Defending Primary-Secondary – The Merits of Hydraulically Separated Boiler Loops

By: Tom Heckbert

Advantages to Primary-Secondary Piping

- Primary-Secondary boiler plants typically have lower upfront capital requirements.
- Primary-Secondary piping easily accommodates hybrid, front-end loaded, and multi-fuel systems.
- Primary-Secondary systems typically require less mechanical room floor space.
- Watertube boilers installed Primary-Secondary in high-temperature applications can also be installed with economizers, to boost efficiency while still taking advantage of lower cost and space requirements.

Advantages of Primary-Only Piping

- Return water is brought to the heat exchanger directly, ensuring the coldest possible return temperatures and maximum condensation.
- Primary-Only systems operate with low return temperatures regardless of load conditions.
- Fewer pumps, and therefore fewer motors, are required for plant operation, reducing electrical consumption.

In recent years, the dominant trend in hydronic boiler plant design has been towards the use of high water-volume “firetube” boilers, installed directly in the system loop, what is known as “Primary-Only” (PO) piping. At its core this change has been driven by the spread of “condensing” boiler technologies, and the fact that PO installation will, in many systems, tend to return slightly cooler water to the boiler plant. This is a desirable trait for condensing efficiency, and while “watertube” boilers installed in a “Primary-Secondary” (PS) configuration also perform very efficiently, the desire to extract ever greater quantities of energy from fossil fuels has motivated the rapid adoption of PO piping and firetube technology. In some ways this trend is welcome, but several truths have often been overlooked in the process: namely, condensing operation may not be possible regardless of the plant design chosen, and PO piping only truly excels where the temperature drop across the system is large.

This paper will examine the logic behind the utilization of PO over PS piping in comparable systems, beginning with a review of the science behind flue gas condensation and the mechanics of hydraulic separation. An examination of how real-world design requirements negate the benefits of PO piping will follow, concluding with a review of some of the advantages of PS piping for design and equipment selection.

The ultimate goal is not to disparage PO piping, for in some cases it will be the most efficient and mechanically logical installation method. Rather, the objective is to establish the reasons why PS must remain an essential part of any designer’s toolkit.

Basics of Condensing Efficiency

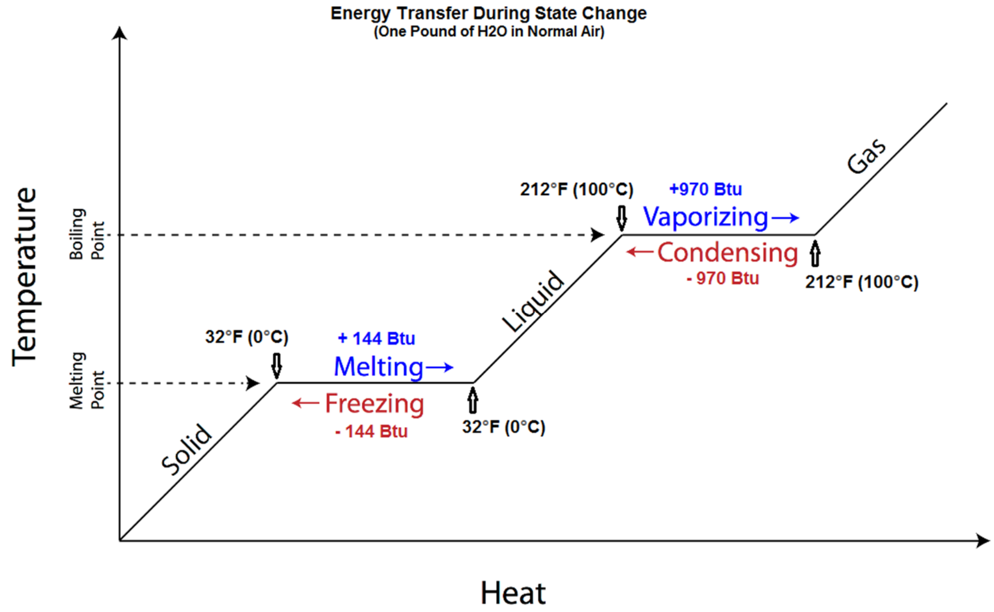
Using approximate values, imagine we combusted a volume of methane (natural gas) equivalent to 1,000 KBtu, using 68°F (20°C) combustion supply air. After the flame the gases will be 2,300°F (1,260°C). If passing these exhaust products through the boiler cools them to 170°F (77°C), we will recover 880 KBtu of the input energy, by collecting 96% of the “sensible” energy (i.e. what we read with a thermometer). If we cool the exhaust to 68°F (20°C), we will recover nearly all of the 1,000 KBtu input and 100% of the sensible energy. However, only 920 KBtu will have been collected through sensible energy transfer. So, how do we account for the other 80 KBtu?

If the combustion of methane with oxygen is perfect (or “stoichiometric”) the only byproducts are carbon-dioxide and water, as can be seen in the chemical formula:

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

Given the temperature after this reaction, this water will begin as steam. As the flue products are cooled during their transit through the heat exchanger, the steam will eventually approach its dew point, which, because the composition of the flue products is not the same as normal air, is typically specified as 131°F (55°C)[1] for methane exhaust.

When the flue gas finally crosses this point, the water vapor will condense, losing energy and changing from a vapor state into a liquid one. During the state change there is no sensible temperature change, but the “latent” energy that was keeping the water vaporized is released as heat, at a rate of 970 Btu per pound of condensate.



For every 1,000 KBtu of methane fuel combusted, there will be the equivalent of 83.5 pounds of steam created (10 US gallons). If all of this water vapor is converted into liquid water, the missing 80 KBtu of energy will be released[2].

With stainless steel or aluminum appliances[3] all that is required to commence useful condensation is to inject water at or below the dew point into the heat exchanger. If the water temperature is only slightly below, then only flue gases very close to the heat exchanger surface will be sufficiently cooled for condensation to take place, so only some of the available latent energy will be recovered. As the temperature drops further, more flue gas will be cooled, and more condensation will occur. Practically any boiler supplied with return temperatures at 68℉ (20℃) will entirely cool the flue gas below dew point causing all available steam to be condensed out, leaving the exhaust dry and cool, putting the combustion efficiency at very nearly 100%.

However, it is important to note that if the water returned to the boiler is 140℉ (60℃) or higher, then ***no condensation is possible***. No part of the flue gas will be cooled below dew point, and all of the steam will find its way into the vent stack. In these conditions, most condensing appliances would have a high-fire exhaust temperature around 170℉ (77℃), meaning the maximum efficiency they can produce will be 88%[4]. This is precisely the same efficiency and performance that would be expected of a non-condensing appliance.

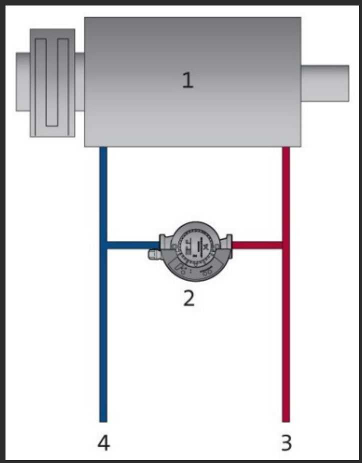
Hydraulic Separation

In the earliest hot water boilers, return flow and temperature controls were not needed. Most were converted steam equipment, with very large water volumes and low water pressure drop. As the industry developed and the demand for smaller footprints grew, boiler designs began shrinking, and the boiler-to-system water volume ratio widened.

The immediate consequence was failures due to “thermal shock”, where large volumes of cold return water would cause part of a heat exchanger to shrink in one area more than in another, causing a stress fracture in an area in between. The most popular solution was to use a bypass pump (also known as a “belly” or “shunt” pump, see side bar) to inject heated water into the boiler return.

The Bypass Pump

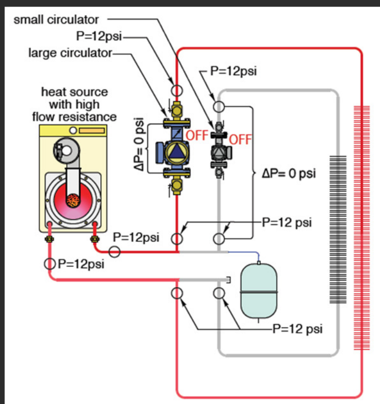
The boiler (1) heats the supply (3), the bypass pump (2) injects heated water into the return (4), increasing its temperature.



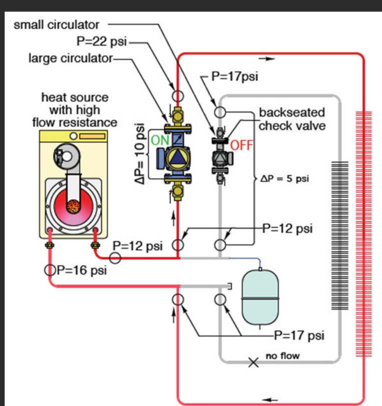
Courtesy: Grundfos

Hydraulic Separation

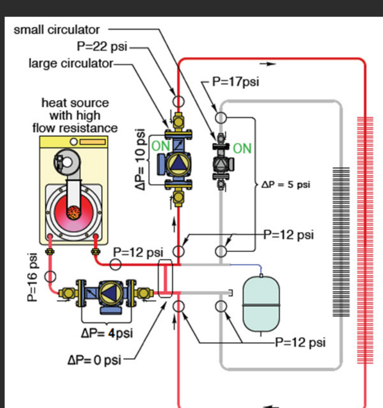
Step 1: The pumps are idle, and the system is at static pressure. Note: the pressure at the expansion tank will not change as pumps activate.



Step 2: The 10psi pump activates. High pressure drop (ΔP) on the boiler heat exchanger creates a 5psi ΔP across the supply/return header. The small pump will not be able to overcome it, meaning no flow.

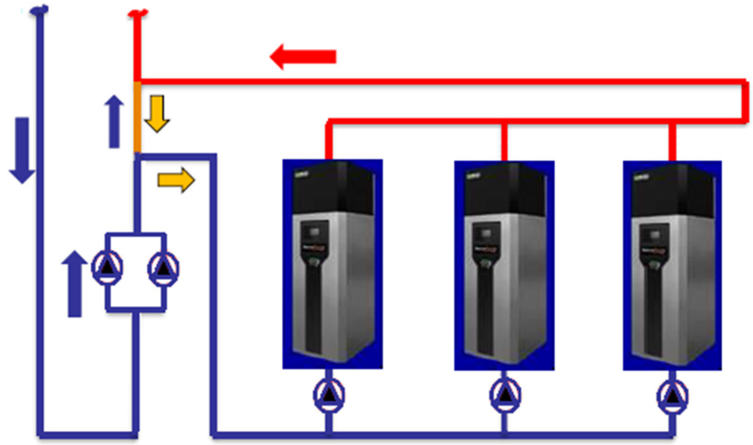


Step 3: Hydraulic separation reduces common piping ΔP so both system loops function. A pump is added to create flow through the boiler.



Images from Caleffi idronics, Issue #15

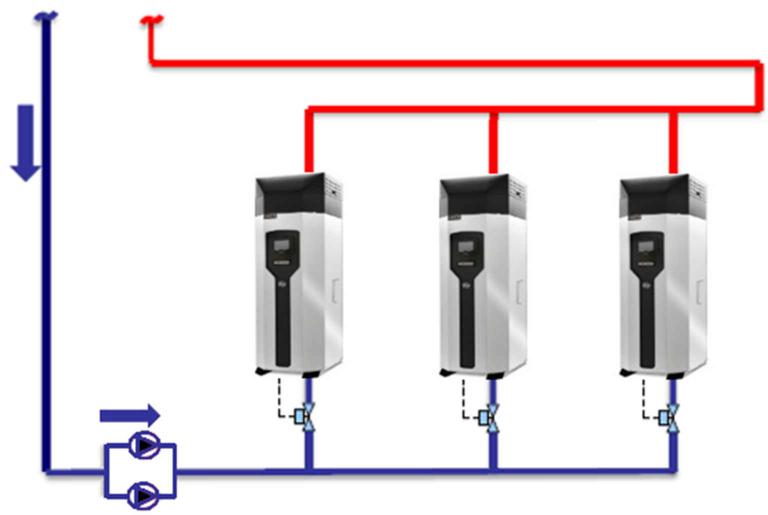
This design allowed the boiler to operate without concern that bursts of cold water would cause thermal stress, but it also boosted the return temperature to levels where condensing would not be possible. Since these early boilers were constructed from carbon steel or cast iron, avoiding condensation was highly desirable. PS piping, a descendent of this concept, offers much the same advantage: as long as the boiler loop flow rate exceeds that of the system loop, heated supply will bypass into the boiler return line (see the yellow arrows below).



Primary-Secondary Piping

PS piping also had the effect of decreasing the pressure drop across the common piping. Where previously the boilers had been an integral component of the system loop, with the bypass pump simply adding circulation on the boilers for temperature protection, in PS piping the boilers became part of a hydraulically distinct (i.e. pressure independent) loop. This “hydraulic separation” (see side bar) allowed heat exchanger pressure drops to increase with no impact on the system operation, which meant boiler size-to-capacity ratios could be tightened even further.

As demands for energy efficiency grew, designers began to look for ways to increase boiler efficiency. One obvious approach was to design condensing heat exchangers suited to the original PO piping designs, this time without a temperature bypass included, to provide the boilers with the coldest possible return temperatures.



Primary-Only Piping

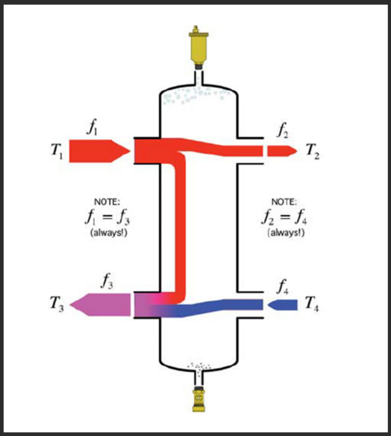
Decoupler Flow

The term “decoupler” is given to the portion of low head loss common piping in a PS system. Common ways to achieve the function of a decoupler are closely-spaced tees (where two tees are closer together than four times the diameter of the pipe they are connected to), buffer tanks and low loss headers. All allow boiler and system side loops to be connected to common piping, without affecting the pressure differential on the other side.

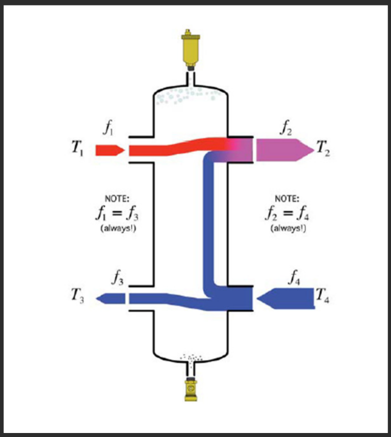
In principle, the pressure at all four connections of a decoupler is the same. Depending on the flow rates on either side of a decoupler, various flow patterns are possible.

If the heat transfer on each side of the decoupler is equal, then the side with the smaller temperature difference (ΔT) will have the higher flow rate.

Higher Boiler Loop Flow



Higher System Loop Flow



Images from Caleffi idronics, Issue #1

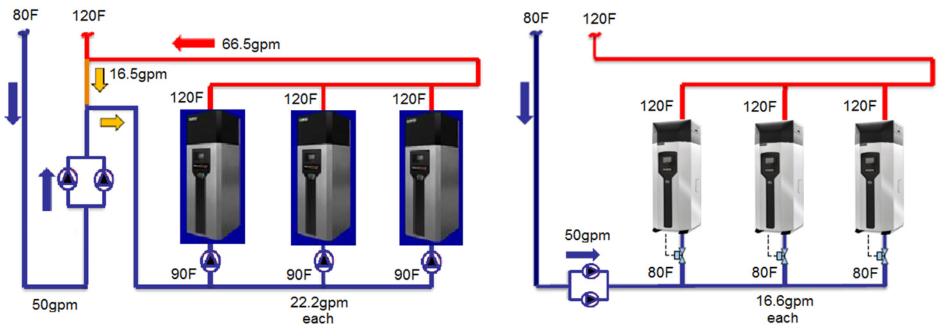
In order to do away with the hydraulic separation of PS piping and move to PO design, these new condensing boilers had to have low water-side head loss, so that the system loops were not forced to share common piping with a large differential pressure (or ΔP). The best way to accomplish this was to change from the popular “watertube” design (where the products of combustion flow through an air-tight chamber containing tubes filled with water), to a “firetube” design (where the exhaust is in the tubes, surrounded by water in a large pressure vessel). Because the ΔP across the pressure vessel in firetube boilers is nearly zero, they allow hydraulic separation of system loops even while being an integral part of their flow path.

An inherent feature of firetube boilers is that they are large, requiring more room per Btu for installation. Typically they are more costly to manufacture than watertube products as a result. This cost is partially offset by the fact that the dedicated boiler circulators required in PS piping are more costly than the motorized isolation valves in PO[5], but the primary motivator for the adoption of firetube boilers is operational cost reduction. Because they do not include a decoupler (see side bar) there is no mixing of boiler return, meaning they will always receive the coldest possible return water, driving the maximum amount of condensing. PO systems also require fewer pumps, so they are expected to consume less electricity than PS systems.

However, the extent to which PS piping will cause decoupler bypass is dependent on the temperature difference (ΔT) and corresponding flow rate requirements across each side of the common piping. Because of their large volume, firetube boilers are typically able to operate with larger ΔT’s across their heat exchangers, but if the system ΔT does not exceed that of the watertube unit, PO and PS will see the same return water temperatures, and the same efficiency. Let’s look at an example.

Primary-Secondary vs. Primary-Only Comparison – Full System Flow

Consider a building with a 1,000 KBtu/h heat load, with a heating plant designed to deliver precisely the right amount of heat. First, imagine the system side loop has a supply and return temperature of 120°F (48.8°C) and 80F (26.6C) respectively, giving a system ΔT of 40°F (22.2°C). The PO boilers can operate with the same ΔT, but the PS boilers will only operate with a maximum 30°F (16.7°C) ΔT.

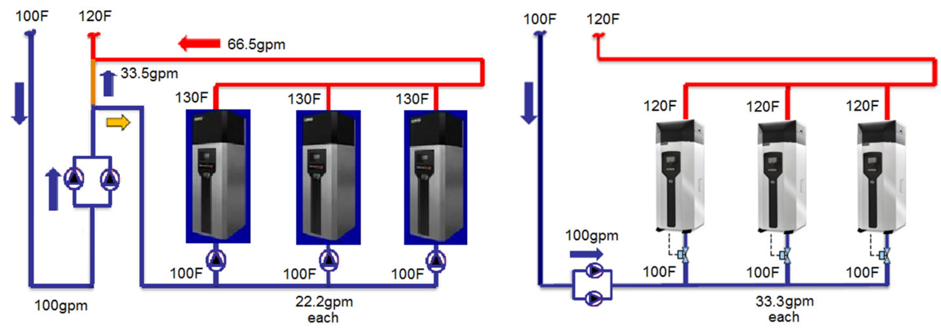


Comparison 1: Primary-Secondary vs. Primary-Only on 40°F (22.2°C) System ΔT

Plainly the PO system has the efficiency advantage at this ΔT. The PS boilers would be operating at around 95% efficiency, but the PO boilers will operate at 97% due to the greater amount of sensible and latent heat energy extracted. In systems designed such that the ΔT will be static, and larger than what the PS plant can achieve, it will not be possible to exceed the efficiency available with PO piping. The larger the ΔT, the greater the decoupler bypass will be, increasing the PO efficiency advantage. In these cases, PO boiler plants should always be considered first.

(Their selection, however, should be based on the system control method, which can mitigate this efficiency advantage, and on plant flexibility requirements, which can make the efficiency gain less of a priority. More on that later.)

Yet many systems are not designed with such large temperature differences. In-floor radiant heating systems, which are an ideal application for condensing technology, are often designed with low temperature drops. The uneven surface temperatures that would result if the water were allowed to cool substantially before returning to the boiler plant would make the floor uncomfortable, and possibly at risk for stress fracture. Temperature profiles as low as 90℉ (32.2℃) supply and 70℉ (21.1℃) return are not uncommon, but a temperature drop of only 20℉ (11.1℃) is typical.

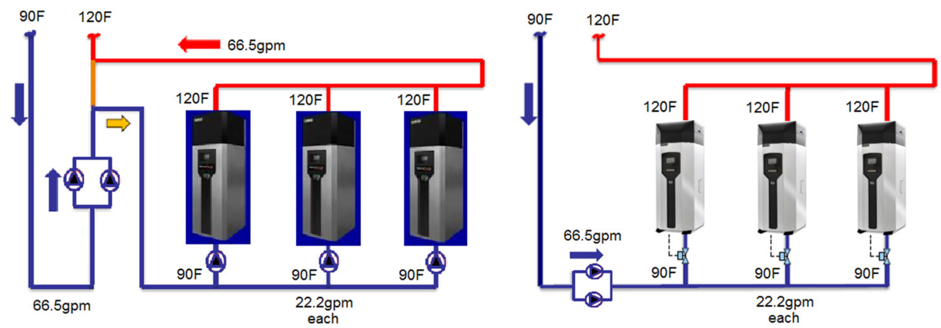


Comparison 2: Primary-Secondary vs. Primary-Only on 20℉ (11.1℃) System ΔT

Most designs should, however, try to utilize a higher ΔT whenever possible. This is because a given heat load can be supplied with 2/3 flow if the ΔT is increased by 50%, and a flow rate decrease will allow a corresponding reduction in the size of pipes, valves, fittings and pumps, reducing costs.

In this spirit, many systems are designed with a 30℉ (16.7℃) ΔT. This temperature difference is within operational limits for most "heat terminal units" (the technical term for the point at which heat is transferred into the desired space), like radiant systems, high-temperature baseboard, unit heaters, etc., such that they are able to deliver the desired amount of heat, with no concern that their transfer efficiency will drop off to unsatisfactory levels.

In a system with a 30℉ (16.7℃) ΔT the PS and PO boiler loops will see effectively identical high-fire performance, with no flow through the decoupler and the coldest return water being injected directed into the boiler.

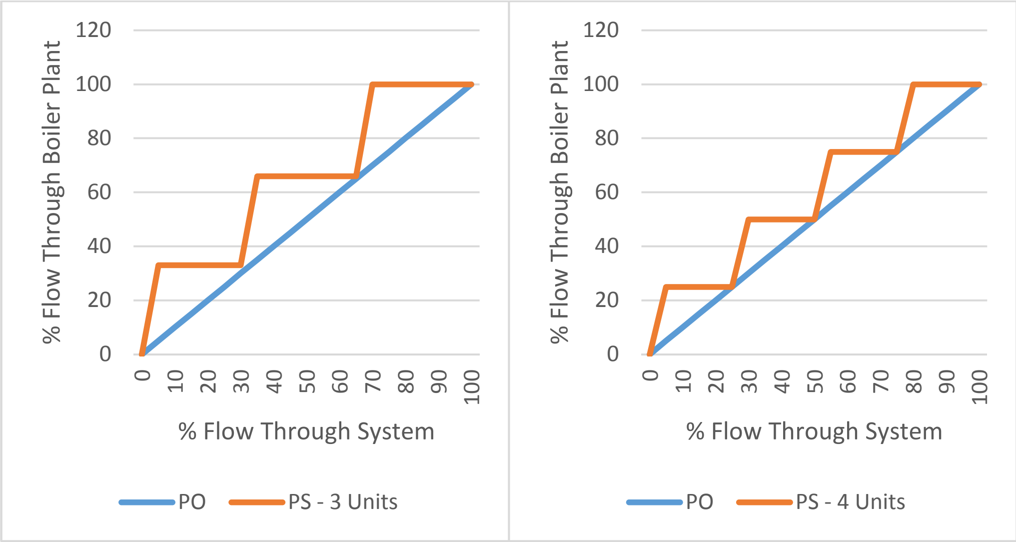


Comparison 3: Primary-Secondary vs. Primary-Only on 30℉ (16.7℃) System ΔT

There are two key points here. First, PO systems always have the potential to exceed the efficiency of PS systems. Return water temperature is the key to efficiency, and at no point will a PS boiler plant receive colder return water than a PO plant. Second, the difference in efficiency between the two systems is defined by the amount of supply bypassed to the return in the decoupler. Any system with a ΔT equal to or less than the operational limit of PS boilers will tend to avoid this bypass, allowing the two piping systems to see comparable if not equal efficiency.

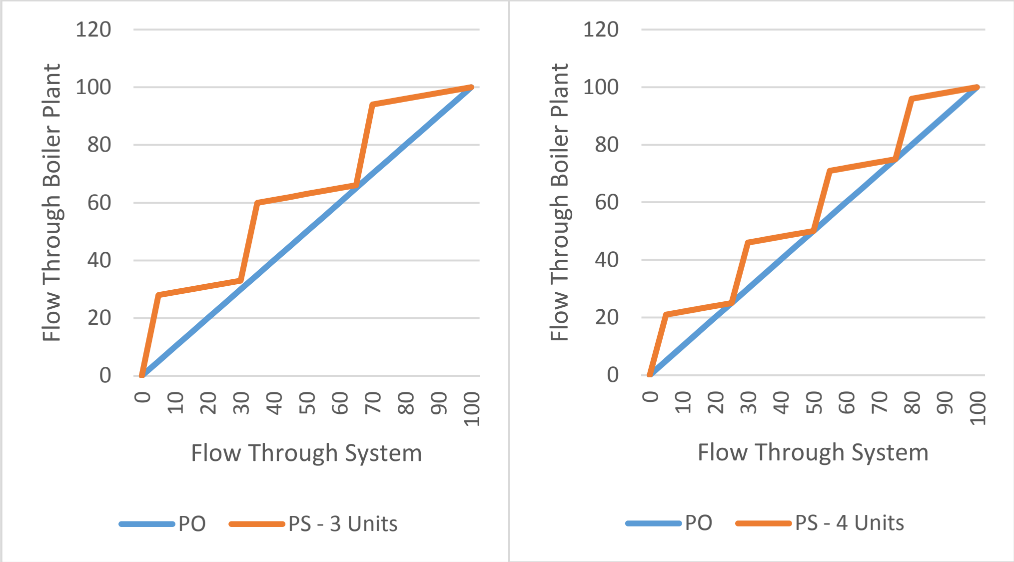
Primary-Secondary vs. Primary-Only Comparison – Variable System Flow

The increasingly common use of variable speed system pumps, which reduce flow in the warmer months of the year, also creates conditions for decoupler bypass. The PS boiler loop will start to outpace the system loop as the system pump begins to slow down. The boiler loop will have higher flow causing bypass until the system flow slows down to the point where the PS boiler plant is able to stage off one boiler and its pump (consequently, the more stages in a PS plant, the easier flow-matching becomes).



Constant Speed Boiler Pumps vs. Variable Speed System Pump

Variable system flow, therefore, tends to favor PO systems. It is possible to reduce some of this effect by using variable speed pumps on the boiler side of the decoupler. While PS watertube boilers will generally have higher minimum flow requirements than firetube, partial setback (i.e. 80% of flow at 20% of firing rate) will allow the PS plant to track the system loop flow rate more precisely, reducing decoupler bypass.



Variable Speed Boiler Pumps vs. Variable Speed System Pump

Remember, the key advantage to PO is maintaining the maximum possible ΔT. In a variable speed system, the point where the decoupler bypass is greatest is in a three unit system at 33% load, when boiler loop flow is double the system flow, meaning the boiler side ΔT will be half that of the system. Because the boiler supply temperature doesn't change, the return temperature necessarily increases.

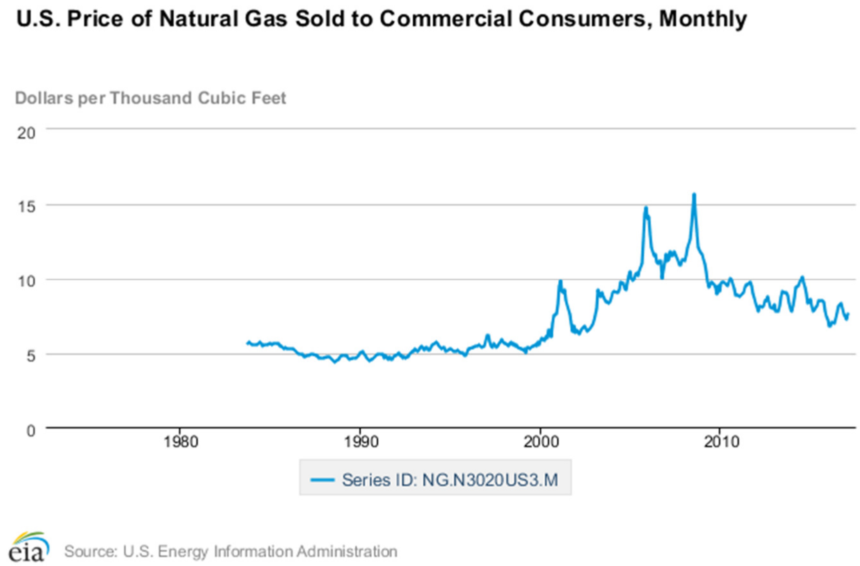
In a system with a 30℉ (16.7℃) ΔT, this will result in a 15℉ (8.4℃) increase in the boiler return temperature, which will correspond to a maximum 3% loss of efficiency. This is because every 10℉ (5.5℃) further into the condensing zone the return temperature drops, the boilers will, in principle, extract 0.5% more sensible and 20% more latent energy, for an overall efficiency gain of about 2%. (If this system is operating above the condensing zone, the efficiency difference is limited to sensible energy gains, or 0.5% per 10℉ (5.5℃).)

This assumes, however, that the system flow varies but the system ΔT does not. This is typical of systems using a static temperature and ΔT, using only two-way valves to stop flow to zones that have their heat demand satisfied. It is not what is expected from systems using outdoor temperature reset, which we will examine later. But first, it is worth noting what the projected fuel savings from the increased efficiency of PO piping really are in this framework.

Projected Savings

Consider a 10,000 KBtu/h system at 33% of load, with return temperatures of 90℉ (32.2℃) and a supply of 120℉ (48.9℃). Based on the 3-unit, constant-speed chart above, the PS plant will bypass a portion of its supply into the return, such that the PS boiler loop ΔT is 15℉ (8.4℃), versus PO boilers operating with the full 30℉ (16.7℃).

At 3,330 KBtu/h the PO boilers (operating at roughly 95% efficiency) should consume roughly 114 ft^3 less natural gas per hour than the PS boilers (with their boosted 105℉ (40.6℃) return and roughly 92% efficiency). Based on the historically highest cost of methane fuel, this efficiency difference would account for a savings of $1.71/hour.

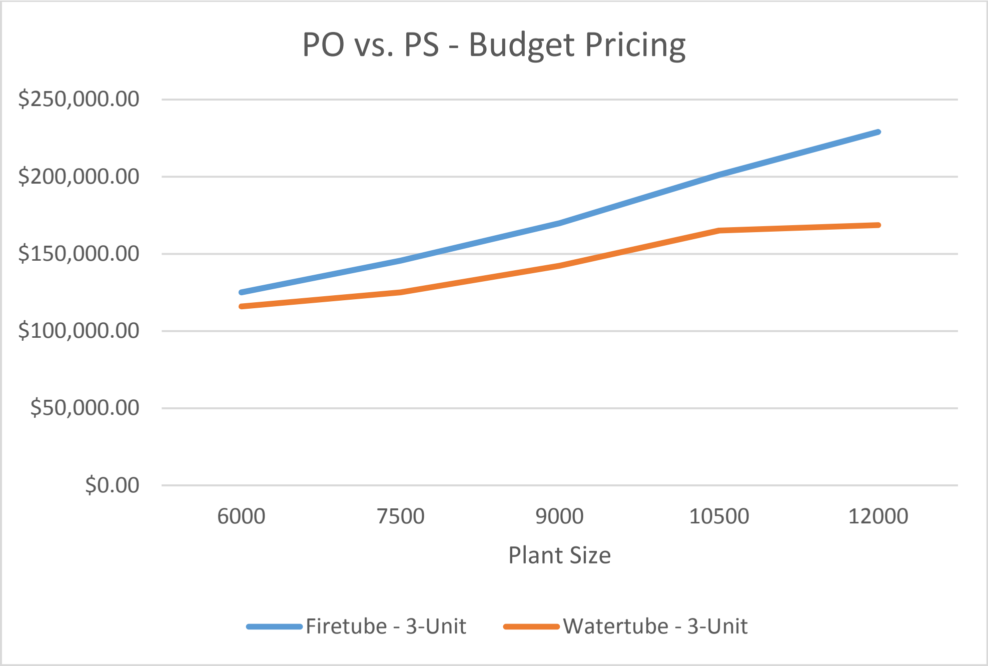


When at 66% of load, the boiler loop flow (now with all boiler pumps running), would exceed the system by 50%, meaning a boiler loop ΔT of 20℉ (11.1℃) compared to the system loop ΔT of 30℉ (16.7℃), resulting in a return temperature of 100℉ (37.8℃) and a 2% efficiency drop for the PS plant. At 6,660 KBtu/h input, the PO boilers (still operating at roughly 95% efficiency) would consume roughly 151 ft^3 less than the PS plant (now operating at roughly 93% efficiency), or roughly $2.27/hour.

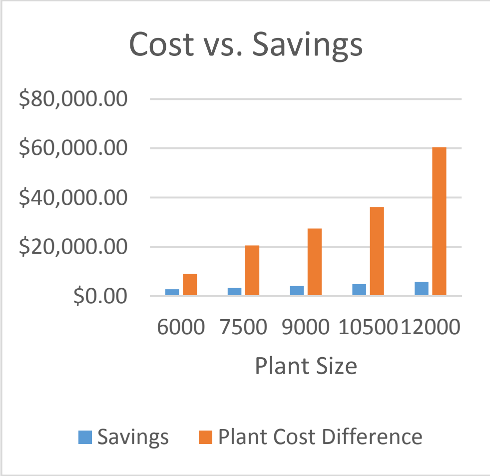
If the boilers operated for six months straight saving $2.27/hour that would amount to $9,915. However, given that the savings are only apparent during the times when there is a mismatch between loop flow rates and that the rate of fuel consumption is lower during part load in the warmer months, meaning less fuel is saved per point of efficiency gained, the actual savings would be much less (projected at around $3,500).

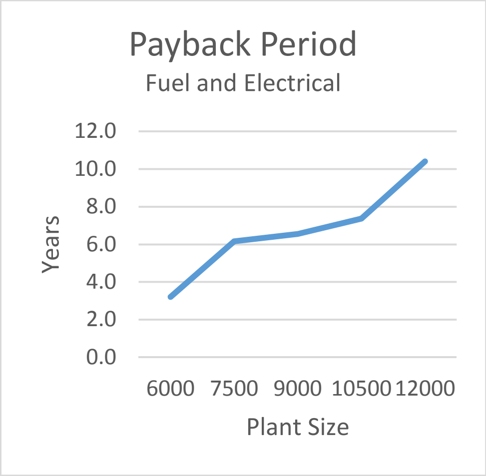
These savings, and the associated electrical savings of not running the boiler pumps (roughly $1,400 at this size range), are worth aiming for, and in that regard the adoption of PO system design is good. However, PO boiler plants themselves come with a higher capital cost. For example, if a 3,500 KBtu/h firetube boiler with motorized isolation valve cost around $67,000, a comparable watertube boiler with pump would cost $55,000, meaning a three boiler plant would have a total cost difference of $36,000. With annual fuel and electrical savings of $4,900, the payback period for this plant would 7.4 years.

As the system scales down, the savings will decrease at a rate commensurate with the load, but the boiler plant itself will not. In other words, boilers half the size will cost more than half as much. This might indicate that larger plants would favor PO systems and firetube boilers: bigger plants mean more savings but not quite as much more cost. However, because PS watertube boilers increase in cost at a lower rate than firetube, the PO system is actually most advantageous in the smaller size range.



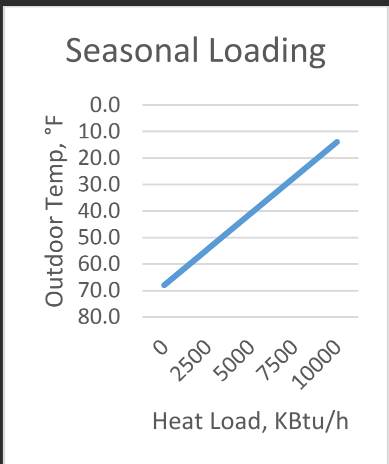
Because the difference in cost between the two plants continues to increase, the PS system sees an ever greater advantage as the plant grows larger. This means a longer payback period for PO systems.





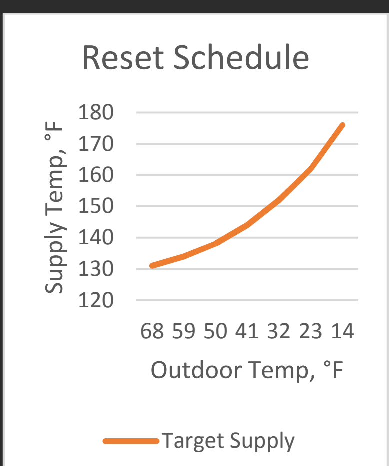
Typical Outdoor Reset Schedule

As the outdoor temperature decreases (shown ascending on the left axis), the building heat load increases (left to right on the bottom axis).



The reason for this is that cooler weather means a greater differential between the fluctuating temperature outdoors, and the target room temperature, which is static. As the difference between these two grows, heat is lost through the building envelope with increasing speed, meaning that more Btu's need to be added every hour.

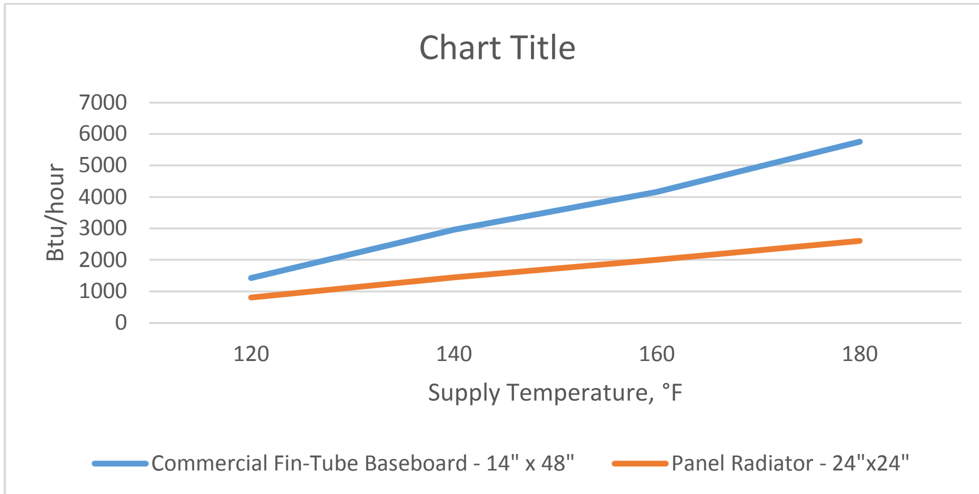
When outdoor reset control is added to a system, the water temperature supplied to a structure is adjusted accordingly. By increasing the supply temperature, heat is delivered to the zone more rapidly, matching the rate of delivery to the rate of heat loss.



In principle, systems with outdoor reset use the same flow rate for all loads, but they cool the water more during higher demand, meaning the ΔT varies according to the load.

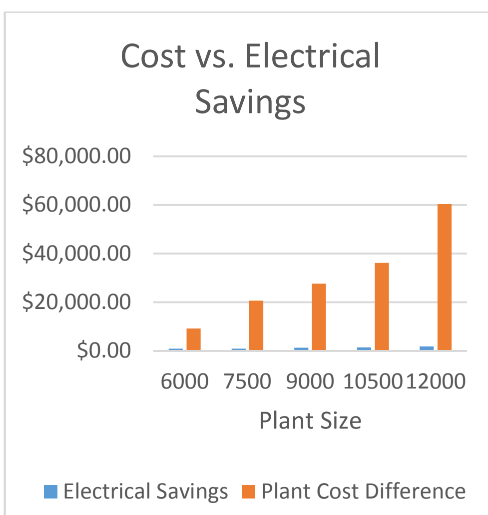
The impact of Outdoor Reset Scheduling

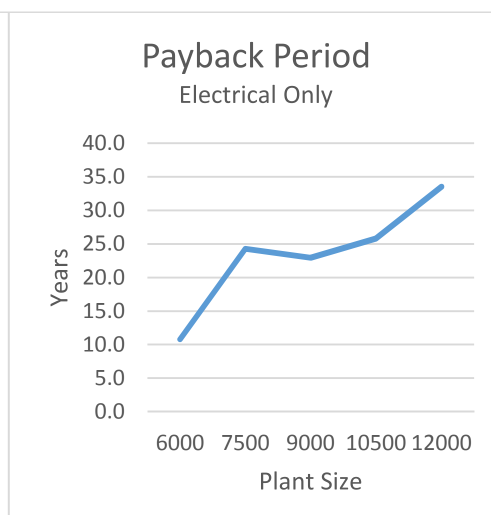
Up until now, the assumption has been that the system was using a static ΔT of 30°F (16.7°C). To prevent the boilers and terminal units from overheating the building[6], such a system has to interrupt flow to the various zones, pulsing heat into the space to maintain the correct room temperature. At 50% load each zone would be open and require flow around half the time, meaning system pumps would modulate down to 50% of their design duty point. When outdoor reset scheduling is added to control a heating system (see side bar), this is no longer the case. Reset scheduling drops the supply temperature according to the current temperature outside the building. When fed with a reduced supply temperature, a given terminal unit will extract heat from the supply more slowly, *cooling the same volume of water less than it had been*.



For example, 1gpm fed through 10' of fin-tube baseboard at 160°F (71.1°C) will deliver 10.4 KBtu/h and be 20.8°F (11.5°C) colder when it exits the baseboard. The same 1gpm fed at 140°F (60.0°C) will deliver 7.4 KBtu/h and be 14.8°F (8.2°C) cooler.

Accordingly, scheduled supply temperatures prevent overheating while maintaining a continuously high flow rate. The volume of water supplied is not changing but the ΔT is shrinking, meaning even if the design had a ΔT larger than the PS boiler maximum, when the system ΔT shrinks the flow will resemble Comparison 2 (Page 6), where the two plants see the same return temperature, and the PS plant supplies *hotter water at a lower flow rate to boost the system up to setpoint*. With equal return temperatures, the only efficiency gain for PO plants will be electrical, greatly extending the payback.





Primary-Only Design Applicability

It is worth asking how many systems are capable of operating with condensing temperature profiles in the first place. Certainly many options exist for mid to low temperature heating, and these are becoming common: radiant heating technologies like in-floor slabs and ceiling or wall panels are found in many commercial buildings. Low temperature baseboard is now more effective and architecturally suitable.

But most commercial systems will struggle to operate with only these types of heat terminals. Most large commercial buildings will have some high-temperature air heating coils, often a significant load in the facility. These coils will be serviced by the boiler plant, limiting the options to reduce supply temperature. This reduces the likelihood of return temperatures below 131°F (55°C), meaning no condensation will take place. PO piping can still be used in these systems, of course, but it will now offer little to no efficiency gain, meaning it will be harder to justify the higher cost, larger footprint, and more limited design options.

Advantages of Primary-Secondary Design – Compatibility

One of the advantages to PS piping is that it allows easy integration of a new boiler plant into existing systems. Because the design allows the use of watertube boilers, the mechanical room footprint required for installation drops substantially. Watertube product options include non-condensing copper heat exchangers, which have an efficiency of 85% to 88%, but which can be installed using any category of venting.

- Cat I = High temperature stack, negative pressure breech
- Cat II = Condensing temperature stack, negative pressure breech
- Cat III = High temperature stack, positive pressure breech
- Cat IV = Condensing temperature stack, positive pressure breech

Since PS boiler plants are hydraulically separated from the system loops, there is no possibility for the equipment selection to impact system performance, meaning the designer does not need perfect information about the system to be sure the boiler selections will work, and they can focus on redundancy and functional requirements.

Advantages of Primary-Secondary Design – Versatility

PS systems allow for easy combinations of different sizes and kinds of equipment, to allow the plant design to be customized to suit the installation. In a PO system, it is typically necessary that all of the selected boilers be of the same size and type.

For example, imagine a system has a large winter heat load, but minimal summer heating requirements. Say the designer picks three 5,000 KBtu/h boilers for the design day heat load with one boiler as a redundant backup (i.e. duty-duty-standby), with two 1,000 KBtu/h boilers selected for summertime operation (i.e. indirect domestic hot water production and HVAC reheat coils). In summer, only the small boilers will operate, reserving the large boilers for emergency back-up.

If the system is designed with a 30°F (16.7°C) ΔT, the various system pumps will produce 665 gpm during the winter and 133 gpm during the summer. In a PO system, there are two requirements for control. First, the boilers must be individually balanced to receive precisely the amount of flow they are suited for. This balancing (which requires increasing the pressure drop across some of the heat exchangers, possibly increasing the ΔP on the common piping to problematic levels) would be difficult, given the various operational combinations that are possible, and expensive, given the size of the balancing valves needed. Second, the control would have to shut down the small boilers in the winter, and the large boilers in the summer, but they could not be fully disconnected from the system, as they may be required for back-up. This control is achievable but complicated, and may require additional control components to make it functional.

A PS system with the same boiler selections would simply use boiler pumps sized for a 30°F (16.7°C) ΔT across each boiler. Any combination of boilers could be activated and their flow rates combined to match the system performance (i.e. one large boiler at 332.5 gpm and one small boiler at 62.5 gpm could be combined for a 395 gpm boiler side flow rate with the same 30°F (16.7°C) ΔT). Balancing would not be required, and the control would not have to deactivate any boilers, it would simply have to prioritize the order in which various units are activated, based on the outdoor temperature at the time.

It would also be possible to select equipment of different types. The three large boilers in this example could be condensing units, designed for operation from fall through to the spring, to take advantage of the efficiency gains from outdoor reset scheduling, while the small units could be low-cost, copper near-condensing units, designed to operate during the summer when only high temperature supply is required for the domestic hot water production and reheat coils.

Advantages of Primary-Secondary Design – Adaptability

Another advantage of PS piping is that the system could be redesigned during its lifetime with minimal disruption. New loops could easily be added on the system side as building expansions are made, provided the common piping header is large enough to accommodate the extra flow. Some of the units in a boiler cascade could be changed to different products as improvements in technology are made, or if larger or smaller units are needed, without requiring any change of the remaining units.

While PO design requires that all the necessary valves and hardware be installed up front, PS design easily accommodates equipment and efficiency upgrades, like the addition of variable speed boiler pumps, when the capital becomes available.

Advantages of Primary-Secondary Design – Flexibility

PS plants also offer flexibility in energy source combinations. Alternative energy sources like air- or water-to-water heat pumps can also be connected to the PS supply and return headers, with easy control integration. Many hospitals require a back-up fuel be stored on site, and in some cases, the amount of fuel available must be equal to 72 hours of uninterrupted high-fire operation. This can require unrealistically massive propane storage volumes, necessitating No.2 fuel oil be chosen as the back-up fuel. But oil cannot be used for condensing operation[7] so they will have to be installed with mixing valves that can make PO piping impossible. Most oil boilers are not suitable for PO installation anyway, meaning that they will require PS piping injected into the system downstream of the condensing plant. This can complicate flow control, and it will also require that a bypass be included to allow system flow to be diverted around the PO plant, so that the return water does not have to flow through the condensing boilers when they are not being used. In a PS system adding oil backup boilers is comparatively simple: when the condensing units are deactivated, the oil boilers start-up and inject heat into the same location the condensing units had been, and the system operation is unaffected.

If using PS boilers because the system operates with higher return temperatures, the watertube boilers can also be supplied with flue gas economizers built into the boiler's exhaust collector. They can allow supplemental loads to be used to generate condensing efficiency even in high temperature systems that are normally limited to a maximum efficiency of 88%. For example, a tank of cold, potable water can be preheated before being sent to the domestic hot water plant. This economizer, sized for 10% of the fuel input, can push the boiler efficiency as high as 98%, and reduce the domestic hot water fuel bill dramatically (the only supplemental equipment required is a storage tank and small pump).

Conclusion

The adoption of Primary-Only systems and firetube boilers is largely a positive sign of the growing market focus on energy efficiency and reducing fuel consumption, whenever possible. Certainly, if striving to extract every point of incremental fuel savings, PO piping offers the maximum theoretical efficiency.

But most systems are an intricate combination of various heat terminals, temperature profiles and hydraulically distinct loops, and Primary-Only piping will not always be the best option for integrating all of these components with the boiler plant. In systems where it is necessary to operate with high supply and return temperatures, no piping method will be able to extract any greater efficiency than another, and the equipment selection should be based on value to the customer.

This means small footprint, low cost watertube technology, proven heat exchanger designs with high serviceability, and a broad range of acceptable applications. The mechanical realties are not changing: Primary-Secondary design is here to stay.

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Notes:

1. *The dew point of 131°F (55°C) is affected by the quality of combustion and the amount of excess air added to stabilize the flame. At stoichiometric combustion the dew point is closer to 140°F (60°C), but under typical operating conditions the dew point is closer to 122°F (50°C), making it harder to reach condensing temperatures. For the purposes of this paper, the 131°F (55°C) threshold will suffice as an average to establish the scientific framework.*

2. *While the carbon dioxide present in the flue products weighs more than twice what the water vapor does, it requires very little energy to keep in a gaseous state, so there is very little latent energy contained there. It is also, to put it mildly, not easily condensed.*

3. *As the water condenses out, other compounds will be pulled with it, some of which will turn the solution acidic, at a pH of about 3.7. In copper and cast iron boilers, the acidity will quickly pit and corrode the heat exchanger, so aluminum (which has no reaction to pH of 4.0) and stainless steel are commonly chosen, as they are best able to survive the corrosive effects.*

4. *88% is possible if the exhaust itself is stainless steel and can accommodate the formation of condensation. These would be Category II and IV vent assemblies. In the case of Category I and III vents, where the exhaust cannot be allowed to condense, typical boiler exhaust temperatures are 225°F (107°C) or higher, limiting efficiency to 85%.*

5. *Motorized isolation valves do not need to be included to make a PO system function, but they keep deactivated boilers from being fed with partially heated water, which increases standby losses. Return water bypass through an inactive unit will also mix down the supply from active ones, making it harder to meet system setpoint.*

6. *While they are often treated as a "call-for-heat", zone thermostats are best thought of as room temperature limiters. Other than design-day the system will always be capable of over-supplying the zones and surpassing the target room temperature. The function of the thermostat is to stop this from happening. Similarly, the real function of outdoor reset is to use colder supply water to make it take longer to overheat a room, giving the thermostat less to do.*

7. *Because oil contains sulphur, oil condensate will contain sulphuric acid, which will destroy any heat exchanger. Using an oil boiler in a low temperature system will, accordingly, require a three way mixing valve to boost the boiler return temperature. The few firetube boilers which use gas and oil dual-fuel burners require low temperature protection when running on the oil backup for this reason. Most will also require that the boiler be washed before changing back to natural gas: the sulphur SO_3 will condense out of the oil flue gas and accumulate on the surface of the boiler at a higher temperature than the H_2O, but the next time water is present it will combine with the sulphur to form sulphuric acid H_2SO_4, which must be washed away immediately.*

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