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Mimer GeoEnergy

Effects of cycling on domestic GSHPs.

Supporting analysis to EA Technology

Simulation / Modelling.

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Report No: C207-R2

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Domestic GSHPs - effects of cycling

1. Introduction

The Department of Energy and Climate Change (DECC) have commissioned EA Technology (EAT) to carry out investigations into the effect of cycling on examples of domestic air source heat pumps (ASHPs) and closed loop ground source heat pumps (GSHPs). This follows on from earlier work carried out by EAT for DECC on the effect of cycling on ASHPs (Green&Knowles2011) and is part of a series of activities commissioned by DECC following the EST monitoring trial of domestic ASHPs and GSHPs throughout the United Kingdom (EST2010, Dunbabin&Wickins2012)

Mimer have been sub-contracted by EAT to provide support for the re-commissioning of the GSHP's ground loops, the selection of a suitable GSHP, and supporting analyses of the experimental measurements made by EAT of the GSHP behaviour. This report describes the simulation and modeling that has been developed based upon the measurements of the GSHP in operation.

The objective has been to set up a simulation model that can reproduce the observed experimental measurements, and can then be used to explore issues that cannot be achieved in the current experimental setup. This report should be read in conjunction with the report prepared by EA Technology (Green 2012)

2. Background to experimental set-up

The work has been undertaken using one of two identical houses (Figure 1) constructed for test purposes at EA Technology's Capenhurst site in the 1990's. The house fitted with radiators has already been used for earlier work on the effect of cycling on an ASHP. This house happens to have had two closed loop GSHP boreholes installed in 1998 to investigate the potential of heat pumps for active and passive cooling in domestic properties using small heat pumps. Although these boreholes have not been used in the intervening years, it was possible that they could be re-commissioned and used to investigate the behaviour of a modern domestic GSHP in heating mode.

Figure 1: The experimental house. (on the left)

3. Heat pump cycling

There are a number of issues associated with the "cycling", ie switching on and off of heat pumps, particularly in the domestic sector.

Electrically driven vapour compression heat pumps, ie the vast majority by far of domestic heat pumps, will usually contain the largest electrical motor present in a UK dwelling. This will be nominally in the range ~1kW electric, up to possibly ~4kW electric, in the single phase environment that covers practically all UK dwellings. For fixed speed compressors, the starting of these motors is characterised by a short (milliseconds), but very high, starting current which can often have an effect on lighting (flicker) or other electrical equipment in the dwelling, or even in adjacent dwellings on the same phase. There are also issues related to the mechanical shock on the compressor, the time to achieve effective lubrication in the compressor's refrigerant circuit, and effects arising from trying to restart a compressor that is still in a hot/high pressure state following a prior run.

As well as these known effects, a query arises over whether there may be impacts on efficiency due to short cycling of the heat pump. An analogy arises with fossil fuelled boilers, where short cycling causes a reduction in efficiency due to the combination of reduced combustion efficiency during start-up as well as having to bring the mass of the combustion chamber and associated heat exchanger up to temperature. In addition, a (closed loop) ground source heat pump, which has highly dynamic temperature behaviour on its source (ie ground) side, may see a decrease in efficiency with increasing run time - due to short term, progressive cooling of the ground loop. This raises the possibility that there may be an optimal run time for a domestic GSHP.

Although there appears to be limited information in the literature regarding the cycling of heat pumps, it is none the less common practice amongst European manufacturers of water-to-water heat pumps (ie the vast majority of European domestic heat pumps) to make recommendations regarding the incorporation of buffer tanks to overcome short cycling. It is also common practice in the chiller industry to find recommendations for buffer tanks to be incorporated on the load side, for the same reason. Typical guidance figures for chillers are a buffering capacity of 10 litres/kW thermal, and for heating heat pumps, up to 40 litres/kW thermal. It should be noted that this practice does not arise for the bulk of American and Canadian domestic heat pumps, because the vast majority of these systems are water-to-air. It should also be noted that the primary reason for buffer tanks on ground source heat pumps is to prevent short cycling, and does not provide any significant thermal storage. The placement of buffer tanks on GSHPs is more flexible than for some ASHPs, where the buffer store is placed immediately downstream of the heat pump, and in some cases is used to provide thermal energy for defrost cycles.

4. Experimental setup

The experimental set up consists of a Kensa 6kW, model H062-S1H, ground source heat pump connected to the closed loop boreholes adjacent to the property. This services the radiators in the house which can be selectively isolated from the total heat distribution system as required in order to induce different cycling times. The system was instrumented with various electricity meters, heat meters, flow meters and temperature sensors - and is comprehensively described in the EAT report (Green 2012)

Later in the experimental programme, a 50 litre buffer tank was introduced between the heat pump and the heat distribution system to allow investigation of the effect of buffering on system efficiency.

After initial commissioning of the GSHP, EAT carried out a series of runs that resulted in different cycling times of the heat pump. Figure 2 is an example of a continuous run over a four hour period. Figure 3 shows the system cycling due to a significant reduction in the "load", achieved by shutting down a number of the radiators in the house. The temperatures displayed in both of these figures are the load side flow and return temperatures and the ground side flow and return temperatures.

Figure 2: Measured ground and load side temperatures for a "continuous run"

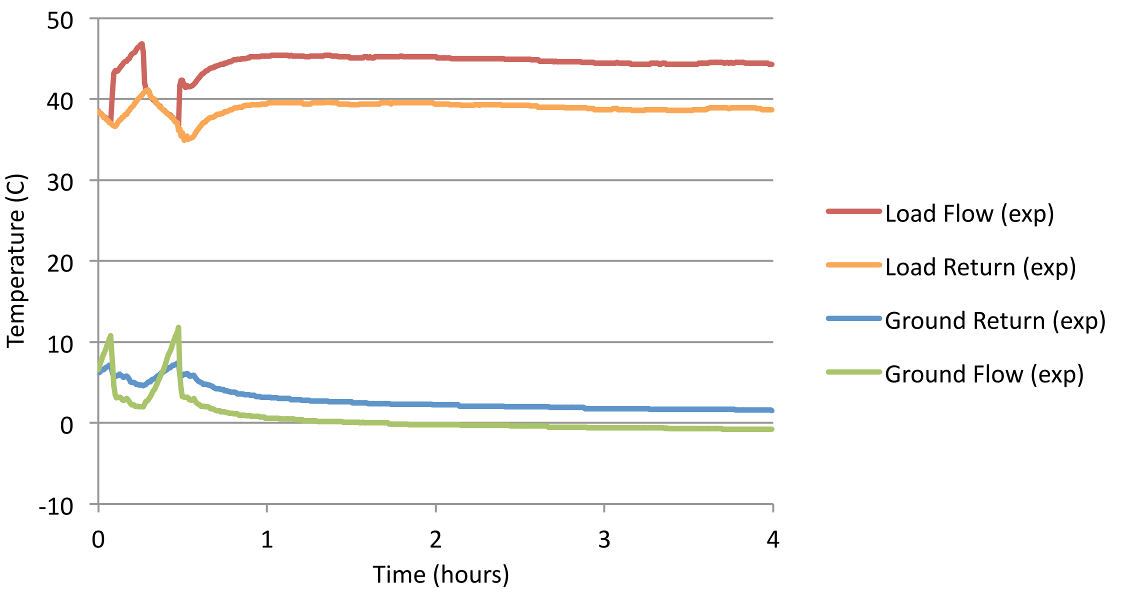
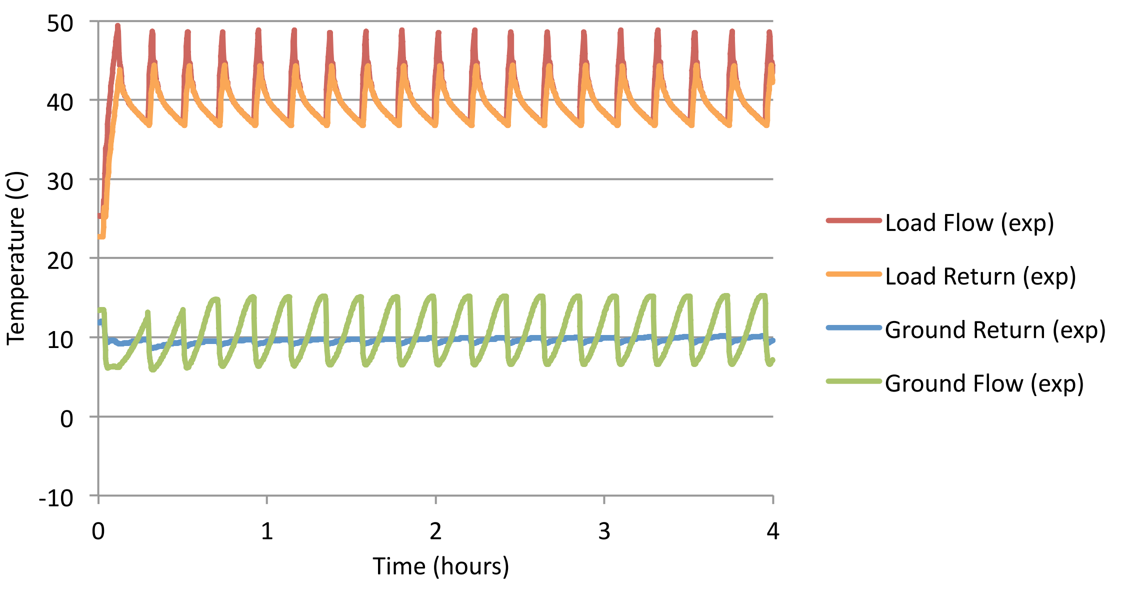


Figure 3: Measured ground and load side temperatures for a "short cycle" run

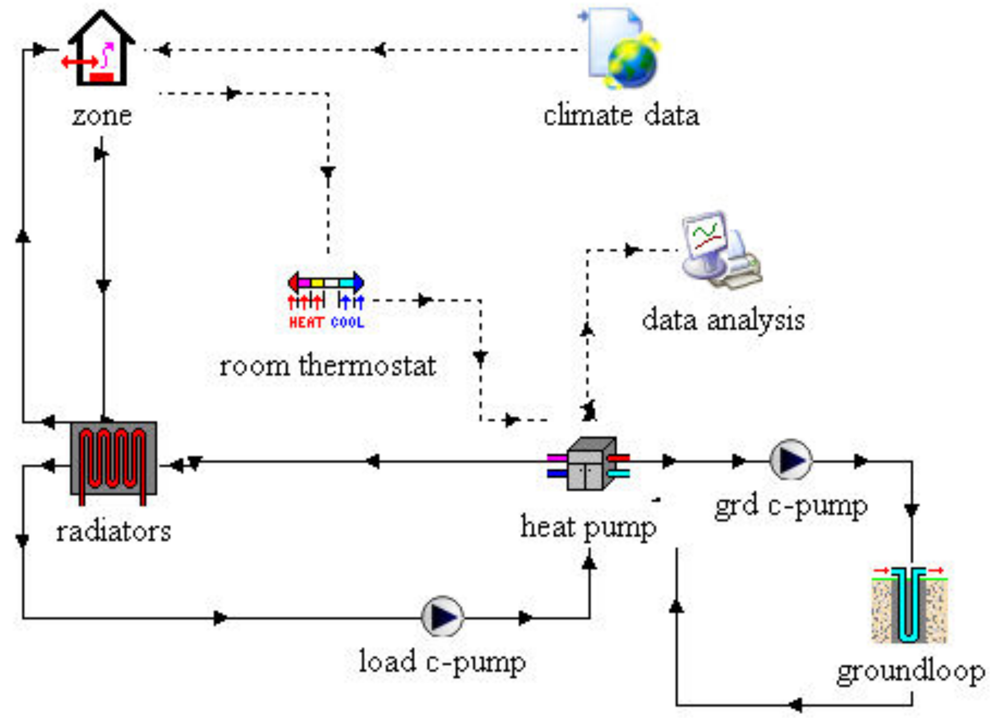


It should be noted that the green temperature trace, whilst being a real measurement, does not reflect the actual ground water "flow" temperature at all times. When the heat pump is off, the temperature appears to reflect either a room temperature or a backflow temperature from inside the heat pump. The ground temperature would only exhibit a reduction from the ground equilibrium temperature (ie circa 11°C at this site) once the heat pump started running, followed by a recovery towards the equilibrium temperature once the heat pump had stopped.

5. The simulation setup

In order to simulate the experimental observations, a model has been built using the Trnsys simulation environment. This allows the construction of a "system" simulator by interconnecting a set of "modules" that replicate the operation of the different components of the system. Version 16 of Trnsys has been used. The main modules that have been used here are the Type 557a representation of a closed loop ground model, and the Type 668 - a model of a water-to-water heat pump. Figure 4 shows the basic model schematic that has been used.

Figure 4: Schematic of the Trnsys model used to simulate the GSHP system.



The ground loop component (Type 557a) needs to be set up with the appropriate geometrical configuration and relevant ground thermal properties - some of which were derived from the thermal testing work reported in the accompanying Part 1 report (Curtis2012).

The heat pump model (Type 668) needs a set of performance measurements related to the heat pump that is being simulated. These establish the performance envelope of the heat pump over a range of ground (source side) temperatures and a range of load side temperatures. Care needs to be taken to provide temperature ranges that exceed any temperatures (both high and low, on source and load sides) that will arise in the simulation, due to the nature of the interpolation method. The associated thermal outputs and COPs of the particular heat pump are interpolated over this temperature field, as the inlet and outlet temperatures vary during the operation of the heat pump. The data used to build this heat pump model were provided by the manufacturer (Kensa) and are based on calorimeter measurements of the performance of this type and model of heat pump (ie the H062-S1H). They are broadly in line with the published data for the heat pump, but do not reflect the performance "allowance" that the EN reporting standard permits.

In building the Trnsys model, the various physical components of the system have been kept as close to reality as possible. Some adjustment of the Type 713, (the heat emitter component) was used to bring the observed temperature behaviour into line with the experimental measurements. Figures 5 and 6 show the modeled behaviour of the "continuous", and "intermittent" operation as measured in Figures 2 and 3. Close agreement is observed.

Figure 5: Ground and load side temperatures from Trnsys for a " continuous run"

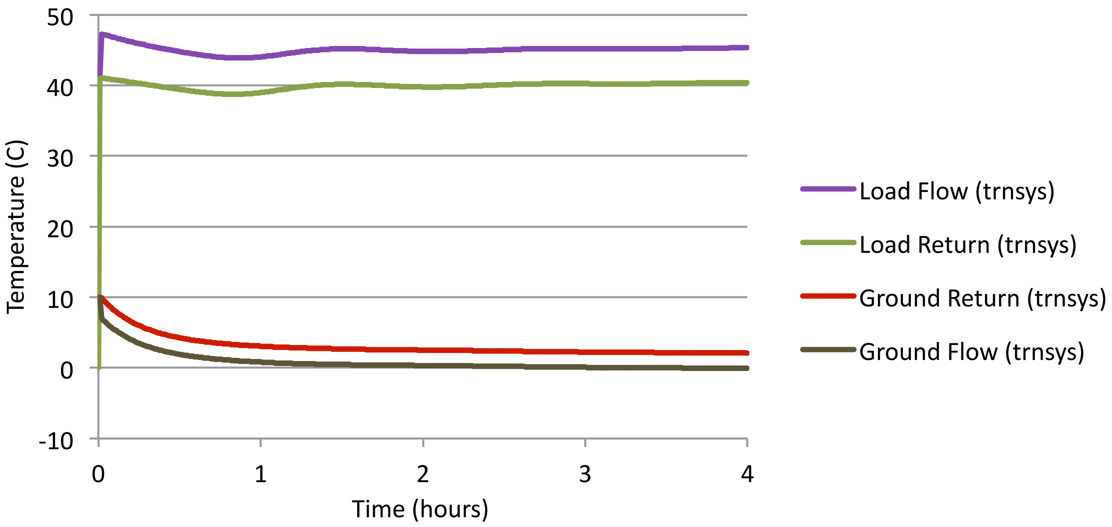
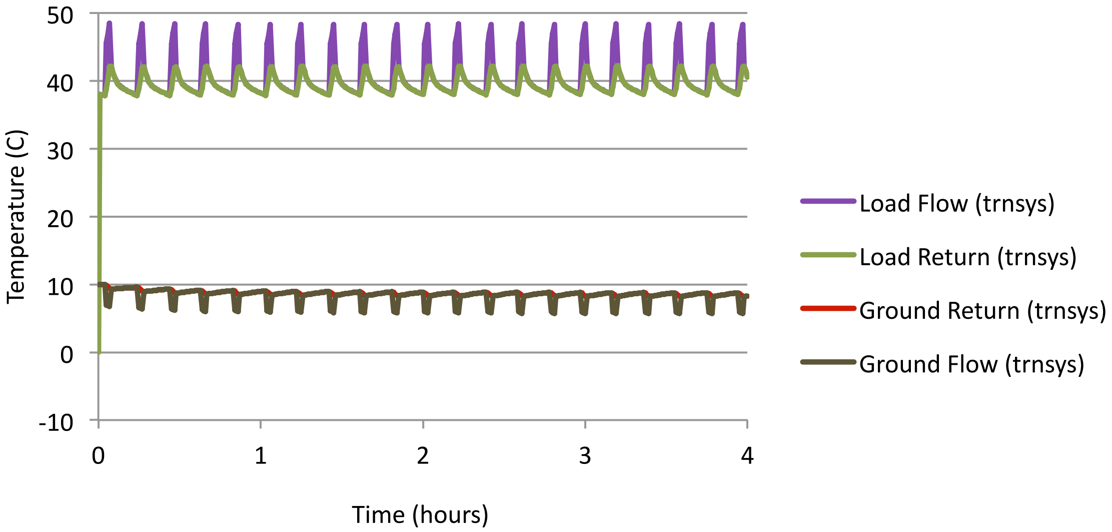


Figure 6: Ground and load side temperatures from Trnsys for a " short cycle" run



Figures 7 and 8 show the overlaid modelled and measured temperatures for the "continuous" and "short" cycle cases respectively.

Figure 7: Comparison of measured and modelled temperatures - Continuous run

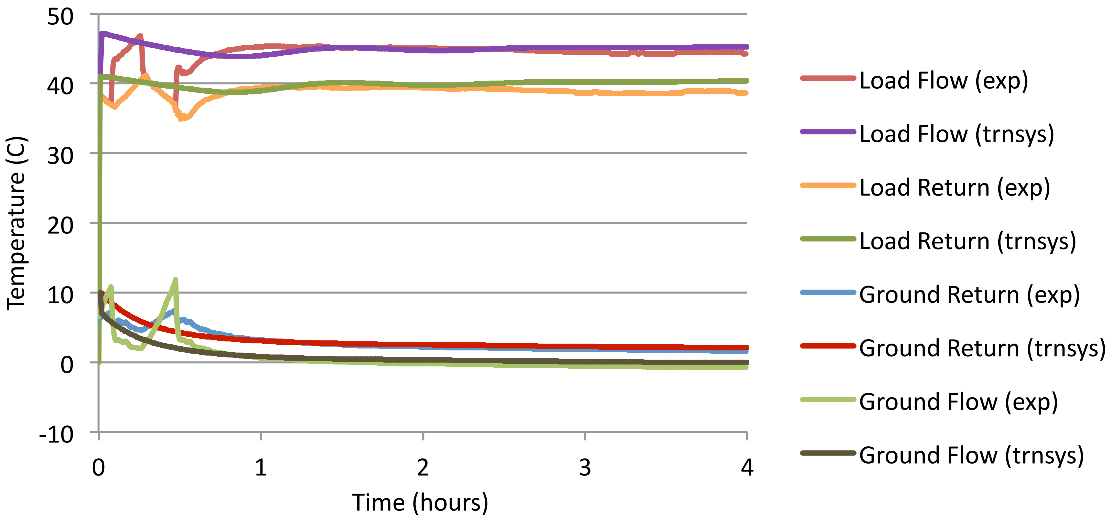
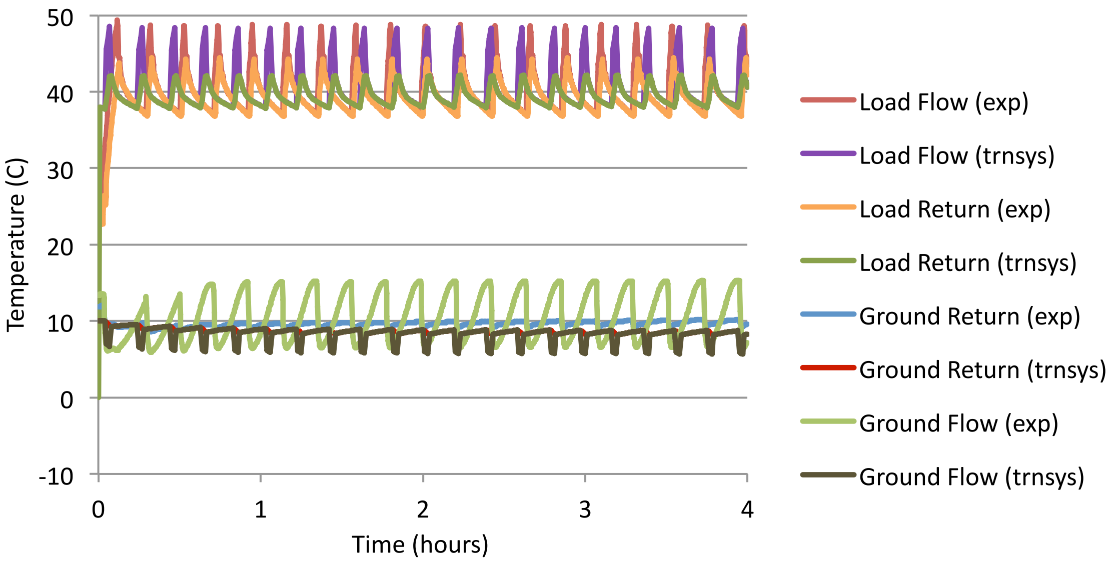


Figure 8: Comparison of measured and modelled temperatures - Short cycle run



Using EAT's experimental measurement of the other cycling "periods", ie for 2,3,4,5 and 7 radiators operating, it was possible to fine tune the Trnsys model to replicate each case in a consistent fashion across the entire "cycling" range.

6. Cycle efficiencies, COPs and SPFs.

As well as replicating the temperature behaviour, it was desirable to match the observed COPs (Coefficient of Performance) or SPFs (System Performance Factor).

Trnsys allows the recording of all energy consumptions associated with operation of the heat pump and associated ancillary equipment. This allows computation of both COPs and overall SPFs over any timescale.

For the purposes of this report, we clarify the use of the terms COP and SPF, as used here. For consistency we have adopted SPFs that conform with definitions proposed by the European SEPEMO project. (www.sepemo.eu). For engineering purposes (rather than pure thermodynamics) COP is usually defined as the ratio of heat output from a heat pump, to the electrical input.

This can be an instantaneous value at any point in time, and for a given heat pump, will vary depending on the source and load side temperatures.

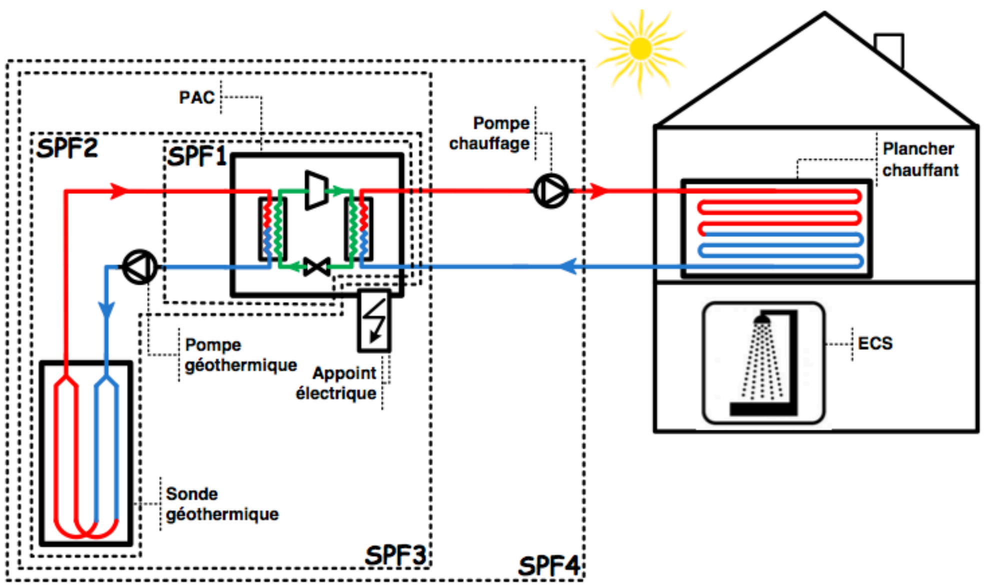
There can be confusion over the use of the term SPF. It is often used to mean Seasonal Performance Factor, ie the average COP over a heating (or cooling) season. However, it can also be used as "System" Performance Factor, to reflect the overall performance of a heat pump system where components other than the basic heat pump itself are included in the evaluation of efficiency.

Figure 9 is taken from SEPEMO, and illustrates the various levels of SPF. An instantaneous value of SPF1 corresponds to the COP of the pure heat pump on its own.

SPF2 includes for the electricity consumed by the ground loop circulation pump. SPF3 includes for any direct electrical boost energy. SPF4 includes for load side circulation pump(s).

In this report, values of SPF1, SPF2, and SPF4 will be encountered. There is no occurrence of SPF3. (ie no supplementary electric heating was used, SPF3 = SPF2). In all cases the SPFs refer to the average value of "heat output / electrical input" over specified periods of time - unless otherwise indicated.

Figure 9: SPF definitions as per SEPEMO (www.sepemo.eu)



Performances measurement (example of a simple geothermal heat pump)

Definition from the SEPEMO project

The efficiency values shown in the EAT report include the electricity used by the source (ground) side circulation pump and the load side circulation pump. The term "experimental SPF" is used, which equates to SPF4, as indicated in this report.

In order to make meaningful comparisons between the measured and simulated results, the impact of the source side and load side circulating pumps on the measured efficiency needs to be taken into account. By subtracting the measured electrical consumption of these pumps, it is possible to compute the average COP of the heat pump alone (SPF1). Figure 10 highlights the difference between the "experimental spf" (SPF4) and the experimentally derived, average COP of the heat pump (SPF1), at different cycle times.

Figure 10: Experimental SPF with increasing radiator 'on' time

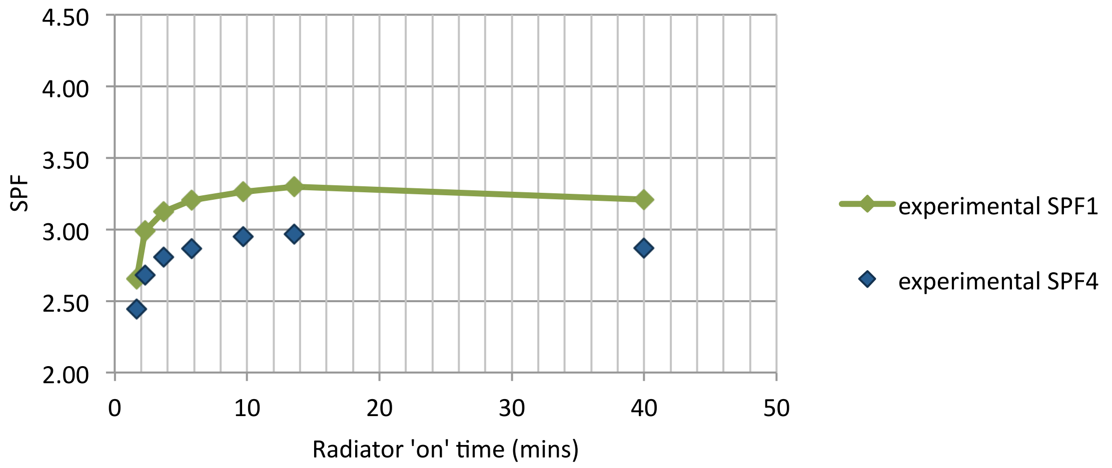
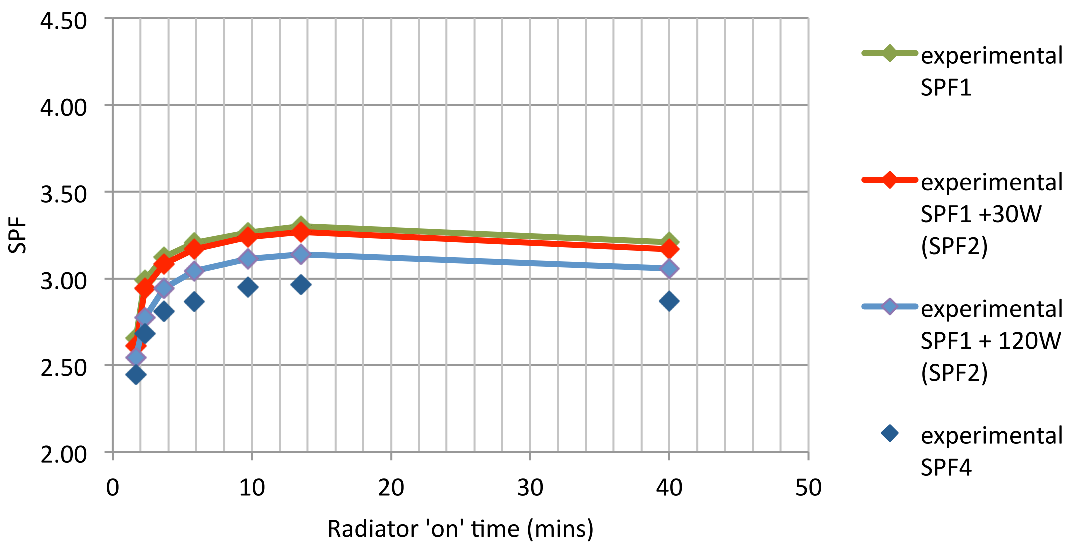


Figure 11 breaks down the impact of the circulating pumps on the SPF in more detail. The top and bottom curves are as for Figure 7, ie the experimentally measured SPF4, that includes all electrical energy used by the pumps, and the “experimental COP”, which is the efficiency of the heat pump alone (SPF1). The "experimental COP + 30W" shows the effect of adding a 30W pump - which typically could be the power rating of a high efficiency ground side circulating pump on a small system. Also shown is the "experimental COP + 120W", curve which could typically be the power of an older ground loop or load side pump. These curves demonstrate the impact that circulation pump power has on the overall SPF of GSHP systems. One of the outcomes of the revision of the MCS installation standard, ie MIS3005, has been to encourage the correct sizing and calculation of ground loop hydraulics so as to minimise the effect of this parasitic energy.

Figure 11: Significance of circ pumps on SPF



Note that it is typical practice to include the ground loop energy in the computation of a GSHP's overall efficiency (ie SPF2). The load side pump is generally considered to be required in any wet heating system and does not usually figure in the quoted efficiency of a GSHP.

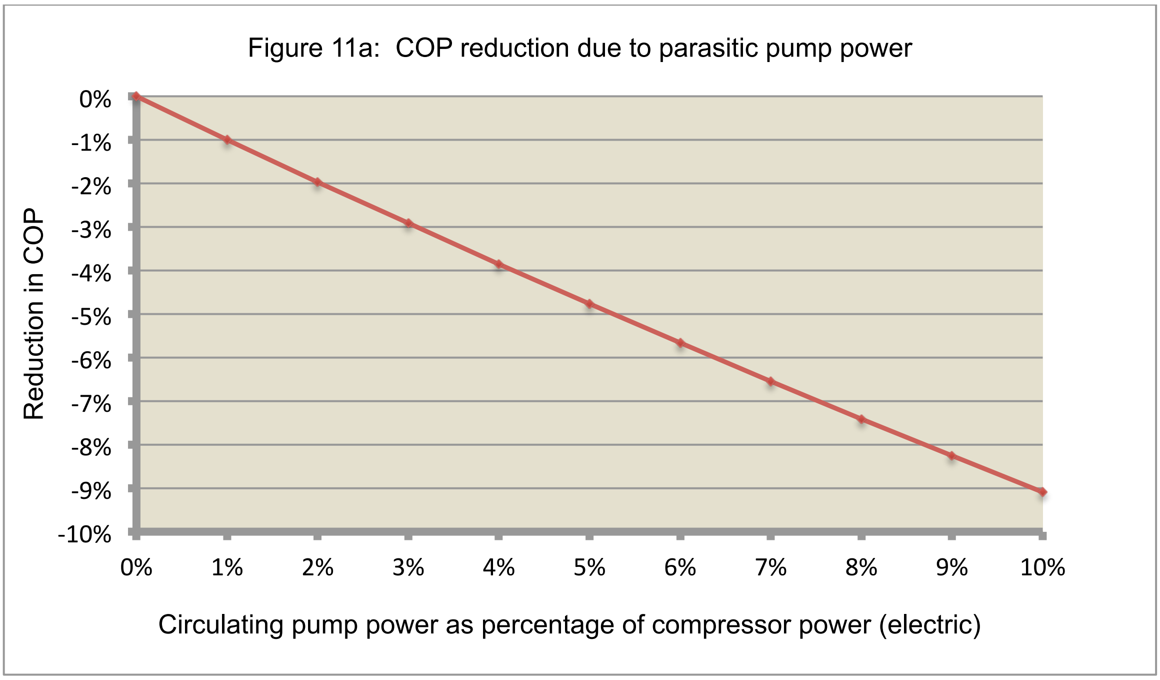
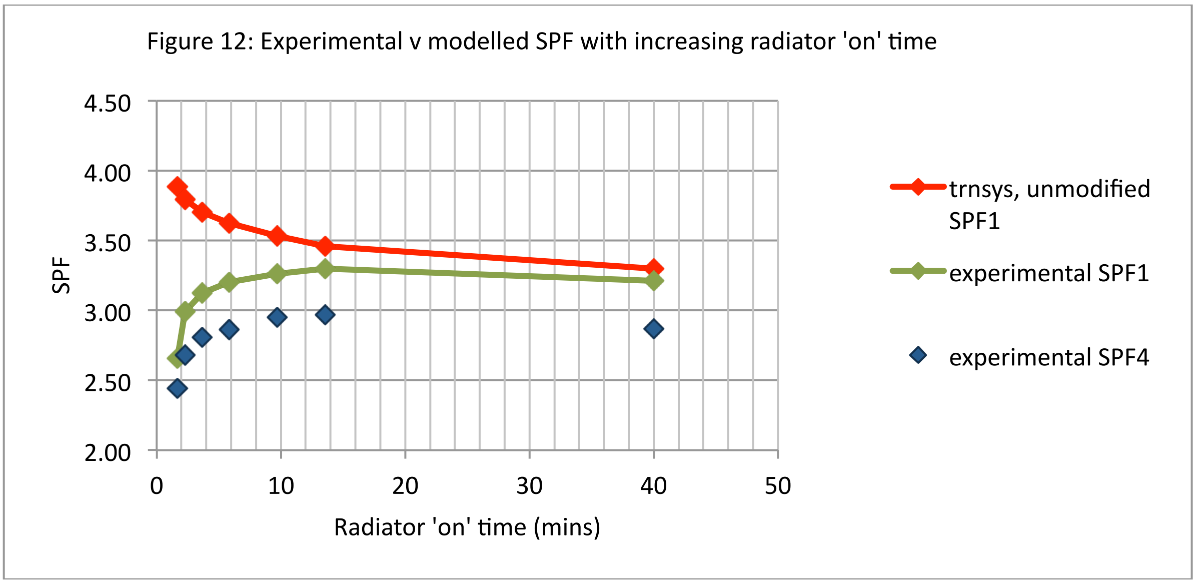


Figure 11a summarises the generic reduction in the COP (SPF1) of a heat pump by the parasitic pump power of a circulating pump. If this is the ground loop pump, then the reduced value represents SPF2. Note that this graph uses the electric power of the circulating pump as a percentage of the electrical power of the heat pump compressor.

Having derived the “experimental COP” (SPF1) as in Figures 10 and 11, Figure 12 shows the comparison with the SPFs obtained from the Trnsys simulations. It is immediately apparent that at short cycle times, the measured efficiency of the heat pump (SPF1) as predicted by the theoretical, simulated model of the heat pump is significantly different to the measured values of SPF1. The high values of the predicted SPFs could be expected because the ground temperature at switch-on will be high, and the load side temperatures will be low. In reality, the expected benefit of these temperatures is not being observed.



It would appear that there are inefficiencies occurring in the heat pump during "start-up". These are thought to divide into two time scales, i) a short period of time where the compressor starts to move refrigerant and the refrigerant cycle gets established, and ii) a longer, overall warming / cooling of the high mass components of the heat pump, notably the compressor itself and the two refrigerant to water heat exchangers.

In order to replicate this behaviour it was necessary to modify the Trnsys Type 668 model of the heat pump. This standard model is built using manufacturer's performance data at a series of load side and source side temperatures. This data will have been generated for the heat pump operating under steady state conditions at different source and load side conditions. In the experiments undertaken here, as in real GSHP applications, the heat pump is in a highly dynamic situation. By altering the Trnsys heat pump component to reduce the performance of the heat pump during "start-up" it was hoped to replicate the observed experimental behaviour.

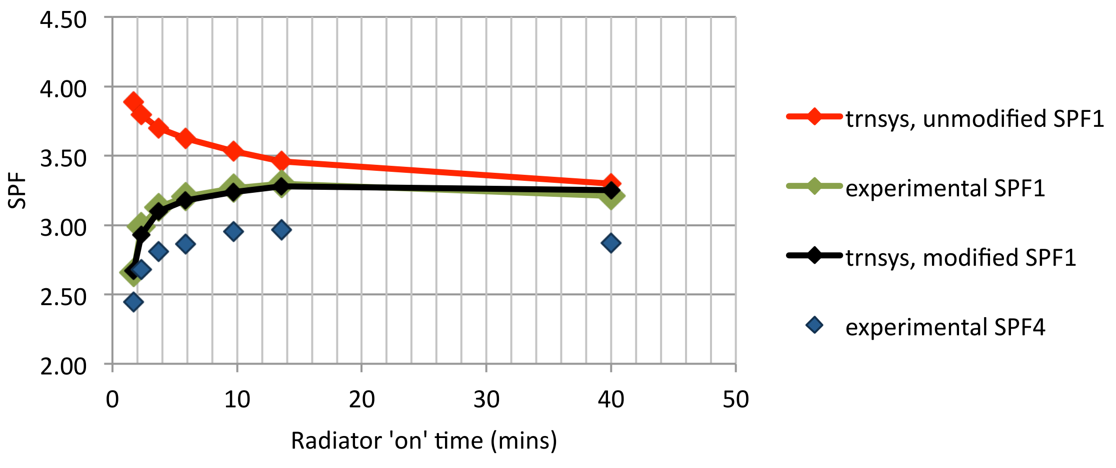
The heat pump model was re-coded such that whenever the compressor starts, its thermal output has a value modified by a (quadratic) polynomial as follows:

Reduced Thermal Output = "Normal" Thermal Output x ($at^2 + bt + c$)

where a, b and c are constants to be determined, and t is the elapsed time that the compressor has run in a cycle. After the "start-up" period has elapsed, the "Reduced Thermal Output" becomes equal to the "Normal Thermal Output" and the Type 668 component reverts to its original method of simulating the heat pump. Note that at all time steps during the simulation, the "Normal" thermal output is computed based on the load and source side temperatures present at that particular time step. By running the simulations through a number of iterations, it was possible to obtain a set of constants (a, b, c,) and a suitable start-up period, that provided a good match to the experimental COP at the different cycling times. (Figure 13)

It should be appreciated that each point on the curves in Figure 13 represents the average COP (SPF1 and SPF4) over several cycles of the specified duration, over a four hour period of operation.

Figure 13: Experimental v modelled SPF with increasing radiator 'on' time

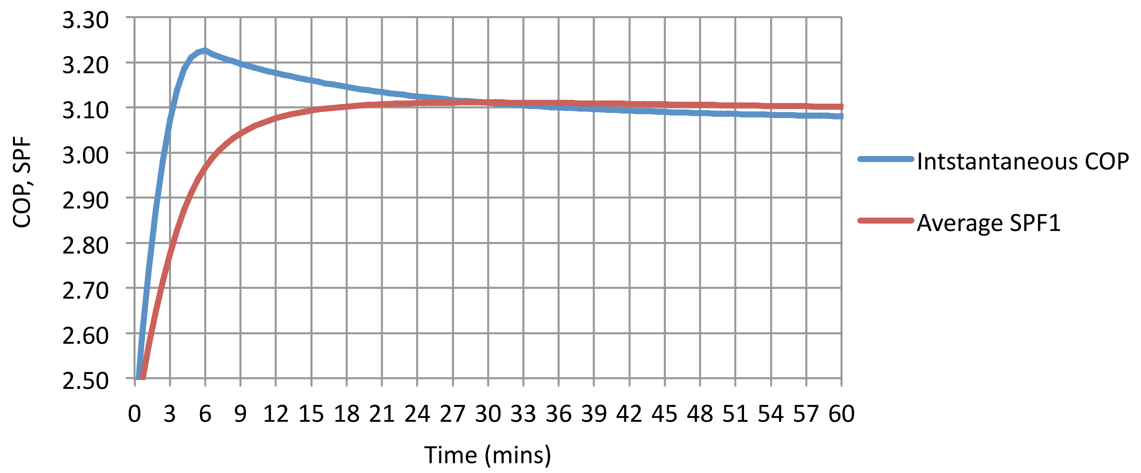


It was found that the degradation in performance needed to take place over a six minute (0.1hr) period in order to match the observed results. It was also found necessary to run the simulation at steps as short as 0.01hr (36 seconds) in order to capture the effect in enough detail. If the simulation is run with significantly larger time steps, it is not possible to see the effect of the performance change as the heat pump starts up. This becomes a significant computation penalty for multi-year simulations of GSHP operation, but none the less replicates the measured performance.

Note that the constants derived here, and the period over which the performance degradation appears to occur, can only be applied to this particular heat pump. Heat pumps with different refrigerant types, different quantities of refrigerant, and different heat exchanger types and sizes may well have different characteristics.

The outcome of this part of the modeling and simulation work has provided a plausible reason for the unexpected early time performance of a water-to-water heat pump during initial start up of the compressor. For this heat pump, it would appear that, for up to the first six minutes, there will be a reduction in instantaneous COP compared to the COP that would be expected under steady state temperature conditions. The shorter the run time, the worse the overall degradation in the average COP for that cycle period will be. As the compressor run time extends beyond six minutes, the effect of poor performance in the early minutes will have a decreasing effect on the average COP of the overall "run". This behaviour is clearly illustrated in Figure 14 where data from a "continuous" (ie a long) run is used to compare the instantaneous COP at any point in the run, with the average COP obtained up to the same point, from the start of the run.

Figure 14: Instantaneous COP and average SPF1 during a long "on" run



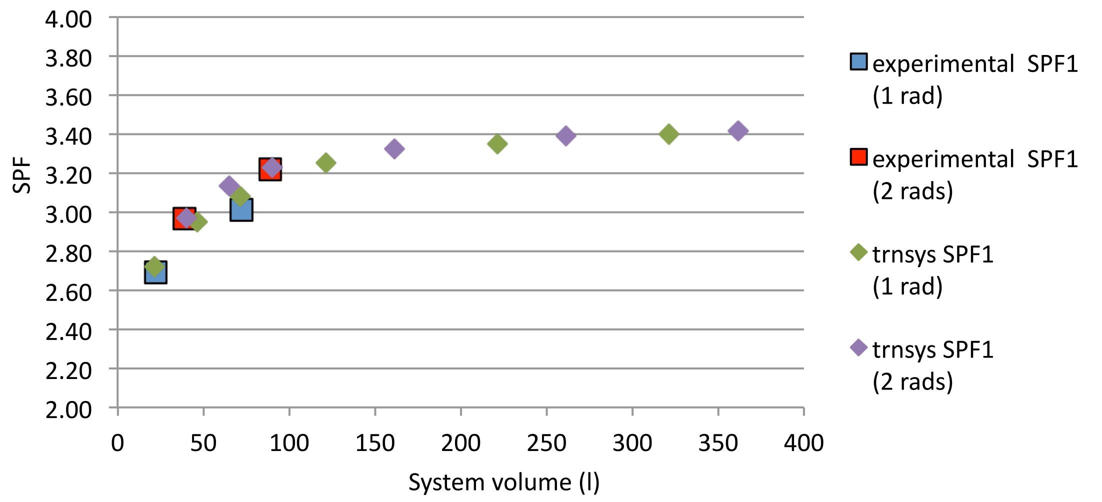
In Figures 10 through 13 it is worth noting that the COP (or SPF) of the final, "long" cycle time appears to be slightly lower than the last of the "short" cycle times. This is very probably a characteristic of a closed loop GSHP, where the ground loop temperature will slowly fall with increasing run time, and hence result in a lower heat pump efficiency. For this reason, GSHPs should be sized such that they do not end up with excessively long, or continuous cycle times in periods of very cold weather. As soon as the heat pump turns off, the borehole and surrounding ground immediately starts to return towards the ground equilibrium temperature. Figure 13 indicates that there is a maximum average COP for this particular GSHP system somewhere around the 30 minute run time.

7. Buffer tanks

In the experimental investigation, the next step was to look at the effect of introducing a buffer tank between the heat pump and the load side (ie the radiators). The set up is described in section 4 of Green 2012. The Trnsys model, with the modified heat pump, was altered to include a representation of a buffer tank.

In Figure 15, there are four experimental points. Short cycle cases with run times of 1.6 minutes (1 radiator) and 2.4 minutes (2 radiators) are shown. These are experimental SPF4 values that have been corrected to SPF1 values by removing the electrical consumption of both circulating pumps. It is seen that the introduction of the 50 litre buffer tank changes the SPFs from ~2.65 to ~3.0 in the case of the single radiator case, and from ~3.0 to ~3.2 in the two radiator case. These are significant improvements in SPF. Note that the x-axis values are "total system volume", ie they include the radiator and pipework volumes as well as the buffer tank volumes.

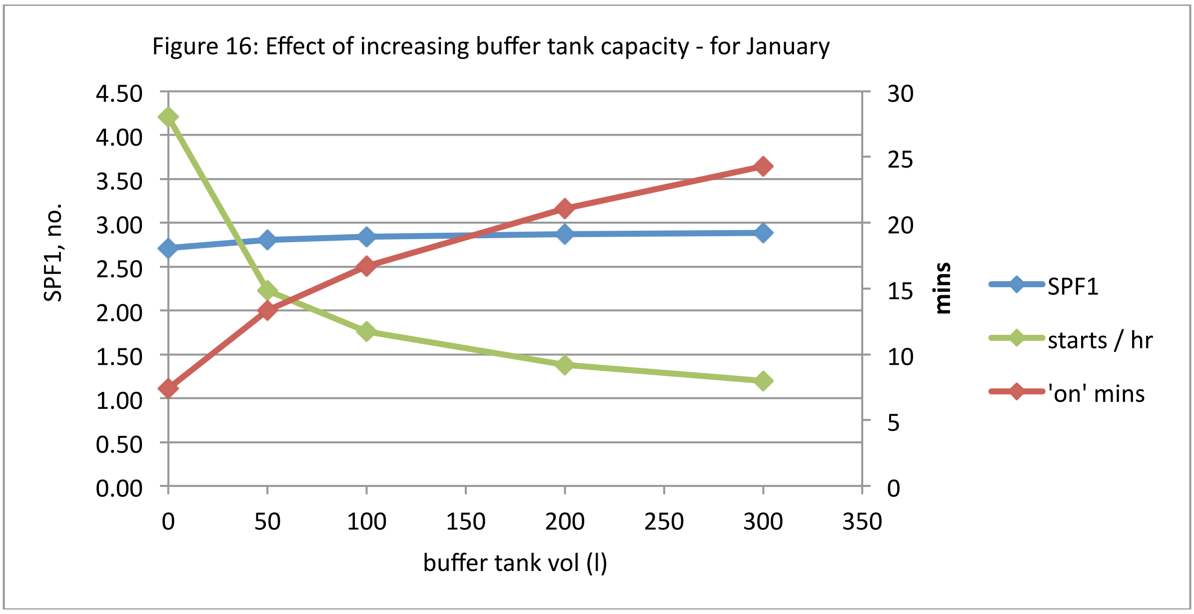
Figure 15: Effect of system volume on SPF

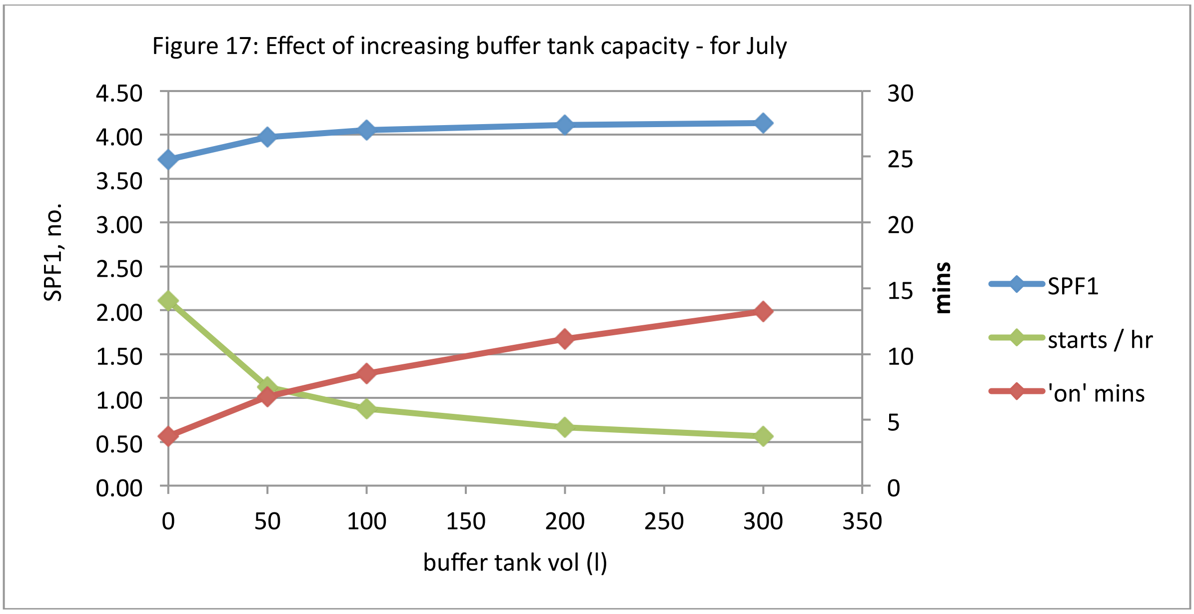


The Trnsys models have been set up to replicate the experimental results of the 1-radiator and 2-radiator systems without a buffer tank. Increasing buffer tank volumes have then been added to the basic system volumes to observe the predicted impact on SPF1 values. These show good agreement at the two 50 litre buffer tank experimental points. There is some indication that the 2-radiator system exhibits slightly higher SPFs than the single radiator system. This may be due to the fact that the experiments were carried out at different times, with potentially differing ground and ambient temperatures. In general though it can be seen that there is significant advantage to be gained by adding a buffer tank to provide a total system volume somewhere in the region of 150 to 200 litres for a heat pump of the size used here (nominally 6kW thermal). In the case studied here, the addition of the 50 litre tank to provide a total system volume in the region 60 to 100 litres delivers most of the potential gain in SPF that can be obtained. This arises from the fact that it will take ~0.11 hours to warm a 60 litre volume through 10°C using 6kW of input. For only a 5°C rise, the volume would need to double to maintain the run time.

It is observed that higher SPFs are predicted here by the Trnsys modelling than for the continuous operation case shown in Figure 14. This is due to the fact that in the latter case, the ground temperature will be falling with long run times, and will have started to pull the SPF value down. In the short run cases (1 and 2-radiators) of Figure 15, this effect is not seen.

In Figures 16 and 17, the effect of changing buffer tank volume on three variables is shown, for two months of the year, ie January and July. The variables are the average COP for the month, the average number of compressor starts per hour and the average compressor run time. For the January case there is only a small change in the average COP for this particular model because the run time at this time of year is already above six minutes on average, and therefore out of the zone where there is performance degradation. The significant value is the change in the number of starts per hour, from somewhere over 4 without a buffer tank, to almost 1 start/hr with a 300 litre buffer tank.





For the case of July, Figure 17, the effect on the COP is more significant, however the overall impact on the annual energy consumption is low, because the total amount of energy required in July is significantly less than January. However, once again, the change in the starts per hour as the runtime increases, is significant. The short cycling is occurring because the heat demand of the house is low in July. The buffer tank eliminates this. The higher SPFs in July compared to January arise from a combined effect of higher average ground temperatures and lower average flow temperatures being required to meet the heating demand.

From Figures 16 and 17, which are for a system run over a period of a month, it could be construed that the value of additional buffering volume, compared to the impact of buffering as shown in Figure 15 for the 1 and 2 radiator configurations, is not very significant. ie the addition of increasing amounts of buffering volume in January is shown to lead to a small increase in SPF. In July the effect is more marked, but still significantly less than was observed in Figure 15.

To understand this, it should be appreciated that the Trnsys simulations used to generate Figures 16 and 17 represent a single zone, single emitter arrangement. The model is not constructed as a multi zone house - with individually controlled radiators.

It should also be noted that as well as system volume, the other factor that controls run times is the control hysteresis, whether on a room thermostat, radiator control valves, or the return water aquastat. Very small control hysteresis by itself can lead to very short run times.

Thus in January (Figure 16) the Trnsys simulation, with the given weather tape, and single zone operation, will generally have long enough run times to overcome the initial start-up penalty on the SPF. Longer run times arising from the additional buffering volume only give rise to slightly improved SPF, but note the significant decrease in the number of starts.

In July there is more significant impact on SPF, because the external weather is having an impact, causing shorter run times even with the single zone representation.

The difference in the impact of adequate buffering volume between the experimental behaviour of the 1 and 2 radiator systems, over 4 hour run periods, (Figure 15) and the behaviour of a whole house model over periods of a month, (Figures 16 and 17) demonstrate the clear need for the following:

Manufacturers / designers / installers need to be aware of the issues that can lead to short cycling of heat pump systems, and take measures to eliminate them. In practical terms this requires that, with a zoned system, there is always enough water volume open to the heat pump under all control conditions, as well as sufficient control hysteresis, to allow for a sufficiently long run time to overcome the start up penalty. Given that there will also be protection on the heat pump to avoid rapid restarts (for electrical network/lifetime issues) the heat pump needs to be able to run for long enough to achieve the required room temperature without tripping out on either high pressure (HP), or return water temperature set point and going in to "delay", which can result in a repetitive start / stop sequence where the heat pump fails to bring the zone to temperature.

There are three approaches that can be taken to achieve this.

i) The level of zoning and sizing of radiator volumes is arranged (at design time) to always present enough volume to the heat pump to offer sufficiently long run times - under all control conditions.

ii) Sufficient control hysteresis is allowed for to provide adequate heat pump run times, whilst maintaining the required level of controlled comfort.

iii) If i) and ii) cannot be achieved through appropriate zone selection, and/or radiator sizing, then additional volume needs to be added to the system through the incorporation of a buffer tank.

Table I shows approximate total system volumes per kW of heat pump thermal output that are required to achieve different run times, with different levels of return water or buffer tank hysteresis. This assumes that no heat loss occurs during the heat up period.

Table I - System water volumes to achieve minimum run times.

System volume/kW (litres)	Water temperature hysteresis (C)	Approximate run time (minutes)
15	10	10
30	5	10
60	5	20
30	10	20
60	10	40

Note that this table only applies to heat pumps with the same characteristics as the unit that has been used in this study. Heat pumps with significantly different compressor types, heat exchanger sizes and types, and refrigerant cycles may exhibit different start-up penalties which may require greater or lesser quantities of buffering.

The Figures in Table I are conservative, and will guarantee the stated heat pump run times. In practice, due to the time taken to heat the additional buffering volume, the emitters will have more time to raise the zone temperature(s) to the desired set point, and lower available system volumes will deliver the minimum runtime required.

8. Heat pump controls

Having established a system model that appears to replicate the experimental observations, the opportunity was taken to investigate the effects of different approaches to heat pump control in a domestic setting.

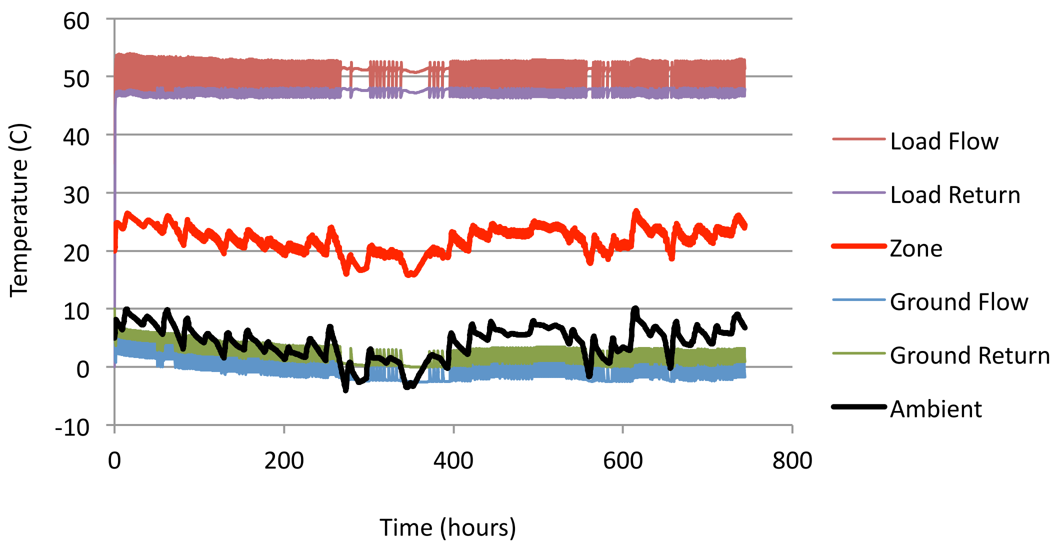
These cases have been run over a period of one month (744 hours) using a weather tape for January in North West England. To establish a reference point, the Trnsys model was set up to require a 45°C flow temperature when the ambient, external temperature was 7°C

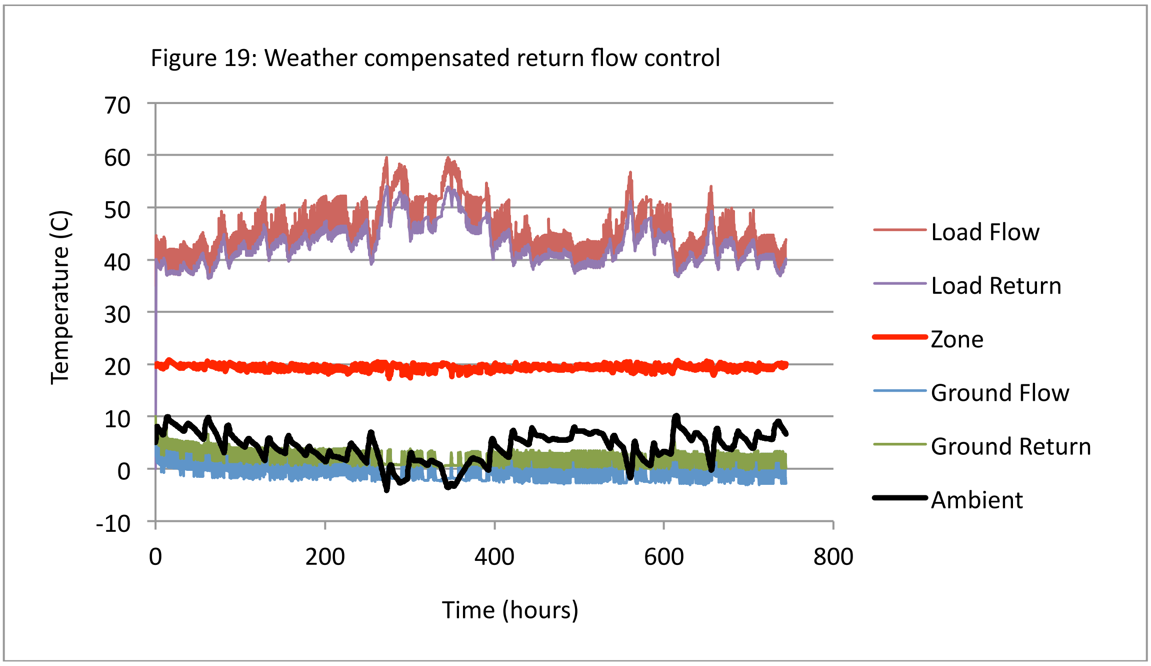
In Figure 18 the control is achieved by using an aquastat on the load side return flow to the heat pump (no room thermostat used). Hence the relatively constant heat pump flow and return temperatures. At this set point, there is very significant variation in the predicted room temperature, with over heating and under heating occurring.

In Figure 19 the return temperature method of control is modified by introducing weather compensation of the aquastat set-point on the return flow temperature. With judicious application of the weather compensation algorithm, significantly better control of the room temperature is achieved.

In the third case, Figure 20, the control is achieved through a room thermostat. This, by definition, results in very close control of the zone temperature. In a UK house using radiators, this method effectively introduces a form of "weather compensation", because the load temperature may only rise to a level where the desired zone temperature is achieved, and the heat pump turns off.

Figure 18: Return (load side) water temperature control





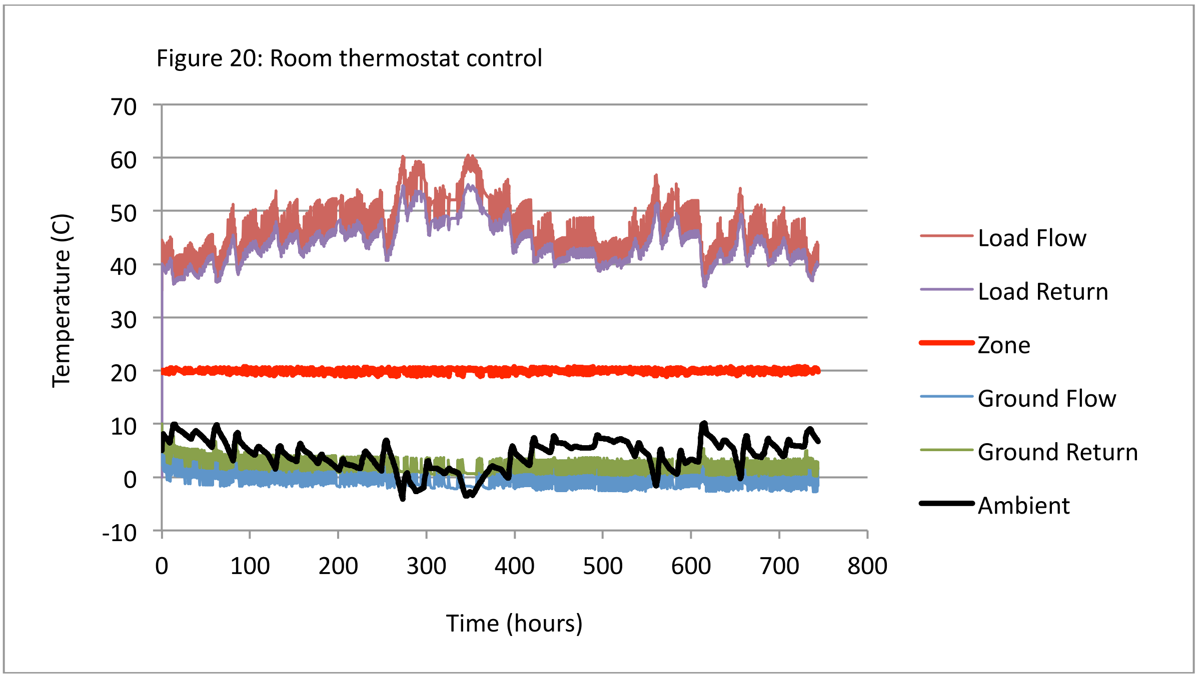


Figure 21 summarises the effects of these three control approaches. For the set point adopted in the return flow method, there is a significant amount of time when the room is over heated. With a lower set point on the return temperature, there would be less overheating, but the effect would be to increase the amount of time that the room was not achieving the desired temperature. The weather compensation of the return temperature is seen to lead to a significant improvement in control, and by definition the room thermostat control leads to the required room temperature at all times.

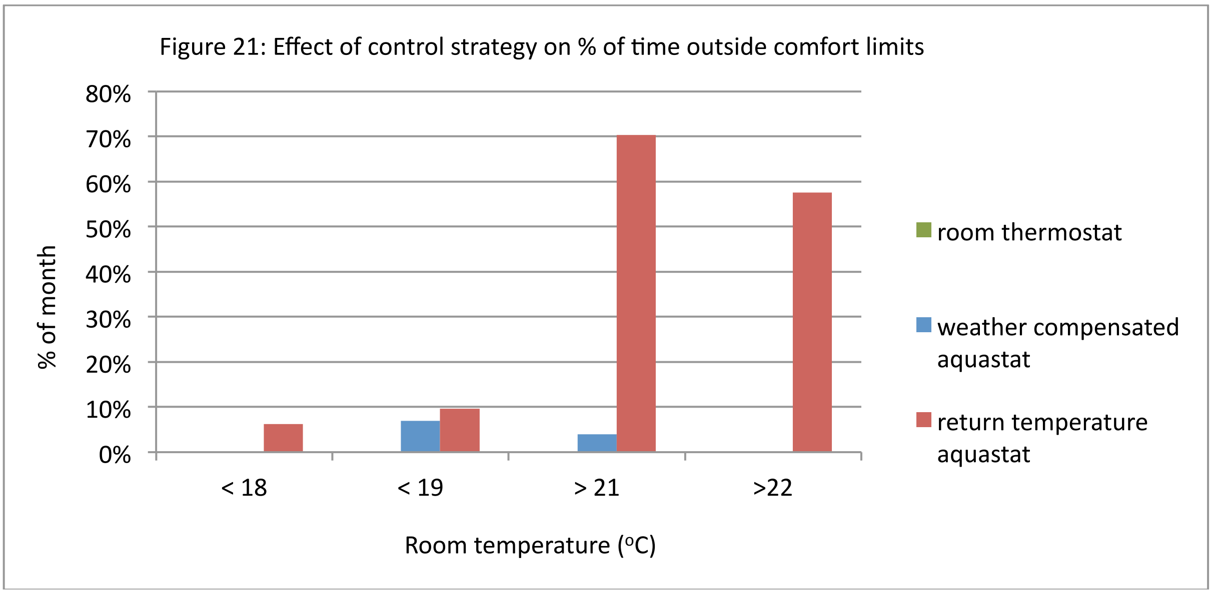
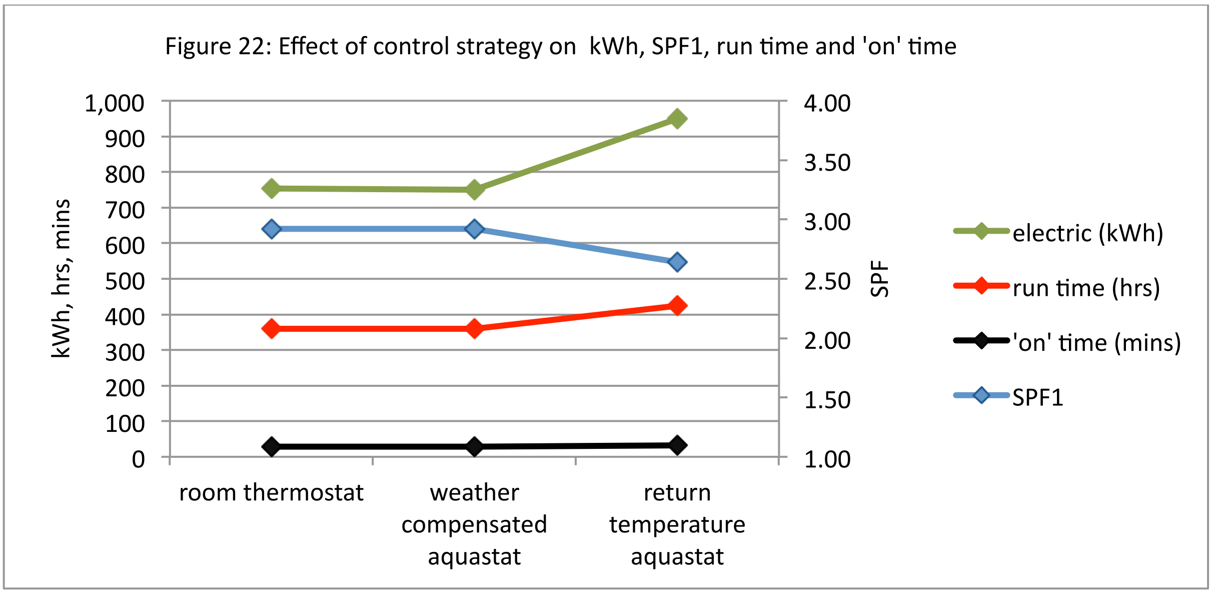


Figure 22 summarises four more factors for these three control strategies, viz COP, kWh of electricity consumed, average run times and the total time that the compressor is "on". The worst case is seen to be the non-weather compensated return flow temperature method of control, which scores the worst on three of the four measures, with minimal changes seen in the 'on' time.



9. Effect of night setback.

In the EST monitoring study (EST2010), it was often reported by end users that they had been told to run the system "continuously". Using the model established here, the final exercise has been to investigate the effect of implementing a night setback control philosophy. Figure 23 shows the Trnsys model that was used to carry out this simulation.

Figure 23: Trnsys model - underfloor heating with night setback

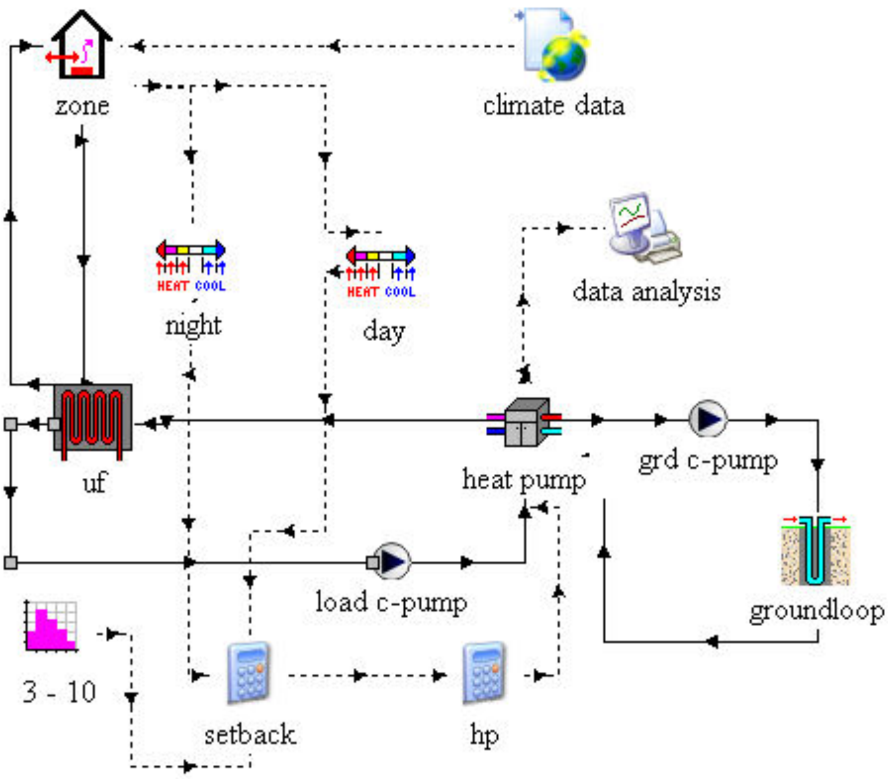
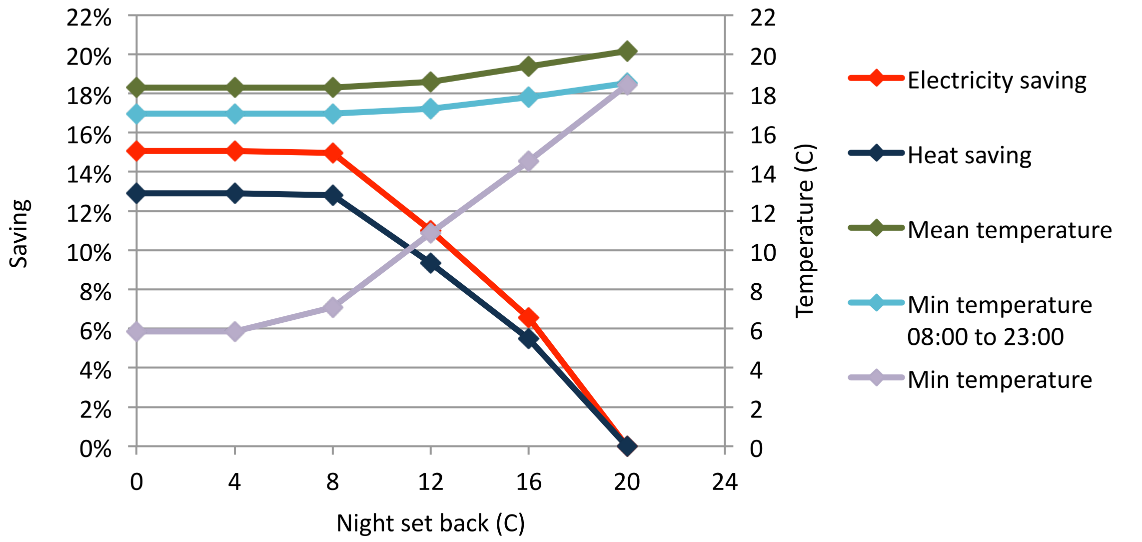


Figure 24 summarises the effect of changes in the night set back temperature. The model has been run using a weather tape for January only. The system has been set up to require a daytime temperature of 20°C, and is then allowed to float down to the set back temperature between the hours 22:00 to 03:00. The emitter system has been defined to replicate underfloor heating, but could easily be modified to investigate the effect using radiators. The figure clearly demonstrates the reduction in heating energy requirements, and hence electrical energy required to drive the heat pump, whilst maintaining the required level of comfort in the house during the daytime. In this case it appears that a night set back of 16°C would still maintain adequate comfort levels whilst saving >6% in running costs. Arguably a night setback of 12°C would be acceptable, with a saving >10%. (Note that no account has been taken of any internal gains in this exercise).

Figure 24: Effect of changing night-time (22:00 - 03:00) setback temperature on comfort levels and energy savings in a single zone , underfloor heated building (uninhabited, January)



10. Other heat pump cycling issues

The previous sections have presented the reasons why short cycling of ground source heat pumps should be avoided arising purely from the thermal and thermodynamic behaviour of the heat pump and associated control philosophies.

Short cycling of heat pump compressors also causes effects on the electrical supply systems, as well as having mechanical and lifetime impact on heat pump components - principally the compressor motor, the compressor itself, and associated electrical contactors.

A heat pump compressor motor will, in most domestic dwellings in the UK, be the largest electrical motor in the property. It is noticeable that the largest electrical consumer items, ie electric lawnmowers and some vacuum cleaners typically do not exceed 2.2kWe. The vast majority of UK dwellings only have a single phase 230VAC supply with a 60 or 100 amp main fuse. The largest heat pumps available for single phase use are nominally ~16kW thermal output and therefore have a 4 - 5 kW electric motor. On start up, these compressors pull significantly higher currents, typically 5 - 8 times the running current for a fraction of a second. It is common practice to fit "soft" starters that will pull this current down to 3-4 times the running current. However, this short spike can often be detected in the form of light flicker and/or interference to some electrical equipment. The effect can also be transmitted to adjacent properties operating on the same electrical phase.

These network issues are thought to have higher impacts in weaker parts of the distribution network. This can be the case in rural, typically overhead line, portions of the system. The tolerable magnitude and frequency of voltage fluctuations for different strengths of network is calculable with reference to detailed guidance provided in an Energy Networks Association engineering document (ENA).

There appears to be a consensus between manufacturers and DNO's that the maximum number of starts per hour should be kept to between 6 and 10, ie the compressor should not start more frequently than on 10 or 6 minute intervals. To achieve this, most heat pumps are fitted with delay timers. Note that these only prevent the compressor from restarting in these intervals, they do not ensure that the compressor will run for any period of time.

An additional reason for fitting delay timers is to allow the refrigeration pressures in the heat pump to re-equalise after a prior run. If the compressor were to re-start in a hot, high-pressure condition, it may pull even more starting current and/or exceed the contactor trip currents, and the compressor will fail to start. It will also put higher mechanical shock loads on the compressor and motor.

Another issue related to compressor start up is the time taken to mobilise lubricant to the compressor bearings. During "off" periods the oil in the refrigerant will make its way to low points in the refrigerant circuit. It takes a short, but finite time to redistribute this lubricant to the compressor bearings on start up. It is therefore advantageous to keep the number of starts to a minimum in order to ensure that the compressor bearings are fully lubricated for as large a portion of the compressor's operating lifetime as possible.

Because of the electrical and mechanical stress placed on the compressor and compressor motor, and associated electrical switching components (contactors), the lifetimes are often determined by the total number of operations, or starts. It is therefore desirable to try to minimise the number of starts per day / week / year - in order to extend the operating life of these major system items.

Whilst it has proved difficult to unearth detailed information regarding the total of number of starts that modern compressors are built to withstand, there would appear to be adopted practice in the heat pump and chiller industry that probably owes its existence to consideration of these items. A number of relevant items and extracts are included in Appendix A.

In the chiller industry it is not uncommon to come across sizing rules-of-thumb of 10 - 20 litres per kilowatt of cooling as the total available buffering volume. In the European heat pump heating industry, figures range upwards from 15 to 40 litres per kilowatt (thermal) of total buffering capacity - to avoid short cycling and provide sufficiently long run times. Given the measurements and modelling carried out here, it can be seen that run times in excess of at least 6-8 minutes are desirable from an efficiency point of view. If the runs times are longer than this, the effect of the initial inefficiency is gradually smeared out. The added complication of a ground source pump is that, it is beneficial after a period of time to stop the heat pump to allow the ground loop temperature to recover. It is worth noting that some microprocessor based heat pump controllers, as well as preventing short cycling in terms of rapid restarts, also try to achieve minimum run times providing that all other safety/pressure related parameters remain within limits.

11. Conclusions.

It has been suggested that short cycling of heat pump compressors is not a "good thing". Some of this may be historical and applied to compressor sizes and technologies that may not be relevant to domestic heating heat pumps in the UK. This study has looked at a number of issues that are associated with short cycling in domestic heat pump systems and provides a basis for suggesting minimum run times for these types of heat pumps.

The study has identified from experimental measurement that the SPF of a GSHP during a "start-up" phase does not match the SPF that would be predicted from calibration measurements derived from steady state operating temperatures. By using a simulation model, and modifying the start up behaviour of the heat pump component, it has been possible to match the measured behaviour of the heat pump at different cycling periods.

It has been possible to alter the simulation component to reflect the observed inefficiency of a ground source heat pump during start up. The correction applied here has not been developed on any physical basis - ie /eg there is no assessment of the thermodynamic cycle during startup or of the time taken to warm and cool the heat exchangers. It is not clear how this behaviour scales with the size of heat pump and or heat exchangers / compressor, or with the type of refrigerant. It is suggested that a combination of effects is occurring, ie a short term inefficiency as the refrigeration cycle is established around the refrigeration circuit, and a longer term effect as the major heat pump components are brought to temperature. Experimental work with other types and sizes of heat pumps could be undertaken to attempt to quantify and classify the effects. However, it is likely that the effect will always be present, and is a significant pointer to requiring that heat pumps do not suffer from very short cycle times. This effect may equally apply to air source heat pumps. There is an indication from other work (Ulhmann&Birch2010&2011) that varying sizes of heat exchangers, and differences in the operation of expansion valves are contributory factors to the startup inefficiency.

In this study it has been shown that a match to observed performance is obtained if a varying degradation in the efficiency is introduced over a six minute period of compressor operation. If the compressor continues to run for a longer period of time, the effect of this initial degradation is gradually lost. This would point to minimum run times for this heat pump of the order of 10 to 15 minutes being desirable, with optimal times being of the order of 30 minutes, with the ground loop at this location.

The incorporation of sufficient overall buffering in a GSHP heat emitter circuit is required to prevent short cycling - particularly as the heating loads on the circuit decrease (ie in the shoulder seasons and summer moths). If it is not possible to ensure that the required buffering is present in the heat distribution and emitter components, at all times of operation, then a buffer tank should be incorporated to offer enough system volume to prevent short cycling. There is a cost and space implication to this. However, the increase in heat pump efficiency, the avoidance of other short cycling impacts and the reduction in the lifetime total number of starts are all benefits worth gaining.

A brief investigation has been made into the impact of controls. It has been shown that the closest form of room temperature control is, not surprisingly, in-zone thermostatic control of the zone air temperature. The poorest form of control appears to be simple non weather compensated heat pump return temperature control. The introduction of weather compensation to the return water temperature is of significant benefit, but this requires careful set up of the compensation algorithm to the particular property, to ensure that it is effective. Zone thermostat control effectively introduces weather compensation by only allowing the flow temperature in a heat pump system to rise to the point at which the zone air temperatures are achieved, and the heat pump is turned off.

A brief investigation of night set back control has also been investigated using the simulation model. It is shown that night set back is a perfectly valid form of control, and that it is not necessary to require that a ground source heat pump should be either "enabled" or required to run, "all the time" - as was often reported in the EST monitoring study. There will be different requirements for the timing, and setting of the set back temperature depending on the insulation and thermal mass of the dwelling, and the type of emitter system.

It would appear that for single phase, single compressor GSHPs, that the number of starts per hour should be restricted to something in the range of six to ten - as a maximum from an electricity distribution perspective. This is to minimise observed electrical interference in the property and adjacent properties that are on the same phase. Because the widespread use of large scale electric motors on the UK distribution network has not been encountered to date, it may be thought worthwhile to investigate the maximum allowable start-up currents that should be allowed for these systems, and any other DNO related electrical requirements that heat pumps should meet before they are rolled out in large numbers across the networks. It should be noted that the widespread, high density adoption of electrically driven heat pumps in a country with a single phase domestic electrical supply is not replicated elsewhere. All of the other EU countries with high prevalence of heating heat pumps have three phase domestic supplies. France is probably the only other EU country with a single phase supply that is beginning to see a significant increase in heat pump take up. Otherwise it may be necessary to look at experiences that other single phase players have had with air conditioning units.

It should be appreciated that several of the issues related to short cycling impacts are overcome with variable speed, inverter driven compressors. Whilst there may still be thermal inefficiencies at startup, there are no impacts from high start up currents, and the motor and compressor do not suffer the same electrical and mechanical shocks on starting. However, variable speed, single phase GSHP systems are still extremely rare. Manufacturers have not made the same attempt to bring these to market as they have with ASHPs, where they have been established in the cooling sector for many years. The possible downside to variable speed compressors is the expense and complexity of the control electronics. None the less, the ability to eliminate the expense, and space requirements of a buffer tank, a possible reduction in ground loop size, and the elimination of high start up currents may all be reasons why a variable speed system may be more attractive in a single phase environment than manufacturers currently recognise.

An interesting outcome of this study arises from the combination of the requirement to prevent electrical / compressor short cycling, and to establish long enough run times to overcome initial startup inefficiencies. Given that the compressor and contactor manufacturers, and the DNO's appear to want to limit the maximum number of starts per hour to between six and ten, and given the desire for a run time of not less than about ten minutes, there is a useful pointer to the maximum number of complete 'on/run/off cycles' per hour of compressor operation. At a maximum of six starts per hour (ie minimum delay of 10 minutes between shut down and a restart), and a ten minute minimum run time, the number of complete 'on/run/off' cycles is ~3. At a maximum of ten starts per hour (6 minutes between shut down and re-start) plus a 10 minute minimum run time, the number of complete cycles is ~4. This provides a useful guide to designers and installers, suggesting that with the compressor delay timer set to between 6 and 10 minutes, they should aim for a **maximum** number of starts per hour to be between 3 and 4.

12. Recommendations

From the investigations carried out here the following recommendations can be made:

- The ground source heat pump used in this investigation delivers reduced thermodynamic performance if operated with very short run times, typically of six minutes or less. It is thought that a similar effect will be seen with other makes and models of GSHPs, but to different extents depending on the size and type of heat exchangers, the refrigerant cycle, expansion valves and thermal mass of major componentry.

- Whilst delay timers are generally incorporated in heat pumps, with delay settings typically of 5 to 10 minutes, these are used to prevent the heat pump from restarting too quickly, and for too many times per hour. This is to prevent nuisance / interference on the electrical supply, and to prevent the heat pump from restarting under high pressure / hot conditions. They do not ensure minimum run times.

It is noted that the Energy Networks Association (ENA) (http://www.energynetworks.org) have recently established a task force to cover network issues related to domestic heat pumps. The remit of the task force is "to collectively examine technical and process issues through engagement with other key stakeholders (including local authorities, heat pump manufacturers, BEAMA, Micropower Council, DECC and Ofgem), and influence decisions concerning the specification and implementation of the heat pumps in order to ensure that network and future Smart Grid requirements are fully considered."

The outcomes of their deliberations should be closely monitored to see if installers and/or manufacturers need to take note of any requirements or recommendations.

- Steps should be taken by the designer/installer, to ensure that under all load side operating conditions, the heat pump will run for a minimum time of the order of ~10 minutes. Longer run times will reduce the effect of the startup "losses" and will lead to higher SPFs. To achieve this on a fixed speed compressor, the designer needs to ensure that there is always enough buffering capacity in the heat distribution system, or add a sufficiently large buffer tank to provide this capacity. The optimal run time predicted for the 6kW system here is of the order of 20 - 30 minutes.

- By reducing short cycling events, and extending run times, the total number of starts per year can be significantly reduced, and the lifetime of the compressor and associated electrical controls will be extended.

- With typical compressor delay times of between 6 and 10 minutes, and a target minimum compressor run time of 10 minutes, designers and installers should aim for **maximum** number of complete 'on/run/off' cycles per hour of between 3 and 4. This should minimise the degradation in startup inefficiency, meet manufacturers' and DNOs' requirements for the maximum starts per hour, and maintain lifetime operating targets for compressor motors and associated electrical contactors.

- Attention should always be paid to the effect of the parasitic pumping energy required to circulate the ground loop antifreeze fluid, and to ensure that this does not have a significant detrimental effect on the overall efficiency of the GSHP system.

- The modified heat pump model that has been developed here to replicate the measurements made on the GSHP system installed at EA Technology, apply to the particular heat pump that was used. Further work would need to be done on other heat pumps of different sizes and types to quantify the degradation in efficiency at start-up. However, it is likely that all heat pumps exhibit this phenomenon, and if anything, larger heat pumps will probably need longer run times to avoid significant degradation in their overall performance.

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Appendix A

Notes related to short cycling of compressors / heat pumps.

The Locomotive

The Key to Avoiding Costly Refrigeration and Air Conditioning Failures is Proper Maintenance and Effective Protection

By James Brogan, The Hartford Steam Boiler Inspection and Insurance Company
‘Many of these controls also monitor voltage and include a function to prevent rapid “short-cycling” of the compressor **(another condition that will significantly shorten motor/compressor life).** Start delay timers will help all systems avoid the destructive effect of rapid cycling of the motor/compressor on and off.’

Danfos
Compressor Protection
Anti Short Cycle

The system must be designed in a way that guarantees a minimum compressor running time to ensure that the motor remains cool and that the oil will return satisfactorily back to the compressor. There must be no more than 12 starts an hour. **A higher number reduces the service life of the compressor**. The use of an anti-short cycle timer is highly recommended. A three minute time out between starts is recommended.

Copeland

COMPRESSOR SHORT CYCLING
AN UNRECOGNIZED PROBLEM

A frequent cause of compressor failure that is seldom recognized or understood is short cycling.

If a compressor failure results, it will either be a motor burn or a lubrication failure, and in both cases it is a near certainty that the cause of failure will be misdiagnosed.

Each time a compressor starts, there is a quick reduction in the suction pressure and therefore the crankcase pressure. The pressure drop causes a reduction in the saturation temperature, resulting in the oil-refrigerant mixture flashing into foam and vapor with the frequent result that a large percentage of the crankcase oil is pumped out of the compressor. If the compressor operates for sufficient time to stabilize the system, the oil will return to the compressor, but if the running period is very short, the oil may still be trapped in the system when the compressor cycles off.

Summary

Regardless of the motor design, excessive short cycling can shorten compressor life, and the service engineer must be alert to malfunctions in system controls which can create short cycling conditions.

Direct response from Copeland regarding short cycling:

We recommend a maximum of 10 starts per hour. There is no minimum off time because scroll compressors start unloaded, even if the system has unbalanced pressures. The most critical consideration is the minimum run time required to return oil to the compressor after start-up. To establish the minimum run time obtain a compressor equipped with a sight tube and install it in a system with the longest connecting lines that are approved for the system. The minimum on time becomes the time required for oil lost during compressor start-up to return to the compressor sump and restore a minimal oil level that will ensure oil pick-up through the crankshaft. Cycling the compressor for a shorter period than this, for instance to maintain very tight temperature control, will result in progressive loss of oil and damage to the compressor. I've attached our guidelines for your reference.

Taken from the Copeland document detailing the use of their scroll compressors:
(Scroll Compressors for Heat Pump Applications ZH12K4E to ZH11M4E, ZH09KVE to ZH48KVE)

5.12 Minimum run time

Emerson Climate Technologies recommends a maximum of 10 starts per hour. There is no minimum off time because scroll compressors start unloaded, even if the system has unbalanced pressures. The most critical consideration is the minimum run time required to return oil to the compressor after start-up. To establish the minimum run time obtain a sample compressor equipped with a sight tube (available from Emerson Climate Technologies) and install it in a system with the longest connecting lines that are approved for the system. The minimum on time becomes the time required for oil lost during compressor start-up to return to the compressor sump and restore a minimal oil level that will ensure oil pick-up through the crankshaft. Cycling the compressor for a shorter period than this, for instance to maintain very tight temperature control, will result in progressive loss of oil and damage to the compressor.

AE17-1262 R1

COMPRESSOR SHORT CYCLING AN UNRECONGIZED PROBLEM

April, 2002

Introduction

A frequent cause of compressor failure that is seldom recognized or understood is short cycling. If a compressor failure results, it will either be a motor burn or a lubrication failure, and in both cases it is a near certainty that the cause of failure will be misdiagnosed.

Each time a compressor starts, there is a quick reduction in the suction pressure and therefore the crankcase pressure. The pressure drop causes a reduction in the saturation temperature, resulting in the oil-refrigerant mixture flashing into foam and vapor with the frequent result that a large percentage of the crankcase oil is pumped out of the compressor. If the compressor operates for sufficient time to stabilize the system, the oil will return to the compressor, but if the running period is very short, the oil may still be trapped in the system when the compressor cycles off.

If this cycle is repeated, the compressor will progressively pump oil from the crankcase, and the entire oil charge can be lost from the crankcase. If the running cycle is short, an oil pressure safety control may not be actuated since it requires at least two minutes run time to trip the heat actuated safety element. Under such conditions the compressor can operate without lubrication to the bearings, with the obvious potential for damage.

A second source of damage can result from liquid refrigerant flooding and loss of refrigerant control. Most expansion valve are quite sluggish in their control characteristics and tend to react slowly to any sudden change in system operating conditions. Under short cycling conditions, the expansion valve may be unable to reach a stable control condition and uncontrolled liquid refrigerant flooding can occur, again posing a threat to the compressor.

Every time the motor cycles on or off, the stator windings try to flex and move. Under prolonged cycling or short cycling conditions, this flexing may eventually create sufficient movement in the windings to scuff the insulation and cause a short. The larger the motor, the more vulnerable it is to winding flexing. With modern motor insulation and varnishes in a properly wound motor, this failure mode is rare, but the potential threat is present in any system subjected to excessive cycling, since the probability is that any motor has a finite life in terms of the number of cycles it can endure.

Short cycling can originate from many sources, and most such problems can be prevented if we understand the reasons behind them.

Discharge Air Thermostat

On larger roof top package air conditioning equipment, short cycling is probably the most common cause of early maintenance problems and compressor failure, to a considerable extent because the specifying engineer fails to specify an operating time delay.

If the controlling thermostat is in the return air stream, the flywheel effect of the conditioned space prevents rapid changes in the return air temperature and short cycling seldom becomes a problem. Unfortunately it is becoming common practice to mount the control thermostat in the discharge air stream. Particularly with large compressors, the abrupt change in cooling capacity as the compressor cycles on and off can create wide swings in discharge air temperature. Compressors equipped with capacity control unloaders can minimize the temperature swing, but factors such as air flow, thermostat setting, and the cooling load may affect the compressor response.

Most manufacturers recognize the problem, but because of competitive pricing an operational time delay is often priced as an optional accessory. Unless the specification clearly requires a time delay, the salesman may be reluctant to push an option which will increase the price, the purchasing agent is frequently interested only in the low bid, and the operating and service engineer are seldom consulted.

In too many cases, after several thousand dollars of maintenance and service expense, the time of delay is eventually added to the system. Obviously there is a need for better communication between specifying engineers and the service and operating personnel who must deal with the practical aspects of everyday operation.

Multizone Hot and Cold Deck Control

A similar problem was frequently encountered when multizone air conditioning units were first introduced with hot and cold deck control. The compressor was controlled by the cold deck thermostat, and obviously the system designer wanted to maintain fairly close temperature control in the cold deck. The problem became acute as a greater portion of the cooling load was satisfied, and the majority of the air flow was shifted to the hot deck. Under light load conditions, compressor cycling could cause tremendous swings in cold deck temperature, with resulting compressor cycling. Time delays in a cold deck system are not conducive to acceptable comfort conditions, and the best solution is hot gas bypass with continuous compressor operation as long as a cooling demand exists.

Automatic Reset High Pressure Control

One of the most frequent types of short cycling failure occurs on systems with air-cooled condensers and automatic reset high pressure controls. If the condenser design is such that the loss of one condenser fan can cause a trip of the high pressure control, on unattended systems it is quite possible that a loss of lubrication failure can occur within a relatively short period of time should a fan motor or fan belt failure occur. The oil is typically pumped from the compressor, the oil pressure safety control is inoperative due to the short operating cycle, and bearing damage can result.

Wherever possible, manual reset high pressure controls are recommended.

Close Differential Control

On any system, air conditioning or commercial refrigeration, where the compressor is controlled by a close differential control, short cycling can be a problem. There really is no magic answer as to an acceptable cycling rate. An adequate run time to stabilize the operating conditions and insure oil return is more important than a long off cycle. The probability is that cycles at three minute intervals will not cause a temperature problem in either the compressor or the contactor.

But the tremendous number of cycles over a period of time that accumulate from short cycling must shorten the life expectancy of both the contactor and the motor, and the benefits of close differential control versus short compressor life must be evaluated on a judgment basis.

The design of the compressor to considerable extent affects its cycle life expectancy. Copelaweld air conditioning and heat pump compressors are spring mounted, with relatively soft mounts for good noise suppression. Spring life of 200,000 cycles would normally be adequate for a 10 year heat pump life. Commercial applications undoubtedly would see more frequent cycling, and 300,000 cycles would be a typical design goal for spring life on commercial welded compressors.
In Copelametic compressors the mounting is external to the compressor, and cycle life would be related to the motor. 500,000 to 1,000,000 cycles might be a typical average life span, with longer life for smaller lower horsepower motors and shorter life for larger horsepower equipment.

Compressor Motor Plug Reversal Tests

In order to validate a three phase motor's capability of surviving under short cycling conditions, Copeland Corporation performs extensive plug reversal tests. The direction of rotation of the motor is reversed at three second intervals on a test stand, putting tremendous stress on the motor windings. This has proven to be a reliable standard to judge a motor's capability of withstanding short cycling stresses.

Summary

Regardless of the motor design, excessive short cycling can shorten compressor life, and the service engineer must be alert to malfunctions in system controls which can create short cycling conditions.