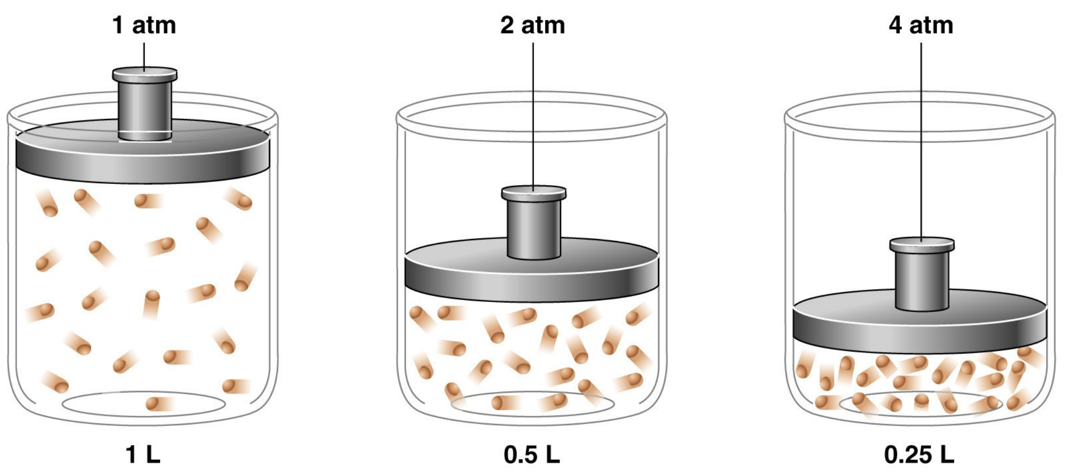
Maths

We will cover - calculation of steam consumption requirements, sizing steam and condensate lines and lots of tables in this section.

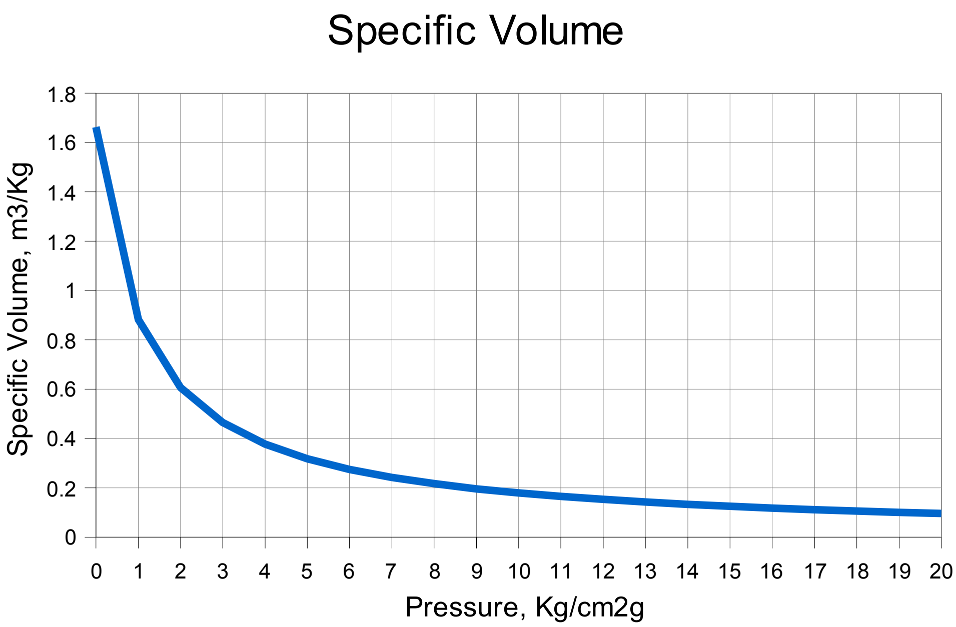
Specific Volume

Specific volume vs. Pressure

We can see below, as the steam pressure increases from 1atm to 4 atm, the density of the steam molecules is increasing. As the specific volume is inversely related to the density, the specific volume will decrease with increasing pressure. We can see the reduced volume in the last jar.



This diagram clearly shows that the greatest change in specific volume occurs at lower pressures, whereas at the higher end of the pressure scale there is much less change in specific volume.



The extract from the steam tables below, shows specific volume, and other data related to saturated steam.

Pressure kg/cm² (g)	Pressure Kg/cm² (abs)	Pressure Bar (abs)	Temp °K	Temp °C	Sp. Vol (Steam) m³ / kg	Density (Steam) kg / m³	Enthalpy of Steam ' h_g ' kcal/kg	Enthalpy of Evap ' h_{fg} ' kcal/kg	Enthalpy of Water ' h_f ' Kcal/kg	Vapour Pressure Bar (a)	Dynamic Viscosity (10)^-6 Pa s
0	1	1	373	100	1.66	0.60	640	539	100	1.02	12.28
0.5	1.53	1.51	384.79	111.64	1.15	0.87	643.98	532.04	111.94	1.51	12.68
1.0	2.03	2.01	393.49	120.34	0.88	1.13	647.01	526.25	120.76	2.01	12.98
2.0	3.03	2.99	406.62	133.47	0.61	1.65	651.34	517.20	134.14	2.99	13.44
2.4	3.43	3.39	410.89	137.74	0.54	1.85	652.68	514.17	138.52	3.39	13.59
3.0	4.03	3.98	416.60	143.45	0.46	2.15	654.41	510.04	144.38	3.98	13.79
4.0	5.03	4.97	424.75	151.60	0.38	2.65	656.77	503.99	152.78	4.97	14.07
5.0	6.03	5.95	431.69	158.54	0.32	3.15	658.65	498.69	159.96	5.95	14.32
6.0	7.03	6.94	437.77	164.62	0.27	3.64	660.20	493.92	166.29	6.94	14.53
7.0	8.03	7.93	443.19	170.04	0.24	4.13	661.51	489.55	171.96	7.93	14.72

At 7 kg/cm^{2}g, the saturation temperature of water is 170°C. More heat energy 'hf' is required to raise its temperature to saturation point at 7 bar g than would be needed if the water were at atmospheric pressure. The table gives a value of 171.96 kcals to raise 1 kg of water from 0°C to its saturation temperature of 170°C.

The heat energy (enthalpy of evaporation 'hfg') needed by the water at 7 bar g to change it into steam is actually less than the heat energy required at atmospheric pressure. This is because the specific enthalpy of evaporation decreases as the steam pressure increases.

However, as the specific volume also decreases with increasing pressure, the amount of heat energy transferred in the same volume actually increases with steam pressure.

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Calculating steam requirements – m cp ΔT

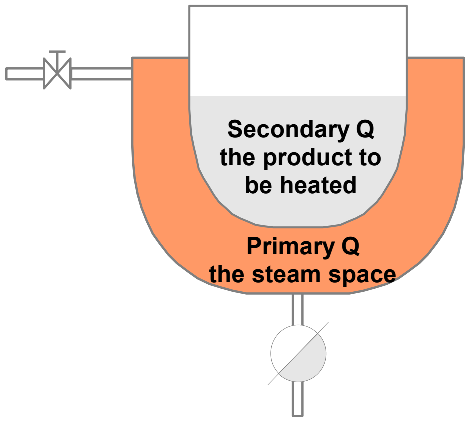
A process needs heat at

- the correct temperature and
- the correct rate of heat transfer

Heat is being generated in the boiler in the form of steam. This heat is being distributed by steam lines to the process. Steam pressure determines the temperature at which heat is supplied, as saturated steam temperature is directly proportional to pressure. We need a ΔT of minimum 15-30°C to have efficient heat transfer (rate of heat transfer).

Consider a heat exchange process. The primary side is the steam space, and the secondary side is the process. Steam is condensing on the primary side into water. It is changing phase into liquid and giving off its latent heat to the process. This is Primary Heat (Q).

Primary Q = m x h_{fg}



Where,
Primary Q = Quantity of heat energy released (in kcals)
m = Mass of steam releasing the heat (in kgs)
hfg = Specific enthalpy of evaporation of steam (in kcals/kg)

On the secondary side, this heat is being used for two things:

- 'heating up' heat - to increase the product temperature to the degree desired

- 'maintainance' heat - to maintain the product temperature as heat is lost by radiation, etc

Secondary Q = m x cp x ΔT

Where,
Secondary Q = Quantity of heat energy absorbed (in kcals)
m = Mass of the substance absorbing the heat (in kgs)
cp = Specific heat capacity of the substance (in kcals / kg °C)
ΔT = Temperature rise of the substance (in °C)

This equation is also modified and used to establish the amount of heat required to raise the temperature of a substance, for a range of different heat transfer processes.

The above equations are very important. As Heat energy is being transferred from the primary to the secondary side, in an ideal condition,

Primary Q = Secondary Q

And this is the equation to calculate the theoretical heat balance of the entire system.

Example 1. Calculate steam flow rate for an autoclave which is heating 10,000 bottles of 1 litre each to a temperature of 120°C in 30 minutes. Steam supply is at 3 kg/cm^{2}g.

Solution. What we are asking for is - what is the mass of steam that is supplied to the autoclave to heat these 10 bottles. This is 'm' on the primary side. First we will calculate the heat absorbed by the bottles (process), ie, secondary Q.

The formula

Secondary Q = m x cp x ΔT

Where,
Secondary Q = Quantity of heat absorbed by the bottles (in kcals)

m = Mass of water in the bottles which is absorbing the heat (in kgs)
= 10,000 bottles X 1lt = 10,000 lt = 10,000 kg

cp = Specific heat capacity of water (in kJ/kg °C) = 1 kcal/kg °C

ΔT = Temperature rise of water (°C) assuming ambient is 30°C
= 120°C – 30°C = 90°C

Gives,
Secondary Q = 10,000 kg x 1 kcal/kg °C x 90°C = 9,00,000 kcal

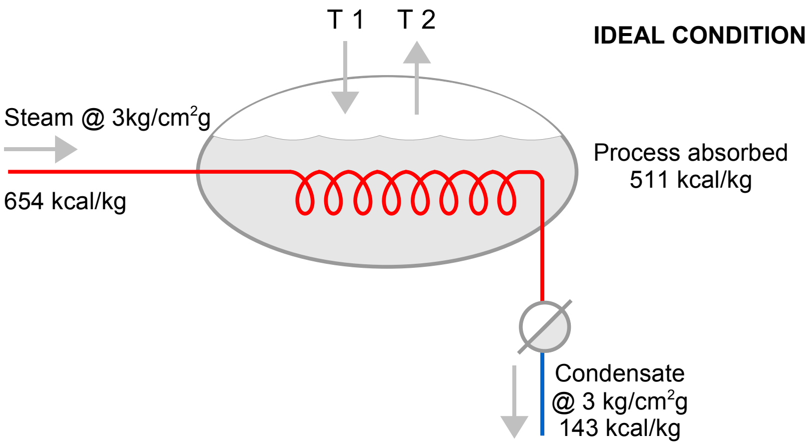
So, 9,00,000 kcal is the heat energy absorbed by this autoclave on the secondary (process) side in 30 minutes. Steam at 3 kg/cm^{2}g has 510 kcal/kg latent heat h_{fg} (from steam tables).

As **Sec Q = Pri Q**,

9,00,000 kcal = m x 510 Kcal/kg
m = 9,00,000 / 510 = 1765 kgs

1765 kgs is the steam required in 30 mins. So, steam flowrate is 1765 X 60/30 = **3530 kgs/hr** for this autoclave.

Suppose steam is supplied to a heat exchanger at 3 kg/cm^{2}g - hg 630 kcal/kg. Condensate is coming out of the traps at 3 kg/cm^{2}g h_{fg} 130 kcal/kg. Ideally, the product should absorb 511 kcal/kg. But, it doesnt. Heat gets absorbed by the heat transfer barriers and is also lost via radiation. So, the actual heat absorbed is less than 511 kcal/kg.



Let us understand what these heat transfer barriers are.

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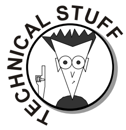
Calculating Heat transfer - U A ΔT

The general heat transfer equation

$$Q = U \times A \times \Delta T$$

Where:
Q = Heat transferred per unit time (kcals/hr)
U = Overall heat transfer coefficient (kcals/hr / m²°C)
A = Heat transfer area (m²)
ΔT = Temperature difference between the primary and secondary fluid (°C)



Q will be a mean heat transfer rate if ΔT is a mean temperature difference LMTD. The highest rate of heat transfer is at steam inlet as the temp diff is highest here, and the outlet has the lowest temp difference, therefore the lowest rate of heat transfer.

The heat transfer coefficient (U)
The heat transfer coefficient basically takes into account all the barriers to effective heat transfer. This can be the deposits of scale, condensate , air film, etc. It can be rust on the steel wall, or chemical reactions between the process and/or steam with the wall. It could be fluid flowrates, the physical nature of fluids, or the orientation of the heat transfer surface itself. All the above play a vital role in transferring heat to the medium, and are summed up in the heat transfer coefficient, U.

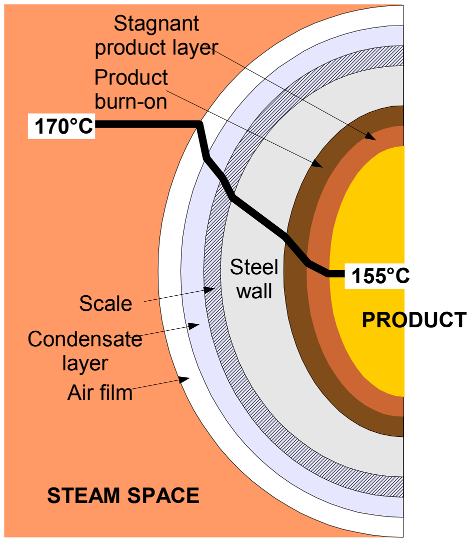


Fig. Barriers that reduce the rate of heat transfer: Metal of the pipe or jacketed pan; air, condensate and scale on the steam side; stagnant product and burn-on on the product side.

Typical Overall Heat transfer coefficient in kcal/hr m² °C

Heating Fluld	Heated Fluld	Convection Free	Convection Forced	Type of Fluld	Type of Apparatus
Liquid	Liquid	125-300	750-1500	Water	Heat exchanger betw een liquids
		25-50	75-200	Oil	
	Gas	5 --- 15	10-50	Water to air	Hot w ater radiators
	Boiling Liquid	500-1500	1000-2000	Water	Saline w ater cooler
		25-100	100-250	Oil	
Gas	Liquid	5—15	10-50	Air to w ater	Air cooled economisers
	Gas	3—10	10—30	Smoke to steam	Super heaters
	Boiling Liquid	5—15	10—50	Smoke to boiling w ater	Boilers
Condensing vapour	Liquid	250-1000	750-4000	Steam to w ater	Condenser feed w ater heaters, oil heaters
		50-150	100-300	Steam to oil	
	Gas	5 --- 10	10-50	Steam to air	Steam pipe in air air re—heaters
	Boiling liquids	1500-1200		Steam to w ater	Evaporation under vacuum
		250-7500		Steam to oil	

Air may be between 1 500 and 3 000 times more resistant to heat flow than steel. Condensate film may be between 100 and 150 times more resistant to heat transfer than a steel heating surface.

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Piping

1. Recommended Velocities

GENERALLY RECOMMENDED VELOCITIES FOR PIPING SYSTEMS	
SERVICE/ APPLICATION	**VELOCITY (m/s)**
Raw or treated water	1 - 5
Boiler feed pump suction	1
Boiler feed pump discharge	3
Saturated steam - HP	25 - 40
Saturated steam - LP	15 - 30
Superheated steam	30 - 50
Flash steam	5 - 15
Condensate	X
Condensate CRPS suction	
Condensate CRPS discharge	
Compressed air	
FD fan discharge	

2. Steam Pipe sizing

Line Sizing Considerations

Line sizing is based on either pressure drop per 100 m or velocity in m/s. Design parameters will sometime vary from plant to plant but, as a rule, allowable pressure drop is 0.115 kg/cm2g per 100 m for runs and less. For pipe runs over 100 m long, 0.03 kg/cm2g per 100 is acceptable. Velocity is normally held at 30 m/s for saturated steam.

In most processes, the warm up loads will be much higher than the running loads. The calculated steam consumption should have a factor of at least 25% extra for the purposes of line sizing.

We have the standard formula,

$$D = 1000 \times \sqrt{\frac{4 \times m \times V}{3600 \times \pi \times c}}$$

Where,
D = Line size in mm
m = Mass flowrate of steam in kg/h
V = Specific volume in m^3/kg
π = a constant 3.14
c = velocity m/s

Example - Saturated Steam line sizing

Find out the line size for mass flowrate of steam 3000 kg/h, at a working pressure of 10.5 kg/cm^{2}g.

We have the formula,

$$D = 1000 \times \sqrt{\frac{4 \times m \times V}{3600 \times \pi \times c}}$$

Our Working pressure is 10.5 kg/cm^2 g

From saturated steam tables, we go down to 10.5 kg/cm^2 g, and across for getting

Saturation temperature: 185.59 ^{0}C

Specific volume: 0.17 m^3/kg

Determination of line size at the three different velocities

Wet or flash steam : 15 - 25 m/s
Saturated steam : 25 - 40 m/s
Superheated steam : 40 + m/s

Condition 1

when velocity is is in the range of 25 m/s hence, actual line size will be

$$D = 1000 \times \sqrt{(4 \times 3000 \times 0.17) / (3600 \times 3.14 \times 25)}$$

$$= 84.96 \text{ mm say } 85 \text{ mm}$$

Condition 2

When velocity is 40 m/s, hence, actual line size should be

$$D = 1000 \times \sqrt{(4 \times 3000 \times 0.17) / (3600 \times 3.14 \times 40)}$$

$$= 67.17 \text{ mm}$$

From the above conditions, the right selection is **80 NB pipe.**

Cross checking for 80 NB (77.93 mm ID pipe), the velocity through it is:

$$c = \frac{(1000)^2 \times 4 \times m \times V}{3600 \times 3.14 \times D^2}$$

$$= \frac{(1000)^2 \times 4 \times 3000 \times 0.17}{3600 \times 3.14 \times (77.93)^2}$$

$$= 30 \text{ m/s}$$

Which is in the permissible range for saturated steam.

NOTE

For saturated steam permisible velocity range is 25 – 40 m/s.

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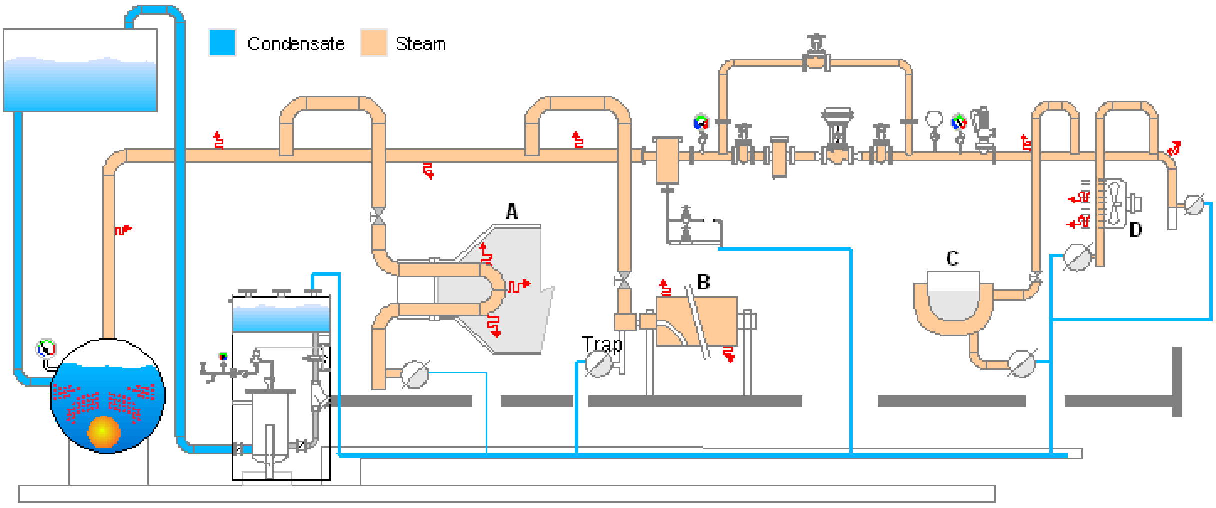
3. Condensate line sizing

First, lets understand the condensate loop from the drawing . Where is it coming from, how it is collected and where is it going?



Proceeding logically, we start with the boiler. It is supplying steam to the process. Process A and B are getting steam at a high pressure and processes C and D are supplied steam after a PRS at a lower pressure.

The steam transfers its heat energy to the process and condenses. The condensate flows from the drain of the process to a trap. The steam space of the plant (like the inside of the pressurized jacketed vessel) and the inside of the trap are at the same pressure. Therefore, the trap must be lower than the process so that condensate flows by gravity to the trap.

You don"t want to lose pressure so this line from drain to trap has to be sized correctly. Each process in the plant may be designed for differing pressures and condensate flow rates, so, the drain connection will not necessarily be the correct size.

Understanding start -up load

When a plant is cold and steam switched on all the processes are at ambient temperature. So, the material to be heated needs a lot of steam at start -up to come to its working temperature or running load. More steam translates to more condensate and the lines to the trap must be sized properly to be able to cope as, the condensing rate of steam is very high. This is start-up load.

Also, the lines had air in them before start -up. The incoming rush of steam carries this air to the trap as well, again loading the trap.

So, pipe sizing and subsequent trap sizing is done based on steam load multiplied by a factor (2 or 3) times running load plus based on process equipment and experience. In some plants the equipment is not used all together , but in phases, depending on process requirement and this needs to be taken into account while sizing return lines.

We also take into account a frictional resistance of 1.4 m bar / meter.

Steel Pipe (mm)	Condensate at Running Load (Kg/hr)
15 mm	1.0
20mm	250
25mm	470
32mm	1020
40mm	1550
50mm	3000
65mm	6050
80mm	9350
100mm	19000

This table has already taken into consideration start-up loads.

Why it is important to size pipes correctly?

Pipe costs go up hugely (disproportionately) as the size increases. So we want to use pipes that can accommodate capacity comfortably, and are not over sized for economic reasons.

Now we move forward in our condensate loop- from the trap outlet to the discharge lines (blue lines).

What is our line carrying now? Besides condensate and air & other gases (on start-up) these discharge lines now also contain flash steam. This is because discharge lines are at (ambient / lower pressure) than the pressurized trap body.

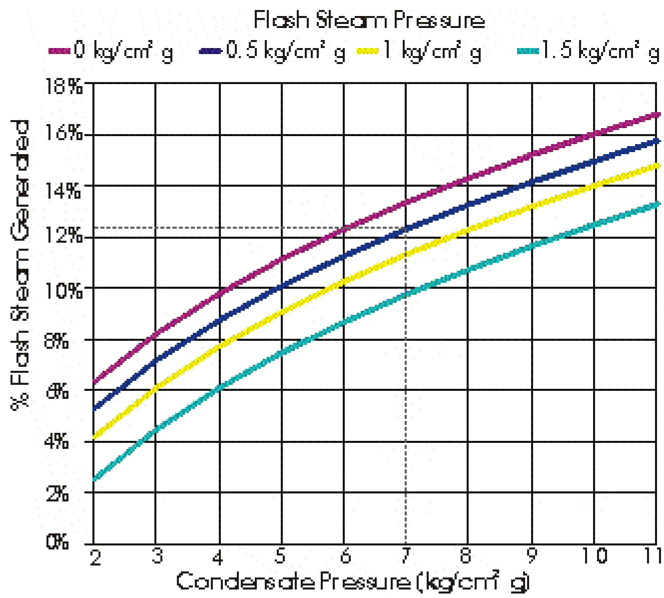
Therefore, in an ideal condition we must discharge first to a flash separator (to recover energy from flash steam).

In the absence of a flash separator, we must, in any case recover the condensate, so the trap discharge lines should go to the receiver of a CRPS or directly to the boiler feedwater tank / deaerator.

At this end of the trap, we again have to consider start up and running load conditions. When the plant starts up the condensate is relatively cool (as the steam discharges much more heat into a cold process) and there will be little or no flash steam (flashing occurs closer to boiling temperature).

But, condensing rate is very high and so is the air content in the pipes. So, the pipes have to be at least the same size as the trap inlet.

As the plant comes to a normal running load, the condensate flow decreases to average running load conditions. But, it's temperature is much higher than at start-up. Flash steam is created as the hot condensate flows from trap at high pressure to discharge lines at low pressures.



Example 4.1. What will be the percentage of flash steam from condensate at 3 kg/cm^{2}g when released to atmosphere ?
From the graph above, we go up on the 3 kg/cm^{2}g line to the 0 kg/cm^{2}g red line. This corresponds to a flash steam percentage of about 8.5%.

Example 4.2. What will be the percentage of flash steam from condensate at 7 kg/cm^{2}g when released to discharge lines with a pressure of 0.5 kg/cm^{2}g ?
Again, from the graph above, we go up on the 7 kg/cm^{2}g line till we touch the blue 0.5 kg/cm^{2}g line. Here the flash percentage is 12.5%.

A Flash steam percentage of 8.5% in example 1 and 12.5% in example 2 seems very trivial. Why are we doing this exercise of finding flash steam percentage ?

This is because the volume of flash steam can be up to 400 times the volume of condensate. Especially, if the pressure difference between the trap body and the condensate return line is high and the condensate is hot. Therefore, it is clear - size your condensate lines on the flash (steam) volume and not on condensate volume.

Let us taken an example again.

Example 4.3.
The jacketed vessel has a condensate load of 1000 kg/h. Pressure at trap is 4 kg/cm^{2}g. The discharge line is at atmospheric pressure. Calculate volume of flash generated.

Looking at the graph we see that 10% of the condensate will flash off. Therefore, 1 kg of condensate discharged via the trap turns into 0.9 kg of water and 0.1 kg of steam in the discharge line.

To calculate volume, of steam, we look at the steam tables and see that specific volume of saturated steam at 0 kg/cm^{2}g is 1.66 m^3 / kg.

Volume of water	= 900 liters = 0.9 m^3/h
Volume of steam	= 100 kg/h X 1.66 m^3/ kg
	= 166 m^3/ hr
Total Volume	= 166.9 m^3/h
% volume of water	= 0.9/166.9 X 100 = 0.54 %
% volume of steam	= 166 /166.9 X 100 = 99.46 %

As we can see, the flash steam is taking up all the volume of the pipe. (In fact, the effect of releasing this huge amount of steam into a small space like the discharge line will be increased pressure - more than atmospheric. Increasing pressure will reduce the pressure differential between trap body and line which will reduce flash steam generated).

Condensate collects at the bottom of the pipe. It grows in thickness and moves at a lower velocity than flash steam. The total mass flow through the line is calculate by adding these two different rates of flow.

So, we have concluded that the sizing of condensate discharge lines is to be done based on the mass flow rate of flash steam.

In our example,
Mass flow rate of flash steam = 0.1 x 1000kg/h = 100kg/h.

Using the pipe capacity chart on page 4.15, we see the size of pipe to take 100 kg/h at discharge line pressure of 0 bar g is 65mm.

(Condensate flowing at the bottom of pipe can cause water hammer so flash steam velocity should be lower than 15 m/s).

Other factors to take into account white pipe sizing return lines:

Flash Separators & Deaerators in the system translates to a higher pressure in the condensate discharge lines. The build up of pressure in the line is because of flash steam in the flash separators and deaerator. For proper functioning of traps be careful to maintain a differential pressure between inlet (process pressure) and outlet (discharge).

It is only if the pipe is undersized for the flow of flash steam at return line pressure, will the back pressure rise so much as to hinder trap operations.

Each section of the pipe must be correctly sized to carry condensate loads and flash steam at acceptable velocities. This will mean little extra pressure is involved (?) and that the discharge from a high pressure trap will not interfere with that from a low pressure trap.

Back pressure
Back pressure on a trap reduces trap capacities, although this becomes noticeable at fairly low upstream pressures. More importantly , it makes air venting and condensate removal tougher at start-up , which can lead to erratic control or waterhammer with temperature controlled equipment.

Causes of backpressure:

- The pressure at the end of the line- either atmospheric or the pressure of the vessel / receiver into which the condensate discharges.

- A trap at low level has to work against "hydrostatic head" to push condensate to an overhead return line. A lift of 1m =0.1 bar increase in back pressure. Similarly, a lift of 5m is 0.5 Bar head the trap has to discharge at.

- Frictional resistance of pipe , bends, etc to the flue flow of condensate , steam and air.

Points to be followed with common return lines.

A plant with carefully sized lines has little to worry about, as far as a common return line for traps goes. A little thought can go a long way to maintain a return line.

(a) The actual connections , for eg, can be swept tees instead of square tees. (This will avoid erosion from high velocity flash steam and blasts of water from blast discharge trap- inverted bucket or thermodynamic trap).

(b) Condensate must not be discharged into a flooded return main. A pumped condensate return main often follows the same route as a steam line and sometimes , this proximity cures plant crew and the discharge from mains drain traps are simply connected to it.

Cool the condensate, or else..

Condensate from mains traps are at saturation temperature. At this temperature, the armount of flash steam released into the low pressure main has huge volume.

This pushes the water already present violently out of the way. Bubbles of flash steam go along the pipe, contact cooler condensate or pipe walls and collapse . This leads to waterhammer.

Therefore, all condensate must be given a chance to cool a little and then released into the return lines.

- Use a generous condensate collection pocket (like in a thermostatic balanced pressure trap) which holds back condensate till it is sub - cooled.
- A continuous discharge float trap also does the job as a steady flow can be more easily absorbed into a flooded line.
- An un- lagged cooling leg of 2-3 m upstream of the trap is another option. It can provide storage volume for condensate so that it can cool down sufficiently before discharge.
- Collecting all condensate from traps into a receiver and then pumping it back is the ideal solutions. It must be remembered that these are only compromises and a gravity fall from trap to receiver is always the aim.

Good Piping Practice Prevents Water Hammer in Steam Systems

One of the most common complaints against steam heat is that a system sometimes develops a hammer-like noise commonly referred to as water hammer. It can be very annoying. However, it may indicate a condition which could produce serious consequences including damaged vents, traps, regulators and piping.

There are two types of water hammer that can occur in steam systems.

- One type is usually caused by the accumulation of condensate (water) trapped in a portion of horizontal steam piping. The velocity of the steam flowing over the condensate causes ripples in the water. Turbulence builds up until the water forms a solid mass, or slug, filling the pipe. This slug of condensate can travel at the speed of the steam and will strike the first elbow in its path with a force comparable to a hammer blow. In fact, the force can be great enough to break the back of the elbow. Steam flowing in a system at 10,000 feet per minute is traveling more than 100 miles per hour. The slug of condensate is carried along by the steam flow.

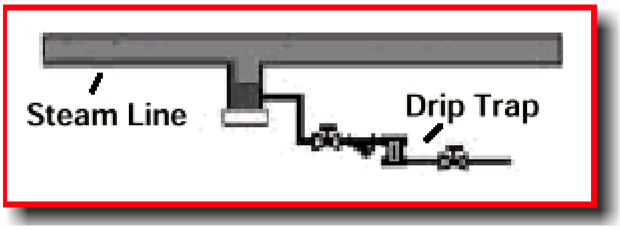
- The second type of water hammer is actually cavitation. This is caused by a steam bubble forming or being pushed into a pipe completely filled with water. As the trapped steam bubble looses its latent heat, the bubble implodes, the wall of water comes back together and the force created can be severe. This condition can crush float balls and destroy thermostatic elements in steam traps. Cavitation is the type of water hammer that usually occurs in wet return lines or pump discharge piping.

A properly piped steam system should not produce water hammer of either type

Correct piping installation guide

Water hammer in steam lines is normally caused by the accumulation of condensate.
Important installation details to prevent water hammer in steam lines include the following:

- Steam pipes must be pitched away from the boiler toward a drip trap station. Drip trap stations must be installed ahead of any risers, at the end of the main and every 300 to 500 feet along the steam piping.



- Drip traps must be installed ahead of all steam regulator valves to prevent the accumulation of condensate when the valve is in a closed position.

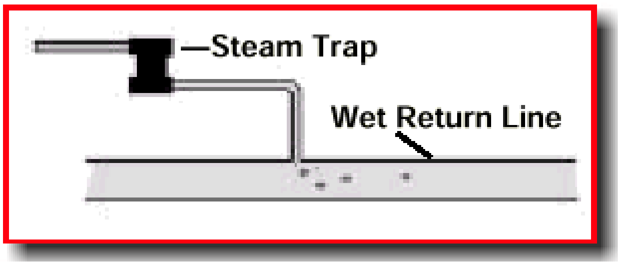
- "Y" Strainers installed in steam lines should have the screen and dirt pocket mounted horizontally to prevent condensate from collecting in the screen area and being carried along in slugs when steam flow occurs.

- All equipment using a modulating steam regulator on the steam supply must provide gravity condensate drainage from the steam traps. Lifts in the return line must be avoided.

Water Hammer in Condensate Return Lines

In most installations, water hammer in condensate return lines is caused by steam pockets forming and imploding. Frequently, the cause is a rise in the discharge line from a trap or a high pressure trap discharging into a low temperature wet return line.

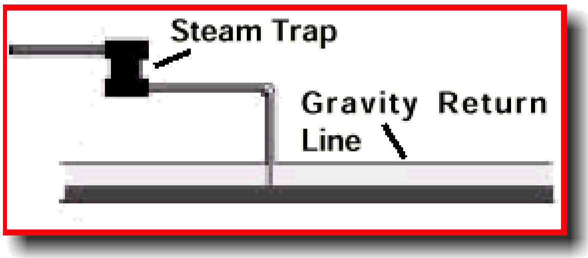
A lift in the return line after the trap will cause water hammer because the temperature of the condensate leaving the trap exceeds 100°C. The high temperature condensate flashes, causing steam bubbles to form. As these steam bubbles are pushed into colder condensate in the return piping, they implode and cause water hammer. The water hammer will normally be worse during start up due to the cold condensate lying in the return piping. As the temperature of the return line increases above 100°C the water hammer often stops. Many industrial applications install lifts to avoid installing additional condensate return systems. When installing a lift, the most commonly used trap is an Inverted Bucket Trap since the open bucket design tolerates moderate water hammer check valve helps isolate the trap from the water hammer forces and prevents back flow of condensate when the steam supply is secured.



When a trap discharges into a wet return line, flashing will occur. Again, these steam bubbles implode causing water hammer. This condition is often found where a high pressure drip trap is connected into a pumped return line with lower temperature condensate. Old steam guides showed the use of a diffuser pipe to break up the high temperature condensate to reduce the size of steam bubbles that occur. The guide showed welding a pipe tangentially in the return line and drilling 1/8 inch holes at least 1 inch apart. Other methods include using a heat exchanger to blend the two temperatures or the use of fin tube radiation to cool the trap discharge.

The most common method used is to install a flash tank on the drip trap discharge allowing the condensate to flash to 100°C and then pumping the cooled condensate into the common return line.

Important installation details to prevent this type of water hammer are listed below.



- Whenever possible, use gravity return lines. Properly sized return lines allow condensate to flow in the bottom portion of the pipe and flash steam to flow in the top portion of the pipe. The top portion also allows efficient air venting during start up of the system.
- Water hammer can occur in pumped discharge lines. A condensate unit is pumping condensate near saturation temperature to an overhead horizontal run and then drops down into a vented boiler feed tank. A negative pressure develops in the horizontal pipe due to the piping drop into the vented receiver. When the pressure falls below saturation temperature, water hammer can occur. A 4 metre vertical drop can allow 88°C condensate to flash and cause water hammer. This condition can be remedied by either creating a back pressure at the low point or by installing a swing check valve open to atmosphere in the horizontal pipe. The swing check will open, allowing air to enter and the vertical water column to drain away.
- This condition can also occur in the boiler feed pump discharge line from a deaerator or pre-heat unit. In many installations, the discharge lines run overhead, a check valve or regulator valve is installed near the boiler, and a check valve is installed at the pump discharge. If the check valve at the pump discharge does not hold tight, condensate drains back to the boiler feed unit, allowing the condensate in the discharge to flash. A steam pocket forms at the high point. The result is water hammer when the pump starts. This can be corrected by replacing the check valve.

Lower condensing pressures at the point of use tend to save energy. and also reduce the amount of flash steam generated when condensate from drain traps is discharging into vented condensate collecting tanks.

It is worth noting that if condensate is continuously dumped to waste, perhaps because of the risk of contamination, less energy will be lost if the condensing pressure is lower.

Returning Condensate

Returning condensate. In determining how the condensate is going to be returned there are basically two considerations:

1. Can the condensate return headers be run below (on a lower floor) the coil outlet trap for gravity drainage and

2. Does the condensate have to be lifted to overhead condensate return piping?

If it is possible to run the return header below the trap outlet this would be the more practical method in regard to the flow of condensate. Regarding other concerns, dropping the trap discharge piping down normally necessitates floor penetrations.

With the condensate dropping down from the trap discharge to the return header the designer doesn't have to be concerned with the lack of lift pressure. However, if there is no alternative but to return the condensate overhead then the designer is going to require a Steamline CRPS. condensate pump. If it is at all possible combine the flow of two or more traps by routing a collection header and running it to the condensate pump.

The steam-powered condensate pump is an equipment item that allows condensate or other liquids to accumulate, by gravity, in the pump chamber under low pressure. The condensate then gets pumped to its destination by steam, air or inert gas pressure.

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Lifting condensate to a higher plane

There are forces the CRPS has to work against, to lift condensate. First, a few terms.

- *Head.* The potential energy of condensate at a given point is called head.
- *Pressure Head.* The pressure the condensate in a pipe exerts at the point.
- *Static head.* This is the vertical height of condensate from the reference point.

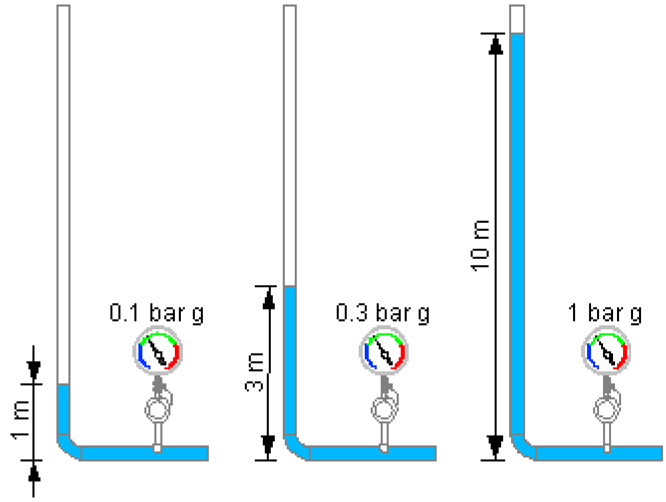
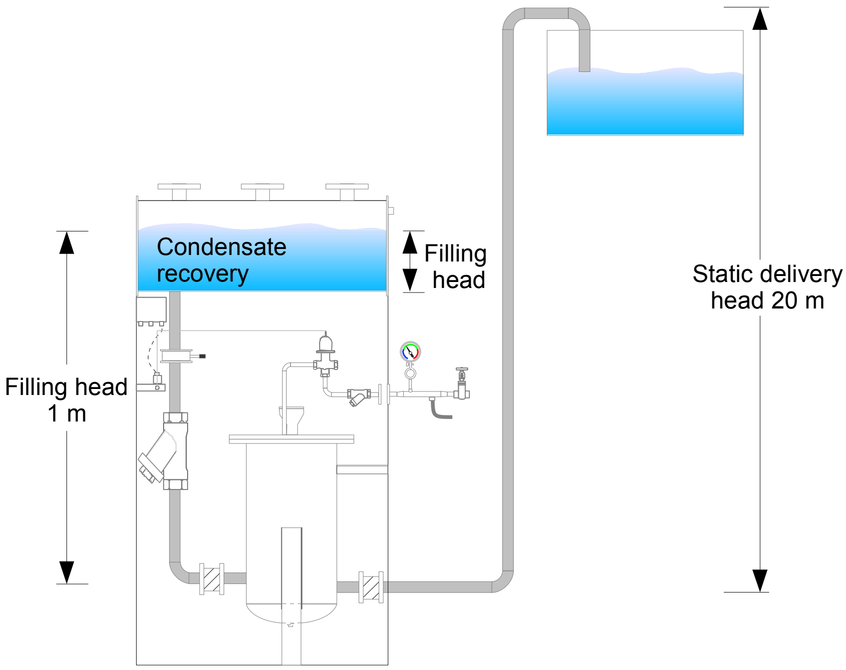


Fig. A static head of 10 metres water column = pressure head of 1 bar g or 1 kg/cm²g.

Below, the CRPS is required to pump to a receiver against a static Delivery Head of 20 metres, or, 2 bar g. It is filling from a head of 1 metre, or, 0.1 bar g. This head of water above inlet connection provides the energy to fill the pump chamber during the filling cycle.



The Steamline CRPS has to work against the delivery head of 20 m. This is because the Suction head pressure is not present in the pump body during pumping and has no effect on the delivery head against which the pump has to operate.

Friction head loss. The energy lost in just trying to move the condensate through the pipe.
We have friction losses through the pipe and the various pipe fittings. So, we take an extra "equivalent length" of pipe fittings. This is added to the actual pipe length, to give total equivalent length.

Total equivalent length = Actual length of pipe + equivalent lenth of fittings

In practice, pipe fittings are not more than more than an additional 10% of the actual pipe length.

Total equivalent length = Actual length + 10%

Table. Condensate Flowrates for dry closed returns

Condensate flowrate (kg/hr) for dry closed returns (Pr. Drop = 0.25 barg / 100m)

Steam Pr. Kg/cm2g		0.35	1.00	2.00	2.00	3.50	4.00	7.00	7.00	7.00	10.50	10.50	10.00
Return Pr. Kg/cm2g		0.00	0.00	0.00	0.35	0.00	0.70	0.00	1.35	2.00	0.00	0.70	3.50
Diff. Pr. Kg/cm2g		0.35	1.00	2.00	1.65	3.50	3.30	7.00	5.65	5.00	10.50	9.80	6.50
Flash steam %		1.7	3.9	6.5	4.9	11.2	7.4	13.3	8.7	7.1	16.4	13.9	7.9
Linesize		Flow possible in Kg/hr											
½	15	646	273	164	286	113	235	80	273	425	65	125	542
¾	20	1139	480	289	504	213	415	141	481	750	115	220	956
1	25	1847	779	469	817	397	673	229	780	1216	186	357	1549
1 ½	40	4352	1837	1106	1925	808	1587	540	1839	2865	438	841	3652
2	50	7172	3026	1823	3172	1270	2615	890	3031	4721	723	1386	6017
2 ½	65	10235	4319	2602	4528	1860	3732	1270	4325	6738	1031	1977	8587
3	80	15803	6669	4018	6991	2858	5762	1962	6678	10404	1593	3054	13259
4	100	27211	11483	6918	12037	5399	9922	3378	11495	17914	2742	5258	22830
5	125	42765	18046	10872	18918	7781	15593	5309	18072	28154	4310	8263	35880
6	150	61757	26061	15701	27320	11252	22518	7667	26098	40658	6225	11933	51814
8	200	106938	45127	27187	47307	19510	38993	13276	45191	70403	10779	20663	89721

4. Air sizing

Volume of compressed air carried by nominal bore pipes at given velocities

Velocity m/s	Volume of Air in m^3/h through pipes of Nominal Bore											
	1/2"	3/4"	1"	1 1/4"	1 ½"	2"	2 1/2"	3"	4"	5"	6"	8"
3.05	1.385	3.12	5.55	8.69	12.49	22.23	34.79	50.06	88.75	138.98	200.25	354.67
3.65	1.663	3.75	6.65	10.4	15	26.64	41.75	60.07	106.74	166.31	239.28	425.95
4.27	1.935	4.38	7.77	12.15	17.48	31.06	48.53	70.09	124.56	193.46	280.01	497.22
4.88	2.223	5.01	8.88	13.86	20.02	35.47	55.49	80.1	142.21	222.31	320.73	568.5
5.49	2.495	5.63	10	15.61	22.57	39.88	62.45	90.11	160.2	249.46	359.76	639.77
6.1	2.766	6.24	11.1	17.31	24.95	44.46	69.41	99.95	178.19	278.31	400.49	711.04
6.71	3.055	6.87	12.2	19.18	27.49	48.87	76.37	109.97	195.16	305.46	439.52	782.32
7.32	3.326	7.5	13.32	20.87	30.04	53.29	83.15	119.98	213.82	332.61	480.25	851.89
7.93	3.598	8.13	14.42	22.57	32.41	57.7	90.28	129.82	230.79	361.46	519.28	923.17
8.54	3.886	8.76	15.54	24.27	34.96	62.11	97.24	140	249.46	388.61	560.01	996.14
9.15	4.158	9.37	16.65	25.96	37.5	66.69	104.2	150.01	266.43	417.46	600.74	1067.41

5. Pressure drop calculations

The study of Pressure Drop calculations is beyond the scope of this paper, but if you would like to know more, please email us at info@steamline.com and we will be happy to send you detailed information on the subject..

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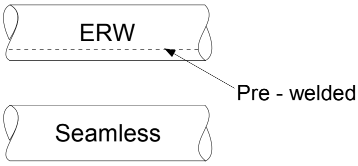
6. Considerations in steam piping

6A. Pipe

Pipe used for steam or condensate is generally of two types, ERW (Electric Resistance Weld) or Seamless. Generally, ERW class C is used for condensate, and seamless Sch40 pipes are used for steam applications.

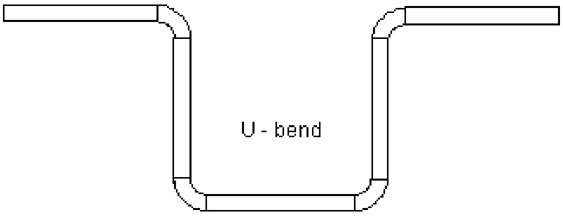


6B. Flanges

Pipe Size	Class	NB	FLANGE O.D	PCD	Dia .OF BORE	FLANGE THICKNESS + RAISED FACE	DIA OF HOLE	NO. OF BOLTS	Ideal Bolt Dia.	Ideal Bolt Length	Steamline Bolt in inch Dia	– Length	Steanline Bolt Metric dia	- Length	Gasket I.D. x O.D.
1/2"	# 150	15	88.9	60.3	22.5	11	15.8	4	12.5	45	1/2"	2"	M 12	50	22.5 x 45
	# 300	15	95.2	66.6	22.5	11	15.8	4	12.5	50.8	1/2"	2"	M 12	50	22 x 50
	# 600	15	95.2	66.6	22.5	14.2	15.8	4	12.5	50	1/2"	2"	M 12	50	22 x 50
3/4"	# 150	20	98.4	69.8	28	12.7	15.8	4	12.5	50	1/2"	2"	M 12	50	28 x 50
	# 300	20	117.4	82.5	28	15.7	19	4	16	63	5/8"	2-1/4"	M 16	65	28 x 63
	# 600	20	117.4	82.5	28	15.7	19	4	16	63	5/8"	2-1/4"	M 16	65	22 x 50
1"	# 150	25	107.9	79.3	35	14.2	15.8	4	12.5	50	1/2"	2"	M 12	50	35 x 63
	# 300	25	123.8	88.9	35	17.5	19	4	16	63	5/8"	2-1/4"	M 16	65	35 x 70
	# 600	25	123.8	88.9	35	17.5	19	4	16	63	5/8"	2-1/4"	M 16	65	35 x 70
1-1/2"	# 150	40	127	98.4	49	17.5	15.8	4	12.5	55	1/2"	2-1/2"	M12	65	49 x 80
	# 300	40	155.5	114.3	49	20.5	22	4	19	77	3/4"	3-1/4"	M 18	85	49 x 92
	# 600	40	155.5	114.3	49	22.2	22	4	19	80	3/4"	3-1/4"	M 18	85	49 x 92
2"	# 150	50	152.4	120.6	62	19	19	4	15.8	70	5/8"	3"	M 16	75	60 x 100
	# 300	50	165.1	127	62	22.3	19	8	16	77	5/8"	3"	M 16	75	62 x 108
	# 600	50	165.1	127	62	25.4	19	8	16	85	5/8"	3-3/4"	M 16	100	62 x 108
3"	# 150	80	190.5	152.4	91	23.8	19	4	15.8	80	5/8"	3"	M 15	75	90 x 130
	# 300	80	209.5	168.2	91	28.4	22	8	19	89	3/4"	3-3/4"	M 18	100	90 x 145
	# 600	80	209.5	168.2	91	31.7	22	8	19	90	3/4"	3-3/4"	M 18	100	90 x 145
4"	# 150	100	228.6	190.5	115	23.8	19	8	15.8	80	5/8"	3"	M 15	75	115 x 170
	# 300	100	254	200	115	31.7	22	8	19	90	3/4"	3-3/4"	M 18	100	115 x 180
	# 600	100	273	215.9	115	38.1	25.4	8	22	109	3/4"	3-3/4"	M 18	100	115 x 180
6"	# 150	150	279.4	241.3	170	25.4	22	8	19	85	3/4"	3-1/4"	M 18	85	170 x 216
	# 300	150	317.5	269	170	36.5	22	12	19	106	3/4"	3-3/4"	M 18	100	170 x 245
	# 600	150	355.6	292.1	170	47.6	28.5	12	25.4	135					
8"	# 150	200	342	298	222	28.4	22	8	19	90	3/4"	3-3/4"	M 18	100	220 x 275
	# 300	200	381	330	222	41	25.4	12	22	120	3/4"	3-3/4"	M 18	100	222 x 305
	# 600	200	419.2	349.2	222	55.5	31.7	12	28	150					
10"	# 150	250	406	361	275	30.2	25.4	12	22	95	3/4"	3-3/4"	M 18	100	275 x 335
	# 300	250	444	387	275	47.7	28	16	25.4	135					275 x 360
	# 600	250	508	431.8	275	63.5	34.9	16	32	165					
12"	# 150	300	482	431	325	31.7	25.4	12	22	100					325 x 405
	# 300	300	520.7	450.8	325	50.8	32	16	28	145					325 x 420
	# 600	300	558.8	488.9	327	66.7	34.9	20	32	175					
	Tab E	300	457.2	406.4	325	25.4	22.2	12	20	82	3/4"	3-1/4"	M 18	85	325 x 380
14"	# 150	350	533	476.2	358	35	28	12	25.4	110					358 x 448
	# 300	350	584	514	358	53.8	32	20	28	145					351 x 480
	# 600	350	603.2	527	359	69.8	38.1	20	34	182					

6C. U-bends

When we cycle the pressures in the boiler because of variations in load (the number of machines using steam), the temperature fluctuates. When the temperature reduces, the steam condenses and becomes wet steam. U-bends help by trapping condensate to prevent water hammer.



Also, the varying temperatures in the steam pipes expand or contract the lines. U-bends help absorb this variation in pipe lengths.

Unfortunately, U-bends also reduce pressure. So, we have to make an intelligent compromise between pressure and dryness.

Piping is a subject on its own, and will be covered in more detail in level 2.

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