

Reducing Demands on the Grid: Do Not Let Rogue Zones and Design Software Defaults Strangle Higher Coefficients of Performance.

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ABSTRACT

Reducing demands on electrical grids requires an "everything is open for negotiation" design approach. Design must include getting into the weeds, from addressing rogue zones to challenging software design defaults. The former is typically associated with architectural flamboyancy, which unnecessarily drives cooling temperatures down and heating temperatures up and is frequently associated with non-compliance with ASHRAE Standard 55 Thermal Environmental Conditions for Human Occupancy. Without challenging the latter, zonal heat gain and loss calculations (especially) in rogue zones may have unreasonable weighting on operating specifications; the results from relying on such items as non-representative outdoor and indoor design conditions and assumptions about enclosure assemblies, including incorrect U values and infiltration assumptions and design program defaults. These and other decisions can drive designs into the ultra-conservative range, often leading to oversizing and inefficiencies (Kilkis, 1998). Based on these assumptions, selected control setpoints can further lead to imbalances, excessive cycling, and COP reduction, placing unnecessary excess demands on electrical grids and creating instability in the indoor thermal environment. An extension of these errors is speculations about fluid characteristics and relying on program default differential temperatures, which drive the flow calculation. In turn, this misguides the selection of pipes and ducts, valves and dampers, which impacts heat terminal units, circulators, and fans. The errors result in inaccurate velocities, pressure drops, and control instability, again all tied back to discomfort and energy use.

To reduce loads on electrical grids, lift and COP must be intentionally targeted by understanding the heat sink, load profiles and appropriate system types; then served by the lowest possible heating supply and highest possible cooling supply. Then, where necessary, the architecture, enclosure, interior elements, and space heat exchanger characteristics can be manipulated to move the required thermal load with the least amount of electrical energy. The final fidelity of the system comes from harmonizing temperatures to their optimal setpoints for simplicity and higher COP (Bean, 2023).

Consequently, all design elements must become negotiable to achieve compliance with ASHRAE Standards, with the byproduct being load reduction on electrical grids, which is only possible with passive solutions first. That is, driving loads down through architecture, enclosure, and interior design, followed by choices in mechanical/electrical systems. Using and optimizing radiative-based systems to solve radiation problems and convective-based systems to solve convection problems.

This paper will demonstrate the benefits of negotiating select design elements and harmonizing temperature profiles to achieve manufacturer-rated COP for reducing grid demands while simplifying control setpoints.

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INTRODUCTION

Traditional design processes work in a top-down hierarchy where clients and architects rely on engineering services to achieve the project's technical objectives. Though robust, the tools afforded to engineers are rarely used to work backward to drive architectural design. Without bottom-up negotiation, many assumptions are made within tools and processes that sacrifice the energy and indoor environmental quality (IEQ) potentials latent in the design process. Regarding the former, it is the preservation and conservation of energy and energy and exergy efficiency in the equipment and systems. The latter is in compliance with IEQ standards defining sound, thermal, air, lighting, odours, and vibration qualities.

This paper uses ASHRAE's Design Graph for Heating and Cooling with Floor and Ceiling Panels (ASHRAE 2023) to represent an inventory of graphical and digital tools utilized throughout the design practice. It is an encompassing example that combines architectural, enclosure, interior, and mechanical/electrical design. Since the tool ultimately dimensions the average fluid temperature for the heat exchanger, it is tied directly to the heat pump lift, and COP thus sits inside the intentions of ASHRAE's energy standards, which ultimately influences the load on electrical grids. Additionally, the tool is linked to thermal comfort (MRT, asymmetry, surface temperatures) and air quality (temperature-driven outgassing and particle generation), so it sits inside the intentions of ASHRAE's IEQ Standards and Guidelines.

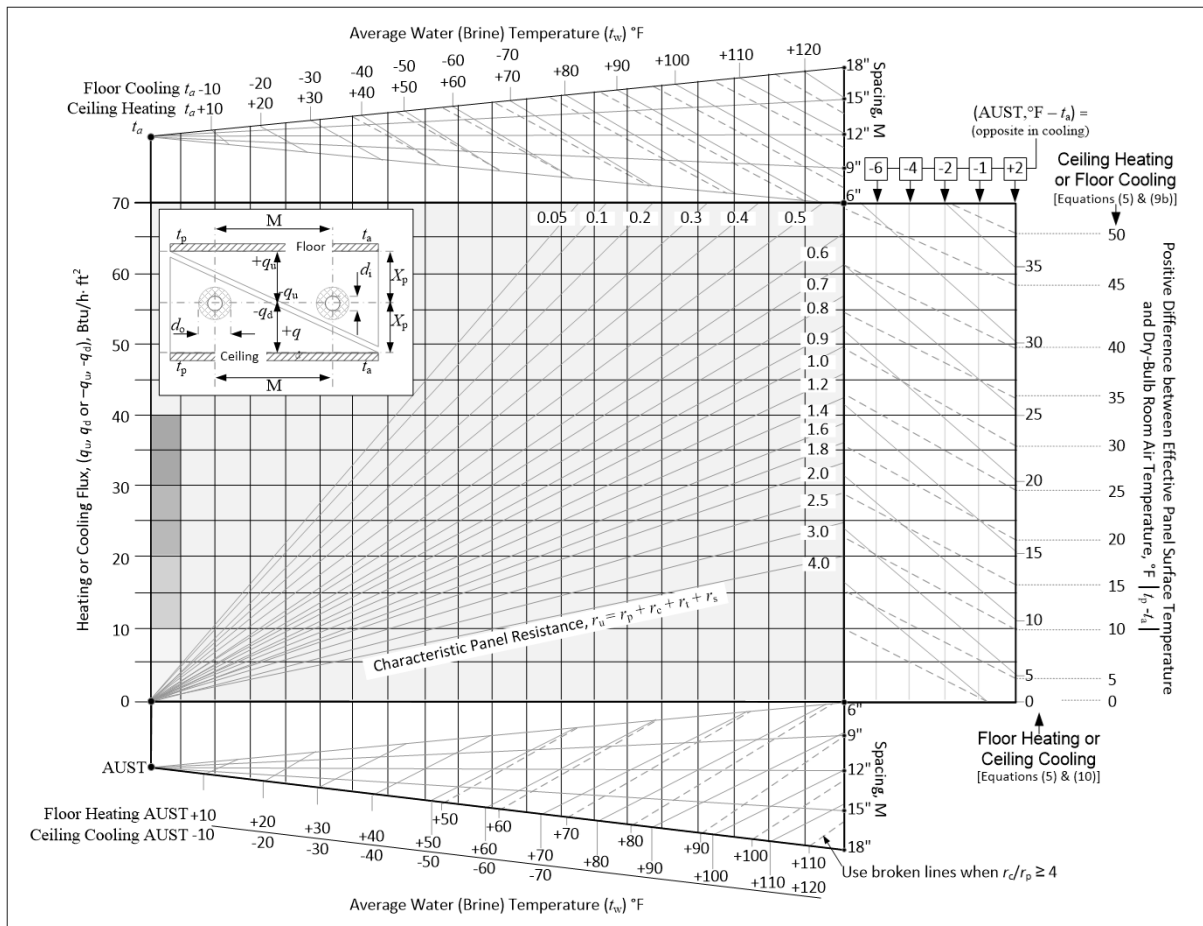


Figure 1 ASHRAE's Design Graph for Heating and Cooling with Floor and Ceiling Panels (for SI version, see ASHRAE 2024, HVAC Systems and Equipment Handbook, Chapter 6:8)

To illustrate fundamental relationships relative to the paper, the elements of the graph of importance are;

1. Flux, heating or cooling, q_u -upwards or q_d -downwards transfer (y-axis, left)
2. Characteristic Panel Resistance (slope lines originating at the xy-axis)
3. Spacing (y-axis right)
4. Average Water temperature (x-axis)

Flux, q_u , q_d , (Btu/h ft² (W/m²))

Flux represents the thermal load over the area of the heat exchanger and can be a direct indicator of the architecture and enclosure performance. When a single heat exchanger is used to condition large areas or multiples of smaller areas, the fidelity of understanding hidden rogue zones is lost, which might negatively influence the overall design. In comparison, using good zoning practices, multiple heat exchangers selected will reveal those atypical zones (e.g. see zones 3 and 9 in Table 2 and Figure 6). The performance of those "rogue" zones needs to be negotiated with the architectural and enclosure design teams so that rather than being the outlier, they can fall in with the typical zones (e.g. see zones 3 and 9 in Table 3 and Figure 6). Then, the performance of the entire inventory of zones should be negotiated to reduce the loads to the lowest possible values. The authors' sample guide in practice is shown in Table 1, Heating Flux Categories for Cold Climates. As a practical illustration, when architectural and enclosure designs result in fluxes of less than 10 Btu/hr ft² (32 W/m²) and when coupled to less resistive (more conductive) panels (<0.5 °F•ft²•hr/Btu (<0.09 m²k/W)) when using tight tube spacings (<8"o.c. (<200mm)), the resulting fluid temperature profile (<100°F (38°C)) enables lower lifts and maximizes COP. Consider the impact on decisions to use air source or ground source heat pumps where the former might max out at approximately -4°F (-20°C) sink temperature and may require auxiliary heat depending on location and season. At approximately 15ft (5m) ground temperature, a stable sink of approximately 50°F (10°C) exists in the latter. Excluding the impact of soil conductivity on heat transfer, the difference in lift for the former could reach 90°F (56°C), at which point auxiliary heat is required, whereas the latter will be less than 50°F (28°C) and not require auxiliary heat. There is a consequential difference in COP between these systems and requirements for auxiliary heat; thus, it cannot be ignored if energy policy concerns include the ever-expanding load on grids.

Table 1. Heating Flux Categories for Cold Climates.

Flux Category Descriptors	IP (Btuh/ft ²)	SI (W/m ²)
Terrible building	>30	>95
Traditional building	20-30	63-95
Transitional building	10-20	32-63
Terrific building	<10	<32

Characteristic Panel Resistance, r_u (°F • ft² • hr/Btu (K•m² /W))

Characteristic panel resistance (aka fin efficiency) describes the resistive (inversely, conductive/convective) features of the heat exchanger (Bean 2024). In this paper, radiant panels are held up as an example, but the principles apply to all heat exchangers in the simplified form of:

$$q = U A \Delta t_{lm}^n \quad (1)$$

Where,

- q = heat transferred
- U = heat exchanger conductivity
- A = heat exchanger surface area
- Δt_{lm} = log mean temperature differential
- n = convection coefficient

With embedded pipe radiant panels, the exchanger is typically fabricated on-site and includes the conductive properties of the construction materials used in the assembly (r_u in Fig 9 and U in Formula 1), including the finished surfaces that the interior designer or decorator specifies. Thus, any surface finishes and assemblies in the exchanger that do not promote the highest conductivity work against the objective of lower lifts and higher COP. Additionally, lower conductive surface finishes (e.g., carpets, engineered woods) are often synthetic and comprised of a smorgasbord of chemicals released through outgassing (i.e.,

VOCs) and particle generation when exposed to sustained shortwave radiation and higher exchanger temperatures. Meanwhile, masonry-type surfaces such as tile, slate, terrazzo, and stained concrete have far more stable characteristics, are more environmentally hygienic, and have conductivity that enables lower heating and cooling temperatures. Additionally, the larger the exchanger's surface area (A), the better the conditions for extracting maximum system efficiencies due to its impact on temperatures (Δt_{lm}), especially when coupled with lower fluxes (q_u , q_d). Increased heat exchanger surface area and fin efficiency should be detailed tactics when minimizing the effects of rogue zones. For this reason, and especially with radiant-based HVAC systems, surface coverings and exchanger assemblies must be negotiable; thus, interior designers and decorators cannot ignore their role in electrification and decarbonization.

Spacing, M (inch (mm))

In any heat exchanger, tube spacing (M) directly impacts fluid temperatures (t_w) and, thus, lift and COP. Wider spacings destroy energy potentials, whereas narrower spacings promote energy potentials. Observe the fluid profiles in Figures 2, 3 and 4, respectfully for spacings of 6"o.c. (150mm), 9"o.c. (225mm) and 12"o.c. (300mm). Whether it has embedded pipes as in radiant systems or rows and fins in an air handler's coil, these characteristics represent a minuscule percentage of the overall mechanical costs yet affect thermal efficacy across the exchanger surface, thus comfort and fluid return temperatures and a lifetime of energy efficiency. The heat exchanger characteristics also contribute to equipment choices (heat pump vs boiler), thus impacting the lifetime exergy efficiency of the system. For these reasons, the heat exchanger designed for space conditioning is the last place to look for cost reductions. A strong argument must be made that a heat pump potential COP should drive design fluid temperatures and, therefore, drive the architecture, enclosure and interior design; otherwise, why invest in the high COP device? If the logic is defensible, then tube spacing defaults in programs should be negotiated and always selected to drive space heating and cooling temperatures into a range of 77.5°F $\pm$ 22.5°F (22.5°C $\pm$ 12.5°C) (Bean 2013). In these conditions, the lowest heat pump lifts and highest COP are achieved.

Average Fluid temperature, t_w (°F (°C))

The average fluid temperature reveals the consequences of the combination of architecture, enclosure, interior design, and tube spacing. The best combinations drive heating fluid temperatures down and cooling temperatures up. For example, note the ten-degree differential between Figure 1 and Figure 3 at 20 Btu/h/sf (63 W/m²) for 6"o.c (150mm) and 12"o.c. (300mm) in bare concrete ($r_u=0$). As this is the average fluid temperature, the actual effect on lift and COP appears on the practitioners' selected or programs defaulted differential temperature in the flow calculation. Any number of combinations of terms (q , r_u , M and t_w) will influence the outcomes.

The ideal optimized combination of design conditions enables heat pumps to achieve their actual rated efficiencies. Thus, the argument must be made to work backwards from lift and COP to drive the heat exchanger log mean temperature differential (Δt_{lm}) and fluid flow temperature differential (dT), characteristic (r_u) and flux (q_u , q_d) via architecture, enclosure and interior design. Within the above framework, the most skilled, persuasive and capable mechanical designers enable load reductions on the grid.

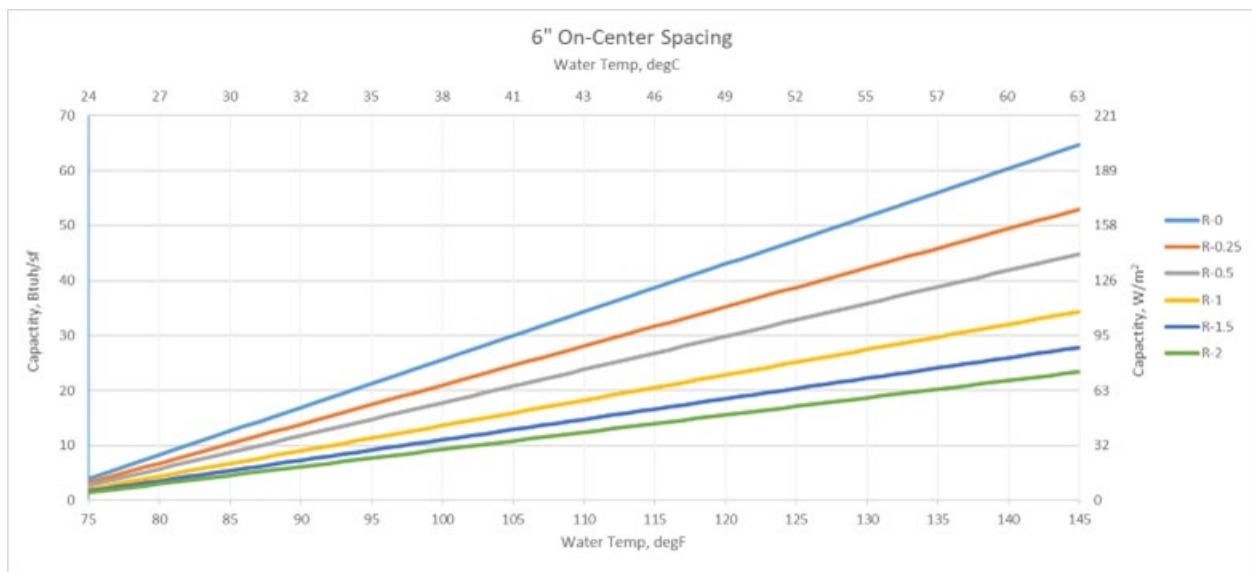


Figure 2 In any heat exchanger, tube spacing (M) directly impacts fluid temperatures (t_w) and, thus, lift and COP. Wider spacings destroy energy potentials, whereas narrower spacings promote energy potentials. The average water temperature for 6"o.c. (150mm) (M) is shown here at various characteristic panel resistances (r_u) for a required flux (q_u).

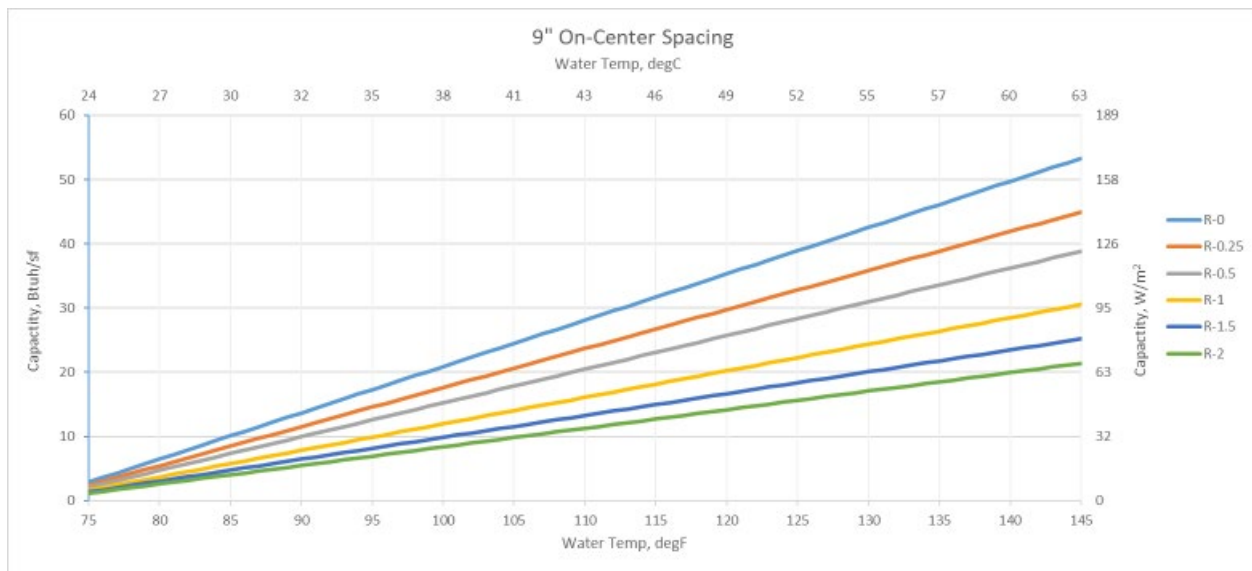


Figure 3 The average water temperature (t_w) for 9"o.c. (225mm) (M) is shown here at various characteristic panel resistances (r_u) for a required flux (q_u).

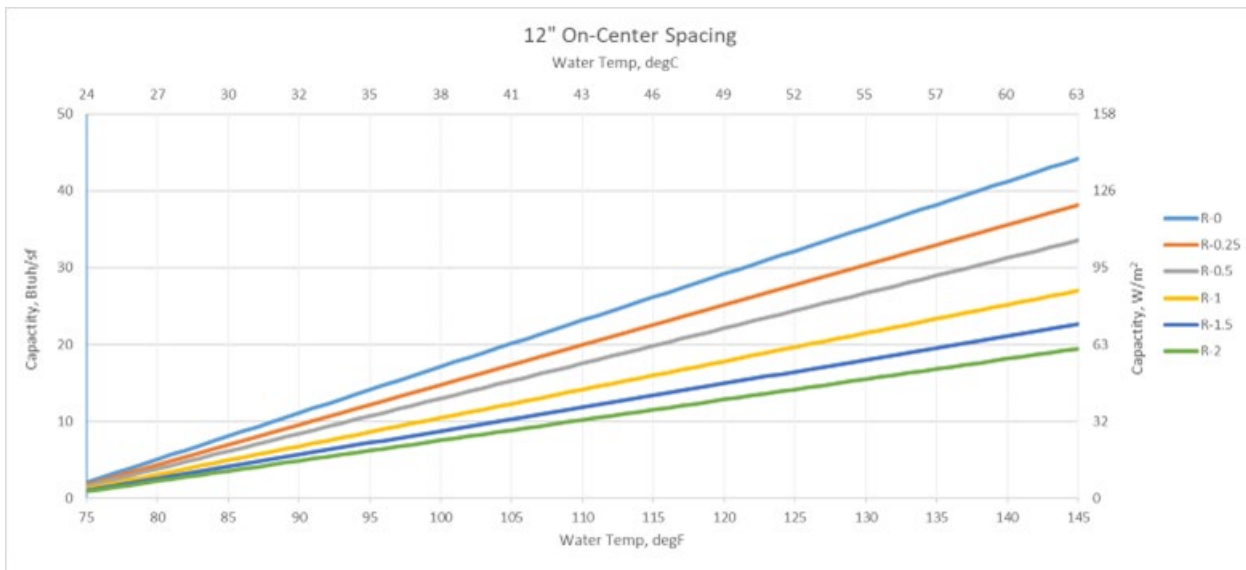


Figure 4 The average water temperature (t_w) for 12"o.c. (300mm) (M) is shown here at various characteristic panel resistances (r_u) for a required flux (q_u).

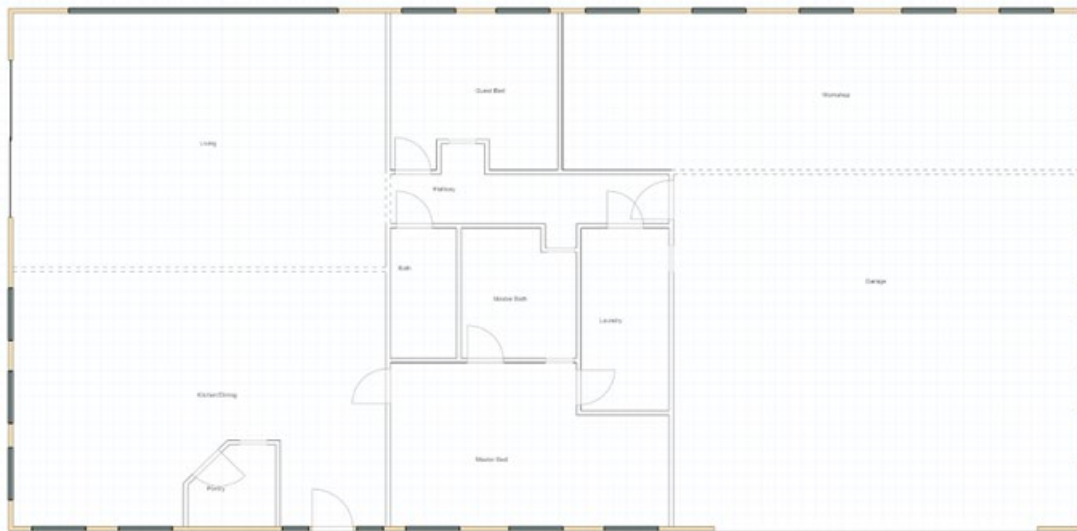


Figure 5 Case study layout for an 11-zone residence

It is typical in the landscapes of design to have an inventory of non-negotiated temperature profiles in a project, which prevents optimization and efficiency and enables complexity and unnecessary capital costs. For the Figure 5 case study layout for an 11-zone residence, Table 2 presents the average water temperature outcomes without negotiating previously discussed elements. Table 3 presents the optimizations by manipulating load (Q), flux (q), panel characteristics (r_u), spacings (M) and differential temperatures (dt). These are compared in Figure 6, a comparison of defaults versus level 1 optimization for harmonizing heat exchanger temperature profiles. Greater harmonization is possible with deeper level 2 or 3 analysis.

It is demonstrated (Table 3 and Figure 6) by negotiating architectural, enclosure, and interior and mechanical design criteria; temperature inventories can be harmonized (closer ranges and flatter data) to optimum setpoints for energy and exergy efficiency and simplicity in controls for capital cost reductions (Bean 2024).

Table 2 Original defaults for average water temperature outcomes without negotiating elements.

Zone	Label	Area A,ft ²	Panel r _u	Spacing M, inch	Zone load Q, Btuh	Flux Q, Btuh/ft ²	t avg, °F	dt °F	t supply, °F	t return, °F
1	Bath	49	0.1	12"	266	5.4	81	20	91	71
2	Garage	809	0	12"	21104	26.1	115	20	125	105
3	Guest Bed	140	2.1	12"	2729	19.5	145	20	155	135
4	Hallway	89	0.7	12"	488	5.5	85	20	95	75
5	Kitchen/Dining	495	0.7	12"	10077	20.4	117	20	127	107
6	Laundry	91	0.1	12"	482	5.3	81	20	91	71
7	Living	537	0.7	12"	12253	22.8	122	20	132	112
8	Master Bath	80	0.1	12"	426	5.3	81	20	91	71
9	Master Bed	231	2.1	12"	4323	18.7	144	20	154	134
10	Pantry	36	0.7	12"	703	19.5	114	20	124	104
11	Workshop	461	0	12"	14992	32.5	126	20	136	116

Conversions: A, 1m²=10.76ft²; r_u, 0.02 K•m² /W = 0.1 °F•ft²•hr/Btu; M, 300mm = 1ft ; t °C = (°F - 32) × 5/9; q 100 W/m²=32 Btu/h•ft²

Table 3 Outcomes with level 1 optimizations manipulating load (Q), flux (q), panel characteristics (r_u), spacings (M) and differential temperatures (dt).

Zone	Label	Area A,ft ²	Panel r _u	Spacing M, inch	Zone load Q, Btuh	Flux Q, Btuh/ft ²	t avg, °F	dt °F	t supply, °F	t return, °F
1	Bath	49	0.1	12"	<u>215</u>	4.4	79	20	89	69
2	Garage	809	0	12"	<u>15788</u>	19.5	104	20	114	94
3	Guest Bed	140	<u>1</u>	<u>6"</u>	<u>2453</u>	17.5	109	<u>10</u>	114	104
4	Hallway	89	0.7	12"	<u>396</u>	4.4	82	20	92	72
5	Kitchen/Dining	495	0.7	<u>6"</u>	<u>9196</u>	18.6	102	20	112	92
6	Laundry	91	0.1	12"	<u>391</u>	4.3	79	20	89	69
7	Living	537	0.7	<u>6"</u>	<u>11432</u>	21.3	107	<u>12</u>	113	101
8	Master Bath	80	0.1	12"	<u>345</u>	4.3	79	20	89	69
9	Master Bed	231	<u>1</u>	<u>6"</u>	<u>3866</u>	16.7	107	<u>10</u>	112	102
10	Pantry	36	0.7	<u>9"</u>	<u>568</u>	15.8	102	20	112	92
11	Workshop	461	0	12"	<u>14992</u>	32.5	126	20	136	116

Conversions: A, 1m²=10.76ft²; r_u, 0.02 K•m² /W = 0.1 °F•ft²•hr/Btu; M, 300mm = 1ft ; t °C = (°F - 32) × 5/9; q 100 W/m²=32 Btu/h•ft²

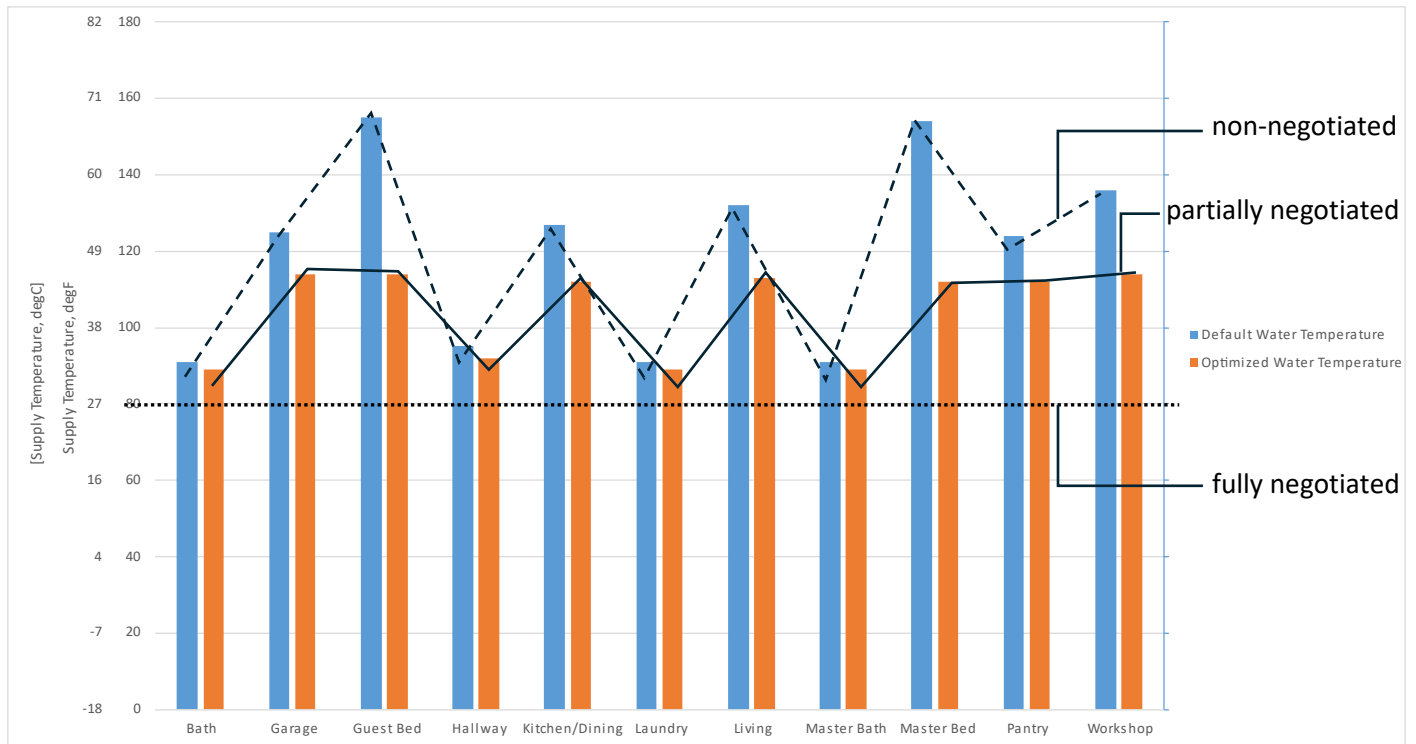


Figure 6 Comparison of defaults (blue data, dashed line, non-negotiated) versus a level 1 optimization (red data, solid line, partially negotiated) for harmonizing heat exchanger temperature profiles. Greater harmonization (closer ranges and flatter data) is possible with deeper level 2 or 3 analysis (dotted line, fully negotiated, one temperature profile of the lowest value).

DISCUSSION

Over 85% of existing building stock in North America is estimated to be under 25,000 ft² (2323 m²) (Mathis, 2016). Building codes do not require licensed engineering professionals in several categories within this sector (e.g. single-family housing, agriculture including barns and shops or conditioned storage facilities); as such, many of the energy and IEQ systems in this inventory have been designed by individuals without formal education in analyzing and optimizing energy systems and indoor environmental quality. The authors estimate that over the past decade, over 4,000,000 buildings in Canada and the United States have HVAC systems designed by undertrained or unqualified practitioners. As property development embraces decarbonization and transitions into electrification, it does so without considering the consequences of a critical mass of practitioners who do not have the skills to push back on architectural, enclosure and interior designs. Though they may have the skills to design basic HVAC systems, the lack of skills to negotiate terms within the first three criteria (architecture, enclosure, interior designs) should cause concern among energy policymakers addressing only the fourth term (HVAC design).

CONCLUSION

If policymakers are committed to electrification and reducing loads on grids, then policy must also lubricate the dialogue from the mechanical practitioners toward the "everything is open for negotiation" design approach. Mechanical designers must understand the optimizations available through architectural, enclosure and interior design, heat exchanger and system design. This skill requires an expanded education curriculum for professional and non-professional practitioners to develop confidence in addressing the architectural rogue zones through to challenging design software defaults.

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