Study And Experimental Analysis Of Horizontal Ground Source Heat Pump System

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Abstract - In modern life as we seen that the requirement of space heating and cooling is growing very fast. As a renewable energy technology, the ground source heat pump system plays an important role for the fulfill the requirement. The ground source heat pump system (GSHPs) also called as the geothermal heat pump system. Ground source heat pump system is highly efficient as compared to other conventional heat pumps. It absorbs heat from underground by pumping water through it. The heat pump then increases the temperature, and the heat is used to provide home heating. The pump needs electricity to run, but the idea is that it uses less electrical energy than the heat is produced. The heat pump performs the same role as a boiler does in a central heating system, but it uses ambient heat from the ground rather than burning fuel to generate heat. GSHP are receiving increasing interest because of their potential to reduce primary energy consumption and thus reduce emissions of greenhouse gases. This paper investigates the performance characteristics of Ground Source Heat Pump system, concentrating on thermodynamic analysis with R-22 as the refrigerant for a heating mode. The main purpose of this paper is to study the energetic potential of the deployment in Dhanbad, India of the Ground Source Heat Pump (GSHP) system for heating mode application. Therefore, a GSHP system using horizontal Ground Heat Exchanger (GHE) has been installed and experimented in BIT Sindri, Dhanbad, India. In the present study based upon the measurements made in the heating mode from the 5th of December till 10th of March 2016-2017, the heat extraction rate from the soil has been found to be, on an average, 2.53 KW. The heating coefficient of performance of the heat pump $(COP)_{HP}$ fluctuates between 2.11 to 3.13. The result shows that overall COP values for the complete system are 5–20% lower than the COP of heat pump. It may be concluded that these performance parameters obtained from the present study are fairly close to those reported by Bi et al. (2004).It is expected that the model present here would be beneficial to everyone dealing with the design, simulation and testing of GSHP systems in future.

Key words: Renewable energy, heat pump, geothermal, GSHP, ground heat exchanger, R22, Space heating, COP.

1.Introduction

As we seen that dependency of our society on fossil fuel such as oil, coal natural gas. These resources are limited, these are created by natural process over millions of years. Burning them to produce energy results in emissions of 'greenhouse gases', including carbon dioxide (CO_2). These gases trap solar radiation in the earths atmosphere and cause undesirable changes in the climate. So our conventional energy sources which are limited in quantity are decreasing continuously and arising a very serious problem and climate change is real threat to our future and also the fossil fuel which we uses in our home are responsible for the emission of carbon dioxide (CO_2) which also contributes to the climate change . Renewable and sustainable energy offers a viable and potent solution to counter the effects of this problem and effects of global warming. By installing any one of the renewable energy technologies, one will be making a major personal contribution to the well being of future generations and could also benefit from lower fuel bills. So, if we describing the options for heating and cooling our house or looking for ways to reduce our energy bills, we may be considering ground-source heat pumps (GSHPs).

The use of ground-source heat pumps (GSHPs) in commercial and residential facilities is one of the best example. The ground source heat pump also known as 'the geothermal heat pump, is a highly efficient renewable source of energy. This technology is widely accepted for both residential as well as commercial buildings. Ground source heat pumps are used for space heating and cooling, as well as water heating. Its great advantage is that it works by concentrating naturally existing heat in the earth, rather than by producing heat through combustion of fossil fuels. Ground source heat pump does not utilizes burning fossil fuel to create heat like furnaces etc. The technology relies on the fact that the Earth (beneath the surface) remains at a relatively constant temperature throughout the year, warmer than the air above it during the winter and cooler in the summer, very much like a cave (Omer, 2008). The Ground source heat pump takes advantage of this by transferring heat stored in the Earth or in ground water into a building during the winter, and transferring it out of the building and back into the ground during the summer or in other words, the ground acts as a heat source in winter and a heat sink in summer.

1.1 Advantages And Disadvantages of GSHPs.

Ground-source heat pumps system is more attractive alternative than conventional air source heat pumps systems. GSHPs have several advantages over air source heat pumps, which are:

- They consume less energy to operate.
- They tap the groundwater which is a more stable energy source than air.
- When the outside temperature is very low they do not require extra energy.
- They use less amount of refrigerant,
- The design is simple and require less maintenance and
- They do not require the unit to be located where it is exposed to weathering
- GSHPs can reduce GHG emissions by 66% or more compared with conventional heating and cooling systems that use fossil fuels.
- EESs use up to 75% less electricity than conventional heating or cooling systems.
- Maintenance costs for this type of technology can be cut in half and operating cost
- reduced one-quarter of that of a conventional system

The main disadvantage is the higher initial capital cost, being about 30–50% more expensive than air source units. This is due to the extra expense and effort to bury heat exchangers in the earth or providing a well for the energy sources. However, once installed, the annual cost is less over the life of the system, resulting in net savings (Madani, 2011). This thesis makes an attempt to investigate the performance evaluation and thermodynamic analysis of GSHP system. An experimental set-up, described here, has been fabricated and tested for the first time on the basis of an academic study performed in BIT Sindri, Dhanbad Jharkhand, India, It consists of an experimental setup of GSHP, with R22 as a refrigerant for heating mode application. The experimental setup mainly includes the ground temperature, the temperature and flow rate of water circulating in GHE and the power consumption of heat pump and circulating pumps. These experimental data are essentially used to evaluate the coefficient of performance of the heat pump and the overall system for continuous operation mode.

1.2 Objective of the present work

The main objective of our present work is to prepare an experimental set up for GSHP for a given control volume. Following objectives have been defined.

- Fabrication of Ground heat exchanger for utilization of geothermal energy.
- Assembling of evaporator for taking heat from circulating water by refrigerant.
- To assess the energy, environmental and economical impact of the system compared to the conventional ASHP systems.
- To determine the energy saving potential of GSHP system for Sindri Dhanbad weather conditions.
- To improve the understanding of geothermal behaviour in the context of climatic conditions of Sindri Dhanbad.
- Different parameters associated with the experiment have to be analysed and variation of COP and heat extraction rate has be depicted with hourly variation over the months of December-February 2016-2017.

2 Description of the experimental set-up:

The aim of the experiments is to investigate the thermal performance analysis of a ground source heat pump. An experimental set up has been designed and fabricated. In this chapter, an experimental study on the feasibility of a ground source heat pump has been carried out. The heating mode of operation of a GSHP is analyzed for the climatic conditions of Sindri, Dhanbad. Coefficients of performance and heating capacity have also been studied in this chapter.
The schematic set-up are illustrated in Fig. 2.1.

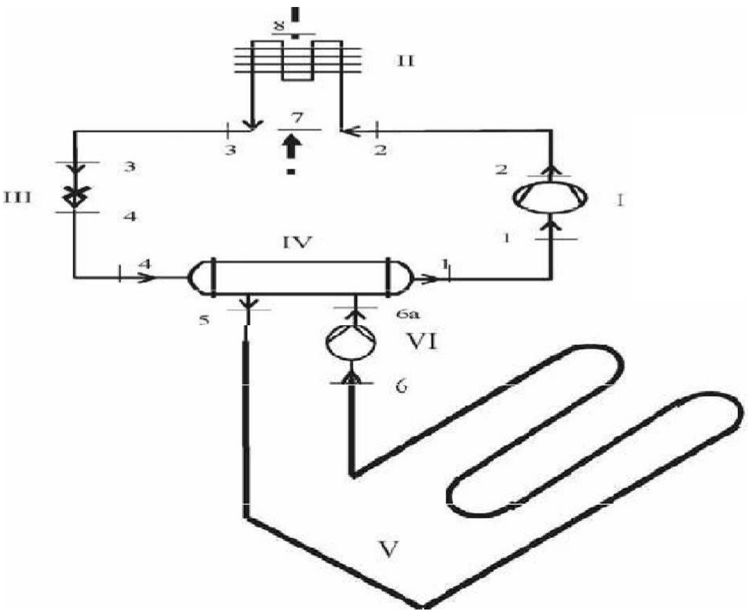


Figure 2.1: Schematic of Experimental set-up

3. Experimental investigations:

In this study, the performance of the system designed under the conditions of Sindri, Dhanbad is analyzed using the measured data. The experiments were conducted on the ground source heat pump system under steady state conditions in the heating mode. In order to analyze the performance of the system under climatic conditions of Sindri, daily average values of 12 measurements between the months December 2016 to February 2017 for the hours between 09:30 to 12:00, with an interval of 15 minutes are used in the calculations.

3.1 Mathematical Model

The present work has been planned towards studying the performance of a ground-source heat pump coupled to a model-sized test room to meet its heating requirements.

The following assumptions are adopted for the mathematical modeling.

a. All processes are steady state and steady flow with negligible potential and kinetic energy effects and no chemical or nuclear reactions.

b. The directions of heat transfer to the system and work transfer from the system are positive.

c. Heat transfer and refrigerant pressure drops in the tubing connecting the components are ignored, since their lengths are short.

d. The compressor operation has an adiabatic efficiency of 80%.

e. The compressor mechanical and the compressor motor electrical efficiencies are 70% and 72%, respectively.

f. Air is an ideal gas with a constant specific heat, and its humidity content is negligible.

g. The power inputs to the fan coil fans and the circulating pumps are negligible compared with the power input to the compressor.

h. The dead state of the refrigerant is taken at ambient temperature, *To* 293.15k and at a pressure, 0 *P* 100 kPa. equal to the saturation pressure of the refrigerant at 0 *T*.

In order to carry out design of GSHP system, experimental data reduction, result analysis, mathematical model and parametric analysis has been described in this section. Under the aforementioned assumptions, the heat input, the rate of exergy decrease, the rate of for a general steady state, steady flow process, the four balance equations are applied to find out the work output irreversibility and the energy and exergy efficiencies.

The mass balance equation can be expressed in rate form as;

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \qquad (3.1)$$

Where $\dot{m}$ the mass flow rate, and the subscript in stands for inlet and out for outlet.

The general energy balance can be expressed as;

$$\dot{E}_{in} = \dot{E}_{out} \qquad (3.2)$$

(Rate of energy transfer in by heat,work and mass) = (Rate of energy transfer out by heat,work and mass)

The general energy balance equation can be written more explicitly as;

$$\dot{Q} + \sum \dot{m}_{in}\, \dot{h}_{in} = \dot{W} + \sum \dot{m}_{out}\, \dot{h}_{out} \qquad (3.3)$$

Where $\dot{Q} = \dot{Q}_{net,in} = \dot{Q}_{in} - \dot{Q}_{out}$ is the rate of net heat input and $\dot{W} = \dot{W}_{net,out} = \dot{W}_{in} - \dot{W}_{out}$ is the rate of net work output, h is the enthalpy per unit mass.

The rate of heat extracted (absorbed) by the unit in the heating mode (the ground heat exchanger load), $\dot{Q}_{e}$ is calculated from the following equation;

$$\dot{Q}_{e} = \dot{m}_{wa}\, C_{p,wa}\, (T_{out,wa} - T_{in,wa}) \qquad (3.4)$$

Where $C_{p,wa}$ is the specific heat of the water–antifreeze solution, $\dot{m}_{wa}$ is the mass flow rate of the water/antifreeze solution and $T_{out,wa}$ and $T_{in,wa}$ is the temperature difference between the outlet and inlet of the GHE.

The heat rejection rate in the condenser may be calculated by;

$$\dot{Q}_{cond} = \dot{m}_{ref}\,(h_2 - h_3) \qquad (3.5)$$

The heat transfer rate in the evaporator may be calculated by;

$$\dot{Q}_{evap} = \dot{m}_{ref}\,(h_1 - h_4) \qquad (3.6)$$

The work input rate to the compressor may be defined as;

$$\dot{W}_{comp} = \frac{\dot{m}_{ref}(h_{2s}-h_1)}{\eta_{i,comp}} \quad (3.7)$$

In case the mass flow rate on the refrigerant side is not measured, the space heating load $\dot{Q}_{sl}$
may be estimated as;

$$\dot{Q}_{sl} = \dot{m}_{air} C_{p,air}(T_{out,air} - T_{in,air}) \quad (3.8)$$

Where $\dot{m}_{air}$ is the mass flow rate of air $C_{p,air}$ is the specific heat of air, $T_{in,air}$ and $T_{out,air}$ are the average air temperatures entering and leaving the fan-coil units, respectively.

For an ideal heat pump system operating between the low and high temperature reservoirs at T_L and T_H, respectively, the maximum heating coefficient of performance, $COP_{cornot,HP}$ is obtained from the Carnot cycle as;

$$COP_{cornot,HP} = \frac{T_H}{T_H - T_L} \quad (3.9)$$

The COP of the heat pump can be calculated as;

$$COP_{HP} = \frac{\dot{Q}_{cond}}{W_{comp}} \quad (3.10)$$

The coefficient of performance of the overall heating system (COP_{sys}), which is the ratio of the condenser load to total work consumptions of the compressor, the pumps (brine and water circulation pumps), and the fan-coil unit (or the condenser fan), may be computed by the following equation;

$$COP_{sys} = \frac{\dot{Q}_{cond}}{W_{comp} + W_{pump} + W_{fc}} \quad (3.11)$$

The earth connection is where heat transfer between the heat pump system and the soil occurs. Refrigerant as a heat-transfer fluid is circulated from the pump, around the tubing, and back to the pump in a closed loop. High density polyethylene (HDPE) pipe material is used as the coil for the ground heat exchanger, which has been used to extract heat from soil. So the length of ground heat exchanger can be determined by following equations, Length of ground heat exchanger (L);

$$L = m_{wa} C_{p,wa} R_{total} \ln\left(\frac{T_{in,wa}}{T_{out,wa}}\right) \quad (3.12)$$

Where, *R Total* is the total thermal resistance and *R Total* ��*Rcon* ��*Rpipe* ��*Rsoil*

Thermal resistance due to convective heat transfer for flow of water in the pipe;

$$R_{con} = \frac{1}{\pi D_i h_{wa}} \quad (3.13)$$

Thermal resistance due to conduction through pipe thickness;

$$R_{pipe} = \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi K_{pipe}} \quad (3.14)$$

Thermal resistance due to conduction in soil;

$$R_{soil} = \frac{1}{S K_{soil}} \quad (3.15)$$

Where, S is the conduction shape factor of the pipe expressed as;

$$S = \frac{1}{\ln\left[\left(\frac{2d}{D_o}\right) + \sqrt{\left(\frac{2d}{D_o}\right)^2 - 1}\right]} \quad (3.16)$$

The sample calculation for the experimental purpose has been done in the section 4.2 for the Data set 1 of 5th December 2016.

3.2 data set 1 (5th December 2016):

The analysis for the designed system has been started from 5th December 2016. For this purpose experimental readings have been taken using hygro thermometer, laboratory thermometer, pressure gauge and energy meter under the climatic conditions of Sindri, Dhanbad. The various parameters used are shown in Table 3.2.1 and the unknown properties of the system are calculated using the equations The mean values of the measured data on 5th December 2016 are given in Table below.

Table 3.2.1: Average measured parameters used in experiment on 5th December 2016

S.N	PARAMETERS	VALUE
01	Temperature of water/antifreeze in GHX at inlet (°C)	10
02	Temperature of water/antifreeze in GHX at outlet (°C)	18
03	Outdoor air temperature (°C)	21
04	Indoor air temperature (°C)	28
05	Pressure at inlet of compressor (bar)	1.4
06	Pressure at outlet of compressor (bar)	12.78
07	Compressor electric current (A)	6.63
08	Brine circulating pump electric current (A)	1.84
09	Condenser fan electric current (A)	1.1
10	Two-phase voltage (V)	220
11	Frequency (Hz)	50
12	Power input to the compressor (kW)	1.12
13	Power input to the brine circulating pump (kW)	0.373
14	Power input to the condenser fan (kW)	0.24

Table 3.2.2: Variation of various temperature of GHE with Time of day.

Time of Day	Water Temperature at the inlet of GHE (C)	Water Temperature at the Outlet of GHE (C)
9:30	14	20
10:00	13	19
10:30	10	17
11:00	11	15
11:30	9	14
12:00	8	14

Figure 3.2.1: Variation of various temperature of GHE with Time of day for 5th December 2016

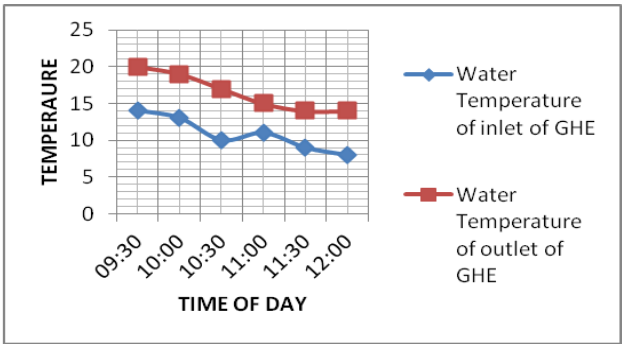
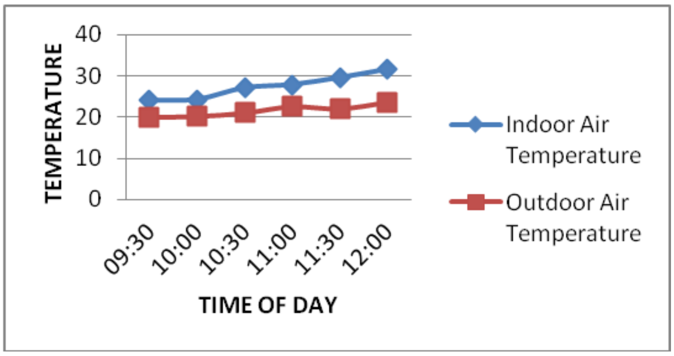


Table 3.2.3: Variation of various temperature of air with Time of day

Time of Day	Indoor Air Temperature (C)	Outdoor Air temperature (C)
9:30	24	20
10:00	24.2	20.2
10:30	27	21
11:00	27.8	22.5
11:30	29.5	22
12:00	31.5	23.5

Figure 3.2.2: Variation of various temperature of air with Time of day for 5th December 2016



3.3 Sample Calculation.

These are the following assumptions were made:

(i) The volumetric efficiency of the compressor was taken to be 85%,
(ii) Compressor isentropic efficiency was taken to be 80%,
(iii) There were no pressure losses in the cycle,

In order to calculate the coefficient of performance of heat pump and system the pressures, temperatures, and required parameters are taken from reading which done on given day. The refrigerating properties from refrigeration charts are taken. For our required results some other properties have been calculated which are enlisted below. The sample calculation has been given below is for 5th December 2016 of average readings after getting steady state (Data set 1, Table: 3.2.1)

Fluid properties:

Given data
Pressure at inlet of compressor (P_1) =1.4 *bar*
Pressure at outlet of compressor (P_2) = 12.78 *bar*

Now from refrigeration table at given data (P_1) we get, (Refrigeration table by S C Jain)
For R-22.
Temperature at inlet of compressor (T_1) = -32 °C
Enthalpy at inlet of compressor (h_{g1}) 391.79 = *KJ Kg*
Entropy at inlet of compressor (s_{g1}) 1.80929 = *KJ Kg K*

Now from refrigeration table at (P_2) we get, (Refrigeration table by S C Jain)

Saturation temperature at outlet of compressor ($T_{2,sat}$) =33 °*C*
Entropy of dry saturated vapour at outlet of compressor ($s_{2,sat}$) =1.70826 *KJ kg K*
Enthalpy of dry saturated vapour at outlet of compressor ($h_{2,sat}$)= 415.207 *KJ Kg*
Enthalpy at outlet of condenser (h_3) = 240.520 *KJ / Kg*

3.3.1 Actual enthalpy at outlet of compressor.

First of all, let us find the temperature (T_2) of superheated vapour at point 2. We know that in isentropic process entropy remains constant, that is

Entropy at inlet of compressor (s_{g1}) = Entropy at outlet of compressor (s_2)

Therefore it can be written in equation form as;

$$S_{g1} = S_{2,sat} + c_p \ln\left(\frac{T_2}{T_{2,sat}}\right)$$

$1.80912=1.70826+1.03 \ l_n(T_2/306)$
$T_2= 337.536$ K

We know that enthalpy at point 2,
$h_2 = h_{2,sat} + c_{pv}(T_2 - T_{2,sat}) = 415.207 + 1.03(337.536-306) = 447.68$ KJ/Kg.

As we know that isentropic efficiency of compressor (η_c) = 85 %(given)
Now using the equation for isentropic efficiency of compressor, the actual enthalpy at compressor outlet can be calculated as:

$$h_{2,act} = \frac{(h_2 - h_1)}{\eta_c} + h_1 = \left(\frac{447.68-391.79}{0.85}\right) + 391.79 = 457.54 \frac{KJ}{Kg}.$$

3.3.2. Heat rejection rate in condenser.

For the calculation of heat rejection rate in condenser, the mass flow rate of the refrigerant should be known. for this we use the following equation.

$$\dot{m} = \frac{W_{comp}}{(h_{2,act} - h_1)} = \frac{1.12}{457.54-391.79} = 0.017 \frac{Kg}{s}.$$

Now from equation (4.5) we may calculate the heat that rejected by condenser to test room

$$Q_{cond} = \dot{m}(h_{2,act} - h_3) = 0.017(457.54 - 240.520) = 369\ KW.$$

3.3.3. Heat transfer rate in evaporator.

The amount of heat that should be transfer in evaporator is calculated by equation (3.6). The heat transfer in evaporator is the amount of heat that extracted by refrigerant from water/antifreeze solution in the evaporator. Therefore, evaporating heat is calculated as,

$$Q_{evap} = \dot{m}\,(h_1 - h_{f4}) = 0.017(391.79 - 240.52) = 2.57 KW.$$

3.3.4. Coefficient of performance of heat pump.

The COP of heat pump may be calculated from equation (4.10), It is the measure of performance of heat pump excluding circulating pump and fan coil unit. Therefore COP is given as,

$$COP_{HP} = \frac{Q_{cond}}{Q_{comp}} = \frac{3.69}{1.12} = 3.29$$

3.3.5. Coefficient of performance of the system.

The COP of the system may be calculated by using equation (3.11).It is the measure of performance of overall system including heat pump, water/antifreeze circulating pump, forced fan. Therefore COP of the system is given as;

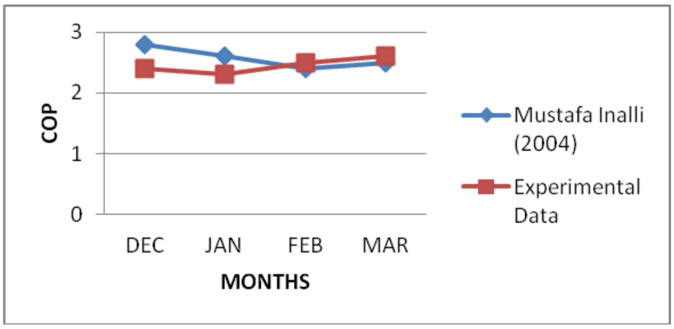
$$COP_{sys} = \frac{Q_{cond}}{W_{comp}+W_{pump}+W_{fc}} = \frac{3.39}{1.1+0.373+0.24} = 2.15$$

4. Results and discussion

4.1 Validation of experimental result

Average best result of COP obtained from the experiments has been validated with the results of Inalli (2004) which shown in Fig. 4.1. the performance of the heat pump unit is low which may be due to a poor design of the system compared to the values given by inalli (2004). But the experimental results are more in the month of February and March due to the effect of local climate. The experimental results are found to be well within the permissible limit as compared to the inalli (2004).

Figure 4.1 validation of experimental result with inalli (2004)



4.2: Average performance parameters

The coefficient of performance of heat pump and system, heat extraction capacity, the heating capacity are the key parameters of this experiment. In this experiment these parameters have been calculated by the equations as explained previously, for this several readings have been taken. Some average experimental data and results are also given here and graphs have been plotted.

Fig 4.2 illustrates the date wise variation of the COPs. It shows a decrease in COP of system and heat pump until 5th January, and then it increases for GHE. The mean values of COPsys and COP_{hp} for GHE are obtained to be 2.2 and 3.11, respectively. The highest COP of the system and COP of heat pump for GHE is found 2.67 and 3.8 in the month of February, while the lowest heating coefficient of the system and coefficient of performance of the heat pump for GHE is found 1.88 and 3.04 in the month of January.

Fig.4.3 depicts the date wise variation of the space-heating rate and heat extraction rate of ground source heat pump system. The mean heat rates rejected by the condenser unit to the room in the heating season for GHE found to be 3.5 kW. The lowest heating capacity for the GHE obtained as 3.2 kW on 5^{th} January while the highest heating capacity for GHE 3.971 kW on 5^{th} of December. Also the mean heat extracted from ground is the heat extracted throught the evaporator is found to be 2.7 kW. The lowest heat extracted on 20th December while highest extraction is found on 15th February through ground heat exchange.

Figure 4.2: variation of various COP with date of experiment

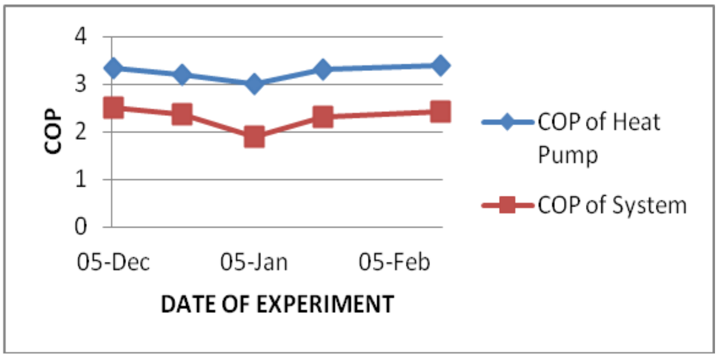
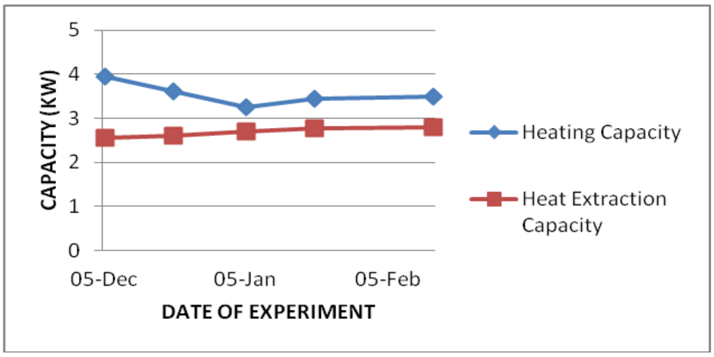


Figure 4.3: variation of various heating capacity with date of experiment



5: Conclusion And Future Scope

An experimental system was installed for investigating the thermal performance of a GSHPS for heating mode. The GSHP systems offer some proven advantages over conventional heating and cooling systems, particularly in terms of efficiency, maintenance costs, and overall operating costs.

5.1: Conclusion

The main conclusions that can be drawn from the present study are listed below:

(a) For the heat pump unit, the values for COP_{HP} has been found to vary from 2.54 to 2.95 while those for overall COP were approximately 5–20 % lower than $COP_{HP.}$

(b) For the circulating pump, the pumping water-antifreeze flow rate was found to be $0.185m^3/h/kW$ of heating capacity. Kavanaugh (1992) suggests that the optimum pumping rates for the circulating pump should range from 0.162 to $0.192m^3/h$ per kW of heating capacity. It may be concluded that the pump selected falls into the acceptable limits.

(c) The relevant soil properties are to be precisely measured before attempting the design of the GHE. Therefore, care may be taken in the design and construction of a ground loop for a heat pump application to ensure long ground loop life and reduce the installation costs.

5.2: Future Scope

The present work experimentally investigated the performance of GSHP system of heating mode for the climatic conditions of Sindri, Dhanbad. In his research we have been discovered many operating and designing parameters, but still there are mamy other issues that may be investigated. Recommended future studies are as follows:

(a) Experimental comparison of various GSHP systems for different range of temperatures and different air flow rates.
(b) Experimental and numerical investigation of performance of GSHP at different operating parameters like low humidity and for high temperature range
(c) Numerical investigation of GSHP system for heat and mass transfer model to optimize the design and operating parameters of ground heat exchanger can also be carried out.
(d) Experimental and numerical investigation of GSHP system by integrating different types of ground heat exchanger in the Indian climatic conditions.

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