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SNOW MELTING AND HEATING SYSTEMS BASED ON GEOTHERMAL HEAT PUMPS AT GOLENIOW AIRPORT, POLAND

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ABSTRACT

A design for a snow melting and heating system for the airport at Goleniow, Poland, is proposed. The system is based on utilization of geothermal energy from the Szczecin region close to Goleniow town. Geothermal water at 40-90˚C is found in this area. The system is based on utilization of heat pumps extracting heat from 40 or 60˚C hot water. Comparison of the performance of different heat pump arrangements in a heating system for 40˚C geothermal fluid is made. Thermal and fluid-flow calculations for the geothermal heat plant are presented here. The production-reinjecting geothermal well system consists of a counter-flow geothermal-heat exchanger and heat pump (electrically driven) with an evaporator located either *directly* after the geothermal heat exchanger (system 1), or with an evaporator located *indirectly* on the return network water (system 2). The system also contains a conventional heat source – a peak gas boiler, which can work together with a geothermal heat exchanger (system 3). The heating plant supplies heat to a distribution network, which consists of parallel-connected heat users with different operational requirements. For the first type of user, a low-temperature radiator heating system, calculations for two different working scenarios were done. In the first scenario, the temperatures of the supply and return water are a function of the outdoor temperature. In the second scenario, the supply and return water temperatures do not depend on the outdoor temperature but are considered to be constant. The second type of user is a tap water / ventilation system running the network water at constant temperature throughout the year. The third type of user is a snow melting system working in a -16 to +3˚C outdoor temperature interval, at variable heat demand values, for melting (55/25˚C) and idling (30/15˚C) time. Geothermal heat is supplied to this system through a heat exchanger.

Each of the three different systems considered in this paper is represented by a relevant schematic diagram, which demonstrates the use of geothermal energy, electrical energy and energy from a gas boiler. In the presented heating systems, the direct heat pump approach is more efficient and economical than the indirect heat pump approach. With purposeful control and by using a snow detector in a snow melting system, the quantity of hot water and operating costs are reduced.

1. INTRODUCTION

Currently almost all of Poland's fuel requirements for heating are met by hard and brown coal. Unfortunately, burning such fossil fuels results in major environmental deterioration. Trying to restrain this rapidly growing environmental pollution, authorities have more and more frequently turned towards alternative energy sources (renewable), among which geothermal energy plays a significant role. Poland is a country rich in geothermal water resources of average enthalpy. The volume of these geothermal waters, having a temperature between ~30 and ~120°C, is estimated at circa 6,500 km^3 (Sokolowski, 1993). Water at temperatures ranging from 50 to 90°C is brought to the surface through boreholes of 1.5-3 km depth. The resources are more or less evenly distributed across a significant part of Poland in specified geothermal basins and sub-basins, which belong to the specific geothermal provinces and regions. The most favourable geothermal conditions can be found in the Polish Lowland, Podhale and Sudety.

Despite such significant energy potential in geothermal resources, extensive exploitation started only one decade ago. Until that time, geothermal water was only used for balneological purposes. Examples of important centres where utilisation of geothermal water for therapeutic treatment is taking place are the spas: Lądek Zdrój, Cieplice, Ciechocinek. Between 1993 and 2001, four large thermal systems based on geothermal energy were constructed and commissioned in Poland (Banska Nizna, Pyrzyce, Mszczonow, Uniejow). Construction of subsequent developments is planned (Zapalowicz et al., 2002).

The project design calculations presented here are based on the utilization of geothermal energy through a heat exchanger and a heat pump in different arrangements. The hot water produced will be used for radiator and tap water heating as well as for heat transfer for the proposed installation of a snow melting system at the Goleniow airport. This will make it possible to reduce emissions of greenhouse gases from conventional fuels.

Pavement snow melting by geothermal water and steam is used in several countries, including Iceland, Japan and the United States. These installations include sidewalks, roadways, ramps, runways, squares, parking areas and bridges. Most commonly it is done with a glycol solution, hot water or steam being circulated in pipes within or below the pavement, using either heat pipes or geothermal fluids. In one instance, hot water was sprinkled directly onto the pavement. This paper will attempt to present the general design requirements for a snow melting system and propose a solution for an airport where geothermal heat is supplied to the water system through heat exchangers. The obvious benefits of such systems are that they eliminate the need for snow removal, provide greater safety for pedestrians and vehicles, reduce the labour in slush removal and make better working conditions at the airport.

2. HEAT PUMPS IN HEATING SYSTEMS

Heat pumps are heat transfer devices that, by using an input of work or heat, are able to reverse the normal direction of passive heat transfer and absorb heat at a low temperature and reject it at a higher temperature. Heat pumps can be used in low-temperature geothermal heating schemes to generally boost the heat output of the fluid, but their particular role in any specific scheme will depend upon the temperatures of the fluids which are being used. Thus, with moderate temperature fluids in the range of 40-70°C, the heat extraction will be dominated by a primary heat exchanger and the heat pumps will usually be connected in a way which extracts additional heat from the geothermal fluid. However, with fluid temperatures less than 40°C, direct heat exchange becomes almost impossible and the heat pump is connected so that it accomplishes all of the heat transfer (Harrison et al., 1990).

2.1 Basic principles

The most common heat pumps are of the vapour compression type where a mechanically driven compressor is used as described below. When a gas is compressed without loss of heat, its temperature and pressure increase because of the work done on the gas by the compressor. Conversely, when a gas expands, its temperature and pressure decrease. Compressing and/or cooling a gas sufficiently can turn it into a liquid. Liquids and gases are separately known as states of matter and together known as fluids.

The conversion of a liquid to a gas is called vaporization or evaporation, and it takes place at constant temperature with absorption of heat from the surrounding environment. The heat absorbed goes into increasing the molecular kinetic energy. The amount of heat required to convert a unit-mass of liquid into its vapour is called its heat of vaporization. In a pan of water boiling on the stove, the liquid water remains at boiling temperature while the heat input from the stove is taken up as heat of vaporization to convert the water to steam. For water, the amount of heat needed is 2258 kJ/kg. When a gas reverts back into liquid, a process known as condensation, it releases heat - its so-called heat of condensation - to its surroundings. The heat of condensation and the heat of vaporization are equal for any given fluid, so the fluid gives up the same amount of heat upon condensation that it took up upon evaporation.

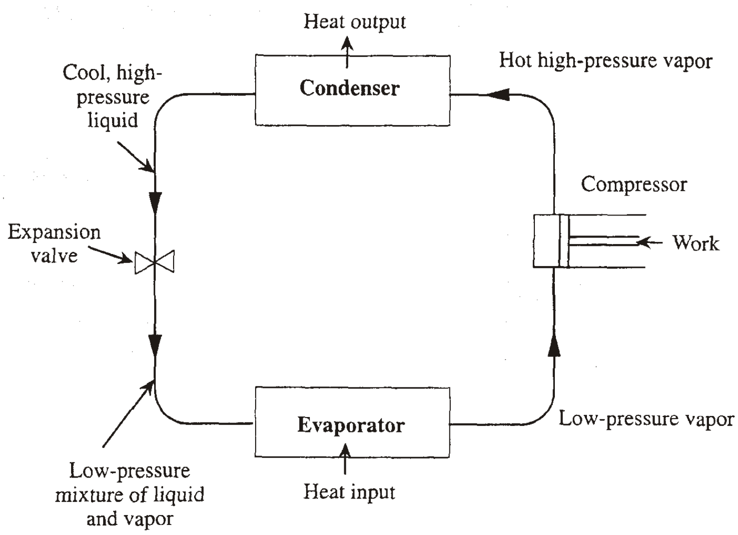

FIGURE 1: Schematic diagram of a heat pump (Wright, 1999)

In a heat pump, a working fluid is contained in a closed, sealed circuit. The working fluid in today's geothermal heat pumps is commonly some kind of hydrofluorocarbon (HFC, chlorine free) or a natural refrigerant like propane, isobutane, ammonia or carbon dioxide. As the working fluid moves around the circuit, it is repeatedly expanded and evaporated in one part of the system, causing cooling and absorption of heat, and compressed and condensed in another part of the system, causing warming and release of heat. The effect is to move, or pump, heat from the part of the system where the fluid is vaporized to the part of the system where the fluid is condensed. Figure 1 illustrates this idea. The compressor does work on the gas, increasing its temperature and pressure. The hot, high-pressure gas flows into the condenser, where it gives up heat to its surroundings by first cooling as a gas and then condensing, releasing its heat of condensation. The fluid exiting the condenser is a cooled, high-pressure liquid. An expansion device releases the pressure on this liquid, which flows into the evaporator as a low-pressure vapour/liquid mixture. In the evaporator, the heat of vaporization of the liquid is absorbed and the resulting low-pressure gas flows back to the compressor, completing the circuit. The heat that appears at the condenser is the sum of the heat absorbed at the evaporator and the heat added by the compressor through the work of the motor running the compressor. Because electricity is used only to run the compressor, the heat pump delivers 3 to 4 times more energy than it consumes (Wright, 1999).

2.2 Basic arrangements

The heat production units in a thermal station fed by geothermal energy for the supply of a district heating network may be the following:

- Primary heat exchangers between the geothermal fluid and the district heating water;
- Heat pumps;
- Heat recovery units of the heat pumps' prime movers;
- Boilers.

Peak load boilers are absolutely necessary when the supply temperature of the district heating network is so high that it cannot be obtained by means of base load units only; otherwise, the decision to install peak load boilers and the choice of the thermal power for these units depends on an economic optimisation calculation (Piatti et al., 1992).

Heat pumps are not single elements like primary heat exchangers or back-up boilers. The evaporators and condensers are located in different parts of the system; by-pass connections of various types are also possible. Consequently, a wide variety of different layouts are possible in geothermal schemes all of which can, in general, perform differently. If attention is focussed on the way in which the heat pump supplies heat in any scheme, then two basic classes of configuration can be identified:

- The heat pump assists the primary heat exchanger, supplying additional heat from the geothermal fluid. This is called the *heat pump assisted (HPA) approach*. In these arrangements the heat pumps are connected in ways which produce additional geothermal heat, over and above what would be obtained from simple heat exchange alone. The geothermal heat transfer is still dominated by the primary heat exchanger and significant heat transfers would be obtained with the heat pump switched off. The basic arrangements are with alternative *direct* or *indirect* evaporator connections.
- The heat pump dominates the geothermal supply and no heat is transferred if the heat pump is not operating. This is called the *heat pump only (HPO) approach*. This arrangement tends to be used when the temperature of the supply fluids, aquifer brines or groundwater, is so low that only insignificantly small heat transfers would be obtained by simple heat exchange alone. The heat is extracted by the evaporator from the geothermal supply fluid either *directly* or across an *auxiliary* heat exchanger. The heat is released to the heating system by the condenser. The only heat transfer path is through the heat pump and no heat is delivered unless the heat pump is working. The heat pump upgrades the heat extracted so that the condenser outlet temperature is higher than the geothermal supply temperature (Harrison et al., 1990).

In the presented project for the airport heating system, configurations based on an electrically-driven heat pump assisted (HPA) with an indirect evaporator, later called indirect evaporator variant, and a heat pump only (HPO) with a direct evaporator, later called direct evaporator variant, as well as a gas boiler were considered.

3. SNOW MELTING SYSTEMS - EXAMPLES

3.1 Iceland

In the old centre of Reykjavík, renewal of roads and pavements has been in progress since before 1990, with snow melting pipes embedded simultaneously. At present, a snow melting area of 30,000 m^2 is a reality; it is expected to be extended to 60,000 m^2 when renewal is complete.

In Reykjavík, geothermal hot water is used for heating houses. The distribution system is partly single pipe and partly double pipe. In the old city, a single pipe system was laid. Geothermal water, after being used for heating houses, was released into the drain system of the city. The return water, at 32°C, contains a lot of energy. While renewing the roads the opportunity for using this water in snow melting was realised. The return water is now led to five control centres for snow melting. The pipe system for the water was laid out in such a way that the centres are interconnected and the water can be distributed to the centres at will.

All snow melting pipes are plastic pipes except for the pipes in control centres, which are made of steel. Mains for supply and return water are laid from control centres and branches are connected to valve boxes where tubes are connected. All tubes are connected with valves. It is very important that these valves be easily accessible so a tube can be disconnected or relieved of air whenever needed. Therefore, concrete valve boxes with cast iron cover plates at final surface level have been used (Figure 2). Thus, access to valves is possible without removing stones from the slab.

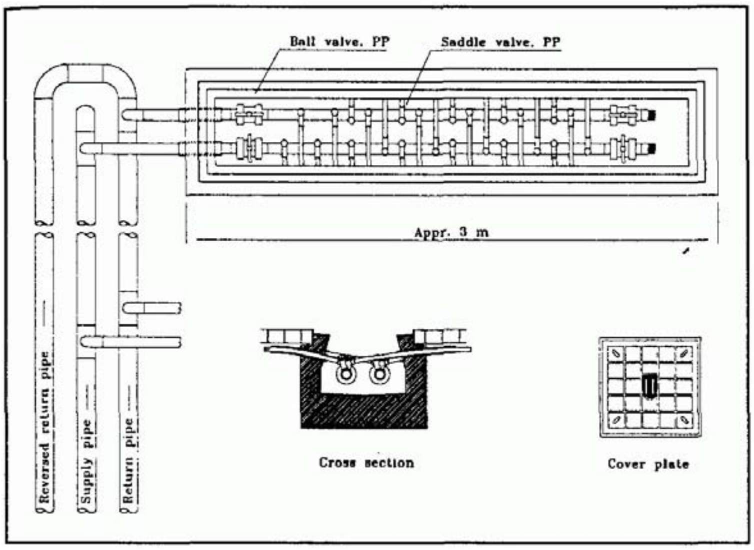

FIGURE 2: Valve box in a snow melting system in Iceland (Ragnarsson, 1997)

The pipe system is a reverse-return system that maintains pressure balance. All snow melting tubes are of equal length, 280 m. The distance between pipes is 0.25 m. Special requirements for more snow melting are met with less distance between pipes, 0.2 m. Tubes are embedded in special sand underneath a stone slab. The sand has higher heat conductivity than ordinary sand. In roads with heavy traffic load from buses, pipes are laid in the asphalt layer (Ragnarsson, 1997).

3.2 Japan

In the Gaia snow-melting system, the Downhole Coaxial Heat Exchanger (DCHE) is used for extracting heat from the earth. The heat exchanger uses a thermally insulated inner pipe and reverse circulation, i.e. the cold fluid flows down the annulus and warmer fluid flows up through the inner pipe, for efficient heat extraction.

The first Gaia snow-melting system was installed in Ninohe, Iwate Prefecture, a city 500 km north of Tokyo, and has been operating successfully since December 1995. The average low temperature for the month of January is -7.1˚C. Annual snowfall in the winter of 1995/96 was 2.9 m. The Gaia system was installed at the downhill section of a curved road with a 9% gradient, in order to prevent accidents by skidding and sliding vehicles in winter. The area covered by the snow-melting system is 4 m wide and 65 m long, covering a total area of 266 m^2. Heating pipes were embedded in the asphalt concrete pavement at 10 cm depth and 20 cm intervals. The material under the road consists of sandy rock and the undisturbed temperature is 22.5˚C at a depth of 150 m. Three DCHEs, each 8.9 cm in outer diameter and 150 m deep, and a 15 kW (electric) heat pump were used. In winter, heat extracted from the earth by DCHEs is transferred to a heat pump. After the temperature is increased by the heat pump, the thermal energy is transmitted to a heating fluid circulating through a network of heating pipes, which melt the snow (as shown in Figure 3). Antifreeze is used as both a heat extraction medium and a heating medium.

In summer, solar heat raises the temperature of the road in which the heating pipes are embedded, up to 30-50˚C. The solar heat is recovered from the road and stored in the earth by connecting the DCHEs and heating pipes directly, and by circulating the fluid in this loop. The warmed fluid in the heating pipes flows into the DCHEs and warms up the surrounding earth, thereby storing the solar heat in the earth. In this way, summertime solar heat is stored for use in the winter for melting snow. This heat storage operation is controlled automatically by referring to the temperatures of the fluid at the bottom, inlet and outlet of DCHEs as well as the road. The Gaia system in Ninohe City demonstrated a reduction of 84% in annual energy consumption compared with systems in Ninohe which use electric heating cables (Morita and Tago, 2000).

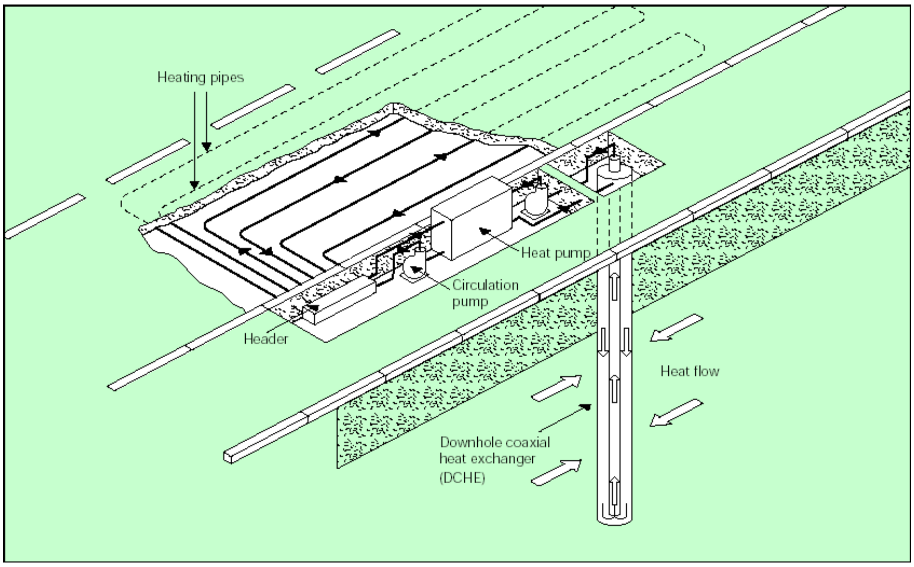

FIGURE 3: Layout of a snow melting system in Japan (Caddet, 2003)

3.3 United States

The oldest geothermal pavement snow-melting system was installed in Klamath Falls, Oregon, in 1948 by the Oregon Highway Department. This is a 450' long section of Esplanade Street approaching a traffic signal on an 8% grade. The grid consisted of ¾" diameter iron pipes placed 3" below the surface of the concrete pavement on 18" centres. The grid system was connected to a geothermal well with the heat transferred through a downhole heat exchanger to a 50-50 ethylene glycol-water solution that circulated at 50 gpm.

By 1997, after almost 50 years of operation, the system failed due to leaks in the grid caused by external corrosion. In the fall of 1998, a contract was issued to reconstruct the bridge deck and highway pavement along with replacing the grid heating system. The top layer of concrete on the bridge deck was removed by hydroblasting and the roadway pavement was entirely removed, and a new crushed rock base added. A ¾" cross-linked polyethylene tubing (PEX) was then used for the grid section, placed in a double overlap pattern resulting in line spacing from 14 to 16" on centre (Figure 4). The PEX pipe was attached to the reinforcing steel within the concrete pavement, providing a cover of about 3" over the pipe within

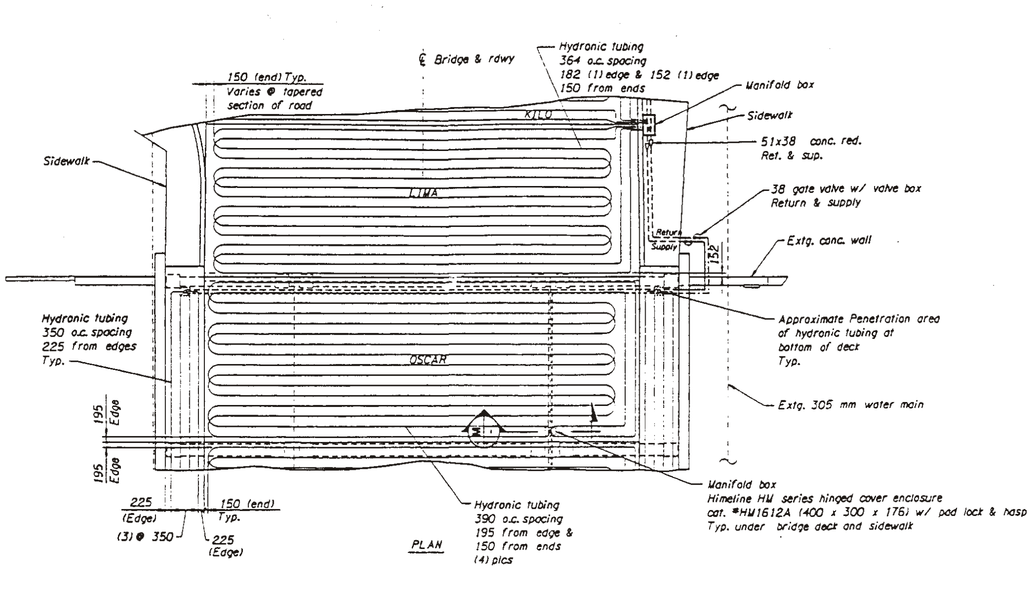

FIGURE 4: Details of a snow melting installation in Klamath Falls, USA (Lund, 1999)

the 7" pavement section. The header pipe, placed along the edge of the roadway consisted of 1.25-2.5" insulated black iron pipe, which in turn was connected to the downhole heat exchanger. The header pipe had brass manifolds placed at about 40' intervals in concrete boxes, to allow for four supply and return PEX pipes to be attached (Lund, 1999).

4. GOLENIOW AIRPORT

4.1 General information

Goleniow airport is located in NW-Poland, near the Baltic Sea and the German border (Figure 5). It is less than 40 km north of Szczecin, at the intersection of two major roads, the A-3 motorway connecting Scandinavia with the southern part of Europe, and the A-6 motorway along the Polish coast of the Baltic Sea to Gdansk. Goleniow town is 5 km from Goleniow airport. The airport, together with the town, forms an important communication junction with railway services to Szczecin, Świnoujście, Kamien Pomorski and Kołobrzeg.

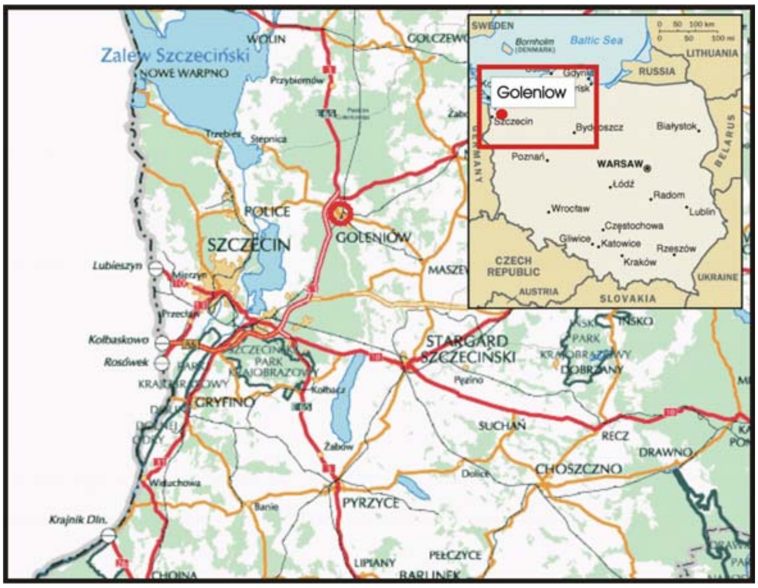

FIGURE 5: Location of Goleniow town and airport

Goleniow town has great prospects for growing tourism, recreation (e.g. sailing, hunting and bird-watching) and co-operation for investment in many economic fields. This is one of the biggest towns of the Szczecin Province and lies in the heart of the Goleniowska Forest on the Ina River. The town has almost 22,500 inhabitants. The landscape of Goleniovian region was moulded by the glacial action, which created the main forms of today's morphological image of the area. It brought a huge mass of different mixed materials: various kinds of gravel, sand, clay, calcareous components and rocks. In the western part of the Goleniowska Plain, there are alleged terraces along the Dąbie Lake and the Oder River, situated at levels from 0.5 m to 25 m (Goleniow, 2002).

The Goleniow airport was a military airport until 1998. It served about 45,000 passengers in 1998. Its location is very advantageous and the management of the airport plans to serve parts of northern Germany. It has at its disposal a new terminal with a capacity for 200,000 passengers per year; runway dimensions are 2,500 x 60 m^2 (Airport, 2002). Investment plans call for the construction of a new automatic meteorological information system, a new air traffic control tower, modernization of the pavement, installation of a secondary surveillance radar system and more (US Department of Commerce, 2002).

4.2 Geothermal conditions

The Szczecin region with Goleniow town is a part of the Szczecin-Lodz Synclinorium which is considered to be one of the best water sources in Poland (Figure 6). The synclinorium area is about 67,000 km^2. The estimated recoverable volume of sub-artesian and artesian geothermal water is 2854 km^3 amounting to

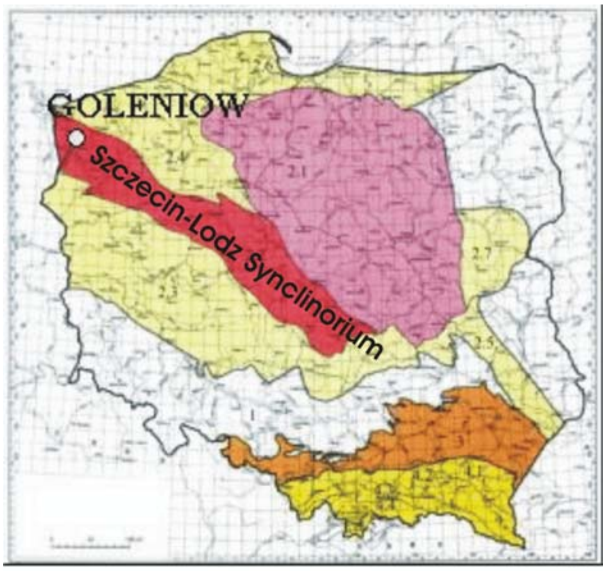

FIGURE 6: Map of the geothermal regions in Poland with Szczecin-Lodz Synclinorium

18,812 million toe (tons oil equivalent). On average, it amounts to 42 million m^3 of water or 280,800 toe/km^2. Geothermal water at 40-90˚C is found in this area at 600-2200 m depth. These strata consist of gray sandstone with high porosity and thickness up to 200 m (Meyer and Kozlowski, 1995).

The town Pyrzyce is located close to Goleniow town, on the same synclinorium. Since 1997 a modern 50 MWt geothermal heat plant has been in operation in Pyrzyce. For the base load it uses 340 m^3/h of 68˚C geothermal fluid as well as two 10 MWt heat pumps that heat the fluid to the 90˚C required by the customers. As a peak heat source, a 22 MWt gas boiler is used. Water is extracted from a depth of about 1500-1650 m, from rock deposits of earlier Jurassic by means of two doublets. Extracted water parameters are: temperature 68˚C, salinity 120 g/l, density 1.062 kg/m^3. The task of the geothermal heat plant in Pyrzyce was to replace 68 local coal boilers in use, and increase the energy efficiency.

From the existing wells in the Szczecin Basin, the following approximate data for the geothermal fluid accessible in Goleniow can be estimated:

- Volumetric flow rate of geothermal water: $\dot{V}_{g\,\min} = 50$ m^3/h; $\dot{V}_{g\,\max} = 150$ m^3/h;
- Temperature of geothermal water: $T_{g\,min} = 40$˚C ; $T_{g\,max} = 100$˚C;
- Average density of geothermal water: $\rho_g = 1.08$ kg / m^3
- Average specific heat of geothermal water at const. pressure: $c_g = 3.81$ kJ/kgK

On the basis of the operation of the Pyrzyce geothermal heat plant and chemical analysis, it is known that the geothermal water from Szczecin Basin is very aggressive. The reason for this is the high mineralization - a high salt content. Also, it is worthwhile to note that there could be danger of microbiological corrosion. Salt and hot geothermal water can be a good environment for bacterial growth. The bacteria, if injected into the bedrock, can silt it up (stick up) making flow impossible.

4.3 Climatic data

The geographical location and surface features are the two most important factors determining the climate of Poland (EPM, 2002). Due to the location of Poland in the temperate latitudes, Arctic air masses dominate. The share of maritime air is greater than of continental air (38-46%), because of the more common, western (oceanic) circulation of air. The terrain pattern forming along the parallels of latitude also aids the flow of humid air masses from the ocean to Poland (75% of atmospheric fronts come from the West).

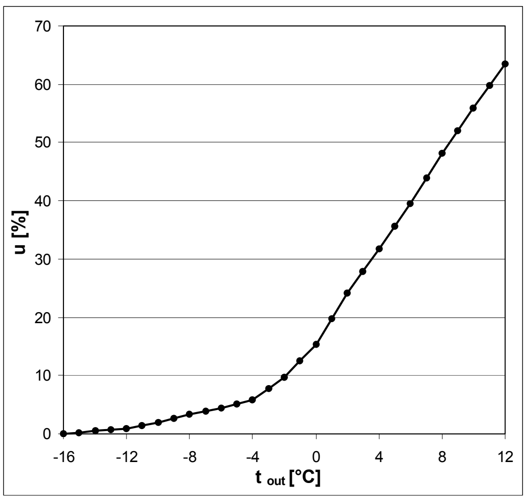

FIGURE 7: Temperature duration curve for the Goleniow area

Because detailed information about weather conditions at Goleniow is not available the estimated temperature duration curve shown in Figure 7 was used. The data is taken from heat load calculation for a geothermal heat plant located near the Goleniow area (Zwarycz, 2000). Table 1 shows average temperatures from the Goleniow area used in the heat demand calculations for snow melting (Airport, 2002). Further assumptions are: relative humidity 80%, average rate of snowfall 2 mm/h and average wind velocity 3.4 m/s.

TABLE 1: Maximum and minimum daily average temperatures (˚C) used for Goleniow area

Months	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**June**	**July**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**
Maximum	-0.6	1.5	6.6	12.2	18.7	20.5	22	22	17.8	13	6.7	3.1
Minimum	-4.8	-4.8	-0.8	1.8	6.9	10.2	11.9	11.7	8.8	5.4	0.9	-2

5. HEAT DEMAND ANALYSIS

5.1 Airport buildings

Heat demand for the airport buildings was calculated based on Polish Standards for public buildings ($U \leq 0.3 W/m^2K$), which define heat demand with the parameters below:

- For geothermal water central heating: 30 W/m^2
- For municipal tap water: 10 W/m^2
- For ventilation/climatization: 20 W/m^2

Table 2 shows heat demand calculated for the airport buildings. The assumed heat demands for tap water and ventilation systems, as one cumulated value, were also taken.

TABLE 2: Heat demand for the Goleniow airport buildings

Buildings		**Heat demand [kW]**				
		Heating season			**Rest of year**	
	Area [m^2]	**Central heating**	**Tap water**	**Ventila-tion**	**Tap water**	**Ventila-tion**
Airport terminal building	2400	72	24	48	24	48
Technical hinterland building	1700	51	17	34	17	34
Fire brigade building	400	12	4	8	4	8
Sum	4500	135	45	90	45	90

5.2 Snow melting system

Heating requirements for snow melting depend firstly on atmospheric factors and secondly on the demand for effectiveness. Atmospheric factors are four, rate of snowfall, air temperature, relative humidity and wind velocity. Effectiveness of snow melting or melting rate means how quickly snow is melted. Heat output and melting rate are dependent on the distance between pipes, the depth of pipes in the ground, fluid flow and the drop in temperature of the fluid in pipes, conductivity of the slab, etc.

The extreme demand for snow melting is to have a constant snow-free surface. This demand is, in most cases, unrealistic because of the operating cost. Using insignificant heat output, and, therefore, an inadequate melting rate, results in melting taking a long time. The time depends on the duration, frequency and quantity of snowfalls. Heat requirements can be great even when there is no snowfall and the earth is free of snow, especially if wind is blowing hard. During freezing periods, it is necessary to keep the surface temperature above freezing point to avoid icing. When snow starts to fall, a sufficient heat resource must be in the soil and the surface layer. The heat melts the snow when it starts to fall and counterbalances a rather slow reaction of the snow melting system. Determining a sufficient heat resource in a snow melting area is dependent on the expected weather condition. If surface temperature is low, snow piles up when snowfall is intense. It takes a long time, one or two months, to build up a resource in cold earth. It can be expensive to preserve an abundant heat capacity in the earth (Ragnarsson, 1997).

Heat is transferred to the snow to change the temperature to the melting point, 0˚C. Heat is required to melt the snow, the heat of fusion. Heat is also required for evaporation. When the surface is free of snow, heat is transformed from the surface to the atmosphere by convection and radiation. This heat loss is dependent on outdoor temperature and wind speed. While snowflakes are being warmed and before they are melted, they act as tiny blankets or insulation. The effect of this insulation can be great. The ratio of the uncovered, or free area A_f to the total area A_t is the free area ratio A_r (general use of variables is explained in Nomenclature).

$$A_r = \frac{A_f}{A_t} \tag{1}$$

To maintain $A_r = 1$, the system must melt the snow so rapidly that accumulation is zero. This is impossible, but for purposes of design, it is permissible to assume $A_r = 1$ as a maximum. For $A_r = 0$, the surface must be completely covered with snow to a depth sufficient to prevent evaporation and heat transfer losses. Research on the insulating effects of snow indicates that, in practice, the free area ratio can be adequately expressed by one of three values: 0, 0.5, and 1.

Chapman and Katunich (1956) derived the general equation for the required slab heat output q_0:

$$q_0 = q_s + q_m + A_r\,(q_e + q_h) \tag{2}$$

The sensible heat q_s to bring the snow to 0˚C is

$$q_s = \frac{sc_p\rho(0-t_a)}{c1} \tag{3}$$

The heat of fusion q_m to melt the snow is

$$q_m = \frac{sh_f\rho}{c1} \tag{4}$$

The heat of evaporation q_e (mass transfer) is

$$q_e = h_{fg}\,(0.005\,V + 0.022)\,(0.625 - p_{av}) \tag{5}$$

The heat transfer q_h (convection and radiation) is

$$q_h = 64.74\,(0.0125\,V + 0.055)\,(t_f - t_a) \tag{6}$$

The solution to Equation 2, however, requires the simultaneous consideration of all four climatic factors, e.g. wind speed, air temperature, relative humidity and rate of snowfall. Annual averages or maximums for the climatic factors should not be used because there is no assurance that they will ever occur simultaneously. Therefore, it is necessary to perform a frequency analysis of the solutions to the equation for all occurrences of snowfall over several years. During operation when the outdoor temperature is above the freezing point, the snow melting system is operated with basic heat output rate, $q_{melting}$. The basic rate is sometimes determined by taking into consideration the output needed to melt snow in freezing periods in the warmest month of operation. In freezing temperatures, 0˚C and below, without snowfall, the system may be idling, which means that some heat, q_{idling}, is supplied to the slab to prevent icing or to secure the start of melting as snow starts to fall. Hot water is injected if house water is not sufficient. Light snowfalls can normally be handled by the idling slab. When snowfall increases, more hot water is added. The effect of snow melting is dependent on the quantum of water and the maximum output the snow melting system is matched for. The maximum output is regulated by the control system and is relative to the demand for snow melting, but limited by acceptable operating cost.

Snow-melting installations are classified as Class I, II, or III (ASHRAE, 1995). Snow-melting systems are generally classified, by urgency of melting, as follows:

- Class I (minimum): Residential walks or driveways; interplant ways or paths;
- Class II (moderate): Commercial sidewalks and driveways; steps of hospitals;
- Class III (maximum): Toll plazas of highways and bridges; aprons and loading areas of airports; hospital emergency entrances.

The difference between a Class I system and a Class II system is the required ability of each system to melt snow. For example, the accumulation of 25 mm of snow in an hour during a heavy storm might not be objectionable for a residential system. On the other hand, a store manager might consider the system inadequate if 13 mm of snow accumulated on the sidewalk in front of the store. In a residential system where initial cost must be kept at a minimum, more frequent snow accumulations may have to be accepted.

Back and edge losses, which increase the required slab output found by Equation 2, must also be considered. These losses can vary from 4 to 50%, depending on factors such as pavement construction, operating temperature, ground temperature, or back exposure. The snow melting pipes are embedded in sand. Sand is a very insulating material, especially when dry. It is, therefore, important to use special sand with good conductivity above the pipes (Ragnarsson, 1997).

TABLE 3: Heat demand for the Goleniow airport snow melting system

	q_0 [W/m^2] class III		**t_m[°C] class III**		**Heat demand [kW] class III**		**Area [m^2]**
	Melting	**Idling**	**Melting**	**Idling**	**Melting**	**Idling**	
Method 1	499	30	44	3	2495	150	5000
Method 2	436	30	39	3	2180	150	5000
Method 3	200	100	40	23	1000	500	5000

For reasons of comparison, three different methods for calculating the heat demand of a snow melting system have been applied. The first used Equations 2 to 6. The second, based on the ASHRAE Handbook, used meteorological data for American cities. In the third method, the Icelandic experience was applied. Table 3 shows calculated required heat load q_0, mean water temperature t_m as well as heat demand for class III installation, assuming relative humidity 80%, average rate of snowfall 2 mm/h and average wind velocity 3.4 m/s. Because of so many assumptions due to not enough (exact) climatic data for Goleniow airport area, calculations by the first method are very approximate. However, the results are comparable to the results of the second method based on experience from the United States. But according to the Icelandic experience, they are too high and result in very high operating costs.

Figure 8 shows calculated values for snow thickness on the ground as a function of time. The assumed snowfall is 3 mm/h and curves for three different values of heat load are

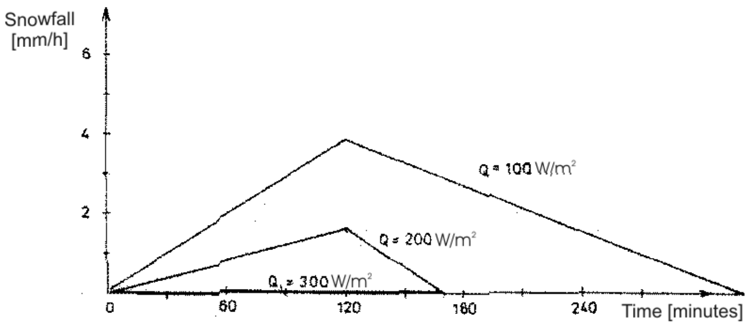

FIGURE 8: Snow thickness on the ground as a function of time (Fjarhitun, 1982)

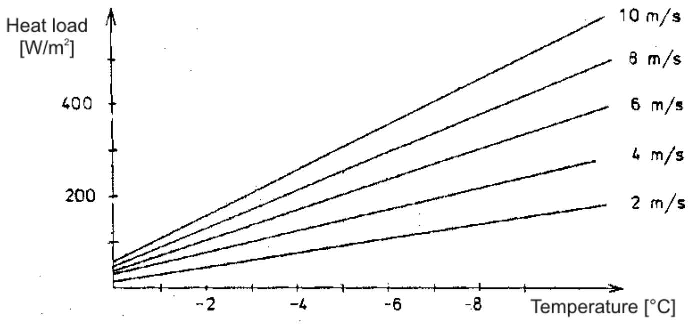

FIGURE 9: Heat load as a function of air temperature and wind velocity (Fjarhitun, 1982)

shown. The figure shows that for 200 W/m^2 heat load the time of melting is about 3 hours and, after 2 hours, 1.6 mm water equivalent of snow stays on the surface. Figure 9 shows that for 3.5 m/s wind velocity and average air temperature -2˚C, the heat load 200 W/m^2 is higher than necessary. The excess heat can be used for back and edge losses. For this reason, values based on the Icelandic literature for a maximum heat load for melting equal to 200 W/m^2 and an idling load equal to 100 W/m^2, were accepted.

6. DESIGN OF A SNOW MELTING SYSTEM

The snow melting system considered here is designed for operation between -16˚C and +3˚C outdoor temperatures. The maximum heat load for melting is assumed to be 200 W/m^2 and supply-return water parameters 55/25˚C. In freezing temperatures (0˚C and below) without snowfall, the system may be idling, which means that some heat is supplied to the slab so that melting begins immediately when snow starts to fall. The idling load is equal to 100 W/m^2 with supply-return water 30/15˚C. The constant water flow rate for both melting and idling operation is 5.7 l/m^{2}h.

A hydronic system design includes selection of the following components:

1. Heat transfer fluid;
2. Pipe system;
3. Fluid heater;
4. Pump(s) to circulate the fluid; and
5. Controls.

In concrete pavement, thermal stress is also a design consideration.

6.1 Heat transfer fluid

There are a variety of fluids suitable for transferring the heat from the fluid heater to the pavement, including brine, oils, and glycol-water. Freeze protection is essential because most systems will not be operated continuously in subfreezing weather. Without freeze protection, power loss or pump failure could cause freeze damage to the pipe system and pavement.

Brine is the least costly heat transfer fluid, but it has a lower specific heat than glycol. The use of brine may be discouraged because of the cost of heating equipment that is resistant to its corrosive potential. Glycols (ethylene and propylene) are used most often in snow-melting systems because of their moderate cost, high specific heat, and low viscosity; ease of corrosion control is another advantage. Automotive glycols containing silicates are not recommended because they can cause fouling, pump seal wear, fluid gelation, and reduced heat transfer. The pipe system should be designed for a periodic addition of an inhibitor. Glycols should be tested annually to determine any change in reserve alkalinity and freeze protection. Only inhibitors obtained from manufacturers of the glycol should be added. Heat exchanger surfaces should be kept below 140˚C, which corresponds to about 280 kPa (gage) steam. Temperatures above 150˚C accelerate the deterioration of the inhibitors (ASHRAE, 1995).

6.2 Pavement construction

Either concrete or asphalt pavement may be used for snow-melting systems. The thermal conductivity of asphalt is less than that of concrete; pipe spacing and temperatures are, thus, different. Hot asphalt may be damaging to plastic snow-melting systems unless adequate precautions are taken.

Concrete slabs containing hydronic snow-melting apparatus must be designed and constructed with a subbase, expansion-contraction joints, reinforcement, and drainage to prevent slab cracking; otherwise, crack-induced shearing or tensile forces would break the pipe. The pipe must not run through expansion-contraction joints, keyed construction joints, or control joints (dummy grooves); however, the pipe may be run under 3 mm score marks (block and other patterns). Control joints must be placed wherever the slab changes size or incline. The maximum distance between control joints for ground-supported slabs should be less than 4.6 m, and the length should be no greater than twice the width, except for ribbon driveways or sidewalks. In ground-supported slabs, most cracking occurs during the early cure. Depending on the amount of water used in a concrete mix, shrinkage during cure may be up to 60 mm per 100 m. If the slab is more than 4.6 m long, the concrete does not have sufficient strength to overcome friction between it and the ground while shrinking during the cure period. If the slabs are poured in two separate layers, the top layer, which contains the snow-melting apparatus, does not usually contribute toward total slab strength; therefore, the lower layer must be designed to provide the total strength. The concrete mix of the top layer should give maximum weather-ability. The compressive strength should be 28-34 MPa; recommended slump is 75 mm maximum and 50 mm minimum.

The pipe may be placed in contact with an existing sound pavement and then covered. Pipe should not be placed over existing expansion-contraction, control, or construction joints. The finest grade of asphalt is best for the top course; stone diameter should not exceed 10 mm. A moisture barrier should be placed between any insulation and the fill. The joints in the barrier should be mopped and the fill made smooth enough to eliminate holes or gaps for moisture transfer. Also, the edges of the barrier should be flashed to the surface of the pavement to seal the ends.

Snow-melting systems should have good surface drainage. When the ambient air temperature is 0˚C or below, runoff from melting snow freezes immediately upon leaving the heated area. Any water that is able to get under the pavement also freezes when the system is de-energized, causing extreme frost heaving. Runoff water should be piped away in drains that are heated or below the frost line. The area to be protected by the snow-melting system must first be measured and planned. For total snow removal, hydronic heat must cover the entire area. In larger installations, it may be desirable to melt snow and ice from only the most frequently used areas. Planning for separate circuits should be considered so that areas within the system can be heated individually, as required.

Where snow-melting apparatus must be run around obstacles, the pipe spacing should be uniformly reduced. Because some drifting will occur adjacent to walls or vertical surfaces, extra heat should be added in these areas, if possible also in the vertical surface. Drainage flowing through the area expected to be drifted tends to wash away some snow (ASHRAE, 1995).

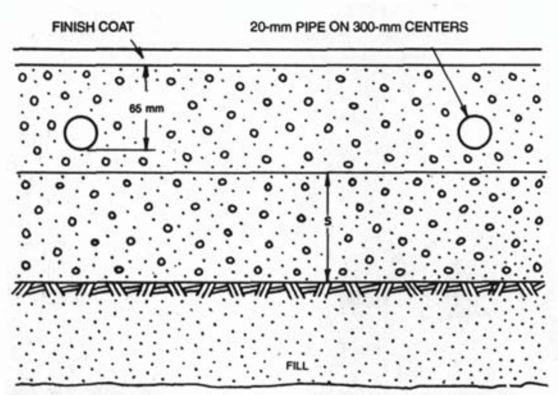

FIGURE 10: Typical pipe layout in a snow melting system

6.3 Pipe system

Pipe systems may be metal or plastic. Steel, iron, and copper pipes have long been used. Steel and iron may corrode rapidly if the pipe is not shielded by a coating and/or cathodic protection. As shown in Figure 10, it is satisfactory to use 20 mm pipe or tube on 300 mm centres as a standard coil. If pumping loads require reduced friction, the pipe size can be increased to 25 mm, but the pavement depth must be increased accordingly. The piping should be supported by a minimum of 50 mm of concrete above and below. This requires a 130 mm pavement for 20 mm pipe and 140 mm for 25 mm pipe.

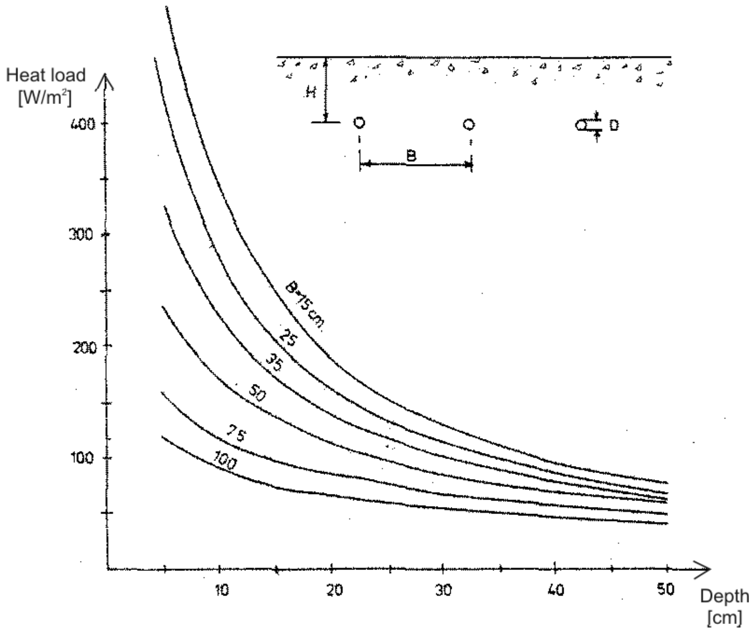

FIGURE 11: Snow melting heat load as a function of pipe depth and distance between them (Fjarhitun, 1982)

Figure 11 shows snow melting heat load as a function of pipe depth and distance between them. It is assumed that the supply and return water temperatures are 35°C and 15°C, respectively, the pipe diameter is 25 mm and the surface temperature 0°C. It is common to put pipes 8 to 10 cm below the surface and to have 25 cm distance between them, then the head load varies between 320 and 270 W/m^2. If the distance between pipes is 40 cm and they are at 10 cm depth the heat load is 220 W/m^2, but then uneven heat transfer at the surface can be expected (Fjarhitun, 1982).

Plastic pipes (polyethylene, polybutylene) are popular due to low material cost, low installation cost, and corrosion resistance. Necessary considerations when using plastic pipes include stress crack resistance, temperature limitations and thermal conductivity. Heat transfer oils should not be used with plastic pipe. Polyethylene and polybutylene pipe is furnished in coils. The smaller sized pipe can be bent to form a variety of heating panel designs without elbows or joints. Mechanical compression connections can be used to connect the heating panel pipe to the larger supply and return piping leading to the pump and fluid heater. Plastic pipe may be fused using the appropriate fittings and fusion equipment. Fusion joining eliminates metallic components and, thus, the possibility of corrosion in the pipe system; it does, however, require considerable installation training. When plastic pipe is used, the system must be designed so that the fluid temperature required will not damage the pipe. If a design requires a temperature above the tolerance of plastic pipe, the heat output will never meet design requirements. A logical solution is to decrease the pipe spacing.

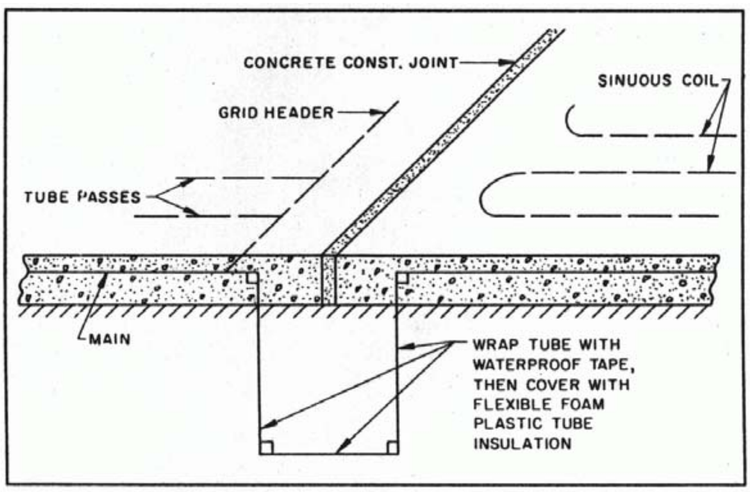

FIGURE 12: Piping details for concrete construction joints

It is a good design practice to avoid passing any embedded piping through a concrete expansion joint; otherwise, the pipe may be stressed and possibly ruptured. A method of protecting piping, that must pass through a concrete expansion joint, from stress under normal conditions is shown in Figure 12. Because the introduction of air causes deterioration of the antifreeze, the pipe system should not be vented to the atmosphere. It should be divided into smaller zones to facilitate and allow isolation when service is necessary. After pipe installation, but before pavement installation, all piping should be air tested to about 700 kPa (gage). This pressure should be maintained until all welds and connections have been checked for leaks. Testing should not be done with water for the following reasons (ASHRAE, 1995):

1. Small leaks may not be observed during pavement installation;
2. Water leaks may damage the concrete during installation;
3. The system may freeze before antifreeze is added; and
4. It is difficult to add antifreeze when the system is filled with water.

6.4 Control

Snow-melting systems can be controlled either manually or automatically. Manual operation is strictly by two-position control; an operator must activate and deactivate the system when snow falls. If the system is not turned off after snowfall, operating cost is increased. If the snow-melting system is not turned on until snow starts falling, it may not melt snow effectively for several hours, giving additional snowfall a chance to accumulate and increasing the time needed to melt the area. Automatic controls provide satisfactory operation because they turn on the system when light snow starts, allowing adequate warm-up before heavy snowfall develops. Operating costs are reduced with automatic turn-off.

Snow detectors monitor precipitation and temperature. They allow operation only when snow is present and may incorporate a delay-off timer. Snow detectors located in the heated area activate the snow-melting system when precipitation (snow) occurs at a temperature below the preset pavement temperature (usually 4˚C). Another type of snow detector is mounted above ground, adjacent to the heated area, without cutting into the existing system; however, it does not detect tracked or drifting snow. Both types of sensors should be located so that they are not affected by overhangs, trees, blown snow, or other local conditions.

To limit energy waste during normal and light snow conditions, it is common to include a remote temperature sensor installed midway between two pipes or cables in the pavement; the setting is adjusted to between 5 and 15˚C. Thus, during mild weather snow conditions, the system is automatically modulated or cycled on and off to keep the pavement temperature at the sensor at set point. The control system may include an outdoor thermostat that turns the system off when the outdoor ambient temperature rises above 2-5˚C, as automatic protection against accidental operation in summer. For optimum operating convenience and minimum operating cost, all of the aforementioned controls should be incorporated in the snow-melting system (ASHRAE, 1995).

6.5 Thermal stress

In general, thermal stresses will cause no problems if the following installation and operation rules are observed:

- Minimize the temperature difference between fluid and the slab surface by maintaining close pipe spacing (Figure 10), a low temperature differential in the fluid (less than 10˚C) and continuous operation (if economically feasible);
- Install pipe within about 50 mm of the surface;
- Use reinforced steel designed for thermal stress if high structural loads are expected.

Thermal shock to the pavement may occur if heated fluid is introduced from a large source of residual heat such as a storage tank, a large pipe system, or another snow melting area. The pavement should be brought up to the temperature by maintaining the fluid temperature differential at less than 10˚C (ASHRAE, 1995).

7. HEATING AND SNOW MELTING NETWORK DESIGN

The purpose of this work was to make the thermal and fluid-flow calculations for a proposed geothermal heat plant at Goleniow Airport in NW-Poland. The geothermal water is utilized as a heat source supplemented with other energy sources, a heat pump (direct and indirect evaporator connection), and a gas boiler. The heating plant supplies heat to a distribution network, which itself consists of parallel-connected heat users featuring different operational requirements.

7.1 Characteristics and assumptions

The main components of the heating system are production–reinjecting geothermal wells, a counter-flow geothermal-heat exchanger, a heat pump (electrically driven) and a gas boiler for peak loads. Three different types of setup are considered:

System 1: Heat is transferred to the distribution network by a heat pump with a closed loop, geothermally heated water on the evaporator side (heat source) and the distribution network water on the condenser side (heat supply).

System 2: Heat is transferred to the distribution network partly by a heat pump working between the return and supply flows and partly by heating the return water after it passes the heat pump evaporator, in the geothermal heat exchanger. A peak gas boiler is located in the radiator heating loop.

System 3: Heat is transferred to the distribution network by the geothermal heat exchanger and a peak gas boiler connected in series. No heat pump is used in this case.

That heat plant is interconnected by means of pipelines and bypasses with three types of users. They have different operational requirements and are connected in parallel:

- The first type of user is a radiator heating system. Two scenarios are considered, in the first, the temperature of the supply water varies as a function of the outdoor temperature; in the second, the supply water temperature is kept constant. It was assumed that during the heating season the inside temperature is kept at +20˚C and the outdoor temperature varies between -16˚C and +12˚C.
- The second type of user is a tap water / ventilation system, running the network water at constant temperature, 50˚C, year round.
- The third type of user is a snow melting system working in a -16˚C to +3˚C outdoor temperature interval, on variable heat demand values for melting (55/25˚C) and for idling (30/15˚C).

For calculations, the following data were assumed:

- Max. volumetric flow rate of geothermal water: $\dot{V}_{g\,\max} = 50\ \mathrm{m^3/h} = 15\ \mathrm{kg/s}$
- Max. temperature of geothermal water
  $T_{g\,max}$ = 40˚C (radiator parameters are 65/50; 65/40; 65/35 or 90/70˚C)
  $T_{g\,max}$ = 60˚C (radiator parameters are 90/70˚C)
- Mean mass density of geothermal water: $\rho_g = 1.08\ \mathrm{kg/m^3}$
- Mean specific heat of geothermal water: $c_g = 3.81$ kJ/kgK
- Max. heat demand of radiator heating: $\dot{Q}_{RH\,\max} = 135$ kW
- Max. heat demand of tap water and ventilation: $\dot{Q}_{TV\,\max} = 135$ kW
- Max. heat demand of snow melting system

  Melting time: $\dot{Q}_{SM\,\max} = 1000$ kW
  Idling time: $\dot{Q}_{SM\,\max} = 500$ kW

Total demand for the thermal power, $\dot{Q}_c$, which was predicted for the design outdoor temperature, t_{out} = -16˚C , was defined by the following relationship:

$$\dot{Q}_C = \dot{Q}_{RH\,\max} + \dot{Q}_{TV\,\max} + \dot{Q}_{SM\,\max} \tag{7}$$

Data for operating time needed for different parts of the system is estimated as shown in Table 4. The heating season is a 232-days period, with 15 snowfall days. However, the snow melting system is in operation for 102 days until t_{out} = 3˚C. Above that temperature, heat is only required for radiators and the tap water / ventilation system. When t_{out} = 12˚C, the heating season is finished and the rest of year (133 days) the system works only for tap water / ventilation demands.

TABLE 4: Operating time data

	Time	
	Days	**Hours**
No snowfall		
$t_{out} > 0$°C	170	4080
$t_{out} \leq 0$°C	47	1128
Snowfall	15	360
Sum for heating season	232	5568
Rest of the year	133	3192
Sum	**365**	**8760**

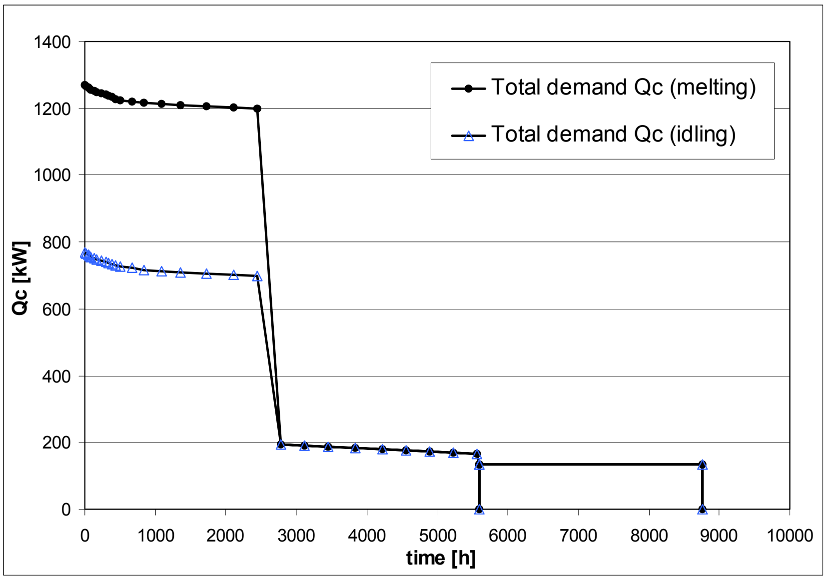

FIGURE 13: The heat demand curve used for Goleniow airport

Because the heat demand curve for Goleniow is not available, a heat demand curve is demonstrated in Figure 13, with the total heat load for the heating season (melting and idling time) and then the rest of the year. This data were taken from similar heat load calculations made for a geothermal heat plant located near the Goleniow area (Zwarycz, 2000).

The total heat energy Q_c [MWh] for one year was estimated from the following formula:

$$Q_c = \sum \Delta Q_i \tag{8}$$

where ΔQ_i [MWh] is heat energy for the elementary time interval, $\Delta\tau$, calculated as follows:

$$\Delta Q_i = \frac{\dot{Q}_1 + \dot{Q}_2}{2} \Delta\tau \tag{9}$$

Geothermal heat energy Q_g, heat pump energy Q_{out} and gas boiler energy Q_{pb} were calculated similarly. Heat demand values for radiator heating, tap water / ventilation and snow melting systems, as well as time occurrences for each outdoor temperature are shown in Appendix I.

7.2 Radiator heating

Heat demand for radiator heating $\dot{Q}_{RH}$ was calculated from the formula (Valdimarsson, 1993):

$$\dot{Q}_{RH} = \dot{Q}_{RH\,max} \frac{20 - t}{20 - t_{out}} \tag{10}$$

And return water temperature from radiators was estimated from the equation:

$$\frac{\dot{Q}_{RH}}{\dot{Q}_{RH\,max}} = \left(\frac{\Delta T_m}{\Delta T_{m0}} \right)^{4/3} \tag{11}$$

Logarithmic temperature difference ΔT_m is defined as

$$\Delta T_m = \frac{(T_{RHs} - T_{RHr})}{\ln\left(\frac{T_{RHs} - T_i}{T_{RHr} - T_i} \right)} \tag{12}$$

The mass flow of the network water for the radiator heating circuit, $\dot{m}_{RH}$, was estimated from the following formula:

$$\dot{m}_{RH} = \frac{\dot{Q}_{RH}}{c_s (T_{RHs} - T_{RHr})} \tag{13}$$

The mass flow of the network water for the tap water / ventilation and snow melting system was calculated in a similar way.

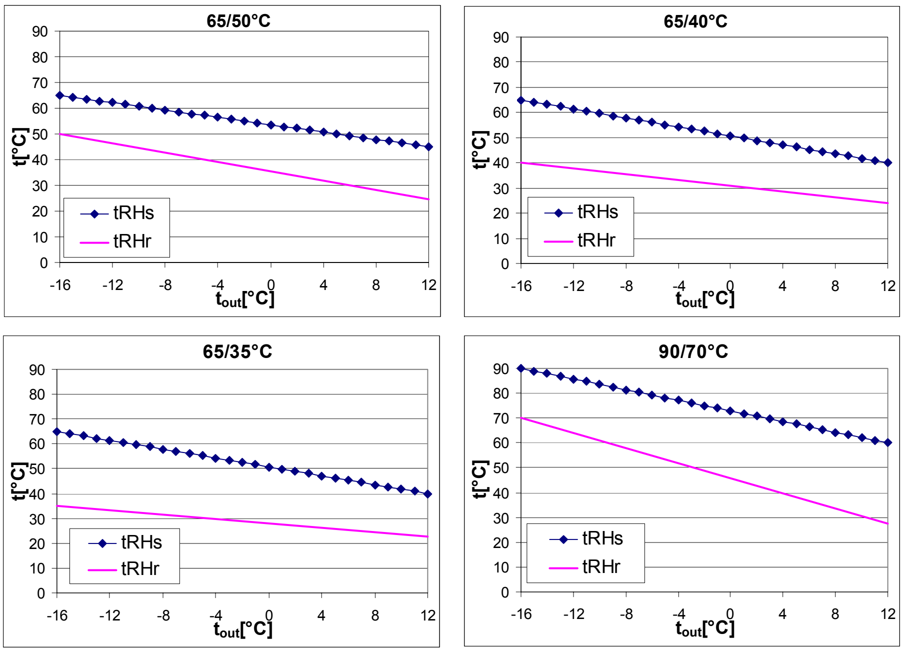

FIGURE 14: Radiator supply and return water temperature variants, first scenario

In the radiator heating system calculations for two different working scenarios were performed. In the first one the temperatures of the supply water is a function of the outdoor temperature. In this case four different sets of supply and return water temperatures at design conditions were considered (65/50, 65/40, 65/35 and 90/70˚C) as shown in Figure 14. In the second scenario, the supply water temperatures do not depend on the outdoor temperature but are considered to be constant. In this case three different sets of supply and return water temperatures at design conditions were considered (65/50, 65/40 and 65/35˚C) as shown in Figure 15. The return water temperature from radiators at different outdoor temperatures is calculated according to the $\Delta T_m / \Delta T_{m0}$ ratio from the heat balance Equation 11.

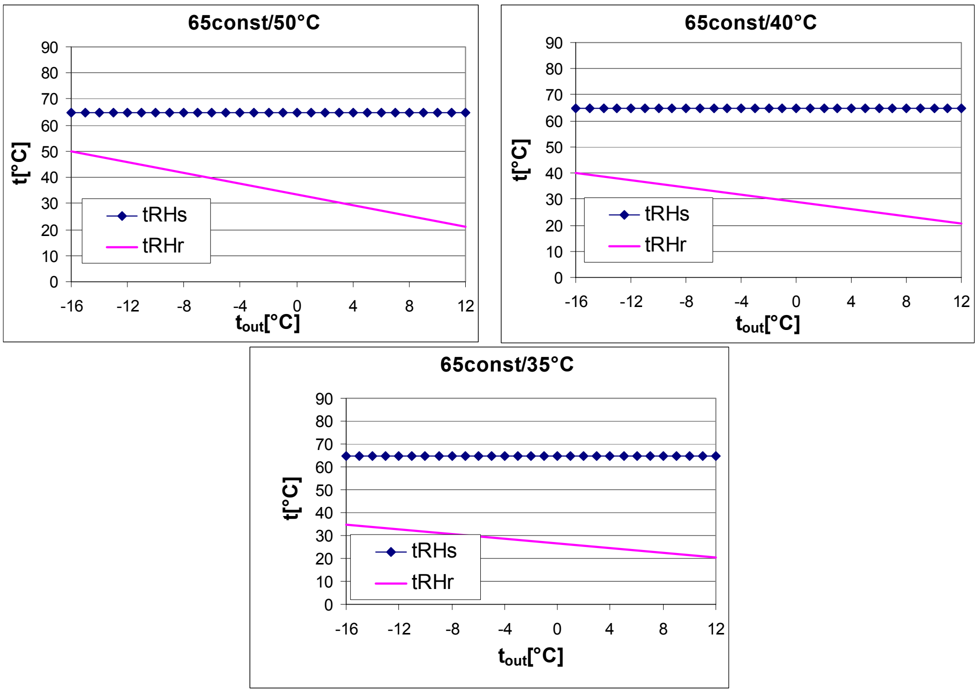

FIGURE 15: Radiator supply and return water temperature variants, second scenario

7.3 Geothermal heat exchanger

For a geothermal-heat exchanger, we assume a multi-plate counter-flow exchanger where the heat capacities for both media are equal to one another, and the temperature difference between them is constant. It was assumed that $\Delta T = 5$ K. In such a case the mean temperature difference equals $\Delta \overline{T}$ = 5 K. The heat flux, the dispersal of the geothermal water via network water flowing through the exchanger, is defined by the following relationship:

$$\dot{Q}_g = \dot{W}_g \, \Delta T_g = \dot{W}_s \, \Delta T_s \tag{14}$$

where $\dot{W}_s = \dot{m}_s c_s$ = Heat capacity of the network water that flows into the heatexchanger; and

$\dot{W}_g = \dot{m}_g c_g$ = Heat capacity of the geothermal water that also goes into that exchanger.

As for the temperature differences: $\Delta T_g = T_{gp} - T_{gi}$, and $\Delta T_s = T_1 - T_3$, these are the respective changes in temperature for mediums circulating in the heat exchanger. For the purposes of our calculations, the mass flow of the network water going into the heat exchanger was regulated in such a way that the following assumed requirement could be satisfied, i.e.:

$$\dot{W}_g = \dot{W}_s \tag{15}$$

Three pipe-joints of the network water – the analytical nodes that are temporarily turned on if necessary, characterize the assumed analytical diagram of the heating system. Exercising the first principle of thermodynamics, and on the assumption that the given node is in steady-state, the medium's temperatures and mass flows were estimated from the equations for energy balance and medium weight.

7.4 Heat pump

After the heat exchange, the working fluid goes through a heat pump unit where it is cooled down. The water heated in the heat pumps' upper heat sources is used for system heating. One of the important parameters characterizing heat pumps is the Coefficient of Performance (*COP*). In heating mode, it describes the ratio of the heating capacity to the input driving power consumption. An approximate value of the *COP* for a real heat pump can be found by using Equation 16:

$$COP = \chi COP_c = \chi \frac{t_2 + 273}{t_2 - t_1} \tag{16}$$

For a real heat pump unit, the χ value equals 0.5-0.6 (Rubik, 1996). The value of χ is not constant and changes with working conditions and heat pump construction (refrigerant, compressor type, etc.). For modern heat pumps χ has high values - near the upper mentioned limit. Consequently, the χ value of 0.6 is assumed in further calculations. Evaporation and condensation temperatures, respectively, for a refrigerant can be calculated according to Equations 17 and 18 (Rubik, 1996):

$$t_1 = t_a - \frac{t_b}{1 - e^{\frac{-t_b}{t_b + t_e}}} \tag{17}$$

$$t_2 = t_d + \frac{t_f}{1 - e^{\frac{-t_f}{t_f + t_c}}} \tag{18}$$

The values for t_c and t_e for heat pumps, based on water as both heat sources, can be assumed as 3-4˚C (Rubik, 1996). Equation 19 is derived by combining Equations 16 (for $\chi = 1$), 17 and 18. It describes the COP_C value as a function of heat source, temperatures, and temperature differences:

$$COP_c = \frac{\left(t_d + \frac{t_f}{1 - e^{\frac{-t_f}{t_f + t_c}}} \right) + 273K}{\left(t_d + \frac{t_f}{1 - e^{\frac{-t_f}{t_f + t_c}}} \right) - \left(t_a - \frac{t_b}{1 - e^{\frac{-t_b}{t_b + t_e}}} \right)} \tag{19}$$

The *COP* has higher values when the temperature differences t_f and t_b are small. According to Equation 16 this conclusion is also true for real heat pumps. Therefore, it can be said that *COP* for heat pumps increases when temperature differences decrease. Further calculations of COP_c by Equation 19 are described assuming $t_c = t_e = 3$˚C.

7.5 System 1

Here, the heat pump assists a geothermal heat exchanger in supplying additional heat to the heating system. As shown in Figure16, the lower heat source for the evaporator is water heated in the geothermal heat exchanger (Te_1), whereas the condenser works between the return, Tc_2, and supply, Tc_1, water from the heating system. The evaporator, using geothermal supply fluid, extracts the heat from a heat exchanger. The heat is released to the system by the condenser. The only heat transfer path is through the heat pump and no heat is delivered unless the heat pump is working. The heat pump upgrades extracted heat so the condenser outlet temperature is higher than the geothermal supply

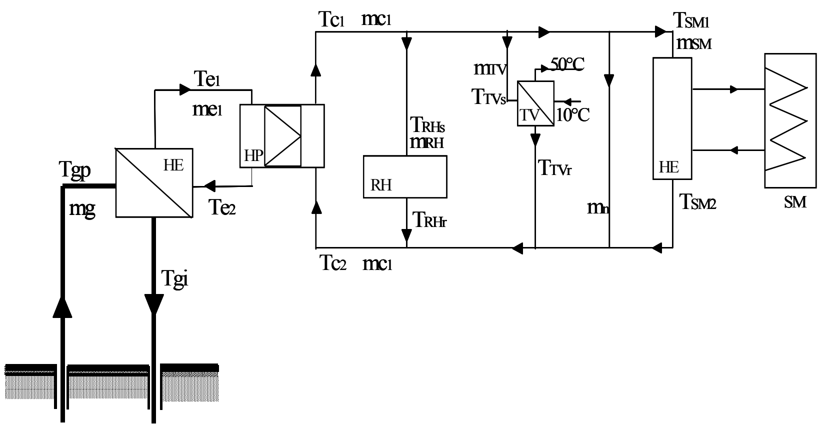

FIGURE 16: Schematic diagram of heating system 1

temperature. For regulation temperature needed before receivers, during melting and especially idling time, quantitative regulation was used. Additional water mass flow was sent through a bypass m_n. This system is without a gas boiler.

To compare how the heat pump works in each of the different radiator parameter scenarios (65/50, 65/40, 65/35˚C), four calculation variants A, B, C, D and T_g = 40˚C were considered. Explanation of those is as follows:

A. Water mass flow mc_1 is constant; the outlet temperature of an upper heat source substance outside the heat pump unit, Tc_1 is equal to 65˚C.
B. Water mass flow mc_1 is constant; outlet temperature of an upper heat source substance outside the heat pump unit Tc_1 is equal to 65˚C when outdoor temperature t_{out} varies between -16 and 3˚C; when t_{out} varies between 4 and 12˚C, Tc_1 = 55˚C.
C. Water mass flow mc_1 is variable; temperature Tc_1 is equal to 65˚C when outdoor temperature t_{out} varies between -16 and 3˚C; when t_{out} varies between 4 and 12˚C, Tc_1 = 55˚C.
D. Water mass flow mc_1 is variable; temperature Tc_1 equals 65˚C.

7.6 System 2

Figure 17 shows heating system 2, where the evaporator is located on the heating system return main. Heat is extracted from the return fluid. This reduces the return temperature to the heat exchanger, and increases the geothermal heat. In the indirect evaporator arrangement, the heat pump and the heat exchanger influence each other. The extra heat, which is transferred due to the cooling action of the heat pump, is proportional to but less than the amount of heat absorbed by the evaporator (Harrison et al., 1990).

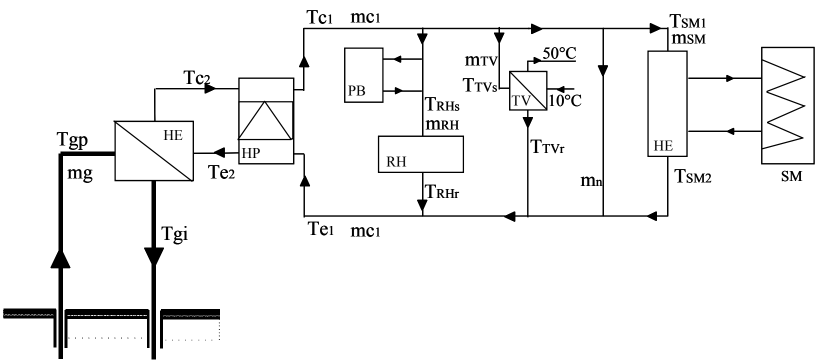

FIGURE 17: Schematic diagram of heating system 2

Calculations similar to system 1 were performed for different radiator parameter scenarios (65/50, 65/40, 65/35˚C), but only for variants A and D and T_g = 40˚C. Extra calculations were also made for T_g = 60˚C and T_g = 40˚C for system 2 and radiator parameters 90/70˚C with a gas boiler as an additional heat source, variants named E and F. Explanations of variants E and F are as follows:

E. Water mass flow mc_1 is constant; temperature Tc_1 equals 70˚C;
F. Water mass flow mc_1 is variable; temperature Tc_1 equals 70˚C.

7.7 System 3

Heating system 3, of which the schematic diagram is presented in Figure 18, consists of a geothermal heat exchanger and a conventional heat source, a gas boiler. This system is without a heat pump. Calculations for radiator

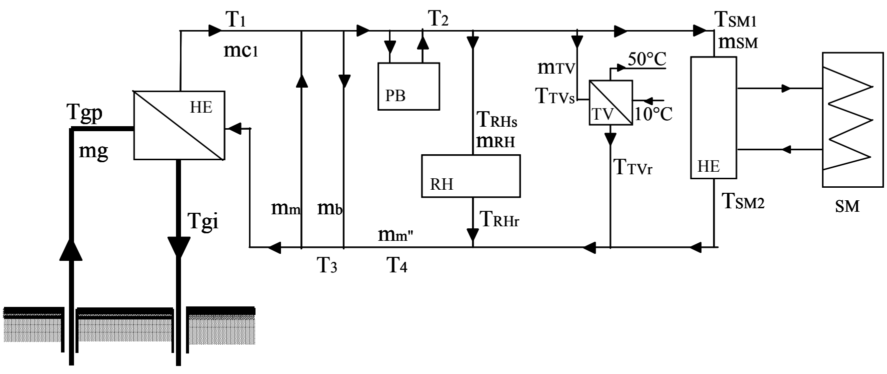

FIGURE 18: Schematic diagram of heating system 3

parameter scenarios (65/50, 65/40, 65/35˚C) and T_g = 40˚C were made. Because the return water temperature from the users was in few cases higher than the supply geothermal water, it was necessary to reduce geothermal water mass flow due to Equation 15 and use the bypasses m_m and m_b.

8. CALCULATED RESULTS

The results of thermal and fluid flow calculations for each of the heating systems described earlier are summarized in Appendix II. There, the results are presented in the form of tables showing how the heat supply is divided between geothermal energy Q_g, electrical energy W, gas energy Q_{pb} and total energy. Moreover, the relevant diagrams have been plotted, and they are presented here.

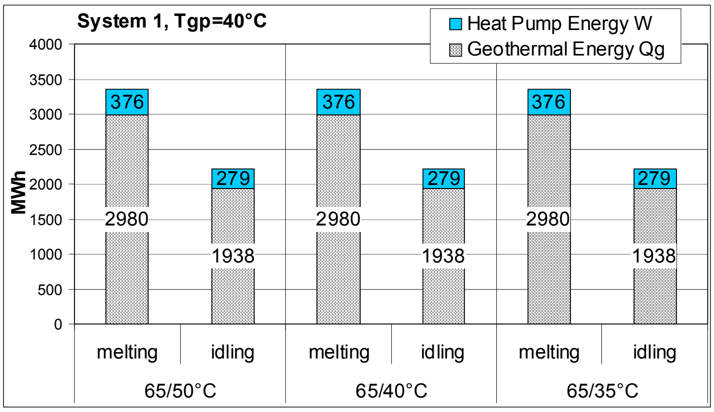

FIGURE 19: Geothermal and electrical energy utilization for different radiator parameters in system 1; melting and idling time

Figure 19 shows the consumption of geothermal and electrical energy obtained for different radiator parameters in System 1. For each set of radiator parameters, two different variants concerning the snow melting system have been calculated, one assuming melting conditions in the whole snow melting period and the other assuming idling conditions in the same period. The diagram shows that the same amount of geothermal energy and electrical power is needed regardless of the radiator parameters. This is because of the constant supply temperature, 65˚C, to all users.

Analysis of average heat pump *COP*-value for each of the calculation variants A, B, C and D (described in Chapter 7) indicate that variant B for 65/50˚C radiator parameters is the best (Figure 20). That particular case is presented in Figure 21.

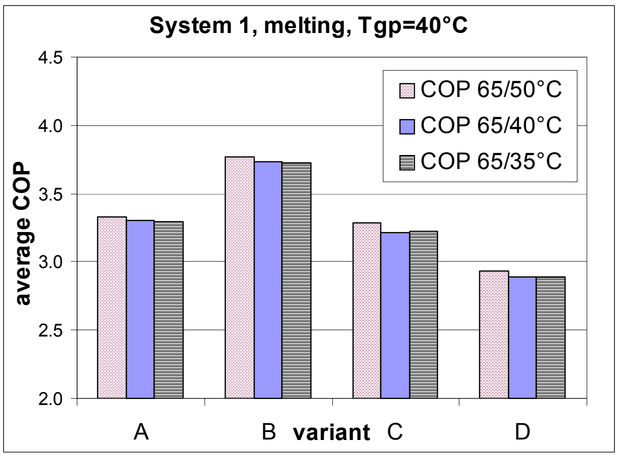

FIGURE 20: Average COP as a function of radiator parameters and calculation variants A, B, C, D

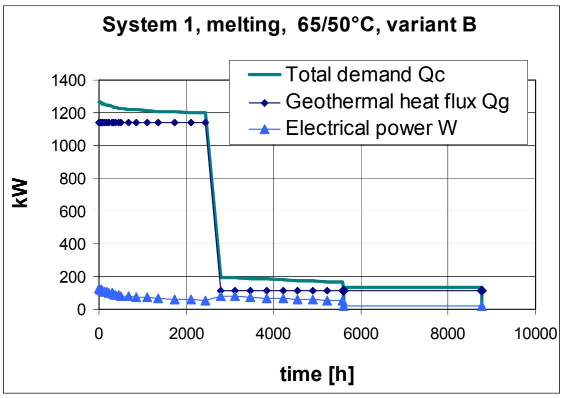

FIGURE 21: Total demand, geothermal heat flux and electricity power as a function of time for system 1; melting time

Figure 22 shows the consumption of geothermal and electrical energy obtained for different radiator parameters in System 2. The calculation variants A and D are shown separately. Regarding snow melting, only the case with melting conditions in the whole snow melting period is considered. Note that variant D is preferable for both 65/50 and 65/40˚C radiator supply / return temperatures, because of lower electricity input.

The diagram also demonstrates that for variant A (constant T_{c1} and m_{c1}), the best radiator temperatures are 65/35˚C with the highest geothermal water utilization and lowest electricity input for the heat pump. That particular example is presented in Figure 23, assuming idling conditions in the whole snow melting period.

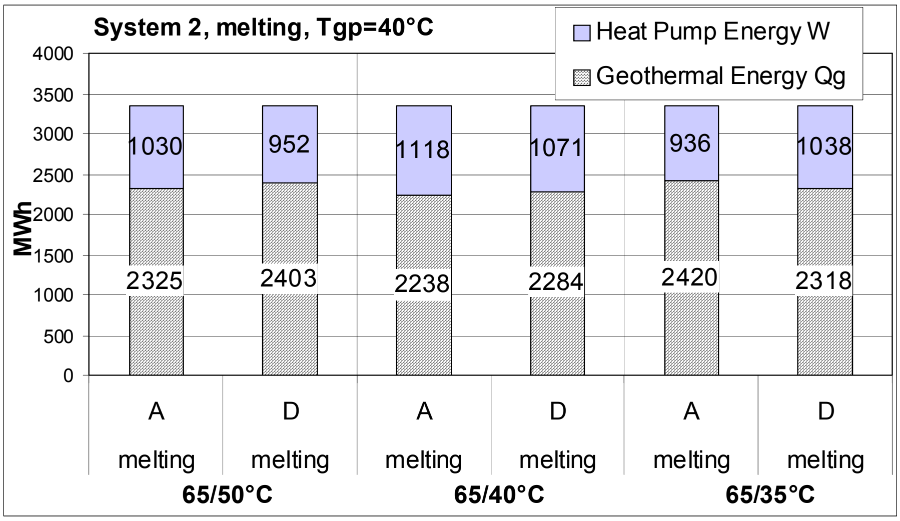

FIGURE 22: Geothermal and electrical energy utilization for different radiator parameters in System 2; melting time

Figure 24 shows calculated energy consumption in System 2 when the radiator supply/return water temperatures are 90/70˚C as commonly used in Poland, i.e. higher than in previous cases. This is obtained by using an additional gas boiler as a heat source. Geothermal water temperatures were $T_{g\ max}$ = 40 and 60˚C. In this example the condenser outlet water temperature is 70˚C. The share of geothermal energy is, of course, much higher when the water is at 60˚C than when it is 40˚C. The figure shows also that when the water temperature is 60˚C, the highest geothermal utilization is obtained in case F with variable water mass flow m_{c1} when air temperature increases above 4˚C (end of operation for the snow melting system).

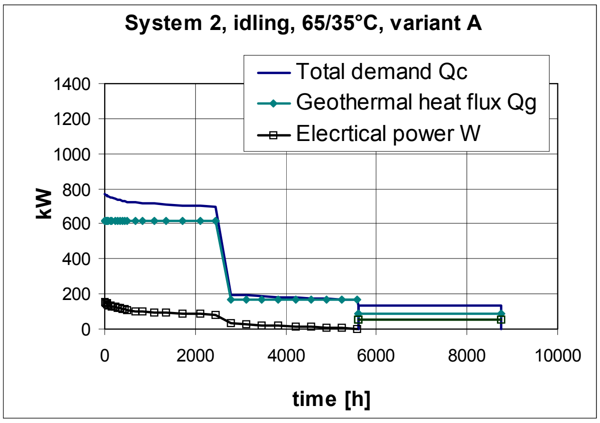

FIGURE 23: Total power demand, geothermal heat flux and electrical power as a function of time for System 2; idling time

Figure 25 shows the total power demand, geothermal heat flux, gas boiler heat flux and electrical power as a function of time for radiator parameters 90/70˚C in System 2. The temperature of the geothermal water is 60˚C and melting conditions are assumed in the whole snow melting period.

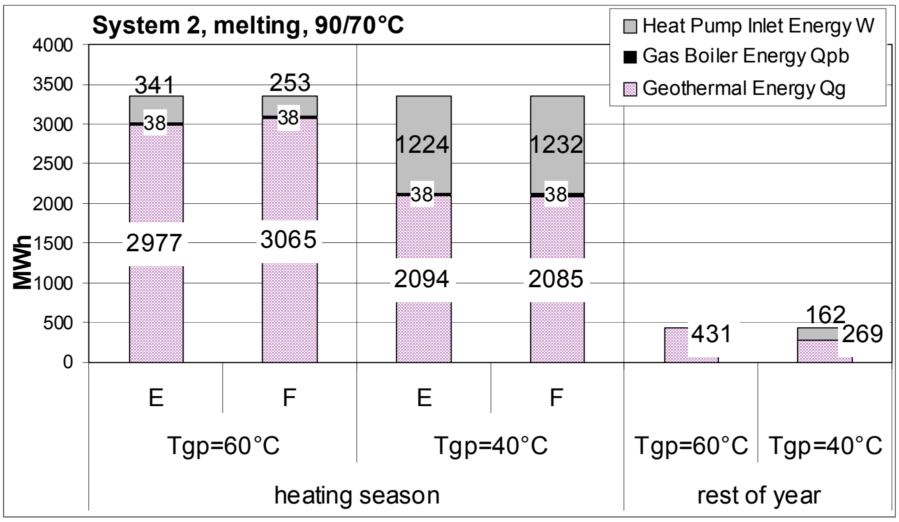

FIGURE 24: Utilization of geothermal, gas and electrical energy for the radiator parameters 90/70˚C in System 2, the temperature of the geothermal water is 40 or 60˚C; melting time

Figure 26 shows the consumption of geothermal energy and energy from the gas boiler in System 3. In this case no heat pump is used. The

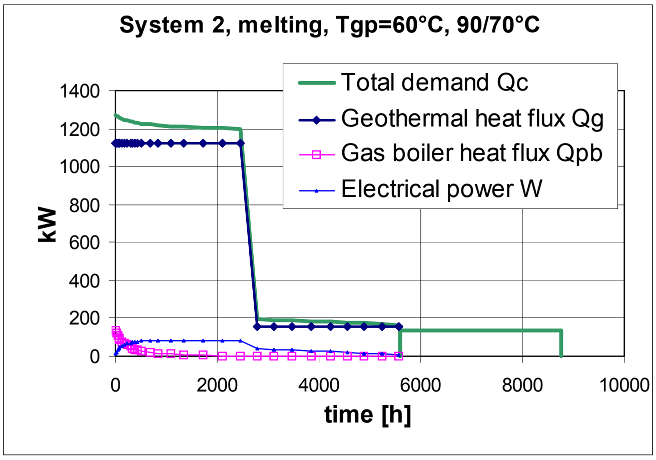

FIGURE 25: Total power demand, geothermal heat flux, gas boiler heat flux and electrical power as a function of time for radiator parameters 90/70˚C in System 2, temperature of geothermal water is 60˚C; melting time

figure shows that the amount of geothermal energy utilized is greater when the supply water temperature to the radiators is constant than when it varies with the outdoor temperature. This is because the temperature of the return water is higher when the supply water temperature varies with the outdoor temperature. In some cases, it is even higher than the temperature of the geothermal water. In that case, the bypass m_m was used for water circulation. When the snow melting system is running under idling conditions and there is no need for the boiler, the second bypass m_b is also used.

The diagrams shown in Figures 27 and 28 were made to compare System 1 and System 2 that have different heat pump arrangements. The radiator supply / return water temperatures 65/40˚C were chosen for this example. The highest utilization of geothermal water is observed in System 1, with the direct heat pump. In System 2, with the indirect heat pump, the utilization of the geothermal water is lower. Similar results were obtained for System 2 using radiator parameters 90/70˚C. The difference in the utilization of the geothermal water in the two cases for System 2 is due to the peak gas boiler that only supplies heat to the users in the case of 90/70˚C supply / return water temperatures.

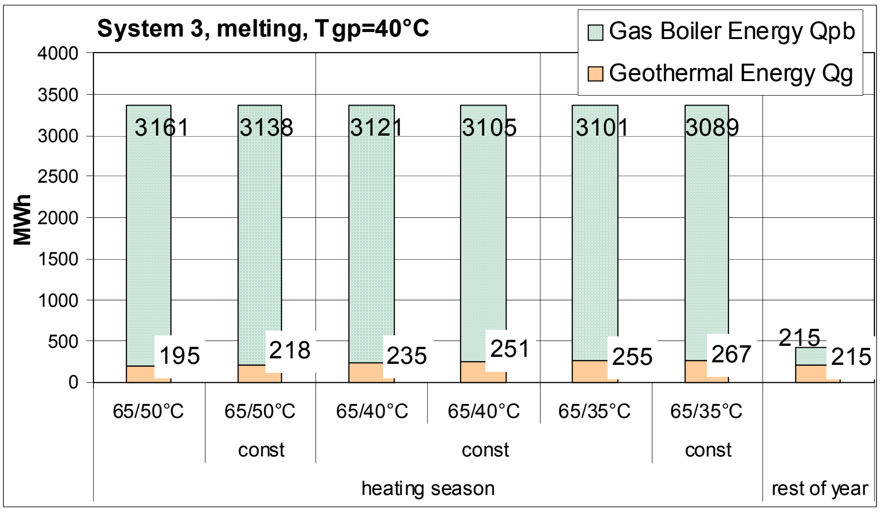

FIGURE 26: Geothermal and gas boiler energy utilization for different radiator parameters in System 3; melting time

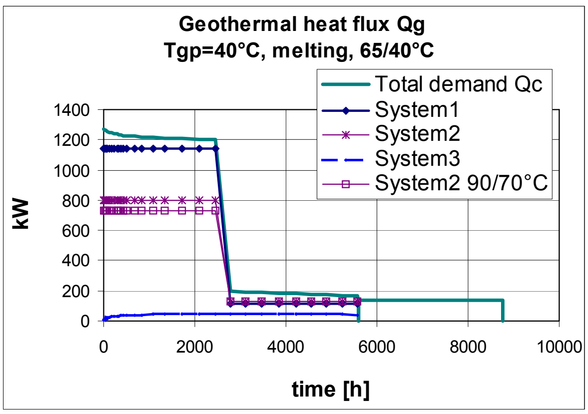

FIGURE 27: Geothermal heat flux as a function of time, comparison for all systems; melting time

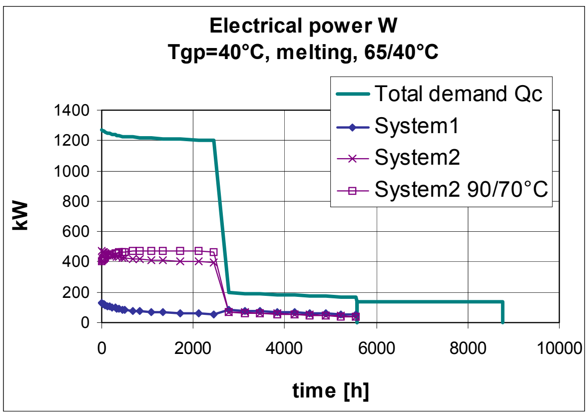

FIGURE 28: Electrical power as a function of time, comparison for all systems; melting time

Because the amount of geothermal heat is connected to the electrical power of the heat pump, a similar utilization diagram was made for electrical power as a function of time (Figure 28). According to the literature, heat pump performance for low temperatures of geothermal water (T_{gp} = 40˚C) is better for System 1, i.e. when using a direct heat pump. This arrangement is usually used when the temperature of the supply water is low as only insignificant heat transfer would be obtained by simple heat exchange alone.

9. ECONOMICAL ASPECTS

In this chapter, calculations on the running costs for the heating systems 1, 2 and 3 are presented. The price of electricity used in the calculations is 0.04 USD/kWh and the price used for natural gas is 0.14 USD/m^3. The costs of subscription and delivery of energy (USD/year) are included in the total cost. The estimated price of geothermal energy is 0.252 USD/kWh.

The total cost of energy (USD/year) for all systems and radiator parameters 65/40˚C for a snow melting system under melting conditions is presented in Figure 29 and in Figure 30 for snow melting system under idling conditions. Comparison of Systems 1 and 2, that use different heat pump arrangements, shows that System 1 has lower total cost per year. This difference is smaller when the snow melting system is under idling conditions. Comparison of System 3 (gas boiler) with the other systems is also shown in Figure 29. The worst case is System 2 for radiator water temperatures 90/70˚C. Such high design temperatures, varying with outside temperature, are commonly used in Poland. The presented cost results show that it is preferable to use low radiator supply and return water temperatures in heat pump-assisted geothermal heating systems.

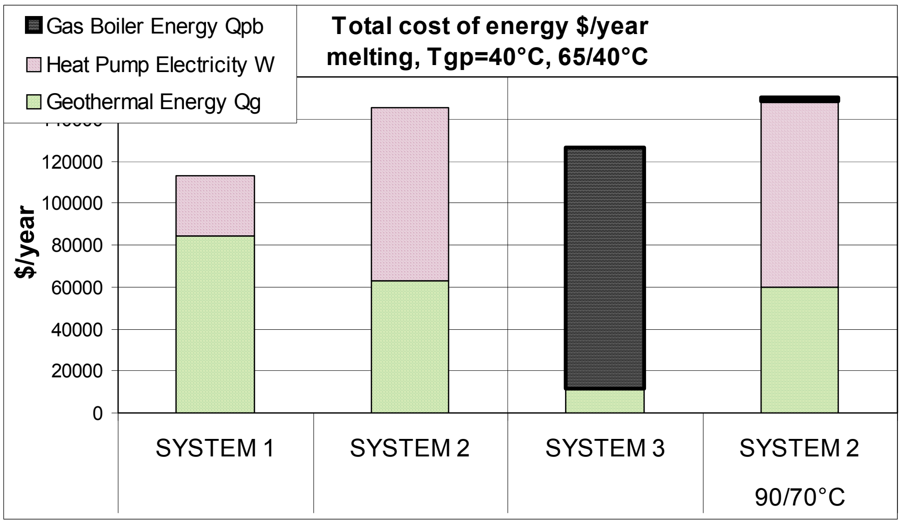

FIGURE 29: Running cost USD/year for heating systems, radiator parameters 65/40˚C, geothermal water temperature 40˚C; idling time

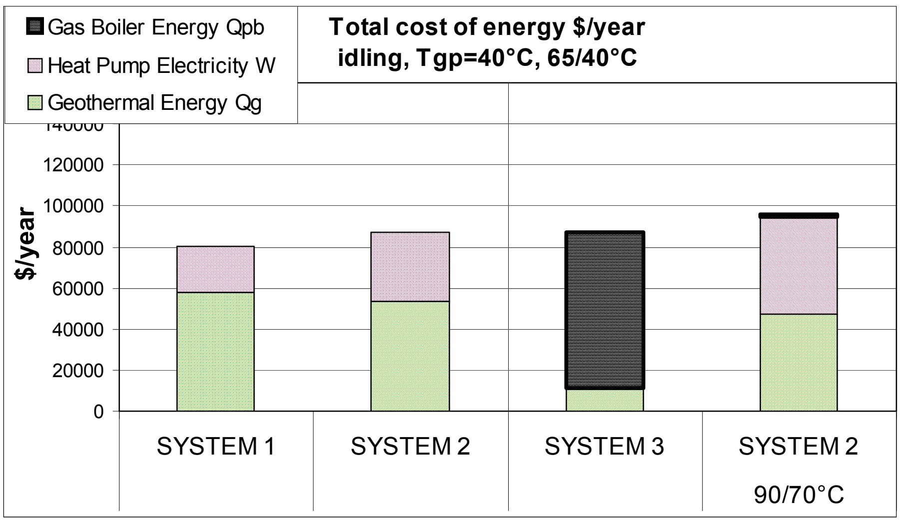

FIGURE 30: Running cost USD/year for heating systems, radiator parameters 65/40˚C, geohtermal water temperature 40˚C; idling time

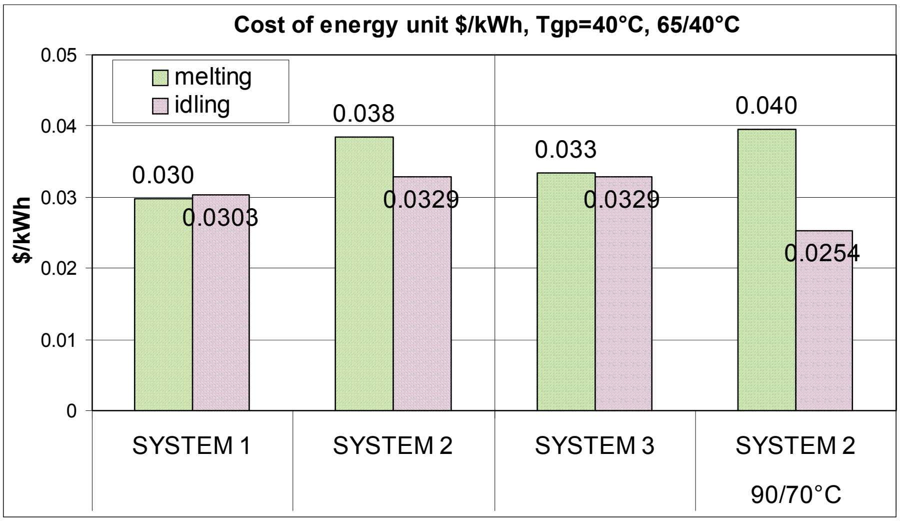

FIGURE 31: Cost of energy unit USD/kWh for the heating systems, radiator parameters 65/40˚C, geothermal water temperature 40˚C; melting and idling time

The cost of energy unit (USD/kWh) for all systems and radiator parameters 65/40˚C is presented in Figure 31. The diagram shows that the best solution is to use System 1. However, in this case a very small economic difference is found between melting and idling operations, as opposed to the situation for System 2. The biggest difference between melting and idling operations is in System 2 for radiator water temperatures 90/70˚C. In that case, a snow melting control system is important in order to decrease the melting time as much as possible.

The economical calculations presented here are very approximate. It is necessary to remember the costs associated with the production of the geothermal water. In the Polish Lowland, the share of geological work and drilling operations ranges from 60 to 80% of the total cost of the construction of a geothermal power plant, depending on the number and depth of the production and injection wells, and, hence, is a major item on the investment list (Zapalowicz, at al., 2002). Important for economical appraisal is also the low consumption of extra energy sources in geothermal heat plants, compared with traditional sources. Finally, it must be remembered that each geothermal utilization project requires individual design and cost calculations.

10. CONCLUSIONS

- The project design calculations presented here are based on the utilization of geothermal energy through a heat exchanger and a heat pump in different arrangements. The hot water produced is proposed to be used for radiator and tap water heating as well as for a snow melting system installed at the Goleniow airport in Poland. This scheme can reduce the annual running costs and also the emission of greenhouse gases from conventional fuel.

- The obvious benefits of a snow melting system are that it greatly reduces the need for snow removal, provides greater safety for pedestrians and vehicles, reduces the labour of slush removal and creates better working conditions at the airport.

- By limiting the maximum heat output of the snow melting system, the use of hot injection water is reduced. When there is snowfall, injection of hot water is needed. Heat output of a maximum 200 W/m^2 is acceptable and the operating cost is low, especially where snow detectors are used. It is crucial to use solid control equipment. By using a computerised control system, it is easy to fulfil the demand for effective snow melting, at low operating costs. With purposeful control and by using a snow detector, the quantity of hot water is expected to be reduced by half, as are the operating costs. Artificial melting of snow on pavements and roads is of great interest to the public. The use of inexpensive geothermal hot water for snow melting is practical.

- Heat pumps in geothermal schemes interact with the rest of the heating system in such a way that their performance must be analysed as part of the overall system. It is important to use temperature or water mass flow regulations in the system.

- For geothermal water temperatures lower than 40˚C, the direct heat pump approach is most advantageous. On the other hand, when the geothermal water temperature is higher than 40˚C, the indirect heat arrangement is preferable. In the presented heating system, based on the critical point 40˚C temperature of geothermal water, the direct heat pump approach (with an evaporator located *directly* after the geothermal heat exchanger) is shown to be more efficient and economical than the indirect heat pump approach (evaporator located *indirectly* on the return network water).

- Utilization of an indirect heat pump approach, for low geothermal water temperature, gives total energy costs higher than in a direct heat pump arrangement. Using radiators with supply/return water temperatures 90/70˚C increases the cost of energy unit. Presented cost is one of the main reasons for decreasing the design radiator supply and return water temperature in geothermal heating systems.

- Selection of low temperature radiator parameters depends not only on heating system efficiency and heat pump *COP*-values. Hydraulic calculations, taking into consideration pipe diameters with insulation, amount of water flowing into the system, regulation solution and size of the circulating pump, are also very important.

- In geothermal heating systems, it is necessary to be aware of high investment costs, mostly due to drilling, as well as low consumption of extra energy sources in geothermal heat plants compared to traditional plants. Each geothermal utilization project requires individual design and cost calculations.

- As an economical and efficient alternative to geothermal water as a heat source, it is possible to use a ground-source heat pump system with borehole heat exchangers. This is a simple device providing a closed circuit for the fluid to take heat from the ground and feed the evaporator, the cold side of the heat pump. The ground is an attractive heat source since its temperature remains near constant throughout the year, except in Polish conditions for the uppermost 15-27 m (Plewa, 1994). A borehole heat exchanger can be of a U-tube type, which is useful where soil moisture is adequate at shallow depth, or a coaxial tube type, especially in the utilization of abandoned wells or deep boreholes.

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NOMENCLATURE

A_f = Free (uncovered) snow area (m^2);
A_r = Snow free area ratio;
A_t = Total snow area (m^2);
a = Function used for outdoor temperatures duration
COP = Coefficient of performance for heat pump;
COP_C = Coefficient of performance for a heat pump working with the Camot's circle;
c_g = Average specific heat of water (kJ/kgK);
c_p = Specific heat of snow (2090 J/kgK);
c_1 = 1000 mm/m x 3600 s/h = 3.6 x 10^6 ;
h_{f0} = Heat of evaporation at the film temperature (kJ/kg);
$\dot{m}$ = Water mass flow (kg/s);
m_{e1} = Water mass flow circulating between geothermal heat exchanger and evaporator (kg/s);
m_{c1} = Water mass flow in system (kg/s);
p_{av} = Vapour pressure of moist air (kPa);
$\dot{Q}$ = Heat demand (kW);
ΔQ_i = Heat demand for the elementary time interval $\Delta\tau$ (MWh);
Q = Energy (MWh);
Q_{in} = Energy taken up by heat pump (MWh);
Q_{out} = Energy delivered by heat pump (MWh);
q_0 = Required slab heat output (W/m^2);
q_e = Heat of evaporation (W/m^2);
q_h = Heat transfer by convection and radiation (W/m^2);
q_m = Heat of fusion (W/m^2);
q_{idling} = Slab heat output for idling time (W/m^2);
$q_{melting}$ = Slab heat output for melting time (W/m^2);
q_s = Sensible heat transferred to snow (W/m^2);
s = Rate of snowfall ($mm_{\text{water equivalent}}/h$);
T = Water temperature (˚C);
T_{e1} = Inlet temperature of evaporator (˚C);
T_{e2} = Outlet temperature of evaporator (˚C);
T_{c1} = Outlet temperature of condenser (˚C);
T_{c2} = Inlet temperature of condenser (˚C);
T_{gp} = Production well water temperature (˚C);
T_{gi} = Injection well water temperature (˚C);
T_i = Indoor temperature (˚C);
T_{SM1} = Snow melting supply temperature (˚C);
T_{SM2} = Snow melting return temperature (˚C);
T_1 = Geothermal heat exchanger outlet water temperature (˚C);
T_2 = Water temperature in system before users (˚C);
T_3 = Water temperature in system after users (melting, idling) (˚C);
T_4 = Water temperature in system after users (idling) (˚C);
ΔT_m = Logarithmic temperature difference (˚C);
ΔT_{m0} = Logarithmic temperature difference at design conditions (˚C);
t = Outdoor temperature (˚C);
t_a = Inlet temperature of a lower heat source substance into a heat pump unit (˚C);
t_{av} = Water film temperature, 0.5 (˚C);
t_b = Heat source substance´s temperature drop in the lower heat source (˚C);
t_c = Temperature difference between the refrigerant condensation temperature and the outlet upper heat source substance´s temperature (˚C);
t_d = Inlet temperature of an upper heat source substance into heat pump unit (˚C);

t_e = Temperature difference between the refrigerant evaporation temperature and the outlet lower heat source substances´ temperature (˚C);
t_f = Heat source substances´ temperature growth in the upper heat source (˚C);
t_m = Mean water temperature in snow melting system (˚C);
t_{out} = Design outdoor temperature, -16˚C;
t_1 = Evaporation temperature for refrigerant in heat pump (˚C);
t_2 = Condensation temperature for refrigerant in heat pump (˚C);
U = Heat transfer coefficient (W/m^{2}K);
$\dot{V}$ = Volumetric flow rate (m^3/h);
V = Wind speed (km/h);
$\dot{W}$ = Network water heat capacity (kJ/K);
W = Heat pump electrical energy (MWh);
W = Heat pump electrical power (kW);

Greek letters

χ = Coefficient that compares a real heat pump COP to the ideal heat pump COP_C working in Carnot's circle at the same working conditions;
ρ_g = Mean mass water density (kg/m^3);
ρ = Density of water equivalent of snow, 998 kg/m^3;
τ = Outdoor temperatures duration (h);

Subscripts

RH = Radiator heating
SM = Snow melting system
TV = Tap water and ventilation
HE = Heat exchanger
HP = Heat pump
PB = Peak boiler
c = Total
b = Bypass b
g = Geothermal
m = Bypass m
m" = Bypass m"
s = Supply
r = Return

REFERENCES

Airport, 2002: *International airport Szczecin-Goleniow*, web page, http://www.airport.com.pl.

ASHRAE, 1995: Snow melting. In: *Heating, ventilating and air conditioning applications*. ASHRAE Applications Handbook 1995, 46.1-13.

Caddet, 2003: *Geothermal and solar heat used to melt snow on roads*. Caddet Renewable Energy Website, web page http://www.caddet-re.org/assets/no76.pdf .

Chapman, W.P., and Katunich, S., 1956: Heat requirements of snow melting systems. *ASHRAE Transactions, 62,* 359.

EPM, 2002: *Description of the climate of Poland*. Environment Protection Ministry, web page, http://www.mos.gov.pl/soe/2a.htm.

Fjarhitun, 1982: *Designing of a snow melting system*. Fjarhitun Eng., Reykjavik, report 013 (in Icelandic), 5 pp.

Goleniow, 2002: *Welcome to Goleniow.* Main web page of Goleniow city, http://www.goleniow.pl/.

Harrison, R., Mortimer, N.D., and Smárason, Ó.B., 1990: *Geothermal heating: A handbook of engineering economics*. Pergamon Press, Oxford, 558 pp.

Lund, J.W., 1999: Geothermal snow melting. In: Popovski, K., Lund, J.W., Gibson, D.J., and Boyd, T.L. (editors), *Direct utilization of geothermal energy*. Geo-Heat Center, Oregon Institute of Technology, Or, 93-98.

Meyer, Z., and Kozlowski, J., 1995: Pyrzyce - first geothermal heating system in Poland. *Proceedings of the World Geothermal Congress 1995, Florence, Italy, 3,* 2269-2272.

Morita, K., and Tago, M., 2000: Operational characteristics of the Gaia snow-melting system in Ninohe, Iwate, Japan. *Geo-Heat Center, Quart. Bull., 21-4,* 5-11.

Piatti, A., Piemonte, C., and Szegö, E., 1992: *Planning of geothermal district heating systems*. Kluwer Academic Publishers, Dordrecht, 308 pp.

Plewa, S., 1994: *Distribution of geothermal parameters on the territory of Poland* (in Polish). Wydawnictwo CPPGSMiE PAN, Krakow, 138 pp.

Ragnarsson, R., 1997: Snow melting - safer cities. *Cold Climate HVAC 1997,* international conference, Reykjavík, 9 pp.

Rubik, M., 1996: *Heat pumps - handbook*. BOINTiE "Instal" (in Polish), Warsaw, 121 pp.

Sokołowski, J., 1993: Polish geothermal resources and their potential utilization in aid of environmental protection. *Technika Poszukiwań Geologicznych, Geosynoptyka i Geotermia, 5-6*, (in Polish).

US Department of Commerce, 2002: *Showcase Europe, Airport development opportunities.* Web page, http://www.sce.doc.gov/documents/market_briefs/aerospace/pdf/Poland-IMI-AirDevelop.pdf.

Valdimarsson, P., 1993: *Modelling of geothermal district heating systems*. University of Iceland, Ph.D. thesis, 315 pp.

Wright, M.P., 1999: *Geothermal heat-pump systems.* UNU G.T.P., Iceland, unpublished lectures.

Zapałowicz, Z., Nowak, W., and Stachel, A., 2002: Condition and perspectives of utilization of geothermal energy in Poland. *4th International Scientific Conference: Effective Production, Transmission and Consumption of Energy, Kosice, Slovakia*, 191-198.

Zwarycz, K., 2000: *The effect of thermal power of the heat receivers connected in series to the distribution network upon the geothermal heat plant operation*. Technical University of Szczecin, Department of Heat Engineering, Szczecin, M.Sc. thesis (in Polish), 201 pp.

APPENDIX I: Heat demand values for radiator heating, tap water / ventilation and snow melting systems, with time occurrences for different outdoor temperatures

						Melting			Idling		
	t_{out} [°C]	**t [h]**	**Dt [h]**	**Function a**	**Q_{TV} [kW]**	**Q_{RH} [kW]**	**Q_{SM} [kW]**	**Q_C [kW]**	**Q_{RH} [kW]**	**Q_{SM} [kW]**	**Q_C [kW]**
Heating season; Tap water-ventilation system runs; Radiator heating system runs; Snow melting system runs	-16	0	0	0	135	135	1000	1270	135	500	770
	-15	21.16	21.16	16.6	135	131	1000	1266	131	500	766
	-14	42.32	21.16	33.2	135	128	1000	1263	128	500	763
	-13	63.48	21.16	49.8	135	124	1000	1259	124	500	759
	-12	84.64	21.16	66.4	135	120	1000	1255	120	500	755
	-11	126.45	41.81	99.2	135	116	1000	1251	116	500	751
	-10	168.77	42.32	132.4	135	113	1000	1248	113	500	748
	-9	232.25	63.48	182.2	135	109	1000	1244	109	500	744
	-8	295.1	62.84	231.5	135	105	1000	1240	105	500	740
	-7	337.42	42.32	264.7	135	101	1000	1236	101	500	736
	-6	379.49	42.07	297.7	135	98	1000	1233	98	500	733
	-5	442.71	63.23	347.3	135	94	1000	1229	94	500	729
	-4	506.19	63.48	397.1	135	90	1000	1225	90	500	725
	-3	674.84	168.65	529.4	135	86	1000	1221	86	500	721
	-2	843.61	168.77	661.8	135	83	1000	1218	83	500	718
	-1	1096.9	253.29	860.5	135	79	1000	1214	79	500	714
	0	1349.7	252.78	1058.8	135	75	1000	1210	75	500	710
	1	1729.6	379.87	1356.8	135	71	1000	1206	71	500	706
	2	2109.2	379.61	1654.6	135	68	1000	1203	68	500	703
	3	2446.6	337.42	1919.3	135	64	1000	1199	64	500	699
	4	2784	337.42	2184	135	60	0	195	60	0	195
	5	3121.4	337.42	2448.7	135	56	0	191	56	0	191
	6	3458.8	337.42	2713.4	135	53	0	188	53	0	188
	7	3838.6	379.74	3011.3	135	49	0	184	49	0	184
	8	4218.3	379.74	3309.2	135	45	0	180	45	0	180
	9	4554.9	336.53	3573.2	135	41	0	176	41	0	176
	10	4893.2	338.31	3838.6	135	38	0	173	38	0	173
	11	5230.6	337.42	4103.3	135	34	0	169	34	0	169
	12	5568	337.42	4368	135	30	0	165	30	0	165
Rest of year	t_{out} [°C]	t [h]	Dt [h]		Q_{TV} [kW]			Q_C [kW]			
	> 12	8760	3192		135			135			

APPENDIX II: Summary of the calculation results for System 1, System 2 and System 3

TABLE 1: System 1

Radiator parameters [°C]	Heating season [MWh]										The rest of year [MWh]				
	Melting					Idling									
	Q_g	Q_{in}	Q_{out}	W	Q_c	Q_g	Q_{in}	Q_{out}	W	Q_c	Q_g	Q_{in}	Q_{out}	W	Q_c
65/50															
A	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
B	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
C	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
D	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
65/50const															
A	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
B	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
C	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
D	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
65/40															
A	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
B	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
C	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
D	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
65/40const															
A	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
B	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
C	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
D	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
65/35															
A	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
B	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
C	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
D	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
65/35const															
A	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
B	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
C	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431
D	2980	2980	3356	376	3356	1938	1938	2217	279	2217	365	365	431	66	431

TABLE 2: System 2

Radiator parameters [°C]	Heating season [MWh]										The rest of year [MWh]				
	Melting					Idling									
	Q_g	Q_{in}	Q_{out}	W	Q_c	Q_g	Q_{in}	Q_{out}	W	Q_c	Q_g	Q_{in}	Q_{out}	W	Q_c
65/50															
A	2325	6605	7635	1030	3356	1909	7327	7635	308	2217	269	54	215	162	431
D	2403	3393	4345	952	3356	1987	4115	4345	230	2217	269	54	215	162	431
65/50const															
A	2325	6605	7635	1030	3356	1909	7327	7635	308	2217	269	54	215	162	431
D	2403	3393	4345	952	3356	1987	4115	4345	230	2217	269	54	215	162	431
65/40															
A	2238	5916	7034	1118	3356	1854	6671	7034	362	2217	269	54	215	162	431
D	2284	2672	3744	1071	3356	1901	3428	3744	316	2217	269	54	215	162	431
65/40const															
A	2238	5916	7034	1118	3356	1854	6671	7034	362	2217	269	54	215	162	431
D	2284	2672	3744	1071	3356	1901	3428	3744	316	2217	269	54	215	162	431
65/35															
A	2420	5947	6883	936	3356	1950	6617	6883	266	2217	269	54	215	162	431
D	2318	2555	3593	1038	3356	1848	3225	3593	368	2217	269	54	215	162	431
65/35const															
A	2420	5947	6883	936	3356	1950	6617	6883	266	2217	269	54	215	162	431
D	2318	2555	3593	1038	3356	1848	3225	3593	368	2217	269	54	215	162	431

TABLE 3: System 2 with an additional gas boiler

Radiator parameters [°C]	Heating season [MWh]												The rest of year [MWh]					
	Melting						Idling											
	Q_g	Q_{pb}	Q_{in}	Q_{out}	W	Q_c	Q_g	Q_{pb}	Q_{in}	Q_{out}	W	Q_c	Q_g	Q_{pb}	Q_{in}	Q_{out}	W	Q_c
90/70 Tgp=60°C																		
E	2977	38	3257	3598	341	3356	1799	38	3219	3598	380	2217	431	0	0	0	0	431
F	3065	38	1700	1953	253	3356	1887	38	1662	1953	292	2217	431	0	0	0	0	431
90/70 Tgp=40°C																		
E	2094	38	7172	8396	1224	3356	1603	38	7820	8396	576	2217	269	0	54	215	162	431
F	2085	38	3325	4558	1232	3356	1595	38	3973	4558	584	2217	269	0	54	215	162	431

TABLE 4: System 3

Radiator parameters [°C]	Heating season [MWh]						The rest of year [MWh]		
	Melting			Idling					
	Q_g	Q_{pb}	Q_c	Q_g	Q_{pb}	Q_c	Q_g	Q_{pb}	Q_c
65/50	195	3161	3356	195	2022	2217	215	215	431
65/50 const	218	3138	3356	218	1999	2217	215	215	431
65/40	235	3121	3356	235	1982	2217	215	215	431
65/40 const	251	3105	3356	251	1966	2217	215	215	431
65/35	255	3101	3356	255	1962	2217	215	215	431
65/30 const	267	3089	3356	267	1950	2217	215	215	431