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EXPERIMENTAL MODEL OF GROUND-SOURCE HEAT PUMP SYSTEM

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1. ABSTRACT

Geothermal heat pump is a new technology in air conditioning systems. This study presents experimentally the procedure of design and installation of a geothermal heat pump model in Baghdad, and to investigate the applicability and efficiency of a ground coupled heating system to reduce the energy consumption. The experimental study was based on using a vertical borehole of depth 7.25 meter with a small unit of water-to-water heat pump for heating an enclosure of dimensions (1.45 x 0. 65 x 0.65) m.

The heat extraction rate from the soil is found to be, on average, (59.03 W/m) of the bore depth, while the required bore depth in meter per kW of heating capacity is obtained as (13.49 m/kW). The entering water temperature to the unit is measured to be 15.1^{o}C and the coefficient of performance of the heat pump COP for the unit is about 2.574.The results are analyzed in terms of hourly and daily operation characteristics.

Keyword: Geothermal heat pump, Ground source heat pump

2. INTRODUCTION

Ground source heat pump systems (GSHPs) are considered as modern and suitable air conditioning and water heating systems. They provide comfort, leading to significant reduction of electrical energy use and demand. Its main advantage is the use of normal ground or ground water temperature between about 5 and 30^{o}C, which are available in all countries of the world [Lund, et al, 2004].

GSHPs come in two basic configurations: ground coupled (closed loop) and ground water (open loop) systems which are horizontally and vertically installed, or in wells and lakes surface water heat pumps

(SWHP) are also installed in a lake. Ground source heat pump systems commonly consist of either water-to-water or (water-to-air) heat pump systems and provide space heating and cooling and water heating with relatively high efficiency.

Many investigators have studied the application of direct utilization of geothermal energy. According to some of them, [Hepbasli, 2005] dealed with the thermodynamic analysis of ground-source (water to water) heat pump system for district heating and he concluded that the value of heat pump unit and whole system were obtained to be 2.85 and 2.64 respectively. A GSHP system [Sekine et al, 2004] was developed using the cast-in-place concrete pile foundations of a building as heat exchangers in order to reduce the initial boring cost. [Haiwen, et al, 2006] and [Mohamad, and Bo, 2008] made a comparison between single and double U-bend pipe heat exchangers. He concluded that the GSHP system with double U-bend buried pipe is often a better solution from technical and economical point of view.

3. GROUND CHARACTERISTICS

Thermal conductivity and diffusivity are two parameters which need to be clarified in order to estimate the likely subsurface temperatures and heat transfer characteristics. The heat transfer to the ground coil will not only be determined by the area for exchange but also these two factors.

4. THERMAL DIFFUSIVITY

Thermal diffusivity is a measure of the ground thermal conduction in relation to thermal capacity. This links the thermal conductivity, specific heat (cp) and density (ρ) according to the relation [Mortaza, 2005]:

$$\alpha_s = \frac{K_s}{cp_{sc}\rho_{sc}} \tag{1}$$

A high thermal diffusivity value is desirable since the heat is rapidly conducted relative to thermal mass [Pharoah, 2007].

In this study, analytical equations have been used for determining the thermal properties, density, thermal conductivity, coefficient of soil, K_s (W /m K) and thermal diffusivity α_s (m^2 /hr). These properties are calculated by the following equations [Mortaza, 2005]:

For silt and clay soils, the thermal conductivity is

$$K_s = 0.14423(0.9\log\omega - 0.2)10^{0.000624\rho_{sd}} \quad (2)$$

and for sand soils

$$K_s = 0.14423(0.7\log\omega + 0.4)10^{0.000624\rho_{sd}} \quad (3)$$

In which ω is the moisture content with weight percentage and ρ_{sd} is the dry density of soil.

$$cp_{sc} = \left[\omega cp_w + (100-\omega)cp_{sd}\right]/100 \quad (4)$$

is the corrected specific heat of soil, and

$$\rho_{sc} = \left[\omega\rho_w + (100-\omega)\rho_{sd}\right]/100 \quad (5)$$

is the corrected density of soil, cp_{sd} is the dry specific heat, assumed to be 0.84 kJ/kg K [Mortaza, 2005].

The mass of a sample of soil from borehole was measured. It is then heated to reject the moisture then its mass was measured again. The difference between the two readings represents the moisture (ω).

Using equations (1 to 5), the physical properties of sand soil (used to fill the vertical borehole) and the coefficient of thermal diffusivity are calculated and presented in table (1)

Table(1): Physical Properties of soil

Parameters	**Values at 7.25 m in depth**
Dry density ρ_{sd} , kg/m ^{3}	1450
Moisture ω % by weight	20
Dry specific heat cp_{sd} , kJ/kg K	0.84
Corrected density ρ_{sc} , kg/m^3	1360
Corrected specific heat cp_{sc} , kJ/kg	1.532
Thermal conductivity K_s , W/m K	1.518
Thermal diffusivity α_s , (m^2 /hr)	0.00262

5. GROUND TEMPERATURE ESTIMATION

The method for sizing the ground heat exchanger requires knowledge of the minimum ground temperature at the ground heat exchanger (GHX) depth.

According to [Ministery of Natural Resources Canada 2001-2005], the minimum ground temperature can be expressed as

$$T_{g,\min} = \overline{T}g - A_s \exp\left[- x_{sd} \sqrt{\frac{\pi}{365\alpha_s}} \right] \qquad (6)$$

Where; $\overline{T}g$ is the mean annual surface temperature, A_s is the annual surface temperature amplitude $(T_{\max} - T_{\min})$.

For horizontal heat exchanger pipe system or a shallow vertical borehole, x_{sd} can be set equal to the average soil depth in equation (6). For a vertical system, this usually becomes a trivial task since the sub-surface ground temperature doses not vary significantly over the course of the year [Ministry of Natural Resources Canada 2001-2005].

6. DESIGN ENTERING WATER TEMPERATURE (Tewt)

The minimum design entering water temperature used in a ground-source heat pump (GSHP) is [Ministry of Natural Resources Canada 2001-2005].

$$T_{ewt,\min} = T_{g,\min} - 7\ ^{o}\mathrm{F} \qquad (7)$$

7. PART LOAD FACTOR

The part load factor for heating can be evaluated as [Ministery of Natural Resources Canada 2001-2005]:

$$F_h = \frac{\overline{q}}{q_{\max}} \qquad (8)$$

Where, $\overline{q}$ is the average load for month, $q_{\max}$ is the peak load for month

8. GROUND HEAT EXCHANGER (GHX) SIZING

The ground heat exchanger sizing is mainly concerned with the determination of the heat exchanger length. The method used in the ground-source heat pump (GSHP) is largely adapted from the

International Ground-Source Heat Pump Association (IGSHPA). The required GHX length [Ministery of Natural Resources Canada 2001-2005], based on heating, L_h, is

$$L_h = q_{h,heat}\left[\frac{\frac{(cop_h - 1)}{cop_h}(R_p + R_s F_h)}{T_{g,\min} - T_{ewt.\min}}\right] \tag{9}$$

Where ; $q_{d,heat}$ is the design heating load, $COPh$ is the design heating coefficient of performance of the heat pump, R_s is the soil thermal resistance, F_h is the part load factor for heating, $T_{ewt,\min}$ is the minimum design entering water temperature (EWT) at the heat pump, R_p is the pipe resistance.

The focus of this study is the ground-source heat pump system with a vertical ground closed-loop heat exchanger configuration. The outer and inner diameters of the tube of the ground heat exchanger are 25 mm and 21 mm, respectively with 6 hours time work in one day.

The design of the ground heat exchanger has been calculated by using the above mentioned equations (6 - 9) and the calculation results of the design are given in the table (2).

Table (2).Calculation Results of Design of the Ground Heat Exchanger for 7.25 Meter Depth Borehole

PARAMETERS	VALUES
Design heating load, $q_{d,heat}$ (W)	500
Design coefficient performance, $COPh$	2.7
Pipe resistance, R_p (m. ^{o}C /W)	0.038
Soil resistance, R_s (m. ^{o}C /W)	0.211
Part load factor, F_h	0.58
Minimum ground temperature, $T_{g,\min}$ (^{o}C)	14.14
Minimum design entering water temperature, $T_{ewt,\min}$ (^{o}C)	7.14
The required Ground Heat Exchanger (GHX)length based on heating, L_h (m)	7.25

8. EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus consists of three separate circuits,

a) The ground coupling circuit

b) The refrigerant circuit, for is a vapor compression system

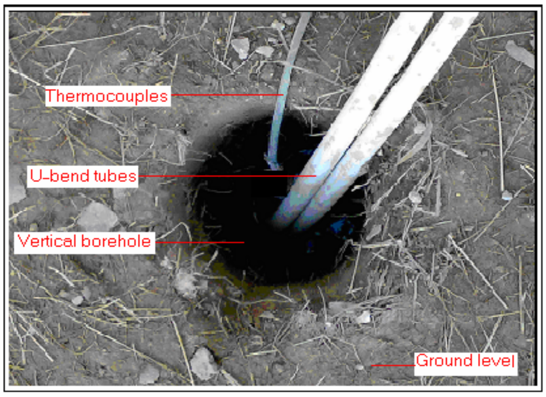
c) The fan-coil circuit in the space condition

Figure (1) shows the layout of test rig which installed in postgraduate laboratory

Figure(1) The experimental rig

a) Ground Coupling Circuit

The ground coil of the heat pump was installed in a bout 7.25 meters depth vertical borehole 150 mm inner diameter as shown in figure (2) and figure (3). The borehole was drilled outside the laboratory beside the wall. The polyethylene tubes used as heat exchanger are of 25 mm outer diameter and 21 mm inner diameter with total length of 17.5 meter U-bend. The bottom-end of the tubes is clamped by a stainless steel clamp to a close return U-bend fitting.



Figure(2) Borehole with Heat Exchanger

Figure (3) Buried Borehole

b) The Refrigerant Circuit

The heat pump refrigeration circuit diagram used is presented in Fig. (1). It consists of four basic components: reciprocating compressor, evaporator, condenser, and capillary tube. The used refrigerant is R12.

c) The Fan Coil Circuit and Space Heating

The fan coil unit consists of radiator with dimensions ($34 \times 27 \times 5$) cm including 25 column of rectangular tubes, axial fan, and pipes. The space heating model has interior dimensions of (65 x 145 x 65) cm and is elevated by (1 m) above laboratory floor. The walls of the experimental model are constructed from the Perspex sheets of (4 mm) thickness.

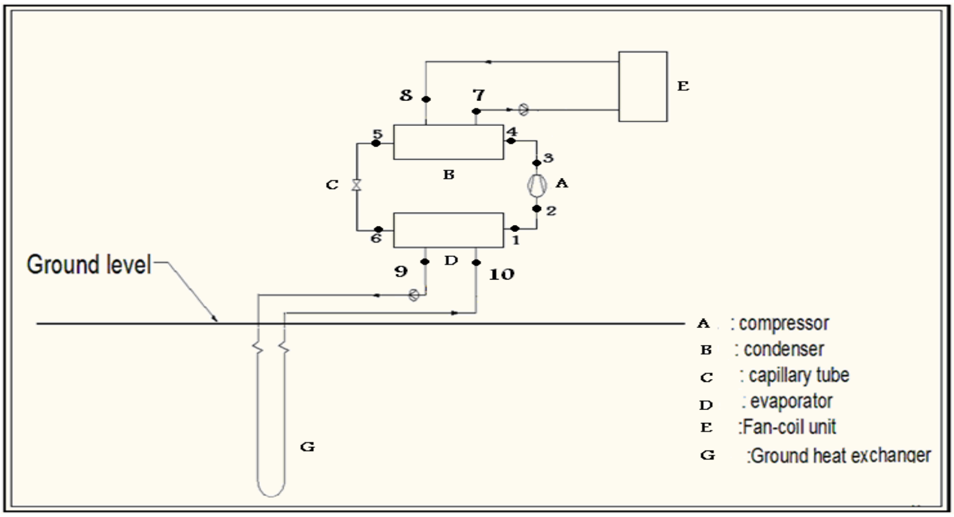
9. METHODS OF TESTING THE PERFORMANCE OF GSHP WITH RESPECT TO TIME

The measurement of the temperature was taken every five minutes of the inlet and outlet water temperature through the evaporator to calculate the heat extracted from the evaporator and ground. The inlet and outlet temperatures of the water passed the condenser were measured to calculate the heating capacity. The inlet and outlet air temperatures were measured in order to calculate the heating space load. The values of the current and voltage for the compressor, fan, and pump were recorded every five minute to calculate their power.

The above measurement are used to evaluate the COP for each hour during the day and for 14 days. The experiments were conducted through 25 January to 9 February 2009.

10. EXPERIMENTAL ANALYSIS AND CALCULATIONS

Figure (4) illustrates the inlet and outlet of each component of the (GSHPs) and figure (5) reveals the experimental P-H diagram of the refrigerant cycle.



Figure(4) The Inlet and Outlet of Each Component of the Ground-Source Heat Pump System (GSHPs)

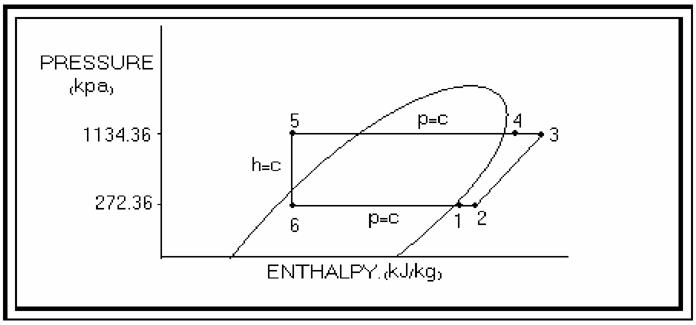


Figure (5): A pressure-enthalpy refrigeration cycle

1- Outlet of evaporator, 2- Inlet of compressor, 3- Outlet of compressor,

4- Inlet of condenser, 5- Outlet of condenser, 6- Inlet of evaporator

The parameters to be calculated based on experimental data are, [Hepbasli, 2005] and [Mortaza, et al, 2005]:

a) The Heat Extracted From Evaporator

$$\dot{Q}_{evap} = \dot{m}_w Cp_w (T_{10} - T_9) \tag{10}$$

Where $\dot{Q}_{evap}$. is the heat extract by the evaporator, $\dot{m}_w$ is the mass flow rate of water passed the evaporator, Cp_w is the specific heat of the water passed the evaporator and $(T_{10} - T_9)$ is the water temperature difference between inlet and outlet evaporator. The heat extract from ground $\dot{Q}_{gh}$ is equal to heat extract of evaporator.

$$\dot{Q}_{evap} = \dot{Q}_{gh} \tag{11}$$

b) The condenser and the Space Heating Load

$$\dot{Q}_{cond} = \dot{m}_w Cp_w (T_7 - T_8) \tag{12}$$

Where $\dot{Q}_{cond}$. is the heating capacity, and $(T_7 - T_8)$ is the water temperature difference between inlet and outlet of condenser, and the space heating load $\dot{Q}_{sh}$ is given by

$$\dot{Q}_{sh} = \rho_a \dot{v}_a Cp_a (T_{a,o} - T_{a,i}) \tag{13}$$

Where ρ_a is the density of the air, $\dot{v}_a$ is the volumetric flow rate of the air, cp_a is the specific heat of the air, and $(T_{a,o} - T_{a,i})$ is outlet and inlet air temperature difference through the fan coil.

c) The Power of the Compressor, Fan, and Pump

$$W_{comp,act} = I_{comp} V_{comp} \cos\phi \tag{14}$$

$$W_{fan,act} = I_{fan} V_{fan} \cos\phi \tag{15}$$

$$W_{pump,act} = I_{pump} V_{pump} \cos\phi \tag{16}$$

where $W_{comp,act}$ is the power of the compressor, $W_{fan,act}$ is the power of the fan, $W_{pump,act}$ is the power of the pump. I, V, and $\cos\phi$, are currents, voltages, power factor, respectively.

d) The Coefficient of Performance (COP) for the Unit and the Whole System.

The energy efficiency of the GSHP unit and the whole system can be defined as follows, respectively:

$$COP_{Hp,act} = \frac{\dot{Q}_{sh}}{W_{comp}} \tag{17}$$

and

$$COP_{system,act} = \frac{\dot{Q}_{sh}}{\sum W_{input}} \tag{18}$$

where $\sum W_{input}$ is the total power rate to the system, which is

$$\sum W_{input} = W_{comp,act} = W_{fan,act} = W_{pump,act} \tag{19}$$

11. RESULTS AND DISCUSSION

Figure (6) shows the variation from the undisturbed ground temperature for various depths. The thermal characteristic of the ground is that a few meters under the surface soil insulate the earth, minimizing the amplitude of the variation in soil temperature in comparison with the temperature in the air above the ground. Heat absorbed by the earth during the summer gets effectively used in the winter, the earth is warmer than the ambient air in the winter and cooler than the ambient air in the summer. This yearly continuous cycle between the air and the soil temperature results in a thermal energy potential that can be harnessed to help heat or cool a building. A Ground-Source Heat Pump (GSHP) transforms this earth energy into a useful energy to heat and cool

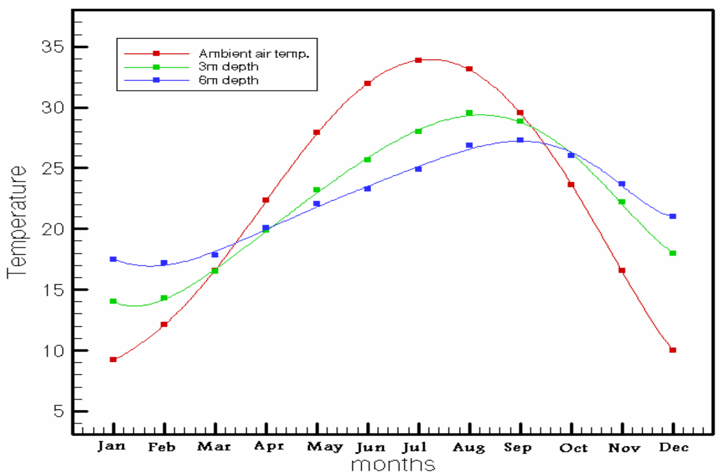


Figure (6) Annual air temperature variation and experimental ground temperature at different depths

Figure (7) shows the average $COP_{Hp,act}$ for each hour during day for hot season. The system operates under thermostatic control and $COP_{Hp,act}$ is nearly constant through the day due to the constant ground temperature. Figure (8) shows the coefficient of performance $COP_{Hp,act}$ on a daily basis for hot season. This figure shows less sensitivity to the ambient conditions. The effect is most noticeable in winter, when at critical times corresponding to cold snaps in the winter; the ground couple system maintains a more constant heating capacity. Sample data from the heating test of the system that verifies the best coefficient of performance (COP) are present in Table (3).

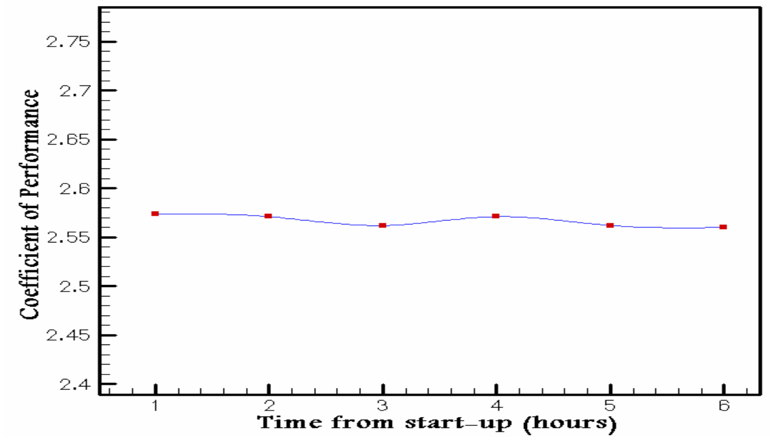


Figure (7) The Experimental Average Hourly Coefficient of Performance ($COP_{Hp,act}$) for One Day.

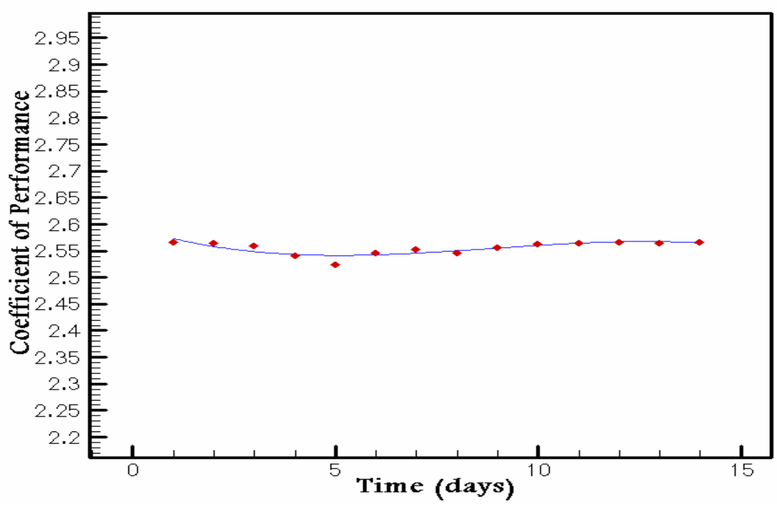


Figure (8): Daily Experimental Coefficient of Performance ($COP_{Hp,act}$) for14 Days .

Table (3) Calculated and Measured Parameters of the Experimental Performance of the System

Item	Value
Measured parameter	
Temperature of water at condenser inlet T_8 °C	30.3
Temperature of water at condenser outlet T_7 °C	35.8
Air temperature at fan coil inlet $T_{a,i}$ °C	20.8
Air temperature at fan coil outlet $T_{a,o}$ °C	25.9
Volumetric flow rate of air $\dot{v}_a$ m^3/s	0.08262
Calculated parameter	
Power input to compressor $W_{comp,act}$ kW	0.1971
Power input to water pump $W_{pump,act}$ kW	0.0422
Power input to fan $W_{fan,act}$ kW	0.0088
Total power input $\sum W_{input}$ kW	0.2481
Heating space $\dot{Q}_{sh}$ kW	0.5074
Heating COP of heat pump unit $COP_{Hp,act}$	2.574
Heating COP of heat pump whole system $COP_{system,act}$	2.045

a) Energetic Evaluation

The heating coefficient of performance of the heat pump for unit $COP_{Hp,act}$ and for the whole system $COP_{system,act}$, were calculated from equations (17) and (18), respectively, and found to be on average (2.574) and (2.045), respectively. By comparison, in a study performed by [Bi et al, 2004], the average heating COP of GSHP system with a heating load of 2298 W is to be 2.83. Also, by comparison with a study performed by [Hepbasli, 2005], the average heating load of 4270 W is to be 2.85. The values of $COP_{Hp,act}$ obtained from the present study agree with [Bi et al, 2004] and [Hepbasli, 2005]. The small difference is referred to the construction design of the rig.

b) Heat Extraction Rate

The key parameter for the ground heat exchanger layout is the specific performance, i.e. heat extraction rate in W / m of the borehole depth. During the hot season, the rate at which heat is extracted from the ground (ground heat exchanger load) was found to be in average (428W) from equation (10). This corresponds to a heat extraction rate of (59.03W/m) of the bore depth, which agrees with heat extraction rate (61.4 W/m) reported by [Hepbasli, 2005].

The heating capacity $\dot{Q}_{cond}$ of the heat pump system was obtained to be (535.6 W). The required borehole length of heating capacity was found to be (13.49) m/kW which agrees with an average value of 11.71 m/kW reported by [Hepbasli, 2005]. The difference between the two values is due to the nature of soil in Baghdad and in Turkey.

11. CONCLUSIONS

1- The values of coefficient of performance for the heat pump unit $COP_{Hp,act}$ and whole system $COP_{system,act}$ were obtained to be 2.574 and 2.054, respectively.

2- The required borehole length in meter per kW of heating capacity $\dot{Q}_{cond}$ was obtained to be (13.49) and the heat extraction rate $\dot{Q}_{evap}$ is (59.03) W/m of bore depth.

3- To obtain the optimum system COP must be adjusted the water flow rate of the ground heat exchanger.

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