Bangin' in the Boiler Plant

by Wayne Kirsner, P.E.

"Each time [the feedwater] pump goes On or Off, it makes a terrible bang, and pipes shakes like creazy... Anyone have a[n] idea how to eliminate (or decrease) water hammering in the steam boiler feeding line...?" Posted on May 22, 2002 at www.boilerroom.com/forum.html

If there's water hammer when a condensate or feedwater pump starts—especially if the fluid being pumped is at or near its saturation temperature—the cause of the hammering is probably "column closure water hammer"[1].

The term "column-closure water hammer" is little known to HVAC engineers and power plant operators because, unlike condensation–induced water hammer, it generally doesn't lead to catastrophic steam ruptures and operator deaths and, thus, has remained largely anonymous in the non-nuclear engineering world. Like its potentially more potent cousin, however, column closure water hammer is a common mechanism of water hammer in steam systems.

In this article, I explain the mechanism of column closure water hammer, and, then, examine the most common causes of water hammer in boiler plants due to both column-closure and condensation-induced water hammer .

Column Closure Water hammer--In a Simple Hydronic System

Figure 1 shows a simple example of an impending column closure water hammer. A pumped return raises 80°F water to a receiver at a height over 33' above the level of the atmospheric tank from which the water is drawn. Prior to the event, the pump has been off and the valve adjacent the upper tank closed (perhaps to prevent the tank from draining back because the check valve at the pump leaks). Problem is, because atmospheric pressure cannot sustain a column of water over 33' high, the leaking check valve allows water to drain from the vertical column until it's less than 33 feet high. This leaves a "void" between the face of the valve and the top of the water column. (Actually the void is a partial vacuum filled with water vapor at the vapor pressure of the 80oF water—about ½ psia).

When the pump is re-started, the column of water will be accelerated into the partial vacuum. Given a long enough void, the column of water can accelerate to its terminal velocity of perhaps 5 to 10 fps before striking the valve face. Upon collision with the valve, the kinetic energy of motion is converted into "potential" energy by compressing the liquid. Specifically, the first "lamina" of water--an imaginary thin disk that makes up the face of the water column—comes to an abrupt halt and compresses like a spring. The same happens with each succeeding lamina of water which crashes into the rear of the compressed lamina in front of it. The appearance, if you could see the laminas' movement in slow motion, is of a compression wave traveling back toward the pump at the speed of sound in water. Because water is stiff stuff, a slight compression causes a huge pressure rise.

A bound on the pressure rise due to water hammer can easily be estimated if the velocity of the fluid is known. An upper bound on the "overpressure" due to water hammer —that is the pressure *above* the ambient pressure in the pipe-- is about 60 psi per 1 foot/sec of water column velocity. (This figure is derived at the end of the article). Thus, if the pump could accelerate the water column to 10 fps, and the void was long enough for the water column to reach its terminal velocity, the overpressure exerted on the valve face and surrounding pipe upon collision could be as high as:

$$10 \text{ fps} * 60 \text{ psi/ fps} = 600 \text{ psi.}$$

For a system with dissolved air in it, however, the above figure is high by a factor of at least 2 to 4. Air dissolved in water will evolve into the partial vacuum formed by the water column separation and act to cushion the water hammer collision[2]. Hence, column closure water hammer will not generally be severe in hydronic systems, UNLESS THEY ARE DEAERATED.

Column-Closure Water hammer -- In a Condensate Return System

In a hot condensate system, water hammer can be severe for two reasons:

First, steam from whence condensate is generated, is often produced from deareated feedwater in which dissolved air has been reduced by a factor of from 20 to 1000 depending on the deareator rating. Consequently, in a condensate system, there can be few non-condensables to evolve into a void to cushion a water hammer.

Secondly, the vapor pressure of condensate, if the condensate is at or near its saturation temperature, allows any void formed in a pipe rise the potential to grow to the entire length of the rise. To see what I mean, let's compare two examples:

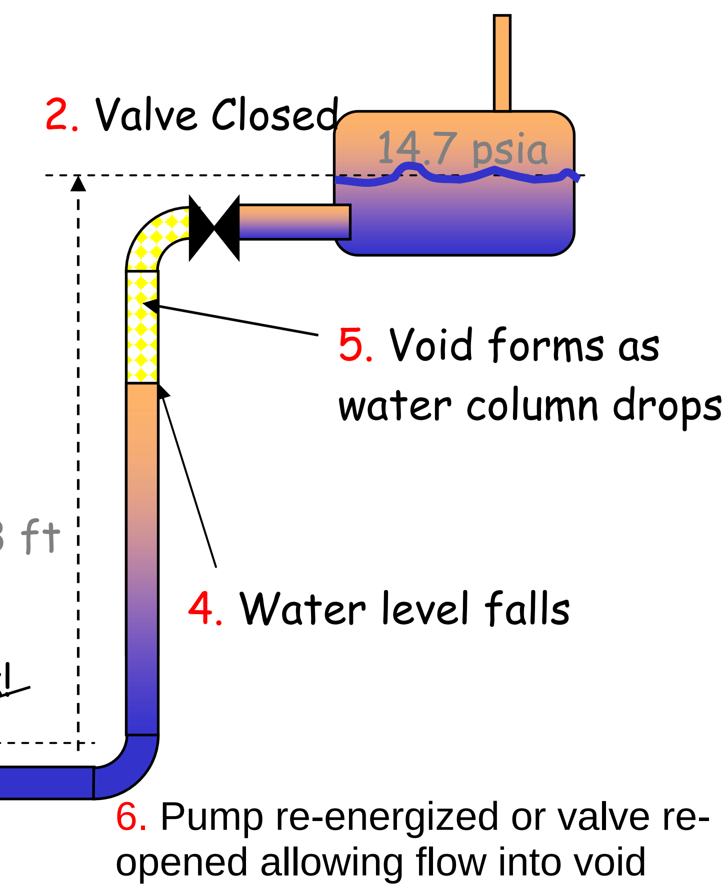
For 80°F cold water which has a vapor pressure of only ½ psia, the atmosphere will support a column of water approximately 33' high because the pressure in the partial vacuum created above the receding water column is almost negligible—only ½ psia.

Hot 200°F condensate, on the other hand, has a vapor pressure of nearly 12 psia. Therefore, the net difference between atmospheric pressure pushing on the low side of the column minus the vapor pressure in the void exerting pressure on the top of the column—2.7 psid-- will allow a void to grow to almost the entire length of the pipe rise less 6.5 feet, calculated like so:

$$h = (14.7 - 12.0) \text{ psi} * 2.4 \text{ '/psi} = 6.5 \text{ ft}$$

Thus, if there's a leak in the check valve (or anything else below the water column), the 200oF water column blocked closed at its top can fall to just 6.5' above the level of water in the lower tank. This results in a longer void that gives a water column more distance in which to accelerate if the pump (or anything else[3]) starts it moving.

What's more, the non-negligible vapor pressure obviates the requirement that the condensate pipe rise be anywhere near 33 feet as was the case for relatively cool water. In the case of 200°F or hotter water, the condensate pipe rise need be less than 7 feet above the level of the lower tank level to facilitate a column closure water hammer-- if there were a leak below the condensate column.



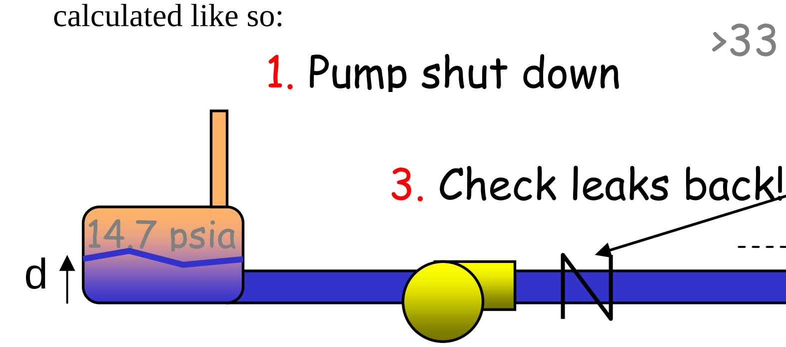


Figure 1. Void formed when a pump is shut down in a valved closed hydronic system lifting over 33 feet. The red numbers show the sequence of events.

[1] Also known as "fluid column separation and rejoining"

[2] Unless the time for air pressure rise is less than 2L/c in which case the air has no impact per F. J. Moody correspondence.

[3] For example, if the upper valve were opened suddenly, a column of water would be driven into the void by the atmospheric pressure in the upper tank.

Returning to the previous example where a pump is able to accelerate water to 10 fps thru a void formed by column separation, and assuming the fluid contains little or no dissolved gas --a 600 psi overpressure is possible. Is that overpressure great enough to break an isolation valve like that represented in Figure 1, or damage other piping components?

600 psi overpressure added to the system pressure would generally not be high enough to break any steel piping components, but it could break a bellows-type expansion joint. It could even break a cast iron valve if the "dynamic load factor" of the collision approached two. (See the sidebar for a description of dynamic load factor).

Dynamic Load factor[4]

The "dynamic load factor" or "impact factor", as it's sometimes known, multiplies the impact felt by the object struck in a collision by a factor of from less than one to up to two. The value of the dynamic load factor depends on the response time of the object struck (i.e. the piping system's natural frequency) compared to the duration of the collision).[5] The duration of the collision depends on the length of the water column being decelerated. The longer the column, the more likely the dynamic load factor will exceed one and approach two. For example, for the bounding case of a very flexible piping system (natural frequency = 4 cps), the dynamic load factor would exceed 1.0 if the run of pipe from tank to valve was over 90 feet long[6] and approach 2.0 if the run of pipe were 270 feet long. At the other extreme, a very stiff piping run with a natural frequency of 40 cps, would require at least a 9 foot long column of water for the dynamic load factor to exceed 1.0, and a column of water greater than 25 feet long to achieve a load factor of 2.0.

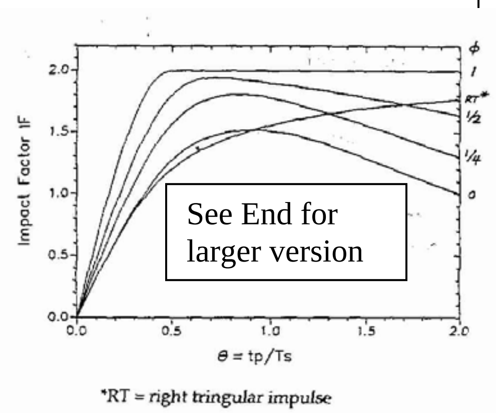


Figure 3.1-5 Impact factor as a function of t_p/T_s and shape of impulse

[4] Reference 2.
[5] The shape of the impulse also matters, but for long columns of water rising in a pipe, the pulse may be considered square.
[6] For "dynamic load factor" to be as great as 2.0, the water column pulse duration, 2L/c, must exceeded1/6 the natural period, Ts, of the system per Wiggert. For a natural freq. of 4 cps, i.e. Ts= .25 sec, c=4300, then L>(c/2) (.25/6) or L > 90 ft.

Accumulation of Flash Steam

To have a column-closure water hammer in a condensate system, it's not necessary to have a leaking check valve or closed isolation valve. If condensate is close to its saturation temperature, all that's needed, besides a rise in the piping, is a section of pipe in which flash steam can accumulate.

Figure 3 depicts a pumped condensate return system where flash steam is trapped in a high point which was inadvertently designed into the system. Raising condensate which is at or near its saturation temperature—a common condition if traps are blowing steam into condensate lines-- can drop condensate pressure below its saturation pressure thereby releasing flash steam. The steam bubbles will rise to a local highpoint and collect there forming a steam pocket. **Figure 3** depicts such a situation after the condensate pump has cycled off.

When the pump cycles on again, the condensate column will be accelerated into the steam pocket. Steam in the pocket will be compressed as the water column advances thereby raising its temperature above that of the surrounding pipe and oncoming condensate. Heat transfer from the steam to the pipe walls and condensate will accelerate due to the increase in temperature thereby causing the flash steam to condense and its volume to collapse. If the rate of steam volume reduction meets or exceeds the rate of the volume consumed by the oncoming water column, there is no "cushioning" of the water hammer strike. (The sidebar at the article's end elaborates).

Without cushioning by either non-condensables or uncondensed steam, the water column can run virtually unimpeded into the water column on the other side of the void causing water to-water water hammer when the two columns collide.

The overpressure of moving water hitting stationary water is ½ that of water hitting a valve or dead-end so the resulting overpressure will be about 30 psi per 1 foot/sec of velocity that the column of water attains. In addition, since the downstream water column is not constrained against movement, the relative velocity of the initial collision will be slightly decreased since pre-compression of the void will start the downstream water column moving before the strike occurs. For these reasons, the water hammer portrayed in Figure 3 will be less forceful than that where a valve is closed and struck.

Water Hammer in a Boiler Feedwater System.

If steam is able to backflow into a boiler feedwater line when the feedwater pump is de-energized, the line will hammer violently. The same can apply to hot boiler feedwater if there's a leak path to a system component at lower pressure. That, *I presume*, is the reason the ASME B31.1 code[7] requires a stop and check valve in the feedwater line, and why it is usually placed at or near the entrance to the boiler. Problem is check valves are unreliable-- they can wear out, be improperly applied, and certain types are highly susceptible to water hammer damage[8].

If steam enters the feedwater line, say due to a low water condition which exposes the feedwater distribution header, hammering can ensue due to either condensation-induced water hammer or, once the feedwater pump is re-energized, column-closure water hammer.

Condensation-induced water hammer can be expected if the feedwater line entering the boiler is horizontal, or even worse, slopes down as it approaches the boiler entrance. Should the water level go low and the check valve leak, steam will enter the distribution header as feedwater drains out. The steam will become entrapped in feedwater, which, if more than 40oF cooler than the steam, can hammer violently.

An incident relayed to me by Bill Lowery of University of North Carolina is a classic case. An off-line boiler had been warmed and pressurized to about 40 psig via plant steam fed thru a heat exchange coil in its mud drum. The water level dropped below the level of the feedwater distribution header in the steam drum so that the spray holes in the header were exposed to steam inside the boiler. This allowed steam to enter the distribution

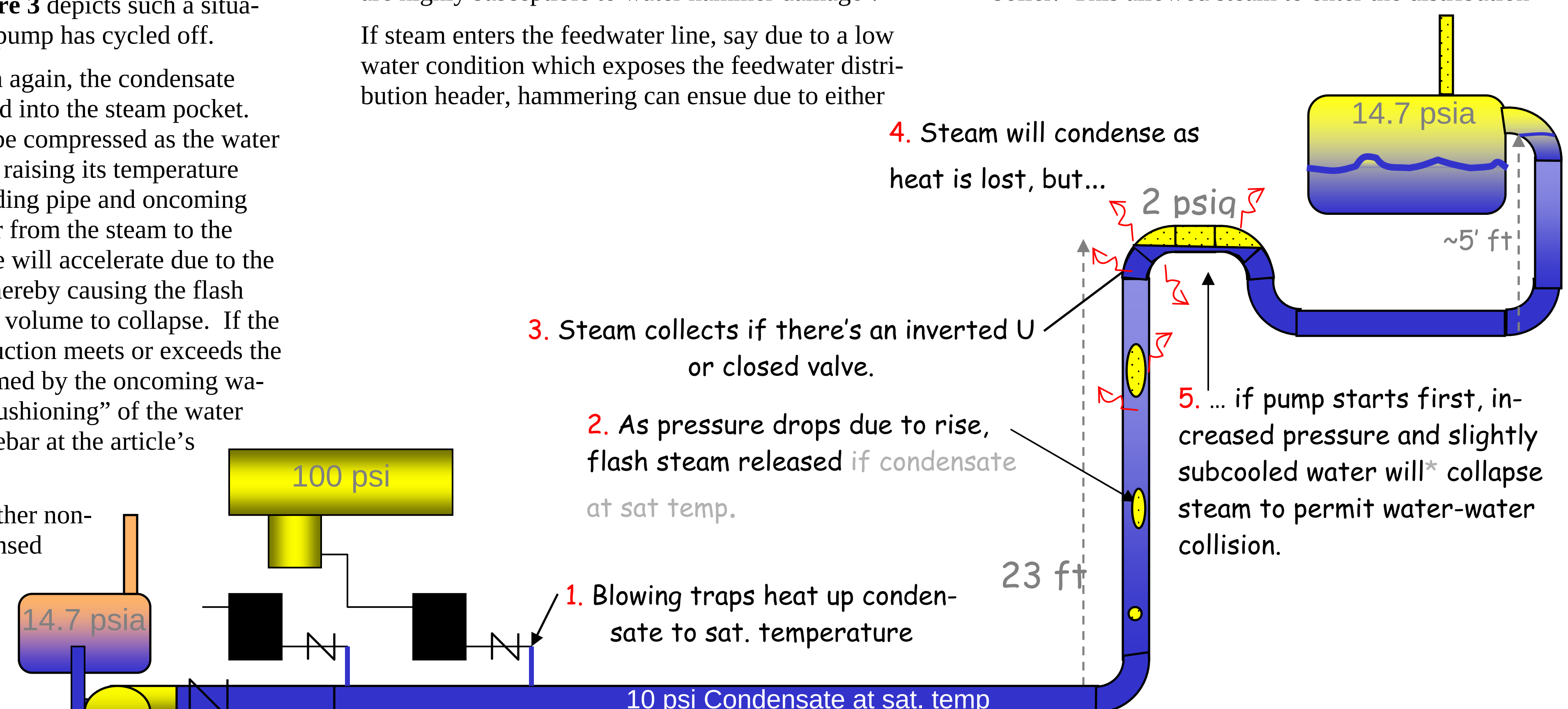


Figure 3. Entrapped Flash Steam in a Condensate Rise

[7] See B31.1-122.1.7
[8] Based on the argument that impact factor for swing type check valves will often be 2.0 due to the small mass of the swing .

header as feedwater from the header drained out into the boiler. **See Figure 4A.**

When operators noticed the low water level, the boiler's feedwater regulator valve was cracked open to refill the boiler. What followed was a succession of sharp bangs severe enough to cause operators to evacuate the boiler portion of the Plant.

Bill Lowery, who investigated the incident, explained to me that the stop/check on the newly erected boiler had been sited at the rear of the boiler (in anticipation of a different boiler configuration) some 22' upstream from the connection to the feedwater input. In addition, the connection of the 6" feedwater line to the boiler was made thru a 6"x 3" concentric reducer as shown in the figure. As a consequence, when the feedwater distribution header was exposed to steam due to the low water condition and allowed to empty, the concentric reducer retained 1.2" of water at the bottom of the 22 ft long horizontal pipe downstream of the stop-check. **(Figure 4A)**. Consequently, the 40 psig steam at 287oF that entered the feedwater line lay atop feedwater that had probably cooled considerably since leaving the deaerator at 235oF before the feedwater pump had been shut off several days earlier. (The horizontal feedwater line might have hammered at this point if induced steam flow due to condensation in the feedwater pipe was sufficient to entrap a steam pocket).

When the feedwater regulator valve was cracked open to restore the water level in the boiler, the 6" feedwater pipe likely did not flow full when the feedwater line turned horizontal after the stop and check valves[9]. This allowed feedwater flowing in thru the check valve to flow along the bottom of the 6" pipe. At the same time, steam contacted by the large turbulent surface of relatively subcooled incoming feedwater would have started rapidly condensing thereby drawing replacement steam from the boiler drum to replace it. In a roughly half-filled horizontal pipe, it doesn't take much relative velocity difference for steam flowing in counterflow over condensate to draw up a wave of condensate (via a Bernoulli-like effect) to plug the pipe and entrap a pocket of steam. See **Figures 4B. & 4C**.

The steam, once entrapped in sufficiently subcooled feedwater, will rapidly condense and collapse to a pressure approaching the vapor pressure of the surrounding feedwater. With 40 psi pressure from the boiler on one side of the water plug, and a low pressure steam void on the other, the plug of water that bridged the top of the pipe will be accelerated into the rapidly condensing steam void ultimately striking the check valve in a water hammer event. After the initial collision, the wave reflection would probably allow the check valve to reopen causing the situation to repeat itself—at least until the feedwater line completely filled with water.

Examination of the check valve after the incident didn't reveal any obvious damage, but to be prudent, the check valve was replaced with a new valve. In addition, to avoid future problems, the feedwater line was re-piped to slope up to the boiler more than ½' per foot so feedwater would no longer drain should the distribution header become exposed, and, even should steam enter the feedwater line due to an upstream leak, the slope was sufficient to suppress water hammer. Bill Lowery tells me that recent controlled tests of this fix under low water conditions in the boiler have proved successful in suppressing waterhammer.

The lessons from this incident are several. The most fundamental is that boiler steam can't be permitted to enter the feedwater system. It did in this case for two reasons:

1. The stop and check valves were not placed at the entrance to the boiler.
2. The water level was allowed to drop below the level of the steam distribution header in a boiler filled with steam when feedwater flow was off.

A less obvious but perhaps more common cause of water hammer in feedwater systems can occur if boiler feedwater is allowed to backflow into the feedwater system due a bad check at the boiler; *and*, a flow path thru the feedwater regulator to the feedwater heater or other boilers at lower pressure is available when feedwater pump(s) are off. This condition requires both a leaky feedwater regulator and faulty check valve at the feedwater pump, if there is one.

Boiler water will be superheated compared to the reduced pressure in the feedwater line if pumps are off and there is a leak path back to a system component at lower pressure. Superheated boiler water will flash allowing steam to accumulate in a high spot, if there is one, or directly contact subcooled feedwater as the flash steam buoyantly rises. In the former case, a column closure water hammer will occur when pumps are re-energized; in the latter case, condensation-induced water hammer is possible.

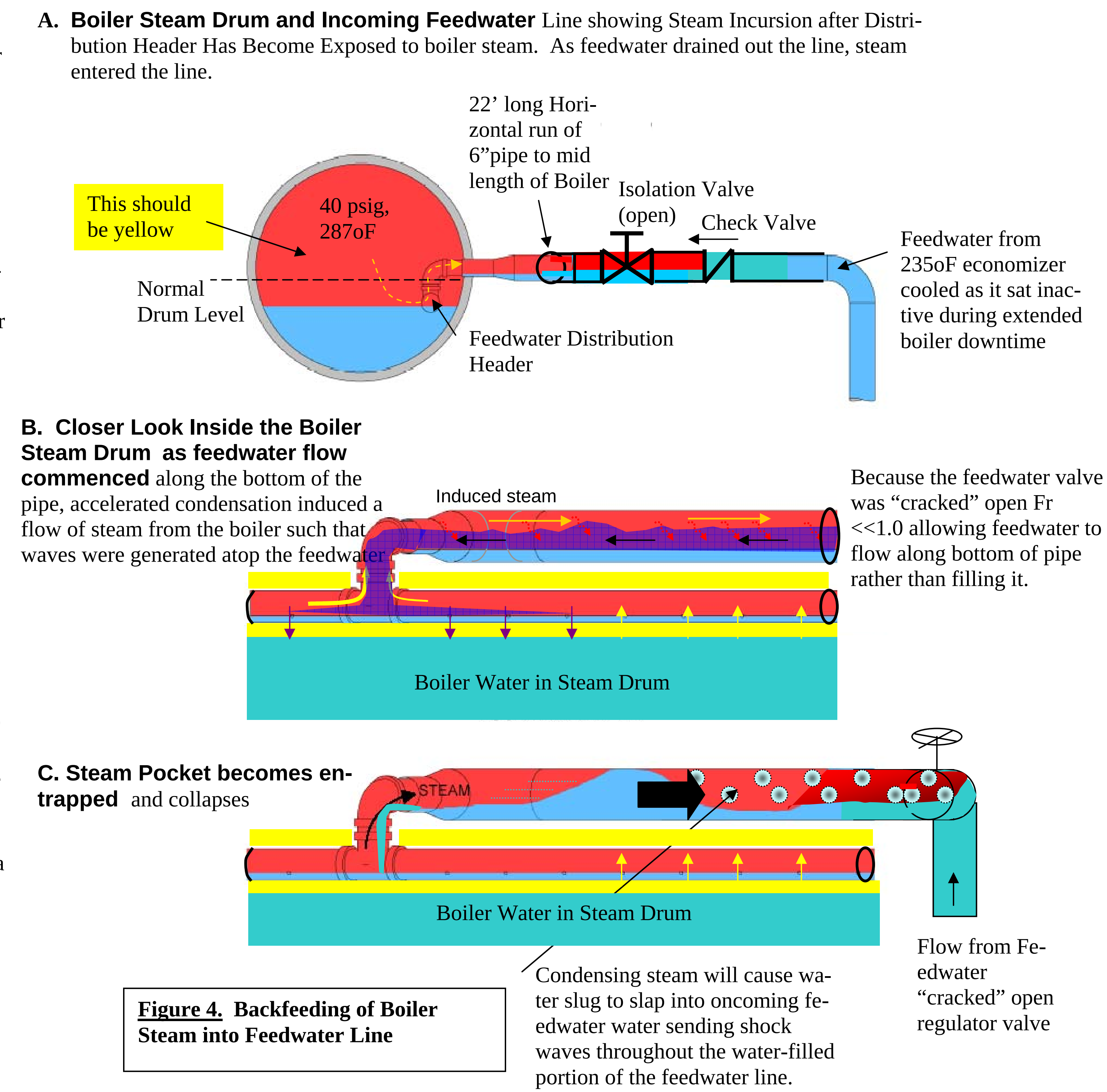


Figure 4. Backfeeding of Boiler Steam into Feedwater Line

[9] Full flow is guaranteed if the Froude No. exceeds 1. This would require a flow velocity of 4 fps in the 6" feedwater line: E.g. : For Fr= v/√gD > 1, v > [32.2 ft/s2 * .5 ft]^1/2 ~ 4 fps

Column Closure and Condensation-Induced Water Hammer—*What You Need To Know*

- First and foremost: Neither mechanism of water hammer is caused by fast moving steam picking up a slug of liquid and hurling it downstream—this description of water hammer is a misconception. Water hammer is caused by fluid accelerating into a void and slamming into something that abruptly stops—not redirects—but STOPS the water column. A fast-moving condensate slug sloshing around an elbow WILL NOT cause water hammer (even though it does impart an impulse force on the elbow's supports at it is redirected around the elbow).

What differentiates a Column-Closure water hammer from a Condensation-Induced water hammer event is the manner in the which the fluid gets moving. To wit--

- A Condensation-Induced water hammer requires a "rapid condensation event" in which an entrapped steam bubble surrounded by subcooled condensate collapses thereby causing the condensate to accelerate and "slap" into the void. The collapse takes place on the order of a microsecond. The motive power for accelerating the water is supplied by the differential between the surrounding steam pressure and the collapsing pressure within the void. High pressures (> ~60 psig) lead to high fluid velocities and powerful water hammer events on the order of several thousand psi. This is the kind of water hammer that shatters steam components and can kill operators.

- Column closure water hammer, on the other hand, requires no subcooling because no "rapid" condensation need take place. The fluid is motivated by a pump or imposed pressure differential accelerating a column of water into a void. The void is created by a either a vacuum generated when a water column separates below a closed valve or dead end, or steam is entrapped in a high spot in a piping system. Column Closure water hammer is generally, though not necessarily, less forceful than condensation-induced water hammer due to the limited velocity attained by a pumped water column.

On more thing…

While the two mechanisms of water hammer I've discussed in this article cover most instances of water hammer in a boiler plant, the internet-posted incident described below the title to this article could have had another, and more pedestrian, explanation.

Water hammer was experienced when the feedwater pump was turned OFF. If the hammer occurs simultaneously with pump shut-off, and, as indicated in a subsequent post, there is only one check valve between the pump and boiler, the hammer could be due to the check valve flapper hanging up. E.g., when the pump shuts off, feedwater line pressure will drop to approximately that of the feedwater tank from which the pump is drawing; boiler pressure will then accelerate feedwater backward thru the hung-up check valve. When the backflow is significant enough to cause the check valve to release and slam shut, the water column in the line will come to an abrupt halt against the check valve with a thud, if not a bang.

The End

References

1. Wiggert and Hatfield, "Liquid Slug Motion in Compliant Piping', Forum on Fluid transients, ASME International Mechanical Engineering Congress and Exposition, Atlanta, GA, Nov 1996 (Also included. in Ref. 4 below)
2. Chou and Griffith, "Admitting Cold Water into Steam Filled Pipes Without Water Hammer due to Bubble Collapse" circa 1988.…"
3. EPRI's "Resolution of Generic Letter 96-06 Waterhammer Issues" User's Manual and Technical Basis Report TR-113594-V1 and V2, December 2000
4. "Water Hammer Handbook for Nuclear Plant Engineers and Operators", EPRI TR-106438, 2856-03, May 1996
5. W. Kirsner, "Condensation-Induced Waterhammer", Heating/Piping/Air Conditioning, Jan 1999

Steam Cushioning of Column-Closure Water Hammer

Consider an oncoming water column moving at 10 fps into a steam filled pipe and striking a dead-end. Calculate the amount of steam compression, and hence backpressure, necessary to raise steam temperature sufficiently to cause steam to condense at a rate so as not to impede the water column. Intial steam pressure assumed 0 psig.

The mass rate of steam condensation so as not to impede the on-coming water column is:

$$m^*_{stm} = \rho \;\; v \;\; A$$

$$= (1/26.8) \text{ lbm/ c.f.} * 10 \text{ fps} * \;A \text{ s.f.} \;=\; 0.37*A \text{ lbm/s}$$

Heat Transfer Required to condense m^*_{stm} at approximately 0 psig:

$$Q^* = 970 \text{ Btu/\#} * 0\,.37A \text{ lbm/s} = 362*A \text{ Btu/s}$$

Let's suppose heat transfer to walls is negligible and only consider heat transferred to the surface of the water disk. (This is probably valid as the column closes in on the dead-end where wall area is small and the compression and condensation becomes most critical. If the water column were striking a second water column, heat transfer Area would = 2A).

$$Q^* = h\,A\,(T_{stm} - T_{cond})$$

Per Peter Griffith, the Chairman of the Expert Panel Review of EPRI's Technical basis Report "Resolution of Generic Letter 96-06 Water hammer Issues , h=64,000 Btu/hrft^{2}F [=17.8 Btu/sft^{2}F] correlated well with simulations and experiments on column closure water hammer events where the water surface is assumed to be highly irregular and wall heat transfer is ignored.

Equating the required heat transfer at 10 fps with the above equation:

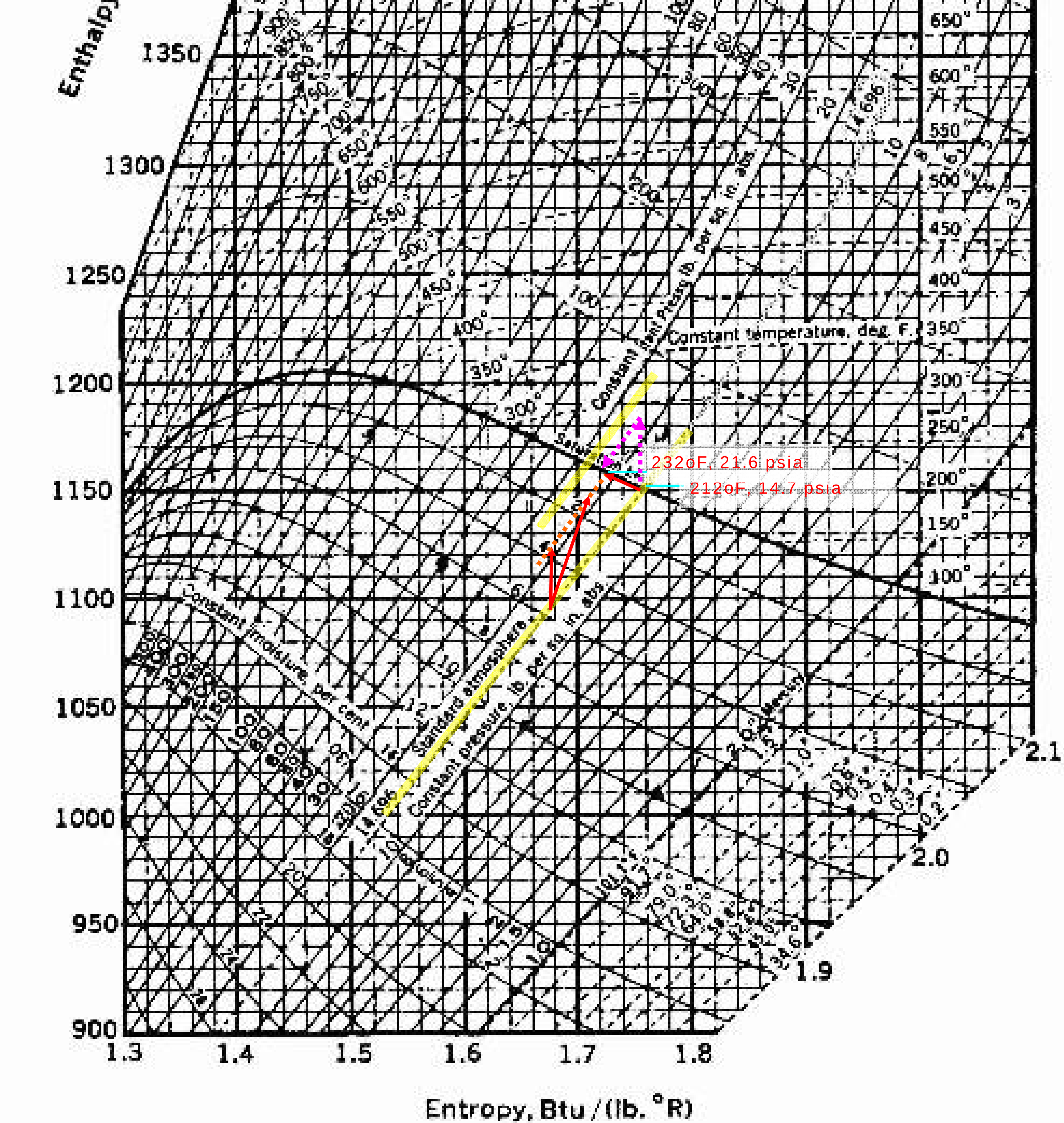
$$362*A \text{ Btu/s} \;=\; 17.8 * A\,(T_{stm} - T_{cond})$$

Therefore for an adequate rate of heat transfer to enable steam to condense in advance of the onrushing condensate:

$$(T_{stm} - T_{cond}) = 20.3^{o}\text{F}$$

The amount of compression, and hence backpressure, to warm steam 20.3oF from 212oF to 232oF is **~8 psi** if the steam remains within the saturation dome (above the dome superheated steam warms at a significantly greater rate with increased pressure). For example, compression from 14.7 psia, 212oF, 96% quality to 22 psia, 98% to 100% quality, warms temperature to 232oF. (It is assumed that steam cannot compress into the superheat region above the steam dome because of the presence of the free water surface. Compressed steam that momentarily enters the superheat region is presumed to evaporate water thereby returning it to the saturation curve and, essentially, riding the curve to the left while evaporating additional steam which makes up for the enthropy debt--see Mollier Diagram at end of article).

The velocity reduction due to backpressure will depend on the magnitude of the pump head or pressure driving force devoted to overcoming pipe friction. For a high-head pump, for example, with say 40 psi available to overcome pipe friction, a backpressure on the scale of 8 psi will reduce the terminal velocity of an accelerated water column by only about 11% ($=1-(32/40)^{.5}$) assuming a relatively flat pump curve. For a pump devoting little head to overcoming pipe friction, on the other hand, more than a few psi of backpressure may reduce water column terminal velocity to the point where water hammer severity is insignificant. (Of course, an iterative calculation is required to calculate the amount of backpressure developed in the void since the critical input to the very first equation above to calculate the rate at which steam must condense is the assumed velocity of the oncoming water column).



Three alternate processes are depicted by the red arrows for compressing atmospheric pressure steam so it warms 20oF as required to provide enough heat transfer to make the steam disappear (i.e., condense) in front of the inrushing water (assumed to be 10fps).

In all cases for compression of wet steam "below the dome" the steam must be compressed to 21.6 psia to warm 20oF. The vertical arrow represents isentropic compression of wet steam; the arrow slanted up and to the right represents compression with entropy increase to the same pressure.

The 3rd case where the red steam arrow follows the saturation curve to the left (entropy decreases!) represents the resultant of two imaginary process arrows—one vertical from the saturation line into the superheat region where the steam warms considerably for a small increase in pressure, then a second process where the superheated steam evaporates moisture from the free surface of the on-rushing water at constant pressure bringing the steam back to the saturation line. The entropy increase of the evaporated water (process arrows not shown) makes the net entropy increase positive.

This 3rd case is confusing in that it entails a small amount of steam evolving from the water surface due to heat transfer from superheated steam whereas the whole premise of the calculation is to find the pressure at which the steam is warmed sufficiently above the onrushing water temperature to condense the steam and thus get it out of the way of the onrushing condensate.

The Joukowski Equation

The over-pressurization due to water hammer, P_o, is equal to :

$$P_o = \rho c v$$

Where:

ρ = density of condensate

c = the speed of sound in water (which is a function of water's stiffness).

v = the velocity reached by the water column

If moving water strikes stationary water, a "½" multiplier arises from the extra springiness of water hitting water. (Essentially the collision puts two "water springs" in parallel effectively halving water's stiffness).

To get an order of magnitude estimate for the over-pressure created in for every I fps of water velocity prior to a collision where water density is roughly 60 #/c.f., and "c" is corrected downward 10% for the coupling of steel pipe walls with the water column, and air cushioning is minimal, overpressurization equals:

$$\mathbf{P_o} = \frac{\text{60\#/c.f. * 4300 fps * 1 fps}}{g_c} = \mathbf{55.6\ psi}$$

Dynamic Load factor[10]

The "dynamic load factor" or "impact factor", as it's sometimes known, multiplies the impact felt by the object struck in a collision by a factor of from less than one to up to two. The value of the dynamic load factor depends on the response time of the object struck (i.e. the piping system's natural frequency) compared to the duration of the collision).[11] The duration of the collision depends on the length of the water column being decelerated. The longer the column, the more likely the dynamic load factor will exceed one and approach two. For example, for the bounding case of a very flexible piping system (natural frequency = 4 cps), the dynamic load factor would exceed 1.0 if the run of pipe from tank to valve was over 90 feet long[12] and approach 2.0 if the run of pipe were 270 feet long. At the other extreme, a very stiff piping run with a natural frequency of 40 cps, would require at least a 9 foot long column of water for the dynamic load factor to exceed 1.0, and a column of water greater than 25 feet long to achieve a load factor of 2.0. The figure shows Impact factor as a function of pulse-to-period ratio "θ" and the "squareness" of the pulse "ϕ" is shown in Figure XX reproduced from Wiggart (Reference 1. and 4.)

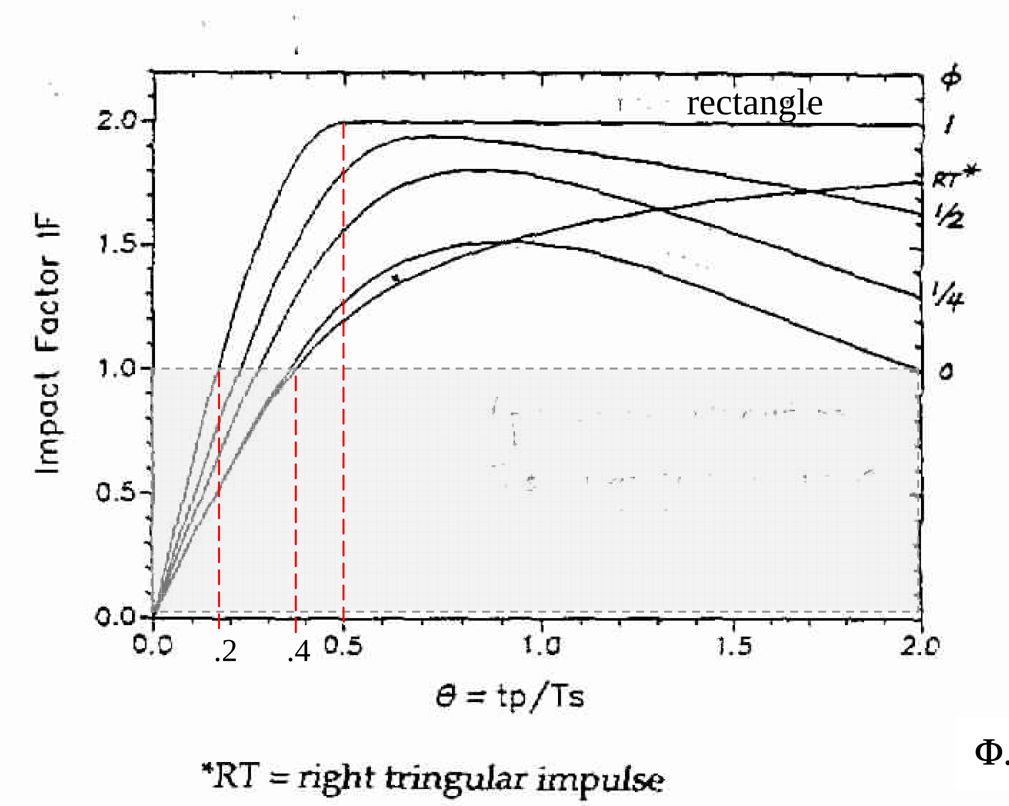


Figure 3.1-5 Impact factor as a function of t_p/T_s and shape of impulse

Figure XX. Impact Factor as a function of pulse-to-period ratio "θ" and the "squareness" of the pulse "ϕ" . Specifically, Φ= peak pulse time/total pulse time so rectanglular pulse =1 and triangular = 0 but 0 on chart represents an isosceles triangular pulse, RT is a right triangle.

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[10] Reference 2.

[11] The shape of the impulse also matters, but for long columns of water rising in a pipe, the pulse may be considered square.

[12] For "dynamic load factor" to be as great as 2.0, the water column pulse duration, 2L/c, must exceeded1/6 the natural period, Ts, of the system per Wiggert. For a natural freq. of 4 cps, i.e. Ts= .25 sec, c=4300, then L>(c/2) (.25/6) or L > 90 ft.