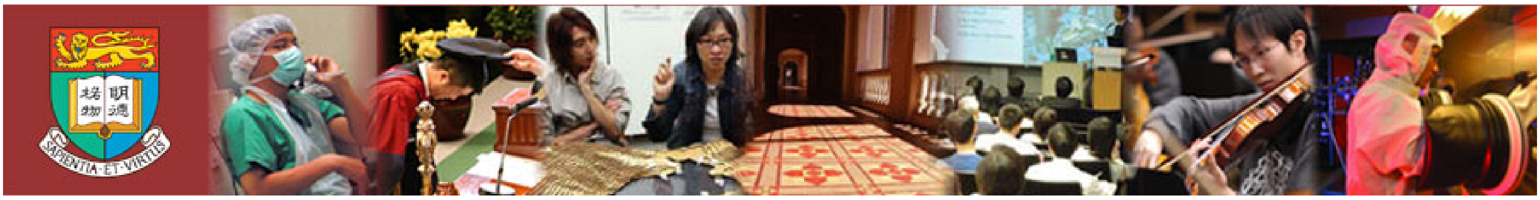

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APPLICATION POTENTIAL OF GROUND-COUPLED HEAT PUMPS FOR MULTI-STOREY OFFICE BUILDINGS IN HONG KONG

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ABSTRACT

In this paper, the application potential of ground-coupled heat pumps (GCHP) in a multi-storey office building in Hong Kong was investigated. It was found that within the limited land area occupied by the building, the GCHP could only handle the cooling load for one floor over a range of the ground thermal conductivities and undisturbed ground temperatures. The year-round energy consumption of the GCHP was compared with those using the conventional vapour-compression chiller systems. An energy saving of at least 13.2% and 2.6% could be achieved against those using an air-cooled and a water-cooled vapour-compression chiller with rated coefficients of performance equal to 3.0 and 5.0 respectively.

INTRODUCTION

With the increasing concern about global warming and the quest for the reduction in carbon emission, the selection of more energy-efficient systems becomes very important. In Hong Kong, around 90% of the total electricity consumption is attributable to buildings and accounts for about 50% of the total carbon emission. In particular, the air-conditioning systems account for the majority of the building energy demand. Hence, the enhancement of energy efficiency of air-conditioning equipment is essential. Ground-coupled heat pumps (GCHP), which employ the ground as the medium of heat exchange with the surroundings, offer a higher energy efficiency than the conventional air-cooled unitary systems (Li and Hong 2010). Fig. 1 shows the general arrangement for such systems. Both cooling and heating can be provided by the GCHP depending on the building applications and the climate of the installation locations. GCHP have been used in Europe and USA for many years (Spitler 2005), and the demand in China is also increasing rapidly (Gao, et al. 2009) in recent years.

In Hong Kong, most of the commercial buildings are multi-storey and employ large-capacity central chilled water air-conditioning systems which enjoy a higher coefficient of performance (*COP*) than that the small-capacity systems can provide. Besides, the allowance of a part-load control in the large-capacity chillers further improves the year-round performance of the air-conditioning plant. In this sense, the energy-saving potential of applying GCHP in a multi-storey office building appears to be challenging, especially in a sub-tropical region like Hong Kong. Moreover, as cooling is predominately required for office buildings in Hong Kong throughout the whole year, the huge amount of unbalanced condenser heat accumulated in the ground will result in deterioration of the capacity of the borehole ground heat exchangers (BHE). Hence, in this research, dynamic simulations are made to compare the energy consumption of GCHP used in such cases with those based on the conventional chilled water air-conditioning systems. Both the air-cooled vapour-compression chiller (ACVCC) and the water-cooled vapour-compression chiller (WCVCC) are used in the analysis in order to assess the application potential of GCHP. Only cooling mode operation is considered.

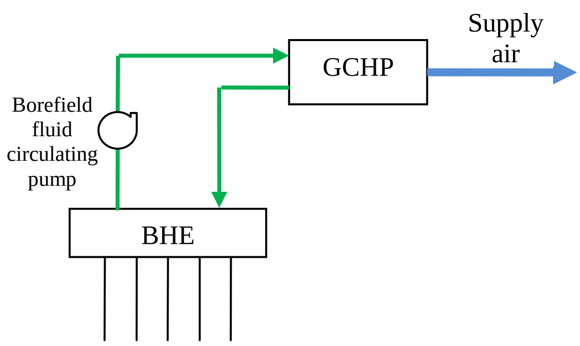

Figure 1 General arrangement of a GCHP system

SIMULATION

In this analysis, a typical twenty-storey office building with a cross-sectional area of 14 m x 14 m and a floor-to-floor distance of 3.6 m is used. Table 1 summarises the various design internal heat sources for each floor in which the supply air conditions are determined based on the typical weather data of Hong Kong (Chan et al. 2006). The design indoor and outdoor conditions are 25.5^{o}C/60%RH and 32.8^{o}C/71%RH respectively. The fresh air amount is 0.01 m^3/s per occupant based on the common practice in Hong Kong. The air-conditioning system only provides cooling to the building with a daily operating schedule from 8:00am to 6:00pm.

Dynamic simulations are performed using a common building simulation tool (TRNSYS 2006). TRNSYS adopts a component-based platform for the formulation of a system. Different components can be added to a system via a graphical interface so that the components can be linked together as desired. Moreover, TRNSYS allows the users to develop their own components for use in the system simulation.

Most of the components required in this study are available in the simulation package, including the ACVCC (Type655), the WCVCC (Type666), the GCHP (Type504) and the building zone (Type56). The rated *COP* of the ACVCC and the WCVCC are 3.0 and 5.0 respectively based on the design temperatures of 7, 30 and 35°C for the return chilled water, entering condenser water and the ambient air. Correction factors are then calculated by the chiller models based on externally supplied data files to determine the chiller performance under other operating conditions. The default files offered by TRNSYS are used for both the ACVCC and the WCVCC. The GCHP model calculates the equipment output by interpolating the performance data supplied through external files derived from the manufacturer's catalogue data (www.carrier.com). The performance data of the GCHP under various operating conditions are listed in the files.

Table 1
Design internal loads for the building zone

TYPE	VALUE
Maximum number of occupants	24
Lighting, W/m^2	17
Computer, W per occupant	230

For the BHE, a new TRNSYS component is developed which is based on a 3-D numerical model (Lee and Lam 2007) using a rectangular coordinate system. All boreholes are assumed to be identical and aligned in a rectangular borefield. Each borehole is then approximated by a square column which is circumscribed by the borehole radius. Two configurations, namely a 2x2 and a 3x3 borefield, are tried with a borehole spacing of 7 m and 4.5 m respectively so that all boreholes are equally-spaced within the 14 m x 14 m land area. Table 2 shows the various parameters used for the borehole unless otherwise specified. A range of the ground thermal conductivities and undisturbed ground temperatures are considered in order to investigate their different effects on the borehole design. The operation of the GCHP is controlled by a room thermostat with hysteresis between 24.5 and 26.5°C. The borefield fluid circulating pump runs when the compressor of the GCHP operates. Meanwhile, the supply air fan of the GCHP is switched on within the entire daily operating schedule.

Table 2
Parameter values used for the boreholes

PARAMETER	VALUE
Insulated length of borehole, m	5
Borehole radius, m	0.055
U-tube inner radius, m	0.013
U-tube outer radius, m	0.016
Distance between tube and borehole centre, m	0.03
Effective ground volumetric heat capacity, kJ/m^{3}K	2,160
Borehole circulating fluid volumetric heat capacity, kJ/m^{3}K	4,190
Borehole circulating fluid thermal conductivity, W/mK	0.614
Borehole circulating fluid dynamic viscosity, kg/ms	0.00086
Pipe thermal conductivity, W/mK	0.4
Grout thermal conductivity, W/mK	1.3

Figs. 2&3 show the general layout of the conventional air-cooled and water-cooled chilled water air-conditioning systems. An air-handling unit (AHU) comprising a cooling coil (Type52) and a supply air fan (Type3), is used to provide cooling to the building zone for each floor. A three-way chilled water valve is employed to regulate the chilled water flow to the cooling coil. A proportional controller is used to control the operation of the water valve based on the zone temperature. The controller outputs a signal from 0 to 1 when the zone temperature ranges from 24.5 to 26.5°C corresponding to the valve changing from a fully closed to a fully open status. To enhance the iteration stability of the system simulation, the chilled water valve opens at least 30%. The operation of the chiller is governed by a return chilled water thermostat between 9 and 12°C. For the WCVCC, a cooling tower (Type51) is employed to supply the condenser water to the chiller. The condenser water pump runs when the chiller operates. The cooling tower is additionally controlled by a return condenser water thermostat between 15 and 20°C. The AHU and the chilled water pump operate throughout the entire daily operating schedule.

In order to investigate clearly the energy-saving potential of a GCHP system, all parasitic energy consumptions from the auxiliary equipment such as the water pumps, the supply air fan and the cooling tower are taken into account. All water pumps are assumed to have an efficiency of 60%. For the cooling tower, the default parameters set by TRNSYS are adopted. The air flowrate of the draft fan is then selected so that the temperature of the condenser water leaving the cooling tower is 30°C when the temperature of the condenser water entering the cooling tower is 35°C. A fan static of 200 Pa is used with a fan efficiency of 65%. Finally, the supply air fan head is taken as 750 Pa with a fan efficiency of 70%.

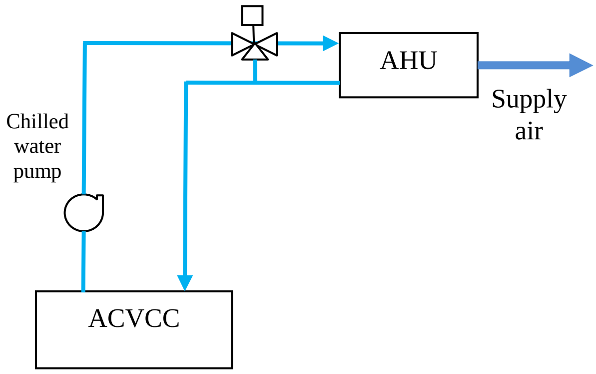

Figure 2 General arrangement of an air-cooled chilled water air-conditioning system

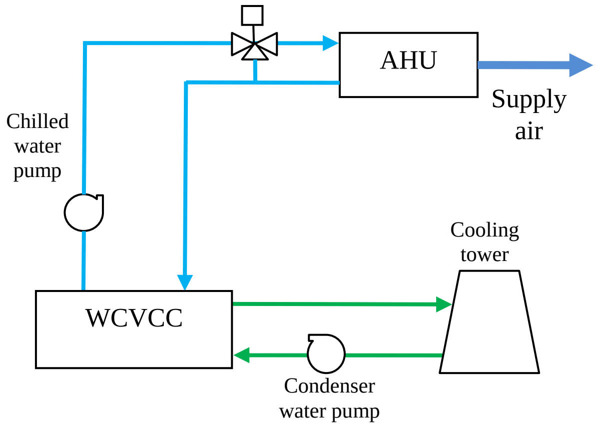

Figure 3 General arrangement of a water-cooled chilled water air-conditioning system

DISCUSSION AND RESULT ANALYSIS

Determination of the system design conditions

Table 3 shows the system design conditions based on the building loads for each floor as indicated in Table 1. The system load included the ventilation load due to the fresh air. The same supply air volume flow rate and the fan static were used for both the conventional AHU and the GCHP. The condenser water and chilled water volume flow rates were then selected so that there was about 5°C temperature change after passing through the chiller.

Table 3
Summarised system design conditions per floor

PARAMETER	VALUE
Peak zone sensible load, kW	18.83
Peak zone latent load, kW	1.19
Peak system load, kW	28.8
Supply air volume flow rate, m^3/s	1.5
Supply air temperature, °C	15.5

Based on the available manufacturer performance data for the GCHP, a single unit could not handle the system cooling load of each floor. Hence, two sets of GCHP's (model RVC048) were used instead which offered a *COP* of 3.23 at a borefield fluid entering temperature of 32.2°C and entering air conditions of 26.67°Cdb/19.44°Cwb including the power consumption from the supply air fan. The chosen total borefield fluid volume flow rate was 1.5 litre/s for each floor. To improve the fluid flow capacity of the 2x2 borefield, a double U-tube was used for each borehole with the best U-tube connection configuration (Zeng et al. 2003) while only a single U-tube was employed for the 3x3 borefield. With this arrangement and the consideration of the specified internal radius for the U-tubes, both configurations could only handle the borefield fluid for two floors at maximum. All boreholes were assumed to be connected in parallel with a single pipe header.

Performance of GCHP at different ground conditions

The capacity of the BHE depends readily on the ground conditions, namely the ground thermal conductivity (k_g) and the undisturbed ground temperature (T_o). Besides, the loading profile applied to the BHE, which relates strongly to the climate of the installed locations, also plays a significant role in influencing the performance. Hence, to investigate the effect of the ground conditions on the performance of GCHP, year-round dynamic simulations were carried out. As the annual air-conditioning load in Hong Kong is cooling-dominated, there will be a surplus of heat injected into the ground which results in a gradual increase in the ground temperature in the absence of groundwater flow. Consequently, a ten-year simulation period was adopted which also allowed the change in the year-round total system energy consumption to be analysed. For a cooling-dominated application, the peak fluid temperature leaving the BHE ($T_{bf,out,max}$) was the key parameter. Hence, the borehole lengths under different ground conditions were selected so that $T_{bf,out,max}$ was around 30°C within the ten-year simulation period as summarised in Figs. 4&5 for the 2x2 and the 3x3 borefield based on the loading for one floor.

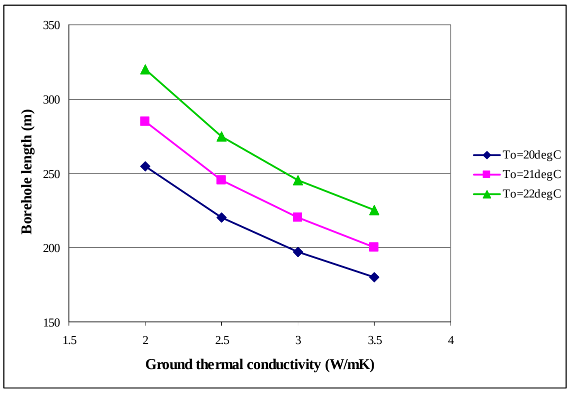

Figure 4 Variation of borehole length with k_g and T_o for a 2x2 borefield serving one floor

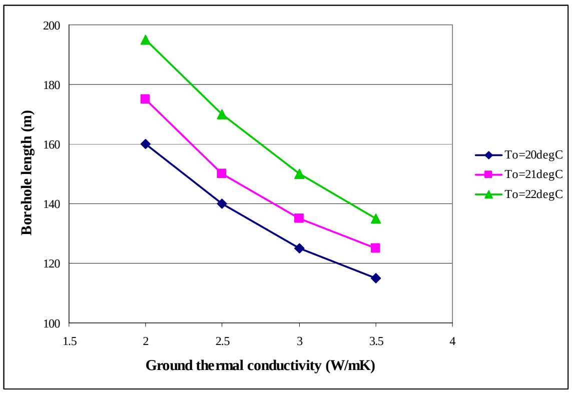

Figure 5 Variation of borehole length with k_g and T_o for a 3x3 borefield serving one floor

From Figs. 4&5, it could be found that a shorter borehole length was required with a lower T_o and a higher k_g. The use of a 2x2 borefield increased the borehole length by at least 59% when compared with those based on a 3x3 borefield. However, when considering the total installed borehole length, the adoption of a 2x2 borefield could lead to a saving of at least 25%. The overall impact on the borehole installation cost depended on the increase in the drilling cost per unit length between deep and shallow boreholes as well as the increase in the material cost for the additional U-tubes inside the boreholes for a 2x2 borefield.

A value of 3.5 W/mK for the ground thermal conductivity was considered high, and if k_g was taken as 2.5 W/mK, the lowest design borehole length was about 140 m which referred to the situation for a 3x3 borefield with an undisturbed ground temperature of 20^{o}C. This was not short indeed, and if the loading for two floors were considered, the resulting design borehole lengths would be at least two times those based on the loading for one floor. Consequently, it was concluded that the BHE could only serve the air-conditioning demand for one floor only.

Figs. 6&7 shows the corresponding total energy consumption from the GCHP and the borefield fluid circulating pump under different ground conditions for the 2x2 and 3x3 borefields respectively. It could be found that the total energy consumption was higher with a higher undisturbed ground temperature. Meanwhile, the use of a 2x2 borefield led to an increase in the total energy consumption ranging from 1.09 to 2.25%, basically due to a higher energy demand from the borefield fluid circulating pump in response to a longer borehole length required. Besides, with a stronger thermal interference between adjacent boreholes for a 3x3 borefield, the fluid temperatures at the early stage of the simulation would need to be lower in order to have the same $T_{bf,out,max}$ at the end of the simulation. Consequently, the *COP* for the GCHP was better when using a 3x3 borefield.

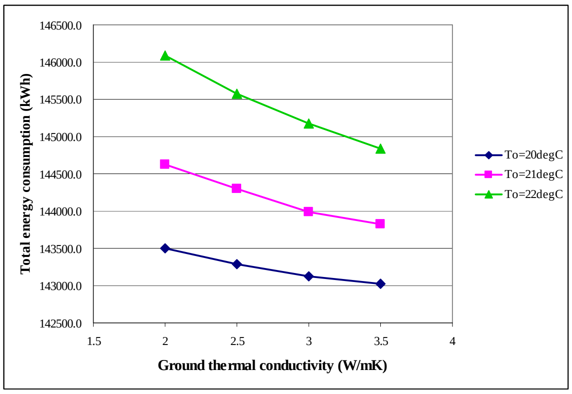

Figure 6 Variation of total energy consumption with k_g and T_o for a 2x2 borefield serving one floor

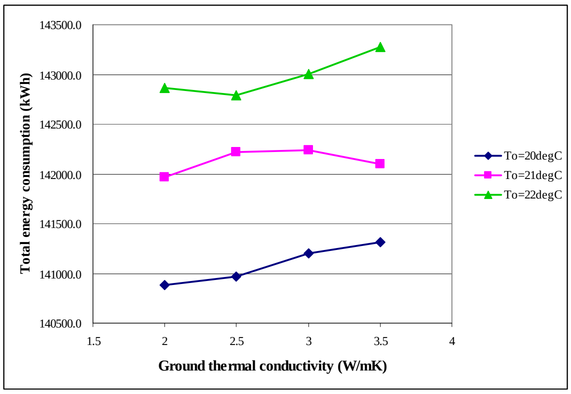

Figure 7 Variation of total energy consumption with k_g and T_o for a 3x3 borefield serving one floor

For a fixed T_o, the effect of k_g on the total energy consumption appeared to be different between the two BHE configurations. The reason was that the change of k_g had two opposite effects on the total energy consumption. With a higher k_g, the required borehole length was shorter which reduced the fluid pump power. However, the fluid temperatures at the early stage of the simulation would be higher in order to achieve the same $T_{bf,out,max}$ at the end of the simulation. This resulted in a higher energy consumption for the GCHP. The overall effect depended on which one was the predominating factor. For a 2x2 borefield with a much longer borehole length, the effect from the fluid pump power was stronger. Hence, the total energy consumption decreased with an increase in k_g. On the other hand, no definite trend was observed when using a 3x3 borefield. Nevertheless, the percentage difference in the total energy consumption was less than 0.34% at various T_o when using a 3x3 borefield, while it was less than 0.87% with the 2x2 borefield.

As the system stops daily after 6:00pm, the fluid temperature inside the borefield will approach the borehole surface temperature. Hence, the borefield fluid temperature at the end of the ten-year simulation indicates the average borehole surface temperature rise after the ten-year operation period. From the various cases investigated, it ranged from

3.54 to 5.14°C for the 2x2 borefield and 4.67 to 6.33°C for the 3x3 borefield. A higher value reached when using the 3x3 borefield was due to the fact that the ground volume within the borefield was smaller with a shorter borehole length. Another reason was the stronger thermal interference from the boreholes with a smaller borehole separation. In general, the average borehole surface temperature increased with the undisturbed ground temperature but decreased with the ground thermal conductivity.

Comparison with conventional central air-conditioning systems

The technical feasibility of a GCHP system depends on its energy-saving potential when compared with that based on a conventional air-conditioning system. To simplify the analysis, it was assumed that the building zones for all the floors were the same. Hence, the total energy consumption for one floor was calculated by dividing the total building energy consumption by twenty. Unlike the BHE in which the performance was highly dynamic, only one-year simulation was needed for the conventional systems, and the overall ten-year energy consumption was determined simply by multiplying the one-year value by ten. The corresponding results for the air-cooled and water-cooled systems were 168357 kWh and 149930 kWh respectively.

By comparing these values with those depicted in Figs. 6&7, it was found that over the ranges of k_g and T_o investigated, energy-saving could be achieved for both borefield configurations. For the ACVCC system, the use of the GCHP could save at least 13.2% and 14.9% of the energy consumption when employing a 2x2 and 3x3 borefield respectively. For the WCVCC system, the corresponding values dropped to 2.6% and 4.4%. The small energy-saving potential necessitated a cheaper way to install the BHE. One possible solution was the adoption of energy piles in which the U-tubes were embedded into the concrete piles. Further study was required to investigate the technical feasibility.

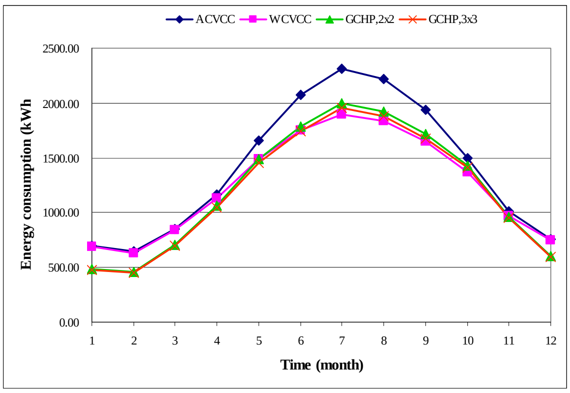

Figure 8 Comparison of monthly total energy consumption for various systems

Fig. 8 compares the monthly total energy consumption for the various systems investigated. For the GCHP, the results were based on the fifth-year simulations with k_g and T_o being 2.0 W/mK and 22°C respectively. It could be found that the GCHP performed better than the WCVCC system during the low-load period. The reason was that the energy consumption from the auxiliary equipment became more significant with the WCVCC system during the said period. Under the operation of the part-load control, the running time for the WCVCC and consequently the condenser water pump and the cooling tower increased. This also led to a higher total energy consumption. Indeed, the year-round running time for the WCVCC was 27% longer than that for the GCHP. Meanwhile, the performance of the GCHP was superior to the ACVCC system throughout the whole year.

CONCLUSION

This paper studied the performance of ground-coupled heat pumps (GCHP) systems applied in a multi-storey office building in Hong Kong. Two configurations, namely a 2x2 and a 3x3 borefield, were tried with a borehole separation of 7 m and 4.5 m respectively. It was found that over the range of the ground thermal conductivities and undisturbed ground temperatures investigated, the required borehole lengths were greater for the 2x2 borefield. However, the total installed borehole lengths was shorter than those based on the 3x3 borefield. Moreover, the total energy consumption was smaller by adopting the 3x3 borefield. Within the limited land area in which the building was accommodated, the GCHP could only handle the cooling load for one floor.

The year-round energy consumption of the GCHP was compared with those using the conventional vapour-compression chiller systems. An energy saving of at least 13.2% and 2.6% could be achieved against those using an air-cooled and a water-cooled vapour-compression chillers with rated coefficients of performance equal to 3.0 and 5.0 respectively when using the 2x2 borefield. Higher values of 14.9% and 4.4% could be reached by employing the 3x3 borefield. The small percentage savings indicated that a cheaper installation method for the ground heat exchangers would be necessary in order to make the GCHP systems economically feasible.

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NOMENCLATURE

COP	Coefficient of performance
k_g	Ground thermal conductivity, mK/W
$T_{bf,out,max}$	Peak fluid temperature the boreholes, °C
T_o	Undisturbed ground temperature, °C

Abbreviations

ACVCC	Air-cooled vapour-compression chiller
AHU	Air-handling unit
BHE	Borehole ground heat exhangers
GCHP	Ground-coupled heat pumps
WCVCC	Water-cooled vapour-compression chiller

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