How to Reduce the Cooling Demand in Office Buildings and Match the Machinery to Heat Pump Design Demand

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ABSTRACT

The most efficient cooling is cooling not needed.

By applying a combination of cooling recuperation in AHUs, indirect evaporative cooling, indoor temperature set point glide depending on ambient temperature the demand for mechanical cooling can be reduced by app. 50%. By combining this with thermal energy storage the necessary machinery could be further reduced, and thereby give a better match for the design performance, when the cooling machinery is used as a heat pump.

Keywords: Energy Efficiency, Demand Reduction, Load Shifting, Heat Pump Performance

1. INTRODUCTION

This paper examines what can be achieved regarding reduction in comfort cooling demand in an office building by adjusting the equipment expected to be there. From there, we investigate what additional equipment is needed to reach the point where the cold demand in terms of compressor swept volume, is matched closely with the design performance of a heat pump covering 95% of the heat energy demand.

During the work on this paper, the focus has drifted from an optimum solution in a new build, to what we can achieve in existing buildings by looking at what is there and tweaking the building control system to reduce the cold demand. We believe that this drifting of focus is of great value, as most buildings are already built.

The additional equipment needed to further reduce the cooling demand are introduced according to the investments necessary to achieve the desired result.

The building to be examined is a 17 000m^2 office building located in London, with 850 people working there.

2. ASSUMPTIONS AND BASIC METHODOLOGY

As in all studies of this nature, we have to make a series of assumptions in order to make the calculations feasible and understandable. This is a fairly simple model, so we have to take care to actually understand the data we put into it.

2.1. The building

As mentioned above, we are examining a building of 17 000 m^2 located in London City, the workplace of 850 people.

In the early phases of a project, where basically nothing apart from the floor space is known, a Norwegian rule of thumb on transmission and infiltration heat loss says app. 23.5 W/m^2 at ambient -20°C and indoor temperature at 22°C, or 0.56 W/m^2/K. We will use this figure later in this paper, when we are estimating the necessary heat pump performance.

2.1.1. Internal heat loads

We assume that the heat loss per person is 100 W, no matter if they are just present or actually working. Each person has a PC with a heat loss of 50W, giving us a total of 150 W/person. At an 80% presence, this will yield an internal heat load from people and their equipment of 102 kW.

In addition, we have heat loads from lighting, app. 5 W/m^2 and 0.5 W/m^2 miscellaneous, like printers, fax machines, coffeemakers etc. The total here will then be 93.5 kW, or both types of heat load totalling 195.5 kW.

2.1.2. Ventilation rates

According to British guidelines, the minimum air change is 6 l/s/person or 21.6 m^3/h/person. This means a minimum design ventilation rate of 18 360 m^3/h. This is a minimum hygienic ventilation rate, based on the human metabolism.

From this, we lock the minimum ventilation rate to 21 000 m^3/h, slightly above the minimum to ensure that we have enough air flow through the building.

However, the necessary ventilation rate to remove the internal loads and maintain a satisfactory indoor climate will be somewhat larger.

A rule of thumb, applied throughout the HVAC-community in Norway, states that a temperature difference of 6K from the air inlet temperature and the room temperature is low, but manageable with regard to the feelings of draft etc.

From the free software “CoolProps” function for humid air calculations “HAPropsSI”, using the average temperature of 19°C and the absolute humidity at 15°C/80%rH we find the heat capacity of the air to be 1.022 kJ/kgK and the density to be 1.203 kg/m^3.

To remove the surplus heat of 195.5 kW we need a mass flow of 31.9 kg/s. This means a volume flow of 95 400 m^3/h from the AHU trough the building, but the inlet volume flow will be 98 200m^3/h. The difference is due to the difference in density due to the drop in temperature.

We therefore find that the necessary summer ventilation rate should be app. 98 200 m^3/h fresh air supply.

2.2. Climate. Ambient conditions

Designing a heating system or a comfort cooling system, only requires climate data for two points, the winter ambient design temperature (WADT) and the summer ambient design condition (SADC). But to design a heat pump installation you need to know how the entire heating system is behaving during the year. We therefore must find annular climate data for a location relevant to the location of the building.

In this case we use Meteonorm 8.0.3 to generate a reference year for London City Airport, with 8 760 data points on ambient temperature and relative humidity. We checked this against the ASHRAE Fundamentals 2017 0.4% weather data for London City, and we strongly suspect there is an error in their database generation, as the point is too low on the humidity. We suspect that the data for the dewpoint has been entered as wet bulb temperature and vice versa. The purple dot in Figure 1 is the point where this correction has been made.

To ensure that the ASHRAE data summer design point is of, we generated a similar scatter, but looking at the ambient enthalpy as a function of temperature.

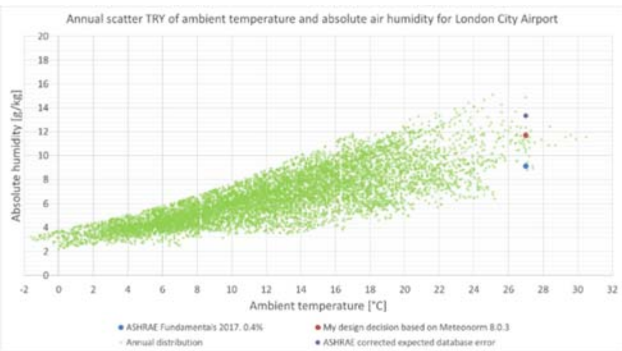

Figure 1. Annual scatter of temperature and absolute humidity. London City Airport

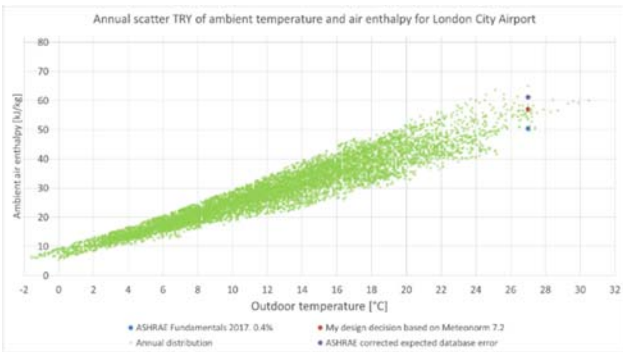

Figure 2. Annual scatter of air enthalpy relative to temperature. London City Airport

From the scatter in Figure 1 we can generate a diagram of the time/temperature interval.

This translates to what in Norway and Denmark is called a temperature duration curve.

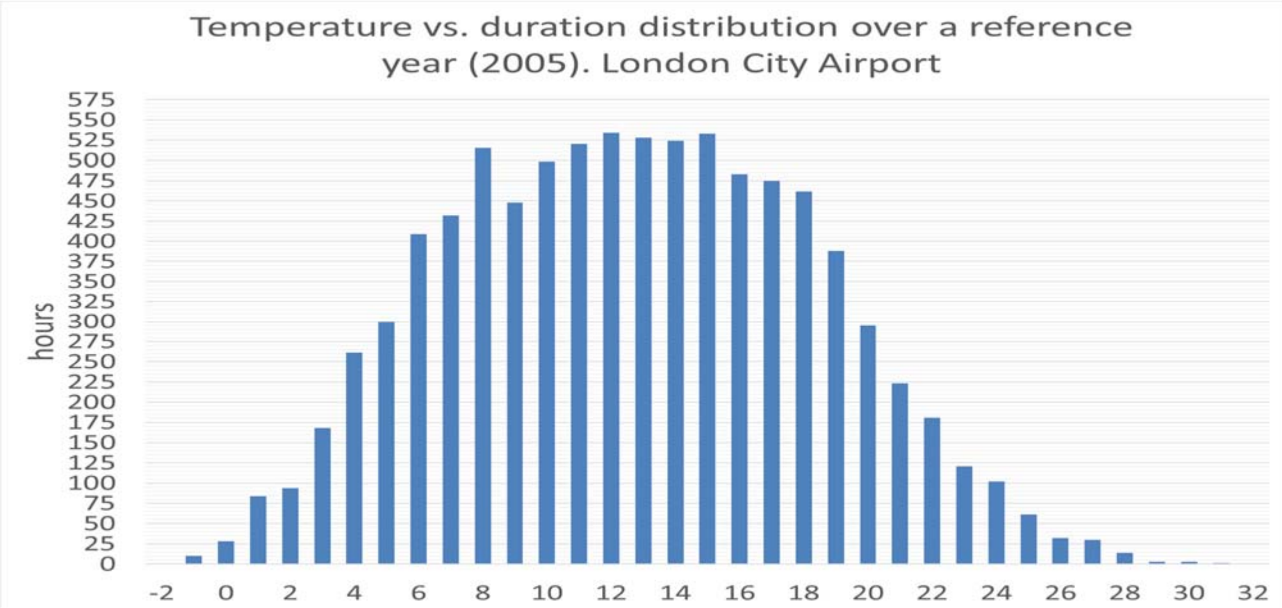

Figure 3. Temperature interval and corresponding duration

The way this curve is used is that at any given point, this is the maximum temperature.

If we want to know how many hours the temperature at London City Airport is below 6°C, you follow the line from 6°C to the curve, and on the x-axis, you can see how many hours the temperature is 6°C or below (1 354 hours).

We can also see that the mean temperature for London City Airport is just above 12°C.

The temperature duration curve would also suggest that the winter ambient design temperature is app. -2°C.

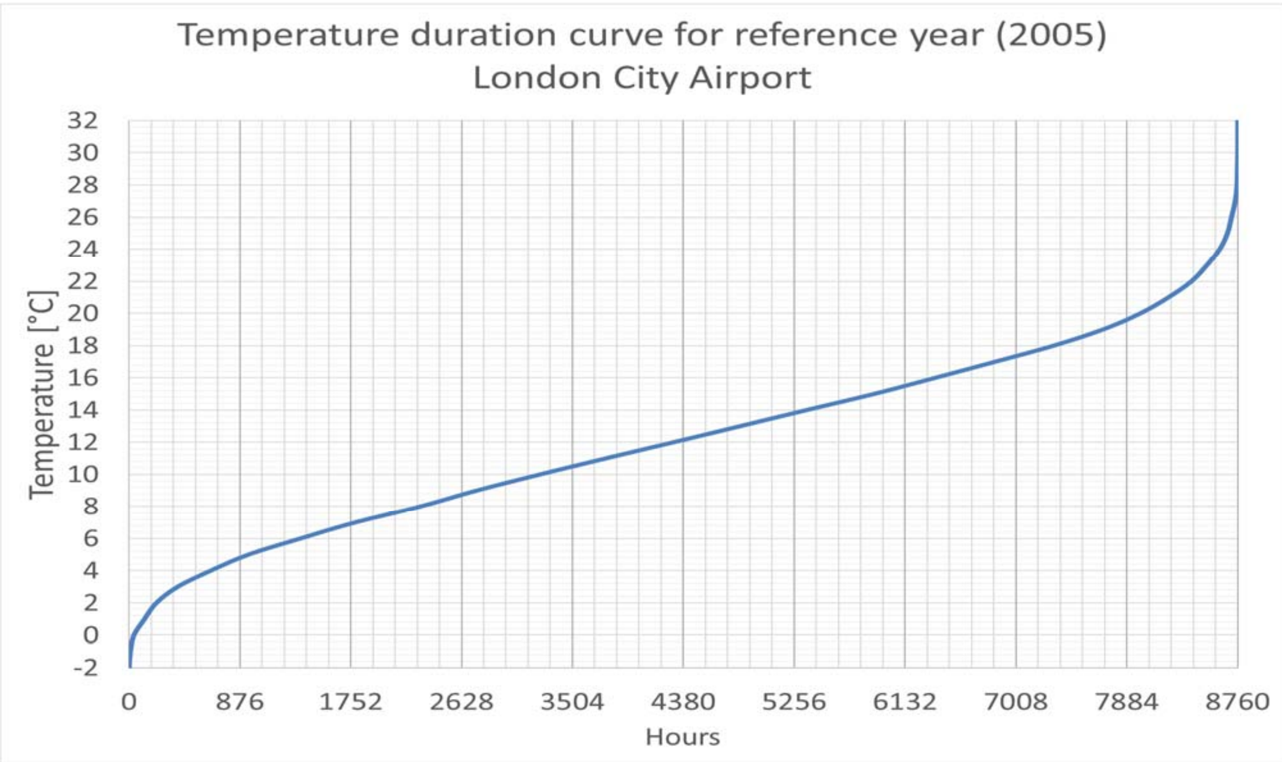

Figure 4. Temperature duration curve for London City Airport

2.3. Building heat and cold demand

2.3.1. Heat demand

From the Norwegian rule of thumb mentioned above, we assess the area specific transmission and infiltration heat demand to be:

$$23.5 \times \frac{22-(-2)}{22-(-20)} \sim 13.4\, W/m^2$$

which translates into a transmission and infiltration heat demand of **228 kW**. It is assumed that this heat demand is linear dependent on ambient temperature. We know this is also dependent on the temperature front moving through the wall but using the temperature duration curve has a tempering effect on the influence of this, and from experience we know that we actually come quite close to real-world results by employing this method.

The design heat demand for ventilation is found from the minimum ventilation rate, the recuperator efficiency of 65% and the design indoor temperature of 22°C. At this point we do not include the ventilation fans contribution to temperature gain, as the design flow is 98 400 m^3/h, and the winter design is 21 000 m^3/h, which means that the fans run with far less power demand in winter than during summer, where we assume to have a temperature gain from fans of app. 1K.

Treating the ventilation recuperator as a normal heat exchanger, with incoming warm air at 22°C and incoming cold air at -2°C, we find that the dT across the recuperator is 15.6K, which means that the supply air temperature after the recuperator will be 13.6°C.

The heat demand to reach 22°C from 13.6°C at an airflow of 21 000 m^3/h is app 60 kW. We design the heating coil to be able to reach isotherm air supply, to avoid a “false” heating demand signal. When the occupancy is low the internal loads are small, triggering temperature readings below the setpoint, which again might trigger higher temperature level in the heating system, which in turn gives unnecessary harder conditions for the heat pump.

The total design heat demand is found to **288 kW**, at -2°C ambient.

2.3.2. Cold demand

Finding the cold demand in ventilation cooling is not as straight forward as determining the heat demand, as not only the temperature of the air both before and after cooling, but the humidity and the temperature of the coiling coil comes into play.

The method applied is a modification of the old method of air mixing. In this method we know the condition of the incoming air, the desired temperature after cooling and the surface temperature of the cooling coil, in this case an evaporator with basically isotherm surface.

In the psychrometric chart, you draw a line from the condition of the incoming air to the coil surface temperature at 100% rH. The end point of the cooling process is found at the desired outlet temperature on the line connecting air inlet and the dew point temperature corresponding to the surface temperature of the coil.

Note that this latter temperature is higher than the temperature of the cooling medium, as the energy must travel from the surface to the medium.

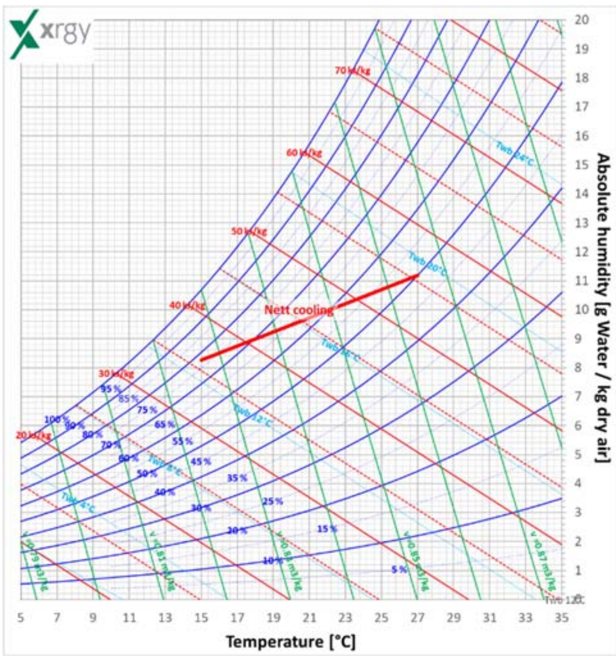

Figure 5. Example of simple assessment of cold demand on isotherm surface

The use of isothermal surfaces gives us a problem when using chilled water as cold bearer. We employ a modification of this method, where we divide the coil into 100 coils, where we assume that this little surface can be regarded as being isotherm.

Our experience is that this simple method usually is within 5% of the actual demand when calculating specific cooling coils.

Each country has its own traditions in these fields, and I am not aware of the British, I employ the Norwegian. 10 – 15 years ago, a cooling coil like the one we are discussing here, would be (unthinkingly) designed for a temperature pair of 7°C/12°C.

As discussed earlier when determining the summer ventilation rate, we want to have the inlet temperature of the air to be 16°C. Due to the pickup of heat, primarily from the fan, we assume that the temperature of the cooling coil is 15°C.

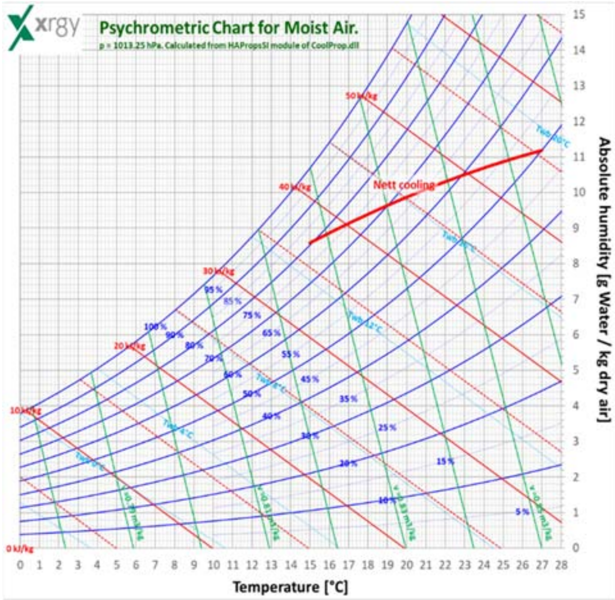

Figure 6. Actual cooling proces 27°C/50%rH to 15°C. Chilled water 7°C/12°C

Using a design ambient condition of 27°C/50%rH, the design cold demand is found from the method mentioned above to be 18.92 kJ/kg or **604 kW**.

This figure alone should be shocking. To meet a cooling demand of 195.5 kW, we have to produce 604 kW of cold, or three times the cooling demand. This is quite a good illustration of the poor efficiency of cooling by ventilation air.

3. COOLING DEMAND REDUCTION WITHOUT INVESTING IN NEW EQUIPMENT

When proposing this, the typical reaction is that it cannot be done.

3.1. Using the recuperator for precooling of supply air

The simple counter question is do they operate their recuperators during the cooling season? There are many instances where they do not.

It is a heat recuperator, and in summer there is no need for additional heating.

However, the recuperator is a heat exchanger, it doesn't matter what direction the heat travels.

If you have an indoor temperature that is lower than the ambient, you can use the outgoing air to precool the supply air.

In the example shown in Figure 7 we have an indoor temperature of 22°C. The recuperator has a temperature efficiency of 65%. This little change in the control system will reduce the cold demand from 604 kW to 461 kW.

No investment, just a few lines added to the building management system.

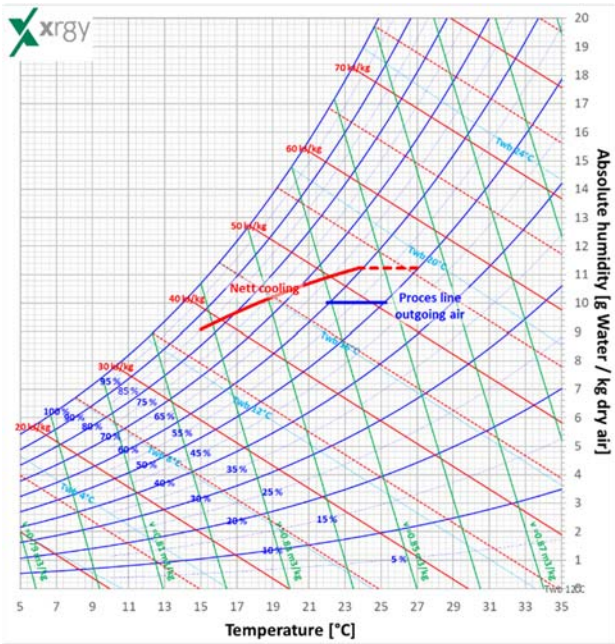

Figure 7. Cold demand reduction by use of recuperator for precooling

3.2. Letting the indoor air float in accordance with the ambient temperature

A well-known challenge with comfort cooling in an office building is that quite often they are freezing cold. There is a mismatch between the desired indoor temperature, and what temperature actually is comfortable. A way to mitigate this is to let the indoor temperature float according to the ambient temperature.

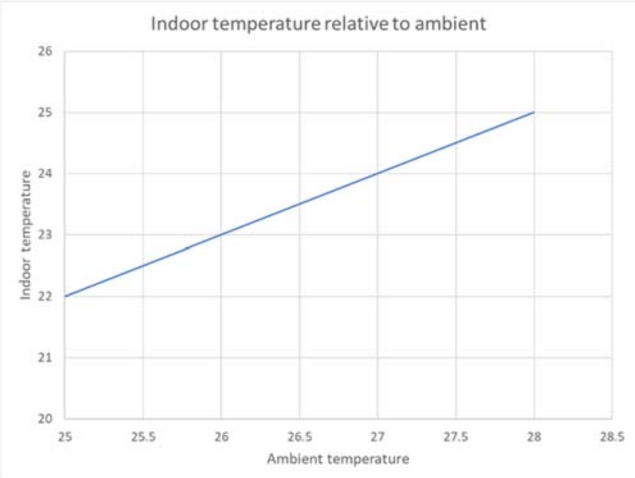

Figure 8. Example of compensation curve for indoor temperature as a function of ambient

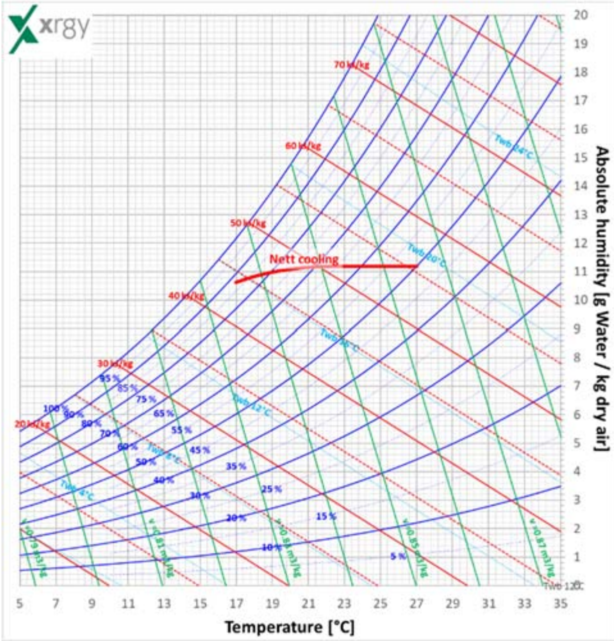

Figure 9. Cold demand by floating indoor temperature

In the example shown in figures 8 and 9, we see that the indoor temperature at 27°C ambient should be 24°C, which in turn means that the temperature of the cooling coil rises from 15°C to 17°C.

This reduces the cold demand from 604 kW to 408 kW, and likely increases the comfort of the occupants. Again, a substantial demand reduction by adding a few lines of code.

3.3. Combining use of recuperator and floating indoor temperature

Now of course the question begs, what happens if we combine the two methods.

Just by using the recuperator and letting the indoor temperature float with the ambient temperature, we achieve a reduction in cold demand to 310 kW, a reduction of 49%.

This can be accomplished without investing in new equipment, just using what we have in a more efficient way, and at the same time probably achieving a better comfort for the occupants.

A reason why we see this marked drop in cold demand, is that as the temperature off the coil rises, the necessary chilled water temperature rises as well, resulting in marked less dehumidification.

In the original situation the sensible part of the cooling process was 66%, whereas in the last the sensible load was 85%. The drop in dehumidification is also very clear to see in the process graphs.

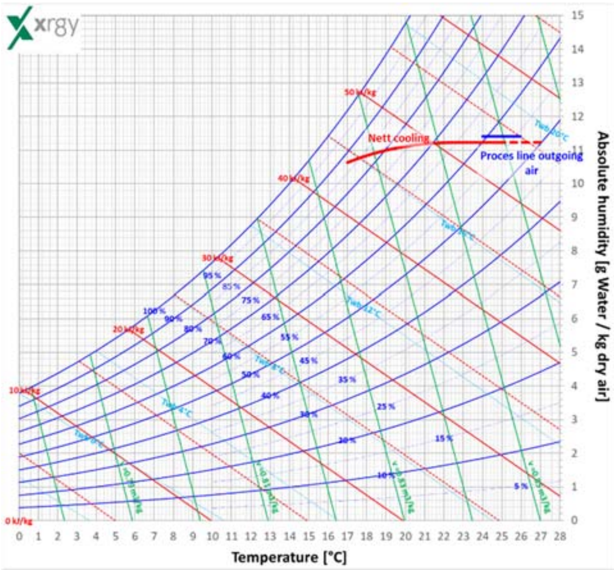

Figure 10. Cold demand by combining use of recuperator and floating indoor temperature

4. HEAT PUMP DESIGN PERFORMANCE

The design performance of a heat pump is based primarily on a desired heat coverage, typically about 95% of the total annual heat demand covered by the heat pump.

Putting the example building and the location in London through this software, including a workday load distribution, shown in Figure 11, gives us an easy look up, shown in Figure 12.

As we can see from this graph, a desired heat energy coverage of 95% mean that we have to install a heat pump with a performance between 130 kW to 140 kW.

When that heat pump is used as a chiller, the cold rate will be somewhere between 160 kW – 180 kW.

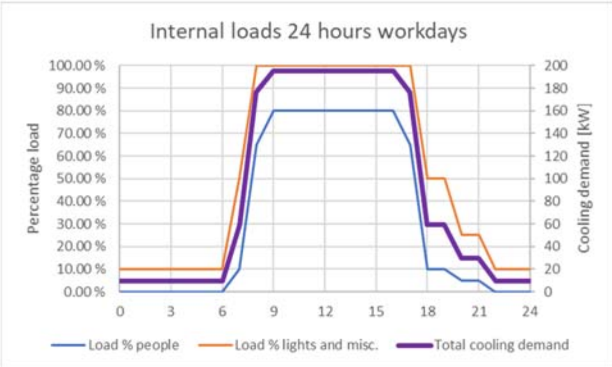

Figure 11. Intern loads during workdays

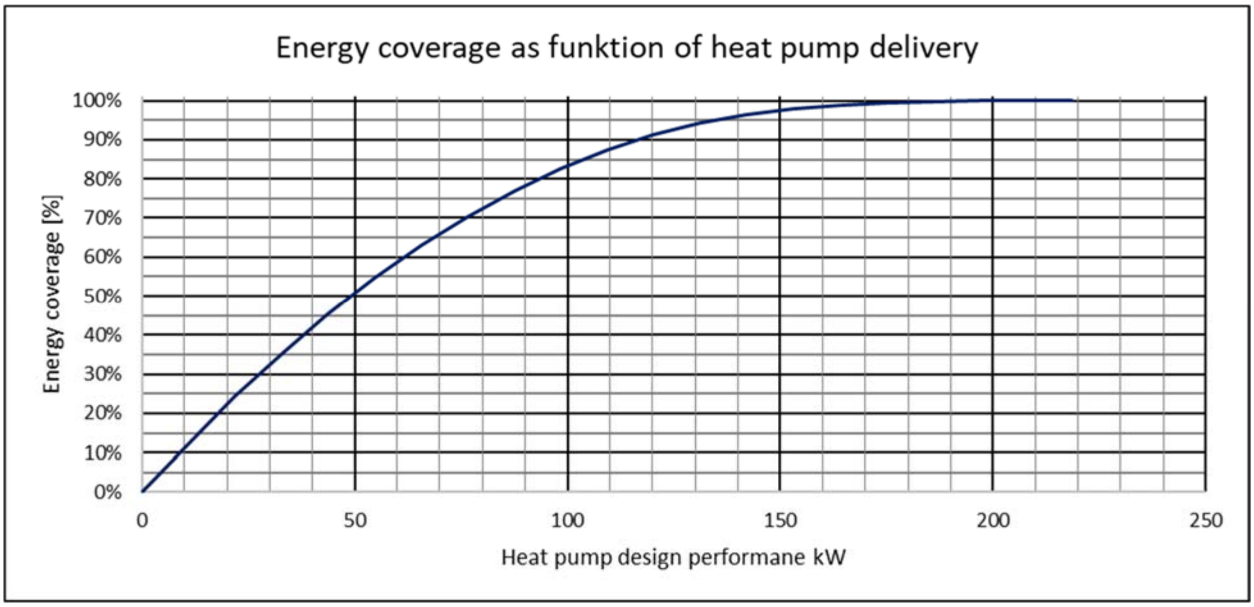

Figure 12. Graph that shows the energy coverage in relation to the heat pump design heat rate

So, looking at the new cold demand of 310 kW, it is clear that we are closer, but we haven't yet earned the right to celebrate. Is there anything we can do to earn it? In my mind, not a lot, unless we bring out the purse and invest.

Conclusion

By using the equipment usually present, we are, in this example, able to reduce the cold demand from 604 kW to 310 kW, a reduction of 49%. We have done this without investing in new equipment, just in lines of code in the building management system.

A reduction this big by just looking hard at what we have, what we can do with it and a slight challenge to tradition is quite surprising, and a clear indication that energy in general is too cheap.