Together Forever

The influence of building materials, furnishings, finishings and a creative solution involving HVAC and interior design. **BY ROBERT BEAN**

Over the decades I have sat in on countless industry meetings and participated in debates over education curriculums for industry members. Typically I sit on the side, most often alone in my beliefs that a large segment of the industry does not actually know what it needs to know. My experience is that when prodded for learning topics, many will ask for more of the same, just presented differently.

Henry Ford probably said it best when he stated: "If I had asked people what they wanted, they would have said faster horses." I am with Henry – educational progress of any consequence is not achieved by "polishing the cannon ball." In this case, the repetitive buffing of knowledge one already possesses.

If you really want to learn you have to follow the advice of business guru Peter Drucker and put yourself in a place where you do not shine and where your lack of knowledge and skills makes you scared, awkward and frustrated. As I say in my classes, learn to embrace the intellectual and emotional pain – it is a sign that you are learning something new. That is often far more useful than taking classes that only confirm what you already know.

So where am I going with this? Well, I have been saying for decades that hydronics and air-based HVAC designers cannot operate in isolation from the world of interior design. There is a non-trivial, consequential relationship between room geometry and finishes contributing to the ergonomics of the very space that the HVAC system is supposed to condition. This relationship directly affects the energy efficiency and indoor environmental quality.

I am going to first illustrate this with the National Research Council of Canada's (NRCC) IA-QUEST[i]; a free, downloadable indoor air quality and emission simulation tool. I will demonstrate this again in a subsequent article in the March 2012 issue using the ASHRAE Nomograph for designing radiant panels. But, before I do this let me be very clear there is a significant difference between the educational and experience requirements of a professional interior designer and interior decorator; something I learned years ago from the folks at ASID[ii].

According to our own Interior Designers of Canada association[iii], "Interior design is about more than just aesthetics. It is about finding creative design solutions for interior environments while supporting the health, safety and well being of occupants and enhancing their quality of life." Does that not just sound like a nice fit with HVAC?

TOOL ASSESSES VOC IMPACT

The IA-QUEST tool was developed, "to help building designers, engineers and managers assess the impact of the volatile organic compound (VOC) emissions from building materials/furnishings on the indoor air quality in buildings." Why is this important? As I have pointed out in previous articles, source control over contaminates is part and parcel of CSA F326 Residential Mechanical Ventilation Systems and ANSI/ASHRAE 62.1(.2) Ventilation For Acceptable Indoor Air Quality.

In fact, it is expected that the HVAC designer should be evaluating potential emission rates to ensure that the designed ventilation system is adequate in consideration of the interior finishes. The IA-QUEST tool, "calculate(s) the concentrations of contaminants that would occur in a ventilated indoor space due to emissions from materials contained within that space."

Sounds like every designer should have a copy of IA-QUEST? But here is what I know to be true, given a choice between learning more about air filters, or floor coverings, the HVAC designer will likely choose to buff up his/her knowledge of filtration instead of learning about VOC emission rates from interior finishes. C'est la vie.

To illustrate the output of the program for floor coverings I have set up the simulation parameters based on *Table 1,* and then for comparison, selected carpeting with underlayment

continued on page 61

Table 1 Simulation Parameters

Room width - 4.5 m
Room length - 7.5 m
Floor area - 33.75 m^2
Room height - 2.44 m
Volume - 82 m^3
ACH, min - unoccupied - 0.05 ach
Ventilation flow rate* - 4.25 m^3/hr
ACH, normal - occupied - 0.3 ach
Ventilation flow rate* - 25.5 m^3/hr
Simulation period - 120 hours
Data output intervals - 15 min

*as calculated by the program

> "If I had asked people what they wanted, they would have said faster horses."
>
> *~ Henry Ford*

and unfinished hardwood from the program's data base. After setting up the ventilation schedule for unoccupied and occupied periods around 600, 900, 1600, 1800 and 2400 hours (*Figure 1*), I ran the calculation to produce comparative emissions for each compound as illustrated in *Figure 2* (hardwood) and *Figure 3* (carpeting). The output units on the y axis are concentrations in mg/m^3 with the x axis showing time in hours.

The uppermost red lines represents the TVOC, or total volatile organic compounds, with subsequent lines representing the various other components emitted during the simulation time. The data "teeth" or "peaks" represent the minimum and normal operational times of the ventilation schedule. Make note of the inventory list of chemicals below the x axis. Not surprisingly, the quantity is much less for the unfinished hardwood than that of the carpet. Likewise, the concentration of chemicals emitted in the former peaks at 0.42 mg/m^3 in comparison to the latter of approximately 2.1 mg/m^3. Due to the quantity of compounds emitted at lower concentrations in carpeting, I have adjusted the output in *Figure 4* so you can see with somewhat better clarity the chemical soufflé that is emitted during the simulation period. *Caveat*: It is not accurate to assume that one simulation is representative of all coverings – most flooring manufacturers have been working diligently to reduce their VOC emissions and their respective data should be used in evaluations.

As with these types of programs, the output values mean nothing unless one can compare them to some value that has meaning to the users of the program, and also in this case, the occupants in the space. Health Canada and other

continued on page 62

Figure 1 Ventilation schedule

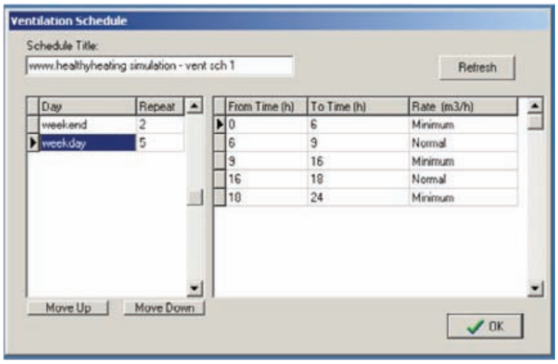


Figure 2 Comparative emissions (hardwood)

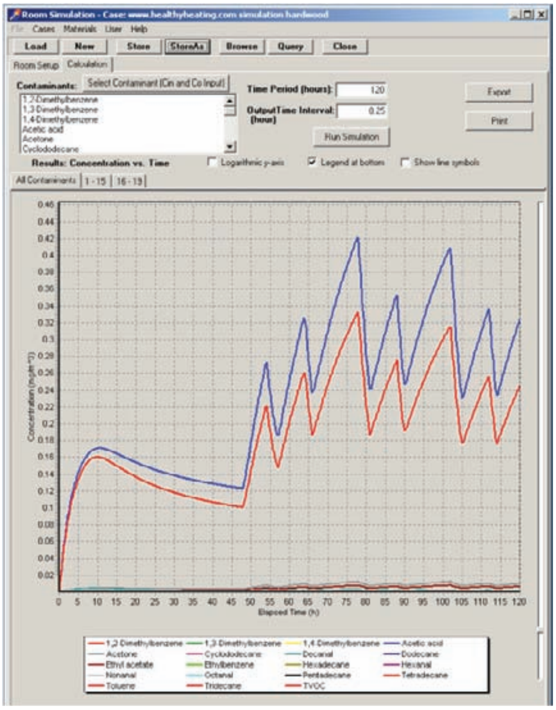
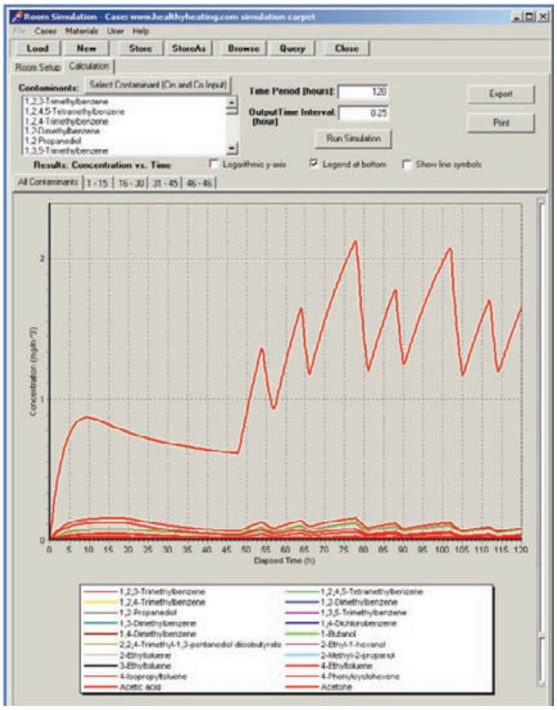


Figure 3 Comparative emissions (carpeting)



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organizations do publish some values that can be used to benchmark against the results, but as anyone competent in the world of IAQ knows, the ultimate test of chemical sensitivity will be the occupants themselves.[iv]

What does this mean? It means there is no shortage of assumptions going on in the world of ventilation design, since many designers never get to talk with the occupants and often do not have access to the schedule of interior finishes and so default to using the ventilation rate prescribed by CSA F326 without considering the influence from interior finishes and need for source control.

In winding up this first part of the demonstration, I want to reiterate that source control à la CSA F326 and ASHRAE 62.1(.2) means having an understanding of interior finishes – meaning interior design is not detached from the world of the HVAC designer. As you will see in the March 2012 article on flooring and radiant systems, neither the hydronics nor the air based designer is immune from this topic – remember this when upgrading your knowledge base. <>

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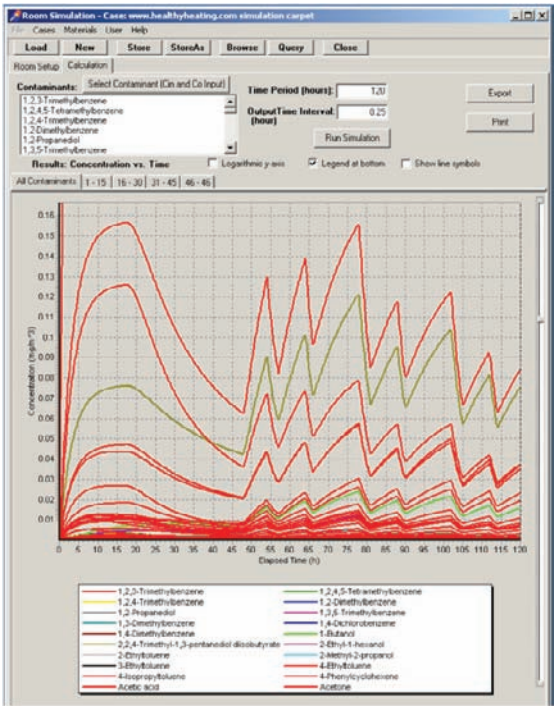
Figure 4 Comparative emissions (carpeting – output adjusted)



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i) http://www.nrc-cnrc.gc.ca/eng/projects/irc/simulation.html
ii) American Society of Interior Designers
iii) http://www.idcanada.org/english/
iv) Charles, K.E. Magee, R.J. Won, D.Y. Lusztyk, E., Indoor Air Quality Guidelines and Standards : Final Report 5.1, Table 6. Guideline Values for Organic Chemicals in Indoor Air (industrial and non-industrial settings) Research Report, NRC Institute for Research in Construction, 204 pp. 36. 2005-03-01

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Part II of my presentation explaining why HVAC designers cannot operate in isolation from interior designers (*see Part I, p.20, HPAC January/February 2012*) will use *Figure 1*, the ASHRAE Design Graph for Sensible Heating and Cooling with Floor and Ceiling Panels. Some of you might look at this and think that the psychrometric chart looks like child's play next to this graph; but like the psychrometric chart, it is simple to use once you know how to navigate around the various components.

So let me start out with the geography of the graph. There are essentially six segments, of which the designers need to establish three through calculation; that being the flux (*Figure 2*); average unconditioned surface temperatures (*Figure 3*); and characteristic panel resistances (*Figure 5*). These and a selection in tube spacing (*Figure 6*) for a heating, cooling or heat/cool drives the temperature difference between the floor and space (*Figure 4*) to emit or absorb the flux at design conditions and the average design fluid temperature in the pipes (*Figure 7*). Your choice in fluid delta t (Δt) will then determine from the average the return temperature and thus the efficiency of the boiler, heat pump, solar system or chiller. It also establishes the upper end of the reset curve for programming your fluid controller. Obviously the factors above are not trivial as all affect the capital and operating cost of the system, the quality of the indoor environment and integrity of the panel materials, which is (again) why radiant should never be treated as a science experiment.

Let's look at each of the steps in using the design graph.

Step 1. Heating or Cooling Flux, Btuh•ft²(W/m²) *(Figure 2)*
Flux in heating is the energy released from the floor through radiation and convection per unit area. In cooling it is the energy absorbed from convection and absorption of long and short wave radiation. At the end of the day the perfor-

Figure 1 ASHRAE Design Graph for Sensible Heating and Cooling with Floor and Ceiling Panels

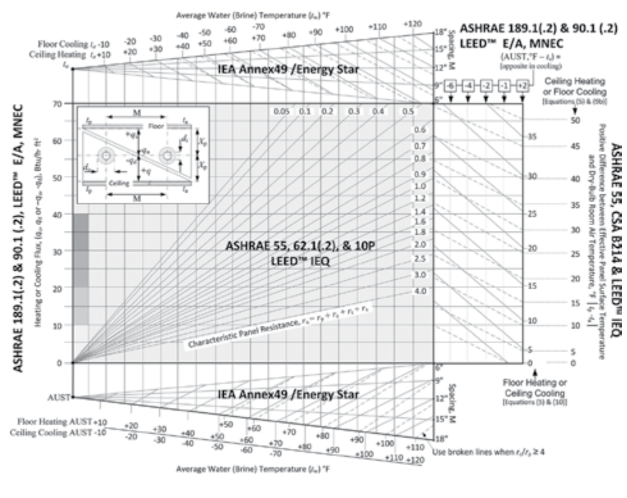


Figure 2 Heating or Cooling Flux, Btuh·ft²(W/m²)

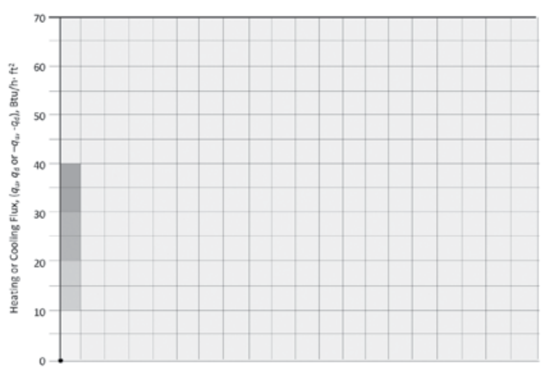
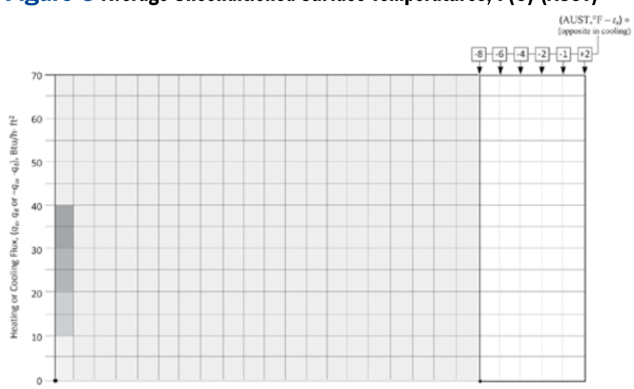


Figure 3 Average Unconditioned Surface Temperatures, F(C) (AUST)



mance of a radiant-conditioned building is judged by its surface flux. High flux represents a bad building, while low flux represents a good building.

I have categorized flux into four regions, as shown in *Table 1*. The short strokes are the higher the flux the hotter the temperatures in heating and the colder the temperatures in cooling. Hot temperatures in heating and cold temperatures in cooling destroy plant efficiency. If you have to operate an embedded pipe radiant system at temperatures typical of baseboards, fan/coils, panel radiators and free standing radiators, you have missed the entire point of using the floor, walls and ceilings for conditioning people and spaces. Low temperatures in heating and high temperatures in cooling are the end game for sustainability – full stop.

Table 1. Categorization Of Building Loads Based On Flux

Category	Flux Btuh • ft^2 (W/m^2)
Terrific	0-10 (0-31)
Transitional	10-20 (31-63)
Traditional	20-30 (63-95)
Terrible	30-40 (95-126)

Step 2. Average Unconditioned Surface Temperatures, F(C) (AUST) (*Figure 3*)

Any surface that is cooler than the heating panel will draw energy out of that heating panel. Likewise, a cooling panel will draw energy out of any warmer surface. All of the unconditioned surfaces (those without pipes) combined represent the cooling or heating load. These surfaces are calculated as an area weighted average for comparison to the conditioned panels (those with pipes) and space temperature for determining the required energy flux and operative comfort conditions. The calculation of AUST is beyond the scope of this article, but for terrific buildings (0-10 Btuh•ft^2 (0-31 W/m^2)) the AUST can be assumed to be at or near space temperatures.

Step 3. Positive Difference Between Effective Panel Surface Temperature and Dry-Bulb Room Air Temperature, F(C) (*Figure 4*)

The greater the flux, the greater the differential required between the panel and space temperature. Surface temperatures are regulated by ASHRAE Standard 55 Thermal

continued on page 130

Figure 4 Positive Difference Between Effective Panel Surface Temperature and Dry-Bulb Room Air Temperature, F(C)

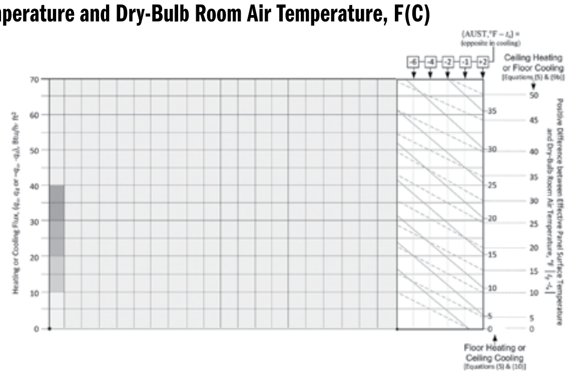


Figure 6 Spacing, in(cm)

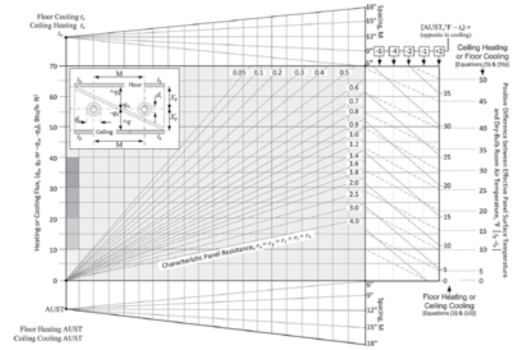


Figure 5 Characteristic Panel Resistance, ft^2· h· F/Btu (m^2· K/W)

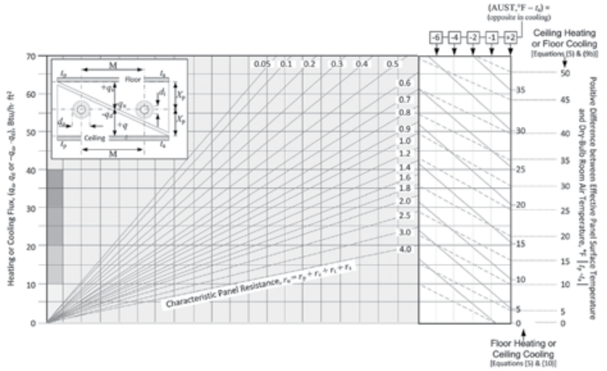
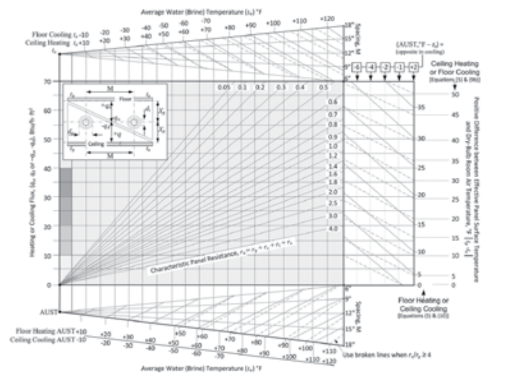


Figure 7 Average Fluid Temperature, F (C)



Environmental Conditions for Human Occupancy based on health and comfort research. In heating 85F (29C) and cooling 66F (19C) are the accepted boundaries for occupants wearing normal footwear. For bare or stocking feet, refer to the 2009 ASHRAE Fundamental Handbook, Section 9.15, Warm or Cold Floors.

Step 4. Characteristic Panel Resistance, ft²•h•F/Btu (m²•K/W) (*Figure 5*)

This is where the meat and potatoes of radiant design take place and the one that is taken for granted especially by the DIY crowd. Radiant panels are on-site fabricated heat exchangers and ultimately the operating temperatures of the fluid are a function of what is known as the fin efficiency of the panel; which is a function of the resistance of: the flooring layers; conductivities of the pipe and encasing materials; pipe surface area; fluid flow characteristics; and log mean temperature differences in the exchanger design. You may have seen some of the finite element analysis (FEA) John Siegenthaler and I have presented on radiant systems. FEA is an excellent way of modelling the heat exchanger to evaluate its efficiency and effectiveness[i]. The calculation procedure again is beyond the scope of this article but the procedures are in the *2008 ASHRAE HVAC Systems and Equipment Handbook,* Section 6.2[ii].

Step 5. Spacing, in(cm) (*Figure 6*)

As noted above, spacing plays a significant role in the effectiveness and efficiency of the panel. It also has a major affect on the surface efficacy, that being the quality of the surface temperature of the panel. To this day it astounds me that individuals try to save money on jobs by reducing the amount of pipe when it has a major affect on operating costs and quality of the system – this despite PEX pipe being the least costly component in a radiant system.

When it comes to heat exchanger designs, pipe is good and the more the better. Since servicing pipe after it is installed is not on anyone's "bucket list," put in the highest quality pipe you can find. Lots of good quality pipe equals low temperatures in heating and high temperatures in cooling, which adds up to the highest efficiency and good comfort.

Step 6. Average Fluid Temperature, F (C) (*Figure 7*)

Average Fluid Temperature is the "keep your eye on the fry" component. Remember it is the average – meaning it is not the supply or return, it is the in-between temperature. That is why they call it the average – go figure. The lower the average in heating and higher in cooling the better. How does this happen? It happens with good buildings with conductive floors and lots of pipe. It is not any more complicated than that.

Here are two examples:

Example 1 Radiant Floor Heating (Figure 8)

1. Entering the graph from the Flux side (25 Btuh·ft²)(79 W/m²), draw a line straight across.
2. Deduct the desired air temperature from the calculated AUST (-2F) (1C) and draw a line straight down to where it intersects the horizontal flux line.
3. Running parallel with the sloped lines for heating, take the line down from the intersection to the boundary and read the differential (approximately 13F)(7C). For a space temp of 70F(21C), the floor temp becomes

Figure 8 Radiant Floor Heating, part 1= 25 Btuh ft² flux, 0.2 floor characteristics, 9-in spacing, 70F space temp, part II = 2.5 floor characteristics

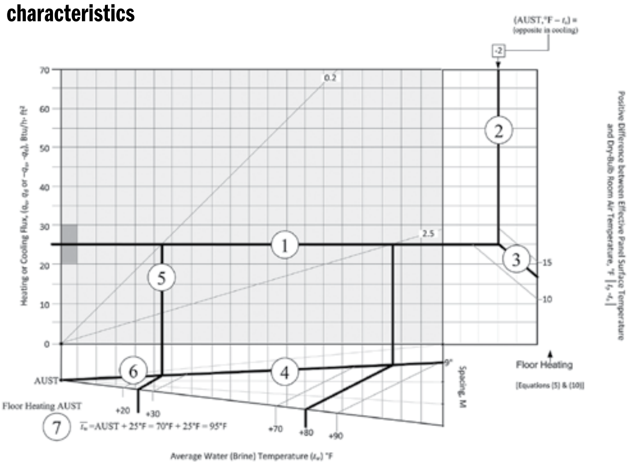
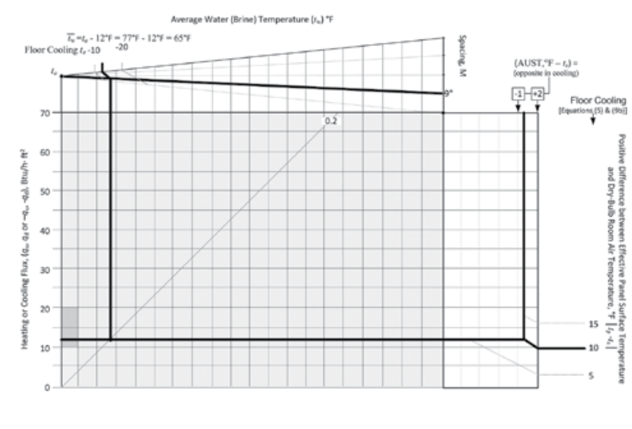


Figure 9 Radiant Floor Cooling, part 1= 12 Btuh ft² flux, 0.2 floor characteristics, 9-in spacing, 77F space temp



70F + 13F = 83F (28C), which is within ASHRAE Standard 55. For those familiar with heat transfer coefficients, you would get similar results where:

[(25 Btuh ft^2)/(2 Btuh ft F)] + 70F = 82.5F (28C)

4. Now select pipe spacing and do not be cheap – more is better! In this case we have selected 9"o.c.(23cm)
5. From the intersection of a hypothetical calculated Characteristic Panel Resistance of 0.2 $ft^2 \cdot h \cdot F/Btu$ (0.03 $m^2 \cdot K/W$), representing conductive low VOC flooring, we take a line down to the 9"(23cm) spacing line.
6. At the above intersection, follow the sloped lines to the boundary and read the value (25F)(14C).
7. Add this value (25F)(14C) to the space temperature of 70F(21C) for an average fluid temperature of 95F(35C).
8. For comparisons do the same with a Characteristic Panel Resistance of 2.5 $ft^2 \cdot h \cdot F/Btu$ (0.5 $m^2 \cdot K/W$) representing a less conductive floor and likely with more VOCs.

Message: The difference between low VOC conductive flooring and less conductive higher VOC flooring is, in this case, the differences between average temperatures of 95F(35C) or 150F(66C) or 55F(31C) spread. Now think about this...with the low VOC conductive flooring and a 20F (11C) delta t, the return temperature becomes 85F(29C); which puts you into the high 90s in combustion efficiency and above a COP 4 for a heat pump – whereas the more resistive flooring puts you out of the heat pump league and drops your combustion efficiency into the mid range regardless of its high efficiency capabilities. All of this is based on 9"(23cm) o.c. which means it could get worse if you decide to go cheap and use wider spacing.

On the plus side, if you do your job and convince the builder and client to improve the building enclosure to get that load down from 25(79W/m^2) to 10(32 W/m^2), you can get that return temp down to 75F(24C) and that does not get any better from an energy efficiency perspective. To put some magnitude around 75F(24C), that would be a nominal 20F(11C) cooler than your blood temp, ergo hot water heating is NOT! There is absolutely nothing hot about these temperatures – tepid yes, but hot no.

Example 2 *Radiant Floor Cooling (Figure 9)*

In this example we are showing a higher performing building with a low flux (approximately 12 Btuh ft^2) (38 W/m^2), and again, a low VOC conductive flooring with the same 9" (23 cm) o.c. spacing. Using the same process except working towards the upper part of the graph for floor cooling, make note that the return temperature going back to the chiller or heat pump would be a nominal 62F (17C) (based on a 5F (2.8C) differential). We all know what cooling with 62F (17C) does to the COPs in your plant. Also make note that the floor temperature for cooling with such high performance is only a nominal 67F (19C) – (or approximately 5F (2.8C) above the return fluid temperature. At 67F (19C) you would be well within ASHRAE Standard 55 and the dew point would be a nominal 60F (15.6C) at 56 per cent RH. By controlling the absolute moisture for microbial control (virus bacteria, moulds), stability of hygroscopic materials (woods) and occupant respiratory and thermal comfort and condensation on the radiant cooling panel becomes a moot point.

THE FINAL WORD

The message in *Part I* regarding the use of the IA-Quest tool for source control over interior finishes is part and parcel of IAQ and CSA F326 and ASHRAE 62.1(.2). It is also in the domain of the interior designer. As you can see from using the ASHRAE design graph in this article, flooring also plays a big role in system efficiency and comfort, as well as IAQ. HVAC and interior design are glued together no matter how you cut it. But here is the upside: clients like their interior designers. In fact, they like them more than more than they like their HVAC contractors (I know that hurts); and the typical client of an interior designer is a woman earning an above average income. You do not have to have a PhD, in public relations or marketing to figure out working with interior design professionals is a no-brainer for the progressive HVAC business owner. <>

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REFERENCES

Figure 1 Design Graph for Heating and Cooling with Floor and Ceiling Panels is reprinted with permission from Section/page 6.8, Panel Heating and Cooling, 2000 ASHRAE Systems and Equipment Handbook.

RESOURCES

i See www.healthyheating.com/Boiler-efficiency.htm

ii See www.healthyheating.com/Effects-of-tube-depth-on-radiant-systems/Effects-of-tube-depth-on-radiant-systems.htm