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Open Heat Exchanger for Improved Heat Efficiency in Geothermal Spas

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ABSTRACT

Hot spas and Jacuzzis are popular in Iceland due to the abundance of reasonably prized geothermal heat available. However the water from the district heating system is too warm to be admitted directly into the spa. For safety reasons the water is mixed with cold water, from 75ºC down to 50ºC, which leads to wasting a large quantity of heat. Therefore a design was suggested that enables the feeding of geothermal water directly into the pot, omitting the step of mixing it with cold water. The idea is to employ an open heat exchanger that transfers much heat from the geothermal water to the bulk water in the spa, before letting it mix with the spa water. A case study was done for one particular spa. Heat load was calculated and measured when the spa was in use, and when it was unused. A design is suggested employing a circular double-plate which is to be placed at bottom of pot. This unit will function as an open heat exchanger feeding district heating water into the pot. Free convection takes place at the up side of the upper plate and forced convection below the upper plate. Heat transfer coefficient for both was calculated. Temperature field in the pool before and after implementation of the open heat exchanger was measured at different points using thermocouples. The measured temperatures were compared to thermal and fluid-dynamic simulation of the temperature and flow fields obtaining good accordance. Results are reasonable and promising for a good design that may considerably reduce the energy expenses for a continuously heated geothermal spa. More detailed measurements were made on the upper plate of the heat exchanger and detailed simulation of the heat exchanger itself was then used to obtain a value for the heat transfer coefficient for the upper plate to the surrounding water. This information was used to make an improved design for the open plate heat exchanger, stating that a diameter of 63 cm and a thickness of 1.5 cm were suggested as final design. Due to economy consideration the recovery time of the implementing of suggested heat exchanger is estimated to 8 months in studied case.

1. INTRODUCTION

In Iceland there is an abundance of geothermal heat, used for district heating in most residential areas. It is quite common for people to have hot spas or geothermally heated outdoor Jacuzzis at their houses, for recreational purposes. However, even in Iceland there are those that use electrically heated Jacuzzis, and the companies that sell those claim that they are cheaper to operate than the geothermal ones. Although that may not be true, due to the fact that the heat is a much cheaper form of energy than electricity, much energy is wasted in the technology currently used for heating the spas.

Currently there is an obvious waste of heat involved in the mixing of hot water in the geothermal district heating system with cold water, before using it as a heat source in the spa. The motivation of this study is therefore to improve energy efficiency in the use of continuously operated geothermally heated spas, as are common in Icelandic homes. As there is no product currently available on the market that makes it possible to emit hot water from the district heating system directly into the spa, the aim of this study is to develop such a unit, an open heat exchanger that releases much of the heat from water into the pool by conduction and free convection, before the water itself enters the pool.

The district heating system provides water at 80°C, which is mixed with cold water at 5ºC to decrease the temperature down to 45-50°C, then this mixed water will go into the hot spa to be used (Figure 1). There are several types of hot spas. All of them have similar structure. Main differences between them are the means of supplying energy to the spa and the source of energy consumed.

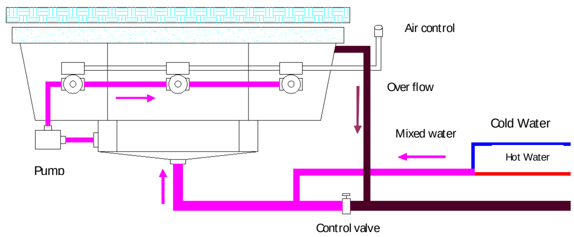

Figure 1: A schematic view of the hot spa.

An investigation of the topic of geothermal spas has been performed at Reykjavik Energy (Reykjavik Energy, 2009) and the results show considerable wasted energy from spas. There is not too much literature available that specifically considers hot spas and small pools, but a spa can be viewed as a swimming pool on a very small scale. Publications on swimming pools and related fields are considered as a reference sources for the literature studies. It was assumed that convection is the most predominant heat transfer mechanism in the spa.

2. THEORY AND METHODS

Heat loss calculation for a spa is the necessary starting point in the study and the only parameter used to describe external conditions is the outdoor temperature. The total heat loss from the spa was calculated to be 11.3 kW on average. The three most important design criteria are strict safety conditions for users with minimum cost and a setup with long service life. A comfortable temperature of water inside of a hot spa is around 40ºC. The maximum possible temperature allowed for water at any location within the hot spa is 50ºC, since higher temperatures are potentially dangerous for users. Water at 75ºC is fed into an open heat exchanger through which it dispels heat to the surrounding water in the spa and is cooled to below 50ºC before it mixes with the water in the spa. The water entering the heat exchanger therefore renews the water in the spa.

According to Figure 2, 0.012 kg/s of water at 75ºC is needed to maintain the desired temperature in the spa. The mass flow would be 0.043 kg/s if the inlet water temperature was 50ºC, which is the present situation for spas that mix hot and cold water.

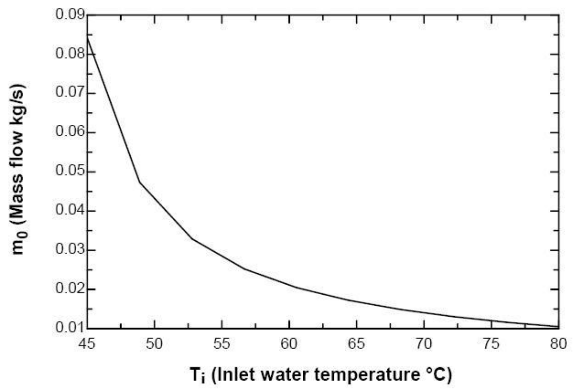

Figure 2: Mass flow change in relation with temperature of inlet water into spa.

The average heat-transfer coefficient from horizontal flat plates is calculated with Equation 1 and constants given in (Holman, 2001).

$$\overline{Nu}_f = C(Gr_f \Pr_f)^m \quad (1)$$

The characteristic dimension for use with these relations has traditionally (McAdams, 1954) been taken as the length of a side for a square, the mean of the two dimensions for a rectangular surface, and 0.9d for a circular disk. References (Lloyd and Moran, 1974) and (Goldstein et al., 1973) indicate that better agreement with experimental data can be achieved by calculating the characteristic dimension with:

$$L = \frac{A}{P} \quad (2)$$

This characteristic dimension is also applicable to unsymmetrical platforms. The essential ingredients of forced convection heat transfer analysis are given by Newton's Law of Cooling:

$$\dot{Q} = hA(T_w - T_\infty) = hA.\Delta T \quad (3)$$

The rate of heat $\dot{Q}$ transferred to the surrounding fluid is proportional to the object's exposed area A, and the difference between the object temperature T_w and the fluid free-stream temperature T_∞. The constant of proportionality h is termed the convection heat-transfer coefficient.

The following analysis was used to determine the design parameters for the open heat exchanger prototype. Temperature under the center plate is described by the following differential equation (Palsson, 2008):

$$c\dot{m}\frac{dT}{dr} + 2\pi rh(T - T_a) = 0 \quad (4)$$

The equation can be solved if the boundary condition T(0) = T_i is given, which is the inflow temperature. It is assumed that the flow is turbulent and the flow velocity as a function of r can be calculated:

$$v = \frac{\dot{m}}{2\pi ry\rho} \quad (5)$$

$$R_e = \frac{\rho vr}{\mu} = \frac{\dot{m}}{2\pi y\mu} \quad (6)$$

$$N_u = 0.332 * R_e^{1/2} P_r^{1/3} \quad (7)$$

$$h = \frac{kN_u}{r} \quad (8)$$

Equation 4 becomes:

$$c\dot{m}\frac{dT}{dr} + 2\pi kN_u(T - T_a) = 0 \quad (9)$$

Then we define

$$\theta = \frac{T - T_a}{T_i - T_a} \quad (10)$$

$$\eta = \frac{r}{R} \quad (11)$$

which results in a dimensionless form of the equation:

$$\frac{d\theta}{d\eta} + \alpha\theta = 0 \quad (12)$$

with

$$\alpha = \frac{2\pi kN_u R}{c\dot{m}} \quad (13)$$

and the boundary condition $\theta(0) = 1$. The solution to (12) is

$$\theta(\eta) = \exp(-\alpha\eta) \quad (14)$$

which can be used to calculate the exit temperature of the water, $\theta(1)$, which then becomes

$$T_e = T_a + (T_i - T_a)\exp\left(-\frac{2\pi kN_u R}{c\dot{m}}\right) \quad (15)$$

The heat flow into the plate is equal to the heat flow from the plate top to the water in the spa. Thus:

$$c\dot{m}(T_i - T_e) = h_t \pi R^2 (T_a - T_p) \qquad (16)$$

Furthermore, the heat balance of the spa itself can be formulated as:

$$c\dot{m}(T_i - T_p) = UA(T_p - T_\infty) \qquad (17)$$

where it is assumed that the water flowing out of the spa has temperature T_p. In order to solve the problem, it is assumed that all parameters are known except $\dot{m}$, which can be found directly from (17), T_e and T_a. After $\dot{m}$ has been found, equations (15) and (16) can be used to obtain T_e and T_a. It is important to note that h_t is generally a function of T_a and T_p. Table 1, illustrates some results from calculations performed with the described method.

Table 1: Experimental data

Parameter	*Value*	*Unit*
Inflow water temp. (Ti)	75	ºC
Spa total heat transfer coefficient	3.5	Kw/m^{2}ºC
Air temp. (T_s)	0	ºC
Mass flow rate (m_0)	0.012	Kg/s
free convection coefficient (h_t)	546.1	w/m^{2}ºC
Rayleigh no. (R_a)	8.776*10^7	
Reynolds no. (R_e)	2153	
Heat loss from spa (Q)	1.628	Kw/m^2
Plate edge temp. (T_e)	43.43	ºC
Plate temp. (T_a)	43.43	ºC
Gap between plates (Y)	0.002	m
Radius of plate (R)	0.15	m

Next step was to decide the dimensions of the plate. It is very important to have a feasible size for the plate due to the possibility of production in future, low cost and ease of use. According to data from modeling and similar experiences, and also regarding to available domestic market, a relatively simple prototype was designed and made. It was decided to use a circular plate with a 30 cm diameter as a prototype for testing. Details of designed plate are shown in figure 3.

2.1 MEASUREMENTS

In order to analyze temperature distribution in the spa, on-line measurements were performed with thermocouples. A sample spa was selected to monitor temperature distribution in the spa. Thermocouples were used to measure temperature and analogue to digital convertor was used to transfer analogue data to digital. Measurement system was installed in spa at a family house in order to monitor eight points in different parts of the spa as well as air temperature. Figure 4 shows the arrangement of measured points. A Matlab code was used to analyze the data gathered in text file format. To make sure that no important information was lost, the sampling frequency was 10 Hz for each point. Then the mean temperature was calculated for 10 seconds and to plot the temperature change as a function of time. The final graphs illustrate mean temperature in 10 seconds, but each point is the mean of 100 data values.

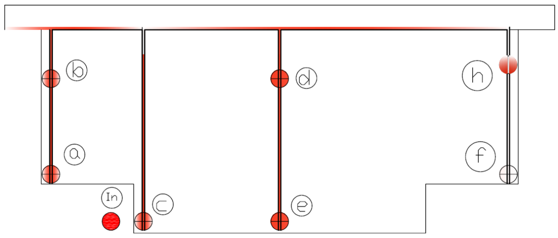

Figure 4: Schematic plan of measured points in the spa and their location.

Figure 5, shows the experiment layout and arrangement of the thermocouples used for the data gathering process.

Figure 5: The experiment layout and arrangement of thermocouples for data gathering process.

2.2 Modeling

Heat transfer can occur by three main methods: conduction, convection, and radiation. Physical models involving conduction, convection only are the simplest, while buoyancy-driven flow or natural convection, and radiation models are more complex.

Fluent solves the energy equation in the following form (Fluent, 2008):

$$\frac{\partial}{\partial t}(\rho E) + \nabla.(\vec{\upsilon}(\rho E + p)) = \nabla.(k_{eff}\nabla T - \sum_j h_j \vec{J}_j + (\overline{\overline{\tau}}_{eff}.\vec{\upsilon})) + S_h \qquad (15)$$

Where k_{eff} is the effective conductivity ($k + k_t$, where k_t is the turbulent thermal conductivity, defined according to the turbulence model being used), and $\vec{J}j$ is the diffusion flux of species j. The first three terms on the right-hand side of Equation 15 represent energy transfer due to conduction, species diffusion, and viscous dissipation, respectively. S_h (energy sources term) includes the heat of chemical reaction, and any other volumetric heat sources that have been defined.

In Equation 15,

$$E = h - \frac{p}{\rho} + \frac{\upsilon^2}{2} \qquad (16)$$

Where sensible enthalpy h is defined for ideal gases as

$$h = \sum_j Y_j h_j \qquad (17)$$

And for incompressible flows as

$$h = \sum_j Y_j h_j + \frac{p}{\rho} \tag{18}$$

In Equations 17 and 18, Y_j is the mass fraction of species j and

$$h_j = \int_{T_{ref}}^{T} C_{p,j} dT \tag{19}$$

Where T_{ref} is 298.15K.

Equation 15 includes the effect of enthalpy transport due to species diffusion. When the pressure-based solver is used, the term (20), is included in Equation 15 by default. When the density-based solver is used, this term is always included in the energy equation.

$$\nabla.(\sum_j h_j \vec{J}_j) \tag{20}$$

To model natural convection inside a closed domain, the solution will depend on the mass inside the domain. Since this mass will not be known unless the density is known, the flow should be modeled in one of the following ways (Fluent, 2008):

Perform a transient calculation. In this approach, the initial density will be computed from the initial pressure and temperature, so the initial mass is known. As the solution progresses over time, this mass will be properly conserved. If the temperature differences in domain are large, this approach must be followed.

Perform a steady-state calculation using the Boussinesq model. In this approach, a constant density will be specified, so the mass is properly specified. This approach is valid only if the temperature differences in the domain are small; if not, the transient approach must be used.

2.2.1 The Boussinesq Model

For many natural-convection flows, the Boussinesq model gives faster convergence rather than setting up the problem with fluid density as a function of temperature. This model treats density as a constant value in all solved equations, except for the buoyancy term in the momentum equation:

$$(\rho - \rho_0)g \approx -\rho_0 \beta (T - T_0) g \tag{21}$$

Where ρ_0, is the (constant) density of the flow, T_0 is the operating temperature, and β is the thermal expansion coefficient. Equation 21 is obtained by using the Boussinesq approximation $\rho = \rho_0(1 - \beta \Delta T)$ to eliminate ρ from the buoyancy term. This approximation is accurate as long as changes in actual density are small; specifically, the Boussinesq approximation is valid when $\beta(T - T_0) << 1$ (Fluent, 2008).

The Boussinesq model should not be used if the temperature differences in the domain are large. In addition, it cannot be used with species calculations, combustion, or reacting flows.

2.2.2 Operating Density

When the Boussinesq approximation is not used, the operating density ρ_0 appears in the body-force term in the momentum equations as $(\rho - \rho_0)g$.

This form of the body-force term follows from the redefinition of pressure as:

$$p'_s = p_s - \rho_0 g x \tag{22}$$

The hydrostatic pressure in a fluid at rest is then

$$p'_s = 0 \tag{23}$$

3. RESULTS AND DISCUSSION

3.1 Monitoring

3.1.1 Monitoring Situation Before Plate Installation

The temperature in the spa is controlled by an electric system which feeds water at a predetermined temperature into the spa. The set point is 37ºC in most cases. This means that when the temperature in the outlet of the spa reaches this value, electronic control system will switch on a pump to feed hot water into the spa. Pumping will stop once the temperature reaches a second set point, which in this case is 40ºC.

Figure 6 shows water temperature in the spa at 8 locations specified in figure 4. The ninth measurement point is air temperature. Point (c) consistently gives the lowest temperature in all of the spa's operation time, as can be seen from the figure. When the outlet temperature in the spa reaches 37ºC, pumping of mixed hot water with temperature around 48°C to 50ºC will begin. It is clear from results that there are points in the spa that have gone below 37°C, because the sensor related to electronic unit is located in the outlet, point (b), which is the closest measurement point to the outlet and always shows the highest temperature in the spa after point (h). This difference between points is minor and is around 1°C to 2°C. Point (win) shows the inlet water temperature. According to results and mentioned description, there are points in the spa where temperature goes down to 36°C and sometimes 35°C. There was one outlet point in the spa, which has an effect on symmetric assumption for temperature distribution. Points (b and h) are confirming a symmetry assumption with very close results, they are almost covered by each other in the plots.

3.1.2 Monitoring with installed plate

After completing the installation of the new system, measurement points were set as before. The only change was that point (win) was used to measure plate edge temperature. Electronic safety unit was adjusted to control operation mode, so it was possible to switch off the old system and start the new one. Then the spa was filled and measurements began.

The temperature of the upper surface of the plate itself is a very important parameter, because it rests in the bottom of the spa and will be touched by the user. It should have an acceptable range of temperature. With the initial arrangement of measuring it was not possible to measure the plate's surface temperature with sufficient accuracy.

By using temperature logger as shown in figures 7 (Appendix I) and 8, plate's temperature was measured. Positions of points win and e were changed according to figure 7 (Appendix I), to measure temperature at two water inflow points.

In figure 7 (Appendix I) the first 400 points show the filling of the spa. As it was expected the minimum temperature was measured at the end, while the maximum temperature was at the center of plate. As mentioned before, 50°C is acceptable at edges of plate, but temperature at the end and middle of plate is relatively high.

After applying several mass flow rates to feed hot water into spa, it was decided to choose 0.03 l/s as an accepted value for mass flow. Figure 9, shows that pumping takes place 67% of the operation time while the mass flow rate is 0.03 l/s. With the assumption of having proportional relation between time and water usage, regardless of weather conditions, it can be calculated that with 0.03 l/s mass flow rate, hot water usage of the spa within 24 hours is equal to 1700 liters.

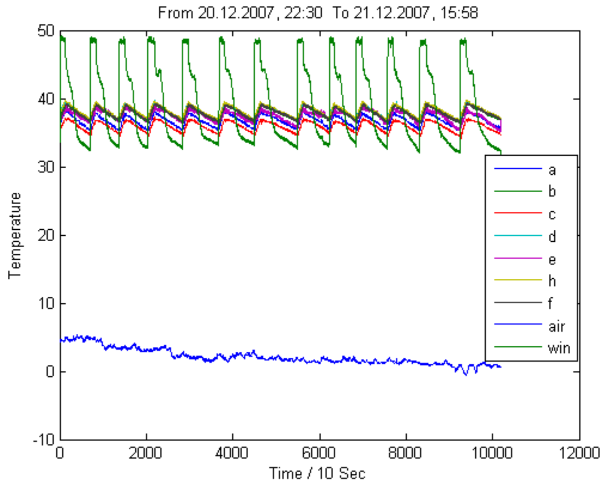

Fig.6: Temperature distribution in the spa along with air temperature while the lid is closed.

Fig.8: Plate logging arrangement.

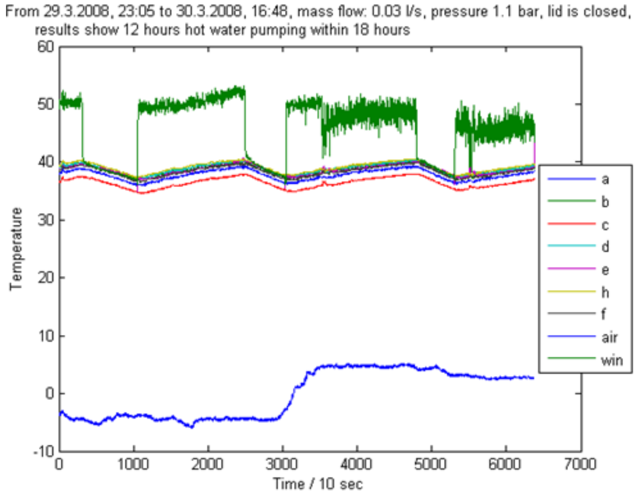

Figure 9: Temperature distribution in the spa with 0.03 l/s mass flow rate.

3.2 Modeling

3.2.1 Modeling of Pot According to Current Situation

A pot according to the specifications in measurement section was modeled by fluent, geometry and meshing were created using Gambit software including the boundary conditions. Then the file was imported to Fluent and heat transfer modeling process was applied. The same method has been applied to model the pot with plate. Modeling results has good agreement with monitoring data. Several mass flow rates were applied for sensitivity analysis of pot regarding this parameter.

Figure 10, shows the cross sectional view of pot regarding to temperature distribution with the mass flow rate of 0.008 kg/s. Two vertical cross sectional lines are shown in this figure. It is interesting to see the effect of water outflow and how interrupts pot's symmetric situation.

Modeling with different mass flow rates proved that temperatures in different zones of pot are sensitive to mass flow rate. This sensitivity is high in some zones. Maximum temperature in top zone equal to 329K with 0.02 kg/s mass flow rate, goes down to 310K when the mass flow rate is adjusted in 0.005 kg/s, this value is 325K, while the mass flow rate is 0.008 kg/s.

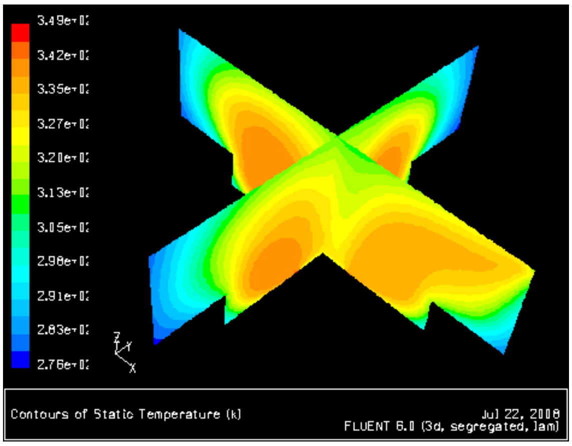

Figure 10: the cross sectional view of pot regarding to temperature distribution (m=0.008 kg/s).

3.2.2 Plate Design and Optimization

A user-defined function (UDF), is a function that user programs, that can be dynamically loaded with the Fluent solver to enhance the standard features of the code. It can be used to define specific boundary conditions, material properties, and source terms for flow regime, as well as specify customized model parameters, initialize a solution, or enhance post-processing (Fluent, 2008). The C program was written to use this ability of Fluent. According to relation between temperature differences and heat transfer coefficient, fluent used this program to reach most accurate values for these parameters.

3.3 Optimization

For sensitivity analysis of parameters, specification of tested plate has been selected as a basic, diameter of plate, gap between upper and lower plates and thickness of upper plate, were the variables that has changed within each round of modeling. Maximum temperature values for water outlet and upper plate are two very important factors from the view of safety. Results look sensitive to all changes, but plate's diameter is the most dominating parameter regarding to design criteria (not exceed more than 323K). According to data from these modeling analysis was done to calculate diameter of plate and thickness of upper plate to fulfill design condition. The results are shown in figure 11. Plate with 63 cm diameter and thickness of 1.5 cm, was suggested as a final design. This size looks large at first step, but regarding to shape and size of floor in pot this design can give pleasant heat through floor to user's feet. The weight will be around 15 kg. For easy removing because of cleaning, it can be built in 4 pieces. Simple linear algebra relation was applied to find the most optimum values for diameter and thickness.

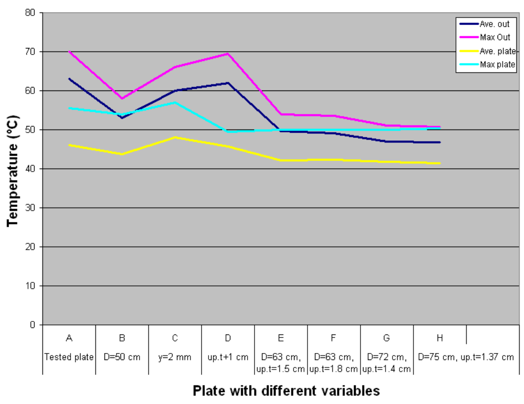

Figure 11: Effect of different variables of plate on temperature distribution.

3.2.3 Modeling of Pot with Plate

The plate with 63 cm diameter and 1.5 cm thickness of upper plate was selected to be modeled by Fluent. Figure 12, shows the temperature distribution in the plate. This model was run just to check the effect of upper plate thickness in the results. 3 mm less thickness didn't have any important change in outlet and surface temperature.

Table 2, shows heat transfer values in different zones of plate and as it can be seen the value of net heat transfer is very small, which proves that the modeling has minimum possible errors.

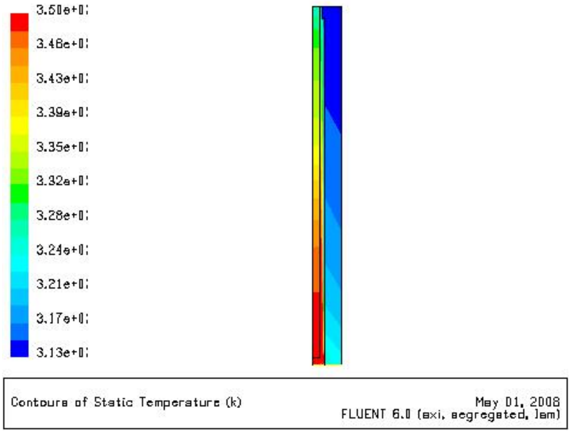

Figure 12: Temperature distribution in the plate.

Table 2: Heat transfer values in different zones of plate.

Zones	Heat Flux report: W
Zone 7 (below)	0
Zone 6 (bs)	-412.93
Zone 10 (in)	6505.10
Zone 9 (out)	-3151.02
Zone 8 (top)	-2932.78
Zone 4 (wall)	3357.63
Zone 14 (wall-shadow)	-3357.61
Net heat-transfer	8.40

The Average temperature for outflow of plate was recorded equal to 322.72k, and for plate surface equal to 315.2k. Figure 13, shows the temperature distribution in outflow of plate. The outflow region is the gap between upper and lower plate.

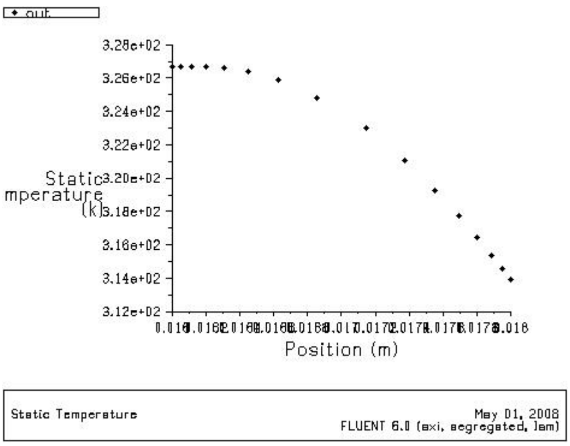

Figure 13: The temperature distribution in outflow of plate

Figure 14, shows the temperature distribution in upper plate of plate.

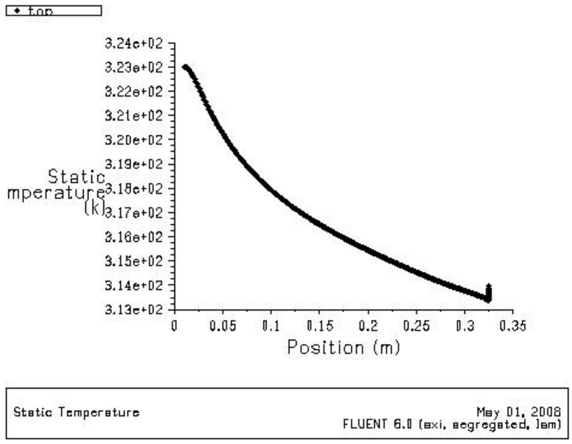

Figure 14: Temperature distribution in upper plate of plate.

4. ECONOMY

The spa which was under monitoring had an analog unit to measure mass flow rate. Then during monitoring, most reliable and accurate data were gathered in table 3.

Table 3: Mass flow rates in spa during different periods.

No.	Time, Date	Mass (m^3)	Duration (hours)	Mass diff. (m^3)
1	22:30, 20.12.2007	1125	17:30	3.6
	15:58, 21.12.2007	1128.6		
2	14:52, 8.3.2008	1532	51	10
	18:00, 10.3.2008	1542		
3	18:07, 10.3.2008	1542	45:40	8
	15:35, 12.3.2008	1550		
Tot			114:10	21.6

Average water consumption is 4.55 m^3 per 24 hours of spa's operation. This amount strongly depends on some parameters like:

- Insulation of the spa
- Weather condition
- Water circulation method, which depends on family's habit and how often they clean their spa.

In the current situation, 80ºC hot water from district heating system is mixed with 5ºC cold water to reach 50ºC, which is then pumped into the spa. With a simple energy balance relation and applying the conservation of mass in water mixing system we get:

$$Q_i = m_1 C_{p1}(T_2 - T_1) = m_2 C_{p2}(T_3 - T_4) \quad (24)$$

resulting in $m_1 = 1.5 * m_2$. According to this result, total of 4.55 m^3 water with 50ºC will be available by mixing 1.82 m^3 cold and 2.73 m^3 hot water with 80ºC.Previously it was concluded that 0.03 l/s mass flow rate is sufficient, and with this flow, spa will use 1.7 m^3 in 24 hours.

All measurements were performed in the same location and the same spa, so if there is any error because of different weather condition, it has been in effect for both operation systems. So the errors due to different weather conditions were assumed to be negligible.

It has been shown that an amount of 1 m^3 of hot water will be saved pr. 24hrs by using the new system in the spa. According to Reykjavik Energy company (Reykjavik Energy, 2008), the price for hot water in Iceland is 70 ISKr/ m^3. So 26316 ISK will be saved within one year. With rough estimate, the plate can be produced by 15000 ISK, and then the investment recovery time, can be calculated for period of less than 8 months. Also compared to the total investment cost involved in installing a spa which is around half million ISkr (Normi hf, 2008), the investment for a new unit can be neglected. These calculations were done according to results from experiment. This amount can be changed according to parameters which were explained previously in this section. But still the result looks very promising. Summary of results are illustrated in table 4.

Table 4. Summary of economical comparison of two systems

Operating type	$\dot{m}_h$ (m^3/ 24 hours)	$\dot{m}_c$ (m^3/ 24 hours)	Cost (ISKr/ m^3)	Total cost
Current system	2.73	1.82	70	191.1
Spa with plate	1.7	0	70	119
Saved per 24 hours (ISKr)				72.1
Unite recovery period (Unite price/ Saved per 24 house), (Days)				208

Note that in Reykjavik, cold water usage is not measured so only hot water usage is considered in table 4.

5. CONCLUSIONS

A novel approach has been suggested for feeding hot water form district heating system directly into hot spas, without prior mixing with cold water. A basic design of an open heat exchanger was made, based on a calculated and measured steady heat load requirement of 11 kW. A circular open plate heat exchanger with a dimension of 30 cm, was constructed and implemented in an outdoor pool, used as a test case. Temperature field in the pool before and after implementation of the open heat exchanger was measured at different points using thermocouples. The measured temperatures were compared to thermal and fluid-dynamic simulation of the temperature and flow fields obtaining good accordance.

The results proved that in principle the open heat exchanger design works, although some design parameters need adjusting in order to meet all criteria on maximum temperature and temperature distribution in the spa. Parts of the upper plate exceeded the set maximum temperature of 50 degrees, and the water flowing out from the sides was also slightly too hot. But the spa was in fact used for days with this configuration.

More detailed measurements were made on the upper plate of the heat exchanger and detailed simulation of the heat exchanger itself was then used to obtain a value for the heat transfer coefficient for the upper plate to the surrounding water. This information was used to make an improved design for the open plate heat exchanger, stating that a diameter of 63 cm and a thickness of 1.5 cm were suggested as final design. This size looks large at first step, but regarding to shape and size of floor in pot this design can give pleasant heat through floor to user's feet.

The conclusion was that for the spa used as a test case, assuming a heat exchanger cost of 15000 ISKr. The recovery time of the investment due to improved heat efficiency is estimated to be 8 months.

In general this is a function of how the spa is used, how often the water is removed for cleaning, how long the lid is kept open etc. The maximum estimated recovery time for a spa in regular use was evaluated as 2 years.

Furthermore, the pressure in district heating network is high enough to feed hot water into the spa so the new system doesn't need a pump.

NOMENCLATURE

A Area.

P Perimeter of the surface.

C Specific heat of water.

$\dot{m}$ Mass flow.

T_a Plate temperature.

H Convection coefficient.

ρ Water density.

y Gap between plate and spa bottom.

R Plate radius.

h_t Free convection coefficient at the plate top.

T_p Water temperature in the spa.

U Heat transfer coeff. between spa and surroundings.

T_∞ Temperature of the surroundings.

m_1 Amount of hot water from district heating system.

m_2 Amount of cold water with 5ºC (kg/s).

T_1 Hot water temperature (80°C).

T_2 Cold water temperature (5°C).

T_3 Desired water temperature to be pumped into spa

$\dot{m}_h$ Mass flow rate for hot water (m^3/ 24 hours)

$\dot{m}_c$ Mass flow rate for cold water (m^3/ 24 hours)

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APPENDIX I

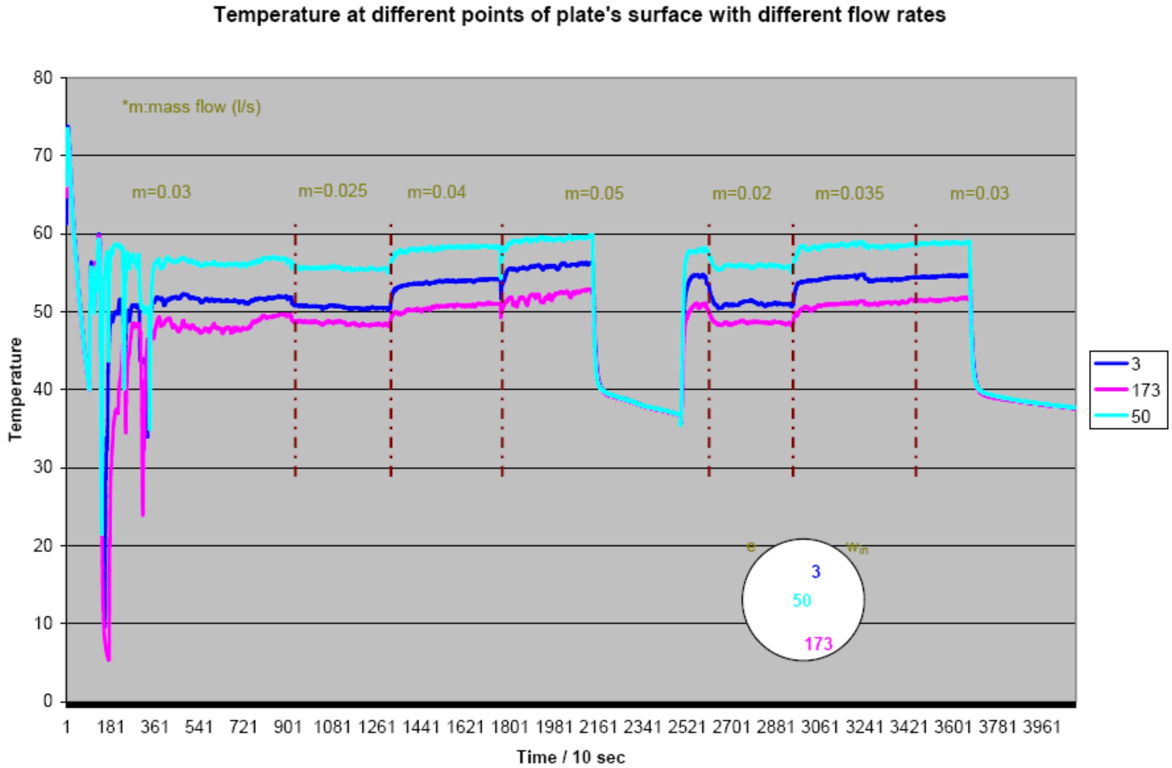

Figure 7: Temperature distribution in the surface of plate at different mass flow rates.