Ground Cooling Potential in a Mediterranean Climate: Fieldwork and Computational Modelling

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The aim of this research was to explore the potential of exploiting ground source cooling, utilising the high thermal mass of the indigenous sedimentary rock, globigerina limestone, typical of the Maltese Islands.

Based on earlier studies of horizontal and vertical pipe configurations, it was established that the vertical U-tube pipe is more efficient. This demonstrated that at 20m depth ground temperatures were around 20°C all year round. This prompted further studies with various refined parameters, this time moving from fieldwork to numerical simulations, facilitating a greater variety of trials.

A mathematical model of the system was developed and implemented into a computational model to simulate the heat transfer processes. This model was calibrated using results from the field tests. The computational model was also used to carry out a more elaborate parametric analysis of the system and to simulate its performance in a variety of configurations. Preliminary results indicate the validity of the numerical model and the applicability of such a system.

The same numerical model results and field tests were then used to estimate the potential of such a ground source heat pump system to curtail energy demand for cooling on running the standard HVAC systems.

Keywords: Ground cooling; Heat pump; Energy-efficiency

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ABSTRACT: The aim of this paper was to explore the potential of exploiting ground source cooling, utilising the high thermal mass of the indigenous sedimentary rock, globigerina limestone, typical of Malta. Based on earlier studies of horizontal and vertical pipe configurations, it was established that the vertical U-tube pipe is more efficient. This demonstrated that at 20m depth ground temperatures were around 20℃ all year round. This prompted further studies with various refined parameters, this time moving from fieldwork to numerical simulations, facilitating a greater variety of trials. A mathematical model of the system was developed, implemented into a computational model to simulate the heat transfer processes. This model was calibrated using results from the field tests. This was also used to carry out a more elaborate parametric analysis of the system and to simulate its performance in a variety of configurations. Preliminary results already indicate the validity of the numerical model and the applicability of such a system. The same numerical model results and field tests were then used to estimate the potential of such a ground source heat pump system to curtail energy demand for cooling on running the standard HVAC systems.
Keywords: ground cooling; heat pump; passive design; energy-efficiency.

INTRODUCTION

The energy legislation of the European Union is based on three pillars, namely security of supply, liberalisation of the energy market and the protection of the environment. Given the increase in security of supply through the Euro-grid and the coming on stream of new wind farms, the first two areas are making great leaps. However this cannot be said of the third pillar.

Environmental protection stems mainly from energy efficiency. In this regard, curtailment of wastage and the fast track improvement in energy efficient systems augur well as energy saving measures in buildings. Ground source heat pumps are but one option. The building industry alone tops up around 40-45% of the national energy bill. There is therefore much ground to be gained by any form of energy efficiency.

BACKGROUND

In a small Mediterranean Island where mild winters and hot dry summers prevail, there is greater energy demand for cooling rather than for heating. In Malta only a nominal effort has been made so far to explore ground source heat exchangers as a form of pre-cooling to conventional environmental control systems (ECS).

By 2020 EU (European Union) countries are aiming to reduce CO_2 emissions by at least 20% below 1990 levels by reducing energy consumption, making use of renewable energy sources, increasing energy efficiency and that every building must have an energy performance certificate and smart metering [1]. Given the fact that as an Island-State, and its fossil-based economy, Malta's obligation is to reach a 10% reduction by 2020 [2].

LITERATURE REVIEW

The Ground Source Heat Pump (GSHP) is a much favoured technology by architects, as this does not disrupt the aesthetics of the façade or skyline. Moreover it requires a smaller plant room than conventional ECS and can be installed in both new construction or retrofitted in older properties. Its design flexibility allows the GSHP to handle even a partial cooling load of a larger building.

Over the last two decades most systems were developed to work with either water or air as the running medium to dissipate heat through both the ground and water (sea or reservoir) as the infinite thermal sink [3]. Experiments pointed towards the use of water in a closed system through the ground. Environmental Authorities were equally reassured of no contamination of the water table by alternative open source systems [4].

The critical parameters affecting the effective heat transfer of a GSHP are pipe diameter, pipe length, pipe material, the water flow rate, the thermal conductivity of the rock, ground temperature at any given depth and time, the thermal capacity of the soil, its water content and naturally, the number of boreholes drilled [5]. In view of the importance of such parameters, before setting up a GSHP, one typical test carried out includes the Thermal Response Test (TRT). This is devised to assess the thermal conductivity of the bedrock, its thermal capacity, ground water effects and natural interactive convection between the water and the pipe work.

EXPERIMENTAL SET-UP

Before setting up such a field test in Malta, precedent studies were assessed, namely from South Korea, Greece and Malta. Essentially these pointed to the use of the ideal pipe diameter, pipe material and flow rates. Table 1 shows pipe parameters for the three case studies evaluated.

Table 1: Pipe parameters for case studies

	University Building, South Korea	City Hall of Pylaia, Northern Greece	Chamber of Parliament, Malta
Pipe type	HDPE	HDPE	MDPE
Pipe diameter (mm)	42	40	32(internal), 40(outer)
Depth of BHE (m)	175	80	140
Number of boreholes	24	21 (3x7)	48
Diameter of borehole (mm)		100	150

As part of a separate research project by Borg [6], an open site was courteously provided on Campus at the University of Malta. The overall setup comprised a 20m long horizontal trench, a 20m deep vertical borehole and a monitoring room, as shown in Figure 1. In both the vertical and horizontal configuration, U-tube pipes were placed, consisting of two pipes connected together by a U-joint. In each configuration, three sets of pipes were placed, namely 10m, 15m and 20m U-tubes, as shown in Figure 1.

The monitoring room was used to record inlet, outlet and ambient temperatures, together with ground temperature at certain depths in the ground, mainly at 1.2m, 5m, 10m, 15m and 20m (Figure 1). In addition, the water flow was also measured.

After obtaining all necessary permits, the 20m long horizontal trench was dug to a depth of 1.2m and a width of 250mm. On the other hand the 20m deep borehole was drilled, having a diameter of 300mm. The rock excavated was all Globigerina limestone.

The pipes used were LDPE (Low Density Polyethylene) pipes, having a diameter of 40mm. Where possible, all the joints used were push fit. The U-tube system was formed by joining two equal lengths of pipe by two elbows and a socket.

The system installed in the monitoring room assessed how good the ground is at absorbing heat transferred from the water heater / condenser or heat exchanger. The experiment is commonly known as TRT (Thermal Response Test).

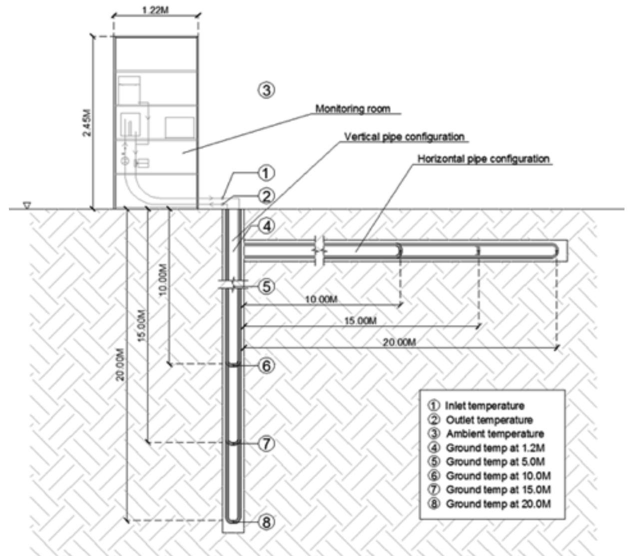

Figure 1: Section through whole system

OUTPUT RESULTS

Overall output results are summarised in Table 2. When comparing the 10m horizontal configuration (550W) with the 20m horizontal configuration (1173W), results show that the highest water outlet temperature lay between 47°C and 49°C. This shows that horizontal configuration lengths at a depth of 1.2m below the ground are inversely proportional to the overall rise in temperature of the system.

On the other hand, comparing the 10m vertical to the 20m vertical configuration, the 20m vertical one was 12.4% more efficient than the 10m one. This clearly shows that the deeper the borehole is drilled below ground level, the greater its efficiency. The significant percentage difference between the 10m vertical and the 20m vertical configurations proves that the 10m vertical one was too close to the ground surface for efficient heat transfer to occur. When comparing the 10m horizontal with the 10m vertical configuration, the latter proved to be 7.17% more efficient. Conversely, comparing the 20m horizontal with the 20m vertical one, the latter turned out to be 20% more efficient. This has a cooling effect of 1.50°C temperature difference.

Table 2: Summary of results

Configuration type		Date of testing	Test duration (hrs)	Power Supplied (W)	Maximum inlet temp (°C)	Maximum outlet temp (°C)	Temp difference (°C)
10m	Vertical	13-15/08/11	23	550	44.61	43.75	0.84
	Horizontal	16-17/08/11	23	550	47.86	47.13	0.74
20m	Vertical	08-10/08/11	47	1173	39.94	38.44	1.5
	Horizontal	28-30/07/11	47	1173	50.49	48.94	1.52
20+10m	Vertical	10-12/08/11	47	1173	40.37	38.68	1.62
	Horizontal	31-02/07/11	47	1173	49.5	48.01	1.48

MATHEMATICAL MODELLING

A mathematical model of the ground source loop used in the experiments was developed in order to be able to simulate a number of parametric variations of the system.

Mathematically the experimental setup can be reduced to a pipeline carrying a fluid at a known inlet temperature having a variable external surface temperature. Also, the pipeline can be discretized into small lengths, over which the outside temperature can be considered as being constant. In this regard a single element is shown in Fig. 2.

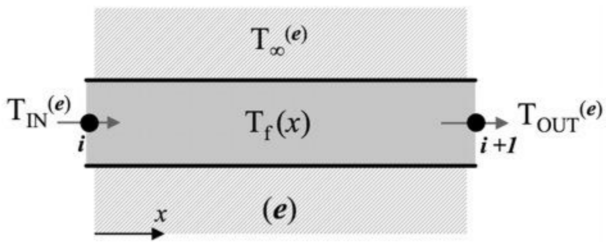

Figure 2: A single element of the discretized pipeline

For each of these elements the inlet temperature, T_{IN}, and the surrounding temperature, T_∞, are known. The temperature distribution within the element can therefore be obtained as a function of distance travelled by the fluid along the pipe, from which the outlet temperature, T_{OUT}, can be computed as follows [7]:

$$T_{OUT}{}^{(e)} = T_\infty{}^{(e)} - \left(T_\infty{}^{(e)} - T_{IN}{}^{(e)}\right) \cdot \exp\left(\frac{-L^{(e)}}{\rho Q c_p R}\right) \quad - (1)$$

In equation (1) the terms ρ, Q and c_p refer to the density, flow rate and specific heat capacity of the fluid, respectively. $L^{(e)}$ denotes the elemental length, which is the total pipe length divided by the number of elements. Finally, R is the total resistance to heat transfer between the fluid and the surrounding soil. In this case this is the sum of three thermal resistances acting in the radial direction of the cross-section.

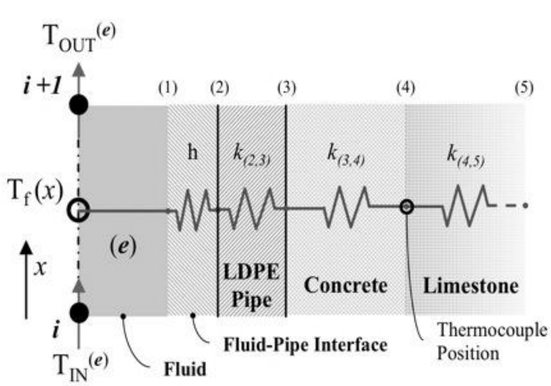

Figure 3: Thermal resistances along the radial system

As shown in Fig. 3, radially the system consists of three solid layers and a fluid-solid pipe-wall interface. The total thermal resistance can be computed by adding the resistances in series [7] as shown in equation (2):

$$R = \frac{1}{\pi D_{(1)} h} + 2\pi \sum_{a=2}^{4} \left[\frac{1}{k_{(a,a+1)}} \ln\left(\frac{D_{(a+1)}}{D_{(a)}}\right)\right] \quad - (2)$$

In equation (2), k is the thermal conductivity of the solid interfaces and D denotes the diametric distance. The numerical subscript (a) refers to the periphery of each interface, as shown in Fig. 3. Finally, h is the heat transfer coefficient at the fluid-pipe interface. This can be expressed in terms of the average *Nusselt number* on the inner area of the pipe. This dimensionless parameter describes the ratio of convective heat transfer at the interface to the conductive heat transfer within the fluid itself [7]. It is related to the heat transfer coefficient as follows:

$$\overline{\mathrm{Nu}_D} = \frac{D_{(1)} h}{k_f} \quad - (3)$$

In equation (3) k_f is the thermal conductivity of the fluid. In many cases this dimensionless parameter is impossible to obtain analytically and empirical results specific to the type of flow are typically used; these would be the result of correlation with a wide range of experimental data. In the case of internal forced convection a number of correlations are available, although some carry errors as large as 25% on the Nusselt number. A more accurate correlation exists that includes the effects of friction at the internal surface of the pipe, reducing the associated uncertainty to 10% [7]. This is referred to as the Gnielinski correlation [8] and is defined as:

$$\overline{\mathrm{Nu}_D} = \frac{(f/8)(\mathrm{Re}_D - 1000)\mathrm{Pr}}{1 + 12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \quad - (4)$$

In the correlation shown in equation (4), Re_D is the Reynolds number using the pipe diameter as a length scale and Pr is the Prandtl number. The Darcy friction factor is denoted by f and is obtained as a function of internal surface roughness based on the Colebrook and White formula [9] along with the Haaland approximation [9], which increases computational efficiency [10]. The Gnielinski correlation has been experimentally validated for the conditions stated below [7], both of which are satisfied by all the computations that were carried out.

$$3000 \le \mathrm{Re}_{D_i} \le 5 \times 10^6$$
$$0.5 \le \mathrm{Pr} \le 2000$$

Finally, it must be noted that the correlation is subject to the assumption that a fully developed turbulent internal flow exists throughout the length of the pipe. Entry lengths for turbulent flows, as in this case, are however typically short [7], and it is reasonable to assume that the average *Nusselt number* obtained from such correlations maintains its validity for the entry region [7].

The elements are easy to inter-relate mathematically since the outlet temperature of an upstream element becomes the inlet temperature of the next downstream element. This was implemented into a computational procedure using a MATLAB® algorithm. Nodal temperatures were obtained by a cubic interpolation of the experimental measurements. An analysis of the discretization was also carried out, with no change in the results beyond 50 elements.

Through the selection of adequate values of thermal conductivity for concrete and limestone, listed in Table 3, the model was calibrated to an accuracy of 0.0078% on the outlet temperature for the experimental parameters. Throughout the modelling process it was also assumed that beyond a distance of 1m from the end of the concrete layer, the surrounding soil is unaffected by the pipeline.

Table 3: Thermal conductivities for concrete and limestone

	Recommended Values [11]	Selected Values
Concrete (Medium)	0.4 – 0.7 $Wm^{-1}K^{-1}$	0.6 $Wm^{-1}K^{-1}$
Limestone	1.3 – 3.3 $Wm^{-1}K^{-1}$	1.6 $Wm^{-1}K^{-1}$

The computational model was designed to compute the final steady-state ground temperature along with the outlet fluid temperature. This is directly comparable to the temperature measured by the thermocouples at $T_{(4)}$ shown in Fig. 3 after the system reached steady-state conditions. This result was computed to act as a form of validation for the model; results are shown in Fig. 4.

It must be noted that the temperature measurements of the soil shown as measured data in Fig. 4, are actually mirrored measurements, such that readings corresponding to 25, 30 and 38m are the same as those taken at 2, 10 and 15m respectively.

One can also observe that the computed steady-state ground temperature is not symmetrical. This is because the fluid near the outlet is cooler than that at the inlet and therefore the soil in its immediate vicinity would tend to also be slightly cooler. Given the limited number of thermocouples available during the experiments, experimental results are not available for direct comparison.

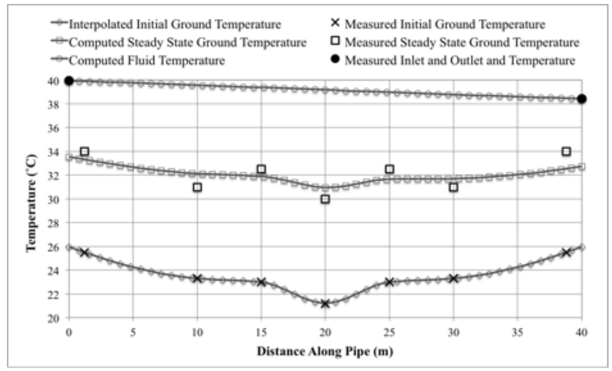

Figure 4: Results from the computational model and experimental measurements

THE EXPERIMENTAL SETUP IN PRACTICE

In cooling mode, the most common use of the experimental setup presented in this paper is to serve directly as a transfer mechanism for the rejection of the heat absorbed from an air conditioned space. Compared to the setup used in this experiment, such a process however generally requires a longer run of pipes. Given the physical limitations of the experiment, rather than rejecting all the heat, the experimental setup was thought of as an add-on to a conventional vapour compression refrigeration cycle air conditioner, providing additional sub-cooling to the traditional process, hence increasing system performance.

VAPOUR REFRIGERATION CYCLE - SUB-COOLING

A conventional vapour refrigeration cycle air conditioner, shown in Fig. 5, makes use of a dedicated refrigerant to transfer heat from one heat sink to another. In cooling mode the evaporator (EVP - Process 4-1), where the refrigerant evaporates absorbing heat from the surroundings, is placed in the area where cooling is required. The condenser (CND - Process 2-3), where the refrigerant condenses rejecting heat to the surroundings, is placed outside. An energy intensive gas compressor (CMP - Process 1-2) drives the refrigerant around the various stages of the cycle.

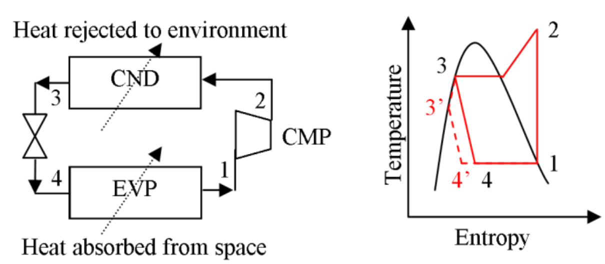

Figure 5: Vapour compression refrigeration cycle

One method which can be used to reduce the overall energy requirement of the compressor is to sub-cool the refrigerant at exit from the condenser [12]. The refrigerant exiting the condenser of a typical vapour refrigeration cycle is generally just wet at the saturation pressure (Point 3). By further sub-cooling the refrigerant (Point 3'), for the same system, the cooling capacity of the refrigeration cycle can be increased, such that the overall energy consumption of the compressor decreases.

INTEGRATING THE MATHEMATICAL MODEL

Using MATLAB®, the mathematical model of the ground source loop, modelled and calibrated using the experimental 20m vertical setup, was integrated within a text-book case scenario of a vapour compression refrigeration cycle air conditioner working on R134A refrigerant, and between an evaporator temperature of -10°C and a condenser temperature of 45°C.

Compared to a conventional vapour compression refrigeration cycle air conditioner, an additional counter flow heat exchanger (HE) was added just after the condenser, shown in Fig. 6. Inside this heat exchanger the refrigerant exiting the condenser is sub-cooled by the water inside the ground source loop flowing in the opposite direction.

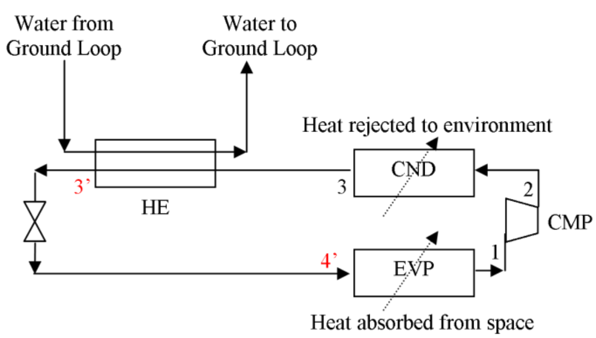

Figure 6: Cycle with additional ground source cooling loop

MODELLING THE HEAT EXCHANGER

The heat exchanger model was modelled using the *NTU* method. The *NTU* method relies on comparing the maximum possible heat transfer between two fluids (q_{max}) inside a heat exchanger to the actual heat transfer between the two fluids (q). This is known as the effectiveness of the heat exchanger (E) and for a counter flow system, such as that used for this particular model the effectiveness can be found using equation (5) [13]:

$$E = \frac{1-exp\left(-(UA/c_r)(1-c_r/c_w)\right)}{1-(c_r/c_w)exp\left(-(UA/c_r)(1-c_r/c_w)\right)} \qquad \text{- (5)}$$

In equation (5) U is the overall heat transfer coefficient, A is the heat exchanger surface area, c_r is the heat capacity of the refrigerant and c_w is the heat capacity of the water inside the ground source loop.

Based on the calculated effectiveness and by knowing the inlet temperatures inside the heat exchanger setup (the inlet temperature of the condensed refrigerant exiting the condenser and entering the heat exchanger is known at 45°C from the vapour compression refrigerating cycle selected, whilst the inlet water temperature of the ground source loop entering into the heat exchanger is calculated from the mathematical model of the 20m vertical ground loop) the exit temperatures of the two fluids can be found, hence the actual heat transfer and the degree of sub-cooling of the refrigerant.

SIMULATIONS AND CALCULATIONS PERFORMED

Using the modelled setup, a set of simulations were performed for different nominal refrigerating power quantities, typical of residential and small commercial cooling loads, where the bulk of vapour compression refrigeration cycle air conditioners are used. Five nominal cooling power ratings were used, namely 2, 3.5, 5, 7.5 and 10kW, and for each the water flow rate inside the ground source loop was varied between 5 l/min and 10 l/min. For each of these power ratings the MATLAB® model was used to calculate the required parameters: heat exchanger effectiveness, exit temperatures and the Coefficient of Performance (COP) of the system. The compressor efficiency was assumed throughout as 85%.

Based on the steady-state performance characteristics obtained from the MATLAB® model, an estimate of the amount of annual energy and CO_2 emissions savings for an individual dwelling in Malta could be calculated using the following considerations:

- The annual average cooling load of a dwelling in Malta is about 1,930 kWh/dwelling/annum [14].
- *Approx.* 1.088 kgCO_2 are emitted per kWh of electricity energy delivered at end-use [15].

For comparison purposes the modelled setup was compared to a conventional vapour compression refrigeration cycle air conditioner operating between the same temperature limits, but without the heat exchanger and the ground source loop providing the additional sub-cooling.

SIMULATION RESULTS

The first important result can be observed from a comparative analysis of the different COP obtained for the different systems modelled and simulated. This is shown in Fig. 7.

Compared to the system without the ground source loop, which has an equivalent COP of 3.51 for the entire range of cooling power ratings simulated, the average COP of the system with the ground source loop increases by 5%, varying between a maximum of 12%

for the 2kW setup with 5 l/min ground source loop water flow rate and a minimum of 2% for the 10kW setup with 10 l/min ground source loop water flow rate. *The average increase for the 3.5 kW setup, the most common cooling power rating for residential units, is of 6%.* Also, it can be observed that for the same ground loop water flow rate, increasing the cooling requirement reduces the improvement in COP obtained, suggesting that the experimental setup has a physical limit beyond which the improvement in COP becomes negligible.

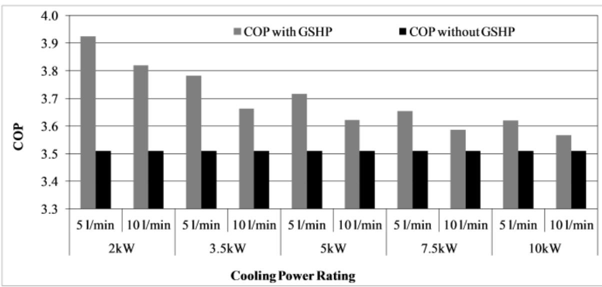

Figure 7: COP for the simulated systems

The change in COP is also reflected in CO_2 emissions. Shown in Fig. 8 are the emissions of the simulated systems supplying the cooling to an individual dwelling in Malta. For an average household the expected emissions reduction compared to the use of a conventional vapour compression refrigeration cycle air conditioner *without* ground source loop would be in the range of a maximum of 11% for the 2kW setup with 5 l/min ground loop water flow rate and a minimum of 2% for the 10kW setup with 10 l/min ground loop water flow rate.

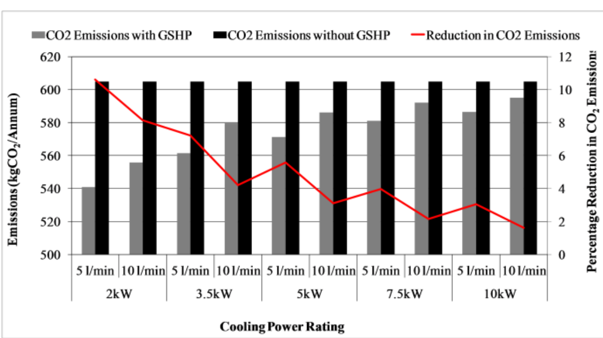

Figure 8: CO_2 emissions for an average dwelling

For a vapour compression refrigeration cycle air conditioner, if one had to assume a uniform COP of 3.51 for a 3.5kW unit, if one had to implement such a system all over Malta, this would result in an average energy cut-back of 1,982,916 kWh/annum and a reduction of CO_2 emissions by 2,180,416 $kgCO_2$/annum.

CONCLUSIONS

Experimental results showed that water circulating through pipes embedded into the ground were not affected by ambient atmospheric air temperatures. Moreover, in all cases the 20m vertical configuration proved to be the most efficient, with a 1.50°C cooling effect. By decreasing flow rates and increasing U-tube lengths, the output temperature would decrease further; hence the system would operate more efficiently.

On the other hand, the mathematical model therefore demonstrated that given the latest state-of-the-art air conditioning technology, (assuming a typical COP of 3.51, for a 3.5kW unit), if one had to implement such a system on a nationwide scale, this would result in an average energy savings of just under 2GWh/annum and a reduction of CO_2 emissions by 2.2×10^6 $kgCO_2$/annum. Hence this proves the potential of GSHP technology for Malta.

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