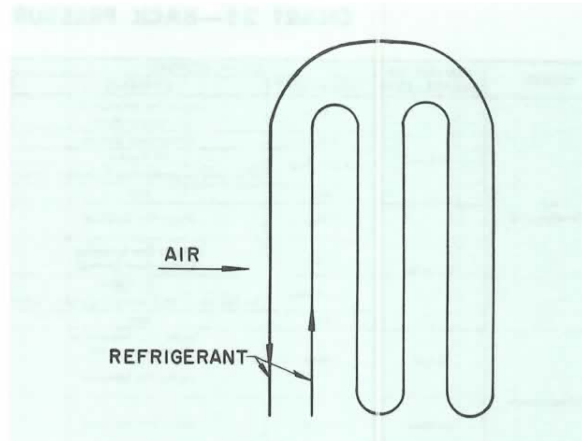


Fig. 85 – Preferred Refrigerant Flow in Coil Circuit to Obtain Superheat (Plan View)

SUCTION LINE

Back Pressure Valves

A conventional type back pressure regulating valve is used in a refrigerating system to maintain a predetermined pressure in the evaporator. A conventional type regulator controls the upstream pressure. The regulator has a spring loaded diaphragm designed to actuate a seat pilot valve. The actuating pressure comes from the evaporator or upstream side of the regulator. When the upstream pressure against the diaphragm is greater than that exerted by the spring, the pilot valve opens and a flow of gas is admitted to the power piston. The piston in turn causes the main port to open. This permits a flow of gas from the upstream side of the valve to the downstream side. When the actuating pressure becomes less than that controlled by the spring pressure, the flow of gas to the power piston is stopped and the regulator closes.

There are many variations of the back pressure regulating valve. Several are described in the following:

1. The compensating type, actuated by air or electricity, varies the suction pressure in accordance with temperature or load demand.
2. The dual pressure regulator is designed to operate at two predetermined pressures without resetting or adjustment; by opening and closing a pilot solenoid, either the low pressure or the high pressure head operates.

Figure 86 shows a simple back pressure regulating valve which is ordinarily used for one of the following purposes:

1. To control evaporator suction pressure in spite of compressor suction pressure variation.
2. To establish evaporator suction pressure when lower compressor suction pressure is demanded by another part of the same system.
3. To prevent evaporator freezing when operating near the freezing temperature.

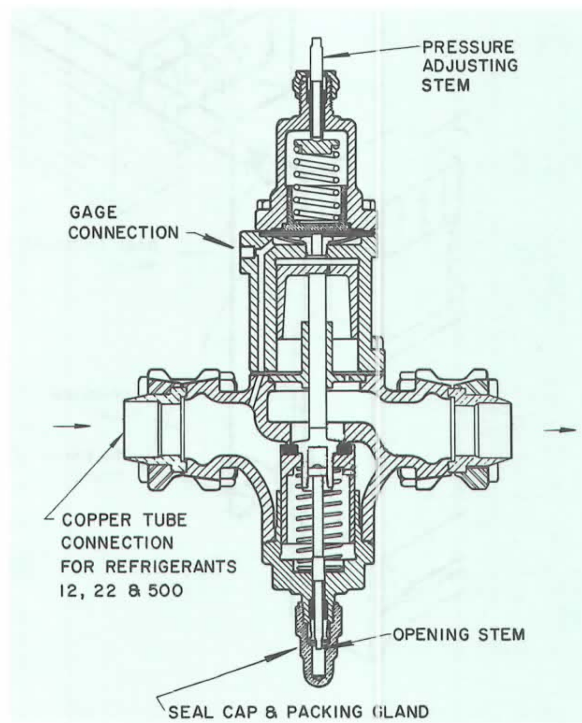


Fig. 86 — Back Pressure Valve

CHART 25—BACK PRESSURE VALVE APPLICATION CHART

SERVICE	NUMBER OF EVAPORATORS	ROOM CONTROL THERMOSTAT	ROOM CONTROL CONTROLS	COMPRESSOR Reduction	CAPACITY Controlled By	BACK PRESSURE VALVE REQUIRED	REASON	REMARKS
Air Conditioning	Single	1-Step	Liquid Line Solenoid or Compressor Motor	None	——	Not Usually	See Notes 1, 2	Analyze for Large % OA Jobs
				50%	Pressure	Not Usually	See Notes 1, 2	Analyze for Large % OA Jobs
		2-Step	Compressor Motors	50%	Temperature	Not Usually	See Notes 1, 2	Analyze for Large % OA Jobs
		Modulating	Air Bypass	None	——	Yes	See Note 1	
			Air Bypass or Modulating Expansion Valve	50%	Pressure	Sometimes	See Note 1	Must Be Analyzed
		Any	Any	3 or more steps	Pressure	Not Usually	See Note 1	Check frosting on last step
	Multiple	1-Step	Liquid Line Solenoid	None	——	Usually	See Note 1	Check frosting at minimum load
				50%	Pressure	Not Usually	See Note 1	Check frosting at minimum load
		2-Step	2 Liquid Line Solenoids (Rarely Used Control)	None	——	Usually	See Note 1	Check frosting at minimum load
				50%	Pressure	Not Usually	See Note 1	Check frosting at minimum load
		Modulating	Air Bypass	None	——	Yes	See Note 1	
			Air Bypass or Modulating Expansion Valve	50%	Pressure	Sometimes	See Note 1	Must be analyzed
		Any	Any	3 or more steps	Pressure	No	See Note 1	
Refrigeration	Single	1-Step	Liquid Line Solenoid or Compressor Motor	None	——	Sometimes	See Note 3	Used on commercial chill rooms only. Back pressure valve is by-passed during chilling operation.
			Liquid Line Solenoid	50%	Pressure	Sometimes	See Note 3	
				3 or more steps	Pressure	No		
	Multiple	1-Step	Liquid Line Solenoid	None	——	Sometimes	See Note 3	Used only on rooms where low limit humidity control is desired.
				50%	Pressure	Sometimes	See Note 3	
				3 or more steps	Pressure	No		
Water Cooling (Flooded Coolers)	Single or Multiple	1-Step	Compressor Motor	None	——	Usually	See Note 4	Check surface temperature at minimum load and on last step.
				50%	Pressure	Sometimes	See Note 4	
				3 or more steps	Pressure	Not Usually	See Note 4	

NOTES:

1: Reason for use of back pressure valve on any air conditioning system is to avoid frosting of coil at light load. Chart applies only to normal applications—refrigerant temperatures at full load of 40°F or above. For full load refrigerant temperatures from 35°-40°, frosting at light load should be checked. For full load refrigerant temperatures below 35°, except where certain of constant load, back pressure valve should be used.

2: Except for jobs using large % OA, this control indicates nearly constant load.

3: Reason for use of back pressure on any refrigeration system is to set a low limit below which room relative humidity cannot drop at light load.

4: Reason for use of back pressure valve on any water cooling system is to avoid freeze-up at light load. Chart applies only to variable loads and to normal applications—leaving water temperatures of 40°F or above. For leaving water temperatures below 40°, analysis should be made to make certain that surface temperature is not lower than 33°, unless certain of constant load, back pressure valve should be used.

3

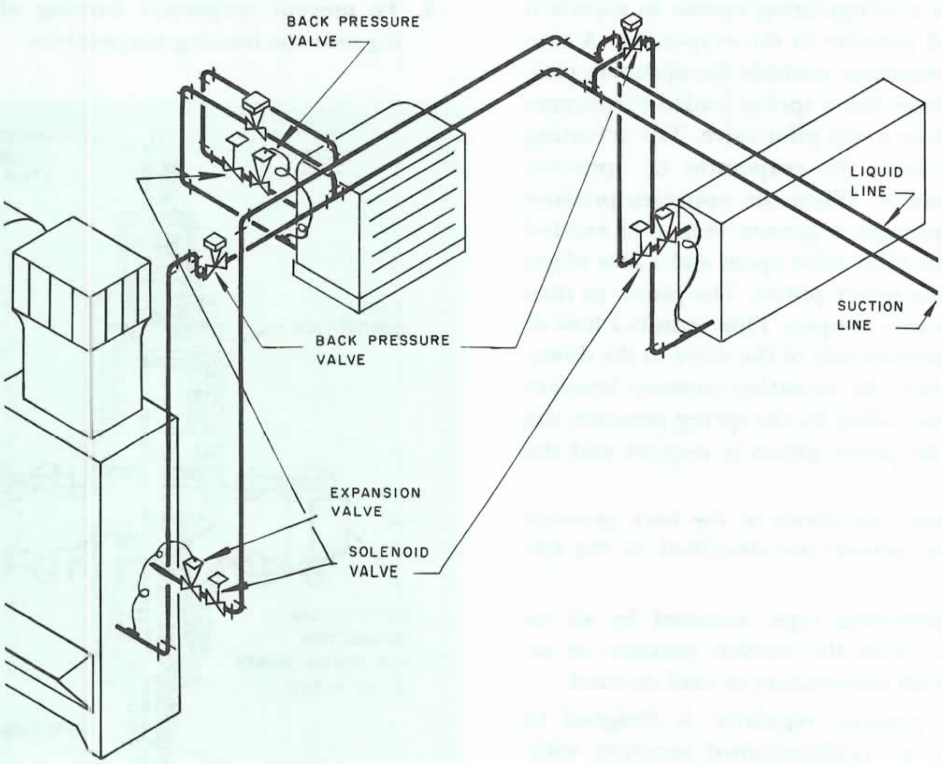


FIG. 87 — INSTALLATION USING BACK PRESSURE VALVES

Chart 25 illustrates the application of back pressure valves for various services such as number and types of evaporators, and types of room and compressor control.

Figure 87 illustrates the location of back pressure valves.

DISCHARGE LINE

Oil Separators

Oil separators reduce the rate of oil circulation. However they are not 100% efficient since some oil always circulates thru the system.

Oil separators are of particular value on certain types of installations such as:

1. Systems requiring a sudden and frequent capacity variation.
2. Systems having extensive pipe lines and numerous evaporators. The large volumes inherent in such systems result in appreciable oil hangup.

There are several objections to oil separators:

1. Oil separators permit some oil to be carried over into the system and, therefore, proper piping design to return oil is still required even though a separator is used.
2. On start-up, gas may condense in the shell of the separator. As a result the separator delivers liquid refrigerant into the crankcase. This in turn increases crankcase foaming and oil loss from the compressor.

During the "off" cycle the oil separator cools down and acts as a condenser for liquid refrigerant that evaporates in the warmer parts of the system. Thus a cool oil separator acts as a liquid condenser during "off" cycles and also on compressor start-up until the separator has warmed up. Large amounts of liquid refrigerant in the crankcase result in poor lubrication and may also result in removing the oil from the crankcase completely.

Figure 88 shows the recommended method for piping an oil separator.

Mufflers

If a muffler is used in the hot gas line, it should be installed in downward flow risers or in horizontal lines as close to the compressor as possible.

The hot gas pulsations from the compressor can set up a condition of resonance with certain lengths of refrigerant piping in the hot gas line. A muffler installed in the compressor discharge aids in eliminating such a condition.

Figure 89 shows a muffler in a hot gas line at the compressor.

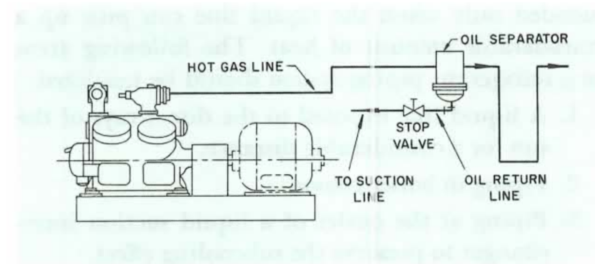


FIG. 88 — OIL SEPARATOR LOCATION

Check Valves

Check valves contribute a relatively large addition to a line pressure drop at full load and must be taken into account in the selection of refrigeration equipment. In addition a check valve cannot be relied upon for 100% shut-off.

Whenever the receiver is warmer than the compressor during shutdown, refrigerant in the receiver tends to boil off and flow back thru the condenser and hot gas discharge line to the compressor where it condenses. If there is sufficient refrigerant in the receiver, liquid refrigerant eventually enters the compressor despite the loop in the hot gas line at the base of the compressor. To prevent this, a check valve should be used *(Fig. 68, page 65).*

In a non-automatic system a hand valve may be used at the inlet to the condenser to manually shut off the flow on shutdown, in which case the pressure drop involved will be much less than that encountered using a check valve.

REFRIGERANT PIPING INSULATION

Liquid lines should not be insulated if the surrounding temperature is lower than or equal to the temperature of the liquid. Insulation is recom-

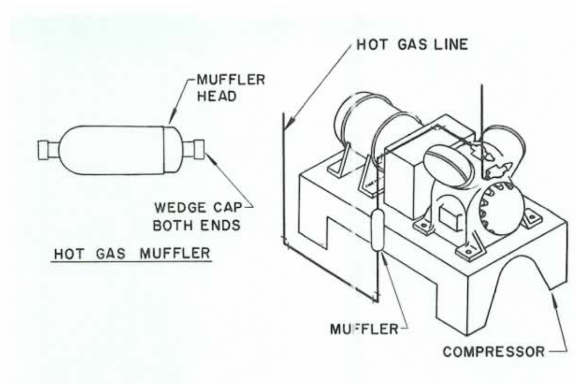


FIG. 89 — HOT GAS MUFFLER AND LOCATION IN HOT GAS LINE

3

mended only when the liquid line can pick up a considerable amount of heat. The following areas in a refrigerant piping system should be insulated:

1. A liquid line exposed to the direct rays of the sun for a considerable distance.
2. Piping in boiler rooms.
3. Piping at the outlet of a liquid suction interchanger to preserve the subcooling effect.

Where liquid and suction lines can be strapped together, a single insulating covering can be used over both lines. This induces an exchange of heat and is desirable from the standpoint of the subcooling effect on the liquid. However excessive superheating of the suction gas can result from too much exchange of heat.

Hot gas lines should not be insulated. Any heat lost by these lines reduces the work to be done by the condenser.

3

Suction lines should be insulated only to prevent dripping where this causes a nuisance or damage. It is generally desirable to have the suction line capable of absorbing some heat to evaporate any liquid which may have entered the suction line from the evaporator. For unusual conditions of high ambient temperatures and simultaneous high relative humidities extra insulation must be applied.

The thickness of insulation required to prevent condensation on the outer surface is that thickness which raises the temperature of the outer surface of the insulation to a point slightly higher than the maximum expected dewpoint of the surrounding air. The external vapor barrier must be made as nearly perfect as possible in order to prevent leakage of vapor into the insulation.

A cellular glass or cellular plastic type of insulation is fast becoming accepted as an ideal insulation. Its cellular structure provides exceptionally high resistance to water and water vapor. The cellular glass, being inorganic, is fire-proof. Cellular plastic which is also available is self-extinguishing.

When located out of doors, insulation must be weatherproofed unless, of course, it is inherently waterproof.

CHAPTER 4. STEAM PIPING

This chapter describes practical design and layout techniques for steam piping systems. Steam piping differs from other systems because it usually carries three fluids – steam, water and air. For this reason, steam piping design and layout require special consideration.

GENERAL SYSTEM DESIGN

Steam systems are classified according to piping arrangement, pressure conditions, and method of returning condensate to the boiler. These classifications are discussed in the following paragraphs.

PIPING ARRANGEMENT

A one- or two-pipe arrangement is standard for steam piping. The one-pipe system uses a single pipe to supply steam and to return condensate. Ordinarily, there is one connection at the heating unit for both supply and return. Some units have two connections which are used as supply and return connections to the common pipe.

A two-pipe steam system is more commonly used in air conditioning, heating, and ventilating applications. This system has one pipe to carry the steam supply and another to return condensate. In a two-pipe system, the heating units have separate connections for supply and return.

The piping arrangements are further classified with respect to condensate return connections to the boiler and direction of flow in the risers:

1. Condensate return to boiler
   a. Dry-return – condensate enters boiler above water line.
   b. Wet-return – condensate enters boiler below water line.
2. Steam flow in riser
   a. Up-feed – steam flows up riser.
   b. Down-feed – steam flows down riser.

PRESSURE CONDITIONS

Steam piping systems are normally divided into five classifications – high pressure, medium pressure, low pressure, vapor and vacuum systems. Pressure ranges for the five systems are:

1. High pressure – 100 psig and above
2. Medium pressure – 15 to 100 psig
3. Low pressure – 0 to 15 psig
4. Vapor – vacuum to 15 psig
5. Vacuum – vacuum to 15 psig

Vapor and vacuum systems are identical except that a vapor system does not have a vacuum pump, but a vacuum system does.

CONDENSATE RETURN

The type of condensate return piping from the heating units to the boiler further identifies the steam piping system. Two arrangements, gravity and mechanical return, are in common use.

When all the units are located above the boiler or condensate receiver water line, the system is described as a gravity return since the condensate returns to the boiler by gravity.

If traps or pumps are used to aid the return of condensate to the boiler, the system is classified as a mechanical return system. The vacuum return pump, condensate return pump and boiler return trap are devices used for mechanically returning condensate to the boiler.

CODES AND REGULATIONS

All applicable codes and regulations should be checked to determine acceptable piping practice for the particular application. These codes usually dictate piping design, limit the steam pressure, or qualify the selection of equipment.

WATER CONDITIONING

The formation of scale and sludge deposits on the boiler heating surfaces creates a problem in generating steam. Scale formation is intensified since scale-forming salts increase with an increase in temperature.

Water conditioning in a steam generating system should be under the supervision of a specialist.

4

TABLE 22—RECOMMENDED HANGER SPACINGS FOR STEEL PIPE

NOM. PIPE SIZE (in.)	DISTANCE BETWEEN SUPPORTS (FT) Average Gradient		
	1" in 10'	½" in 10'	¼" in 10'
¾	9	—	—
1	13	6	—
1¼	16	10	5
1½	19	14	8
2	21	17	13
2½	24	19	15
3	27	22	18
3½	29	24	19
4	32	26	20
5	37	29	23
6	40	33	25
8	—	38	30
10	—	43	33
12	—	48	37
14	—	50	40
16	—	53	42
18	—	57	44
20	—	60	47
24	—	64	50

NOTE: Data is based on standard wall pipe filled with water and average number of fittings.

Courtesy of Crane Co.

PIPING SUPPORTS

All steam piping is pitched to facilitate the flow of condensate. *Table 22* contains the recommended support spacing for piping pitched for different gradients. The data is based on Schedule 40 pipe filled with water, and an average amount of valves and fittings.

PIPING DESIGN

A steam system operating for air conditioning comfort conditions must distribute steam at all operating loads. These loads can be in excess of design load, such as early morning warmup, and at extreme partial load, when only a minimum of heat is necessary. The pipe size to transmit the steam for a design load depends on the following:

1. The initial operating pressure and the allowable pressure drop thru the system.
2. The total equivalent length of pipe in the longest run.
3. Whether the condensate flows in the same direction as the steam or in the opposite direction.

The major steam piping systems used in air conditioning applications are classified by a combination of piping arrangement and pressure conditions as follows:

1. Two-pipe high pressure
2. Two-pipe medium pressure
3. Two-pipe low pressure
4. Two-pipe vapor
5. Two-pipe vacuum
6. One-pipe low pressure

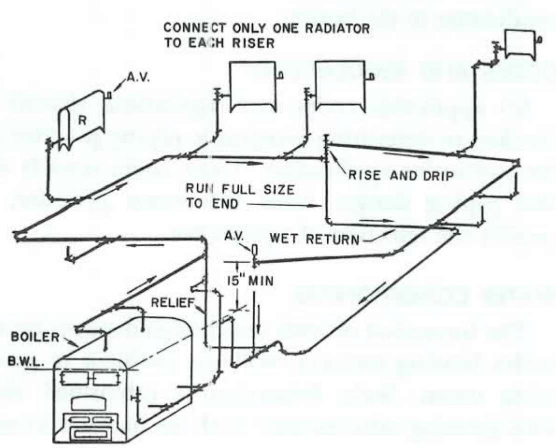


FIG. 90 — ONE-PIPE, UPFEED GRAVITY SYSTEM

ONE-PIPE SYSTEM

A one-pipe gravity system is primarily used on residences and small commercial establishments. *Fig. 90* shows a one-pipe, upfeed gravity system. The steam supply main rises from the boiler to a high point and is pitched downward from this point around the extremities of the basement. It is normally run full size to the last take-off and is then reduced in size after it drops down below the boiler water line. This arrangement is called a wet return. If the return main is above the boiler water line, it is called a dry return. Automatic air vents are required at all high points in the system to remove non-condensable gases. In systems that require long mains, it is necessary to check the pressure drop and make sure the last heating unit is sufficiently above the water line to prevent water backing up from the boiler and flooding the main. During operation, steam and condensate flow in the same direction in the mains, and in opposite direction in branches and risers. This system requires larger pipe and valves than any other system.

The one-pipe gravity system can also be designed as shown in *Fig. 91*, with each riser dripped separately. This is frequently done on more extensive systems.

Another type of one-pipe gravity system is the down-feed arrangement shown in *Fig. 92*. Steam flows in the main riser from the boiler to the building attic and is then distributed throughout the building.

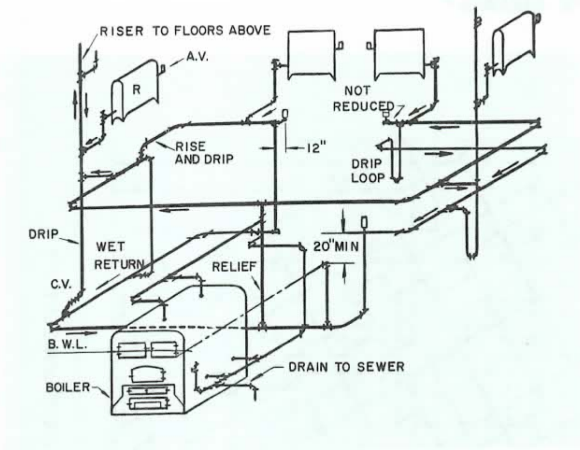


FIG. 91 — ONE-PIPE GRAVITY SYSTEM WITH DRIPPED RISERS

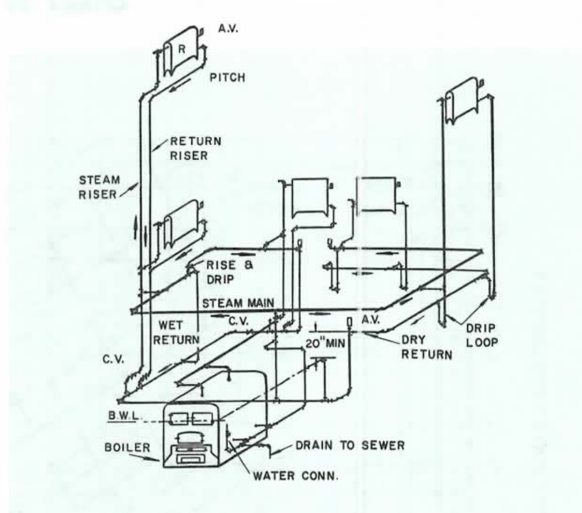


FIG. 93 — TWO-PIPE GRAVITY SYSTEM

TWO-PIPE SYSTEM

A two-pipe gravity system is shown in *Fig. 93*. This system is used with indirect radiation. The addition of a thermostatic valve at each heating unit adapts it to a vapor or a mechanical vacuum system. A gravity system has each radiator separately sealed by drip loops on a dry return or by dropping directly into a wet return main. All drips, reliefs and return risers from the steam to the return side of the system must be sealed by traps or water loops to insure satisfactory operation.

If the air vent on the heating unit is omitted, and the air is vented thru the return line and a vented condensate receiver, a vapor system as illustrated in *Fig. 94* results.

The addition of a vacuum pump to a vapor system classifies the system as a mechanical vacuum system. This arrangement is shown in *Fig. 95*.

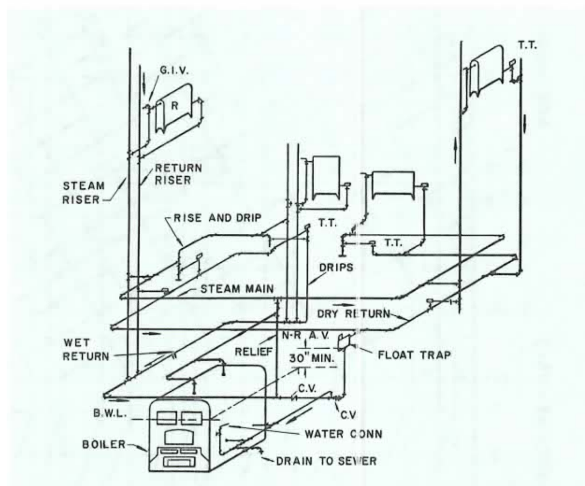


FIG. 94 — VAPOR SYSTEM

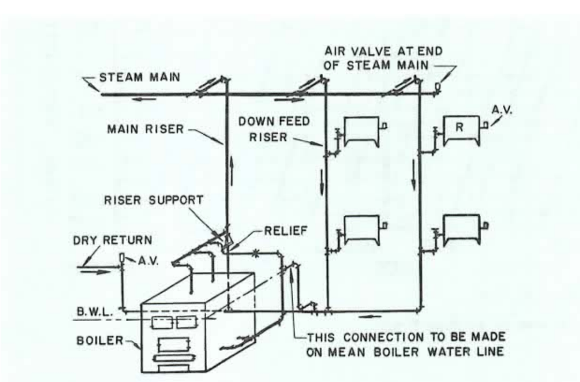


FIG. 92 — ONE-PIPE, DOWNFEED GRAVITY SYSTEM

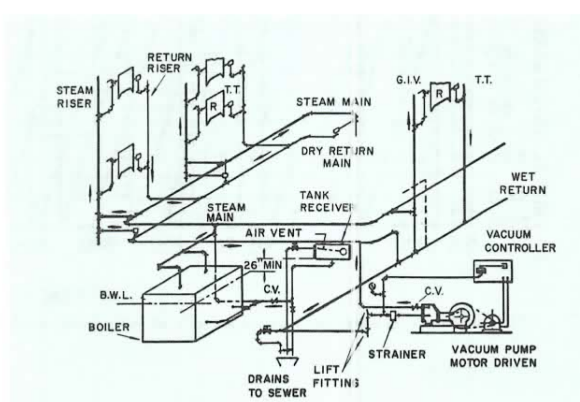
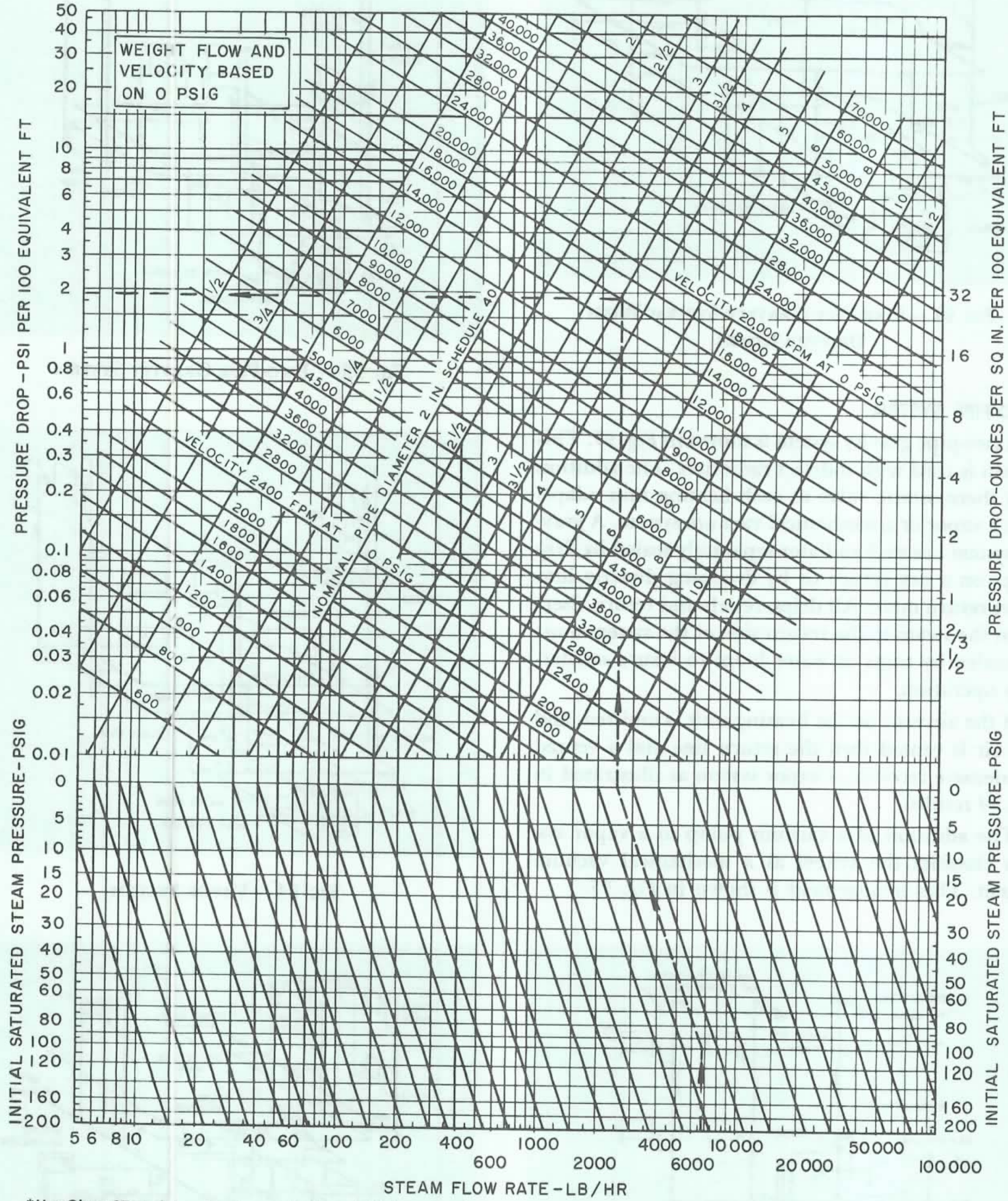


FIG. 95 — MECHANICAL VACUUM SYSTEM

Figures 90 thru 95, courtesy of Kent's Mechanical Engineer's Handbook, Power Volume, Eleventh Edition, John Wiley & Sons Publisher.

CHART 26—PIPE SIZING*



*Use *Chart 27* to determine steam velocity at initial saturated steam pressures other than 0 psig.

Charts 26 and 27 from Heating Ventilating Air Conditioning Guide 1959. Used by permission.

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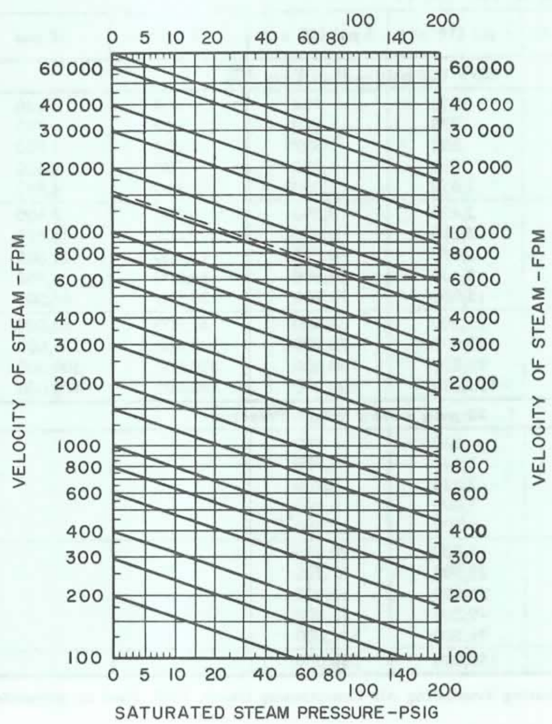
CHART 27—VELOCITY CONVERSION*



*See *Example 3, page 89,* for use of chart

PIPE SIZING

GENERAL

Charts and tables have been developed which are used to select the proper pipe to carry the required steam rate at various pressures.

Chart 26 is a universal chart for steam pressure of 0 to 200 psig and for a steam rate of from 5 to 100,000 pounds per hour. However, the velocity as read from the chart is based on a steam pressure of 0 psig and must be corrected for the desired pressure from *Chart 27*. The complete chart is based on the Moody friction factor and is valid where condensate and steam flow in the same direction.

Tables 23 thru 28 are used for quick selection at specific steam pressures. *Chart 26* has been used to tabulate the capacities shown in *Tables 23 thru 25*. The capacities in *Tables 26 thru 28* are the results of tests conducted in the ASHAE laboratories. Suggested limitations for the use of these tables are shown as notes on each table. In addition, *Table 28* shows the total pressure drop for two-pipe low pressure steam systems.

RECOMMENDATIONS

The following recommendations are for use when sizing pipe for the various systems:

Two-Pipe High Pressure System

This system is used mostly in plants and occasionally in commercial installations.

1. Size supply main and riser for a maximum drop of 25-30 psi.
2. Size supply main and risers for a maximum friction rate of 2-10 psi per 100 ft of equivalent pipe.
3. Size return main and riser for a maximum pressure drop of 20 psi.
4. Size return main and riser for a maximum friction rate of 2 psi per 100 ft of equivalent pipe.
5. Pitch supply mains ¼ in. per 10 ft away from boiler.
6. Pitch return mains ¼ in. per 10 ft toward the boiler.
7. Size pipe from *Table 23.*

Two-Pipe Medium Pressure System

This system is used mostly in plants and occasionally in commercial installations.

1. Size supply main and riser for a maximum pressure drop of 5-10 psi.
2. Size supply mains and risers for a maximum friction rate of 2 psi per 100 ft of equivalent pipe.
3. Size return main and riser for a maximum pressure drop of 5 psi.
4. Size return main and riser for a maximum friction rate of 1 psi per 100 ft of equivalent pipe.
5. Pitch supply mains ¼ in. per 10 ft away from the boiler.
6. Pitch return mains ¼ in. per 10 ft toward the boiler.
7. Size pipe from *Table 24.*

Two-Pipe Low Pressure System

This system is used for commercial, air conditioning, heating and ventilating installations.

1. Size supply main and risers for a maximum pressure drop determined from *Table 28,* depending on the initial system pressure.
2. Size supply main and riser for a maximum friction rate of 2 psi per 100 ft of equivalent pipe.
3. Size return main and riser for a maximum

TABLE 23—HIGH PRESSURE SYSTEM PIPE CAPACITIES (150 psig)

Pounds Per Hour

PIPE SIZE (in.)	PRESSURE DROP PER 100 FT							
	⅛ psi (2 oz)	¼ psi (4 oz)	½ psi (8 oz)	¾ psi (12 oz)	1 psi (16 oz)	2 psi (32 oz)	5 psi	10 psi
	SUPPLY MAINS AND RISERS				130 - 180 psig—Max Error 8%			
¾	29	41	58	82	116	184	300	420
1	58	82	117	165	233	369	550	790
1¼	130	185	262	370	523	827	1,230	1,720
1½	203	287	407	575	813	1,230	1,730	2,600
2	412	583	825	1,167	1,650	2,000	3,410	4,820
2½	683	959	1,359	1,920	2,430	3,300	5,200	7,600
3	1,237	1,750	2,476	3,500	4,210	6,000	9,400	13,500
3½	1,855	2,626	3,715	5,250	6,020	8,500	13,100	20,000
4	2,625	3,718	5,260	7,430	8,400	12,300	19,200	28,000
5	4,858	6,875	9,725	13,750	15,000	21,200	33,100	47,500
6	7,960	11,275	15,950	22,550	25,200	36,500	56,500	80,000
8	16,590	23,475	33,200	46,950	50,000	70,200	120,000	170,000
10	30,820	43,430	61,700	77,250	90,000	130,000	210,000	300,000
12	48,600	68,750	97,250	123,000	155,000	200,000	320,000	470,000
	RETURN MAINS AND RISERS				1 - 20 psig - Max Return Pressure			
¾	156	232	360	465	560	890		
1	313	462	690	910	1,120	1,780		
1¼	650	960	1,500	1,950	2,330	3,700		
1½	1,070	1,580	2,460	3,160	3,800	6,100		
2	2,160	3,300	4,950	6,400	7,700	12,300		
2½	3,600	5,350	8,200	10,700	12,800	20,400		
3	6,500	9,600	15,000	19,500	23,300	37,200		
3½	9,600	14,400	22,300	28,700	34,500	55,000		
4	13,700	20,500	31,600	40,500	49,200	78,500		
5	25,600	38,100	58,500	76,000	91,500	146,000		
6	42,000	62,500	96,000	125,000	150,000	238,000		

pressure drop determined from *Table 28*, depending on the initial system pressure.

4. Size return main and riser for a maximum friction rate of ½ psi per 100 ft of equivalent pipe.
5. Pitch mains ¼ in. per 10 ft away from the boiler.
6. Pitch return mains ¼ in. per 10 ft toward the boiler.
7. Use *Tables 25 thru 27* to size pipe.

Two-Pipe Vapor System

This system is used in commercial and residential installations.

1. Size supply main and riser for a maximum pressure drop of 1/16 - ⅛ psi.
2. Size supply main and riser for a maximum friction rate of 1/16 - ⅛ psi per 100 ft of equivalent pipe.
3. Size return main and supply for a maximum pressure drop of 1/16 - ⅛ psi.
4. Size return main and supply for a maximum friction rate of 1/16 - ⅛ psi per 100 ft of equivalent pipe.
5. Pitch supply ¼ in. per 10 ft away from the boiler.
6. Pitch return mains ¼ in. per 10 ft toward the boiler.
7. Size pipe from *Tables 25 thru 27*.

Two-Pipe Vacuum System

This system is used in commercial installations.

1. Size supply main and riser for a maximum pressure drop of ⅛ - 1 psi.
2. Size supply main and riser for a maximum friction rate of ⅛ - ½ psi per 100 ft of equivalent pipe.
3. Size return main and riser for a maximum pressure drop of ⅛ - 1 psi.
4. Size return main and riser for a maximum friction rate of ⅛ - ½ psi per 100 ft of equivalent pipe.
5. Pitch supply mains ¼ in. per 10 ft away from the boiler.
6. Pitch return mains ¼ in. per 10 ft toward the boiler.
7. Size pipe from *Tables 25 thru 27*.

One-Pipe Low Pressure System

This system is used on small commercial and residential systems.

TABLE 24—MEDIUM PRESSURE SYSTEM PIPE CAPACITIES (30 psig)

Pounds Per Hour

PIPE SIZE (in.)	PRESSURE DROP PER 100 FT					
	⅛ psi (2 oz)	¼ psi (4 oz)	½ psi (8 oz)	¾ psi (12 oz)	1 psi (16 oz)	2 psi (32 oz)
	SUPPLY MAINS AND RISERS			25 - 35 psig - Max Error 8%		
¾	15	22	31	38	45	63
1	31	46	63	77	89	125
1¼	69	100	141	172	199	281
1½	107	154	219	267	309	437
2	217	313	444	543	627	886
2½	358	516	730	924	1,033	1,460
3	651	940	1,330	1,628	1,880	2,660
3½	979	1,414	2,000	2,447	2,825	4,000
4	1,386	2,000	2,830	3,464	4,000	5,660
5	2,560	3,642	5,225	6,402	7,390	10,460
6	4,210	6,030	8,590	10,240	12,140	17,180
8	8,750	12,640	17,860	21,865	25,250	35,100
10	16,250	23,450	33,200	40,625	46,900	66,350
12	25,640	36,930	52,320	64,050	74,000	104,500
	RETURN MAINS AND RISERS			0 - 4 psig - Max Return Pressure		
¾	115	170	245	308	365	
1	230	340	490	615	730	
1¼	485	710	1,025	1,285	1,530	
1½	790	1,155	1,670	2,100	2,500	
2	1,575	2,355	3,400	4,300	5,050	
2½	2,650	3,900	5,600	7,100	8,400	
3	4,850	7,100	10,250	12,850	15,300	
3½	7,200	10,550	15,250	19,150	22,750	
4	10,200	15,000	21,600	27,000	32,250	
5	19,000	27,750	40,250	55,500	60,000	
6	31,000	45,500	65,500	83,000	98,000	

From Heating Ventilating Air Conditioning Guide, 1959. Used by permission.

1. Size supply main and riser for a maximum pressure drop of ¼ psi.
2. Size supply main and risers for a maximum friction rate of $1/16$ psi per 100 ft of equivalent pipe.
3. Size return main and risers for a maximum pressure drop of ¼ psi.
4. Size return main and risers for a maximum friction rate of $1/16$ psi per 100 ft of equivalent pipe.
5. Pitch supply main ¼ in. per 10 ft away from the boiler.
6. Pitch return main ¼ in. per 10 ft toward the boiler.

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TABLE 25—LOW PRESSURE SYSTEM PIPE CAPACITIES

Pounds Per Hour

CONDENSATE FLOWING WITH THE STEAM FLOW

NOM. PIPE SIZE (in.)	PRESSURE DROP PER 100 FT													
	$1/16$ psi (1 oz)		⅛ psi (2 oz)		¼ psi (4 oz)		½ psi (8 oz)		¾ psi (12 oz)		1 psi		2 psi	
	SATURATED PRESSURE (PSIG)*													
	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12
¾	9	11	14	16	20	24	29	35	36	43	42	50	60	73
1	17	21	26	31	37	46	54	66	68	82	81	95	114	137
1¼	36	45	53	66	78	96	111	138	140	170	162	200	232	280
1½	56	70	84	100	120	147	174	210	218	260	246	304	360	430
2	108	134	162	194	234	285	336	410	420	510	480	590	710	850
2½	174	215	258	310	378	460	540	660	680	820	780	950	1,150	1,370
3	318	380	465	550	660	810	960	1,160	1,190	1,430	1,380	1,670	1,950	2,400
3½	462	550	670	800	990	1,218	1,410	1,700	1,740	2,100	2,000	2,420	2,950	3,450
4	726	800	950	1,160	1,410	1,690	1,980	2,400	2,450	3,000	2,880	3,460	4,200	4,900
5	1,200	1,430	1,680	2,100	2,440	3,000	3,570	4,250	4,380	5,250	5,100	6,100	7,500	8,600
6	1,920	2,300	2,820	3,350	3,960	4,850	5,700	7,000	7,200	8,600	8,400	10,000	11,900	14,200
8	3,900	4,800	5,570	7,000	8,100	10,000	11,400	14,300	14,500	17,700	16,500	20,500	24,000	29,500
10	7,200	8,800	10,200	12,600	15,000	18,200	21,000	26,000	26,200	32,000	30,000	37,000	42,700	52,000
12	11,400	13,700	16,500	19,500	23,400	28,400	33,000	40,000	41,000	49,500	48,000	57,500	67,800	81,000

*The weight-flow rates at 3.5 psig can be used to cover sat. press. from 1 to 6 psig, and the rates at 12 psig can be used to cover sat. press. from 8 to 16 psig with an error not exceeding 8 percent.

Tables 23 thru 25 from Heating Ventilating Air Conditioning Guide 1959. Used by permission.

TABLE 26—RETURN MAIN AND RISER CAPACITIES FOR LOW PRESSURE STEAM SYSTEMS—POUNDS PER HOUR

PIPE SIZE (in.)	PRESSURE DROP PER 100 FT.																	
	1/32 psi (1/2 oz)			1/24 psi (2/3 oz)			1/16 psi (1 oz)			1/8 psi (2 oz)			1/4 psi (4 oz)			1/2 psi (8 oz)		
	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac	Wet	Dry	Vac	Wet*	Dry	Vac
								RETURN MAINS										
¾						42			100			142			200			283
1	125	62		145	71	143	175	80	175	250	103	249	350	115	350		120	494
1¼	213	130		248	149	244	300	168	300	425	217	426	600	241	600		255	848
1½	338	206		393	236	388	475	265	475	675	340	674	950	378	950		385	1,340
2	700	470		810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000		830	2,830
2½	1,180	760		1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350		1,410	4,730
3	1,880	1,460		2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350		2,585	7,560
3½	2,750	1,970		3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000		3,780	11,300
4	3,880	2,930		4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000		5,550	15,500
5	6,090	4,600		7,880	5,250	7,880	9,680	5,870	9,680	13,700	7,560	13,700	19,400	8,420	19,400		8,880	27,300
6	8,820	6,670		12,600	7,620	12,600	15,500	8,540	15,500	22,000	10,990	22,000	31,000	12,200	31,000		12,800	43,800
8	15,200	11,480		21,700	13,120	21,700	26,700	14,700	26,700	37,900	18,900	37,900	53,400	22,800	53,400		25,500	75,500
10	24,000	18,100		34,300	20,800	34,300	42,200	23,200	42,200	59,900	29,800	59,900	84,400	33,300	84,400		35,400	114,400
								RETURN RISERS										
¾		48			48	113		48	175		48	249		48	350		48	494
1		113			113	244		113	300		113	426		113	600		113	848
1¼		248			248	388		248	475		248	674		248	950		248	1,340
1½		375			375	815		375	1,000		375	1,420		375	2,000		375	2,830
2		750			750	1,360		750	1,680		750	2,380		750	3,350		750	4,730
2½		1,230			1,230	2,180		1,230	2,680		1,230	3,800		1,230	5,350		1,230	7,560
3		2,250			2,250	3,250		2,250	4,000		2,250	5,680		2,250	8,000		2,250	11,300
3½		3,230			3,230	4,480		3,230	5,500		3,230	7,810		3,230	11,000		3,230	15,500
4		4,830			4,830	7,880		4,830	9,680		4,830	13,700		4,830	19,400		4,830	27,300
5		7,560			7,560	12,600		7,560	15,500		7,560	22,000		7,560	31,000		7,560	43,800
6		10,990			10,990	21,700		10,990	26,700		10,990	37,900		10,990	53,400		10,990	75,500
8		18,900			18,900	34,300		18,900	42,200		18,900	59,900		18,900	84,400		18,900	114,400
10		29,800			29,800			29,800			29,800			29,800			29,800	

*Vac values may be used for wet return mains.

7. Size supply main and dripped runouts from *Table 25.*

TABLE 27—LOW PRESSURE SYSTEM PIPE CAPACITIES

Pounds Per Hour

CONDENSATE FLOWING AGAINST STEAM FLOW

PIPE SIZE (in.)	TWO-PIPE SYSTEM		ONE-PIPE SYSTEM		
	Vertical	Horizontal	Up-feed Supply Risers	Vertical Con-nectors	Riser Run-outs
A	B*	C†	D‡	E	F
¾	8	—	6	—	7
1	14	9	11	7	7
1¼	31	19	20	16	16
1½	48	27	38	23	16
2	97	49	72	42	23
2½	159	99	116	—	42
3	282	175	200	—	65
3½	387	288	286	—	119
4	511	425	380	—	186
5	1,050	788	—	—	278
6	1,800	1,400	—	—	545
8	3,750	3,000	—	—	—
10	7,000	5,700	—	—	—
12	11,500	9,500	—	—	—
16	22,000	19,000	—	—	—

*Do not use Column B for pressure drops less than 1/16 psi/100 ft of equivalent length. Use *Chart 26, page 84.*

†Pitch of horizontal runouts to riser should be not less than ½ in./ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one pipe size larger than called for in this table.

‡Do not use Column D for pressure drops less than 1/24 psi/100 ft of equivalent run except on sizes 3 in. and larger. Use *Chart 26, page 84.*

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8. Size undripped runouts from *Table 27, Column F.*
9. Size upfeed risers from *Table 27, Column D.*
10. Size downfeed supply risers from *Table 25.*
11. Pitch supply mains ¼ in. per 10 ft away from boiler.
12. Pitch return mains ¼ in. per 10 ft toward the boiler.

Use of Table 28

Example 1 — Determine Pressure Drop for Sizing Supply and Return Piping

Given:

Two-pipe low pressure steam system
Initial steam pressure — 15 psig
Approximate supply piping equivalent length — 500 ft
Approximate return piping equivalent length — 500 ft

Find:

1. Pressure drop to size supply piping
2. Pressure drop to size return piping

Solution:

1. Refer to *Table 28* for an initial steam pressure of 15 psig. The total pressure drop should not exceed 3.75 psi in the supply pipe. Therefore, the supply piping is sized for a total pressure drop of 3.75 or 3/4 psi per 100 ft of equivalent pipe.
2. Although ¾ psi is indicated in *Step 1, Item 4* under the two-pipe low pressure system recommends a maximum of ½ psi for return piping. Therefore, use ½ psi per 100 ft of equivalent pipe.

$$\text{Return main pressure drop} = \tfrac{1}{2} \times \frac{500}{100} = 2.5 \text{ psi.}$$

4

Friction Rate

Example 2 illustrates the method used to determine the friction rate for sizing pipe when the total system pressure drop recommendation (supply pressure drop plus return pressure drop) is known and the approximate equivalent length is known.

Example 2 — Determine Friction Rate

Given:

Four systems
Equivalent length of each system — 400 ft
Total pressure drop of systems — ½, ¾, 1, and 2 psi

Find:

Friction rate for each system.

Solution:

SYSTEM NUMBER	SYSTEM EQUIV. LENGTH (ft)	TOTAL SYSTEM PRESS. DROP (psi)	FRICTION RATE FOR PIPE SIZING (per 100 ft)
1	400	½	(400/100) (x) = ½ x = ⅛
2	400	¾	(400/100) (x) = ¾ x = 3/16
3	400	1	(400/100) (x) = 1 x = ¼
4	400	2	(400/100) (x) = 2 x = ½

Use of Charts 26 and 27

Example 3 — Determine Steam Supply Main and Final Velocity

Given:

Friction rate — 2 psi per 100 ft of equivalent pipe
Initial steam pressure — 100 psig
Flow rate — 6750 lb/hr

TABLE 28—TOTAL PRESSURE DROP FOR TWO-PIPE LOW PRESSURE STEAM PIPING SYSTEMS

INITIAL STEAM PRESSURE (psig)	TOTAL PRESSURE DROP IN SUPPLY PIPING (psi)	TOTAL PRESSURE DROP IN RETURN PIPING (psi)
2	½	½
5	1¼	1¼
10	2½	2½
15	3¾	3¾
20	5	5

Find:

1. Size of largest pipe not exceeding design friction rate
2. Steam velocity in pipe.

Solution:

1. Enter bottom of *Chart 26* at 6750 lb/hr and proceed vertically to the 100 psig line (dotted line in *Chart 26*). Then move obliquely to the 0 psig line. From this point proceed vertically up the chart to the smallest pipe size not exceeding 2 psi per 100 ft of equivalent pipe and read 3½ in.
2. The velocity of steam at 0 psig as read from *Chart 26* is 16,000 fpm. Enter the left side of *Chart 27* at 16,000 fpm. Proceed obliquely downward to the 100 psig line and horizontally across to the right side of the chart (dotted line in *Chart 27*). The velocity at 100 psig is 6100 fpm.

Example 4 illustrates a design problem for sizing pipe on a low pressure, vacuum return system.

Example 4 — Sizing Pipe for a Low Pressure, Vacuum Return System

Given:

Six units
Steam requirement per unit — 72 lb/hr
Layout as illustrated in *Figs. 96 thru 98*
Threaded pipe and fittings
Low pressure system — 2 psi

4

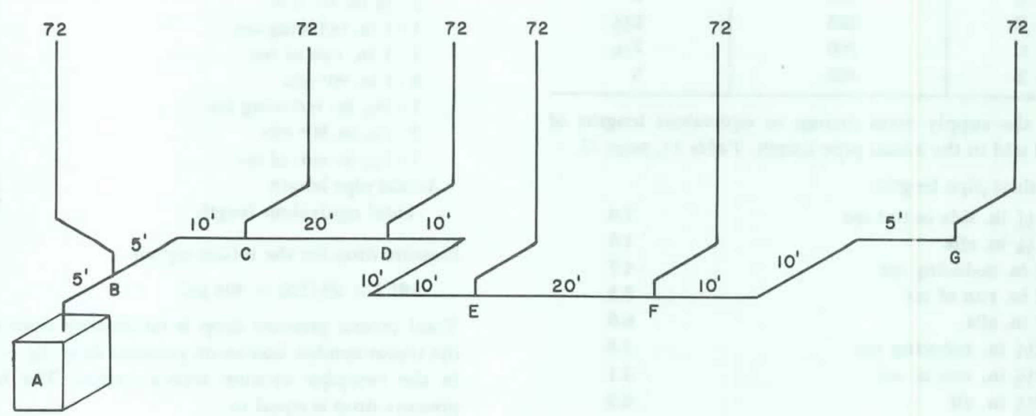


FIG. 96 — LOW PRESSURE STEAM SUPPLY MAIN

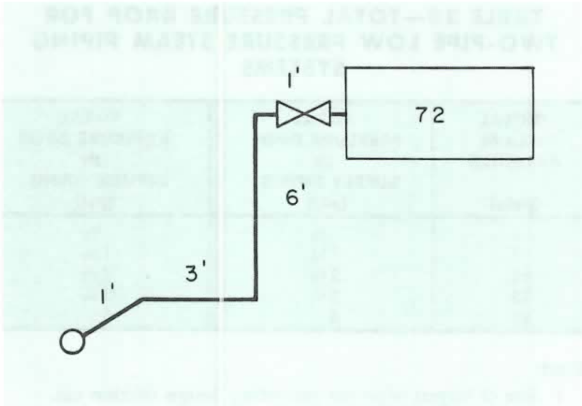


FIG. 97 — LOW PRESSURE RUNOUT AND RISER

Find:

Size of pipe and total pressure drop

Note: Total pressure drop in the system should never exceed one-half the initial pressure. A reasonably small drop is required for quiet operation.

Solution:

Determine the design friction rate by totaling the pipe length and adding 50% of the length for fittings:

$$115 + 11 + 133 = 259$$
$$259 \times .50 = 130$$
$$\overline{389} \text{ ft equiv length}$$

Check pipe sizing recommendations for maximum friction rate from *"Two-Pipe Vacuum System," Item 2,* ⅛-½ psi. Check *Table 28* to determine recommended maximum pressure drop for the supply and return mains (½ psi for each).

Design friction rate = $1/3.89 \times (1/2 + 1/2) = 1/4$ psi per 100 ft. The supply main is sized by starting at the last unit "G" and adding each additional load from unit "G" to the boiler; use *Table 25*. The following tabulation results:

SECTION	STEAM LOAD (lb/hr)	PIPE SIZE (in.)
F - G	72	1¼
E - F	144	2
D - E	216	2
C - D	288	2½
B - C	360	2½
A - B	432	3

Convert the supply main fittings to equivalent lengths of pipe and add to the actual pipe length, *Table 11, page 17.*

Equivalent pipe lengths	
1 - 1¼ in. side outlet tee	7.0
2 - 1¼ in. ells	4.6
1 - 2 in. reducing tee	4.7
1 - 2 in. run of tee	3.3
2 - 2 in. ells	6.6
1 - 2½ in. reducing tee	5.6
1 - 2½ in. run of tee	4.1
1 - 2½ in. ell	6.0
2 - 3 in. ells	15.0
1 - 3 in. reducing tee	7.0
Actual pipe length	115.0
Total equivalent length	172.6 ft.

Pressure drop for the supply main is equal to the equivalent length times pressure drop per 100 ft:

$$172.6 \times .25/100 = .43 \text{ psi}$$

This is within the recommended maximum pressure drop (1 psi) for the supply.

The branch connection for *Fig. 97* is sized in a similar manner at the same friction rate.

From *Table 27* the horizontal runout pipe size for a load of 72 lb is 2½ in. and the vertical riser size is 2 in.

Convert all the fittings to equivalent pipe lengths, and add to the actual pipe length.

Equivalent pipe lengths	
1 - 2½ in. 45° ell	3.2
1 - 2½ in. 90° ell	4.1
1 - 2 in. 90° ell	3.3
1 - 2 in. gate valve	2.3
Actual pipe length	11.0
Total equivalent length	23.9 ft.

Pressure drop for branch runout and riser is

$$23.9 \times .25/100 = .060 \text{ psi}$$

The vacuum return main is sized from *Table 26* by starting at the last unit "G" and adding each additional load between unit "G" and the boiler.

Each riser — 72 lb per hr, ¾ in.

SECTION	STEAM LOAD (lb/hr)	PIPE SIZE (in.)
F - G	72	¾
E - F	144	¾
D - E	216	1
C - D	288	1
B - C	360	1¼
A - B	432	1¼

Convert the return main fittings to equivalent pipe lengths and add to the actual pipe length, *Table 11, page 17.*

Equivalent pipe lengths	
1 - ¾ in. run of tee	1.4
5 - ¾ in. 90° ells	7.0
1 - 1 in. reducing tee	2.3
1 - 1 in. run of tee	1.7
2 - 1 in. 90° ells	3.4
1 - 1¼ in. reducing tee	3.1
3 - 1¼ in. 90° ells	6.9
1 - 1¼ in. run of tee	2.3
Actual pipe length	133.0
Total equivalent length	161.1 ft.

Pressure drop for the return equals

$$161.1 \times .25/100 = .404 \text{ psi}$$

Total return pressure drop is satisfactory since it is within the recommended maximum pressure drop (⅛ — 1 psi) listed in the two-pipe vacuum return system. The total system pressure drop is equal to

$$.430 + .060 + .404 = .894 \text{ psi}$$

This total system pressure drop is within the maximum 2 psi recommended (1 psi for supply and 1 psi for return).

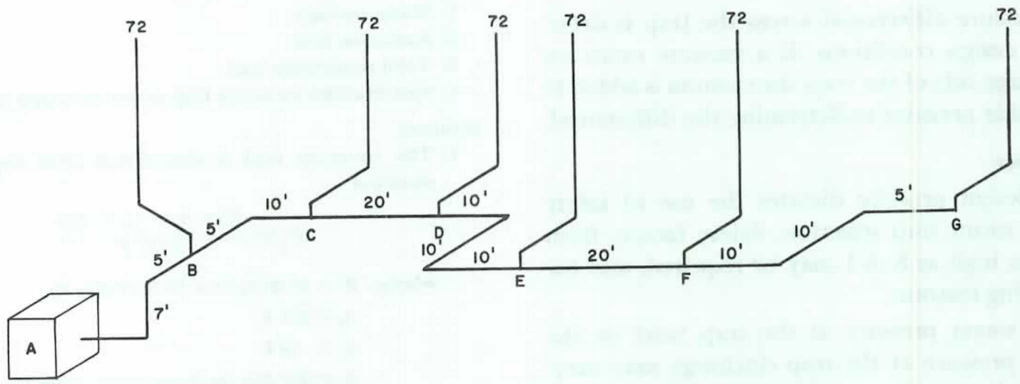


FIG. 98 — LOW PRESSURE VACUUM RETURN

PIPING APPLICATION

The use and selection of steam traps, and condensate and vacuum return pumps are presented in this section.

Also, various steam piping diagrams are illustrated to familiarize the engineer with accepted piping practice.

STEAM TRAP SELECTION

The primary function of a steam trap is to hold steam in a heating apparatus or piping system and allow condensate and air to pass. The steam remains trapped until it gives up its latent heat and changes to condensate. The steam trap size depends on the following:

1. Amount of condensate to be handled by the trap, lb/hr.
2. Pressure differential between inlet and discharge at the trap.
3. Safety factor used to select the trap.

Amount of Condensate

The amount of condensate depends on whether the trap is used for steam mains or risers, or for the heating apparatus.

The selection of the trap for the steam mains or risers is dependent on the pipe warm-up load and the radiation load from the pipe. Warm-up load is the condensate which is formed by heating the pipe surface when the steam is first turned on. For practical purposes the final temperature of the pipe is the steam temperature. Warm-up load is determined from the following equation:

$$C_1 = \frac{W \times (t_f - t_i) \times .114}{h_l \times T}$$

where:

C_1 = Warm-up condensate, lb/hr

W = Total weight of pipe, lb *(Tables 2 and 3, pages 2 and 3)*

t_f = Final pipe temperature, F (steam temp)

t_i = Initial pipe temperature, F (usually room temp)

.114 = Specific heat constant for wrought iron or steel pipe (.092 for copper tubing)

h_l = Latent heat of steam, Btu/lb (from steam tables)

T = Time for warm-up, hr

The radiation load is the condensate formed by unavoidable radiation loss from a bare pipe. This load is determined from the following equation and is based on still air surrounding the steam main or riser:

$$C_2 = \frac{L \times K \times (t_f - t_i)}{h_l}$$

where:

C_2 = Radiation condensate, lb/hr

L = Linear length of pipe, ft

K = Heat transmission coefficient, Btu/(hr) (linear ft) (deg F diff between pipe and surrounding air), *Table 54, Part I*

t_f, t_i, h_l explained previously

The radiation load builds up as the warm-up load drops off under normal operating conditions. The peak occurs at the mid-point of the warm-up cycle. Therefore, one-half of the radiation load is added to the warm-up load to determine the amount of condensate that the trap handles.

4

Pressure Differential

The pressure differential across the trap is determined at design conditions. If a vacuum exists on the discharge side of the trap, the vacuum is added to the inlet side pressure to determine the differential.

Safety Factor

Good design practice dictates the use of safety factors in steam trap selection. Safety factors from 2 to 1 to as high as 8 to 1 may be required, and for the following reasons:

1. The steam pressure at the trap inlet or the back pressure at the trap discharge may vary. This changes the steam trap capacity.
2. If the trap is sized for normal operating load, condensate may back up into the steam lines or apparatus during start-up or warm-up operation.
3. If the steam trap is selected to discharge a full and continuous stream of water, the air could not be vented from the system.

The following guide is used to determine the safety factor:

DESIGN	SAFETY FACTOR
Draining steam main	3 to 1
Draining steam riser	2 to 1
Between boiler and end of main	2 to 1
Before reducing valve	3 to 1
Before shut-off valve (closed part of time)	3 to 1
Draining coils	3 to 1
Draining apparatus	3 to 1

When the steam trap is to be used in a high pressure system, determine whether or not the system is to operate under low pressure conditions at certain intervals such as night time or weekends. If this condition is likely to occur, then an additional safety factor should be considered to account for the lower pressure drop available during night time operation.

Example 5 illustrates the three concepts mentioned previously in trap selection—condensate handled, pressure differential and safety factor.

Example 5 — Steam Trap Selection for Dripping Supply Main to Return Line

Given:

Steam main — 10 in. diam steel pipe, 50 ft long
Steam pressure — 5 psig (227 F)
Room temperature — 70 F db (steam main in space)
Warm-up time — 15 minutes
Steam trap to drip main into vacuum return line (2 in. vacuum gage design)

Find:

1. Warm-up load.
2. Radiation load.
3. Total condensate load.
4. Specifications for steam trap at end of supply main.

Solution:

1. The warm-up load is determined from the following equation:

$$C_1 = \frac{W \times (t_f - t_i) \times .114}{h_l \times T}$$

where: $W = 40.48$ lb/ft × 50 ft (*Table 2*)
$t_f = 227$ F
$t_i = 70$ F
$h_l = 960$ Btu/lb (from *steam tables*)
$T = .25$ hr

$$C_1 = \frac{2024 \times (227 - 70) \times .114}{960 \times .25}$$
$$= 150 \text{ lb/hr of condensate}$$

2. The radiation load is calculated by using the following equation:

$$C_2 = \frac{L \times K \times (t_f - t_i)}{h_l}$$

where: $L = 50$ ft
$K = 6.41$ Btu/(hr) (linear foot) (deg F diff between pipe and air) from *Table 54, Part I*
$t_f = 227$ F
$t_i = 70$ F
$h_l = 960$ Btu/lb (from *steam tables*)

$$C_2 = \frac{50 \times 6.41 \times (227 - 70)}{960}$$
$$= 52 \text{ lb/hr of condensate}$$

3. The total condensate load for steam trap selection is equal to the warm-up load plus one half the radiation load.

Total condensate load $= C_1 + (\frac{1}{2} \times C_2)$
$= 150 + (\frac{1}{2} \times 52)$
$= 176$ lb/hr

4. Steam trap selection is dependent on three factors: condensate handled, safety factor applied to total condensate load, and pressure differential across the steam trap.

The safety factor for a steam trap at the end of the main is 3 to 1 from the table on this page. Applying the 3 to 1 safety factor to the total condensate load, the steam trap would be specified to handle 3 × 176 or 528 lb/hr of condensate.

The pressure differential across the steam trap is determined by the pressure at the inlet and discharge of the steam trap.

Inlet to trap = 5 psig
Discharge of trap = 2 in. vacuum (gage)

When the discharge is under vacuum conditions, the discharge vacuum is added to the inlet pressure for the total pressure differential.

Pressure differential = 6 psi (approx)

Therefore the steam trap is selected for a differential pressure of 6 psi and 528 lb/hr of condensate.

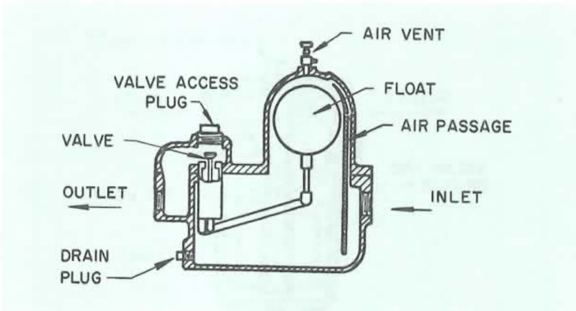


FIG. 99 — FLOAT TRAP

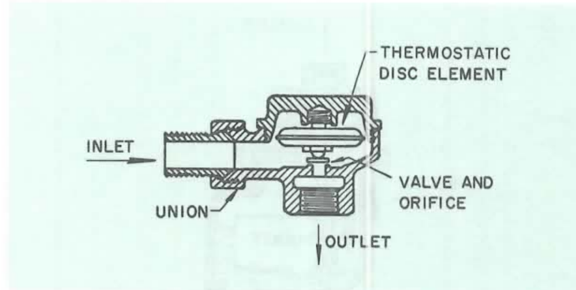


FIG. 101 — THERMOSTATIC TRAP, DISC TYPE

STEAM TRAP TYPES

The types of traps commonly used in steam systems are:

Float	Flash
Thermostatic	Impulse
Float & thermostatic	Lifting
Upright bucket	Boiler return or
Inverted bucket	alternating receiver

The description and use of these various traps are presented in the following pages.

Float Trap

The discharge from the float trap is generally continuous. This type *(Fig. 99)* is used for draining condensate from steam headers, steam heating coils, and other similar equipment. When a float trap is used for draining a low pressure steam system, it should be equipped with a thermostatic air vent.

Thermostatic Trap

The discharge from this type of trap is intermittent. Thermostatic traps are used to drain condensate from radiators, convectors, steam heating coils, unit heaters and other similar equipment. Strainers are normally installed on the inlet side of the steam trap to prevent dirt and pipe scale from entering the trap. On traps used for radiators or convectors, the strainer is usually omitted. *Fig. 100* shows a typical thermostatic trap of the bellows type and *Fig. 101* illustrates a disc type thermostatic trap.

When a thermostatic trap is used for a heating apparatus, at least 2 ft of pipe are provided ahead of the trap to cool the condensate. This permits condensate to cool in the pipe rather than in the coil, and thus maintains maximum coil efficiency.

Thermostatic traps are recommended for low pressure systems up to a maximum of 15 psi. When used in medium or high pressure systems, they must be selected for the specific design temperature. In addition, the system must be operated continuously at that design temperature. *This means no night setback.*

4

Float and Thermostatic Trap

This type of trap is used to drain condensate from blast heaters, steam heating coils, unit heaters and other apparatus. This combination trap *(Fig. 102)* is used where there is a large volume of condensate which would not permit proper operation of a thermostatic trap. Float and thermostatic traps are used in low pressure heating systems up to a maximum of 15 psi.

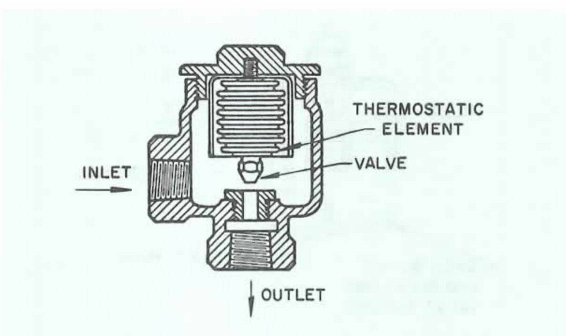


FIG. 100 — THERMOSTATIC TRAP, BELLOWS TYPE

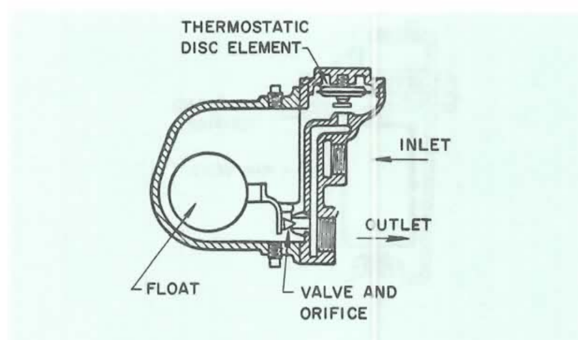


FIG. 102 — TYPICAL FLOAT AND THERMOSTATIC TRAP

Figures 99 thru 102 from Heating Ventilating Air Conditioning Guide 1959. Used by permission.

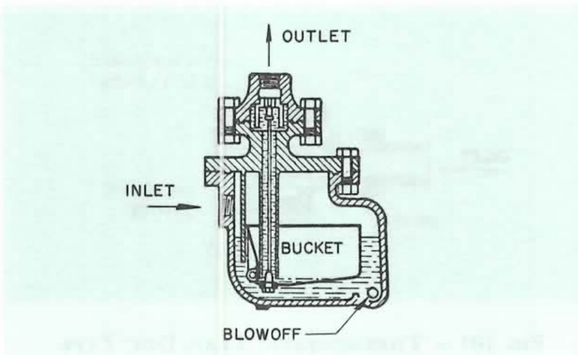


FIG. 103 – UPRIGHT BUCKET TRAP

For medium and high pressure systems, the same limitations as outlined for thermostatic traps apply.

Upright Bucket Trap

The discharge of condensate from this trap *(Fig. 103)* is intermittent. A differential pressure of at least 1 psi between the inlet and the outlet of the trap is normally required to lift the condensate from the bucket to the discharge connection. Upright bucket traps are commonly used to drain condensate and air from the blast coils, steam mains, unit heaters and other equipment. This trap is well suited for systems that have pulsating pressures.

4

Inverted Bucket Trap

The discharge from the inverted bucket trap *(Figs. 104 and 105)* is intermittent and requires a differential pressure between the inlet and discharge of the trap to lift the condensate from the bottom of the trap to the discharge connection.

Bucket traps are used for draining condensate and air from blast coils, unit heaters and steam heating coils. Inverted bucket traps are well suited for draining condensate from steam lines or equipment where an abnormal amount of air is to be discharged and where dirt may drain into the trap.

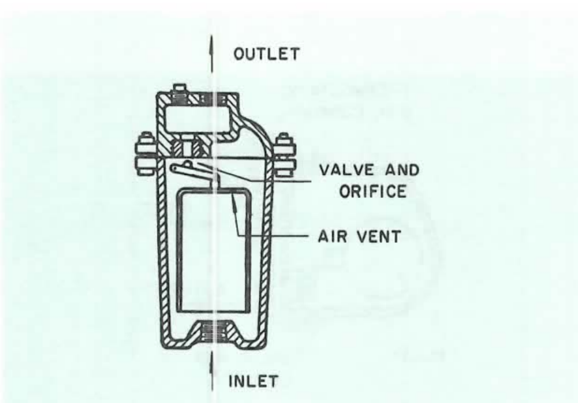


FIG. 104 – INVERTED BUCKET TRAP

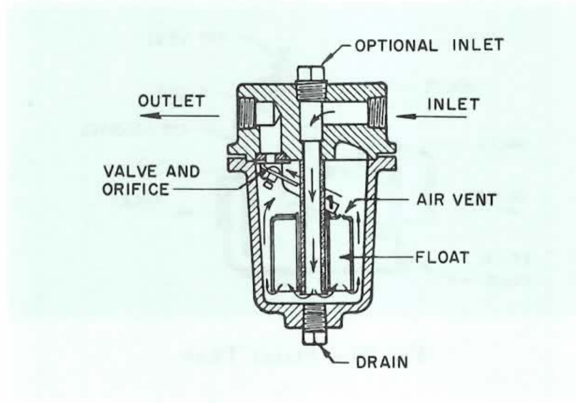


FIG. 105 – INVERTED BUCKET TRAP WITH GUIDE

Flash Trap

The discharge from a flash trap *(Fig. 106)* is intermittent. This type of trap is used only if a pressure differential of 5 psi or more exists between the steam supply and condensate return. Flash traps may be used with unit heaters, steam heating coils, steam lines and other similar equipment.

Impulse Trap

Under normal loads the discharge from this trap *(Fig. 107)* is intermittent. When the load is heavy, however, the discharge is continuous. This type of trap may be used on any equipment where the pressure at the trap outlet does not exceed 25% of the inlet pressure.

Lifting Trap

The lifting trap *(Fig. 108)* is an adaption of the upright bucket trap. This trap can be used on all steam heating systems up to 150 psig. There is an

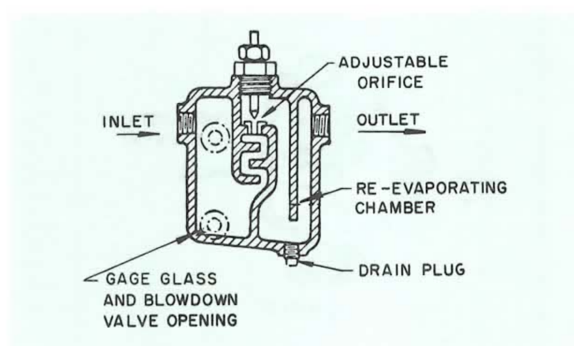


FIG. 106 – FLASH TRAP

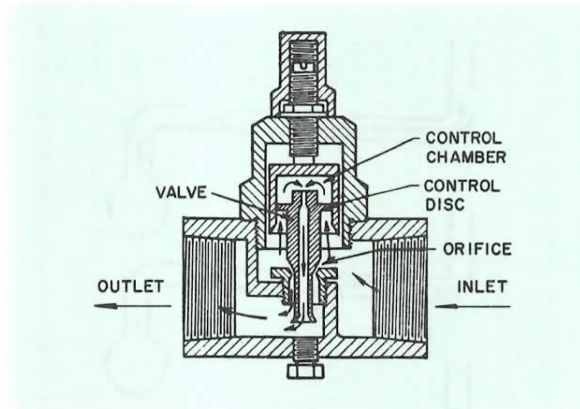


FIG. 107 — IMPULSE TRAP

auxiliary inlet for high pressure steam, as illustrated in the figure, to force the condensate to a point above the trap. This steam is normally at a higher pressure than the steam entering at the regular inlet.

Boiler Return Trap or Alternating Receiver

This type of trap is used to return condensate to a low pressure boiler. The boiler return trap *(Fig. 109)* does not hold steam as do other types, but is an adaption of the lifting trap. It is used in conjunction with a boiler to prevent flooding return mains when excess pressure prevents condensate from returning to the boiler by gravity. The boiler trap collects condensate and equalizes the boiler and trap pressure, enabling the condensate in the trap to flow back to the boiler by gravity.

CONDENSATE RETURN PUMP

Condensate return pumps are used for low pressure, gravity return heating systems. They are normally of the motor driven centrifugal type and have a receiver and automatic float control. Other types

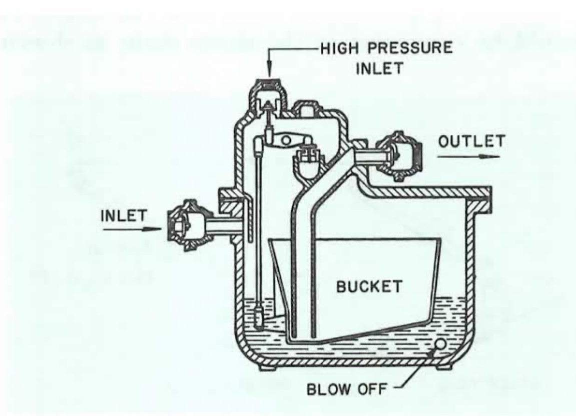


FIG. 108 — LIFTING TRAP

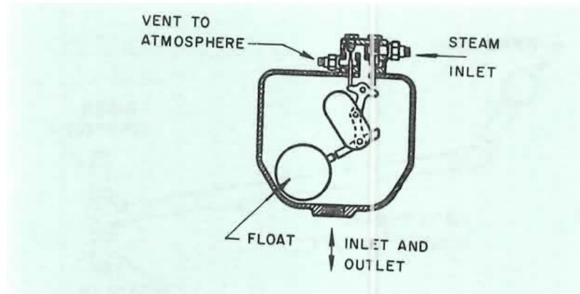


FIG. 109 — BOILER RETURN TRAP OR ALTERNATING RECEIVER

of condensate return pumps are the rotary, screw, turbine and reciprocating pump.

The condensate receiver is sized to prevent large fluctuations of the boiler water line. The storage capacity of the receiver is approximately 1.5 times the amount of condensate returned per minute, and the condensate pump has a capacity of 2.5 to 3 times normal flow. This relationship of pump and receiver to the condensate takes peak condensation load into account.

4

VACUUM PUMP

Vacuum pumps are used on a system where the returns are under a vacuum. The assembly consists of a receiver, separating tank and automatic controls for discharging the condensate to the boiler.

Vacuum pumps are sized in the same manner as condensate pumps for a delivery of 2.5 to 3 times the design condensing rate.

PIPING LAYOUT

Each application has its own layout problem with regard to the equipment location, interference with structural members, steam condensate, steam trap and drip locations. The following steam piping diagrams show the various principles involved. The engineer must use judgment in applying these principles to the application.

Gate valves shown in the diagrams should be used in either the open or closed position, *never for throttling*. Angle and globe valves are recommended for throttling service.

In a one-pipe system gate valves are used since they do not hinder the flow of condensate. Angle valves may be used when they do not restrict the flow of condensate.

All the figures show screwed fittings. Limitations for other fittings are described in *Chapter 1*.

Figures 103 thru 109 from Heating Ventilating Air Conditioning Guide 1959. Used by permission.

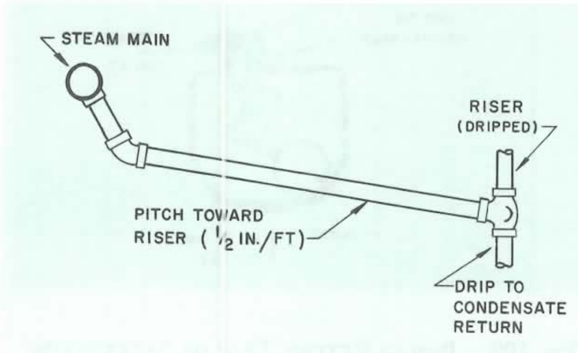


FIG. 110 — CONNECTION TO DRIPPED RISER

Steam Riser

Figures 110 and 111 illustrate steam supply risers connected to mains with runouts. The runout in *Fig. 110* is connected to the bottom portion of the main and is pitched toward the riser to permit condensate to drain from the main. *This layout is used only when the riser is dripped.* If a dry return is used, the riser is dripped thru a steam trap. If a wet return is used, the trap is omitted.

Fig. 111 shows a piping diagram when the steam riser is not dripped. In this instance the runout is connected to the upper portion of the steam main and is pitched to carry condensate from the riser to the main.

Prevention of Water Hammer

If the steam main is pitched incorrectly when the riser is not dripped, water hammer may occur as illustrated in *Fig. 112*. Diagram *"a"* shows the runout partially filled with condensate but with enough space for steam to pass. As the amount of condensate increases and the space decreases, a wave motion is started as illustrated in diagram *"b"*. As the wave or slug of condensate is driven against the turn in the pipe (diagram *"c"*), a hammer noise is

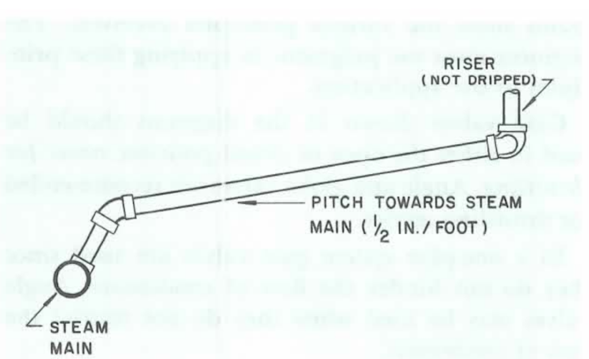


FIG. 111 — CONNECTION TO RISER (NOT DRIPPED)

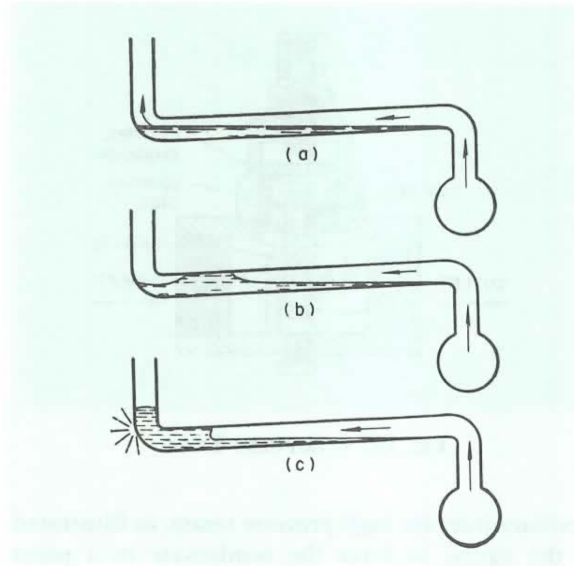


FIG. 112 — WATER HAMMER

caused. This pounding may be of sufficient force to split pipe fittings and damage coils in the system.

The following precautions must be taken to prevent water hammer:

1. Pitch pipes properly.
2. Avoid undrained pockets.
3. Choose a pipe size that prevents high steam velocity when condensate flows opposite to the steam.

Runout Connection to Supply Main

Figure 113 illustrates two methods of connecting runouts to the steam supply main. The method using a 45° ell is somewhat better as it offers less resistance to steam flow.

Expansion and Contraction

Where a riser is two or more floors in height, it should be connected to the steam main as shown

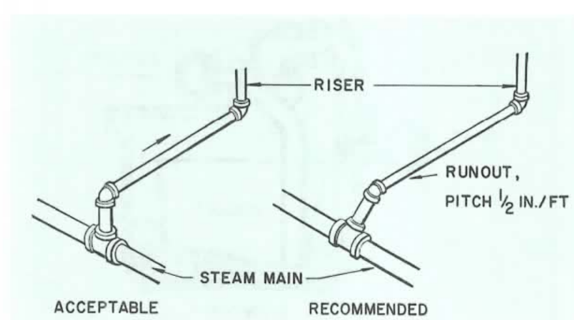


FIG. 113 — RUNOUT CONNECTIONS

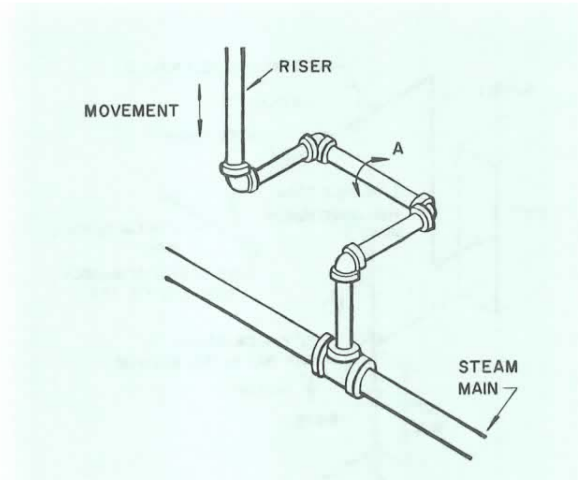


Fig. 114 — Riser Connected to Allow For Expansion

in *Fig. 114*. Point *(A)* is subject to a twisting movement as the riser moves up and down.

Figure 115 shows a method of anchoring the steam riser to allow for expansion and contraction. Movement occurs at *(A)* and *(B)* when the riser moves up and down.

Obstructions

Steam supply mains may be looped over obstructions if a small pipe is run below the obstruction to take care of condensate as illustrated in *Fig. 116*. The reverse procedure is followed for condensate return mains as illustrated in *Fig. 117*. The larger pipe is carried under the obstruction.

Dripping Riser

A steam supply main may be dropped abruptly to a lower level without dripping if the pitch is downward. When the steam main is raised to a higher level, it must be dripped as illustrated in *Fig. 118*.

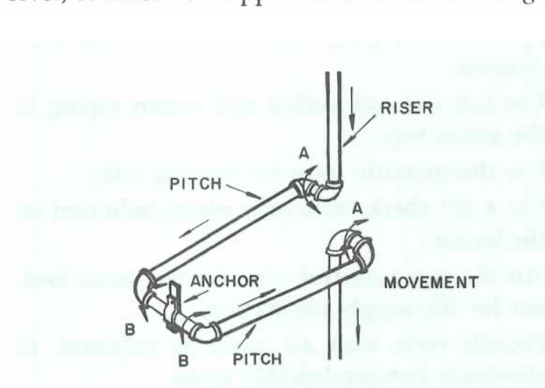


Fig. 115 — Riser Anchor

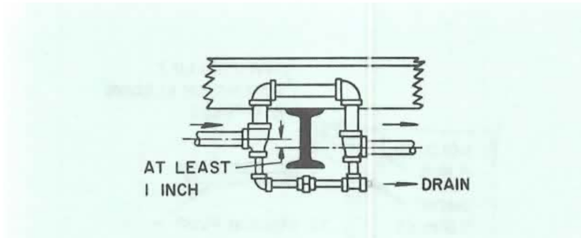


Fig. 116 — Supply Main Loops

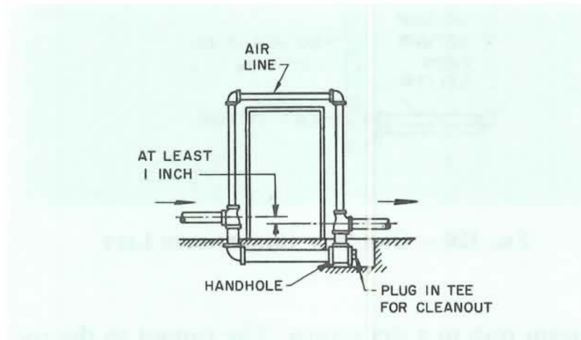


Fig. 117 — Return Main Loop

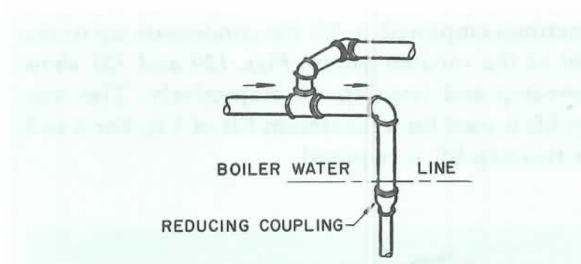


Fig. 118 — Dripping Steam Main

This diagram shows the steam main dripped into a wet return.

Figure 119 is one method of dripping a riser thru

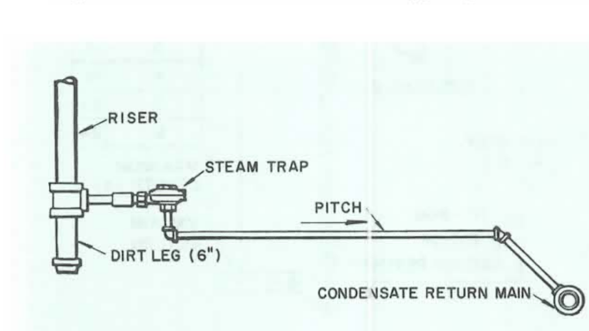


Fig. 119 — Riser Dripped to Dry Return

Figures 116 thru 118 from Heating Ventilating Air Conditioning Guide 1959. Used by permission.

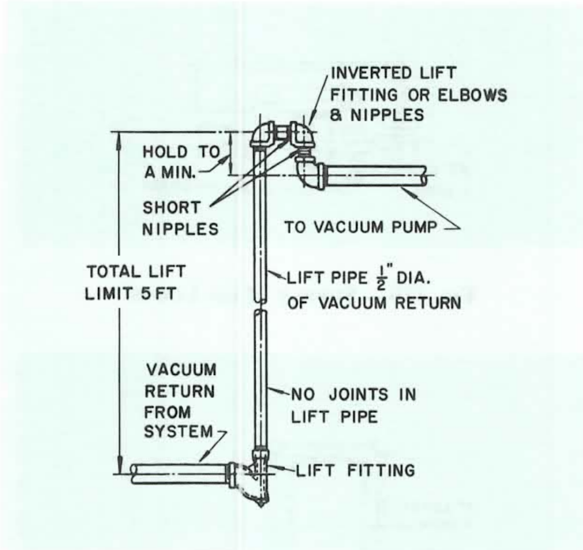
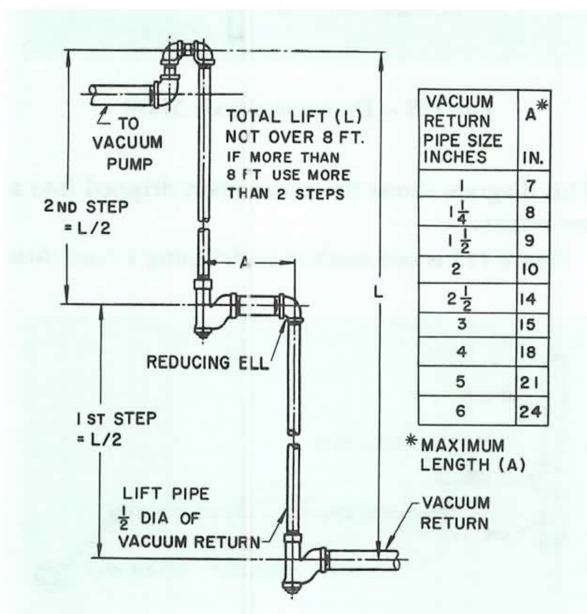


FIG. 120 — ONE-STEP CONDENSATE LIFT

a steam trap to a dry return. The runout to the return main is pitched toward the return main.

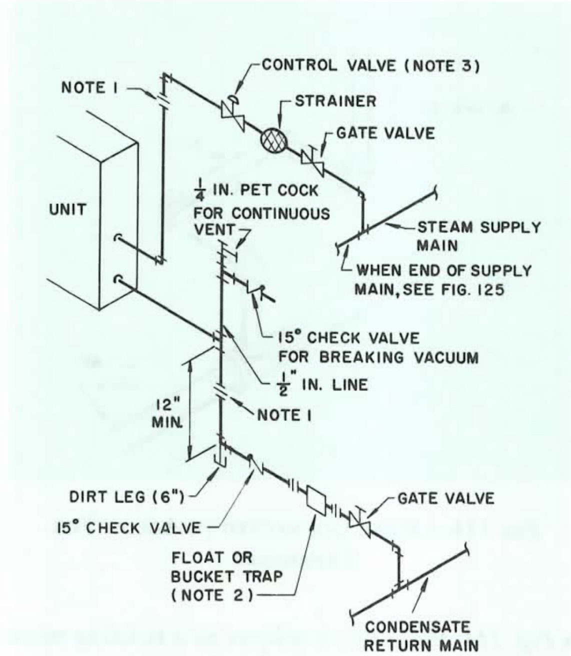
Vacuum Lift

As described under vacuum systems, a lift is sometimes employed to lift the condensate up to the inlet of the vacuum pump. *Figs. 120 and 121* show a one-step and two-step lift respectively. The one-step lift is used for a maximum lift of 5 ft. For 5 to 8 ft a two-step lift is required.



VACUUM RETURN PIPE SIZE INCHES	A* IN.
1	7
1¼	8
1½	9
2	10
2½	14
3	15
4	18
5	21
6	24

FIG. 121 — TWO-STEP CONDENSATE LIFT



NOTES:

1. Flange or union is located to facilitate coil removal.
2. Flash trap may be used if pressure differential between steam and condensate return exceeds 5 psi.
3. When a bypass with control is required, see *Fig. 126*.
4. Dirt leg may be replaced with a strainer. If so, tee on drop can be replaced by a reducing ell.
5. The petcock is not necessary with a bucket trap or any trap which has provision for passing air. The great majority of high or medium pressure returns end in hot wells or deaerators which vent the air.

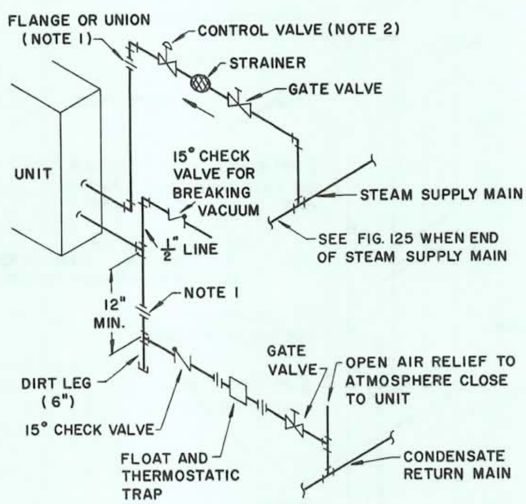
FIG. 122 — HIGH OR MEDIUM PRESSURE COIL PIPING

Steam Coils

Figures 122 thru 131 show methods of piping steam coils in a high or low pressure or vacuum steam piping system. The following general rules are applicable to piping layout for steam coils used in all systems:

1. Use full size coil outlets and return piping to the steam trap.
2. Use thermostatic traps for venting only.
3. Use a 15° check valve only where indicated on the layout.
4. Size the steam control valve for the steam load, not for the supply connection.
5. Provide coils with air vents as required, to eliminate non-condensable gases.
6. Do not drip the steam supply mains into coil sections.

Figures 120 and 121 from Heating Ventilating Air Conditioning Guide 1959. Used by permission.



NOTES:

1. Flange or union is located to facilitate coil removal.
2. When a bypass with control is required, see *Fig. 126*.
3. Check valve is necessary when more than one unit is connected to the return line.
4. Dirt pocket is the same size as unit outlet. If dirt pocket is replaced by a strainer, replace tee with a reducing ell from unit outlet to trap size.

FIG. 123 — SINGLE COIL LOW PRESSURE PIPING GRAVITY RETURN

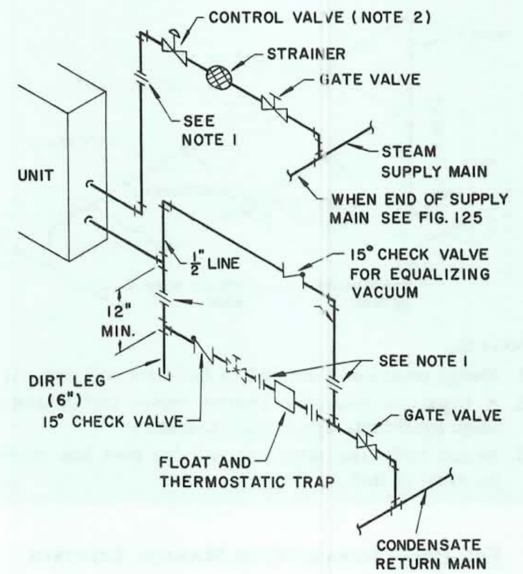
7. Do not pipe tempering coils and reheat coils to a common steam trap.
8. Multiple coils may be piped to a common steam trap if they have the same capacity and the same pressure drop and if the supply is regulated by a control valve.

Piping Single Coils

Figure 122 illustrates a typical steam piping diagram for coils used in either a high or medium pressure system. If the return line is designed for low pressure or vacuum conditions and for a pressure differential of 5 psi or greater from steam to condensate return, a flash trap may be used.

Low pressure steam piping for a single coil is illustrated in *Fig. 123*. This diagram shows an open air relief located after the steam trap close to the unit. This arrangement permits non-condensable gases to vent to the atmosphere.

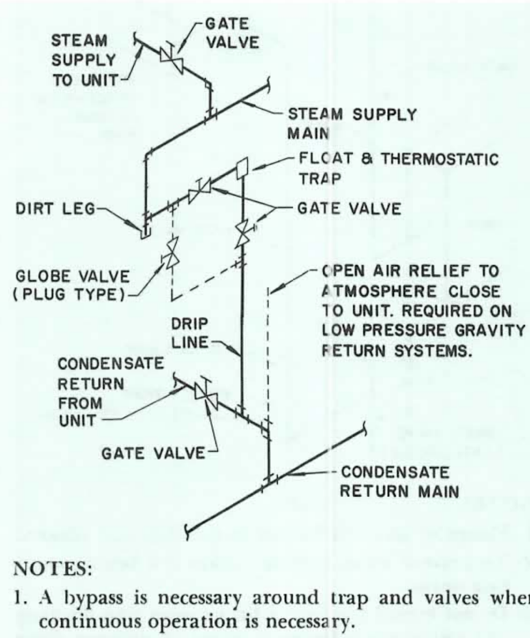
Figure 124 shows the piping layout for a steam coil in a vacuum system. A 15° check valve is used to equalize the vacuum across the steam trap.



NOTES:

1. Flange or union is located to facilitate coil removal.
2. When a bypass with control is specified, see *Fig. 126*.
3. Check valve is necessary when more than one unit is connected to the return line.

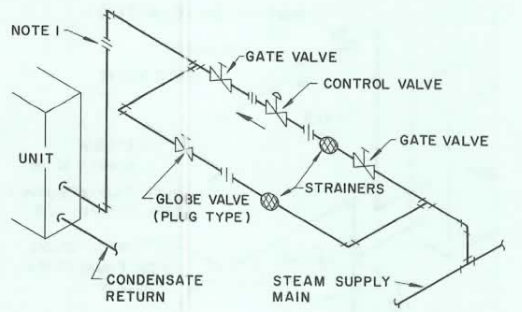
FIG. 124 — VACUUM SYSTEM STEAM COIL PIPING



NOTES:

1. A bypass is necessary around trap and valves when continuous operation is necessary.
2. Bypass to be the same size as trap orifice but never less than ½ inch.

FIG. 125 — DRIPPING STEAM SUPPLY TO CONDENSATE RETURN



NOTES:

1. Flange or union is located to facilitate coil removal.
2. A bypass is necessary around valves and strainer when continuous operation is necessary.
3. Bypass to be the same size as valve port but never less than ½ inch.

FIG. 126 — BYPASS WITH MANUAL CONTROL

Dripping Steam Supply Main

A typical method of dripping the steam supply main to the condensate return is shown in *Fig. 125*.

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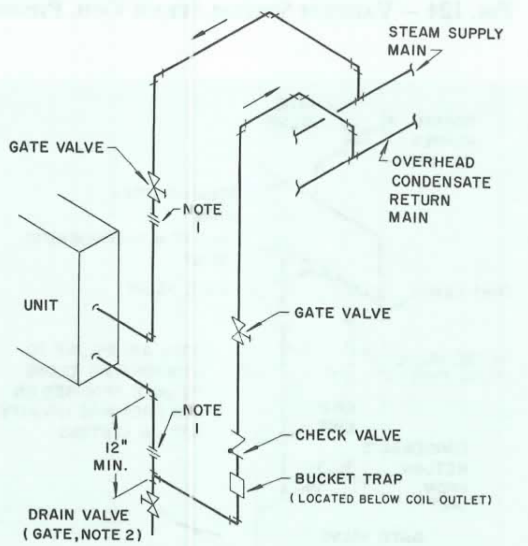
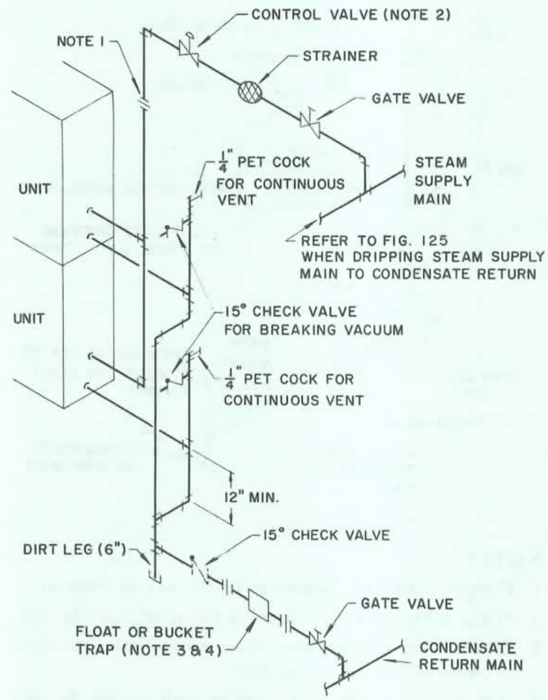


NOTES:

1. Flange or union is located to facilitate coil removal.
2. To prevent water hammer, drain coil before admitting steam.
3. Do not exceed one foot of lift between trap discharge and return main for each pound of pressure differential.
4. *Do not use this arrangement for units handling outside air.*

FIG. 127 — CONDENSATE LIFT TO OVERHEAD RETURN



NOTES:

1. Flange or union is located to facilitate coil removal.
2. When bypass control is required, see *Fig. 126*.
3. Flash trap can be used if pressure differential between supply and condensate return exceeds 5 psi.
4. ***Coils with different pressure drops require individual traps. This is often caused by varying air velocities across the coil bank.***
5. Dirt pocket may be replaced by a strainer. If so, tee on drop can be replaced by a reducing ell.
6. The petcock is not necessary with a bucket trap or any trap which has provision for passing air. The great majority of high pressure return mains terminate in hot wells or deaerators which vent the air.

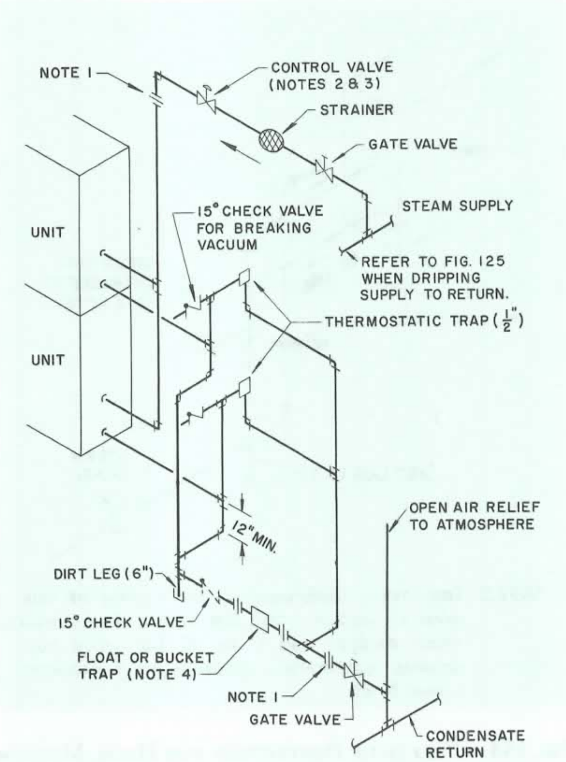
FIG. 128 — MULTIPLE COIL HIGH PRESSURE PIPING

Steam Bypass Control

Frequently a bypass with a manual control valve is required on steam coils. The piping layout for a control bypass with a plug type globe valve as the manual control is shown in *Fig. 126*.

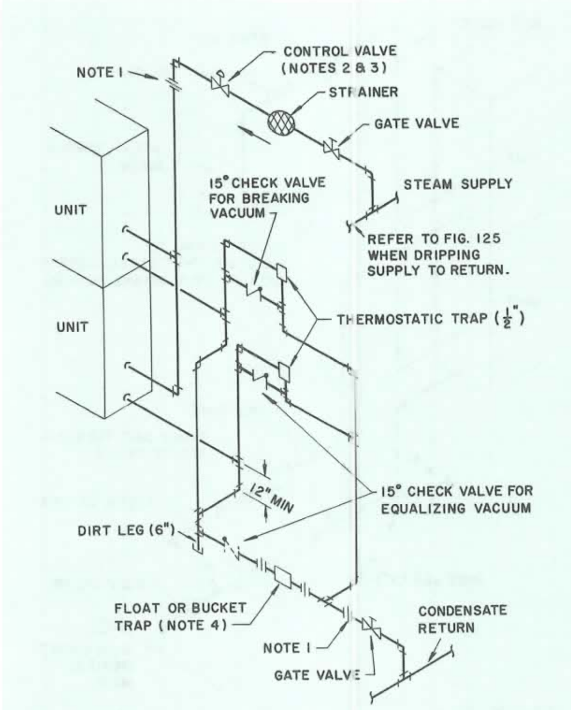
Lifting Condensate to Return Main

A typical layout for lifting condensate to an overhead return is described in *Fig. 127*. The amount of lift possible is determined by the pressure differential between the supply and return sides of the system. The amount of lift is not to exceed one foot for each pound of pressure differential. The maximum lift should not exceed 8 ft.



NOTES:
1. Flange or union is located to facilitate coil removal.
2. See *Fig. 131* when control valve is omitted on multiple coils in parallel air flow.
3. When bypass control is required, see *Fig. 126*.
4. ***Coils with different pressure drops require individual traps. This is often caused by varying air velocities across the coil bank.***

Fig. 129 — Multiple Coil Low Pressure Piping Gravity Return



NOTES:
1. Flange or union is located to facilitate coil removal.
2. See *Fig. 131* when control valve is omitted on multiple coils in parallel air flow.
3. When bypass control is required, see *Fig. 126*.
4. ***Coils with different pressure drops require individual traps. This is often caused by varying air velocities across the coil bank.***

Fig. 130 — Multiple Coil Low Pressure Vacuum System Piping

Piping Multiple Coils

Figures 128 thru 131 show piping layouts for high pressure, low pressure and vacuum systems with multiple coils. If a control valve is not used, each coil must have a separate steam trap as illustrated in *Fig. 131*. This particular layout may be used for a low pressure or vacuum system.

If the coils have different pressure drops or capacities, separate traps are required with or without a control valve in the system.

Boiler Piping

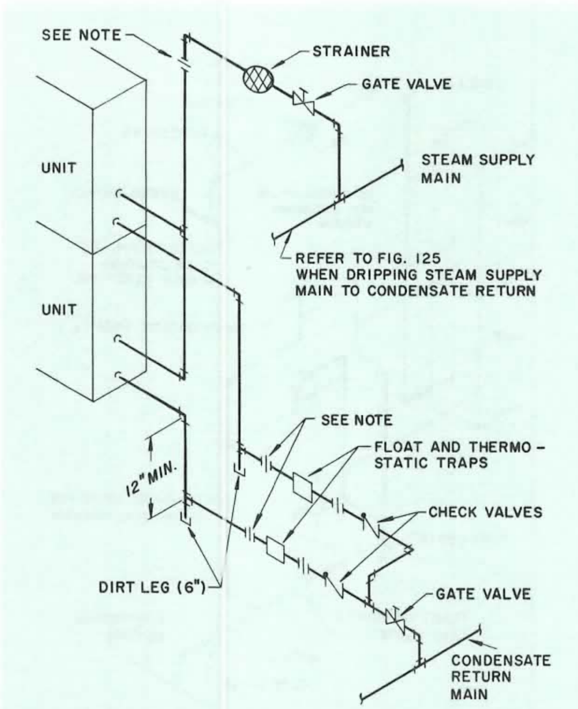
Figure 132 illustrates a suggested layout for a steam plant. This diagram shows parallel boilers and a single boiler using a "Hartford Return Loop."

FREEZE-UP PROTECTION

When steam coils are used for tempering or preheating outdoor air, controls are required to prevent freezing of the coil.

In high, medium, low pressure and vacuum systems, an immersion thermostat is recommended to protect the coil. This protection device controls the fan motor and the outdoor air damper. The immersion thermostat is actuated when the steam supply fails or when the condensate temperature drops below a predetermined level, usually 120 F to 150 F. The thermostat location is shown in *Fig. 133*.

The 15° check valve shown in the various piping diagrams provides a means of equalizing the pressure within the coil when the steam supply shuts off. This check valve is used in addition to the immersion thermostat. The petcock for continuous venting removes non-condensable gases from the coil.



NOTE: Flange or union is located to facilitate coil removal.

Fig. 131 – Low Pressure or Vacuum System Steam Traps

Non-condensable gases can restrict the flow of condensate, causing coil freeze-up.

On a low pressure and vacuum steam heating system, the immersion thermostat may be replaced by a condensate drain with a thermal element (*Fig. 134*). The thermal element opens and drains the coil when the condensate temperature drops below 165 F. Condensate drains are limited to a 5 lb pressure.

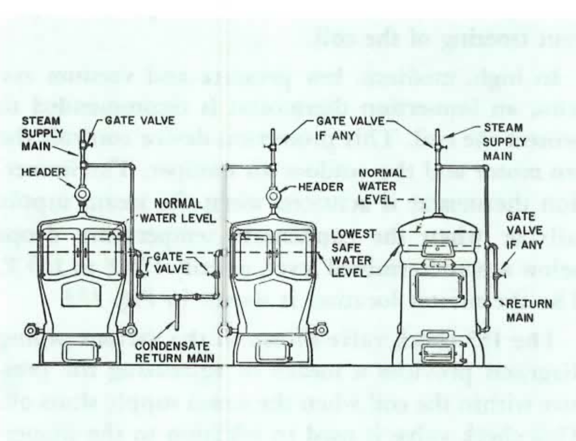
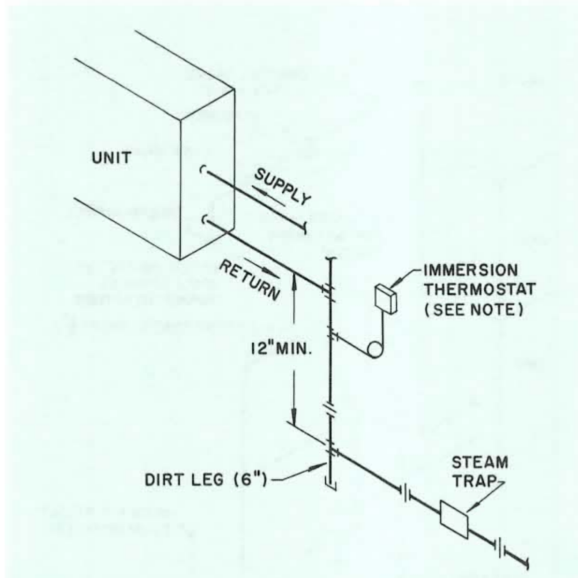
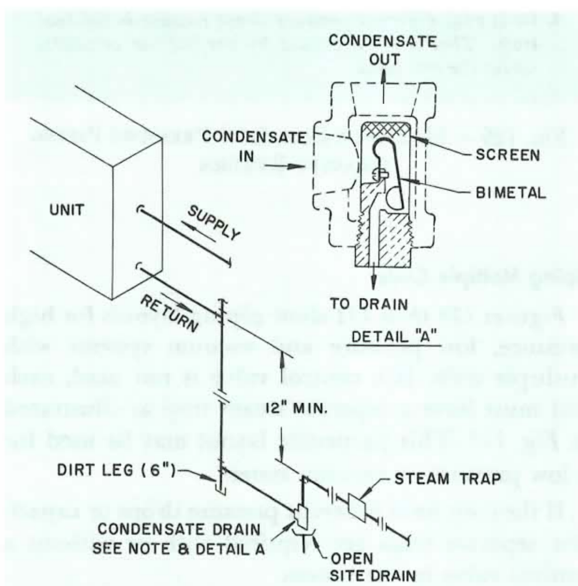


Fig. 132 – "Hartford" Return Loop



NOTE: Immersion thermostat is for control of outdoor air dampers and fan motor. Thermostat closes damper and shuts off fan when condensate temperature drops below a predetermined level.

Fig. 133 – Freeze-up Protection for High, Medium, Low Pressure, and Vacuum Systems



NOTE: Condensate drain drains coil when condensate temperature drops below a predetermined level.

Fig. 134 – Freeze-up Protection for Low Pressure and Vacuum Systems

The following are general recommendations in laying out systems handling outdoor air below 35 F:

1. Do not use overhead returns from the heating unit.
2. Use a strainer in the supply line to avoid collection of scale or other foreign matter in distributing orifices of nonfreeze preheater.
3. Refer to Part 2, Chapter 1, "Preheat Coils"; and Part 6, Chapter 4, "Coil Freeze-Up Protection."

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