USE OF VERY LOW TEMPERATURE GEOTHERMAL WATER IN RADIATOR HEATING SYSTEMS

Thorbjorn Karlsson,
Faculty of Engineering, University of Iceland,
Hjarðarhagi 6, IS-107 Reykjavik, Iceland
and
Arni Ragnarsson,
Geothermal Division, Orkustofnun (National Energy Authority),
Grensásvegur 9, IS-108 Reykjavik, Iceland

Key words - geothermal energy, direct use, district heating, heat exchangers, utilization economy

***ABSTRACT:** This paper deals with the utilization of geothermal water of very low temperature. Experience gained in Iceland over 50 years of utilizing water in the 55 - 80°C temperature range in radiators with a focus on the low temperatures is described. The theory of performance of hot water radiators is reviewed and the method of evaluating the annual hot water requirements for space heating purposes is outlined, both in direct throughflow district heating systems as well as more complicated district heating systems with heat exchangers at users' stations. The question of sizing the radiators and individual thermostatic controls is discussed as well as the optimization of temperature programs for radiators with and without heat exchanger substations. It is shown that by properly sizing the radiator and equipping it with the required controls one can utilize water directly even at inlet temperatures as low as 55°C. By operating the system at these low temperatures and with a large temperature drop, heat losses from the system and pumping requirements are reduced. Measured values of supply and return temperatures in a sample of houses in various geothermal district heating systems in Iceland are presented which confirm that large temperature drops in radiators can be achieved in spite of low inlet temperatures.*

1 INTRODUCTION

As a general rule the geothermal fluid pumped from drilled wells in geothermal areas around the world is unsuitable for direct use or consumption due to unfavourable fluid chemistry. Exceptions from this are found in geothermal district heating (DH) systems in Iceland, which may be roughly divided into the following four categories:

1. ***Direct throughflow systems.*** In most geothermal DH systems in Iceland the geothermal fluid is pure enough and not too hot (≤ 80°C) to allow its pumping directly from the boreholes through the customers' heating systems. The returning water is disposed of through the local wastewater system.
2. ***Mixed circulation and throughflow systems.*** In some areas the geothermal fluid is too hot (≥ 90°C) to be safe for direct use, yet pure enough for direct use. DH systems using this water are built partly with a single pipe throughflow distribution system and partly a double supply and return pipe system. The cool return water, mixed with the hot geothermal fluid, reduces its temperature to a safe level (≈ 80°C) before it is pumped to the users.
3. ***Residential heat exchanger systems.*** The geothermal fluid is unsuitable for direct use in residential heating systems. In such cases a heat exchanger may be installed at each user which then serves as a boiler for a closed loop house heating system.
4. ***Closed circulation DH systems.*** The geothermal fluid, unsuitable for direct use, is piped to a central heat exchanger station from which the heating fluid is circulated to the users through a closed loop distribution system. Geothermal systems of this type are an exception in Iceland but several electrically heated systems are in operation outside geothermal areas.

When geothermal energy is being utilized the main objective is to make the maximum possible use of the available energy of the fluid. In simple terms this means that the temperature of the geothermal fluid is brought down as far as possible before it is returned to its source or wasted. In Iceland, where most geothermal DH is by category 1, this has led to radiator design requirements where the temperature drop through radiators is much larger than is customary elsewhere. For example, with geothermal fluid supply temperature of 80°C a common design is a return temperature of 40°C at outdoor design temperature of -15°C, a socalled 80/40/-15 radiator design. This results in considerably larger radiators than in fuel fired European DH systems where a 90/70/-15 radiator design is common.

The paper describes a study of geothermal DH systems belonging to categories 1 and 3. The ***direct throughflow system*** is studied for various fixed values of the geothermal fluid temperature, ranging from 90°C down to ***60°C*** with an assumed radiator design which varies with the fixed supply temperature. The annual water flow required for heating is determined as a function of the supply water temperature and of the building design heating load for typical weather conditions as they are observed in Iceland.

The study of the performance of ***residential heat exchanger systems*** is more complicated than that of the direct throughflow system. The analysis is based on the same weather conditions and radiator design as for the direct throughflow systems and limited to small brazed plate heat exchangers commonly used in Iceland.

2 REVIEW OF RADIATOR THEORY

Space heating in Iceland is traditionally done by hot water radiators installed in the space to be heated. The heat transferred from the radiator water to the surrounding space is given by the equation

$$Q = A k_r \Delta T_m, \tag{1}$$

where Q = heat transferred, [W], A = radiator surface area, [m²], k_r= radiator heat transfer coefficient, [W/(m²°C)], and ΔT_m = logarithmic mean difference between radiator water temperature and surrounding space temperatures, ["C]. This difference (LMTD) is given by the equation

$$\Delta T_m = \frac{T_s - T_r}{\ln \dfrac{T_s - T_i}{T_r - T_i}}, \tag{2}$$

where T_s = radiator supply water temperature, T_r = radiator return water temperature, and T_i = surrounding air temperature. From experience it is found that the LMTD value varies with the radiator heat load according to an empirical equation on the form

$$\frac{Q}{Q_o} = \left(\frac{\Delta T_m}{\Delta T_{mo}}\right)^n,$$

where $()_o$ indicates design conditions, in Iceland usually based on an outdoor air temperature of -15°C. and the exponent n is a weak function of the radiator size and shape, ranging from 1.25 to 1.35. The German standard DIN 4703 (1988) gives n = 1.3. Combining Eqs. (1) through (3) gives the relationship between the radiator heat transfer coefficient and the heat load:

$$\frac{k_r}{k_{ro}} = \frac{Q}{Q_o}\frac{\Delta T_{mo}}{\Delta T_m} = \left(\frac{\Delta T_m}{\Delta T_{mo}}\right)^{n-1} \tag{4}$$

Eq. (2) shows that a lower value of T_r results in a lower value of the LMTD for constant values of T_s and T_i. Consequently as seen from Eq. (1) if ΔT_m is reduced for a given value of the required heat Q, the radiator size A must be increased. In a throughflow system with a constant geothermal supply water temperature T_{gs} equal to T_s, the necessary radiator size is determined from Eq. (1) at design conditions. In this paper the radiator design is assumed to follow the equation

$$T_s/T_r/T_{ao} = T_{gs}/0.5T_{gs}/\text{-}15, \tag{5}$$

where T_{ao} = design outdoor air temperature (-15°C). For a geothermal supply water temperature of 80°C this formula gives the commonly used 80/40/-15 radiator system previously mentioned.

3 WEATHER CONDITIONS

The geothermal water quantity required to supply the needed heat energy for space heating depends on the weather pattern in the area under study. Annual geothermal water requirements are evaluated from the annual frequency distribution of the daily mean temperature in Reykjavik corrected for the effects of winds and sun. Data from the Reykjavik Municipal DH Service indicate better correlations between geothermal water use and corrected temperatures than measured values (Jónsson and Jónsson, in press). The temperature so adjusted is called the equivalent daily mean temperature.

The necessary weather data for this purpose were furnished by the Icelandic Meteorological Office as presented in Fig. 1. It shows the annual frequency distribution of the equivalent daily mean temperatures averaged over the period from 1981 through 1990.

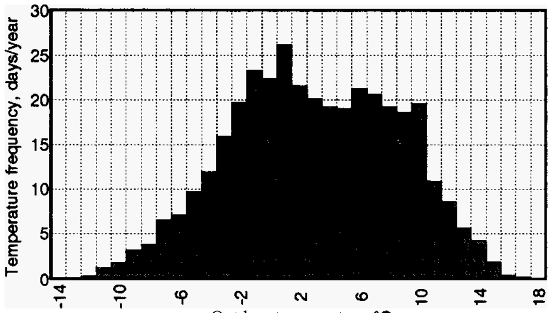

Fig. 1 Frequency distribution of equivalent daily mean outdoor temperature (corrected for the influence of winds and sun) in Reykjavik, Iceland, averaged over the period 1981 - 1990.

4 DIRECT THROUGHFLOW SYSTEMS

The direct throughflow DH system is very simple in construction as shown schematically in Fig. **2.** The geothermal water is pumped directly to the user entering his radiators at a temperature of T_s, returning at a temperature of T_r, and subsequently disposed of through the local wastewater system.

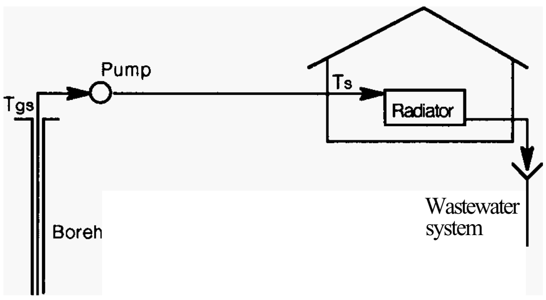

Fig. 2 Direct throughflow geothermal DH system - simple line diagram.

Analysis of the direct throughflow geothermal DH system is based on the assumption that the supply water temperature entering the radiators equals the geothermal fluid temperature: $T_s = T_{gs}$. The radiator design is given by Eq. *(5)* and assuming a design heat load Q_o, given at the outdoor temperature T_{ao}, the heat load at any other outdoor temperature T_a is given by the equation

$$Q = Q_o \frac{T_a - T_i}{T_{ao} - T_i}. \tag{6}$$

The LMTD for this load is found from Eq. (3):

$$\Delta T_m = \Delta T_{mo} \left(\frac{Q}{Q_o} \right)^{1/n}, \tag{7}$$

from which the return temperature is found from Eq. **(2).** With both supply and return temperatures known the geothermal water flow is then determined from the equation

$$m = \frac{Q}{c_p (T_s - T_r)}, \tag{8}$$

where c_p = specific heat of water, assumed constant over the temperature interval being considered. By going systematically through all temperature values shown in Fig. 1 and adding together the total water flow used for heating at each value the annual flow of geothermal water needed is obtained. This is done for all supply temperature values to be studied.

5 RESIDENTIAL HEAT EXCHANGER SYSTEMS

Fig. 3 shows a simple line drawing of a residential system connected with a geothermal DH distribution system through a heat exchanger. The figure shows that the geothermal water enters the heat exchanger at a temperature T_{gs}. The returning fluid at a temperature T_{gr} is disposed of through the local wastewater system as before.

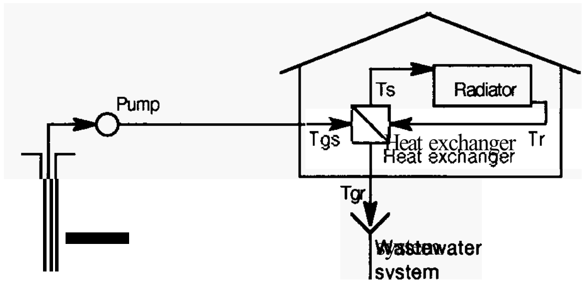

Fig. 3 Residential heat exchanger type geothermal DH system - simple line diagram.

Of the many types of heat exchangers available the brazed plate type is the most widely used heat exchanger for residential systems in Icelandic geothermal DH systems. The main reasons are that (a) the plate heat exchanger requires less space in terms of m³/kW than other types, (b) the high turbulence in plate exchangers, even at Reynold's numbers as low as 10 to **400,** results in very effective heat transfer characteristics and inhibits fouling of heat transfer surfaces, and (c) the brazed type is of a much lighter construction and easier to handle than other types.

Fig. **4** shows the flow scheme for a plate heat exchanger of the type commonly used in DH house systems in Iceland. The flow pattern for each fluid, the primary or geothermal fluid and the secondary or radiator fluid, is of the socalled 4/4 pass type where each fluid passes four times through the heat exchanger. The flow arrangement is largely counterflow where the flow direction in one channel runs in opposite direction to that of the other fluid in the two adjacent channels. Performance calculations of these heat exchangers are based on counteflow heat transfer which for small units with relatively few plates calls for correction factors to the calculated counterflow LMTD values (Raju and Bansal, 1981). As the number of plates is increased the counterflow assumption is closer to reality resulting in the LMTD correction factor approaching unity .

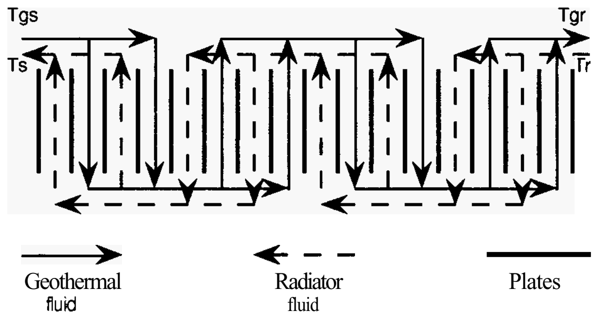

Fig. 4 Flow pattern through a 4/4 type plate heat exchanger system.

For a heat exchanger in which each fluid runs through in m passes (m/m type) and each stream (geothermal and radiator fluids) runs through n channels in each pass the total number of plates, p, making up the heat exchanger (including end plates) is

$$p = 2mn + 1. \tag{9}$$

The heat exchanger in Fig. **4** has $m = 4$ and $n = 2$, making $p = 17$ plates.

The analysis of the performance of residential heat exchanger systems calls for the evaluation of optimum operating conditions for such systems. The optimum is defined as that value of the radiator supply temperature which gives the lowest possible temperature of the returning primary (geothermal) fluid.

Available information on design and performance of plate heat exchangers is rather limited. Methods to evaluate the film coefficients in plate heat exchangers consist of using equations developed for circular tubes using an equivalent diameter for the noncircular passage. The formula used **is** the socalled Sieder-Tate equation valid for turbulent flow and found in any standard heat transfer text:

$$\mathrm{Nu} = K \mathrm{Re}^x \mathrm{Pr}^y (\mu/\mu_w)^z \tag{10}$$

where $\mathrm{Nu} = hD_e/\lambda$ = Nusselt number, [-], $\mathrm{Re} = D_e G/\mu$ = Reynolds number, [-], $\mathrm{Pr} = c_p\mu/\lambda$ = Prandtl number, [-], h = film coefficient of heat transfer at plate surface, [W/(m²°C)], λ = thermal conductivity of fluid, [W/(m°C)], K = constant, G = mass velocity of fluid in channel, [kg/(m²s)], μ = dynamic viscosity of fluid, [Pa s], μ_w = dynamic viscosity of fluid at plate wall, [Pa s], c_p = specific heat of fluid, [J/ (kg°C)], D_e = equivalent diameter of channel, [m], (x,y,z) = exponents, [-], taken as constants = (0.667,0.333,0.14).

The heat transfer coefficients h_g and h_r (subscripts $()_g$ and $()_r$ referring to the geothermal water (primary) and radiator water (secondary) sides, respectively) are evaluated from the Sieder-Tate equation based on average flow temperature on respective sides. The overall heat transfer coefficient is then given by the equation

$$\frac{1}{U} = \frac{1}{h_g} + \frac{1}{h_r} + R_p\ , \tag{11}$$

where U = overall heat transfer coefficient, [W/(m²°C)], R_p = plate heat resistance including fouling $= R_f + t/\lambda_p$, [m²°C/W], R_f = heat resistance due to fouling, [m²°C/W], t = plate thickness, [m], λ_p = thermal conductivity of plate material, [W/(m°C)].

With the value of U determined, the heat transferred from one fluid to the other is given by the equation

$$Q = UA\Delta T_{he}, \tag{12}$$

where A = total heat transfer area of the heat exchanger = $A_p(p-2)$, [m²], A_p = heat transfer area of each plate, [m²], ΔT_{he} = heat exchanger LMTD, [°C], given by the equation

$$\Delta T_{he} = \frac{(T_{gs}-T_s)-(T_{gr}-T_r)}{\ln\dfrac{T_{gs}-T_s}{T_{gr}-T_r}} \tag{13}$$

where T_{gs} = geothermal fluid temperature entering heat exchanger, [°C], T_{gr} = geothermal fluid temperature leaving heat exchanger, [°C], T_s = radiator fluid temperature leaving heat exchanger, [°C], T_r =radiator fluid temperature entering heat exchanger, [°C].

Two additional equations relate the heat transferred and the properties of the two fluids:

$$Q = \dot{m}_g c_{pg}(T_{gs}-T_{gr}), \tag{14a}$$

$$Q = \dot{m}_r c_{pr}(T_s - T_r), \tag{14b}$$

where $\dot{m}_g$ = mass flow of geothermal fluid, [kg/s], $\dot{m}_r$ = mass flow of radiator fluid, [kg/s], c_{pg} = specific heat of geothermal fluid [J/ (kg°C)] evaluated at the geothermal fluid mean temperature in the heat exchanger, $T_{gm} = (T_{gs}+T_{gr})/2$, c_{pr} = specific heat of radiator fluid, [J/ (kg°C)], evaluated at the radiator fluid mean temperature in the heat exchanger, $T_{rm} = (T_s + T_r)/2$.

Eqs. (10) through (14) together with the previously discussed radiator equations provide the necessary tools to solve the heat exchanger problem as described below.

Suppliers in Iceland specify a nominal output [W] of plate heat exchangers for geothermal use at given primary and secondary fluid temperature differentials, usually specified as follows:

$$T_{gs}/T_{gr} - T_r/T_s = 80/40 - 35/75. \tag{15}$$

From this information it is relatively straightforward to evaluate the constant K of the Sieder-Tate Eq. (10) based on the average flow temperatures of T_{gm} = 60°C on the primary side and T_{rm} = 55°C on the secondary side and the nominal heat exchanger output Q_{he}. The same constant is assumed to apply to both sides.

Now the performance of the heat exchanger connecting the geothermal DH system to a radiator house heating system is evaluated. It is assumed that the radiators are designed for a specified performance at design outdoor conditions as given by Eq. (5). Available weather data as described previously provide a basis for the evaluation of the annual geothermal fluid consumption for heating. The procedure, outlined by Karlsson (1992), is as follows:

1. The design heat load for the building in question, Q_o, is given. For an outdoor temperature T_a the heat load is found from Eq. (6). The returning radiator water temperature T_r is then found for an assumed value of the radiator supply temperature T_s. With the water specific heat c_{pr} evaluated at the average water temperature the necessary radiator flow $\dot{m}_r$ is obtained. The radiator side is now fully defined and the average plate temperatures on both sides, T_{gp} and T_{rp}, are easily found by use of Eq. (10).
2. Assuming a given geothermal water supply temperature T_{gs}, Eq. (10) is applied to the primary (geothermal fluid) side of the heat exchanger together with Eqs. (11) and (12) and computed values from section 1 above. Solution of the resulting equation provides the value of the leaving temperature, T_{gr}. Thereby the necessary geothermal fluid flow is easily found.
3. Running through a series of values for the radiator water supply temperature shows that the value of T_{gr} reaches a minimum value at a given value of T_s which represents the optimum value. Going this way through all outdoor temperatures occurring throughout the year it is a simple matter to generate the optimum operating conditions for the heat exchanger under all possible conditions.

6 PRESENTATION OF NUMERICAL RESULTS

6.1 Optimum Radiator Supply Temperature

In order to determine the operating characteristics of heat exchangers we consider four sizes of 4/4 brazed plate heat exchangers manufactured in Sweden by Cetetherm AB, all of type Cetepac 310, with 33, 41, 57 and 81 plates respectively. Rated heat output and other numerical parameters are presented in Table 1. The fouling factor chosen is for brackish water as reported by Alfa-Laval (1969).

Table 1 Numerical parameters and other specifications used for numerical analysis of heat exchangers, manufactured by Cetetherm AB, Type Cetepac 310.

General data:	Plate material			Stainless steel type AISI 316
	Plate thickness, t			0.35 mm
	Fouling factor, R_f			$75\cdot10^{-6}$ m²°C/W
	Plate heat conductivity, λ_p			15.0 W/(m°C)
	Space between plates, b			2.0 mm
	Width of plates, w			100.0 mm
	Heat transfer area/plate, A_p			0.025 m²
No. of plates, p	33	41	57	81
Nominal output, Q_{he} kW	9.0	13.0	21.5	35.0

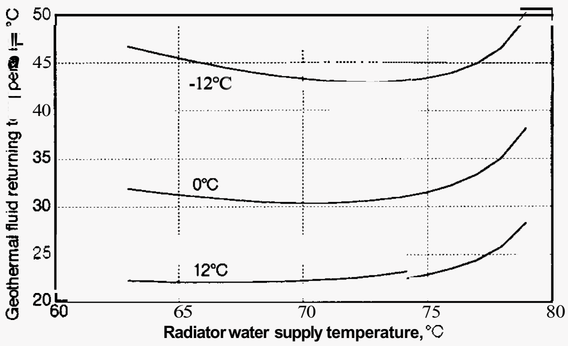

Fig. 5 Returning geothermal fluid temperature as a function of radiator water supply temperature at various outdoor temperatures. Geothermal water supply temperature = 80°C.

Figs. 5 - 7 show results for a heat exchanger with 41 plates operating in a building with a design heat load of 13 kW, equal to the rated output, and operating at a constant supply temperature of 80°C. Fig. 5 shows for a fixed outdoor temperature how the geothermal fluid return temperature varies as the radiator water supply water temperature is changed. As the supply temperature is increased from 63°C the geothermal return temperature is reduced to a minimum value from where it increases again as the supply temperature increases further. The figure also shows that the higher the outdoor temperature is, the lower will be the supply temperature at which the lowest geothermal temperature value occurs. Since the low point means the maximum extraction of the geothermal fluid energy this shows that optimal operating conditions call for variations in radiator water supply temperature with outdoor temperature.

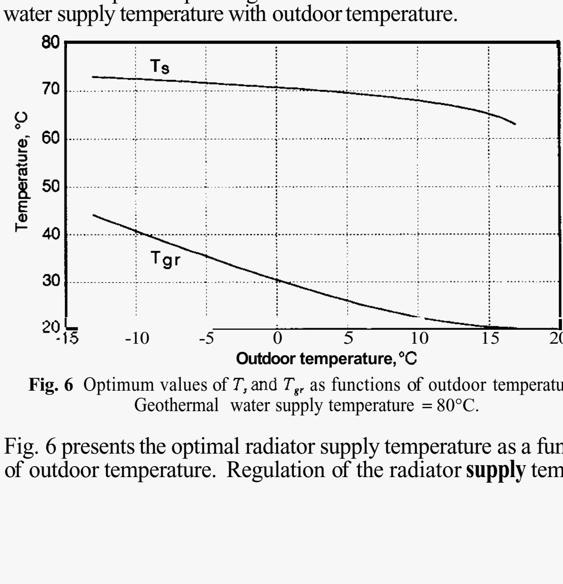

Fig. 6 Optimum values of T_s and T_{gr} as functions of outdoor temperature. Geothermal water supply temperature = 80°C.

Fig. 6 presents the optimal radiator supply temperature as a function of outdoor temperature. Regulation of the radiator supply temperature according to this curve will ensure the optimal operating conditions for the heat exchanger. The figure also presents the corresponding variation of the geothermal water returning temperature.

It is seen from Fig. **6** that the optimum radiator supply temperature shows relatively little variation throughout the range of outdoor temperatures experienced in Iceland. The question then arises how much geothermal water is required if the radiator supply water temperature is set at a fixed value throughout the year. The results are plotted in Fig. **7** where the annual geothermal fluid use in metric tons/year is plotted as a function of a constant value of radiator supply temperature ranging from **63** through **79 °C.** The curve has a minimum value of **953.4** metric tons/year of geothermal fluid required at $T_s = 70.6$°C. Compare this with the annual fluid required when operating under optimum conditions as shown by Fig. 6, which turns out to **be 952.6** metric tons/year. In practical terms the difference is nil.

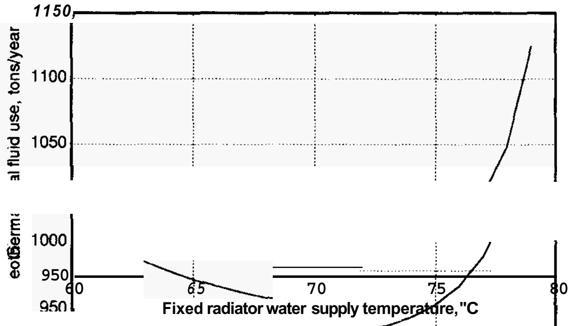

Fig. 7 Annual geothermal fluid use at fixed radiator water supply temperature. Geothermal water supply temperature = 80°C.

6.2 Varying Heat Load and Heat Exchanger Size

The above results apply to a heat exchanger of a given size operating in a building of a given design heat load. Strength limitations of the radiator systems limit the allowable pressure drop across the heat exchanger to 30 kPa which puts a limit on the heat load for a given heat exchanger. On this basis the maximum output for each of the four heat exchanger sizes was evaluated with the results shown in Fig. **8** as a function of the geothermal supply temperature. For all cases the analysis was carried out assuming a constant value of the radiator water supply temperature, chosen so as to keep the annual geothermal flow at a minimum.

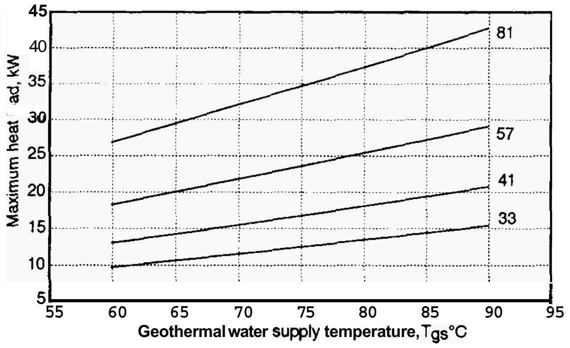

Figure 8 Maximum heat exchanger output at varying geothermal water supply temperature. Numbers by each curve indicate number of plates.

On basis of these results an analysis of the annual geothermal water requirements of heat exchanger systems was carried out for the four

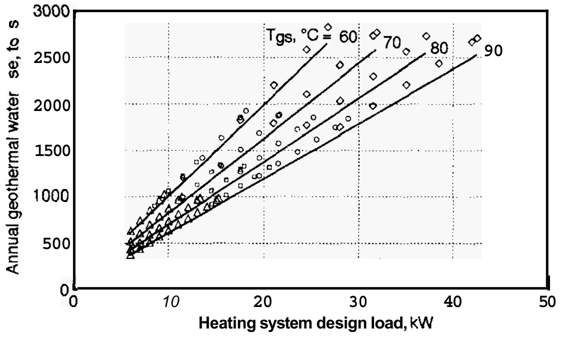

Fig. 9 Annual geothermal water requirements for heat exchanger systems at varying heating system design loads. The dots show heat exchanger values whereas the solid lines indicate the geothermal water required in a direct throughflow system.

sizes considered with a fixed radiator supply temperature selected to give minimum annual geothermal water use. Fig. 9 shows the calculated values of the annual water flow needed for heating buildings of varying sizes evaluated in accordance with the methods discussed. The radiator sizes are designed by the rules previously described [see Eq. (5)] based on the geothermal supply temperature. Inspection of the results reveals that the four sizes of heat exchangers studied cover very well the range of design heat loads from *6* kW up to 25 to 45 kW depending on the geothermal temperature. The solid lines drawn in the figure indicate the direct throughflow water requirements for the respective geothermal temperatures. It **is** seen that the heat exchanger systems require from **7** *to* **9%** more geothermal water than do the direct throughflow systems.

7 RESULTS OF MEASUREMENTS

7.1 Measurements in Direct Throughflow Systems

Most of the geothermal DH systems in Iceland are direct throughflow systems. The most common tariff system is based on three components: a) connection charge, b) fixed annual charge, and c) variable charge based on the quantity of water used. Energy metering is not suited for direct throughflow systems as it does not encourage customers to maximize the heat energy extracted from the water before it is returned. **A** tariff system based on maximum flow restriction (MFR) was widely used in earlier days, but today this method is mostly restricted to rural areas. In that system the user chooses the maximum flow rate equal to the expected demand during the coldest winter period, with the restriction achieved by a differential pressure control valve.

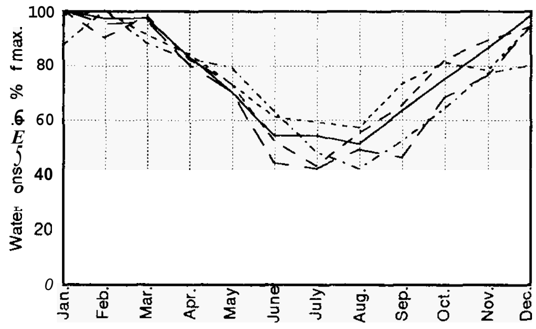

Fig.10 Distribution of relative monthly water consumption for five **VFM** systems.

Water consumption and energy utilization have been measured in a sample of houses in 14 DH systems in Iceland (Ragnarsson, 1991 a,b). The measurements were performed in 20 - 50 houses in each system or a total of 340 houses. Five of the systems used volumetric flow meters (VFM) and nine used MFR at the time of the investigation, but today most of them have been converted to VFM. The supply and return temperatures were measured five times in each house over a period of one year and a volumetric flow meter reading made at the same time.

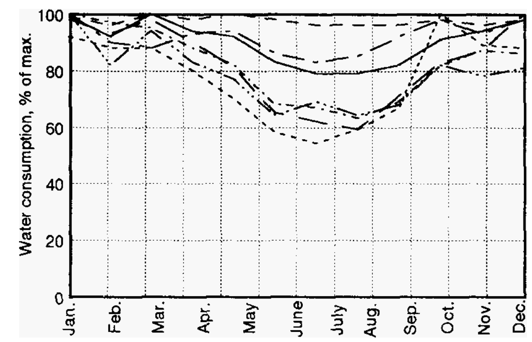

Fig.11 Distribution of relative monthly water consumption for seven MFR systems.

Fig. 10 shows the relative distribution of the total water consumption during one year for the VFM systems, and Fig. 11 shows the same for the MFR systems. **As** can be seen from Fig. 10 the water consumption is reduced considerably during the summer months reflecting fairly well the outdoor temperature variations. Fig. 11 shows that for the MFR systems the seasonal distribution varies considerably from one system to another. In general seasonal variations have only a small influence on the water consumption in these systems compared with that of the VFM systems.

From the water consumption distribution for VFM systems which all show similar behaviour (Fig. 10) an equation has been derived correlating the water use in a given period of the year to an estimated annual use. If each day of the year is numbered from 1 to 365 and the water consumption from day No. d_1 to day No. d_2 is known, the annual water consumption is estimated as

$$\text{Annual water flow [m}^3\text{/year]} = V/C, \tag{16}$$

where V = volume of water consumed during the period from d_1 to d_2 [m³] and C is a function of the two dates given by the expression

$$C = 0.00274(d_2 - d_1) + 0.0515[\cos(0.0172 d_2 + 4.25) - \cos(0.0172 d_1 + 4.25)]. \tag{17}$$

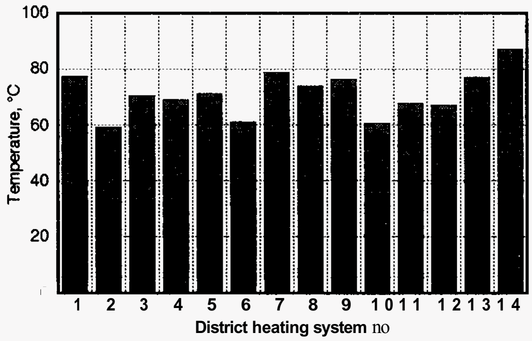

Fig. 12 Average values of supply (total column heights) and return (dark shaded columns) temperatures and utilized temperature drop (light shade) of the 14 DH systems considered. Systems 1 - 5 denote VFM and 6 - 14 are MFR systems

Results of measurements of supply and return temperatures for the sample of houses are given in Fig. 12, which shows average values for each DH system. The supply water temperature varies from 59 to 87°C and the return temperature varies from 28 to 44.5°C. The utilized temperature drop, shown in Fig 12 as the difference between supply and return temperatures, varies from 25 to 42.5°C.

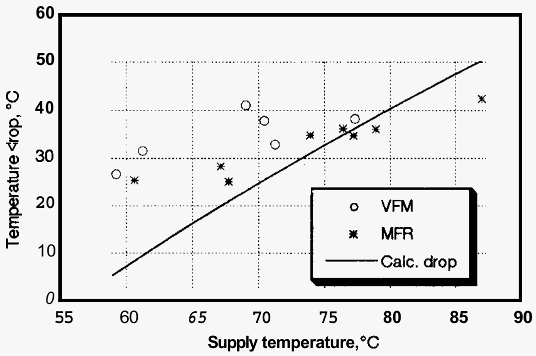

Fig. 13 Temperature drop as related to geothermal supply temperature Dot shapes indicate the DH system type and the solid line shows the theoretical temperature drop for an 80/40/-15 radiator system at design conditions.

The same results are shown in Fig. 13, where the temperature drop is correlated to the supply temperature for each DH system. For comparison the figure also shows a theoretical curve giving the calculated drop for a system based on the standard 80/40/-15 radiator design working at different supply temperature conditions. The calculations are based on design radiator heat load (-15°C). Measurements show that the temperature drop is larger in systems with a high supply temperature than in systems with a low supply temperature as could be expected. The influence of the supply temperature is, however, much smaller than the theoretical curve shows, mainly because users in systems with low supply temperatures install larger radiators than the standard 80/40/-15 design in order to improve the utilization of the geothermal water. Fig. 13 shows clearly the influence of the tariff system on the energy utilization. In VFM systems the temperature drop is in general larger than in MFR systems at similar supply water temperature.

From the water consumption and temperature measurements the annual average energy consumption per m³ of heated space has been evaluated for each house in the VFM system. The results, presented in Fig. 14, show that the energy use varies from 62 to 84 kWh per m³ of heated space. It is noted that the difference has a clear correlation to the heating cost in the respective DH systems which varies considerably.

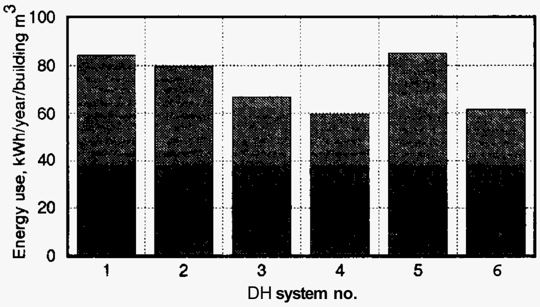

Fig. 14 Measured annual energy use of 6 VFM systems.

7.2 Measurements in Heat Exchanger Systems

No wide ranging data collection programs have so far been organized to study the geothermal residential heat exchanger DH systems in Iceland. These systems constitute a relatively recent addition to the types operating in Iceland and it may be that the inherent difference between these and the simple throughflow systems making them worthy of scrutiny has not become clear to their operators. One of the present authors (Karlsson), however, has for the last three years recorded on a regular basis the temperatures and flow volume of geothermal water in his home in Seltjamames, a suburb of Reykjavik, Iceland. This little town of about 4500 inhabitants operates its own geothermal DH system which draws its hot water from 2500 m deep wells. The water is unsuitable for direct throughflow utilization, so each user has a heat exchanger installed in his house. The results of the measurements made for the year 1991 are shown in Figs. 15 and 16, giving monthly averages throughout the year of temperatures and geothermal water used.

The Seltjamames DH system has an added complexity by the fact that the geothermal water produced from the boreholes is at a temperature of about 115°C. This makes it necessary to use a part of the returning geothermal flow for cooling down the supply water to the users. The system is therefore a combination of categories 2 and 3 (see Introduction).

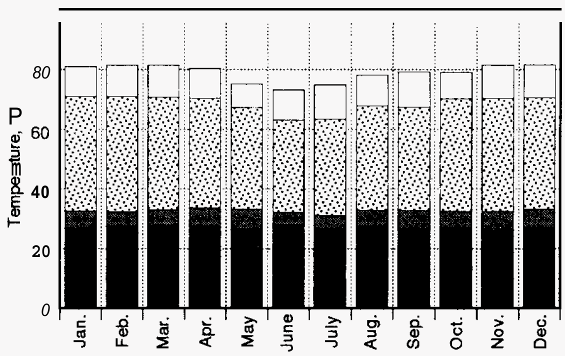

Fig. 15 Average monthly supply and return temperatures in a heat exchanger system. The four values overlapping for each month show from bottom the values of T_r, T_{gr}, T_s, and T_{gs}. Measured data during 1991 in a private home in Seltjamames, Iceland.

The four temperature values shown for each month in Fig. 15 represent as indicated the radiator return temperature, the geothermal return temperature, the radiator supply temperature, and the geothermal supply temperature. It appears that the assumption of a constant temperature value of the incoming geothermal water is not quite true for the Seltjamames system. A constant temperature value from the boreholes is not far from reality, but cooling of the water as it is distributed to the user is always present. This cooling is increased in summer when demand is reduced as is clear from the figure. In Seltjamames there is an added cooling factor with a part of the returning geothermal water being mixed with the borehole water which results in an even more pronounced cooling of the water. The result is that the geothermal supply temperature, which in the cold season, January through April and September through December, is fairly constant and close to 80°C, is lowered during the summer season, May through August, reaching a low average value of about 73°C during the month of June.

This house system is controlled by keeping the radiator water supply temperature T_s at a constant value close to or slightly above 70°C, based on $T_{gs} \approx 80$°C in accordance with the results obtained in

Section 6.2. However, as the geothermal water supply temperature is decreased, the radiator supply temperature is reduced more or less in tune with the geothermal value. The control is therefore achieved by maintaining an approximately constant temperature difference between geothermal and radiator supply temperatures.

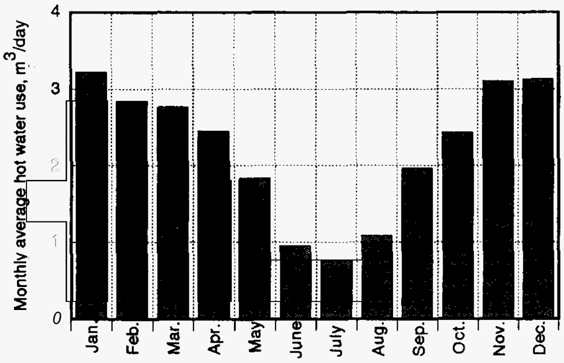

Fig. **16** Average monthly geothermal water use in a residential heat exchanger system. Measured data during 1991 in a private home in Seltjamames, Iceland.

Fig. 16 shows the monthly averages of daily geothermal water used by the system (hot tap water heating included). Fig. 1 seems to indicate that throughout the average year there is no day in which some heating is not needed judging from the equivalent temperature. In reality it is not quite that bad, since there are sunny days during the warmest period in which it is not necessary to start the heating systems. In all months, however, some heating will be called for from time to time, but there is a very marked difference for example between the requirements in January and July as clearly seen in Fig. 16.

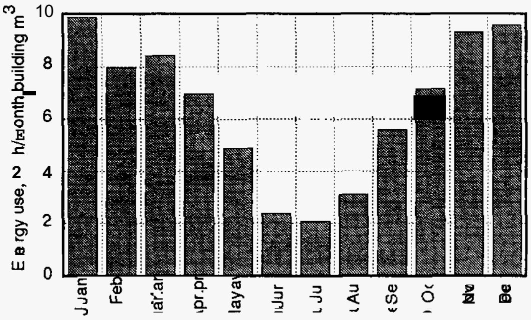

Fig. **17** Monthly energy use in a residential heat exchanger system. Measured data during 1991 in a private home in Seltjamames, Iceland

The monthly specific use of energy, shown in Fig. 17, has been evaluated from the data of Figs. 15 and 16. The seasonal variations follow more or less those found in Fig. 16 resulting in a total annual energy use of 77.4 kWh/building m³. This result is comparable with the results from the direct throughflow systems shown in Fig. 14, even though the heat exchanger system is found to be from 7 to 9 per cent less effective than direct throughflow systems.

8 DISCUSSION AND CONCLUSION

This paper has dealt with Icelandic geothermal DH systems which in some respects are unique and their design and construction different from geothermal DH systems elsewhere in the world. Many of the systems utilize water of rather low temperatures where 60 - 70 °C or even lower are not uncommon and temperatures exceeding 80°C are an exception. Most homes in the country are heated by means of hot water radiators and for the majority the geothermal water is pure enough to be piped directly through the radiators and subsequently disposed of through local wastewater systems.

In a few places the geothermal water is not suitable for direct throughflow utilization due to its chemistry leading to corrosion of the thin walled steel radiators. In these cases the users have been forced to install heat exchangers in their heating systems where heat is transferred from the geothermal fluid to a more suitable fluid circulating in a closed loop piping system feeding the radiators.

It is shown how the low temperature geothermal energy is utilized in both of the above systems simply by making the radiators larger than is customary in hot water radiator systems, for example in continental Europe. This will bring the temperature of the radiator water down to low values resulting in good utilization of the geothermal heat even for low supply temperatures. The large temperature drop leads to less water being circulated to the users and thereby to reduced pumping requirements and smaller pipe dimensions in the distribution system.

Control of the direct throughflow system is very simple since only a room temperature sensor actuating a flow control valve on the supply pipe feeding each radiator is needed. The same control mechanism is required for the heat exchanger systems, but additional control mechanism is necessary to control the geothermal water flow to the heat exchanger. It is shown that by proper selection of radiator supply temperature the annual geothermal water flow may be kept at a minimum. This represents the most economical operating conditions and the optimum radiator supply temperature so determined is found to be a slowly varying function of the outdoor temperature. Calculations show, however, that it is possible to select a constant value for the radiator supply temperature which results in a geothermal flow within one per cent of the most economical flow. This leads to the conclusion that there is no reason to install a complex control mechanism where the radiator supply temperature is controlled by the outdoor temperature. Instead it is sufficient to control the geothermal water flow to the heat exchanger by a fixed radiator supply temperature. It should be borne in mind, however, that these speculations are based on the assumption of a constant supply temperature of the geothermal fluid which is not quite true in practice.

The results of measurements of the energy utilization of Icelandic direct throughflow DH systems show a distinct correlation between the economy of water use and the type of tariff system used. In systems with tariff based on maximum flow restriction (MFR) seasonal variations in water use are in many cases found to be insignificant in comparison with those of systems with tariff based on volumetric flow metering (VFM). It is for this reason that most DH systems in Iceland have abandoned the MFR tariff method in favour of the more economical VFM method.

Very limited data are available from the operation of residential heat exchanger DH systems in Iceland. The numerical analysis presented in section *6* has indicated that a supply temperature to radiators approximately 10°C below that of the geothermal water will result in the most economical operating conditions for the temperatures in question. Section 7.2 presents some preliminary data from a private home heating system where such conditions were maintained approximately. The results indicate that a fairly good utilization economy of the geothermal heat was achieved.

9 REFERENCES

Alfa-Laval (1969). Thermal Handbook. The Alfa-Laval/de Laval Group, Sweden, Table T-11, p. 143.

Deutsches Institut fur Normung (1988). DIN 4703 - ***Raumheizkorper, Teil 3 - Begriffe, Grenzabmaße, Umrechnungen, Einbauhinweise.*** Beuth Verlag GmbH, Burggrafenstraße 6, 1000 Berlin.

Jónsson, G. R., and Jónsson, V. K. (in press). Equivalent Outdoor Temperature for District Heating Systems. Paper accepted for publication in ***ASHRAE Transactions.***

Karlsson, T. (1992). ***Heat Exchangers in District Heating Users' Substations*** (in Icelandic). Report No. VD-VSS-92004, Engineering Department, University of Iceland, October 1992.

Ragnarsson, A., (1991a). ***Energy Utilization in Icelandic District Heating Systems*** (in Icelandic). Paper presented at the 11th annual meeting of the Icelandic District Heating Association, Keflavik, Iceland, May 1991.

Ragnarsson, A., (1991b). ***Energy Utilization in Icelandic District Heating Systems and the Influence of Different TariffSystems*** (in Icelandic). Paper presented at the Energy 91 Conference, held in Reykjavik, Iceland, November 1991. Printed in the Conference Proceedings, pp 639 - 649.

Raju, K. S. N., and Bansal, J. C. (1981). Design of Plate Heat Exchangers. In: ***Low Reynolds Number Heat Exchangers.*** Proc. Fourth NATO Advanced Study Institute on Heat Transfer. Sponsored by NATO Scientific Affairs Division, Ankara, Turkey, July 13 - 24, 1981. Hemisphere Publishing Corp., New York 1983.