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Achievement of geothermal energy using ground heat exchanger in Ma'en†

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Abstract

Geothermal energy in Jordan is a low-emission and renewable source that could provide households and commercial buildings with heating and cooling. Access to this 'free' energy may be available just a few feet underground. Thus, the objectives of this research are designing a ground heat exchanger that utilizes geothermal energy in heating to exchange the primary geothermal fluid with a secondary clean fluid, determining the feasibility of designing a ground heat exchanger system to pump geothermal energy under the weather conditions of the Ma`en area in Jordan and calculating the amount of energy saved. The design procedure involves applying energy and momentum equations around the geothermal fluid circuit. The FLUENT software program is used to calculate the ground heat exchanger parameters and the amount of energy saved. Finally, the feasibility study shows that the Geoexchange systems represent a savings to homeowners of around 70% in the heating mode, and up to 50% in the cooling mode compared to conventional fossil fuel systems.

Keywords: Geothermal energy; Economics, Capital cost; Ground heat exchanger; Heat pump; Ma'en

1. Introduction

Jordan is a key country in the Middle East. Despite being adjacent to several oil-rich countries, Jordan struggles to secure its resources of energy, especially when prices of oil increase. A large portion of its budget is spent on importing oil from various countries. The problem is aggravated year after another due to the growth in population and increase in electricity demand. Industrial development requires more fuel consumption and continuous operation of power plants; therefore, the search for alternative energy sources has become an imminent issue in Jordan. Renewable energy sources are fundamentally different from fossil fuel or nuclear power plants because of their widespread occurrence and abundance [1]. The primary advantage of many renewable energy sources is their lack of greenhouse gas and other emissions in comparison to fossil fuel combustion. Most renewable energy sources are low carbon emitters and do not introduce any risks such as nuclear waste [2].

Renewable energy sources can potentially supply several times the present world energy demand. They can enhance energy markets, secure long-term sustainable energy supplies, and reduce local and global atmospheric emissions. They can also provide commercially attractive options to meet specific needs for energy services (particularly in developing countries and rural areas), create new employment opportunities and offer possibilities for local manufacturing of equipment [3].

2. Geothermal resources in Jordan

Geothermal investigations revealed a rich geothermal potential in low enthalpy resources spread amongst several geothermal fields. The geothermal gradient map of Jordan shows two distinct regions of high geothermal gradients of up to 50 °C/km, as shown in Fig. 1. The first region is in the immediate vicinity of the east Dead Sea escarpment, where many springs discharge thermal water originating from the Lower Cretaceous Sandstone, forming three main geothermal fields. These fields are the Mukhiebeh thermal springs, Zara and Zarqa Ma'in and Afar and Burbeitta thermal springs. The second is near the border between Syria and Iraq. In this region, several thermal wells discharge water from the Upper Cretaceous Limestone. In both regions, there are many wells (shallow and deep) discharging thermal water, such as the shallow wells near Queen Alia airport, North Shuneh well and Mukheibeh well field [4].

The heat of thermal springs and wells is attributed to one or more of the following reasons [4]:

(1) Cooling of young bodies of basalt.

(2) Deep circulation of water in a normal geothermal gradient.

(3) Circulation of water in an abnormally high geothermal gradient related to crystal spreading across the Dead Sea Rift.

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Fig. 1. Geothermal Location in Jordan [4].

(4) Friction associated with lateral movement of active faults related to the Dead Sea Rift.

(5) Radioactivity: For example, the radiogenic heat production due to U238 disintegration series in the sandstone aquifer complex.

(6) Heat-storing horizontal (thermal blanket) consisting partly of dry sandstone overlain by marls with heat conductivity of only about half that of wet sandstone results in a temperature gradient of about twice the gradient of the whole sequence, maintaining herewith a constant heat flow. Based on the above reasons, it seems that the thermal water originates as groundwater in the Paleozoic sandstone to the east of the Dead Sea Rift escarpment. The water is heated by deep circulation in a moderately elevated geothermal gradient (may be 50 °C/km) which would imply circulation depths of about 2 km. The water moves towards the rift, ascends through fractures and is cooled by mixing with local ground waters before emerging as thermal springs [4].

2.1 Geothermal utilization in Jordan

2.1.1 Present utilization:

Geothermal energy is one of the alternative sources of energy which could be utilized for different purposes. Jordan is blessed with this energy source in several parts of the country. Thermal water in Jordan has been used directly as curative water. The therapeutic value of thermal waters in Jordan has been recognized since ancient times. Generally, thermal water has various properties in temperature and curative abilities. For example, the thermal water of Zara and Zarqa Ma'in springs is quite useful in treating Osteo Arthritis, Degenerative Disc and Post Traumatic disorders. The thermal water of North Shuneh is good for cervical spondylosing, while the thermal water of Afra and Burbeitta is quite good in treating Degenerative Disc and Post Traumatic problems. In the Zarqa Ma'in area, thermal water is utilized for medical purposes at a modern spa constructed in the area [4].

2.1.2 Future utilization

Thermal water in Jordan can be used in several direct uses such as Ref. [4]:

(1) Heating Greenhouses: The existing greenhouses in Jordan are of the unheated plastic tunnel type. The outside temperature drops to about zero or below in winter (November-February). Thus, greenhouses become unusable. Thermal water can heat the greenhouses to avoid such a problem [4].

(2) Fish Farming: At present, several farms producing tilapia exist in Jordan. For example, the Arab Fish Company farm consists of some 40 basins. It produces between 20 and 55 tons of tilapia per year. In the winter, temperatures cannot be maintained at a sufficient level to ensure the survival of the fingerlings and allow the growth of the fish. Certain thermal water sources, such as that at North Shuneh well, would enable new projects to be developed using geothermal energy to maintain the water in the basins at a sufficient temperature in the winter. The North Shuneh well discharges 700 m^3/h of thermal water at 57°C. It has great potential, part of which could be used to feed a pilot fish farm breeding tilapia and possibly other species so as to diversify production [4].

(3) Refrigeration by Absorption: The method of refrigeration by absorption allows one to create positive cold temperatures (from + 0.5 to + 5°C) for conservation of fruit and vegetables or negative cold temperatures (for example, -25°C) for conservation of meat or fish [4].

3. Design of heating system in Ma`en

A geothermal heat pump or ground source heat pump is a central heating and/or cooling system that pumps heat to or from the ground. It uses the earth as a heat source (in winter) or a heat sink (in summer). This design takes advantage of the moderate temperatures in the ground to boost efficiency and reduce the operational costs of heating and cooling systems, and may be combined with solar heating to form a geo-solar system with even greater efficiency. Geothermal heat pumps are also known by a variety of other names, including geoexchange, earth energy or water-source heat pumps [1]. Ground source heat pumps harvest a combination of geothermal power and heat from the sun when heating, but work against these heat sources when used for air conditioning [9].

Like a refrigerator or air conditioner, these systems use a heat pump to force the transfer of heat. Heat pumps can transfer heat from a cool to a warm space, against the natural direction of flow, or they can enhance the natural flow of heat from a warm area to a cool one. The core of the heat pump is a loop of refrigerant pumped through a vapor-compression refrigeration cycle that moves heat. Heat pumps are always more efficient at heating than pure electric heaters, even when extracting heat from cold winter air. But unlike an air-source heat pump, which transfers heat to or from the outside air, a ground source heat pump exchanges heat with the ground. This is much more energy-efficient because underground temperatures are more stable than air temperatures throughout the year. Seasonal variations decrease with depth and disappear below seven meters due to thermal inertia [2]. Like a cave, the shallow ground temperature is warmer than the air above during the winter and cooler than the air in the summer. A ground source heat pump extracts ground heat in the winter (for heating) and transfers heat back into the ground in the summer (for cooling). Some systems are designed to operate in one mode only, heating or cooling, depending on climate [9].

The setup costs are higher than for conventional systems, but the difference is usually returned in energy savings in 3 to 10 years. System life is estimated at 25 years for inside components and 50+ years for the ground loop. As of 2004, there are over a million units installed worldwide providing 12 GW of thermal capacity per year. About 80,000 units are installed in the USA and 27,000 in Sweden [9].

3.1 Ground heat exchanger

Heat pumps provide wintertime heating by extracting heat from a source and transferring it to the building. In theory, heat can be extracted from any source, no matter how cold, but a warmer source allows higher efficiency. A ground source heat pump uses the shallow ground as a source of heat, thus taking advantage of its seasonally moderate temperatures [10].

In the summer, the process can be reversed so the heat pump extracts heat from the building and transfers it to the ground. Transferring heat to a cooler space takes less energy, so the cooling efficiency of the heat pump benefits from the lower ground temperatures [10].

Shallow horizontal heat exchangers experience seasonal temperature cycles due to solar gains and transmission losses to ambient air at ground level. These temperature cycles lag behind the seasons because of thermal inertia, so the heat exchanger can harvest heat deposited by the sun several months earlier. Deep vertical systems rely heavily on the migration of heat from surrounding geology, unless they are recharged annually by exhaust heat from air conditioning [10].

Ground source heat pumps must have a heat exchanger in contact with the ground or groundwater to extract or dissipate heat. This component accounts for a third to half of the total system cost. Several major design options are available for these, which are classified by fluid and layout [10]. Direct exchange systems circulate refrigerant underground, closed loop systems use a mixture of anti-freeze and water, and open loop systems use natural groundwater [9].

3.2 Thermal efficiency

The net thermal efficiency of a heat pump should take into account the efficiency of electricity generation and transmission, which are typically about 40%. Since the system moves 3 to 5 times more heat energy than the electric energy it consumes, the total energy output is much greater than the input. This results in net thermal efficiencies greater than 100% for most electricity sources. Traditional combustion furnaces and electric heaters can never exceed 100% efficiency, but heat pumps provide extra energy by extracting it from the ground [9].

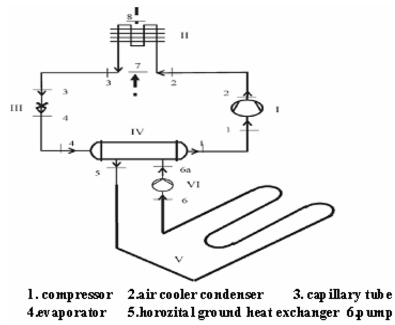

Fig. 2. Geothermal system.

3.3 Ground heat exchanger energy balance

For a general steady-state, steady-flow process, three balance equations, namely mass, energy and exergy balance equations, are employed to find the following parameters:
- Heat input
- Rate of exergy decrease
- Rate of irreversibility
- Energy and exergy efficiencies [5].

Fig. 2 shows the schematic diagram of the geothermal model system. The temperature of the potential resources in the Zara area (100 to 120°C) enables one to envisage the creation of refrigerated warehouses for the conservation of fruit and vegetables (temperature maintained between +0.5 and +5°C) [4].

The ground heat exchanger was used to pump geothermal fluid to the HVAC facilities of the Ma`en Hotel.

3.4 Energetic modeling

In general, the mass balance equation can be expressed in the rate form as Ref. [6]:

$$\sum m_{in} = \sum m_{out} \ . \tag{1}$$

The general energy balance is expressed below as the total energy input equal to total energy output, with all energy terms as follows:

$$Q + \sum m_{in} h_{in} = W + \sum m_{out} h_{out}$$
$$\overset{\circ}{W} = \overset{\circ}{W}_{net,out} = \overset{\circ}{W}_{out} - \overset{\circ}{W}_{in} \ . \tag{2}$$

() is the rate of net work output, and h is the specific enthalpy [7].

$$Q + \sum m_{in} h_{in} = W + \sum m_{out} h_{out} \ . \tag{3}$$

Assuming no changes in kinetic and potential energies with no heat or work transfers, the energy balance can be simplified to reflect only flow enthalpies [5]:

$$\sum m_{in}h_{in} = \sum m_{out}h_{out} . \quad (4)$$

The rate of heat extracted (absorbed) by the unit in the heating mode (the ground heat exchanger load), $\overset{\circ}{Q}_e$, is calculated from the following equation [5].

$$Q_e = m_{wa}C_{p,wa}(T_{out,wa} - T_{in,wa}) . \quad (5)$$

The heat rejection rate in the condenser is calculated by [5]:

$$Q_{cond} = m_{ref}(h_2 - h_1) . \quad (6)$$

The heat transfer rate in the evaporator is:

$$Q_{evap} = m_{ref}(h_1 - h_4) . \quad (7)$$

The work input rate to the compressor is:

$$W_{comp} = \frac{m_{ref}(h_{2s} - h_1)}{\eta_{i,comp}\eta_{m,comp}} . \quad (8)$$

In case the mass flow rate on the refrigerant side is not measured, the space heating load $\overset{\circ}{Q}_{sl}$ may be estimated as [5]:

$$\begin{aligned} Q_{s1} &= m_{air}C_{p,air}(T_{out,air} - T_{air,in}) \\ m_{air} &= \rho_{air}V_{air} \end{aligned} . \quad (9)$$

The COP of the GSHP can be calculated as [5]:

$$COP_{HP} = \frac{Q_{cond}}{W_{comp}} . \quad (10)$$

The coefficient of performance of the overall heating system (COP_{SYS}), which is the ratio of the condenser load to total work done by the compressor, the pumps (brine and water circulation pumps), and the fan-coil unit (or the condenser fan), may be computed by the following equation [5]:

$$COP_{sys} = \frac{Q_{cond}}{W_{comp} + W_{Pumps} + W_{fc}} . \quad (11)$$

The instantaneous usable energy collected by solar collector can be calculated as [5]:

$$Q = m_{wa}C_{P,wa}(T_{out} - T_{in}) . \quad (12)$$

3.5 Exergetic modeling

The general exergy rate balance can be expressed as follows [5]:

$$Ex_{heat} - Ex_{work} + Ex_{mass,in} - Ex_{mass,out} = Ex_{dest} \quad (13)$$

and, more explicitly:

$$\sum(1 - \frac{To}{Tk})Q_k - W + \sum m_{in}\psi_{in} - \sum m_{out}\psi_{out} = Ex_{dest} . \quad (14)$$

The specific exergy (flow exergy) of refrigerant (or water) is calculated by [5]:

$$\psi_{r,w} = (h - ho) - To(s - so) . \quad (15)$$

The total flow exergy of air is calculated from [5]:

$$\begin{aligned} \psi_a = {} & (C_{p,a} + \omega C_{p,v})To\{(\frac{T}{To}) - 1 - In(\frac{T}{To})\} \\ & + (1 + 1.6078\omega)RaToIn(\frac{p}{po}) \\ & + RaTo[(1 + 1.6078\omega)In[(1 + 1.6078\omega_o)/(1 + 1.6078\omega)] \\ & + 1.6078\omega In(\omega/\omega_o) \end{aligned} . \quad (16)$$

where the specific humidity ratio is: $\omega \equiv \frac{m_\upsilon}{m_a}$.

The exergy rate is calculated by [5]:

$$Ex = m\psi .$$

For exergy destruction (or irreversibility), the entropy generation S_{gen} is calculated first and used in the following equation:

$$i = Ex_{dest} = T_o S_{gen} . \quad (17)$$

The exergy destructions in the heat exchanger (condenser and evaporator), ground heat exchanger, pump, expansion valve, and solar collector are calculated, respectively, as follows [5]:

3.6 Heat exchanger

$$Ex_{dest,HE} = \sum Ex_{in} - \sum Ex_{out} = Ex_{dest} . \quad (18)$$

4. Ground heat exchanger

$$Ex_{dest,grh} = m_{wa}(\psi_{in} - \psi_{out}) + Q(1 - \frac{To}{T_{ground}}) . \quad (19)$$

4.1 Energetic modeling

In general, the mass balance equation can be expressed in the rate form as [8]:

$$\sum m_{in} = \sum m_{out} . \tag{20}$$

The general energy balance can be expressed as the total energy input equal to total energy output $\left(\overset{\bullet}{E}_{in} = \overset{\bullet}{E}_{out}\right)$ with all energy terms as follows:

$$Q + \sum m_{in} h_{in} = W + \sum m_{out} h_{out} \tag{21}$$

where $\left(\dot{Q} = \dot{Q}_{in,net} = \dot{Q}_{in} - \dot{Q}_{out}\right)$ is the rate of net heat input, and ($\overset{\circ}{W} = \overset{\circ}{W}_{net,out} = \overset{\circ}{W}_{out} - \overset{\circ}{W}_{in}$) is the rate of net work output, and the specific enthalpy [5].

Assuming no changes in kinetic and potential energies with no heat or work transfers, the energy balance can be simplified to flow enthalpies as follows:

$$\sum m_{in} h_{in} = \sum m_{out} h_{out} . \tag{22}$$

In our work, the total rate of heat can be determined by the following equation:

$$Q_T = Q_{solar} + Q_{geothermal} \tag{23}$$

where

$$\begin{array}{l} Q_{solar} = m_{wa} Cp_{wa} (T_{out,2,wa} - T_{out,1,wat}) \\ Q_{geo} = m_{wa} Cp_{wa} (T_{out,1,wa} - T_{in,wa}) \end{array} . \tag{24}$$

4.2 Heat exchanger

Heat exchangers are devices built for efficient heat transfer from one fluid to another and are widely used in engineering processes. By applying the first law of thermodynamics to a heat exchanger working at steady-state condition, we obtain [11]:

$$\sum m_i \Delta h_i \equiv 0.$$

- Recuperative type, in which fluids exchange heat on either side of a dividing wall.
- Regenerative type, in which hot and cold fluids occupy the same space containing a matrix of material that works alternatively as a sink or source for heat flow.
- Evaporative type, such as a cooling tower in which a liquid is cooled or evaporated in the same space as the coolant.

The best type of heat exchanger, which is the most common in practice, is the recuperative type, and the design of our heat exchanger will be based on the counter flow type [11].

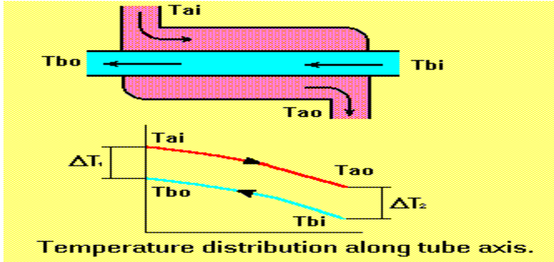

Fig. 3. Counter-flow heat exchanger.

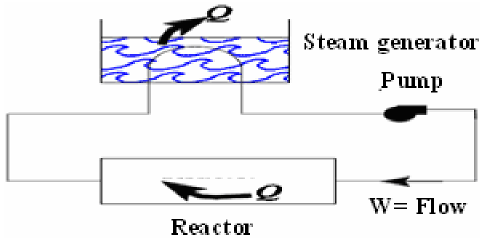

Fig. 4. HTS simplified schematic.

4.3 Counter-flow heat exchanger

Fig. 3 shows a fluid flowing through a pipe and exchanging heat with another fluid through an annulus surrounding the pipe. In a counter-flow heat exchanger, fluids flow in opposite directions, the specific heat capacity of fluids is constant, and it can be shown that [12] $\frac{dQ}{dt} \equiv UA\Delta T$

where

$\frac{\partial Q}{\partial t}$ represents rate of heat transfer between two fluids, and

$$\Delta T \equiv \frac{(\Delta T_1 - \Delta T_2)}{\ln\left(\frac{\Delta T}{\Delta T_2}\right)} . \tag{25}$$

5. Model theory

5.1 Reactor heat balance

Heat is generated by nuclear fission, transferred to a moving heat transport medium and carried by this medium to the steam generators for steam production. This is indicated in Fig. 4 [13].

Performing an energy balance around the reactor, the energy coming out of the reactor equals the energy going in plus the reactor energy generation. Thus [13],

$$Wh_o \equiv Wh_t + Q \qquad \text{or} \qquad Q \equiv W\left(h_o - h_t\right).$$

5.2 Steam generator heat transfer

Neglecting minor factors such as pump heat, piping heat losses, pump gland seal leakage and miscellaneous heat losses via auxiliary systems, the power transferred to the steam gen-

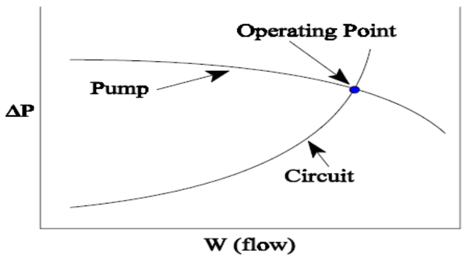

Fig. 5. Circuit losses and pump head vs. flow.

erator is Q (kW) [13].

The heat transfer at any point in the steam generator is given by Fourier's law:

$$dQ \equiv U\left(T_p - T_s\right)dA .$$

U is a function of flow, temperature, the amount of boiling (quality), the physical layout, heat exchanger tube material and the degree of curding or fouling in the steam generator [13].

Thus, the total heat transfer is:

$$Q \equiv \int_Q dQ \equiv \int_A U\left(T_p - T_s\right)dA_s .$$

5.3 Primary side flow

A final primary heat transport system relation is needed to complete this approximate picture. The primary side flow is determined by a balance between the head generated by the primary pumps and the circuit head losses due to friction, as shown in Fig. 5 [13].

$$\Delta P_{pump} \equiv \Delta P_{circuit}$$

It can be approximated by a power series:

$$\Delta P_{pump} \equiv A_o + A_1 W + A_2 W^2 + ... \qquad (26)$$

The circuit losses obey the classical velocity squared relationship to a first order approximation [13]:

$$\Delta P_{circuit} \equiv KW^2$$

where K can be a complex function of material properties and pipe geometry details.

5.4 Secondary side flow

The secondary side steam flow can be calculated by an energy balance on the secondary side of the boiler (similar to that done for the reactor) [13]:

$$Q \equiv W_{steam}\left(h_{steam} - h_{feedwater}\right).$$

The feed water temperature (hence enthalpy) is given by the turbine manufacturer. The steam temperature (hence enthalpy) is set by the controlled steam pressure. Thus, the steam flow is [13]:

$$W_{steam} \equiv \frac{Q}{\left(h_{steam} - h_{feedwater}\right)}.$$

5.5 Approximate solution

The primary heat transport approximate conditions are set, then, by the simultaneous solution of the energy balance at the core, the energy balance at the steam generator and the momentum balance around the circuit. The secondary side is quantified by an energy balance at the steam generator [13].

5.6 Secondary side

FLUENT software program was used to produce the result under the climate of Jordan weather (Ma'en area).

In summary [13]: $Q = W\left(h_o - h_t\right)$

$$Q = \frac{UA}{C_p}\left[\frac{h_{o+}h_t}{2} - h_s\right] = UA\left[\frac{T_{o+}T_t}{2} - T_s\right] \qquad (27)$$

$$\Delta P_{pump} = \Delta P_{circuit}$$

$$Q = W_{steam}\left(h_{steamo} - h_{feedwatert}\right)$$

$$\rightarrow h_o \equiv \frac{Q}{W} + h_i .$$

Substituting into Eq. (27),

$$Q \equiv \frac{UA}{C_P}\left[\frac{h_o + h_1}{2} - h_s\right] \equiv UA\left[\frac{T_o - T_1}{2} - T_s\right]$$

$$\frac{Q}{W} \equiv \frac{UA}{C_P W}\left[\frac{Q}{2W} + h_i - h_s\right] \equiv \frac{UA}{W}\left[\frac{Q}{2C_P W} + T_i - T_s\right]. \qquad (28)$$

Solving for h_i gives:

$$h_t \equiv \frac{Q}{W}\left[\frac{C_P W}{UA} - \frac{1}{2}\right] + h_s \equiv \left[\frac{QC_P}{UA} + h_s - \frac{Q}{2W}\right]$$

$$T_t \equiv \frac{Q}{W}\left[\frac{W}{UA} - \frac{1}{2C_P}\right] + T_s . \qquad (29)$$

Since all parameters, Q, W, Cp, A, U, etc., are positive quantities, the reactor inlet enthalpy (and hence the inlet temperature) will rise as flow increases, will rise as secondary side temperature and enthalpy increase and may go up or down as power changes [13].

The reactor outlet enthalpy (h_o) is directly related to h_i.

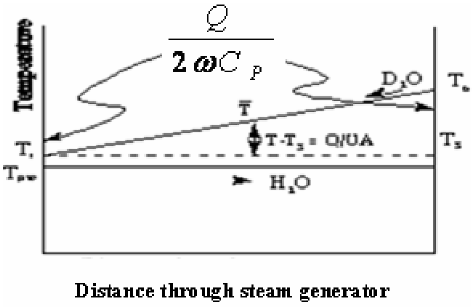

Fig. 6. Temperature variations.

Thus,

$$T_o \equiv \frac{Q}{W}\left[\frac{W}{UA}+\frac{1}{2C_P}\right]+T_s$$

$$h_t \equiv \frac{Q}{W}+h_i \equiv \frac{Q}{W}+\frac{Q}{W}\left[\frac{C_P W}{UA}-\frac{1}{2}\right]+h_s \equiv \equiv \frac{Q}{W}\left[\frac{C_P W}{UA}+\frac{1}{2}\right]+h_s \quad (30)$$

The average enthalpy in the core and the steam generator is [13]:

$$h_{aver} \equiv \frac{h_o+h_i}{2} \equiv \frac{Q}{W}\left(\frac{C_P W}{UA}\right)+h_s \equiv \left(\frac{QC_P}{UA}\right)-h_s$$

$$T_{aver} \equiv \frac{T_o+T_i}{2} \equiv \frac{Q}{UA}+T_s \quad (31)$$

which means that T_{averg} is a simple linear function of the reactor power, Q.

Fig. 6 shows the spread variation of temperature through the steam generator.

On the other hand, in **FLUENT**, the heat exchanger core is treated as a fluid zone with momentum and heat transfer. Pressure loss is modeled as a momentum sink in the momentum equation, and heat transfer is modeled as a heat source in the energy Eq. [14].

5.7 Stream-wise pressure drop

In the heat exchanger model, pressure loss is modeled using the porous media model in **FLUENT**. The porous media inputs are automatically set based on user inputs to the heat exchanger model. The stream-wise pressure drop can be expressed as [14]:

The stream-wise pressure gradient

$$\frac{\partial \rho}{\partial s} = \frac{1}{2} f \rho_m U_{A_m}^2 .$$

The pressure loss coefficient is computed from

$$f \equiv \left(K_c+1-\sigma^2\right)-\left(K_e+1-\sigma^2\right)+f_c \frac{A}{A_c} .$$

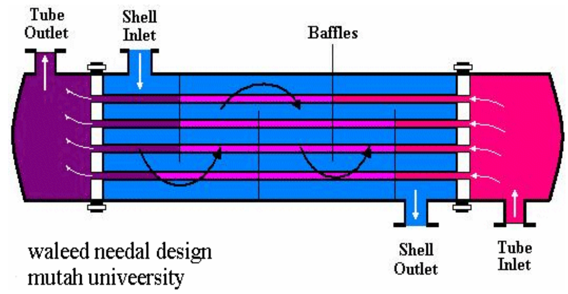

Fig. 7. The shell and tube design.

It is necessary to specify the core friction coefficient and exponent when setting up the heat exchanger model.

The Reynolds number in the equation is defined as

$$\mathrm{Re}_{\min} \equiv \frac{\rho_m U_{A_{\min}} D_h}{\mu_m} .$$

For a heat exchanger core, the hydraulic diameter can be defined as $D_h \equiv 4L\left(\frac{A_c}{A}\right)$.

Note that U_{A_m} can be calculated from $UA_{\min} \equiv \frac{U}{\sigma}$.

5.8 Heat rejection

Heat rejection is computed for each cell within a macro and added as a source term to the energy equation for the air flow. The heat transfer for a given cell is computed from

$$q_{cell} \equiv \varepsilon\left(m^\circ C_p\right)_{air}\left(T_{in}-T_{cell}\right) . \quad (32)$$

The total heat rejection from the heat exchanger core is computed as the sum of the heat rejection from all the macros:

$$q_{total} \equiv \sum_{macrol} q_{macrol} .$$

The coolant inlet temperature to each macro is computed based on the energy balance of the coolant flow. For a given macro,

$$q_{macro} \equiv \varepsilon\left(m^\circ C_p\right)_{coolent}\left(T_{in}-T_{cell}\right) \quad (33)$$

where T_{in} and T_{out} are the inlet and outlet temperatures of the coolant in the macro, respectively. The value of T_{out} then becomes the inlet temperature to the next macro.

6. Calculation

Fig. 7 shows the shell and tube design.

All calculations are inserted in the appendix

Table 1. Annual greenhouse gas savings from using a ground source heat pump instead of a high-efficiency furnace in a detached residence [9].

Country	Electricity CO_2 Emissions intensity	GHG savings relative to		
		Natural gas	*Heating oil*	*Electric heating*
Canada	223 ton/GWh	2.7 ton/yr	5.3ton/yr	3.4 ton/yr
USA	676 ton/GWh	0.5ton/yr	2.2 ton/yr	10.3 ton/yr
China	839 ton/GWh	1.6 ton/yr	1.0 ton/yr	12.8 ton/yr

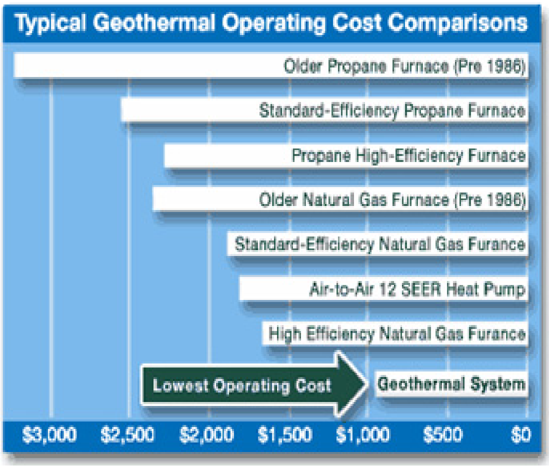

Fig. 8. Typical geothermal operating cost comparisons [14].

6.1 Payback period for your geothermal heat pump system

When a geothermal system is installed in a new home, the monthly savings in operating costs will generally offset the additional monthly cost in the mortgage, resulting immediately in a monthly positive cash flow. Energy savings is only one of the many benefits of a geothermal system [14].

Fig. 8 shows typical geothermal operating cost comparisons with other alternative heating and central air conditioning systems.

6.2 Environmental impact

Ground source heat pumps are the most energy-efficient, environmentally clean, and cost-effective space conditioning systems available. Heat pumps offer significant emission reductions potential, particularly where they are used for both heating and cooling and where electricity is produced from renewable resources [9].

Ground-source heat pumps have unsurpassed thermal efficiencies and produce zero emissions locally, but their electricity supply almost always includes components with high greenhouse gas emissions. Their environmental impact, therefore, depends on the characteristics of the electricity supply. The GHG emissions savings from a heat pump over a conventional furnace, which is shown in Table 1, can be calculated based on the following formula [9]:

$$GHG \cdot Saving \equiv HL\left(\frac{FI}{AFUE \times 1000\frac{kg}{ton}} - \frac{EI}{COP \times 3600\frac{\sec}{hr}}\right). \tag{34}$$

- HL = seasonal heat load ≈ 80 GJ/yr for a modern detached house in the northern USA
- FI = emissions intensity of fuel = 50 kg(CO_2)/GJ for natural gas, 73 for heating oil
- AFUE = furnace efficiency ≈ 95% for a modern condensing furnace
- COP = heat pump coefficient of performance ≈ 3.2 seasonally adjusted for northern USA heat pump
- EI = emissions intensity of electricity ≈ 200-800 ton(CO_2)/GWh, depending on region

Ground-source heat pumps always produce less greenhouse gases than air conditioners, oil furnaces, and electric heating, but natural gas furnaces may be competitive depending on the greenhouse gas intensity of the local electricity supply. In countries like Canada and Russia with low emitting electricity infrastructure, a residential heat pump may save 5 tons of carbon dioxide per year relative to an oil furnace, or about as much as taking an average passenger car off the road. Yet, in countries like China or USA, which are highly reliant on coal for electricity production, a heat pump may result in 1 or 2 tons more carbon dioxide emissions than a natural gas furnace [9].

7. Conclusion

Several investigations of geothermal energy in Jordan have taken place over the last three decades. The geothermal gradient map of Jordan shows two distinct regions of high geothermal gradients of up to 50°C/km, where many springs discharging thermal water are available. These fields include the Mukhiebeh thermal springs, Zara and Zarqa Ma'in and Afar and Burbeitta thermal springs, etc. In the Dead Sea and Zarqa Ma'in area, there is a modern spa where thermal water is utilized for medical purposes. Thus, several hotels and tourist resorts have been built to encourage tourist to come to Jordan. These hotels and tourist resorts used fossil fuel (diesel) as the main fuel for heating applications. Consequently, introducing the extracted geothermal energy from these springs into the energy mixture of these heating systems is vital and strategic. The following achievements are deduced as a result of introducing the ground heat exchanger into the hotels' heating systems:

- Provision of 40% to 60% of total energy consumption.
- Reduced boiler size.
- Reduced emissions and enhanced environmental protection.

Nomenclature

$\dot{m}$: Air is the mass flow rate of air
Δh_i : Change of specific enthalpy
Cp, : Air is the specific heat of air
U : Overall heat transfer coefficient
V : Air is the volumetric flow rate of air,
A : Area of the tube
Q_k : The heat transfer rate
Δ T : Logarithmic mean temperature difference
W : The work rate,
H : The enthalpy,
c : The flow exergy,
S : The entropy,
F : Stream wise pressure gradient
W : Coolant mass flow rate (kg/s);
ρ_m : Mean air density
ho : Core exit enthalpy (kJ/kg);
U_{A_m} : Air velocity at the minimum flow area
hi : Core inlet enthalpy (kJ/kg);
σ : Minimum flow to face area ratio
Q : Reactor power transferred to the coolant (kJ/s or kW)
K_c : Entrance loss coefficient
U : Overall heat transfer coefficient
Ko : Exit loss coefficient
A : Heat transfer area
A : Air-side surface area
T_p : Primary side temperature
Ac : Minimum cross-sectional flow area
T_p : Secondary side temperature
f_c : Core friction factor
ε : Heat exchanger effectiveness
a : Core friction coefficient
$m^{\circ}C_P$: Flow rate specific heat
b : Core friction exponent
T_{in} : Coolant inlet temperature
$\mathrm{Re}_{\min}$: Reynolds number
T_{cell} : Cell temperature/
ρ_m : Mean air density
D_h : Hydraulic diameter
μ_m : Mean air viscosity
U_{A_m} : Air velocity

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Appendix

A.1 Calculation, assumptions and restrictions

The following assumptions are made in the heat exchanger model:

- The heat exchanger effectiveness, ε, is defined for a complete heat exchanger, and can be applied to a small portion of the heat exchanger represented by a computational cell.
- The air capacity rate, $(\dot{m}.C_p)_{air}$, is less than the coolant capacity rate.
- The cell temperature, T_{cell}, (i.e., the cell central value) can be used instead of the temperature of the fluid entering the cell.
- Flow acceleration effects are neglected in calculating the pressure loss coefficient.
- The coolant is restricted to a single phase [22-23].

Sample calculation:-
First, we will calculate shell side inlet nozzle pressure drop
@
Nozzle diameter = 75 mm
Density at inlet temperature = 981,6327 kg/m^3
Inlet mass flow rate = 36000 kg/hr
Then we calculate pressure drop and find it = 3913.1353 pas
Shell side outlet nozzle pressure drop
@
Nozzle diameter = 75 mm
density at inlet temperature = 995.6502 kg/m^3
inlet mass flow rate = 36000 kg/hr
then we calculate pressure drop and find it = 1286.0438 pas
tube side inlet nozzle pressure drop
@
nozzle diameter = 100 mm
Density at inlet temperature = 996.5161999 kg/m^3
Inlet mass flow rate = 18000 kg/hr
Then we calculate pressure drop and find it = 203.2521 pas
tube side outlet nozzle pressure drop
@
Nozzle diameter = 100 mm
density at inlet temperature = 965.6981999 kg/m^3
Inlet mass flow rate = 18000kg/hr
Then we calculate pressure drop and find it = 205.5196 pas
Using fluid pumping power calculator
@
Mass flow rate = 10 kg/s
Density = 981,6327 kg/m3
Pumping efficiency = 0.80
we find that pumping power = 0,05 kw
Shell side data
Temperature
From = 63 C
T_O = 30 C
FLOW RATE = 10 kg/s

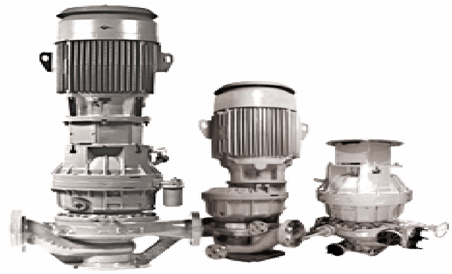

Fig. A.1. Sundyne centrifugal pumps.

Specific heat capacity = 4181j/kg.k
Thermal conductivity = 0,58 w/m k
Viscosity = 0.00061675 kg /ms
Density = 988.64145 kg/m^3
Fouling factor = 0.00035 m^2 C/w
Specific enthalpy = 194.735 kj/kg
Maximum pressure drop = 3913.1353 pas
Tube side data
Temperature
From = 27 C
TO = 55 C
FLOW RATE = 5kg/s
Specific heat capacity = 4181j/kg.k
Thermal conductivity = 0,58 w/m k
Viscosity = 0.0007035 kg /ms
Density = 988,64145 kg/m^3
Fouling factor = 0.00018 m^2 C/w
Specific enthalpy = 171.965 kj/kg
Maximum pressure drop = 1286.0438 pas
Then using programes to calaulate Q
Q= 1379.73 KW
Then assumed overall heat transfer coefficient
U= 200 W/m^{2}C
We select heat exchanger type : fixed – tube plate exchanger layout
No of shell passes 65
No of tube passes 50
Ds = 10 mm
Outside diameter OD = 25 mm
Pitch diameter = 5 mm
Tube length = 2 m
Inside diameter = 15 mm
BWG = 1
TUBE thickness = 5 mm
Tube arrangement is square
We choice counter current flow
Then we can calculate the log mean temperature difference
T lm = 5,0977 C
True temperature difference use R and S value to find the temperature correction factor
R= 1.1786
S= 0.7778
Estimate temperature correction = 16.34511

Consequently, centrifugal and Rotary Positive Displacement pumps are selected. Sundyne centrifugal pumps are best for process and general industry applications and come in direct drive and integrally geared configurations.

A.2 Total cost of this work

We select heat exchanger type: fixed – tube plate exchanger layout.

Pump cost that can do this work equal 75 JD.

We needed net work of pipes at about 60 meter length and its cost about 250 JD.

We need valves and anther connecting parts in cost equal 35 JD.

Design heat exchanger will tack about 500 JD.

Other cost as man worker 300 JD.

TOTAL COST =1220 JD.

This work can save about half the energy consumed at heating in winter and cooling in summer.

The original cost of energy of the hotel = 6500 JD/YEAR.

After this process, the cost will decrease to 3000 JD/YEAR.

Consequently, the operational cost will be:

$$Q = mC_P(T_2 - T_1)$$
$$Q_{load} = 10 * 42000 * 30 = 12600000 \text{ J} = 12600 \text{ KJ}.$$

While the operational cost before installing the heat exchanger was

$$Q_{load} = 10 * 42000 * 60 = 26200000 \text{ J} =26200 \text{ KJ}.$$

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