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ENGINEERING

EQUIVALENTS & CONVERSION FACTORS

A MULTIPLY	B BY	C TO OBTAIN
Atmospheres	14.697	Pounds per sq. in.
Atmospheres	1.033	Kilograms per sq. cm
Atmospheres	29.92	Inches of mercury
Atmospheres	760	Millimeters of mercury
Atmospheres	407	Inches of water
Atmospheres	33.90	Feet of water
Barrels (petroleum)	42	Gallons
Barrels per day	0.0292	Gallons per minute
Bars-G	14.5	Pounds per sq. in.
Centimeters	0.3937	Inches
Centimeters	0.03281	Feet
Centimeters	0.01	Meters
Centimeters	0.01094	Yards
Cubic centimeters	0.06102	Cubic inches
Cubic feet	7.48055	Gallons
Cubic feet	0.17812	Barrels
Cubic feet per second	448.833	Gallons per minute
Cubic inches	16.39	Cubic centimeters
Cubic inches	0.004329	Gallons
Cubic meters	264.17	Gallons
Cubic meters per hour	4.40	Gallons per minute
Feet	0.3048	Meters
Feet	0.3333	Yards
Feet	30.48	Centimeters
Feet of water	0.882	Inches of mercury
Feet of water	0.433	Pounds per sq. in.
Gallons (U.S.)	3785	Cubic centimeters
Gallons (U.S.)	0.13368	Cubic feet
Gallons (U.S.)	231	Cubic inches
Gallons (Imperial)	277.4	Cubic inches
Gallons (U.S.)	0.833	Gallons (Imperial)
Gallons (U.S.)	3.785	Liters
Gallons of water	8.328	Pounds (at 70°F)
Gallons of liquid per minute	500 x Sp. Gr.	Pounds per hr liquid (at 70°F)
Gallons per minute	0.002228	Cubic feet per second
Horsepower (boiler)	34.5	Pounds water per hr. evaporation
Horsepower (boiler)	33479	Btu per hour
Inches	2.54	Centimeters
Inches	0.0833	Feet
Inches	0.0254	Meters
Inches	0.02778	Yards
Inches of mercury	1.133	Feet of water
Inches of mercury	0.4912	Pounds per sq. in.
Inches of mercury	0.0345	Kilograms per sq. cm
Inches of water	0.03613	Pounds per sq. in.
Inches of water	0.07355	Inches of mercury
Kilograms	2.205	Pounds
Kilograms	0.001102	Short tons (2000 lbs.)
Kilograms per minute	132.3	Pounds per hour
Kilograms per sq. cm	14.22	Pounds per sq. in.
Kilograms per sq. cm	0.9678	Atmospheres
Kilograms per sq. cm	28.96	Inches of mercury
Kilopascals	0.145	Pounds per sq. in.
Liters	1000	Cubic centimeters
Liters	0.2642	Gallons
Liters per hour	0.0044	Gallons per minute
Meters	3.281	Feet
Meters	1.0936	Yards
Meters	100	Centimeters
Meters	39.37	Inches
Megapascals	145.0	Pounds per sq. in.
Pounds	0.0005	Short tons (2000 lbs.)
Pounds	0.4536	Kilograms
Pounds	0.000454	Metric Tons
Pounds	16	Ounces
Pounds per hour	6.32/M.W.	Cubic feet per minute
Pounds per hour liquid	0.002/Sp. Gr.	Gallons per minute liquid (at 70°F)
Pounds per sq. in.	27.684	Inches of water
Pounds per sq. in.	2.307	Feet of water
Pounds per sq. in.	2.036	Inches of mercury
Pounds per sq. in.	0.0703	Kilograms per sq. cm
Pounds per sq. in.	51.71	Millimeters of mercury
Pounds per sq. in.	0.7037	Meters of water
Specific Gravity (of gas or vapors)	28.97	Molecular Wt. (of gas or vapors)
Square centimeters	0.1550	Square inches
Square inches	6.452	Square centimeters
Tons (short ton 2000 lbs.)	907.2	Kilograms
Tons (short ton 2000 lbs.)	0.9072	Metric Tons
Tons (metric) per day	91.8	Pounds per hour
Water (cubic feet)	62.3	Pounds (at 70°F)
Yards	0.9144	Meters
Yards	91.44	Centimeters

This table may be used in two ways:
(1) Multiply the unit under column A by the figure under column B; the result is the unit under column C.
(2) Divide the unit under column C by the figure under column B; the result is the unit under column A.

CAPACITY FORMULAS FOR STEAM LOADS

Definition of Terms and Units:

Q_S = Steam Load or Steam Capacity (lbs/hr)
E = Heat Load (Btu/hr)
m = Amount of water to cool per time (lbs/hr)
C_P = Specific Heat of fluid being heated (Btu/(lb-°F))
C_{P1} = Specific heat of solid being heated (Btu/(lb-°F))
ΔT = Temperature rise (°F)
ΔT_1 = Condensate Temp. - Temp. Set Point (°F)
ΔT_2 = Temperature difference (°F) (Temp. set point - temp. of cooling water)
LH = Latent Heat of Saturated Steam (Btu/lb)
s.g. = Specific gravity of fluid
Q_W = Flow rate of water (GPM)
Q_L = Flow rate of liquid (GPM)
Q_{air} = Flow rate of air (CFM or ft^3/min)
G = Volume of liquid to be heated (gallons)
t = Time to heat product (hours)
W = Weight of material (lbs)
p_{air} = Density of air (lb/ft^3)
500 = 60 min/hr x 8.33 lbs/gal (convert GPM of water to lbs/hr)

APPROXIMATE FORMULAS	EXACT FORMULAS
When Heating Water with Steam $Q_S = \frac{Q_W}{2} \times \Delta T$	$Q_S = \frac{Q_W \times \Delta T \times 500}{LH}$
When Heating Fuel Oil with Steam $Q_S = \frac{Q_L}{4} \times \Delta T$	$Q_S = \frac{Q_L \times \Delta T \times C_p \times 500 \times s.g.}{LH}$
When Heating Air Coils with Steam $Q_S = \frac{Q_{air}}{900} \times \Delta T$	$Q_S = \frac{Q_{air} \times 0.24 \times p_{air} \times 60(min/hr) \times \Delta T}{LH}$
When Heat Load (Btu/hr) is Known $Q_S = \frac{E}{1000}$	$Q_S = \frac{E}{LH}$

When Boiler Output (H.P.) is Known

Q_S = Boiler H.P. x 34.5

Heating Water in Open-Top Tank with Direct Steam Injection

$$Q_S = \frac{G \times \Delta T \times 8.33\ (lbs/gal)}{LH \times t}$$

When Square Feet Equivalent Direct Radiation (EDR) is Known

$$Q_S = \frac{EDR}{4}$$

Heating Liquid in Jacketed Kettles

$$Q_S = \frac{G \times s.g. \times C_P \times \Delta T \times 8.33\ (lbs/gal)}{LH \times t}$$

Condensate Cooling using Water

Step 1: $E = m\Delta T_1$
Step 2: $Q_S = E/\Delta T_2$
Step 3: Water required: $Q_W(GPM) = Q_S/500$

Heating Solids by Direct Steam Injection into Chamber (Platens, Autoclaves, etc.)

$$Q_S = \frac{W \times C_{p1} \times \Delta T}{LH \times t}$$

FORMULAS FOR CONTROL VALVE SIZING FOR LIQUIDS

The following formulas for Control Valve Sizing assume turbulent flow based on liquids similar in viscosity to water, and pipe sizes equal to the size of the valve ports, with no attached fittings.

C_V = Valve Flow Coefficient
Q = Volumetric Flow Rate of Liquid (US GPM)
P_1 = Absolute Inlet Pressure (psia)
P_2 = Absolute Outlet Pressure (psia)
ΔP = Pressure Drop (psi) = $P_1 - P_2$
G = Specific Gravity of the Liquid
P_V = Vapor Pressure of the Liquid

For Normal Flow:

When: $\Delta P < K_C (P_1 - P_V)$:

$$Q = C_V\sqrt{\frac{\Delta P}{G}}$$ Flow Rate based on C_V and ΔP.

$$C_V = Q\sqrt{\frac{G}{\Delta P}}$$ C_V required based on Flow Rate and ΔP.

$$\Delta P = \left[\frac{Q}{C_V}\right]^2 G$$ Pressure drop across valve based on Flow Rate and C_V.

Potential for Cavitation

Cavitation can occur when the pressure inside the control valve drops below the vapor pressure (P_V) of the liquid. Cavitation should be avoided because it restricts flow rate, generates noise and may reduce life expectancy of internal components. When $\Delta P < K_C (P_1 - P_V)$, the Standard Flow Equation will predict performance. When $\Delta P \geq K_C (P_1 - P_V)$, cavitation may occur and the accuracy of the normal flow equation may be reduced.

F_L = The Valve Pressure Recovery Factor.
For Globe Style Control Valve. **F_L** = 0.9

K_C = 0.65 **F_L^2** Based on when a 2% reduction of normal flow rate occurs. (0.65 proportionality conconstant is used for conservative determination of cavitation)

P_V = Vapor Pressure of the Liquid (psia).
(see chart for water at various temperatures.)

Valve Sizing Example:

A control valve is needed that will handle a maximum flow rate of 100 GPM of water @ 180°F. Since the temperature of the water is elevated, cavitation becomes a concern. Determine the maximum pressure drop across the control valve before cavitation will occur. Based on this maximum pressure drop, determine the required minimum C_V value of the control valve.

Conditions of Service:
Q = 100 GPM
T = 180°F Water
P_1 = 50 psig = 64.7 psia

(1) To Determine the ΔP across the valve when cavitation could potentially occur, use the formula $\Delta P_C = K_C (P_1 - P_V)$.

$$\Delta P_C = K_C (P_1 - P_V)$$
$$= 0.53 (64.7 - 7.51)$$
$$= \mathbf{30} \text{ psi}$$

K_C = 0.65 F_L^2 = 0.65 $(0.9)^2$ = 0.53
(for globe value)

P_V = 7.51 @ 180°F for water
(see chart)

(2) Determine the minimum C_V of the control valve at the maximum ΔP of 30 psi.

$$C_V = Q\sqrt{\frac{G}{\Delta P}}$$
$$= 100\sqrt{\frac{0.972}{30}}$$
$$= \mathbf{18}$$

G = 0.972 @ 180°F for water
(see chart)

Water Physical Properties		
Temp. (°F)	**G (Ref. to 60°F)**	**P_V (psia)**
32	1.001	0.09
40	1.001	0.12
50	1.001	0.18
60	1.000	0.26
70	0.999	0.36
80	0.998	0.51
90	0.996	0.70
100	0.994	0.95
120	0.990	1.69
140	0.985	2.89
160	0.979	4.74
180	0.972	7.51
200	0.964	11.5
212	0.959	14.7

NOTE: Since a minimum C_V of 18 was calculated, we could choose a 1-1/2" HB globe style control valve which has a C_V = 22.

FORMULAS FOR VALVE SIZING

Control Valve Sizing for Saturated Steam

The following formulas for Valve Sizing are based on ISA Standard 75.01.01-2007 (60534-2-1 Mod).
The formulas assume pipe sizes equal to the size of the valve ports, with no attached fittings.

Cv = Valve Flow Coefficient
ΔP = Pressure Drop = P_1 - P_2
P_1 = Absolute Inlet Pressure (psia)
P_2 = Absolute Outlet Pressure (psia)
W = Saturated Steam Flow (lbs/hr)
T_1 = Steam Inlet Temperature (°R) (see table)
Z_1 = Steam Compressibility Factor (see table)

γ = Heat Capacity Ratio for Steam = 1.3 (0-300 psig)
F_γ = Heat Capacity Ratio Factor for Steam = $\gamma/1.4 = 1.3/1.4 = 0.93$
x = Pressure Drop Ratio = $\Delta P/P_1$
ΔP_{cr} = Critical Pressure Drop
x_T = Critical Pressure Drop Ratio for Air
x_{cr} = Critical Pressure Drop Ratio for Steam
= $\Delta P_{cr} / P_1 = F_\gamma x_T = 0.93 x_T$

For Sub-critical Flow

$$Cv = \frac{W}{82\, P_1 (1 - x/3x_{cr})\sqrt{x/(T_1 Z_1)}}$$

When $\Delta P/P_1 < x_{cr}$: $x = \Delta P/P_1$

For Critical Flow

$$Cv = \frac{W}{54.6\, P_1 \sqrt{x_{cr}/(T_1 Z_1)}}$$

When $\Delta P/P_1 \geq x_{cr}$

For single-ported globe valve with flow-to-open seating arrangement:
$x_T = \mathbf{0.72}$ $x_{cr} = 0.93x_T = \mathbf{0.67}$

$$Cv = \frac{W}{82\, P_1 (1 - x/2)\sqrt{x/(T_1 Z_1)}}$$

When $\Delta P/P_1 < \mathbf{0.67}$

$$Cv = \frac{W}{54.6\, P_1 \sqrt{0.67/(T_1 Z_1)}}$$

When $\Delta P/P_1 \geq \mathbf{0.67}$

Example: Determine the Cv Value for a Control Valve with 60 psig Inlet Steam Pressure, and 30 psig Outlet Pressure with a Flow Rate of 4,000 lbs/hr.

W = 4,000 lbs/hr
P_1 = 60 psig = 74.7 psia
P_2 = 30 psig = 44.7 psia
ΔP = 30 psi
x = $\Delta P/P_1$ = 30/74.4 = 0.40
x_{cr} = $0.93x_T$ = 0.93 x 0.72 = 0.67
Since $x < x_{cr}$ flow is sub-critical
T_1 = 767 (from table)
Z_1 = 0.955 (from table)

$$Cv = \frac{W}{82\, P_1 (1 - x/2)\sqrt{x/(T_1 Z_1)}}$$

When $\Delta P/P_1 < 0.67$

$$Cv = \frac{4{,}000}{82\,(74.7)(1 - 0.4/2)\sqrt{0.4/(767 \times 0.955)}}$$

$$= \frac{4{,}000}{114.6} = \mathbf{35}$$

Saturated Steam Table

P_1 psig	P_1 psia	T_1 (°R)	Z_1
0	14.7	672	0.985
10	24.7	699	0.978
20	34.7	718	0.973
30	44.7	734	0.968
40	54.7	746	0.963
50	64.7	757	0.959
60	74.7	767	0.955
70	84.7	776	0.951
80	94.7	784	0.947
90	104.7	791	0.943
100	114.7	798	0.940
110	124.7	804	0.936
120	134.7	810	0.933
130	144.7	815	0.930
140	154.7	821	0.927
150	164.7	826	0.923
160	174.7	830	0.920
170	184.7	835	0.917
180	194.7	839	0.915
190	204.7	843	0.912
200	214.7	848	0.909
210	224.7	851	0.906
220	234.7	855	0.903
230	244.7	859	0.901
240	254.7	862	0.898
250	264.7	866	0.895
260	274.7	869	0.893
270	284.7	872	0.890
280	294.7	875	0.888
290	304.7	879	0.885
300	314.7	882	0.883

Properties of Saturated Steam

Pressure	Temp. (°F)	Heat (BTU/lb) Sensible	Heat (BTU/lb) Latent	Heat (BTU/lb) Total	Volume (ft^3/lb) Condensate	Volume (ft^3/lb) Steam
in Hg vac						
25	133	101	1018	1119	0.01626	143.3
20	161	129	1002	1131	0.01640	75.41
15	179	147	991	1138	0.01650	51.41
10	192	160	983	1143	0.01659	39.22
5	203	171	976	1147	0.01666	31.82
(PSIG)						
0	212	180	970	1151	0.01672	26.80
1	215	184	968	1152	0.01674	25.21
2	219	187	966	1153	0.01676	23.79
3	222	190	964	1154	0.01679	22.53
4	224	193	962	1155	0.01681	21.40
5	227	195	961	1156	0.01683	20.38
6	230	198	959	1157	0.01685	19.46
7	232	201	957	1158	0.01687	18.62
8	235	203	956	1159	0.01689	17.85
9	237	206	954	1160	0.01690	17.14
10	239	208	953	1160	0.01692	16.49
12	244	212	950	1162	0.01696	15.33
14	248	216	947	1163	0.01699	14.33
16	252	220	944	1165	0.01702	13.45
18	255	224	942	1166	0.01705	12.68
20	259	228	940	1167	0.01708	11.99
22	262	231	937	1168	0.01711	11.38
24	265	234	935	1169	0.01713	10.83
25	267	236	934	1170	0.01715	10.57
26	268	237	933	1170	0.01716	10.33
28	271	240	931	1171	0.01719	9.874
30	274	243	929	1172	0.01721	9.459
32	277	246	927	1173	0.01723	9.078
34	279	249	925	1174	0.01726	8.728
35	281	250	924	1174	0.01727	8.563
36	282	251	923	1174	0.01728	8.404
38	284	254	922	1175	0.01730	8.104
40	287	256	920	1176	0.01733	7.826
42	289	258	918	1177	0.01735	7.566
44	291	261	916	1177	0.01737	7.323
45	292	262	916	1178	0.01738	7.208
46	294	263	915	1178	0.01739	7.096
48	296	265	913	1178	0.01741	6.883
50	298	267	912	1179	0.01743	6.683
55	303	272	908	1180	0.01748	6.230
60	307	277	905	1182	0.01753	5.837
65	312	282	901	1183	0.01757	5.491
70	316	286	898	1184	0.01761	5.184
75	320	291	895	1185	0.01766	4.911
80	324	295	892	1186	0.01770	4.665
85	328	298	889	1187	0.01774	4.444
90	331	302	886	1188	0.01778	4.242
95	335	306	883	1189	0.01782	4.059
100	338	309	881	1190	0.01785	3.891
105	341	312	878	1190	0.01789	3.736
110	344	316	876	1191	0.01792	3.594
115	347	319	873	1192	0.01796	3.462
120	350	322	871	1192	0.01799	3.340
125	353	325	868	1193	0.01803	3.226
130	356	328	866	1194	0.01806	3.119
135	358	331	864	1194	0.01809	3.020
140	361	333	861	1195	0.01812	2.927
145	363	336	859	1195	0.01815	2.839

(continued)

Pressure (PSIG)	Temp. (°F)	Heat (BTU/lb) Sensible	Heat (BTU/lb) Latent	Heat (BTU/lb) Total	Volume (ft^3/lb) Condensate	Volume (ft^3/lb) Steam
150	366	339	857	1196	0.01818	2.756
155	368	341	855	1196	0.01821	2.678
160	371	344	853	1196	0.01824	2.605
165	373	346	851	1197	0.01827	2.535
170	375	349	849	1197	0.01830	2.469
175	377	351	847	1198	0.01833	2.407
180	380	353	845	1198	0.01835	2.347
185	382	355	843	1198	0.01839	2.291
190	384	358	841	1199	0.01841	2.237
195	386	360	839	1199	0.01844	2.185
200	388	362	837	1199	0.01847	2.136
205	390	364	836	1200	0.01850	2.089
210	392	366	834	1200	0.01852	2.044
215	394	368	832	1200	0.01855	2.001
220	395	370	830	1200	0.01857	1.960
225	397	372	829	1201	0.01860	1.920
230	399	374	827	1201	0.01863	1.882
235	401	376	825	1201	0.01865	1.845
240	403	378	823	1201	0.01868	1.810
245	404	380	822	1202	0.01870	1.776
250	406	382	820	1202	0.01873	1.744
255	408	384	818	1202	0.01875	1.712
260	409	385	817	1202	0.01878	1.682
265	411	387	815	1202	0.01880	1.652
270	413	389	814	1203	0.01882	1.624
275	414	391	812	1203	0.01885	1.596
280	416	392	811	1203	0.01887	1.570
285	417	394	809	1203	0.01889	1.544
290	419	396	808	1203	0.01891	1.520
295	420	397	806	1203	0.01894	1.497
300	422	399	805	1203	0.01896	1.473
310	425	402	802	1204	0.01901	1.428
320	428	405	799	1204	0.01906	1.386
330	430	408	796	1204	0.01910	1.346
340	433	411	793	1204	0.01915	1.309
350	436	414	790	1204	0.01919	1.273
360	438	417	787	1204	0.01923	1.240
370	441	420	785	1204	0.01927	1.207
380	443	423	782	1205	0.01932	1.177
390	446	426	779	1205	0.01936	1.148
400	448	428	777	1205	0.01940	1.120
450	460	441	764	1205	0.01961	0.9992
500	470	453	752	1205	0.01980	0.9010
550	480	464	740	1204	0.02000	0.8195
600	489	475	729	1203	0.02019	0.7509
650	497	485	718	1203	0.02038	0.6922
700	505	494	707	1202	0.02056	0.6415
750	513	504	697	1200	0.02074	0.5971
800	520	512	687	1199	0.02092	0.5580
900	534	529	667	1196	0.02128	0.4922
1000	546	545	648	1192	0.02164	0.4390
1250	574	581	601	1182	0.02256	0.3410
1500	598	614	556	1169	0.02352	0.2740
1750	618	644	510	1155	0.02456	0.2248
2000	637	674	463	1137	0.02572	0.1864
2250	654	703	413	1116	0.02707	0.1554
2500	669	734	358	1092	0.02871	0.1293
2750	683	766	295	1061	0.03097	0.1062
3000	696	805	211	1016	0.03465	0.0835

DRAINING CONDENSATE FROM STEAM MAINS OR STEAM SUPPLY LINES

Charts Assume All Pipes are Insulated (with 80% efficiency)

Warm Up Loads in Pounds of Condensate per hour per 100 ft. of Steam Main

Warm Up Loads are based on a 1 hour warm up time

Outside Temperature at 70˚F. Based on Sch. 40 Pipe up to 250 PSI; Sch. 80 above 250 PSI; Sch. 120, 5″ & Larger, above 800 PSI.

Steam Pressure (PSIG)	Pipe Size														0˚F Correction Factor †
	2″	2 1/2″	3″	4″	5″	6″	8″	10″	12″	14″	16″	18″	20″	24″	
0	6.2	9.7	12.8	18.2	24.6	31.9	48	68	90	107	140	176	207	308	1.50
5	6.9	11.0	14.4	20.4	27.7	35.9	48	77	101	120	157	198	233	324	1.44
10	7.5	11.8	15.5	22.0	29.9	38.8	58	83	109	130	169	213	251	350	1.41
20	8.4	13.4	17.5	24.9	33.8	44	66	93	124	146	191	241	284	396	1.37
40	9.9	15.8	20.6	90.3	39.7	52	78	110	145	172	225	284	334	465	1.32
60	11.0	17.5	22.9	32.6	44	57	86	122	162	192	250	316	372	518	1.29
80	12.0	19.0	24.9	35.3	48	62	93	132	175	208	271	342	403	561	1.27
100	12.8	20.3	26.6	37.8	51	67	100	142	188	222	290	366	431	600	1.26
125	13.7	21.7	28.4	40	55	71	107	152	200	238	310	391	461	642	1.25
150	14.5	23.0	30.0	43	58	75	113	160	212	251	328	414	487	679	1.24
175	15.3	24.2	31.7	45	61	79	119	169	224	265	347	437	514	716	1.23
200	16.0	25.3	33.1	47	64	83	125	177	234	277	362	456	537	748	1.22
250	17.2	27.3	35.8	51	69	89	134	191	252	299	390	492	579	807	1.21
300	25.0	38.3	51	75	104	143	217	322	443	531	682	854	1045	1182	1.20
400	27.8	43	57	83	116	159	241	358	493	590	759	971	1163	1650	1.18
500	30.2	46	62	91	126	173	262	389	535	642	825	1033	1263	1793	1.17
600	32.7	50	67	98	136	187	284	421	579	694	893	1118	1367	1939	1.16
800	38	58	77	113	203	274	455	670	943	1132	1445	1835	2227	3227	1.16
1000	45	64	86	126	227	305	508	748	1052	1263	1612	2047	2485	3601	1.15
1200	52	72	96	140	253	340	566	833	1172	1407	1796	2280	2767	4010	1.14
1400	62	79	106	155	280	376	626	922	1297	1558	1988	2524	3064	4440	1.13
1600	71	87	117	171	309	415	692	1018	1432	1720	2194	2786	3382	4901	1.13
1750	78	94	126	184	333	448	746	1098	1544	1855	2367	3006	3648	5285	1.13
1800	80	97	129	189	341	459	764	1125	1584	1902	2427	3082	3741	5420	1.13

Running Loads in Pounds of Condensate per hour per 100 ft. of Steam Main

Outside Temperature at 70˚F.

Steam Pressure (PSIG)	Pipe Size														0˚F Correction Factor †
	2″	2 1/2″	3″	4″	5″	6″	8″	10″	12″	14″	16″	18″	20″	24″	
10	6	7	9	11	13	16	20	24	29	32	36	39	44	53	1.58
30	8	9	11	14	17	20	26	32	38	42	48	51	57	68	1.50
60	10	12	14	18	24	27	33	41	49	54	62	67	74	89	1.45
100	12	15	18	22	28	33	41	51	61	67	77	83	93	111	1.41
125	13	16	20	24	30	36	45	56	66	73	84	90	101	121	1.39
175	16	19	23	26	33	43	53	66	78	86	98	107	119	141	1.38
250	18	22	27	34	42	50	62	77	92	101	116	126	140	168	1.36
300	20	25	30	37	46	54	68	85	101	111	126	138	154	184	1.35
400	23	28	34	43	53	63	80	99	118	130	148	162	180	216	1.33
500	27	33	39	49	61	73	91	114	135	148	170	185	206	246	1.32
600	30	37	44	55	68	82	103	128	152	167	191	208	232	277	1.31
800	36	44	53	69	85	101	131	164	194	214	244	274	305	365	1.30
1000	43	52	63	82	101	120	156	195	231	254	290	326	363	435	1.27
1200	51	62	75	97	119	142	185	230	274	301	343	386	430	515	1.26
1400	60	73	89	114	141	168	219	273	324	356	407	457	509	610	1.25
1600	69	85	103	132	163	195	253	31	375	412	470	528	588	704	1.22
1750	76	93	113	145	179	213	278	347	411	452	516	580	645	773	1.22
1800	79	96	117	150	185	221	288	358	425	467	534	600	667	800	1.21

† For outdoor temperatures of 0˚F, multiply load value selected from table by correction factor shown.

STEAM PROPERTIES & FLOW CHARACTERISTICS

STEAM CAPACITY TABLES

This chart provides a simple method for sizing steam pipes with velocities in the range of 7,000 to 10,000 ft/min.
(Example: a 1″ pipe with 100 PSIG steam pressure has a flow rate of 672 lbs/hr at a velocity of 7250 ft/min.)

STEAM CAPACITY – Flow in lbs/hr

Pressure (PSIG)	Temp. (°F) (sat.)	FULL-PORT VALVE or PIPE SIZE																
		1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
		VELOCITY (FPM)																
		7062	7094	7125	7187	7250	7312	7375	7500	7625	7750	7875	8000	8250	8500	9000	9500	10000
250	406	176	324	518	916	1498	2615	3591	6018	8731	13700	18620	24360	39470	58730	107700	179200	267700
200	388	143	264	423	748	1223	2135	2932	4913	7128	11190	15200	19880	32230	47950	87910	146300	218500
175	378	127	235	375	664	1086	1895	2603	4361	6328	9931	13490	17650	28610	42560	78040	129800	194000
150	366	111	205	328	580	948	1655	2273	3810	5528	8675	11790	15420	24990	37180	68170	113400	169500
125	353	95	175	280	496	811	1415	1943	3256	4724	7414	10070	13180	21360	31780	58260	96940	144800
100	338	79	145	232	411	672	1173	1612	2701	3919	6150	8356	10930	17720	26360	48330	80410	120100
90	331	72	133	213	377	617	1076	1478	2477	3594	5641	7665	10030	16250	24180	44330	73760	110200
80	324	66	121	194	343	561	979	1345	2254	3270	5132	6973	9122	14780	22000	40330	67100	100300
70	316	59	109	175	309	505	881	1211	2029	2943	4619	6277	8211	13310	19800	36300	60400	90240
60	308	53	97	155	274	449	783	1076	1803	2616	4105	5577	7296	11820	17590	32260	53670	80190
50	298	46	85	136	240	392	684	940	1575	2286	3587	4874	6376	10330	15380	28190	46900	70080
40	287	39	72	116	205	335	585	803	1346	1953	3066	4166	5449	8831	13140	24090	40080	59890
30	274	33	60	96	170	278	485	666	1115	1618	2539	3451	4514	7315	10880	19960	33200	49610
25	267	29	54	86	152	249	434	596	999	1449	2274	3090	4042	6551	9747	17870	29730	44430
20	259	26	47	76	134	219	383	526	881	1279	2006	2726	3566	5780	8600	15770	26230	39200
15	250	22	41	66	116	190	331	455	763	1107	1737	2360	3087	5003	7444	13650	22710	33930
10	240	19	35	55	98	160	279	384	643	933	1464	1990	2603	4218	6276	11510	19150	28610
5	228	15	28	45	79	130	227	311	522	757	1188	1615	2112	3423	5093	9339	15540	23220
0	212	11	21	34	60	97	170	233	391	568	891	1210	1583	2566	3818	7000	11650	17400

This table represents steam loss thru an orifice on a failed open steam trap, assuming that 25% of the flow consists of condensate.

STEAM FLOW – thru various orifice diameters discharging to atmosphere (0 PSIG) in lbs/hr

Orifice Diameter (Inches)	Inlet Pressure (PSIG)												
	2	5	10	15	25	50	75	100	125	150	200	250	300
1/32	0.31	0.47	0.58	0.70	0.94	1.53	2.12	2.70	3.30	3.90	5.10	6.30	7.40
1/16	1.25	1.86	2.30	2.80	3.80	6.10	8.50	10.80	13.20	15.60	20.30	25.10	29.80
3/32	2.81	4.20	5.30	6.30	8.45	13.80	19.10	24.40	29.70	35.10	45.70	56.40	67.00
1/8	4.50	7.50	7.40	11.20	15.00	24.50	34.00	43.40	52.90	62.40	81.30	100.00	119.00
5/32	7.80	11.70	14.60	17.60	23.50	38.30	53.10	67.90	82.70	97.40	127.00	156.00	186.00
3/16	11.20	16.70	21.00	25.30	33.80	55.10	76.40	97.70	119.00	140.00	183.00	226.00	268.00
7/32	15.30	22.90	28.70	34.40	46.00	75.00	104.00	133.00	162.00	191.00	249.00	307.00	365.00
1/4	20.00	29.80	37.40	45.00	60.10	98.00	136.00	173.00	212.00	250.00	325.00	401.00	477.00
9/32	25.20	37.80	47.40	56.90	76.10	124.00	172.00	220.00	268.00	316.00	412.00	507.00	603.00
5/16	31.20	46.60	58.50	70.30	94.00	153.00	212.00	272.00	331.00	390.00	508.00	627.00	745.00
11/32	37.70	56.40	70.70	85.10	114.00	185.00	257.00	329.00	400.00	472.00	615.00	758.00	901.00
3/8	44.90	67.10	84.20	101.00	135.00	221.00	306.00	391.00	478.00	561.00	732.00	902.00	1073.00
13/32	52.70	78.80	98.80	119.00	159.00	259.00	359.00	459.00	559.00	659.00	859.00	1059.00	1259.00
7/16	61.10	91.40	115.00	138.00	184.00	300.00	416.00	532.00	648.00	764.00	996.00	1228.00	1460.00
15/32	70.20	105.00	131.00	158.00	211.00	344.00	478.00	611.00	744.00	877.00	1144.00	1410.00	1676.00
1/2	79.80	119.00	150.00	180.00	241.00	392.00	544.00	695.00	847.00	998.00	1301.00	1604.00	1907.00

SIZING STEAM PIPES • Steam Velocity Chart (Schedule 40 pipe)

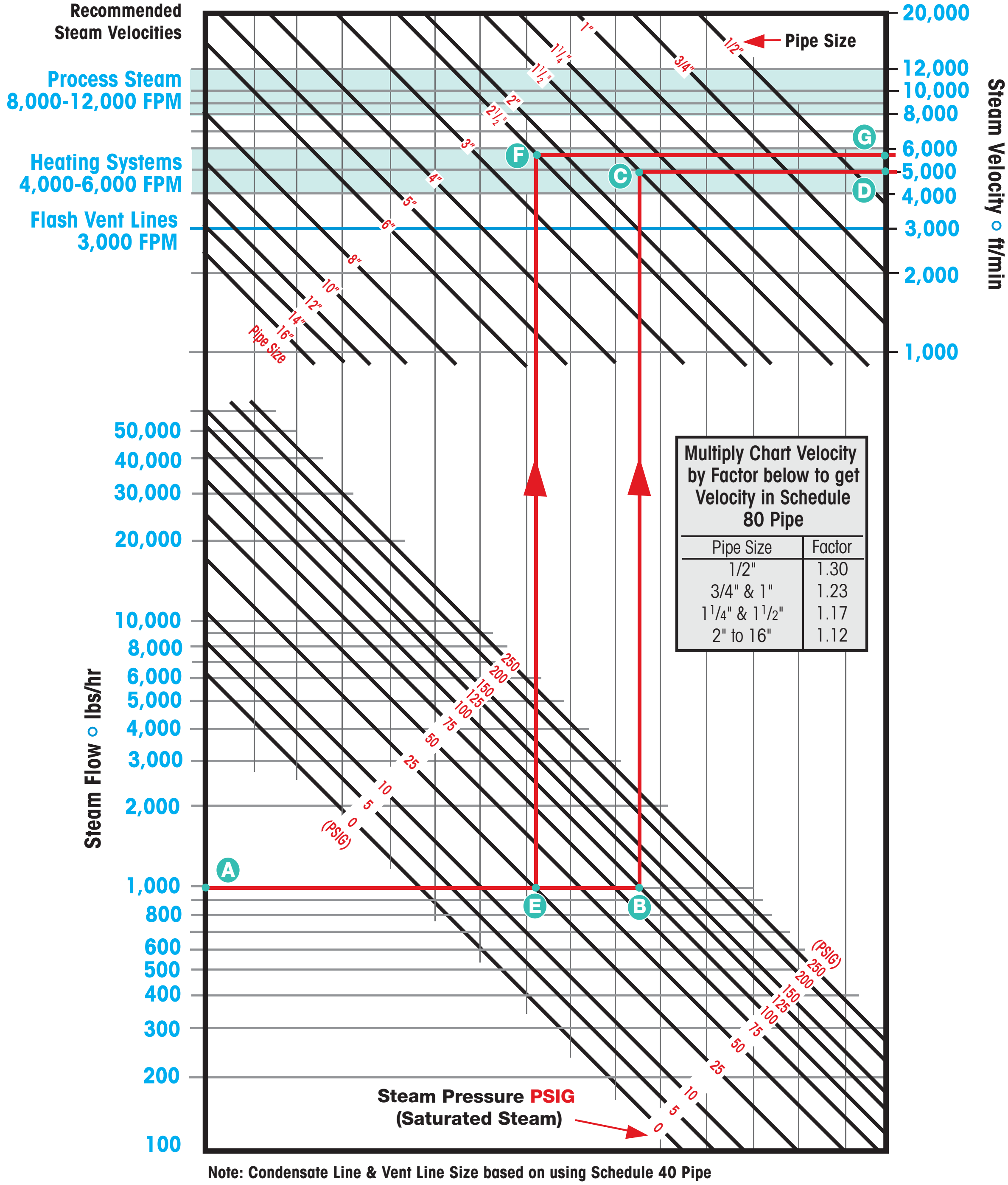
Saturated steam lines should be sized for a steam velocity of 4800 to 7200 ft/min.

Piping on pressure reducing stations should be sized for the same steam velocity on both sides of the regulator. This usually results in having a regulator smaller than the piping and having larger piping on the downstream side of the regulator.

Example using Steam Velocity Chart:
100 PSIG Inlet Pressure to control valve; 25 PSIG Outlet Pressure; 1,000 lbs/hr flow rate; Determine pipe size required.

Upstream Piping:
Enter Velocity Chart at A 1000 lbs/hr. Follow line to B 100 PSIG Inlet Pressure. Follow line vertically upwards to C $1^1/2''$ Pipe Diameter. Steam Velocity at D shows 4800 ft/min.

Downstream Piping:
Enter Velocity Chart at A 1000 lbs/hr. Follow line to E 25 PSIG Outlet Pressure. Follow line vertically upwards to F $2^1/2''$ Pipe Diameter. Steam Velocity at G shows 5500 ft/min.



Pipe Size	Factor
1/2"	1.30
3/4" & 1"	1.23
$1^1/4$" & $1^1/2$"	1.17
2" to 16"	1.12

Multiply Chart Velocity by Factor below to get Velocity in Schedule 80 Pipe

Note: Condensate Line & Vent Line Size based on using Schedule 40 Pipe

SIZING OF CONDENSATE RETURN LINE, FLASH TANK DIAMETER & VENT LINE

- Velocity of Flash Steam in Condensate Return Lines should be between 4000 and 6000 ft/min.
- Velocity in Flash Tank should be less than 600 ft/min.
- Velocity in a Vent Pipe should be less than 4000 ft/min.

Example: **A steam trap with a 150 PSIG steam inlet pressure is being discharged into a flash tank operating at 20 PSIG. The condensate load on the trap is 3200 lbs/hr.**

Problem:

(1) Determine the size of the condensate return line from the trap to the flash tank based on velocities of 4,000 - 6,000 ft/min.

(2) Determine the diameter of the flash tank based on velocities less than 600 ft/min.

(3) Determine the size of the vent line on the flash tank based on velocities less than 4000 ft/min.

Solution:

The accepted practice of determining condensate return pipe sizing is to base the size of the return pipe on the amount of flash steam in the return line. This is due to the fact that the volume of flash steam is over 1,000 times greater than the equivalent volume of liquid condensate. Therefore, the flash steam is the dominant factor affecting flow in the return line. We must first calculate the amount of flash steam produced.

From the **Percent Flash Steam Table** we find that 11.8% of the condensate will flash into steam.
Therefore .118 X 3200 = 377 lbs/hr of flash steam will be produced in the condensate return line and flash tank.

Enter **Condensate Line, Flash Tank & Vent Line Sizing** chart at A **377 lbs/hr**.

Move horizontally to point B 20 PSIG Flash Tank Pressure.

Move vertically upwards to point D to determine a **5" Flash Tank Diameter** is needed to keep velocities less than 600 ft/min.

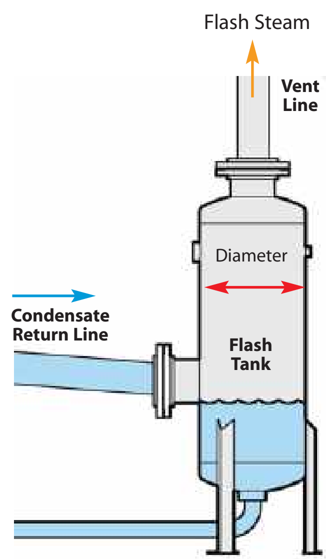
Continue to move vertically to point E to determine that the Vent Line on the Flash Tank should be **2" Diameter** in order to keep velocities less than 4,000 ft/min.

Continue to move vertically to point C to determine that the **Condensate Line should be 1 1/2" Diameter** in order to maintain condensate return line velocities between 4000 and 6000 ft/min.

PERCENT (%) FLASH STEAM TABLE

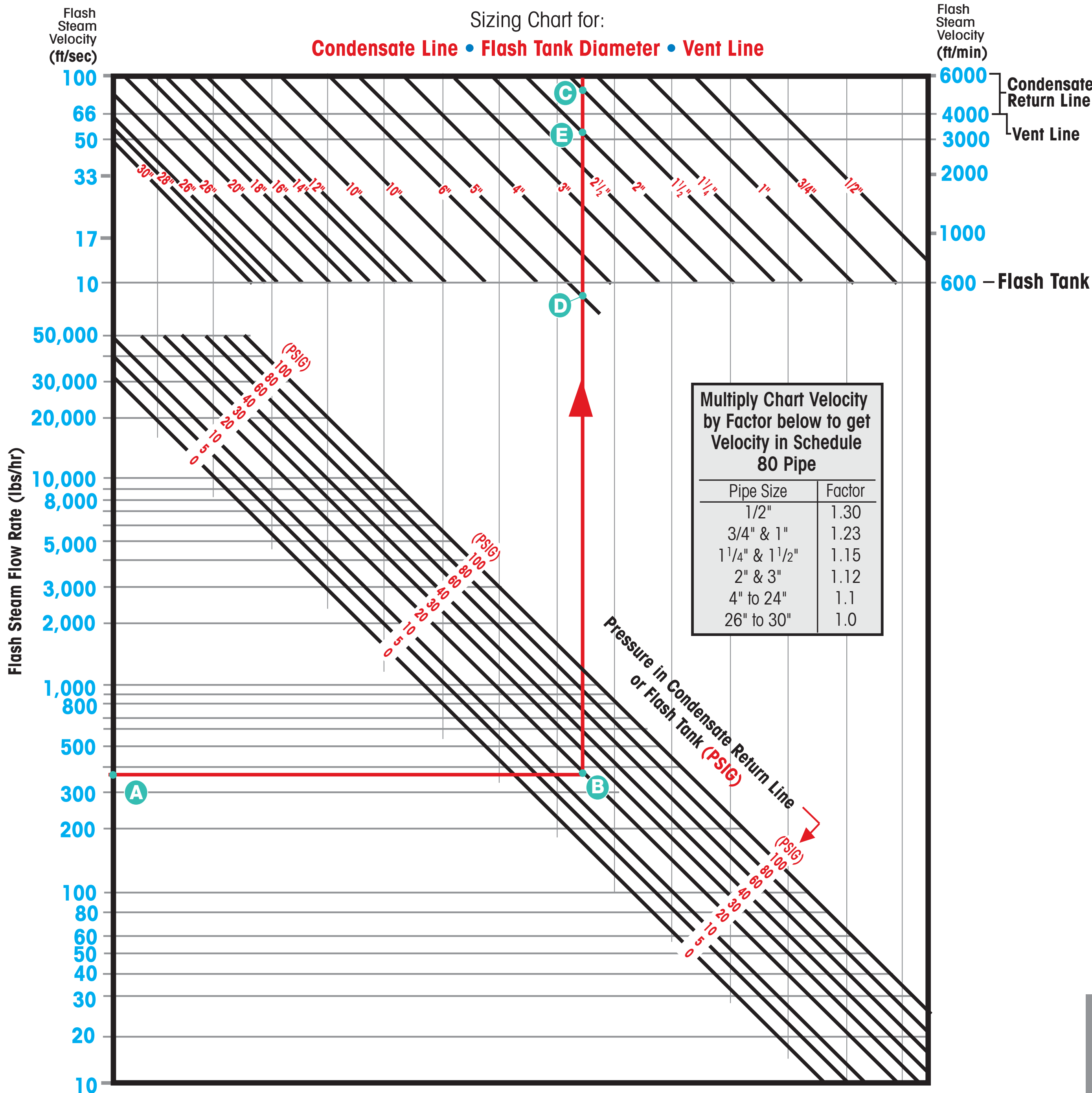
Percent Flash Steam produced when condensate is discharged to atmosphere (0 PSIG) or into a flash tank controlled at various pressures

Condensate Pressure (PSIG)	Flash Tank Pressure (PSIG)								
	0	5	10	20	30	40	60	80	100
5	1.6	0.0							
10	2.9	1.3	0.0						
15	3.9	2.4	1.1						
20	4.9	3.3	2.1	0.0					
30	6.5	5.0	3.7	1.7	0.0				
40	7.8	6.3	5.1	3.0	1.4	0.0			
60	10.0	8.5	7.3	5.3	3.7	2.3	0.0		
80	11.8	10.3	9.1	7.1	5.5	4.2	1.9	0.0	
100	13.3	11.8	10.6	8.7	7.1	5.8	3.5	1.6	0.0
125	14.9	13.5	12.3	10.4	8.8	7.5	5.3	3.4	1.8
150	16.3	14.9	13.7	11.8	10.3	9.0	6.8	4.9	3.3
200	18.7	17.3	16.2	14.3	12.8	11.5	9.4	7.6	6.0
250	20.8	19.4	18.2	16.4	14.9	13.7	11.5	9.8	8.2
300	22.5	21.2	20.0	18.2	16.8	15.5	13.4	11.7	10.2
350	24.1	22.8	21.7	19.9	18.4	17.2	15.1	13.4	11.9
400	25.6	24.2	23.1	21.4	19.9	18.7	16.7	15.0	13.5



SIZING OF CONDENSATE RETURN LINE, FLASH TANK DIAMETER & VENT LINE

- Velocity of Flash Steam in Condensate Return Lines should be between 4000 and 6000 ft/min.
- Velocity in Flash Tank should be less than 600 ft/min.
- Velocity in a Vent Pipe should be less than 4000 ft/min.



Multiply Chart Velocity by Factor below to get Velocity in Schedule 80 Pipe

Pipe Size	Factor
1/2"	1.30
3/4" & 1"	1.23
1 1/4" & 1 1/2"	1.15
2" & 3"	1.12
4" to 24"	1.1
26" to 30"	1.0

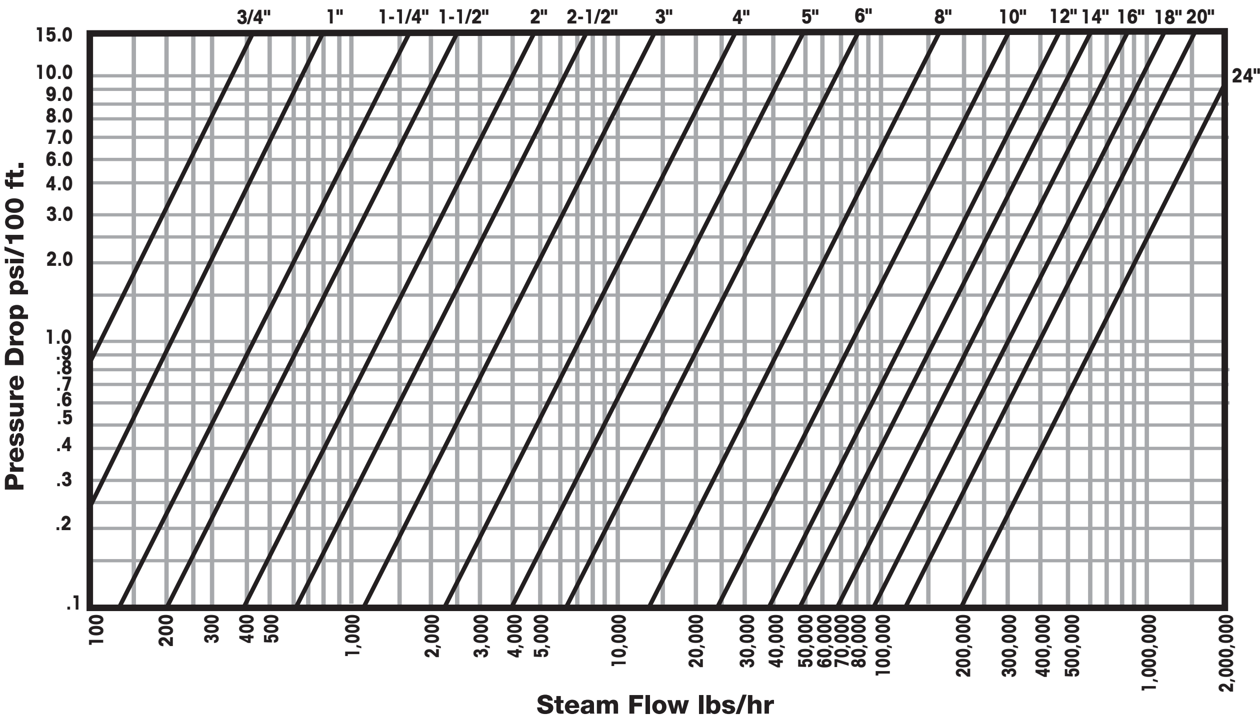
ENGINEERING

PRESSURE DROP IN SCHEDULE 40 PIPE

100 PSIG Saturated Steam

For other pressures use correction factors

psi	0	0	5	10	15	20	30	40	60	75	90	100	110	125	150	175	200	225	250	300	350	400	500	600
factor	6.9	6.0	5.2	4.3	3.6	3.1	2.4	2.0	1.5	1.3	1.1	1.0	0.92	0.83	0.70	0.62	0.55	0.49	0.45	0.38	0.33	0.29	0.23	0.19



Flow of Water thru Schedule 40 Steel Pipe

Pressure Drop per 1,000 Feet of Schedule 40 Steel Pipe

Flow Rate (GPM)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)	Velocity (ft/s)	Pressure Drop (PSI)
	1"																	
1	0.37	0.49	**1 1/4"**															
2	0.74	1.70	0.43	0.45	**1 1/2"**													
3	1.12	3.53	0.64	0.94	0.47	0.44												
4	1.49	5.94	0.86	1.55	0.63	0.74												
5	1.86	9.02	1.07	2.36	0.79	1.12	**2"**											
6	2.24	12.25	1.28	3.30	0.95	1.53	0.57	0.46										
8	2.98	21.1	1.72	5.52	1.26	2.63	0.76	0.75	**2 1/2"**									
10	3.72	30.8	2.14	8.34	1.57	3.86	0.96	1.14	0.67	0.48								
15	5.60	64.6	3.21	17.6	2.36	8.13	1.43	2.33	1.00	0.99	**3"**							
20	7.44	110.5	4.29	29.1	3.15	13.5	1.91	3.86	1.34	1.64	0.87	0.59	**3 1/2"**					
25			5.36	43.7	3.94	20.2	2.39	5.81	1.68	2.48	1.08	0.67	0.81	0.42				
30			6.43	62.9	4.72	29.1	2.87	8.04	2.01	3.43	1.30	1.21	0.97	0.60	**4"**			
35			7.51	82.5	5.51	38.2	3.35	10.95	2.35	4.49	1.52	1.58	1.14	0.79	0.88	0.42		
40					6.30	47.8	3.82	13.7	2.68	5.88	1.74	2.06	1.30	1.00	1.01	0.53		
45					7.08	60.6	4.30	17.4	3.00	7.14	1.95	2.51	1.46	1.21	1.13	0.67		
50					7.87	74.7	4.78	20.6	3.35	8.82	2.17	3.10	1.62	1.44	1.26	0.80		
60							5.74	29.6	4.02	12.2	2.60	4.29	1.95	2.07	1.51	1.10	**5"**	
70							6.69	38.6	4.69	15.3	3.04	5.84	2.27	2.71	1.76	1.50	1.12	0.48
80							7.65	50.3	5.37	21.7	3.48	7.62	2.59	3.53	2.01	1.87	1.28	0.63
90	**6"**						8.60	63.6	6.04	26.1	3.91	9.22	2.92	4.46	2.26	2.37	1.44	0.80
100	1.11	0.39					9.56	75.1	6.71	32.3	4.34	11.4	3.24	5.27	2.52	2.81	1.60	0.95
125	1.39	0.56							8.38	48.2	5.42	17.1	4.05	7.86	3.15	4.38	2.00	1.48
150	1.67	0.78							10.06	60.4	6.51	23.5	4.86	11.3	3.78	6.02	2.41	2.04
175	1.94	1.06							11.73	90.0	7.59	32.0	5.67	14.7	4.41	8.20	2.81	2.78
200	2.22	1.32	**8"**								8.68	39.7	6.48	19.2	5.04	10.2	3.21	3.46
225	2.50	1.66	1.44	0.44							9.77	50.2	7.29	23.1	5.67	12.9	3.61	4.37
250	2.78	2.05	1.60	0.55							10.85	61.9	8.10	28.5	6.30	15.9	4.01	5.14
275	3.06	2.36	1.76	0.63							11.94	75.0	8.91	34.4	6.93	18.3	4.41	6.22
300	3.33	2.80	1.92	0.75							13.02	84.7	9.72	40.9	7.56	21.8	4.81	7.41
325	3.61	3.29	2.08	0.88									10.53	45.5	8.18	25.5	5.21	8.25
350	3.89	3.62	2.24	0.97									11.35	52.7	8.82	29.7	5.61	9.57
375	4.16	4.16	2.40	1.11									12.17	60.7	9.45	32.3	6.01	11.0
400	4.44	4.72	2.56	1.27									12.97	68.9	10.08	36.7	6.41	12.5
425	4.72	5.34	2.72	1.43									13.78	77.8	10.70	41.5	6.82	14.1
450	5.00	5.96	2.88	1.60	**10"**								14.59	87.3	11.33	46.5	7.22	15.0
475	5.27	6.66	3.04	1.69	1.93	0.30									11.96	51.7	7.62	16.7
500	5.55	7.39	3.20	1.87	2.04	0.63									12.59	57.3	8.02	18.5
550	6.11	8.94	3.53	2.26	2.24	0.70									13.84	69.3	8.82	22.4
600	6.66	10.6	3.85	2.70	2.44	0.86									15.10	82.5	9.62	26.7
650	7.21	11.8	4.17	3.16	2.65	1.01	**12"**										10.42	31.3
700	7.77	13.7	4.49	3.69	2.85	1.18	2.01	0.48									11.22	36.3
750	8.32	15.7	4.81	4.21	3.05	1.35	2.15	0.55									12.02	41.6
800	8.88	17.8	5.13	4.79	3.26	1.54	2.29	0.62	**14"**								12.82	44.7
850	9.44	20.2	5.45	5.11	3.46	1.74	2.44	0.70	2.02	0.43							13.62	50.5
900	10.00	22.6	5.77	5.73	3.66	1.94	2.58	0.79	2.14	0.48							14.42	56.6
950	10.55	23.7	6.09	6.38	3.87	2.23	2.72	0.88	2.25	0.53							15.22	63.1
1,000	11.10	26.3	6.41	7.08	4.07	2.40	2.87	0.98	2.38	0.59							16.02	70.0
1,100	12.22	31.8	7.05	8.56	4.48	2.74	3.16	1.18	2.61	0.68	**16"**						17.63	84.6
1,200	13.32	37.8	7.69	10.2	4.88	3.27	3.45	1.40	2.85	0.81	2.18	0.40						
1,300	14.43	44.4	8.33	11.3	5.29	3.86	3.73	1.56	3.09	0.95	2.36	0.47						
1,400	15.54	51.5	8.97	13.0	5.70	4.44	4.02	1.80	3.32	1.10	2.54	0.54						
1,500	16.65	55.5	9.62	15.0	6.10	5.11	4.30	2.07	3.55	1.19	2.73	0.62						
1,600	17.76	63.1	10.26	17.0	6.51	5.46	4.59	2.36	3.80	1.35	2.91	0.71	**18"**					
1,800	19.98	79.8	11.54	21.6	7.32	6.91	5.16	2.98	4.27	1.71	3.27	0.85	2.58	0.48				
2,000	22.20	98.5	12.83	25.0	8.13	8.54	5.73	3.47	4.74	2.11	3.63	1.05	2.88	0.56				
2,500			16.03	39.0	10.18	12.5	7.17	5.41	5.92	3.09	4.54	1.63	3.59	0.88	**20"**			
3,000			19.24	52.4	12.21	18.0	8.60	7.31	7.12	4.45	5.45	2.21	4.31	1.27	3.45	0.73		
3,500			22.43	71.4	14.25	22.9	10.03	9.95	8.32	6.18	6.35	3.00	5.03	1.52	4.03	0.94	**24"**	
4,000			25.65	93.3	16.28	29.9	11.48	13.0	9.49	7.92	7.25	3.92	5.74	2.12	4.61	1.22	3.19	0.51
4,500					18.31	37.8	12.90	15.4	10.67	9.36	8.17	4.97	6.47	2.50	5.19	1.55	3.59	0.60
5,000					20.35	46.7	14.34	18.9	11.84	11.6	9.08	5.72	7.17	3.08	5.76	1.78	3.99	0.74
6,000					24.42	67.2	17.21	27.3	14.32	15.4	10.88	8.24	8.62	4.45	6.92	2.57	4.80	1.00
7,000					28.50	85.1	20.08	37.2	16.60	21.0	12.69	12.2	10.04	6.06	8.06	3.50	5.68	1.36
8,000							22.95	45.1	18.98	27.4	14.52	13.6	11.48	7.34	9.23	4.57	6.38	1.78
9,000							25.80	57.0	21.35	34.7	16.32	17.2	12.92	9.20	10.37	5.36	7.19	2.25
10,000							28.63	70.4	23.75	42.9	18.16	21.2	14.37	11.5	11.53	6.63	7.96	2.78
12,000							34.38	93.6	28.50	61.8	21.80	30.9	17.23	16.5	13.83	9.54	9.57	3.71
14,000									33.20	84.0	25.42	41.6	20.10	20.7	16.14	12.0	11.18	5.05
16,000											29.05	54.4	22.96	27.1	18.43	15.7	12.77	6.60

PIPE, FITTING & FLANGE SPECIFICATIONS

PIPE DATA TABLE

Pipe Size (in.)	Outside Diameter (in.)	Weight Class	Carbon Steel Schedule	Stainless Steel Schedule	Wall Thickness (in.)	Inside Diameter (in.)	Circum. (Ext.) (in.)	Circum. (Int.) (in.)	Flow Area (sq. in.)	Weight of Pipe (lbs/Ft.)	Weight of Water (lbs/Ft.)	Gallons of Water per Ft.	Section Modulus	Pipe Size (in.)
1/8	0.405	–	–	10S	.049	.307	1.27	.96	.074	.19	.032	.004	.00437	1/8
		STD	40	40S	.068	.269		.85	.057	.24	.025	.003	.00523	
		XS	80	80S	.095	.215		.68	.036	.31	.016	.002	.00602	
1/4	0.540	–	–	10S	.065	.410	1.70	1.29	.132	.33	.057	.007	.01032	1/4
		STD	40	40S	.088	.364		1.14	.104	.42	.045	.005	.01227	
		XS	80	80S	.119	.302		.95	.072	.54	.031	.004	.01395	
3/8	0.675	–	–	10S	.065	.545	2.12	1.71	.233	.42	.101	.012	.01736	3/8
		STD	40	40S	.091	.493		1.55	.191	.57	.083	.010	.0216	
		XS	80	80S	.126	.423		1.33	.141	.74	.061	.007	.0255	
1/2	0.840	–	–	5S	.065	.710	2.64	2.23	.396	.54	.172	.021	.0285	1/2
		–	–	10S	.083	.674		2.12	.357	.67	.155	.019	.0341	
		STD	40	40S	.109	.622		1.95	.304	.85	.132	.016	.0407	
		XS	80	80S	.147	.546		1.72	.234	1.09	.102	.012	.0478	
		–	160	–	.187	.466		1.46	.171	1.31	.074	.009	.0527	
		XXS	–	–	.294	.252		.79	.050	1.71	.022	.003	.0577	
3/4	1.050	–	–	5S	.065	.920	3.30	2.89	.665	.69	.288	.035	.0467	3/4
		–	–	10S	.083	.884		2.78	.614	.86	.266	.032	.0566	
		STD	40	40S	.113	.824		2.59	.533	1.13	.231	.028	.0706	
		XS	80	80S	.154	.742		2.33	.433	1.47	.188	.022	.0853	
		–	160	–	.219	.612		1.92	.296	1.94	.128	.015	.1004	
		XXS	–	–	.308	.434		1.36	.148	2.44	.064	.008	.1103	
1	1.315	–	–	5S	.065	1.185	4.13	3.72	1.103	.87	.478	.057	.0760	1
		–	–	10S	.109	1.097		3.45	.945	1.40	.409	.049	.1151	
		STD	40	40S	.133	1.049		3.30	.864	1.68	.375	.045	.1328	
		XS	80	80S	.179	.957		3.01	.719	2.17	.312	.037	.1606	
		–	160	–	.250	.815		2.56	.522	2.84	.230	.027	.1903	
		XXS	–	–	.358	.599		1.88	.282	3.66	.122	.015	.2136	
1 1/4	1.660	–	–	5S	.065	1.530	5.22	4.81	1.839	1.11	.797	.096	.1250	1 1/4
		–	–	10S	.109	1.442		4.53	1.633	1.81	.708	.085	.1934	
		STD	40	40S	.140	1.380		4.34	1.495	2.27	.649	.078	.2346	
		XS	80	80S	.191	1.278		4.02	1.283	3.00	.555	.067	.2913	
		–	160	–	.250	1.160		3.64	1.057	3.76	.458	.055	.3421	
		XXS	–	–	.382	.896		2.81	.630	5.21	.273	.033	.4110	
1 1/2	1.900	–	–	5S	.065	1.770	5.97	5.56	2.461	1.28	1.066	.128	.1662	1 1/2
		–	–	10S	.109	1.682		5.28	2.222	2.09	.963	.115	.2598	
		STD	40	40S	.145	1.610		5.06	2.036	2.72	.882	.106	.3262	
		XS	80	80S	.200	1.500		4.71	1.767	3.63	.765	.092	.4118	
		–	160	–	.281	1.338		4.20	1.406	4.86	.608	.073	.5078	
		XXS	–	–	.400	1.100		3.46	.950	6.41	.420	.049	.5977	
2	2.375	–	–	5S	.065	2.245	7.46	7.05	3.958	1.61	1.72	.206	.2652	2
		–	–	10S	.109	2.157		6.78	3.654	2.64	1.58	.190	.4204	
		STD	40	40S	.154	2.067		6.49	3.355	3.65	1.45	.174	.5606	
		XS	80	80S	.218	1.939		6.09	2.953	5.02	1.28	.153	.7309	
		–	160	–	.344	1.687		5.30	2.241	7.46	.97	.116	.9790	
		XXS	–	–	.436	1.503		4.72	1.774	9.03	.77	.092	1.1040	
2 1/2	2.875	–	–	5S	.083	2.709	9.03	8.51	5.764	2.48	2.50	.299	.4939	2 1/2
		–	–	10S	.120	2.635		8.28	5.453	3.53	2.36	.283	.6868	
		STD	40	40S	.203	2.469		7.76	4.788	5.79	2.07	.249	1.064	
		XS	80	80S	.276	2.323		7.30	4.238	7.66	1.87	.220	1.339	
		–	160	–	.375	2.125		6.68	3.546	10.01	1.54	.184	1.638	
		XXS	–	–	.552	1.771		5.56	2.464	13.69	1.07	.128	1.997	
3	3.500	–	–	5S	.083	3.334	11.00	10.47	8.730	3.03	3.78	.454	.744	3
		–	–	10S	.120	3.260		10.24	8.347	4.33	3.62	.434	1.041	
		STD	40	40S	.216	3.068		9.64	7.393	7.58	3.20	.384	1.724	
		XS	80	80S	.300	2.900		9.11	6.605	10.25	2.86	.343	2.225	
		–	160	–	.438	2.624		8.24	5.408	14.32	2.35	.281	2.876	
		XXS	–	–	.600	2.300		7.23	4.155	18.58	1.80	.216	3.424	
4	4.500	–	–	5S	.083	4.334	14.14	13.62	14.75	3.92	6.39	.766	1.249	4
		–	–	10S	.120	4.260		13.38	14.25	5.61	6.18	.740	1.761	
		STD	40	40S	.237	4.026		12.65	12.73	10.79	5.50	.661	3.214	
		XS	80	80S	.337	3.826		12.02	11.50	14.98	4.98	.597	4.271	
		–	120	–	.438	3.624		11.39	10.31	19.00	4.47	.536	5.178	
		–	160	–	.531	3.438		10.80	9.28	22.51	4.02	.482	5.898	
		XXS	–	–	.674	3.152		9.90	7.80	27.54	3.38	.405	6.791	
5	5.563	–	–	5S	.109	5.345	17.48	16.79	22.44	6.36	9.72	1.17	2.498	5
		–	–	10S	.134	5.295		16.63	22.02	7.77	9.54	1.14	3.029	
		STD	40	40S	.258	5.047		15.86	20.01	14.62	8.67	1.04	5.451	
		XS	80	80S	.375	4.813		15.12	18.19	20.78	7.88	.945	7.431	
		–	120	–	.500	4.563		14.34	16.35	27.04	7.09	.849	9.250	
		–	160	–	.625	4.313		13.55	14.61	32.96	6.33	.759	10.796	
		XXS	–	–	.750	4.063		12.76	12.97	38.55	5.61	.674	12.090	

PIPE DATA TABLE *(continued)*

Pipe Size (in.)	Outside Diameter (in.)	Weight Class	Carbon Steel Schedule	Stainless Steel Schedule	Wall Thickness (in.)	Inside Diameter (in.)	Circum. (Ext.) (in.)	Circum. (Ext.) (in.)	Flow Area (sq. in.)	Weight of Pipe (lbs/Ft.)	Weight of Water (lbs/Ft.)	Gallons of Water per Ft.	Section Modulus	Pipe Size (in.)
		–	–	5S	.109	6.407		20.13	32.24	7.60	13.97	1.68	3.576	
		–	–	10S	.134	6.357		19.97	31.74	9.29	13.75	1.65	4.346	
		STD	40	40S	.280	6.065		19.05	28.89	18.97	12.51	1.50	8.496	
6	6.625	XS	80	80S	.432	5.761	20.81	18.10	26.07	28.57	11.29	1.35	12.22	6
		–	120	–	.562	5.501		17.28	23.77	36.39	10.30	1.24	14.98	
		–	160	–	.719	5.187		16.30	21.15	45.35	9.16	1.10	17.81	
		XXS	–	–	.864	4.897		15.38	18.84	53.16	8.16	.978	20.02	
		–	–	5S	.109	8.407		26.41	55.51	9.93	24.06	2.88	6.131	
		–	–	10S	.148	8.329		26.17	54.48	13.40	23.61	2.83	8.212	
		–	20	–	.250	8.125		25.53	51.85	22.36	22.47	2.69	13.39	
		–	30	–	.277	8.071		25.36	51.16	24.70	22.17	2.66	14.69	
		STD	40	40S	.322	7.981		25.07	50.03	28.55	21.70	2.60	16.81	
8	8.625	–	60	–	.406	7.813	27.10	24.55	47.94	35.64	20.77	2.49	20.58	8
		XS	80	80S	.500	7.625		23.95	45.66	43.39	19.78	2.37	24.51	
		–	100	–	.594	7.437		23.36	43.46	50.95	18.83	2.26	28.14	
		–	120	–	.719	7.187		22.58	40.59	60.71	17.59	2.11	32.58	
		–	140	–	.812	7.001		21.99	38.50	67.76	16.68	2.00	35.65	
		XXS	–	–	.875	6.875		21.60	37.12	72.42	16.10	1.93	37.56	
		–	160	–	.906	6.813		21.40	36.46	74.69	15.80	1.89	38.48	
		–	–	5S	.134	10.482		32.93	86.29	15.19	37.39	4.48	11.71	
		–	–	10S	.165	10.420		32.74	85.28	18.65	36.95	4.43	14.30	
		–	20	–	.250	10.250		32.20	82.52	28.04	35.76	4.29	21.15	
		–	30	–	.307	10.136		31.84	80.69	34.24	34.96	4.19	25.57	
		STD	40	40S	.365	10.020		31.48	78.86	40.48	34.20	4.10	29.90	
10	10.750	XS	60	80S	.500	9.750	33.77	30.63	74.66	54.74	32.35	3.88	39.43	10
		–	80	–	.594	9.562		30.04	71.84	64.43	31.13	3.73	45.54	
		–	100	–	.719	9.312		29.25	68.13	77.03	29.53	3.54	53.22	
		–	120	–	.844	9.062		28.47	64.53	89.29	27.96	3.35	60.32	
		XXS	140	–	1.000	8.750		27.49	60.13	104.13	26.06	3.12	68.43	
		–	160	–	1.125	8.500		26.70	56.75	115.64	24.59	2.95	74.29	
		–	–	5S	.156	12.438		39.08	121.50	20.98	52.65	6.31	19.2	
		–	–	10S	.180	12.390		38.92	120.57	24.17	52.25	6.26	22.0	
		–	20	–	.250	12.250		38.48	117.86	33.38	51.07	6.12	30.2	
		–	30	–	.330	12.090		37.98	114.80	43.77	49.74	5.96	39.0	
		STD	–	40S	.375	12.000		37.70	113.10	49.56	49.00	5.88	43.8	
		–	40	–	.406	11.938		37.50	111.93	53.52	48.50	5.81	47.1	
12	12.750	XS	–	80S	.500	11.750	40.06	36.91	108.43	65.42	46.92	5.63	56.7	12
		–	60	–	.562	11.626		36.52	106.16	73.15	46.00	5.51	62.8	
		–	80	–	.688	11.374		35.73	101.64	88.63	44.04	5.28	74.6	
		–	100	–	.844	11.062		34.75	96.14	107.32	41.66	4.99	88.1	
		XXS	120	–	1.000	10.750		33.77	90.76	125.49	39.33	4.71	100.7	
		–	140	–	1.125	10.500		32.99	86.59	139.67	37.52	4.50	109.9	
		–	160	–	1.312	10.126		31.81	80.53	160.27	34.89	4.18	122.6	
		–	–	5S	.156	13.688		43.00	147.15	23.07	63.77	7.64	23.2	
		–	–	10S	.188	13.624		42.80	145.78	27.73	63.17	7.57	27.8	
		–	10	–	.250	13.500		42.41	143.14	36.71	62.03	7.44	36.6	
		–	20	–	.312	13.376		42.02	140.52	45.61	60.89	7.30	45.0	
		STD	30	–	.375	13.250		41.63	137.88	54.57	59.75	7.16	53.2	
		–	40	–	.438	13.124		41.23	135.28	63.44	58.64	7.03	61.3	
14	14.000	XS	–	–	.500	13.000	43.98	40.84	132.73	72.09	57.46	6.90	69.1	14
		–	60	–	.594	12.812		40.25	128.96	85.05	55.86	6.70	80.3	
		–	80	–	.750	12.500		39.27	122.72	106.13	53.18	6.37	98.2	
		–	100	–	.938	12.124		38.09	115.49	130.85	50.04	6.00	117.8	
		–	120	–	1.094	11.812		37.11	109.62	150.79	47.45	5.69	132.8	
		–	140	–	1.250	11.500		36.13	103.87	170.28	45.01	5.40	146.8	
		–	160	–	1.406	11.188		35.15	98.31	189.11	42.60	5.11	159.6	
		–	–	5S	.165	15.670		49.23	192.85	27.90	83.57	10.02	32.2	
		–	–	10S	.188	15.624		49.08	191.72	31.75	83.08	9.96	36.5	
		–	10	–	.250	15.500		48.69	188.69	42.05	81.74	9.80	48.0	
		–	20	–	.312	15.376		48.31	185.69	52.27	80.50	9.65	59.2	
		STD	30	–	.375	15.250		47.91	182.65	82.58	79.12	9.49	70.3	
16	16.00	XS	40	–	.500	15.000	50.27	47.12	176.72	82.77	76.58	9.18	91.5	16
		–	60	–	.656	14.688		46.14	169.44	107.50	73.42	8.80	116.6	
		–	80	–	.844	14.312		44.96	160.92	136.61	69.73	8.36	144.5	
		–	100	–	1.031	13.938		43.79	152.58	164.82	66.12	7.93	170.5	
		–	120	–	1.219	13.562		42.61	144.50	192.43	62.62	7.50	194.5	
		–	140	–	1.438	13.124		41.23	135.28	233.64	58.64	7.03	220.0	
		–	160	–	1.594	12.812		40.26	128.96	245.25	55.83	6.70	236.7	

PIPE DATA TABLE *(continued)*

Pipe Size (in.)	Outside Diameter (in.)	Weight Class	Carbon Steel Schedule	Stainless Steel Schedule	Wall Thickness (in.)	Inside Diameter (in.)	Circum. (Ext.) (in.)	Circum. (Ext.) (in.)	Flow Area (sq. in.)	Weight of Pipe (lbs/Ft.)	Weight of Water (lbs/Ft.)	Gallons of Water per Ft.	Section Modulus	Pipe Size (in.)
		–	–	5S	.165	17.67		55.51	245.22	31.43	106.26	12.74	40.8	
		–	–	10S	.188	17.62		55.37	243.95	35.76	105.71	12.67	46.4	
		–	10	–	.250	17.50		54.98	240.53	47.39	104.21	12.49	61.1	
		–	20	–	.312	17.38		54.59	237.13	58.94	102.77	12.32	75.5	
		STD	–	–	.375	17.25		54.19	233.71	70.59	101.18	12.14	89.6	
		–	30	–	.438	17.12		53.80	230.30	82.15	99.84	11.96	103.4	
18	18.00	XS	–	–	.500	17.00	56.55	53.41	226.98	93.45	98.27	11.79	117.0	18
		–	40	–	.562	16.88		53.02	223.68	104.87	96.93	11.62	130.1	
		–	60	–	.750	16.50		51.84	213.83	138.17	92.57	11.11	168.3	
		–	80	–	.938	16.12		50.66	204.24	170.92	88.50	10.61	203.8	
		–	100	–	1.156	15.69		49.29	193.30	207.96	83.76	10.04	242.3	
		–	120	–	1.375	15.25		47.91	182.66	244.14	79.07	9.49	277.6	
		–	140	–	1.562	14.88		46.73	173.80	274.22	75.32	9.03	305.5	
		–	160	–	1.781	14.44		45.36	163.72	308.50	70.88	8.50	335.6	
		–	–	5S	.188	19.62		61.65	302.46	39.78	131.06	15.71	57.4	
		–	–	10S	.218	19.56		61.46	300.61	46.06	130.27	15.62	66.3	
		–	10	–	.250	19.50		61.26	298.65	52.73	129.42	15.51	75.6	
		–	20	–	.375	19.25		60.48	290.04	78.60	125.67	15.12	111.3	
		STD	30	–	.500	19.00		59.69	283.53	104.13	122.87	14.73	145.7	
20	20.00	XS	40	–	.594	18.81	62.83	59.10	278.00	123.11	120.46	14.44	170.4	20
		–	60	–	.812	18.38		57.73	265.21	166.40	114.92	13.78	225.7	
		–	80	–	1.031	17.94		56.35	252.72	208.87	109.51	13.13	277.1	
		–	100	–	1.281	17.44		54.78	238.83	256.10	103.39	12.41	331.5	
		–	120	–	1.500	17.00		53.41	226.98	296.37	98.35	11.79	375.5	
		–	140	–	1.750	16.50		51.84	213.82	341.09	92.66	11.11	421.7	
		–	160	–	1.969	16.06		50.46	202.67	379.17	87.74	10.53	458.5	
		–	–	5S	.188	21.62		67.93	367.25	43.80	159.14	19.08	69.7	
		–	–	10S	.218	21.56		67.75	365.21	50.71	158.26	18.97	80.4	
		–	10	–	.250	21.50		67.54	363.05	58.07	157.32	18.86	91.8	
		STD	20	–	.375	21.25		66.76	354.66	86.61	153.68	18.42	135.4	
		XS	30	–	.500	21.00		65.97	346.36	114.81	150.09	17.99	177.5	
22	22.00	–	60	–	.875	20.25	69.12	63.62	322.06	197.41	139.56	16.73	295.0	22
		–	80	–	1.125	19.75		62.05	306.35	250.81	132.76	15.91	366.4	
		–	100	–	1.375	19.25		60.48	291.04	302.88	126.12	15.12	432.6	
		–	120	–	1.625	18.75		58.90	276.12	353.61	119.65	14.34	493.8	
		–	140	–	1.875	18.25		57.33	261.59	403.00	113.36	13.59	550.3	
		–	160	–	2.125	17.75		55.76	247.45	451.06	107.23	12.85	602.4	
		–	–	5S	.218	23.56		74.03	436.10	55	188.98	22.65	96.0	
		–	10	10S	.250	23.50		73.83	433.74	63	187.95	22.53	109.6	
		STD	20	–	.375	23.25		73.04	424.56	95	183.95	22.05	161.9	
		XS	–	–	.500	23.00		72.26	415.48	125	179.87	21.58	212.5	
		–	30	–	.562	22.88		71.86	411.00	141	178.09	21.35	237.0	
24	24.00	–	40	–	.688	22.62	75.40	71.08	402.07	171	174.23	20.88	285.1	24
		–	60	–	.969	22.06		69.31	382.35	238	165.52	19.86	387.7	
		–	80	–	1.219	21.56		67.74	365.22	297	158.26	18.97	472.8	
		–	100	–	1.531	20.94		65.78	344.32	367	149.06	17.89	570.8	
		–	120	–	1.812	20.38		64.01	326.08	430	141.17	16.94	652.1	
		–	140	–	2.062	19.88		62.44	310.28	483	134.45	16.12	718.9	
		–	160	–	2.344	19.31		60.67	292.98	542	126.84	15.22	787.9	
		–	–	5S	.250	29.50		92.68	683.49	79	296.18	35.51	172.3	
		–	10	10S	.312	29.38		92.29	677.71	99	293.70	35.21	213.8	
30	30.00	STD	–	–	.375	29.25	94.25	91.89	671.96	119	291.18	34.91	255.3	30
		XS	20	–	.500	29.00		91.11	660.52	158	286.22	34.31	336.1	
		–	30	–	.625	28.75		90.32	649.18	196	281.31	33.72	414.9	

ENGINEERING

MAXIMUM ALLOWABLE WORKING PRESSURES (PSIG) FOR SEAMLESS CARBON STEEL PIPE

Nominal Pipe Size (in.)	Maximum allowable working pressure at -20 to 650 °F ▲												
	SCH 10	SCH 20	SCH 30	STD WALL	SCH 40	SCH 60	XH	SCH 80	SCH 100	SCH 120	SCH 140	SCH 160	XXH
1/2	-	-	-	1694	1694	-	3036	3036	-	-	-	4551	9223
3/4	659	-	-	1450	1450	-	2589	2589	-	-	-	4505	7531
1	1065	-	-	1578	1578	-	2601	2601	-	-	-	4290	7150
1 1/4	556	-	-	1069	1069	-	1941	1941	-	-	-	3001	5593
1 1/2	486	-	-	1004	1004	-	1821	1821	-	-	-	3091	5114
2	388	-	-	903	903	-	1659	1659	-	-	-	3225	4475
2 1/2	431	-	-	1214	1214	-	1936	1936	-	-	-	2963	4936
3	346	-	-	1094	1094	-	1773	1773	-	-	-	2933	4405
3 1/2	303	-	-	1023	1023	-	1671	1671	-	-	-	-	-
4	269	-	-	974	974	-	1598	1598	-	2243	-	2868	3858
5	284	-	-	888	888	-	1475	1475	-	2123	-	2791	3485
6	239	-	-	833	833	-	1473	1473	-	2038	-	2738	3414
8	225	543	628	770	1038	1343	1343	1649	2068	2388	2715	2605	-
10	224	434	578	723	723	1070	1070	1311	1641	1975	2406	2754	-
12	219	366	534	630	696	1033	898	1305	1653	2009	2295	2735	-
14	333	451	573	573	693	999	816	1311	1690	2013	2341	2675	-
16	291	395	500	500	711	980	711	1305	1638	1975	2378	2669	-
18	258	350	538	444	725	1013	631	1303	1648	1998	2303	2665	-
20	233	399	568	399	693	995	568	1299	1653	1970	2338	2663	-
22	211	-	-	363	-	-	515	-	-	-	-	-	-
24	194	331	541	331	683	1004	471	1295	1664	2003	2309	2656	-
26	-	-	-	306	-	-	435	-	-	-	-	-	-
30	209	376	488	265	-	-	376	-	-	-	-	-	-
36	-	-	-	220	-	-	314	-	-	-	-	-	-
42	-	-	-	189	-	-	269	-	-	-	-	-	-

▲ For allowable working pressures at higher temperatures, multiply values listed above by the following factors:

Grade A

Temperature	700 °F	750 °F	800 °F	850 °F	900 °F
Multiply by	0.971	0.892	0.750	0.708	0.417

Grade B

Temperature	700 °F	750 °F	800 °F	850 °F	900 °F
Multiply by	0.956	0.853	0.720	0.620	0.333

ENGINEERING

FLANGE STANDARDS – Dimensional Data in inches

125 lb. CAST IRON — ANSI STANDARD B16.1

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	–	–	4 1/4	4 5/8	5	6	7	7 1/2	8 1/2	9	10	11	13 1/2	16	19
Thickness of Flange (min)[a]	–	–	7/16	1/2	9/16	5/8	11/16	3/4	13/16	15/16	15/16	1	1 1/8	1 3/16	1 1/4
Diameter of Bolt Circle	–	–	3 1/8	3 1/2	3 7/8	4 3/4	5 1/2	6	7	7 1/2	8 1/2	9 1/2	11 3/4	14 1/4	17
Number of Bolts	–	–	4	4	4	4	4	4	8	8	8	8	8	12	12
Diameter of Bolts	–	–	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	7/8	7/8

[a] 125 lb. Cast Iron Flanges have plain faces (i.e. not raised faces).

250 lb. CAST IRON — ANSI STANDARD B16.1

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	–	–	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	17 1/2	20 1/2
Thickness of Flange (min)[b]	–	–	11/16	3/4	13/16	7/8	1	1 1/8	1 3/16	1 1/4	1 3/8	1 7/16	1 5/8	1 7/8	2
Diameter of Raised Face	–	–	2 11/16	3 1/16	3 9/16	4 3/16	4 15/16	5 11/16	6 5/16	6 15/16	8 5/16	9 11/16	11 15/16	14 1/16	16 7/16
Diameter of Bolt Circle	–	–	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	15 1/4	17 3/4
Number of Bolts	–	–	4	4	4	8	8	8	8	8	8	12	12	16	16
Diameter of Bolts	–	–	5/8	5/8	3/4	5/8	3/4	3/4	3/4	3/4	3/4	3/4	7/8	1	1 1/8

[b] 250 lb. Cast Iron Flanges have a 1/16” raised face which is included in the flange thickness dimensions.

150 lb. BRONZE — ANSI STANDARD B16.24

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	3 1/2	3 7/8	4 1/4	4 5/8	5	6	7	7 1/2	8 1/2	9	10	11	13 1/2	16	19
Thickness of Flange (min)[c]	5/16	11/32	3/8	13/32	7/16	1/2	9/16	5/8	11/16	11/16	3/4	13/16	15/16	1	1 1/16
Diameter of Bolt Circle	2 3/8	2 3/4	3 1/8	3 1/2	3 7/8	4 3/4	5 1/2	6	7	7 1/2	8 1/2	9 1/2	11 3/4	14 1/4	17
Number of Bolts	4	4	4	4	4	4	4	4	8	8	8	8	8	12	12
Diameter of Bolts	1/2	1/2	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	7/8	7/8

[c] 150 lb. Bronze Flanges have plain faces (i.e. not raised faces) with two concentric gasket-retaining grooves between the port and the bolt holes.

300 lb. BRONZE — ANSI STANDARD B16.24

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	3 3/4	4 5/8	4 7/8	5 1/4	6 1/2	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	–	–
Thickness of Flange (min)[d]	1/2	17/32	19/32	5/8	11/16	3/4	13/16	29/32	31/32	1 1/16	1 1/8	1 3/16	1 3/8	–	–
Diameter of Bolt Circle	2 5/8	3 1/4	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	–	–
Number of Bolts	4	4	4	4	4	8	8	8	8	8	8	12	12	–	–
Diameter of Bolts	1/2	5/8	5/8	5/8	3/4	5/8	3/4	3/4	3/4	3/4	3/4	3/4	7/8	–	–

[d] 300 lb. Bronze Flanges have plain faces (i.e. not raised faces) with two concentric gasket-retaining grooves between the port and the bolt holes.

FLANGE STANDARDS – Dimensional Data in inches (continued)

150 lb. STEEL — ANSI STANDARD B16.5

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	–	–	4	4 5/8	5	6	7	7 1/2	8 1/2	9	10	11	13 1/2	16	19
Thickness of Flange (min)[e]	–	–	7/16	1/2	9/16	5/8	11/16	3/4	13/16	15/16	15/16	1	1 1/8	1 3/16	1 1/4
Diameter of Raised Face	–	–	2	2 1/2	2 7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter of Bolt Circle	–	–	3 1/8	3 1/2	3 7/8	4 3/4	5 1/2	6	7	7 1/2	8 1/2	9 1/2	11 3/4	14 1/4	17
Number of Bolts	–	–	4	4	4	4	4	4	8	8	8	8	8	12	12
Diameter of Bolts	–	–	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	7/8	7/8

[e] 150 lb. Steel Flanges have a 1/16" raised face which is included in the flange thickness dimensions.

300 lb. STEEL — ANSI STANDARD B16.5

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	–	–	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	17 1/2	20 1/2
Thickness of Flange (min)[f]	–	–	11/16	3/4	13/16	7/8	1	1 1/8	1 3/16	1 1/4	1 3/8	1 7/16	1 5/8	1 7/8	2
Diameter of Raised Face	–	–	2	2 1/2	2 7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter of Bolt Circle	–	–	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	15 1/4	17 3/4
Number of Bolts	–	–	4	4	4	8	8	8	8	8	8	12	12	16	16
Diameter of Bolts	–	–	5/8	3/4	5/8	3/4	3/4	3/4	3/4	3/4	3/4	3/4	7/8	1	1 1/8

[f] 300 lb. Steel Flanges have a 1/16" raised face which is included in the flange thickness dimensions.

400 lb. STEEL — ANSI STANDARD B16.5

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	3 3/4	4 5/8	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10	11	12 1/2	15	17 1/2	20 1/2
Thickness of Flange (min)[g]	9/16	5/8	11/16	13/16	7/8	1	1 1/8	1 1/4	1 3/8	1 3/8	1 1/2	1 5/8	1 7/8	2 1/8	2 1/4
Diameter of Raised Face	1 3/8	1 11/16	2	2 1/2	2 7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter of Bolt Circle	2 5/8	3 1/4	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	7 7/8	9 1/4	10 5/8	13	15 1/4	17 3/4
Number of Bolts	4	4	4	4	4	8	8	8	8	8	8	12	12	16	16
Diameter of Bolts	1/2	5/8	5/8	5/8	3/4	5/8	3/4	3/4	7/8	7/8	7/8	7/8	1	1 1/8	1 1/4

[g] 400 lb. Steel Flanges have a 1/4" raised face which is included in the flange thickness dimensions.

600 lb. STEEL — ANSI STANDARD B16.5

PIPE SIZE	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
Diameter of Flange	3 3/4	4 5/8	4 7/8	5 1/4	6 1/8	6 1/2	7 1/2	8 1/4	9	10 3/4	13	14	16 1/2	20	22
Thickness of Flange (min)[h]	9/16	5/8	11/16	13/16	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 3/4	1 7/8	2 3/16	2 1/2	2 5/8
Diameter of Raised Face	1 3/8	1 11/16	2	2 1/2	2 7/8	3 5/8	4 1/8	5	5 1/2	6 3/16	7 5/16	8 1/2	10 5/8	12 3/4	15
Diameter of Bolt Circle	2 5/8	3 1/4	3 1/2	3 7/8	4 1/2	5	5 7/8	6 5/8	7 1/4	8 1/2	10 1/2	11 1/2	13 3/4	17	19 1/4
Number of Bolts	4	4	4	4	4	8	8	8	8	8	8	12	12	16	20
Diameter of Bolts	1/2	5/8	5/8	5/8	3/4	5/8	3/4	3/4	7/8	7/8	1	1	1 1/8	1 1/4	1 1/4

[h] 600 lb. Steel Flanges have a 1/4" raised face which is included in the flange thickness dimensions.

ENGINEERING

PIPE FITTING & FLANGE SPECIFICATIONS

FITTING STANDARDS & SPECIFICATIONS

Class or Material	Dimensions	Material Spec.	Galvanizing	Thread	Pressure Rating	Federal/Other
Malleable Iron Fittings						
Class 150/PN 20	ASME B16.3•	ASTM A-197	ASTM A-153	ASME B120.1+	ASME B16.3•	ASME B16.3**
Class 300/PN 50	ASME B16.3•	ASTM A-197	ASTM A-153	ASME B120.1+	ASME B16.3•	
Malleable Iron Unions						
Class 150/PN 20	ASME B16.39•	ASTM A-197	ASTM A-153	ASME B120.1+	ASME B16.39•	ASME B16.39***
Class 250	ASME B16.39•	ASTM A-197	ASTM A-153	ASME B120.1+	ASME B16.39•	
Class 300/PN 50	ASME B16.39•	ASTM A-197	ASTM A-153	ASME B120.1+	ASME B16.39•	
Cast Iron Threaded Fittings						
Class 125	ASME B16.4•	ASTM A-126 (A)	ASTM A-153	ASME B120.1+	ASME B16.4•	ASME B16.4u
Class 250	ASME B16.4•	ASTM A-126 (A)	ASTM A-153	ASME B120.1+	ASME B16.4•	ASME B16.4u
Cast Iron Plugs & Bushings						
	ASME B16.14•	ASTM A-126 (A)	ASTM A-153	ASME B120.1+	ASME B16.14•	WW-P-471
Cast Iron Drainage Threaded Fittings						
	ASME B16.12•	ASTM A-126 (A)	ASTM A-153	ASME B120.1+	ASME B16.12•	
Cast Iron Flanges & Flanged Fittings						
Class 125 (1"-12")	ASME B16.1•	ASTM A-126 (A) or (B)	ASTM A-153	ASME B120.1+	ASME B16.1•	ASME B16.1•
Class 125 (14" & up)	ASME B16.1•	ASTM A-126 (B)	ASTM A-153	ASME B120.1+	ASME B16.1•	ASME B16.1•
Class 250 (1"-12")	ASME B16.1•	ASTM A-126 (A) or (B)	ASTM A-153	ASME B120.1+	ASME B16.1•	ASME B16.1•
Class 250 (14" & up)	ASME B16.1•	ASTM A-126 (B)	ASTM A-153	ASME B120.1+	ASME B16.1•	ASME B16.1•
Forged Steel Threaded Fittings						
Class 2000, 3000, 6000	ASME B16.11•	ASTM A105, ASTM A182, ASTM A350		ASME B120.1+	ASME B16.11•	
Pipe Nipples						
Steel Pipe - welded	ASTM A733	ASTM A53 Type F or Type E		ASME B120.1+		WWN 351
Steel Pipe - seamless (High Temperature)	ASTM A733	ASTM A106 Gr.B		ASME B120.1+		WWN 351
Brass		ASTM B43		ASME B120.1+		WWN 351

• an American National standard (ANSI)
+ ASME B120.1 was ANSI B2.1
u Formerly WW-P-501
** Formerly WW-P-521
*** Formerly WW-U-531

ENGINEERING

STANDARD CLASS PRESSURE-TEMPERATURE RATINGS ANSI/ASME B16.34

Working Pressure by Classes	Temperature (°F)	A 216 WCB (a)	A 352 LCB (d)	A 216 WCC (a) A 352 LC2 (d) A 352 LC3 (d) A 352 LCC (e)	A 217 WC1 (b) A 352 LC1 (d)	A 217 WC4 (h) A 217 WC5 (i)	A 217 WC6 (j)	A 217 WC9 (j)	A 217 C5	A 217 C12	A 351 CF3 (f) A 351 CF8	A 351 CF3M (g) A 351 CF8M	A 351 CF8C	A 351 CN7M (l)
		Working Pressure in PSI												
150 LB.	-20 to 100	285	265	290	265	290	290	290	290	290	275	275	275	230
	200	260	250	260	260	260	260	260	260	260	235	240	245	215
	300	230	230	230	230	230	230	230	230	230	205	215	225	200
	400	200	200	200	200	200	200	200	200	200	180	195	200	185
	500	170	170	170	170	170	170	170	170	170	170	170	170	170
	600	140	140	140	140	140	140	140	140	140	140	140	140	140
	650	125	125	125	125	125	125	125	125	125	125	125	125	125
	700	110	110	110	110	110	110	110	110	110	110	110	110	110
	750	95	95	95	95	95	95	95	95	95	95	95	95	95
	800	80	80	80	80	80	80	80	80	80	80	80	80	80
	850	65	65	65	65	65	65	65	65	65	65	65	65	–
	900	50	50	50	50	50	50	50	50	50	50	50	50	–
	950	35	35	35	35	35	35	35	35	35	35	35	35	–
	1000	20	20	20	20	20	20	20	20	20	20	20	20	–
	1050	–	–	–	–	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	–
	1100	–	–	–	–	–	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	–
	1150	–	–	–	–	–	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	–
	1200	–	–	–	–	–	15(1)	20(1)	20(1)	20(1)	20(1)	20(1)	20(1)	–
	1250	–	–	–	–	–	–	–	–	–	20(1)	20(1)	20(1)	–
	1300	–	–	–	–	–	–	–	–	–	20(1)	20(1)	20(1)	–
	1350	–	–	–	–	–	–	–	–	–	20(1)	20(1)	20(1)	–
	1400	–	–	–	–	–	–	–	–	–	20(1)	20(1)	20(1)	–
	1450	–	–	–	–	–	–	–	–	–	15(1)	20(1)	20(1)	–
	1500	–	–	–	–	–	–	–	–	–	10(1)	15(1)	15(1)	–
300 LB.	-20 to 100	740	695	750	695	750	750	750	750	750	720	720	720	600
	200	675	655	750	680	750	710	715	750	750	600	620	635	555
	300	655	640	730	655	730	675	675	730	730	530	560	590	525
	400	635	620	705	640	705	660	650	705	705	470	515	555	480
	500	600	585	665	620	665	640	640	665	665	435	480	520	470
	600	550	535	605	605	605	605	605	605	605	415	450	490	455
	650	535	525	590	590	590	590	590	590	590	410	445	480	450
	700	535	520	570	570	570	570	570	570	570	405	430	470	445
	750	505	475	505	530	530	530	530	530	530	400	425	460	440
	800	410	390	410	510	510	510	510	510	510	395	415	455	430
	850	270	270	270	485	485	485	485	440	485	390	405	445	–
	900	170	170	170	450	450	450	450	355	450	385	395	430	–
	950	105	105	105	280	345	380	380	260	370	375	385	385	–
	1000	50	50	50	165	215	225	270	190	290	325	365	365	–
	1050	–	–	–	–	190	140	200	140	190	310	360	360	–
	1100	–	–	–	–	–	95	115	105	115	260	325	325	–
	1150	–	–	–	–	–	50	105	70	75	195	275	275	–
	1200	–	–	–	–	–	35	55	45	50	155	205	170	–
	1250	–	–	–	–	–	–	–	–	–	110	180	125	–
	1300	–	–	–	–	–	–	–	–	–	85	140	95	–
	1350	–	–	–	–	–	–	–	–	–	60	105	70	–
	1400	–	–	–	–	–	–	–	–	–	50	75	50	–
	1450	–	–	–	–	–	–	–	–	–	35	60	40	–
	1500	–	–	–	–	–	–	–	–	–	25	40	35	–

STANDARD CLASS PRESSURE-TEMPERATURE RATINGS ANSI/ASME B16.34 (continued)

Working Pressure by Classes	Temperature (˚F)	A 216 WCB (a)	A 352 LCB (d)	A 216 WCC (a) A 352 LC2 (d) A 352 LC3 (d) A 352 LCC (e)	A 217 WC1 (b) A 352 LC1 (d)	A 217 WC4 (h) A 217 WC5 (i)	A 217 WC6 (j)	A 217 WC9 (j)	A 217 C5	A 217 C12	A 351 CF3 (f) A 351 CF8	A 351 CF3M (g) A 351 CF8M	A 351 CF8C	A 351 CN7M (l)
		Working Pressure in PSI												
400 LB.	-20 to 100	990	925	1000	925	1000	1000	1000	1000	1000	960	960	960	800
	200	900	875	1000	905	1000	950	955	1000	1000	800	825	850	740
	300	875	850	970	870	970	895	905	970	970	705	745	785	700
	400	845	825	940	855	940	880	865	940	940	630	685	740	640
	500	800	775	885	830	885	855	855	885	885	585	635	690	625
	600	730	710	805	805	805	805	805	805	805	555	600	655	605
	650	715	695	785	785	785	785	785	785	785	545	590	640	600
	700	710	690	755	755	755	755	755	755	755	540	575	625	595
	750	670	630	670	710	710	710	710	710	710	530	565	615	585
	800	550	520	550	675	675	675	675	675	675	525	555	610	575
	850	355	355	355	650	650	650	650	585	650	520	540	590	–
	900	230	230	230	600	600	600	600	470	600	510	525	575	–
	950	140	140	140	375	460	505	505	350	495	500	515	515	–
	1000	70	70	70	220	285	300	355	255	390	430	485	485	–
	1050	–	–	–	–	250	185	265	190	250	410	480	480	–
	1100	–	–	–	–	–	130	150	140	150	345	430	430	–
	1150	–	–	–	–	–	70	140	90	100	260	365	365	–
	1200	–	–	–	–	–	45	75	60	70	205	275	230	–
	1250	–	–	–	–	–	–	–	–	–	145	245	165	–
	1300	–	–	–	–	–	–	–	–	–	110	185	125	–
	1350	–	–	–	–	–	–	–	–	–	85	140	90	–
	1400	–	–	–	–	–	–	–	–	–	65	100	70	–
	1450	–	–	–	–	–	–	–	–	–	45	80	55	–
	1500	–	–	–	–	–	–	–	–	–	30	55	45	–
600 LB.	-20 to 100	1480	1390	1500	1390	1500	1500	1500	1500	1500	1440	1440	1440	1200
	200	1350	1315	1500	1360	1500	1425	1430	1500	1500	1200	1240	1270	1115
	300	1315	1275	1455	1305	1455	1345	1355	1455	1455	1055	1120	1175	1045
	400	1270	1235	1410	1280	1410	1315	1295	1410	1410	940	1030	1110	960
	500	1200	1165	1330	1245	1330	1285	1280	1330	1330	875	955	1035	935
	600	1095	1065	1210	1210	1210	1210	1210	1210	1210	830	905	985	910
	650	1075	1045	1175	1175	1175	1175	1175	1175	1175	815	890	960	900
	700	1065	1035	1135	1135	1135	1135	1135	1135	1135	805	865	935	890
	750	1010	945	1010	1065	1065	1065	1065	1065	1065	795	845	920	880
	800	825	780	825	1015	1015	1015	1015	1015	1015	790	830	910	865
	850	535	535	535	975	975	975	975	880	975	780	810	890	–
	900	345	345	345	900	900	900	900	705	900	770	790	865	–
	950	205	205	205	560	685	755	755	520	740	750	775	775	–
	1000	105	105	105	330	425	445	535	385	585	645	725	725	–
	1050	–	–	–	–	380	275	400	280	380	620	720	720	–
	1100	–	–	–	–	–	190	225	205	225	515	645	645	–
	1150	–	–	–	–	–	105	205	140	150	390	550	550	–
	1200	–	–	–	–	–	70	110	90	105	310	410	345	–
	1250	–	–	–	–	–	–	–	–	–	220	365	245	–
	1300	–	–	–	–	–	–	–	–	–	165	275	185	–
	1350	–	–	–	–	–	–	–	–	–	125	205	135	–
	1400	–	–	–	–	–	–	–	–	–	95	150	105	–
	1450	–	–	–	–	–	–	–	–	–	70	115	80	–
	1500	–	–	–	–	–	–	–	–	–	50	85	70	–

STANDARD CLASS PRESSURE-TEMPERATURE RATINGS ANSI/ASME B16.34 (continued)

Working Pressure by Classes	Temperature (°F)	A 216 WCB (a)	A 352 LCB (d)	A 216 WCC (a) A 352 LC2 (d) A 352 LC3 (d) A 352 LCC (e)	A 217 WC1 (b) A 352 LC1 (d)	A 217 WC4 (h) A 217 WC5 (i)	A 217 WC6 (j)	A 217 WC9 (j)	A 217 C5	A 217 C12	A 351 CF3 (f) A 351 CF8	A 351 CF3M (g) A 351 CF8M	A 351 CF8C	A 351 CN7M (l)
		Working Pressure in PSI												
900 LB.	-20 to 100	2220	2085	2250	2085	2250	2250	2250	2250	2250	2160	2160	2160	1800
	200	2025	1970	2250	2035	2250	2135	2150	2250	2250	1800	1860	1910	1670
	300	1970	1915	2185	1955	2185	2020	2030	2185	2185	1585	1680	1765	1570
	400	1900	1850	2115	1920	2115	1975	1945	2115	2115	1410	1540	1665	1445
	500	1795	1745	1995	1865	1995	1925	1920	1995	1995	1310	1435	1555	1405
	600	1640	1600	1815	1815	1815	1815	1815	1815	1815	1245	1355	1475	1365
	650	1610	1570	1765	1765	1765	1765	1765	1765	1765	1225	1330	1440	1350
	700	1600	1555	1705	1705	1705	1705	1705	1705	1705	1210	1295	1405	1335
	750	1510	1420	1510	1595	1595	1595	1595	1595	1595	1195	1270	1385	1320
	800	1235	1175	1235	1525	1525	1525	1525	1490	1525	1180	1245	1370	1295
	850	805	805	805	1460	1460	1460	1460	1315	1460	1165	1215	1330	–
	900	515	515	515	1350	1350	1350	1350	1060	1350	1150	1180	1295	–
	950	310	310	310	845	1030	1130	1130	780	1110	1125	1160	1160	–
	1000	155	155	155	495	640	670	805	575	875	965	1090	1090	–
	1050	–	–	–	–	565	410	595	420	565	925	1080	1080	–
	1100	–	–	–	–	–	290	340	310	340	770	965	965	–
	1150	–	–	–	–	–	155	310	205	225	585	825	825	–
	1200	–	–	–	–	–	105	165	135	155	465	620	515	–
	1250	–	–	–	–	–	–	–	–	–	330	545	370	–
	1300	–	–	–	–	–	–	–	–	–	245	410	280	–
	1350	–	–	–	–	–	–	–	–	–	185	310	205	–
	1400	–	–	–	–	–	–	–	–	–	145	225	155	–
	1450	–	–	–	–	–	–	–	–	–	105	175	125	–
	1500	–	–	–	–	–	–	–	–	–	70	125	105	–
1500 LB.	-20 to 100	3705	3470	3750	3470	3750	3750	3750	3750	3750	3600	3600	3600	3000
	200	3375	3280	3750	3395	3750	3560	3580	3750	3750	3000	3095	3180	2785
	300	3280	3190	3640	3260	3640	3365	3385	3640	3640	2640	2795	2940	2615
	400	3170	3085	3530	3200	3530	3290	3240	3530	3530	2350	2570	2770	2405
	500	2995	2910	3325	3105	3325	3210	3200	3325	3325	2185	2390	2590	2340
	600	2735	2665	3025	3025	3025	3025	3025	3025	3025	2075	2255	2460	2275
	650	2685	2615	2940	2940	2940	2940	2940	2940	2940	2040	2220	2400	2250
	700	2665	2590	2840	2840	2840	2840	2840	2840	2840	2015	2160	2340	2225
	750	2520	2365	2520	2660	2660	2660	2660	2660	2660	1990	2110	2305	2200
	800	2060	1955	2060	2540	2540	2540	2540	2485	2540	1970	2075	2280	2160
	850	1340	1340	1340	2435	2435	2435	2435	2195	2435	1945	2030	2220	–
	900	860	860	860	2245	2245	2245	2245	1765	2245	1920	1970	2160	–
	950	515	515	515	1405	1715	1885	1885	1305	1850	1870	1930	1930	–
	1000	260	260	260	825	1065	1115	1340	960	1460	1610	1820	1820	–
	1050	–	–	–	–	945	684	995	705	945	1545	1800	1800	–
	1100	–	–	–	–	–	480	565	515	565	1285	1610	1610	–
	1150	–	–	–	–	–	260	515	345	380	980	1370	1370	–
	1200	–	–	–	–	–	170	275	225	260	770	1030	855	–
	1250	–	–	–	–	–	–	–	–	–	550	910	615	–
	1300	–	–	–	–	–	–	–	–	–	410	685	465	–
	1350	–	–	–	–	–	–	–	–	–	310	515	345	–
	1400	–	–	–	–	–	–	–	–	–	240	380	255	–
	1450	–	–	–	–	–	–	–	–	–	170	290	205	–
	1500	–	–	–	–	–	–	–	–	–	120	205	170	–

Note: For welding end valves only.
(1) Flanged end ratings terminate at 1000˚F.

Footnotes:
a) Permissible, but not recommended for prolonged usage above about 800˚F.
b) Permissible, but not recommended for prolonged usage above about 850˚F.
d) Not to be used over 650˚F.
e) Not to be used over 700˚F.
f) Not to be used over 800˚F.
g) Not to be used over 850˚F.
h) Not to be used over 1000˚F.
i) Not to be used over 1050˚F.
j) Not to be used over 1100˚F.

ENGINEERING

INTRODUCTION TO STEAM TRAPS

WHAT IS A STEAM TRAP AND WHAT DOES IT DO?

A steam trap is an automatic valve that allows condensate, air and other non-condensable gases to be discharged from the steam system while holding or trapping the steam in the system. Several different types of steam trap technologies exist to accomplish this extremely critical and necessary task.

WHY ARE STEAM TRAPS REQUIRED?

For any steam system to operate properly the condensate, air and other non-condensable gases such as carbon dioxide must be removed from the steam system. This is the purpose of the steam trap.

CONDENSATE:

When steam releases its heat energy, the steam reverts back to water. This occurs in a heat exchanger making hot water, in a radiator heating a room, or in a steam pipe transferring steam. This water, technically referred to as *condensate*, must be removed from the system or the system would back up with water. The removal of condensate from the steam system is considered the primary function of the steam trap.

AIR:

Air exists in all steam pipes prior to system start-up when the system is cold. This air must be bled out of the piping system so that the steam can enter and eventually reach the designated process applications. If the air is not removed, the steam will effectively be blocked from entering the steam pipes by the residual air. In addition to blocking the steam, air acts as an insulator to heat transfer. Even after the system is filled with steam, small amounts of air can re-enter the system thru various paths such as boiler water make-up systems, vacuum breakers and air vents.

NON-CONDENSABLE GASES:

Gases other than air such as carbon dioxide exist inside steam systems. These non-condensable gases must also be separated from the steam and removed from the system for all processes to operate properly. In addition to inhibiting steam flow and proper heat transfer, carbon dioxide can be very corrosive to components in the system.

STEAM TRAP GENERAL APPLICATION CATEGORIES:

DRIP APPLICATIONS:

Drip applications are by far the most common application for steam traps. This application refers to removing the condensate that forms in steam lines when steam loses its heat energy due to radiation losses. Traps used in these applications are referred to as *drip traps*. Generally speaking, traps used for these applications require relatively small condensate capacities and don't normally need to discharge large amounts of air. (Air removal is the primary function of air vents and process traps located throughout the system.) The most common trap choices for drip applications are *thermodynamic* for steam pressures over 30 PSIG, and *float & thermostatic* for pressures up to 30 PSIG. Inverted bucket traps are also commonly used for drip trap applications due to their ability to handle large amounts of dirt and scale often found in this type of application.

PROCESS APPLICATIONS:

Process trap applications refer to removing condensate and air directly from a specific heat transfer process such as a heat exchanger that could be making hot water or a radiator heating a room. Traps used in these applications are referred to as *process traps*. Generally speaking, traps used for process applications require larger condensate handling capability and also need to be able to discharge large amounts of air. The most common trap choices for process applications are *float & thermostatic* traps and *thermostatic* traps. Both are known for their excellent condensate and air handling capabilities. In contrast, thermodynamic traps and inverted bucket traps, which have poor air handling ability, would normally make a poor choice for process applications.

TRACING APPLICATIONS:

Steam tracing refers to using steam to indirectly elevate the temperature of a product using jacketed pipes or tubing filled with steam. A typical application would be wrapping a pipeline containing high viscosity oil with tracing tubing. The steam inside the tubing heats the oil to lower its viscosity, allowing it to flow easily thru the pipeline. Similar to any steam applications, a steam trap must be used on the end of the steam tubing to discharge unwanted condensate. Steam traps used in these applications are referred to as *tracing traps*. The most common trap choice for tracing applications is the *thermostatic* type.

The **Thermodynamic Disc Trap** is simple and compact and an excellent choice for a wide variety of drip applications. They excel in drip applications of pressures ranging from 30 psig to high pressure applications exceeding 3,000 psig, including superheated steam. The ½" TD600L is suitable for most drip applications, and offers reduced size discharge orifice holes which are preferable in terms of performance, longevity, and efficiency.

The design of modern **Thermostatic** Bellows Traps allows these traps to be used on a wide variety of applications, from general service drips to small-to-medium process heating applications with relatively constant loads. The welded stainless steel bellows is extremely rugged which prevents failure from waterhammer and corrosion, making these traps suitable for demanding industrial service. Also, because thermostatic traps subcool condensate, the condensate discharged generates less flash steam which may be advantageous in certain installations. For these reasons, Thermostatic Bellows Traps can be considered as a primary selection or as an alternative to other styles.

The **Float & Thermostatic Trap** is the primary choice of steam trap for process applications. They are excellent at discharging air from the system during start-up and offer a wide range of capacities to accommodate the vast majority of process heating applications. Their design allows them to immediately respond to changing condensate loads and pressures, which is a typical requirement of continuous heating process applications where control valves are used to modulate steam flow. Available F&T models range from designs for low pressure heating to high-pressure and capacity industrial applications requiring cast steel or stainless steel.

DRIP LEG DESIGN

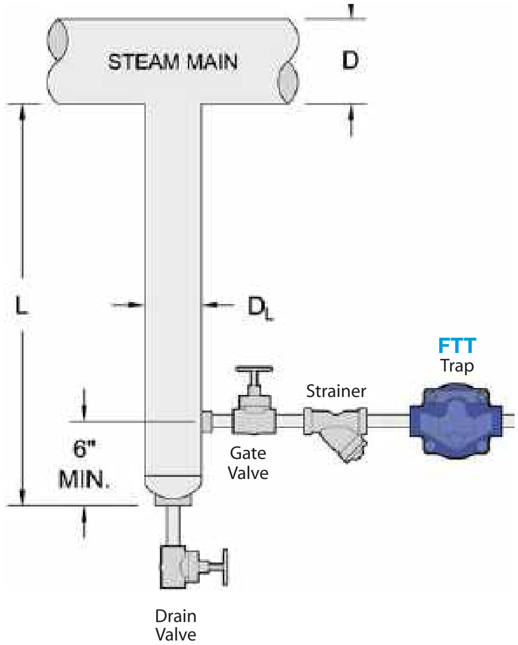
PURPOSE: Drip Legs are used for removing condensate from steam transmission and distribution lines. This helps ensure high quality steam for use in various plant applications and also will prevent damaging and dangerous waterhammer.

OPERATION: As steam travels at high velocity through piping, condensate forms as the result of piping heat losses and/or improper boiler control resulting in condensate carryover. Drip legs are therefore located at points where condensate may accumulate to allow for drainage by gravity down to a steam trap for proper discharge from the system. Since condensate drains by gravity, drip legs must be located on the bottom of piping and designed with diameters large enough to promote collection.

INSTALLATION GUIDELINES: (see Figure 1)

- For drainage of steam transmission and distribution lines, drip legs should be located at bends in piping (direction changes), low points, end of line, and in straight run of piping every 200 feet.
- For protection of equipment such as regulators and control valves, drip legs should be installed directly ahead of the regulating or control valve line.
- Proper steam trap selection for drip applications is dependent upon application requirements, such as pressure, number of and distance between installed steam traps, ambient conditions, start-up requirements, etc. A commonly accepted practice is to use float & thermostatic (F&T) steam traps for low pressure steam systems up to 30 PSIG, and thermodynamic steam traps for steam pressures over 30 PSIG.
- Because condensate drainage from steam systems is dependent upon gravity, drip leg diameter is critical for optimum removal – larger is better. Collection leg diameter (D_L) is recommended to be the same size as the steam main (D), up to 4”. For steam mains above 4”, the collection leg diameter may be half the diameter of the main, but not less than 4”. The length (L) of the drip leg for systems with automatic start-up should be a minimum of 28” to provide approximately 1 PSI head pressure. The length (L) of the drip leg for systems with supervised start-up should be 1.5 x D_L, but not less than 8”.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- A drain valve is included at the bottom of the collection leg for manual discharge of condensate during supervised start-up. The drain valve should be located at least 6” below the steam trap line.
- An isolation valve and strainer should be installed before the steam trap. The isolation valve simplifies maintenance of the trap and the strainer protects the trap from any dirt, debris or scale in the line.

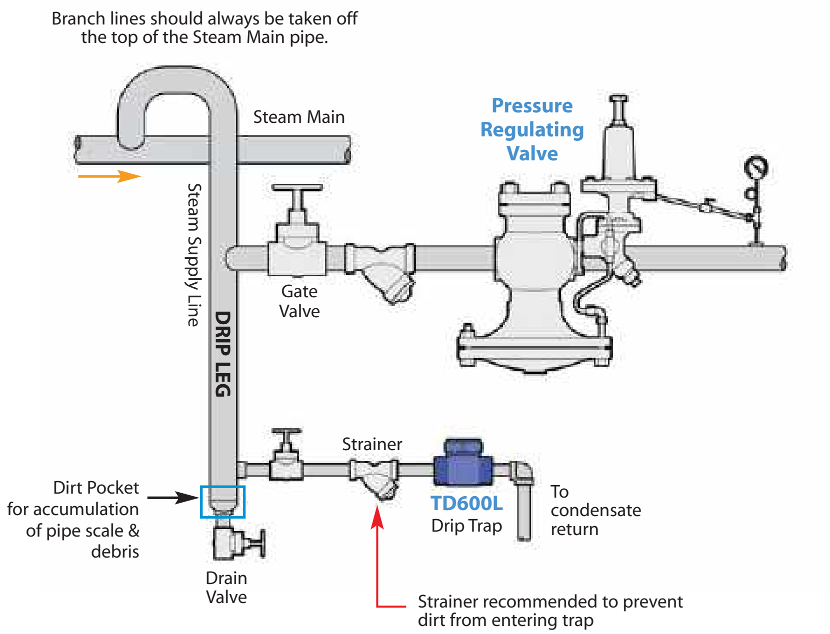
Figure 1: Proper DRIP LEG Designs



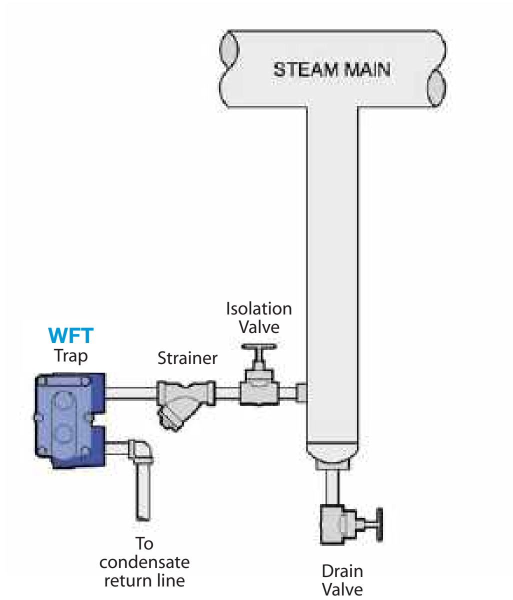
DRIP LEG DESIGN CRITERIA:

1) Locate prior to valves, bends in pipe (direction changes), low points, end of line and straight piping runs (max. 200 ft. apart).
2) Diameter:
   - Drip leg diameter (D_L) to be equal to steam main diameter (D) for steam main sizes up to 4”
   - Drip leg diameter (D_L) may be half the steam main diameter (D) for steam main sizes over 4”, but not less than 4”
3) Length (L):
   - For systems with automatic start-up, L to be 28” minimum (= 1 PSI minimum head pressure)
   - For systems with supervised start-up, L to be 1.5 x D_L, but not less than 8”

DRIP LEG Before Regulator or Control Valve



DRIP LEG Draining Steam Main



ENGINEERING

PROCESS TRAP GUIDELINES – Gravity Drainage

PURPOSE: For removing condensate from below steam heat transfer equipment to ensure optimum heating under various load conditions.

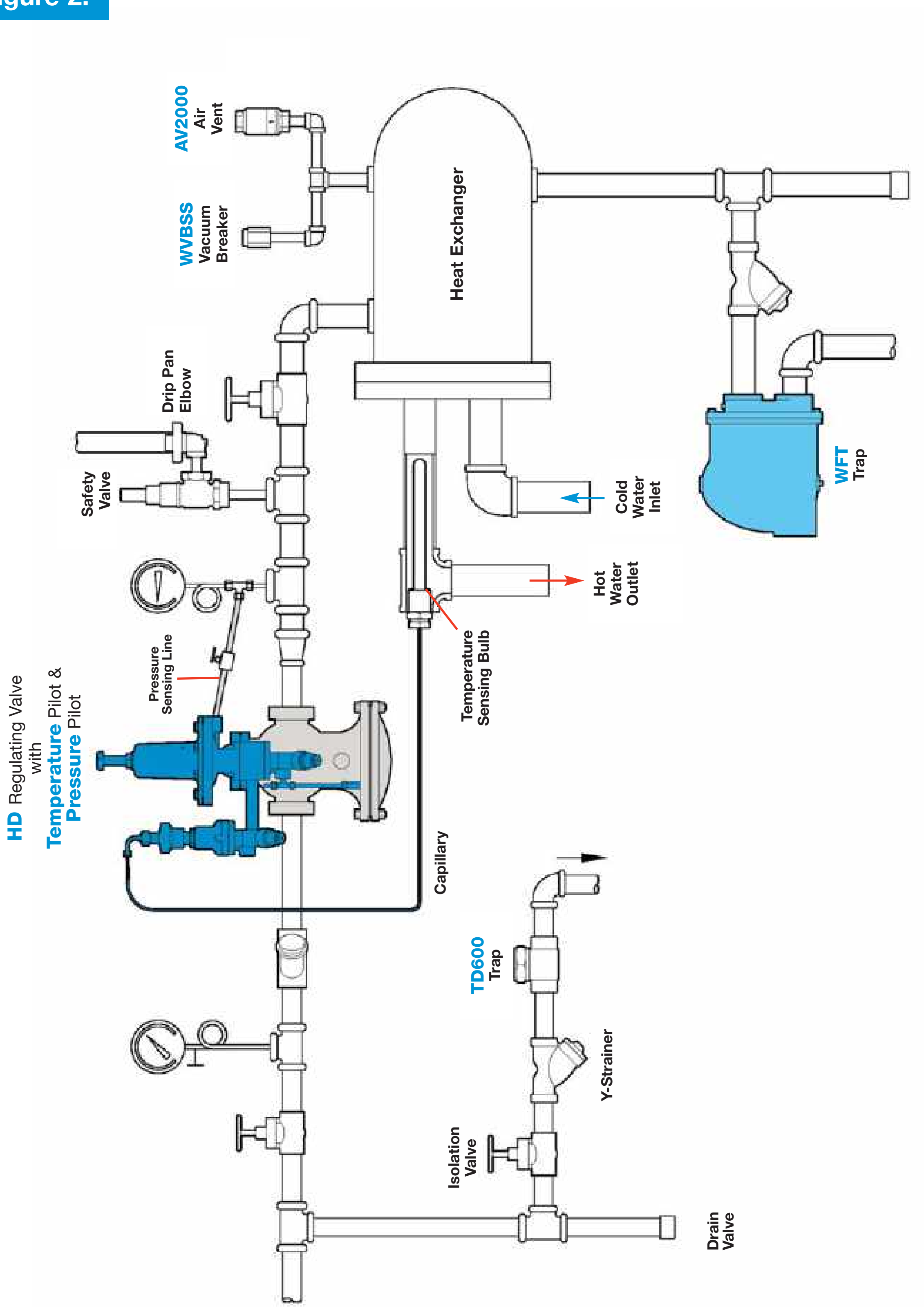
OPERATION: Steam used to heat product such as water in a heat exchanger condenses to liquid after passing though the heat exchanger and releasing its heating energy. To ensure optimum heating, this condensate is removed through an adequately sized drip leg and steam trap properly selected for the application and installed below the equipment. A Float and Thermostatic (F&T) steam trap is often an appropriate choice due to its modulating discharge and air venting capability.

INSTALLATION GUIDELINES: (see Figure 2)

- Selection and sizing of the process steam trap is critical to proper operation. A safety load factor (SLF) is applied to accommodate load variations, as well as high start-up requirements. Consult appropriate sections of this catalog or the factory for guidelines regarding proper process steam trap selection and sizing.
- The collecting leg to the process trap should be no smaller than the designed condensate outlet of the heat transfer equipment. Note that some steam trap technologies such as thermostatic require extended distance between the heat exchanger and steam trap to allow for back-up of subcooled condensate.
- The process trap should be located 2.3 feet (28”) below the condensate outlet of the heat exchanger to provide a minimum of 1 PSI head pressure.
- The drip leg and steam trap prior to the regulating valve protect the valve from condensate, as well as ensure the best quality steam for heat transfer. Note the take-off from the top of the steam main to avoid condensate that would collect on the bottom of the main piping.
- The vacuum breaker and auxiliary air vent located at the top of the heat exchanger vessel promotes proper drainage and optimum heat transfer. The vacuum breaker allows system equalization with atmospheric air to allow gravity condensate drainage when vacuum is formed from condensing steam. The air vent improves heat-up times and overall heat transfer by expelling accumulated air on start-up.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- An isolation valve and strainer should be installed before any steam trap. The isolation valve simplifies maintenance of the trap and the strainer protects the trap from any dirt, debris or scale in the line.

PROCESS TRAP GUIDELINES – Gravity Drainage

Figure 2:



Shell & Tube Heat Exchanger with Gravity Drainage of Condensate

PROCESS TRAP GUIDELINES – Syphon Drainage

PURPOSE: For removing condensate from steam heat transfer equipment when the steam trap is to be installed *above* the point where condensate will collect.

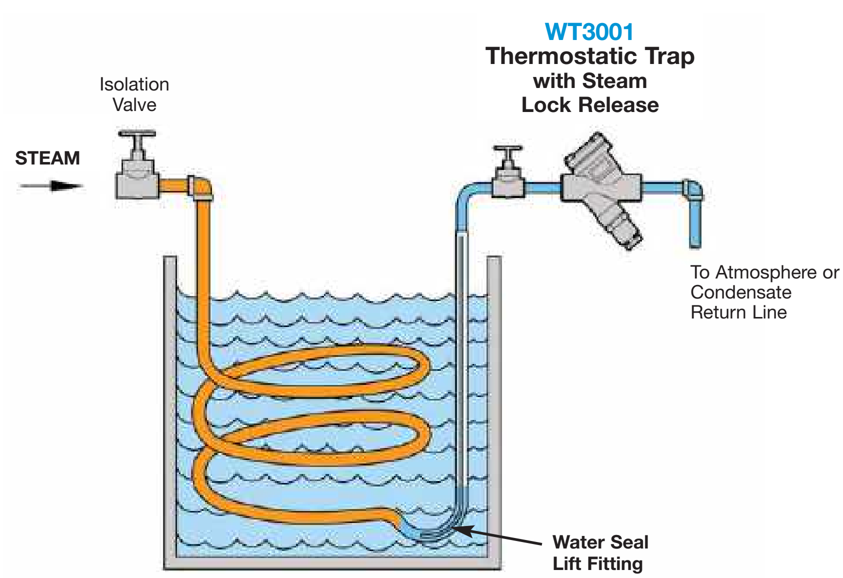
OPERATION: When steam is used to heat liquid in a tank with a submerged coil or a rotary drum dryer, gravity drainage to the steam trap is not possible. For these applications, it is necessary to install the steam trap above the drain point of the equipment by creating a syphon lift to allow for proper condensate drainage.

INSTALLATION GUIDELINES: (see Figure 3)

- There are two critical requirements to ensure proper operation of syphon lift process drainage systems: A water seal lift fitting and a steam trap with a function to prevent steam lock (often referred to as Steam Lock Release or SLR).
- The lift fitting on a submerged coil provides a water seal to stop steam from pushing past the condensate and reaching the steam trap, preventing a vapor-lock condition of the trap.
- Steam Lock Release (SLR) is provided on the steam trap to ensure the syphon lift remains continuous by preventing steam from becoming trapped – or locked – between the cavity of the steam trap and incoming condensate. The SLR function allows any small portion of trapped steam to be automatically removed from the system, allowing continuous drainage.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- An isolation valve and strainer should be installed before any steam trap. The isolation valve simplifies maintenance of the trap and the strainer protects the trap from any dirt, debris or scale in the line.

Figure 3: **SUBMERGED COIL FOR HEATING LIQUID**

Steam Lock Release Mechanism must be used when trap is positioned above condensate level.



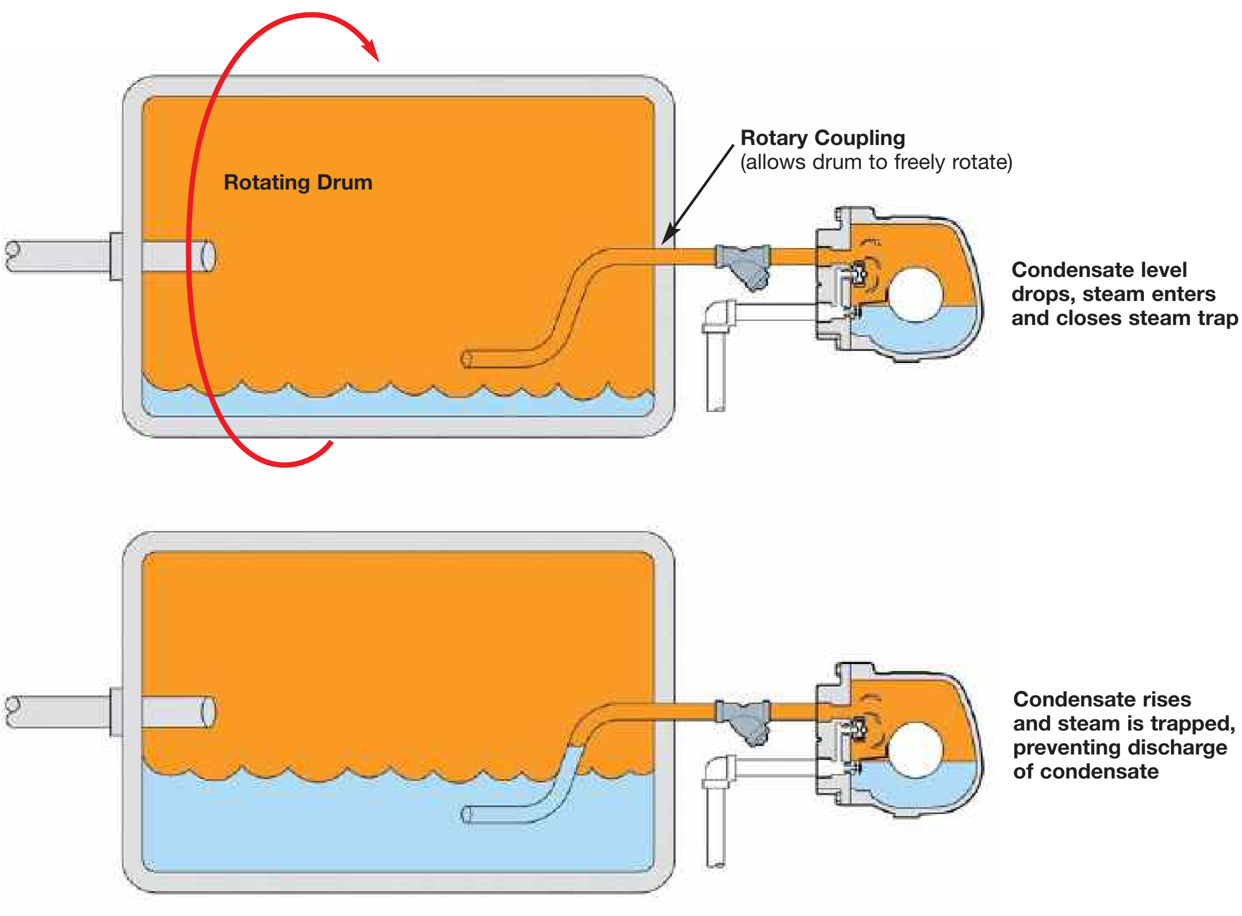
ENGINEERING

ROTATING STEAM DRYER

Commonly found in the Paper Making industry, a rotating piece of equipment offers a unique challenge of removing the condensate. Steam inside a rotating drum cylinder is used to heat product such as sheets of paper over the outside surface of the drum. The steam pressure pushes the condensate up through the pipe to the steam trap. Since the pipe that the condensate is traveling through is surrounded by steam, an issue can develop that will "**Steam Lock**" the trap causing the trap to stay closed, allowing the condensate to build up inside the rotating drum (Figure 4). By placing a **Steam Lock Release** feature on the Steam Trap, a small amount of steam will be constantly discharged through the trap. This allows condensate to reach the steam trap which causes it to open and function properly. This steam lock release feature is available on ALL F&T and Thermostatic traps and should be considered on this type of application.

Figure 4: Rotating Steam Dryer Illustrating **"Steam Lock"**

Steam Lock Release Option
must be used when trap is positioned above condensate level.



ENGINEERING

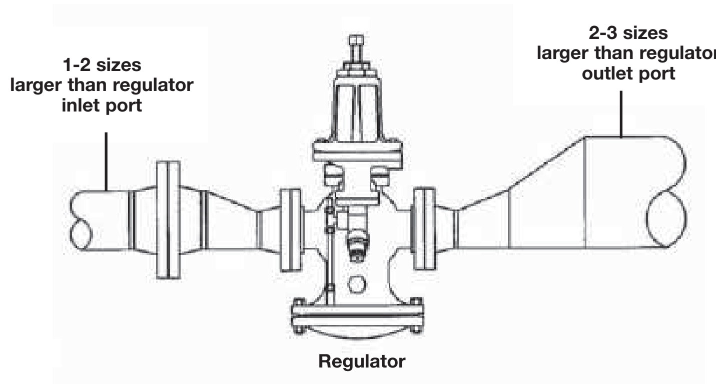
Regulator Application & Installation Notes

The following are considerations for all steam regulator installations, as system operation is dependent upon proper design, installation, start-up and maintenance procedures:

Inlet & Outlet Pipe Sizing

Improperly sized piping can contribute to excessive noise in a steam system. Make certain inlet and outlet piping to the regulator is adequately sized for the flow, velocity and pressure requirements.

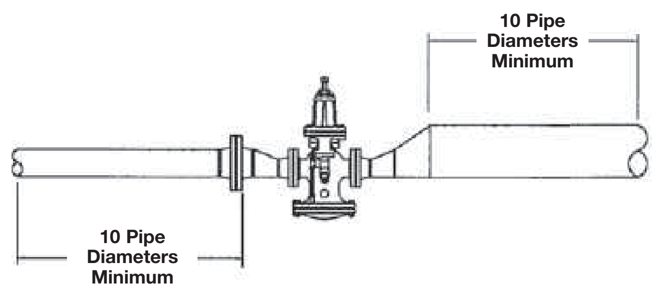
Inlet piping can be 1-2 sizes larger and outlet piping 2-3 sizes larger than the connection ports of a properly sized regulator.



Straight Run of Pipe Before and After the Valve

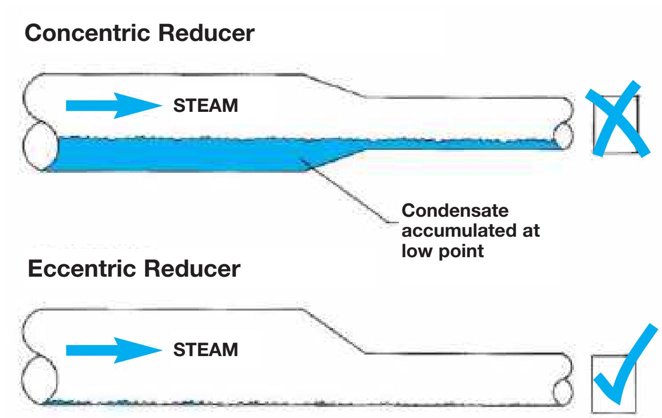
Pipe fittings, bends and other accessories contribute to fluid turbulence in a system which can result in erratic control. To limit this and ensure optimum system operation, follow recommended guidelines for minimum straight run lengths of pipe before and after a regulator.

Note: Any isolation valves or pipeline accessories should be full-ported.



Reducer Selection

Concentric pipe reducers should be avoided on the inlet side of regulators as they can allow entrained condensate to collect, potentially leading to damaging and dangerous waterhammer. Therefore, when reducers are required in the steam piping to accommodate properly sized valves and pipes, <u>use e*ccentric reducers on regulator inlets*</u> *and concentric or eccentric reducers on regulator outlets.*

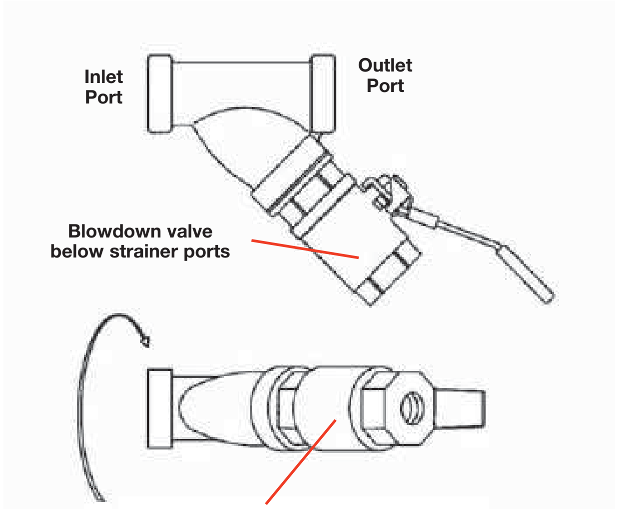


ENGINEERING

Strainers with Blowdown Valves

Regardless of any filters provided on a regulator, a strainer with blowdown valve is recommended before (upstream of) all regulator installations. Pipeline debris and scale can damage internal valve components, potentially leading to poor operation and/or failure.

Note: Consider strainer orientation to avoid collection of condensate (see diagram).

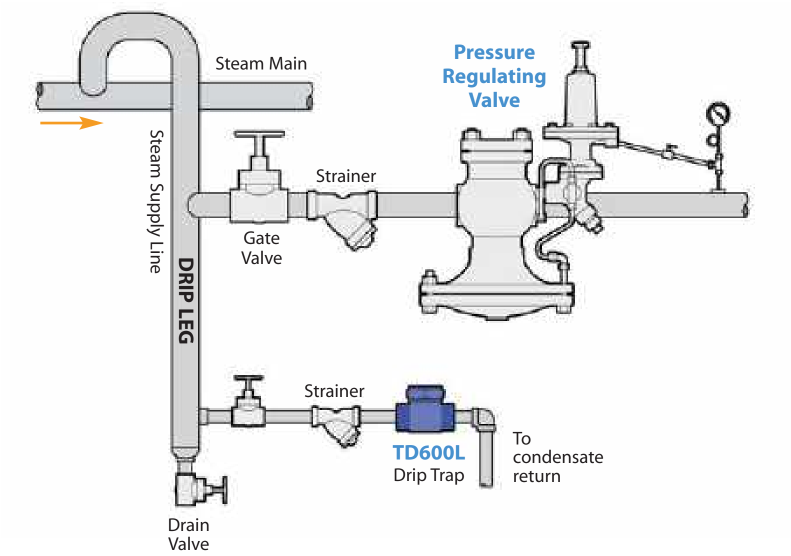


Blowdown valve oriented at same elevation as strainer ports to avoid condensate collection

Drip Legs & Steam Traps

To prevent condensate accumulated during shutdown from possibly damaging the regulator or piping at start-up, an adequately sized drip leg with steam trap should be installed prior to all regulators. This will also help protect the regulator during normal operation.

Note: Separators may be necessary when boiler carryover or "wet" steam is a concern.



Proper Start-up & Maintenance Procedures

It is important to follow good start-up practices to avoid operational complications and potential system damage. Starting a steam system too quickly or using an improper sequence may lead to a potentially hazardous working environment. Lack of system maintenance over time can also contribute to this situation.

It is imperative to develop proper start-up and maintenance procedures and train personnel on the importance of following them at all times.

Consult equipment manufacturers for specific guidelines, if necessary.

PRESSURE REDUCING STATION • Using Spring-Loaded Pilot

PURPOSE: For reducing system inlet pressure to a constant outlet pressure.

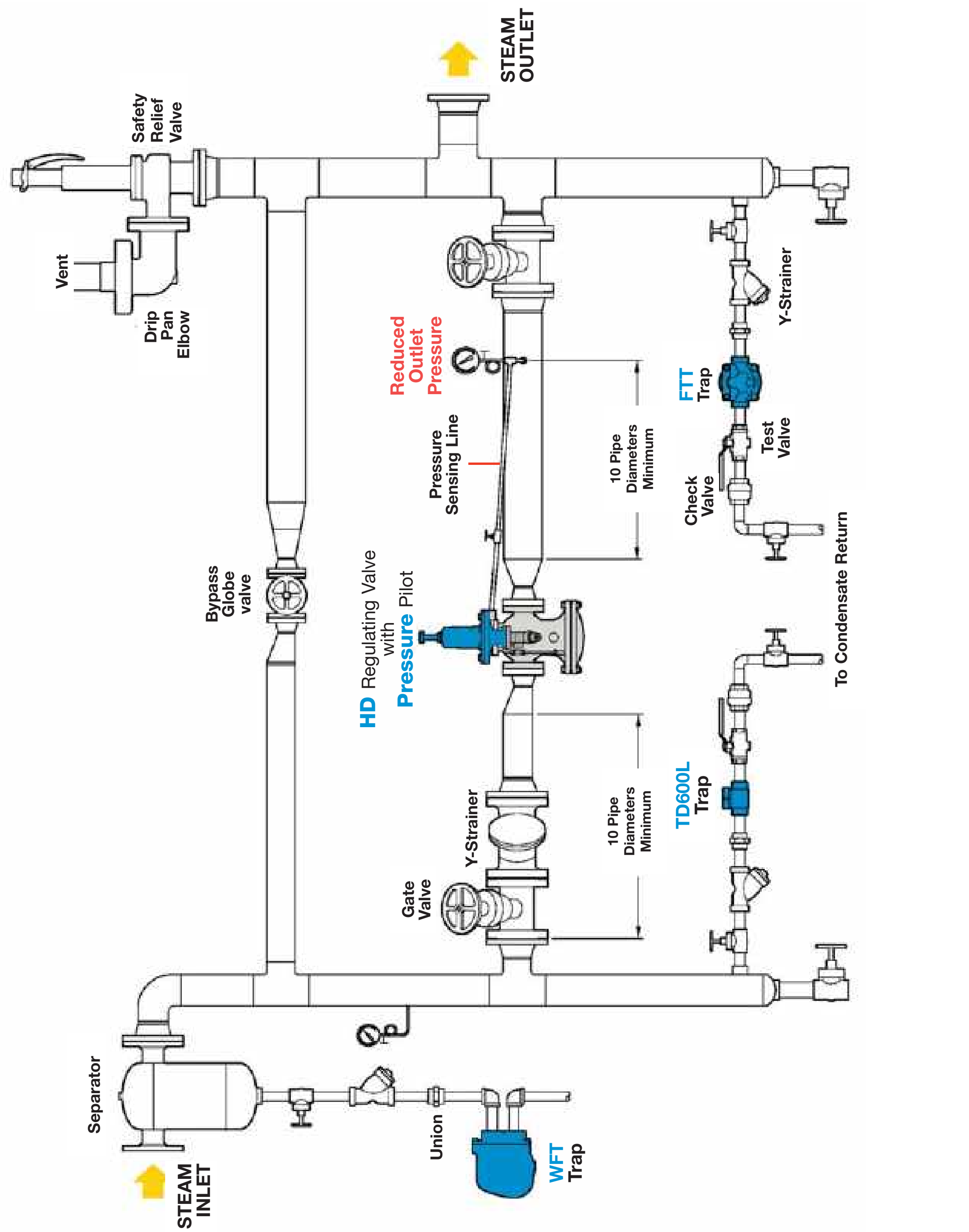
OPERATION: The pressure reducing valve (PRV) can be easily adjusted to set the desired outlet pressure and modulates to maintain that pressure setting. The PRV requires no external power source.

INSTALLATION GUIDELINES: (see Figure 5)

- This example depicts a pilot-operated steam PRV, whereby an external sensing line is required to sense downstream pressure. The end of the sensing line is placed away from the turbulent flow of the valve outlet. This helps to improve accuracy of the set pressure. Set pressure is adjusted by turning a screw on the pilot to increase or decrease compression on a balancing spring.
- For optimum operation and service life, maintain recommended minimum piping straight runs before and after the PRV. Inlet pipe diameters could be 1-2 sizes larger and outlet pipe diameters 2-3 sizes larger than the end connections of an appropriately sized PRV. The purpose of increasing the pipe size downstream of the regulator is to keep the steam velocity constant on both sides of the regulator.
- The pressure sensing line should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While the separator shown upstream is appropriate for protection of the PRV, it is not always required as a properly sized drip leg with steam trap may sufficient. It is recommended for systems where steam is known to be “wet” and the entrained moisture could affect valve performance and/or result in component damage.
- Consider installing a properly sized bypass line with globe valve to provide continuous operation should regulator maintenance be required.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. The SRV needs to handle the complete volume of steam from the regulator and bypass loop. Consult the factory for appropriate SRV sizing guidelines.

PRESSURE REDUCING STATION • Using Spring-Loaded Pilot

Figure 5:



SINGLE STAGE Pressure Reducing Station using Spring-loaded Pilot
(HD Regulator Applications)

ENGINEERING

PRESSURE REDUCING STATION • Using Air-Loaded Pilot

PURPOSE: For reducing system inlet pressure to a constant outlet pressure when valve is located in a remote location and/or using air pressure for control is desired.

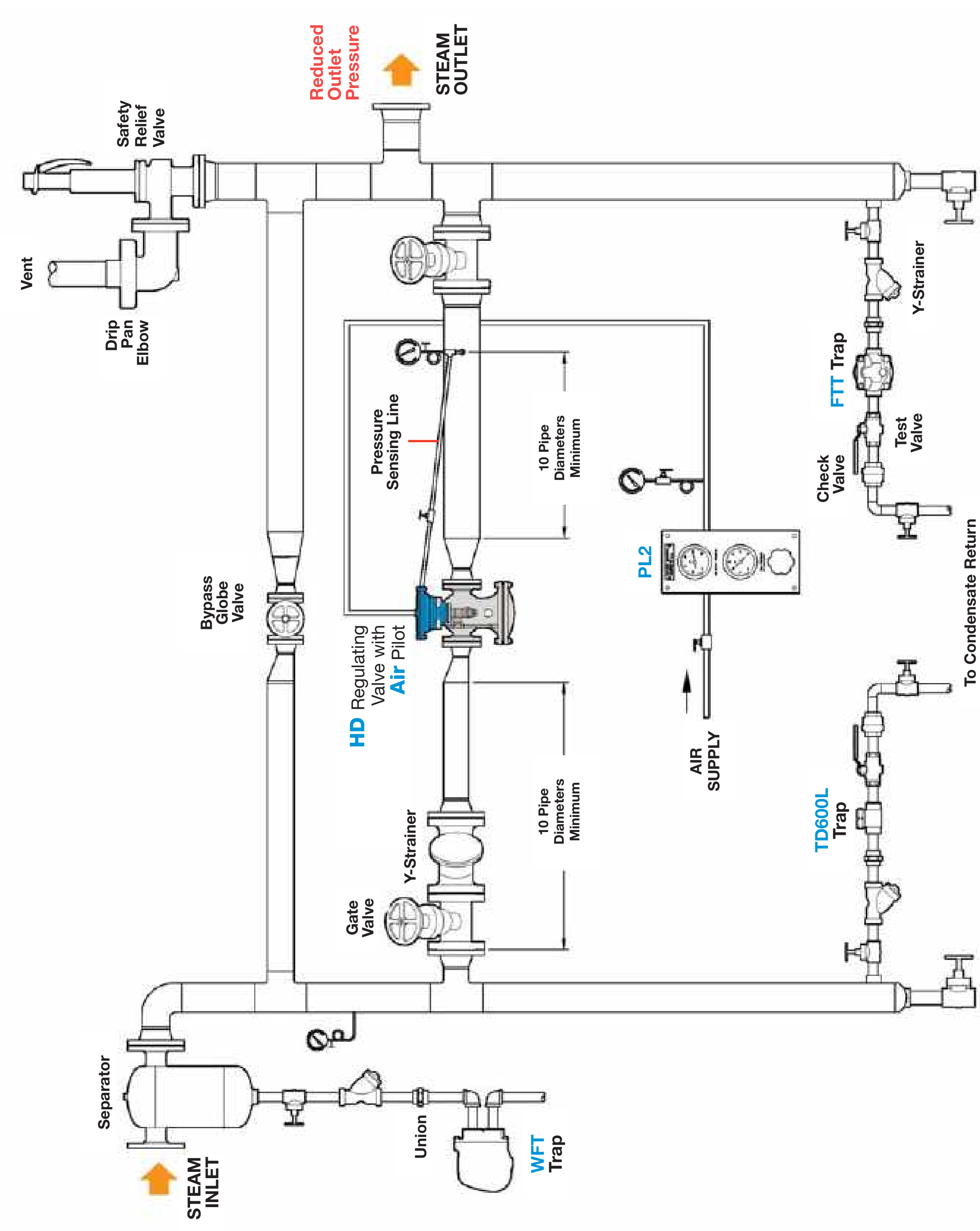
OPERATION: This combination of HD regulating valve and PA-pilot allows air to be used to control outlet pressure in lieu of the spring of a standard PP-pilot. Using air allows for simple adjustment of control pressure when valve is installed in a remote and/or difficult to access location.

INSTALLATION GUIDELINES: (see Figure 6)

- The desired set outlet pressure will determine the specific PA-Pilot required as well as the air supply pressure to attain the set pressure. Consult the appropriate section of this catalog or the factory for selection guidelines.
- For optimum operation and service life, maintain recommended minimum piping straight runs before and after the PRV. Inlet pipe diameters could be 1-2 sizes larger and outlet pipe diameters 2-3 sizes larger than the end connections of an appropriately sized PRV. The purpose of increasing the pipe size downstream of the regulator is to keep the steam velocity constant on both sides of the regulator.
- The pressure sensing line should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While the separator shown upstream is appropriate for protection of the PRV, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be “wet” and the entrained moisture could affect valve performance and/or result in component damage.
- Consider installing a properly sized bypass line with globe valve to provide continuous operation should regulator maintenance be required.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. The SRV needs to handle the complete volume of steam from the regulator and bypass loop. Consult the factory for appropriate SRV sizing guidelines.

PRESSURE REDUCING STATION • Using Air-Loaded Pilot

Figure 6:



Pressure Reducing Station with Air-Loaded Pilot for Remote Installations
(HD Regulator Applications)

PRESSURE REDUCING STATION • 2-Stage (Series) for High Pressure Turndown

PURPOSE: For reducing steam system inlet pressure to a constant outlet pressure when the pressure drop exceeds the recommended operation of a single-stage pressure regulating valve (PRV). This will help reduce overall velocity, resulting in less noise and improved equipment service life.

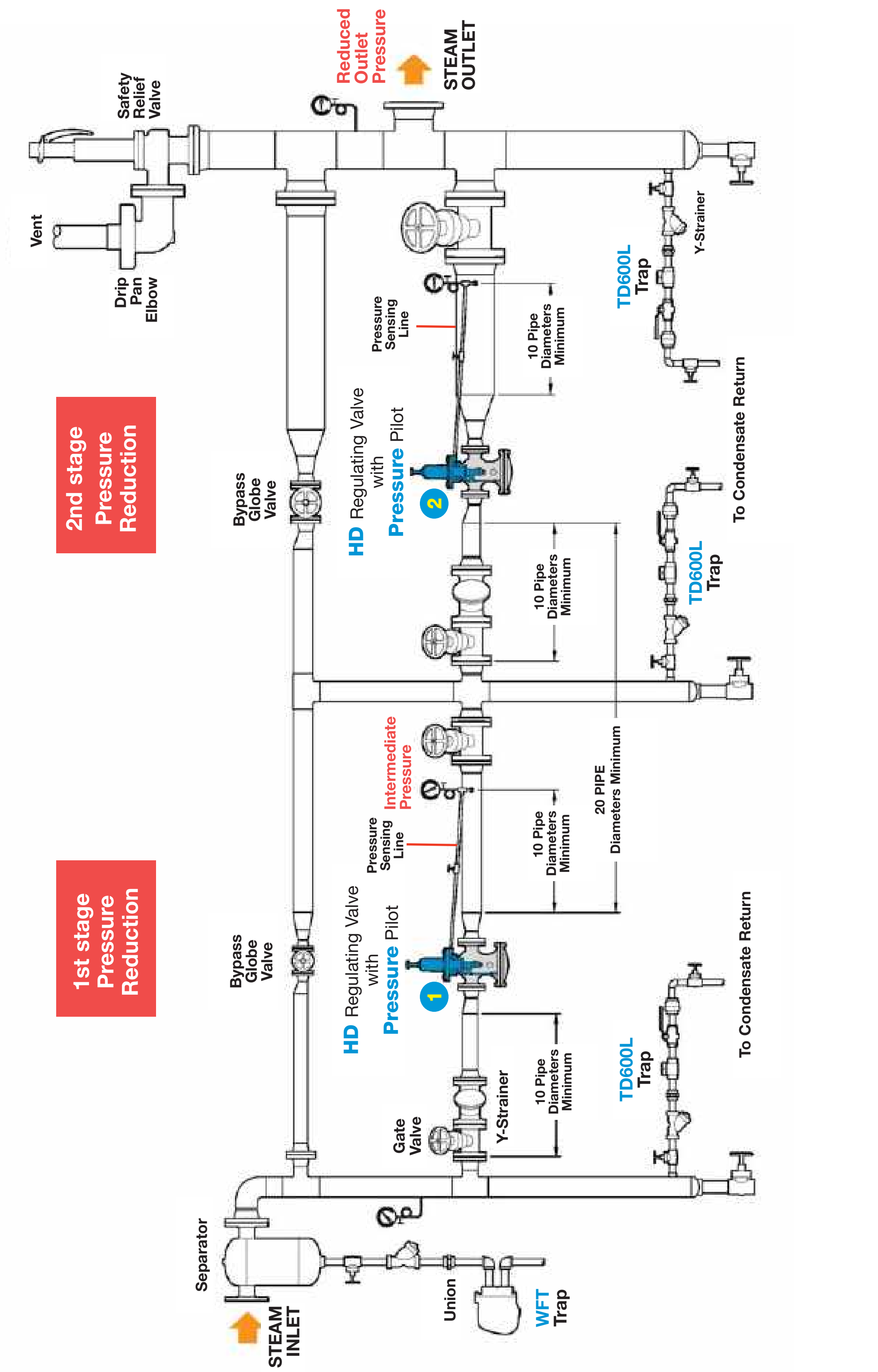
OPERATION: The 1st stage PRV is selected to reduce the initial steam pressure to a reasonable pressure between the initial inlet and desired outlet delivery pressure. This intermediate pressure is typically selected to ensure that each PRV is within recommended turndown guidelines. However, it is also possible there will be a use for steam at a specific intermediate pressure, which must be considered when evaluating turndown and sizing guidelines. The 2nd stage PRV, installed in series with the 1st stage, then reduces pressure to the final outlet delivery pressure. Individual valve setting and operation is the same as for single-stage applications.

INSTALLATION GUIDELINES: (see Figure 7)

- This example depicts a two-stage (series) pilot-operated steam PRV pressure reducing station using HD Regulators with Pressure Pilot. An external sensing line is required to sense downstream pressure from each regulator. The end of each sensing line is placed away from the turbulent flow at the valve outlet. This helps to improve accuracy of the set pressures. Set pressure for each PRV is adjusted by turning a screw on the pilot to increase or decrease compression on a balancing spring.
- For optimum operation and service life, maintain recommended minimum piping straight runs before and after the PRV. Inlet pipe diameters could be 1-2 sizes larger and outlet pipe diameters 2-3 sizes larger than the end connections of an appropriately sized PRV. The purpose of increasing the pipe size downstream of the regulator is to keep the steam velocity constant on both sides of the regulator.
- Each pressure sensing line should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While the separator shown upstream is appropriate for protection of the PRVs, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be “wet” and the entrained moisture could affect valve performance and/or result in component damage.
- Consider installing a properly sized bypass line with globe valve on each stage, to provide continuous operation should regulator maintenance be required.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. The SRV needs to handle the complete volume of steam from the regulator and bypass loop. Consult the factory for appropriate SRV sizing guidelines.

PRESSURE REDUCING STATION • 2-Stage (Series) for High Pressure Turndown

Figure 7:



Two-Stage (Series) Pressure Reducing Station
(HD Regulator Applications)

PRESSURE REDUCING STATION • Parallel for High Flow Turndown

PURPOSE: For reducing steam system inlet pressure to a constant outlet pressure when steam flow rates vary widely. This will help improve system rangeability resulting in more accurate control.

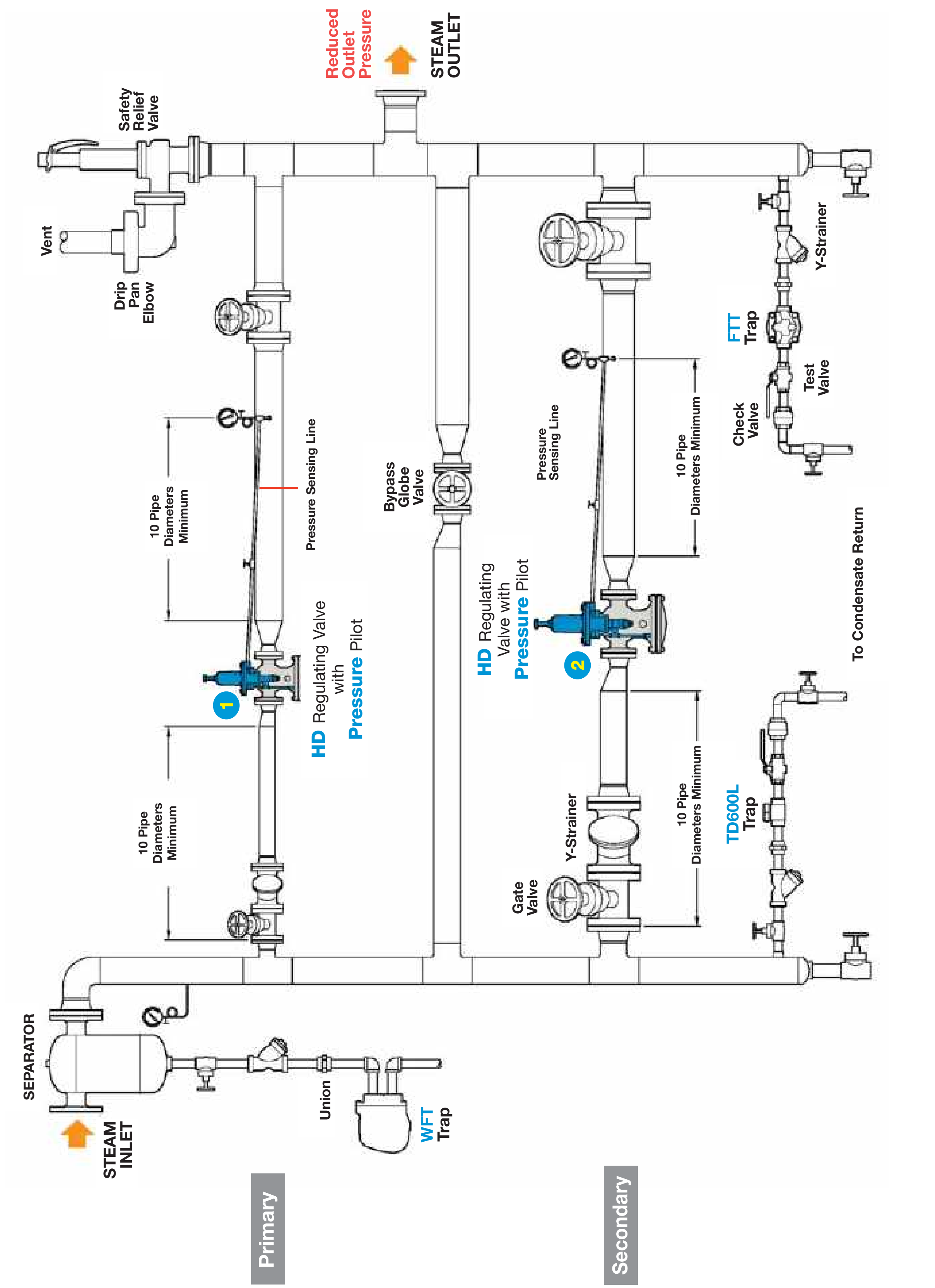
OPERATION: Because regulators are simple, self-powered devices which do not rely on an external control signal to determine valve steam position, they may not have the flow rangeability of control valves. Therefore, if a system has large flow variations (such as summer and winter loads), multiple regulators should be considered. Typically referred to as a 1/3 - 2/3 system, one valve may be sized for approximately 1/3 of the total maximum load demand and a larger valve for the remaining 2/3. When full load is required, both valves will be open and regulating. The small regulator is typically the primary valve and is set at a pressure 2 psi higher than the larger secondary valve. This allows the small regulator to be the only one flowing when demand is low. When flow increases and the small valve cannot keep up with the demand, the downstream pressure will begin to drop which will allow the larger secondary valve to open in order to help satisfy the demand. Although the smaller regulator is commonly selected as the primary valve, either the smaller or larger regulator may be set as the primary valve based on anticipated load demand requirements. The primary valve should always be set a minimum of 2 psi above the secondary valve.

INSTALLATION GUIDELINES: (see Figure 8)

- This example depicts a parallel pilot-operated steam PRV pressure reducing station using HD Regulators with Pressure Pilot. An external sensing line is required to sense downstream pressure from each regulator. The end of each sensing line is placed away from the turbulent flow at the valve outlet. This helps to improve accuracy of the set pressures. Set pressure for each PRV is adjusted by turning a screw on the pilot to increase or decrease compression on a balancing spring.
- Proper setting of the valves is key to proper operation. The chosen primary valve should be set at a pressure approximately 2 PSI higher than that of the secondary valve.
- For optimum operation and service life, maintain recommended minimum piping straight runs before and after the PRV. Inlet pipe diameters could be 1-2 sizes larger and outlet pipe diameters 2-3 sizes larger than the end connections of an appropriately sized PRV. The purpose of increasing the pipe size downstream of the regulator is to keep the steam velocity constant on both sides of the regulator.
- Each pressure sensing line should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While the separator shown upstream is appropriate for protection of the PRV, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be "wet" and the entrained moisture could affect valve performance and/or result in component damage.
- Consider installing a properly sized bypass line with globe valve to provide continuous operation should regulator maintenance be required.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. The SRV needs to handle the complete volume of steam from the regulator and bypass loop. Consult the factory for appropriate SRV sizing guidelines.

PRESSURE REDUCING STATION • Parallel for High Flow Turndown

Figure 8:



PARALLEL Pressure Reducing Station
(HD Regulator Applications)

ENGINEERING

PRESSURE REDUCING STATION • for Combination High Pressure & High Flow Turndown

PURPOSE: For reducing steam system inlet pressure to a constant outlet pressure when flow conditions vary widely combined with a high pressure drop (i.e. higher than the recommended range of a single-stage regulator).

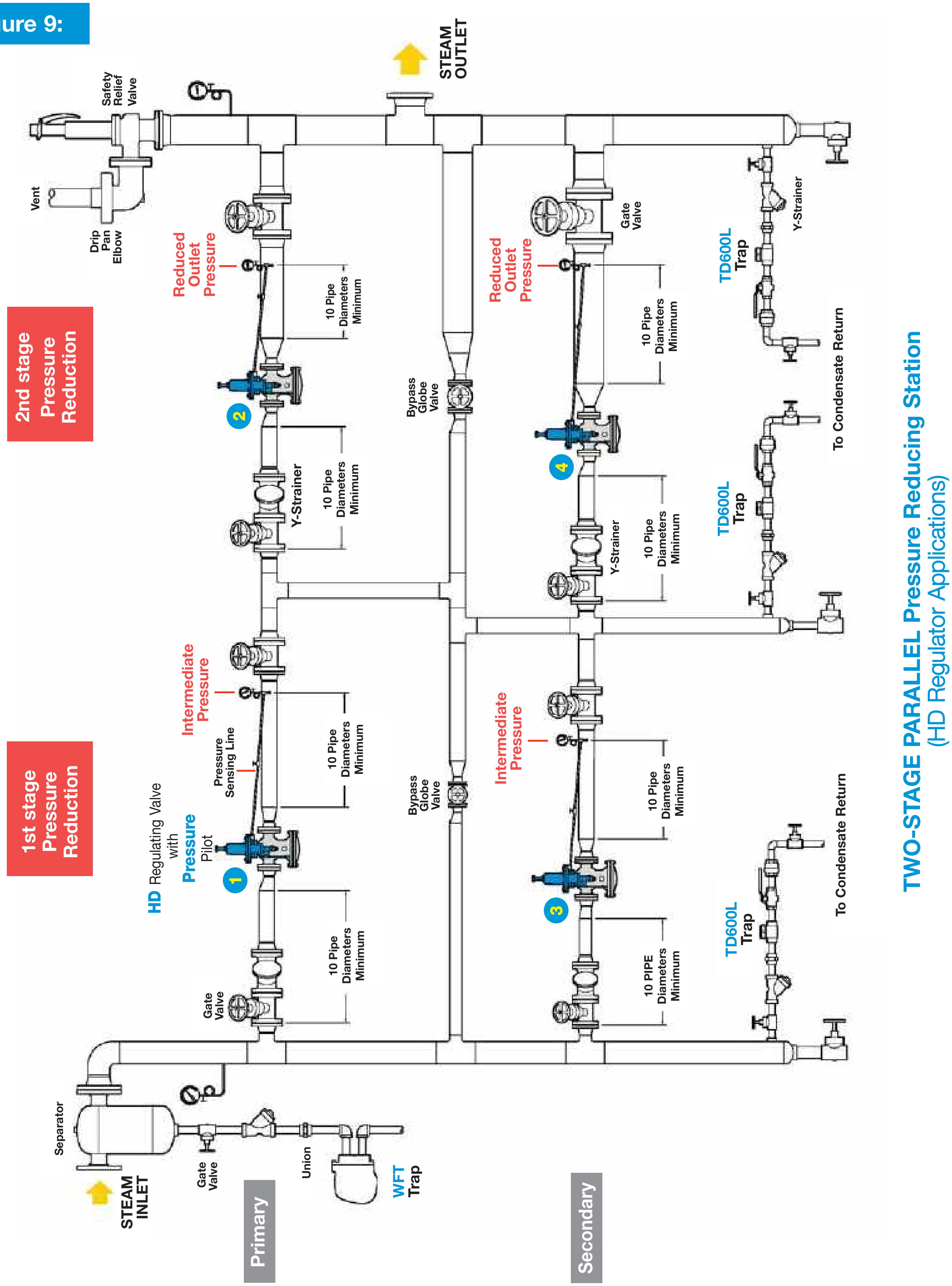
OPERATION: This system is a combination of Two-Stage (Series) and Parallel pressure reducing stations and operates based on the individual principles of each system. Each series of valves will be sized to handle a portion of the total maximum load demand, typically 1/3 and 2/3 of the total anticipated flow. If the smaller series of valves is determined to operate as the primary, then the 2nd stage valve will be set 2 psi higher than the 2nd stage valve in the secondary series. This allows the primary series to be the only one flowing when demand is low. When flow increases and the primary series cannot keep up with demand, the downstream pressure will being to drop which will allow the larger secondary series of valves to open in order to help satisfy the demand.

INSTALLATION GUIDELINES: (see Figure 9)

- This example depicts a two-stage parallel pilot-operated steam PRV pressure reducing station using HD Regulators with Pressure Pilot. An external sensing line is required to sense downstream pressure from each regulator. The end of each sensing line is placed away from the turbulent flow at the valve outlet. This helps to improve accuracy of the set pressures. Set pressure for each PRV is adjusted by turning a screw on the pilot to increase or decrease compression on a balancing spring.
- Proper setting of the valves is key to proper operation. The chosen 1st stage primary valve should be set at a pressure approximately 2 PSI higher than that of the 1st stage secondary valve.
- For optimum operation and service life, maintain recommended minimum piping straight runs before and after the PRV. Inlet pipe diameters could be 1-2 sizes larger and outlet pipe diameters 2-3 sizes larger than the end connections of an appropriately sized PRV. The purpose of increasing the pipe size downstream of the regulator is to keep the steam velocity constant on both sides of the regulator.
- Each pressure sensing line should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While the separator shown upstream is appropriate for protection of the PRVs, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be “wet” and the entrained moisture could affect valve performance and/or result in component damage.
- Consider installing a properly sized bypass line with globe valve on each stage, to provide continuous operation should regulator maintenance be required.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. The SRV needs to handle the complete volume of steam from the regulator and bypass loops. Consult the factory for appropriate SRV sizing guidelines.

PRESSURE REDUCING STATION • for Combination High Pressure & High Flow Turndown

Figure 9:



ENGINEERING

REGULATING VALVE APPLICATIONS

TEMPERATURE CONTROL • of Heat Exchanger with Pressure Limiting Pilot

PURPOSE: For accurately controlling both temperature of a product being heated in heat transfer equipment as well as limiting the pressure of the incoming steam, providing optimum heat transfer characteristics.

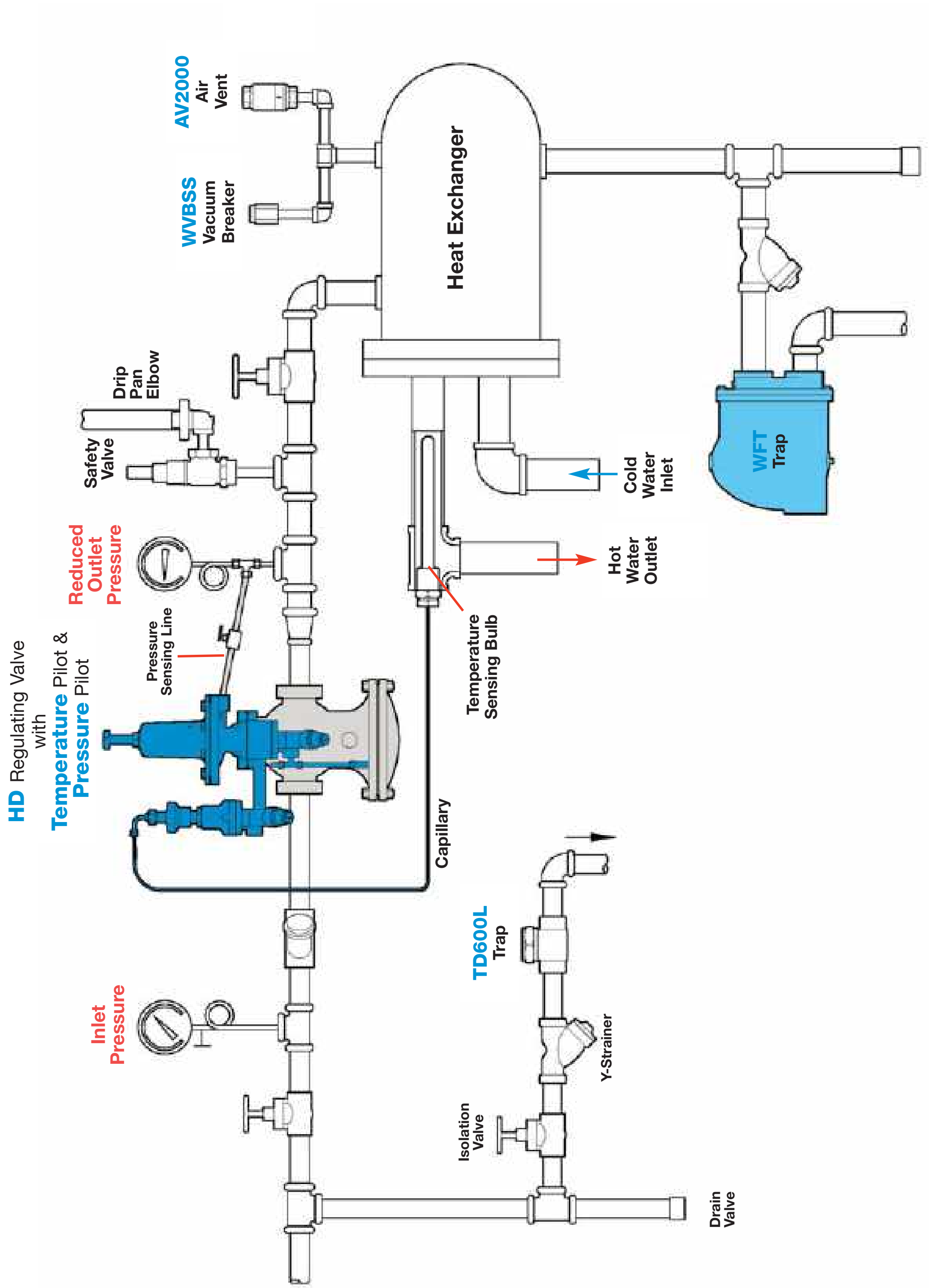
OPERATION: When a pilot-operated HD valve is selected, a single valve can be used for both pressure and temperature control when equipped with a PP-Pilot and PT-Pilot. As temperature at the sensing bulb falls below set point, the valve begins to modulate open to supply steam for heating. Supply pressure to the heat exchanger is them limited by adjusting the pressure pilot to the recommended value for optimum heat transfer and/or a limiting pressure of the heat transfer equipment. The HD Regulator with combination PT & PP Pilots requires no external power source.

INSTALLATION GUIDELINES: (see Figure 10)

- The temperature and pressure pilots should be set individually, starting slowly and gradually with the PT-pilot.
- Care should be given to the installation of the temperature sensing bulb to ensure full immersion in the liquid. The sensing bulb should be placed as close as possible to the heat exchanger vessel to ensure accurate temperature control of the process fluid.
- For optimum operation and service life, maintain recommended minimum piping straight runs before and after the Regulator. Inlet pipe diameters could be 1-2 sizes larger and outlet pipe diameters 2-3 sizes larger than the end connections of an appropriately sized Regulator. The purpose of increasing the pipe size downstream of the regulator is to keep the steam velocity constant on both sides of the regulator.
- The pressure sensing line should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While a separator is appropriate for protection of the Regulator, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be "wet" and the entrained moisture could affect valve performance and/or result in component damage.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- The vacuum breaker and auxiliary air vent located at the top of the heat exchanger vessel promotes proper drainage and optimum heat transfer. The vacuum breaker allows system equalization with atmospheric air to allow gravity condensate drainage when vacuum is formed from condensing steam. The air vent improves heat-up times and overall heat transfer by expelling accumulated air on start-up.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. Consult the factory for appropriate SRV sizing guidelines.

TEMPERATURE CONTROL • of Heat Exchanger with Pressure Limiting Pilot

Figure 10:



Temperature Control of a Heat Exchanger with Pressure Limiting
(HD Regulator Applications)

ENGINEERING

TEMPERATURE CONTROL
of a BATCH PROCESS with Electrical Time Sequence Programmer (Solenoid Pilot)

PURPOSE: For accurately controlling temperature of a batch process where on-off operation is to be electronically controlled.

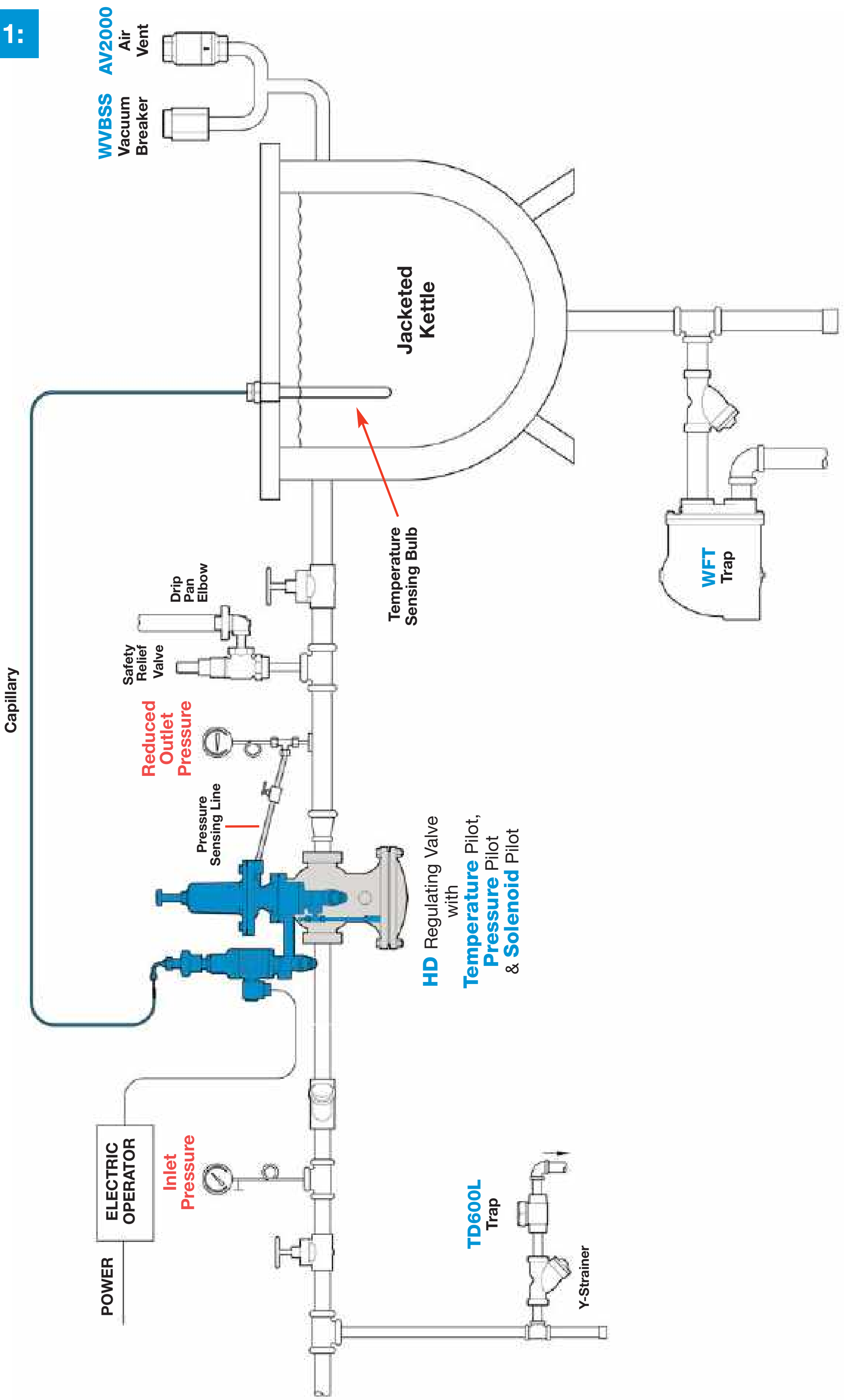
OPERATION: Operation is similar to that of the pressure and temperature combination pilot-operated regulator whereby the temperature (PT) pilot senses the temperature of the heated product (e.g. water) and appropriately modulates the flow of steam. Pressure is limited by the pressure (PP) pilot. The solenoid valve (PS-pilot) is electronically activated to control on-off operation of the batch process.

INSTALLATION GUIDELINES: (see Figure 11)

- The temperature and pressure pilots should be set individually, starting slowly and gradually with the PT-pilot.
- For optimum operation and service life, maintain recommended minimum piping straight runs before and after the PRV. Inlet pipe diameters could be 1-2 sizes larger and outlet pipe diameters 2-3 sizes larger than the end connections of an appropriately sized Regulator. The purpose of increasing the pipe size downstream of the regulator is to keep the steam velocity constant on both sides of the regulator.
- The pressure sensing line should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While a separator is appropriate for protection of the Regulator, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be “wet” and the entrained moisture could affect valve performance and/or result in component damage.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- The vacuum breaker and auxiliary air vent located at the top of the jacketed kettle vessel promotes proper drainage and optimum heat transfer. The vacuum breaker allows system equalization with atmospheric air to allow gravity condensate drainage when vacuum is formed from condensing steam. The air vent improves heat-up times and overall heat transfer by expelling accumulated air on start-up.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. Consult the factory for appropriate SRV sizing guidelines.

TEMPERATURE CONTROL
of a BATCH PROCESS with Electrical Time Sequence Programmer (Solenoid Pilot)

Figure 11:



AUTOMATIC TEMPERATURE CONTROL of a BATCH PROCESS with Electrical Time Sequence Programmer (Solenoid Pilot)
(HD Regulator Applications)

REGULATING VALVE APPLICATIONS

TEMPERATURE CONTROL of a SEMI-INSTANTANEOUS HEATER using a Self-Contained Temperature Regulating Valve

PURPOSE: For accurate control of the temperature of a product being heated when the benefits of a self-contained regulator are required.

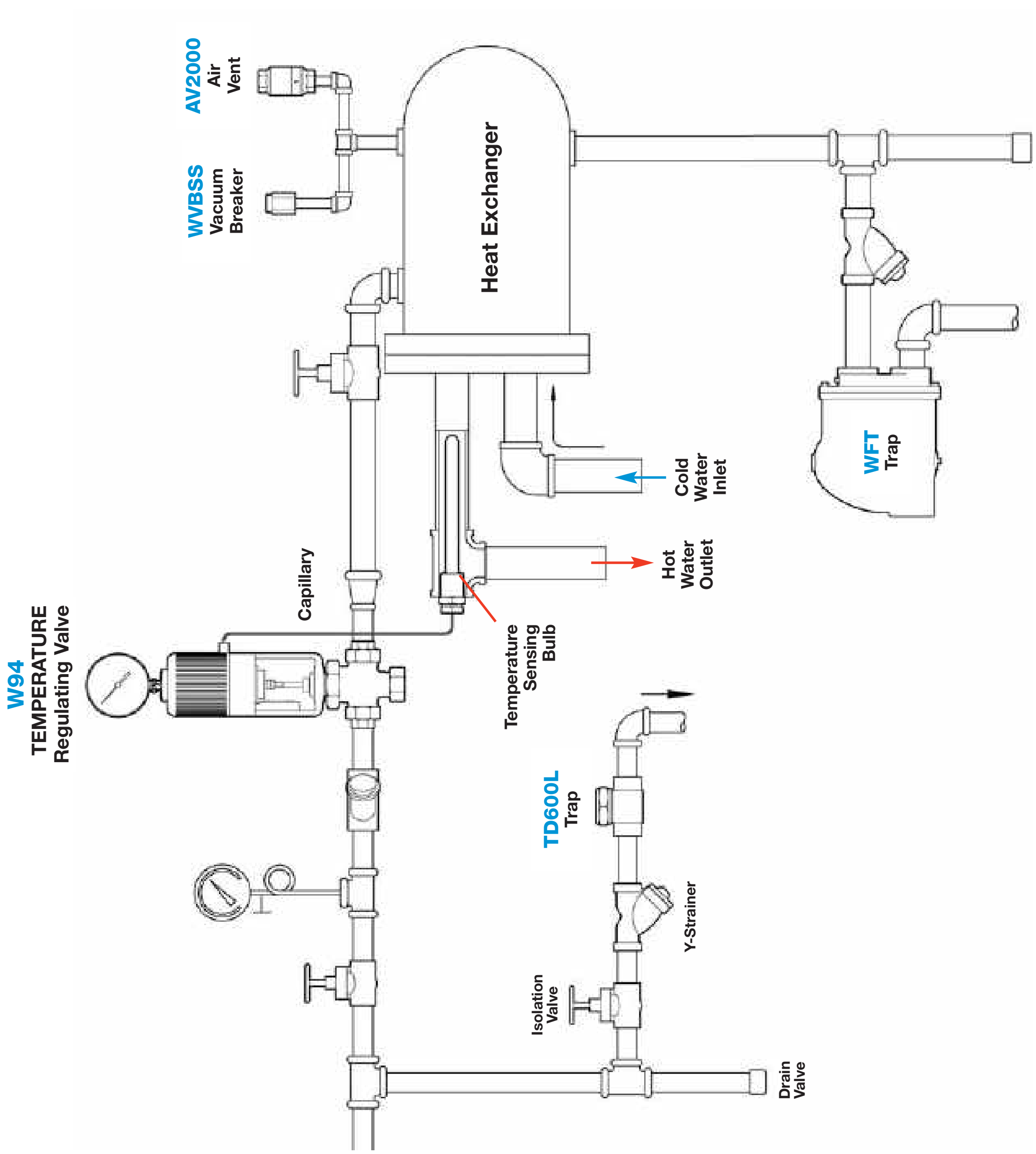
OPERATION: A self-contained temperature regulating valve (TRV) such as the W94, offers response times and characteristics suitable for semi-instantaneous heating applications. The temperature sensing bulb senses the temperature of the liquid being heated and allows modulation of the valve for appropriate supply of steam.

INSTALLATION GUIDELINES: (see Figure 12)

- Care should be given to the installation of the temperature sensing bulb to ensure full immersion in the liquid. The sensing bulb should be placed as close as possible to the heater tank to ensure accurate temperature control of the process fluid.
- All pressure sensing lines should slope downwards, away from the regulator, to prevent condensate from entering the pilot.
- Eccentric reducers, if required, are used on valve inlets to prevent accumulation of condensate which could become entrained with high-velocity steam, possibly resulting in dangerous waterhammer.
- While a separator is appropriate for protection of the Regulator, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be "wet" and the entrained moisture could affect valve performance and/or result in component damage.
- Consider low-cracking pressure (1/4 PSI opening pressure) check valves after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- The vacuum breaker and auxiliary air vent located at the top of the heater tank promotes proper drainage and optimum heat transfer. The vacuum breaker allows system equalization with atmospheric air to allow gravity condensate drainage when vacuum is formed from condensing steam. The air vent improves heat-up times and overall heat transfer by expelling accumulated air on start-up.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. The SRV needs to handle the complete volume of steam from the regulator and bypass loop. Consult the factory for appropriate SRV sizing guidelines.

TEMPERATURE CONTROL of a SEMI-INSTANTANEOUS HEATER using a Self-Contained Temperature Regulating Valve

Figure 12:



Semi-Instantaneous Hot Water Heater with W94 Temperature Regulator
(Temperature Regulator Applications)

ENGINEERING

CONDENSATE DRAINAGE • using Pump-Trap

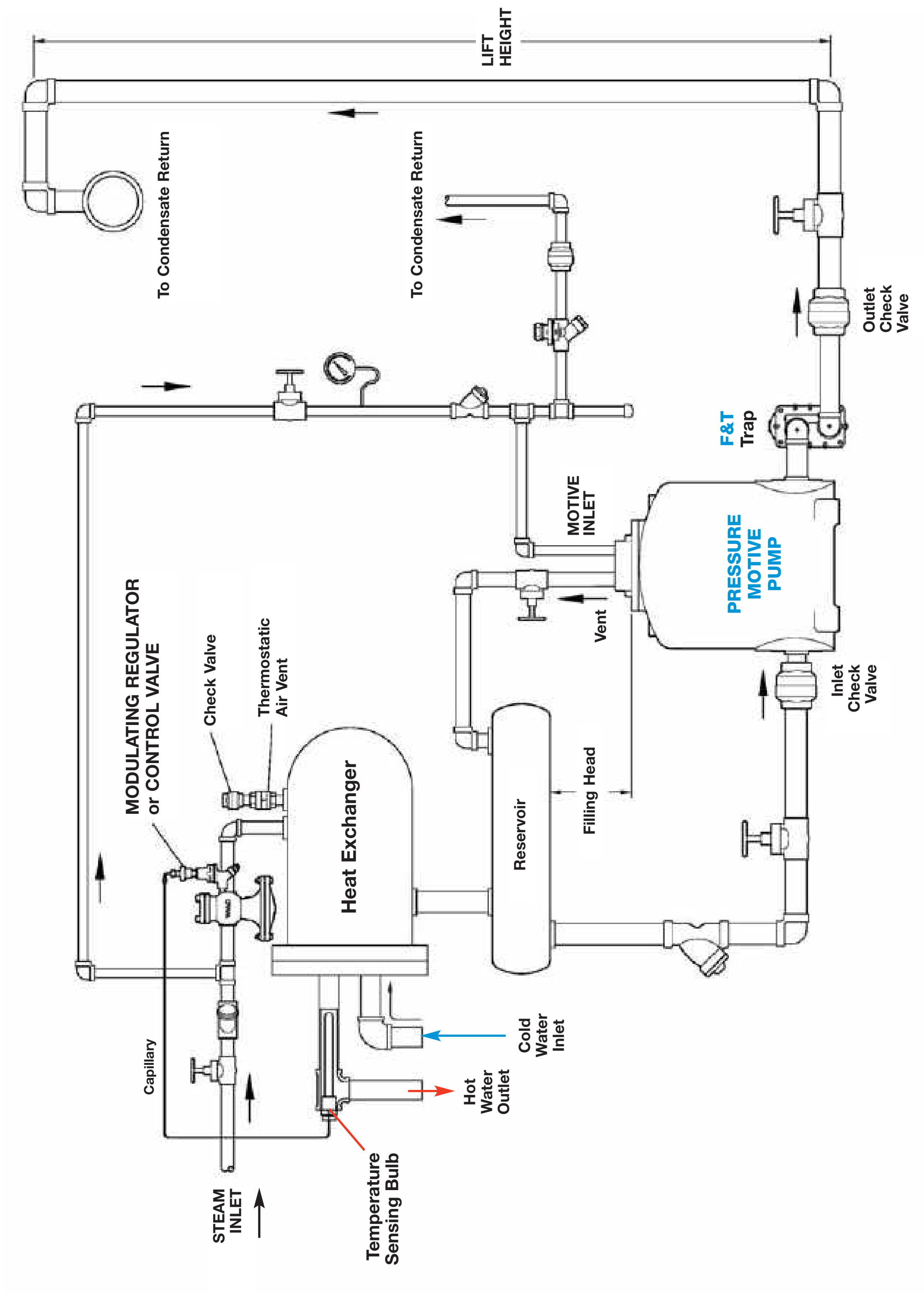
PURPOSE: For removing condensate from below steam heat transfer equipment when a modulating valve is used for control, and condensate discharge is elevated and/or pressurized, resulting in Stall condition.

OPERATION: The Pressure Motive Pump (PMP) is used to overcome the stall condition that exists when steam feeding a single piece of heat transfer equipment is controlled by a modulating steam valve and steam pressure falls below system back pressure as the valve closes. A steam trap is required after the PMP to prevent the loss of live steam when the system is under positive pressure. Operating as a closed loop provides an energy-efficient system by eliminating the need to vent flash steam.

INSTALLATION GUIDELINES: (see Figure13)

- Proper installation and piping of the pump vent line is critical to ensure the system operates correctly. Follow guidelines or consult factory for additional information.
- Maintain proper fill head above the top of the pump to ensure proper function of the pump and system. A suitably sized reservoir or oversized piping should be installed ahead of the pump for accumulation of condensate during the pump's discharge cycle (i.e. when not filling).
- The steam trap after the pump must be sized in conjunction with the pump to ensure proper function as a system. Improper sizing may result in reduced capacity leading to condensate back-up, poor heat transfer and potentially dangerous waterhammer. Consult appropriate sections of this catalog or the factory for guidelines regarding proper sizing of the pump-trap combination.
- While a separator is appropriate for protection of the Regulator, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be "wet" and the entrained moisture could affect valve performance and/or result in component damage.
- Low-cracking pressure (1/4 PSI opening pressure) check valves should be installed after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- The thermostatic air vent installed on the heat exchanger promotes optimum heat transfer. The air vent improves heat-up times and overall heat transfer by expelling accumulated air on start-up. When properly sized and installed, the pump-trap combination can operate in sub-atmospheric (i.e. vacuum) conditions; therefore, a vacuum breaker should not be used.

Figure 13:



Drainage of a Single Source of Condensate for a Closed Loop System
(Pump-Trap Applications)

ENGINEERING

PMP & PUMP-TRAP APPLICATIONS

CONDENSATE DRAINAGE from Below Grade • using Pump-Trap

PURPOSE: For drainage of condensate from below process equipment where fill head is limited due to height restrictions and the pump must be installed below grade.

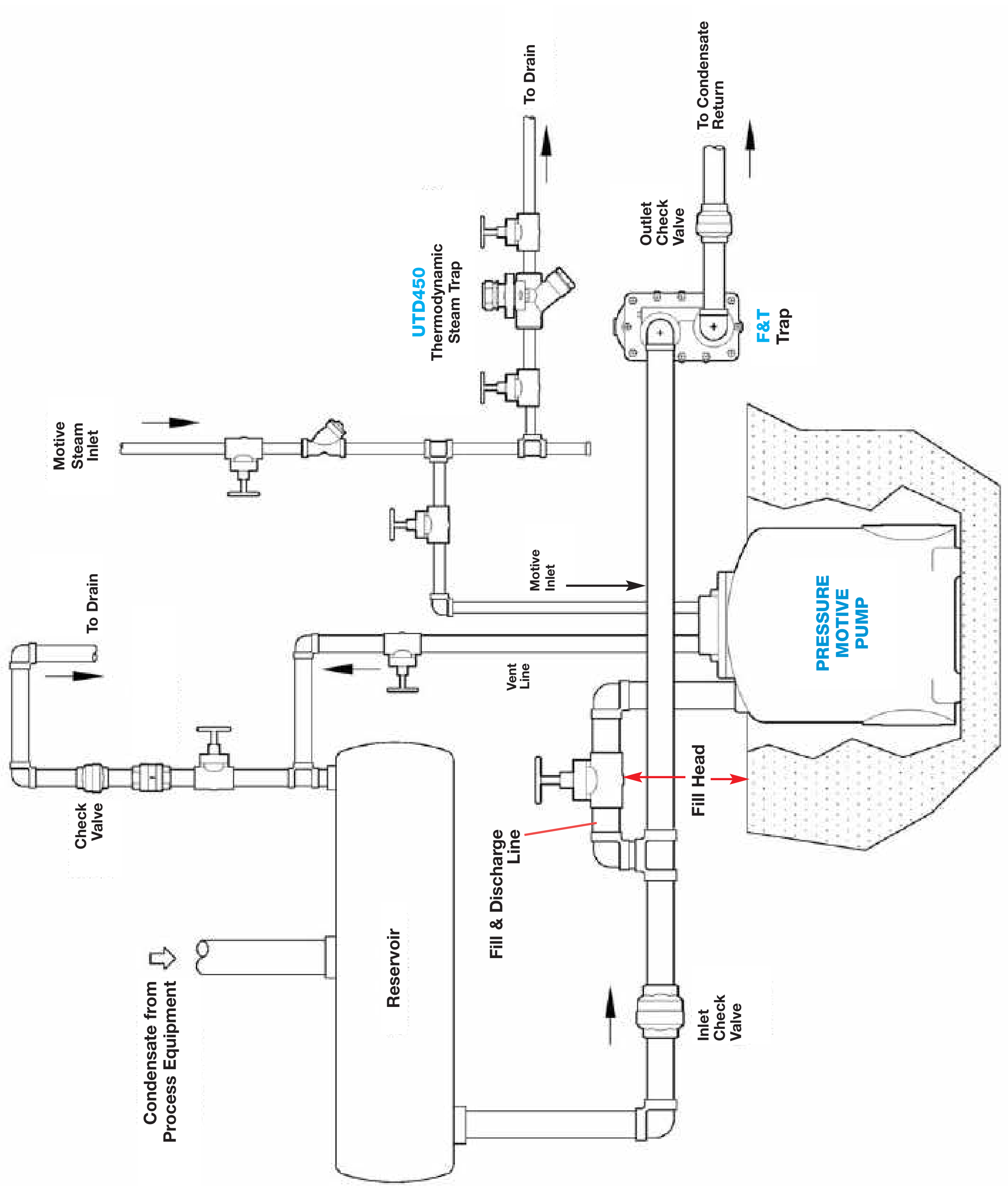
OPERATION: When fill head is restricted and it is more suitable to create a pit below grade than reposition process equipment, the Pressure Motive Pump (PMP) may be modified so both condensate inlet and outlet connections are on top to limit the necessary pit size. When stall exists, condensate will accumulate between the inlet and outlet check valves and eventually drain into and fill the PMP tank. Once the PMP fills and its mechanism trips, high pressure motive steam will enter the pump tank and force condensate back out the same connection. The check valves will direct the flow of pumped condensate into the return piping.

INSTALLATION GUIDELINES: (see Figure 14)

- The positioning of the check valves and PMP fill/discharge line are the key elements which allow the system to function properly. The check valves dictate the proper direction of condensate flow for both fill and discharge cycles of the PMP. The PMP fill/discharge line should be taken off the top, as shown, so condensate only accumulates and fills the pump during stall.
- Proper installation and piping of the pump vent line is critical to ensure the system operates correctly. Follow guidelines or consult factory for additional information.
- Maintain proper fill head above the top of the pump to ensure proper function of the pump and system. A suitably sized reservoir or oversized piping should be installed ahead of the pump for accumulation of condensate during the pump's discharge cycle (i.e. when not filling).
- The steam trap after the pump must be sized in conjunction with the pump to ensure proper function as a system. Improper sizing may result in reduced capacity leading to condensate back-up, poor heat transfer and potentially dangerous waterhammer. Consult appropriate sections of this catalog or the factory for guidelines regarding proper sizing of the pump-trap combination.
- Low-cracking pressure (1/4 PSI opening pressure) check valves should be installed after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.

CONDENSATE DRAINAGE from Below Grade • using Pump-Trap

Figure 14:



Drainage of Condensate from BELOW GRADE for a Closed Loop System in Situations with Minimal Fill Head
(Pump-Trap Applications)

CONDENSATE DRAINAGE using Vertical Reservoir and Pump-Trap

PURPOSE: For drainage of condensate from below process equipment where fill head is limited due to height restrictions and a horizontal reservoir cannot be installed.

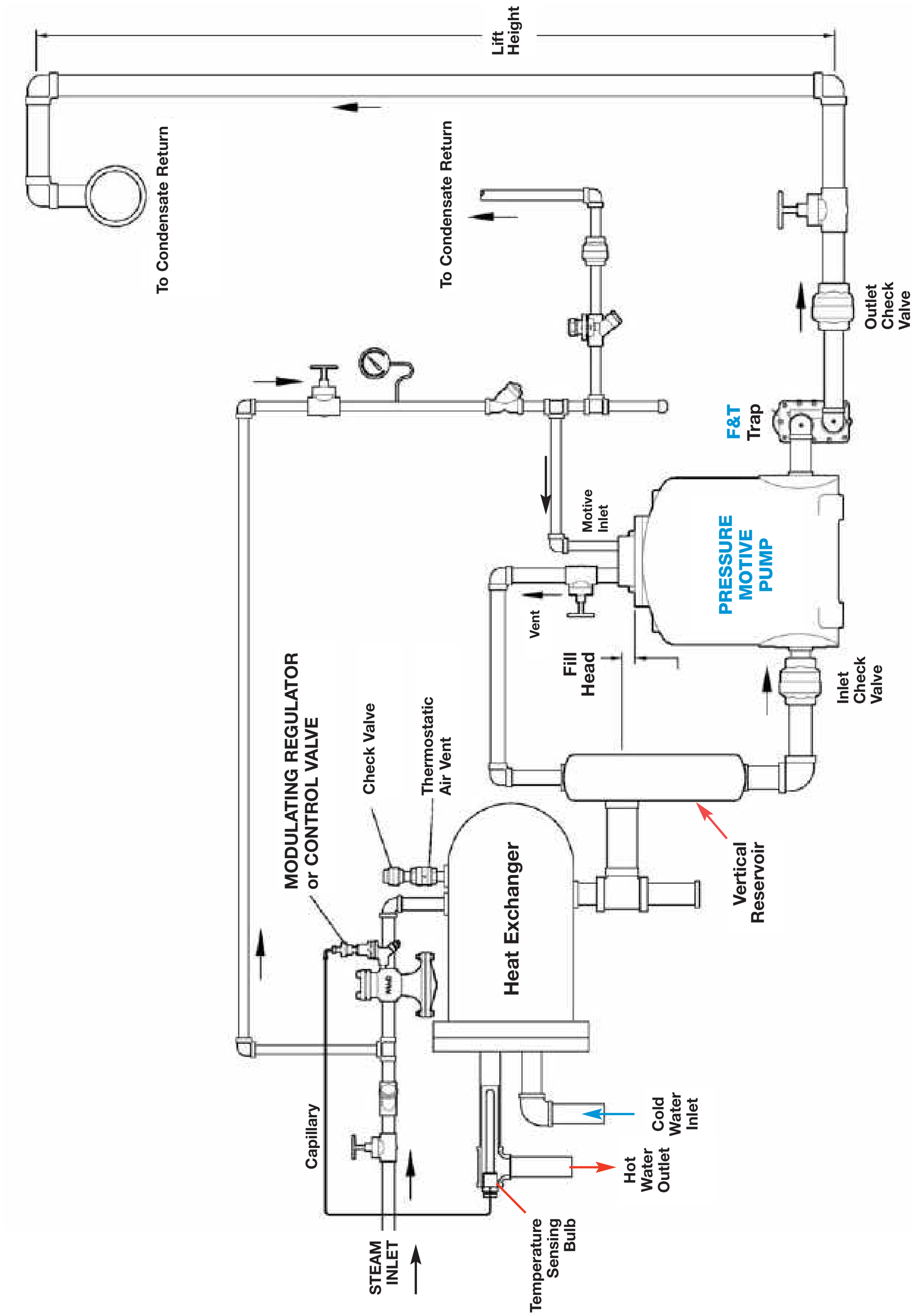
OPERATION: This system functions similarly to the system shown on page 455. However, when fill head is restricted due to heat exchanger height above ground, consider a vertical reservoir in lieu of a horizontal reservoir. This would accommodate condensate back-up as well as provide sufficient vapor space for the adequate venting of the pump while providing sufficient fill head to ensure proper operation of the pump.

INSTALLATION GUIDELINES: (see Figure 15)

- The vertical reservoir must be properly designed and installed to allow adequate condensate back-up during the pump's discharge cycle (i.e. when not filling), unobstructed venting of the pump, as well as sufficient fill head to ensure proper pump and system operation. Consult factory for additional assistance.
- Proper installation and piping of the pump vent line is critical to ensure the system operates correctly. Follow guidelines or consult factory for additional information.
- The steam trap after the pump must be sized in conjunction with the pump to ensure proper function as a system. Improper sizing may result in reduced capacity leading to condensate back-up, poor heat transfer and potentially dangerous waterhammer. Consult appropriate sections of this catalog or the factory for guidelines regarding proper sizing of the pump-trap combination.
- Low-cracking pressure (1/4 PSI opening pressure) check valves should be installed after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system. .
- The thermostatic air vent located on the heat exchanger promotes optimum heat transfer. The air vent improves heat-up times and overall heat transfer by expelling accumulated air on start-up. When properly sized and installed, the pump-trap combination can operate in sub-atmospheric (i.e. vacuum) conditions; therefore, a vacuum breaker should not be used.

CONDENSATE DRAINAGE using Vertical Reservoir and Pump-Trap

Figure 15:



Drainage of Condensate from HEAT EXCHANGER positioned Close to the Ground
(Pump-Trap Applications)

ENGINEERING

FLASH STEAM RECOVERY

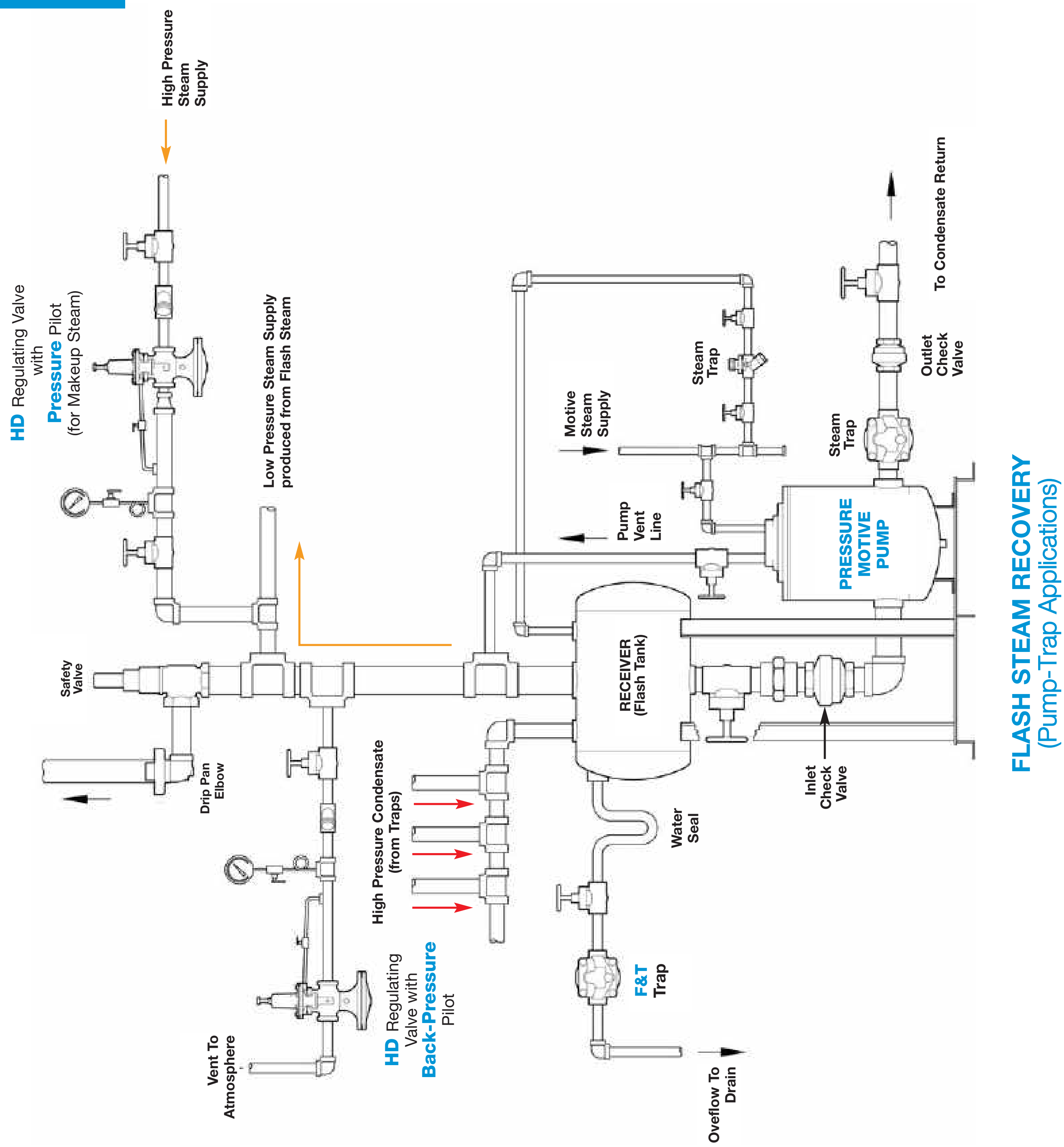
PURPOSE: For recovering flash steam from multiple condensate sources and drainage of the condensate when the total system back pressure is greater than the total of the individual source pressures.

OPERATION: Condensate at various pressures collects in a receiver (flash tank), equalizing the pressures to that of the flash tank. This allows drainage by gravity into the Pressure Motive Pump (PMP), filling the PMP until the internal mechanism reaches its upper trip point and activates the motive steam used for pumping. The flash steam generated from the high pressure condensate may be used to supplement other applications for optimum energy efficiency. The pressure in the receiver tank is maintained by a back pressure regulator and protected by a safety relief valve.

INSTALLATION GUIDELINES: (see Figure 16)

- The key element for proper system operation is the sizing of the receiver tank and receiver vent connection, which must accommodate the flash steam. Consult appropriate sections of this catalog or the factory for guidelines regarding proper sizing of the receiver tank and receiver vent connection.
- Proper installation and piping of the pump vent line is critical to ensure the system operates correctly. Follow guidelines or consult factory for additional information.
- Careful consideration should be given to sizing of the auxiliary components such as the back pressure regulator and safety relief valve.
- Maintain proper fill head above the top of the pump to ensure proper function of the pump and system. A suitably sized receiver or oversized piping should be installed ahead of the pump for accumulation of condensate during the pump's discharge cycle (i.e. when not filling).
- The steam trap after the pump must be sized in conjunction with the pump to ensure proper function as a system. Improper sizing may result in reduced capacity leading to condensate back-up, poor heat transfer and potentially dangerous waterhammer. Consult appropriate sections of this catalog or the factory for guidelines regarding proper sizing of the pump-trap combination.
- While the separator shown upstream is appropriate for protection of the PRV, it is not always required, as a properly sized drip leg with steam trap may be sufficient. It is recommended for systems where steam is known to be "wet" and the entrained moisture could affect valve performance and/or result in component damage.
- Low-cracking pressure (1/4 PSI opening pressure) check valves should be installed after steam traps when discharging into condensate return lines. Check valves eliminate the possibility of condensate backing up through the steam trap into the system.
- A safety relief valve (SRV) is appropriate where applicable codes dictate their requirement, or anywhere protection of downstream piping and equipment from over-pressurization is desired. Consult the factory for appropriate SRV sizing guidelines.

Figure 16:



ENGINEERING

REMOVAL OF WATER OR CONDENSATE FROM A PIT

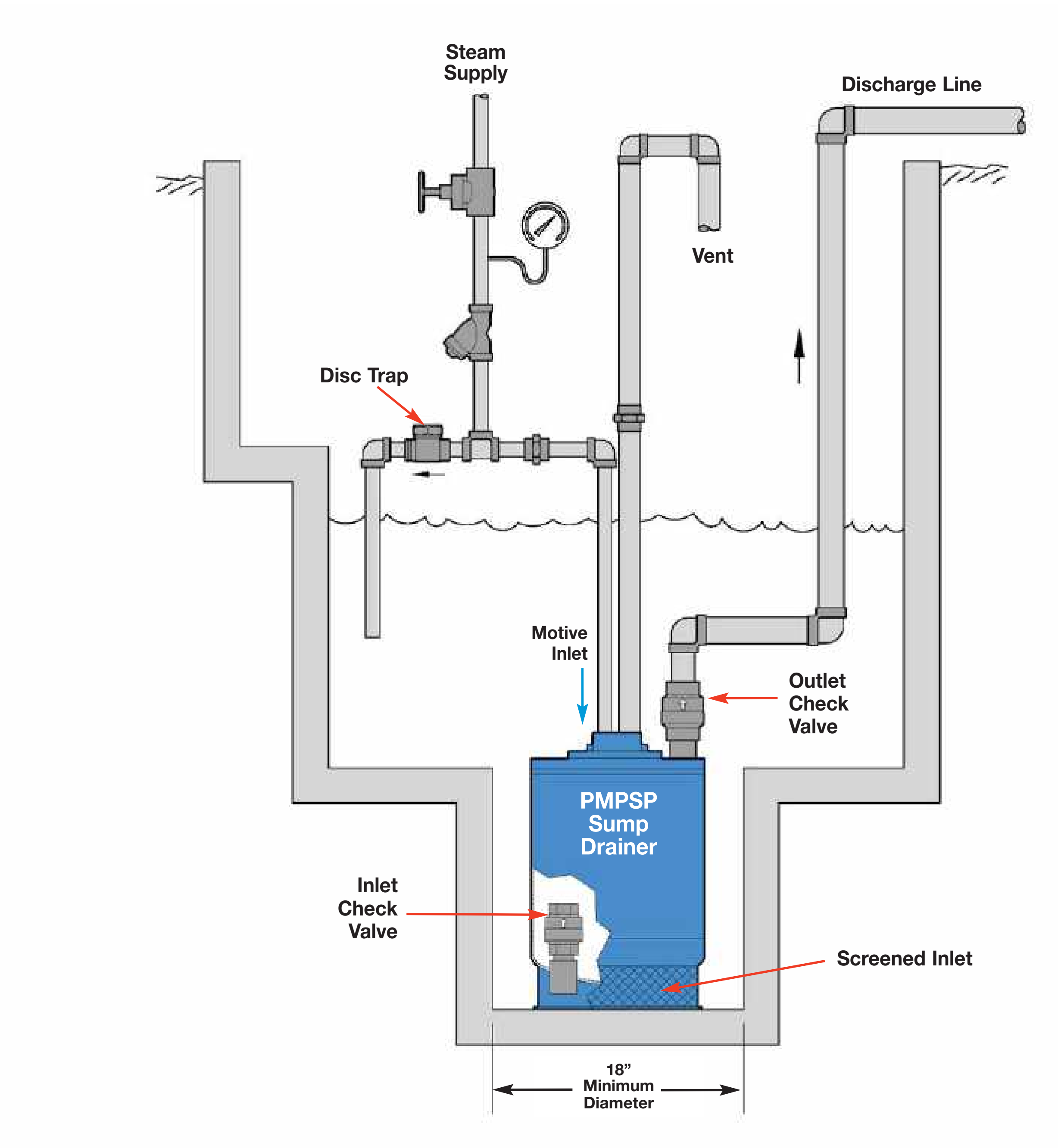
PURPOSE: For drainage of water and condensate from collection pits – especially with minimal horizontal space.

OPERATION: Water enters the inlet check valve through a screened area at the bottom of the PMPSP Sump Drainer. After the pump fills, the internal mechanism is actuated and the water is discharged from the pump by motive steam or compressed air or other gas.

INSTALLATION GUIDELINES: (see Figure 17)

- Make certain vent line is unobstructed and allowed to discharge directly to atmosphere.
- Other compressed gases, such as nitrogen, may be used as a motive source.
- Pit diameter should be at least 18” to ensure proper installation and operation.
- Proper installation and piping of the pump vent line is critical to ensure the system operates correctly. Follow guidelines or consult factory for additional information.
- Note that liquid level in the pit must rise above the pump to allow proper function.

Figure 17: Sump Drainer: "The Pit Boss"



PMPSP Sump Drainer ("The Pit Boss")

HEAT EXCHANGER FORMULAS & EXAMPLE

Formulas for Heat Exchanger System using a Modulating Control Valve

Definition of Terms and Units:

E = Mean Heat Transfer Rate or Heat Load (Btu/hr)

$\mathbf{E_D}$ = Design Heat Load (Btu/hr)

U = Overall Heat Transfer Coefficient (Btu/(hr-ft^2-°F))

A = Heat Transfer Surface Area of Heat Exchanger (ft^2)

$\mathbf{\Delta T_M}$ = Mean Temperature Difference between Steam and Water (°F)

$\mathbf{Q_W}$ = Volumetric Flow Rate of Water (GPM)

$\mathbf{Q_S}$ = Steam Load or Steam Capacity (lbs/hr)

$\mathbf{C_p}$ = Specific Heat Capacity of Water (Btu/(lb-°F))

$\mathbf{T_S}$ = Saturated Steam Temperature (°F)

$\mathbf{T_B}$ = Back pressure Equivalent Saturated Steam Temperature (°F)

$\mathbf{T_o}$ = Outlet Water Temperature (°F)

$\mathbf{T_i}$ = Inlet Water Temperature (°F)

$\mathbf{\Delta T_W}$ = Temperature Rise of Water (°F) = $\mathbf{T_o - T_i}$

$\mathbf{T_{WM}}$ = Mean Water Temperature (°F) = $\mathbf{(T_o + T_i)/2}$

LH = Latent Heat of Saturated Steam (Btu/lb)

$\mathbf{P_1}$ = Control Valve Inlet Pressure (PSIA)

$\mathbf{P_2}$ = Control Valve Outlet Pressure (PSIA)

$\mathbf{\Delta P}$ = Control Valve Differential Pressure (PSI) = $\mathbf{P_1 - P_2}$

$\mathbf{C_v}$ = Control Valve Flow Coefficient

Formula 1: Mean Heat Transfer Rate (E) of Heat Exchanger

$$E = U\,A\,\Delta T_M$$

The Heat Transfer Rate **E** (in Btu/hr) that takes place in a Heat Exchanger (HX) is a function of the Surface Area **A** (ft^2), the average temperature difference $\mathbf{\Delta T_M}$ (°F) between the steam and water, and the overall heat transfer coefficient **U**. The above formula can be used to calculate the heat loads for a HX based on the steam temperature inside the HX shell. This formula, when solved for **A**, can be used to size the HX (see **Formula 2**). Typical **U** values used for a steam to water HX range from 120 for stainless steel to over 200 for copper.

Formula 2: Heat Transfer Surface Area (A) of Heat Exchanger

$$A = \frac{E_D}{U\,\Delta T_M}$$

This formula is used to calculate the surface area (size) of the heat exchanger's internal tube or plates based on the design (maximum) heat load ($\mathbf{E_D}$) and average temperature difference ($\mathbf{\Delta T_M}$) between the steam and water. Since $\mathbf{\Delta T_M}$ is directly proportional to the steam pressure inside the HX shell, the specific steam pressure used to heat the water at $\mathbf{E_D}$ will determine the HX size. From the above formula, it can be seen that $\mathbf{\Delta T_M}$ is inversely proportional to **A** (the surface area). Therefore, the higher the steam pressure, the smaller the HX size, and vice versa.

Formula 3: Mean Temperature Difference (ΔT_M) between Steam and Water

$$\Delta T_M = \frac{(T_S - T_o) + (T_S - T_i)}{2}$$

This formula gives the average of the temperature differences between the steam and water at the outlet of the HX ($\mathbf{T_s - T_o}$) and at the inlet of the HX ($\mathbf{T_s - T_i}$).

Formula 4: Saturated Steam Temperature (T_S) as function of Mean Temperature Difference

$$T_s = \Delta T_M + T_{WM} \qquad \text{Where, } T_{WM} = (T_o + T_i)/2$$

This formula is derived by solving **Formula 3** for $\mathbf{T_S}$. It is useful for determining the steam temperature when the mean temperature difference ($\mathbf{\Delta T_M}$) is known. For example, the steam temperature at minimum load can be determined by solving **Formula 1** for $\mathbf{\Delta T_M}$ when $\mathbf{E = E_{min}}$, and then substituting $\mathbf{\Delta T_M}$ into the above formula. Once $\mathbf{T_S}$ is known, the pressure inside the HX shell can be determined from the Saturated Steam Table.

Formula 5: Heat Load (E)

$$E = Q_w \times 500 \times C_p \times \Delta T_w = Q_w \times 500 \times (T_o - T_i) \qquad [C_p = 1.0 \text{ Btu/(lb-°F)}]$$

The above formula shows that the heat load for the HX depends on the water flow rate ($\mathbf{Q_W}$) and the water temperature rise ($\mathbf{\Delta T_w = T_o - T_i}$).

ENGINEERING

Formulas for Heat Exchanger System using a Modulating Control Valve

Formula 6: **Steam Load (Q_S) as function of Heat Load**

$$Q_S = \frac{E}{LH}$$

The steam load or capacity (**Q_S** in lbs/hr) is dependent on the heat load (**E** in Btu/hr) and the latent heat (**LH** in Btu/lb) the steam contains. The Latent Heat of saturated steam is dependent on the steam pressure. Consult the Saturated Steam Table in Engineering Section. LH is typically approximated to 1,000 Btu/lb.

Formula 7: **Steam Load (Q_S) as function of Water Flow Rate**

$$Q_S = \frac{Q_W \times 500 \times (T_o - T_i)}{LH} \qquad Q_S = \frac{Q_W \times \Delta T_W}{2}$$ (approximation for LH = 1,000 Btu/lb)

This formula is derived by substituting the right side of **Formula 5** for **E** in **Formula 6**. It can be used for calculating the steam load directly from the flow rate of water to be heated.

Formula 8: **Water Flow Rate (Q_w) as function of Heat Load**

$$Q_W = \frac{E}{500 \times (T_o - T_i)}$$

This formula is derived by solving **Formula 5** for **Q_w**. It is useful for determining the water flow rate thru the HX at the stall point (**$Q_{w\text{-}stall}$**). This is explained in the following HX example (see part M).

Formula 9: **Percent Stall Load**

$$\% \text{ Stall Load} = \frac{T_B - T_{WM}}{T_S - T_{WM}} \times 100 \qquad \text{Where } T_{WM} = \frac{T_o + T_i}{2}$$

This formula is used to calculate the percentage of Full Heat Load (**E_D**) at which heat exchanger stall will occur. Since water flow rate is proportional to heat load (see **Formula 8**), the % Stall Load can be used to calculate the water flow rate at stall (see **Formula 10**).

Formula 10: **Water Flow Rate at Stall ($Q_{w\text{-}stall}$)**

$$Q_{w\text{-}stall} = Q_{w\text{-}full\ load} \times (\%\ \text{Stall Load})/100$$

Where, **$Q_{w\text{-}full\ load}$** = Water flow rate at design (maximum) heat load (**E_D**) = Maximum water flow rate

This formula is used in conjunction with **Formula 9** to calculate the water flow rate at which heat exchanger stall will occur without having to know the size of the HX.

Formula 11: **Control Valve Steam Capacity (Q_S) at Sub-Critical Flow**

For $\Delta P < 0.42\ P_1$:

11a: $$Q_S = 2.1\ C_v \sqrt{\Delta P (P_1 + P_2)}$$

11b: $$C_v = \frac{Q_S}{2.1 \sqrt{\Delta P (P_1 + P_2)}}$$

These formulas are applied when the pressure drop across the control valve (**ΔP**) is less than the critical pressure drop (**0.42** P_1).

Formula 12: **Control Valve Steam Capacity (Q_S) at Critical Flow**

For $\Delta P \geq 0.42\ P_1$:

12a: $$Q_S = 1.71\ C_v\ P_1$$

12b: $$C_v = \frac{Q_S}{1.71\ P_1}$$

When the pressure drop across the valve (**ΔP**) is greater than or equal to the critical pressure drop (**0.42 P_1**), the steam capacity (**Q_S**) depends only on the valve inlet pressure (**P_1**). The flow rate at this condition is called the critical flow. For a constant inlet pressure, the critical flow is the maximum capacity of the valve. The above formulas are derived from **Formula 11a** by using the critical pressure drop (**$\Delta P = 0.42\ P_1$**) and differential pressure (**$\Delta P = P_1 - P_2$**) formulas to eliminate **ΔP** and **P_2** from the equation.

Note: Formulas 11 and 12 are simplified versions of the steam flow equation.

ENGINEERING

Heat Exchanger Example:

Heating Water with Steam using a Modulating Control Valve

Basic overview of system:

A shell and tube heat exchanger (**HX**) is used to heat 100 GPM of water from 50°F to 140°F using saturated steam at 100 PSIG to the inlet side of the control valve. A modulating control valve, in conjunction with a temperature sensor and electronic controller, is used to regulate the flow of steam into the HX. At the design load of 100 GPM, the valve will supply the HX with 50 PSIG steam. At times of lower demand, the flow rate of water can be reduced to a minimum of 25 GPM. The HX is constructed with stainless steel and has an overall heat transfer coefficient of 120 Btu/(hr-ft^2-°F). The condensate produced from the condensing steam in the HX will drain thru a float-type steam trap located directly below the exchanger outlet and into a condensate return line with total back pressure of 10 PSIG.

OBJECTIVES: (see Figure 18)

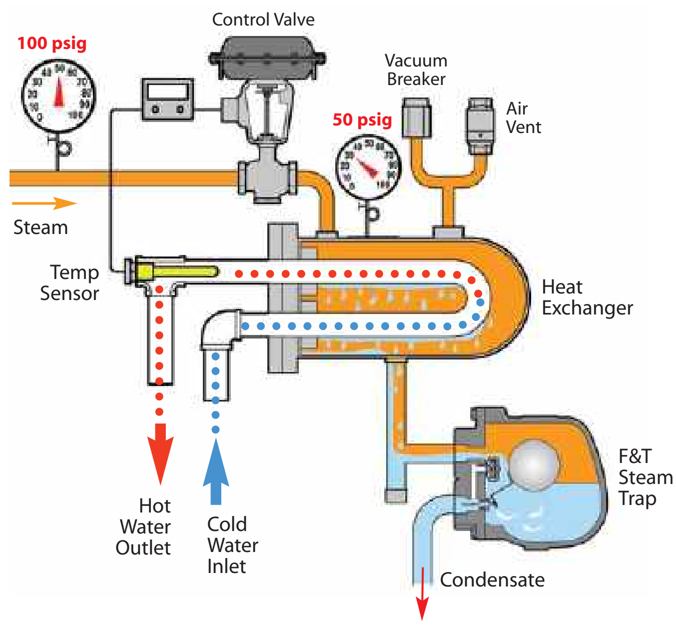
1) Select an appropriately sized **HX** that will effectively heat water from an estimated start temperature of 50°F to a final temperature of 140°F. The system must operate effectively in the flow rate range of 25 GPM to 100 GPM.

2) Select the appropriately sized **Control Valve** to effectively deliver steam to the HX.

3) Select the appropriately sized **Steam Trap** for draining condensate from the HX. The selection is based on steam pressure and condensate load in the HX.

4) Discuss advantages of using a **Pumping Trap** so the steam system can operate in vacuum during low demand, why a pumping trap may be a necessity if the condensate return line has back pressure or the condensate must be lifted after the HX, and how to select the proper size Pumping Trap.

Figure 18: **Heat Exchanger System with Steam Trap at Maximum Load**



HEAT EXCHANGER SIZING

The basic formula describing the heat transfer in a heat exchanger is $\mathbf{E = U \times A \times \Delta T_M}$, where **E**(Btu/hr) is the average heat transfer rate, **U** is the overall heat transfer coefficient, **A**(ft^2) is the heat transfer surface area (size) of the HX and $\mathbf{\Delta T_M}$ is the average temperature difference between the steam and water being heated.

A) What is the design heat load (E_D) for this application?

The first step in sizing a HX is to calculate the maximum heat load (Btu/hr) required to heat the water. The design heat load (**E$_D$**) of the heat exchanger is the maximum heat load that needs to be transferred by the steam to the water based on the given conditions. The maximum heat load occurs at the maximum water flow, which is 100 GPM. Using ***Formula 5***:

$$\mathbf{E_D = Q_w \times 500 \times C_p \times \Delta T_w}$$

$$= 100 \text{ GPM} \times \frac{500 \text{ lbs/hr}}{1 \text{ GPM}} \times 1.0 \text{ Btu/(lb-°F)} \times (140\text{°F} - 50\text{°F})$$

$$= 50{,}000 \text{ lbs/hr} \times 1.0 \text{ Btu/(lb-°F)} \times 90\text{°F}$$

$$= \mathbf{4{,}500{,}000 \text{ Btu/hr}}$$

Heat Exchanger Example:

Heating Water with Steam using a Modulating Control Valve

B) What is the mean temperature difference (ΔT_M) between the steam and the water being heated?

From the HX formula we can see that in order to determine the size of the HX required to heat the water, we must first know the steam temperature (which is directly related to steam pressure) in the HX during the period of maximum demand. The steam pressure in the HX is dependent on the pressure drop across the control valve. **For optimal control in heating applications, it is typical to target a 50% pressure drop across the valve at the maximum steam load.** Therefore, at full load, the pressure drop across the control valve is 50 PSIG and the steam pressure supplied to the heat exchanger is also 50 PSIG.

As the steam (primary fluid) passes thru the heat exchanger, it transfers its latent heat energy to the water (secondary fluid) and condenses without a change in temperature. Therefore, the condensate leaving the heat exchanger is at the same temperature as the steam entering. From the saturated steam table, the steam temperature (T_S) of 50 PSIG saturated steam is 298°F. The water inlet temperature (T_i) is 50°F and the water outlet temperature (T_o) is 140°F.

We now have enough information to calculate the mean temperature difference between the steam (primary fluid) and water (secondary fluid). ***Formula 3*** is used to calculate the mean temperature difference (ΔT_M) which is the average of the temperature differences at both ends of the HX:

$$\Delta T_M = \frac{(T_S - T_o) + (T_S - T_i)}{2} = \frac{(298 - 140) + (298 - 50)}{2} = \frac{158 + 248}{2} = \frac{406}{2} = \mathbf{203°F}$$

C) What is the Overall heat transfer coefficient (U) of the heat exchanger?

The U value of the HX depends on several factors, including type of HX, the quality of the steam used, if any fouling is expected, if the flow of water is turbulent or laminar, and the material of construction. The higher the U value, the better the heat transfer, and the smaller the HX needs to be. Typical U values range from 120 for a stainless steel HX to over 200 for copper. For this example, a Stainless Steel HX was selected for longevity purposes and so a U value of 120 will be used to determine the HX size.

D) What is the minimum heat transfer surface area (A) of the heat exchanger that can meet the design heat load?

The size of a HX is dependent on the steam pressure inside its shell. The higher the steam pressure, the smaller the HX for a given heat load. 50 PSIG was chosen because the supply pressure is 100 PSIG and this gives a 50% pressure drop across the control valve, as previously discussed. If a lower steam pressure is used, this would require a larger HX, and vice versa.

In a heat exchanger, the mean heat transfer rate is proportional to the mean temperature difference between the two fluids, as given by ***Formula 1***. Rearranging this equation gives ***Formula 2***, where **E** has been replaced by E_D, the design heat load. Using ***Formula 2*** and the mean temperature difference determined above, gives the heat transfer surface area:

$$A = \frac{E_D}{U\,\Delta T_M} = \frac{4{,}500{,}000 \text{ Btu/hr}}{120 \text{ Btu/(hr-ft}^2\text{-°F)} \times 203°\text{F}} = \mathbf{185\ ft^2}$$

Therefore, for a perfectly sized heat exchanger, the heat transfer area of the tube is 185 square feet. In practice, the heat exchanger is usually oversized by at least 15% to account for fouling of the heating surfaces over time or to allow for an increase in the maximum heat load.

Heat Exchanger Example:

Heating Water with Steam using a Modulating Control Valve

CONTROL VALVE SIZING

E) What is the flow of steam (steam capacity) thru the control valve at the design heat load?

Formula 6 gives the mass flow rate of steam based on the heat load and the latent heat of saturated steam (LH). From the saturated steam table, the latent heat of 50 PSIG steam is 912 Btu/lb. Therefore, the steam capacity is:

$\mathbf{Q_S = E_D / LH}$ = 4,500,000 Btu/hr / 912 Btu/lb = **4,934 lbs/hr**

F) How must the control valve be sized?

The valve must be sized for the maximum steam capacity of the application, which occurs at the maximum (design) heat load of the heat exchanger. We first need to determine if the pressure drop across the valve at the maximum flow rate is above or below the critical pressure drop, so that we can apply the correct formula:

Valve Inlet Pressure (P_1) = Steam Supply Pressure = 100 PSIG + 14.7 = **114.7 PSIA**

Valve Outlet Pressure (P_2) = Heat Exchanger Pressure = 50 PSIG + 14.7 = **64.7 PSIA**

Differential Pressure (ΔP) = $P_1 - P_2$ = 114.7 – 64.7 = **50 PSI**

Critical Pressure Drop ($\Delta P_{critical}$) = 0.42 P_1 = 0.42 (114.7) = **48.2 PSI**

Since the pressure drop across the valve (50 PSI) is greater than the critical pressure drop (48.2 PSI), the steam flow thru the valve is critical. Therefore, we apply ***Formula 12b*** to calculate the valve coefficient:

$\mathbf{C_v = Q_S / (1.71 \times P_1)}$ = 4,934 / (1.71 x 114.7) = 4,934 / 196.1 = 25.2

Therefore, the control valve must have a flow coefficient of at least 26.

G) What Watson-McDaniel Control Valve should be selected for this application?

Refer to the Control Valves section of this catalog. The Watson McDaniel **HB-Series 2-Way Pneumatic Control Valve** is used for heating and cooling applications. Since this is a heating application with steam, a Normally-Closed, Air-To-Open (ATO) valve should be selected. (This is a fail-safe feature in case the air signal to the valve actuator becomes interrupted. If the air signal is lost, the valve will automatically close and block the flow of steam.)

Referring to the HB Control Valve Selection chart, a full-port valve with Cv value of 42 should be selected. The Model Number for this valve with NPT ports is **HB-17-N-ATO**. This valve has a 2” NPT connection size, stainless steel body and trim, and a pressure-temperature rating of 300 PSIG @ 450°F.

Heat Exchanger Example:
Heating Water with Steam using a Modulating Control Valve

H) What is the maximum close-off pressure of the control valve selected in Part G?

From the HB Control Valve Selection chart, the maximum close-off pressure for the selected valve is 85 PSI ΔP if no positioner is used, and 135 PSI ΔP if a positioner is used.

I) For the selected control valve, is a positioner required to completely shut off the flow of steam to the heat exchanger?

When the control valve is completely closed, the pressure drop across the valve is at its maximum value:

$$\begin{aligned} \Delta P_{MAX} &= \text{Steam Supply Pressure} - \text{Heat Exchanger Pressure} \\ &= 100 \text{ PSIG} - 0 \text{ PSIG} \\ &= 100 \text{ PSI} \end{aligned}$$

Thus, the control valve must have a close-off pressure capability of at least 100 PSI. Without a positioner, the maximum close-off pressure of the valve is 85 PSI. Therefore, a valve positioner is necessary to provide the required closing force to the actuator diaphragm.

In a normally-closed valve, the valve is held closed by a spring force. The spring pressure is set so that the valve will stay closed against an inlet pressure of 85 PSIG. The opening action is performed by the 3-15 PSIG air signal to the actuator diaphragm. When the air signal is 15 PSIG, the valve will completely open against the spring pressure. When the air signal is 3 PSIG, the valve will stay closed provided that the inlet pressure does not exceed 85 PSIG. If the inlet pressure exceeds 85 PSIG, the valve will open and a positioner will then be required to decrease the air signal pressure below 3 PSIG to allow the valve to fully close.

HEAT EXCHANGER FORMULAS & EXAMPLE

Heat Exchanger Example:

Heating Water with Steam using a Modulating Control Valve

STEAM TRAP SIZING

J) What type and size of steam trap should be chosen for this application?

Initially we will assume that the condensate from the HX is being discharged to a condensate return line at atmospheric pressure (0 PSIG). At full-load, the steam pressure at the inlet of the trap is 50 PSIG. We have already calculated the maximum condensate load that would be generated at this pressure to be around 5,000 lbs/hr (see Part E). If this was the only set of conditions, selection of the steam trap would be fairly simple. We simply look at the capacity chart and choose a steam trap that will pass at least 5,000 lbs/hr at 50 PSI differential pressure. (See the capacity chart below for the FTE Series Float & Thermostatic steam traps.)

However, at different flow rates of water, the HX will have very different pressures. At a flow rate of ~58 GPM, the pressure in the HX drops to 0 PSIG and the HX is still producing condensate at the rate of ~2,700 lbs/hr. This condensate still needs to be drained from the HX but how can this happen with NO differential pressure? The purpose of having an extended drip leg under the HX is to give the trap a certain amount of head pressure. If the trap is mounted 14 inches below the HX, this will correspond to ½ PSI of head pressure. The trap must then be able to pass at least 2,700 lbs/hr of condensate at ½ PSI ΔP.

CAPACITIES – *Condensate (lbs/hr)*

				Differential Pressure (PSI)															
Model Code	PMO (PSIG)	Pipe Size	Orifice Size	1/4	1/2	1	2	5	10	15	20	30	50	75	100	125	200	250	300
FTE-20-17-N*	20	2″	.937″	6100	7800	9300	11800	15900	19500	22500	26000								
FTE-50-17-N	50	2″	2.125″	12800	16900	20100	25300	33000	40200	43500	46000	47800	52500						
FTE-50-18-N	50	2½″	2.125″	20400	25700	31000	37000	46300	55100	60300	65100	72000	82100						
FTE-125-18-N	125	2½″	2.125″	20400	25700	31000	37000	46300	55100	60300	65100	72000	82100	90400	97700	105000			
FTE-200-16-N	200	1½″	.375″	950	1350	1900	2200	2700	3300	3900	4400	5300	6400	7600	8500	9400	11900		
FTE-200-17-N	200	2″	.75″	2700	4100	5700	7400	9900	11800	13400	14400	16400	19000	21500	23000	24500	29200		
FTE-200-18-N	200	2½″	1.5″	7200	12300	17400	21500	27600	32600	36000	39300	43100	49200	54700	58800	61900	74000		
FTES-50-18-N	50	2½″	2.125″	20400	25700	31000	37000	46300	55100	60300	65100	72000	82100						
FTES-125-18-N	125	2½″	2.125″	20400	25700	31000	37000	46300	55100	60300	65100	72000	82100	90400	97700	105000			
FTES-300-18-N	300	2½″	1.5″	7200	12300	17400	21500	27600	32600	36000	39300	43100	49200	54700	58800	61900	74000	86000	100550

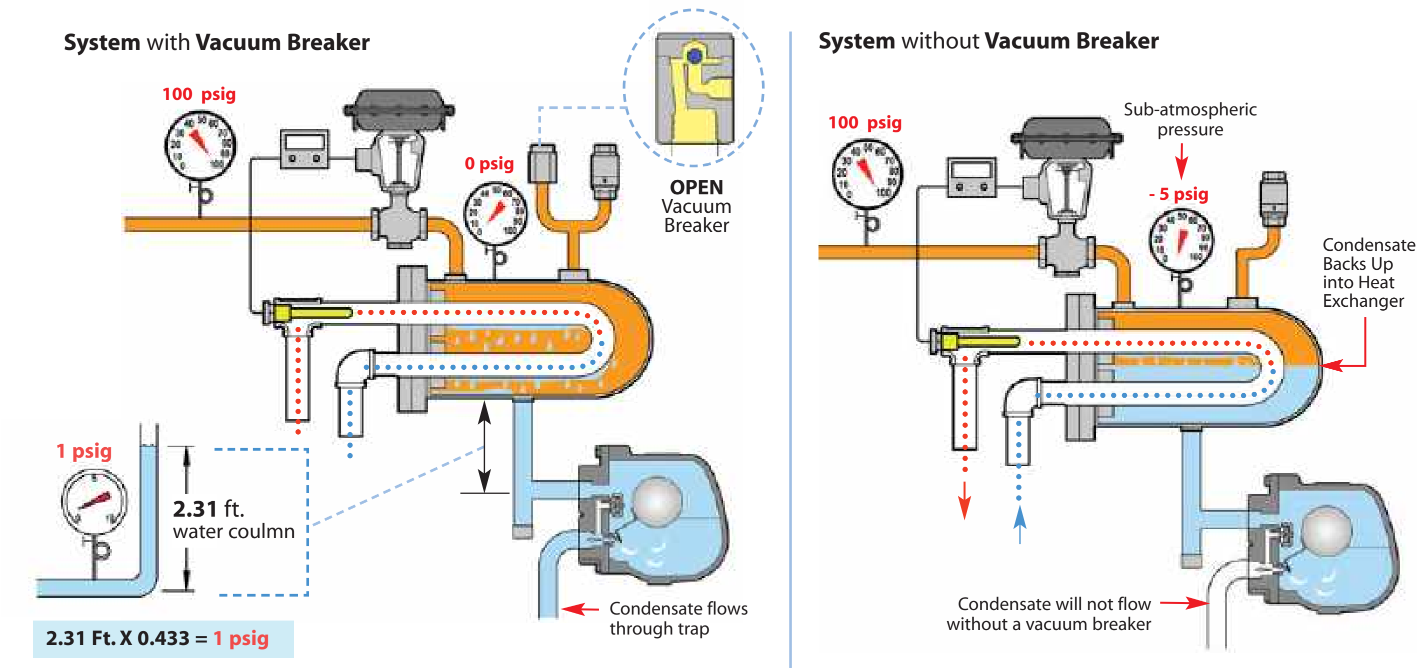
** Single seat orifice. All others are double seated.*

As a general rule, for HX applications using steam pressures over 30 PSIG, the steam trap should be sized for 2.5X maximum condensate load at full differential pressure. Therefore, to provide an appropriate safety margin, we must select a trap that can pass 2.5 x 5,000 = 12,500 lbs/hr of condensate at 50 PSI ΔP. In addition, the steam trap must be able to handle the maximum possible inlet pressure which is 100 PSIG (the steam supply pressure). Referring to the FTE capacity chart above; the best trap to select is the **FTE-200-17-N**. This trap can pass 19,000 lbs/hr at 50 PSI ΔP and 4,100 lbs/hr at ½ PSI ΔP, which meets the above criteria.

Heat Exchanger Example:
Heating Water with Steam using a Modulating Control Valve

Now, what happens when the water flow rate thru the HX reduces to the point that the steam pressure goes into vacuum? This occurs at water flow rates below about 58 GPM down to the minimum of 25 GPM. Since the HX will be operating in vacuum, the condensate would not effectively drain regardless of the steam trap size chosen. This is why a vacuum breaker must be added to the top of the HX. The vacuum breaker draws in air to neutralize the pressure in the HX which allows the condensate to drain (see Figure 19).

Figure 19: **Heat Exchanger System with Steam Trap shown with and without Vacuum Breaker**



The HX can be properly drained using a steam trap as long as there is no BACK PRESSURE in the condensate return line. If there is back pressure in the condensate return line, a Pumping Trap must be used.

Heat Exchanger Example:
Heating Water with Steam using a Modulating Control Valve

PUMP-TRAP SIZING

K) If the condensate return line has a total back pressure of 10 PSIG, can a steam trap be used to drain the heat exchanger?

At full-load conditions, the steam pressure is 50 PSIG and the condensate load is ~5,000 lbs/hr. Since the total back pressure of the return line is 10 PSIG, the differential pressure across the steam trap is 40 PSI. An appropriately sized steam trap can handle this situation. However, when the steam pressure is reduced to 10 PSIG or lower, due to lower heat demand, the differential pressure across the steam trap will be 0 PSIG or less. Without positive differential pressure across the trap, the condensate cannot drain from the HX. During this situation, the condensate will back up into the HX shell. Therefore, a steam trap will not be effective in discharging condensate from the HX under **all** conditions.

L) If the water flow rate is reduced, less heat energy per unit of time is needed to heat the water and therefore the heat load will also reduce. This will cause a reduction in steam flow and pressure in the heat exchanger. If the steam pressure falls to or below the system back pressure, the condensate will begin to back up into the heat exchanger, causing the system to stall. Why is it important to prevent stall from occurring?

Condensate flooding the heat exchanger space will cause poor temperature control, accelerated corrosion and potentially damaging waterhammer. These factors can cause rapid or premature failure of the unit, leading to costly repairs and downtime.

M) For the heat exchanger size (surface area) calculated in Part D, what is the flow rate of water at which stall will occur?

We will use two methods to calculate the water flow rate at stall and then compare the two methods.

Method 1: Based on Heat Exchanger Size

Stall occurs at the point where the steam pressure equals the back pressure. The steam pressure at stall is therefore 10 PSIG. From the saturated steam table, this is equivalent to a steam temperature (**Ts**) of 239°F. ***Formula 3*** can now be used to calculate the mean temperature difference between the steam and water:

$$\Delta T_M = \frac{(T_S - T_o) + (T_S - T_i)}{2} = \frac{(239 - 140) + (239 - 50)}{2} = \frac{99 + 189}{2} = \frac{288}{2} = 144°F$$

The heat load at stall is then calculated from ***Formula 1***:

$\mathbf{E_{stall} = U\ A\ \Delta T_M}$ = 120 Btu/(hr-ft^2-°F) x 185 ft^2 x 144°F = 3,196,800 Btu/hr

Finally, the volumetric flow rate of water at stall is calculated from ***Formula 8:***

$$\mathbf{Q_{w\text{-}stall} = E_{stall} / [500 \times (T_o - T_i)]}$$

$$= \frac{3{,}196{,}800}{[500 \times (140 - 50)]} = \frac{3{,}196{,}800}{(500 \times 90)} = \frac{3{,}196{,}800}{45{,}000}$$

= **71 GPM**

Method 2: Based on % Stall Load Formula

T_S = Steam temperature at full-load = 298°F (50 PSIG steam)

T_B = Back pressure equivalent saturated steam temperature = 239°F (10 PSIG steam)

T_{WM} = Mean water temperature = $\frac{T_o - T_i}{2} = \frac{140 + 50}{2} =$ 95°F

Using ***Formula 9***:

$$\textbf{\% Stall Load} = \frac{T_B - T_{WM}}{T_S - T_{WM}} \times 100 = \frac{239 - 95}{298 - 95} \times 100 = .71 \times 100 = 71\%$$

The water flow rate at stall is then calculated using ***Formula 10***:

$\mathbf{Q_{stall} = Q_{w\text{-}full\ load} \times (\%\ Stall\ Load)/100}$ = 100 GPM x 71/100 = **71 GPM**

Heat Exchanger Example:
Heating Water with Steam using a Modulating Control Valve

Comparison of Methods:

Both methods gave the same result for the water flow rate at which stall will occur: 71 GPM. This means that at flows at or below 71 GPM, the steam pressure in the system is insufficient to push the condensate thru the steam trap and into the return line. The condensate will therefore back up into the heat exchanger unless something is done to prevent it.

The main difference between the two methods is that the heat exchanger size was needed to calculate the stall flow rate using **Method 1**, but not needed using **Method 2**.

N) How can stall (condensate back-up) be prevented?

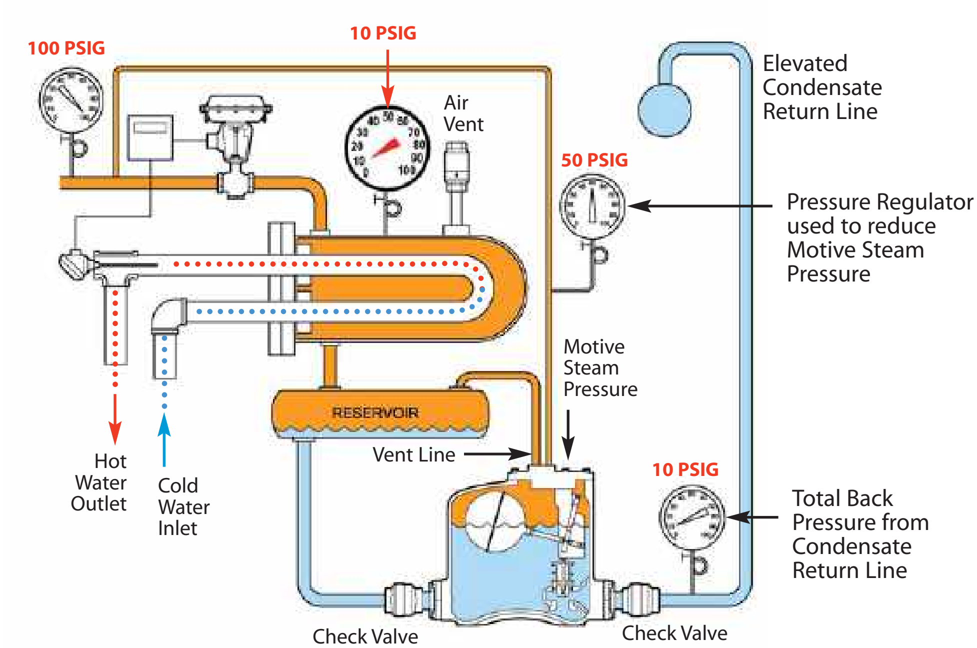
Stall can be prevented by replacing the steam trap with a pump-trap (i.e., a pressure motive pump and steam trap combination). Pump-traps are available with either internal or external steam traps, depending on capacity requirements.

When there is sufficient steam pressure to push the condensate thru the trap, the pump is not used and the pump-trap is operating in **trap mode**. The condensate will pass thru the pump body and thru the trap. The trap must be sized to handle the condensate at full-load conditions as well as when the trap differential pressure is slightly above 0 PSI (i.e., just above the stall point). In addition, the orifice size of the trap should be optimized to handle the high instantaneous discharge flow rate when the pump is operating. This will reduce the discharge time of the pump and its overall fill/discharge cycle. Watson McDaniel pump-traps have the trap size optimized for all conditions.

When the steam pressure drops to or below the back pressure, the condensate will start to fill the pump tank. When the float in the tank reaches the upper trip point, the mechanism will open the steam valve while simultaneously closing the vent valve. High pressure steam will then force the condensate thru the trap and into the condensate return line. Check valves are used with the pump to prevent the backflow of condensate. When the pump is emptied, the float mechanism will then simultaneously close the steam valve and open the vent valve so the pump can fill on the next cycle. When the pump is being used, the pump-trap is operating in **pump mode**. The pump must be sized to handle the condensate load at the stall point. That is, when the steam pressure is equal to the back pressure.

When sizing Pressure Motive Pumps in closed-loop return systems, a condensate **reservoir** should be installed on the inlet side of the pump and below the HX, as shown in Figure 20. This will enable the condensate to collect while the pump is in the discharge cycle, thus preventing liquid backup into the HX. The reservoir should be located 12" above the top of the pump tank to provide adequate filling head. The reservoir must have sufficient size (volume) to provide adequate vapor space for the condensate to collect during the pump's discharge cycle and for the pump to vent during its filling cycle. The vent line also acts as a balancing line to allow condensate to drain into the pump tank while the HX is operating in vacuum.

Figure 20: Heat Exchanger System with Pump-Trap at Stall Load



Heat Exchanger Example:
Heating Water with Steam using a Modulating Control Valve

O) If a pump-trap is used to prevent stall, what capacity must the pump have?

The maximum condensate load that the pump must discharge occurs at the stall point (i.e., when the steam pressure is equal to the total back pressure of the condensate return line). This can be determined from the steam load at the stall point, using ***Formula 6***. The heat load at stall was determined in Part M to be 3,196,800 Btu/hr. The steam temperature at stall was also determined in Part M to be 239˚F. From the steam table, the latent heat of steam at 239˚F is 953 Btu/lb. The steam capacity is:

$\mathbf{Q_S = E_{stall} / LH}$ = 3,196,800 Btu/hr / 953 Btu/lb = **3,354 lbs/hr**

The maximum condensate load at stall conditions is therefore 3,354 lbs/hr and the pump must be sized to remove condensate at this rate.

P) What Watson-McDaniel Pump-Trap should be selected for this application?

Referring to the pump-trap capacity chart when operating in **Pump Mode**, it can be seen that model **WPT3** (pump with external trap mounted on common base) can meet the condensate load at stall (3,354 lbs/hr) when the motive steam pressure is 50 PSIG and the total back pressure is 10 PSIG. Under these conditions, this model has a maximum capacity of 4,080 lbs/hr. Since the steam supply pressure is 100 PSIG, a pressure regulator can be used to reduce the pressure to 50 PSIG for the motive steam line.

MINIMUM LOAD & OPERATION IN VACUUM

Q) What is the minimum heat load of the application?

The minimum heat load occurs at the minimum water flow of 25 GPM. Using ***Formula 5***:

$$\mathbf{E_{min} = Q_w \times 500 \times C_p \times \Delta T_w}$$
$$= 25 \text{ GPM} \times \frac{500 \text{ lbs/hr}}{1 \text{ GPM}} \times 1.0 \text{ Btu/(lb-°F)} \times (140°\text{F} - 50°\text{F})$$
$$= 12{,}500 \text{ lbs/hr} \times 1.0 \text{ Btu/(lb-°F)} \times 90°\text{F}$$
$$= \mathbf{1{,}125{,}000 \text{ Btu/hr}}$$

R) What is the steam temperature in the heat exchanger at the minimum load?

Use ***Formula 1*** to calculate the mean temperature difference between the steam and water:

$\mathbf{\Delta T_M = E_{min} / (U\ A)}$ = 1,125,000 Btu/hr / (120 Btu/(hr-ft^2-°F) x 185 ft^2) = 50.7°F

The steam temperature is then given by ***Formula 4***:

$$\mathbf{T_S} = \mathbf{\Delta T_M + T_{WM}} = \mathbf{\Delta T_M + \frac{T_o + T_i}{2}} = 50.7 + \frac{140 + 50}{2} = 50.7 + \frac{190}{2} = 50.7 + 95 = \mathbf{146°F}$$

S) What is the steam pressure in the heat exchanger at the minimum load?

From the steam table (using extrapolation), the steam pressure corresponding to 146°F saturated steam is 22.7 in Hg Vacuum which is equivalent to -11.1 PSIG. Therefore, the steam pressure inside the heat exchanger is below atmospheric pressure.

This is another advantage in the use of a pump-trap. If a steam trap is used to drain condensate, the system could not operate in vacuum since the condensate would never drain out. Therefore, a vacuum breaker is used which essentially mixes the steam with air to achieve the proper temperature differential for a particular size HX. This added air facilitates corrosion by forming carbonic acids. Some of this air is eventually discharged thru the air vent on top of the HX but some mixes with the condensate. A pump-trap can discharge condensate when the HX is operating in vacuum, which precludes the use of a vacuum breaker and thus air is prevented from entering the system.

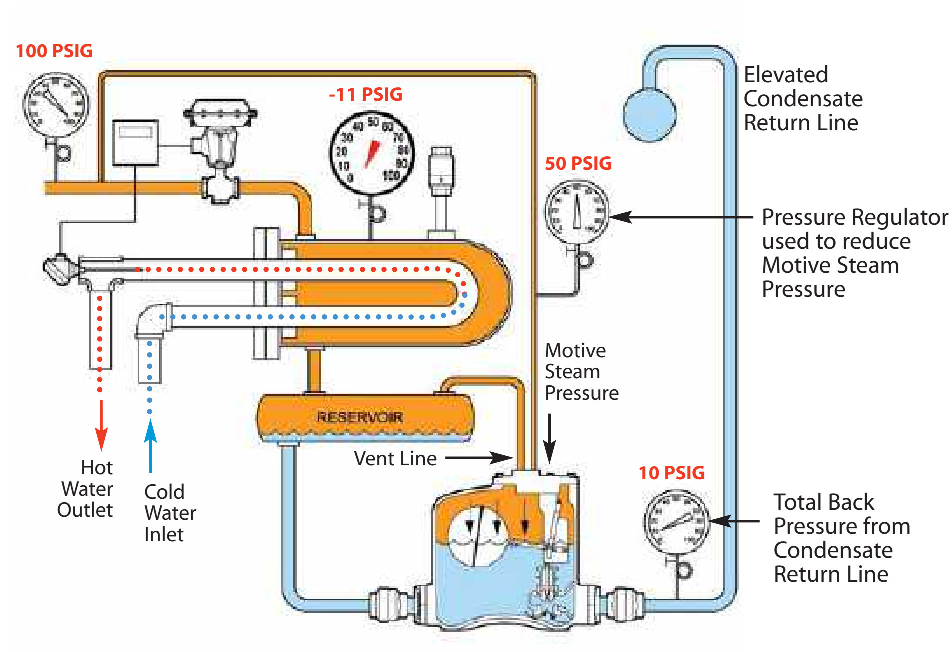
Heat Exchanger Example:
Heating Water with Steam using a Modulating Control Valve

T) What is the flow of steam (steam capacity) thru the control valve at the minimum load?

From the steam table (using extrapolation), the latent heat of steam (LH) corresponding to 146°F saturated steam is 1,011 Btu/lb. Using ***Formula 6***, the steam capacity is:

$Q_S = E_{min} / LH$ = 1,125,000 Btu/hr / 1,011 Btu/lb = **1,113 lbs/hr**

Figure 21: **Heat Exchanger System with Pump-Trap at Minimum Load**



SUMMARY of HEAT EXCHANGER SYSTEM

The following table summarizes the above results and shows how the heat load and the pressure, temperature, latent heat and flow of steam vary as a function of the water flow rate. It can be seen that the system is operating in **Trap Mode** between water flow rates of 100 and 71 GPM, and in **Pump Mode** between 71 and 25 GPM. Also, at flow rates below ~58 GPM, the steam pressure inside the HX is below atmospheric pressure (0 PSIG).

Flow Rate Water (GPM)	Heat Load (Btu/hr)	Steam Usage (lbs/hr)	Steam Pressure in HX (PSIG)	Steam Temp in HX (°F)	Latent Heat of Steam (Btu/lb)	Condensate Generated (lbs/hr)	Trap Differential Pressure (PSI)	System Condition	
100	**4,500,000**	**4,934**	**50**	**298**	**912**	**4,934**	**40**	Trap Mode	**(Maximum Heat Load)**
94.7	4,262,400	4,633	40	287	920	4,633	30		
88.3	3,973,800	4,278	30	274	929	4,278	20		
80.9	3,640,800	3,873	20	259	940	3,873	10		
71.0	**3,196,800**	**3,354**	**10**	**239**	**953**	**3,354**	**0**	(Stall Point)	**Steam Pressure = Back Pressure**
57.7	2,597,400	2,678	0	212	970	2,678	---	**Pump Mode**	
47.9	2,153,400	2,191	-5	192	983	2,191	---	(Vacuum)	
25	**1,125,000**	**1,113**	**-11**	**146**	**1,011**	**1,113**	---		**(Minimum Heat Load)**

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