Patterson-Kelley

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Short-Cycle Prevention For Double-Digit Savings
(Part 1: Fundamentals & Hydronics)

Abstract: Most hydronic heating boilers short-cycle. This creates mechanical problems and wastes energy. Surprisingly, most of this short-cycling is designed into modern systems. Numerous customers, including many engineers and contractors, have come to us looking for a deeper understanding of the phenomenon and for a definitive solution. This paper addresses these concerns, develops theory, and presents a bullet-proof solution to the short-cycling problem. The paper is presented in three parts.

What's Wrong With Short Cycling?

There are two problems with short cycling, one mechanical, one economic.

The mechanical problem comes from the effects of rapid cycling on boiler components. The burner material for instance, rapidly heats and cools, and sometimes cannot run long enough to dry out. This can create stress and corrosion failures. Gas valves can see decades of use in a few short months. There also tend to be nuisance shutdowns and unexplained flame failures with flame programmer fault codes that have no easily identifiable cause. If you want to make a thirty year boiler fail in five years and drive you nuts in the interim, short-cycle it.

The economic problem is less widely known and appreciated. There is an old rule-of-thumb which says that a short cycling boiler achieves fifteen efficiency points less than the lowest efficiency achieved in non-short-cycling low fire. An atmospheric flex-tube boiler, for instance, that achieves 72% efficiency at low fire will see 57% efficiency in short cycling mode. The loss of fuel efficiency is staggering. If you want an energy efficient boiler plant design, there is often more to be gained from short-cycle prevention than from choosing an ultra-high efficiency boiler.

The Load Profile

The facts are these: design heating loads almost never occur, and boilers spend nearly all of their operating hours only partially loaded. Table 1 shows BIN-HOUR data for Wilkes-Barre, PA, a location showing a typical distri-bution of weather hours for a city in the country's heating zone.

A BIN-HOUR chart like this is made by on counting the number of hours spent during the year at each outdoor temperature. The data is tabulated by creating bins which span three degrees, e.g., from 62˚F to 64˚F, from 60˚F to 62˚F, from 58˚F to 60˚F, and so on, down to the minimum-recorded outdoor temperature. Obviously, each location will have its own distinctive data array. (The data is available from the National Weather Service, and is normally part of the database furnished with energy analysis software programs.) All the hours spent within each bin are entered for the month in which they occur, and the total for all months is shown in the TOTAL ("TOT") column. The result is a chart of how many hours are spent

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Wilkes-Barre, Pennsylvania

AVG T	BIN RANGE			TOT	J	F	Mr	A	My	Jn	Jl	A	S	O	N	D	% TOT	CUM %	% LOAD
63	62	to	64	335				23	45	46	40	45	102	23	11		5.22	5.22	2.78
61	60	to	62	308				20	45	45	48	40	59	40	10	1	4.80	10.02	5.56
59	58	to	60	200				17	39	33	13	42	36	16	3	1	3.12	13.14	8.33
57	56	to	58	273	4		4	29	71	36	22	21	32	45	8	1	4.26	17.40	11.11
55	54	to	56	351	8	4	6	38	68	32	10	17	46	89	31	2	5.47	22.87	13.89
53	52	to	54	285	5	3	28	46	42	16	6	2	38	51	47	1	4.44	27.32	16.67
51	50	to	52	232	2	3	16	35	38	11	1		32	64	27	3	3.62	30.93	19.44
49	48	to	50	238	1	7	18	52	22	5			33	42	47	11	3.71	**34.64**	**22.22**
47	46	to	48	258		4	27	45	23	1			28	58	51	21	4.02	38.67	25.00
45	44	to	46	227	2	2	30	55	22				8	53	39	16	3.54	42.20	27.78
43	42	to	44	287	4	5	69	76	24				12	48	30	19	4.47	46.68	30.56
41	40	to	42	180	3	8	56	42	10				5	26	15	15	2.81	**49.49**	**33.33**
39	38	to	40	273	8	20	50	50	32				4	40	41	28	4.26	53.74	36.11
37	36	to	38	337	24	23	80	46	20				4	41	60	39	5.25	59.00	38.89
35	34	to	36	257	24	31	39	28	11					18	59	47	4.01	63.00	41.67
33	32	to	34	313	46	50	53	8	4					10	69	73	4.88	67.88	44.44
31	30	to	32	360	50	62	71	9						9	68	91	5.61	73.50	47.22
29	28	to	30	298	44	79	45	12						4	49	65	4.65	78.14	50.00
27	26	to	28	187	45	32	34	6							22	48	2.92	81.06	52.78
25	24	to	26	207	85	28	28								13	53	3.23	84.28	55.56
23	22	to	24	142	62	24	32								1	23	2.21	86.50	58.33
21	20	to	22	180	45	45	30								7	53	2.81	89.30	61.11
19	18	to	20	155	37	47	17								1	53	2.42	91.72	63.89
17	16	to	18	105	38	33	7									27	1.64	93.36	66.67
15	14	to	16	113	38	47	3									25	1.76	**95.12**	**69.44**
13	12	to	14	85	46	22	1									16	1.33	96.45	72.22
11	10	to	12	54	18	31										5	0.84	97.29	75.00
9	8	to	10	52	21	26										5	0.81	98.10	77.78
7	6	to	8	38	22	14										2	0.59	98.69	80.56
5	4	to	6	21	17	4											0.33	99.02	83.33
3	2	to	4	16	12	4											0.25	99.27	86.11
1	0	to	2	17	14	3											0.27	99.53	88.89
-1	-2	to	0	13	9	4											0.20	99.73	91.67
-3	-4	to	-2	13	6	7											0.20	99.94	94.44
-5	-6	to	-4	4	4												0.06	100.00	97.22
-7	-8	to	-6														0.00	100.00	100.00
				6,414															

Table 1. A BIN-HOUR chart for Wilkes-Barre, PA showing the average number of hours spent at each outdoor temperature in the range of temperatures for which heating is normally required. A similar chart can be made for virtually any location.

at each outdoor temperature by month and for a typical year.

There are several things worth noting. First, note how few hours occur at the lowest temperatures, i.e., as you approach the outdoor temperature that is used to calculate the heating load. Design conditions are relatively rare, and in many years those conditions don't even occur. Second, note that the hour distribution is multi-modal, i.e., the most commonly occurring temperatures occur here and there across the data.

Note the three columns at the right of the table.

1. "%TOT" lists the percentage of total hours occurring within each bin. The 335 hours occurring in the 63˚F bin represent 5.22% of total hours; the 180 hours occurring in the 41˚F bin are 2.81% of total hours.

2. "CUM%" lists the cumulative percentage of hours as we move from the maximum outdoor temperature at which heating is required down to the design temperature. Thus,

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for instance, the 63˚F bin covers 335 hours which are 5.22% of total hours. We add the 308 hours occurring in the 61˚F bin (which represent 4.80% of total hours), and have covered 10% of total hours with these two bins. The process continues until we have accounted for 100% of total hours. Note that 95% of total hours are covered by the time we reach the 15˚F bin, well above the design temperature. Another way to say the same thing is that the hours between 15˚F and the design temperature represent only 5% of total hours.

3. "%LOAD" converts outdoor temperature to percent of design heating load. These numbers are the result of an interesting thought experiment to which we now turn.

Is it possible to directly relate outdoor temperature to percent of design heating load? The short answer is, no. Building loads differ widely because of variations in architecture, use of space, magnitude of internal heat gains, and so on. But consider this thought experiment. Imagine a building in which all heating zones are perimeter zones, where there is no solar heat gain, and which is heated with 100% outdoor air. In such a case the heating load would be proportional to the difference between the indoor and outdoor temperatures. The load on a typical building clearly cannot be larger than this, and would certainly be a mere fraction of it. This represents a worst case scenario by a large measure. It answers this question: what must the load be less than?

In this imaginary scenario, the heating load for each bin is shown in the "%LOAD" column. With this column added, Table 1 now juxtaposes bin temperature, bin-hours, cumulative percentage of hours, and percent of design heating load. The result is interesting. The median outdoor temperature, the temperature at which there are as many hours above as below, occurs just below the 41˚F bin. The heating load at that point is about 1/3 of the design load, which means that at least 50% of total heating hours see a load that's only 1/3 or less of design. 95% of total hours are covered by the time the outdoor temperature falls into the 15˚F bin, at which point the heating load is only 70% of design. This means that 30% of total heating plant capacity is installed for only 5% of total hours. Remember that the load cannot be larger than this, and is certainly a lot less. No wonder boilers short-cycle. Even if it's only part of the overall problem, boilers are just too big for the real world loads they serve most of the time.

The result of this thought experiment is a clearer understanding of a simple fact: a boiler plant spends nearly all of its time operating at part load, and that actual loads are nearly always a small fraction of the design load. In fact, the loads are so small so often that system performance at design load should rank dead last among the designer's concerns. What matters is the performance of the boiler plant across a range of part load conditions.

This simple understanding is complicated by the fact that design loads do occur, but not in the expected way. The entire boiler plant is energized, not because the outdoor temperature drops, but because heating systems are shut down during unoccupied hours, over the weekend and on holidays, and the system pickup loads can be massive, though usually for a short period of time.

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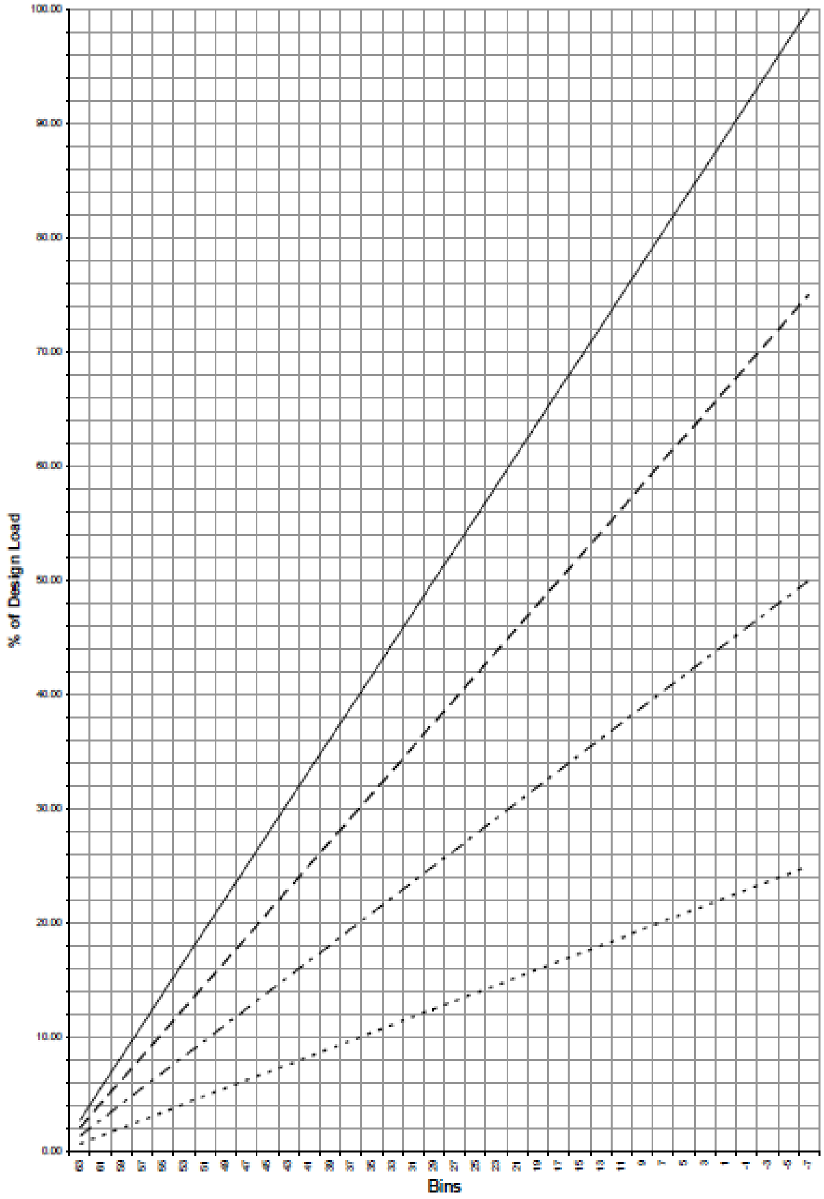

Figure 1. The maximum theoretical load (100%) along with some load curves drawn at fixed percentages.

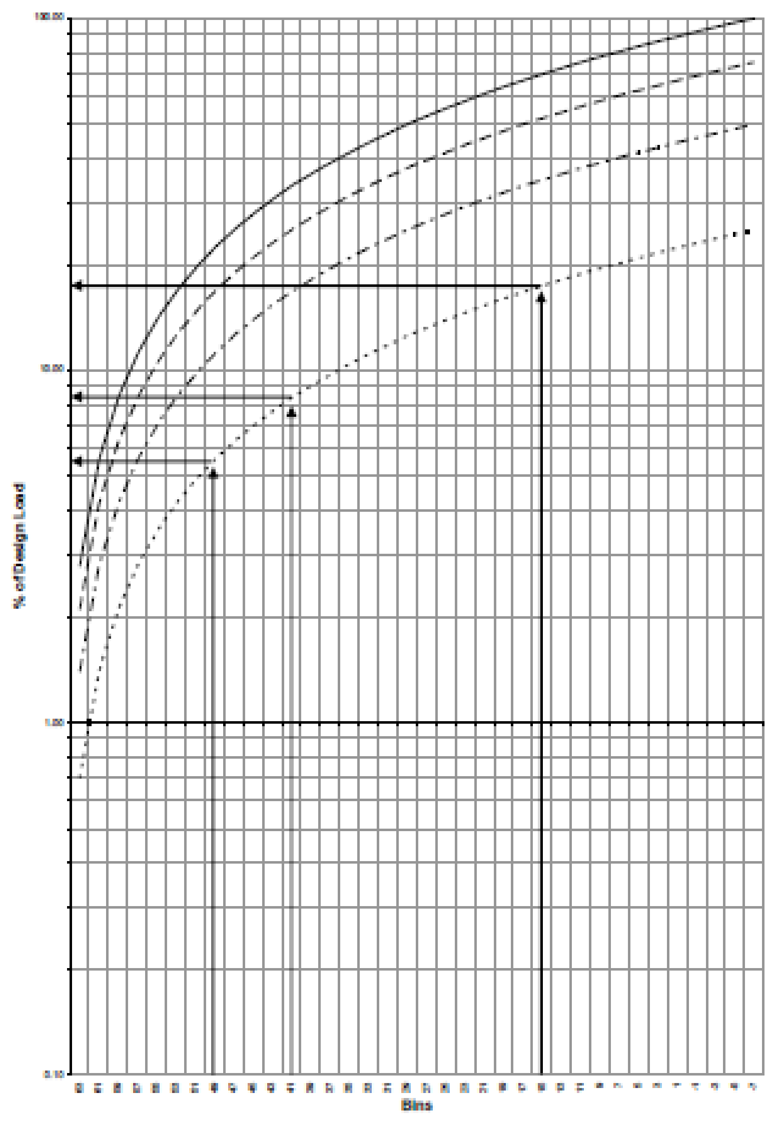

Figure 2. This graph is Figure 1 but on a logarithmic scale. Follow the arrows up from the 15°, 41° and 49° bins to the 25% line, then left to read percent of design heating load for each outdoor temperature.

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Since most of the hours are spent at the low end of the curves in Figure 1, redrawing Figure 1 with a logarithmic scale makes it easier to read. The result is Figure 2, a graphical representation of the results of our thought experiment. It is important to remember that this is a thought experiment, and we should not therefore over-interpret the results. It is unlikely, for instance, that the heating load will be linearly related to outdoor temperature as shown here (the curve is more likely parabolic). What we're after here is a simpler understanding: what's the order of magnitude?

With that said, consider Table 1 and Figure 2. In the CUM% column we find that we've covered about one third of our heating hours by the time the outdoor temperature has dropped through the 49˚F bin, about one half of our heating hours by the time the outdoor temperature has dropped through the 41˚F bin, and about 95% of our heating hours by the time the outdoor temperature has dropped through the 15˚F bin. At these three temperatures, the maximum theoretical loads are about 22%, 33% and 70% respectively. Now follow the arrows in Figure 2. Assume that the actual load is only 25% of these theoretical maximums. We enter the graph at 49˚F, go up to the 25% curve, then left to read a value somewhere between 4% and 5%. Doing the same for the 41˚F bin yields a value between 8% and 9%, and the 15˚F bin yields something just under 11%.

What does this mean? It means that the load is less than 5% of design at least one third of the time, less than 9% of design at least one half of the time, and less than 11% of design for 95% of total heating hours. Are real heating loads really that small? Yes they are. Once past the morning warm-up, they dive to nearly nothing - or at least a small fraction of the design heating load. John Honeck, an engineer with our Minneapolis based representative, Blesi-Evans, took this thought experiment a step further. He took the plans of an existing 100,000 sq. ft. commercial building in Minnesota and used commercially available energy analysis software to estimate the load based on actual internal heat gains. By late morning on a design day - which in this case is -25˚F - the load was only 8% of design. Since the heating system is in place and reheat coil configuration and surface areas are known, he reverse-engineered the system to determine the water temperature required to satisfy the actual load. By late morning on a -25˚F design day the result was 130˚F.

This should not surprise anyone who spends time in boiler plants after they have been commissioned and turned over to their owners. A large junior high school in suburban Chicago operates with only one of six boilers firing on a 10˚F day; a massive high school in suburban Chicago operates with less than 10% of boiler plant capacity by mid-day on a design day. The examples are too numerous to list, and it's time to acknowledge this as engineering fact: boiler plants spend nearly all their operating hours serving small partial loads and should be engineered to do so efficiently. One of the reasons boilers short-cycle is that the boilers designed into systems are simply too large for the loads they serve. This is one reason boilers short-cycle, but not the only reason. It gets worse.

The Boiler-System Interaction

Engineers (and most, if not all, boiler manufacturers) often overlook the short cycling effects of their system designs. A distinction must be made between what equipment is *capable* of doing in the test lab and what a system *allows* it to do in the field. It is clear to us at Patterson-Kelley that many, if not most, systems force boilers to short-cycle because of the way the hydronics and controls are designed. There are deep flaws in the way the industry thinks about piping and control (and this applies with equal force to the boiler manufacturers). We consider five older but common piping arrangements to illustrate the point. We then turn to what is being promoted by at least two manufacturers today as the state-of-the-art. All are flawed from the standpoint of engineering fundamentals, and a new approach to boiler plant design is clearly called for.

All of the following examples ignore at least one of two engineering facts. Fact #1: if you put energy into a system, it must be carried away from the point of input at least as fast as it is being introduced. This means that the designer must consider the relationship between a system's minimum energy consumption rate and a boiler's minimum heat output. Where the boiler's minimum heat output exceeds the system's minimum heat consumption rate, something must be done in system design to create a balance. Fact #2: energy wants to do work, so there must always be a minimum water mass available for the boiler to work on. Remember too that boilers can't read blueprints; they do what systems make them do, not what you want them to do.

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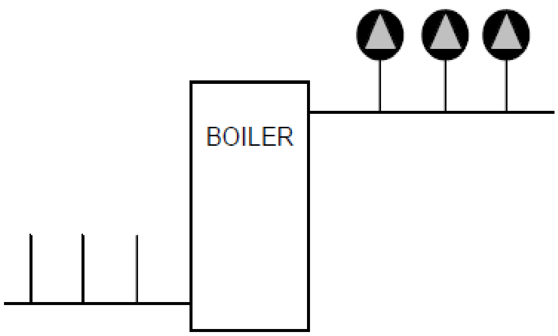
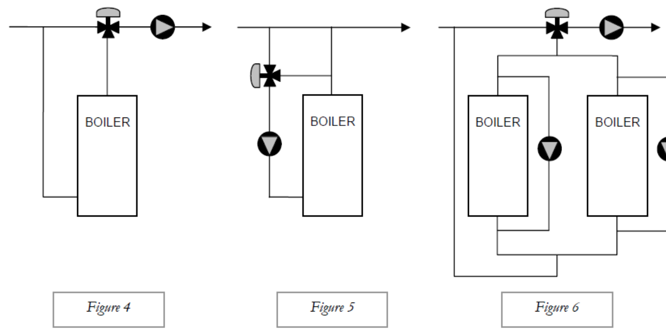


Figure 3. There are thousands of these systems out there, and nearly all become problem jobs when new boilers are installed.

Consider the case of a boiler paired with thermostatically controlled zone pumps. There are thousands of these systems in the field, many of them in churches and apartment buildings. The arrangement is shown in Figure 3. The problem is that the original boiler was a high mass design with a large water content. If the boiler is now replaced with a low mass design, a good move from an energy standpoint, short-cycling occurs. When the boiler fires, its heat output is in the piping RIGHT NOW! Even if it has a modulating burner, whatever heat output is generated is in the piping RIGHT NOW! If only one or two small zone pumps are running, the system flow is not adequate to carry the heat away from the boiler fast enough to prevent the boiler's own (small) water content from rising in temperature to where the boiler's operating limit is tripped off. Once the boiler trips off, this same small flow rate is adequate to quickly cool the boiler's small water content to where the boiler's operating limit is tripped on. Loads from only one or two zones occur more frequently than any other type of load in these systems, so the short cycling condition becomes chronic.



Figures 4, 5, 6. The intent of these designs was to provide outdoor reset control. All cause the boiler(s) to short-cycle.

Consider the case of a boiler paired with a three-way control valve for outdoor reset control. There are thousands of these systems out there in buildings of all types. The arrangement is shown in Figure 4. When the boiler is off, the valve strokes to add more boiler water. With a low mass, low water content boiler, even a slight opening of the boiler side valve port causes the boiler to turn on. Once the main flame is established and heat is being generated, the heat is too much and the temperature overshoots the valve controller setpoint. This causes the valve to begin closing. As it does so the temperature rises in the boiler, shutting it off; but now the hot side input is inadequate, so the valve's boiler side port opens, quickly cooling the boiler and turning the boiler back on. In other words, the boiler short-cycles.

With an older, high mass boiler with its large water content, the control problem was not this serious. Beyond the inefficiency of the older boiler designs, most old boilers required relatively high flow rates to prevent damage from unwanted thermal expansion and contraction. When energy is added in a boiler, it wants to do work; if it can't do the work of heating water, if there is not enough system flow and velocity across the boiler's metal surfaces to cool them at the same rate at which they are being heated, the added energy looks for some other work to do. Most manufacturers dealt with this problem by installing a blend pump between the discharge and inlet to create the flow that the system is not providing. This solves the mechanical/metallurgical problem, the need for turbulent cooling flow across hot metal surfaces, but did nothing to keep the boiler from short-cycling.

Another troublesome system is shown in Figure 5. This

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arrangement is intended to control the temperature of the water entering the boiler to prevent condensation in non-condensing boilers. The problem is that the response rate of valve and boiler are not the same, and the rangeability of the control valve is greater than that of the burner. When the boiler is off, the valve strokes to recirculate water from boiler discharge to boiler supply. When the boiler fires, its heat is added RIGHT NOW! The valve actuator tries to stroke toward the other end of its range, but barely gets moving when the boiler cycles off due to the recirculation of hot water. The water content of the system inside the three-way valve is so small that it takes just a few seconds to either cool it down (starting the boiler) or heat it up (stopping the boiler).

A more recent design is shown in Figure 6. A number of boiler manufacturers quickly endorsed this arrangement, seeing in it a solution to their flow and return water temperature concerns in three-way valve systems. In fact, both of these concerns are dealt with by this design. The short-cycling problem, though, remains, with a dynamic that mimics the one detailed above for Figure 4. The speed with which this design was embraced by manufacturers, however, speaks volumes about how far the industry is from really understanding, and then solving, the short cycling-problem.

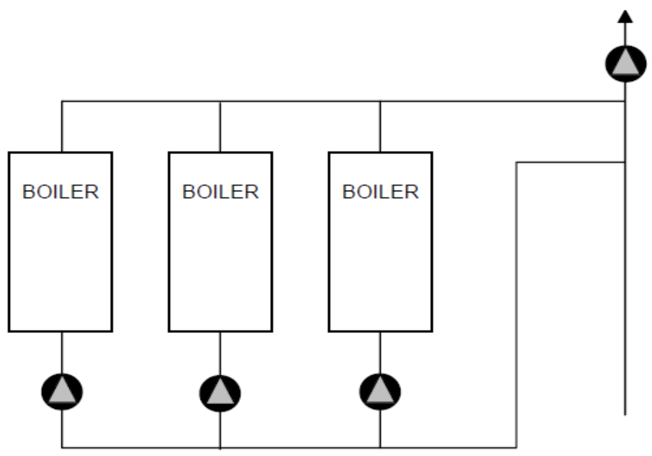

Figure 7. Long recommended by low mass boiler manufacturers, including Patterson-Kelley, this arrangement may have outlived its usefulness. System requirements have changed dramatically since this was first introduced 20 years ago.

Primary-secondary designs are quite common as well, and Figure 7 shows a typical arrangement. These systems can work reasonably well in large, constant flow heating systems. In recent years, however, three-way control valves have been replaced with two-way valves at the terminal units, and constant speed system pumps have been replaced with variable speed pumps. The ASHRAE energy code now mandates variable speed pumping in many applications. Note how each boiler gets its own constant speed pump. Note also how the boilers are piped in parallel in a secondary piping circuit which ties into the system header.

There are two issues here. First, if the boilers are too big for the partial loads, their minimum energy input might be more than is required to raise the temperature of the system water to its setpoint. Since the sensor for the boiler sequencing control is in the system header upstream of the main system circulating pump, it doesn't take long for the excess heat to reach the sensor and cycle off the boiler. When this happens, the sensor sees the loss of heat and immediately calls for more. Second, if system flow is reduced by the action of terminal unit control devices and the main system pump's speed controller, the boiler pump's flow rate might exceed the system flow rate. Flow will reverse in the common piping between the two primary-secondary connections. This rapidly raises the boiler's entering water temperature, and even a small increase in temperature might well trip the boiler's operating limit. Engineers and contractors have often dealt with this problem by moving the boiler sequencing sensor to the return side of the system during commissioning. This works sometimes but not always, and has the disadvantage of giving up any hope of controlling the system's supply water temperature.

Is This Really The State Of The Art?

For some time the primary-secondary arrangement shown in Figure 7 has been the most commonly used method of applying low mass boilers. At least two of the industry's manufacturers have based their marketing attack on their ability to eliminate the boiler circulating pumps and the associated primary-secondary piping. Their claim has been that their boilers are flow-insensitive, that they can fire without damage with no minimum flow. The result is a system like that shown in Figure 8. The boilers are piped in parallel and accept whatever flow the system provides. They have modulating burners with very good to excellent turndown characteristics and are designed for condensing service. A sequencing panel monitors temperature in the common

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piping and modulates the burners to maintain the desired temperature.

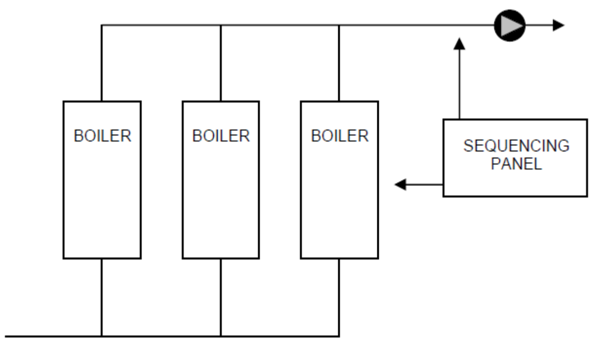

Figure 8. Currently being promoted as the state of the art, this condensing boiler system can be said to miss the point entirely. Either it will short-cycle, or the boilers will be forced to operate at their lowest possible combustion efficiency. Understanding why takes some analysis, but reveals enduring engineering lessons that lead directly to a new design method.

The legendary Gil Carlson once said to me that the purpose of studying hydronics is to develop an intuitive sense of what happens inside a pipe when you turn on a pump. I would add this: the key to designing a boiler plant is to combine this sense with an understanding of what happens when energy is injected into the flowing water at one rate and is extracted at another rate, and where energy input and energy extraction display two very different patterns with respect to time. (A conjecture: at some point someone will work out the linear algebra associated with the control problem arising from these differing rates and establish as mathematical fact that the systems we are installing today are, as currently configured, inherently uncontrollable.)

Consider Figure 8 and let's make some assumptions. The system design load is 1,000 MBH. The owner has asked for two boilers to handle the load (500 MBH output each) plus standby (an additional boiler with an output of 500 MBH). Such a system used to be designed with a flow rate corresponding to a 20˚F△T, or 100 GPM. In this system each boiler will see only 33 GPM under design conditions. Modern systems, however, utilize two way control valves (except for a limited number of three way valves which provide minimum pump flow), and apply a variable speed drive to the main system circulating pump. We know that a common system load is less than 10% of the design load, but also that minimum flow is going to be somewhere around 30% of design. Under these conditions, which will occur during nearly all operating hours, the boilers will each see a 10 GPM flow rate and a very small system temperature drop.

Today's systems aren't designed with a flow corresponding to a 20˚F△T, they're designed with a flow rate corresponding to a 40˚F△T. Thus our flow under design conditions is 50 GPM, not 100 GPM. The boilers see 17 GPM at design conditions. If the main system pump slows to minimum speed, the resulting system flow rate is 15 GPM. Each boiler now sees only 5 GPM. The system temperature drop is still quite small, though somewhat larger than before. The boiler manufacturer says, no problem: we don't require any flow.

The central problem in the industry's discussion of this subject is a matter of mangled rhetoric, and involves a significant miscommunication between the engineer and the manufacturer. The designers question means one thing; the manufacturer's answer means something entirely different. The designer wants to know whether there is any significance to boiler flow rate (there is); the manufacturer's tells the designer that his boiler doesn't care what the flow rate is (a boiler issue, not a system issue). The answer doesn't come to grips with the question.

The question is not whether the boiler needs more flow than the system is likely to provide in order to operate without mechanical or metallurgical damage; the question is what these variable flow rates do to the thermal efficiency of the boiler plant. The question is not whether the boiler will fail; the question is whether the boiler will short-cycle. The question is not whether the boiler manufacturer needs them; the question is whether a system with more optimal boiler flow rates will outperform a system with the random boiler flow rates caused by widely varying system energy extraction rates. The question is not whether the boiler "requires" circulating pumps; the question is whether a system that has them will outperform a system that doesn't. In other words, you might have to remind the boiler salesman, as you might have occasion to tell your teenagers at home, "It's not about you."

Evaluate the issue now based on conditions which obtain during the preponderant majority of operating hours: the system pump at low or minimum flow, 5 GPM system flow

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per boiler, a low mass and low water content boiler design with a large burner modulation range, a system load which is only 8% of design, and a very narrow system ΔT. We add to this the method used to stage and modulate the boilers used in these systems: we measure system supply water temperature and modulate the boilers to maintain temperature at the measuring point. What happens?

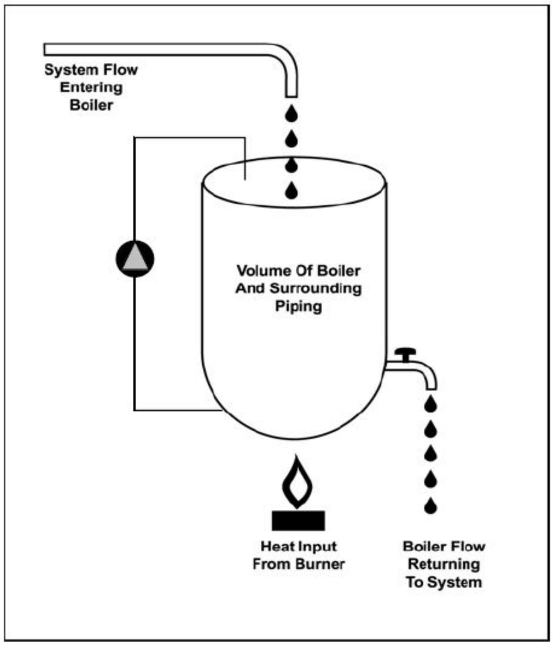

Figure 9. Most modern systems mimic the operation of a bucket with a burner, and the system shown in Figure 8 is no exception. There's no home for the heat.

In situations like this, a simplified conceptual analogy - a model - can be helpful. Most modern low mass boilers can be thought of as "a bucket with a burner" (as in Figure 9). Imagine a bucket with a water content of 10 or 15 gallons. We drain 5 GPM and replace the lost water with 5 GPM at a temperature that is within 2˚F to 4˚F of the desired bucket temperature (if we fall below this temperature the burner fires, if we rise above this temperature the burner turns off). Now turn on the burner and allow it to operate at its lowest possible input setting. How long does it take to satisfy the temperature requirement? And how long does it take, if the burner turns off, to cool the bucket down to where the burner must again fire? The time cycle is calculable (as a problem in related rates), and the answer is a time span expressed in seconds. But wait, this too gets worse.

Figure 8 puts three of these buckets in parallel piped to supply and return headers. Let's assume that the temperature rise within the on-line bucket doesn't cycle the bucket off on its own limit control. The sequencing panel is blissfully unaware of what's happening in the individual boilers. It wants the water to be at setpoint at the sensor location RIGHT NOW! The water flowing through the off-line buckets is not heated. If only one bucket is firing, the net temperature rise across the three bucket system as a whole is only one third of the rise across the fired bucket, and the mix temperature is what the header sensor sees. If this temperature rise does not satisfy the control system, the on-line burner comes out of low fire and begins to modulate higher. It doesn't know to stay in low fire! Now the question becomes whether the on-line burner will be turned off first by its own internal limit control or by the sequencing panel; and then whether one of these controls will be saying one thing when the other control is saying just the opposite.

The conventional wisdom has it that this is where burner modulation becomes important, and the more the better. That's true as far as it goes, but how do we make the boiler stay in low fire if we are measuring a mix temperature in the header? Modulation alone doesn't address that problem: it's an issue of how modulation is controlled. Furthermore, all commercially available boiler sequencing controls - especially those intended for use on modulating boilers - produce their control output based on the offset from set-point at the header sensor location. There is nothing to stop the boiler from firing to a high temperature if that's what's required to instantaneously satisfy the set-point requirement. Assuming that we can get past what appears to be a poorly conceived control strategy - poorly conceived from a short-cycling standpoint - there is another concern which arises from a consideration of heat transfer fundamentals: is it really true that we can modulate a boiler across a wide enough range to make all these problems go away.

Heat Transfer Considerations

The short answer is, no. To understand why requires that we attend to some heat transfer fundamentals.

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Figure 10 is a schematic representation of the heat transfer process that occurs at a boiler tube wall. As every engineer at one time learned, both radiant and convective heat transfer occur on the gas-side of the boiler. Once the tube wall is heated, it heats the inner boundary layer, and this boundary layer heats the water. This is the case for every watertube type boiler. A firetube boiler, in which the fire is in the tube and the water is outside the tube, works in a similar way except radiant heat transfer occurs at the furnace wall, and the balance of gas-side heat transfer surface sees convective heat transfer. The key to understanding modulation is to understand the heat transfer process on both the gas-side and the water side.

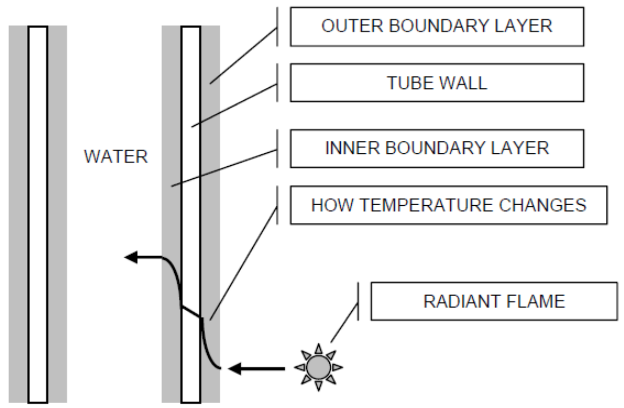

Figure 10. This is the heat transfer process at the boiler tube. The story of modulation is the story of what happens during this process. Water and gas flows must remain turbulent or efficiency drops dramatically.

Water-side. Radiant flame heats the tube wall, and heats gas by convection. How hot is the metal? Finned tubing might have a fin edge temperature of 850˚F. The temperature of the tube wall itself depends upon whether anything prevents heat transfer to the inner boundary layer and then into the water. Now every hydronic system has a fill pressure. Consider a piping system with a vertical height of 35 ft.. The pressure at the top needs to be 5 PSI for air control, so to fill the system from the bottom requires that the system makeup valve be set to provide a system side pressure of just over 20 PSI. This will be sufficient to raise the water to the top of the system and provide 5 PSI at the top. 25 PSI is the factory setting for many of the industry's standard fill valves, so let's assume that the system sees 25 PSI in the boiler room.

What happens when we put water under a pressure of 25 PSI against a piece of metal at, say, 400˚F? It boils. The pressure required to prevent this from happening is simply not found in any hydronic system. This isn't necessarily a problem; in fact, it's expected and beneficial to a point. Very fine steam bubbles form at the tube wall, move away from the tube wall and collapse, and enhance the heat transfer process. You can see this at the bottom of a pot of water on the stove. Without knowing why, professional chefs all learn to dip a spoon in a pot of water that's being heated to boil it more quickly. Breaking up the boundary layer is essential to efficient heat transfer, and it's the turbulent water flow inside the tube that does this and allows the boiler to benefit from this boundary layer process.

Now, reduce the water flow and allow the flow to become transitional or laminar. Suddenly the boundary layer grows and becomes an insulating layer of superheated steam - no longer an enhancer of heat flow, but a highly efficient insulator which prevents heat transfer. Furnace temperatures vary with burner modulation, but the range is narrower than one might suspect. The temperature of the metal in the tube wall varies in a similar manner. Even under light system loads, furnace temperatures remain high, and remain hot enough to boil water under the relatively low system static pressurization. In a variable flow heating system the water flow may well be insufficient to prevent the formation of this insulating boundary layer of superheated steam. The burner may well turn down to very low input, but the loss of water flow due to the reduction in system flow might well, therefore, reduce efficiency and raise the stack temperature. This is not to say that the boiler will be damaged (though this is possible in some designs). The point is rather that the loss of flow might well alter the efficiency of heat transfer and raise the stack temperature, indicating that a progressively smaller fraction of total energy input is making its way into the water.

Fire-side. Fire-side processes are also important, particularly in fire-tube designs. The cross sectional area of the boiler's gas passageways is fixed. As the burner modulates downward, the gas-side flow will go from turbulent to

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transitional to laminar. This transition alters the fire-side U-value. This inner change can be observed as rising stack temperature, an unmistakable indication that, once again, a progressively smaller fraction of total energy input is making its way into the water. The fire-side flow in one of the industry's more popular firetube type condensing boilers, for instance, goes laminar at 40% input. Below this point the stack temperature rises steadily. This same boiler is marketed as a high turndown device. Their claims for the capability of their burner are accurate, though by no means exhaustive. Without knowing the effect of these heat transfer processes on overall boiler operating efficiency, an engineer might unknowingly design a system that achieves its lowest efficiency under light loading - exactly the condition which characterizes the preponderant majority of operating hours.

These processes establish the physical limits of the usefulness of burner modulation. It is mechanically possible to modulate burner input across a very wide range. What is useful from the standpoint of cycle efficiency is only part of what is mechanically possible. This is not to underestimate the importance of using modulating burners in modern systems; but it is important to understand that it is not a panacea. There is nothing so discouraging to an old boiler pro than to see an owner pay a premium for condensing boiler equipment, see it installed in a variable flow, variable temperature, heating system with small partial loads, and watch it operate near low fire with a 395˚F stack temperature when making 130˚F water. You might well have achieved a better result with a cheaper, less capable, boiler.

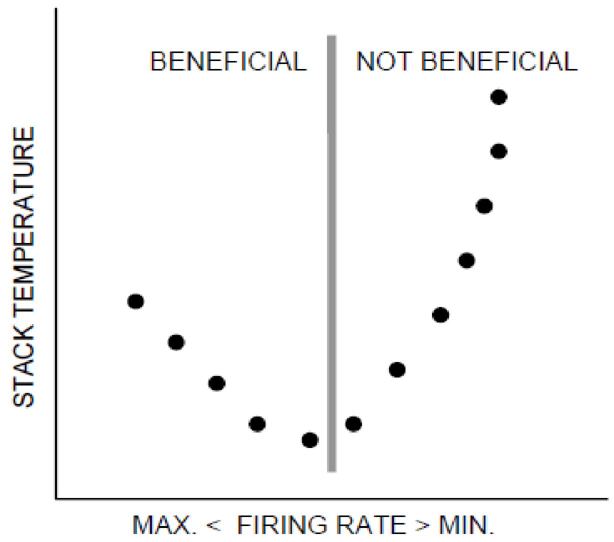

Figure 11. Stack temperature falls with declining burner input in modern boiler designs, indicating rising efficiency. Too much input reduction can lead to a loss of efficiency. It's a poor solution if a high turndown boiler copes with small partial loads by operating on the right hand side of this curve.

Figure 11 shows what's happening as seen from its effect on stack temperature. The most recent boiler designs achieve their highest efficiency at low fire and partial loading. If the heat transfer processes are not understood and considered in boiler design and application, and the boiler modulates down to where the efficiency of heat transfer at the tube wall is lost, high priced boilers can produce low efficiency results. The key to modulation is to use it across the range in which is produces rising efficiency, and to avoid firing the boiler at all in the range in which this is not the case. The question then becomes whether this much turndown - good turndown - is enough to make the boiler track the micro-loads which characterize modern systems.

Some manufacturers have dealt with the water side boundary layer problem by incorporating a boiler recirculation pump into their hardware. It's inside the cabinet, so it is not evident to the observer that they too feel the need to use a "boiler pump." The effect of this approach can be to exacerbate the short-cycling problem. Figure 12 shows why this is so. The situation is as before: the system pump at low or minimum flow, a three boiler system with 5 GPM system flow per boiler, a low mass and low water content boiler design with a large burner modulation range, a system load which is only 8% of design, and a very narrow system ΔT. Now, add an internal circulation pump as shown in Figure 12. Heated water is mixed with the water returning from the system, thus narrowing the temperature difference between the system return water and the desired supply temperature. If the boiler would short-cycle before, it certainly will now, and the boiler will modulate down to where it produces a rising stack temperature. This approach is counter productive.

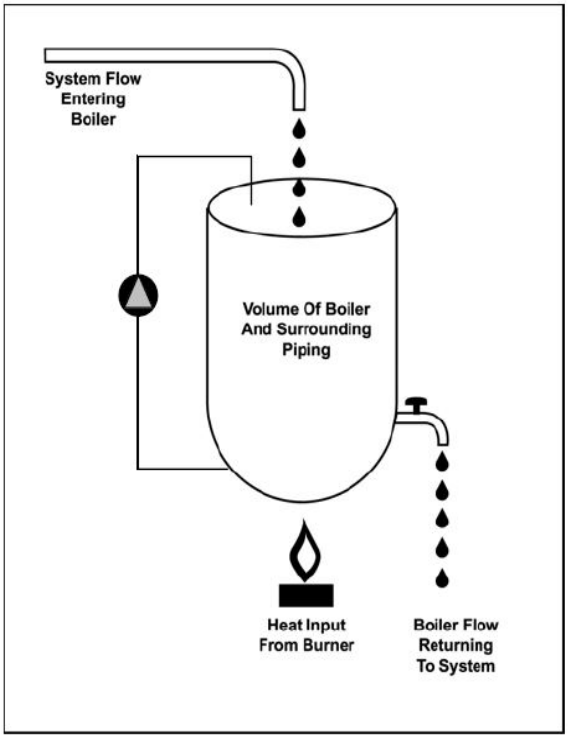

Figure 12. How not to solve the problem. The addition of an internal boiler circulation pump can increase short cycling, or at a minimum, force the boiler to operate in the least efficient part of its modulation range.

The Industry Situation

It is useful to take an inventory of the engineering facts thus far uncovered by this analysis.

✓ Once past morning warmup, heating loads are really micro-loads - loads at single digit percentages of the design load - in modern buildings.
✓ Traditional methods of achieving water temperature reset encourage, rather than prevent, boiler short-cycling.
✓ Boiler plant designs that allow boiler flow to be established by a variable speed system pump are as prone to short-cycling as traditional designs (and probably more so), or at a minimum, force boilers to modulate into a range of operation in which they are inefficient due to the heat transfer processes described above.
✓ Variable speed system pumps require a minimum flow to avoid mechanical damage (often 30% of maximum flow). This flow is often more than the system requires under light load conditions (e.g., loads at 8% of design). The result is a narrowing of system ΔT. One traditionally sized boiler at its most efficient low fire setting is often more heat than the system needs or can handle without short-cycling.
✓ Boilers can be designed to be flow insensitive, i.e., to be fired with minimal flow without mechanical or metallurgical damage. This does not mean that they achieve their best, or even acceptable, efficiency under these low flow conditions.
✓ The system flow resulting from optimal terminal unit performance and the boiler flow that optimizes the boiler's thermal efficiency are almost never the same.
✓ Boiler plant sequencing panels that modulate boil-ers based on the net supply header temperature encourage, rather than cure, short-cycling.
✓ Short-cycling wastes a significant amount of fuel, at least 15% and often 50% or more. In Patterson-Kelley's experience, eliminating it has saved as much as 80%. Short-cycling is not a simple phenomenon, and arises from the interactions that take place between two or more of the processes described above. There are system conditions which arise from optimal performance of the heating system's terminal units, and there are system conditions which are optimal for efficient boiler plant performance.

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These two sets of conditions are rarely, are perhaps only accidentally, the same. What is optimal for the former is usually not optimal for the latter, and vice versa. Attempts by designers to accommodate the needs of the boiler when designing system hydronics can, and often do, compromise the performance of their systems. Ignoring the needs of the boiler creates short-cycling, and the energy lost from it often serves to undo the gains made by state-of-the-art system designs. It seems to me that the needs of the system's terminal units come first because they ultimately determine occupant comfort and the efficiency of energy utilization in the occupied space. It's as though there are really two systems being designed - the heating system and the boiler plant - and that their requirements are always different, usually different enough to make a difference, and therefore ultimately irreconcilable.

A New Approach To Designing Boiler Plants

Central to the argument of this paper are two core beliefs derived from the foregoing analysis: first, that optimizing water flow at the boiler enhances thermal efficiency; second, that every boiler has an input range across which it achieves its highest combustion efficiency. Forcing the boiler to operate with an optimum water flow rate and within the most efficient part of its modulation range - at the same time! - is the skeleton key to maximizing cycle efficiency and, therefore, optimizing overall system energy consumption. Too many planets have to come into alignment for a designer to reliably depend upon the heating system to create conditions optimal for boiler performance, and vice versa. The designer should, in my view, design the heating system with complete disregard for the boiler, and then design the boiler plant with complete disregard for the needs of the system; and then the designer should connect these two systems in a way that doesn't interfere with the performance of either. How can this be done without adding complexity to the design process?

The proposed design process here begins by drawing a buffer tank in the middle of the page. The hydronic system is then designed as though it contains no boilers, and is connected to two nozzles on the buffer tank. Next. boilers need to be thought of differently. With their low mass, low water content and small size, they are almost like shell-and-tube heat exchangers with burners on them. Therefore, think of them as "gas fired heat exchangers," and install them as side-arm heaters connected to the buffer tank. Pump away from the buffer tank to the two systems (heating system and boiler plant) near the top and return water from the two systems near the bottom. The buffer tank becomes the control point for measuring system water temperature and sequencing the boilers.

Figure 13 shows the result in schematic form. The flow rates of the two systems are determined independently, and the boiler's water-side pressure drop is accommodated by the head of the boiler circulating pump(s), and is never included in the system pump head. In an ideal world, the boiler manufacturer would know the optimum water flow rate for each point in the boiler's modulating range, and the boiler itself would be able to control the speed of the boiler circulator to use only the minimum amount of energy necessary to maximize heat transfer efficiency inside the boiler. In fact, Patterson-Kelley has incorporated this capability into the control panel of its new MACH series boiler. The industry's pump suppliers are not yet ready to support this capability in the US, though they are doing so in Europe and have done so for many years. Patterson-Kelley is making the assumption that the industry will head in this direction, sooner rather than later, as it is the natural and logical next step in the evolution of boiler plant design.

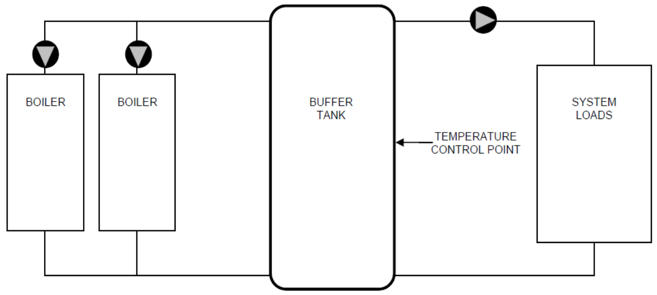

Figure 13. The basic system layout. A variable speed pump serves system loads. Variable speed boiler circulators serve the boilers. The flow paths cross and mix in the buffer tank, which provides an ideal place for measuring the system water temperature.

Observe several things about this system. First, the buffer tank de-couples the system from the boiler plant as in primary-secondary piping. No flow occurs in either system unless the respective pump forces such flow to occur. Second, the flow paths cross in the buffer tank and mixing occurs, making the buffer tank an ideal place to measure the system's water temperature. Third, the buffer tank is located on the suction side of all pumps, so it makes an ideal place for the attachment of the system's compression tank and makeup water valve. Fourth, the expansion in the cross sectional area of the flow path produces a reduction in velocity. En-trained air will most certainly separate here. One approach to air control would therefore be to install a small circulating pump and one of the new generation of air separators as a side-stream device that draws water from the top centerline of the tank and re-injects it at a point near the bottom of the tank. This will probably improve air removal, reduce the cost of the air separation system, and pay for the buffer tank.

Notice also that the designer has total freedom in system design. Any type of system design now becomes acceptable as the boiler plant no longer imposes any restrictions. Retrofit applications are more easily designed and accomplished as the designer no longer needs to consider how changes in boiler plant pressure drop might affect the existing system balance. Boiler sequencing and staging control becomes less critical: it's more like controlling a conventional domestic water heating system. Simpler and less expensive control devices and less elaborate sequences of operation will do a better job of maintaining desired system supply water temperatures.

An accidental benefit of this approach is that a number of challenging system problems are easier to solve. Consider the case of a two-pipe heating-cooling changeover system. Certainly, we will see fewer of these systems in the future as new energy codes discourage or prohibit their use in commercial buildings. Nevertheless, there are many of these in existence and engineers will encounter them as older boiler plants require upgrade and replacement.

One of the problems designers nearly always face in designing such systems is how to arrange mechanical room piping so that the flow rates are right for heating and cooling. Cooling season flow rates are higher than heating season flow rates. The approach shown in Figure 14 makes this part of the design simple. Two system pumps could operate in parallel for cooling, and one could be turned off for heating. The changeover valve on the right side of the buffer tank simply diverts flow through a balancing valve to reset the balance point for heating season operation. This is as simple as it gets. The changeover valves on the left of the tank send water either to the boilers or to a chiller. The buffer tank now serves both the boilers and the chillers. That should please the chiller manufacturer because chiller short-cycling is as deadly for chillers as boiler short-cycling is for boilers, and chiller manufacturers have been actively promoting, if not requiring, the installation of buffer tanks for years.

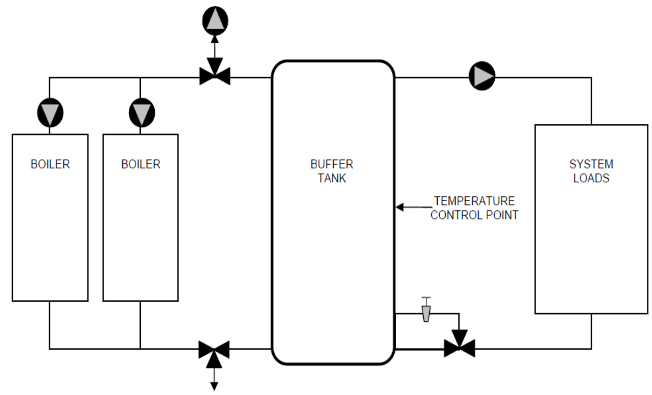

Figure 14. The technique shown in Figure 13 is applied to the two-pipe changeover system. Now three changeover valves are all it takes to change from heating to cooling and rebalance the system to provide the correct system flow rate.

A boiler of a given input must do its work on a minimum mass of water. That's one of those obvious truths that become obvious once someone points it out. Whatever the minimum incremental energy input of a boiler, one thing must happen if the boiler is not to short-cycle: the heat must be carried away from the boiler fast enough to keep the operating limit from tripping off. One of our Patterson-Kelley representatives has a term for this: he calls it creating a "home for the heat." Figure 9, our "bucket with a burner," addresses the same issue. The purpose of the buffer tank is to provide the minimum required heat sink for the boiler when the system cannot be relied upon to provide it.

How big does the bucket have to be to prevent short-cycling? Take the boiler's minimum energy input in BTU's,divide by 60 to get BTU's per minute. This is the

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minimum amount of energy the boiler puts out in one minute. Divide by the weight of water (8.3 lbs./gal.), and divide again by some tolerable temperature difference, say, 20˚F. The result is the required thermal mass the boiler must work on for every minute of run time. This minimum required mass can be provided in one of three ways: a small bucket with a large replacement flow rate; a large bucket operating across a wide △T with a small replacement flow rate; or through a combination of bucket size, △T and replacement flow rate sufficient to swallow whatever the boiler is producing, and have it carried away into the system during the minimum run time.

A modulating burner can make this bucket smaller, but doesn't make it go away. Remember that modulation is only useful across the range of input for which it produces increasing efficiency. Remember too that many modulating boilers don't start at their lowest input setting, but operate at something higher for a brief period for flame stabilization before going to low fire. It's important to know exactly how the boiler operates before deciding what value to use as your minimum energy input.

The sizing formula is simple:

Volume = [t x (QMIN INPUT - QMIN EXTRACT] / [500 x △T],

where

t = minimum desired run time in minutes
QMIN INPUT = minimum boiler energy input
QMIN EXTRACT = minimum system energy ex traction rate
500 = 8.3 lbs./gal. x 60 min./hr.
△**T** = the allowable tank temperature drop.

QMIN EXTRACT for most systems under partial load is so close to zero, it might as well be zero. Zero is the most often used value for estimating purposes. **QMIN INPUT** varies with boiler type and design. Because one often does not know which manufacturer will be the successful bidder for the boiler order, one should generally set this parameter at 50% of the output of one boiler. The results are surprisingly small. For a 4,000 MBH boiler plant configured with four 1,000 MBH condensing boilers, I would use 50% of the output of one boiler as QMIN INPUT. If the boiler has a nominal 92% efficiency rating (1,000,000 x .50 x .92 = 460,000) and I allow a 20˚F temperature difference, the required volume is only 46 gallons for every minute of desired run time. Now, the designer faces a simple question: does the system always give me this much mass for the boiler to work on in any of the three ways listed above?

Note that you can take the tank, turn it on its side, then make it long and skinny while maintaining the required volume. Hang it from the ceiling and it becomes an oversized header, another way of accomplishing the same thing. All that's important is that the system connections be made as shown in Figure 13: returns at one end, supplies at the other, sensing point in between. Note also that tank size determines cycle times and, therefore, cycle efficiency. The system designer is, therefore, in total control of what happens when the system is commissioned. Cycle times and efficiencies are as the system designer makes them.

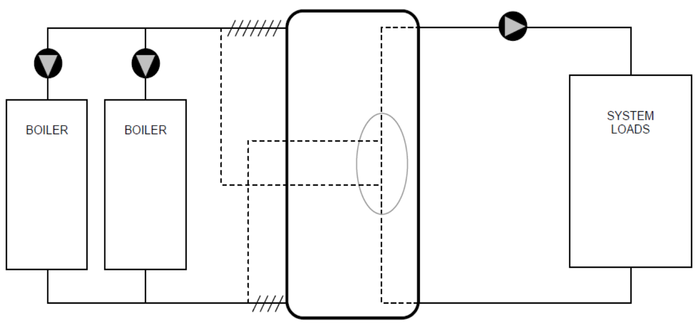

Figure 15. Remove the buffer tank and replace it with the dotted piping and the system become classic primary-secondary with variable speed pumps in both systems. Note the part of the system shown inside the oval. You must understand what happens here.

What if you determine that there will always be an adequate home for the heat? Some larger systems will do this and a buffer tank will not be necessary. In that case, remove the tank and make the piping connections shown as dotted lines in Figure 15. What you get is a primary-secondary piping system, but ideally with variable speed pumps in both the system (primary) and boiler plant (secondary).

Something important emerges here. In the end, the hydronic component of the short-cycling phenomenon has

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to do with happens inside the oval. Precisely here, in these 18" of pipe, the whole story unfolds: the interaction effects of piping, control method, and system volume create short-cycling. It's where the boiler's energy enters the system and where the system flow carries it away from the point of injection. These 1" of pipe also act like a "bucket with a burner." All the proposed approach to system design demands is that we: (a.) evaluate and thoroughly understand what's happening at this exact point in the system, and (b.) adjust the volume of this exact point in the system so that there is always a home for the heat. Period.

This increase in volume, where it is required, can take the form of a buffer tank, an oversized header, or a system flow rate that is always sufficient to carry away whatever the boiler puts into the system. All accomplish the same thing. Burner modulation can and does minimize the volume or system flow rate requirement, but modulation should only be applied on the left side - and never the right side - of the curve shown in Figure 11. Changing the way boiler modulation is controlled and the way boilers are sequenced in multi-boiler systems can also minimize the volume and system flow rate requirement. Understanding what's possible in this regard, and how such strategies are limited in application, requires another paper (see Part II). What's important is that we do nothing with our control strategy that undoes the progress made by fixing the hydronics.

Final Considerations

1. Understanding the energy effect of short-cycling is sometimes made easier by thinking of the issue in terms of the boiler's operating cycle. During pre- and post-purge, the boiler is a reverse heat exchanger that takes heat from the system and throws it up the chimney. When the flame is established, the boiler first replaces this purged energy and then replaces the energy consumed by the heating system. The higher the percentage of time spent turning on and shutting down, the higher the percentage of total heat that goes up the chimney.

2. Light load devices are often found in fire pump systems, house pump systems, and chiller plants. Walk into a boiler room and what do you see? The boilers are all large, all the same size, and all too big for the micro loads. Why don't we use jockey boilers? It seems that the boiler plant is the only building mechanical system that does not include a light load machine. It's time to change that and make a light load boiler a standard part of every boiler plant.

3. Some engineers have reminded us that there will be radiation and convection losses from the buffer tank (if one is required by the system), and that this represents a permanent parasitic energy loss. That's true; but the energy savings achieved to date have been so dramatic that we now dismiss this objection as unimportant. Remember that improving cycle efficiency produces savings of 15% to 50%, and often more. If the cost of achieving this is a 0.25% loss at the tank, it's a price that should be paid. Remember too all of the secondary advantages identified above. It is certainly possible to design a system that connects the buffer tank to the system only when it's needed, only during times when the system does not provide a "home for the heat," but doing so adds cost and complexity, and the owner's future fuel bills will depend upon the understanding and skill of the service technicians who will come and go in the years ahead. It is likely that many of them will not understand what the system does, why it must do it, how it does it, and how to optimize its performance. We should pause and reflect before giving up the bullet-proof characteristic of the approach proposed here.

4. This approach is bullet-proof: it works no matter whose equipment gets purchased by the contractor.

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