Case Study

Relationship between heat pump & Radiant systems performance and carbon saving – a case study

A simple and relevant scenario

This case study is centred around the use of heat pumps as the heat source and banks of radiant panels as the heat emitters to provide comfortable classroom conditions in schools designed and built in line with the requirements of building bulletins 87 and 101.

Heat pumps have traditionally been associated with medium or low water temperature operation and hence have been deemed suitable only for low temperature emitters such as underfloor heating systems. The advent of high temperature heat pumps opens up the possibility of combining these with more effective emitters, one of which is radiant ceiling panels which provide increased flexibility due to their modularity and, as a result of their low thermal inertia, provide quick response and heat-up times. Unlike conventional combinations of heat sources and emitters, the energy efficiency of the heat pump/radiant panel combination will increase at part load conditions, a feature which is explored in detail below and converted to a quantifiable saving in carbon dioxide emissions.

Heating load calculations are conventionally performed based on worst case scenarios for space heat losses. While this worst case condition may apply for a few hours each year its use in sizing equipment invariably leads to oversized conventional equipment with the attendant problems of inefficient part load performance and control issues when the equipment is operating distant from the conditions against which it is selected to run. One solution is to design the equipment against conditions which occur more commonly rather than as a worst case and rely upon a back-up for the most severe hours; while imminently sensible this is a situation which would require a change in accepted practice and is not considered in this study.

The combination of high temperature heat pumps and radiant panels can circumvent the above arguments allowing equipment to be sized conventionally against worst case conditions in the full knowledge that its efficiency will increase and control will not be compromised at part load, under which the equipment will operate for 99% of the heating season. As radiant panel outputs increase almost linearly with mean water temperature their outputs are directly linked to the control of the heat pump's water delivery temperature throughout the performance range. The excess in available panel area at part load, rather than being a hindrance now becomes a positive benefit when coupled to the high temperature heat pump as it allows the heat pump to operate at lower water supply temperatures and hence at a higher level of energy efficiency. Conversely, and beneficially, the relatively high supply water temperatures associated with the latest range of heat pumps allows economical numbers of radiant panels to be selected against the worst case design conditions.

The following analysis is aimed at demonstrating the annual carbon dioxide savings that would be expected to accrue compared to the use of conventional boiler technology. While it is only strictly valid for the case study in question a good approximation of the available savings for other situations can be assessed by straight comparison with the design heat source requirements. Biomass boilers are not used in the comparison as the decision between Biomass and competing technologies cannot be made on the basis of explicit emission savings; other less obvious factors need to be considered.

The case study: design conditions and equipment selection

The heating system in question is intended to provide the heating requirements for a number of classrooms within a school. These could represent the entirety of a small school or could be an individual zone of a larger complex.

Two scenarios are studied; the first has a design heat loss of 100kW and the second a design heat loss of 200kW

Design space temperature	19°C
Design outside temperature	-5°C
Design fabric loss 1	12kW
Design ventilation loss 1	88kW
Design total heat loss 1	100kW
Design fabric loss 2	24kW
Design ventilation loss 2	176kW
Design total heat loss 2	200kW
Design hot water F/R	65/60°C
Radiant panel output	410W/m2
Area of panel required 1	244m2
Area of panel required 2	488m2
Total length of 600mm wide panels 1	407m
Total length of 600mm wide panels 2	813m
No. of 3m long panels required 1	136
No. of 3m long panels required 2	271
Prana HT heat pump output at design air/water temperatures 1	100kW
Prana HT heat pump output at design air/water temperatures 2	200kW

(Specification for case study)

Table 1. Design conditions and equipment selection

The selection of panels in 3m modules is given as an example only and actual panels sizes will vary will the layout of the room and suspended ceiling. The selection of 19°C as a space temperature is based on the use of a radiant heating system; such systems can operate successfully at lower space temperatures as the increased temperatures of the room surfaces provide a higher ‘operative’ temperature than would be available for a warm air heating system. Space air temperatures would typically be 2 to 3°C lower for a comfortable radiant heating system.

The heat loss has been split between fabric and ventilation losses. The fabric losses are simply a function of the internal and external design temperatures chosen and the U values of the walls etc. The design ventilation losses are calculated from the stipulation for a design value of 8 litres/s per occupant. The building regulations specify the above value and the required U values for new buildings and as a result the majority of the design heat loss will always be attributable to ventilation losses. These ventilation losses are often offset at source by the incorporation of hot water heater batteries within the ventilation inlets, this analysis assumes, however, that all the losses are offset by the radiant panel system.

The heat source selected is a high temperature heat pump capable of generating hot water up to 65 degrees. The heat pumps are installed with a buffer tank and integral circulating pump comprising the primary circuit and ensuring full flow through the heat exchanger. A secondary pumping system is employed on the other side of the buffer tank to supply the radiant panel runs. The heat pump is rated at 100kW so 2 off of the units are required for the 200kW system. These 2 heat pumps will supply a common buffer tank and be controlled by a common weather-compensated controller.

Actual/seasonal system performance

At the design conditions given above the combination of heat pumps and radiant panels will provide the required heating capacity against the worst case conditions. These conditions require that the heat pumps generate hot water at the maximum temperature available i.e 65°C. At this temperature the coefficient of performance of the heat pumps (COP = useful heat output/rate of energy input) is low, registering a value of only around 2.1. Figure 1 identifies the variation in COP for the AWHT heat pump with variation in the ambient air temperature and the water supply temperature.

The COP for any heat pump is always maximised thermodynamically by minimising the temperature difference between the supply water into which heat is transferred and the ambient air from which heat is absorbed. This variation in COP is dampened when using AWHT heat pumps due to the Enhanced Vapour Injection feature of the scroll compressors used. This technology allows for injection of high pressure/temperature vapour midway through the compression process and allows a single compressor to attain compression ratios which would normally be associated with multi stage compression. The high compression ratio achieved using EVI ensures that the compressor can work efficiently even when source and sink temperatures are wide apart.

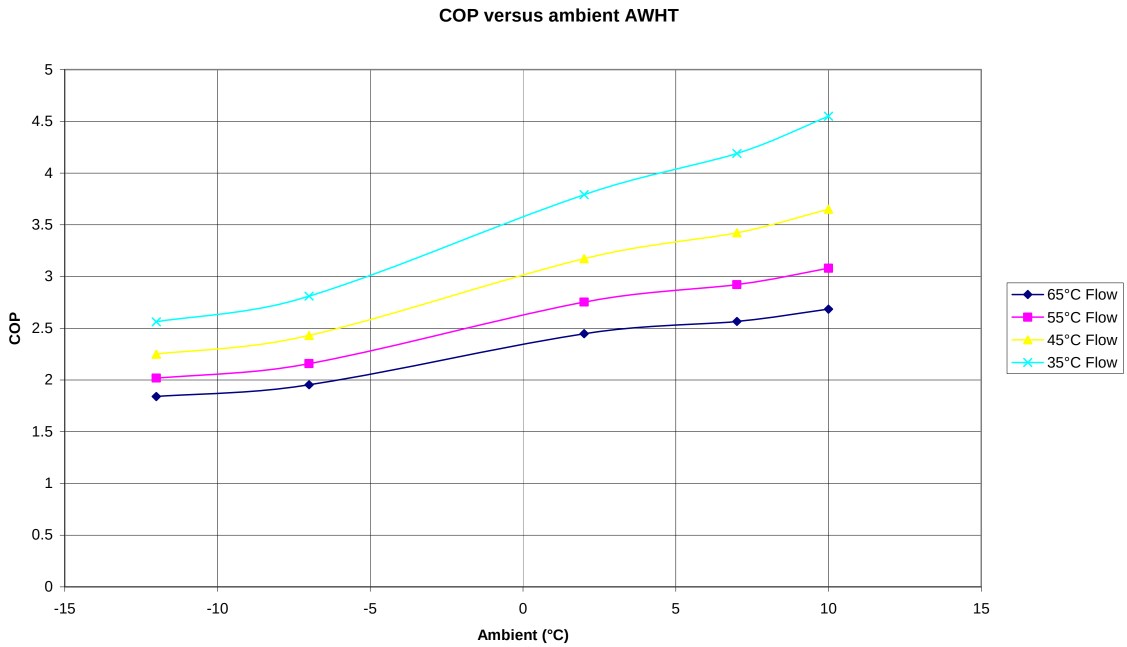


Figure 1. COP at a range of operating conditions

The rate at which the heat pump absorbs electrical energy for its operation remains reasonably flat as the temperature of the ambient air increases, the rate at which the heat pump generates heating energy in the form of hot water, however, increases considerably with increasing ambient temperature hence the increase in the COP, which compares the latter to the former, when exposed to milder conditions.

If the heat pump delivered its heat constantly at the worst case design condition then it would operate at a constant COP of around 2.1 i.e. it would be just over twice as efficient as an electric heating system. As the ambient temperature increases, however, the heating load is reduced allowing the heat pump to supply water at a lower temperature and permitting operation at higher COP levels. These lower supply temperatures are sufficient for the radiant heating system to maintain the room at the design comfort temperature due to the reduced room heat losses.

In order to undertake a meaningful analysis of the overall energy requirements and applicable carbon emissions it is necessary to quantify the performance parameters of the heating system across the range and duration of conditions that would be expected throughout the annual heating season. Before doing this it is instructive to take an overview of the techniques used by the heat pump controller to match the supply water temperature to the instantaneous heat load.

Figure 2 shows the so called ‘heating curve’ for the heat pump. The heating curve represents an algorithm built into the heat pump’s controller which automatically varies the supply water temperature in response to a change in the ambient temperature. There are several heating curves incorporated in the controller and the selection of the correct one depends on the type of emitter being used. These curves are characterised to allow the heat output from the emitters to vary inversely with the ambient temperature and reducing heating load. The curve shown is almost linear and characterised to suit a radiant heating system.

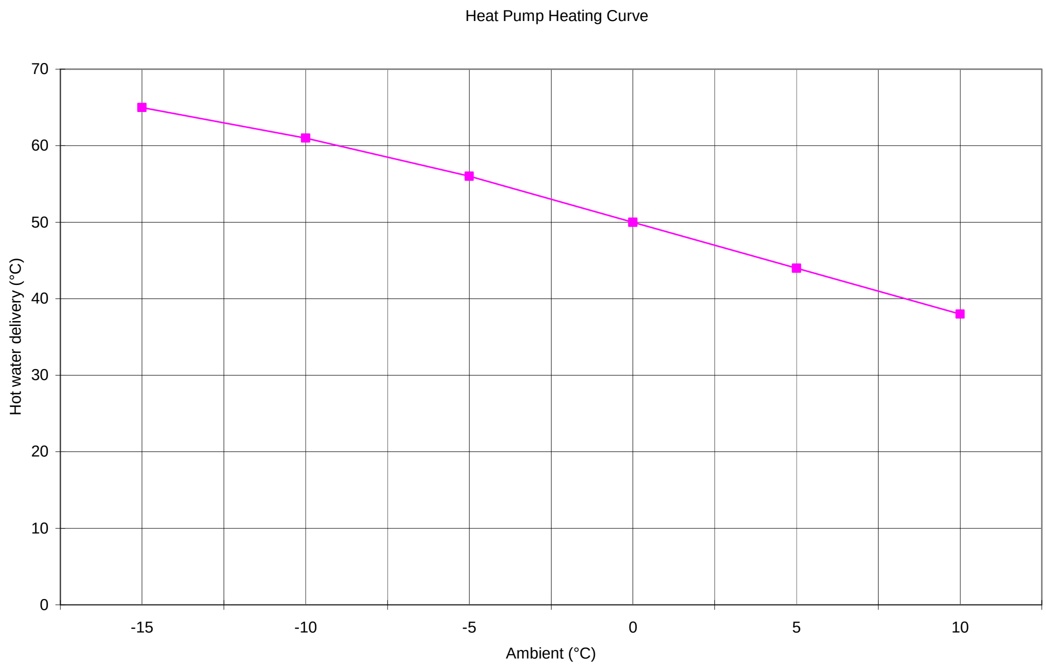


Figure 2. Heat pump heating curve to suit radiant heating system

Essentially, the heating system output is determined by the heating curve operating on the sensed ambient temperature. In this way the heat pump system is able to maximise its efficiency (COP) by reducing the temperature of the water supplied to the heating circuit, emitters piped into the system should not be directly controlled by room thermostats if the energy savings of this control method are to be fully realised. A room temperature sensor is supplied with the heat pump control system and its sphere of influence on the water delivery temperature can be increased from a default setting of 0% should fine tuning of the system be required. This is at odds with conventional heating systems whereby the emitters are controlled by valves or dampers which react to a space mounted thermostat or temperature sensor.

To assess the annual cost of heating in terms of carbon emissions an accurate record of ambient temperatures and their duration throughout the heating season is required. This data is available from several sources and figure 3 shows the ‘bin’ data for a UK location (London) based on historically recorded data. This data separates the full range of recorded temperatures into a number of discrete ‘bins’ against which are logged the number of hours p.a. when the temperature falls within this bin. The data is intended to represent as closely as possible, within the constraints of the available data formats, the ambient temperature spectrum which will portray only the heating season and only those periods of occupation. This has been taken as the middle third of the day whenever the ambient temperatures fall below 13°C and is only appropriate for public and commercial premises; a domestic dwelling would be more properly represented by data for the hours not covered by the above.

As shown in the chart, the worst case condition which was taken as -5°C in the design specification, occurs for only a very few hours p.a. and that the majority of the operating hours are towards the higher ambient temperature end of the spectrum. As the ambient temperature increases the water delivery temperature from the heat pump will reduce in line with figure 2 and the COP will increase in line with figure 1

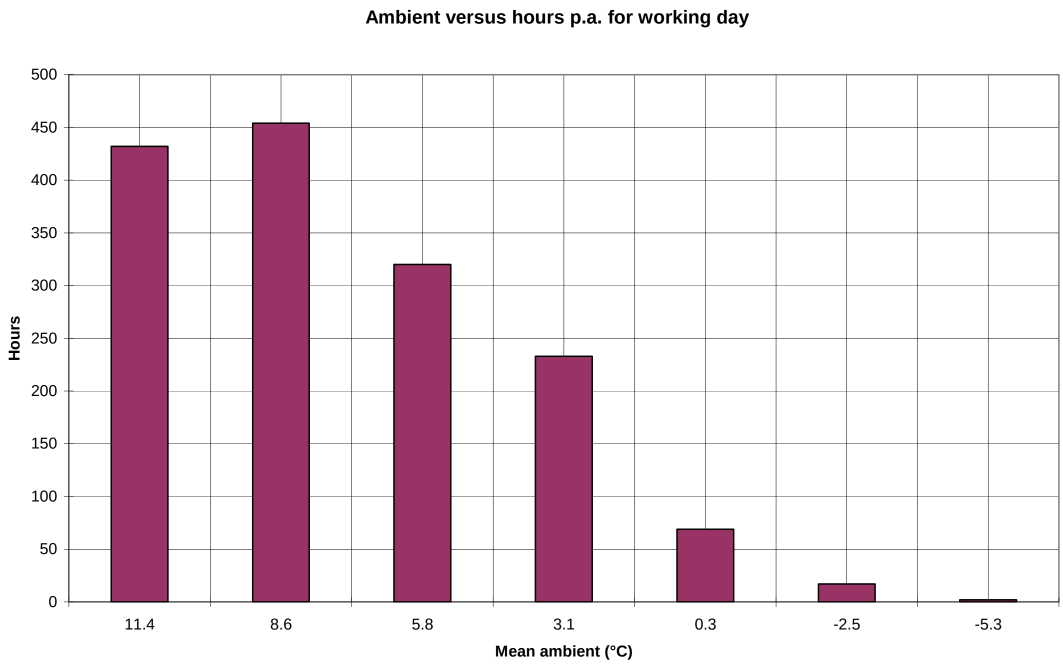
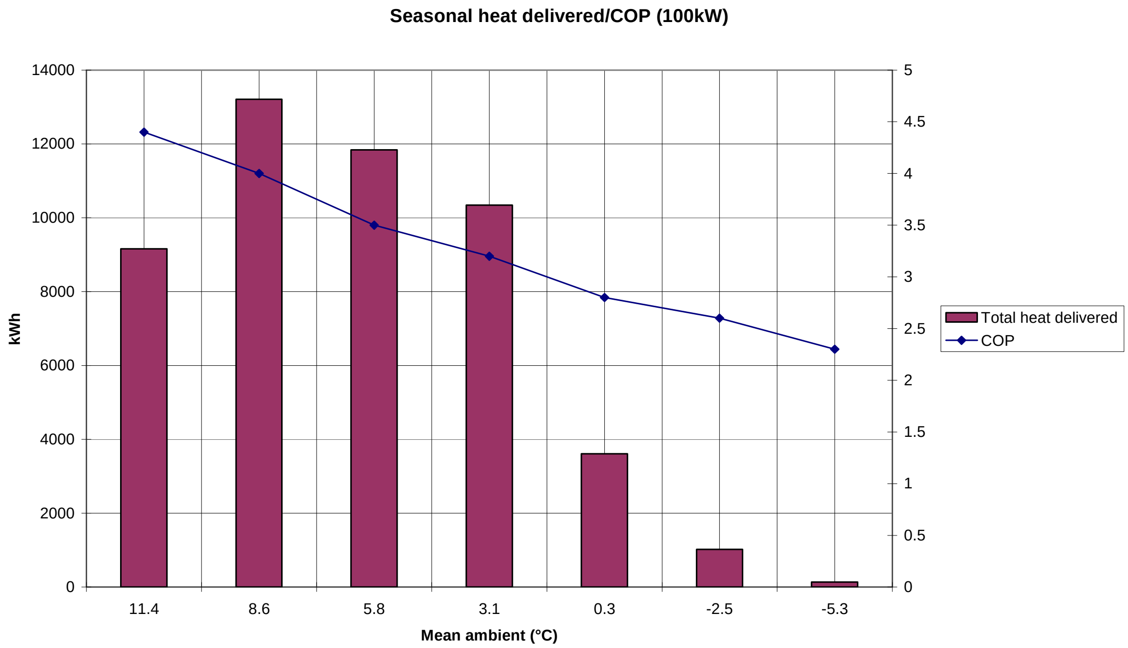


Figure 3. Frequency of ambient air temperatures during occupied periods of heating season

The total delivered heat in kWh for each of the prescribed bins is shown in figure 4 along with the operating COP for that condition. The correct value of COP for each bin is obtained by determination of the supply water temperature from figure 2 and the corresponding COP from figure 1.



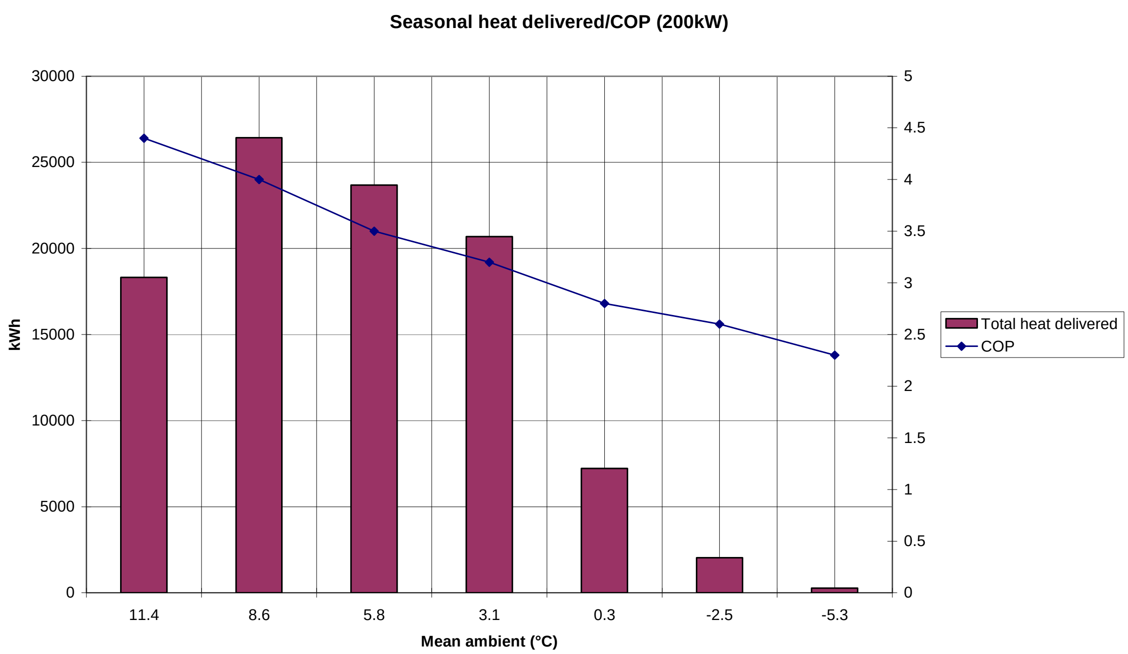


Figure 4. Total heat delivered and heat pump COP for heating season

The total annual heat load obtained by summing the columns in figure 4 is 49321kWh for the 100kW installation and 98643kWh for the 200kW installation. The COP for the heat pumps varies across the heating season from 2.1 for the worst case up to 4.3 at the warmest ambient condition. The weighted mean value for the COP from figure 4 is 3.8 for both scenarios. Combining the total heat supplied and the weighted mean COP gives a value for the electrical energy expended in supplying this heat as 12979kWh and 25959kWh respectively.

The heat loads and heat delivered above are based on actual ventilation losses that would be expected. The worst case design ventilation load is based on the stipulated rate of ventilation i.e. 8 litres/s as prescribed in building bulletin 101. This document states, however, that the actual ventilation rate should be equal to 5 litres/s in order to maintain acceptable CO_2 levels. The heat losses above are based on full occupancy and an actual ventilation rate of 5 litres/s, these losses will be conservative due to the assumption of full occupancy.

Carbon savings – the bottom line

Using current government figures for the CO_2 emission associated with the national grid of 0.562 $kgCO_2/kWh$ the emissions associated with the heat pump supplying an annual 49321kWh of heat will be 0.562 x 49321/3.8 = 7294 $kgCO_2$. For the 200kw scenario the emissions will be equal to 0.562 x 98643/3.8 = 14589 $kgCO_2$. This can be compared to the alternative of gas fired boilers using the following assumptions for boiler seasonal efficiency and government emissions figures.

Condensing boiler: Maximum seasonal efficiency = 88%, CO_2 emissions @ 0.206 $kgCO_2/kWh$, total emissions = 0.206/0.88 x 49321 (98643) = 11546 (23091) $kgCO_2$

Conventional boiler: Maximum seasonal efficiency = 75%, CO_2 emissions @ 0.206 $kgCO_2/kWh$, total emissions = 0.206/0.75 x 49321 (98643) = 13547 (27094) $kgCO_2$

Normalised emissions:
Air source heat pump = 1
Condensing boiler = 1.58
Conventional boiler = 1.86

From the above, carbon savings of 37% will be achieved compared to a modern, efficient condensing boiler. This figure is somewhat higher than rule of thumb savings quoted for air source heat pumps which indicate savings of between 20 and 30%. The increased savings are not only associated with the efficiency of the heat pumps being proposed but are also a result of the detailed analysis of the actual operating conditions. An analysis which was based on 24 hour operation would show a somewhat reduced saving but would be inappropriate for the situation being studied.

While boilers of all types show some variations in efficiency at different loads, these variations are small compared to the variations in heat pump COP. Heat pumps should be treated differently to boilers if an accurate assessment of the savings is to be made. Mean values of COP do not adequately quantify the potential savings.

The emissions quantified above are somewhat simplified for the benefit of the analysis; they do not take into consideration emissions that result from pumping equipment of other system losses. As these are likely to be of an equal order for the various systems considered this will not invalidate the exercise.