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Modeling of Low Temperature Geothermal District Heating Systems[#]

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ABSTRACT

In this work, low temperature geothermal district heating systems with heat pumps have been studied and compared with fuel-oil boiler heating systems for intermittent and continuous regimes according to the optimum indoor air temperature and operational cost. Izmir Institute of Technology (IZTECH) Campus is taken as a case study. Various heat pump and boiler configurations are studied to meet required duty. Operational cost analysis for each alternative is conducted. According to the results, for IZTECH Campus the best alternative, which gives the optimum indoor air temperature and the lowest operational cost, is heat pump continuous regime.

Key Words: District heating systems; Heat pumps; Geothermal energy; Modeling of district heating systems.

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1. INTRODUCTION

District heating systems (DHSs) may be defined as the heating and/or cooling of two or more structures from a central heat source. The thermal energy is distributed through a network of insulated pipes consisting of supply and return mains. Heat can be provided through the use of conventional boilers that burn conventional fuels such as oil, natural gas, or coal, or from cogeneration plants that produce both electricity and heat. DHSs may also utilize renewable resources such as geothermal, biomass, or waste heat resources such as industrial waste heat. Fossil fuel peaking or back up is often an integral part of DHSs (Bloomquist, 2001).

Geothermal DHSs have several potential advantages such as reduced fossil fuel consumption and heating cost. On the other hand, air quality is improved. Additionally, the fire hazard of individual buildings is reduced, because combustion does not occur in the buildings.

The methods by which heat is extracted from geothermal fluid depend strongly on temperature of the fluid and nature of the heating application. There are two basic methods of heat extraction, which are used in heating applications. Direct heat exchange and heat pumps (HPs). The use of HPs is often considered when the fluid temperature is too low for heat transfer to occur by direct heat exchange (Harrison et al., 1990).

IZTECH Campus has a geothermal resource at 33°C, which is classified as low temperature geothermal resource. Thus in this study heat pump heating systems (HPHSs) have been considered. Fuel boiler heating system (FBHS), which represents the existing heating system, is considered as an alternative to the HPHS. Two heating strategies have been adopted to both alternatives: Intermittent and continuous. Indoor temperature of the buildings is the main control parameter of the heating simulations. Mathematical models were derived; the programs using Matlab (The Mathworks, 2002) and EES (F-Chart Software, 2002) have been written and run using hourly weather data. Nearly 125 different heating regimes with various parameters have been studied to determine the best heating system.

2. IZTECH CAMPUS AND THE EXISTING HEATING SYSTEM

The construction of the buildings in the Campus was started on November 1994. Currently number of the existing buildings has reached to 15 with 50730 m^2-floor area and the Campus is still under development. Individual heating, ventilation and air conditioning (HVAC) systems are employed at each department.

On the other hand, a geothermal resource outcropped as a hot spring with a temperature of 33–35°C in the Campus. Exploration studies started in 1995 and initial geochemistry studies indicated that reservoir temperature would be as high as 60–100°C (Giese et al., 2000). The resource is exploited in 2002 by drilling 5 gradient wells, one of which is completed as a production well at 33°C temperature. The expected flowrate is 30–40 kg/s and it is assumed as 30 kg/s in the calculations.

In this study, the Campus buildings are considered into two groups: existing buildings and new buildings, which are under construction at present or that will be built in the future. Total heat load for existing and new buildings is calculated as 3662 kW and 4830 kW, respectively. Campus heat load totals 8492 kW.

3. MODELING OF DISTRICT HEATING SYSTEMS

Campus DHS is modelled according to macroscopic, dynamic model depending on black box approach and using heating equipment, heat loss, building energy storage and heat pump models. Temperature drop in the pipes is omitted. Thus, pipe-cooling is not considered. Heating system is simulated according to constant flowrate and variable return water temperature.

3.1. Building Heat Loss Model

The heat loss is mainly a function of the outdoor air temperature. By taking the outdoor temperature as a primary influencing factor for the weather, the heat loss model becomes:

$$\dot{Q}_{\text{loss}} = U_b \cdot A_b \cdot (T_i - T_o) \tag{1}$$

3.2. Heating Equipment Model

The heating equipment (radiator, fan coil, etc.) transfers heat from the district heating water to the indoor air. The input signals to the heating equipment model are indoor temperature, water flow and building supply temperature. The output signals are heat supply and return temperatures. The heat transferred from the water is written as:

$$\dot{Q}_{\text{supply}} = \dot{m} \cdot C_p \cdot (T_s - T_r) \tag{2}$$

The cooling of the district heating water is a nonlinear function of the operational and design parameters. Water return temperature from a building is determined by the performance of the heating equipment and can be written as

$$T_r = f\,(T_s, T_i, \dot{m}, T_{s0}, T_{i0}, \dot{m}_0\,, T_{r0}) \tag{3}$$

In addition to Eq. (2), the rate of heat transferred from the waterside to the ambient air can be expressed as:

$$\dot{Q}_{\text{supply}} = U_{\text{heq}} \cdot A_{\text{heq}} \cdot LMTD_{\text{heq}} = \dot{Q}_{\text{heq}} \tag{4}$$

where the logarithmic mean temperature difference can be calculated as follows:

$$LMTD_{\text{heq}} = \frac{(T_s - T_i) - (T_r - T_i)}{\ln((T_s - T_i)/(T_r - T_i))} \tag{5}$$

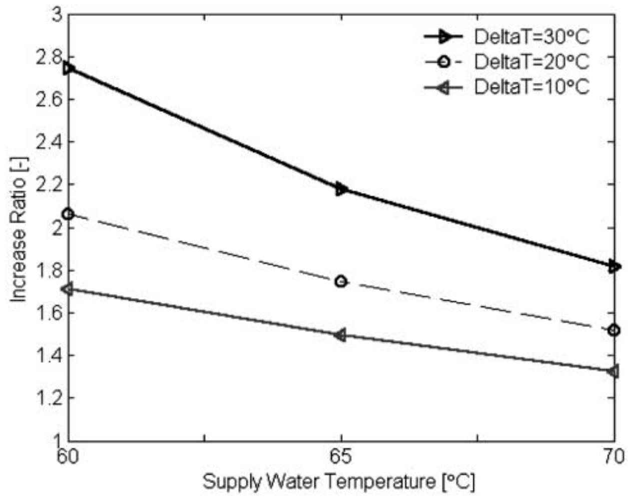

Figure 1. Increase ratio versus supply water temperature depending on various temperature differences between supply and return water temperature.

Performance of the building radiator or fan coil system depends on the supply and return water temperatures, or

$$\text{Performance} = \frac{\dot{Q}_{\text{heq}}}{\dot{Q}_{\text{heq0}}} = \left(\frac{LMTD_{\text{heq}}}{LMTD_{\text{heq0}}}\right)^{(n)} \tag{6}$$

where the index zero refers to the reference conditions. The reference condition of the existing space heating equipments (radiators, fan coil etc.) is 90–70°C (Valdimarsson, 1993). The value of n can be determined experimentally. This value is taken as 1.35 for radiator and 1 for fan coil (Valdimarsson, 1993; Yildirim, 2003).

Depending on the performance of the heating equipment based on the reference conditions, it could be necessary to add extra heating equipment for various supplies or return temperatures. As heating equipment fan coils are considered since they can be used for cooling as well. Performance of the fan coils is evaluated depending on various supply water temperatures and temperature difference (ΔT) between supply and return water temperatures based on the reference conditions. Figure 1 gives the increase ratio of the fan coil size. The need to extra heating equipment increases with increasing ΔT and decreasing supply water temperature.

3.3. Building Energy Storage Model

By assuming all heated parts of the building to be heated at uniform indoor temperature at all times, the building can be modelled as a single heat capacity element. A differential equation is then written relating the net heat flow to the building to time derivative of the indoor temperature and the building heat capacity. Then the building energy storage can be described by Valdimarsson (1993)

$$\frac{dT_i}{dt} = \frac{1}{C}\dot{Q}_{\text{net}} = \frac{1}{C}(\dot{Q}_{\text{supply}} - \dot{Q}_{\text{loss}}) \tag{7}$$

In steady state approach, T_i is taken as constant at design indoor temperature (20°C) but in the dynamic approach indoor temperature is calculated from Eq. (7).

3.4. Heat Pump Model

HPs are not single elements like primary heat exchangers or back-up boilers. The evaporators and condensers are located in different parts of the system and also by-pass connections of various types are possible. Consequently a wide variety of different layouts are possible in geothermal schemes all of which can, in general, perform differently.

If attention is focused on the way in which the HP supplies heat in any scheme, then two basic classes of configuration can be identified. The HP assists the primary heat exchanger, supplying additional heat from the geothermal fluid. This is called the heat pump assisted (HPA) approach. The HP dominates the geothermal supply and no heat is transferred if the HP is not operating. This is called the heat pump only (HPO) approach. As a general rule if geothermal fluid temperature is greater than 40°C, "HPA" layout gives better performance, otherwise "HPO" layout is recommended (Harrison et al., 1990).

HPO type heat pump, which is given in Fig. 2, is considered for the Campus DHS since geothermal fluid temperature is 33°C. Because of the corrosive effects of this fluid, a heat exchanger is also considered to protect HP unit where geothermal fluid passes through heat exchanger rather than evaporator.

According to Fig. 2 the HP heat flows can be written as:

$$\dot{Q}_{\text{cond}} = \dot{m} \cdot C_p(T_s - T_r) \tag{8}$$

$$\dot{Q}_{\text{eva}} = \dot{m}_w \cdot C_p \cdot (T_{\text{in}} - T_{\text{out}}) \tag{9}$$

The coefficient of efficiency (COP) is often used to measure the performance of a HP. The COP for an idealized Carnot HP can be related to the temperature in the condenser and evaporator

$$\text{COP}_{\text{car}} = \frac{T_{\text{cond}}}{T_{\text{cond}} - T_{\text{eva}}} \tag{10}$$

Although in general COP for a HP is defined as the ratio of the heat released to the work input

$$\text{COP} = \frac{\dot{Q}_{\text{cond}}}{\dot{W}} = \frac{\dot{Q}_{\text{cond}}}{\dot{Q}_{\text{cond}} - \dot{Q}_{\text{eva}}} \tag{11}$$

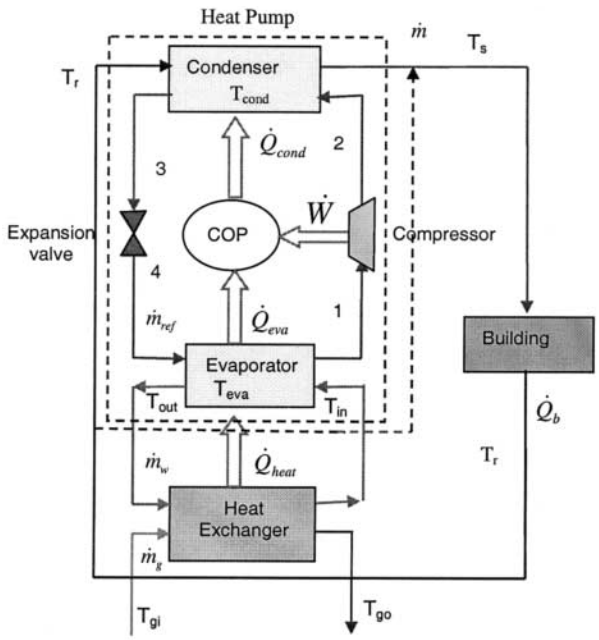

Figure 2. Considered heat pump only (HPO) district heating system.

It is also often assumed that thermal and mechanical losses in the cycle reduce the performance further to about 50% of the theoretical value, or the COP for a practical HP becomes (Harrison et al., 1990)

$$\mathrm{COP} = 0.5 \cdot \mathrm{COP}_{\mathrm{car}} \tag{12}$$

3.4.1. Heat pump heating system simulation program's algorithm

The program is written in Matlab (The Mathworks, 2002), for its relative flexibility and ease of programming. According to weather data of Izmir, heating season starts in October and ends in April, where the heating system is turned off during the rest of the year. Hence, simulation is conducted only for this time period (October to April, 3874 h) ignoring the rest of the year.

The parameters to be determined for the heat pump district heating system (HPDHS) design are as follows;

- Condenser outlet temperature (T_s),
- Geothermal fluid flowrate ($\dot{m}_g$),
- COP of the heat pump.

First step is to determine the condenser outlet temperature and supply water temperature (T_s). If the value of the flow ($\dot{m}$) is not zero, HPDHS runs. Then return temperature from the radiators (T_r) is calculated by an iterative technique.

To calculate HP capacity, evaporator outlet temperature (T_{out}) should be calculated according to supply water temperature. To do so, some assumptions had been made. Initially, evaporator outlet temperature is assumed. Using Eqs. (8)–(12) exact evaporator outlet temperature can be calculated by iteration. Then HP capacity and geothermal outlet temperature is calculated.

Hundred and eight heating alternatives have been studied for various geothermal fluid temperature and flowrate, supply water temperature and ΔT using steady state approach at 0°C outdoor and 20°C indoor design temperature. Figure 3 exhibits COP values for various geothermal fluid temperature (25–50°C), flowrate (60–120 kg/s) and supply water temperature (60–70°C) for $\Delta T = 30$°C. It is obvious from the figure that COP is increased with increasing geothermal fluid flowrate and temperature, but decreased with increasing supply temperature. For heating applications COP is generally between 3.5 and 4. For desired COP value, Fig. 3 indicates the possible supply temperature and flowrate for known geothermal fluid temperature. For 33°C fluid temperature and the highest COP value of 4, Fig. 3 gives a required flowrate and supply temperature as 120 kg/s and 60°C, respectively. Assuming average geothermal flowrate of each well is 30 kg/s, the required number of wells is 4.

Using obtained design parameters above, heating system is simulated by dynamic approach. The Campus is predominantly occupied and therefore heated only during the working hours (09:00–17:00), which is referred to as intermittent

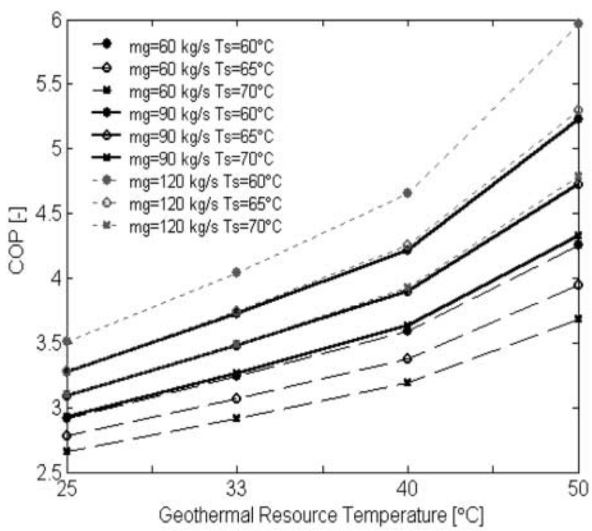

Figure 3. Coefficient of performance (COP) versus geothermal resource temperature depending on various supply temperatures and geothermal flowrate ($\Delta T = 30$°C).

heating. The simulation has been carried out both for intermittent and continuous heating, for which the system runs 24 h a day continuously without stop.

In the simulation program intermittent heating system was turned on one hour earlier (08:00) and turned off at the end of this period (17:00). For the calculation of indoor temperature Eq. (7) can be written in matrix form as:

$$\left[\frac{dT_i}{dt}\right] = \left[-\frac{U_b \cdot A_b}{C}\right][T_i] + \left[\frac{U_b \cdot A_b}{C} \quad \frac{\dot{m} \cdot C_p \cdot (T_s - T_r)}{C}\right]\begin{bmatrix} T_o \\ 1 \end{bmatrix} \tag{13}$$

Equation (13) can be solved by discrete method (Nappa, 2002).

4. HEATING SYSTEM ALTERNATIVES FOR THE CAMPUS

Two main alternatives, which are HP and FB are considered. Various sub-alternatives have been studied but only 3 of which for each option are given in Table 1. Outdoor design temperature is taken as 0°C for each alternative (MMO, 2002).

In large HP schemes it is common to use a number of separate HP units grouped together so that they operate effectively as one HP. The advantages of using a group of smaller HPs as opposed to one large one include: improved performance because each unit works with a smaller temperature difference between the evaporator and condenser; and improved reliability and easier control because individual units can be shut down independently (Harrison et al., 1990).

In this study, one HP unit is employed for each well. Thus, total number of the HPs is 4. HPs are connected in parallel and are operated depending on outdoor temperature (T_o), which is given in Table 1. Three sub-alternatives are evaluated for HP option. First 2 alternatives are simulated for intermittent, the last one is for continuous regime. Design parameters for all HP alternatives are 33°C geothermal fluid, 60°C supply and 30°C return water temperature.

FB option is operated in the similar way. For fuel boiler district heating system (FBDHS) heating system supply temperature is taken as the same as the boiler set temperature. The boiler set temperatures, which are recommended by Demirdöküm depending on outdoor temperature, are used (Dağsöz, 1998).

Table 1. Considered heating alternatives.

Alternative No.		1	2	3
Heating type		Intermittent	Intermittent	Continuous
Increase ratio		1	2.8	1
Number of operated heat pump or boiler units	$T_o <= 5°C$	4	4	4
	$5°C < T_o <= 10°C$	3	4	3
	$10°C < T_o <= 13°C$	2	4	2
	$13°C < T_o < 18°C$	1	2	1

5. APPROXIMATE OPERATIONAL COST OF THE HEATING SYSTEM

In the calculations, the cost of annual energy consumption is calculated approximately by:

$$\mathrm{Cost}_{\mathrm{tot}} = \mathrm{Cost}_{\mathrm{hen}} + \mathrm{Cost}_{\mathrm{cp}} + \mathrm{Cost}_{\mathrm{wp}} \tag{14}$$

For HP:

$$\mathrm{Cost}_{\mathrm{hen}} = \frac{W_{\mathrm{annual}}}{\eta_{\mathrm{el}}} \cdot P_{\mathrm{el}} \tag{15}$$

The electricity unit cost is 0.09\$/kWh (March 2003) (TEDAS, 2002). Electric motor efficiency is assumed to be 0.8.

For FB:

$$\mathrm{Cost}_{\mathrm{hen}} = m_{\mathrm{fuel}} \cdot P_{\mathrm{fuel}} = \frac{Q_{\mathrm{boiler}}}{H_u . \eta_{\mathrm{boil}}} \cdot P_{\mathrm{fuel}} \tag{16}$$

Specific heat capacity of the fuel oil is 11.27 kWh/kg and fuel cost is 0.53\$/kg (March 2003) (Petrol Ofisi, 2002). In the calculations boiler efficiency is taken as 0.8.

For circulation pumps:

$$\mathrm{Cost}_{\mathrm{cp}} = \frac{\dot{m} \cdot g \cdot h_p}{1000 \cdot \eta_m \cdot \eta_p} \cdot P_{\mathrm{el}} \tag{17}$$

Total dynamic head of circulation pump (h_p) has been taken as 40 m for city water circulation pump (CWCP) and 65 m for geothermal fluid circulation pump (GFCP). Motor and pump efficiencies are assumed as 0.85 and 0.7, respectively. Equation (17) can be used also for well pumps where h_p is taken as 100 m.

6. RESULTS AND DISCUSSION

Each sub-alternative is simulated by using hourly outdoor temperature data of Izmir city. In Fig. 4, variations of indoor and outdoor temperature during the working hours are shown for HPDHS. As it can be seen from the figure, the best indoor condition with existing heating equipment size is obtained by Alternative 3, which is operated for continuous regime. Alternative 1 exhibits poor indoor air temperature. To improve the indoor air temperature of the buildings, fan coil size should be increased. Increase ratio is limited to 2.8 times since further increase in fan coil size caused some complications in geothermal return temperature (Alternative 2).

In Fig. 5 duration curve of the HP compressor work is shown. While intermittent regime requires a large heating system size, the size for continuous regime decreases drastically. FB alternatives are simulated in the same way with HP option as given above. Variations of the indoor and outdoor temperatures during the working period and duration curve of boiler capacity for FBDHS alternatives are shown in Figs. 6 and 7, respectively. The most significant difference from HP option

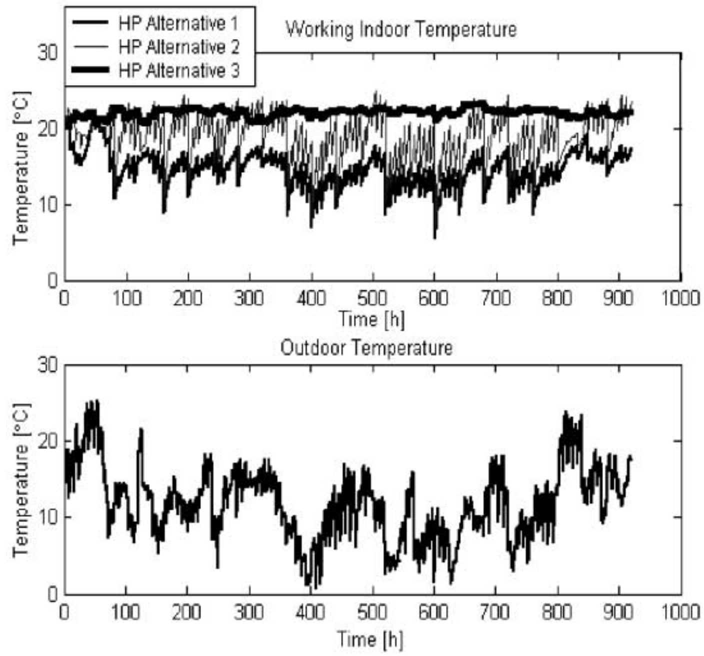

Figure 4. Variations of indoor and outdoor temperature during the working period for heat pump heating system alternatives.

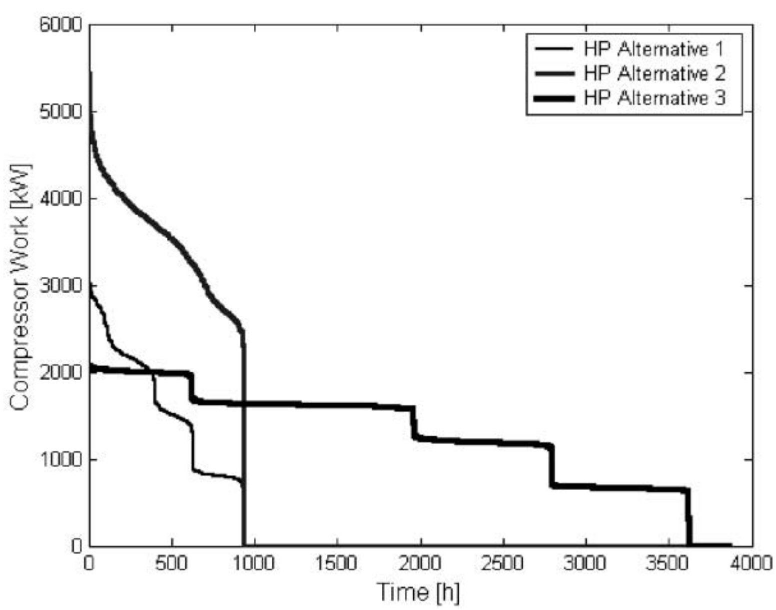

Figure 5. Duration curve of compressor work.

is the boiler capacity. The capacity does not change for Int. and Cont. regimes as can be seen from Fig. 7. Therefore, higher indoor temperature is obtained for the continuous regime.

Indoor conditions of the buildings such as degree hour (DH), average indoor temperature (AIT) and average deviation (AV) for each considered alternatives are given in Table 2. Cost of annual energy consumption of considered heating alternatives are given in Table 3. For Cont. regime, operational cost of HPDHS is 40% more economic than FBDHS.

7. CONCLUSIONS

In this study, HP and FB district heating alternatives were investigated for IZTECH Campus, based on indoor air temperature and system operation cost. According to the results, intermittent heating systems yield poor indoor conditions. Heating requirements of the buildings cannot be obtained by running a heating system only during working hours. To achieve comfortable indoor conditions, heating should be started much earlier than the start of the working hours. That means an increase in system capacity, which also means an increase in fuel consumption. To use larger heating equipment improves the indoor conditions of the

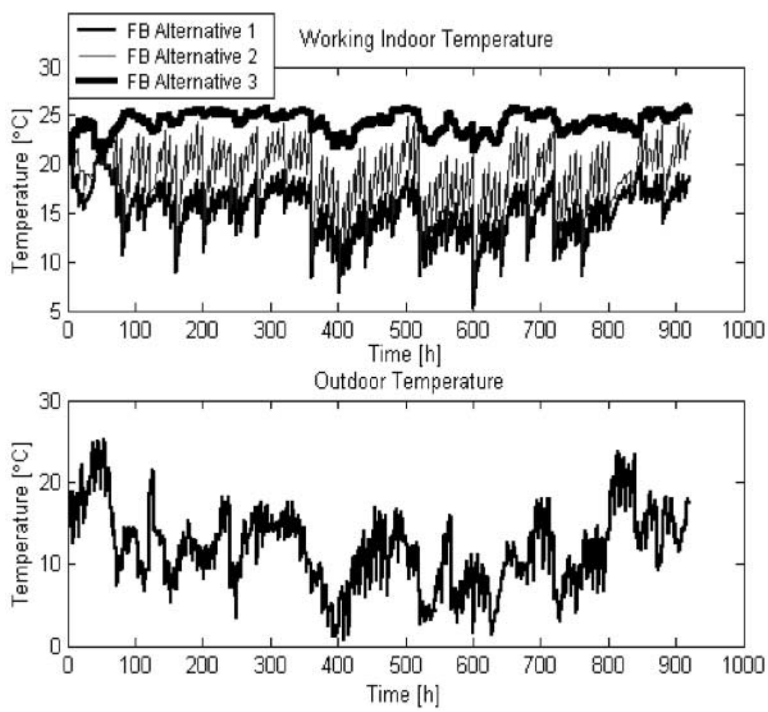

Figure 6. Variations of indoor and outdoor temperature during the working period for fuel boiler heating system alternatives.

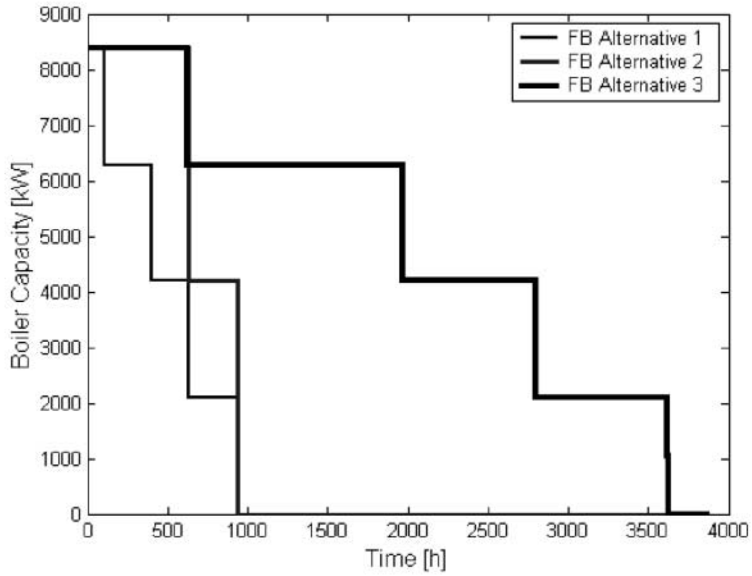

Figure 7. Duration curve of boiler capacity.

Table 2. Indoor conditions of the buildings for different heating alternatives.

	Working hours			All heating season		
Alternatives	DH	AIT (°C)	AD (°C)	DH	AIT (°C)	AD (°C)
HP Alt. 1	4950	14.6	5.4	24,982	13.5	6.5
HP Alt. 2	1511	19.0	1.0	13,317	16.9	3.1
HP Alt. 3	0	22.0	−2.0	140	22.0	−2.0
FB Alt. 1	4395	15.2	4.8	23,009	14.1	5.9
FB Alt. 2	2795	17.2	2.8	17,668	15.5	4.5
FB Alt. 3	0	24.3	−4.3	140	24.3	−4.3

Table 3. Cost of annual energy consumption for the different heating alternatives.

	Circulation pump electricity (USD)		Well pump electricity (USD)	Heat pump electricity (USD)	Boiler fuel (USD)	Total cost (USD)
Alternatives	GWCP	CWCP				
HP Alt. 1	5984	3741	9206	171,500	0	190,431
HP Alt. 2	9039	10,474	13,907	370,480	0	403,900
HP Alt. 3	26,085	14,538	40,131	555,990	0	636,744
FB Alt. 1	0	5611	0	0	254,360	259,971
FB Alt. 2	0	15,711	0	0	384,250	399,961
FB Alt. 3	0	21,806	0	0	1,108,800	1,130,606

buildings. Since heating equipment design temperature is an important issue for heating systems, heating equipment should be selected for low supply and return temperatures for new buildings, and the heating equipment size in existing buildings should be increased. For IZTECH Campus, the best alternative, which gives the optimum indoor air temperature and the lowest operational cost, is HP continuous regime. In the stage of actual design, investment cost of the system should also be considered.

NOMENCLATURE

A	Heat transfer area (m^2)
C	Building heat capacity (kJ/°C)
COP	Coefficient of efficiency of HP
Cost	Cost of annual energy consumption (USD)
C_p	Specific heat capacity of the fluid (kJ/kg°C)
g	Gravitational acceleration constant (9.81 m/s^2)
h_p	Total dynamic head (TDH) of pump (m)
H_u	Specific heat capacity of the fuel oil (kWh/kg)
LMTD	Logarithmic temperature difference (°C)
m_{fuel}	Annual fuel oil consumption (kg)
$\dot{m}$	Flow rate of the fluid (kg/s)
n	Performance coefficient of heating equipment
P_{el}	Unit selling cost of electricity (USD/kWh)
P_{fuel}	Unit selling cost of fuel oil (USD/kg)
Q_{boiler}	Annual energy consumption of fuel boiler (kWh)
$\dot{Q}$	Heat transfer rate (kW)
t	Time (s)
T	Temperature (K)
U	Overall heat transfer coefficient (kW/m^2°C)
W_{annual}	Annual energy consumption of HP (kWh)
$\dot{W}$	Net HP inlet power (kW)

Greek Letters

η	Efficiency
Δt	Time step (s)

Subscripts

0	Reference condition
b	Building
boil	Boiler
car	Carnot
cond	Condenser

cp	Circulation pump
el	Electric motor
eva	Evaporator
fuel	Fuel-oil
g	Geothermal fluid
heat	Heat exchanger
hen	Heat engine
heq	Heating equipment
i	Indoor or inlet
loss	Loss
m	Motor
net	Net
o	Outdoor or outlet
p	Pump
r	Return
ref	Refrigerant
s	Supply
supply	Supplied
tot	Total
w	Water between evaporator and heat exchanger
wp	Well pump

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