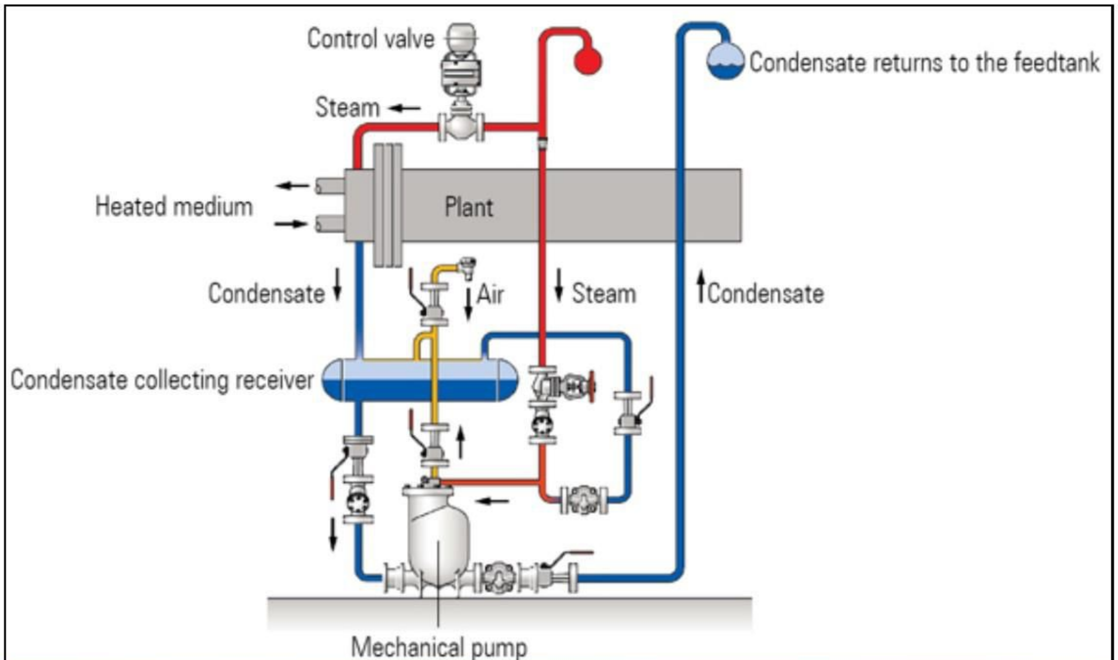

CALCULATING STEAM FLOWS

AND

SIZING EQUIPMENT

CONTROL SPECIALTIES, INC.

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Calculating Steam Flows and Sizing Equipment

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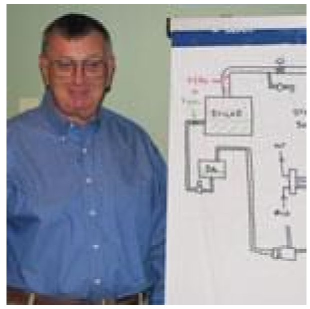
BIOGRAPHY OF EDWARD MOSCHETTI

Edward brings over 50 years of knowledge and experience to the manufacturing, medical, university, and commercial arenas. His education background includes a degree in physics from Muhlenberg College. The last 30 years have been devoted to providing solutions to industrial facilities geared around plant utilities and processes. He has spearheaded energy teams in major corporations resulting in the savings of energy dollars as well as building/designing small to medium size boiler facilities.



In two of our prior steam related eBooks, we covered boiler plant operations and steam distribution and the primary equipment in a steam system. In this eBook, our primary focus is on utilizing steam and providing you with the tools necessary to size and select the proper equipment for your steam system. In our first three chapters, we discuss energy and how it relates to the BTU required to manufacture products, heat buildings, generate electricity, etc. Once you have a clear understanding as to how we put steam to work, we will take the next nine chapters to address the important topic of calculating steam flows and sizing and selecting critical steam using equipment. A steam system in total is a large investment so it only makes sense to operate a system for the very best return. We hope the next twelve chapters will assist you in making the correct equipment selections, thus making your system operate at its optimum!

Chapter 1 –Energy and the BTU

Energy: "The capacity or power to do work such as the capacity to move an object (of a given mass), by the application of force. **Energy** can exist in a variety of forms, such as electrical, mechanical, chemical, thermal, or nuclear, and can be transformed from one form to another."

http://dictionary.reference.com/browse/energy

Without energy we could not exist. If you go back in history, civilization has advanced our understanding to harness energy to improve our lives. From the early discovery of fire to provide warmth, light, and self-protection; to the ability to transform raw materials into useful food sources to current day multiple energy sources.

Energy is illusive. You cannot go to an energy store to buy a box of BTU's, watts, or horsepower. The quest to control energy sources has led to wars and many other upsetting events. The current hot topic is climate change, and its connection with a fossil (carbon) based set of fuel sources. Our sun produces light energy from nuclear fusion. In early earth history, plants took sunlight energy and through photosynthesis produced plant matter. Over eons of time, the plant matter decayed and changes in the structure of the earth buried much of the matter to produce flammable materials such as coal, oil, natural gas and similar materials. If you ponder just a moment, the fossil carbon based fuels created by sunlight so many eons ago provides the energy sources to power our current standards of living. Only time, rational thinking, and lots of innovation will lead us to supply the energy needs of the world in the future.

Steam provides a very convenient method to transport large quantities of energy for final uses to produce food, warmth, products, power transportation, generate electric power and so many other uses.

In our previous EBooks, Boiler Plant Operations and Tips and Steam Distribution and Primary Equipment, we covered the production and distribution of steam to do useful work. For this work process to be productive, we need clean dry steam. Process problems, in many cases, track back to issues in boiler operations or the steam distribution system. As we proceed with this EBook, we will assume that you have clean dry steam for your steam consuming applications requiring thermal energy.

To utilize energy we need to be able to understand its properties, quantify them, and then put them to work. Thermal energy (heat) is hard to measure other than something is hot or cold. The measurement decided on was the British Thermal Unit (BTU), which was defined as the amount of energy to raise the temperature of one pound of water one degree Fahrenheit. If you wish to be a home scientist, then measure one pound of water into a sauce pan, put a thermometer in the pan of water, turn on the heat and when the temperatures rises by one degree, the energy content of the water increases by one BTU. In the reverse process, if the water temperature were to drop one degree Fahrenheit by placing a bit of ice in the water, then one BTU of work was performed by melting a bit of ice. With this bit of kitchen science, we have created a heat transfer media available worldwide, at no cost, and predictable in its properties which we call steam - water in vapor form.

Energy Source	Unit	Energy Content (*Btu*)
Electricity	1 Kilowatt-hour	3412
Butane	1 Cubic Foot (cu.ft.)	3200
Coal	1 Ton	28000000
Crude Oil	1 Barrel - 42 gallons	5800000
Fuel Oil no.1	1 Gallon	137400
Fuel Oil no.2	1 Gallon	139600
Fuel Oil no.3	1 Gallon	141800
Fuel Oil no.4	1 Gallon	145100
Fuel Oil no.5	1 Gallon	148800
Fuel Oil no.6	1 Gallon	152400
Diesel Fuel	1 Gallon	139000
Gasoline	1 Gallon	124000
Natural Gas	1 Cubic Foot (cu.ft.)	950 - 1150
Heating Oil	1 Gallon	139000
Kerosene	1 Gallon	135000
Pellets	1 Ton	16500000
Propane LPG	1 Gallon	91330
Propane gas $60^{o}F$	1 Cubic Foot (cu.ft.)	2550
Residual Fuel Oil [1)]	1 Barrel - 42 gallons	6287000
Wood - air dried	1 Cord	20000000
Wood - air dried	1 pound	8000

http://www.engineeringtoolbox.com/energy-content-d_868.html

For reference purposes from our EBook Boiler Plant Operations a quick review of the steam tables –

Gauge Pressure (psig)	Temperature (^oF)	Specific Volume Saturated Vapor (ft^3/lb)	Enthalpy		
			Saturated Liquid (Btu/lb)	Evaporated (Btu/lb)	Saturated Vap (Btu/lb)
25 (Inches Mercury Vacuum)	134	142	102	1017	1119
20 (Inches Mercury Vacuum)	162	73.9	129	1001	1130
15 (Inches Mercury Vacuum)	179	51.3	147	990	1137
10 (Inches Mercury Vacuum)	192	39.4	160	982	1142
5 (Inches Mercury Vacuum)	203	31.8	171	976	1147
0 [1)]	212	26.8	180	970	1150
1	215	25.2	183	968	1151
2	219	23.5	187	966	1153
3	222	22.3	190	964	1154
4	224	21.4	192	962	1154
5	227	20.1	195	960	1155
6	230	19.4	198	959	1157
7	232	18.7	200	957	1157
8	233	18.4	201	956	1157
9	237	17.1	205	954	1159
10	239	16.5	207	953	1160
12	244	15.3	212	949	1161
14	248	14.3	216	947	1163
16	252	13.4	220	944	1164
18	256	12.6	224	941	1165
20	259	11.9	227	939	1166
22	262	11.3	230	937	1167
24	265	10.8	233	934	1167
26	268	10.3	236	933	1169
28	271	9.85	239	930	1169
30	274	9.46	243	929	1172
32	277	9.1	246	927	1173
34	279	8.75	248	925	1173
36	282	8.42	251	923	1174
38	284	8.08	253	922	1175
40	286	7.82	256	920	1176
42	289	7.57	258	918	1176
44	291	7.31	260	917	1177
46	293	7.14	262	915	1177
48	295	6.94	264	914	1178
50	298	6.68	267	912	1179

Use the following link if you wish to view the full table:

http://www.engineeringtoolbox.com/saturated-steam-properties-d_273.html

It helps to have some pressure temperature points handy, so long ago I memorized the following which has been a big help over the years –

- Steam at 0 psig is 212° F
- Steam at 25 psig is 250° F
- Steam at 50 psig is 298° F
- Steam at 100 psig is 338° F
- Steam at 150 psig is 366° F
- Steam at 200 psig is 388° F

Note that as the pressure increases, the corresponding temperatures rise at a <u>slower</u> rate.

Using this information, we will look at how to calculate steam flows for a wide variety of applications by using simple mathematics and short cuts to simplify steam flow calculations.

Chapter 2 – Calculating Steam Flows-The Long Formula

Calculating steam flows can appear complex at first, but with a few shortcuts we can simplify the calculation process.

The basic equation to calculate the BTU's required to heat any substance is as follows –

$$BTU = M\ x\ SH\ x\ \Delta T$$

Where - BTU = British Thermal Units

M =Mass or weight in pounds of material to be heated

SH = Specific Heat of material to be heated

$\Delta T =$ Final temperature minus the initial temperature

As an example let's assume we wish to heat 100 gallons of water from 60° F to 160° F. One gallon of water weighs 8.34 lbs (100 gallons = 834 lbs) and has a specific heat of 1.0. From the formula-

$$BTU = 100\ gallons\ x\ 8.34\ lb\ x\ 1.0\ x\ 100$$

$$BTU = 83{,}400$$

Steam flows are measured in pounds per hour (lb/hr). To arrive at a heat flow rate, we need to modify our basic BTU equation –

$$\boldsymbol{BTU = (M\ x\ SH\ x\ \Delta T) \div (T)}$$

Where - T = Time in hours

The steam flow rate is a function of the total BTU's required over the amount of time to achieve the temperature change. A few examples –

You wish to heat the 100 gallons of water from 60° to 160° F in –

1.0 hour = 83,400 BTU/Hr

30 Minutes (.5 Hr) = 166,800 BTU/Hr

15 Minutes (.25 Hr) = 333,600 BTU/Hr

1.5 hours = 55,600 BTU/hr

2.0 hours = 41,700 lb/hr

The point of showing these examples is to show that speed costs money. How so? For many steam calculations, we can use an average of 1 pound of steam contains 1,000 BTU's of thermal energy.

$$1\ lb\ of\ steam - 1000\ BTU$$

In our example, steam flows from 15 minutes at 333.6 lb/hr to 2 hours at 41.7 lb/hr. This is a large range of flow differences depending on your time period required.

To further translate how important knowing the time required to meet your requirement

$$1\ Boiler\ Horsepower = 34.5\frac{lbs}{hr}of\ steam$$

In our example, the range of boiler loads would be 9.7 BHP for 15 minutes to 1.2 BHP for 2 hours, or a ratio of 8.1 : 1.

As we head further into steam flow calculations, time required to accomplish the process is a very important, and in many cases over looked, component to arriving at the required steam flows. I raise this point because, in so many cases over years of my experience in helping a customer to select the right equipment, we would go over the data to arrive at steam total flow requirements and then ask the question; "in what time period?" The typical answer was almost always a very short time period. When converted to steam flows for equipment costs and boiler loading, the customer was always shocked, and came back with a substantial reduction in time required to accomplish the heating of the material. Production rate is virtually always answered faster until dollars and cents come into play. Speed impacts steam flows so always work with a realistic time requirement, or present the options if the decision rests with others.

The enthalpy (also called latent) heat of steam is the number of BTU's of heat required to convert one pound of water to one pound of steam. The amount of latent heat required is a function of the temperature, and therefore the pressure of the steam. A quick table will help illustrate the relationship –

- Steam at 0 psig is 212°F with a latent heat of 970 BTU's per pound
- Steam at 25 psig is 250°F with a latent heat of 934 BTU's per pound
- Steam at 50 psig is 298°F with a latent heat of 912 BTU's per pound
- Steam at 100 psig is 338°F with a latent heat of 880 BTU's per pound
- Steam at 150 psig is 366°F with a latent heat of 857 BTU's per pound
- Steam at 200 psig is 388°F with a latent heat of 837 BTU's per pound

The reverse process occurs that heat delivered to your requirement is the latent heat for the pressure of the steam in your process. Note that low pressure steam has a much higher latent or available heat per pound of steam than high pressure steam. The difference between 200 psig and 25 psig as an example is ~ 10.4%.

We've used rules of thumb before so as a starting point look at your final process temperature and add 50°F to arrive at the temperature of the steam, and therefore the pressure. As an example, our process final temperature required was 200°F which when adding 50°F would yield a steam temperature of 250°F or 25 psig.

Chapter 3 - Calculating Steam Flows-Quick Formula's

Calculating steam flows using the long formulas is accurate but can be challenging with units of measure and other pertinent data required. Is this kind of accuracy required? If you are engineering and designing a new facility, or process, from the ground up the answer would tend to be yes since you have a lot of specification data available from suppliers and other sources. On the other hand, if you are upgrading or making needed repairs to a current system, you may not have all the specification data available or it might be obsolete with the current requirements.

Steam system design on installed systems requiring upgrade and improvements just might not require that kind of precision. We will examine in this chapter a series of +/- 5/10% formulas which will get the job done with sufficient precision based on one assumption which is the latent heat of vaporization (enthalpy) remains constant at 1000 BTU's per pound of steam. If you are concerned with this concession, in over 50 years of upgrading steam equipment, the formulas work and I cannot ever recall a situation where I saw any cause for concern when we started the system back into operation.

Heating Water with Steam

A common use for steam is heating water. Using a simple formula, and a few conversions, let's examine how we can do a great variety of quick calculations and save you a lot of time and effort.

A very important conversion is - $\mathbf{500\frac{lbs}{hr}(steam) = 1\ GPM\ of\ water\ flow}$

The quick formula for heating water with steam is –

$$\boldsymbol{Q = (GPM\ \div 2)\ x\ \Delta T}$$

Q = lb/hr of steam

GPM = Water flow in gallons per minute

$\Delta \boldsymbol{T} =$ Final temperature – initial temperature

As an example, we wish to heat 10 gpm of water from 60 F to 160 F

$$Q = (10\ \div 2)x\ 100$$

$$Q = 5\ x\ 100$$

$$Q = 500\ lb/hr$$

Simple formulas can be very powerful, like Einstein with E = MC^2 defining the relationship between energy and mass.

So using our simple formula, the new problem is we would like to heat 150 gallons of water in 15 minutes from 60°F to 160°F. If we divide 150 gallons by 15 minutes we get 10 gpm. The temperature rise is 100°F, so we have the same problem to solve as we just did above.

Another common problem is heating water in a vessel. To arrive at numbers we can use, we will use geometry to calculate the water volume, and knowing the time required to complete the process we can use our simple formula again.

Another conversion from my aquarium days:

$$\mathbf{231\ cubic\ inches = 1\ gallon\ of\ water}$$

For a rectangular vessel $Volume = length\ x\ Width\ x\ height$

For a round vessel $Volume =\ \pi\ x\ radius^2\ x\ length\ (or\ height)$

Assume a rectangular vessel 5' x 10'x 8' which is 60" x 120" x 96" = 7,296 cubic inches or 31.6 gallons.

Assume a tank 4' in diameter x 10' tall which is the radius squared times the height. The math would be 24" x 24" x 120" = 69,120 cubic inches or 299.2 gallons.

In either example, divide by the required time in minutes to arrive at GPM, divide by 2 and enter the temperature difference for lb/hr of steam.

By always reducing the variables in heating water to a rate of rise in GPM and knowing the temperature difference you can calculate the steam flow quickly.

Heating other Liquids with Steam

Water has a specific heat of 1.0. Let's assume we want to heat 10 GPM of olive oil from 60°F to 160°F. The specific heat (thank you Google) is 1.97. Using our water example we came up with 500 lb/hr of steam but the specific heat of olive oil is 1.97. The ratio of water's specific heat to olive oil is –

$$(Water\ Specfic\ Heat) \div (Olive\ Specfic\ Heat) = Correction\ Factor$$

$$1 \div 1.97 = .51\ Correction\ Factor$$

For correcting our water example for olive oil the steam flow would be –

500 lb/hr x .51 = 255 lb/hr

So revising our correction factor for any liquid -

$$(Water\ Specific\ Heat) \div (New\ Liquid\ Specific\ Heat) = Correction\ Factor$$

Our simple formula for water, which can be manipulated for many different problems, can be adjusted for any liquid by using a specific heat correction factor. Sort of amazing, if you think about it, how much you can do with a simple equation.

Heating Air with Steam

Heating air with steam is another common application for both comfort heating as well as many process applications in industry.

The quick formula for heating air with steam is –

$$Q = (CFM \div 800)\ x\ \Delta T$$

Q = Steam flow in lb/hr

CFM = Cubic feet per minute

ΔT = Final temperature minus initial temperature

As an example, assume you wish to heat 16,000 CFM of air 20°F to 120°F.

$$Q = (16{,}000 \div 800)\ x\ 100$$

$$Q = 2{,}000 \frac{lb}{hr} of\ steam$$

As in the heating water with steam quick formula, if you take the steam flow calculated you can also quickly convert to BTU's per hour since both equations assume the heat content of steam to be 1,000 BTU's per pound. Using the air example, 200 lb/hr is equal to 200,000 BTU's per hour.

This formula can be used for many heating air applications. As an example, we wish to estimate how much heat would be required to heat a space. Assume the dimensions of the space are 20' wide by 50' long and 15' high. The volume of the space would be 15,000 cubic feet. A reasonable assumption is the space air volume would be subject to four air changes per hour. If we divide 15,000 by 4, we get an equivalent air flow of 3,750 cubic feet per hour. If the temperature increase required for each air change is 20°F, the steam required to heat the space would be ~94 lb/hr or 94,000 BTU/hr. If we planned to use steam, or gas fired unit heaters, two 50,000 BTU/hr units should do the job. To be precise, you should do the long formula calculations for heating a space, but I suspect your final numbers will be close to the quick formula estimate.

In HVAC, as well as process applications, steam coils are used to heat air. If the original specification data is missing on air flow data, then this method might help solve your problem. The quantity of air moving in a duct can be calculated from –

$$CFM = Velocity\ x\ Face\ Area$$

As an example, assume you have a duct handling process air being heated from 60°F to 160°F with a height of the duct 3' and the width 6'. The area of the duct is 18 square feet which would also be very close to the face area of the coil – so use the duct area to be safe.

If you have an air velocity meter, you can measure the air velocity in feet per minute. If you don't have a meter, typically HVAC ducts are designed for velocities of 300 to 600 feet per minute. Ducts handling process air are usually designed for velocities in the range of 500 to 1,000 feet per minute. You will have to make a judgement call, but to be safe, I would tend to go with the higher velocity. In our example for process air, I would use 1,000 feet per minute.

$$CFM = 1{,}000 \frac{feet}{minute} x\ 18\ square\ feet$$

$$CFM = 18{,}000$$

Using our estimating formula –

$$Q = (CFM \div 800)\ x\ \Delta T$$

$$Q = (18{,}000 \div 800)\ x\ 100$$

$$Q = 2{,}250\ lb/hr$$

A comment on steam design engineering with all the data available, as compared to what you must deal with in so many cases with fixing an existing installation with little or no data available. Long ago when I was learning the ropes of steam systems, an old timer suggested that solving steam problems is a bit like cooking; part recipe and part judgement call. If you are faced with the judgement call, play with safer estimates if faced with unknown choices. You will never face an unhappy person if you over engineered a bit and it works. If you go on the low range of estimates and the fix does not work, I promise you will make that mistake just once – this is from my personal experience!

Other Estimating Options

Some other conversions that can be used to help calculate, or estimate, steam flows are as follows –

- If you use 1,000 BTU's of energy = 1 pound of steam, any rating for steam consuming equipment providing a BTU rating can be used to arrive at a steam flow. As an example, a unit heater rated at 100,000 BTU/hr would consume 100 lb/hr of steam.
- If you are heating a product, or anything in contact, with a heated metal surface, then each square foot of metal surface will radiate about 35,000 BTU/hr of thermal energy if it is at least 50°F hotter than the material being heated. As an example, a platen press with a heated area of 20 square feet would radiate about 700,000 BTU/hr which would point at a steam consumption of about 700 /b/hr. This is a Hail Mary last resort estimator and surprisingly has served me well over many years when no other data is available.
- One boiler horsepower (BHP) is equal to 34.5 lb/hr of steam output. For smaller steam systems, look at the boiler size in HP and if the majority of the steam is being directed to a few final uses, then you might be able to estimate steam flows based on the estimated division of the boiler output to each application. A Hail Mary approach which can work in a no data available situation.
- If the steam to your application is being fed through a control valve, you can use the line size of the control valve as a possible sizing option. Take the line size installed and look up for most any brand of pilot operated pressure reducing

valve for the steam flow at the supply pressure and to the lowest delivered pressure to gain an estimate of steam consumption.

- In really desperate situations, you can take the line size of the steam line feeding the equipment and look on a steam flow table showing steam velocities and use a figure at about 5,000 feet per minute.

The operative word on using this approach is that this will provide an *estimate* and should be used accordingly and with caution. The more hard data you have to work with, the more accurate your calculations will be.

Chapter 4 – Sizing Steam Piping

The arteries of a steam system are the piping system. To properly size a steam line you will need the following information –

- The quantity of steam to be carried in the pipe
- The operating pressure of the pipe
- The permissible pressure drop in the pipe
- The distance of the pipe.

From the Engineering Tool Box Website this chart will allow you to make a line size selection based on the capacity and pressure -

http://www.engineeringtoolbox.com/sizing-steam-pipes-d_266.html

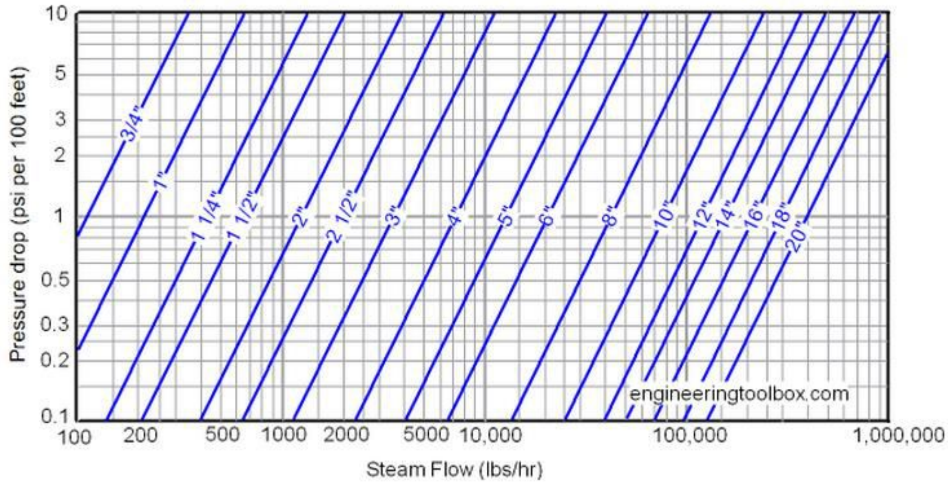
Capacity *(lb/hour)*															
Pressure ***(psi)***	**Steam Velocity** ***(ft/sec)***	**Pipe Size** ***(inch)***													
		1/2"	3/4"	1"	1 1/4"	1 1/2"	2"	2 1/2"	3"	4"	5"	6"	8"	10"	12"
5	50	12	26	45	70	100	190	280	410	760	1250	1770	3100	5000	7100
	80	19	45	75	115	170	300	490	710	1250	1800	2700	5200	7600	11000
	120	29	60	110	175	245	460	700	1000	1800	2900	4000	7500	12000	16500
10	50	15	35	55	88	130	240	365	550	950	1500	2200	3770	6160	8500
	80	24	52	95	150	210	380	600	900	1500	2400	3300	5900	9700	13000
	120	35	72	135	210	330	590	850	1250	2200	3400	4800	9000	14400	20500
20	50	21	47	82	123	185	210	520	740	1340	1980	2900	5300	8000	11500
	80	32	70	120	190	260	520	810	1100	1900	3100	4500	8400	13200	18300
	120	50	105	190	300	440	840	1250	1720	3100	4850	6750	1300	19800	28000
30	50	26	56	100	160	230	420	650	950	1650	2600	3650	6500	10500	14500
	80	42	94	155	250	360	655	950	1460	2700	3900	5600	10700	16500	23500
	120	62	130	240	370	570	990	1550	2100	3950	6100	8700	16000	25000	35000
40	50	32	75	120	190	260	505	790	1100	1900	3100	4200	8200	12800	18000
	80	51	110	195	300	445	840	1250	1800	3120	4900	6800	13400	20300	28300
	120	75	160	290	460	660	1100	1900	2700	4700	7500	11000	19400	30500	42500
60	50	43	95	160	250	360	650	1000	1470	2700	3900	5700	10700	16500	24000
	80	65	140	250	400	600	1000	1650	2400	4400	6500	9400	17500	27200	38500
	120	102	240	410	610	950	1660	2600	3800	6500	10300	14700	26400	41000	58000
80	50	53	120	215	315	460	870	1300	1900	3200	5200	7000	13700	21200	29500
	80	83	190	320	500	730	1300	2100	3000	5000	8400	12200	21000	33800	47500
	120	130	290	500	750	1100	1900	3000	4200	7800	12000	17500	30600	51600	71700
100	50	63	130	240	360	570	980	1550	2100	4000	6100	8800	16300	26500	35500
	80	102	240	400	610	950	1660	2550	3700	6400	10200	14600	26000	41000	57300
	120	150	350	600	900	1370	2400	3700	5000	9100	15000	21600	38000	61500	86300
120	50	74	160	290	440	660	1100	1850	2600	4600	7000	10500	18600	29200	41000
	80	120	270	450	710	1030	1800	2800	4150	7200	11600	16500	29200	48000	73800
	120	175	400	680	1060	1520	2850	4300	6500	10700	17500	26000	44300	70200	97700
150	50	90	208	340	550	820	1380	2230	3220	5500	8800	12900	22000	35600	50000
	80	145	320	570	900	1250	2200	3400	4900	8500	14000	20000	35500	57500	79800
	120	215	450	850	1280	1890	3400	5300	7500	13400	20600	30000	55500	85500	120000
200	50	110	265	450	680	1020	1780	2800	4120	7100	11500	16300	28500	45300	64000
	80	180	410	700	1100	1560	2910	4400	6600	11000	18000	26600	46000	72300	100000
	120	250	600	1100	1630	2400	4350	6800	9400	16900	25900	37000	70600	109000	152000

This chart from the Engineering Tool Box website will allow for a factor of pressure drops on various line size runs –

http://www.engineeringtoolbox.com/steam-pipe-pressure-drop-d_1129.html

Pressure drop in steam pipes can be estimated with the diagrams below.

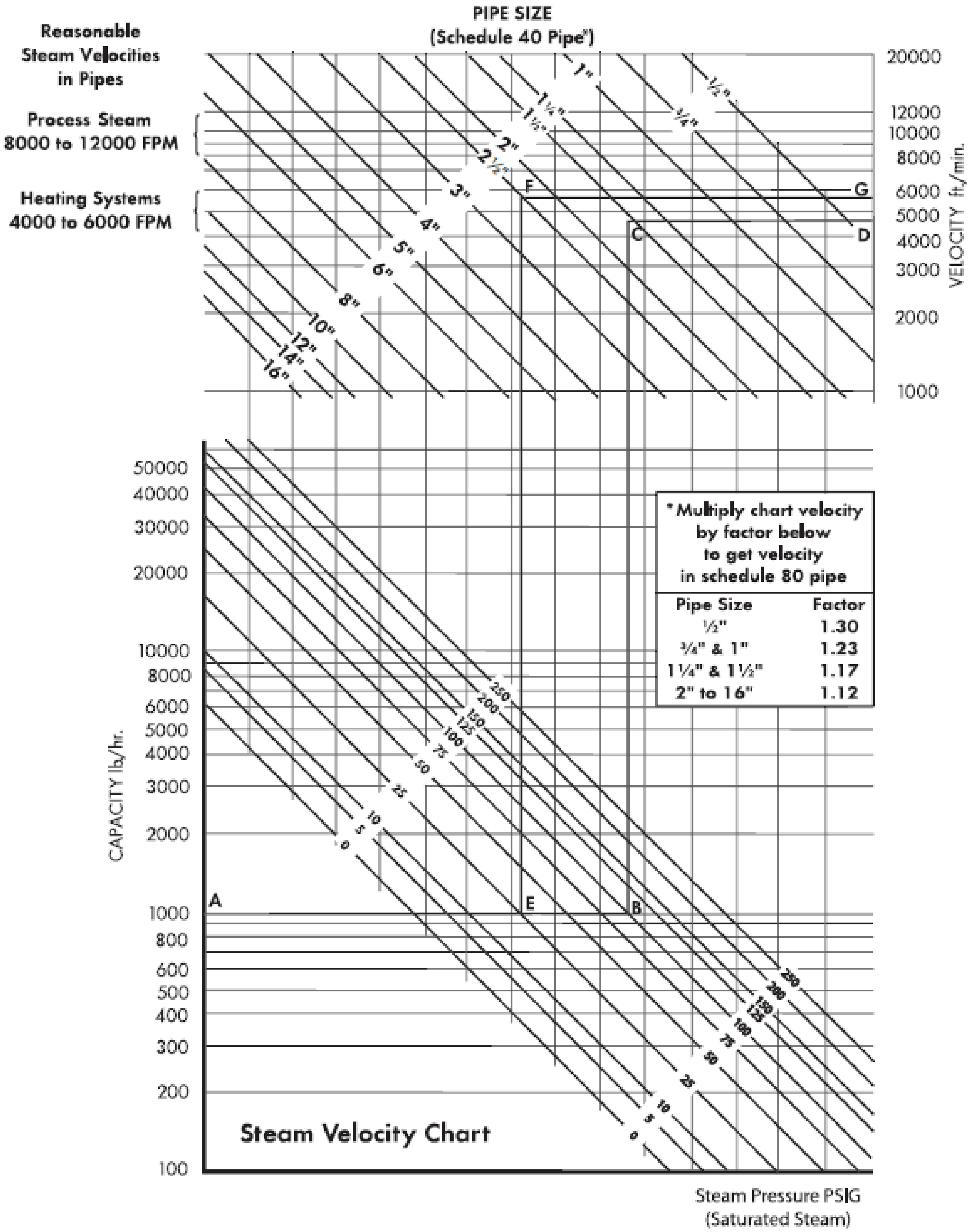
Imperial Units



The diagram is made for steam with pressure *100 psi*. For other pressures use correction factors:

Pressure *(psi)*	0	5	10	30	60	90	100	110	150	200	250	300
Factor	6.9	5.2	4.3	2.4	1.5	1.1	1.0	0.92	0.70	0.55	0.45	0.38

TABLE 4 STEAM VELOCITY CHART



From the charts some added comments –

- Steam line velocities, depending on the source, will sometimes be in feet per minute or as shown on the chart, feet per second. To change from one to the other, multiply or divide by 60 as required. As an example, 6,000 feet per minute would be 100 feet per second.
- Steam line velocities determine the pressure drop per 100 feet of pipe with higher velocities decreasing steam line size, but increasing pressure drop per 100 feet. You might have to do a few option selections juggling velocity and pressure drop for the proper sized line for your application.
- Velocity also determines how much noise a steam line will generate. High velocity steam lines can generate a high pitched "singing" sound which might not be acceptable in your application.
- Velocities in the 4,000 to 6,000 feet per minute range (~ 60-100 feet per second) are suggested for very low noise levels, such as in HVAC systems.
- Velocities in the 8,000 to 12,000 feet per minute range (~130-200 feet per second range) will generate ever increasing noise levels. In general, I would suggest you not exceed 120 feet per second (7,200 feet minute) if possible.

An example will help illustrate the process. Assume you wish to design a steam line to carry 10,000 lb/hr of steam at a pressure of 100 psig. Steam line noise is not an issue for this steam line selection. The line is to run 300'. The process would be as follows –

- Since larger pipe costs more to purchase and install, let's use 120 feet per second as our design velocity.
- Consulting the first chart (Capacity) a 5" steam line will carry 15,000 lb/hr and a 4" steam line will carry 9,100 lb/hr. Since 5" pipe is considered an odd size, you either have to accept higher pressure drop with a 4" line or lower pressure drop with a 6" line. To check which might be the better selection, let's examine both on the pressure drop chart
- For a 4" line the pressure drop is 2 psig per 100' of pipe run or 6 psig for 300'. For a 6" line the pressure drop is .25 psig per 100' or .75 psig for 300'.
- Although either selection might work the 4" line, if you require 100 psig at your point of use, will mean you will have to increase system pressure. Increased pressure adds fuel costs, so over the long run the 6" line is a better choice. Since piping is expensive to purchase and install, the 6" line has reserve capacity in the event you ever add equipment or increase steam flows.
- This is a great example of steam design that is part engineering and part judgement!

With the steam line supply now sized for your application, let's add a new design element to our project. The actual steam pressure required for the equipment which will use 10,000 lb/hr is 30 psig, so the steam pressure at point of use will be reduced by a steam control valve from 100 psig to 30 psig. The line run from the steam PRV will be 50 feet to the equipment inlet.

- From our capacity chart 10,000 lb/hr at 30 psig and 120 feet per second will require an 8" steam line.
- Pressure drop for a 50 foot run will be .1 psig with a correction factor of 2.4 for 30 psig. Multiplying yields a pressure drop of .24 psig for 100 feet of pipe, or about 50% for 50 feet with an estimated pressure drop of ~.12 psig

As with electrical distribution systems, transmitting steam at higher pressures to minimize line size runs, be aware that as you reduce pressures at the point of use you will need to increase line sizes. For most installations of a steam pressure reducing valve, the discharge line size from the PRV will be larger than the supply steam line with larger pressure drops increasing line sizes by quite a bit. You will need to run the same checks on control valves with pressure drops higher than 10-20% of the inlet steam pressure.

Chapter 5 – Sizing Steam Equipment

If you search the web for sizing steam equipment, you can quickly be overwhelmed with a lot of information, which in some cases, leads you to the conclusion that this is a complicated process or requires input into complex calculators. Like many things in life, you can make it as complicated or simple as suits your fancy. I tend to like simple and rely on basic principles.

Let's first define the steam equipment we will address in this chapter –

- Pressure Reducing Valves
- Back Pressure Control Valves
- Temperature Control Valves
- Safety Valves
- Flow Meters
- Steam Traps
- Rotary Joints
- Condensate Pumps
- Other Specialty Valves

Flow is the amount of mass or volume passing from one point to another. Mass flow is typically measured in pounds per hour (lb/hr) for steam and condensate flows are measured in volumetric form in gallons per minute (gpm). The conversion formula for steam between mass and volumetric flow is –

$$500\,\frac{lb}{hr} = 1\ gallon\ per\ minute$$

Physics is the branch of science that defines flow with a general formula as follows –

$$Flow = Area\ x\ Velocity\ x\ Density\ x\ Friction\ Correction\ x\ Square\ Root\ of\ Pressure\ Drop$$

The area, velocity, and density components define the volume of flow while the square root of the pressure drop defines the rate function. Put in simple terms, for any device the rate of flow is proportional to the characteristics of the device multiplied by the square root of the pressure drop. Logic would lead to the conclusion that higher pressure drops impart more energy, which results in higher flow rates.

Trying to calculate the characteristics of any device is a formidable challenge to an empirical format which was devised to deal with the area and friction characteristics. It is called the flow coefficient or C_v of a device.

Pressure drop is defined as the difference of the absolute supply pressure minus an absolute downstream pressure with atmospheric pressure being 14.696 PSIA which is rounded off to 14.7 PSIA. Pressure gauges measure in PSIG which negate atmospheric pressure. A few examples include –

$$14.7\ PSIA = 0\ PSIG$$

$$114.7\ PSIA = 100\ PSIG$$

$$114.7\ PSIA - 113.7\ PSIA = 1\ PSIA$$

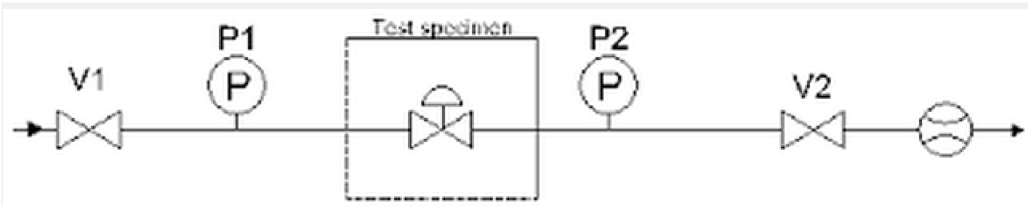
$$100\ PSIG - 99\ PSIG = 1\ PSIG$$

If measurement for pressure is made at the same atmospheric pressure, then the numerical pressure difference is the same numerical value.

C_v is defined as the flow of water through any valve or device at a pressure drop of 1 PSIG. Expressed as an equation –

$$C\mathrm{v} = \mathrm{GPM}/\sqrt{\Delta \mathrm{P}}$$

The test stand would be as follows –



The device to be tested for a C_v factor is placed in a test stand with valves V1 and V2 being used to control the pressure drop between PI and P2 (example 1 PSIG). Flows can be metered or weighed to arrive at results. With a C_v factor for a valve, the flow for other pressure drops can be calculated by re-arranging the C_v formula as follows –

$$GPM = Cv\ x\ \sqrt{\Delta P}$$

As an example, a water control valve has a C_v of 10. What would be the flow of the valve from 25 psig to atmospheric pressure (0 psig)?

$$GPM = Cv\ x\ \sqrt{\Delta P}$$

$$GPM = 10\ x\ \sqrt{25}$$

$$GPM = 50$$

Using the C_v for water flow calculations works great. Using C_v for steam flow calculations is more complicated and requires complex formulas which can be found in engineering reference books or on the web - not my first choice following the principal for keeping things simple.

Most manufacturers have done extensive tests of their equipment and provided flow tables for their equipment. We will use flow tables to size our equipment.

To size any valve or related piece of equipment you need three pieces of information –

- The required flow in lb/hr for steam and gpm for water or condensate
- The supply pressure in psig
- The discharge pressure in psig

Chapter 6 Sizing Pressure Reducing Control Valves

Steam pressure reducing valves are offered by many manufacturers worldwide and the capacity tables are virtually identical for all of them. I will use the chart for an Armstrong GP 2000 as an example to walk you through the process –

Capacities for Steam

GP-2000, GP-2000K-1, GP-2000K-3, GP-2000K-6, GD-2000K, GP-2000R, GP-2000CS Capacities—Steam lb/hr											
lb/hr											
Inlet	Outlet	Connection Size in									
psig		1/2	3/4	1	1-1/4	1-1/2	2	2-1/2	3	4	6
C_v Factor		5	7.2	10.9	14.3	18.8	32	60	78	120	250
15	8	201	290	438	575	756	1,287	2,413	3,137	4,826	10,055
	3	250	361	546	716	942	1,603	3,005	3,907	6,010	12,521
20	13	219	316	478	628	825	1,404	2,633	3,423	5,267	10,972
	3	313	451	683	896	1,178	2,006	3,761	4,889	7,521	15,669
25	18	236	340	515	676	889	1,513	2,837	3,688	5,673	11,819
	3 - 5	339	489	740	971	1,276	2,172	4,073	5,295	8,146	16,972
30	23	252	363	550	721	948	1,614	3,026	3,934	6,052	12,609
	3 - 7	382	550	833	1,093	1,437	2,446	4,586	5,962	9,172	19,109
40	33	281	405	613	804	1,057	1,799	3,373	4,385	6,747	14,056
	25	395	569	861	1,130	1,486	2,529	4,741	6,164	9,483	19,756
	3 - 12	468	673	1,020	1,338	1,758	2,993	5,612	7,296	11,224	23,384
50	42	327	471	713	936	1,230	2,094	3,927	5,105	7,853	16,361
	30	491	707	1,071	1,405	1,847	3,143	5,894	7,662	11,788	24,557
	3 - 17	553	797	1,206	1,582	2,080	3,540	6,638	8,630	13,276	27,659
60	51	373	537	814	1,067	1,403	2,389	4,479	5,823	8,958	18,662
	45	471	679	1,028	1,348	1,773	3,017	5,657	7,355	11,315	23,572
	35	586	843	1,277	1,675	2,202	3,748	7,027	9,135	14,053	29,278
	3 - 22	639	920	1,392	1,827	2,401	4,088	7,664	9,963	15,328	31,934
75	63	471	678	1,026	1,346	1,769	3,012	5,647	7,341	11,295	23,530
	55	593	854	1,292	1,696	2,229	3,794	7,114	9,249	14,229	29,643
	45	703	1,012	1,532	2,010	2,643	4,499	8,435	10,966	16,871	35,148
	4 - 30	767	1,104	1,672	2,193	2,884	4,908	9,203	11,964	18,406	38,347
100	85	595	857	1,298	1,703	2,239	3,811	7,145	9,289	14,291	29,773
	75	751	1,081	1,636	2,147	2,822	4,804	9,007	11,709	18,014	37,529
	60	914	1,316	1,992	2,614	3,436	5,849	10,967	14,257	21,934	45,696
	5 - 42	981	1,412	2,138	2,805	3,687	6,276	11,768	15,299	23,536	49,034

The vertical information shows pressure drops and capacities while the horizontal information relates to sizes and C_v factors.

Let's examine the information for a 1" valve in some detail –

- The C_v factor is 10.9 and can be used if desired to compare it to other steam PRV's. You can also use this information to run through formulas to arrive at capacities for conditions not shown on the chart.
- Up to four capacities are provided for a specific inlet pressure.
- As an example, at a supply pressure of 60 psig, capacity data is provided for reduced pressures of 51, 45, 35 and 3-22 psig.

Flow through a valve for steam will max out due to velocity considerations at ~ 50% of the absolute inlet pressure to the valve. A supply pressure of 60 PSIG is 74.7 PSIA and 50% of that value is 37.4 PSIA or 22 psig. This is called the critical pressure drop and is defined as that pressure drop for which no additional flow will occur.

As an example, assume we need to size a steam PRV to pass 1,800 lb/hr from 75 to 45 psig. Using the chart a 1" valve would pass 1,532 lb/hr.

If you have some other pressure drop not covered in the chart, you can use the C_v value and find a correct formula for your conditions. You can also extrapolate the data on the chart to get a close value of the valve size.

As an example, assume we need 2,500 lb/hr from 85 psig down to 50 psig.

- Using the chart, select the inlet pressure on the chart which is just under your supply pressure requirement; in this example it would be 75 psig.
- Select a down steam pressure just under your requirement of 50 psig which would be 45 psig.
- The capacity of a 1 ½" valve from 75 psig to 45 psig is 2,643 lb/hr.

The ratio of flow rates for different pressures is about the ratio for the square root of the pressure drop.

- The capacity of a 1 ½" valve at 30 psig pressure drop (75 - 45 psig) is 2,643 lb/hr.
- The desired pressure drop is 35 psig (85 – 50 psig).
- The square root of 30 is 5.47 and the square root of 35 is 5.92 or an increase ratio of 1.08. The capacity of the PRV for this application would be 1.08 x 2643 lb/hr or 2,854 lb/hr. Is this precisely correct? No. Is it close enough in the real world for 99%+ of steam PRV applications? Yes.

Remember to check your steam line downstream pipe size on a chart to make sure it is of the proper size. The greater the pressure drop in a steam PRV, the greater the downstream line size will be as compared to the supply line size.

Back pressure control valves will follow the same sizing format.

Temperature Control Valves

Sizing a steam temperature control valve will follow the same type of flow charts and process but with one major difference; you will not know the downstream pressure in most applications. If the equipment you are supplying steam to has vendor technical data specifying the supply pressure to the equipment, by all means use that value.

If you don't know the steam pressure, here is the method I was taught 50 years ago by the old timers I learned from and has served me well over the years. As we learned with a steam PRV, critical pressure drop across a steam control valve is ~50% of the absolute inlet pressure.

The rule of thumb I learned, and have used with great success, is to use 25% of the inlet pressure in psig to select a safe downstream pressure for a temperature control valve.

As an example, you need a TRV (temperature regulating valve) to supply 2,500 lb/hr to a process at 100 psig supply steam pressure. 25% of 100 is 25 psig which then leads to a downstream pressure of 75 psig. Now you can follow the PRV method to select a valve to pass 2,500 lb/hr of steam from 100 psig to 75 psig.

Chapter 7 Sizing Safety Valves

Safety valves for steam can be set to meet one of three requirements.

V Stamp which applies to all ASME Section I valves installed on boilers operating at a pressure greater than 15 psig.

HV stamp which applies to all ASME Section IV valves installed on boilers operating at 15 psig or less.

UV stamp which applies to all ASME Section VIII valves installed on unfired steam using equipment.

For unfired vessels, the UV Stamp Section VIII sizing tables are used to size and select a safety valve. Accumulation is the amount of pressure change in percent of set pressure required for the valve to go to full capacity.

For unfired vessels (Section VIII), safety valves must be set to prevent the pressure from rising 10% or 3 psig, whichever is greater above the maximum operating pressure of the vessel. As an example, assume a pressure vessel has a maximum pressure rating of 100 psig. 10% of 100 psig is 10 psig and since it is greater than 3 psig, then 90 psig would be the safety valve setting. At 90 psig the valve would begin to open and be 100% open at 100 psig. The capacity of the safety valve would be based on the maximum capacity of the control valve feeding the vessel and not the design capacity of the system.

Safety valves for steam are offered by a number of companies and I will use the sizing chart for the Kunkle series 6000 safety valves as an example. Other manufacturers' capacity charts will offer similar information.

Kunkle uses letters to designate the orifice size. For the 6000 series it ranges from D to J. The selection process for a steam safety valve for a non-fired application is as follows:

- Determine the set, or relief pressure for the valve. If feeding a vessel it will be the ASME stamped maximum pressure, or it could be some other component in the system with a lower maximum pressure rating. Check carefully and always use the lowest component or vessel pressure rating in your system as your safety valve set pressure.

- Determine the maximum capacity of steam which could be flowing to the system. As an example, assume you have calculated a steam load of 1,500 lb/hr and then selected a control valve which will pass at least that amount of steam flow. In virtually every application, the capacity of the steam control valve will be higher for the valve selected and that is the flow which must be used for the safety valve selection - not the design flow. If the control valve you have selected for this application is 1,800 lb/hr, then 1,800 lb/hr and not 1,500 lb/hr would be the required safety valve capacity.
- The set pressure of the valve is not the maximum pressure you can transmit to the system and will be determined by the accumulation requirement for the safety valve. Set pressure is 10% or 3 psig whichever is greater. So for a vessel with a maximum pressure, the safety valve will start to open at 90 psig and be fully open at 100 psig. Any vessel with a maximum operating pressure of 30 psig or less will be governed by the minimum 3 psig rule. Any vessel above 30 psig will be governed by the 10% rule so keep this in mind if you are operating code vessels close to 10% of the rated vessel pressure.

Capacities

Non-code[1] and ASME Section VIII Steam (U.S., lb/h) Flow Coefficient = 0.878						
Set Pressure (psig)	Orifice Area, in²					
	D (0.121)	E (0.216)	F (0.336)	G (0.554)	H (0.863)	J (1.414)
3	87	155	242	398	621	1017
4	100	178	277	457	711	1166
6	121	215	335	552	860	1409
8	137	245	382	629	980	1606
10	152	271	422	695	1083	1775
15	179	319	497	819	1276	2091
20	206	368	573	944	1471	2410
25	234	417	649	1070	1666	2730
30	261	466	725	1195	1861	3050
35	291	520	808	1333	2076	3401
40	321	573	892	1470	2291	3753
45	351	627	975	1608	2505	4105
50	381	681	1059	1746	2720	4456
55	411	734	1143	1884	2934	4808
60	442	788	1226	2022	3149	5160
65	472	842	1310	2159	3364	5511
70	502	896	1393	2297	3578	5863
75	532	949	1477	2435	3793	6215
80	562	1003	1560	2573	4008	6566
85	592	1057	1644	2710	4222	6918
90	622	1110	1727	2848	4437	7270
95	652	1164	1811	2986	4651	7621
100	682	1218	1895	3124	4866	7973

As an example, you have a vessel with a maximum rated steam pressure of 80 psig and a control valve which feeds it with a maximum capacity (not the design flow which will be less) of 2,450 lb/hr. Verify the chart is for ASME Section VIII and in our example a "G" orifice would meet your requirement.

KUNKLE SAFETY VALVE PRODUCTS
SERIES 6000

Specifications

Model Number1	Orifice	Connections ANSI Standard Inlet in	Inlet [mm]	Outlet in	Outlet [mm]	Valve Dimensions in [mm] A in	A [mm]	B in	B [mm]	C in	C [mm]	Approximate Weight lb	[kg]
60**DC#	D	1/2	[12.7]	3/4	[19.0]	2 1/8	[54]	1 5/8	[41]	6 1/2	[165]	1 1/2	[0.7]
60**DD#[2]	D	3/4	[19.0]	3/4	[19.0]	2 1/8	[54]	1 5/8	[41]	6 1/2	[165]	1 3/4	[0.8]
61**DC#	D	1/2	[12.7]	—		—		—		6 1/2	[165]	1 1/4	[0.6]
60**ED#	E	3/4	[19.0]	1	[25.4]	2 3/8	[60]	1 3/4	[44]	7 1/2	[191]	2 1/2	[1.1]
60**EE#[2]	E	1	[25.4]	1	[25.4]	2 1/2[4]	[64]	1 3/4	[44]	7 5/8[5]	[194]	2 3/4	[1.2]
61**ED#	E	3/4	[19.0]	—		—		—		7 1/2	[191]	2 1/4	[1.0]
62**ED#	E	3/4	[19.0]	1 1/4	[31.75]	2 7/8	[73]	1 3/4	[44]	7 1/2	[191]	2 3/4	[1.2]
60**FE#	F	1	[25.4]	1 1/4	[31.8]	2 5/8	[67]	2	[51]	8 1/2	[216]	3 1/2	[1.6]
60**FF#[2]	F	1 1/4	[31.8]	1 1/4	[31.8]	2 7/8	[73]	2	[51]	8 3/4	[222]	3 3/4	[1.7]
61**FE#	F	1	[25.4]	—		—		—		8 1/2	[222]	3 1/4	[1.5]
62**FE#	F	1	[25.4]	1 1/2	[38.0]	2 7/8	[73]	2	[51]	8 1/2	[222]	3 3/4	[1.7]
60**GF#	G	1 1/4	[31.8]	1 1/2	[38.0]	3 1/8	[79]	2 3/8	[60]	9 5/8	[244]	5 1/2	[2.5]
60**GG#[2]	G	1 1/2	[38.0]	1 1/2	[38.0]	3 3/8	[86]	2 3/8	[60]	10	[254]	5 3/4	[2.6]
61**GF#	G	1 1/4	[31.8]	—		—		—		9 5/8	[244]	5	[2.3]
62**GF#	G	1 1/4	[31.8]	2	[51.0]	3 3/8	[86]	2 1/4	[57]	9 5/8	[244]	5 3/4	[2.6]
60**HG#	H	1 1/2	[38.0]	2	[51.0]	3 5/8	[92]	2 3/4	[70]	10 5/8	[270]	7 3/4	[3.5]
60**HH#[2]	H	2	[51.0]	2	[51.0]	4 1/8	[105]	2 3/4	[70]	11 1/8	[283]	8	[3.6]
61**HG#	H	1 1/2	[38.0]	—		—		—		10 5/8	[270]	7 1/4	[3.3]
62**HG#	H	1 1/2	[38.0]	2 1/2	[64.0]	3 7/8	[98]	3	[76]	10 5/8	[270]	8	[3.6]
60**JH#	J	2	[51.0]	2 1/2	[64.0]	4 1/4	[108]	3 3/8	[86]	13 5/8	[346]	15 1/2	[7.0]
60**JJ#[2]	J	2 1/2	[64.0]	2 1/2	[64.0]	4 1/2	[114]	3 3/8	[86]	14	[356]	15 3/4	[7.2]
61**JH#	J	2	[51.0]	—		—		—		13 5/8	[346]	15	[6.8]
62**JH#	J	2	[51.0]	3	[76.0]	4 5/8	[117]	3 3/8	[86]	13 5/8	[345]	15 1/2	[7.0]

The 6000 series is offered in a number of trim packages - see if you are interested-

http://www.control-specialties.com/admin/uploads/kunkle_6010.pdf

The key issue is that the “G” orifice is offered in four pipe sizes; 1 ¼” x 1 ½”, 1 ½” x 1 ½”, 1 ¼” x no threads and 1 ¼” x 2”. Lower steam pressures will require larger steam line sizes, so make your final pipe size selection based on the discharge pressure from the safety valve.

Every other brand of safety valve made will have similar engineering information in their data sheets which will follow the Kunkle example we have used.

Safety valves become a very important part of your steam system so, if you are uncertain on selecting a new or replacement valve, consult a reliable source for guidance.

More information can be found at the Engineering Tool Box website –

http://www.engineeringtoolbox.com/safety-valves-high-pressure-steam-d_833.html

Chapter 8 Sizing Flow Meters

There are many types of flow meters available, those suitable for steam applications include:

- Orifice plate flowmeters
- Turbine flowmeters (including shunt or bypass types)
- Variable area flowmeters
- Spring loaded variable area flowmeters
- Direct in-line variable area (DIVA) flowmeter
- Pitot tubes
- Vortex shedding flowmeters

Each of these flowmeter types has its own advantages and limitations. To ensure accurate and consistent performance from a steam flowmeter, it is essential to match the flowmeter to the application.

All flow meters rely on the velocity in the steam line to arrive at a steam flow. In the great majority of cases, the meter size will not be the pipe size you are planning to measure the flow in your facility. Check carefully the specifications for the meter type and brand you are considering for selecting the proper sized meter. In many cases, the meter pipe size will be smaller to provide more accurate readings and better turn down.

A great reference document on meter types can be found at the EPA website –

http://www.epa.gov/chp/documents/wbnr011013_ierna.pdf

Chapter 9 Sizing Steam Traps

A steam trap, in a simple definition, is an automatic valve which will vent non-condensables and condensate from a piece of steam consuming equipment and stop the loss of live steam as it enters the steam trap. Steam traps fail prematurely due to improper sizing and selection, with oversizing being high on the list of trouble-makers.

To size a steam trap you will need the following information.

- Steam traps come in two basic modes of operation. Applications which have the steam supplied by a modulating control valve should use a float and thermostatic (F&T) trap. Constant pressure applications can use a blast on/off steam trap which include inverted bucket traps, disc or thermodynamic traps and thermostatic traps. I do not consider orifice traps to meet the requirement of a steam trap, so if you plan to use this type of device, consult the supplier for suggestions on sizing.
- The condensate flow to the steam trap, which will be in almost every case the steam flow to the device if you are draining equipment. Refer to the steam tables in Chapter if you are draining steam lines and mains.
- The supply pressure to the trap which for modulating control valves can be tricky to pin down and not an issue for constant pressure applications.
- The back pressure on the discharge side of the trap, which should be carefully examined or measured with a pressure gauge if possible.

Safety factors will vary based on the steam trap type. This table from the Engineering Toolbox has a great summary.

Steam Trap Type	Safety Factor
Balanced Thermostatic Steam Trap	3
Bimetallic Steam Trap	2.5
Float Steam Trap	2
Inverted Bucket Steam Trap	2.5
Liquid Expansion Steam Trap	3
Thermodynamic Steam Trap	1.5

http://www.engineeringtoolbox.com/steam-trap-safety-factor-d_1144.html

Steam traps on constant pressure applications are easier to size and select, so we will explore the sizing process for this type of application in detail and then expand to modulating control valve applications. From the Engineering Toolbox website, this chart provides an overview of steam trap operational characteristics.

Type of Steam Trap	Operation				Normal Failure Mode
	No or little load	Light Load	Normal Load	Heavy Load	
Float & Thermostatic	No Action	Usually continuous. May cycle.	Usually continuous. May cycle.	Continuous	Closed
Inverted Bucket	Small Dribble	May dribble	Intermittent	Continuous	Variable
Bi-metal Thermostatic	No Action	Usually Dribble Action	May blast at high pressures	Continuous	Open
Impulse	Small Dribble	Usually continuous with blast at high loads	Usually continuous with blast at high loads	Continuous	Open
Thermodynamic Disc	No Action	Intermittent	Intermittent	Continuous	Open

If you wish to delve further into steam trap types see our EBook “Steam Distribution and Primary Equipment” or this link to the Engineering Toolbox site which covers steam trap operation.

http://www.engineeringtoolbox.com/steam-traps-d_282.html

If you want the short version recommendations based on my 50 years of experience –

- For all applications where the steam supply is modulated use an F&T trap.
- For constant pressure applications you can use any of the on/off blast style traps which by default is all other non F&T traps.

Capacity charts for mechanical traps are different than charts for other types of traps. We’ll examine a typical capacity chart for a non-mechanical steam trap first. We’ll use a Spirax Sarco TD-52 as an example.

Non-Mechanical Chart

Capacities

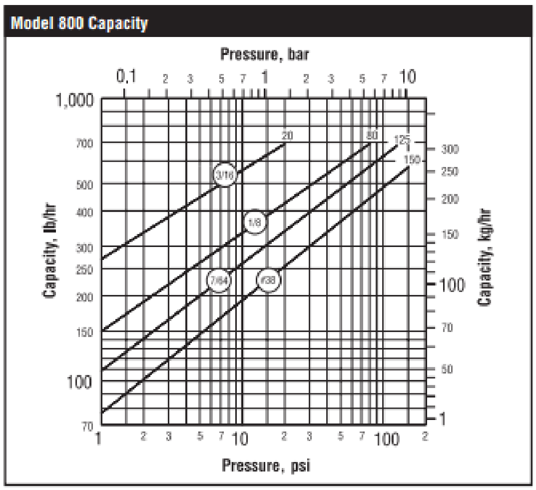
Pounds of condensate per hour continuous discharge at saturated steam temperature to atmosphere

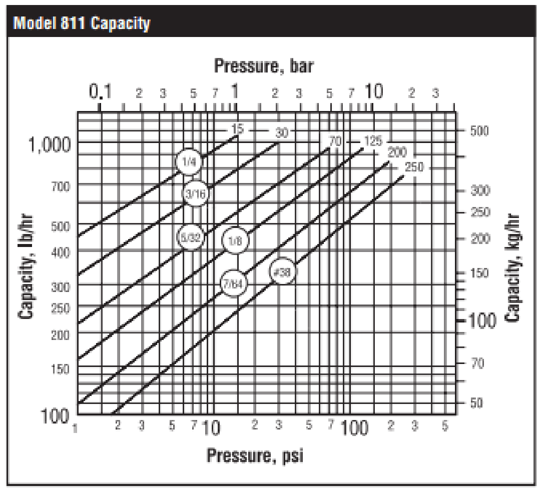
Inlet Pressure psig	barg	3/8" TD52 1/2" TD52L	1/2" TD52 3/4" TD52L	TD52 3/4"	TD52 1"
3.5	*.24*	180	300	405	640
5	*.34*	185	310	420	670
10	*.69*	190	345	470	725
20	*1.4*	200	410	560	865
30	*2.1*	215	465	640	980
50	*3.5*	245	575	810	1200
75	*5.2*	305	700	1000	1470
100	*6.9*	370	810	1160	1750
150	*10.3*	500	1000	1450	2200
200	*13.8*	610	1140	1670	2600
250	*17.2*	700	1270	1900	2900
300	*20.7*	790	1410	2100	3250
350	*24.1*	880	1530	2250	3500
400	*27.6*	960	1630	2430	3780
450	*31.0*	1050	1730	2600	4020
500	*34.5*	1100	1830	2750	4250
550	*37.9*	1160	1910	2900	4450
600	*41.4*	1250	2000	3050	4700

Trap sizes and model numbers are listed across the top and capacities listed as a function of the supply pressure. As an example, a ½" TD-52 or ¾" TD-52L has a capacity of 810 lb/hr.

As an example of a mechanical steam trap, we will use an Armstrong 800 inverted bucket trap. Mechanical traps use a float or inverted bucket coupled to a lever arm to open and close a valve seat. The force which this mechanism can generate is a function of the buoyancy of the float or bucket combined with the leverage provided by the arm. This force is fixed. Flow through an orifice is based on the pressure differential with higher pressure differentials generating higher flows. Since the buoyancy is fixed, the seat size for higher pressures has to be reduced in size, but the higher pressure differentials will generate more flow. Mechanical traps reduce the seat size as pressures are increased. This allows one mechanism, and therefore one trap, to operate over a wide range of pressures. Note for an 800 trap, seat sizes can range from 3/16" to a number 38 drill size based on the supply pressure or pressure differential across the trap. Note that the 3/16" orifice is good for a maximum pressure differential of 20 psig while the number 38 drill size is good for a maximum of 150 psig. The body material of the trap will determine the maximum operating pressure of the trap to contain the pressure; and the seat size will determine the maximum differential pressure operating range. <u>If the differential pressure on the seat selected exceeds the maximum ∆P of the seat, the trap will lock shut. Be aware of this when selecting a mechanical trap or if you increase operating pressure on existing traps.</u>

Mechanical Trap Chart





Steam traps on constant pressure applications are easier to size and select, so we will explore the sizing process for this type of application in detail and then expand to modulating control valve applications.

A few examples might best illustrate the process of picking a constant pressure trap.

- A unit heater rated at 100,000 BTU/hr and operating at a supply pressure of 30 psig requires a steam trap. The condensate will be discharged to a vented condensate pump receiver located below the unit heater. From our load calculation section, a heater output of 100,000 BTU/hr will require a steam flow of 100 lb/hr. The supply pressure is 30 psig and the back pressure is 0 psig. We will use a thermodynamic trap with a safety factor of 1.5 times the design flow. Consult a chart for the desired trap which will discharge 150 lb/hr at a supply pressure of 30 psig
- A unit heater rated at 100,000 BTU/hr and operating at a supply pressure of 30 psig requires a steam trap. The condensate will be discharged to condensate return with no elevation changes uphill and going to a deaerator operating at 7 psig. The pressure drop on the return line run back to the DA is 3 psig. Using our safety factor, select a constant pressure trap capable of operating at 30 psig body pressure, but size the trap on a 20 psig supply pressure due to the ∆P across the trap being 20 psig.

- A unit heater rated at 100,000 BTU/hr and operating at a supply pressure of 30 psig requires a steam trap. The condensate will be discharged to condensate return with elevation changes uphill, and going to a deaerator operating at 7 psig. If you cannot eliminate the elevation changes from where the trap ties into the return line, then use a condensate pump or risk water hammer in the return lines.

Always lift condensate from a steam trap to the highest point and then ensure that the return line makes no further elevation changes. If an elevation change uphill is required, install a condensate pump at this point.

All steam traps will lift condensate and the amount of lift is a function of the supply pressure on the steam trap. A pressure on 1 psig is 27.7” on the water column. As a safe rule of thumb, a steam trap will lift condensate 1 foot for each pound of available pressure. Note that on any modulating steam supply application that the supply pressure to the trap will be 0 psig, so do not lift condensate on any modulating steam supply application - use a condensate pump.

Sizing a steam trap when a modulating control valve is supplying the steam to the equipment adds a major unknown variable-the downstream pressure from the control valve to the equipment using steam.

From our discussions earlier, we discussed the concept of critical pressure drop which occurs when the velocity of the steam through the valves equals the speed of sound. Critical ΔP occurs at ~ 50% of the absolute inlet pressure. As an example, at 100 psig adding 14.7 brings us to 114.7 psia. 50% of 114.7 is about 57.4 psia or about 42.7 psig. At critical ΔP further decreasing the downstream pressure will result in no further flow through the valve.

A typical capacity chart for a control valve will contain 3-5 values for any supply pressure in addition to the capacity for various ΔP values. Using the chart shown below, a 1” valve with a Cv of 10.9 would have the following flow capacities-

- 100 to 85 psig = 1,298 lb/hr
- 100 to 75 psig = 1,636 lb/hr
- 100 to 60 psig = 1,992 lb/hr
- 100 to 5-42 psig = 2,138 lb/hr (critical ΔP)

If you know with certainty the pressure drop across the valve by all means use that value to begin the steam trap selection. If you are uncertain of the pressure drop and want to avoid undersizing the steam trap consider using the critical ΔP value as the required trap capacity. Do you risk possibly being oversized? Yes! I can assure you that no one will be upset with an oversized trap but with an undersized trap you will quickly hear about equipment and production issues. Since you should always use a modulating steam trap, no harm is done to the trap or the process application. I have followed this concept over my 50 years of sizing steam traps and never received a call about undersized traps.

Capacities for Steam

GP-2000, GP-2000K-1, GP-2000K-3, GP-2000K-6, GD-2000K, GP-2000R, GP-2000CS Capacities—Steam lb/hr											
lb/hr											
Inlet	Outlet	Connection Size in									
psig		1/2	3/4	1	1-1/4	1-1/2	2	2-1/2	3	4	6
C_V Factor		5	7.2	10.9	14.3	18.8	32	60	78	120	250
15	8	201	290	438	575	756	1,287	2,413	3,137	4,826	10,055
	3	250	361	546	716	942	1,603	3,005	3,907	6,010	12,521
20	13	219	316	478	628	825	1,404	2,633	3,423	5,267	10,972
	3	313	451	683	896	1,178	2,006	3,761	4,889	7,521	15,669
25	18	236	340	515	676	889	1,513	2,837	3,688	5,673	11,819
	3 - 5	339	489	740	971	1,276	2,172	4,073	5,295	8,146	16,972
30	23	252	363	550	721	948	1,614	3,026	3,934	6,052	12,609
	3 - 7	382	550	833	1,093	1,437	2,446	4,586	5,962	9,172	19,109
40	33	281	405	613	804	1,057	1,799	3,373	4,385	6,747	14,056
	25	395	569	861	1,130	1,486	2,529	4,741	6,164	9,483	19,756
	3 - 12	468	673	1,020	1,338	1,758	2,993	5,612	7,296	11,224	23,384
50	42	327	471	713	936	1,230	2,094	3,927	5,105	7,853	16,361
	30	491	707	1,071	1,405	1,847	3,143	5,894	7,662	11,788	24,557
	3 - 17	553	797	1,206	1,582	2,080	3,540	6,638	8,630	13,276	27,659
60	51	373	537	814	1,067	1,403	2,389	4,479	5,823	8,958	18,662
	45	471	679	1,028	1,348	1,773	3,017	5,657	7,355	11,315	23,572
	35	586	843	1,277	1,675	2,202	3,748	7,027	9,135	14,053	29,278
	3 - 22	639	920	1,392	1,827	2,401	4,088	7,664	9,963	15,328	31,934
75	63	471	678	1,026	1,346	1,769	3,012	5,647	7,341	11,295	23,530
	55	593	854	1,292	1,696	2,229	3,794	7,114	9,249	14,229	29,643
	45	703	1,012	1,532	2,010	2,643	4,499	8,435	10,966	16,871	35,148
	4 - 30	767	1,104	1,672	2,193	2,884	4,908	9,203	11,964	18,406	38,347
100	85	595	857	1,298	1,703	2,239	3,811	7,145	9,289	14,291	29,773
	75	751	1,081	1,636	2,147	2,822	4,804	9,007	11,709	18,014	37,529
	60	914	1,316	1,992	2,614	3,436	5,849	10,967	14,257	21,934	45,696
	5 - 42	981	1,412	2,138	2,805	3,687	6,276	11,768	15,299	23,536	49,034

Steam traps are a broad topic and cover a wide range of applications. In summary, to size a steam trap you need –

- The supply pressure to the steam trap.
- The downstream pressure from the trap.
- The condensate flow through the trap.
- The type of trap best suited to your application.

Following are two links to allow you to further explore information to aid in your selection and installation process.

Best practices for steam trap installation-

http://www.plantengineering.com/single-article/best-practices-for-steam-trap-installation/f18967ab7c05120f5f03dccdaf629b00.html

“The Steam Trap Handbook“ by James F MacCauley P.E. is a great reference manual covering steam traps in great detail. The link below will take you to an online copy of the book by Google Books. The reference manual is part of my library on steam traps.

https://books.google.com/books?id=O1U95ImkT94C&pg=PA136&lpg=PA136&dq=sizing+steam+traps&source=bl&ots=kfVHDJcLiJ&sig=qZYafxOkVq0dffqC42rhxiMJtdw&hl=en&sa=X&ved=0CCEQ6AEwATgUahUKEwjopPWV8KvIAhXKHx4KHcycABc#v=onepage&q=sizing%20steam%20traps&f=false

Chapter 10 – Sizing Condensate Pumps

A steam system is a loop with the final leg of the loop being the condensate return system. For direct injection steam systems, no condensate return is possible so we can close out any discussions.

As a steam trap discharges condensate, two things occur –

- Flash steam is generated as the hot condensate flows to a lower pressure.
- The remainder of the cooler condensate is then available for return to the boiler plant.

To produce steam, we heat feedwater under pressure inside a boiler until it boils off and forms steam. As the heat energy of the steam is used in a process application, the heat of vaporization is transferred to the process and is about 1,000 BTU's per pound of steam condensed. Consult a steam table for further details if you need more information. What then remains is condensate at the corresponding temperature and pressure upstream of the steam trap. A quick refresher on the pressure / temperature relationship can be seen in this short table –

- Steam at 0 psig is 212°F
- Steam at 25 psig is 250°F
- Steam at 50 psig is 298°F
- Steam at 100 psig is 338°F
- Steam at 150 psig is 366°F
- Steam at 200 psig is 388°F

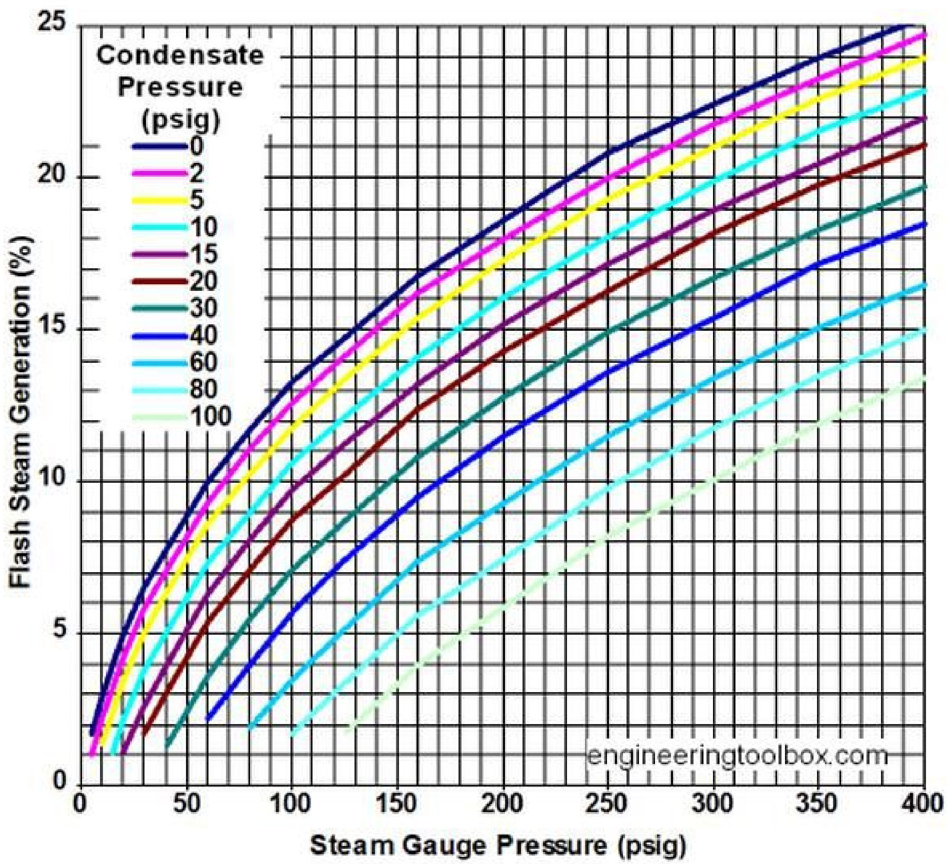
See this link at Engineering Tool Box's website for the full range of pressure / temperature relationships –

http://www.engineeringtoolbox.com/saturated-steam-properties-d_273.html

To understand the basics of flash steam, let's assume that a steam trap is discharging condensate at a supply pressure of 100 psig to 0 psig (atmospheric pressure). On the upstream side of the trap, the condensate is at a temperature of 338°F. As the 338°F condensate flows through the steam trap, the condensate must now exit at 212°F, which is the boiling point of water at 0 psig. To allow the transformation to occur, a portion of the 338°F condensate has to flash off (flash boil), and the thermal energy required to boil off this portion of the condensate reduces the energy in the condensate and lowers the temperature to 212°F.

The same process occurs in a car radiator which, if the radiator cap is quickly removed when the cooling system is up to temperature, will result in a geyser of steam.

Flash charts and tables do the mathematics and provide a flash percentage for any given supply and return pressures. In our example, going from 100 to 0 psig results in a flash rate of 13.3%. See the chart or table below from the Engineering Tool Box website.



Flash Steam generated from Condensate *(%)*											
Steam Pressure before the Steam Trap *(psig)*	Condensate Pressure after the Trap *(psig)*										
	0[1)]	2	5	10	15	20	30	40	60	80	100
5	1.7	1									
10	2.9	2.2	1.4								
15	4	3.2	2.4	1.1							
20	4.9	4.2	3.4	2.1	1.1						
30	6.5	5.8	5	3.8	2.6	1.7					
40	7.8	7.1	6.4	5.1	4	3.1	1.3				
60	10	9.3	8.6	7.3	6.3	5.4	3.6	2.2			
80	11.7	11.1	10.3	9	8.1	7.1	5.5	4	1.9		
100	13.3	12.6	11.8	10.6	9.7	8.8	7	5.7	3.5	1.7	
125	14.8	14.2	13.4	12.2	11.3	10.3	8.6	7.7	5.2	3.4	1.8
160	16.8	16.2	15.4	14.1	13.2	12.4	10.6	9.5	7.4	5.6	4
200	18.6	18	17.3	16.1	15.2	14.3	12.8	11.5	9.3	7.5	5.9
250	20.6	20	19.3	18.1	17.2	16.3	17.7	13.6	11.2	9.8	8.2
300	22.7	21.8	21.1	19.9	19	18.2	16.7	15.4	13.4	11.8	10.1
350	24	23.3	22.6	21.6	20.5	19.8	18.3	17.2	15.1	13.5	11.9
400	25.3	24.7	24	22.9	22	21.1	19.7	18.5	16.5	15	13.4

http://www.engineeringtoolbox.com/flash-steam-generation-d_278.html

If you want to explore flash steam, see this link for a short tutorial for the basic physics-

http://www.engineeringtoolbox.com/flash-steam-generation-d_425.html

Two basic types of condensate pumps have evolved for steam systems –

- Motor driven pumps, which offer high capacities at lower costs, but are prone to cavitation issues due to pumping hot condensate.
- Steam Powered pumps, which in some applications can deal with high temperature condensate, but cost more as compared to a motor driven pump.

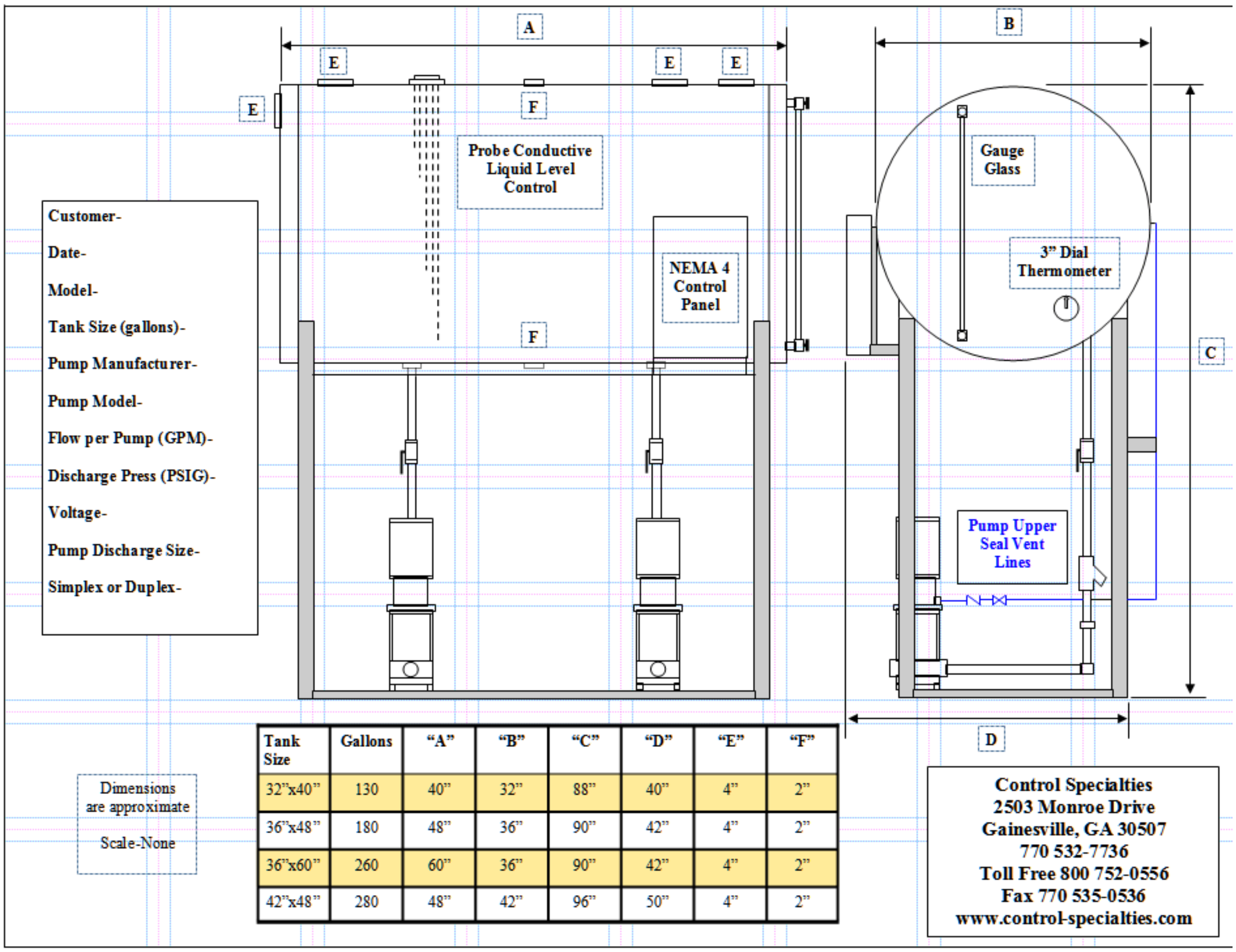
Electric Condensate Pumps

The electric condensate pump package has advantages such as:

- Small package size
- Higher range of capacities
- Excellent for returning cooler condensate
- Inexpensive initial purchase price
- Low inlet height requirements
- Can operate against high back pressure

Unfortunately, they also have several disadvantages, including high maintenance cost on hot condensate applications due to NPSH issues, electrical safety in wet areas, and units must have a 2:1 safety factor on the receiver for condensate cooling. Typically, the standard receiver is too small to allow condensate to cool down to 190°/195°F so it is best to go to a much larger receiver.

Another example of electric condensate pumps for hot condensate is to use a multistage centrifugal pump such as the Grundfos CR series pumps and an oversized horizontal tank for flashing off steam and allowing for a cooling area for the hot condensate.



Tank Size	Gallons	"A"	"B"	"C"	"D"	"E"	"F"
32"x40"	130	40"	32"	88"	40"	4"	2"
36"x48"	180	48"	36"	90"	42"	4"	2"
36"x60"	260	60"	36"	90"	42"	4"	2"
42"x48"	280	48"	42"	96"	50"	4"	2"

Steam Powered Pumps

They have advantages including:

- No seals, motors or impellers thus lower maintenance
- Sized for actual condensate load
- No need for a 2:1 safety factor
- Can be used in closed loop, single trap applications
- Simple to install and repair

- Can return hotter condensate in closed-loop, single trap applications

Steam powered pumps have a capacity limit for all brands, which is limited by the amount of condensate that can flow through the inlet supply line to the pump. As an example, 3" supply and 2" discharge pumps will have capacities in the range of ~10,000 to 13,000 lb/hr and will vary on the filling heads for all manufacturers. If the pumps are handling condensate from multiple traps, a receiver must be used ahead of the pump and be vented to atmosphere to prevent flashing inside the pump on the fill cycle. Sizing of this receiver, or pipe selection, must be done with care. The other piping to and from the pump, as well as the steam supply, must be done properly. It is best to consider buying a skid package from the manufacturer to avoid installation issues. In my experience, a steam powered pump package will cost about twice as much as a comparable motor driven pump. This is a generalization, so each type of pump should be considered for your application, and a choice made based on a cost benefit analysis.

Condensate Pump Sizing

To size a condensate pump you will need to know the required flow from the pump and the required discharge pressure from the pump. The flow to the pump is the steam consumption to your application in lb/hr. The conversion is –

$$\mathbf{500}\frac{\boldsymbol{lbs}}{\boldsymbol{hr}}(\boldsymbol{steam}) = \mathbf{1}\ \boldsymbol{GPM\ of\ water\ flow}$$

For motor driven condensate pumps, you will need a flow GPM. Steam powered pumps are sized based on the lb/hr of condensate.

The required discharged pressure is a total of the static head, or lift, on the pump, friction losses back to the point you wish to pump condensate, and any pressure in the destination point which needs to be overcome.

A comprehensive example, assuming condensate is being pumped from a group of traps discharging to the pump and being returned to a deaerator in the boiler room, is as follows –

- Total condensate flow 10,000 lb/hr
- Total lift or static head on the pump discharge 30'
- Total friction losses for condensate return lines 15' of head loss
- Pressure in deaerator 5-7 psig

Let's deal with the required discharge pressure from the pump first –

- 1 PSIG = 28" (2.33') of water column.

- The total static head is 30’ of vertical lift and 15’ of friction head loss in the return piping for a total head on the pump of 45’. Converting 45’ of head is equal to ~19.3 psig which we will call 20 psig. Add on a deaerator pressure of 7 psig and the calculated head on the pump is 27 psig.
- For an electric pump selection, the design flow is 20 gpm (10,000 lb/hr ÷ 500 lb/hr).
- For a steam powered pump, the required flow is 10,000 lb/hr of condensate.
- You could, if desired, calculate the flash loss and reduce the pump flows by that amount. I do not make this adjustment to allow for added safety margin in selecting a pump since an undersized condensate pump is a very expensive fix.

For an electric pump my selection specifications would be –

- Design flow of 20 gpm with a 2:1 duty cycle on the pump would equal 40 gpm.
- With a total calculated discharge pressure on the pump of 27 psig I would suggest a safety factor of 1.5:1 so the required head on the pump would be 40.5 psig minimum.
- Summary of the specifications would be 40 gpm at a minimum discharge pressure of 40 psig.
- Installing a globe valve on the discharge side of the pump will allow control of the run time of the condensate pump.

For a steam powered pump my selection specifications would be –

- Design flow to the pump 10,000 lb/hr
- Calculated total head on the pump required is 27 psig, and allowing for at least a safety factor of 1.5, a minimum steam pressure to the pump of 40 psig. If the steam pressures are much higher, I would suggest a steam PRV on the steam supply to the pump which will allow you to adjust the rate of discharge from the pump.
- Steam powered pumps for a single drain point can also be put in series with a steam trap and allows for a zero vent loss installation.

Picking the right condensate pump for your application requires a total look at the costs of equipment, condensate temperatures, and making sure the installation is properly engineered. For applications with condensate temperatures from the traps in the 250°-275°F range, an electric condensate pump with an oversized receiver can well serve your needs. For higher temperatures, a steam powered pump or skid package might better serve your requirements.

Both electric and steam powered pumps can be provided in duplex packages. If your application is important to your facility, I would suggest that you go to a duplex installation to avoid downtime.

Recovering condensate is a money saver in water chemicals with payback times usually well under one year. With water consumption becoming a major issue, returning condensate can yield large reductions. As an example, a boiler making 10,000 lb/hr of steam requires 20 gpm of feedwater or 1,200 gallons per hour. If your facility, as an example, operates the steam system 6,000 hours a year, that works out to 7,200,000 gallons annually. If you are able return 80% of the feedwater, you are dropping water consumption by 5,760,000 gallons annually.

Be wise and be green and return condensate.

Chapter 11 - Flash Tank and Vertical Flash Tank Sizing

In Chapter 10 we covered condensate pumps and discussed the issue of hot condensate impacting both electric and steam powered pumps. Flash steam occurs when water goes from a higher pressure to a lower pressure and flashes off the excess temperature to lower the water temperatures to match the system pressure.

The process of flashing condensate can be used to both cool condensate for pumping and also generate opportunities to recover flash steam at lower pressures, which can be used elsewhere in your facility. Using flash steam works best for heating water or air in a process, or application, which is continuous. A great example in the boiler plant is to use flash steam to preheat boiler feedwater.

Flash steam can be recovered by two basic devices –

- A horizontal flash tank
- A vertical flash receiver

Each type has application uses with the horizontal style tank working best for large flows and operations with electric condensate pumps. Vertical flash receivers work well on lower flows; therefore, they tend to work best with single or closely grouped applications. These are broad generalizations and either can be made to work for most applications.

Horizontal Flash Tanks

We'll examine each system and start with horizontal flash tanks. Referring to Table 1 below, enter the vertical steam pressure column which shows the steam pressure to the steam traps. Then move to the right and select the square footage of flash area required per 1,000 lb/hr of condensate. As an example, 10,000 lb/hr of condensate going from 100 psig to 0 psig would require 2.85 square feet of surface area per 1,000 lb/hr or 28.5 square feet of 10,000 lb/hr.

The 28.5 square feet of area is the largest area of a horizontal tank and would be at the half-full point. The 28.5 square feet is the area defined as a rectangle formed by the tank diameter times the tank length. As a general rule of thumb, design a tank length which is about twice as long as its diameter. This would work out to 3 diameters or in our example works out to 9.5' (28.5 ÷ 3).

If we round off to 10' for the tank diameter, the tank length would be 20' providing a flash area of 30 square feet. A bit larger in the tank size is better than going short on surface area.

FLASH TANKS

TABLE 1 FLASH TANK IN SQ. FT. = DIAMETER x LENGTH OF HORIZONTAL TANK FOR 1,000 LB. CONDENSATE PER HOUR BEING DISCHARGED

Steam Pressure PSIG	Flash Tank Pressure PSIG										
	0	2	5	10	15	20	30	40	60	80	100
400	5.41	4.70	3.89	3.01	2.44	2.03	1.49	1.15	.77	.56	.42
350	5.14	4.45	3.66	2.81	2.28	1.91	1.38	1.07	.70	.51	.37
300	4.86	4.15	3.42	2.62	2.11	1.75	1.26	.96	.62	.44	.31
250	4.41	3.82	3.12	2.39	1.91	1.56	1.11	.85	.52	.37	.25
200	3.99	3.40	2.80	2.12	1.68	1.37	.97	.72	.43	.28	.18
175	3.75	3.20	2.61	1.95	1.57	1.25	.87	.64	.38	.23	.15
160	3.60	3.08	2.50	1.86	1.46	1.19	.80	.59	.34	.21	.12
150	3.48	2.98	2.41	1.80	1.40	1.14	.77	.56	.31	.19	.10
140	3.36	2.86	2.31	1.72	1.35	1.05	.72	.52	.29	.16	.08
130	3.24	2.76	2.23	1.65	1.29	1.02	.67	.49	.26	.14	.07
120	3.12	2.65	2.15	1.57	1.22	.97	.61	.44	.23	.12	.04
110	2.99	2.52	2.05	1.50	1.15	.91	.58	.40	.20	.09	.02
100	2.85	2.41	1.92	1.40	1.07	.85	.53	.36	.16	.06	
90	2.68	2.26	1.81	1.30	.99	.77	.48	.31	.13	.05	
80	2.52	2.12	1.67	1.18	.90	.68	.42	.25	.09		
70	2.34	1.95	1.55	1.08	.81	.61	.35	.20	.04		
60	2.14	1.77	1.39	.96	.70	.52	.27	.14			
50	1.94	1.59	1.22	.81	.58	.41	.20	.08			
40	1.68	1.36	1.02	.67	.44	.30	.11				
30	1.40	1.10	.81	.50	.29	.16					
20	1.06	.81	.55	.28	.12						
12	.75	.48	.28								
10	.62	.42	.23								

Table 2 allows you to then calculate the required vent line size based on the square footage of flash area in our example. A flash area of 28.5 square feet would require a vent line size of 4" since 3 ½" is now obsolete. The flash steam generated would be 13.3% of the inlet flow or for 10,000 lb/hr 1,330 lb/hr.

TABLE 2 VENT LINE SIZE FOR HORIZONTAL FLASH TANKS

Area in Sq. Ft. *	Vent Pipe Size
Less than 3.2	1″
3.2 to 5.5	1¼″
5.5 to 7.4	1½″
7.4 to 12.0	2″
12.0 to 17.5	2½″
17.5 to 27	3″
27 to 36	3½″
36 to 47	4″
47 to 73	5″
73 to 105	6″
106 to 140	7″
140 to 185	8″
185 to 300	10″
300 to 420	12″

* Area in sq ft equals diameter of tank in feet multiplied by length of tank in feet.

TABLE 3 PERCENT FLASH

Steam Pressure PSIG	Flash Tank Pressure										
	0	2	5	10	15	20	30	40	60	80	100
5	1.7	1.0	0								
10	2.9	2.2	1.4	0							
15	4.0	3.2	2.4	1.1	0						
20	4.9	4.2	3.4	2.1	1.1	0					
30	6.5	5.8	5.0	3.8	2.5	1.7	0				
40	7.8	7.1	6.4	5.1	4.0	3.1	1.3	0			
60	10.0	9.3	8.6	7.3	6.3	5.4	3.6	2.2	0		
80	11.7	11.1	10.3	9.0	8.1	7.1	5.5	4.0	1.9	0	
100	13.3	12.6	11.8	10.6	9.7	8.3	7.0	5.7	3.5	1.7	0
125	14.8	14.2	13.4	12.2	11.3	10.3	8.6	7.4	5.2	3.4	1.5
160	16.8	16.2	15.4	14.1	13.2	12.4	10.6	8.5	7.4	5.5	4.0
200	18.6	18.0	17.3	16.1	15.2	14.3	12.8	11.5	9.3	7.3	5.0
250	20.6	20.0	19.3	18.1	17.2	16.3	14.7	13.6	11.2	9.8	8.2
300	22.7	21.8	21.1	19.9	19.0	18.2	16.7	15.4	13.4	11.8	10.1
350	24.0	23.3	22.6	21.6	20.5	19.8	18.3	17.8	15.1	13.5	11.9
400	25.3	24.7	24.0	22.9	22.0	21.1	19.7	13.5	16.5	15.0	13.4

Percent flash for various initial steam pressures and flash tank pressures.

Some examples of horizontal flash tank installations are shown as follows –

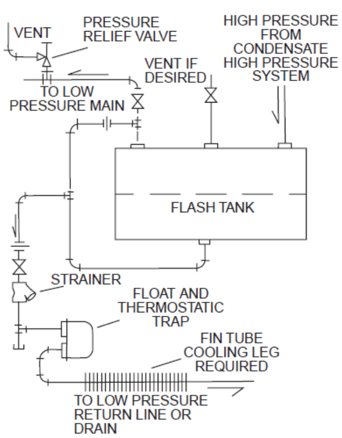


Figure 1 *A typical flash tank piping diagram discharging to atmosphere. NOTE: Omit trap if condensate is discharged into vented pump receiver.*

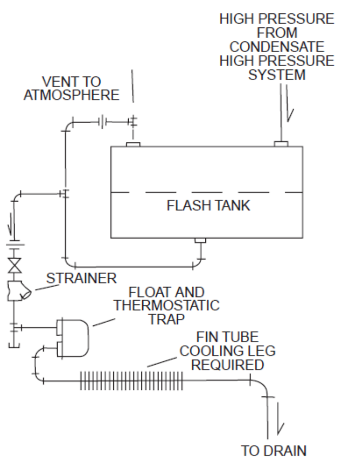


Figure 2 *A typical flash tank piping diagram with flash discharging to low-pressure steam system.*

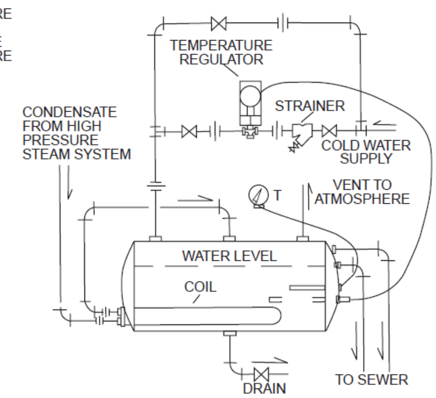


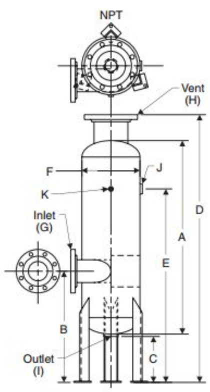
Figure 3 *This diagram depicts a combination flash tank installation with subcooling condensate.*

The design can be altered to provide for flash steam waste heat recovery by installing a back pressure control valve on the vent as shown in Figure 1. Over the past 50 years I have used this old but conservative data to design and install many waste heat recovery systems with never any performance issues.

Vertical Flash Receivers

Vertical flash receivers are much more compact in design, and less costly to purchase and install as compared to a horizontal flash tank. For systems where condensate temperature and electric pump NPSH is a critical issue, I would proceed with caution since the vertical tank does not have the large condensate storage area present in a horizontal tank.

An example of a vertical flash receiver from Armstrong International is as follows –



Features

- ASME coded and stamped vessels
- Standard pressure rating 150 psi (other pressure ratings available upon request)
- Standard models are designed and sized to cover a wide range of applications and loads
- Flash vessels are designed to provide low velocity flash steam with no water carryover
- Quick payback for flash recovery investment
- **Special tanks available upon request**

For a fully detailed certified drawing, refer to CDF #1023.

Physical Data—Standard Design Model VAFT								
Model No.	AFT-6		AFT-8		AFT-12		AFT-16	
	in	mm	in	mm	in	mm	in	mm
A	36	914	36	914	40	1,016	48	1,219
B	21	533	21	533	23	584	26	660
C	9-1/2	241	9-1/2	241	9-1/2	241	9-1/2	241
D	51	1,295	52	1,321	55-3/8	1,407	63-1/2	1,613
E	36	914	36	914	40	1,016	48	1,219
F	6	150	8	203	12	305	16	406
G	2	50	3	80	4	102	6	150
H	2-1/2	65	4	102	6	150	6	150
I	1-1/2	40	1-1/2	40	2	50	2	50
J	3/4	20	1	25	1-1/2	40	2	50
K	1/2	15	1/2	15	1/2	15	1/2	15

NOTE: Connections "G" and "H" are 150 lb. flanges. All others are NPT. All flash tanks are ASME coded for 150 psig (10 bar). Special sizes available upon request.

Capacities—Standard Design Model VAFT				
Model No.	Maximum Condensate Load		Maximum Flash Load	
	lb/hr	kg/hr	lb/hr	kg/hr
AFT-6	2,000	907	500	227
AFT-8	5,000	2,268	1,000	454
AFT-12	10,000	4,536	2,000	907
AFT-16	20,000	9,072	3,000	1,361

Flash Steam Savings Analysis

Part I: Determining the amount of flash steam produced

A. Condensate Load A = ________ lb/hr.

B. Annual hours of operation B = ________ hrs/yr.

C. Steam Cost C = ________ $/1,000 lbs.

D. Flash steam percentage from chart (on page 264) D = ________ %

E. Flash steam produced:

D x A = flash steam produced E = ________ lb/hr.

Part II: Determining dollar value of the flash steam

F. Annual flash steam savings:

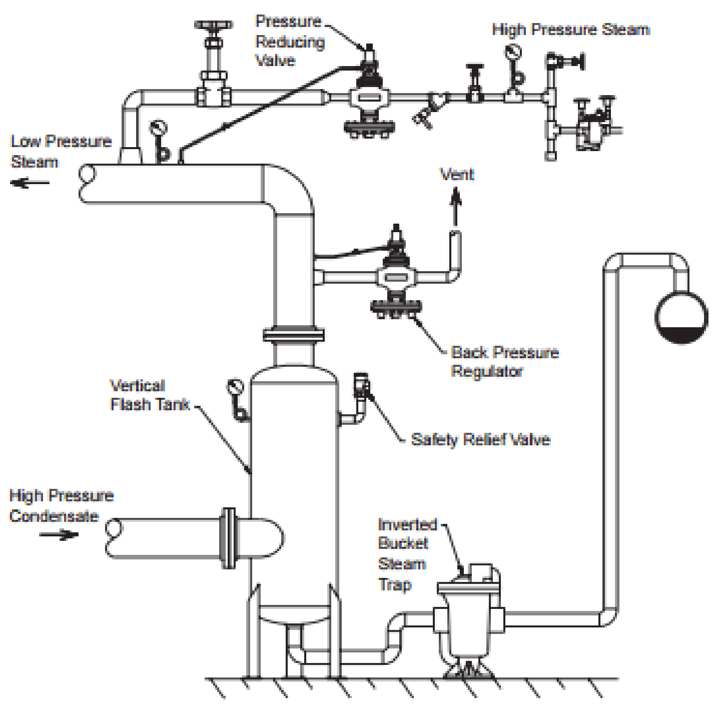
$F = \frac{E \times B \times C}{1{,}000}$ F = ________ $/yr.

A vertical flash receiver is a piece of pipe welded into a vertical tall narrow tank to ASME code specifications. Flash occurs as hot condensate enters with the flash steam rising up and condensate falling to the lower point of the receiver.

Sizing it from a chart –

Capacities—Standard Design Model VAFT				
Model No.	Maximum Condensate Load		Maximum Flash Load	
	lb/hr	kg/hr	lb/hr	kg/hr
AFT-6	2,000	907	500	227
AFT-8	5,000	2,268	1,000	454
AFT-12	10,000	4,536	2,000	907
AFT-16	20,000	9,072	3,000	1,361

In our example a 10,000 lb/hr condensate flow would require an AFT-12 vertical flash receiver which is fabricated from a section of 12"pipe which is about 40" long.



A typical installation with waste heat recovery is shown with piping details. I've used vertical flash receivers in many applications with great success. If used with an electric condensate pump, make sure you provide for an oversized receiver tank to allow for condensate pump cool down.

From a great article in Chemical Engineering magazine, there is a lot of great information on flash system design –

http://www.chemeng.queensu.ca/courses/integratedDesign/Resources/documents/flashsystemdesign.pdf

Flash recovery systems can yield substantial savings and will typically payback in about one year.

Chapter 12– Final Thoughts

A well-engineered steam system is a great way to provide large quantities of thermal energy for use in your facility. Investments in equipment are substantial and energy costs on an annual basis are substantial. An old rule of thumb is that the cost to fuel a boiler annually is about 4 times the cost of the boiler!

A steam system in distress is an energy and pollution mess, including wasting large volumes of water. A steam system should be monitored at least on a monthly basis for proper operations and efficiency.

Steam system design is a topic that could encompass volumes of books to cover the entire range of applications. Consider this as a quick reference document and use the web for lots more information. Be informed when using the web and compare suggestions as well as equipment options.

See our other EBooks on steam and feel free to contact us more information if we can be of help.