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Thermodynamic and Thermoeconomic Analysis of Combined Geothermal Space Heating and Thermal Storage Using Phase Change Materials

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ABSTRACT

The present work discusses utilization of geothermal energy for space heating in combination with phase change energy storage system. Thermodynamics and thermoeconomics of the combined heating and thermal storing system were studied to show the scope of energy storage and cost savings. The analysis was done taking second law efficiency into consideration since the second law represents true flow of energy. The space heating is based on geothermal water that is used in a plate type heat exchanger to give heat to the secondary fluid which is passed through the radiators for room heating. Heat supply to the storage system was provided from the geothermal water at the exit of the heat exchanger. A computational model of the combined space heating and thermal storage system was developed and used to perform thermodynamic studies of the heat storage process and heating system efficiency at different times and ambient temperatures. The basis for these studies is daily variations in heating demand that is higher during the night than during the day. The results show the scope of utilization of phase change material for low ambient temperature conditions. Under proper conditions sufficient amount of exergy is stored during charging period at low ambient temperature to fulfill the day time heat load requirement. Under these conditions the cost flow rate of exergy storage is found to be lower than the radiator heating cost flow rate. Thus, the usage of exergy storage at low ambient temperature for heating at higher ambient temperature makes significant contribution in cost saving. The paper makes references too many studies on exergy analysis and thermal energy storage systems using phase change materials.

1. INTRODUCTION

Space heating is one of the well-known applications of geothermal energy utilization for decades. The usage of geothermal energy for space heating provides an economical and a non-polluting way for achieving human comforts. In order to study a thermodynamic systems performance which can either involve heating or power generation, second law of thermodynamics plays an important role. The second law helps in better understanding of energy flow processes in addition to first law of thermodynamics. Exergy is the maximum theoretical useful work obtainable as the systems interact to equilibrium, heat transfer occurring with the environment only. Several studies have been conducted on exergy analysis of buildings. The concept of low exergy systems for heating and cooling have been proposed in IEA ECBCS Annex 37(2000). An exergetic life cycle assessment for resource evaluation in the built environment was conducted by Meester et al (2009). Shukuya and Komuro (1996) applied concepts of entropy and exergy for investigating relation between building, passive solar heating and environment. Various results about patterns of human exergy consumption in relation to various heating and cooling systems were given by Saito and Sukaya (2001). Conclusions about inadequacy of energy conservation concept for understanding important aspects of energy utilization processes were made by Yildiz and Gungör (2009). The second law analysis is important in order to have an efficient utilization of the available resource.

1.1 Phase Change Material

One of the best solutions for the problem when supply and demand are out of phase is the usage of energy storage materials. The key to effective utilization of renewable energy sources is efficient and economical energy storage systems. One of the efficient ways of storing heat is the application of phase change materials (PCM). Such systems absorb and release heat energy as latent heat of the storing material with a change of phase. Various advantages of PCM over other storage systems helped gained its importance over the years. The study done by Adebiyi and Russell (1987) concluded two advantages derived from using a phase change material in a thermal energy storage system design. First advantage was the increase in second law efficiency of the system as compared to the one using sensible heat storage system. The second major advantage concluded was the reduced amount of storage material required. Since PCM is based on the principle of storing heat as latent energy of phase change, the energy stored is far higher than stored by sensible heating systems which stores energy equal to their specific heat capacity.

Various analysis of the latent heat storage based on the first law of thermodynamics can be found in various literature. The first law analysis helps us to obtain a workable design but not an optimum one. In order to consider the effect of time and the temperature at which heat is supplied second law analysis is required. According to Bejan et al. (1996) an optimal system which a designer can develop having the least irreversibility is based on minimization of entropy generation. The application of second law analysis for studying latent heat storage was studied by Bjurstrom and Carlsson (1985) and Adebiyi and Russell (1987) and later added by Bejan (1996). A study was done by El-Dessouky and Al-Juwayhel (1997) for investigating the effect of different variables on the entropy generation number defined by Bejan et al. (1996). The analysis considered the case with the storage material exchanging heat at a constant melting point. For analysis two commonly available storing materials, paraffin wax and calcium chloride were considered with air or water as the heating fluid.

A number of materials exists which can be used as PCM over a wide range of temperatures. A list of such materials can be found in the literature (Abhat, 1983). The selection of a PCM for energy storage is based on the fulfillment of the criteria such as high latent

heat capacity, non-corrosiveness, high thermal conductivity, and it should be non-toxic without deposition or super cooling. The transition temperature of the phase change material is decided by the room temperature required.

1.2 Importance of PCM to Geothermal Energy Storage

Studies on analysis of PCM, both numerical and experimental, have been reported in the literature (Farid et al., 2004). The work is mainly focused on solar energy storage. It is important to know the need to store geothermal energy where the mass flow itself can be controlled. The answer to the question depends upon the conditions for storage application. For an existing heating system using radiators and heat exchangers, an important fact discussed in the literature (Karlsson and Ragnarsson, 1995) is the variation of geothermal exit temperature with outside ambient temperature for a fixed design network. It is found that as the ambient temperature decreases, the exit temperature of geothermal heating network increases. The increase in exit temperature reflects increase in useful heat loss. In terms of second law efficiency, the decrease in ambient temperature causes increase in exergy loss to the atmosphere. Surely the application of energy storage can provide a means of saving energy which can be used in the off peak time when the demand is low. Stored heat from the geothermal exit fluid at night can be used for fulfilling day time requirement. The other advantage of thermal storage for geothermal application is found in areas with limited flow and having high heat demand. Heat energy in such cases can be stored during off peak or day time and can be utilized for peak load demands during night. Managing the heating from limited flow will also help save on drilling cost by reducing the need for new wells.

1.3 Thermoeconomics

Thermoeconomic or exergoeconomic analysis considers both thermodynamic and economic principles for improving systems performance. Exergoeconomic analysis helps in quantifying the exergy losses in terms of monetary losses. According to Bejan et al. (1996) exergoeconomics is defined as the branch of engineering that combines exergy analysis and economic principles to provide the system designer or operator with information not available through conventional energy analysis and economic evaluations, but crucial to the design and operation of a cost effective system. Application of exergoeconomic analysis can be found in the literature for processes involving heat transfer and power generation. Exergetic cost analysis for space heating using ground source heat pump system was done by Jingyana et al (2010). The work shows the sensitivity of various sub-systems to unit exergy cost. Application of exergoeconomic analysis for geothermal district heating system was presented by Oktay and Dincer (2009) and reviewed by Hepbasli (2010) using energy, exergy and economic aspects. Performance evaluation of geothermal heating system and case studies were done by Kecebas (2011). Effect of reference temperature on thermoeconomic evaluation of geothermal heating system was studied by Kecebas (2013).

In this context the current work proposes the usage of phase change material for geothermal energy storage combined with a radiator heating system. Thermodynamic and thermoeconomic aspects of the proposed system are studied taking second law efficiency of the system into consideration.

2. THERMODYNAMIC MODELING

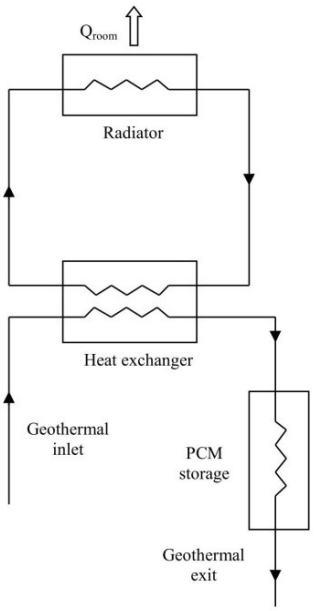

Figure 1: Schematic diagram of combined heating and storing system.

Figure 1 shows the schematic diagram of the combined space heating and thermal storage system. The geothermal water is first passed through a heat exchanger giving heat to the secondary fluid. The secondary fluid passes through the radiator heating system giving heat to the room and then returns back to the heat exchanger forming a closed loop. The geothermal fluid, after exiting the heat exchanger, passes through the phase change thermal storage system. The geothermal fluid gives heat to the phase change material and then exits out of the system. Thermodynamic modeling of the combined system is described below.

2.1 Heating System

The current analysis assumes usage of indirect method of space heating with radiators and plate type heat exchangers as mentioned in Karlsson and Ragnarsson (1995). The geothermal fluid gives heat to the secondary fluid which is passed through the radiators for room heating. The radiators manufactured are designed for fixed design conditions at assumed room temperature. For the temperature other than the design temperature, the heat load is calculated as:

$$\dot{Q}_T = \dot{Q}_{des}\left(\frac{T_{amb}-T_{room}}{T_{des}-T_{room}}\right) \tag{1}$$

The value of logarithmic mean temperature difference (LMTD) between the radiator and the room at design conditions is given as:

$$LMTD_R = \frac{T_{R,in}-T_{R,out}}{ln\left(\frac{T_{R,in}-T_{room}}{T_{R,out}-T_{room}}\right)} \tag{2}$$

Knowing the logarithmic mean temperature difference for the design condition, the logarithmic temperature difference at different conditions is calculated as:

$$\frac{\dot{Q}_T}{\dot{Q}_{des}} = \left(\frac{\mathrm{LMTD}_R}{\mathrm{LMTD}_{R,des}}\right)^n \tag{3}$$

The value of the exponent *n* is 1.3 (Anon, 1977).

On obtaining the logarithmic mean temperature difference, the value of radiator outlet temperature can be calculated using equation 2.

Obtaining the temperature at radiator outlet, the mass flow rate of fluid required through the radiator is found using the equation given below:

$$\dot{m} = \frac{\dot{Q}_T}{C(T_{in}-T_{out})} \tag{4}$$

Calculation of the above parameters for the radiators makes us able to fully describe the radiator or secondary fluid side of the system. The output parameters from the radiators as well as the geothermal fluid inlet temperature are input parameters for the heat exchanger. In general, plate type heat exchangers are used for residential heating due to its advantages of compactness, high heat transfer rate and ease of construction and maintenance over shell and tube type heat exchangers. For calculation of heat transfer coefficients through plate heat exchangers, the current analysis uses the relation suggested by Incropera et al. (2007) for flow through circular pipes with diameter equal to the hydraulic diameter of a non-circular channel of the heat exchanger through which the geothermal and the secondary fluids pass. The relation is given as:

$$Nu = 0.0296Re^{0.8}Pr^{0.333} \tag{5}$$

Calculating the convective heat transfer coefficient for both radiator side and geothermal side fluid, overall heat transfer coefficient can be determined.

The amount of heat transferred through the heat exchanger is then given as:

$$\dot{Q_T} = A_{HX} \cdot U_{HX} \cdot \mathrm{LMTD}_{\mathrm{HX}} \tag{6}$$

Knowing the value of heat exchange required and the area of the heat exchanger selected for the specific design, the logarithmic mean temperature difference of the heat exchanger can be calculated from the above equation. The geothermal exit temperature from the heat exchanger can then be calculated from the following equation:

$$LMTD_{HX} = \frac{(T_{geo,in}-T_{R,in})-(T_{geo,out}-T_{R,out})}{ln\left(\frac{T_{geo,in}-T_{R,in}}{T_{geo,out}-T_{R,out}}\right)} \tag{7}$$

Knowing the return water temperature from the above equation, the mass flow rate of geothermal fluid can be found using equation 4.

2.2 Phase Change Material

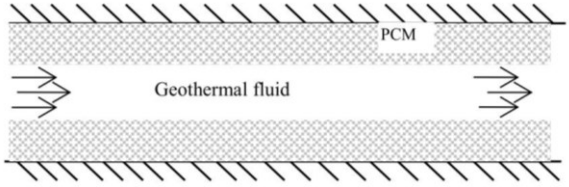

Figure 2: Configuration of storage system.

The current work focuses on analysis of PCM combined with a geothermal space heating system taking fixed values of design parameters such as storage component length, diameters and mass of phase change material. The aim of the study is to calculate the amount of energy and exergy saving using PCM combined with geothermal space heating. The mass of phase change material was assumed to be 400 kg. The mass of the storage material and other physical properties were kept constant during the analysis. The heat storage system has two concentric cylinders with inner and outer diameters of 0.02 and 0.045 m respectively. The heat source fluid flows through the inner cylinder and the storage material is filled in the annulus between the cylinders as shown in figure 2.

The small outer diameter of the cylinders makes such arrangement to be kept along the floor or walls of a room in a loop similar to radiator floor heating arrangement. The parameters such as diameters, length and phase change material weight can have optimized values as per the room dimensions and heat load. It is to be made clear that the analysis does not claim the current cylindrical arrangement of storage system as the optimum arrangement. The PCM material used is calcium chloride. The properties of the PCM material and the heating fluid (water) are taken from El-Dessouky and Al-Juwayhel (1997). Analysis for the PCM using air and water as heating fluid was discussed in the corresponding literature considering water and air as ideal fluid. For calculation of exergy terms this assumption does not validate fully for water. Exergy analysis for PCM was also done by Bjurstrom and Carlsson (1985). Relations were derived for the amount of exergy stored in the PCM and final temperature of the storage material. The current work uses the equations derived by Bjurstrom and Carlsson (1985) for analysis of the storage system. The main assumptions used in the analysis are:

- There is no heat exchange between the storage system and the surrounding area during charging process.
- The heat exchanger wall is assumed to have no resistance to heat transfer.
- Thermo-physical properties of the PCM and flowing fluid are assumed to be constant.
- Since the analysis is done using phase change material with a low transition temperature, the temperature difference between the transition temperature and room temperature is small, hence all stored exergy is assumed to be used for room heating.
- The heat transfer coefficient on the fluid (geothermal water) side (h_f) is calculated using equation 5 assuming flow through the cylinder to be turbulent.

The heat transfer coefficient between the wall and storage material is calculated according to the relation developed by Yanadori and Masuda (1989) based on experimental data.

$$h_m = \frac{2\lambda_s}{0.4\left(D_{inner} ln\left(\frac{D_{outer}}{D_{inner}}\right)\right)} \tag{8}$$

Calculating the convective heat transfer coefficient for the geothermal fluid and the phase change material side, the overall heat transfer coefficient for the PCM storage is calculated as:

$$\frac{1}{U_s} = \frac{1}{h_f} + \frac{1}{h_m} \tag{9}$$

In the above equation resistance due to the wall is neglected as it is small in comparison to the convective heat transfer coefficients.

The Number of Transfer Units on the fluid side is given by:

$$NTU = \frac{-U_s \pi D_{inner} L_s}{\dot{m}_{geo} c_{geo,f}} \tag{10}$$

For the analysis the initial temperature of the storage material and inlet temperature of the heat source is assumed. The procedure used for calculation of the final storage temperature and the amount of exergy stored is found from equations given by Bjurstrom and Carlsson (1985). The procedure used is:

A dimensionless time Ω is defined as:

$$\Omega = \frac{\dot{m}_{geo} c_{geo,f}}{M_{pcm} C_{pcm}} t \tag{11}$$

For the analysis temperatures are made dimensionless using the following equation:

$$\theta = \frac{T - T_o}{T_o} \tag{12}$$

Where T_o= reference ambient temperature (K).

The temperature efficiency (r) of the heat exchanger is given as:

$$r = 1 - \exp(-NTU) \tag{13}$$

Since the phase change material exhibit three stage absorbing: sensible heat in solid phase from initial temperature to transition stage, latent heat during transition phase and then sensible heat in liquid phase, an equivalent heat capacity is defined by the relation:

$$M_s C_{pcm}(T_l - T_b) = M_s C_s(T_m - T_b) + M_s H + M_s C_l(T_l - T_m) \tag{14}$$

or

$$\theta_l - \theta_b = \alpha(\theta_m - \theta_b) + \omega(\theta_l - \theta_b) + \sigma(\theta_l - \theta_m) \quad (15)$$

The constants α, ω and σ are given as:

$$\alpha = \frac{C_s}{C_{pcm}} \quad (16)$$

$$\omega = \frac{H}{C_{pcm}(\theta_l - \theta_b)} \quad (17)$$

$$\sigma = \frac{C_l}{C_{pcm}} \quad (18)$$

For heating at temperature below phase transition temperature, the instantaneous storage temperature as a function of time is given as:

$$\theta_s = \theta_b + (\theta_i - \theta_b)\left(1 - exp^{-r\Omega'}\right) \quad (19)$$

where Ω' is given as:

$$\Omega' = \frac{\Omega}{\alpha} \quad (20)$$

For heating the PCM above the transition temperature, the equation for instantaneous storage temperature and geothermal fluid outlet temperature is given as:

$$\theta_s = \theta_m + (\theta_i - \theta_m)\left(1 - exp^{-r\Omega''}\right) \quad (21)$$

$$\theta_e = \theta_i(1 - r) + \theta_m r \quad (22)$$

where Ω'' is given as:

$$\Omega'' = \frac{1}{\sigma}\left(\theta - \left(\frac{\alpha}{r} ln\left(\frac{\theta_i - \theta_b}{\theta_i - \theta_m}\right)\right) - \frac{\omega}{r}\left(\frac{\theta_l - \theta_b}{\theta_i - \theta_m}\right)\right) \quad (23)$$

where second and third terms in the brackets represent the dimensionless terms required for bringing PCM to phase transition temperature and its completion.

For PCM where the storage temperature exceeds the transition temperature, the amount of energy stored in the PCM is given by:

$$Q_{stored} = M_{pcm}C_s(\theta_m - \theta_b) + M_{pcm}H + M_{pcm}C_l(\theta_s - \theta_m) \quad (24)$$

3. SYSTEM ANALYSIS

3.1 Exergy Analysis

3.1.1 Heating System

On determining the heat load, the amount of exergy required for the heating is determined. In order to determine the exergy required, the quality factor of the room air is to be estimated. Since heat energy is a form of low grade energy, the amount of exergy present in the heat energy is determined by the quality factor which is to be estimated by means of Carnot efficiency given as

$$Y_{q,room} = 1 - \frac{T_{amb}}{T_{room}} \quad (25)$$

The amount of exergy required for satisfying the heat load demand is given as

$$\dot{\varepsilon}_{room} = Y_{q,room} \cdot \dot{Q}_T \quad (26)$$

The amount of exergy given by the geothermal fluid is calculated as:

$$\dot{\varepsilon}_{geo} = \dot{m}_{geo}\left[\left(h_{geo,in} - h_o\right) - T_{amb}\left(s_{geo,in} - s_o\right)\right] \quad (27)$$

The value of enthalpy and entropy of the secondary radiator fluid as a function of temperature and pressure are calculated using the relations given by Cooper and Dooley (2007).

3.1.2 Storage System

The amount of exergy stored in the PCM is given by Bjurstrom and Carlsson (1985)

$$\varepsilon_{stored} = M_{pcm}C_{pcm}T_{amb}\alpha\left[(\theta_m - \theta_b) - ln\left(\frac{1+\theta_m}{1+\theta_m}\right)\right] + \omega(\theta_l - \theta_b)\frac{\theta_m}{1+\theta_m} + \sigma\left[(\theta_s - \theta_m) - ln\left(\frac{1+\theta_s}{1+\theta_m}\right)\right] \quad (28)$$

3.2 Exergoeconomic Analysis

The total exergy given to a system as fuel exergy is transformed into product exergy and some part goes as exergy destruction and exergy loss. The increase in efficiency causes increase in exergy output for given input but also causes increase in cost required to

improve the system. For exergoeconomic analysis, cost is assigned for every exergy flow in a system. The cost is proportional to the amount of exergy the flow contains. The product cost is defined according to the fuel cost, capital expenditure and other operation and maintenance costs required for production or services. The general equation is given as:

$$\dot{C}_P = \dot{C}_F + \dot{Z}^{Cl} + \dot{Z}^{OM} \tag{29}$$

The equation signifies that the total cost associated with the product is the sum of the fuel cost, capital investment and the other costs related to operation and maintenance of the system producing the product.

A general cost balance equation in terms of cost per unit exergy for an ith component having heat and work interactions with the surroundings can be represented in terms of costs per unit exergy as:

$$\sum c_{out,i}\dot{\mathcal{E}}_{out,i} + c_{w,i}\dot{W}_i = \sum c_{in,i}\dot{\mathcal{E}}_{in,i} + c_{Q,i}\dot{\mathcal{E}}_{Q,i} + \dot{Z}_i^{Cl} + \dot{Z}_i^{OM} \tag{30}$$

The inlet cost in the above equation is obtained from that of exit cost of the previous component. For the first component, the inlet cost is the cost at which the fuel is supplied. Hence the above equation can be solved for the unknowns.

3.2.1 Purchase Equipment Cost

For real life applications, equipment cost can be obtained from vendor's catalogue. Generally, it is not possible to obtain detailed cost of every component for every design condition. For such cases the literature provides some useful sources. Mathematical charts and relations build from past experiences are available giving cost values in terms of different parameters such as design and geometry. The simplest way of estimating product cost is using exponential law which defines the product cost as an exponential function of the size of the component. The relation is given as follows:

$$I = I_r \left(\frac{S}{S_r}\right)^e \tag{31}$$

Such relation is assumed to be valid for a given range of equipment size. A general sixth tenth rule is used for any equipment by taking the value of the exponent e as 0.6. Different values of reference costs and their size along with the type of component were given by Boehm (1987). Because of various economic factors, the cost always changes with time. The above relation is for finding the cost from the reference cost of the indexed year. The obtained cost is brought to the current year cost by using conversion relation given as:

$$Reference\ year\ cost = original cost \cdot \frac{reference\ year\ cost\ index}{cost\ index\ for\ year\ for\ which\ calcualtion\ was\ made} \tag{32}$$

The cost index used in the above equation takes into consideration the inflation in the cost of material, equipment and labour. Various cost indices are available in the literature. The current analysis uses Marshall and Swift Cost Index (Marchall et al. 2009) for indexing the equipment cost. The exergoeconomic analysis requires levelized costs of the equipment. For converting investment cost into levelized costs, Capital Recovery Factor (CRF) calculated for the interest rate of *i* % with *N* years of life and *t* hours of annual operation is used given by the following equation:

$$CRF = \left(\frac{i(1+i)^N}{(1+i)^N - 1}\right)\left(\frac{1}{t \cdot 3600}\right) s^{-1} \tag{33}$$

The other factor to be taken into consideration while calculating component cost is operation and maintenance cost. The analysis assumes 2% of the investment cost of each component as the operation and maintenance cost (β). The cost flow rate for a given ith component is then calculated as:

$$\dot{Z}_i = \frac{I_i(CRF + \beta)}{t} \tag{34}$$

3.2.2 Exergy Costing

Guidelines for obtaining equations for different streams can be found in the literature (Bejan et al., 1996). The present case mainly involves usage of heat exchanging components. The balancing equations for the components used are described below.

(1) Heat exchanger

In this system a heat exchanger is used to deliver heat coming from geothermal inlet stream to radiator exit stream, which is the product stream. Hence the following equations are obtained:

$$\dot{C}_{geo,in} - \dot{C}_{HX,out} + \dot{C}_{R,out} - \dot{C}_{R,in} = -\dot{Z}_{\mathrm{HX}} \tag{35}$$

Also, since the fuel side is that of geothermal stream, hence its cost per unit exergy remains constant. The equation is given as

$$\frac{\dot{C}_{geo,in}}{\dot{\mathcal{E}}_{geo,in}} = \frac{\dot{C}_{HX,out}}{\dot{\mathcal{E}}_{HX,out}} \tag{36}$$

(2) Radiator

The cost associated with the radiator is charged to the product which is the exergy flowing out from the radiator for room heating. The equation obtained is

$$\dot{C}_{R,in} - \dot{C}_{R,out} - \dot{C}_{heating} = -\dot{Z}_R \quad (37)$$

Also, no exergy is added to the inlet stream, hence the cost per unit exergy remains same, and hence the equation obtained is

$$\frac{\dot{C}_{R,in}}{\dot{\varepsilon}_{R,in}} = \frac{\dot{C}_{R,out}}{\dot{\varepsilon}_{R,out}} \quad (38)$$

(3) Storage system

The cost associated with the storage is charged to the product which is the exergy stored in the system. The equation obtained is

$$\dot{C}_{HX,in} - \dot{C}_{geo,out} - \dot{C}_{storage} = -\dot{Z}_{storage} \quad (39)$$

Also, no exergy is added to the inlet stream, hence the cost per unit exergy remains same, and hence the equation obtained is

$$\frac{\dot{C}_{HX,out}}{\dot{\varepsilon}_{HX,out}} = \frac{\dot{C}_{geo,out}}{\dot{\varepsilon}_{geo,out}} \quad (40)$$

The above linear equations can be solved simultaneously to obtain the unknown variables.

4. RESULTS AND DISCUSSIONS

In order to have an optimized heating system, low temperature of geothermal fluid at the exit of the heat exchanger is required signifying high heat exchange in the radiator. With the aim of improving thermodynamic efficiency, the heating system design was simulated by parallel addition of heat storage using phase change material for different ambient temperatures and for different time durations of the heat storing process in the phase change material. The design load for the radiator heating arrangement assumed was 35 kW at a temperature of -15°C. The room temperature is assumed to be 20°C. The inlet temperature of the geothermal water was assumed to be 80°C.

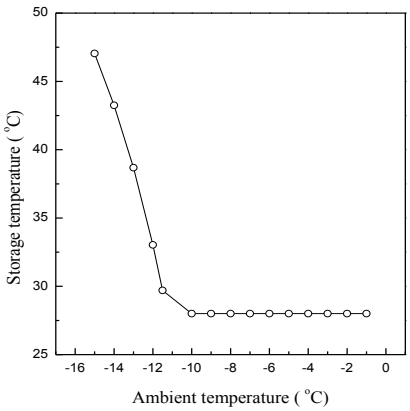

Figure 3: Variation of storage temperature with ambient temperature after an hour of operation.

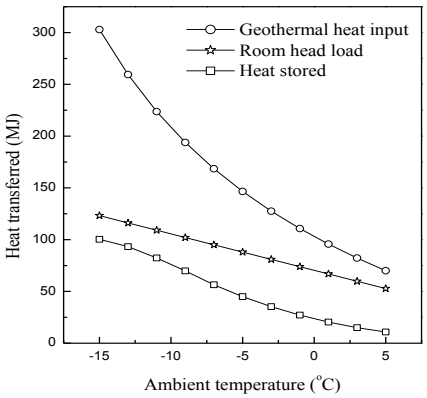

Figure 4: Variation of heat transferred with ambient temperature after an hour of operation.

Figure 3 shows the variation of storage temperature with ambient temperature after an hour of heat transfer process. It is found that at lower ambient temperature heat gained by phase change material is high enough making phase change material above transition temperature early. This occurs due to high temperature and mass flow rate of geothermal fluid at storage inlet. With increase in ambient temperature, storage inlet temperature and mass flow rate of the geothermal fluid decreases causing less heat transfer rate

between the geothermal fluid and the phase change material. This causes phase change material to be still in the transition stage after the same interval of time as that for lower temperature.

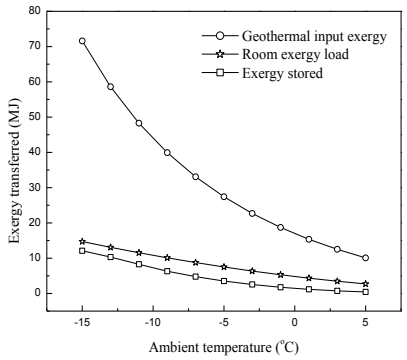

Figure 5: Variation of exergy changes with ambient temperature.

Figure 4 shows the variation of heat changes for different ambient temperatures after an hour of heat storing process keeping all other parameters constant. With the increase in ambient temperature, heat input from the geothermal fluid decreases. This occurs due to low geothermal fluid exit temperature at heat exchanger exit at high ambient temperature. The mass flow rate of the geothermal fluid required also decreases with the increase in ambient temperature which also causes decrease in heat input from the geothermal fluid. Room heat load shows continuous decrease as the ambient temperature increases. Amount of heat gained by phase change storage is higher at lower temperature and decreases with increase in ambient temperature. At lower ambient temperature, the phase change material is above transition stage. The amount of heat stored is higher as high mass flow rate of geothermal fluid at lower temperature also adds to high heat storage. The amount of heat stored at lower ambient temperature is the summation of latent heat of material and the sensible heat stored up to the storage temperature of material at that time period. The exergy changes also show similar trends as shown in Figure 5. Figure 4 and 5 show significant contribution made by phase change storage in energy saving at low ambient temperatures. The graphs shows that the energy and exergy stored are small in comparison to the total input but are significant in comparison to the room heat load.

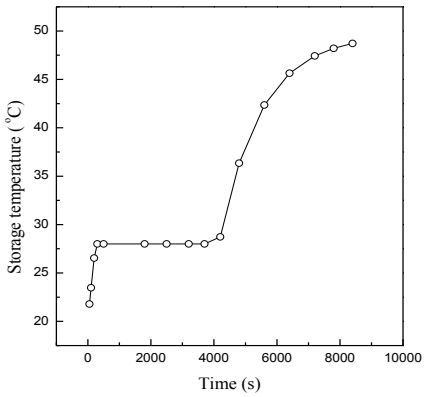

Figure 6: Variation of storage temperature with time at - 10 °C ambient temperature.

Figure 6 shows the variation of the phase change material storage temperature with time during charging assuming outside ambient temperature of -10°C. The initial temperature of the phase change material is assumed to be equal to the room temperature. With increase in storage time, storage temperature increases until the transition temperature occurs. Since the temperature difference between the heat source fluid and the storage temperature is high, heat transfer is high causing rapid increase in temperature. On reaching the transition temperature, heat transfer occurs at constant temperature until the latent heat of the material is absorbed. After the transition phase is over, storage temperature again starts increasing. The initial phase of the storage temperature shows high rate of increasing temperature and then starts decreasing as the storage temperature becomes closer to the heat transfer fluid inlet temperature.

Figure 7 shows the variation of total heat given by the geothermal heat source fluid, room heat load and stored heat with time. With an increase in time duration the room heat increases constantly as the ambient temperature is fixed and hence we have a fixed heat load. Total geothermal heat input from the geothermal heat source also increases linearly since for constant room heat load mass, flow rate of geothermal fluid also remains constant and the inlet hot fluid temperature is also fixed. The constant increase is reflected by the constant slopes of the geothermal heat input and room heat load in Figure 7. On the other hand the total heat stored increases constantly until the transition stage is complete and after that it decreases until the total heat stored becomes constant. The total heat stored shows a constant slope during the latent heat absorption. After that sensible heat storage starts taking place and the rate of heat storage decreases as the storage temperature starts approaching the heat source fluid temperature. It can be seen from

the graph that the room heat load requirement is much smaller than the total heat input from the geothermal fluid. Addition of phase change storage system makes significant heat saving comparable to the room heat load requirement. The significant amount of heat stored can be enough for satisfying the heat load requirement during the day time when the heat load is less.

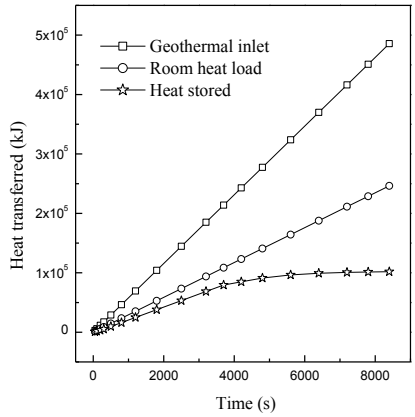

Figure 7: Variation of total heat transferred with time at - 10 °C ambient temperature.

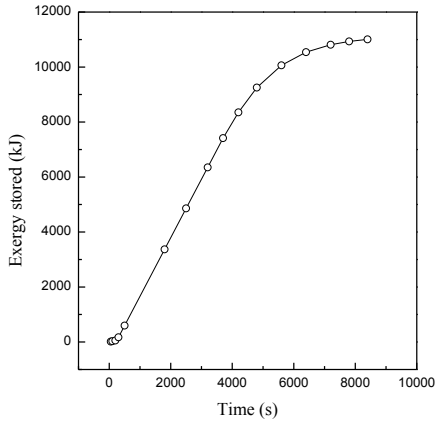

Figure 8: Variation of total exergy stored with time at - 10 °C ambient temperature.

Figure 8 shows the variation of total exergy stored in phase change material with time. Initial phase of the storage process shows constant increase in exergy accumulation in the phase change storage process. This exergy accumulation process increases constantly until latent heat storage takes place. After the phase transition is over, sensible heating starts. The exergy accumulation in the phase change storage then starts decreasing as the phase change storage temperature approaches geothermal fluid inlet temperature.

Figure 9 shows the variation of fractional exergy stored as a function of heat stored. The initial and end part of the curve represents heat storage due to sensible heating and the middle constant range represents latent heat storage. It is seen from the graph that major exergy accumulation takes place only in the latent heat change process and sensible heating of phase change material does not contributes much to exergy storage. Hence it can be concluded from the graph that latent heat storage makes a major contribution to heat storage than sensible heat.

Figure 10 shows the variation of cost flow per unit exergy at different ambient temperature. The inlet cost flow rate of the geothermal fluid is associated with that of well and pumping cost. Since the well cost varies from place to place, the current analysis assumes well cost of 0.5 million USD and 0.1USD/kWh for the pumping cost with a total required head of 200 m from the pump and 7,000 hours of operation annually. The discharge from the well is assumed to be 12 kg/s. The exergy flow rate from a geothermal fluid depends upon the ambient temperature. With increase in ambient temperature, exergy per unit mass flow rate of geothermal fluid decreases as the reference temperature increases. Also, the increase in ambient temperature causes decrease in mass flow rate through the well as heat load decreases. These two factors cause cost flow rate of the geothermal fluid to increase with ambient temperature. Cost flow rate of exergy supplied by the radiator to the room increases with ambient temperature. As the ambient temperature increases, heat load required decreases causing exergy flow rate required to decrease. The decrease in exergy flow rate does not affect the purchase cost associated with the heat exchanger and radiator since the size of the equipment depends upon the design conditions for the minimum ambient temperature. A similar variation is found for the cost flow rate of exergy stored in the phase change storage system. The increase is found to be greater at higher ambient temperature. This occurs because as the ambient temperature increases, heat exchanger exit temperature and mass flow rate decreases causing less exergy flow rate at the heat exchanger exit. The decrease in exergy flow rate causes increase in cost flow per unit exergy as the purchasing cost

remains the same as that for minimum ambient design conditions. An important observation as noted from the graph is the difference in cost flow rates of the radiator heating and the stored exergy. It can be seen from the graph that the cost flow rate of stored exergy is lower than the radiator heating at lower ambient temperature. Since with the increase in ambient temperature cost flow rate of radiator heating increases, the amount of exergy stored at lower temperature having low cost flow rate can be used at higher temperatures resulting in cost saving.

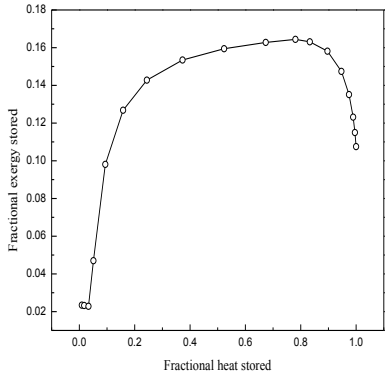

Figure 9: Variation of fractional exergy stored with fractional heat stored.

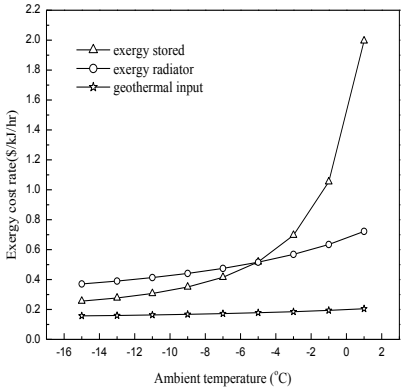

Figure 10: Variation of exergy cost flow rate per unit exergy with ambient temperature.

Table 1 shows the exergy and cost flow rate per unit exergy at different state points in the combined heating and storing system at -4°C ambient temperature. The values of cost flow rates per unit exergy shows that each system including geothermal well and pumping, the heating system and the heat storage system makes a significant contribution to overall product cost flow rate that is cost of room heating and storing exergy. Also, the low value of exergy storage cost in comparison to room heating using radiator system at higher ambient temperature shows advantage of storing exergy at low ambient temperature.

Table 1: Exergy and cost flow rates at -4°C ambient temperature.

State points	Exergy flow rate (kW)	Cost per unit exergy($/kJ hour)
Geothermal inlet	19.88	0.181
Storage inlet	11.56	0.181
Storage exit	7.24	0.181
Radiator inlet	16.24	0.218
Radiator exit	8.72	0.218
Room heating	4.08	0.370
Storage	3.28	0.540

5. CONCLUSION

Thermodynamic and exergoeconomic analysis of space heating using geothermal energy was performed. The cost involved with drilling and pumping also contributes to the total cost for space heating using geothermal energy. The cost becomes significant when the flow rate requirement is high due to high heat load requirement. The variation of the heating system efficiency with ambient temperature was studied. The heating system efficiency was found low at lower ambient temperature. Hence thermal losses become significant at lower temperature.

A parallel combination of phase change storage system with heating system was studied. Heat supplied to the storage system was provided from the exit of the heat exchanger. Thermodynamic studies of the heat storage process at different time and ambient temperatures were done. Higher heat storage was observed at low ambient temperature. The rate of exergy storage during the latent heat transition period was found to be larger than sensible heat storage. A sufficient amount of exergy was stored during charging period at low temperatures which could be used to fulfill the day time heat load requirement.

An exergoeconomic analysis of the combined heating and storage system was done. At low ambient temperature, cost flow rate of the exergy storage was found to be lower than the radiator heating cost flow rate. For the assumed value of investment cost flow rates, the heat storage using phase change material was found to be more economical than radiator heating till -5°C. The cost flow rate of radiator heating exergy increases with increase in ambient temperature. The usage of exergy storage at low ambient temperature for heating at higher ambient temperature makes significant contribution in cost savings.

NOMENCLATURE

A	=	area (m^2)
c	=	cost flow per unit exergy ($/kW)
C	=	specific heat capacity (kJ/kgK)
$\dot{C}$	=	cost flow rate ($/s)
D	=	diameter (m)
h	=	specific enthalpy (kJ/kg), convective heat transfer coefficient ($W/m^2 K$)
H	=	latent heat capacity of phase change material (kJ/kg)
I	=	component cost ($)
L	=	length (m)
$LMTD$	=	logarithmic mean temperature difference
$\dot{m}$	=	mass flow rate (kg/s)
M	=	mass (kg)
Nu	=	Nusselt number
NTU	=	Number of transfer units
P	=	power (kW)
Pr	=	Prandtl number
$\dot{Q}$	=	heat flow rate (kW)
Re	=	Reynolds number
s	=	specific entropy (kJ/K)
S	=	equipment size
t	=	time (s)
T	=	temperature (K)
U	=	overall heat transfer coefficient ($W/m^2 K$)
$\dot{W}$	=	rate of work (kW)
x	=	thickness (m)
$\dot{Z}$	=	levelized cost

Greek Letters

$\dot{\varepsilon}$	=	rate of exergy (kW)
λ	=	thermal conductivity (W/mK)
θ	=	dimensionless temperature
Ω	=	dimensionless time

Subscripts and Superscripts

amb	=	ambient
b	=	initial state of phase change material
Cl	=	capital
D	=	destruction
des	=	design
e	=	exit
f	=	fluid
F	=	fuel
geo	=	geothermal
HX	=	heat exchanger
i	=	i^{th} component
in	=	inlet
$inner$	=	inner
l	=	liquid phase
L	=	loss
m	=	transition phase
o	=	reference state
out	=	outlet
$outer$	=	outer
OM	=	operation and maintenance
P	=	product

q	=	heat
pcm	=	phase change material
r	=	reference
room	=	room
R	=	radiator
s	=	solid phase
storage	=	phase change material storage system
T	=	temperature

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