

WHEN THE HEAT RISES



Why Commercial HVAC Systems Struggle During Extreme Heat : Understanding Outdoor Design Temperature and Why Commercial HVAC Systems Struggle During Extreme Heat

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EXECUTIVE SUMMARY

Every commercial HVAC system in Northeast Ohio was sized against a specific climatic number. In Cleveland, that number is **89.7°F**.

It is called the outdoor design temperature, and it comes from ASHRAE's climatic design data. It is not the hottest temperature Cleveland has ever seen, and it is not a forecast. It is a **probability**: a dry-bulb temperature that, averaged over the climate record, is **exceeded roughly 35 hours in a year** of 8,760.

Which means your building's cooling system was engineered to hold the line right up to a temperature that gets beaten about thirty-five hours a year — and to run efficiently for the other eight thousand seven hundred. That is not a corner cut. **That is what correct engineering looks like.**

When outdoor temperatures climb past the design point, two things happen at once, and they work against each other.

Building cooling load rises. Heat pushes through the roof, walls, and glass in proportion to the temperature difference, and code-mandated outdoor air must be cooled from whatever the outdoor temperature happens to be. Every degree adds load.

Equipment capacity falls. A cooling system does not create cold — it moves heat from inside the building to the outdoor air. As that air gets hotter, rejecting heat into it gets harder. Published manufacturer data shows commercial rooftop capacity declining by roughly half a percent to eight-tenths of a percent for every degree above the industry's 95°F rating point — while the compressor draws steadily *more* power.

The two curves converge at the design point and cross just past it. In a conceptual example — a 50,000 square foot Cleveland office, built from published ASHRAE equations and manufacturer performance data, with every input and assumption disclosed in the Appendix — a **97°F afternoon leaves a correctly sized system a few percent short** of what the building is asking for. Nothing has failed. The arithmetic simply ran out.

Field measurement is consistent with this. A Pacific Gas & Electric study monitored commercial rooftop units at one-minute intervals and observed that about 20% of them ran **continuously** at peak conditions — which, as its authors put it, is **"as one would expect for a properly sized unit."**

Continuous operation in extreme heat is the signature of a healthy, right-sized system — not a fault. It is what a system with no surplus left does.

But this cuts both ways, and you should know it before you read further

The same field study found that **more than 40% of commercial rooftop units are oversized by more than 25%**. Run our model on one of those, and it **holds setpoint comfortably through 100°F** — with about 11% capacity still to spare at 97°F.

So if your building is oversized — and the odds say it is — **and it still cannot cool on a 97°F afternoon, the weather is not your explanation. Something is wrong with the equipment.**

This paper explains the physics. **Section 6 gives you a framework to tell when the physics is not the answer.** It reduces to one question: *is the building falling behind only when it is brutally hot outside, or is it also falling behind on an ordinary summer afternoon?*

The first is physics. The second is a service call.

Two numbers determine what your building can do in July: its design temperature and its installed capacity. Most owners cannot state either. Both are knowable.

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SECTION ONE

What Is an Outdoor Design Temperature?

Ask a facility manager how their cooling system was sized and the honest answer is usually: *someone decided how many tons the building needed*. Ask **how** that decision was made, and the trail leads back to a single climatic number.

Engineers design against statistics, not weather

ASHRAE publishes climatic design data for more than 9,000 weather stations in Chapter 14 of the *Handbook—Fundamentals*. For each station it gives cooling design dry-bulb temperatures at three probability levels: **0.4%, 1%, and 2%**. These are annual cumulative frequencies of occurrence. ASHRAE is explicit:

"The 0.4, 1.0, 2.0, and 5.0% values are exceeded on average 35, 88, 175, and 438 h per year, respectively, for the period of record."

Two points of precision matter, and most trade literature gets them wrong. These are **exceedance** values — the temperature is *beaten* about 35 hours a year at the 0.4% level, not "reached." And ASHRAE says **on average, for the period of record**. Any individual summer can exceed it considerably more. The design condition is a long-run statistic, not a promise about this year.

STATION	0.4% DESIGN DB	1%	2%
Cleveland Hopkins Intl.	89.7 °F	87.2 °F	84.6 °F
Akron-Canton Airport	88.8 °F	86.3 °F	84.0 °F

Table 1. ASHRAE Handbook—Fundamentals (2021), Ch. 14. Mean coincident wet-bulb at the 0.4% condition: Cleveland 73.6°F, Akron 72.8°F.

ASHRAE publishes all three percentiles and **deliberately declines to say which to use**: "The designer, engineer, or other user must decide which set(s) of conditions and probability of occurrence apply to the design situation under consideration."

Commercial practice commonly sizes to the 0.4% condition. So the working number for a Cleveland commercial building is roughly **90°F**. On a 95°F afternoon your building is operating five degrees past the condition it was engineered for. On a 97°F afternoon, seven.

Your cooling system was not designed to defeat the hottest day of the decade. It was designed to hold the line on a temperature that gets exceeded about thirty-five hours a year — and to run efficiently for the other eight thousand seven hundred.

What “correctly sized” actually means

Equipment is selected to **meet the design load** — with a modest margin, not a generous one. In practice that means installed capacity somewhere near the load itself: enough to hold the line at the design condition, and not so much that the system runs badly for the rest of the year.

We are deliberately not going to hand you a standard-mandated percentage here. The relevant commercial standards are not freely available, we have not read them, and **we are not going to cite a standard we have not read.** What we *can* do is show you exactly how much the sizing decision matters — which we do, at length, in Section 3 and the Appendix.

The essential point survives without any appeal to authority: **the margin at the design condition is small on purpose.** A system with generous headroom at 90°F is a system that short-cycles, fails to dehumidify, and wears itself out for the other 8,725 hours of the year.

Why not simply install a bigger unit?

This is the first question every building owner asks, and it deserves a real answer.

WHY NOT JUST BUY A BIGGER UNIT?

It short-cycles. ENERGY STAR: air conditioners reach peak efficiency only after roughly ten minutes of running. “When operating time increases from 5 to 9 minutes, efficiency improves 17 percent.” An oversized unit satisfies the thermostat before it ever gets efficient.

It cannot dehumidify. “In order for air conditioners to dehumidify... they have to cycle long enough for moisture to condense on the coils and drain away. With oversized units, short-cycling reduces the amount of condensation that drains off the coils and **even allows some moisture to evaporate back into the air.**” The result is a building that is cold and clammy.

It wears out faster. Every start is mechanical and electrical stress. The Department of Energy links frequent cycling directly to control failure.

It costs more — twice. Higher first cost, and higher demand charges, which are set by your peak. NREL found right-sizing delivered an average **14% reduction in upfront capital cost** across two case-study buildings.

And it hides problems. “Other maintenance problems, such as dirty filters, leaky ducts, and improper refrigerant charge, **are masked by the large output of oversized units.**” Oversizing does not just cost money. It blinds you.

This is not hypothetical. When PG&E instrumented commercial rooftop units in the field, it found that although **“25% over sizing is generally considered by the HVAC design and construction professions to be a safe and acceptable practice,”** in fact **“over 40% of the units are greater than 25% over sized”** and about 10% were

oversized by more than half. In at least 40% of cases, the unit could have been **half the size**.

The industry already tried buying bigger. It is a large part of why buildings are uncomfortable — and, as Section 3 will show, it is also why so many buildings that *should* be coping in a heat wave are not.

And your building is probably not carrying the load you think

There is a widespread belief that modern offices have overwhelmed their 1990s cooling systems with computers and equipment. **The measured evidence says the opposite.**

NREL metered plug loads across commercial buildings and found that designers in the late 1980s and 1990s sized systems for plug load densities of **3 to 5 watts per square foot**, based on nameplate ratings. Actual measured office plug loads today average about **0.28 W/ft²**, with peaks around 0.50 — **lower by a factor of five to ten** than what design and lease documents assume. Laptops replaced desktops. LCDs replaced CRTs. Efficiency standards did their work.

Your 1995 building was very likely sized for an internal load it never actually carried. **That is another reason legacy commercial buildings tend to be oversized rather than starved** — and another reason adding tonnage is rarely the answer to a comfort complaint.

SECTION TWO

How Commercial Cooling Load Is Built

“Cooling load” is the total rate at which heat must be removed from a building to hold a setpoint. It arrives through several independent pathways — and **only some of them care what the outdoor temperature is.**

COMPONENT	WHAT DRIVES IT	MOVES WITH OUTDOOR TEMP?
Roof conduction	How hot it is outside, how much roof, how well insulated	Yes — linearly
Wall & glazing conduction	Same, for walls and glass	Yes — linearly
Outdoor air (sensible)	How much fresh air code requires, and how hot it is	Yes — linearly
Infiltration	Air leaking in, and how hot it is	Yes
Solar gain through glass	Sunlight coming through windows	No — independent
Outdoor air (latent)	Moisture that must be removed from fresh air	Tracks humidity, not temperature
People, lighting, plug load	Occupancy and installed watts	No — independent
Supply fan heat	Energy the fan itself adds to the airstream	No — independent

Table 2. The governing equations for each are given in the Appendix.

The roof is the villain, and it is not close

Most people assume the roof matters because hot air sits on it. The truth is more aggressive.

A sunlit surface does not exchange heat with the air at the *air's* temperature. It exchanges heat at what ASHRAE calls the **sol-air temperature** — the temperature the outdoor air *would have to be*, with no sun at all, to drive the same heat into the surface.

ASHRAE publishes sol-air values for July at 40°N latitude. **Cleveland sits at 41.4°N — this table is not an analogy, it applies directly.** At solar noon, with the outdoor air at 90°F and the building held at 75°F:

SURFACE	SOL-AIR TEMP	DRIVING ΔT	MULTIPLE OF AIR ΔT
Dark roof (horizontal)	162 °F	87 °F	5.8×
Light-colored roof	122 °F	47 °F	3.1×
Dark west wall	114 °F	39 °F	2.6×
Dark north wall	101 °F	26 °F	1.7×
<i>Plain outdoor air</i>	90 °F	15 °F	1.0×

Table 3. ASHRAE sol-air temperature table, July 21 at 40°N. These values assume a dark surface in light wind. A reflective roof, or a windy day, reduces the effect substantially — which is precisely why roof color is a lever, and why it appears in Section 7.

A dark commercial roof under a July sun behaves as though the outdoor air were 162°F. The driving temperature difference across it is nearly **six times** what the thermometer would suggest.

And in a single-story building — the warehouse, the strip center, the suburban office, the school wing — **the roof area equals the floor area.** It is the largest surface the building has, it is aimed at the sun, and it is running at that multiplier.

On a 90°F afternoon, the thermometer reads 90. Your dark roof is behaving as though it were 162.

One honest caveat: that 5.8× figure is a ratio, and ratios shrink as their denominator grows. At 97°F outdoor the same dark roof runs at about 4.3× the air-to-room difference — still enormous, but the multiplier itself does not grow with the heat. The absolute roof load does.

Outdoor air: the load you are not allowed to turn off

Every commercial building is required by code to bring in outdoor air. ASHRAE Standard 62.1 sets the minimum — for a typical office, **5 cfm per person plus 0.06 cfm per square foot** of floor area.

You cannot opt out. It is a code obligation, and it means that on the hottest afternoon of the year your building is **required** to draw in thousands of cubic feet per minute of the hottest air available and cool it down.

There is a timing point worth knowing, too. Outdoor-air load hits the cooling coil the moment the thermometer moves. Roof heat arrives late — the roof soaks up sun at noon and hands it to the building around four o'clock. When the temperature spikes, you feel the outdoor-air load immediately, and the roof load from that same spike is still on its way.

The insight that carries the rest of the paper

In the conceptual 50,000 ft² Cleveland office specified in the Appendix, the temperature-driven components add up to a slope of about **11,500 Btu/h for every degree** of outdoor temperature.

But the temperature-*independent* components — solar gain, lighting, people, plug load, fan heat, and dehumidification — total roughly **975,000 Btu/h** and **do not move at all** when the outdoor temperature changes.

At the 89.7°F design condition, that building's total load is about **1,145,000 Btu/h (roughly 95 tons)** — and **about 85% of it is completely indifferent to the outdoor temperature**.

*Cooling load rises steeply in absolute terms — but modestly in percentage terms. Because such a large share of the load is fixed, the total load curve has a **high intercept**, not a shallow slope. Total load climbs only about **1% per degree**. On its own, that is manageable. It is not on its own — and Section 3 is about the other half.*

The Scissors: Where Load and Capacity Cross

It is not a switch. It is a margin.

The common mental model of air conditioning is binary: *the AC works, or the AC is broken*. Nearly every frustrated call to a service contractor during a heat wave begins from that assumption.

The engineering reality is a **margin that narrows** — continuously, predictably, and by an amount you can calculate.

Correct sizing does not mean comfortable headroom. It means enough capacity to hold the line at the design condition — and not so much that the system runs badly for the rest of the year.

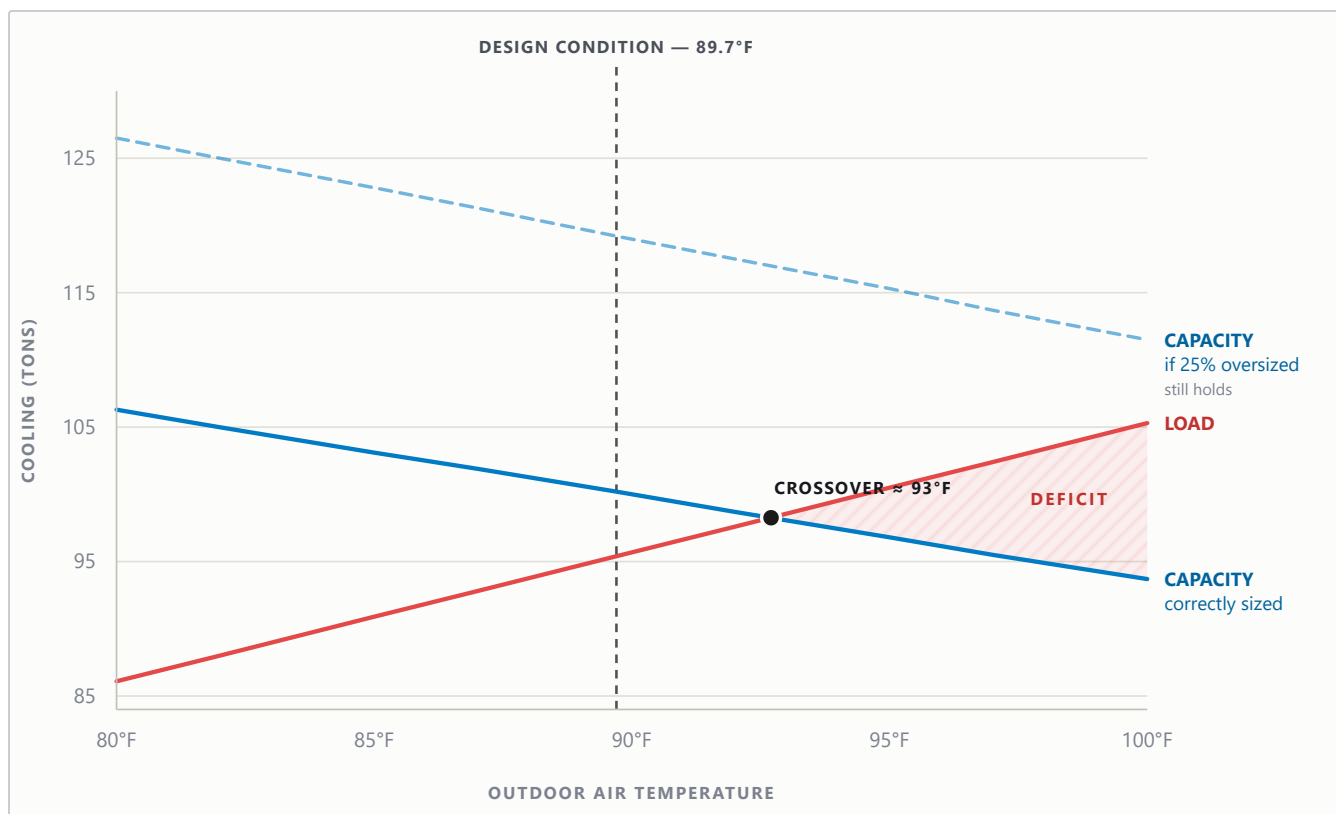
The scissors

Two things happen simultaneously above the design temperature, and the reason buildings fall behind is that they **compound**.

The load rises — about **1.0% of the design load per degree**, in our conceptual example.

The capacity falls — about **0.7% of the design load per degree**, applying the derate slope that published manufacturer tables actually show. (Section 4 explains the physics.)

Together, the deficit opens at roughly 1.7% of the design load for every degree — about **60% of it from load rising, 40% from capacity falling**. Both blades of the scissors are real. Neither alone explains what building owners experience.



— Building cooling load — Capacity — correctly sized — Capacity — 25% oversized

Chart 1. Load rises with every degree; capacity falls with every degree. For a correctly sized system they cross just past the design condition, and the shaded gap beyond is cooling the building demands but the equipment cannot deliver. **Note the third line.** A system that is 25% oversized — which field studies find describes more than 40% of real commercial rooftop units — keeps surplus capacity across this entire range. If *that* building cannot cool at 97°F, the explanation is not the weather.

*Conceptual example. Built from ASHRAE load equations and published manufacturer performance data — **plus a number of stated assumptions**, all disclosed and flagged as such in Appendix A. **The crossover shown assumes capacity is anchored at the design condition; anchoring instead at the AHRI 95°F rating point moves it to ≈95°F. Both readings are given in Appendix A, note 5.** The crossover is in any case far more sensitive to how the equipment was sized than to any of the physics plotted here. Illustrative of a relationship. Not a substitute for a site-specific load calculation.*

OUTDOOR	LOAD	CAPACITY	MARGIN	RESULT
80 °F	86	106	+23%	Comfortable. System cycles.
85 °F	91	103	+13%	Comfortable. System cycles.
89.7 °F	95	100	+5%	DESIGN CONDITION. Holds setpoint, running nearly flat out.
~93 °F	—	—	0%	CROSSOVER. The margin is gone.
95 °F	100	97	–4%	Falling behind.
97 °F	102	96	–7%	Short. Setpoint drifts.
100 °F	105	94	–11%	Short by a ninth.

Table 4. Conceptual example — 50,000 ft² single-story 1990s-era office, Cleveland. Load and capacity in tons. Every input and assumption disclosed in the Appendix. Illustrative of a relationship — not a universal result.

*One honest caveat on the 97°F row: the size of the deficit depends on whether “capacity equal to 105% of the design load” is measured at the design condition or at the AHRI 95°F nameplate rating. Both are ordinary readings, and they give **3% to 7% short at 97°F**. We publish the range because we cannot honestly tell you which one describes your building. (Appendix, note 5.)*

Nothing broke between the 89.7°F row and the 97°F row. No filter clogged. No compressor failed. **Seven degrees happened.**

The crossover is a range, not a number — and this matters more than the number does

It would be easy to read “about 93°F” off that table and treat it as a fact about buildings. **It is not.** It is a fact about *this* building, and the largest single thing it depends on is **how the equipment was sized** — a choice a designer made years ago, not a property of the weather.

HOW THE EQUIPMENT WAS SIZED	CROSSOVER
Sized at 100% of design load — no margin at all	≈ 90 °F
Our example — 105%	≈ 93 °F
Sized at 115% of design load	≈ 98 °F
25% oversized — describes >40% of real buildings	≈ 104 °F

Table 5. About fourteen degrees of spread, from the sizing decision alone. That is wider than the entire effect this paper describes.

Which means the honest answer to “at what temperature will my building fall behind?” is: **it depends on your building, and specifically on a decision someone made when it was built.** Anyone who hands you a single universal number — **including anyone quoting this paper** — is overselling.

And now the part that matters most: the oversized building

Section 1 established that **more than 40% of commercial rooftop units are oversized by more than 25%**. That is not the exception. In the field, it is the most common building there is.

So run the model on one.

OUTDOOR	CORRECTLY SIZED	25% OVERSIZED
89.7 °F (design)	+5% — holds	+25% — holds easily
95 °F	–4% — drifts	+15% — holds
97 °F	–7% — drifts	+11% — holds
100 °F	–11% — drifts	+6% — still holds
Crossover	≈ 93 °F	≈ 104 °F

Table 6. The most common building in the field is not the one this paper's physics describes.

An oversized system does not fall behind at 97°F. It has capacity to spare. So if an oversized building cannot cool on a 97°F afternoon — and most commercial buildings are oversized — the weather is not the explanation. Something is broken.

This is the most important paragraph in the paper, and it is the one a contractor is least likely to tell you.

Everything else here explains why a *correctly sized* system runs out of margin in extreme heat. That explanation is true, well-supported, and should change how you interpret a hot afternoon. But it applies to a building that was sized correctly — and the field data says **most were not**.

If your building was generously sized (and the odds favor it), the physics in this paper **buys you no excuse whatsoever**. Section 6 is how you find out what is actually wrong.

Why a small deficit feels like a catastrophe

When capacity falls short of load, a building does not simply “stay a bit warm.” The indoor temperature rises **until the load drops enough to match the capacity available**. The building finds a new equilibrium — at a higher temperature.

But recall from Section 2: **only about 15% of this building’s cooling load is temperature-sensitive at all**. The solar gain, the lights, the people, the computers, the fan heat, the dehumidification — none of it gets smaller because the room got warmer.

So to shed a deficit, the *temperature-sensitive* part of the load must fall by enough to cover the whole gap. And since that part is small, the indoor temperature has to move **a lot**.

OUTDOOR	SYSTEM SHORT BY	INDOOR DRIFTS TOWARD
95 °F	4%	~79 °F
97 °F	7%	~82 °F
100 °F	11%	~87 °F

Table 7. These are **upper bounds**, not predictions. As the space warms, the air returning to the coil gets warmer and more humid, and the equipment **recovers some capacity** — a self-limiting effect this model does not include. Thermal mass delays the drift further. Real buildings do better than this table. The **mechanism** is the point, not the values.

A 7% capacity shortfall does not make a building 7% less comfortable. The physics is not gentle.

That is why heat waves feel less like a gradual decline and more like a building giving up. Because most of a commercial cooling load does not care how warm the room gets, a modest capacity shortfall forces a large temperature drift.

SECTION FOUR

The Second Curve: Why Capacity Falls as It Gets Hotter

Chart 1 has two lines. Section 3 was about the one going up — the one everyone expects. This section is about the one coming down, which almost nobody does.

A cooling system does not make cold

An air conditioner is not a cold-generator. It is a heat pump — a machine that picks heat up inside your building and puts it down outside.

Which raises an obvious question: what happens when *outside* is hot?

Rejecting heat into 85°F air is straightforward. Rejecting that same heat into 105°F air is much harder — you are trying to push heat into a place that is already hot. The machine still works. It just cannot move as much.

That is the entire concept. What follows is the mechanism, for readers who want it.

THE MECHANISM, AND A CORRECTION WORTH MAKING

Optional. The paragraph above is the whole story for practical purposes.

The condenser must reject the heat absorbed indoors *plus* the work the compressor added, and it can only do that by running hotter than the air around it. So as outdoor air temperature rises, the **condensing temperature is forced up with it.**

Now follow the refrigerant. It leaves the condenser as a liquid at that elevated temperature. Passing through the expansion valve, some of it **flashes immediately to vapor** — and refrigerant that has already boiled contributes little further cooling in the evaporator. The hotter the liquid arriving at the valve, the larger that flash fraction, and **the less cooling each pound of refrigerant can do.**

That is the dominant loss. Each pound of refrigerant simply does less work when it starts out hotter.

The correction. The usual explanation in the trade says that rising head pressure reduces refrigerant **mass flow**, and that this is why capacity falls. **Laboratory measurement says otherwise.** Southern California Edison tested six commercial rooftop units at ambient temperatures up to 130°F and reported that as capacity fell, suction pressure *drifted upward*, the refrigerant became denser, and **only the units from one of the manufacturers tested failed to show an increase in mass flow rate.**

The capacity loss is real. The popular mechanism for it is not. **The refrigerant is not moving more slowly — each pound of it is doing less.**

We ran the thermodynamics independently for R-410A. From 95°F to 115°F ambient, the cooling delivered per pound of refrigerant falls by **13–15%**, while compression work per pound rises **22–25%**. Published manufacturer data shows total capacity falling **9–16%** across the same range — the three models in our reference list lose 16.2%, 12.4%, and 9.2% respectively.

Those bands overlap. The collapse of cooling-per-pound is, on its own, large enough to account for the great majority of the observed derate — theory and measurement land in the same territory, which is the best evidence we have that the mechanism above is right.

*We stop short of claiming it accounts for all of it. If mass flow genuinely rises, capacity should fall by slightly **less** than the per-pound figure. And the measured spread across just three units (9% to 16%) is wider than the theoretical band (13% to 15%), so the two cannot be cleanly resolved against each other. We would rather leave that open than narrow a range in our own favor.*

What the manufacturers actually publish

These are not estimates. These are performance tables for the equipment on Northeast Ohio roofs.

OUTDOOR AIR	GROSS CAPACITY	COMPRESSOR POWER	CAPACITY ÷ KW
85 °F	207,700 Btu/h	10.44 kW	19.9
95 °F — AHRI rating point	193,100 Btu/h	11.78 kW	16.4
105 °F	177,700 Btu/h	13.34 kW	13.3
115 °F	161,900 Btu/h	15.16 kW	10.7

Table 8. Lennox Emergence 15-ton rooftop unit, Bulletin 210734, at 6,000 cfm and 67°F entering wet bulb. Between 95°F and 115°F this unit loses **16.2% of its cooling capacity** while drawing **28.7% more power**. Its effective efficiency falls by roughly **35%**.

Across three published models from two manufacturers, commercial rooftop capacity falls roughly **0.5% to 0.8% per degree** above the 95°F rating point — and the derate *steepens* as it gets hotter. Our model uses the mean of the three, about 0.63%/°F, and the Appendix shows what happens at either end of the range.

The nameplate is a rating, not a promise

The cooling capacity stamped on your rooftop unit is certified under **AHRI Standard 340/360**. That standard rates capacity at a specific, defined condition: **95°F outdoor air**, with 80°F dry-bulb / 67°F wet-bulb entering the evaporator.

Above 95°F, the standard specifies something different. It defines a “**Maximum Operating Condition**” of **115°F** — a requirement that the unit *keep running*. Not a statement of how much it will deliver.

YOUR 20-TON UNIT IS NOT ALWAYS A 20-TON UNIT

On a hotter day, you have less than you bought.

AHRI rates cooling capacity at 95°F. **It publishes no rated capacity above 95°F**. The standard your equipment is certified to requires it to *survive and operate* up to 115°F — it makes no claim about how much cooling it produces there, because performance above the rating point is, by the structure of the standard, outside the rating basis.

The number on the nameplate is not a guarantee about a 97°F afternoon. It is a measurement taken at 95°F. This is documented, published, and entirely expected by the people who built the machine.

A related trap: “nominal tonnage” is a model-line label, not a measurement. Lennox’s own table above shows a “15-ton” unit producing 16.1 tons at the rating point. Do not do arithmetic with the number in the model name.

The part your CFO should read

At 95°F the unit produces 193,100 Btu/h for 11.78 kW. At 115°F it produces 161,900 Btu/h for 15.16 kW.

Less cooling. More power. Every hour.

Efficiency falls by about a third precisely when the building needs cooling most — and precisely when the utility is setting your **demand charge**, which is billed on your peak. The hottest afternoon of the year is simultaneously your worst performance and your most expensive hour.

SECTION FIVE

Why Systems Run Continuously During Heat Waves

Cycling is what surplus looks like

A cooling system cycles off when it has **caught up** — when it has removed heat faster than the building gained it, driven the space to setpoint, and has capacity left over. Cycling is the visible evidence of **surplus capacity**.

At the design condition, a correctly sized system has very little surplus. So it barely cycles. It runs. And past the design condition, when load exceeds capacity, it *cannot* catch up — so it cannot shut off. **A system running continuously in a heat wave is not failing to shut off. It has nothing left to shut off with.**

The field data

Pacific Gas & Electric instrumented commercial rooftop units with one-minute interval monitoring across more than 18,000 hours of operation, during a period that in many cases **exceeded ASHRAE 1% design conditions**:

*"About 20% of the monitored units ran continuously under peak load conditions, **as one would expect for a properly sized unit.**"*

And the mirror image:

*"**Frequent cycling, especially under peak load conditions, is a reliable indicator of roof top unit over sizing.**"*

Nearly 60% of the monitored units were cycling three or more times per hour at peak conditions — typically six minutes on, sixteen minutes off.

Scope, honestly stated: this is 1998 data from Northern California, drawn from 145 usable datasets of 250 monitored units, most of them around 5 tons. The mechanism generalizes. The specific percentages are theirs, not ours.

MYTH VS. REALITY

MYTH

If my AC never shuts off, something is broken.

REALITY

A properly operating system may run continuously during extreme heat because cooling demand has met or exceeded available capacity. Field monitoring found that continuous operation at peak conditions is what a **correctly sized** unit does.

MYTH

It's not holding 72°F, so it's failing.

REALITY

If your system was sized to hold setpoint at an 89.7°F design condition, then holding within a few degrees at 97°F **is** good performance. The relevant question is not whether the building hit setpoint on the worst day of the year. It is whether it holds setpoint on an **ordinary** summer day.

MYTH

A bigger unit would have prevented this.

REALITY

A unit large enough to hold setpoint on the hottest hour of the decade would be oversized for the other 8,725 hours — short-cycling, dehumidifying poorly, masking maintenance faults, and costing more to buy and run. More than 40% of commercial rooftop units **already** exceed the industry's own 25% oversizing allowance, and it has not made buildings comfortable.

MYTH

It's hot out, so of course my building can't keep up.

REALITY

Not if your system is oversized — and most are. A 25%-oversized system holds setpoint comfortably at 97°F with capacity to spare. If your building is generously sized and still cannot cool, the heat is not your problem. **Go to Section 6.**

SECTION SIX

When Continuous Operation Does Indicate a Real Problem

Everything to this point has explained why a healthy system struggles in extreme heat. **This section exists so that explanation is never used as an excuse — including by us.**

THE QUESTION THAT SEPARATES PHYSICS FROM FAILURE

Is the building struggling **only** when it is brutally hot outside — or does it also struggle on an ordinary summer afternoon?

Only during genuine extreme heat, with recovery when the weather breaks? That is the design margin doing what it was engineered to do. This paper explains it.

Also struggling on a routine 82°F July afternoon? The design temperature has nothing to do with it. An 82°F day is nearly eight degrees *below* the Cleveland design condition. Whatever margin your building has, it has it in abundance there. That is a mechanical problem, and it will not resolve when the heat wave ends.

The second question, and it is nearly as important

Was your system generously sized?

If you have more installed tonnage than a current load calculation would call for — and the field data says **more than 40% of buildings do** — then the physics in this paper **does not apply to you at 97°F**. An oversized system has surplus at 97°F. It should be coping. If it is not, something is wrong.

Signals consistent with normal full-load operation

- The unit runs continuously **only** during genuine extreme heat
- Supply air is still measurably cold, for the current indoor humidity
- The building holds within a few degrees of setpoint and **recovers overnight**
- Performance returns to normal when outdoor temperatures do
- Other buildings nearby are experiencing the same thing at the same time

Signals of genuine mechanical failure

Warm or lukewarm supply air. The clearest signal. If the temperature drop across the cooling coil has collapsed, the system is not removing heat.

But have it measured correctly. The “18–22°F temperature split” that every technician quotes is **not a universal**. The correct target depends on indoor humidity, and it **collapses as humidity rises** — at 75°F return air with a muggy 71°F wet bulb, the correct target split is about **11.7°F, not 20°F**.

If a technician tells you the split “looks fine” — or “looks bad” — **ask what the return wet bulb was**. A tech who measured only dry bulb has told you very little. And on a rooftop unit with an economizer, what they measured may have been mixed air rather than return air, which shifts the target again. This is a screening tool, not a substitute for direct airflow measurement.

A frozen evaporator coil. Ice on a cooling coil in July is **always** a fault — never “the system working too hard.” It points to low airflow or low refrigerant charge. Note the trap: those underlying faults are *most likely to surface precisely when the unit runs long and hard*. A coil that freezes during a heat wave is a genuine failure that the heat wave **exposed**, not caused.

Short cycling during a heat wave. Read Section 5 again. At peak load, a healthy right-sized unit should be *running*. If yours is switching off and on, it is either badly oversized or tripping on a fault.

Low refrigerant charge. A collapsed temperature split across the evaporator “indicates that the compressor is not extracting load from the air stream, which indicates a high possibility of a low refrigerant charge.” And an important framing point: **refrigerant is not consumed**. A system low on charge has almost always sprung a leak — and a leak is a failure, not a maintenance item.

A dirty condenser coil. This interacts directly with everything in Section 4. A fouled condenser cannot reject heat efficiently, which drives condensing temperature and pressure up — *the same mechanism* that high outdoor temperature imposes. A dirty coil is therefore **a self-inflicted heat wave**, stacked on top of the real one. In high ambient, the combination is what trips the high-pressure switch.

This is often the honest answer to “*why did my unit die on the hottest day of the year?*” The coil was already dirty. The 97°F afternoon simply revealed it.

A stuck economizer. This one hides for years and almost nobody checks it. Economizers fail at a remarkable rate — across eleven field studies, failure rates average **about 60%**. Most commonly they fail **closed**, quietly costing you free cooling every spring and fall.

But when one fails **open**, it becomes actively destructive in hot weather. In a simulated building in Bakersfield, California, an economizer stuck fully open **added 84% to the summer peak load**. The magnitude would be smaller in Cleveland’s milder climate — but the mechanism is identical, and the unit will appear hopelessly undersized on hot days, because it is being force-fed outdoor air while trying to cool the building.

A CONTRACTOR WHO RECOMMENDS MORE TONNAGE WITHOUT A LOAD CALCULATION

If your building struggles on a 97°F day and someone proposes a bigger unit, ask two questions:

1. What is my building's design temperature?
2. May I see the load calculation?

If they cannot produce one, the recommendation is a guess — and what you would be buying is every penalty in the callout in Section 1: short cycling, poor dehumidification, masked faults, higher demand charges, shorter equipment life.

Adding tonnage is the most expensive way to be wrong about this, and it is very often the wrong answer. **Especially given that your building is more likely than not to be oversized already.**

SECTION SEVEN

What Building Owners Should Do

Know your two numbers

Most building owners cannot state their building's **design temperature** or its **installed cooling capacity**. These determine what your building can and cannot do in July, and both are knowable.

HOW TO FIND YOUR TWO NUMBERS

Installed capacity is on the equipment nameplate and in the original mechanical drawings. Walk the roof and add up the tonnage. *(Remember Section 4: nominal tonnage is a label, not a measurement — but it is close enough to tell you what order of magnitude you are working with.)*

Design temperature is the ASHRAE 0.4% cooling dry-bulb for your nearest weather station. Cleveland is 89.7°F; Akron-Canton is 88.8°F.

The third number — the load your building actually carries today — requires a load calculation. If nobody has performed one since the building was constructed, that number is genuinely unknown. And it is the one that tells you whether your equipment is right-sized, oversized, or has quietly been outgrown.

Set expectations before the heat wave, not during it

The most damaging thing about a heat wave in a commercial building is often not the temperature — it is the **surprise**. Tenants and staff who have been told in advance that the building may run a few degrees warm during genuine extreme heat respond very differently from those who conclude, at 3 p.m. on a Thursday, that the building is broken.

A one-paragraph email in May is worth more than a service call in July.

Protect the capacity you already own

Everything in Section 4 says capacity is scarce on hot days. Every item below is capacity you have **already paid for** and are at risk of throwing away:

- **Condenser coils.** A dirty condenser imposes the same physics as a heat wave. Cleaning it is the most direct way to protect hot-weather capacity.
- **Filters and airflow.** DOE: 5–15% of energy consumption, and low airflow is a leading cause of frozen coils.
- **Refrigerant charge.** Low charge almost always means a leak. Find it.
- **Economizers.** Field failure rates average around 60%. If nobody has checked yours in the last few years, the statistics say it is more likely broken than not.

Understand where load creep is real

Section 1 showed that the “modern offices overwhelmed old systems” story is a myth — measured plug loads today are far *below* what 1990s designers assumed.

Where load creep is real is at the zone level. The storage closet that became a server room. The open floor that got densified. The tenant who installed a commercial kitchen. NREL’s data shows offices *with data centers* running three to four times the plug load of a plain office.

That is a zoning and air-distribution problem, not a tonnage problem. The instinct to solve it by adding tonnage will make the humidity and cycling penalties in Section 1 *worse*, not better. The right fix is usually to get the cooling to the right *place*, not to buy more of it.

Reduce the load before you buy more capacity

The cheapest ton of cooling is the one you never have to produce. The highest-leverage moves are usually on the **load side**:

- **Roof reflectivity.** Section 2's table is the argument: a light-colored roof runs at roughly a 3.1× effective temperature multiple instead of a dark roof's 5.8×. On a single-story building this is the largest surface you own. *(In our conceptual example, switching from dark to light cuts the design load by about 9 tons — and it is why we show that sensitivity in the Appendix rather than quietly leaving it out.)*
 - **Glazing film or shading.** Solar gain through glass is a fixed load that does not care about the outdoor temperature — which means you carry it at *every* condition, not just the hot ones.
 - **Controls and scheduling.** Pre-cooling the building's thermal mass ahead of a forecast heat event shifts load out of the worst hours.
 - **A planned setpoint strategy.** A communicated 2°F relaxation during a genuine extreme-heat event is vastly better than an unplanned 8°F drift.
-

CONCLUSION

What This All Comes Down To

Outdoor design temperature is not an excuse. It is an **engineering boundary condition** — and a knowable one.

Commercial HVAC systems in Northeast Ohio are sized against a statistical condition: roughly **90°F** in Cleveland, a temperature exceeded on average about **35 hours a year**. Below that line, a correctly sized system has margin. At that line it has little — because having little there is what correct sizing means.

Above that line, two curves move against each other. Load rises with every degree. Capacity **falls** with every degree, because a cooling system must reject heat into outdoor air that is itself getting hotter. In our conceptual example the deficit opens at roughly **1.7% of the design load per degree**.

Seven degrees past the design point, a correctly sized building can be short several percent of the cooling it needs — with nothing broken and nothing to repair.

Continuous operation during extreme heat is therefore, very often, the sign of a system **working**. When commercial rooftop units were monitored at peak conditions, about 20% ran continuously — which, in the words of the engineers who ran the study, is “as one would expect for a properly sized unit.”

But honesty cuts in two directions

More than 40% of commercial rooftop units are oversized by more than 25%. Those systems **do not run out of margin at 97°F**. They have capacity to spare. If a generously sized building cannot cool on a hot afternoon, the physics in this paper explains nothing about it — **and a contractor who offers that explanation is not being straight with you.**

Likewise, a building that cannot cool on an ordinary 82°F afternoon is not a victim of the weather. It has a problem, and that problem has a name: a fouled condenser, a leaking circuit, a failed economizer, a collapsed airflow path.

The right question is not “*why isn’t my air conditioning keeping up?*”

It is: is my building’s cooling capacity actually matched to the load it carries today — and do I know what happens to both when the temperature climbs past ninety?

That is an answerable question. And it is worth answering **before** the next heat wave, not during it.

About Air Temp Mechanical

Serving Northeast Ohio since 1978.

Air Temp Mechanical is a commercial-only mechanical contractor. We do not service residential systems, and the focus is deliberate: commercial and industrial HVAC and refrigeration are engineering disciplines, not scaled-up home comfort.

Commercial HVAC · Commercial Refrigeration · Healthcare · Industrial · Preventive Maintenance ·
Emergency Service

If you do not know your building's design temperature or its installed capacity margin, that is a conversation worth having in April — not at 3 p.m. on the hottest Thursday in July.

We are glad to walk a roof, review a load calculation, and tell you plainly what your building can and cannot do when it gets hot — **including when the honest answer is that nothing is wrong with it.**

air-tempmech.com · Northeast Ohio

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ASHRAE (2021). *ASHRAE Handbook—Fundamentals*, Ch. 14, "Climatic Design Information." Station data: Cleveland Hopkins International Airport (WMO 725240); Akron-Canton Airport (WMO 725210).

Air Conditioning Contractors of America. *Outdoor Design Conditions Guide*, quoting ACCA Manual J, 8th ed., v2.0. (*Manual J governs residential work; cited here for its explicit treatment of design conditions and safety factors.*)

A retraction, printed rather than buried. An earlier draft of this paper cited ACCA **Manual S** for a "95–115% of design load" commercial sizing window. That was wrong twice over: Manual S is ACCA's *Residential* Equipment Selection standard, and we had not verified the window against a primary source in any case. Rather than swap in another standard we have not read, we removed the appeal to authority entirely. The sizing margin used in our conceptual example is declared as an assumption in the Appendix, and its consequences are shown as a sensitivity band instead of asserted as a rule. We print this note because a paper that asks you to check its arithmetic should also show you where it got its own arithmetic wrong.

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ASHRAE. *Radiant Time Series (RTS) Method*, Online Supplemental Material to *Principles of HVAC*, 9th ed. © 2021 ASHRAE. This document states that its content mirrors ASHRAE *Handbook—Fundamentals* Ch. 18, and is the source consulted for the sol-air definition, the sol-air table, and the load equations reproduced here.

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Lennox Industries. *Energence Ultra High Efficiency Rooftop Units*, Bulletin No. 210734, p. 25. Model LCH180U4M, 6,000 cfm, 67°F entering wet bulb. **Gross** capacity.

Carrier Corporation. *WeatherMaker 48TC Product Data*, Form 48TC-7-16-03PD Rev. A. Models 48TC*D12 and 48TC*D14, 80°F EDB / 67°F EWB. **Gross** capacity.

AHRI Standard 340/360-2019 (I-P), *Performance Rating of Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment*, Table 5. Standard Rating Condition (cooling): 95.0°F outdoor entering air; 80.0°F DB / 67.0°F WB indoor entering air. Maximum Operating Condition: 115°F.

Proctor Engineering Group. *Temperature Split — Airflow Check Procedure*. Adapted from Carrier, CheckMe!™, and California Title 24. **Note: developed for residential and light-commercial split systems.** On a commercial rooftop unit with an economizer, the air entering the coil may be mixed rather than return air, which shifts the target. The principle — that the correct split depends on entering humidity — holds regardless.

Heinemeier, K. (2014). "Free Cooling: At What Cost?" *ACEEE Summer Study*, 3:121. UC Davis, Western Cooling Efficiency Center. (*The 84% peak-load figure is from a simulated building in Bakersfield, California — a hot, dry climate. The magnitude in Northeast Ohio would be smaller; the mechanism is the same.*)

ANSI/ASHRAE/ACCA Standard 180, *Standard Practice for Inspection and Maintenance of Commercial Building HVAC Systems*.

Additional supporting reference: U.S. DOE / Pacific Northwest National Laboratory, *Building America Solution Center: Design for Extreme Heat* — "Typical air-conditioning systems are designed for standard local conditions and may not have the full capacity needed during an extreme heat event."

APPENDIX A

Conceptual Example: Assumptions, Method, and Limits

Every figure in Section 3 comes from the model below. It is published in full so a competent engineer can reproduce Chart 1 — and challenge it.

This is an illustrative example, not a universal result. It exists to demonstrate a *relationship*. It is not a substitute for a site-specific load calculation.

Which of these numbers are sourced, and which are ours

We think this distinction matters more than any individual value, so we are making it explicit rather than typesetting everything to look equally authoritative:

- **ASHRAE / NREL / MFR** — from a published standard, laboratory study, or manufacturer performance table
- **DERIVED** — computed by us from the above
- **ASSUMPTION** — **chosen by us.** Reasonable, disclosed, and open to challenge. Not sourced.

PARAMETER	VALUE	BASIS
Floor / roof area	50,000 ft ² , single story	ASSUMPTION
Roof U-value	0.048 (≈ R-20, existing construction — not current code)	ASSUMPTION
Glazing U / SHGC	0.50 / 0.35, 30% window-to-wall	ASSUMPTION
Glazing solar irradiance	100 Btu/h·ft ² — produces ~10% of total load	ASSUMPTION
Occupancy	250 people (5 per 1,000 ft ² , office default)	ASHRAE 62.1
Outdoor air	4,250 cfm (5 cfm/person + 0.06 cfm/ft ²)	ASHRAE 62.1
Lighting power density	1.20 W/ft ² — an existing 1990s office (current code is ~0.75)	ASSUMPTION
Plug load density	0.50 W/ft ² — measured office peak, not a nameplate estimate	NREL
Supply fan power	1.0 W/cfm (typical RTU range 0.8–1.25)	ASSUMPTION
Supply airflow	38,150 cfm — iterated to self-consistency at 400 cfm/ton	DERIVED
Humidity ratio difference	34.0 gr/lb — from Cleveland's 0.4% design vs 75°F/50% RH	DERIVED
Equipment selection margin	105% of design total load	ASSUMPTION
Capacity derate	0.63%/°F — the mean of three published tables (0.46 / 0.62 / 0.81)	MFR

Table A1. Conceptual example inputs. Rows tinted red are our own assumptions, not sourced values.

Results at the 89.7°F design condition

- Total cooling load: **≈ 1,145,000 Btu/h (about 95 tons)** — a load density of about **525 ft²/ton**
- Share of load independent of outdoor temperature: **about 85%**
- Load sensitivity: **about 1.0% of design load per °F**
- Capacity derate: **about 0.7% of design load per °F**
- **Combined: the deficit opens at roughly 1.7% of design load per °F** — about 60% load, 40% capacity
- Crossover for this example: **about 93°F**

Sensitivity — why we publish a band, not a decimal point

Felts & Bailey report that actual delivered capacity varies by **±10% of nominal**. Publishing a crossover temperature to a decimal place on top of a ±10% input would be false precision. Here is the real spread — and note which input dominates:

SCENARIO	CROSSOVER
Sized at 100% of design load — no margin	≈ 90 °F
Our example — 105%	≈ 93 °F
Sized at 115% of design load	≈ 98 °F
25% oversized — describes >40% of real buildings	≈ 104 °F
Shallowest published derate (−0.46%/°F)	≈ 93 °F
Steepest published derate (−0.81%/°F)	≈ 92 °F

Table A2. Read this carefully. **The equipment-sizing decision moves the crossover across a fourteen-degree range. The choice of manufacturer derate moves it by one.**

The thing this paper is about — the physics of load rising and capacity falling — is real, but it is not the dominant variable in whether your building copes. How someone sized your equipment, years ago, matters more.

That is not a weakness in the argument. **It is the argument**, and it is why Section 6 exists.

Roof color also moves the numbers. Substituting a light-colored roof (+32°F sol-air adder instead of +72°F) drops the design load from about 95 tons to about **86 tons**. Section 7 recommends reflective roofs; honesty requires showing that doing so changes our own model.

Method notes and limitations — read these

Sol-air raises the intercept, not the slope. The solar term in the sol-air equation is *added on*, not multiplied in, so a one-degree rise outdoors produces a one-degree rise in sol-air temperature. Envelope conduction therefore tracks outdoor dry-bulb essentially one-for-one during a sustained hot spell.

This is a steady-state model, and it is NOT the current ASHRAE method. ASHRAE’s Radiant Time Series method uses a 24-term lagged convolution and would **not** sum solar-noon roof gain, peak glazing solar, and peak (mid-afternoon) dry-bulb simultaneously — as we do here. Our approach therefore **overstates the coincident peak somewhat**. We use it because it is transparent and reproducible by hand. **It should not be described as “the ASHRAE method.”**

Latent load is treated as temperature-independent. It tracks the outdoor humidity ratio, not dry-bulb. This is physically correct, though outdoor humidity and temperature are statistically correlated in practice.

Capacity derate is linearized at $-0.63\%/^{\circ}\text{F}$ — the mean of the three published tables (-0.46 , -0.62 , -0.81). The true curve is slightly concave; the derate *worsens* as it gets hotter, so a linear model is mildly conservative at high temperatures. *(An earlier draft used $-0.77\%/^{\circ}\text{F}$ and described it as “mid-range.” It was not: it sat beside the steepest table we had found, and it biased the result by about 22% in the direction that made our own argument look stronger. It has been corrected to the actual mean.)* We also extrapolate that slope *below* 95°F ; the manufacturer tables support it (Lennox’s 85°F row implies $-0.76\%/^{\circ}\text{F}$ below the rating point), but a straight line across 80 – 100°F is our simplification, not theirs.

The sizing margin is anchored at the design condition, and that is a choice with consequences.

“Capacity equal to 105% of the design load” — measured *at what outdoor temperature?* We anchor at the 89.7°F design condition, which is what a careful selection at design conditions produces. But equipment is often selected off an **AHRI 95°F nameplate** instead, and below 95°F a unit delivers *more* than nameplate — so that reading gives the building more capacity at design. Anchored at design, the margin at 97°F is **-7%** and the crossover $\approx 93^{\circ}\text{F}$. Anchored at AHRI, it is **-3%** and $\approx 95^{\circ}\text{F}$. **We publish both because we cannot honestly tell you which describes your building.** An earlier draft picked the design anchor silently — and it was, once again, the reading that made our own effect look bigger.

Equipment is sized against the design load, not from nominal tonnage. Nominal tonnage is a model-line label: Lennox’s own table shows a “15-ton” unit producing 16.1 tons gross at AHRI conditions.

Capacity is modeled as a function of outdoor temperature only — and that is a real limitation.

Actual DX capacity also depends on the air *entering* the evaporator coil, which is why AHRI specifies indoor entering conditions alongside the 95°F outdoor condition. We do not model entering-air effects. A more rigorous treatment would.

The equilibrium drift table gives upper bounds, not predictions. We hold capacity constant as the space warms. In reality a warmer, more humid return-air stream raises the entering wet bulb and the coil **recovers some capacity**, limiting the drift. Thermal mass delays it further. Real drift is smaller than the table shows.

We claim physical linearity, not empirical linearity. Measured load-versus-temperature curves for real buildings tend to bend *upward*, because solar irradiance and humidity — though physically independent of dry-bulb — are statistically correlated with it.

This model was not fitted to any measured building. Our load density of about $525\text{ ft}^2/\text{ton}$ is lighter than the 300 – $400\text{ ft}^2/\text{ton}$ rules of thumb often quoted; that is expected, because this is an envelope-light building and because those rules of thumb encode exactly the inflated internal-load assumptions Section 1 dismantles.

Chart 1 is generated by the model, not drawn by hand. Every constant it uses — including the crossover marker and the end-label positions — is emitted by the model. An earlier draft had a chart whose numbers had quietly drifted out of agreement with its own appendix, and a later one repeated the mistake in the label layer. Both are fixed.

The model source code is available on request. If you find an error in it, we would genuinely like to know.