OPTIMIZATION OF HEAT EXCHANGERS FOR GEOTHERMAL DISTRICT HEATING

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ABSTRACT

This paper analyzes the optimal configuration and operating parameters of a heat exchanger in a geothermal district heating system. An optimization algorithm is presented for the non-linear constrained problem to maximize the annual net profit for a system of counter-flow heat exchangers. Several parameters that affect the net profit are examined, including the mass flow rates of working fluids and heat transfer area, which both directly affect the outgoing temperatures. The performance of the heat exchanger and fuel savings by reducing fuel consumption to generate heat are modeled within the problem formulation. Also, power input to the pump for fluid circulation is included. By formulating these multiple parameters over a wide range of design conditions, the algorithm presents a useful new design tool for the improvement of heat exchanger networks in geothermal systems.

OPTIMISATION D'UN ÉCHANGEUR DE CHALEUR POUR LE CHAUFFAGE GÉOTHERMIQUE URBAIN

RÉSUMÉ

Nous analysons dans cet article la configuration optimale et les paramètres d'opération d'un échangeur de chaleur dans un système de chauffage géothermique urbain. Nous présentons un algorithme d'optimisation pour le problème de contrainte non-linéaire, afin de maximiser le profit annuel net pour un système d'échangeurs de chaleur à contre-courant. Plusieurs des paramètres qui affectent le profit net sont examinés, incluant les taux du débit massique des fluides en opération et le secteur d'échange de chaleur, qui tous les deux affectent directement les températures à la sortie. Dans la formulation du problème, nous avons modélisé la performance de l'échangeur de chaleur et les économies de combustible en réduisant la consommation de combustible pour la production de la chaleur. Nous avons inclus également la puissance d'entrée à la pompe pour l'écoulement du fluide. En formulant ces multiples paramètres sur une large gamme de conditions de conception, l'algorithme présente un outil de conception nouveau et utile pour l'amélioration des réseaux d'échangeurs de chaleur dans les systèmes géothermiques

1. INTRODUCTION

Heat exchangers are important components of geothermal systems, industrial processes such as chemical refineries, vehicles and many other applications. Geothermal heating is a source of energy that is a technically-proven, cost effective source of electrical and thermal energy. Recent estimates show that over 6,000 MW electrical and 8,200 MW thermal energy from geothermal heating are currently developed throughout the world [1–3]. Geothermal resources can be divided into three broad groups: low, moderate-temperature, and high temperature systems. Systems at a temperature of 90 °C or lower are called low-temperature geothermal sources, which are best suited for space heating and cooling [3–5]. Geothermal sources between 90 °C and 150 °C are called moderate-temperature. High temperature geothermal sources are generally up to 250 °C and these are typically used for power generation. Generation of electricity in geothermal power plants is the most common use of high-temperature geothermal sources. In most cases, the reservoir is covered with impermeable rocks that prevent hot fluids from easily reaching the surface, so they are kept under pressure. This includes superheated steam, steam mixed with water, or hot water only, depending on the hydro-geological structure and temperature of the rocks. To reach those sources, wells are normally drilled into the reservoir to extract hot fluids [6].

Geothermal district heating has been one of the fastest growing segments of the geothermal space heating industry. It accounts for over 75 percent of all space heating from geothermal sources worldwide [7]. The U.S. currently has the largest geothermal production capacity [8]. The primarily low-temperature sources are used for heating and cooling applications for space and district heating, greenhouse heating, fish farming, process heating, snow melting, swimming pool heating, and other industrial applications. For most moderate-temperature sources, electricity generation is not economical due to the low thermal efficiency. Geothermal energy is more effective when used directly rather than conversion to electricity, as the direct use of geothermal heat in processes like heating and cooling can replace the burning of fossil fuels. Another recent application for geothermal energy is hydrogen liquefaction for transportation and storage. In the long term, district heating systems will likely prove to be more cost effective in comparison to these other applications [9–12].

In geothermal heat exchanger systems, there are two types of fluid streams moving through a network of heat exchangers, i.e., hot/cold water and the geothermal fluid. In this paper, heat from geothermal brine is transferred to circulating water that flows through a heat exchanger. Due to many simultaneous design criteria and objectives, optimization algorithms are useful design tools to improve performance of heat exchangers[13]. Sogut and Durmayaz [14] analyzed a solar heat engine and maximized the power density by separating irreversibilities of the solar concentrating collector and heat exchangers. Bojic [15] optimized the heating and cooling loads of a building using an energy-module network and dynamic programming. The system consisted of an air-to-air heat pump, heat-recovery exchanger, air-to-earth heat exchanger, and an air-mixing device. Morton [16] optimized a heat-exchanger network using a nonlinear program methodology to minimize the energy cost. A design optimization tool based on genetic algorithms was developed by Kumar et al. [17] for ground-air heat exchangers. The optimization algorithm used a search procedure to generate possible design solutions, which are then evaluated in terms of passive heating and cooling of the building. Results were presented for the sizing of an earth-to-air heat exchanger in a non air-conditioned residential building. Wang et al. [18,19] developed a novel Collaboration Pursuing Method for multidisciplinary design optimization of thermofluid systems. Gholap et al. [20] developed a multi-objective optimization procedure to obtain the optimal values for design variables, in order to minimize the energy consumption and material costs in heat exchanger for refrigerators.

In this paper, the method of Sequential Quadratic Programming (SQP) is used for heat exchanger optimization [21]. In particular, a counter-flow plate heat exchanger is analyzed for a geothermal system. The optimization extends the past study by Dagdas [5] to maximize the net annual profit of the heat exchanger in a geothermal district heating system.

2. PROBLEM DESCRIPTION

Consider a counter-flow heat exchanger depicted in Fig. 1. In a past study by Dagdas [5], only one design variable (heat transfer area) was considered for optimization of the net annual profit in the geothermal system. This paper extends the work of [5] to develop a more comprehensive formulation with additional design variables and realistic constraints. For example, the power required to circulate the brine fluid is considered. As the heat transfer area increases within the heat exchanger, the net input power to the pump also increases due to additional pressure losses, so this factor should be included in the overall optimization. The total annual profit takes into account the amount of fossil fuel required to heat a boiler and energy savings by replacing the energy source with a geothermal heating system, as well as the amount of energy required by the pump to circulate geothermal fluid in the system.

In the counter-flow heat exchanger of Fig. 1, two streams are shown, with a hot fluid or geofluid (geothermal fluid) entering the heat exchanger and leaving the other side at a lower temperature. The cold fluid is circulating water, which enters the heat exchanger and leaves from the other side with a higher temperature. In a counter-flow plate heat exchanger, there is no interaction and mixing between the two fluids. This can be advantageous with respect to the overall heat exchange. In the problem formulation, the inlet temperatures are known, and the parameters to be optimized are the heat transfer area and mass flow rates of the circulating water and geofluid streams. The model is formulated for typical low temperature geothermal resources at around 80 °C geothermal temperatures. The objective function of the problem is defined to establish the optimum values of the heat transfer area, mass flow rate of hot water, and mass flow rate of cold water, in order to maximize the annual net profit of a geothermal district heating system [7,22].

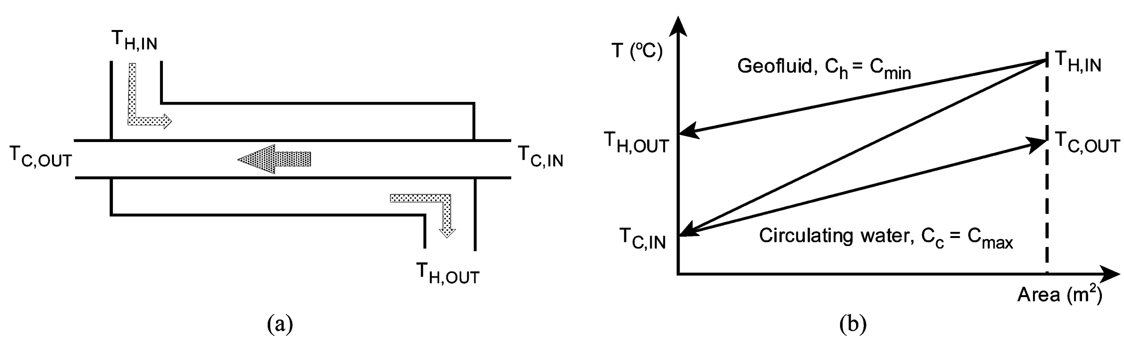


Fig. 1. (a) Schematic and (b) temperature profiles of a counter-flow heat exchanger

3. HEAT AND FLUID FLOW FORMULATION

In the geothermal system, a counter-flow heat exchanger (HE) is used to transfer energy from the geofluid to water. To formulate the net profit of the geothermal system, the heat exchanger investment costs and power used for circulating pumps will be subtracted from the annual cost savings. The following is adopted in the analysis: 1) mass flow rate of the circulating water stream that enters the heat exchanger is larger than the geothermal brine stream, and 2) the heat rejected and dissipated to the environment from the heat exchanger is neglected.

The mass balance can be expressed as,

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

where $\dot{m}$ is the mass flow rate, and the subscripts "*in*" and "*out*" refer to inlet and outlet steams, respectively.

The thermal capacities of the circulating water and geofluid water, C_c and C_h, respectively, are defined as,

$$C_c = \dot{m}_s \cdot C_p \tag{2}$$

$$C_h = \dot{m}_{geo} \cdot C_p \tag{3}$$

where $\dot{m}_s$ and $\dot{m}_{geo}$ are the mass flow rates of the water and geofluid, respectively, and C_p is the specific heat.

The annual net profit of the system is maximized by examining different methods of optimization. The objective function to be maximized for the system is represented as follows,

$$NK = YPT - Ca \tag{4}$$

where NK is the annual net profit of the geothermal system [\$/year]. YPT is the annual savings, which is obtained by subtracting the work of the pump that circulates the geofluid through the heat exchanger, from the annual amount of natural gas required to generate heat for the boiler, instead of heat supplied by the geofluid. Ca, the capital recovery investment, is the annual investment cost for the geothermal heat exchanger. Equation (4) is expressed as the annual cost of fuel, subtracted from the capital recovery investment.

The variable YPT is defined as,

$$YPT = \left(\dot{Q} - BP\right) \cdot H \cdot F \frac{3600}{LHV \cdot \eta_k} \qquad [\$/year] \tag{5}$$

where $\dot{Q}$ is the heat transfer rate from geothermal brine to the water in the heat exchanger, H is the number of operational hours of the plant, F is the fuel cost, LHV is the lower heating value of the fuel, BP is the pump power, and η_k is the efficiency of the boiler. Thus, the annual investment cost of the heat exchanger can be expressed as

$$Ca = I_c \cdot A_{sur} \cdot CFR \tag{6}$$

where I_c is the investment cost of a unit plate heat exchanger area, and A is the area of heat transfer in the heat exchanger. The cost recovery factor, CRF, can be determined from [23]

$$CRF = \frac{(1+i)^n . i}{(1+i)^n - 1} \tag{7}$$

where i and n are the interest rate and heat exchanger lifetime, respectively.

To obtain the fossil-fuel energy consumption, the heat transfer rate, $\dot{Q}$, generated from the geothermal system must be calculated and substituted into Eq. (5). This corresponds to the amount of fuel required to generate the same amount of power. The heat transfer rate from the geofluid to the circulating water in the heat exchanger is

$$\dot{Q} = C_{hot} \cdot (T_{h,in} - T_{h,out}) \tag{8}$$

where $T_{h,in}$ and $T_{h,out}$ represent the inlet and outlet temperatures of the hot water in the system, respectively.

The heat transfer area within the heat exchanger is given by,

$$A_{sur} = \pi DL \tag{9}$$

where D and L are the diameter and length [m] of the pipe in the heat exchanger system. In [5], the heat transfer area was used as a constraint. However in this paper, one of the two independent variables (D or L) will be assumed constant, while the other is used as a constraint. This approach is more realistic and accurate because the optimal results will change for different lengths and diameters, even if the heat transfer area remains constant. For instance, the optimum value for the diameter could approach infinity and the length approaches zero, or vice versa, which satisfies a particular heat transfer area constraint when both are multiplied together, yet the results would be unrealistic and infeasible. In this paper, the diameter of the pipe will be held constant and the length will be used as a constraint. This approach is adopted because the diameter of the pipe is implicitly used in the formulation of the pump power input, so the diameter is needed to calculate the power. The model could be readily modified to formulate the problem with the length of the pipe held constant, while the diameter is used instead as the constraint.

In the heat exchanger, the heat transfer rate of the circulating water is equal to the amount of heat loss by the geofluid stream, i.e.,

$$\dot{Q} = C_{cold} \cdot (T_{c,out} - T_{c,in}) \tag{10}$$

where $T_{c,in}$ and $T_{c,out}$ represent the inlet and outlet circulating water temperatures, respectively.

The total heat transfer rate can be expressed as follows,

$$\dot{Q} = U \cdot A_{sur} \cdot \Delta T_{mean} \tag{11}$$

where ΔT_{mean} is the mean logarithmic temperature of the two streams and it is defined as,

$$\Delta T_{mean} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln((T_{h,in} - T_{c,out})/(T_{h,out} - T_{c,in}))} \tag{12}$$

It is assumed that the mass flow rate of the circulating water is larger than the mass flow rate of the geofluid. Also, the thermal capacity of the circulating water is larger than the thermal capacity of the geofluid, $C_{hot} < C_{cold}$. As a result, the variation of temperatures within the heat exchanger can be depicted by Fig. 1b. Since the thermal capacity of the geofluid is smaller than the circulation fluid, its temperature difference must be larger than the circulating water. As a result, the fluid with a smaller thermal capacity has a larger slope in the temperature profile. Also, it is assumed that the total heat transferred by the geofluid stream, Eq. (3), equals the total heat gain by the circulation water, Eq. (4). Therefore the heat transfer rate can be obtained as follows [24],

$$\dot{Q} = C_{hot} \cdot (T_{h,in} - T_{h,out}) = C_{cold} \cdot (T_{c,out} - T_{c,in}) \tag{13}$$

Solving for $T_{c,in}$ in Eq. (13) and substituting into Eq. (12), allows Eq. (11) to be rewritten as,

$$1 - \frac{C_{hot}}{C_{cold}} = \frac{C_{hot}}{U \cdot A_{sur}} \cdot \ln\left[\frac{(T_{h,in} - T_{c,in}) - (C_{hot}/C_{cold})(T_{h,in} - T_{h,out})}{T_{h,out} - T_{c,in}}\right] \tag{14}$$

where U is the overall heat transfer coefficient of the heat exchanger. Equation (14) can be simplified to,

$$T_{h,out} = \frac{T_{h,in} \cdot (1 - C_{hot}/C_{cold}) - T_{cin} \cdot (1 - \exp(U \cdot A_{sur}(1/C_{hot} - 1/C_{cold})))}{\exp(U \cdot A(1/C_{hot} - 1/C_{cold})) - C_{hot}/C_{cold}} \tag{15}$$

or alternatively,

$$\frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,in}} = \frac{1 - \exp(U \cdot A_{sur}(1/C_{hot} - 1/C_{cold}))}{C_{hot}/C_{cold} - \exp(U \cdot A_{sur}(1/C_{hot} - 1/C_{cold}))} \tag{16}$$

which represents the correlation between the hot and cold water temperatures, and the heat transfer area of the heat exchanger.

3. FORMULATION OF HEAT EXCHANGER EFFECTIVENESS

The heat exchanger efficiency, ε, where $0 \leq \varepsilon \leq 1$, is defined as,

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} \tag{17}$$

The variable $\dot{Q}_{\max}$ is the maximum heat transfer rate from the geofluid to the circulation water, if the heat transfer area was infinite. The outlet temperature of the fluid stream with a smaller heat capacity value is equal to the other fluid's inlet temperature, so that $T_{h,out} = T_{c,in}$. As a result, $\dot{Q}_{\max}$ becomes,

$$\dot{Q}_{\max} = C_{cold} \cdot (T_{h,in} - T_{c,in}) \tag{18}$$

If $C_{hot} > C_{cold}$, the outlet temperature of the geofluid is equal to the inlet temperature of the circulation water,

$$\dot{Q}_{\max} = C_{cold} \cdot (T_{h,in} - T_{c,in}) \tag{19}$$

Alternatively, $\dot{Q}_{\max}$ can be written as,

$$\dot{Q}_{\max} = C_{\min} \cdot (T_{h,in} - T_{c,in}) \tag{20}$$

Since the mass flow rate of the geofluid stream is smaller than the water mass flow, it has a smaller thermal capacity and Eq. (20) can be rewritten as,

$$C_{\min} = C_{hot} \tag{21}$$

Consequently, substituting Eq. (24) into Eq. (23) yields,

$$\dot{Q}_{\max} = C_{hot} \cdot (T_{h,in} - T_{c,in}) \tag{22}$$

Substituting Eq. (22) into Eq. (17), the heat exchanger effectiveness becomes,

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_{hot} \cdot (T_{h,in} - T_{h,out})}{C_{hot} \cdot (T_{h,in} - T_{c,in})} = \frac{C_{cold} \cdot (T_{c,out} - T_{c,\mathrm{in}})}{C_{hot} \cdot (T_{h,in} - T_{c,in})} \tag{23}$$

Simplifying equation (26),

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{(T_{hin} - T_{hout})}{(T_{hin} - T_{cin})} \tag{24}$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{1 - \exp(U \cdot A_{sur}(1/C_{hot} - 1/C_{cold}))}{C_{hot}/C_{cold} - \exp(U \cdot A_{sur}(1/C_{hot} - 1/C_{cold}))} \tag{25}$$

The effectiveness of the heat exchanger is written as a function of the number of transfer units, $NTU = U \cdot A_{sur}/C_{\min}$, which is a dimensionless parameter, and the heat capacity ratio, $R = C_{\min}/C_{\max}$. NTU expresses the size of the heat exchanger. Using these definitions, the following correlation is obtained,

$$U \cdot A_{sur} \cdot \left(\frac{1}{C_{hot}} - \frac{1}{C_{cold}} \right) = NTU \cdot (1 - R) \tag{26}$$

Substituting Eq. (26) into Eq. (25), the effectiveness of the heat exchanger becomes,

$$\varepsilon = \frac{1 - e^{NTU(1-R)}}{R - e^{NTU(1-R)}} \tag{27}$$

Then the heat transfer rate in the geothermal system can be obtained as,

$$\dot{Q} = \dot{m}_{geo} \cdot C_p (T_{h,in} - T_{c,in}) \left(\frac{1 - \psi}{\dot{m}_g/\dot{m}_s - \psi} \right) \tag{28}$$

where

$$\psi = \exp\left(\frac{UA_{sur}}{\dot{m}_g C_p} \left(1 - \frac{\dot{m}_g}{\dot{m}_s} \right) \right) \tag{29}$$

where $T_{h,in}$, and $T_{c,in}$ are the inlet temperatures of the hot and cold water, respectively. Equation (28) is a function of area and mass flow rates of the hot and cold water.

4. COST ANALYSIS OF HEAT EXCHANGERS IN A GEOTHERMAL DISTRICT HEATING SYSTEM

In this section, the cost analysis of the heat exchanger in a geothermal district heating system is performed to determine the maximum annual profit. When conventional fossil fuel energy is used for district heating or cooling, a certain amount of fuel is required to operate the systems. Thus, in a geothermal system, there can be a considerable fuel cost savings, compared to a conventional fossil fuel system. Adding a heat exchanger to a geothermal system, instead of a boiler, provides a financial savings by eliminating the fuel cost. Also, there are no emissions or greenhouse gases in the geothermal system, and it is cleaner than a conventional heating system. At least two pumps are required to circulate the hot and cold water within the heat exchanger. So the annual savings are obtained by subtracting the energy required to drive the pumps, from the annual fuel saving.

More specifically, the annual saving is the annual fuel cost used to produce the same amount of heat energy. The specific fuel consumption, b_e [kg/kWh], of a conventional district heating system can be expressed as follows,

$$b_e = \frac{3600}{LHV \cdot \eta_k} \tag{30}$$

where LHV and η_k are the lower heating value of the fuel and the boiler efficiency, respectively. Therefore, the specific annual fuel consumption, B_e, in units of kg/kW·year, is

$$B_e = H \cdot \frac{3600}{LHV \cdot \eta_k} \tag{31}$$

To acquire a heating rate of $\dot{Q}$, the annual fuel consumption, B, is

$$B = \dot{Q} \cdot H \cdot \frac{3600}{LHV \cdot \eta_k} \tag{32}$$

Since at least two pumps are required, the annual cost savings of the geothermal system is obtained by subtracting the investment cost, including the cost for operating the pumps and cost of equipment, from the annual fuel consumption.

The heat transfer area has two main unknowns: length and diameter of piping. It is necessary to know the diameter of the pipe, in order to calculate the power of the pump. This power input is calculated by finding the head loss and subsequently the power of the pump. By correlating the wall shear stress, τ, for flow conditions in either laminar or turbulent flow, the head loss in the pipe flow can be determined. The shear stress depends on other flow variables and fluid properties as follows,

$$\tau = F(\rho, V, \mu, D, \in) \tag{33a}$$

where ε is the wall-roughness height, and V is the velocity of the flow inside the pipe [m/sec]. Thus the dimensional analysis tells us that,

$$\frac{8\tau}{\rho V^2} = f = F(R_{ed}, \frac{\in}{d}) \tag{33b}$$

The dimensionless parameter f is called the Darcy Friction Factor.

Also the relation of head loss and wall shear stress can be expressed as follows,

$$\Delta z + \frac{\Delta p}{\rho g} = h_f = \frac{2\tau}{\rho g}\frac{\Delta L}{r} \tag{33c}$$

where Δz is the height elevation difference of the inlet and outlet of the pipe, h_f is the head loss, r is the pipe radius, g is gravitational force, and p is the pressure inside the pipe.

Combining Eqs. (33b) and (33c), the desired expression for pipe head loss can be defined as follows [25],

$$h_f = f\frac{LV^2}{D(2g)} \tag{33}$$

The friction factor can be expressed as,

$$f = \frac{64}{\mathrm{Re}_D} \tag{34}$$

where Re_D is the Reynolds number, which depends on the pipe diameter. Also, the power of the pump can be expressed as follows,

$$BP = \frac{\gamma Q h_f}{\eta_p} \tag{35}$$

Expanding equations (33) to (35) in terms of the mass flow rate, diameter, area and friction factor, it can be shown that the power required to drive the shaft for a centrifugal pump (in units of Watts) becomes,

$$BP = \frac{8fL\dot{m}_{geo}^3}{D^5\rho^2\pi^2\eta_P} \tag{36}$$

The power is required to estimate the annual net profit of the system. When a conventional system is used for heating, the revenue of annual fuel consumption is equal to the annual money saved in a geothermal system, over the same period. Hence, the previous result of annual savings (in units of \$/year) of a geothermal system can be rewritten as follows,

$$YPT = B_eF\left(\dot{Q} - BP\right) - Ca \tag{37}$$

where F is the fuel cost [\$/kg]. With the above, an objective function and optimization formulation can be derived to maximize the annual savings.

The objective function is determined by finding Q from Eq. (28) and then substituting it into Eqs. (4) and (5) to find the annual cost savings. This results in the following, which must be maximized

$$NK = HF\frac{3600}{LHV\cdot\eta_k}\left\{\dot{m}_{geo}C_p(T_{hi} - T_{ci})\frac{1-\psi}{\left(\dot{m}_g/\dot{m}_s - \psi\right)} - \frac{8fL\dot{m}_{geo}^3}{D^5\rho^2\pi^2\eta_P}\right\} - I_cA_{sur}CRF \tag{38}$$

subject to the following constraints:

$$\begin{cases} A > 0 \\ 200 > \dot{m}_{geo} > 10 \\ 600 > \dot{m}_s > \dot{m}_{geo} \\ T_{h,in} > T_{c,in} \\ T_{h,out} > T_{c,in} \end{cases} \tag{39}$$

Equations (38) and (39) represent the objective function to be maximized and associated constraints, respectively. In the next section, the above constraints will be imposed, as well as others that represent other problem scenarios.

5. RESULTS AND DISCUSSION

The method of Sequential Quadratic Programming (SQP) is used to solve the non-linear constrained optimization problem for the counter-flow heat exchanger outlined in Equations (38) and (39) [21]. Problem parameters, constraints and other conditions are listed in Tables 1 to 8. The geofluid source temperature is 76 °C (see Table 1). There are many geothermal sources around the world in this temperature range. Different constraints for the circulating mass flow and pipe diameter are examined in this section. The results are compared to experimental data and previously reported data [5], in order to validate the formulations. Despite its relative simplicity, the double-pipe heat exchanger is still a widely used design in a number of geothermal and industrial systems, so it is worthwhile to develop improved algorithms for this configuration. The current study extends the model of Dagdas [5] to include more design variables and realistic design conditions and constraints. These include pump power to circulate brine fluid, its dependence on heat transfer area and pressure losses, amount of fossil fuel required to heat the boiler and resulting energy savings.

In the following results, the non-linear constrained problem will be divided into the following three cases. Firstly, validation of the results will be performed by applying the new generalized formulation to the simplified case examined previously by Dagdas [5], without pumping power and a single constraint involving only the heat transfer area. Then, the formulation will be extended to new cases with more realistic conditions and constraints (i.e., variable length and constant diameter of pipe) and input parameters like the pump input power, which affect the overall economics of the system optimization. In the second case, results will be obtained for three constraints and lower mass flow rates of the hot and cold water, with varying pipe diameters. For the third case, a higher mass flow rate for the hot and cold streams will be examined, along with a larger pipe diameter.

5.1 Single Constraint

Consider the first case where the total heat transfer area is the only constraint on the heat exchanger. For the operating data in Tables 1 and 2, the optimal value of heat transfer area is determined and compared against past data [5]. Results in Table 3 and Fig. 2a illustrate a close agreement between the current predictions and past data by Dagdas [5]. This close agreement provides useful validation of the formulation. In Fig. 2a, the profit reaches a maximum at about a 1,600 m^2 heat transfer area. At lower areas, more energy input is required to achieve the same outlet temperature, while excessive pumping power lowers the profit at high areas of heat transfer.

The predicted results for the outlet temperatures are shown in Fig. 2b. These outlet temperatures were obtained for the optimal area. The geofluid outlet temperature decreases and

Table 1 Problem parameters

	Values
Inlet temperature of geofluid	76 °C
Specific heat capacity of water, C_p	4.186 kJ/kg
Overall heat transfer coefficient, U	5.100 kW/m^{2}K
Boiler efficiency, η_k	0.85
Pump efficiency, η_p	0.85
Cost of heat exchanger per unit, I_c	350 \$/m^2
Lower heating value of fuel, H_u	41900 kJ/kg
Operational hours	3000 hrs
Fuel cost, F	0.50 \$/kg
Heat exchanger life, n	15 years
Interest rate, i	0.10

Table 2 Problem parameters for case 1 (single constraint)

Parameters	Value
Area of heat transfer, A	$1 < A < 2000$ m^2
Inlet temperature of geofluid, $T_{h,in}$	76°C
Inlet temperature of circulating water, $T_{c,in}$	50°C
Mass flow rate of geofluid, m_{geo}	222 kg/s
Mass flow rate of circulating water, m_s	540 kg/s

Table 3 Comparisons between predicted and past data [5]

Parameters	Predicted	Dagdas [5]
Outlet temperature of circulating water, $T_{c,out}$	50	50.08°C
Outlet temperature of geofluid, $T_{h,out}$	61.0	60.58°C
Optimum area of heat transfer, A	1611.696	1612 m^2
Objective function, NK	4,276,779	4,276,385 \$

Table 4 Problem constraints for case 2

Constraint	Minimum	Maximum
Length of the inner pipe, L	1	200 m
Mass flow rate of the geofluid, m_{geo}	1	20 kg/s
Mass flow rate of the circulating water, m_s	1	15 kg/s

Table 5 Additional input values for case 2

Parameters	Values
Inlet temperature of circulation water, T_{cin}	30°C
Inlet temperature of geofluid, T_{hin}	76°C

Table 6 Results with three constraints for different pipe diameters

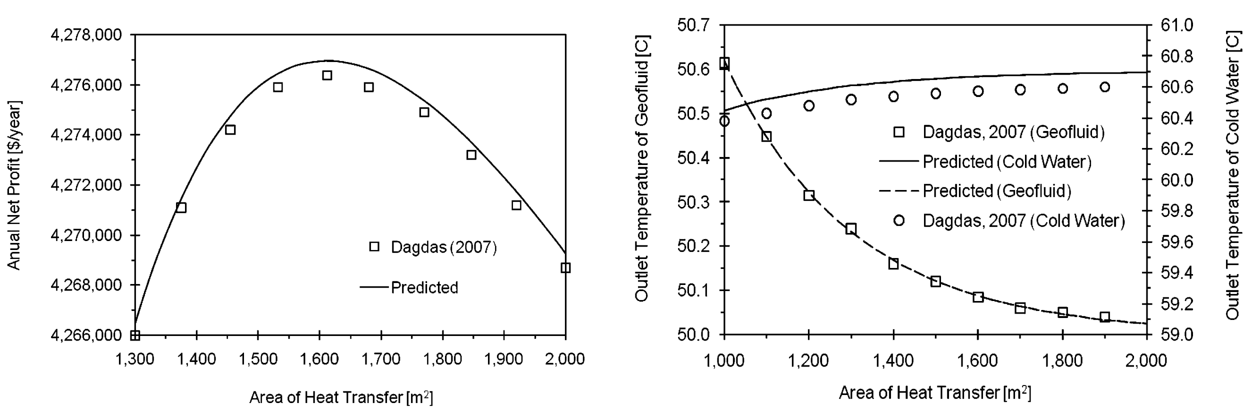
Pipe Diameter [Inches]	Objective Function [$/year]	Geofluid Flow Rate [kg/s]	Water Flow Rate [kg/s]	Length [m]	Cold Outlet Temperature [^{o}C]
1.5	43075	1.87	15	31.86	35.7
2.5	113715	4.89	15	56.75	45.0
3.0	157733	6.75	15	71.25	50.69
5.0	340906	13.96	15	146.84	72.84

Table 7 Constraint values for case 2

Constraints	Minimum	Maximum
Length of the inner pipe, L	1	200 m
Mass flow rate of the geofluid, m_{geo}	1	90 kg/sec
Mass flow rate of the circulating water, m_s	1	100 kg/sec

Table 8 Results for case 3 with a larger mass flow rate and pipe diameter

Pipe Diameter [Inches]	Objective Function [$/year]	Geofluid Flow Rate [kg/s]	Water Flow Rate [kg/s]	Length [m]	Cold Outlet Temperature [°C]
8	1095494	46.68	100	195	53.86



(a) (b)

Fig. 2. (a) Annual net profit (Dagdas 2007) and (b) outlet temperatures of the geofluid and cold water

the water outlet temperature rises, when the area increases because of higher overall heat exchange between the fluid streams. The outlet temperature of cold water increases by about 10 °C. Although the system is operating efficiently, it is not using the maximum energy availability of the geothermal fluid. As a result, there are limited practical applications for this system, due to the relatively small difference between inlet and outlet temperatures of the water. By using a longer pipe or higher flow rate, the outlet temperature can be increased significantly, which will make the system more efficient.

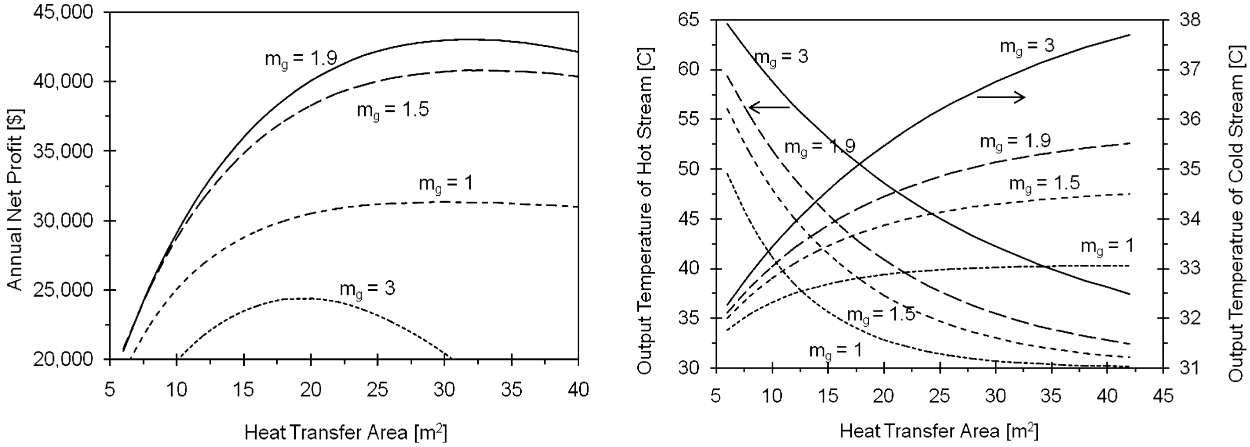
5.2 Three Constraints with Low Mass Flow Rates

This first case provides useful validation against past data [5], but in practice it is impractical to only use the total heat transfer area as a single constraint, since the pipe diameter can be very large or unrealistically small to achieve the same constant area, by adjusting the pipe length accordingly. Thus, the second case considers a fixed pipe diameter (which can be altered separately), with three constraints and a lower mass flow rate. Three constraints involving the pipe length and mass flow rates of the hot and cold water will be considered to maximize the annual net profit. Different pipe diameters and an inlet temperature of 30 °C are used for the circulating water. The problem constraints are listed in Table 4. Initially, 20 assumed values for the design variable were used within the allowable range of the constraints. The constant values for the input temperatures of the hot and cold water are presented in Table 5. Table 6 presents the optimum results for different pipe diameters. In all cases, the results satisfied the Karush-Kuhn-Tucker conditions (KKT), and gradient and Hessian approximations [21]. As a result, an optimum was achieved for the converged results. The optimum value for the flow rate of water reached its maximum allowable value. Only the maximum value of the mass flow rate of circulating water was plotted.

Figures 3 and 4 illustrate the annual net profit vs. heat transfer length for pipe diameters of 1.5in and 2.5in, respectively. From Fig. 3a, the optimum values of 1.9kg/s and 31.86m for the mass flow rate of the hot fluid and length, respectively, yield the maximum annual net profit. At the optimum points, Figs. 3b and 4b show the corresponding outlet temperature profiles for the hot and cold streams. Figure 5a shows the annual net profit vs. heat transfer length in a 3in pipe. The annual net profit for different mass flow rates increases rapidly, until it reaches a maximum value above the heat transfer length of 60m, after which is starts decreasing. For a mass flow rate of 6.7kg/s and pipe length of 71.25m, the annual net profit reaches a maximum value. Figure 5b indicates that a larger mass flow rate yields a higher outlet temperature of the circulating water. The annual net profit for a mass flow rate of 8kg/s is smaller than the optimal mass flow rate of 6.7kg/s. Figure 6 shows a contour plot of the cost objective function vs. heat transfer length and mass flow rate, respectively. The graph depicts the same trends that were observed previously for the objective function. From additional results that were obtained for a pipe diameter of 5in, a higher mass flow rate leads to a higher outlet temperature and lower annual net profit.

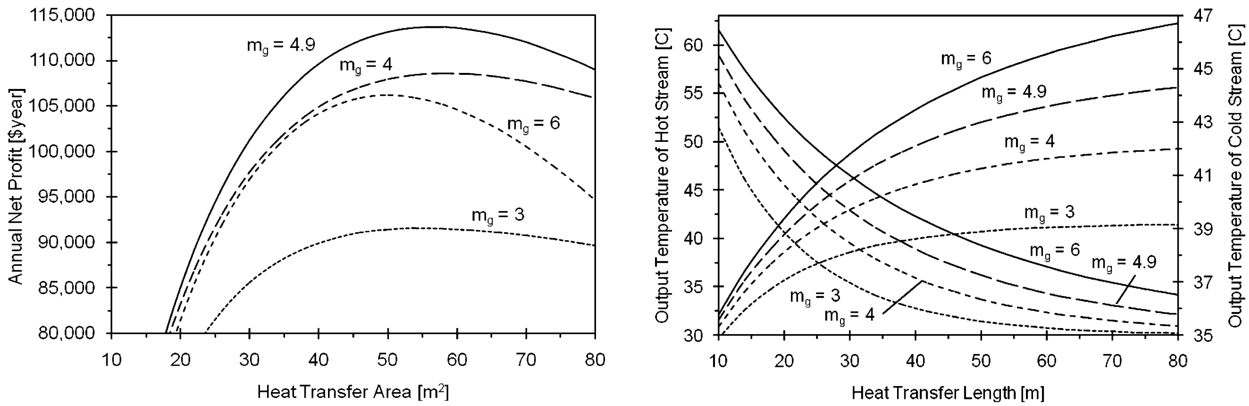
5.3 High Mass Flow Rates with a Given Pipe Diameter

Finally, the third case examines a larger mass flow rate and pipe diameter. Different maximum values for the mass flow rates and a pipe diameter of 8in were considered. The model data for inlet temperatures of hot and cold water is given in Table 5. Since a higher mass flow rate is considered, a larger annual net profit and more energy savings are achieved. Table 7 shows the constraint limitations, while results from the simulations are presented in Table 8. This system can be used in many practical applications, since the outlet cold temperature is



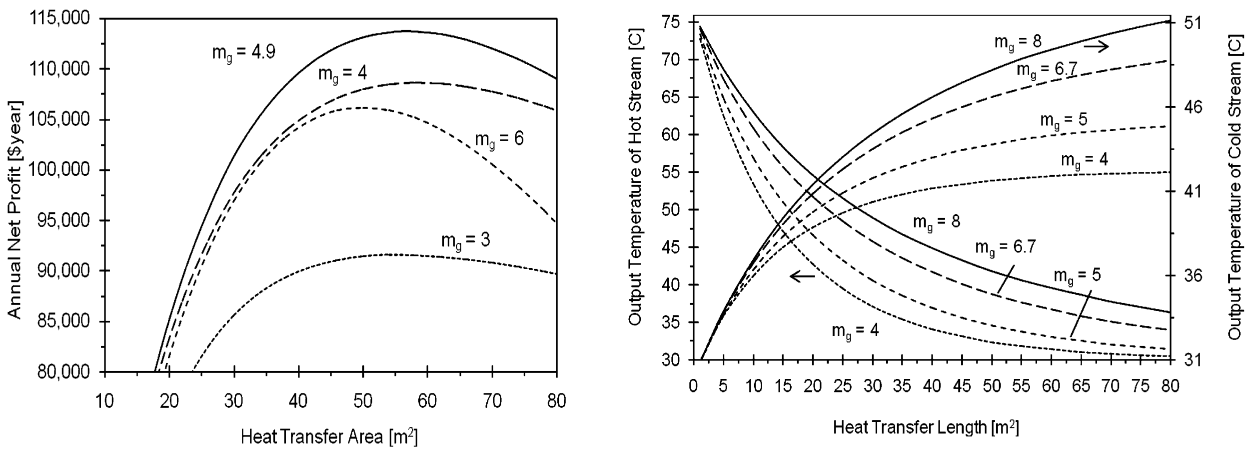
(a) (b)

Fig. 3. (a) Annual profit and (b) outlet temperatures for a 1.5 inch pipe diameter



(a) (b)

Fig. 4. (a) Annual profit and (b) outlet temperatures for a 2.5 inch pipe diameter



(a) (b)

Fig. 5. (a) Annual profit and (b) outlet temperatures for a 3 inch pipe diameter

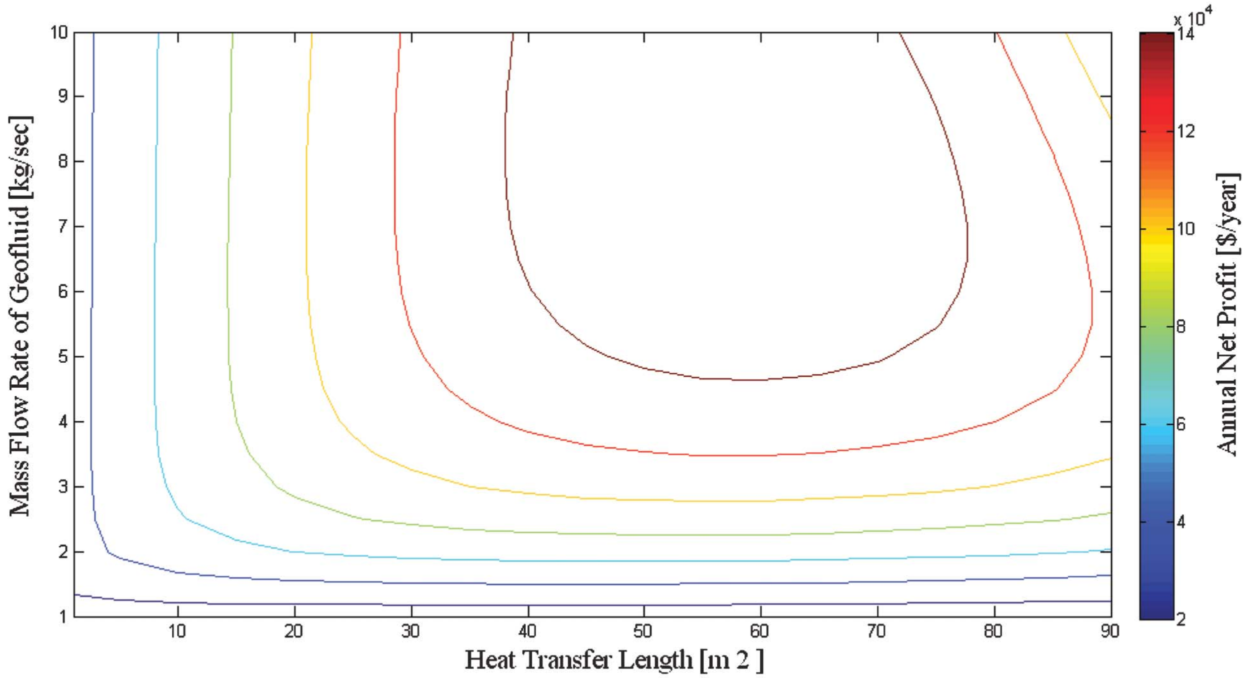


Fig. 6: Contours of the net profit at varying mass flow rates for a 3 inch pipe diameter

increased by around 20 °C. The outlet temperature can be further increased by lengthening the pipe or increasing the geofluid mass flow rates.

Although a number of past studies have been conducted for double-pipe heat exchanger optimization, this paper has presented an improved formulation that includes and examines a wider range of design parameters, such as flow rates, geometrical configurations, pump power input and cost analysis. This can have useful practical applicability to geothermal systems, space and district heating, process heating in chemical industries and other applications, where the double-pipe heat exchanger is commonly used.

6. CONCLUSIONS

Optimization studies were performed to maximize the net annual operating profit of a heat exchanger in a geothermal district heating system. Predicted results were compared successfully against past data for validation of the formulation. It was shown that the geofluid outlet temperature decreases and the cold water outlet temperature rises, when the area increases because of higher overall heat exchange between the fluid streams. The model formulation and system constraints provide a more realistic design tool for optimizing the system performance of heat exchangers. Furthermore, the results have provided useful new data for the improvement of geothermal energy systems.

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NOMENCLATURE

A	heat transfer area (m^2)
b_e	specific fuel consumption (kg/kWh)
B_e	annual fuel consumption (kg/kWyr)
BP	pump power (W)
C_p	specific heat (W/°C)
C	thermal capacity ($\dot{m}C_p$)
Ca	capital recovery investment ($/year)
CRF	cost recovery factor
D	diameter (m)
f	Darcy friction factor
F	fuel cost ($)
H	number of operational hours
h_f	head loss
i	interest rate
I_c	investment cost ($)
L	length (m)
LHV	lower heating value (kJ)
$\dot{m}$	mass flow rate (kg/s)
n	heat exchanger lifetime in years
NK	annual net profit ($/year)
NTU	number of transfer units
P	pressure (Pa)
R	heat capacity ratio
$\dot{Q}$	heat transfer rate (W)
r	pipe radius (m)
Re	Reynolds number
T	temperature (°C)
U	overall heat transfer coefficient (Wm^2 °C)
V	fluid velocity (m/s)
YPT	annual savings ($/year)
Δz	height elevation difference (m)

Greek

ε	heat exchanger effectiveness, or wall roughness (mm)
γ	specific gravity (kg/m^2s^2)
η_k	boiler efficiency
η_p	pumpr efficiency
μ	dynamic viscosity (kg/ms)
ρ	density (kg/m^3)

Subscripts

c	cold
D	diameter
g	geofluid
h	hot

in	inlet
max	maximum value
mean	mean value
out	outlet
s	circulating water