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Fuel Consumption Minimization in Greenhouses by Employing an Innovative System Taking Advantage of Solar and Geothermal Energies

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Replacing fossil flues by renewable and sustainable energies have been the concern of scientists working in this area over the recent decades. Furthermore, greenhouses play a key role in producing various crops year-round and in bohemian climates. For this objective, the greenhouses employ heating, ventilating and air conditioning (HVAC) systems to provide the favourable conditions for the considered crop. At the moment, generally, the heating demand of greenhouses is provided by air heaters which burn gasoil as their main fuel. In this work, an innovative combined configuration of solar-geothermal heat has been proposed in order to reduce the amount of fuel consumption in the greenhouses. A comprehensive thermo economic analysis has been accomplished to demonstrate the proposed system effectiveness. The economic survey approach at this step was Net Present Value (NPV). The NPV results prove the super satisfactory performance of the proposed system with a payback period of 6.5 years.

Keywords: Greenhouse, Geothermal Energy, Solar Energy, Multi-Node Storage Tank, NPV

1 Introduction

Greenhouse employs sun rays to foster vegetables, fruits and flowers more efficient than traditional cultivation approaches, year-round and even in bohemian climates. The intensity of energy usage in greenhouses has been studied by many authors generally [1, 2] and even for specific crops (tomato [3, 4], grape [5], and strawberry [6]). All the aforementioned studies demonstrated the excessive amount of energy consumption in the greenhouses.

The researches show that in spite of the significant amount of solar energy absorbed into the greenhouses by passive solar system, huge amount of fuel is still needed for reaching the desirable temperature of crop in many days over the year. Currently, generally, the heating demand of greenhouses is provided by air heaters burning gas oil as main fuel in Iran (and probably other countries). Ambient air is taken in by a Forced draft (F. D.)

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Fan and an air heater heats it. This air is then distributed in the greenhouse by secondary HVAC systems such as ducts and wicket gates [1].

Many studies have been done to find opportunities and methods to decrease the fuel consumption of greenhouses. These studies led to many proposals aimed at improving the thermal behaviour of greenhouses such as more efficient cover materials, better framework shapes and etc. [7, 8]. Many other researches also focused on employing alternative energies in the greenhouses [9-15]. Among all the renewable energy sources, geothermal energy and solar energy have been considered most.

The geothermal energy is one of the most important and promising sources of renewable energy stored in the Earth. The geothermal energy is cost effective, reliable, sustainable, and environmentally friendly [16]. Geothermal resources may be classified by type of rock formation/form of water and temperature, ranging from less than 20 °C to over 300 °C [17].

Resources above 150 °C are normally used for electric power generation and those below 150 °C are typically used in direct-use projects for heating and cooling [18]. In of the first studies in this field, Rafferty [19] studied the feasibility of employing geothermal energy in the greenhouses, covering peak heat loss calculations for the most common construction methods. Bakos et al. [20] presented the second work in this area by doing an analysis of an extended heating system which uses low temperature water or direct geothermal fluid to heat a greenhouse. Karytsas et al. [21] evaluated low enthalpy heating of greenhouses employing geothermal heat in various points of the world. Ghosal and Tiwari [22] also presented a mathematical modeling for greenhouse heating by using geothermal energy.

On the other hand, the solar energy is another important source of renewable energy. Solar energy technologies have been being employed in many applications such as heating, cooling, photovoltaic and agricultural [23]. In this regard, Maghsood [24] presented the first comprehensive work to study solar energy parameters in plastic covered greenhouses. Van Bavel et al. [25], then, did a comprehensive energy analysis on fluid-roof greenhouses. Santamouris et al. [26] and Hasson [27] are two other important references presented about passive techniques of employing solar energy in greenhouses. The active techniques, in which solar thermal collectors are employed, have been proposed to be used in greenhouses in various methods. Willits et al. [28] were the pioneers of employing active solar systems in greenhouses by proposing a solar storage system in greenhouses. Zabeltits [29], later, proposed a thorough proposal of employing solar energy for heating greenhouses by various approaches such as separate solar collectors, solar collectors integrated in the greenhouse, and the use of the greenhouse itself as a solar collector. Several more works in this field also could be addressed [30-33]. Farzaneh-Gord et al. [34] also presented another approach of solar energy storage in greenhouses by evaluating the energy performance of a greenhouse in Iran as the case study.

This paper, however, presents another innovative system taking advantage of both solar and geothermal energies to improve the greenhouse energy performance. To the knowledge of the authors this approach has not been discussed in the literature before and this is the first time that combination of both solar and geothermal energies in greenhouses is proposed. The thermo economic performance of the proposed system has also been compared with the performance of the other already proposed systems employing either solar energy or geothermal energy individually.

2 The Case Study

The proposed twofold system was employed to evaluate the thermal behaviour of Dashte-Minoo greenhouse set as the case study of this work. Dashte-Minoo greenhouse is located between Azadshahr city and Gonbadekavoos city in the north-east of Iran. The geographical location of the considered greenhouse set is in latitude of 37° and longitude of 55°. This greenhouse set includes 10 similar greenhouses with the same dimensions. These greenhouses produce cucumber as the main and year-round crop. The most suitable temperature for growing cucumber has been addressed 20 ºC [35]. The employed cover material is twin Polycarbonate with 8 mm thickness playing a remarkable role to both isolate the greenhouse space from the environment and collecting the solar rays simultaneously. Although this area is among the worst areas of Iran in terms of solar radiation absorption, it is still a very appropriate area for hosting greenhouses with average solar radiation intensity around 4.5 kWh/m^2. day for a horizontal surface [36]. Figures (1-a) and (1-b) show respectively the location of the case study on the map including the sun shine potential of different areas of Iran and dimensions of each greenhouse in the case study.

3 The Case Study Heating Demand in the Conventional Configuration

As it was explained before, in general, the heating demand of greenhouses in the conventional heating method is provided by air heaters burning gasoil as main fuel in Iran (and probably other countries as well). Figure (2) shows the schematic diagram of heating preparation for the greenhouses by conventional heating method. As the figure shows, the system takes advantage of an F. D. fan that intakes the ambient air. This air then goes through an air heater burning gasoil to meet the desired temperature. Afterward, the warmed air is distributed in the greenhouse space by secondary HVAC systems such as diffusers and etc.

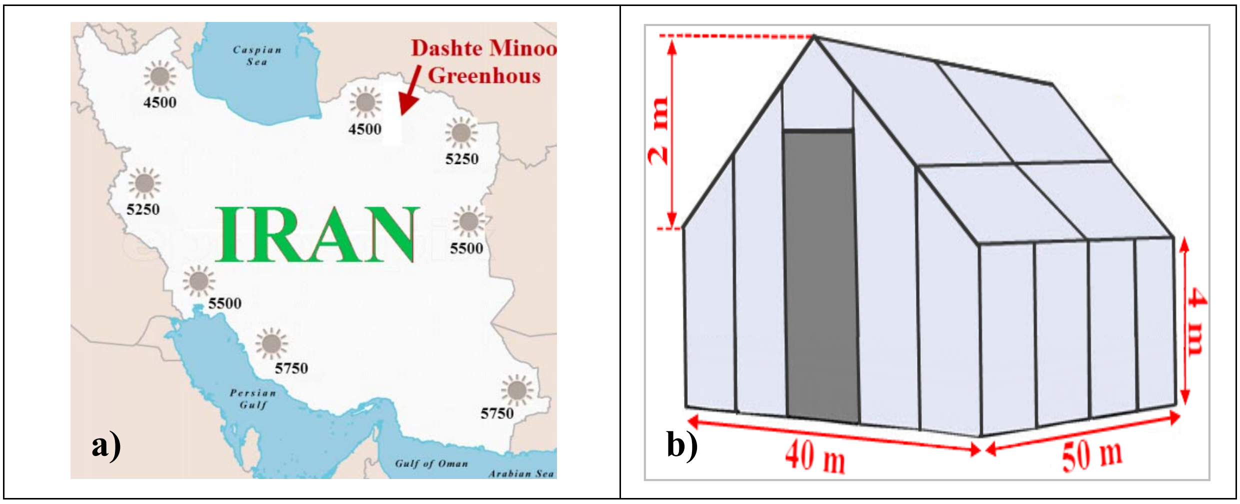

Figure 1 a) Iran's solar radiation absorption potential and b) the case study dimensions

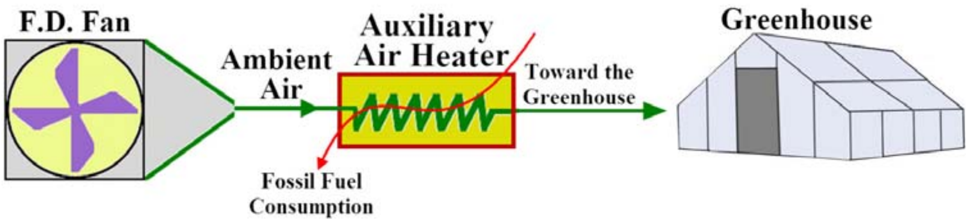

Figure 2 The schematic diagram of heating preparation in conventional method

The major problem is that the fuel consumption of air heater to provide the required heat is too much. Following, a detailed energy analysis formulation associated with the above layout is presented. The energy balance on the greenhouse structure could be given as below:

$$\dot{Q}_{GR} = \dot{Q}_{lost} + \dot{Q}_{vent} - \dot{Q}_{solar} \tag{1}$$

Where, $\dot{Q}_{GR}, \dot{Q}_{lost}, \dot{Q}_{vent}, \dot{Q}_{solar}$ are the required heat of greenhouse to stay at the favorable temperature, the overall heat lost from the greenhouse to the environment and soil, the heat lost from the greenhouse due to ventilations and the absorbed solar energy into the greenhouse due to greenhouse effect, respectively. All these parameters are calculated individually below.

The first parameter on the right side of equation 1 is the overall heat lost from the greenhouse through the walls and field. This item can be calculated by the following equation.

$$\dot{Q}_{lost} = \sum_{i=1}^{n} U_i A_i (T_{in} - T_a) \tag{2}$$

Where, U_i, A_i, T_a and T_{in} are respectively the overall heat transfer coefficient, the heat transfer rout area, the ambient temperature and the internal greenhouse temperature which is supposed to be equal to 20 ^oC over the whole year [36]. Table (1) details the values of U and A of all the heat lost routs for each greenhouse [36].

Knowing the ambient temperature as well as the above table information, one can easily calculate the overall heat loss of greenhouse to the environment.

The second parameter on the right side of equation 1 is the heat lost from the greenhouse due to the ventilation. The value of this item for each greenhouse could be obtained as:

$$\dot{Q}_{vent} = N \cdot V \cdot \rho \cdot c_p \cdot (T_{in} - T_a)/3600 \tag{3}$$

Where, c_p, ρ, V and N represent respectively the air constant pressure thermal capacity (1.005 kJ/kg.^{o}C), the air density at the considered temperature (0.995 kg/m^3), the greenhouse air volume and the greenhouse air switching frequency during an hour (2 times per hour for winter and 20 times per hour for summer, based on the information reported in [36]).

Table 1 U and A values for the heat lost routs of each greenhouse

Rout	A (m^2)	U (W/m^2. ^{o}C)
Ceiling	2010	0.625
Southern and Northern Walls	200	0.625
Eastern and Western Walls	200	0.625
Field	2000	0.5

The given value by equation 3 must be divided by 3600 in order to give the instantaneous energy lost value.
The last term on the right side of equation 1 is the absorbed and stored solar heat into the greenhouse due to the greenhouse effect. Equation 4 calculates the amount of absorbable solar energy by a surface with 1 m^2 area:

$$S = I_b R_b (\tau\alpha)_b + I_d (\tau\alpha)_d \left(\frac{1+Cos\beta}{2}\right) + I \cdot \rho_g (\tau\alpha)_g \left(\frac{1-Cos\beta}{2}\right) \quad (4)$$

Where, I, I_d, I_b and S are the total available solar radiation, the diffuse component of solar radiation, the beam component of solar radiation and the absorbed solar radiation by the slopped surface, respectively. The Greek symbols β and (τα) in this equation represent the surface slope angle and the average absorption-transmission coefficient of the corresponding surface and its cover. Actually, the average absorption-transmission coefficient for the greenhouse is a functional of the greenhouse cover material, the physical properties of internal elements of the greenhouse and the radiation angle. Regarding the greenhouse content, the absorption coefficient for the internal elements of the greenhouse is proposed to be considered 0.7 while they are exposed to normal radiations [36]. Detailed information about solar radiation calculation on an arbitrary surface is available in [37-39].

Equation 4 must be employed to calculate the amount of solar energy entered and absorbed in the greenhouse from all sides except the northern wall of greenhouse. It is because this wall doesn't contribute for solar radiation collection due to being completely deviated from the south. Table (2) presents the values of β for all the walls and the ceiling of each greenhouse.

Employing equation 4 and the above table, one can obtain the absorbable solar flux value from each rout. Finally, equation 5 should be applied to calculate the overall heat absorbed by the greenhouse internal elements due to the greenhouse effect.

$$\dot{Q}_{solar} = (S_{south} \times A_{south}) + (S_{east} \times A_{east}) + (S_{north} \times A_{north}) + (S_{west} \times A_{west}) + (S_{ceiling} \times A_{ceiling}) \quad (5)$$

Calculating all the three parameters on the right side of equation 1, one now could be able to calculate the total energy that each greenhouse requires to stay at the desired temperature. The amount of fuel that the air heater burns to provide this energy could be given by the following equation.

$$\dot{m}_f = \frac{\dot{Q}_{GR}}{LHV.\eta_h} \quad (6)$$

Where, η_h, LHV and $\dot{m}_f$ represent respectively the air heater thermal efficiency, the fuel lower heating value and the fuel mass flow rate. In this work, the air heater thermal efficiency and gasoil lower heating value are considered 45 MJ/kg and 50%, respectively [37].

Table 2 The slop angle values for different walls of the greenhouse

Rout	β
Southern	90
Ceiling	0
Eastern	90
Western	90

4 The Proposed Design

Regarding this fact that the target temperature in the greenhouse is relatively low (20 ºC), both of the renewable energy sources proposed for this system are designed to provide low temperature heats. Therefore, the employed solar collectors for the solar heater system are flat plate solar collectors and also the boreholes of the geothermal heating systems are supposed to be delved up to middle depths. It bears mentioning that the heating task is for when the ambient temperature is below 20 ºC. As low temperatures can be harmful for the crop, temperatures higher than the appropriate value could harm the cultivation efficiency as well. Therefore, ventilation (and even cooling if necessary) should be done in the greenhouse in the warm months of the year. Figure (3) illustrates the schematic diagram of proposed system aimed at taking advantage of solar and geothermal energies simultaneously.

As the figure shows, the system takes advantage of three heat exchangers i.e. the geothermal heat exchanger (G.H.E), the solar heat exchanger (S.H.E) and the conventional air heater (A.A.H). In fact, four various cases have been considered in the proposed system. The first case is when the ambient temperature is much less than the desired value. Thus, the required energy of greenhouse is in a high range. In this case, all the three heaters should be employed to rise the air temperature up to the favorable value.

For this objective, the intake air by the F. D. fan enters the G.H.E supplied by a few geothermal boreholes. The G.H.E water goes down through the boreholes and receives heat from the Earth. The water coming back from the boreholes delivers the received heat to the intake air passing the heat exchanger. After this step, the air enters the S.H.E and its temperature goes up to higher values. It should be noted that the S.H.E also is a shell and tube heat exchanger supplied by a solar storage tank. Actually, the solar storage tank stores the solar heat collected by the flat plate collector series in the form of hot water. So that, the collected solar heat during the day must be stored in the form of hot water and this hot water can be reclaimed for usage at either nights or cloudy hours. The double preheated air then enters the A.A.H that is an auxiliary system in the proposed system up to the favorable temperature. This air is distributed through the greenhouse space by some secondary HVAC systems. This case is probably the time that the proposed system is more efficient than other times as the both solar and geothermal systems are working at their full load.

The second possible case is when the intake air can meet the desired temperature after passing both the G.H.E and S.H.E. Therefore, no more heating is required. In this case, the A.A.H should be turned off and out of circuit. The dashed line parallel with the A.A.H in the figure is the rout that the air should pass in this case.

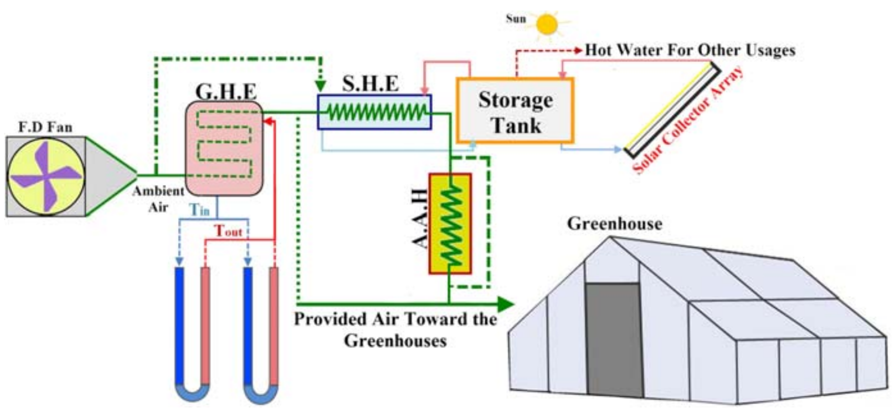

Figure 3 The schematic diagram of proposed system to utilize simultaneous solar and geothermal energies

The third case is when the ambient temperature is not much less than the desired temperature so that the heating demand is not too much. For such cases the geothermal system could not be effective because the achievable temperatures from middle depth geothermal boreholes are less than the target temperature. This issue will be explained thoroughly in the geothermal formulation section. This is actually the main reason that the solar thermal system is proposed to accompany the geothermal heating system in this work. Therefore, for such a case, the S.H.E should be employed to provide the required heat. The dash double dot line (– • •) in the figure shows the rout that the air has to pass for this state. After receiving the required heat, the air has to pass the bypass path parallel with the A.A.H. One should note that for the sake of simplification the system is designed to employ the stored solar energy during night hours (i.e. from the sunset to the next sunshine). This assumption is reasonable as the heating demand over nights is much more than heating demand during day hours and the stored solar heat can be employed without over-heating risk.

The last case refers to the times that not only heating demand is zero, but also the air temperature is so high that cooling and ventilating should be done in order for leveling the greenhouse air temperature. As the ventilating and cooling process is out of this work scopes, the geothermal system contribution is the only parameter in cooling and ventilating that will be reported in the result section. The dotted line after the G.H.E shows the rout that the air has to pass for this case (i.e. directly toward the greenhouse and the secondary HVAC systems). It is also noteworthy that the collected and stored solar heat at these times can be employed for other applications such as human usages in the greenhouse site where more than 20 people are working all day long. Following, more detailed technical information about the employed geothermal system and solar heater system is presented.

4.1 The Solar Heater System Details

For the solar thermal systems using the flat plate type collectors, one of the most important factors to determine is the collector slop angle and direction. Evidently, in the north hemisphere the collectors should be placed toward south and the angle between the collector plate and horizon is recommended to be chosen the latitude of the place plus a value in range of 10-15. The previous study of the authors show that for this Area of Iran, the most appropriate slop angle for the flat plate solar collectors is 45^o [15]. The amount of initial investment should be determined based on a specific thermo economic tradeoff approach detailed in [38]. As the system proposing in this work takes advantage of two different renewable sources and each of them has its own costs and benefits, inevitably, the proposal should be divided into two improvement suggestions. Thus, the solar heater unit is proposed first and the optimal cost of capital for this system is calculated. Then, the second suggestion (geothermal heating unit) is presented for the one-step improved configuration. Now, one can easily calculate the optimal cost of capital for the geothermal proposal.

As employing solar heating system in greenhouses has already been surveyed by the authors, the technical details of the work are supposed to be just the same of the previous work as the optimal values. Detailed description about the data and the method of obtaining this data is available in [15]. Table (3) presents the technical information related to the employed solar thermal system.

Table 3 Properties of employed solar collectors

Characteristic	Information
Collector Length	200 Cm
Collector Wide	95 Cm
Collector Thickness	9.5 Cm
Cover Matter	Glass
Cover Thickness	4 mm
Absorber Plate Thickness	0.5 mm
Tubes Inner Diameter	10 mm
Tubes outer Diameter	12 mm
Tubes space	150 mm
Plate Area	1.51 m^2
Plate Matter	Copper

4.2 The Geothermal System Details

The annual average ambient temperature for the target location has been reported equal to 18.9 ^oC [40]. This temperature is uniformly available for all days of the year after depth of 100 m in this city. After this depth, obviously, the more depths for the borehole, the more energy is obtainable from the Earth. Therefore, for the ambient temperatures less than this, the geothermal system could be employed for heating task and otherwise it could be usable for cooling and ventilating auxiliary system. For systems taking advantage of geothermal energy, there are numbers of important issues to be considered at the simulation. They are the number, the depth and the diameter of boreholes, the distance between the boreholes, the operating fluid mass flow rate flowing into each borehole and the optimal volume of G.H.E. About the depth and the distance of boreholes, it is recommended to select the value of BH factor from 0.05 to 0.2. Where, the factor BH is defined as the ratio of distances between the boreholes to the depth of each borehole [41]. The value of BH adopted for this work is 0.1.
As it was claimed before, after depth of 5-6 m the Earth's temperature tends to stay on a constant value and the value rises slightly by the rate of 30 $^oC/km$. Naturally, even considering a constant temperature for the soil, as the depth of borehole increases the achievable heat also grows because the heat transfer area goes up. However, the growth in the cost corresponding with each borehole increases by increasing the depth. The companies working in this area in Iran claim that the deep boreholes up to 150 m depth can be delved for the price of 25 USD/m including all the complementary necessary equipment such as the heat exchanger and piping costs [42]. As delving boreholes with depths more than 150 m requires more advanced facilities and is more difficult, the price goes up for such cases. As the low temperature heat is required and economic considerations are the pivotal factors in this project, the depth 150 m also was chosen as the ideal depth of boreholes in this work.
Considering the ideal depth for the boreholes as well as the adopted value for the BH factor, thus, the distance between the boreholes of the proposed system in Dashte-Minoo greenhouse set should be 15 m. About the most suitable diameter of the boreholes, it could be said that borehole diameter is an engineering question determined by heat transfer issues. In general, the smaller the diameter of the borehole, the greater the heat transfer efficiency. It is assumed that a smaller borehole diameter is also less likely to permit aquifer contamination by water movement through the borehole. However, on the other hand, bigger diameters for the geothermal boreholes lead to more heat transfer area and consequently more heat transfer rate in a unit of borehole length. Actually, the recommended range of geothermal borehole diameters by the experts is in range of 90 mm to 190 mm [43].

Table 4 The proposed geothermal system details for Dashte-Minoo greenhouse set

Operating Fluid	Water
Operating Fluid Mass Flow Rate	0.14 kg/s
Operating Fluid Velocity	0.3 m/s
Pipe Material	High Density Polyethylene
Pipe Thermal Conductivity Coefficient	0.42 W/K
Pipe Diameter	0.026 m
Pipe Thickness	0.006 m
Grout Thermal Conductivity Coefficient	1.25 W/K
Heat Exchanger Type	Shell and Tube
Earth's Thermal Conductivity Coefficient	1.5 W/K
Earth's Thermal Diffusion Coefficient	0.0778 m^2/day
Heat Exchanger Coil Length	112 m
Heat Exchanger Coil Diameter	0.1 m
Heat Exchanger Volume	12.5 m^3

The diameter of boreholes chosen for this work was 150 mm. The selection of the appropriate number for the boreholes depends on the amount of required energy as well as the amount of initial investment. Where, the G.H.E volume depends on the number of boreholes. Obviously, the heat exchanger with more capacity and volume requires more investments and based on the advice coming from the reference companies it is a linear functional of the boreholes lengths. Therefore, the number of boreholes (consequently the volume of heat exchanger) should be defined on a basis of a thermo economic analysis. The proposed layout for the boreholes in the case study location is a rectangular shape field with a regular 15 m gap between the boreholes and the recommended number of boreholes for this whole greenhouse set is 35 boreholes, resulted from a thermo-economic evaluation. Eventually, table (4) details the proposed systems information for Dashte-Minoo greenhouse set.

5 The Proposed System Energy Analysis

To analyse the thermal behaviour of the system, one should perform the balance of energy of the whole system. Knowing the total energy required of the greenhouses, one could easily determine each of the three heating systems contributions. For this objective, according to figure (3), four control volumes should be adopted and the first law of thermodynamics should be written for each one.

5.1 Geothermal Heating System

As figure (3) shows, the first control volume includes the G.H.E and the underground-vertical heat exchanger (boreholes). The energy balance equation for this control volume could be given by [37-39]:

$$m_w c_w \frac{dT_w}{dt} = \dot{Q}_{GHX} - \dot{Q}_{a-1} \tag{7}$$

Where, m_w, c_w and T_w are the mass, thermal capacity and temperature of working fluid of the system (water). Also, $\dot{Q}_{GHX}$ and $\dot{Q}_{a-1}$ refer to the absorbed heat from the earth by the operating fluid and the gained heat by the intake air in the first step of preheating (i.e. the G.H.E), respectively.

Various types of geothermal heat exchangers for taking advantage of the Earth's heat are available such as horizontal and vertical.

Regarding the fact that the vertical heat exchangers have more efficiency, occupy less room and require less energy for pumping the working fluid comparing to the horizontal ones, the employed underground heat exchangers in this study are vertical type. The rate of absorbed heat by the underground-vertical heat exchanger highly depends to the thermal and physical properties of soil, applied tube and material of the borehole structure. Over the simulation process, all the aforementioned properties are considered to be constant over time, but the temperature of entering water into the underground-vertical heat exchanger as the only variable item. The total heat absorbed by the underground-vertical heat exchanger set could be given by the following equation [16]:

$$\dot{Q}_{GHX} = q \cdot N_b \cdot H \tag{8}$$

Where, N_b and H are the number of boreholes and the effective length of each borehole, respectively. q also is the absorbable heat for the unit effective length of tube by one borehole individually and could be obtained by the following equation [44]:

$$q = \frac{\dot{m}_f \cdot c_f \cdot (T_{out} - T_{in})}{H} \tag{9}$$

In which, $\dot{m}_f$, c_f, T_{out} and T_{in} are the mass flow rate, thermal capacity, outlet temperature and inlet temperature of working fluid, respectively. The amount of T_{out} could be calculated as [19- 22]:

$$T_{out} = T_{gr} + \left[\sum_{\lambda=1}^{n_t} \frac{q_\lambda - q_{\lambda-1}}{4 \cdot \pi \cdot k_{gr}} \cdot \left(\int_{\frac{1}{\sqrt{4\alpha(t_\lambda - t_{\lambda-1})}}}^{\infty} I_e \cdot \frac{I_{ls}(H.S, D.S)}{H.S^2} dS \right) \right] + q \cdot R_b - \frac{q \cdot H}{2 \cdot \dot{m}_b \cdot C_b} \tag{10}$$

Where, λ again counts the hourly time steps as all the calculation processes are mainly done on a basis of hourly periods. Therefore, q_λ represents the absorbed q in time step λ and $q_{\lambda-1}$ does that for time step λ-1. T_{gr}, k_{gr}, α also refer to the ground temperature, thermal conductivity coefficient and diffusion coefficient of the borehole, respectively.

Defining *S* as the Laplace transform variable in the short-term response, h=H.S, d=D.S and D as the ineffective length of tubes in the borehole , the other factors in the above equation could be given by the following equations step by step.

$$I_{ls}(h,d) = 2\, ierf(h) + 2\, ierf(h+2d) - ierf(2h+2d) - ierf(2d) \tag{11}$$

Where, from mathematics we've already defined the error function erf(z) (which is a normalized form of the Gaussian function) as below:

$$erf(z) = \frac{2}{\sqrt{\pi}} \int_0^z e^{-z^2} dz \tag{12}$$

Subsequently, we define:

$$ierf(z) = z \cdot erf(z) - \frac{1}{\sqrt{\pi}}(1 - e^{-z^2}) \tag{13}$$

The function I_e also could be given as below:

$$I_e(s) = \frac{1}{N}\sum_{i=1}^{N}\sum_{j=1}^{N} e^{-r_{i,j}^2 s^2} \tag{14}$$

Here $r_{i,j}$ denotes the radial distance between borehole i and j ($i \neq j$). The contribution from the own heat source of the borehole i is obtained for the radial distance r_b. The last important item in this equation is R_b which refers to the overall thermal resistance of borehole and may be achieved by:

$$R_b = \frac{1}{2}(R_{conv} + R_{cond}) + R_{grt} \tag{15}$$

Where, the indices conv, cond and grt refer to the convectional heat resistance coefficient of the working fluid within the tubes, the conduction heat resistance coefficient of the tubes and the conduction heat resistance coefficient of the boreholes walls. These parameters can be obtained by the following equations, respectively:

$$R_{conv} = \frac{1}{\pi\, d_i\, h_f} \tag{16}$$

$$R_{cond} = \frac{ln\left(d_o/d_i\right)}{2\cdot\pi\cdot k_p} \tag{17}$$

$$R_{gr} = \frac{1}{k_{gr}\cdot\beta_0\cdot\left(d_b/d_o\right)^{\beta_1}} \tag{18}$$

In the above equations, d_i, d_o and d_b refer to the internal and external diameter of the tube and the diameter of borehole, respectively. h_f, k_p and k_{gr} also represent the convection and convective heat transfer coefficients for the fluid, the tube and the walls of the borehole s, respectively. β_0 and β_1 also are form factors which could be found from the corresponding tables reported in [45].

Employing the detailed formulation above, one could make a system of equation of two equations-two passives including equations 9 and 10. This system of equation would be calculable employing computational and numerical methods. The employed computational method in this work is a modified version (by authors) of the already available function in Matlab software named Guadgk.

5.2 The Solar Heat Exchanger

Based on what the schematic diagram shows, the second control volume which should be considered is the S.H.E. It contains three possible inlet flows and two exit flows. However, they all are not supposed to be in use simultaneously. Firstly, if the air enters the G.H.E to be preheated, then, the direct canal of air to the S.H.E should be close. Reversely, if there is no need to geothermal heat, then this path should be close and the direct path is open. Secondly, the paths of water incoming/outgoing from/to the solar storage tank are close during the day as this source is supposed to be used over nights only. Thirdly, the outlet air path also is the other stream outgoing from the control volume. Equation 19 presents the energy balance on the control volume associated with the S.H.E.

$$m_w c_w \frac{dT_w}{dt} = \dot{Q}_{st} - \dot{Q}_{a-2} \tag{19}$$

Where, $\dot{Q}_{st}$ and $\dot{Q}_{a-2}$ are respectively the heat supplied to the control volume by the solar storage tank and the gained heat by the intake air stream in the second step of preheating (i.e. the S.H.E).

5.3 The Solar Storage Tank

The obtainable energy rate from a flat plate solar collector could be given by:

$$\dot{Q}_u = A_c \cdot F_R \cdot [S - U_l (T_{fi} - T_a)] \tag{20}$$

Where, F_R, T_{fi}, T_a, A_c and U_l are the collector removal factor, inlet working fluid temperature, ambient temperature, absorption surface area and the total heat transfer coefficient for the collector, respectively. Also S is absorbed solar flux by flat plate collector per one square meter area and can be obtained from equation below [37-39]:

$$S = (\tau\alpha)_{av} . I_T \tag{21}$$

Where, $(\tau\alpha)_{ave}$ and I_T are respectively, average absorption-transmission coefficient of collector and radiated solar flux on slopped collector. The functional dependence of the collector efficiency on the meteorological and system operation values can be represented by the following equation [37]:

$$\eta_i = 0.78 - 1.4\frac{(T_{pm} - T_a)}{I_T} - 0.09\frac{(T_{pm} - T_a)^2}{I_T} \tag{22}$$

The collector efficiency has been calculated based on En-12975-2 Standard [46]. The most reliable and accurate method of energy analysis of the storage tank is considering a multi-node tank. As water with higher temperature is less dense, the warmer layers of water lie in upper levels of the storage tank. In a multi-node tank, the tank is divided into N different divisions with different temperatures and densities. The energy balance on the whole storage tank can be reached by writing the first law of thermodynamics law for each division. Therefore, an N differential equations set is formed and has to be solved simultaneously. Evidently, more node numbers lead to more accurate results, however, it leads to a more complicated system of equation. N=3 has been recommended as the best choice [23]. Consequently, a three-node tank has been chosen in this work. Figure (4) illustrates the schematic of a three-node storage tank.

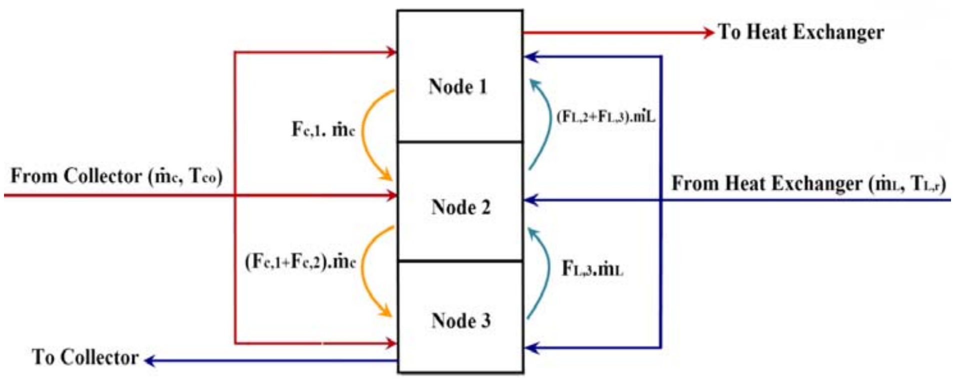

Figure 4 The schematic of a three- node storage tank

T_{co}, $\dot{m}_c$, T_L, T_o, and $\dot{m}_L$ are respectively the water temperature and mass flow rate incoming into the storage tank from collectors, the water temperature outgoing to the S.H.E from the storage tank, the water temperature outgoing from the storage tank toward the collectors and the mass flow rate of incoming back water from the S.H.E into the storage tank. Defining coefficient F_i^c, one can specify which node receives water from the collector. This coefficient could be defined as below:

$$F_i^c = \begin{cases} 1 & \text{if} \quad i = 1 \ \text{and} \ T_{co} > T_{s,i} \\ 1 & \text{if} \quad T_{s,i-1} \geq T_{co} > T_{s,i} \\ 0 & \text{if} \quad i = 0 \ \text{or} \ \text{if} \ i = N+1 \\ 0 & \text{otherwise} \end{cases} \tag{23}$$

Where, $T_{s,i}$ is the i-esime node temperature. Also, defining the coefficient F_i^L, one can determine which node receives the water coming back from the S.H.E:

$$F_i^L = \begin{cases} 1 & \text{if} \quad i = 1 \ \text{and} \ T_{L,r} > T_{s,i} \\ 1 & \text{if} \quad T_{s,i-1} \geq T_{L,r} > T_{s,i} \\ 0 & \text{if} \quad i = 0 \ \text{or} \ \text{if} \ i = N+1 \\ 0 & \text{otherwise} \end{cases} \tag{24}$$

The net mass flow rate from node i-1 to node i can be given by:

$$\begin{cases} \dot{m}_{m,1} = 0 \\ \dot{m}_{m,1} = \dot{m}_c \sum_{j=1}^{i-1} F_j^c - \dot{m}_L \sum_{j=i+1}^{N} F_j^L \\ \dot{m}_{m,N+1} = 0 \end{cases} \tag{25}$$

Eventually, i-esime node energy balance could be written as below:

$$m_i \frac{dT_{s,i}}{dt} = \left(\frac{UA}{C_p}\right)_i \left(T_a - T_{s,i}\right) + F_i^c \dot{m}_c \left(T_{co} - T_{s,i}\right) + \dot{Q}_{st} + \begin{cases} \dot{m}_{m,i}\left(T_{s,i-1} - T_{s,i}\right) & \text{if} \quad \dot{m}_{m,i} > 0 \\ \dot{m}_{m,i+1}\left(T_{s,i} - T_{s,i+1}\right) & \text{if} \quad \dot{m}_{m,i+1} < 0 \end{cases} \tag{26}$$

As it was explained before $\dot{Q}_{st}$ is the energy rate flowing from the storage tank into the heat exchanger.

$$\dot{Q}_{st} = F_i^L \dot{m}_L \left(T_{L,r} - T_{s,i}\right) \tag{27}$$

For solving equation 26 numerical methods are required and in this work Runge-Kutta method has been employed [23].

5.4 The Energy Balance on the A.A.H

The last control volume that should be considered is the A.A.H. One knows that the air flowing into the greenhouse set should carry the total thermal energy $\dot{Q}_{GR}$.

Therefore, the heating duty of A.A.H can be calculated as:

$$\dot{Q}_{a-3} = \dot{Q}_{GR} - \dot{Q}_{a-1} + \dot{Q}_{a-2} \tag{28}$$

All the parameters on the right hand of the equation above have been comprehensively discussed before. Thus, the consuming fuel mass flow rate in the proposed system can be calculated as:

$$\dot{m}_f = \frac{\dot{Q}_{a-3}}{LHV \cdot \eta_h} \tag{29}$$

6 Economic Assessment Approaches

In this work, net present value (NPV) economic analysis method was employed. The approach is used to assess the effectiveness of proposed system. The NPV method is a comprehensive method including all complementary costs such as inflation and R&M costs. Overall, the NPV is defined as below:

$$NPV = \sum_{t=1}^{N} \frac{R_t}{(1+i)^t} \tag{30}$$

Where, I, t and R_t introduce the interest/inflation rate, the time of cash flow and the net cash flow respectively [34]. Employing the above equation, one could calculate the payback period of the project.

7 Results and Discussion

As the key parameter of the calculation process, figure (5) shows the greenhouse location ambient temperature in (2013). As expected the least entrance temperatures belong to January and February with averages less than 5°C and the highest temperatures in June and July with averages more than 31°C. Based on what explained about the system performance details, for the temperatures over the red line in the graph (20 °C), the proposed system could not be effective enough as the greenhouse heating demand is zero.

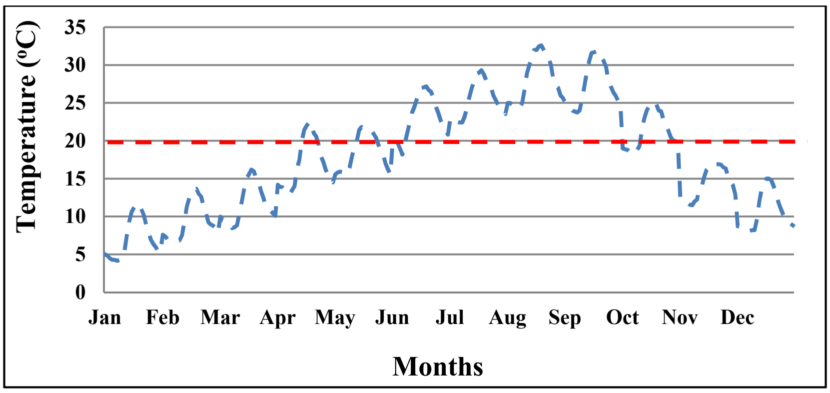

Figure 5 The monthly-hourly average ambient temperature in 2013

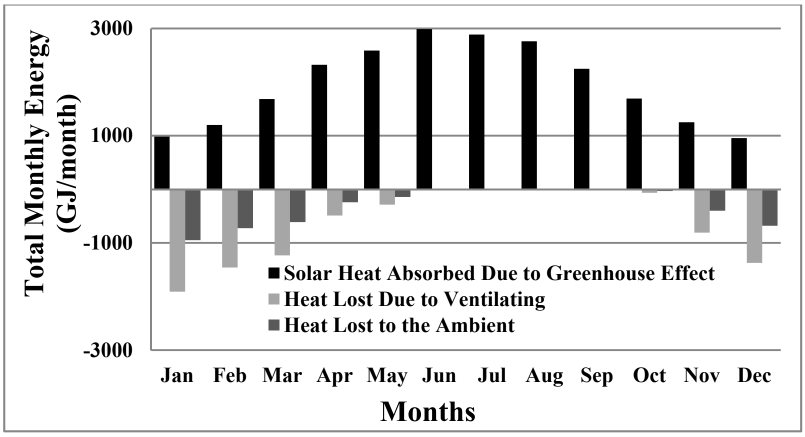

Figure 6 The total monthly energy value of the greenhouse in 2013

Figure (6) shows the amount of various effective parameters contributing in the total energy demand of the greenhouses. Clearly, the solar energy absorbed by the intrnal elements of the greenhouse has a positive value and on the other hand the both ventilating and heat transfer losts to the ambient have negative values. Naturally, during the times that the ambient temperature is lower, all the parameters meet lower values and in warmer months both negative and positive values tend to higher values.

The investigations show that the slope angle 45 °C is the most optimal angle of the solar collector set inclination for this region of Iran. Figure (7) presents the the total monthly absorbed solar radiation and the useful energy delivered to the working fluid by one collector in the proposed system.

Regarding the figure, the values corresponding with both of the considering parameters fluctuate sharlpy, while it was expected to gain more energy during summer and warmer months of the year. As the reason, it can be expressed that the considered slope angle is much more close to the inclination angle most appropriate for winter rather than the value associated with summer. As this factor is highly effective on the amount of absorbable solar energy the maximum value is related to one of the coldest months of the year i.e. March.

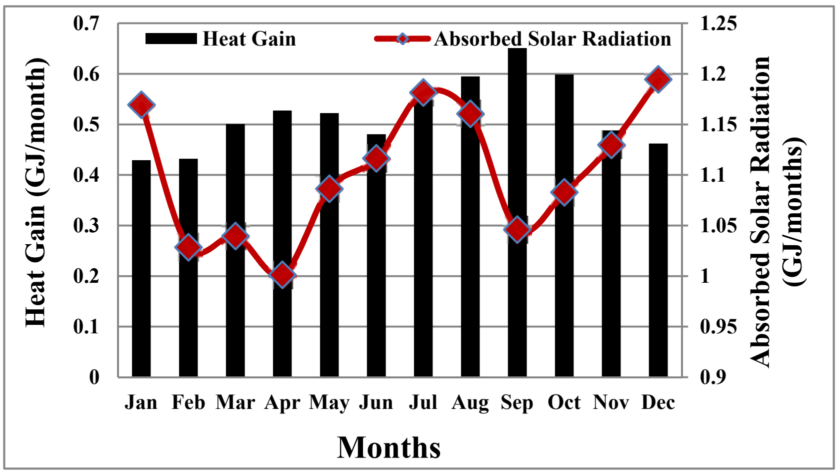

Figure 7 The total monthly absorbed solar energy and heat gain by one collector

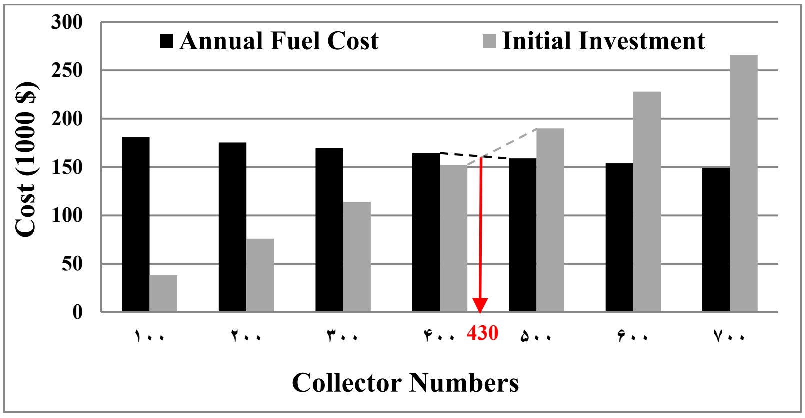

Figure 8 initial investment determinations for the solar heater system

Based on the initial investment determination approach thoroughly discissed in [37-39], the optimal solar system includes 430 flat plate collectors and a huge storage tank with 43 m^3 volume. The cost of capital for this system is 163,000 USD. Figure (8) shows how this system arised as the optimal investment for the first step of modification i.e. proposing the solar thermal heater.

It should be noted that the volume of the storage tank is proportional to the number of collectors so that for each collector module the storage tank volume should be 100 lit.

This advise has come from the previous study of the authors [33]. One should note that the fuel considered for preparing this figure is the consuming natural gas of this area of Iran. The natural gas is named Torkamn, with density of 0.578 kg/m^3 and the lowering heating value of 48.1 MJ/kg [47]. The natural gas price also has been considered based on the current natural gas global price 0.25 $\$/m^3$. Also, the solar system price is 380 USD for each collector and its belongings including a 100 lit storage tank. The seller company undertakes the whole installation process for free as the purchase condition.

Figure (9) shows the water mass flow rate between the storage tank and the S.H.E and the temperature of each node in the storage tank in a monthly-hourly averaged manner. The mass flow rates equal to zero refer to sunshine hours during the day in which the solar energy is supposed to be stored.

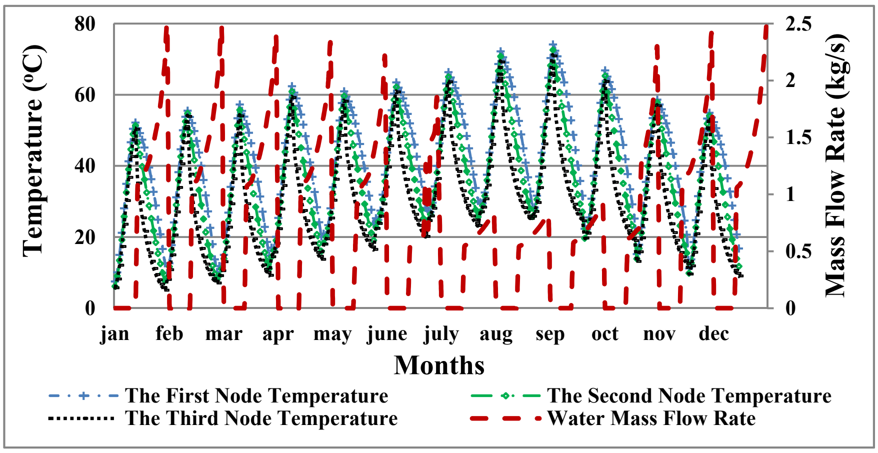

Figure 9 Different nodes temperatures in storage tank and water mass flow rate between storage tank and S.H.E

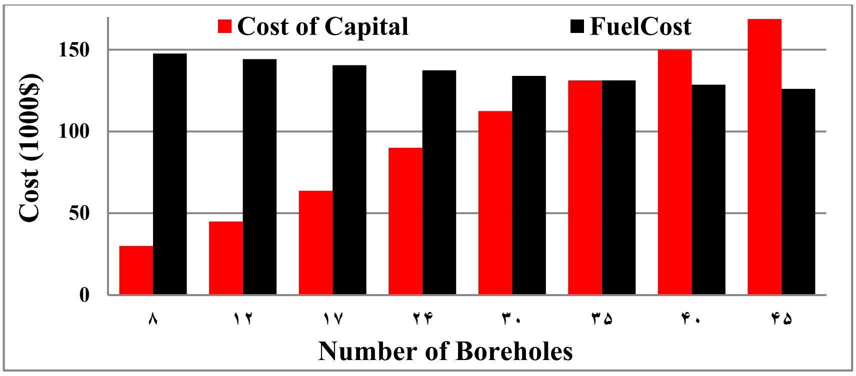

Figure 10 Initial investment for the geothermal heater system

Expectedly, higher storage temperatures are achievable during summer. The storage temperature could rise up to about 75 ºC for August and September. On the other hand, higher mass flow rates are required to supply the desired energy in cold months of the year where water flow rates around 2.5 kg/s are sometimes required in both Jnaruary and February.
The next step is defining the optimal initial investment of the goethermal heater system including the number of boreholes and the capacity of the storage tank and other belongings. According to figure (10), the optimum cost of capital for this system is 131,000 USD including 35 boreholes and a shell and tube heat exchanger with 17.5 m3 volume.

One should note that the total life time of such geothermal systems is considered 25 years. Over time, the geothermal boreholes lose their efficiency little by little. So that, the best acheievable performance of each borehole is in the first year of its application.

Figure (11) shows the total annual obtainable heat by the proposed geothermal borehole set of series including 35 boreholes. Based on what the figure shows, the total annual benefit of the first year of application is about 24,000 USD.

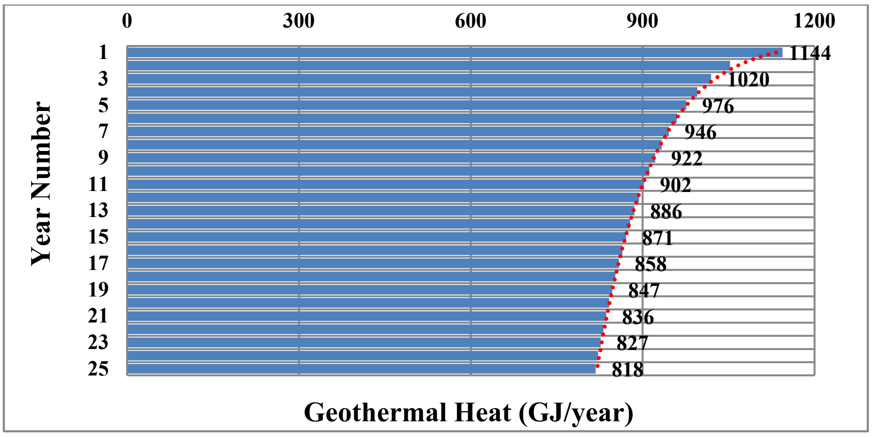

Figure 11 The total annual obtainable heat from the geothermal heat including 35 boreholes over time

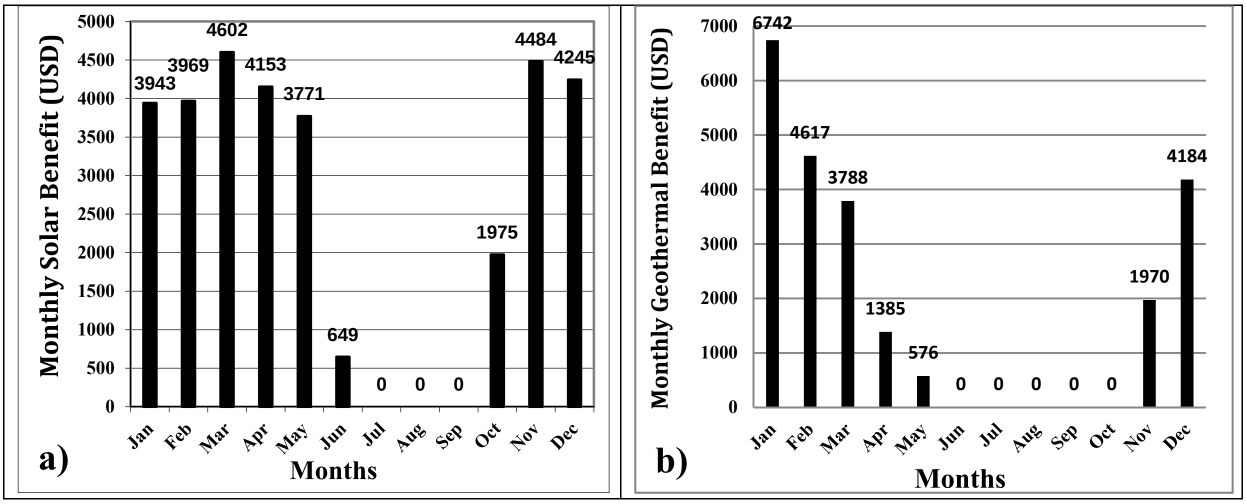

Figure 12 Monthly achievable revenue in the case study by the proposed solar (a) and geothermal (b) systems

As the figure shows, although the total obtainable heat from the proposed geothermal heat decreases over time, it is still efficient enough at the end of the 25th year by over 818 GJ/year. Note that the figure only illustrates the geothermal heat provided by this constraint that only air stream with temperatures less than 16 ^{o}C enter the geothermal heater.
Otherwise, the overall obtainable heat for all the considered years could be more than these values. The value used for preparing this was the average value of the whole life time.
Figures (12-a) and (12-b) show the total monthly obtainable revenue employing the proposed solar and geothermal heating systems, respectively.
According to the figures, expectedly, during summer the achievable benefit for both cases is zero as the air temperature is more than 20. It is noteworthy that the air exchange per hour for winter and cold months of the year has been considered two times per hour and for summer this value increases up to 20 times per. In solar heating system, the maximum achievable benefit is more than 4600 USD which occurs in March while the best performance of the geothermal system belongs to January in which the acheivable benefit exceeds 6700 USD.

Figure (13) also shows the three different heaters (G.H.E, S.H.E, A.A.H) contributions in providing the required heat in each month of the year.

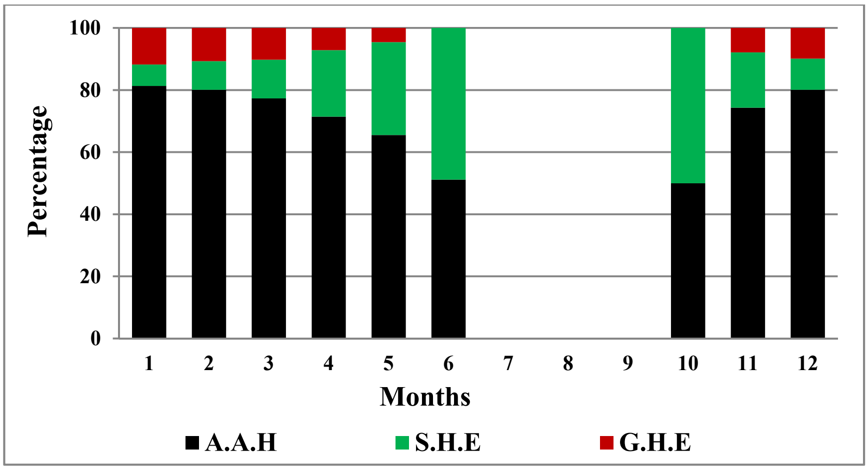

Figure 13 The contribution of each heater in energy preparation during each month

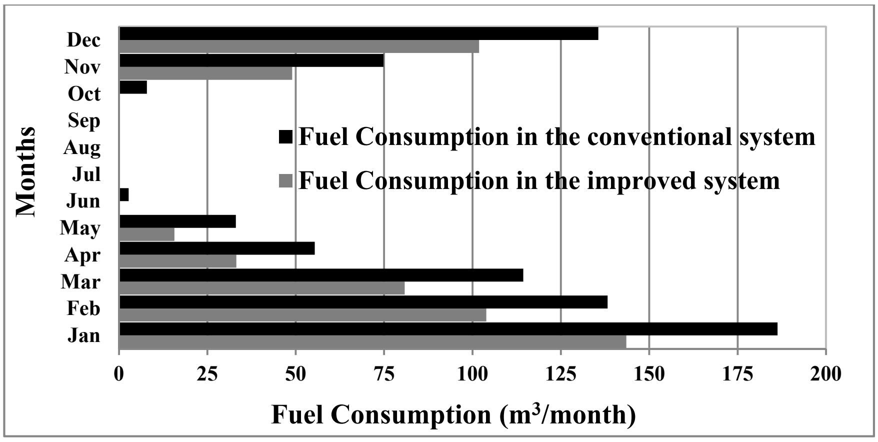

Figure 14 Fuel Comsumption conparison between the proposed and conventional systems

Reagrding the figure, in the colder months of the year the contribution of the geothermal system rises, while for warmer months of the year the contribution of the solar system for providing the required heat increases.

Figure (14) compares the total monthly fuel consumption of the greenhouses in the conventional and the proposed configurations. The figure shows that the maximum fuel consumption reduction takes place in January, February and December by 42, 34 and 33 km^3/month providences, respectively.

Finally, figure (15) shows the results of NPV analysis considering the inflation rate equal to the current inflation rate of USD and R&M costs equal to 5% of the total capital cost. As the figure proves, the payback period of the proposed system in the case study is less than 6.5 years. Considering the useful life time of 25 years for the both solar and geothermal system, thus, employing the proposed system is highly recommended.

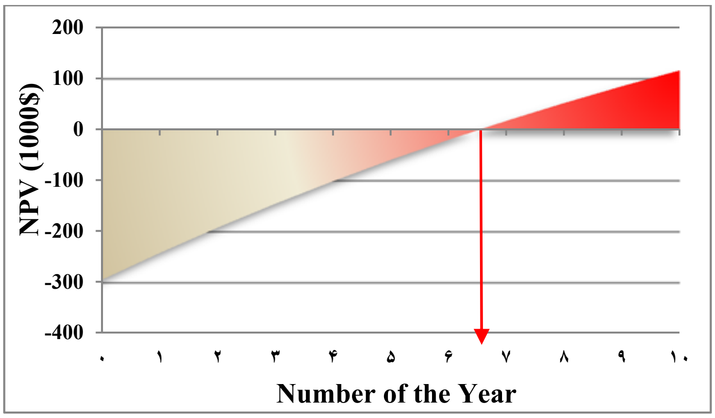

Figure 15 NPV analysis results

8 Conclusion

There are many places in industry which have high potential to be equipped to renewable and sustainable energies to modify the fuel usage pattern. Greenhouses are agricultre places in which huge amount of fossil fuel is used annually. This research proves that employing a combined system taking advantage of geothermal system and solar system can considerably improve the fuel consumption pattern of greenhouses.

The proposed system employs 430 flat plate collector and a solar storage tank with 43 m^3 volume in addition to a geothermal system including 35 boreholes and a shell and tube heat exchanger with 17.5 m^3 volume.

The total cost for the proposed system is 294,000 USD and based on the NPV approach the payback period is 6.5 years. Pay attention to this fact that the typical greenhouse of this work, is located in the north of Iran with moderate climate, the proposed system can even be more efficient for the central area of Iran with very warmer weather.

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Nomenclature

c_{pw}	Specific heat capacity of the water in the tank	(W/kg-K)
F_R	Removal factor of the collector	
I	Total heat transfer coefficient	(W/m^2)
$\dot{m}_f$	Mass flow rate of fuel consumed by the heater	(kg/s)
m_{st}	Mass of exist water in the storage tank	(kg)
$\dot{m}$	Mass flow rate	(kg/s)
$\dot{m}_{NG}$	Mass flow rate of the natural gas	(kg/s)
N	Air change number per hour	
$\dot{Q}_{GR}$	Greenhouse energy demand	(kW)

$\dot{Q}_{lost}$	Total heat lost from the greenhouse walls	(kW)
$\dot{Q}_{vent}$	Total ventilation heat lost from the greenhouse	(kW)
$\dot{Q}_{solar}$	Total solar heat gain by the greenhouse	(kW)
N_b	Number of borehole	
q	Heat transfer rate per unit length	(kW/m)
$r_{i,j}$	Radial distance between borehole i and j	(m)
R	Thermal resistance	(m^2. ºC /W)
S	Absorbable solar flux	(W/m^2)
T_a	Ambient temperature	(ºC-K)
T_{NG-1}	Natural gas temperature before heater	(ºC-K)
T_{NG-2}	Natural gas temperature after heater	(ºC-K)
T_w	Temperature of water in the tank	(ºC-K)
T_{st}	Temperature of water in the storage tank	(ºC-K)
T_w	Temperature of water in the tank	(ºC-K)
T_{fi}	Collector inlet working fluid temperature	(ºC-K)
T_{pm}	The average temperature of the collector	(ºC-K)
U_l	Total heat transfer coefficient	(W/m^2-K)
V	The greenhouse volume	(m^3)
$\dot{W}_{rev}$	Rate of maximum obtainable work	(W)
$\dot{W}_T$	Rate of maximum producible work by the turbine	(W)
y_i	Molar ratio	

Greek symbols

η_h	Heater thermal efficiency
η_i	Collector efficiency
$(\tau\alpha)_{ave}$	Average absorption-transmission coefficient of collector
β	Collector slop angle
γ	Collector azimuth angle

چکیده

در طی چند دهه اخیر، جایگزینی منابع سوخت های فسیلی با انرژی های تجدید پذیر و پایدار یکی از اصلی ترین دغدغه های دانشمندان و متخصصین این امر بوده است. علاوه بر این، گلخانه نقش کلیدی در تولید محصولات مختلف در طول سال و در آب و هوای غیرمتعارف بازی می کند. از اینرو، سیستم های گرمایشی و تهویه مطبوع در گلخانه ها برای ایجاد شرایط مطلوب محصول مورد نظر بکار گرفته می شوند.

درحال حاضر، به طور کلی گرمایش مورد نیازگلخانه ها توسط هیترهای گازوئیل سوز سنتی تامین می گردد. در این پژوهش، یک طرح نوآورانه و ترکیبی از انرژی های خورشیدی و زمین گرمایی به منظور کاهش میزان مصرف سوخت در گلخانه ها ارائه شده است. همچنین، به منظور نشان دادن اثربخشی سیستم پیشنهادی، یک تجزیه و تحلیل جامع اقتصادی حرارتی روی آن انجام می گیرد.

برای انجام آنالیز اقتصادی سیستم مورد نظر، روش ارزش خالص فعلی مورد استفاده قرار می گیرد و نتایج این ارزیابی اثبات کننده عملکرد فوق العاده رضایت بخش سیستم پیشنهادی با دوره بازگشت سرمایه ۶٫۵ سال می باشد.