Direct GeoExchange Cooling for the Australian Square Kilometer Array Pathfinder

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Direct Geoexhange Heat Pumps (DGHPs) provide chilling via refrigerant-carrying copper loops buried in the ground which act as a condenser and achieve higher efficiency than equivalent air source heat pumps because of the ground's constant heat capacity. DGHPs are particularly suited to desert environments with more extreme ambient temperatures. The Australian Square Kilometer Array Pathfinder (ASKAP) radio telescope will be an array of 36 x 12-m diameter parabolic dish antennae situated in the WA desert each requiring 5 – 7 kW_{th} cooling for computer equipment. A direct geoexchange system installed at the CSIRO facility in Marsfield, NSW provides chilling to prototype equipment via a 160 L water buffer. Preliminary results indicate a Coefficient of Performance (COP) 5 for chilling water to 15 C. We describe the results from this prototype in detail.

Keywords: Direct Geoexchange, Direct Use, Geothermal Heat Pump

Direct Geoexchange

The emerging geothermal industry in Australia is focused on producing electricity ("indirect use") and should begin delivering substantial results over the next decade. *Direct Use* geothermal energy is more efficiently available as it entails only one energy conversion (absorption or radiation of heat), rather than the several that occur in electricity generation and usage, with each step losing a percentage on conversion. Geothermal Heat Pumps (GHPs) are a direct use of geothermal energy which involve circulating a fluid (water, brine or refrigerant) through earth loops (poly pipe or copper) a few tens of metres deep (Fig. 1, Payne et al. 2008).

Direct Geoexchange Heat Pumps (DGHPs) which circulate refrigerant through copper loops have greater efficiency than water-loop GHPs because:

- copper is more thermally conductive than insulating plastic;
- latent heat transfers directly with the ground on evaporation or condensation; and
- an intermediate water-to-refrigerant heat exchanger between the ground loops and compressor is not required.

DGHPs transfer heat via 30-metre deep, 75-mm diameter bore holes compared with 100-metre deep, 150-mm diameter bores used for water-loop GHPs. A continuous, closed loop of copper piping is inserted and sealed with a thermally conductive grout (cement). Below 5 metres the Earth remains at a stable temperature all-year-round (16-17^{o} C at Marsfield, NSW). The smaller temperature difference between the heat source/sink and the building or water to be heated/cooled results in lower head pressure and energy requirement of the compressor compared to conventional heating and cooling systems. A "desuperheater" may be employed in chilling applications to further optimise the performance and produce hot water or air which can be used usefully elsewhere.

Figure 1: Copper earth loop, liquid & vapour manifold.

ASKAP

The Australian Square Kilometer Array Pathfinder (ASKAP) radio telescope will be an array of 36 x 12-m diameter parabolic dish antennae situated in Boolardy WA (Fig. 2) with construction due to commence in early 2010 and is a precursor to the Square Kilometer Array (SKA) of several thousand 12-m dishes. A prototype of the Electronics Systems (ES) and Phased Array Feed (PAF) package to be cooled has been installed at the CSIRO Australian Telescope National Facility (ATNF) headquarters in Marsfield, NSW. These two components should not exceed 20 - 30^{o} C under all operating and weather conditions and require an estimated heat dissipation of 5 – 7 kWth. Various cooling methods have been considered and the extreme desert temperatures and high price of diesel generated power strongly favour a DGHP solution. The results of testing of a DGHP are summarised in this paper.

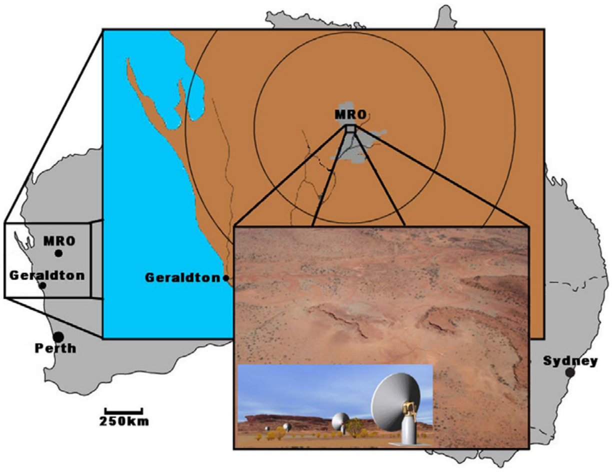

Figure 1: Location of ASKAP

System Design

Cooling is provided by a 10.5 kWth DGHP which circulates refrigerant (R407C) through 4 x 30-m copper earth loops to a refrigerant-to-water, braised-plate heat exchanger. Water is circulated through this heat exchanger into a 160 L buffer tank and is chilled to 7 – 15^{o} C (the “primary” circuit). Water is circulated from this buffer tank to the PAF & ES to deliver the required cooling to the heating load (the “secondary” circuit). The design of the secondary circuit is beyond the scope of this paper – here the performance as a function of thermal load is explored. A controller achieves the specified water temperature.

Earth Loop Installation

The 10.5 kW_{th} system requires 4 x 30-m copper earth loops (½-inch vapour and ¼-inch liquid) with PVC insulation on the upper 15-m of the liquid line to minimise heat transfer between the two and stop flashing of the liquid refrigerant. 75-mm diameter holes are drilled vertically 30-m deep in a square configuration with at least 3-m spacing between them to allow sufficient thermal diffusion through the ground. After insertion of the loops the holes are filled with a thermally conductive (geothermal) grout (e.g. 111-mix, Therm-Ex, IDP-357, Barotherm) ensuring that there are no air pockets. The liquid & vapour lines from the earth loops are braised to their respective manifolds (½-inch liquid & 7/8-inch vapour) using a 15% silver braising alloy. The lines are insulated with ½-inch non-corrosive insulation material (e.g. Armaflex, Insul-Lock) and run towards the heat pump (compressor) with a maximum length of 40-m. Figure 3 shows the drilling and site at Marsfield.

Buffer Tank, Heat Exchanger & Flow Rates

For chilling in the primary circuit, the braised-plate heat exchanger acts as an evaporator and it is crucial to have sufficient water flow over it to fully evaporate the refrigerant else performance will be compromised. The change in water temperature across the heat exchanger should not exceed 5 C and is ideally below 3 C. A minimum flow rate of 0.6 L/s (36 L/min) is required to achieve this. It has proven important to have large diameter (1.5-inch) water pipe between the evaporator and buffer tank. An alternative design has a refrigerant coil evaporator dwelling in the buffer tank and eliminates the need for a circulation pump – this will be investigated.

Heat Pump Design & Refrigeration

Figure 3: Installation of earth loops at CSIRO Marsfield.

The internal design of the heat pump refrigerant system is illustrated in Figure 4. The components are described in turn.

Compressor: A 3-phase Scroll compressor drives the heat pump with compression ratio of 4. Its efficiency is a function of the evaporating and condensing temperatures (pressures).

Active Charge Control (ACC): This:

- prevents liquid refrigerant from reaching the compressor by acting as a reservoir;
- evaporates refrigerant to keep the system properly charged and to eliminate superheat;
- improves volumetric efficiency, reduces power draw, and lets compressor run cooler;
- enables passage of oil entrained in refrigerant;
- indicates refrigerant level via 3 sight glasses.

Liquid Flow Control (LFC): This is an efficient Thermal Expansion Valve (TXV) which:

- Sets proper refrigerant flow rate based on condenser (upstream) operating conditions;
- Ensures zero sub-cooling so condenser is fully active;

- Reduces compressor discharge pressure and lowers power requirement;
- Prevents vapor from "blowing through".

Oil Separator: Oil lubricates the compressor and the oil separator acts with the ACC to prevent oil from migrating down the earth loops.

Dryer: This filters and dries the refrigerant before it enters the LFC.

Reverse Cycle & Check Valve: This reverses the refrigerant flow between heating & cooling modes. For this specific cooling-only application this is redundant hardware and a specific GHP will be designed for the final application.

Refrigerant: R407C [HFC azeotrope: R32 (23%) + R125 (25%) + R134a (52%)]. Future work will explore other HFC and hydrocarbon refrigerants.

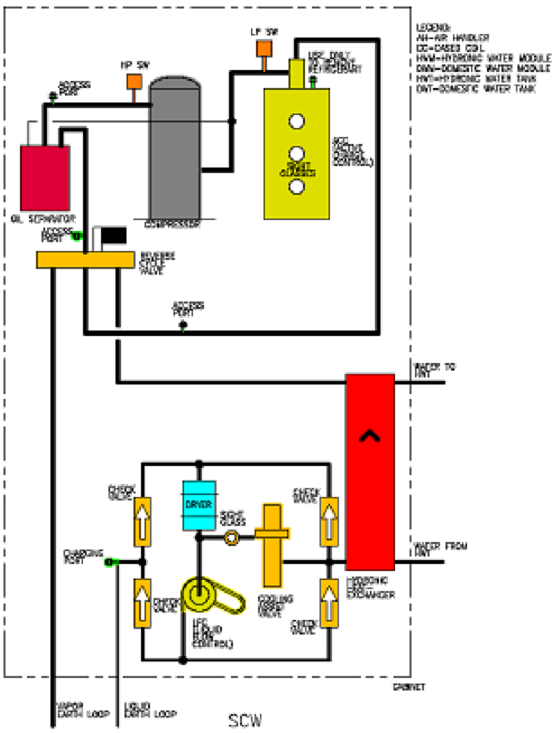

Figure 4: Internal components of DGHP.

Expected Performance

The Air-Conditioning, Heating and Refrigeration Institute (AHRI) has a well-established standard for DGHPs: ANSI/ARI Standard 870, 2001. Also, DGHPs are EnergyStar rated and endorsed by the Environmental Protection Authority (EPA). The DGHP manufacturer, EarthLinked, provides performance tables which are derived from both the Scroll compressor's performance and field trials. For an earth temperature of 16^{o} C and chilling water to 15^{o} C, the expected Coefficient of Performance (COP = thermal energy removed/ electrical energy input) is 5.1. The theoretical maximum (Carnot cycle) performance for cooling is given by (Tcond –Tevap)/Tevap where Tcond is the condensing temperature and Tevap the evaporating temperature (Kelvin) and is 8.4 for Tcond = 37 ^{o} C and Tevap = 4 ^{o} C which correspond to the above conditions.

Methodology

To find the most efficient means of cooling, the COP is measured as a function of the controllable aspects of the system. The key controllable variables are:

Buffer Set Point ($T_{b\text{-}set}$): the temperature to which the buffer tank is controlled to be chilled – it is measured at the primary outlet of the buffer tank.

Buffer Maximum ($T_{b\text{-}max}$): the buffer tank temperature at which the controller switches on the chilling DGHP.

Cabinet Set Point ($T_{cabinet}$): the temperature to which the cabinet is cooled.

Secondary Load (P_{load}): the thermal power which is simulated with a set of heaters.

Tank Load (P_{tank}): the element of the tank can be turned on to act as a thermal load to the system.

Other input variables include:

Ambient Temperature: this includes both the wet (T_{wet}) and dry (T_{dry}) bulb temperatures.

Environmental Gain (P_{envt}): the tank & pipes are insulated but there is still environmental gain.
Tank Volume (V_{tank}): the buffer tank volume.

Water Volume (V_{water}): volume of water in the system: tank, primary & secondary pipes.

Output variables include:

Compressor Electrical Power (P_{comp}): the electrical power used by the compressor.

Cycle Time (t_{cycle}): the time between compressor start-ups.

Compressor Time (t_{comp}): the time the compressor is on during a cycle.

Duty Cycle (D): the percentage of time the compressor is on (t_{comp}/t_{cycle}).

Earth Loop Temperatures ($T_{5\text{–}30m}$): The temperatures measured at 5, 10, 15, 20, 25 & 30m.

Refrigerant liquid & vapour temperature (T_{liq}, T_{vap}): the temperatures to and from the earth loops

Suction, Head & Return Pressures (P_{head}, P_{suc}, P_{ret}): The pressures in & out of the compressor and returning from the earth loops.

Results

The default input values are: $T_{b\text{-}set}$ = 15^{o} C, $T_{b\text{-}max}$ = 17^{o} C, $T_{cabinet}$ = 23^{o} C, P_{load} = 5.2 kW_{th}, P_{tank} = 0,

V_{tank} = 160 L, V_{water} = 201 L. The ambient temperature varies between 10 and 35^o C during the test periods and future results will be calibrated against this. An estimate of the environmental gain was determined by cooling the buffer to 15^o C, leaving the system off and measuring the time taken (t_{envt}) for the temperature to increase a known amount (ΔT_{test}). $P_{envt} = C_{pw}.V_{water}.\rho_w. \Delta T_{test}/t_{envt}$ where C_{pw} = 4.2 kJ/(kg.K) is the specific heat of water, ρ_w = 1 kg/L is the density of water. It was found that for ΔT_{test} = 2 K, t_{envt} = 135 min giving P_{envt} = 208 W.

Figure 5 shows the temperature and pressure outputs for the above input conditions. From these results, the COP can be derived. We find t_{cycle} = 1645 sec, t_{comp} = 737.5 sec, giving D = 44.83%. There is a temperature overshoot of at least 0.75 K below $T_{b\text{-}set}$ and above $T_{b\text{-}max}$ giving ΔT = 3.5 K. The power required to reduce the system water temperature is $P_{water} = C_{pw}.V_{water}.\rho_w. \Delta T/t_{comp}$ = 4.01 kW_{th}.

The electricity consumed by the compressor is measured over a known number of cycles to give average power consumption. For this run, E = 210 MJ is consumed over n_{cycle} = 87.17 cycles (39.8 hr) giving an average power $P_{comp} = E/(t_{comp}.n_{cycle})$ = 3.268 kW_{elec}.

$COP = [P_{water} + (P_{load} + P_{envt})/D]/P_{comp}$ = 5.0. Note that D accounts for the fact that the load and environment are constantly delivering heat to the system which must be dissipated by the heat pump whilst on.

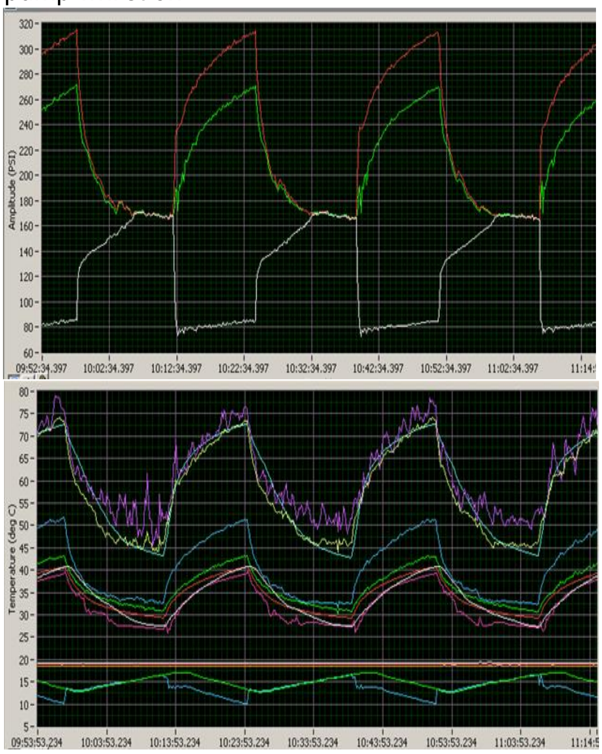

Figure 5: Temperature (top) and Pressure (bottom). From top to bottom, the lines are: T_{vap}, $T_{30\text{-}m}$, $T_{25\text{-}m}$, $T_{20\text{-}m}$, $T_{15\text{-}m}$, $T_{10\text{-}m}$, $T_{5\text{-}m}$, T_{liq}, spares, $T_{tank\text{-}out}$, $T_{tank\text{-}in}$. P_{head}, P_{ret}, P_{suc}.

Using this method, the COP is calculated as a function of load and DGHP configuration. Figure 6 shows a summary of average COP values as a function of secondary thermal load and Figure 7 displays COP as a function of the buffer water temperature.

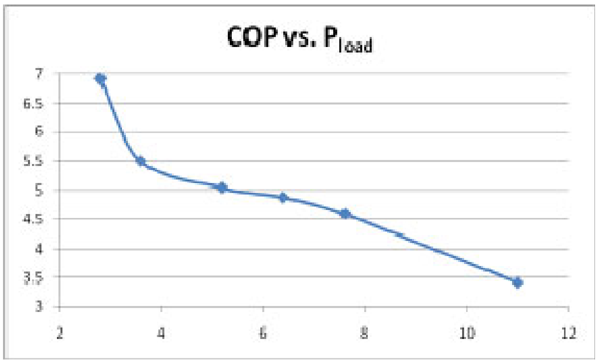

Figure 6: COP vs. P_{load}.

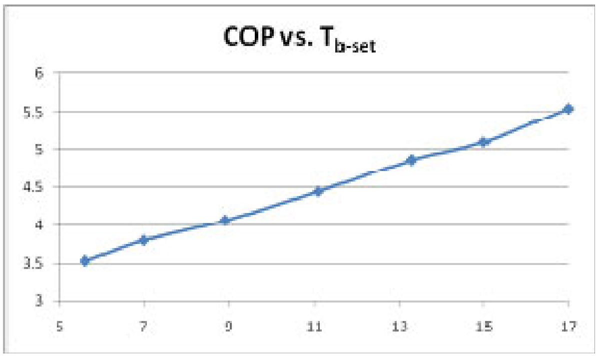

Figure 7: COP vs. $T_{b\text{-}set}$.

It is clear that the system should not be designed to run with a 100% duty cycle. An initial configuration of the system had the heat pump unit 30 m away from the earth loops. After moving the DGHP to within 4 m of the earth loops, the pressure drop across the earth loops was 5 – 15 PSI lower. Also, the manifold pit was initially exposed for testing and thus about 15 m of the copper earth loops was exposed to air and thus not efficiently dissipating heat. Backfilling the manifold pit boosted the performance.

Earth Loop Temperature

The ground temperature is measured every 5 m down one of the bore holes via thermocouples. As seen in Figure 5, after the compressor starts, the ground temperatures rise asymptotically towards saturation at which thermal output matches thermal dissipation in the ground. The no load condition is shown in Figure 8 along with the maximum load (11 kW_{th}) with saturated ground temperatures in Figure 9. In Figure 8 note that the manifold temperatures track the ambient temperature overnight as the pit was still open at the time this data was taken. From the no load graph it is clear that there are anomalous offsets in the ground temperatures. The precise explanation for these is as-yet undetermined. It is suspected that the thermocouples used are unsuitable for use with the grout. In the absence of further details, the temperatures are calibrated by these offsets.

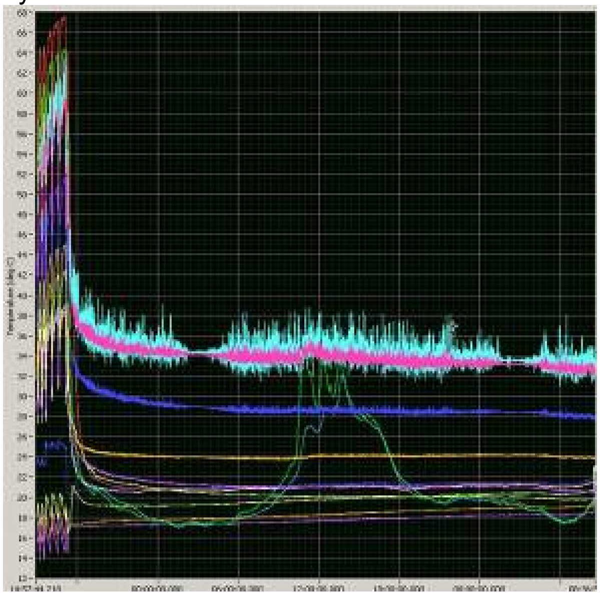

Figure 8: Temperatures for no load condition.

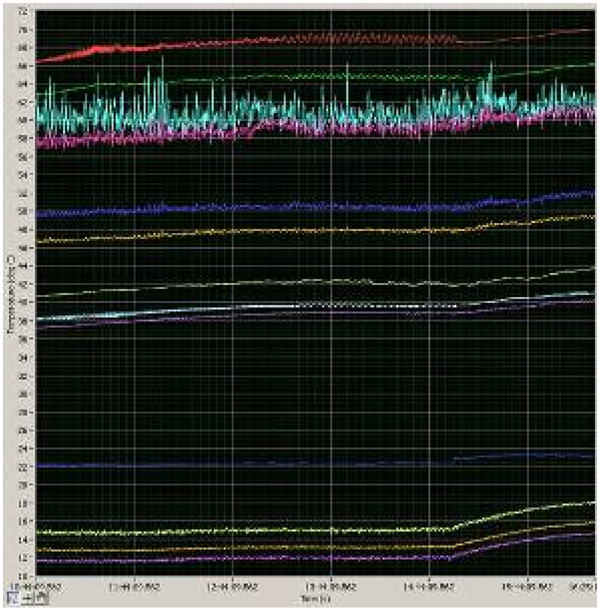

Figure 9: Temperatures for P_{load} = 11 kW_{th}.

Conclusions & Further Work

DGHPs are an efficient solution for heating and cooling and are particularly suitable for cooling in a desert environment with no power infrastructure. It has been shown that a DGHP achieves a COP exceeding 5 when chilling water to 15^{o} C in the primary circuit of a cooling system for a pedestal of the ASKAP – the prototype installation is at Marsfield NSW with a ground temperature of 16 - 17^{o} C. Furthermore, it has been demonstrated that COP decreases as a function of load in the secondary circuit (which determines the duty cycle) and buffer water temperature. Important factors which influence performance include the distance from earth loops to heat pump and the coverage of the manifolds.

Further results will be obtained for this configuration and additional experiments include:

- Use of a flooded evaporator within the buffer tank to eliminate the primary circulation pump.
- Use of a desuperheater between the compressor head and the earth loops.
- Use of alternative refrigerants.

These results are representative of expected behaviour in the field site at Boolardy, WA though slight variations will result from:

- An expected ground temperature of 20-21^{o} C at Boolardy.
- Different thermal conductivity of the ground.
- Different baseline thermal gain to the system.

Figure 10: Location of desuperheater, compressor & buffer tank.

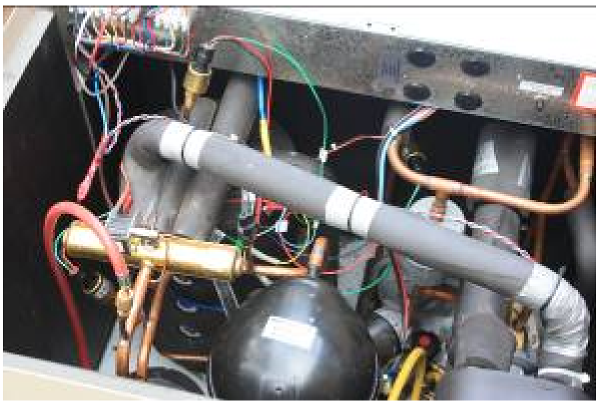

Figure 11: View inside heat pump unit.

References

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