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Low Temperature Geothermal District Heating System Design. Case Study: IZTECH Campus, Izmir, Turkey

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ABSTRACT

Izmir Institute of Technology (IZTECH), founded in 1992, has a geothermal resource in the border of the Campus with a temperature of 33°C. At present, the Campus is still under development and each faculty building has an individual fuel-fired boiler heating system.

District heating system design consists of two parts: heating system design and piping network design. Piping network design is given in another paper at World Geothermal Congress, titled as "Piping Network Design for IZTECH Campus Geothermal District Heating System, Izmir, Turkey", extensively (Yildirim *et al.*, 2004). In this study, heating system design of IZTECH Campus is given in detailed. Because the production well has low geothermal fluid temperature; heat pump district heating system (HPDHS) is considered to be the best option. As an alternative, boiler-heating system is considered for comparison. Each heating system is simulated using hourly outdoor temperature data. For the simulations, the main control parameter is the indoor temperature of the buildings. Mathematical models are derived using Matlab and EES programs. Various heating regime alternatives have been studied for HPDHS for the various condenser outlet temperature and geothermal fluid flowrate and two of these alternatives are given in this study.

Furthermore, economic analysis has been done for each heating system alternative depending on investment and operational costs. According to the results of the economical analyses, while HPDHS has the highest investment cost with 3,040,125 US$, it has minimum operational cost. The alternatives are evaluated according to internal rate of return (IRR) method, which shows the profit of the investment. The results indicate that, the HPDHS has minimum 3.02% profit comparing with the fuel boiler district heating system (FBDHS) at the end of the 20-year period.

1. INTRODUCTION

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 energy 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 district heating system (GDHS) has several potential advantages. Using GDHSs, fossil fuel consumption and heating costs are reduced. On the other hand, air quality is improved. Additionally, the fire hazard of individual buildings is reduced, because of 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. The use of heat pumps 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 have been considered. Also FBDHS is planned to represent the existing heating system of the Campus. Three heating scenarios are considered depending on the heating period of the buildings in the Campus. 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. The investment and operational costs of the heating system alternatives, which are HPDHS and FBDHS, are calculated. Then, the investment costs are analysed according to internal rate of return (IRR) method (Yildirim and Gokcen, 2003, Erdogmus, 2003).

2. IZTECH CAMPUS AND EXISTING HEATING SYSTEM

The construction of the buildings of the Campus was started on November 1994. Number of the existing buildings has reached to 15 with 50,730-m^2-floor area and the Campus is still under development. Individual HVAC (Heating, Ventilation and Air Conditioning) systems are employed at each department. On the other hand, IZTECH Campus has a geothermal resource. Exploration studies in the field started in 1995. In 2002, 5 gradient wells were drilled and one of which is production well 33°C temperature and 30 kg/s flowrate.

Campus total heat load was determined as 11,207 kW (3,662 kW for existing buildings and 7,545 kW for the buildings, which are under construction/planned) in this study.

3. MODELING OF DISTRICT HEATING SYSTEM

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

- **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:

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

- **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 supplied and return temperature.

The heat transferred from the water is written as:

$$\dot{Q}_s = \dot{m}.C_p.(T_s - T_r) \tag{2}$$

The cooling of the district heating water is a non-linear 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

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

Addition to Eq. (2), in this case the rate of heat transferred from the waterside to the ambient air can be expressed in the following, equation:

$$\dot{Q}_s = U_{heq}.A_{heq}.LMTD_{heq} = \dot{Q}_{heq} \tag{4}$$

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

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

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

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

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. 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 supply or return temperature.

Fan-coils are considered as heating equipment since they can be used for cooling as well. Performance of the fan-coils is evaluated depending on various supply water temperature and ΔT (temperature difference between supply and return water temperature) based on the reference conditions. Figure 1 gives the increase ratio of the fan-coils. The need to extra heating equipment increases with increasing ΔT and decreasing supply water temperature.

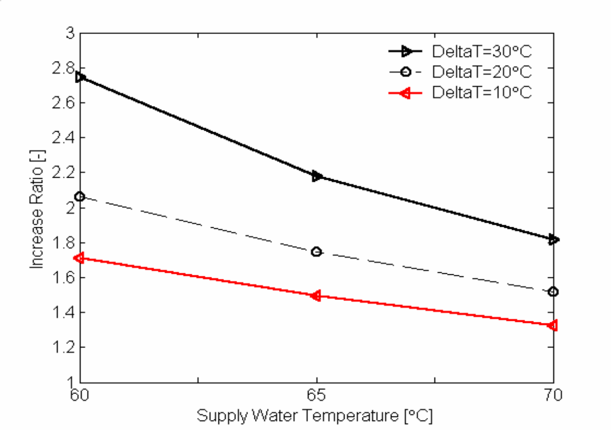

Figure 1: Increase ratio of fan-coils versus supply water temperature depending on various temperature differences between supply and return water temperature.

- **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 becomes as described in Eq. (7) (Valdimarsson, 1993).

$$\frac{dT_i}{dt} = \frac{1}{C}\dot{Q}_{net} = \frac{1}{C}(\dot{Q}_s - \dot{Q}_l) \tag{7}$$

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 by Eq. 11, which is an improved version of Eq. (7).

3.1. Heat Pump Model

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 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 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.

- 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.

As a general rule if,

- T_{gi} >40°C 'HPA'
- T_{gi}<40°C 'HPO' layouts are recommended (Harrison *et al.*, 1990).

HPO type heat pump is considered for the Campus DHS because existing geothermal temperature is 33°C at present. The considered heat pump heating system is shown in Figure 2. Because of the corrosion effects of geothermal fluid, a heat exchanger is also considered. Geothermal fluid passes through heat exchanger rather than evaporator.

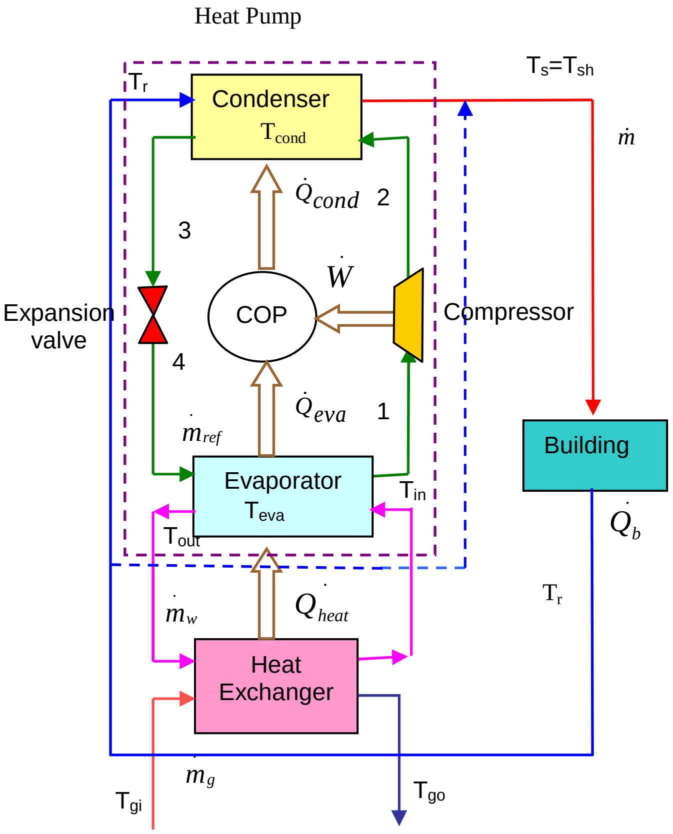

Figure 2: Considered HPO DHS.

According to Figure 2 the heat pump heat flows can be written as:

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

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

COP is the coefficient of the efficiency of the heat pump. The Carnot efficiency of the heat pump can be defined as the ratio of the heat released to work input. It is also often assumed that the thermal and mechanical losses in the cycle reduce the performance further to about 50% of the theoretical value. The COP becomes (Harrison *et al.*, 1990)

$$COP = 0.5 COP_{car} \tag{10}$$

3.1.1. Algorithm of Simulation Program

The parameters to be determined for the heat pump heating system (Figure 2) design are as follows;

1. Condenser outlet temperature (T_{sh}),
2. Geothermal fluid flowrate ($\dot{m}_g$),
3. Coefficient of the performance (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 (m) is not zero, heat pump heating system runs. Then return temperature from the radiators (T_r) is calculated by an iterative technique.

To calculate heat pump capacity, evaporator outlet temperature (T_{out}) should be calculated according to the supply water temperature. To be able to do that evaporator outlet temperature should be assumed. Then using Eq. (8, 9, 10) exact evaporator outlet temperature can be calculated by iteration. Then heat pump capacity and geothermal outlet temperature are calculated.

- **Condenser outlet temperature**

It is desirable to have a minimum temperature difference between condenser inlet and outlet to obtain high COP values for heat pumps. COP values versus geothermal flowrate are plotted in Figure 3 for various condenser outlet temperatures (40-55°C) at 35°C condenser inlet temperature and 33°C geothermal fluid temperature (T_{gi}).

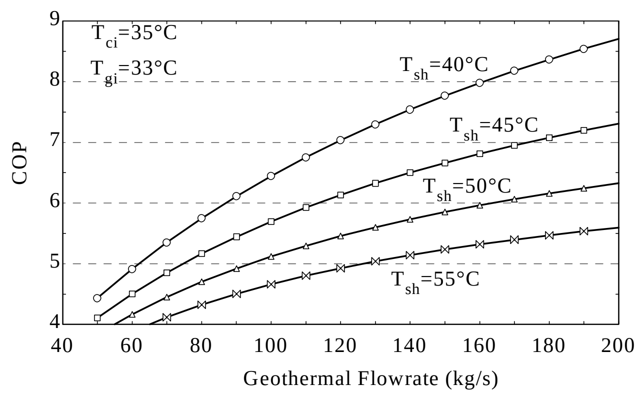

Figure 3: Relationship between geothermal flowrate and COP.

COP value increases with increasing geothermal fluid flowrate and decreasing condenser outlet temperature. But, there is a trade-off between condenser outlet temperature and economy of the system. Low condenser outlet temperature causes reduction in heating equipment performance, while increasing in the flowrate and network pipe diameter. But large heat pump units have high COP values with small temperature difference between supply (condenser outlet) and return (condenser inlet) temperatures.

In manufacturer's catalogues COP value is given as 5-8 for large heat pump units. An example for 45°C supply and 35°C return temperature (ΔT=10°C), COP is around 6 for large heat pump capacities. On the other hand, for 55°C supply and 35°C return temperature (ΔT=20°C), COP is around 4 and capacity of the heat pumps is small. Thus, the number of heat pump units is increased for large temperature differences. This also causes an increase in investment and operational costs. Therefore, there is a trade-off between COP and condenser outlet temperature. For a COP around 6, 45°C condenser outlet temperature gives the best result.

- **Geothermal fluid flowrate**

For a specified condenser outlet temperature, relationship among geothermal fluid flowrate, COP and heat pump capacity is shown in Figure 4. With a chosen COP of 6.2, Figure 4 gives a geothermal flowrate of 120 kg/s and a heat pump capacity of 1,877 kW. Required number of the production wells is 4 to meet the 120 kg/s geothermal flowrate requirement.

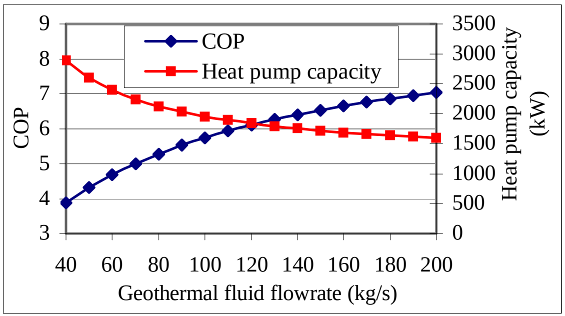

Figure 4: Relationship between geothermal fluid flowrate, COP and compressor work for 45°C condenser outlet temperature.

Depending on 45°C condenser outlet temperature, heating system is selected to have 45°C supply/35°C return temperature (ΔT=10°C). Consequently, flowrate in the Campus loop, is calculated as 179.4 kg/s with Eq.(2).

After determining the condenser outlet temperature and geothermal flowrate, heat pump heating system is simulated according to building heat loss, heating equipment, building energy storage and heating system models, which are explained in the previous sections.

For IZTECH Campus DHS, 4 separate heat pump units of the same capacity are considered because of the improved performance, reliability and operational flexibility (Harrison *et al.*, 1990). Each heat pump, which is employed with one HEX, is fed by each production well and heat pumps are operated depending on the outdoor temperature.

- If outdoor temperature is between 0-5°C, all heat pumps,
- If outdoor temperature is between 5-10°C, 3 heat pumps,
- If outdoor temperature is between 10-13°C, 2 heat pumps,
- If outdoor temperature is between 13-18°C, only one heat pump will be operated.

System is simulated using a control system with constant flowrate and variable return water temperature.

To calculate indoor temperature, Eq. (7) can be written in different matrix formation as:

$$\left[\frac{dT_i}{dt}\right] = \left[-\frac{U_b \cdot A_b}{C}\right]\left[T_i\right] + \begin{bmatrix}\frac{U_b \cdot A_b}{C} & \frac{\dot{m} \cdot C_p \cdot (T_s - T_r)}{C}\end{bmatrix}\begin{bmatrix}T_o \\ 1\end{bmatrix} \quad (11)$$

Equation 11 can be solved easily by discrete method (Nappa, 2000).

3.2. Fuel Boiler Model

Individual heating systems in the Campus are operated manually by technicians. Each technician turns on/off the system and changes the boiler set temperature according to his experience. Thus, the buildings in each group are heated in a different way.

Obtaining the required heat depends on the running time of the boiler and boiler set temperature. Fuel consumption of the boiler changes drastically depending on the boiler set temperatures. Thus, to obtain the best heating regime and boiler set temperature as a function of outdoor temperature, some alternatives are simulated. For fuel boiler DHS supply temperature is taken same as boiler set temperature. Results of the simulations of FBDHS are given in Table 1.

Table 1: Results of the simulations of FBDHS.

Alt. No	Boiler set temperature (°C)	Average indoor temperature during the working hours (°C)	Fuel-oil consumption of the boiler heating system (kg)
1	Tb_set=80	24.3	899,540
2	Tb_set=60	21.7	684,048
3	To<=5°C, Tb_set=80 5°C <To<=10°C, Tb_set=70 10°C <To<=14°C, Tb_set=60 14°C <To<=17°C, Tb_set=50	20.7	662,886
4	-3°C <To<=0°C, Tb_set=90 0°C <To<=3°C, Tb_set=81.6 3°C <To<=6°C, Tb_set=72.9 6°C <To<=9°C, Tb_set=63.8 9°C <To<=12°C, Tb_set=54.2 12°C <To<=15°C, Tb_set=43.7 15°C <To<18°C, Tb_set=31.7	20.12	618,500

Conventional heating systems are designed for peak load. Thus, the total heat load of the system is taken as 11,207 kW and heating equipment (fan-coil, radiator, etc.) are designed for supply and return temperatures of 90°C and 70°C, respectively.

4. RESULTS

4.1. Heating System Design

<u>For HPDHS</u>

- The peak flowrate is 120 kg/s and annual flowrate requirement is 281,124 tons. The average necessary flowrate is calculated as 74 kg/s.

- Geothermal fluid return temperature varies between 10 and 21°C.

- For steady state approach indoor temperature is assumed constant at balance temperature, 20°C. But for dynamic approach indoor temperature is calculated using Eq. (11). Figure 5 exhibits the indoor and outdoor temperature variations throughout the heating season.

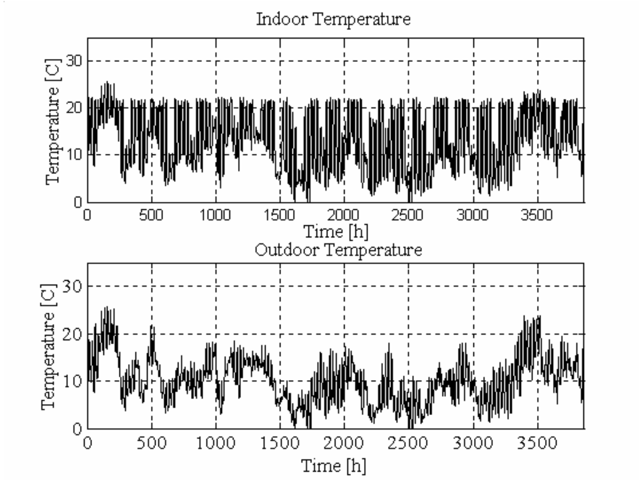

Figure 5: Variation of indoor and outdoor temperatures during the heating season.

For FBDHS

The Alternative 4 (Table 1), which uses a boiler set temperature recommended by Demirdokum (Dagsoz, 1998), is the best heating regime for FBDHS with least fuel consumption and best indoor temperature around 20°C (Figure 6).

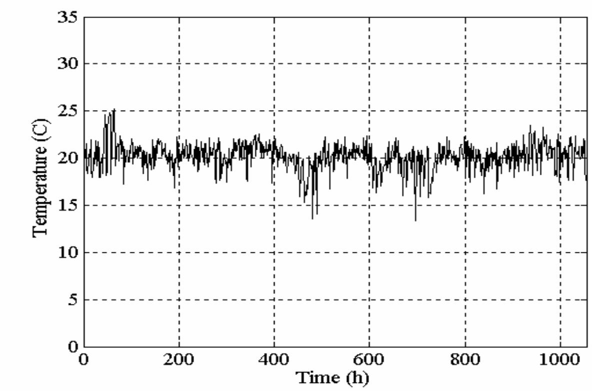

Figure 6: Variation of indoor temperatures for the best alternative of FBDHS.

4.2. Economical Analysis

For operational cost, 3 heating scenarios are considered depending on the heating period of the buildings in the Campus. While for Scenario 1 all buildings in the campus are heated between 8.00 a.m. and 17.00 p.m. during the week, for Scenario 2 the buildings are considered to be heated between 8.00 a.m. and 20.00 p.m. during the week. Various heating periods are considered for Scenario 3. According to Scenario 3, while the office buildings are heated between 8.00 a.m. and 20.00 p.m. during the week, Medical Centre, Sport Centre, Library are heated longer than office buildings, staff houses and dormitories are heated 24 hours a day. Annual heating requirements are calculated for each scenario according to degree-hour method as 5,129,892 kWh, 6,897,293 kWh and 9,612,556 kWh, respectively. Depending on the annual heating requirements of the buildings, annual operational costs are calculated. Total investment and operational cost of the heating system alternatives are given in Table 1 and the alternatives are evaluated according to internal rate of return (IRR) method**,** which shows the profit of the investment.

Table 2: Total investment and operational cost of the heating system alternatives for each scenario.

Alternative No	Total Investment Cost (US$)	Annual Operational Cost (US$/year)		
		Scenario 1	Scenario 2	Scenario 3
Alternative 1 (HPDHS)	3,040,125	127,843	171,889	239,556
Alternative 2 (FBDHS)	1,068,301	358,664	482,234	672,076

Investment and operational costs of the alternatives are given in Table 2. The Table indicates that HPDHS has maximum investment cost and it is approximately 3 times of FBDHS investment cost. The largest portions of investment cost for HPDHS and FBDHS are heat pump units and control systems, respectively.

The Table exhibits that operational cost of HPDHS is three times lower than FBDHS. The electricity consumption cost of the heat pumps, circulation and well pumps constitutes about 84% in total operational cost of the HPDHS. FBDHS, fuel oil cost has the largest portion of the total operational cost.

For the IRR calculations, differences between investment, operational and amortization cost of the alternatives are used. The amortization life is considered as 20 years. In IRR calculation, annual operational costs of the systems are assumed constant during the 20-year and difference between the operational costs is considered as profit. Cash flow is the difference between annual profit and amortization cost of the systems. For the Scenario 1, which is similar with the real case of the Campus, Alternative 1 and 2 are compared for amortization cost and the cash flow at the end of 20-year. The cash flow of the Alternative 1 (HPDHS) is 672,772 US$ depending on the Alternative 2 (FBDHS). IRR is calculated as 3.02% for Scenario 1.

5. CONCLUSIONS

In the study, HPDHS and FBDHS alternatives are investigated for IZTECH Campus.

For the Scenario 1, which is one of the heating scenarios and similar with the real heating case of the Campus, the HPDHS has 3.02% profit at the end of the 20-year period comparing with FBDHS. According to the results HPDHS is more attractive than FBDHS.

This study considers only heating system design and economical analyses for heating requirements. But each building is also equipped with cooling system. While considered HPDHS can be used for cooling requirements as well, for FBDHS, chillers should be installed to the system. Thus, the investment cost of the boiler heating systems increases. That means the HPDHS could be more attractive than FBDHS if cooling requirements of the buildings are considered.

NOMENCLATURE

A	heat transfer area, [m^2]
C	building heat capacity, [kJ/°C]
COP	coefficient of efficiency of heat pump [-]
C_p	specific heat capacity of the fluid, [kJ/kg°C]
LMTD	logarithmic temperature difference, [°C]
$\dot{m}$	flow rate, [kg/s]
n	performance coefficient of heating equipment [-]
$\dot{Q}$	heat transfer rate, [kW]
t	time, [s]
T	temperature, [°C]
U	overall heat transfer coefficient, [$kW/m^2°C$]
$\dot{W}$	net heat pump inlet power, [kW]

Greek Letters

Δt : Time step [s]

Subscripts

0	reference condition
b	building
car	carnot
cond	condenser
eva	evaporator
g	geothermal fluid
heat	heat exchanger
heq	heating equipment
i	indoor, inlet
l	loss
o	outdoor, outlet
r	return
ref	refrigerant
s	supply
w	water

REFERENCES

Bloomquist, R.G.: Geothermal district energy system analysis, design, and development. In *European Summer School on Geothermal Energy Applications*, The International Geothermal Training Centre of the University of Oradea, Romania, 213-253, (2001).

Dagsoz, A. K.: Chapter 19, Technical publication No:6 of Demirdokum Inc, 268-281, 1998 (in Turkish).

Erdogmus, A.B.: Economic Assessment of Balçova-Narlıdere Geothermal District Heating System, *M. Sc. Thesis*, Izmir Institute of Technology, 179 pp, 2003.

F-Chart Software: EES Engineering Equation Solver, F-Chart Software internet website, http://www.fchart.com/ees/ees.shtml., 2004.

Harrison, R., Mortimer, N. D., Smarason, O. B.: Geothermal Heating A handbook of Engineering Economics. 558 pp, 1990.

Nappa, M.: District heating modelling. University of Iceland, report, 47 pp, 2000.

The MathWorks: Matlab program, The MathWorks internet website, http://www.mathworks.com/products/, 2004.

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

Yildirim, N.: District heating system of IZTECH Campus and its integration to the existing system, *M.Sc Thesis*, Izmir Institute of Technology, 157 pp, 2003.

Yildirim, N., Gokcen, G. (2003): Modelling of Low Temperature Geothermal District Heating Systems, *Proceedings*, the First International Exergy, Energy and Environment Symposium (IEEES-1), Izmir, Turkey, (2003).

Yildirim, N., Gokcen, G., Toksoy, M.: Piping Network Design for Iztech Campus Geothermal District Heating System, Izmir Turkey, *Proceedings*, World Geothermal Congress, Antalya, Turkey, (2005).