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**Analysis on Potential of Energy Demand Reduction for Heating and Cooling
with Underground U Type Heat Exchanger
(A Case Study in Commercial Building in Pulchowk)**

by

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ABSTRACT

In the world, there is an aim to promote sustainable energy thinking and implementation. In commercial buildings with significant cooling and heating loads, it is critical to consider long-term solutions in the design phase that help to reduce energy usage. Recently, the variable refrigerant flow has a high Coefficient of Performance (COP) in the heating, ventilation, and air conditioning area, which has made enormous progress in demand for energy reduction. However, the conventional commercial HVAC system has a lower COP than the ground source heat pump system (GSHP), so this study presents a heating and cooling load calculation and details analysis of system design and financial Analysis of GSHP to existing commercial VRF systems.

In this study heating load for the winter season and the cooling load for the summer season of the commercial building are calculated and also the required bore length for the GSHP system. It is found that the cooling load for the project is almost 2.67 times the heating load and the required bore length for cooling is 1.5 times higher than the heating length.

For designing the system ground floor is taken into consideration and the Samsung DVMS water system is selected for GSHP looping. So an outdoor unit of 20HP is selected and an indoor unit 2units of 8TR and one unit of 4TR indoor unit duct able type is selected.

The initial investment is high for the GSHP system as compared to VRF and other split and VRF systems which is almost double of VRF systems. Because of the high expense of drilling and installing the looping system, the cost is significant. However, because COP is more valuable in a long-running hour, it has a shorter payback period and a higher internal rate of return.

For the fluid heat transfer to the ground from the ANSYS software it is found that in heating conditions there is a change of temperature by around 9°C and for cooling conditions, there is a change of temperature around 14°C .

As a result, there is a noticeable temperature difference, resulting in improved system performance. Under this study, financial analysis is carried out in two scenarios: 18 hours per day and 24 hours per day. In scenario 1, there is an energy demand reduction by 35307KWhr annually and in scenario 2, there is an annual saving of energy 47076KWhr annually. For scenario 1, the payback period is of 12.3years with an

internal rate of return of 14% and for scenario 2, the payback period is 7.7 years with an internal rate of return of 20%.

So the above result shows GSHP is feasible for the project having high cooling and heating load having a high running hour. The findings reveal that GSHP offers better potential for energy savings and greenhouse gas reduction.

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List of Acronyms and Abbreviations

AC:	: Air-Conditioning
ASHRAE:	: American Society of Heating, Refrigerating and Air-Conditioning Engineers
BTU	: British Thermal Unit
GHGS	: Green House Gas
GSHP	: Ground Source Heat Pump
GWth	: Gigawatts-Thermal
HAP	: Hourly Analysis Programme
HVAC	: Heating Ventilation and Air Condition
NRREP	: National Rural Energy Programme
R& D	: Research and Development
DF	: Discounting Factor
NPV	: Net Present Value
IRR	: Internal Rate of Return
PBP	: Pay Back Period
TR	: Ton of Refrigeration
IRR	: Internal Rate of Return
PBP	: Pay Back Period
VRF	: Variable Refrigerant Flow
QC	: Quality Control
MW:	: Mega Watt

CHAPTER ONE: INTRODUCTION

1.1 Background

Globally, there is an enormous increase in energy usage patterns, which has increased and resulted in energy supply issues, concerns, and other environmental issues. Since global warming has become one of the most challenging issues, we need to find the most efficient means of consuming energy. Also, it has impacted the pattern of global energy consumption, with the use of non-renewable resources to meet this need being a major contributor to global warming.

Buildings and home functions have gradually grown their contribution to energy use, leading key sectors like transportation and industry (Pérez-Lombard et al., 2008).

The economic strategy for long-term growth, as the largest energy consumer, buildings must undoubtedly promote efficiency and reasonable energy use. (Sarbu et al., 2014). It is believed that the hugest and most opportunity for cost-saving in terms of the energy sector is buildings. Furthermore, research has shown that energy-saving strategies are the most cost-effective way to minimize GHG emissions. Currently, heat accounts for over 80% of the energy demand for space heating and cooling in homes and utility buildings. (Sarbu et al., 2014). For cooling purposes, the demand for it is leaping each year.

Population growth, comfort levels, rising demand for building services, time spent inside buildings, etc. ensure that energy demand will continue to rise in the future which has enhanced the importance of building efficiency to include in policy at the national and international levels (Pérez-Lombard et al., 2008).

Geothermal energy is renewable energy that is created underground and can be utilized for heating or converted into electricity for many uses (Donatini, 2019). Geothermal power has huge growth potential in the world. Heat up to 10000 meters of the earth's surface is thought to contain 50000 times more energy than all of the world's oil and gas reserves (Shere, 2013). Improved energy efficiency with the application of renewable energy plays a major role in the existing building for heating and cooling purpose. Geothermal energy can be collected whenever it is available and used whenever it is needed, even in a different season. For instance, geothermal heat exchange can be done in different seasons for heating and cooling whenever needed.

Geothermal energy is particularly important because it is a renewable and clean energy practice and results in the reduction of energy demand.

Ground Source Heat Pump (GSHP) systems employ the ground as a heat source as well as a heat sink for space heating and cooling, and they transfer heat with the subsurface environment to offer cooling and heating for a growing variety of applications (Bose et al., 2002). The GSHP technology can offer higher efficiency for air-conditioning systems (AC) because the underground environment provides higher temperatures for heating and cooling and experience fewer temperature fluctuations than ambient air temperature change. All around the world, GSHP systems have been used in residential and commercial buildings. It is estimated that the global installation of GSHP systems has increased steadily in recent years, with annual increases ranging from 10% to 30%. These systems have a wide range of uses, have profited from worldwide collaboration, and are being acknowledged as a cost-effective energy conservation standard (Bose et al., 2002).

As a result, a rigorous investigation into the usage of geothermal energy for heating and cooling is required.

1.2 Problem Statement

Energy consumption for heating and cooling is increasing. Due to the increasing demand for commercial and residential fuels and electricity is increasing. In the current scenario in Nepal, heating, and cooling is done in the traditional way and HVAC system which are producing Green House gas and depleting the environment. In the present context, energy saving is the most financial energy alternative. So the implementation of energy-efficient technology with geothermal heating and cooling could be the best option to cope with the current situation. Geothermal energy will be green and reduce energy demand and the emission of greenhouse gases. So we need to analyze and research the potential of energy demand reduction for heating and cooling purposes in domestic applications using geothermal energy with a case study in the context of Nepal. There may be various potential for energy demand reduction for heating and cooling which need to be studied carefully. So further study and research on geothermal are needed to evaluate the end uses of commercialization and Quality Control (QC) mechanism in the case of Nepal.

1.3 Objectives

1.3.1 Main Objectives

- i. To analyze the potential of energy demand reduction for heating and cooling with an underground U type heat exchanger

1.3.2 Specific Objectives

- i. To calculate the building's heating and cooling load
- ii. To analyze the U Type of Heat exchange, calculate the effective length and diameter
- iii. To design system and modeling of heating and cooling system.
- iv. To calculate cost-benefit analysis with the existing system

1.4 Scope

The contributions of this study are:

- i. To identify the major heat transfer medium to the space.
- ii. To identify the potential of heat transfer reduction to the building
- iii. To develop the system for geothermal heating and cooling to the building.
- iv. To identify the potential of energy demand reduction with Geothermal Heating and Cooling
- v. To analyze the cost-benefit analysis with the system

1.5 Hypothesis of the Project

The following hypothesis will form the basis of this project's work to achieve the objectives.

- i. The maximum heat transfer medium and heat transfer reduction method will be identified to minimize heating and cooling load.
- ii. The effective bore length and system size will be developed.
- iii. Reduce energy demand in the building.
- iv. This system will reduce the energy cost of the building.

1.6 Limitation

During the calculation of space heating and cooling load in the building wattage of light is taken as per required from design and electrical appliances are taken reference similar to as of general office. In process of calculation required vertical bore length for heating and cooling, the ground temperature is taken reference value. In this heating and cooling system for heat transfer in-ground as a condensing unit, only a single U-type heat exchanger is implemented. Instead of TRNSYS, the ANSYS simulation program is used to analyze the heat transfer to the ground.

CHAPTER TWO: LITERATURE REVIEW

2.1 Geothermal Energy

Geothermal energy is the heat energy in the earth's crust or liquid that fills the holes and cracks in the rock, sand, and gravel. Calculations and studies reveal that the earth, which began in a molten state, would have cooled and solidified many thousands of years ago without any further energy input than that provided by the sun. Radioactive decay inside the ground is supposed to be the most important source of geothermal energy (Bullard, 1973).

Energy is accumulated at high temperatures near the earth's surface due to plate motion or volcanic activities. In locations where geological circumstances and the existence of water permit, conduction and convection are also used to transport energy from deeper layers of the crust to the surface. Geothermal energy may be extracted in one of two ways: liquid-dominated or vapor-dominated. Since there is variation in radioactive decay, volcanic activity, fluid circulation rock conductivities, etc., varied areas have various heat fluxes (from the crust to the surface) and temperatures at different depths. The temperature rises roughly 13.7°F per 1000 feet of depth on average, and gradients ranging from 5 to 27°F per 1000 ft are frequent. It is accepted that the most attractive and feasible economic resources are those having higher-than-average temperature gradients and/or heat flow rates. However, if specific geological characteristics are present, places with normal slopes may be important resources (ASHRAE Technical Committee, 2015).

The supplies of geothermal energy, their obtaining or recovery may differ in a significant way. Geothermal energy exists in a variety of subterranean habitats, and accessing it might necessitate a variety of procedures and technologies for that procedure before the resource can be retrieved for production (Miller, 2006).

2.2 The Geothermal Potential Energy in the World

Geothermal energy is produced when heat from deep below the ground is transmitted to water pockets in the form of hot springs. Such a source may not naturally occur on the earth's surface, necessitating drilling to obtain hot water for use in a variety of applications. People have known about hot water for thousands of years and have used it for heating, cleaning, agriculture, aquaculture, industry, and other various purposes. Heat and thermal energy that is produced and saved inside the earth is known as

geothermal energy. The geothermal energy of the earth is supposed to arrive from the planet's initial creation (20%) and the radioactive decay of minerals (80 %)(Turcotte, D.L. & Shubert G., 2002).

Since geothermal power can generate electricity its necessity and demand are rising day by day. Around 8000 MW (In 1999) of energy was produced throughout the world (in twenty-two countries), with the majority of countries being developing ones. Also, in the same year, the US produced approximately 3000 MW of geothermal energy, more than twice as much as wind and solar combined. In addition, geothermal power plants provide about 6% of the electricity in Utah and 10% of the electricity in Nevada (Encyclopedia.com, 2018).

The temperature of a geothermal resource determines its particular applications. The highest temperature from the geothermal resources (150°C or higher) is often used to generate electricity or power. From the low geothermal temperature resources, we can directly use them in applications in agricultural uses, commercial and residential structures, industrial operations, and recreation activities. GHPs can effectively provide heating and cooling effects in shallow-earth environments with generally stable ground temperatures (Miller, 2006).

From 1999, starting with 28 nations, countries using geothermal energy directly increased to 58 countries in 2000, 72 countries in 2005, and 78 countries in 2010. As per the information to the World Geothermal Congress Indonesia (Bali) in 2010 used thermal plant for direct usage in the latter months of 2009 was 48,493 MW, nearly a 72 percent growth over 2005 statistics, rising at the compound at 11.4 percent yearly with a capacity factor of 0.28. GSHP account for 47.2 percent of total thermal energy used, with 25.8% going to bathing and swimming, 14.9 percent going to space heating (of which 85 percent is district heating), 5.5 percent going to greenhouses and open ground heating, 2.8 percent going to industrial process heating, 2.7 percent going to aquaculture pond and raceway heating, and 0.4 percent going to agricultural drying., and 0.5 percent for snow melting. 250 million barrels (38 million tonnes) of the nonrenewable source of energy i.e. oil were saved (Lund & Toth, 2021).

In 2020, the global capacity for geothermal generation for daily heating purposes – is expected to expand by 2.4 gigawatts-thermal (GWth) (approximately 8%), reaching 32 GWth. (REN21, 2021).

Geothermal resources are used for energy in two ways (similar to solar and bioenergy), either as a source of electricity or as a source of heat for different "direct-use" thermal applications, such as industrial and building heating (REN21, 2021). Geothermal heating and power generating systems have the advantage of being available all day in a year with only scheduled maintenance (Insight et al., 1982).

2.3 The Geothermal Potential Energy in Nepal

The National Rural Energy Initiative (NRREP) was established in 2012 as a five-year program to provide renewable energy (wind energy) subsidies rather than doing research and development. However, geothermal energy has long been overlooked, which may be due to a lack of road access to geothermal locations, as well as skilled staff, financial resources, and economic feasibility for power generation or other purposes, among other factors. (Ranjit, 2015).

By 2003, thirty-two hot water springs sites are situated using geothermal with a water temperature up to 50° Centigrade (Centre for Energy Studies, 2012). These hot springs are utilized for medicinal purposes, and further research is needed to find out what else these sources may be used for (Shrestha et al., 2003). The geothermal energy in Nepal is mainly extracted in the form of hot water which is mainly used for cleaning or bathing purposes (hot springs and bath ponds) only. Research by Ranjit in 2010 has suggested that the Sadhu Khola – Jomsom area in Nepal's central region has a large geothermal reservoir (Ranjit, 2010). Without some pilot-scale demonstration project, the concept of using low-temperature water for a purpose other than bathing cannot be utilized (Raza et al., 2011).

2.4 Geothermal Heat Pump

The supplies of geothermal energy and their recovery process may widely vary. Since the availability of this energy also varies in different locations, and habitats, we have to develop a wise method, procedure, equipment, and technology to make it useful for the purpose. Heat energy collected within commercial buildings and stored seasonally in the land during the summer cooling procedure is utilized to keep buildings warm in the winter at very low electricity costs. In addition, because the temperature of the underlying soil is lower than the surrounding air in the summer, GHPs cool buildings more efficiently in comparison to conventional air conditioners (Liu et al., 2019).

At a depth of 30 feet, temperatures stay generally stable, ranging between 10°C and 15°C. In most areas, this means that soil temperatures are warmer than air temperatures in the winter, and in the summer, soil temperatures are colder than the air (Miller, 2006), Hence, this constant temperature is used by GHP technologies to store extra heat from the source and then release it as needed for cooling or heating. GHP systems may be utilized almost everywhere to heat and cool houses and buildings, as well as deliver hot water.1) a system of pipes that are buried in the ground (Ground heat exchanger), exchanging heat directly with groundwater or immersed in a surface water body 2) a system to deliver energy, such as ductwork of HVAC system and 3) an effective geothermal heat pump, which transfers thermal energy between the ground heat exchanger and the delivery system. The ground heat exchanger transmits heat from the ground to a fluid (for examples water, antifreeze agent)

2.5 Types of Ground Heat Exchanger

Ground heat exchangers are found in various sizes and shape 4.9es. Heat exchangers which are closely looped are used in the vast majority of GHP systems in the United States (84 percent); vertical closed-loop systems account for slightly more than half of the total, whereas horizontal closed-loop systems account for around half. The remaining GHP systems (16%) are open-loop or closed-loop and utilize groundwater or surface water (Liu et al., 2019; Lund & Toth, 2021).

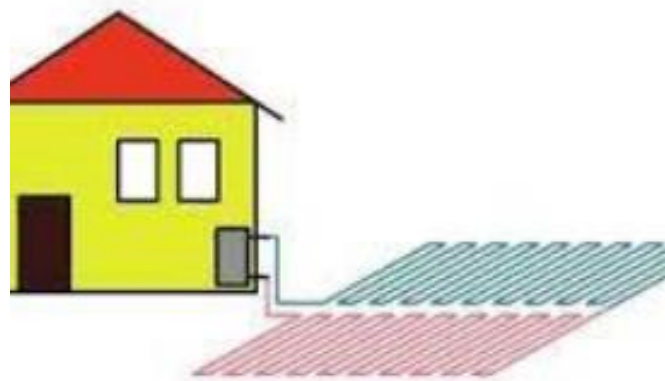


Figure 2.1 Horizontal Loop

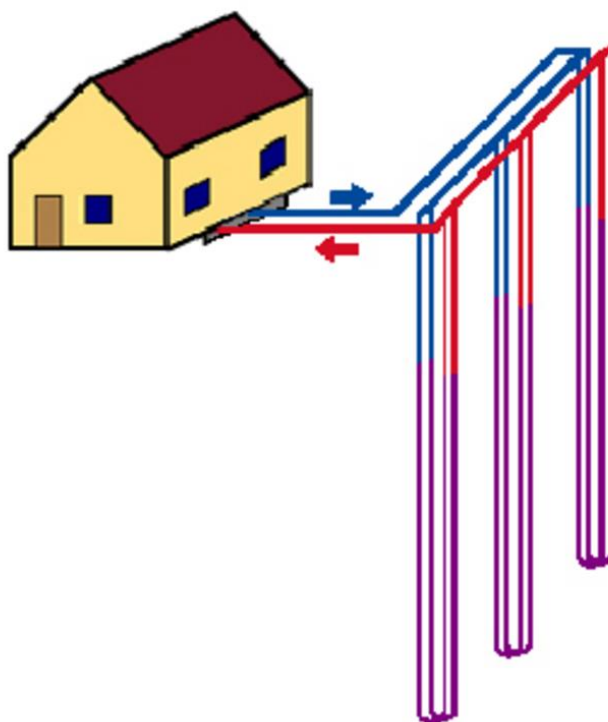


Figure 2.2 Vertical Heat Exchange

2.6 Geothermal Heat Pump Principle

Geothermal heat pumps are also known as ground source heat pumps, are outstanding renewable energy inventions that may be utilized in both residential and commercial structures. It's a small compact setup that includes a compressor, heat exchanger, control unit, etc. Several factors influence the size of a geothermal heat pump, including climate, installation costs, insulation, soil condition, land availability, heating and cooling demand, and so on. The horizontal, vertical, and pond/lake geothermal heat pumps are all closed-loop systems, while the fourth is an open-loop device.

The heat pump circulates fluid or refrigerant through the pipes throughout the winter, and the temperature of the fluid or refrigerant is raised by the heat from heated rock, soil, or water. The heated fluid is then pumped back into the building, where the heat exchanger may transmit the heated fluid to the building's heating system, warming the air and increasing the temperature. The cooled fluid is then returned to the earth, where it is reheated before being used to heat the structure. Similarly, in summer the same process occurs but here the fluid absorbs the heat in the air of the building or space into the ground i.e. it continually remove hot air in the building and hence cools the space or building.

The Vapor Compression Cycle is used by the geothermal heat pump to transmit heat from one area to another. During the summer, when we need to cool a building or area, the refrigerant or the fluid coming from the evaporator is compressed, raising the temperature and pressure. The heated refrigerant or fluid next passes into the condenser, where it loses heat and condenses back to liquid. The refrigerant with high pressure is expanded by reducing the pressure via an expansion valve. As a result, the cold fluid or refrigerant travels to the heat exchanger or evaporator, where the heat is exchanged, i.e., the cold fluid captures and cools the heat of the building. The fluid is returned to the compressor, and the cycle is repeated. The reversing valve converts the interior coil to condenser mode and the subterranean pipework to evaporator mode throughout the winter.

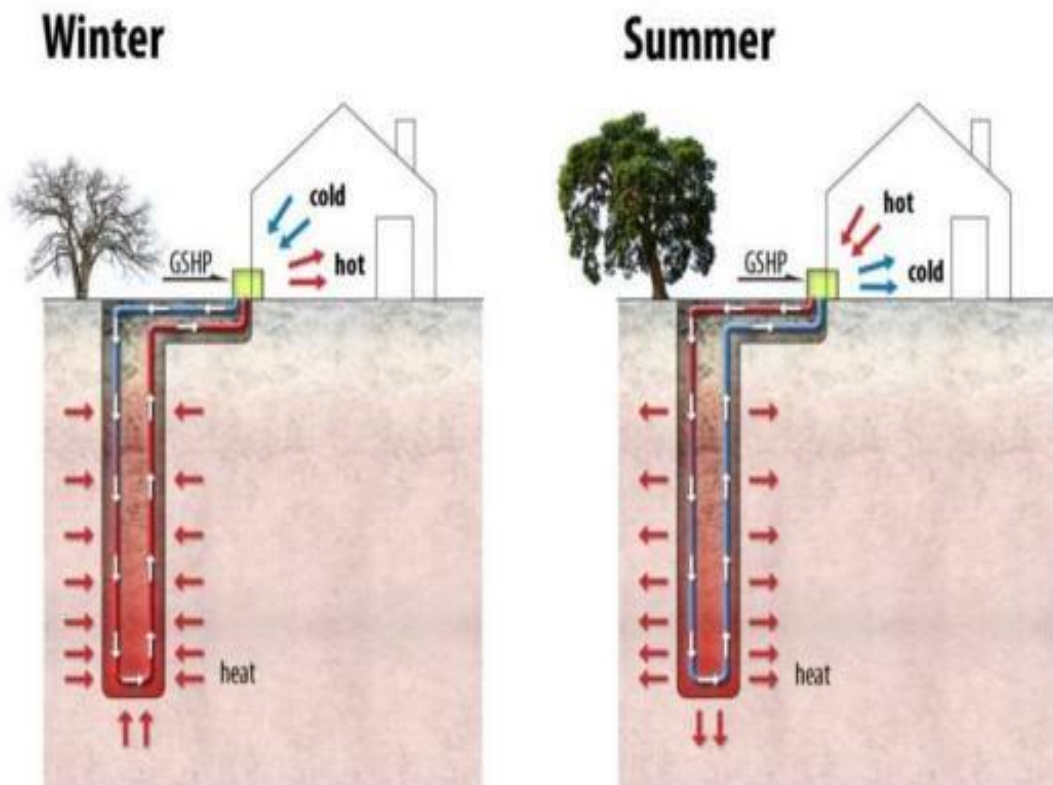


Figure 2.3 Principles of direct geothermal heating and cooling (Ian W Johnston et al, 2012)

2.7 Human Comfort

The condition of a contented or satisfied mind in a thermal environment is known as human comfort. The requirement and need for individuals to feel comfortable while utilizing air conditioning is rising as the atmospheric condition or environment is changing day by day. Because huge buildings consume more energy and have a shorter

lifespan, electrical air conditioning may not be a wise solution, hence a central air conditioning system is installed for those structures (Sahu, 2014).

The requirement for additional air conditioning in the building for human comfort is also influenced by the structure or building orientation. The American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., and the United States Department of Housing and Urban Development calculated that except in the country's extreme southern latitudes, heating and cooling loads for a typical residential structure at various orientations were shown to be higher for north than south orientations.

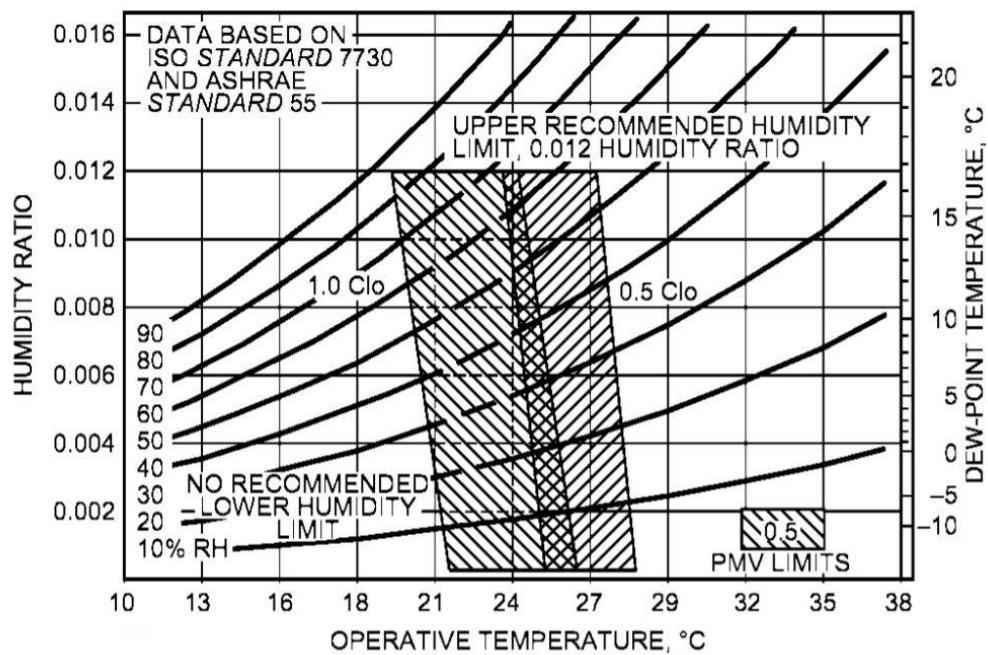


Figure 2.4 Summer and Winter Comfort Zones

Source: (ASHRAE Technical Committee, 2009)

Besides various environmental or external factors like temperature, thermal radiation, humidity, and airspeed, personal factors like personal condition and clothing impact human comfort. Since the thermal radiation and the airspeed are less controllable the impact of temperature and humidity is studied. The above chart considers operative temperature and humidity with airspeed ≤ 0.2 m/s for people wearing 0.1 (winter clothing) to 0.5clo (summer clothing). Both low and high humidity can create discomfort for humans. ASHRAE explains that for human comfort, there is no lower humidity limit and the maximum recommended upper humidity limit is 0.012 humidity ratio. Generally for designing an HVAC system comfort temperature is taken from 20 °C to 27°C. However, ASHRAE explains the optimum operative temperature for

designing both summer and winter comfort zones where the comfort varies from the clothing insulation level.

2.8 U Type Heat Exchanger

Among various heat exchanger tools, the U type is widely used because of its simple design, construction, better heat exchange performance, lesser tube joints, high load-bearing, fewer leakages, etc. When there is a temperature differential between the two sides of tubes, heat conduction begins, interfering with the soil's capacity to operate as a heat sink (Pei & Zhang, 2016). Though vertical double U type heat exchangers exchange more heat than single U type heat exchangers, thermal interference or resistance is more likely to occur with double U type heat exchangers (De Carli et al., 2010).

2.9 Double Glazing Glass

Two panes of glass are put into a window frame in double glazing glass, with inert gases such as argon supplied between the panes to boost insulation and reduce heat transmission, ensuring human comfort or providing respite from high-temperature conditions (Lohia & Dixit, 2015).

Research done by (Forughian & Taheri Shahr Aiini, 2017) found that double-glazed windows are extremely good and effective in isolating the inside temperature from the outside, saving both electrical and heat energy.

2.10 Procedure for Calculating Space Design Cooling Load

2.10.1 Cooling Load Temperature Difference (CLTD)

This technique was introduced by ASHRAE in 1979 for the calculation of total heat gains via a building envelope to properly design and size HVAC equipment. It denotes the difference in temperature between inside and outside air, which also includes the heating effects of solar radiation. It may be described by the theoretical temperature differential established for each wall or roof to provide the same heat load for the exposed surface, which accounts for the combined impact of radiation, conductive storage, and other factors.

$$CLTD=q/UA$$

Where, U=Design heat transfer coefficient, A= Surface area, q=conduction through any medium

$$\text{CLTD}_{\text{corrected}} = (\text{CLTD} + \text{LM}) * K + (78 - T_R) + (t_o - 85) \quad \text{Equation 2.1}$$

Where,

Average outside temperature $(t_o) = \text{OA} - (\text{DR}/2)$ or $T_{\text{avg}} = T_{\text{max}} - (\text{Daily range} / 2)$

LM= Latitude month correction

K=color adjustment factor

K=1.0 for dark-colored

K=0.83 medium colored

K=0.65 light-colored

$78 - T_R$ = temperature correction

$t_o - 85$ = correction design temperature of outdoor

2.10.2 Heat Gain through Roof

$$q = U * A * \text{CLTD} \quad \text{Equation 2.2}$$

Where

U=Transmission factor

A=Areas

CLTD=Temperature difference heat transfer factor

2.10.3 Heat Gain through Walls

$$q = U * A * \text{CLTD} \quad \text{Equation 2.3}$$

Equation 2.4 Heat Gain through Walls

Where

U=Transmission factor

A=surface area

CLTD= Temperature difference heat transfer factor

2.10.4 Heat Gain through Glass

Conduction

$$q = U * A * \text{CLTD} \quad \text{Equation 2.5}$$

Where

U=heat transfer coefficient

A=surface area

CLTD= Temperature difference heat transfer factor

Solar Radiation

$$q = A * \text{SC} * \text{SHGF} * \text{CLF} \quad \text{Equation 2.6}$$

A=area according to design

SC=coefficients of shading for a different combination

SHGF=heat transfer factor for different direction

CLF=load factor for cooling

2.10.5 Heat Gain through Internal Lights

$$q = \text{input} * \text{BF} * 3.4 \quad \text{Equation 2.7}$$

Where input=total load

BF=ballast factor

2.10.6 Heat Gain through People

Sensible Heat Gain

$$Q_s = \text{No} * \text{Sens HG} * \text{CLF} \quad \text{Equation 2.8}$$

Where

No=No of people in Space

HG=Heat Gain

CLF=heat transfer factor according to a time interval and density of people for cooling load

Latent Heat Gain

$$Q_l = \text{No} * \text{Lat HG} \quad \text{Equation 2.9}$$

Where

No=No of people Equation 2.10

Lat HG=Latent heat Gain

2.10.7 Heat Gain through Appliances

Sensible Heat Gain

Q_s =Suggested sensible heat transfer

Latent Heat Gain

Q_l =Suggested latent heat transfer

Heat transfer through Ventilation and Infiltration

Sensible Heat Gain

$$Q_s = 1.1 * \text{CFM} * \text{dT} \text{ (Btu/hr)} \quad \text{Equation 2.11}$$

Latent Heat Gain

$$Q_l = 4840 * \text{CFM} * \text{dW} \quad \text{Equation 2.12}$$

Where dw=difference in humidity ratio

2.11 Procedure for Calculating Space Design Heating Load

2.11.1 Heat Gain through Roofs, Walls, and Glass

$$Q=U*A*TD \quad \text{Equation 2.13}$$

Where

U=heat transfer factor for the design

A=space area

TD=Difference of temperature between inside design and outside design

2.11.2 Heat Gain through Infiltration and Ventilation

Sensible Heat Gain

$$Q_s=1.1*CFM*dT \text{ (Btu/hr)} \quad \text{Equation 2.14}$$

Latent Heat Gain

$$Q_l=4840*CFM*dW \quad \text{Equation 2.15}$$

2.12 Procedure for Calculating Required Bore Length for Heat Exchanger

2.12.1 The Equation for Calculation of Ground Heat Transfer Depth

The Ingersoll approach may be used to build closed-loop ground heat exchangers. This approach is a per-unit-length variation of a simple steady-state heat transfer equation.

$$q = \frac{L_{\text{bore}}(T_g - T_w)}{R_{\text{ov}}} \quad \text{Equation 2.16}$$

Where

q = transfer of heat

L_{bore} = depth for heat transfer

t_g = temperature underground surface

t_w = fluid temperature in the system

R_{ov} = underground surface resistance

The efficiency of the ground source heat system determines and the consideration of heat transfer between ground and building (EER or COP_c):

$$\frac{Q_{\text{cond}}}{Q_{\text{lc}}} = \frac{\text{EER}+3.412}{\text{EER}} \quad \text{Equation 2.17}$$

$$\frac{Q_{\text{cond}}}{Q_{\text{lc}}} = \frac{\text{COP}_c+1.0}{\text{COP}_c} \quad \text{Equation 2.18}$$

Where

q_{cond} = heat exchange to ground

q_{lc} = space cooling load, Btu/h (W)

EER = ratio of energy efficiency

COP_c = performance for space cooling

In heating mode, the ground source heat pump and ancillary system is provided to the building. As a result, the evaporator removes heat from the earth.

$$\frac{Q_{\text{evap}}}{Q_{\text{lh}}} = \frac{\text{COPh}-1}{\text{COPh}} \quad \text{Equation 2.19}$$

Where,

Q_{evap} = rate of heat transfer of heat pump evaporator

Q_{lh} = calculation of space heating,

COP_h = efficiency of heating of space

The equivalent full-load hours in cooling (EFLH_c) and heating (EFLH_h) are used to calculate the net annual heat transfer rate (q_a) (EFLH_h). EFLH values for various locales and building types may be found in the (Kavanagh, 2014)

$$Q_a = \frac{Q_{\text{cond}} \cdot \text{EFLH}_c + Q_{\text{evap}} \cdot \text{EFLH}_h}{8760} \quad \text{Equation 2.20}$$

The resultant equation for cooling ground heat exchanger bore length is as follows:

$$L_c = \frac{Q_a R_{ga} + Q_{\text{cond}} (R_b + \text{PLFm} R_{gm} + F_{sc} R_{gst})}{t_g - \frac{\text{ELT} + \text{LLT}}{2} + t_p} \quad \text{Equation 2.21}$$

The required length for heating is

$$L_h = \frac{Q_a R_{ga} + Q_{\text{evap}} (R_b + \text{PLFm} R_{gm} + F_{sc} R_{gst})}{t_g - \frac{\text{ELT} + \text{LLT}}{2} + t_p} \quad \text{Equation 2.22}$$

Where

F_{sc} = heat interference factor for supply and return design

L_c = depth for cooling, ft (m)

L_h = depth for heating, ft (m)

PLF_m = load factor for the system

q_a = system heat transfer underground loop, Btu/h (W)

R_{ga} = ground resistance—yearly pulse

R_{gst} = ground resistance of the system—less interval pulse,

R_{gm} = ground resistance under earth surface— one month pulse

R_b = bore resistance of the system,

t_g = temperature below certain height from the ground, °F (°C)

t_p = temperature imbalance after certain time from heat transfer of the sytem, °F (°C)

ELT = supply fluid temperature of the system

LLT = return fluid temperature of the system

The period of utilization, size of pipe, and ground thermal heat transfer must all be connected in the Fourier number(F_o), according to Carslaw and Jaeger (1947) (Kavanagh, 2014)

$$F_o = \frac{4\alpha g\tau}{d^2} \quad \text{Equation 2.23}$$

In three intervals of time, a ten-year (3650 days) interval of q_a , a short time (30 days) pulse of q_m , and a four hours (0.167 days) pulse of q_d , can be used to describe a system. The definition of three times is

$$\tau_1 = 3650$$

$$\tau_2 = 3650 + 30 = 3680,$$

$$\tau_f = 3680.167 \text{ days}$$

After that fourier value is calculated concerning below value:

$$F_{of} = 4\alpha\tau_f/d^2$$

$$F_{o1} = 4\alpha(\tau_f - \tau_1)/d^2$$

$$F_{o2} = 4\alpha(\tau_f - \tau_2)/d^2$$

The figure below is then used to calculate the G value for an individual of the Fourier data during each heat pulse, the three equivalent thermal resistance values are

$$\text{calculated from } R_{ga} = \frac{G_f - G_1}{k_g} \quad \text{Equation 2.24}$$

$$R_{gm} = \frac{G_1 - G_2}{k_g} \quad \text{Equation 2.25}$$

$$R_{gst} = \frac{G_2}{k_g}$$

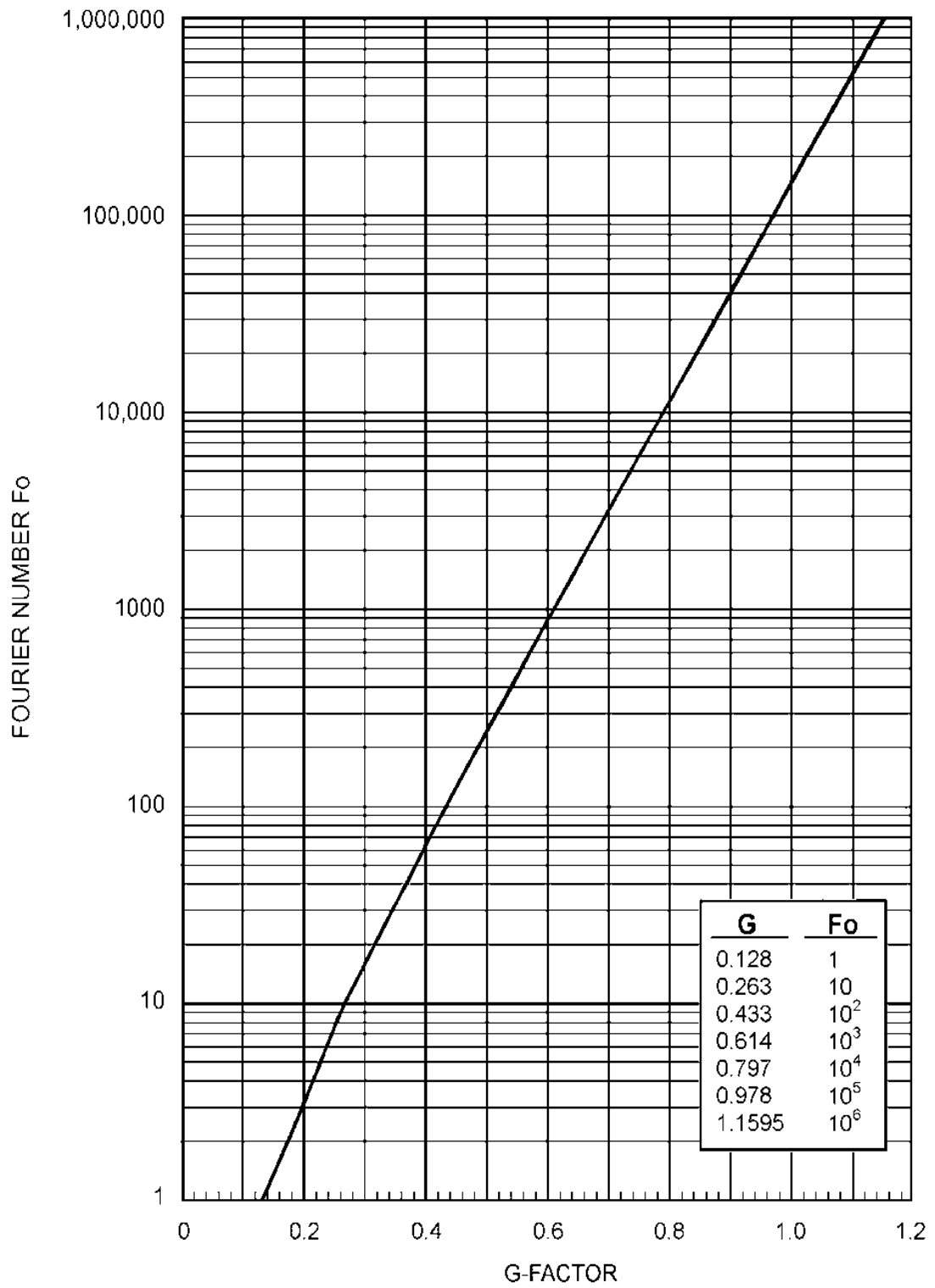


Figure 2.5 Fourier/G-Factor Graph for Ground Thermal Resistance(Kavanagh, 2014)

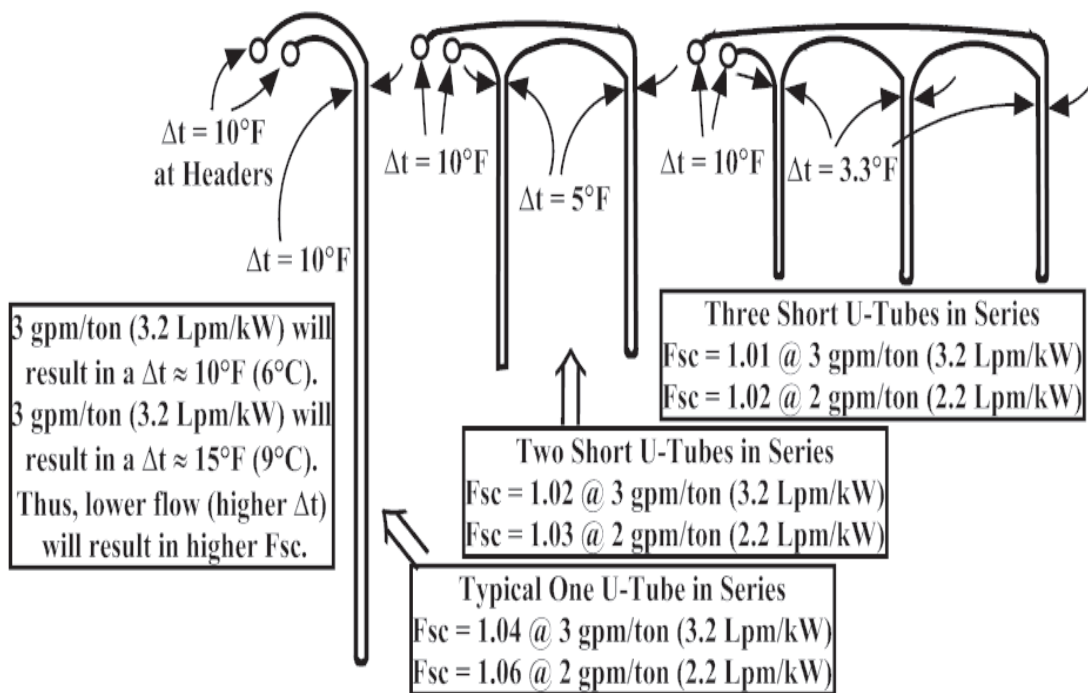


Figure 2.6 Heat Interference Factor (Kavanagh, 2014)

2.13 Hourly Analysis Program (HAP 4.9)

It's a flexible tool for planning commercial HVAC systems as well as energy analyses. HAP 4.9 is used to determine the design cooling and heating loads for the system's rooms, zones, and coils. Further, it determines required airflow rates of space, sizes of air circulation fans, chillers, boilers, etc. It stimulates hour-by-hour activities of all heating and air conditions systems in buildings, plants, lighting appliances, etc. (Corporation, 2016).

2.14 ANSYS

ANSYS may be used to simulate HVAC system operation in both summer and winter scenarios. Further, it is very helpful to design comfortable space conditions for both seasons (Popovici, 2017). It helps in creating a virtual prototype and calculating the heating, ventilation, and air condition (HVAC) of buildings for better analysis of energy in the space.

CHAPTER THREE: RESEARCH METHODOLOGY

This part indicates the complete methods and processes carried out to complete the research objectives. This part is divided into major works. i.e. data collection to calculate building heating load and cooling load, selection of indoor and outdoor system, calculation of required bore length for heating and cooling system. Heat flow analysis of U Type of heat exchanger under the ground was done using ANSYS software and all the technical parameters pipe length, diameter, and heat flow are calculated. The data for heating and cooling load calculation was collected from architecture engineering, site visit, and discussion with the Project Manager. And for ANSYS simulation data were collected from the soil test center and different articles for ground temperature. Hence the system is optimized with geothermal heating and cooling. To conduct effective length, ANSYS simulation will be done and analyzed heat flow to the ground and analyze inlet and outlet temperature for heating and cooling conditions.

For the verification of the heating and cooling load for the building Hourly Analysis Program(HAP 4.9) software will be used for the analysis of the heating and cooling load in the different time frames.

The general methodology for this research can be visualized as:

1. Data collection of the building details.
2. Calculation of Heating and Cooling Load.
3. Calculation of Heating and Cooling Load with HAP 4.9 Software
4. Design of Heating and Cooling with Ground Source Heat Pump System.
5. Calculation of vertical required bore length for heating and cooling.
6. Heat flow analysis with ANSYS software.
7. Heating load and cooling load analysis with HAP 4.9 software.
8. Cost-benefit analysis with the existing system.

This complete structure of the above methodology includes the answer to the research question, data collection, GSHP system development, simulation setup, result, conclusion, and recommendation. At first, the necessary theories and related literature are studied, and then the necessary resources to complete the research are to be finalized. The resources may be the data, model development approach, and software selection. Reviewing the articles and journal papers gives insights into the degree to

which any works or research or studies are put forward by scholars from the past to the present. The different theories and models needed for giving the right track to this thesis work are to be studied first.

3.1 Data Collection of the Building Details

To calculate heating load and cooling load, we need to identify floor layout, building orientation, and building materials. For the floor layout and building design, we collected the data from the architecture engineer and got details on the site layout, and orientation of the building. From the architecture engineer we collected.

1. Building height, floor height, and floor area
2. Window design
3. Building Orientation

For building material, we discussed with the owner and project manager to get the proper information. So with the discussion, we collected

1. Wall type and thickness
2. Slab Details
3. Glass details
4. Light details
5. Appliance details

3.2 Calculation of Heating and Cooling Load

To calculate heating and cooling load for the building is calculated based on ASHRAE standards and different required values are taken reference from the ASHRAE data table.

3.3 Design of System with Ground Source Heat Pump System

In finalizing the heating and cooling system in the building details calculation is done and finalized the system size requirement is. According to the required system size, the indoor unit and outdoor unit will be finalized and selected as per standard with the review of the technical catalog. With the review of different technical catalogs and design calculations, an optimized system will be selected.

3.4 Calculation of Vertical Required Bore Length for Heating and Cooling

After calculating the heating load and cooling load for the building and selecting GSHP for the building. To extract and absorb heat from the ground proper length will be required for the proper function of the system. Analyzing the soil type, ground temperature, moisture content, and other important parameter required length for heating and cooling is calculated. But for the system whichever is the greater will be selected in the design.

3.5 Heat Flow Analysis with ANSYS Software

After finalizing the system and required length for the ground U type heat exchanger under the ground surface, the temperature difference in liquid entering temperature and liquid leaving temperature is analyzed with different water speed and mass flow rates. With this analysis in the software, we can have a clear view of the effectiveness of the system with different mass flows and speeds of the liquid inside the U-type heat exchanger. So the analysis of the software will assist to design the optimized system for Geothermal heating and cooling.

3.6 Heating Load and Cooling Load Analysis with HAP 4.9 Software

The Hourly Analysis Program (HAP 4.9) is a computer program that aids engineers in the design of a building's HVAC system. It aids in load estimation and system design. It also aids in the simulation of building energy usage, cost, and system completion. So in this research HAP 4.9 software will be used for analyzing different loads in different time intervals and assist to finalize the system for space heating and cooling in the building.

3.7 Procedure to Calculate Heating and Cooling Load

To calculate cooling and heating load in HAP 4.9 software following data should be accurately input to get the result

i. Weather

For this case study, Pulchowk commercial building is taken into consideration and in the HAP 4.9 software in weather properties region Asia, location Kathmandu, Nepal is selected. So the software automatically gets the weather data and details are shown in APPENDIX A.

ii. Project Libraries

In Project Libraries, we need to enter the data

- Schedules

According to HAP 4.9 software schedule is created for an appliance, lights, people, and fans. In schedule Hour Profile and Assignments are created for the calculation process and the figure is shown in APPENDIX B

- Walls

For walls, according to our wall type, our project wall type is selected in the HAP 4.9 software, details are shown in APPENDIX C

- Roofs, Windows, Doors, and Shades

Similarly, details of roofs, windows, doors, and shades are selected for the HAP 4.9 software.

iii. Spaces

After creating the profile project libraries space is defined in the system. In the spaces we need to enter the project date, details are revealed in APPENDIX D

- General Data
- Internals Data
- Walls, Windows, Doors
- Roofs, Skylights
- Infiltration
- Floors
- Partitions

iv. Systems

After completing the procedure weather, project libraries, and spaces we need to select the system, system components, zone components, and sizing data. After completing all this process system will calculate all details and generate the reports, details are revealed in APPENDIX E.

3.8 Cost-Benefit Analysis with the Existing System

In the comparison of the technical catalog of the GSHP system and HVAC system, the GSHP system has a high COP compared to the normal system. So GSHP system is energy efficient and reduces the energy demand of the building and will have a cost-benefit implementation of the GSHP system. So detailed cost-benefit analysis will be calculated with this system concerning recent commercial HVAC systems.

3.9 Financial Analysis

Before investing capital in a proposed project, a financial study is required. As a result, a financial study was conducted to determine its financial viability. Three major financial decision factors were analyzed in this study to see if investing in a Ground Source Heat Pump is feasible or not in terms of a financial point of view. The suggested scheme's payback duration, internal rate of return, and net present value were all taken into consideration as financial indicators.

$$\text{Discounted Payback Period (DPP)} = \frac{\ln\left(\frac{1}{1 - \frac{I}{Ar}}\right)}{\ln(1+r)} \quad \text{Equation 3.1}$$

Where I= initial investment

A=Annual returns

r = Discount rate

$$\text{NPV} = -I + \frac{F1}{(1+r)} + \dots + \frac{Fn}{(1+r)^n} \quad \text{Equation 3.2}$$

Where, I= initial investment

F=Future Values

r = Discount rate

The additional cost between the VRF system and DVMS GSHP system is the investment, and the annual returns are the energy savings from reduced energy consumption. The additional investment for the GSHP system was calculated using the rate provided by the Samsung dealer and regarding the technical catalog of Samsung, energy demand reduction is calculated and according to NPV, IRR, and the payback period is calculated. For the discount rate, 12% was considered according to Nepal Rastra Bank(NRB, 2020).

CHAPTER FOUR: RESULT AND DISCUSSION

The proposed methodology which is enlisted above has been implemented to conduct the research. Heating and cooling load have been calculated for the design of the Ground Source Heat Pump system.

4.1 Calculation of Cooling Load

In process of calculating the cooling load, the monthly temperature pattern for Kathmandu valley is found.

Table 4.1 Monthly temperature pattern for Kathmandu (Bhujju et al., 2020)

Month	Max DBT	Min DBT	Max WBT	Min WBT
Jan	77.2	52.2	70.6	51.7
Feb	79.2	54.2	71.6	53.7
Mar	82.4	57.4	74.8	56.9
Apr	83.6	58.6	75.0	58.1
May	86.0	61.0	76.0	60.5
Jun	88.0	63.0	78.0	62.5
Jul	89.0	64.0	78.0	63.5
Aug	89.0	64.0	78.0	63.5
Sep	87.0	62.0	77.0	61.5
Oct	84.8	59.8	75.8	59.3
Nov	80.6	55.6	73.8	55.1
Dec	78.2	53.2	71.8	52.7

Project Details

For the calculation of heating and cooling load, the case study, Pulchowk commercial building is taken into consideration. Details are mentioned in Table 4.2.

Table 4.2 Project Details

Project	Commercial Buildings
Location	Pulchowk
Latitude	28
Longitude	85

Table 4.3 Group Type and U Value Details (ASHRAE Technical Committee, 1997)

Group No.	Description of Construction	Weight (lb/ft ²)	U-Value Btu/(hr.ft ² . F)	Heat Capacity Btu/(ft ² . F)	Code Numbers of Layers (See Table 3.11)
4-in. Face Brick+(Brick)					
C	Air Space+4-in. Face Brick	83	0.358	18.3	A0, A2, B1, A2, E0
D	4-in. Common Brick	90	0.415	18.4	A0, A2, C4, E1, E0
C	1-in. Insulation or Air Space+4-in. Common Brick	90	0.174-0.301	18.4	A0, A2, C4, B1/B2, E1, E0
B	2-in. Insulation+4-in. Common Brick	88	0.111	18.5	A0, A2, B3, C4, E1, E0
B	8-in. Common Brick	130	0.302	26.4	A0, A2, C9, E1, E0
A	Insulation or Air Space+8-in. Common Brick	130	0.154-0.243	26.4	A0, A2, C9, B1/B2, E1, E0

Table 4.4 Wall Group

S.N	Description	Wall Group
1	8 in common brick	B

4.1.1 Calculating Cooling Load for Ground Floor

To calculate heat transfer from different mediums like the wall, glass, and floor cooling load temperature difference is taken into consideration to incorporate that heat transfer from different medium takes a certain time. So CLTD for Calculating Cooling Load from Sunlit Walls are calculated.

Group B Walls

Table 4.5 Cooling Load Temperature Difference

Direction/Time	1	2	3	4	5	6	7	8	9	10	11	12
N	15	14	14	13	12	11	11	10	9	9	9	8

NE	19	18	17	16	15	14	13	12	12	13	14	15
E	23	22	21	20	18	17	16	15	15	15	17	19
SE	23	22	21	20	18	17	16	15	14	14	15	16
SE	21	20	19	18	17	15	14	13	12	11	11	11
SW	27	26	25	24	22	21	19	18	16	15	14	14
W	29	28	27	26	24	23	21	19	18	17	16	15
NW	23	22	21	20	19	18	17	15	14	13	12	12

Direction/Time	13	14	15	16	17	18	19	20	21	22	23	24
N	9	9	9	10	11	12	13	14	14	15	15	15
NE	16	17	18	19	19	20	20	21	21	21	20	20
E	21	22	24	25	26	26	27	27	26	26	25	24
SE	18	20	21	23	24	25	26	26	26	26	25	24
SE	11	12	14	15	17	19	20	21	22	22	22	21
SW	13	13	14	15	17	20	22	25	27	28	28	28
W	14	14	14	15	17	19	22	25	27	29	29	30
NW	12	11	12	12	13	15	17	19	21	22	23	23

To calculate month correction (LM) for the latitude of Pulchowk and design month for 24°. The value is tabulated in table 4.6 below.

Table 4.6 Latitude Month Correction

Month	N	NE/NW	E/W	SE/SW	S	HOR
Dec	-5	-8	-7	3	13	-13
Jan/Nov	-4	-9	-6	3	13	-11
Feb/Oct	-4	-6	-3	3	10	-7
March/Sep	-3	-3	-1	1	4	-3
Apr/Aug	-2	0	-1	-1	-3	0
May/Jul	1	2	0	-3	-6	1
Jun	3	3	0	-4	-6	1

Source: (ASHRAE,1980)

After the calculation of Latitude month cooling load temperature is required to calculate heat gain. So the calculation of cooling load temperature difference is tabulated in the given table 4.7.

The reference month is July and the temperature considered for calculating is taken as degree centigrade.

Table 4.7 CLTD Correction

S.N	Direction	CLTD	K	LM	To	Ti	CLTDcorr
1	East	27	1	0	29	22	30.1
2	West	30	1	0	29	22	33.1
3	North	15	1	3	29	22	21.1
4	South	22	1	-6	29	22	19.1

1) Heat Gain Through Wall

To calculate heat gain through the wall, the CLTD value calculated in table 8 is used to calculate heat gain through the wall. CLTD value differs in each direction and heat gain through the wall in each direction is calculated and shown in table 4.8.

Table 4.8 Heat Gain through Wall

Item	Orientation	Area	Factor(U)	CLTDcor	Heat Gain(BTU/HR)
Wall	East	0	0.302	30.1	0
Wall	West	350	0.302	33.1	3498.67
Wall	South	695	0.302	21.1	4428.68
Wall	North	750	0.302	19.1	4326.15

Table 4.9 Heat Gain through Glass

2) Heat Gain Through Glass

i. Conduction

Item	Orientation	Area	Temperature Difference	Factor(U)	Heat Gain(BTU/HR)
Glass	East	360	11.8	1	4248
Glass	South	340	11.8	1	4012

Glass	West	10	11.8	1	118
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ii.Radiation

Item	Orientation	Area	Heat Gain	Factor(U)	Heat Gain(BTU/HR)
Glass	East	360	165	1	59400
Glass	South	340	27	1	9180
Glass	West	10	165	1	1650

Table 4.10 Heat Gain through Door

Item	Orientation	Area	Temperature Difference	Factor(U)	Heat Gain(BTU/HR)
Glass	West	40	11.8	1	472
Radiation					
Item	Orientation	Area	Heat Gain	Factor(U)	Heat Gain(BTU/HR)
Glass	West	40	165	1	6600

Table 4.11 Heat Gain through Light

Length	58.5	ft
Breadth	37	ft
Area	2164.5	sq ft
Lux	500	
Watt	8	W
Luminaire efficiency	120	lumen per watt

To calculate the required number of light $N=EA/(\phi*UF*MF)$ equations are used where utilization factor(UF) and maintenance factor(MF) is taken into account. Using $UF=0.6$ and $MF=0.8$. The required no of light is 186.

Total wattage required =1488 wattage.

Heat gain through light =6324 Btu/hr.

3) Heat Gain from People

To calculate heat gain from people occupancy density method is used and the value is taken from the activity level of the commercial mall value. The details value is shown in the table 4.12 below.

Table 4.12 Heat Gain through People

People	74
Latent heat	155
Sensible heat	245
Total latent heat	11470
Total sensible heat	18130
Total heat(Btu/hr)	29600

4) Heat Gain through Appliances

To calculate heat gain through appliances generally, office appliances are taken into consideration. The list of appliances, power consumption, and quantity are shown in table 4.13.

Table 4.13 Heat Gain through Appliance

S.N	Description	Wattage	Qty	Total Wattage
1	Desktop	77	2	154
2	Printer	130	1	130
3	Dispenser	350	1	350
4	Oven	1030	1	1030

For Coffee Maker,

Table 4.14 Sensible and Latent Heat Gain

Sensible heat	1050	watt
Latent heat	1540	btu/hr

Total wattage of the appliance=2714

Sensible Heat Gain from appliance=9227.6

Latent heat Gain from appliance=1540

Total heat gain (Btu/hr)=10767.6

5) Heat Gain through Ventilation

In the commercial building, there will be a transfer of heat in the space through ventilation. For heat load through ventilation, the occupancy density method is used to

estimate the total no of people in the space. From the Ashrae handbook occupancy density of 40 per 1000 sq ft is taken into consideration

- i. For Sensible heat gain

$$Q_s = 1.1 * CFM * dT \text{ (Btu/hr)}$$

- ii. For Latent Heat Gain

$$Q_l = 4840 * CFM * dW$$

Where dW= difference in humidity ratio

Details calculation is shown in table 4.15

Table 4.15 Heat Gain through Ventilation

CFM/Person	7.5
Occupancy density per 1000 sq. ft.	40
Area	1836
No of people	73.44
Total no of people	74
CFM as standard for floor	0.06
CFM for floor	110.16
CFM for people	555
Total CFM	665.16

Sensible heat Gain= 8633.7768 Btu/hr

For Latent Heat Gain

Table 4.16 represents the dry-bulb temperature and wet bulb temperature of the outside condition

Table 4.16 Outside Climatic Details

Outside		
DBT	WBT	HR
85	75	0.01648

Table 4.17 represents dry bulb temperature and wet bulb temperature of inside space condition

Table 4.17 Inside Climatic Details

Inside		
DB	RH	HR
76	50	0.009
dW		0.00748
Latent heat gain		24080.92

6) Heat Gain through Infiltration

In process of calculating heat gain through infiltration, the air change method is taken into consideration. Details calculation is shown in table 4.18.

Table 4.18 Floor details

Area	1836	sq ft
Height	10	ft
Volume	18360	ft ³

No of airchange =0.7(ASHRAE,1980)

CFM=Volume*Na/60

Where Na=Number of air change

The required CFM for the system is 241.2

Sensible Heat Gain=4241.16

Table 4.19 For Outside Condition

DB	WB	HR
85	75	0.01648

Table 4.20 For Inside Condition

DB	RH	HR
76	50	0.009
dW		0.00748
Latent heat gain		7754.72544

7)Heat Gain Through Floor

Length	51	ft
Breadth	36	ft

Area	1836	sq ft
Floor-type	5" heavy weight concrete	
Group Type	E	
U value	0.585	(Ashrae,1980)

Cooling Load Temperature Difference for Calculating Cooling Load from Sunlit Walls
Table 4.21 CLTD for Group E Walls

Direction/Time	1	2	3	4	5	6	7	8	9	10	11	12	13	14
N	12	10	8	7	5	4	3	4	5	6	7	9	11	13
NE	13	11	9	7	6	4	5	9	15	20	24	25	25	26
E	14	12	10	8	6	5	6	11	18	26	33	26	38	37
SE	15	12	10	8	7	5	5	8	12	19	25	31	35	37
SE	15	12	10	8	7	5	4	3	4	5	9	13	19	24
SW	22	18	15	12	10	8	6	5	5	6	7	9	12	18
W	25	21	17	14	11	9	7	6	6	6	7	9	11	14
NW	20	17	14	11	9	7	6	5	5	5	6	8	10	13

Direction/Time	15	16	17	18	19	20	21	22	23	24
N	15	17	19	20	21	23	20	18	16	14
NE	26	26	26	26	25	24	22	19	17	15
E	36	34	33	32	30	28	25	22	20	17
SE	37	36	34	33	31	28	26	23	20	17
SE	29	32	34	33	31	29	26	23	20	17
SW	24	32	38	43	45	44	40	35	30	26
W	20	27	36	43	49	49	45	40	34	29
NW	16	20	26	32	37	38	36	32	28	24

(Source: ASHRAE,1980)

In heat gain through floor latitude month is calculated and for horizontal floor horizontal value from data, the table is taken into consideration. The value is tabulated in table 4.22.

Table 4.22 Latitude Month Horizontal Value

Month	HOR
Dec	-13

Jan/Nov	-11
Feb/Oct	-7
March/Sep	-3
Apr/Aug	0
May/Jul	1
Jun	1

Source (ASHRAE,1980)

Similar to the heat gain through the wall, the cooling load temperature difference is calculated for the heat gain through the roof. Details are shown in Table 4.23.

Table 4.23 CLTD Correction for Horizontal

S.N	Direction	CLTD	K	LM	To	Ti	CLTD _{corr}
1	Horizontal	49	0.585	1	29	22	32.35

For calculation of CLTD month reference, June/July is taken into consideration.

So Heat gain through the floor is 34745.84

Now the summary of the heat gain for the ground floor is summarized in table 4.24.

Table 4.24 Summary of Heat Gain

S.N	Heat Source	Sensible Heat	Latent Heat
1	Glass	78608	
2	Wall	12253.499	
3	Glass Door	7072	
4	Light	6324	
5	People	12691	11470
6	Appliances	9227.6	1540
7	Ventilation	8633.7768	24080.92051
8	Infiltration	4241.16	7754.72544
9	Down floor	34745.841	
10	Total	173796.8768	44845.64595
11	Total	218642.5228	
12	Tonnage required	20.04223125	

4.1.2 Calculating Cooling Load for First Floor

1) Heat Gain Through Wall

Item	Orientation	Area	Factor(U)	CLTDcor	Heat Gain(BTU/H R)
Wall	East	0	0.302	30.1	0
Wall	West	350	0.302	33.1	3498.67
Wall	South	410	0.302	21.1	2612.6
Wall	North	750	0.302	19.1	4326.15

2) Heat Gain Through Glass

i. Conduction

Item	Orientation	Area	Temperature Difference	Factor(U)	Heat Gain(BTU/H R)
Glass	East	360	11.8	1	4248
Glass	South	340	11.8	1	4012
Glass	West	10	11.8	1	118

ii. Radiation

Item	Orientation	Area	Heat Gain	Factor(U)	Heat Gain(BTU/H R)
Glass	East	360	165	1	59400
Glass	South	340	27	1	9180
Glass	West	10	165	1	1650

3) Heat Gain Through Door

1) Heat Gain Through Glass

i.

Conduction

Item	Orientation	Area	Temperature Difference	Factor(U)	Heat Gain(BTU/H R)
Glass	West	40	11.8	1	472

ii. Radiation

Item	Orientation	Area	Heat Gain	Factor(U)	Heat Gain(BTU/H R)
Glass	West	40	165	1	6600

4) Heat Gain Through Light

Length	58.5	ft
Breadth	37	ft
Area	2164.5	sq ft
Lux	500	
Watt	8	W
Luminaire efficiency	120	lumen per watt

To calculate the required number of light $N = EA / (\phi * UF * MF)$ equations are used where utilization factor (UF) and maintenance factor (MF) is taken into account. Using $UF = 0.6$ and $MF = 0.8$. The required no of light is 186.

Total wattage required = 1488 wattage.

Heat gain through light = 6324 Btu/hr.

5) Heat Gain from People

To calculate heat gain from people occupancy density method is used and the value is taken from the activity level of the commercial mall value. The details value is shown in the table below

People	74
Latent heat	155
Sensible heat	245
Total latent heat	11470
Total sensible heat	18130
Total heat(Btu/hr)	29600

6) Heat Gain through Appliances

To calculate heat gain through appliances generally, office appliances are taken into consideration. The list of appliances, power consumption, and quantity are shown.

Table 4.25 List of Appliances

S.N	Description	Wattage	Qty	Total Wattage
1	Desktop	77	2	154

2	Printer	130	1	130
3	Dispenser	350	1	350
4	Oven	1030	1	1030

For Coffee Maker,

Sensible heat	1050	watt
Latent heat	1540	BTU/hr

Total wattage of the appliance=2714

Sensible Heat Gain from appliance=9227.6

Latent heat Gain from appliance=1540

Total heat gain (Btu/hr) =10767.6

7) Heat Gain through Ventilation

Details calculation is shown in the table

CFM/Person	7.5
Occupancy density per 1000 sq. ft.	40
Area	1836
No of people	73.44
Total no of people	74
CFM as standard for floor	0.06
CFM for floor	110.16
CFM for people	555
Total CFM	665.16

Sensible heat Gain= 8633.7768 Btu/hr.

For Latent Heat Gain

The table represents dry bulb temperature and wet bulb temperature of the outside condition.

Outside		
DBT	WBT	HR
85	75	0.01648

The table represents dry bulb temperature and wet bulb temperature of inside space condition.

Inside		
DB	RH	HR
76	50	0.009
dW		0.00748
Latent heat gain		24080.92

8) Heat Gain through Infiltration

In process of calculating heat gain through infiltration, the air change method is taken into consideration. Details calculation is shown in table 4.26.

Table 4.26 Dimensions Details of Floor

Area	1836	sq. ft.
Height	10	ft.
Volume	18360	ft ³

No of air charge =0.7(ASHRAE,1980)

$$CFM=Volume*Na/60$$

Where Na=Number of air change

The required CFM for the system is 241.2

According to equation

$$\text{Sensible Heat Gain}=4241.16$$

For outside condition

DB	WB	HR
85	75	0.01648

For inside condition

DB	RH	HR
76	50	0.009
dW		0.00748
Latent heat gain		7754.72544

Summary of Heat Gain for the first floor

S.N	Heat Source	Sensible Heat	Latent Heat
1	Glass	78608	
2	Wall	10437.42	

3	Glass Door	7072	
4	Light	6324	
5	People	18130	11470
6	Appliances	9227.6	1540
7	Ventilation	8633.77	24080.92
8	Infiltration	4241.16	7754.72
9	Total	142673.95	44845.64
10	Total	187519.6	
11	Tonnage required	15.62	

4.1.3 Calculating Cooling Load for Second and Third Floor

Similarly, with the above calculation procedure for cooling load, the cooling load for the second and third floors is 14.85TR and 14.85TR respectively.

4.1.4 Calculating Cooling Load for the Fourth Floor

On this floor heat gain from the roof is taken into consideration. So details load calculations for this floor are

1) Heat Gain Through Wall

Item	Orientation	Area	Factor(U)	CLTDcor	Heat Gain(BTU/HR)
Wall	East	0	0.302	30.1	0
Wall	West	350	0.302	33.1	3498.67
Wall	South	410	0.302	21.1	2612.6
Wall	North	750	0.302	19.1	4326.15

2) Heat Gain Through

Glasses

i.

Conduction

Item	Orientation	Area	Temperature Difference	Factor(U)	Heat Gain(BTU/HR)
Glass	East	360	11.8	1	4248
Glass	South	340	11.8	1	4012
Glass	West	10	11.8	1	118

ii. Radiation

Item	Orientation	Area	Heat Gain	Factor(U)	Heat Gain(BTU/HR)
Glass	East	360	165	1	59400
Glass	South	340	27	1	9180
Glass	West	10	165	1	1650

3) Heat Gain through Door

1) Heat Gain through Wooden door

Item	Orientation	Area	CLTD corr	Factor(U)	Heat Gain(BTU/HR)
Glass	West	40	33.1	0.3	397.2

4) Heat Gain through Light

Length	51	ft
Breadth	36	ft
Area	1836	sq ft

Lux	500	
Watt	8	W
Luminaire efficiency	120	lumen per watt

$$\text{Number} = EA / (\phi * UF * MF)$$

UF	0.6
MF	0.8
N=	185.0787347
	186
Total wattage(Watt)	1488
Heat gain through light	
Heat gain(Btu/hr)	6324

5) Heat Gain through People

People	10
Latent heat	155
Sensible heat	245
Total latent heat	1550
Total sensible heat	2450
Total heat	4000

6) Heat Gain through Appliances

S.N	Description	Wattage	Qty	Total Wattage
1	Induction	879	1	879
2	Refrigerator	322	1	322
3	Dispenser	350	1	350
4	Oven	1030	1	1030

For coffee maker

Sensible heat(W)	1050
Latent heat (Btu/hr.)	1540

Total wattage	3631
Heat gain from sensible heat	12345.4
Latent heat gain	1540
Total (Btu/hr.)	13885.4

7) Heat Gain through Ventilation

CFM/Person	5
Occupancy density (per 1000 sq. ft.)	5
Area	1836
No of people	9.18
Total no of people	10
CFM as standard for floor	0.06
CFM for floor	110.16
CFM for people	50
Total CFM	160.16

Sensible Heat Gain: 2078.88 Btu/hr.

For Latent Heat Gain

The table represents dry bulb temperature and wet bulb temperature of the outside condition.

Outside		
DBT	WBT	HR
85	75	0.01648

The table represents dry bulb temperature and wet bulb temperature of inside space condition.

Inside		
DB	RH	HR
76	50	0.009
dW		0.00748
Latent heat gain		5798.3

8) Heat Gain through Infiltration

In process of calculating heat gain through infiltration, the air change method is taken into consideration. Details calculation is shown in the table.

Area	1836	sq ft
Height	10	ft
Volume	18360	ft ³

No of airchange =0.7(ASHRAE,1980)

$$\text{CFM}=\text{Volume}*\text{Na}/60$$

Where Na=Number of air change

The required CFM for the system is 241.2

According to equation

$$\text{Sensible Heat Gain}=4241.16$$

For outside condition

DB	WB	HR
85	75	0.01648

For inside condition

DB	RH	HR
76	50	0.009
dW		0.00748
Latent heat gain		7754.72544

Cooling Load Temperature Difference for Calculating Cooling Load from flat roof
For suspended ceiling

Table 4.27 CLTD for suspended ceiling

1	Steel Sheet with 1-in.	9	0.13 4	2	0	-2	-3	-4	-4	-1	9	2 3	3 7	5 0
	(or 2-in.) insulation	(10)	(0.09)											

2	1-in. wood with 1-in. insulation	10	0.11 5	2 0	1 5	1 1	8	5	3	2	3	7	1 3	2 1
3	4-in. h.w. concrete	20	0.13 4	1 9	1 4	1 0	7	4	2	0	0	4	1 0	1 9
4	2-in. h.w. concrete with 1-in. insulation	30	0.13 1	2 8	2 5	2 3	2 0	1 7	1 5	1 3	1 3	1 4	1 6	2 0
5	1-in. wood with 2-in. insulation	10	0.08 3	2 5	2 0	1 6	1 3	1 0	7	5	5	7	1 2	1 8
6	6-in. h.w. concrete	26	0.10 9	3 2	2 8	2 3	1 9	1 6	1 3	1 0	8	7	8	1 1
7	2-in. wood with 1-in. insulation	15	0.09 6	3 4	3 1	2 9	2 6	2 3	2 1	1 8	1 6	1 5	1 5	1 6
8	8-in. h.w. concrete	33	0.09 3	3 9	3 6	3 3	2 9	2 6	2 3	2 0	1 8	1 5	1 4	1 4
9	4-in. h.w. concrete with 1-in. (or 2-in.) insulation	53 (54)	0.12 (0.09)	3)	2)	2)	2)	2)	2)	2)	2)	2)	2)	2)
10	2.5-in. wood with 2-in. insulation	15	0.07 2	3 5	3 3	3 0	2 8	2 6	2 4	2 2	2 0	1 8	1 8	1 8
11	Roof Terrace System	77	0.08 2	3 0	2 9	2 8	2 7	2 6	2 5	2 4	2 3	2 2	2 2	2 2
12	6-in. h.w. concrete with 1-in. (or 2-in.) insulation	77 (77)	0.12 (0.08)	2)	2)	2)	2)	2)	2)	2)	2)	2)	2)	2)
13	4-in. wood with 1-in. (or 2-in.) insulation	19 (20)	0.08 (0.06)	3)	3)	3)	3)	3)	2)	2)	2)	2)	2)	2)

1	Steel Sheet with 1-in. (or 2-in.) insulation	6 2	7 1	7 7	7 8	7 4	6 7	5 6	4 2	2 8	1 8	1 2	8 5	1 5	- 4	7 8	8 2	2. 50
2	1-in. wood with 1-in. insulation	3 0	4 0	4 8	5 5	6 0	6 2	6 1	5 8	5 1	4 4	3 7	3 0	2 5	1 7	2 6	6 0	4. 11
3	4-in. h.w. concrete	2 9	3 9	4 8	5 6	6 2	6 5	6 4	6 1	5 4	4 4	3 6	3 0	2 4	1 7	0 6	6 5	4. 83
4	2-in. h.w. concrete with 1-in. insulation	2 0	3 5	3 9	3 3	4 6	4 7	4 6	4 4	4 4	4 1	3 8	3 5	3 2	1 8	1 3	4 7	6. 94
5	1-in. wood with 2-in. insulation	2 5	3 3	4 1	4 8	5 3	5 7	5 7	5 6	5 2	4 6	4 0	3 4	2 9	1 8	5 7	5 2	4. 21

	insulation																		
6	6-in. h.w. concrete	1 6	2 2	2 9	3 6	4 2	4 8	5 2	5 4	5 4	5 1	4 7	4 2	3 7	2 0	7 4	5 7	4 4	6. 17
7	2-in. wood with 1-in. insulation	1 8	2 1	2 5	3 0	3 4	3 8	4 1	4 3	4 4	4 4	4 2	4 0	3 7	2 1	1 5	4 4	2 9	6. 89
	insulation																		
8	8-in. h.w. concrete	1 5	1 7	2 0	2 5	2 9	3 4	3 8	4 2	4 5	4 6	4 5	4 4	4 2	2 1	1 4	4 6	3 2	7. 51
9	4-in. h.w. concrete with 1-in. (or 2-in.) insulation	2 4	2 7	2 9	3 2	3 4	3 6	3 8	3 8	3 8	3 7	3 6	3 4	3 3	1 9	2 0	3 8	1 8	11 .5 8
10	2.5-in. wood with 2-in. insulation	2 0	2 2	2 5	2 8	3 2	3 5	3 8	4 0	4 1	4 1	4 0	3 9	3 7	2 1	1 8	4 1	2 3	6. 98
	insulation																		
11	Roof Terrace System	2 3	2 3	2 5	2 6	2 8	2 9	3 1	3 2	3 3	3 3	3 3	3 3	3 2	2 2	2 2	3 3	1 1	16 .3 6
12	6-in. h.w. concrete with 1-in. (or 2-in.) insulation	2 3	2 5	2 6	2 8	3 0	3 2	3 3	3 4	3 4	3 4	3 4	3 3	3 2	2 1	2 0	3 1	1 4	16 .2 6
	insulation																		
13	4-in. wood with 1-in. (or 2-in.) insulation	2 1	2 2	2 2	2 4	2 5	2 7	3 0	3 2	3 4	3 5	3 6	3 7	3 6	2 3	2 1	3 7	1 6	9. 64
14	1-in. (or 2-in.) insulation																		

Source: ASHRAE,1980

Calculating Latitude month correction

Month	HOR
Dec	-13
Jan/Nov	-11
Feb/Oct	-7
March/Sep	-3
Apr/Aug	0
May/Jul	1
Jun	1

Source: ASHRAE, 1980

Calculating cooling load temperature difference for roof taking month reference June/July

S.N	Direction	CLTD	K	LM	To
1	Horizontal	65	0.585	1	29

So with the above procedure heat gain through the roof =49654.7

Summary of cooling load

S.N	Heat Source	Sensible Heat	Latent Heat
1	Glass	78608	
2	Wall	10437.422	
3	Door	397.2	
4	Light	6324	
5	People	2450	1550
6	Appliances	12345.4	1540
7	Ventilation	2078.8768	5798.304512
8	Infiltration	4241.16	7754.72544
9	Roof	49654.71225	
10	Total	166536.7711	16643.02995
11	Total	183179.801	
12	Tonnage required	15.26498342	

The summary of the cooling load of each floor is tabulated in the table below.

S.N	Floor	Tonnage Required
1	Ground	20.04
2	First	15.63
3	Second	14.85
4	Third	14.85
5	Fourth	15.26
6	Total	80.63

4.2 Calculation of Heating Load

4.2.1 Calculation of Heating Load for Ground Floor

In calculating cooling load, a cooling load temperature difference is taken into consideration but while calculating heating temperature difference is calculated and taken into account during the calculation process.

Design Inside Temperature(Ti)	77.2
Design Outside Temperature(To)	52.2
Temperature Difference(TD)	25

The above temperature is mentioned in the Fahrenheit unit and the temperature difference is found to be 25⁰F.

Description of a wall in the project

S.N	Description	Wall Group
-----	-------------	------------

1	8 in common brick	B
---	-------------------	---

Now calculating heat loss through a wall

Item	Orientation	Area	Factor(U)	TD	Heat Gain(BTU/HR)
Wall	East	0	0.302	25	0
Wall	West	350	0.302	25	2642.5
Wall	South	410	0.302	25	3095.5
Wall	North	750	0.302	25	5662.5

2) Heat Loss Through Glass

i. Conduction

Item	Orientation	Area	Temperature Difference	Factor(U)	Heat Gain(BTU/HR)
Glass	East	360	25	1	9000
Glass	South	340	25	1	8500
Glass	West	10	25	1	250

Here, in heat loss, only heat loss through conduction is taken into consideration.

3) Heat Loss through Glass Door

Item	Orientation	Area	Temperature Difference	Factor(U)	Heat Gain(BTU/HR)
Glass	West	40	25	1	1000

4) Heat Gain through Light

Length	51	ft
Breadth	36	ft
Area	1836	sq ft

Lux	500	
Watt	8	W
Luminaire efficiency	120	lumen per watt

Number=EA/(Ø*UF*MF)

UF	0.6
MF	0.8
N=	185.0787347

	186
Total wattage(Watt)	1488
Heat gain (Btu/hr.)	6324

5) Heat Gain through Appliances

S.N	Description	Wattage	Qty	Total Wattage
1	Desktop	77	2	154
2	Printer	130	1	130
3	Dispenser	350	1	350
4	Oven	1030	1	1030

For coffee maker,

Sensible heat(Watt)	1050
Latent heat (Btu/hr.)	1540

Total heat gain from appliance

Total wattage	2714
Heat gain from sensible heat	9227.6
Latent heat gain	1540
Total (Btu/hr.)	10767.6

6) Heat Loss through Ventilation

Calculating CFM

CFM/Person	7.5
Occupancy density (per 1000 sq. ft.)	40
Area	1836
No of people	73.44
Total no of people	74
CFM as standard for floor	0.06
CFM for floor	110.16
CFM for people	555
Total CFM	665.16

From the above CFM, sensible heat gain is 18291.9Btu/hr.

For Latent Heat Gain

Outside Condition		
DB	WB	HR
45	42	0.00496

Inside Condition		
DB	RH	HR
72	50	0.00837
dW		0.00341

Latent heat gain=10978 Btu/hr

7) Heat Loss through Infiltration

Volume details of the area

Area(Sq ft)	1836
Height(Ft)	10
Volume(Ft3)	18360

Calculating CFM with air change method, where $CFM = Volume * Na / 60$

Where Na=No of air change

So required CFM=214.2

Sensible heat gain =4241.16 Btu/hr.

For latent heat outside condition

DB	WB	HR
45	42	0.000496

For latent heat inside condition

DB	RH	HR
72	50	0.00837
dW		0.007874

So heat transfer through latent is 3535.23 Btu/hr.

9) Heat Loss through Floor

The floor type is heavy-weight concrete and 5 inches. Its falls on the group type E and from ASHRAE its U value is 0.585 and of floor area 1836 sq ft.

Calculating heat transfer through the floor, the heat transfer through the floor is 26851.5 Btu/hr.

Summary of heat transfer through a different source

S.N	Heat Source	Sensible Heat	Latent Heat	Remark
1	Glass	17750		Heat loss
2	Wall	11400.5		Heat loss
3	Glass Door	1000		Heat loss
4	Light	6324		Heat Gain
5	Appliances	9227.6	1540	Heat Gain
6	Ventilation	18291.9	10978.067	Heat loss
7	Infiltration	4241.16	3535.2425	Heat loss
8	Down floor	26851.5		Heat loss

So the heat loss of the system is 94048.37 and heat gain is 17091.6. The net heat required for the system is 76956.77 Btu/hr which is 6.41TR.

4.2.2 Calculation of Heating Load for First, Second, Third, and Fourth Floor

With a similar process heating load is calculated and summaries for the different floors are

S.N	Floor	Tonnage Required
1	Ground	6.413064099
2	First	6.111258715
3	Second	5.411061634
4	Third	5.411061634
5	Fourth	6.83788002
6	Total	30.1843261

4.3 Calculation of Heating and Cooling Load from HAP 4.9 Software

In the HAP 4.9 software, the ground floor is considered and the heating and cooling load for the ground floor is 6.2TR and 19.0TR. In the HAP 4.9 software system sizing summary, hourly air system loads, hourly zone loads, and system psychometrics is calculated and a details report is shown in Annex.

4.4 Design of Ground Source Heat Pump System

4.4.1 Selection of Indoor Unit

For the design first floor is taken into consideration and the cooling load required is 19.42TR. So duct able type three indoor unit is selected of size 8TR, 8TR, and 4TR. Details are shown in the figure below

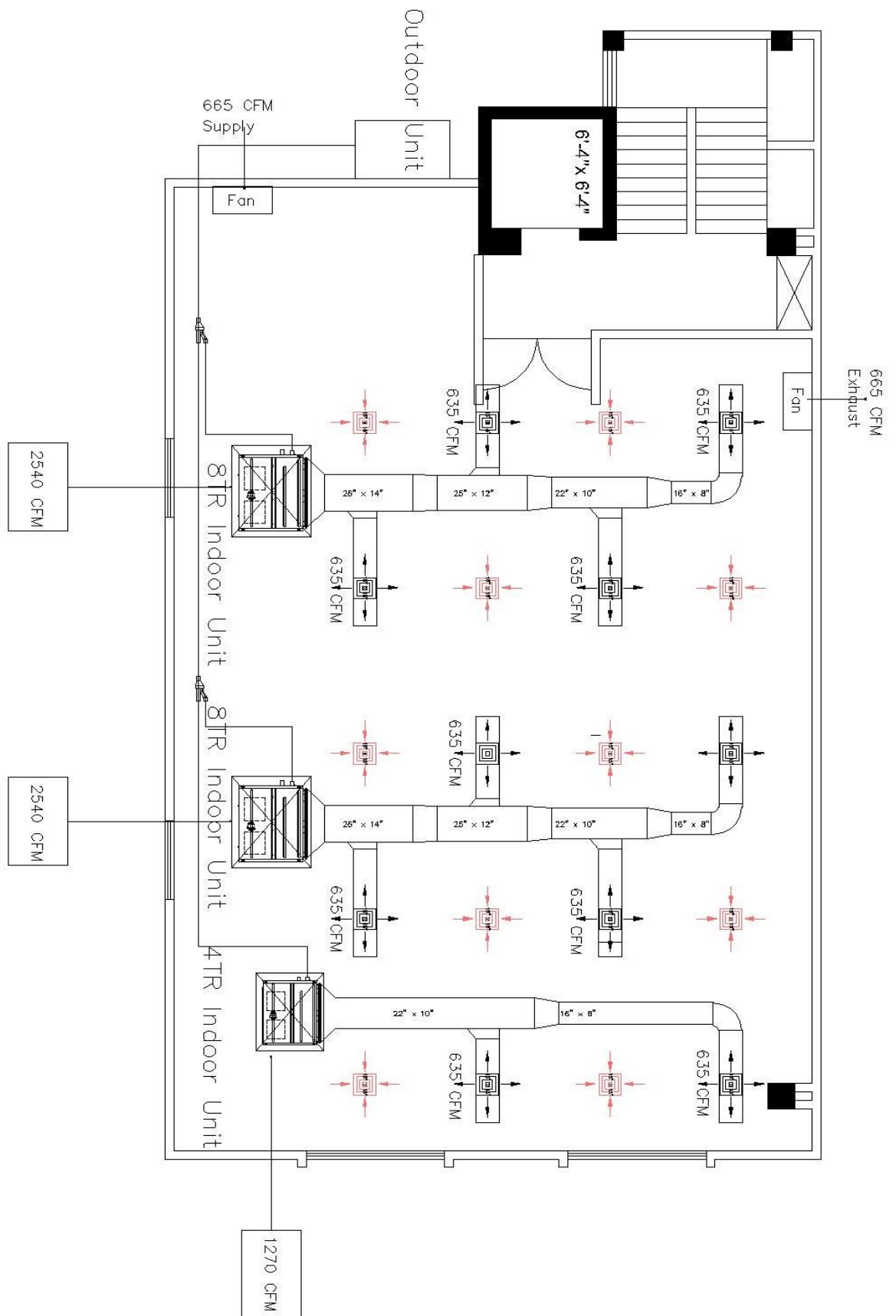


Figure 4.1 Systematic Diagram of Indoor Units

4.4.2 Selection of Outdoor Unit

For the outdoor unit, the DVMS Water system is selected. System specification model no AM200FXWANR/EU and technical specification of the system is:

HP -20

Cooling KW-56

Heating KW-6

4.5 Calculation of Vertical Length for Heating and Cooling

4.5.1 Length for Cooling

To calculate vertical bore length of cooling 16 TR load is taken into consideration

Cooling load(Btu/hr)	192000
Heating load(Btu/hr)	84000

For the design month maximum temperature month is taken into consideration i.e. July and August

Design Month	July and August
Part load factor(PLFm)	0.32

For the drilling of vertical bore length, a 5-inch diameter is selected and of U type HDPE pipe diameter is 1 inch.

Borehole diameter	5 inch
U pipe diameter	1 inch

Geothermal Heat Pump Design Manual states that the supply fluid temperature should be 20 to 30 degrees Fahrenheit warmer than the ground temperature for cooling and 10 to 20 degrees Fahrenheit cooler for heating (McQuay, 2002).

Ground Temperature(Tg)	57.2
Entering liquid temperature	87.2
Leaving liquid temperature	102.2

From the technical catalog COP of the DVMS water system is analyzed and described in the table.

COP cooling	5.2
COP Heating	5.8

Equivalent Full load hours in Cooling(EFLHc)	
EFLHc	1155

Equivalent Full load hours in Heating(EFLHh)	
EFLHh	585

Heat Transfer Rate in cooling and heating and net annual heat to and from the ground	
Cooling load	192000
Qcond	228923.0769

Heating load	84000
Qevap	69517.24138

Calculating net annual heat transfer to and from the ground, the result is shown in the table below

Qa(Btu/hr)	-25540.93238
------------	--------------

Calculating thermal resistances of the ground for the three prescribed heat pulses

Thermal diffusivity(α)	1.39
T1	3650
T2	3680
Tf	3680.167
Diameter of bore(in)	5
Diamter of U pipe(in)	1

Fof	117859.5563
Gf	0.978

Fo1	966.1162752
G01	0.614

Fo2	5.3482752
Go2	0.1955

Ground Conductivity(k)(Btu/h.ft.F)	1.7
Rga	0.214117647

Rgm	0.246176471
Rgst	0.115

Calculating the required flow through each U tube for the required cooling

Flow/U tube(gpm)	1.024
Gpm	2

So for the system, 2 GPM is taken.

For the system grout value of 0.75 is taken into consideration. For the resistance using interpolation, resistance is calculated

Grout value	Thermal resistance
0.4	0.39
0.8	0.23

Using Interpolation

Rb is calculated at 0.25.

For the system, the short circuit loss factor and temperature are taken into consideration and shown in the table

Short circuit heat loss factor	1.06
Temperature penalty	2

So calculating the required bore length for cooling using equation 16

The required length for cooling is 3060.25ft. So using a 5*4 array of the grid. The required length for each bore is 153 ft.

4.5.2 Length for Heating

To calculate vertical bore length of cooling 16 TR load is taken into consideration

Cooling load(Btu/hr)	192000
Heating load(Btu/hr)	84000

For the design month maximum temperature month is taken into consideration i.e.

July and August

Design Month	July and August
Part load factor(PLFm)	0.31

For the drilling of vertical bore length, a 5-inch diameter is selected and of U type HDPE pipe diameter is 1 inch

Borehole diameter	5 inch
U pipe diameter	1 inch

Ground Temperature(T_g)	57.2
Entering liquid temperature	42.2
Leaving liquid temperature	51.2

From the technical catalog COP of DVMS water system is analyzed and described in the table

COP cooling	5.2
COP Heating	5.8

Equivalent Full load hours in Cooling(EFLHc)	
EFLHc	1155

Equivalent Full load hours in Heating(EFLHh)	
EFLHh	585

Heat Transfer Rate in cooling and heating and net annual heat to and from the ground

Cooling load	192000
Q_{cond}	228923.0769

Heating load	84000
Q_{evap}	69517.24138

Calculating net annual heat transfer to and from the ground, the result is shown in the table below

Q_a (Btu/hr)	-25540.93238
----------------	--------------

Calculating thermal resistances of the ground for the three prescribed heat pulses

Thermal diffusivity(α)	1.39
T1	3650

T2	3680
Tf	3680.167
Diameter of bore(in)	5
Diamter of U pipe(in)	1

Fof	117859.5563
Gf	0.978

Fo1	966.1162752
G01	0.614

Fo2	5.3482752
Go2	0.1955

Ground Conductivity(k)(Btu/h.ft.F)	1.7
Rga	0.214117647
Rgm	0.246176471
Rgst	0.115

Calculating the required flow through each U tube for the required cooling

Flow/U tube(gpm)	1.71
Gpm	2

So for the system, 2 GPM is taken.

For the system grout value of 0.75 is taken into consideration. For the resistance using interpolation, resistance is calculated

Grout value	Thermal resistance
0.4	0.39
0.8	0.23

Using Interpolation

Rb is calculated at 0.25.

For the system, the short circuit loss factor and temperature is taken into consideration and shown in the table

Short circuit heat loss factor	1.06
Temperature penalty	2

So calculating the required bore length for heating using equation 17

The required length for cooling is 2055.2ft. So using a 5*4 array of the grid. The required length for each bore is 102 ft.

4.5.3 Systematic Arrangement of Bore Hole

The total bore length required for cooling and heating is 3060.25 ft and 2055.19ft respectively. 5*4 array is selected with a spacing of 3m. The required breadth and length is 30.36 ft and 40.2ft respectively and of an area of 1220.472 sq ft.

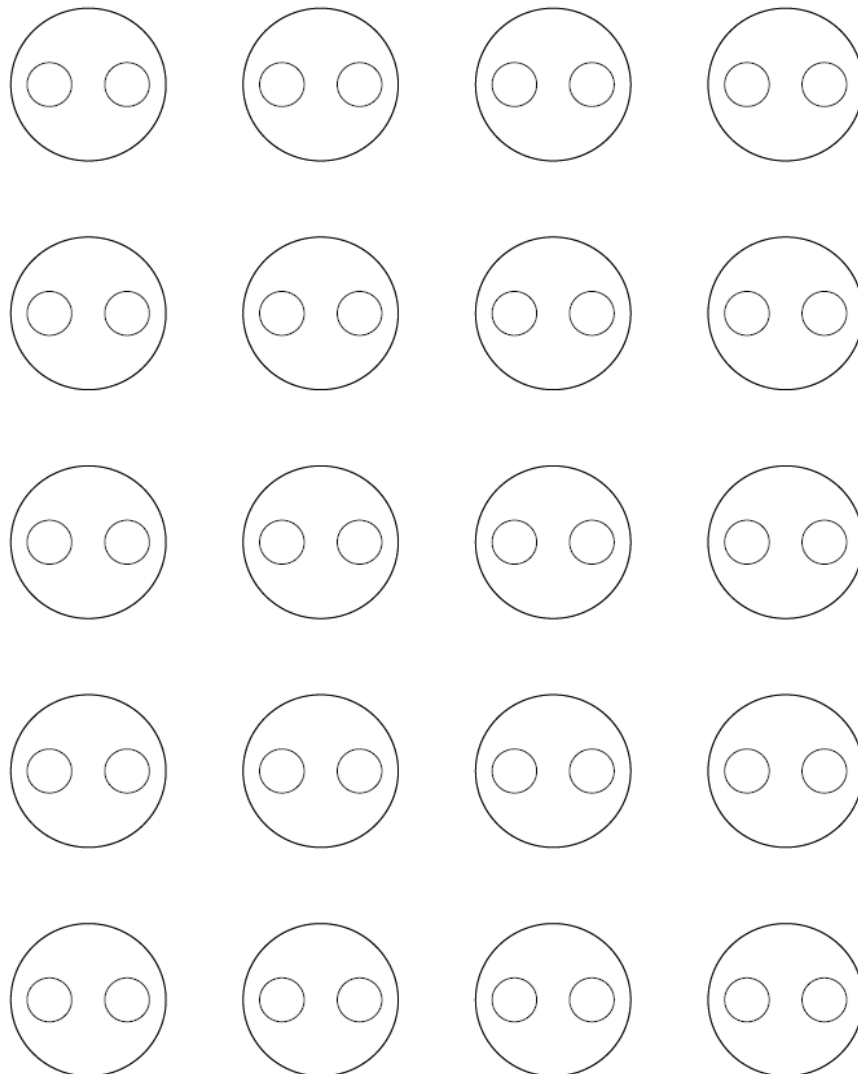


Figure 4.2 Showing Bore-hole Layout

4.6 Analysis of Outlet Temperature from Pipe through U Type Pipe

4.6.1 Analysis of Cooling Condition

Boundary Condition

- The flow of liquid: 2 GPM
- Ground Temperature: 14.02°C

For inlet temperature, the input temperature is given 30°C.

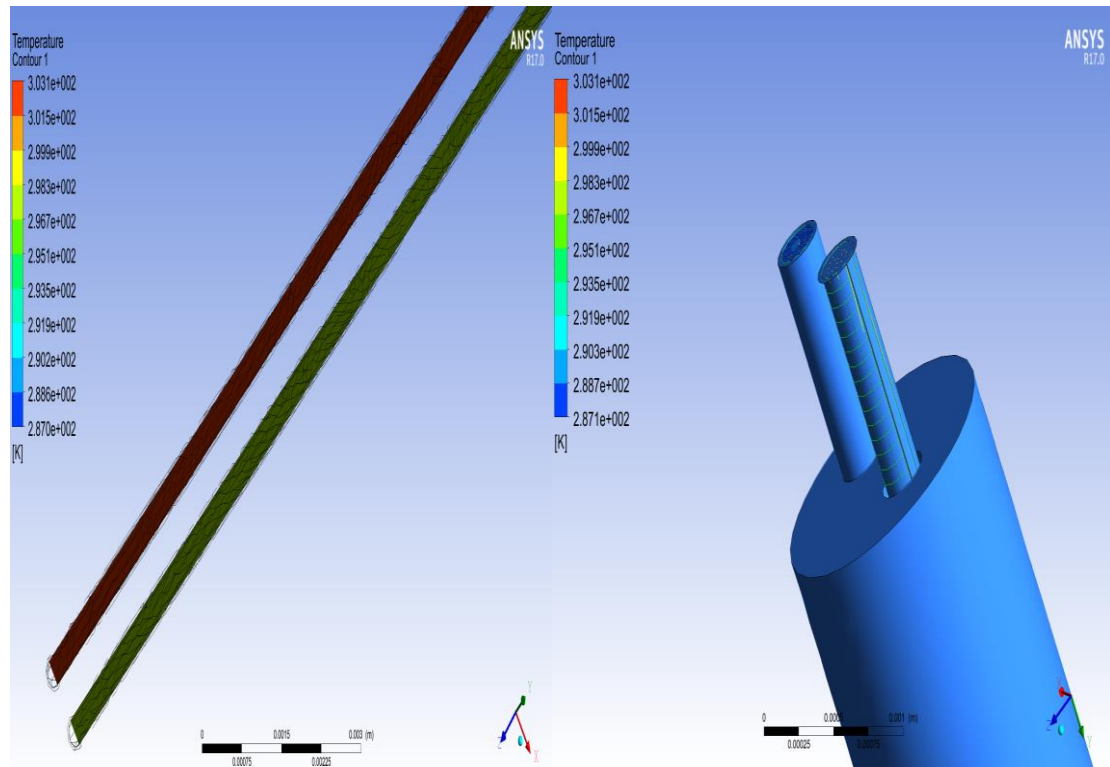


Figure 4.3 Analysis of Cooling Condition

From the analysis of outlet temperature in cooling conditions temperature in the outlet decreased by around 14.

4.6.2 Analysis for Heating

Boundary Condition

- The flow of liquid: 2 GPM
- Ground Temperature: 1.02°C

For inlet temperature, input temperature is given 5°C

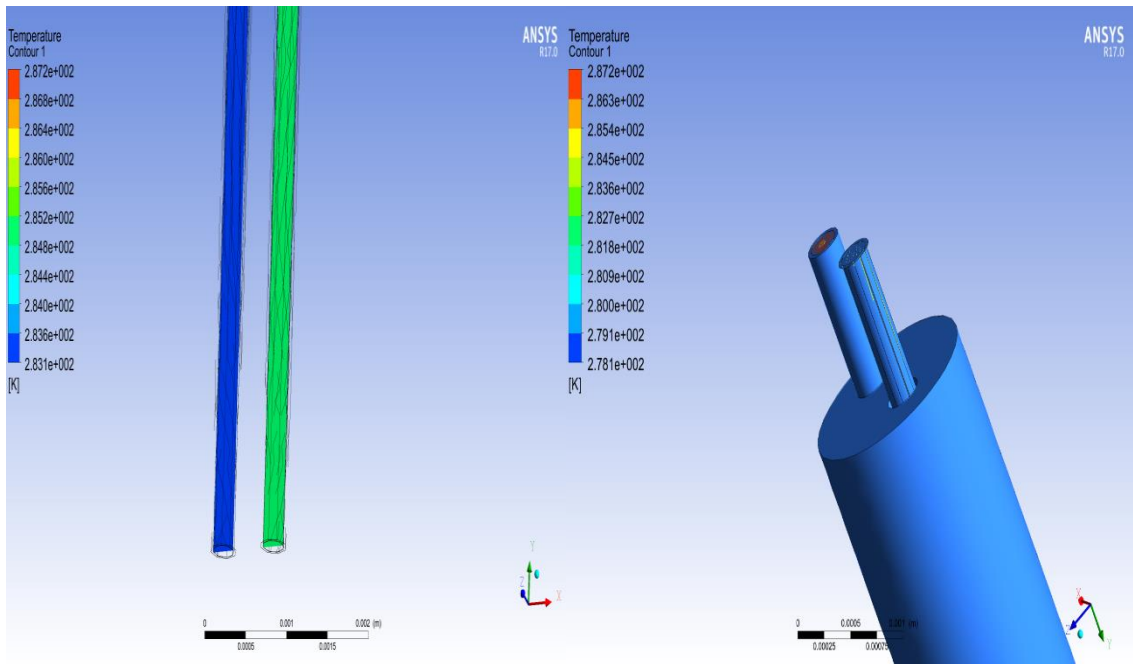


Figure 4.4 Analysis for Heating

For heating, the inlet temperature is given 5°C and in the outlet, the temperature change is around 9°C.

4.7 Systematic Drawing

In the building for heating and cooling with ground source heat pump system, according to the calculation for the indoor unit, we will be requiring 20TR suitable type indoor unit is implemented and details of the calculation are shown in the figure. For the heat exchange with the ground surface 20HP Samsung DVMS water system is used with a 5" diameter borehole of 153 feet in depth and an HDPE pipe of 1" diameter is used for U type heat exchanger. The total floor 5*4 grid is used that is total of 20 boreholes is required for the system. The details of the systematic diagram are shown in the figure below and calculation of HAP 4.9 in APPENDIX F.

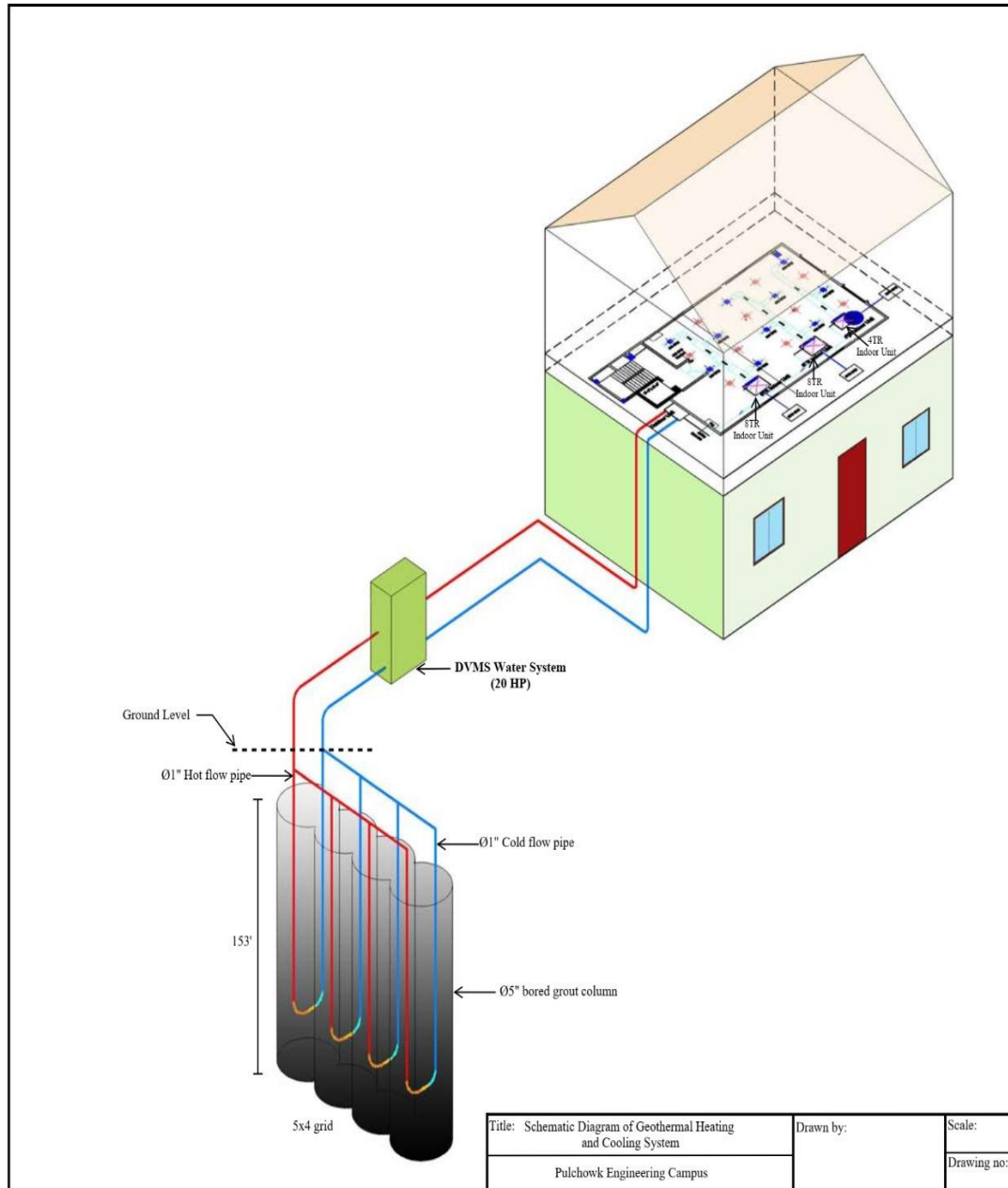


Figure 4.5 Systematic Drawing

4.8 Financial Analysis

In this section, the comparative cost analysis is analyzed between the existing excessively used VRF system and the GSHP system. In a comparison of this system due to the difference in COP, there is an energy demand reduction. With these energy saving in this section NPV, IRR, and Payback period is calculated.

Initial Investment

VRF system

The initial investment cost with the outdoor unit, indoor unit, and installation cost is revealed in table 4. 28, and details of cost are attested in Appendix B

Table 4.28 Investment Cost of VRF System

Details	VRF SYSTEM
System Cost	2903817.5

Ground Source Heat Pump System

In the Ground Source Heat Pump System, the indoor system cost and bore cost are shown in table 4. 29. In this system, there is an additional cost of bore cost which is appreciated high. Details of the GSHP system are attested in Appendix C

Table 4.29 GSHP Investment Cost

Details	DVMS WATER SYSTEM
System Cost	2389667.5
Bore Cost	3264462.05
Total Cost	5654129.55

For VRF System

While performing the financial analysis two scenarios is implemented in this section.

1. Scenario 1

In scenario 1 running an hour 18 hours is considered and loading of 100%. So operating cost is shown in table 7

Table 4.30 Scenario 1

Loading	100%
Running Hour Per Year	6570hr
KW hr VRF	92390.625
VRF Operating Cost	1108688

2. Scenario 2

3. In scenario 2 running an hour 24 hours is considered and loading of 100%. So operating cost is shown in table 8

Table 4.30 Scenario 2

Loading	100%
Running Hour Per Year	8760hr
KWhr VRF	123187.5
VRF Operating Cost	1478250

For GSHP system

While performing the financial analysis two scenarios is implemented in this section

1. Scenario 1

In scenario 1 running an hour 18 hours is considered and loading of 100%. So operating cost is shown in table 7

Table 4.31 Scenario 1

Loading	100%
Running Hour Per Year	6570hr
KWhr GSHP	57083.61
GSHP Operating Cost	685003

2. Scenario 2

3. In scenario 2 running an hour 24 hours is considered and loading of 100%. So operating cost is shown in table 4.32

Table 4.32 Scenario 2

Loading	100%
Running Hour Per Year	8760hr
KWhr GSHP	76111.48
VRF Operating Cost	913338

For Scenario 1

In a comparison between VRF and GSHP systems, there is

Annual Energy Saving: 35307 KWhr

Annual Operating Cost Saving: Rs 423684

Discount Rate : 12 % (NRB, 2020)

Additional Investment: : 2750312.05

So with the above data

The net present value is Rs 414371.75, an internal rate of return of 14%, and a payback period of 12.3 years.

For Scenario 2

In comparison between VRF and GSHP systems, there is

Annual Energy Saving : 47076 KWhr

Annual Operating Cost Saving : Rs 564912

Discount Rate : 12 % (NRB, 2020)

Additional Investment : 2631943.3

So with the above data

Net present value is Rs 1469266.34, an internal rate of return of 20%, and a payback period of 7.7 years.

Details financial analysis of two systems VRF system and GSHP system of two scenarios is attested in Appendix B and Appendix C

CHAPTER FIVE: CONCLUSION AND RECOMMENDATION

5.1 Conclusion

This research provides details system of the GSHP system for commercial buildings. For the commercial buildings heating and cooling load is calculated. The cooling and heating load for the commercial building is calculated at 80.63TR and 30.18TR respectively. These results show that the cooling load in Puchowk is almost 2.67 times higher than the heating load. The ground temperature is 14.02. So due to the higher load for cooling the lower ground temperature favors the cooling. So for the GSHP system, vertical bore length is calculated. The required bore length for cooling and heating are 4460.57ft and 1925.94ft. By arranging the array of 5*4bore holes, the individual length of the borehole is 153 ft. for cooling and 102ft for heating. So these results show that there is extra potential for the heating system in commercial buildings.

In this study heating and cooling load is also calculated by HAP 4.9 software. In HAP 4.9 software only the ground floor is taken into consideration. From the manual calculation for the ground floor, there is a cooling and heating load of 20.04TR and 6.4TR respectively. From HAP 4.9 it is calculated that the cooling and heating load for the ground are 19.0TR and 6.16TR respectively. Regarding manual calculation, the HAP 4.9 calculation result is 5.2% less in cooling load and 3.75% less in heating load.

For the GSHP system for the ground floor, the DVMS water system of the 20HP outdoor unit is selected and 2 units of 8TR indoor unit with duct are selected and one unit of 4TR indoor unit with duct is selected.

In ANSYS with the flow of liquid in U type, change in temperature is analyzed and for cooling condition, there is the temperature variation of 14⁰C and heating condition there is the temperature variation of 9⁰C

The financial analysis of the GSHP system is compared to the VRF system. The initial investment of the VRF system is NRs 2903817.5 and the initial investment cost of the DVMS Water system is NRs 5654129.6. The initial investment cost of GSHP is higher due to the boring of the vertical length of the U type of Heat exchanger. The cost of a borehole system is around 5654129.55 which is almost 57.7% of the total cost of the system. Due to the greater COP of the GSHP system, in scenario 1 running of 18hr per day annually there is a reduction of energy consumption by 35307KWhr. In scenario 2 running of 24hr per day, there is a reduction of energy consumption by 47076KWhr

annually. So this shows that the GSHP system will be feasible for the sector where there is high consumption of heating and cooling. In scenario 1 there is the payback period of 12.3 years with an IRR of 14% and in scenario 2 there is the payback period of 7.7 years with an IRR of 20%.

The above results clarify that the GSHP system will be feasible in commercial projects where there is high consumption of heating and cooling application. Implementation of GSHP systems assists to reduce energy demand resulting in lesser emission of greenhouse gases and results in good internal rate of return.

5.2 Recommendations

For GSHP ground temperature plays a vital role in the system, so the ground temperature should be properly measured for the area where heating and cooling application are required. In the GSHP system, almost 57.7% of the cost is of the borehole, so an optimized procedure should be implemented to reduce borehole costs and make them financially feasible.

In the building, most heat transfers occur from the window and wall. So to reduce heating and cooling load instead of normal glass it is recommended to use double glazing glass with a UPVC frame and insulation materials between the walls can be used to reduce the building heating and cooling load.

It is recommended to use the GSHP system instead of the current HVAC system where the running hour of the HVAC system is high. In an area having more running hours, there is a high internal rate of return and which also helps to reduce energy demand and greenhouse gas emissions.

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APPENDICES

APPENDIX A Weather Properties

Weather Properties - [Katmandu]

Design Parameters | Design Temperatures | Design Solar | Simulation

Region:	Asia/Pacific	Atmospheric Clearness Number	1.00
Location:	Nepal	Average Ground Reflectance	0.20
City:	Katmandu	Soil Conductivity	0.800 BTU/hr/ft/F
Latitude:	27.7 deg	Design Clg Calculation Months	Jan to Dec
Longitude:	-85.2 deg	Time Zone (GMT +/-)	5.8 hours
Elevation:	4388.0 ft	Daylight Savings Time	<input type="radio"/> Yes <input checked="" type="radio"/> No
Summer Design DB	89.0 °F	DST Begins	Apr 1
Summer Coincident WB	78.0 °F	DST Ends	Oct 31
Summer Daily Range	25.0 °F	Data Source:	
Winter Design DB	33.0 °F	User Modified	
Winter Coincident WB	27.3 °F		

OK Cancel Help

APPENDIX B Schedule Hour Profile and Assignments

Schedule Properties - [Appliance]

Schedule Type | Hourly Profiles | Assignments

Schedule Name:

Schedule Type:

- Fractional (People, Lighting, Equipment, Misc. Sensible, Misc. Latent, Ventilation Airflow, Service Hot Water Usage, Misc. Electric, Misc Fuel)
- Fan/Thermostat
- Utility Rate Time-of-Day

OK Cancel Help

APPENDIX C Selection of Wall Type

Wall Properties - [external 8" wall]

Wall Assembly Name: external 8" wall

Outside Surface Color: Dark Absorptivity: 0.900

Layers: Inside to Outside	Thickness in	Density lb/ft ³	Specific Ht. BTU/lb/F	R-Value hr-ft ² -F/BTU	Weight lb/ft ²
Inside surface resistance	0.000	0.0	0.00	0.68500	0.0
8-in common brick	8.000	120.0	0.20	1.58730	80.0
Cement Plaster	0.500	61.0	0.20	0.12000	5.1
Outside surface resistance	0.000	0.0	0.00	0.33300	0.0
Totals	8.500			2.73	85.1

Overall U-Value: 0.367 BTU/hr/ft²/F

OK Cancel Help

APPENDIX D Defining Project Library Space (Space Properties)

Space Properties - [General Details]

General | Internals | Walls, Windows, Doors | Roofs, Skylights | Infiltration | Floors | Partitions

Name: **General Details**

Floor Area: **1836.0** ft²

Avg Ceiling Height: **10.0** ft

Building Weight: **70.0** lb/ft²

Light Med. Heavy

OA Ventilation Requirements

Space Usage: **RETAIL: Mall common area**

OA Requirement 1: **7.5** CFM/person

OA Requirement 2: **0.06** CFM/ft²

Space usage defaults: ASHRAE Std 62.1-2007
Defaults can be changed via View/Preferences.

OK Cancel Help

APPENDIX E System Details

The image shows a software dialog box titled "Air System Properties - [VAV Package]". It has a standard Windows-style title bar with a close button. Below the title bar is a tabbed interface with four tabs: "General", "System Components", "Zone Components", and "Sizing Data". The "General" tab is currently selected and highlighted. The main area of the dialog contains four labeled input fields:

- Air System Name:** A text box containing the value "VAV Package".
- Equipment Type:** A dropdown menu with "Packaged Rooftop Units" selected.
- Air System Type:** A dropdown menu with "VAV" selected.
- Number of Zones:** A text box containing the value "1".

At the bottom of the dialog, there are three buttons: "OK", "Cancel", and "Help".

APPENDIX F HAP 4.9 Result

Air System Information

Air System Name	VAV Package	Number of zones	1
Equipment Class	PKG ROOF	Floor Area	1836.0 ft ²
Air System Type	VAV	Location	Katmandu, Nepal

Sizing Calculation Information

Calculation Months	Jan to Dec	Zone CFM Sizing	Peak zone sensible load
Sizing Data	Calculated	Space CFM Sizing	Individual peak space loads

Central Cooling Coil Sizing Data

Total coil load	19.0 Tons	Load occurs at	Aug 1500
Total coil load	228.1 MBH	OA DB / WB	89.0 / 78.0 °F
Sensible coil load	124.5 MBH	Entering DB / WB	78.1 / 67.7 °F
Coil CFM at Aug 1500	5876 CFM	Leaving DB / WB	55.0 / 54.2 °F
Max block CFM at Jul 2000	6830 CFM	Coil ADP	52.4 °F
Sum of peak zone CFM	6830 CFM	Bypass Factor	0.100
Sensible heat ratio	0.546	Resulting RH	59 %
ft ² /Ton	96.6	Design supply temp.	55.0 °F
BTU/(hr-ft ²)	124.3	Zone T-stat Check	1 of 1 OK
Water flow @ 10.0 °F rise	N/A	Max zone temperature deviation	0.0 °F

Supply Fan Sizing Data

Actual max CFM at Jul 2000	6830 CFM	Fan motor BHP	0.00 BHP
Standard CFM	5814 CFM	Fan motor kW	0.00 kW
Actual max CFM/ft ²	3.72 CFM/ft ²	Fan static	0.00 in w/g

Outdoor Ventilation Air Data

Design airflow CFM	831 CFM	CFM/person	11.24
CFM/ft ²	0.45 CFM/ft ²	CFM/person

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Aug 1500 COOLING OA DB / WB 89.0 °F / 78.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 33.0 °F / 27.3 °F		
ZONE LOADS	Details	Sensible (BTU/ hr)	Latent (BTU/h r)	Details	Sensible (BTU/ hr)	Latent (BTU/h r)
Window & Skylight Solar Loads	700 ft ²	12649	-	700 ft ²	-	-
Wall Transmission	1755 ft ²	4227	-	1755 ft ²	23827	-
Roof Transmission	0 ft ²	0	-	0 ft ²	0	-
Window Transmission	700 ft ²	7272	-	700 ft ²	28257	-
Skylight Transmission	0 ft ²	0	-	0 ft ²	0	-
Door Loads	50 ft ²	1632	-	50 ft ²	1850	-
Floor Transmission	1836 ft ²	7617	-	1836 ft ²	0	-
Partitions	0 ft ²	0	-	0 ft ²	0	-
Ceiling	0 ft ²	0	-	0 ft ²	0	-
Overhead Lighting	2190 W	6945	-	0	0	-
Task Lighting	9180 W	29114	-	0	0	-
Electric Equipment	2714 W	8616	-	0	0	-
People	74	14597	15170	0	0	0
Infiltration	-	3244	8578	-	8572	0
Miscellaneous	-	0	39007	-	0	0
Safety Factor	20% / 20%	19183	12551	0%	0	0
>> Total Zone Loads	-	115095	75306	-	62506	0
Zone Conditioning	-	114791	75306	-	49322	0
Plenum Wall Load	0%	0	-	0	0	-

Plenum Roof Load	0%	0	-	0	0	-
Plenum Lighting Load	0%	0	-	0	0	-
Return Fan Load	5876 CFM	0	-	831 CFM	0	-
Ventilation Load	831 CFM	9749	28301	831 CFM	24651	0
Supply Fan Load	5876 CFM	0	-	831 CFM	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads		- 124540	103607		- 73972	0
Central Cooling Coil		- 124540	103609		- 0	0
Terminal Reheat Coils		- 0	-		- 73972	-
>> Total Conditioning		- 124540	103609		- 73972	0
Key:	Positive values are clg loads			Positive values are htg loads		
	Negative values are htg loads			Negative values are clg loads		

APPENDIX G Cost Details of VRF System

Project: Office Building					
Location: Pulchowk					
Date: 2022/2/18					
Sub: Bills of Quantities: VRF Air-conditioning works					
ABSTRACT OF COSTS					
HVAC/AC WORKS (VRF System-Samsung Brand)					
SUPPLY & INSTALLATION OF AIR-CONDITIONING SYSTEM			OFFERED RATE & AMOUNT		
S.N	OUTDOOR, INDOOR UNITS & ACCESSORIES	Qty	Units	Rate (NRs)	Amount (NRs)
				In figure	
A.	High Side Work				
1	<p><u>Outdoor air-cooled VRF/VRV condensing units.</u></p> <p>Supply of a digital compressor with a VRF/VRV system (with R-410 A) that may be connected to several indoor units. The outside unit will be a factory-assembled weatherproof shell made of thick gauge mild steel panels that have been epoxy-coated. The item must be factory connected and tested with all of the appropriate controls. The compressor must have a digital scroll type controller that is very efficient. Anti-corrosion resin must be applied to the aluminum fins. To guarantee steady operation with extended refrigerant pipes, the unit must be supplied with an oil recovery</p>				

	system. The unit must be able to work with a BMS system. The Capacities, Refrigerant, etc, shall be as follows:				
a.	Nominal Cooling Capacity: 56 kW Model (AM200FXVAGH/EU)- 20 HP	1	Sets	1,350,000	1,350,000
	Nominal Heating Capacity: 63 kW				
	Power Source: 3 Phase 4 wire system: 380-415 V, 50 Hz				
2	Indoor VRF Ceiling Cassette Unit/ High Wall:				
	Supply, install and test of Ceiling Suspended unit, matched to the outdoor unit, with corrosion-resistant galvanized sheeting, 3-speed fan, inbuilt condensate pump, conveniently accessible and cleanable filter, connections for remote control of fan and space temperature, and all mounting and attaching accessories				
a.	Nominal Cooling Capacity: 28 kW (HSP Duct)	2	Sets	320,000	640,000
	Nominal Heating Capacity: 31.5 Kw (AM280FNHDEH/EU) - 8 TR				
	Power Source: 1 Phase: 220 V, 50 Hz				

b.	Nominal Cooling Capacity: 14 kW (MSP Duct)	1	Sets	135,000	135,000
	Nominal Heating Capacity: 16 kW (AM140FNMDEH/EU) - 4 TR				
	Power Source: 1 Phase: 220 V, 50 Hz				
3	Supply, installation, testing, and commissioning of, Branch kit with insulation				
	(Model for Y joint) size-2 (for both suction/liquid line) (MXJ- YA2512M)	1	Sets	11,000	11,000
	(Model for Y joint) size-3 (for both suction/liquid line) (MXJ- YA2815M)	1	Sets	13,500	13,500
4	Corded Remote controller with controls of temperature set up airflow mode (MWR-SH00N)	3	Sets	7500	22,500
Sub Total (A)					2,172,000
B.	Low Side Works				
1	Pressure, supply, and installation Interconnecting refrigerant copper pipework between each set of interior and outdoor units are tested and commissioned.				
	9.52	15	Rmt	850.00	12,750
	12.7	20	Rmt	1,150	23,000
	15.88	15	Rmt	1,350	20,250
	19.05	20	Rmt	1,850	37,000

	22.2	25	Rmt	2,450	61,250
	28.58	25	Rmt	2,950	73,750
2	Supply and installation of closed-cell elastomeric nitrile rubber tubular insulation for refrigerant copper pipe between each set of interior and outdoor units (19mm/13mm thick).				
	9.52	15	Rmt	110	1,650
	12.7	20	Rmt	120	2,400
	15.88	15	Rmt	130	1,950
	19.05	20	Rmt	150	3,000
	22.2	25	Rmt	150	3,750
	28.58	25	Rmt	200	5,000
3	INDOOR UNIT CONDENSATE PIPE				
a.	CPVC drain pipe complete with fittings, MS supports, and hangers, and 9mm thick close cell insulation was provided and installed.	3	Lot	4,500	13,500
4	Installation, Testing & Commissioning of Indoor unit.	3	Sets	4,000	12,000
5	Loading and Unloading of Outdoor Unit	1	Sets	22,500	22,500
6	Additional refrigerant R410A	1	Lot	22,500	22,500

7	Vacuum and Pressure Testing of Entire System	1	Lot	12,500	12,500
8	Supply and install of Control Wire for communication between IDU & ODU.	3	Sets	3,000	9,000
9	Control Panel to Operate Outdoor Units with Voltage Preventer ,single Phase preventor (OV, UV, SC)	1	Sets	60,000	60,000
Sub Total (B)					397,750
Total(A+B)					2,569,750
13% Vat					334,067
Grand Total(with VAT)					2,903,817

APPENDIX H Cost Details of DVMS Water System

Project: Office Building					
Location: Pulchowk					
Date: 2022/2/18					
Sub: Bills of Quantities: DVMS Water Air Conditioner Work					
ABSTRACT OF COSTS					
HVAC/AC WORKS (VRF System- Samsung Brand)					
SUPPLY & INSTALLATION OF AIR-CONDITIONING SYSTEM			OFFERED RATE & AMOUNT		
S.N	OUTDOOR, INDOOR UNITS & ACCESSORIES	Qty	Units	Rate (NRs)	Amount (NRs)
				In figure	
A.	High Side Work				
1	<u>DVMS WATER Outdoor Unit,</u>				
a.	Nominal Cooling Capacity: 56 kW Model (AM200MXWANR/EU)- 20 HP	1	Sets	895000	895000
	Nominal Heating Capacity: 63 kW				
	Power Source: 3 Phase 4 wire system: 380-415 V, 50 Hz				
2	Indoor VRF Ceiling Cassette Unit/ High Wall:				
	Ceiling Suspended unit, matched to outdoor unit, with galvanized sheeting treated for corrosion, 3- speed fan, inbuilt condensate pump mandatory, readily accessible and cleanable filter, connections for remote control of fan and space temperature, all mounting and fastening accessories				

a.	Nominal Cooling Capacity: 28 kW (HSP Duct)	2	Sets	320000	640000
	Nominal Heating Capacity: 31.5 Kw (AM280FNHDEH/EU) - 8 TR				
	Power Source: 1 Phase: 220 V, 50 Hz				
b.	Nominal Cooling Capacity: 14 kW (MSP Duct)	1	Sets	135000	135000
	Nominal Heating Capacity: 16 Kw (AM140FNMDEH/EU) - 4 TR				
	Power Source: 1 Phase: 220 V, 50 Hz				
3	Supply, installation, testing, and commissioning of, Branch kit with insulation				
	(Model for Y joint) size-2 (for both suction/liquid line) (MXJ- YA2512M)	1	Sets	11000	11000
	(Model for Y joint) size-3 (for both suction/liquid line) (MXJ- YA2815M)	1	Sets	13500	13500
4	Corded Remote controller with controls of temperature set up airflow mode (MWR-SH00N)	3	Sets	7500	22500
Sub Total (A)					1,717,000
B.	Low Side Works				
1	Pressure, supply, and installation Interconnecting refrigerant copper pipework between each set of interior and outdoor units is tested and commissioned.				
	9.52	15	Rmt	850.00	12,750.00
	12.7	20	Rmt	1,150.00	23,000.00
	15.88	15	Rmt	1,350.00	20,250.00
	19.05	20	Rmt	1,850.00	37,000.00

	22.2	25	Rmt	2,450.00	61,250.00
	28.58	25	Rmt	2,950.00	73,750.00
2	Supply and installation of closed-cell elastomeric nitrile rubber tubular insulation for refrigerant copper pipe between each set of interior and outdoor units (19mm/13mm thick).				
	9.52	15	Rmt	110	1,650
	12.7	20	Rmt	120	2,400
	15.88	15	Rmt	130	1,950
	19.05	20	Rmt	150	3,000
	22.2	25	Rmt	150	3,750
	28.58	25	Rmt	200	5,000
3	INDOOR UNIT CONDENSATE PIPE				
a.	CPVC drain pipe complete with fittings, MS supports, and hangers and 9mm thick close cell insulation was provided and installed.	3	Lot	4,500	13,500
4	Installation, Testing & Commissioning of Indoor unit.	3	Sets	4,000	12,000
5	Loading and Unloading of Outdoor Unit	1	Sets	22,500	22,500
6	Additional refrigerant R410A	1	Lot	22,500	22,500
7	Vacuum and Pressure Testing of Entire System	1	Lot	12,500	12,500
8	Supply and install of Control Wire for communication between IDU & ODU.	3	Sets	3,000	9,000
9	Control Panel to Operate Outdoor Units with Voltage Preventer ,single Phase preventor (OV, UV, SC)	1	Sets	60,000	60,000

Sub Total (B)	397,750
Total(A+B)	2,114,750
13% Vat	274,918
Grand Total(with VAT)	2,389,668

APPENDIX I Financial Analysis of Scenario 1

Particulars /Years	0	1	2	3	4	5	6
Saving		423684	423684	423684	423684	423684	423684
Net Profit	- 27503 12	423684	423684	423684	423684	423684	423684
DF	1.00	0.89	0.80	0.71	0.64	0.57	0.51
PV of cashflow	- 27503 12	378289. 29	337758. 29	301569. 90	269258. 84	240409. .68	214651. 50
Cumulative PV	- 27503 12	- 2372022 .71	- 2034264 .42	- 1732694 .52	- 1463435 .68	- 122302 6	- 100837 4.50

7	8	9	10	11	12	13	14
423684	423684	423684	423684	423684	423684	423684	423684
423684	423684	423684	423684	423684	423684	423684	423684
0.45	0.40	0.36	0.32	0.29	0.26	0.23	0.20
191653. 12	171118. 86	152784. 70	136414. 91	121799. 03	108749. 13	97097. 44	86694. 14
- 816721. 37	- 645602. 51	- 492817. 82	- 356402. 91	- 234603. 88	- 125854. 75	- 28757. 31	- 57936. 83

15	16	17	18	19	20
423684	423684	423684	423684	423684	423684
423684	423684	423684	423684	423684	423684
0.18	0.16	0.15	0.13	0.12	0.10
77405.48	69112.04	61707.18	55095.69	49192.58	43921.95

135342.31	204454.35	266161.53	321257.22	370449.80	414371.75
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Discount Rate	12%
Initial Investment	2750312
Present Value	3164683.753
NPV	414371.7525
IRR	14%
PBP	12.33171

APPENDIX J Financial Analysis of Scenario 2

Particulars/ Years	0	1	2	3	4	5
Saving		564912	564912	564912	564912	564912
Net Profit	- 275031 2	564912	564912	564912	564912	564912
DF	1	0.89	0.80	0.71	0.64	0.57
PV of Cashflow	- 275031 2	504385.7 1	450344.3 9	402093.2 0	359011.7 9	320546.2 4
Cumulative PV	- 275031 2	- 2245926. 29	- 1795581. 90	- 1393488. 69	- 1034476. 91	- 713930.6 7

6	7	8	9	10	11	12
564912	564912	564912	564912	564912	564912	564912
564912	564912	564912	564912	564912	564912	564912
0.51	0.45	0.40	0.36	0.32	0.29	0.26
286202.00	255537.50	228158.4 8	203712.9 3	181886.5 5	162398.7 0	144998.8 4
- 427728.67	- 172191.17	55967.32	259680.2 5	441566.7 9	603965.4 9	748964.3 3

13	14	15	16	17	18	19	20
564912	564912	564912	564912	564912	564912	564912	564912
564912	564912	564912	564912	564912	564912	564912	564912
0.23	0.20	0.18	0.16	0.15	0.13	0.12	0.10
129463. 25	115592. 19	103207.3 1	92149.38	82276.24	73460.92	65590.11	58562.60

878427. 58	994019. 77	1097227. 08	1189376. 46	1271652. 70	1345113. 63	1410703. 74	1469266. 34
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Discount Rate	12%
Initial Investment	2750312
Present Value	4219578.337
NPV	1469266.337
IRR	20%
PBP	7.754699823

APPENDIX K Originality Report

