SZÉKELY ENGINEERING

EXPLANATIONS & EXAMPLES

Thermodynamics and Knowing Where to Start *or*, Why You Must Know What You Know – Cold.

Any of you who may have spent some time on my website may have not only noticed that I'm a one-person shop providing full MEP/SP engineering services, but may also have inferred that I started my professional life as an electrical designer/draftsman. I took and passed the licensing exam after given only three days' notice that I could sit for it in 1981on the basis of 12 years of, acceptable to the State Board for Engineering and Land surveying, supervised engineering experience.

That experience was pretty evenly split between commercial consulting engineering firms and industrial heavy construction design-build outfits, where infinitesimally few of my colleagues were pure draftsmen who worked from sketches or markups of drawings. Almost all of us designed as we drew, where the distinction amongst electrical designers in the heavy construction world was between physical and "wiring" design, where the former would lay out unit substations, switchgear, motor control centers and the like, while the latter would deal with sizing feeders and branch circuits, and control wiring diagrams, with some, like myself, doing both to the extent of not only laying out all the equipment and designing the lighting power, and HVAC and process motor control systems, but also the short-circuit calculations from the 34,500 volt unit substation to the lights and receptacles to ensure something called selective coordination and keep equipment from exploding under short circuit conditions.

Shortly after I hung out my shingle I got dragged into doing my own Mechanical, Plumbing, and Fire Protection engineering after discovering early in my career that the moonlighters I'd hired to perform those services were more proficient in making mistakes and missing deadlines, than they were in timely submission of well-prepared engineering design packages. I had taken the licensing examinations in the days before one had to stick to a specialty such as Electrical or Structural Engineering in the Principles and Practices portion of the exam, and one could pick and choose among the various disciplines in which to solve engineering problems, which is what I did, knowing just enough about Mechanical Engineering to later get in trouble.

I discovered, as I expect has every engineer on the face of the planet, that the level of mathematics used in engineering doesn't, unless one is working on the bleeding edge of known technology, get much past algebra and trigonometry. To date, on only one occasion, did I see a need to attempt to apply calculus.

Furthermore, discovering in the unaccredited correspondence schooling which prepared me for the licensing examination that HVAC heat loss/heat gain calculations and sprinkler hydraulic calculations did not even rise to the level of the complex numbers of alternating current short-circuit calculations, I settled fairly quickly into my role as a one-person shop and routinely, when designing baseboard hydronic heating systems, selected baseboards based on their 1 GPM flow rate rather than their 4 gpm flow rate as the latter provided only a 6% or so increased heat output at the cost of an increase in flow resistance varying approximately with the square of the flow rate.

Most manufacturers however list fintube heat outputs based on a minimum velocity through the piping of 3 feet per second rather than 1 or 4 GPM. This is because lower velocities do not scrub away enough of the stagnant boundary layer on the inside of the fintube to allow for sufficient heat transfer from the water to the tube.

Not bothering to investigate and calculate the relationship between velocity and flow rate in various fintube pipe sizes, I merrily went along selecting fintube from catalog pages showing 1 GPM and 4 GPM flow rates, picking the former so as to minimize pumping horsepower and piping sizes, completely forgetting about the difference between temperature and heat I've so often harped about in these writings.

This came back to bite me on a three-story building where I'd called for a 1 gpm flow rate in each of 12 hydronic circuits to serve a design load of 417,000 BTUH. Had I stopped to consider that, even at the slightly inaccurate 500 x GPM x Δt calculation for ***heat*** content carried/transferred, a return ***temperature*** about 42 degrees lower than the average supply temperature along with a 4 GPM flow rate would have been required to satisfy the load, I could have bumped up the flow rate in each circuit to 2 GPM to satisfy the actual load, because half of the total load was ventilation air, served by electric duct heaters, and the specified (multi-speed) pumps could handle 2 GPM with no change.

But, before how 1 GPM bit me, how *did* that formula come about? You may remember from your days in High School Physics or earlier issues of my newsletters that a British Thermal Unit is the amount of *heat* it takes to raise the *temperature* of a pound of water by one degree Fahrenheit.

Since a gallon of room temperature water weighs about 8⅓ pounds and a flow of 1 gallon per minute is 60 gallons per *hour*, the amount of British Thermal Units per *hour* (BTUH) imparted per gallon per degree 8⅓ x 60 = 500 times the temperature difference in degrees Fahrenheit so the amount of heat given up by the water is 500 x the flow in gallons per minute x the difference in the temperature of the water between when it entered a heat transfer device such as a fintube radiator and when it left it to go to the next radiator in a series circuit to return to the boiler to be reheated at the end of a circuit.

Most of us *assume* a 20˚F Δt in the formula, the "magic" 20 degrees Fahrenheit, which is *why* I think it came back to bite me.

That is, notwithstanding the facts that at peak load, even 2 GPM and 42˚F Δt, the return water would be (given the standard 180˚F hydronic fintube average supply temperature), *warmer* than the 130˚F at which condensation causes damage to conventional boiler, and that I'd selected a boiler with a 505,000 BTUH net capacity, where, as stated earlier, only about half of the load was served by the baseboard fintube, I was pulled into a lawsuit against the uninsured Architect as a third party defendant, where my insurance company was put on the hook for the beef between the Owner and the Architect because of the "inadequacies" of my design, where the actual inadequacy turned out to be a grossly oversized boiler.

Thermodynamic heat transfer is not only *not* arithmetically proportional, as evidenced by the 6% increase in fintube heat output for a 400% increase in flow rate, it's also dependent on more than one factor, among which are the temperature difference between the heat source and receiver, and the rate of supply of heated source mass.

Or, in English, how much heated water is being supplied and the difference in temperature between the heating water and the room, not to mention heat transfer coefficients between various materials and substances such a copper piping, aluminum fins, air, people, furniture, etc.

The upshot of all this is that at the 1 GPM of my design, because of these complexities of heat transfer, the water would likely have returned to the boiler 35 rather than 42 degrees lower in temperature (making condensation damage even less than an issue) with the result that the fintube might not have satisfied the design load (had it been responsible for the whole rather of it than half of it) for a 70˚F indoor/outdoor design Δt.

So, had I checked it, the 42˚F temperature drop "predicted" by my application of a general rough formula to my 12 GPM total flow, might have made me go to a 2 GPM per circuit flow, giving the Owner nothing really to look to hang his hat on (which would not have necessarily left me out of it). The suit, by the way, happened well before the work was completed and tested.

The point I took from all this, as pointed out by John Siegenthaler in one of his Caleffi hydronics webinars, *Heat Transfer in Hydronics Systems*, (46:06 into a 1:32:50 presentation) is that you can lose sight of intuitive reality when burying yourself in formulae. John's example shows a calculated Δt of 201˚F to satisfy a 50,000 BTUH load supplied by 101˚F water at 0.5 GPM – not bloody likely.

Another point in John's webinar is that most of the heat delivered by hydronic fintube is convective rather than radiant, which makes sense given that the purpose of the fins on fintube "radiators" is to heat the air they're in contact with.

Radiant flooring on the other hand, relies on lots of water flowing slowly through lots of tubing, with flow rates of at about 0.5 GPM of 140˚F water so you don't burn your feet when walking barefoot, with a Δt of 10˚F or more, and *now* you're in the return temperature region where a return mixing valve or condensing boiler is probably required, with GPM and Δt information as hard to find in radiant floor system catalog data, as is heat output vs GPM in fintube catalog data.

Sterling, however, puts out a good integration of their fintube data on a single page, here.