District Heating System of IZTECH Campus and Its Integration to the Existing System

By

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To

My mother and my father

…

ABSTRACT

İzmir Institute of Technology (IZTECH), founded in 1992, is the third state university of İzmir. At present IZTECH Campus has individual fuel boiler heating systems for each faculty building and the Campus is still under development. But the Campus has also a geothermal source. In 2002, 5 gradient wells were drilled. Of these, one well has a geothermal fluid of 33°C is obtained but the actual flowrate of the geothermal fluid has not been measured yet. The aim of this Thesis is to investigate this source whether it can be used for district heating application for IZTECH Campus.

Mainly two heating system types have been considered;

- Heat pump heating system (HPHS) (using a renewable energy source, geothermal energy),
- Fuel boiler heating system (FBHS) (using a conventional energy source, fuel-oil).

HPHS is considered as HPO type since the existing geothermal fluid temperature is low. While HPHS is considered only as district, FBHS is considered as district and individual. Each heating system is simulated using hourly outdoor temperature data. For these heating simulations, the main control parameter is the indoor temperature of the buildings. Mathematical models are derived using Matlab [16] and EES [17] programs.

Various heating regime alternatives have been studied for HPHS for the various condenser outlet temperature and geothermal fluid flowrate. Consequently, the heating regime with 35°C condenser inlet and 45°C condenser outlet temperature with 120 kg/s geothermal fluid flowrate considered to be the best option.

FBHSs are also simulated for various boiler set temperatures. Boiler set temperatures have been recommended by Demirdöküm [39], is the best alternative with the least fuel consumption and best indoor temperature around 20°C.

Besides heating system simulations, piping network simulation is made using the software Pipelab [18]. The piping network of the Campus has been considered with two loops as geothermal and Campus. Each loop contains supply and return main. The location of the heat centre and the pressure loss per unit length are common design parameters for economy of the system. Therefore, several alternatives have been studied and because of the lowest investment and operational cost, Alternative 3, where the heat centre is in the middle of the Campus, is considered to be the best option with target pressure loss of 150 Pa/m. For installation type of piping network, underground (buried) pipeline installation is selected.

Furthermore, economic analysis has also been done for each heating system alternative depending on investment and operational costs. For operational cost, 3 heating scenarios are considered depending on the heating period of the buildings in the Campus. According to the results of economical analyses, while heat pump district heating system (HPDHS) has the biggest 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 according to the fuel boiler district heating system (FBDHS) at the end of the 20-year period. The profit increases with increasing operating period of the heating systems.

ÖZ

1992 yılında kurulan İzmir Yüksek Teknoloji Enstitüsü İzmir'in 3. devlet üniversitesidir. Gelişim içerisindeki kampüste her fakülte kendisine ait fuel-oil yakıtlı ısıtma sistemine sahiptir. Kampüs aynı zamanda bir jeotermal kaynağa sahiptir. 2002 yılında 5 gradyen kuyusu açılmış ve bunların birinden 33°C sıcaklığında jeotermal akışkan elde edilmiştir fakat akışkanın gerçek debisi henüz ölçülmemiştir. Tezin ana amacı, bu kaynağın İzmir Yüksek Teknoloji Enstitüsü Kampüsü için bölgesel ısıtma sistemi uygulamasında kullanılıp kullanılmayacağının araştırılmasıdır.

Başlıca iki tip ısıtma sistemi düşünülmüştür;

- Isı pompalı ısıtma sistemi (IPIS) (yenilenebilir bir enerji kaynağı kullanarak- jeotermal enerji),
- Kazanlı ısıtma sistemi (KIS) (konvensiyonel bir enerji kaynağı kullanarak-fuel-oil).

Mevcut jeotermal akışkan sıcaklığı düşük olduğundan IPIS, HPO tipinde düşünülmüştür. IPIS sadece bölgesel olarak düşünülürken, KIS bölgesel ve bağımsız olarak iki alternatif altında incelenmiştir. Her bir ısıtma sistemi saatlik dış hava sıcaklıkları kullanılarak simule edilmiştir. Simulasyonlarda ana kontrol parametresi binaların iç hava sıcaklığıdır. Matematik modeller, Matlab ve EES programları kullanılarak oluşturulmuştur.

IPIS'nde, farklı kondenser çıkış sıcaklıkları ve jeotermal akışkan debileri için, çeşitli ısıtma rejimi alternatifleri çalışılmıştır. Sonuçta, 35°C kondenser giriş, 45°C kondenser çıkış sıcaklıkları 120 kg/s jeotermal akışkan debisi ile birlikte en iyi seçenek olarak belirlenmiştir.

Tezde KISleri de, değişik kazan set sıcaklıkları için simule edilmiştir. Demirdöküm tarafından önerilen kazan set sıcaklıkları, düşük fuel-oil tüketimi ve 20°C'nin etrafında en iyi iç hava sıcaklığı ile en iyi alternatiftir.

Isıtma sistemi simulasyonunun yanında, boru hattı simulasyonu da Pipelab programı kullanılarak yapılmıştır. Boru hattı kampüs ve jeotermal olmak üzere iki çevrimden oluşmaktadır. Besleme ve dönüş hatlarını içeren her çevrim toprak altında gömülüdür. Sistemin ekonomisi için ısı merkezinin yeri ve birim boru boyundaki basınç kaybı genel dizayn parametreleridir. Bu nedenle çeşitli alternatifler çalışılmış ve en

düşük yatırım ve işletme maliyetlerinden dolayı ısı merkezi kampüsün ortasında olan 3. alternatif 150 Pa/m basınç düşümü hedefi ile en iyi seçenek olarak düşünülmüştür.

Ayrıca, her bir ısıtma sistemi alternatifi için yatırım ve işletme maliyetlerine bağlı olarak ekonomik analiz de yapılmıştır. İşletme maliyeti için, kampüsteki binaların ısıtma periyotlara bağlı olarak 3 ısıtma senaryosu düşülmüştür. Ekonomik analizlerin sonuçlarına göre, ısı pompalı bölgesel ısıtma sistemi (IPBIS) 3,040,125 US$ ile en büyük yatırım maliyetine sahipken, en düşük işletme maliyetine de sahiptir. Alternatifler, sistemlerin yatırımın karlılığını gösteren iç karlılık oranı (IKO) metoduna göre değerlendirilmiştir. Ekonomik analiz sonuçları, IPBIS'nin 20 yıllık periyodun sonunda fuel oil kazanlı bölgesel ısıtma sistemine (FKBIS) göre en az %3.02 oranında karlı olduğunu göstermektedir. Isıtma sistemlerinin işletme periyodunun artması ile kar yükselmektedir.

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NOMENCLATURE

a	: Heat load factor
A	: Heat transfer area, m^2
ac	: Amortization coefficient
A_p	: Cross section area of the pipe (m^2)
BV	: Book value (US$)
C_{air}	: Thermal mass of the air (kJ/°C)
CEP	: Coefficient of energy performance
COP	: Coefficient of performance of heat pump
Cost	: Cost of annual energy consumption (US$)
C_p	: Specific heat capacity of the fluid (kJ/kg°C)
C_v	: Specific heat of the air in the constant volume (0,718 kJ/kg°C)
D	: Diameter of the pipe (m)
f	: Friction factor
g	: Gravitational acceleration constant (9.81 m/s^2)
h	: Enthalpy (kJ/kg-K)
HDD_y	: Annual heating degree day
HDH_y	: Annual heating degree hour
h_p	: Total dynamic head (TDH) of pump (m)
H_u	: Specific heat capacity of the fuel oil (kWh/kg)
L	: Length of the pipe (m)
LMTD	: Logarithmic temperature difference (°C)
m	: Mass (kg)
$\dot{m}$	: Flow rate of the fluid (kg/s)
M	: Water mass (kg)
n	: Heating equipment coefficient
n_{hd}	: Number of heating degree-hours in one day
N_{hp}	: Number of existing heat pump unit in the heating system
n_{hp}	: Number of operating heat pump unit in the heating system
n_{hy}	: Number of heating degree-hours in one year
n_{month}	: Number of the heating month in the year

n_y : Number of the year

P_{el} : Unit selling cost of electricity (US$/kWh)

P_{fuel} : Unit selling cost of fuel oil (US$/kg)

P_g : Actual rate of heat transfer (kW)

Q_{aanual_boiler} : Annual energy consumption of fuel boiler (kWh)

Q_c : Total compressor work of heat pump (MWh)

Q_g : Geothermal supply which would by obtained by heat exchanger alone (MWh)

Q_{gh} : Combined supply of heat pump and heat exchanger (MWh)

Q_y : Annual heating energy (kWh)

$\dot{Q}$: Heat transfer rate (kW)

t : Time (s)

T : Temperature (°C)

U : Overall heat transfer coefficient (kW/m^2°C)

$V_{building}$: Volume of the building (m^3)

w : Velocity of the fluid (m/s)

W_{annual} : Total heat pump energy (kWh)

W_{el} : Total electric power input of heat pump motor (kWh)

$\dot{\mathbf{W}}$: Net heat pump inlet power (kW)

Z : Number of heating day in the month

Greek Letters

ΔP_{pipe} : Pressure drop of the pipe (Pa)

Δt : Time step (s)

ρ : Density of the fluid (kg/m^3)

ρ_{air} : Density of the air at 100 kPa and 20°C (1,188 kg/m^3)

η : Efficiency

Subscripts

0	: Reference condition
1	: Evaporator outlet
2	: Compressor outlet
3	: Condenser outlet
4	: Evaporator inlet
air	: Air
b	: Boiler outlet
bal	: Balance
boiler	: Boiler
building	: Building
car	: Carnot
cond	: Condenser
DIN	: DIN 4701 outdoor
el	: Electricity
elc	: Electric motor
eva	: Evaporator
fuel	: Fuel-oil
g	: Geothermal fluid
g0	: Maximum geothermal fluid
h	: Heating system
he	: Heating equipment
hi	: Heating temperature difference
heat	: Heat exchanger
i	: Indoor or inlet
in	: Evaporator water inlet
loss	: Loss
motor	: Motor
net	: Net
o	: Outdoor or outlet
out	: Evaporator water outlet
pump	: Pump
r	: Return

ref : Refrigerant

s : Supply

sh : Condenser outlet

supply : Supplied

w : Water between evaporator and heat exchanger

Superscripts

$\dot{}$ (dot) : Quantity per unit time

Abbreviations

DHS : District heating system

FBDHS : Fuel boiler district heating system

FBHS : Fuel boiler heating system

GDH : Geothermal district heating

GDHS : Geothermal district heating system

HEX : Heat exchanger

HPA : Heat pump assisted

HPDHS : Heat pump district heating system

HPHS : Heat pump heating system

HPO : Heat pump only

IFBHS : Individual fuel boiler heating system

IRR : Internal rate of return

IZTECH : İzmir Institute of Technology

Chapter 1

INTRODUCTION

District heating may be defined as the heating and/or cooling of two or more structures from a central heat source. Heat may be provided in the form of either steam or hot water and meet process, space or hot water requirements. 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. District heating systems 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 district heating systems [1].

Geothermal district heating systems (GDHSs) have several potential advantages. Using geothermal district heating systems, fossil fuel consumption and heating costs are reduced. On the other hand, air quality is improved. Additionally, the fire hazard of individual buildings, because of combustion does not occur in the buildings.

Geothermal district heating (GDH) has been given increasing attention in many countries during the last decade and many successful GDH projects have been reported. (Experience by researchers and engineers plays an important role in the design and control of geothermal district heating systems) [2]. Lund and Freeston [3][4] have reviewed the worldwide application of geothermal energy for direct utilization. They reported that an estimate of the thermal energy used at the beginning of 2000 is 190,699 TJ/yr (52,972 GWh/yr), and the distribution of the thermal energy used by category is 42% for bathing and swimming pool heating, 23% for space heating (42,926 TJ/yr), 12% for geothermal heat pumps, 9% for greenhouse heating, 6% for aquaculture pond and raceway heating, 5% for industrial applications, 2% for other uses and less than 1% each for agricultural drying, snow melting and air conditioning. Energy use growth in space heating since 1995 is 12%, or 2.3% annually. About 75% of the 42,926 TJ/yr utilization for space heating is estimated for district heating and the remainder for individual space heating. The majority of the district heating systems (DHSs) are in Europe, where the leaders are France and Iceland. The United States, on the other hand, dominates the individual space heating systems use, which typical of Klamath Falls, Oregon, and Reno, Nevada. Other countries with extensive district heating systems are China, Japan and Turkey. The share of Turkey in the worldwide thermal energy use is about 12.10% [5].

Turkey has a place among the first seven countries in abundance of geothermal resources around the world [6,7]. The General Directorate of Mineral Research and Exploitation (MTA) is conducting in charge of reconnoitre and exploration studies in Turkey. MTA initiated inventorial works and chemical analyses of the hot springs and mineral waters in 1962 [8]. The existence of more than 600 hot springs indicates that Turkey has an important geothermal energy potential, as illustrated in detail elsewhere [9]. The data accumulated since 1962 show that the estimated geothermal power and direct use potential are about 4,500 MWe and 31,500 MWt, respectively. However, only 2.3% of this potential has so far been utilized [10-12].

Turkey is among the first five leader countries in its geothermal direct use applications. The DHS applications were started with large scale, city based GDHSs. In this regard, city based geothermal district heating applications can be categorized in two groups, namely:(a) low temperature applications and (b) high temperature applications. There is one low temperature large-scale GDH application installed in Kirsehir, while high temperature applications are in larger numbers [5].

The investigations on geothermal energy in the country gained speed in the 1970s. Geothermal district heating applications have started in 1987 with the heating of 600 residences in Balikesir, Gonen. However, the utilization of geothermal energy could not show a fast increase due to scaling problems up to the early 1980s. Since then, important developments have been recorded in geothermal energy utilization. Recently, geothermal direct use applications have reached up to 52,000 residences equivalence of geothermal heating [5,7,12,13].

İzmir Institute of Technology (IZTECH), which was founded in 1992, is situated in Gülbahçe Campus. The Campus is under development, existing buildings are employed with individual HVAC (Heating, Ventilation and Air Conditioning) systems. In Chapter 2, IZTECH Campus and the existing heating system is introduced briefly. Since Gülbahçe geothermal field is in the vicinity of the Campus, the main purpose of this Thesis is to investigate the possibility of installing a GDHS.

Several explorations have been done in Gulbahce geothermal field since 1995. One of which was a geochemical study conducted by Giese [14]. Geothermometry was applied to determine the reservoir temperature using chemical analysis of the outcrop water, which is at 35°C. The chemical analysis showed that geothermal fluid has scaling and corrosion potential. The contents of Li and Si were interpreted to be generated by the reservoir temperature of 60°C in minimum but to be maybe more than 100°C. In 2002, 5 gradient wells were drilled in the field. One of which was production well with a depth of 260 m and a temperature of 33°C.

The methods by which heat is extracted from geothermal fluid depend strongly upon the temperature of the fluid and the nature of the heating application. There are two basic methods of heat extraction 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 [15].

In the Thesis, two types of heating systems are studied namely; (a.) Heat pump district heating system (HPDHS), and (b.) Boiler (conventional) heating system. Because of the production well has low geothermal fluid temperature, 33°C, HPDHS is considered to be the best option. Boiler heating system is considered as district heating due to compare considered HPDHS and also as individual to represent the existing heating system.

The Campus is mostly used during working hours and the existing heating system runs only during this period. Thus, intermittent heating strategy has been applied for the proposed heating systems. For intermittent heating simulation, the heating period is considered to be between 9.00 a.m. and 17.00 p.m. hours. In the simulation, the system is turned on one hour earlier and turned off at the end of the period. For these heating simulations, the main control parameter is the indoor temperature of the buildings. Mathematical models are derived using Matlab [16] and EES [17] programs using hourly outdoor temperature data. Besides heating system simulations, a network simulation is made using the software Pipelab [18], which uses the Matlab program as a basis.

Weather characteristic is one of the major criteria in designing a heating system. Chapter 3 gives weather characteristics of Izmir city and also energy estimating methods, which are used for determining the size of heating system.

Chapter 4 describes the geothermal district heating systems and major equipments, in particular heat pump heating systems.

In Chapter 5, considered district heating alternatives are explained and the simulation procedures are given. Depending on these alternatives, heating system and network is designed in Chapter 6 and an economical analysis of the alternatives is given. Finally, the results and some recommendations are made in Chapter 7.

Chapter 2

IZTECH CAMPUS

2.1 General Information

IZTECH was founded in 1992 as the third State University of İzmir in order to use, develop and produce advanced technologies.

Figure 2.1: Location of IZTECH Campus

The Campus is located in Urla, about 40 km. west of downtown Izmir with a highway connection. Facing the blue waters of the Aegean Sea, it has a total area of 3,500 ha. A location of the Campus is shown in Figure 2.1.

IZTECH is, comprised of three Faculties, those of Engineering, Architecture, and Science. IZTECH started offering 4 year undergraduate programs since the Academic Year 1998-1999. In the fall semester of the 2002-2003 academic year, the number of students is 1,418, 913 students are in the undergraduate programs and 505 students are in the graduate programs. The total number of staff is 698 with 406 academics [19].

2.2 The Existing Heating System And Heating Load

The construction of the Campus buildings of the Campus was started on November 1994. The number of the existing buildings reached to 15 with a floor area of 50,730 m^2. At present, 8 buildings, dormitories and staff houses are under construction. Total number of buildings is planned as nearly 110. Buildings have different construction properties and elements. Existing buildings and their heat loads are given in Table 2.1. The total heat load of the existing buildings is about 3,662 kW.

Table 2.1: Total heat load of the existing buildings

	Building Name	Heat Load (kW)	Total Building Usage Area (m^2)	Total Building Usage Volume (m^3)
Engineering Faculty	Main Building	162	2,852	9,412
	Classrooms Building	127	1,870	6,171
	Laboratories	135	1,948	6,428
	Laboratories	160	2,913	9,613
	TOTAL	584	9,583	31,624
	Mechanical. Engineering Laboratories	193	2,141	7,065
	Mechanical. Engineering Laboratories	228	1,805	5,957
	TOTAL	421	3,946	13,022
Architecture Faculty	Studio Building	310	4,800	17,280
	Main Building	272	4,897	17,629
	TOTAL	582	9,697	34,909
Science Faculty	Main Building	213	3,538	11,675
	Laboratories	203	3,276	10,811
	Laboratories	186	3,606	11,900
	TOTAL	602	10,420	34,386
Rectorship	Rectorship Building	353	2,994	9,880
	Precidency Building	442	5,190	17,127
Cafeteria		412	4,700	23,500
Incubator Building		267	4,200	31,500
CAMPUS TOTAL		**3,662**	**50,730**	**195,948**

The existing heating system in the Campus consists of individual fuel boilers employed five heat centres. Table 2.2 gives information about these heat centres. Figure 2.2 exhibits one of these centres, Engineering Faculty Heat centre.

Table 2.2: General information of the existing heating system in the Campus.

Heat centre		Number of Fuel Boiler	Capacity of Fuel Boiler (kW)	Total Capacity of Fuel Boilers (kW)	Capacity of Fuel Tank (tone)
1	Rectorship Building	1	465	465	20
2	Presidency Building	2	291	582	25
3	Faculty of Engineering	3 1	349 465	1,512	21
4	Faculty of Science	1 1	465 698	1,163	20
5	Faculty of Architecture	2	465	930	21

Figure 2.2: A view of the Heat centre in the Faculty of Engineering

Each heating system runs manually by a technician, who is responsible from each heat centre. Generally, heating systems are turned on when working hour, which is between 9.00 a.m. and 17.00 p.m, starts. But running period of the heating system and set temperature of the fuel boiler change depending on the outdoor temperature. Fuel consumption of the heating system is affected by heating regime, but it is not measured properly. The fuel consumption measurements are listed in Table A 1 in Appendix A. As it can be seen from the Table, consumption measured until April 2002. Therefore, at present it is quite difficult to say, exactly, how much fuel consumed annually in each heating system. Figure 2.3 displays annual fuel consumptions of the Campus for 2000 and 2001.

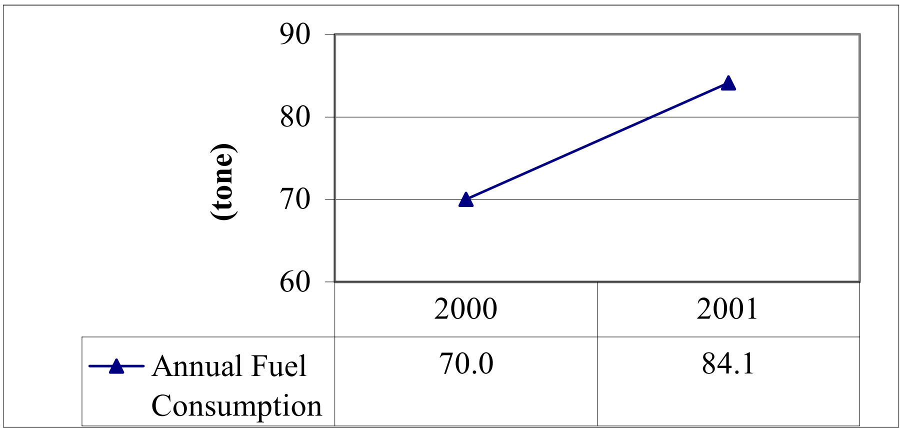


Figure 2.3: Fuel consumption of the Campus.

Electricity consumption of the faculties is shown in Figure 2.4. Figure exhibits an increase in electricity consumption through the years. But, while Architecture Faculty and Science Faculty are showing a slight increase, electricity consumption of Engineering Faculty increases drastically and has the highest share in the Campus. This is because the Engineering Faculty has the largest building area and employes more staff and students than the other faculties as it is given in Table 2.3.

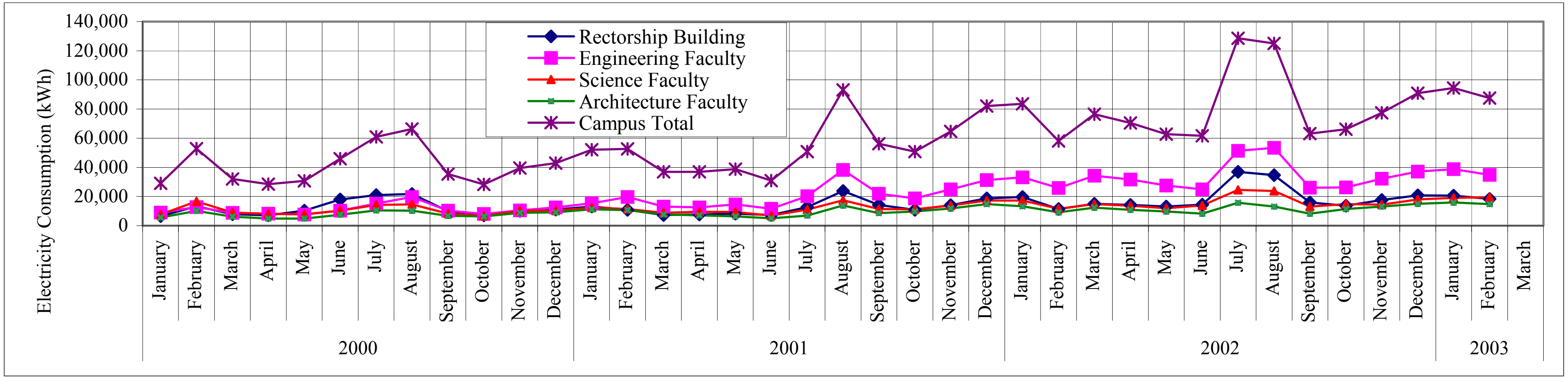


Figure 2.4: Electricity consumption of IZTECH Campus (Appendix A Table A 1).

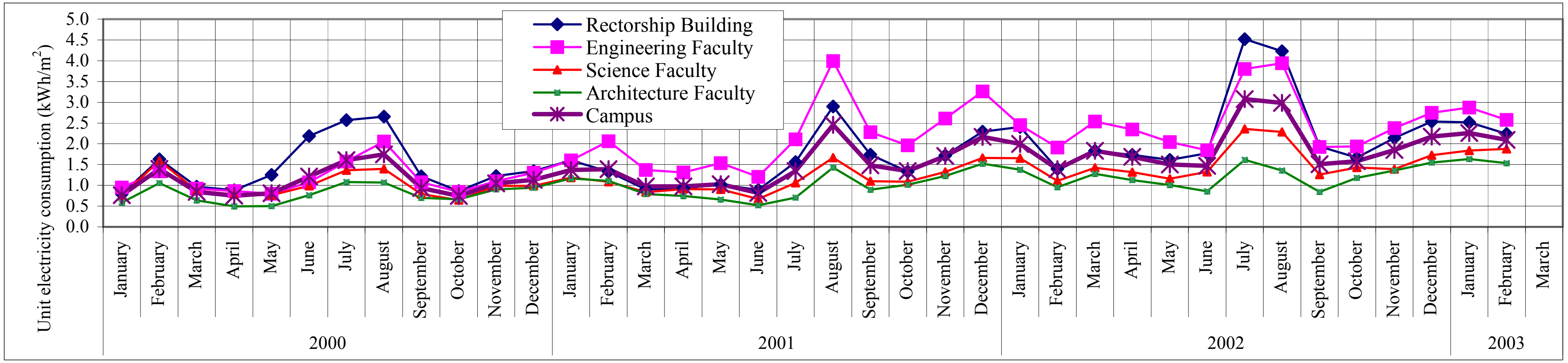


Figure 2.5: Unit electricity consumption per square meter of IZTECH Campus (Appendix A Table A 1).

Figure 2.5 shows the unit electricity consumption (kWh/m^2) of the Campus. Unit electricity consumption values of the Rectorship and Presidency Buildings are generally very close to the Campus average, except summer periods. That means they can represent the unit electricity consumption of the Campus. Architecture and Science Faculty have lower unit electricity consumption than the Campus average while Engineering Faculty has higher.

Table 2.3: Campus data in 2001.

	Unit	Rector. Build.	Pre. Build.	Eng. Faculty	Science Faculty	Arch. Faculty	Campus Total
Heat Load	(kW)	353	442	1,004	602	582	2,983
Conditioned Building Area	(m^2)	2,994	5,190	13,529	10,420	9,697	41,830
Number of Staff		78	143	119	93	105	538
Number of Students			82	364	152	256	854
Number of Computer		49	107	207	94	115	572
Number of Printer		37	31	43	31	39	181
Annual Electricity Consumption	(kWh)	149,011		242,339	140,525	114,187	646,062
Annual Water Consumption	(tone)	5,839		11,384	10,410	5,499	33,132
Annual Fuel Consumption	(tone)	19		28	20	18	84

To determine the seasonal electricity consumption and the relationship between monthly electricity consumption and outdoor temperature, Figure 2.6 is plotted based on 2002 data. This figure indicates that in winter electricity consumption increases with decreasing outdoor temperature except February. The electricity consumption decreased in February 2002 because of the one-week religious holiday. Electricity consumption is nearly constant, when heating and cooling systems are not operated (spring and autumn) and the consumption increases drastically during the summer period because of the cooling systems, which are turned on in July and August.

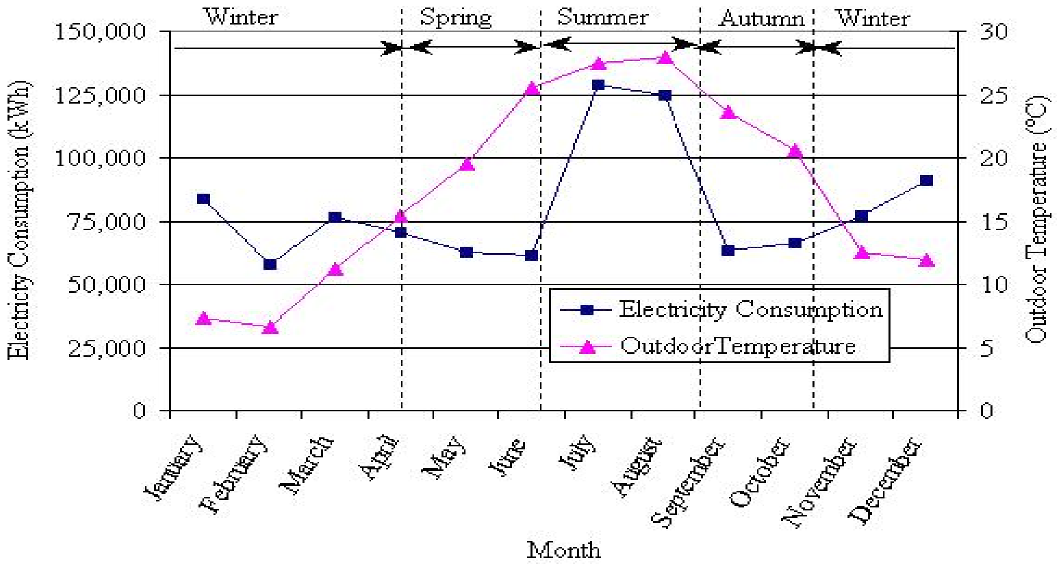
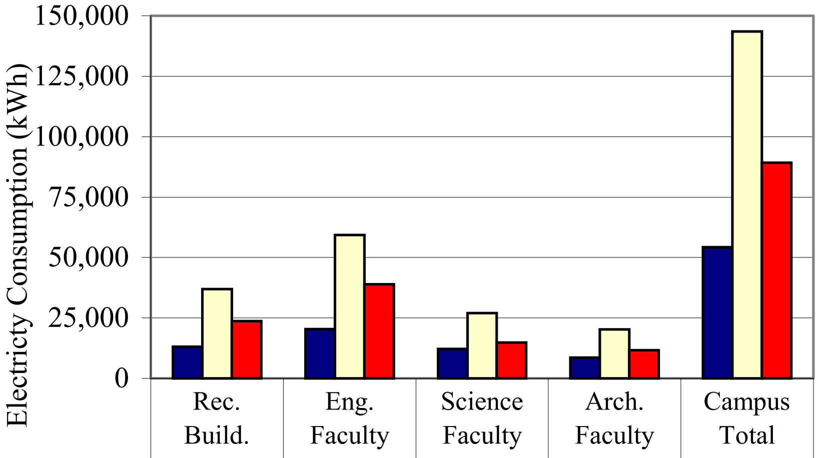


Figure 2.6: Relationship between electricity consumption and outdoor temperature for 2002 (Appendix A Table A 2).

Effect of the cooling systems on the electricity consumption is illustrated in Table 2.4 and Figure 2.7. Electricity consumption of the Campus increases nearly 164% during the summer season.

Table 2.4: Effect of the cooling systems on the electricity consumption

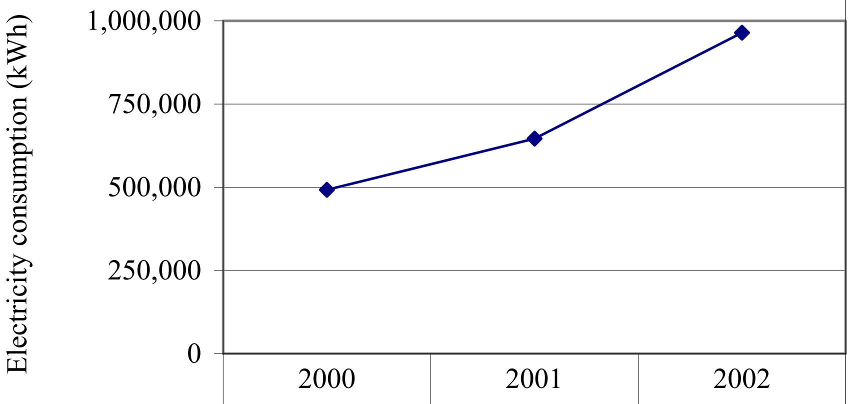
Case	Electricity Consumption (kWh)						
	Date	Working Day	Rec. Build.	Eng. Faculty	Science Faculty	Arch. Faculty	Campus Total
Cooling systems are not operated	June (04-18.6.2001)	10	3,716	5,956	3,816	2,576	16,064
	June (18.6-02.7.2001)	10	3,850	6,020	3,432	2,670	15,972
	July (02-16.07.2001)	10	4,090	6,267	3,672	2,430	16,459
	July (16-20.07.2001)	5	1,492	2,177	1,260	892	5,821
	TOTAL	35	13,148	20,420	12,180	8,568	54,316
Cooling systems are operated	July (22-27.07.2001)	5	3,184	5,761	3,036	884	12,865
	July (27.7-15.8.2001)	12	14,288	21,112	10,680	9,312	55,392
	August (15-27.08.2001)	8	9,736	17,236	7,368	5,326	39,666
	August (27.8-10.9.2001)	8	9,704	15,251	5,896	4,720	35,571
	TOTAL	33	36,912	59,360	26,980	20,242	143,494
Electricity consumption difference		(kWh)	23,764	38,940	14,800	11,674	89,178
		%	180.7	190.7	121.5	136.3	164.2



	Rec. Build.	Eng. Faculty	Science Faculty	Arch. Faculty	Campus Total
when cooling systems are not operated (kWh)	13,148	20,420	12,180	8,568	54,316
when cooling systems are operated (kWh)	36,912	59,360	26,980	20,242	143,494
Electricity consumption difference (kWh)	23,764	38,940	14,800	11,674	89,178

Figure 2.7: Effect of the cooling systems on the electricity consumption

Annual electricity consumption of the Campus by years is shown in Figure 2.8. According to the data annual increase ratio of the electricity consumption is 31% in 2001 and 49% in 2002. The ratio is increasing by years with the development of the Campus.



	2000	2001	2002
Annual Electricity Consumption (kWh)	492,361	646,061	964,082

Figure 2.8: Annual electricity consumption of the Campus.

Chapter 3

WEATHER ANALYSIS AND ENERGY ESTIMATING METHODS

The main input signal to a district heating simulation model is the weather. When a district heating system is being designed, the characteristic climate condition is the main design criterion, along with the consumer behavior. The purpose of the system is to fulfil the wishes of the consumer regardless of the weather situation. The influence of weather on the operation of district heating systems is mainly through the outdoor temperature [20].

In simulating a DHS first step is to determine the heating energy requirements of the heating-cooling system. The energy requirements and fuel consumption of the heating and/or cooling systems have a direct impact on the cost of the system. Prediction of energy requirements may be difficult as many variables are involved for a long time. Weather data is important information for prediction of the heat requirements. Depending on outdoor temperature energy estimating methods, such as heat load factor, degree-day and degree-hour methods are used to predict heat requirement of the systems.

Energy estimation often lead to an economic analysis that aims to establish the cost effectiveness of conservation measures. Thus, a through energy analysis provides intermediate data, such as time of energy usage and maximum demand, so that utility charges can be accurately estimated. [21].

İzmir, located in the western part of Turkey, has a Mediterranean Climate. According to a study conducted by Turkish Society of HVAC & Sanitary Engineers [22], 1993 was determined as a typical year for Izmir, between 1976 and 1996. Typical year can be explained as the most representative year in terms of outdoor temperature in the investigated time period.

Figure 3.1 shows the outdoor air temperature duration curve of the year 1993 for İzmir.

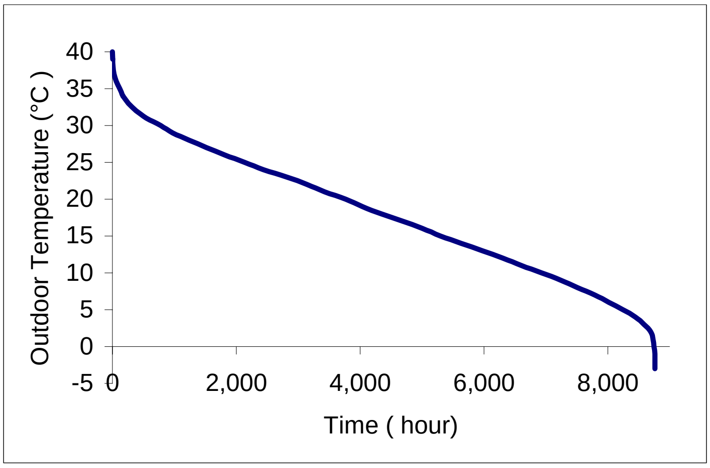


Figure 3.1: Outdoor air temperature duration curve for İzmir [23].

Maximum, minimum and average outdoor temperatures of this typical year are shown in Figure 3.2.

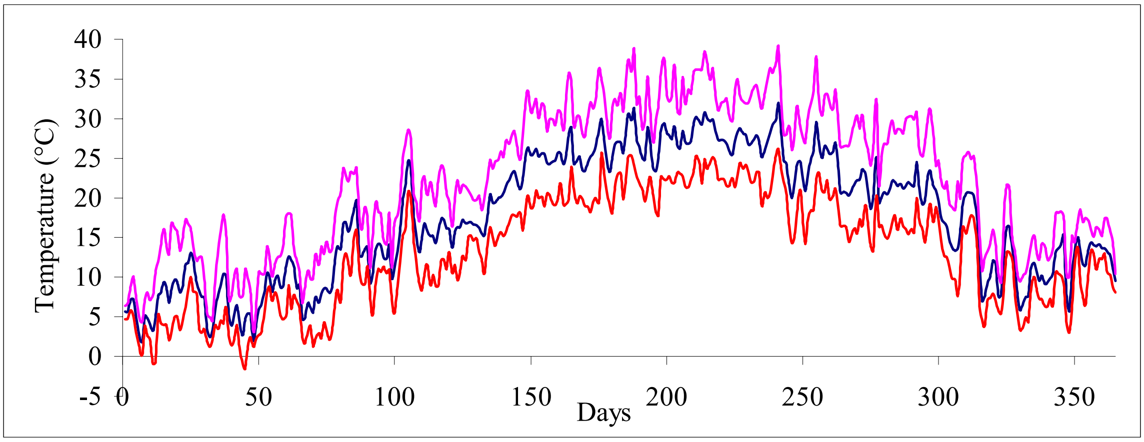


Figure 3.2: Daily maximum, minimum and average outdoor air temperatures of the year 1993 [23].

3.1 Heating-Cooling Energy Estimating Methods

3.1.1 Heat Load Factor

Size of the systems, such as heating, cooling and HVAC systems, solar collectors, cooling tower, are selected depending on peak load which is calculated according to the coldest design temperature. Peak load depends on the weather conditions for heating applications completely and for cooling applications partly. Thus, temperature variations in the season are considered for annual energy analyses of the systems. Analyse of temperature variations are done by heat load factor, which is calculated by outdoor temperature for 24 hours, or daily or monthly average outdoor temperatures in the heating season. Heat load factor can be explained simply as a ratio of average heat load of the month to peak load of the system and it calculates as:

$$a = \frac{T_{bal} - T_o}{T_{bal} - T_{DIN}} \tag{3.1}$$

T_{DIN} is the lowest two-day average outdoor temperature, which is seen at least ten times during the last 20 year-period according to DIN 4701 standard for determination of the outdoor design temperature [22]. This value is 0°C for İzmir [24].

According to a study, performed by the Turkish Society of HVAC & Sanitary Engineers, outdoor design temperature can be taken as 1.6°C by 99% percentile or 0.3°C by 99.6% percentile for İzmir. That means the encountered outdoor temperature is below 0.3°C only for 35 hours in a year. These values are given in Table 3.1 [22].

Table 3.1: Outdoor design temperature for İzmir.

	DIN 4701	99% Percentile	99.6% Percentile
Outdoor design temperature (°C)	0	1.6	0.3

Annual energy requirement of the system for heating can be calculated as [25]:

$$Q_y = \sum_{i=1}^{n_{month}} (24 \cdot Z_i \cdot Q_0 \cdot a_i) \tag{3.2}$$

3.1.2 Degree-Day And Degree-Hour Methods

There are various methods for the calculation of monthly or annual energy consumption; however, degree-hour or degree-day methods are the simplest. The degree-day method can provide a simple estimate of annual energy loads, which can be quite accurate if the indoor temperature is constant. Degree-hours and degree-days are conventionally calculated for a balance (or a base) temperature. The degree-hour method is one of the proper methods to use in order to forecast consumption of fuel in residential heating. Degree-days or degree-hours for a balance temperature of 18°C in Europe or 18.3°C in the United States have been widely tabulated, based on the observation that this has represented average conditions in typical buildings.

The rate of energy consumption of a heating system is

$$\dot{Q}_h = \frac{U_{building} \cdot A_{building}}{\eta_h} \cdot [T_i - T_o(t)]^+ \tag{3.3}$$

The plus sign superscript on the bracket indicates that only positive values are to be counted. The value of T_i depends on the type and use of the building, degree of activity of occupants, duration of occupancy, presence of radiant heat sources such as large lighting loads or glass exposures, and outdoor design conditions.

The yearly energy consumption is the time integral of the instantaneous consumption over the heating or cooling season. If the simplifying assumption is made that UA/η_h is constant then the annual heating consumption can be written as an integral:

$$Q_y = \frac{U_{building} \cdot A_{building}}{\eta_h} \cdot \int [T_i - T_o(t)]^+ \cdot dt \tag{3.4}$$

In the practice, this integral is approximated by a sum of averages over short time intervals (hourly or daily), and the result is called degree-hours or degree-days.

If the heating temperature is defined as

$$T_{hi} = (T_{bal} - T_o)^+ \tag{3.5}$$

yearly heating degree-days can be expressed as

$$HDD_y = \sum_{i=1}^{n_{hy}} T_{hi} \tag{3.6}$$

Here T_{bal} is the balance temperature. T_{bal} is defined as the temperature at which internal heat generation (from the sun, occupants, lights, etc.) just balances transmission and infiltration losses [26].

For 20°C balance temperature and 0°C design outdoor temperature, the degree-day value for İzmir is found, as 1737.8, according to [22]. These values are shown in Appendix A, Table A 4.

One can obtain the yearly heating degree-hours in the following form [26].

$$HDH_y = \sum_{i=1}^{n_{hd}} T_{hi} \tag{3.7}$$

Annual heat requirement can be estimated from the following equations by the help of degree-day and degree-hour methods;

$$Q_y = \frac{24 \cdot Q_0 . HDD_y}{T_{bal} - T_{DIN}} \tag{3.8}$$

$$Q_y = \frac{Q_0 . HDH_y}{T_{bal} - T_{DIN}} \tag{3.9}$$

Chapter 4

GEOTHERMAL DISTRICT HEATING SYSTEMS

Geothermal district heating system is defined as the use of one or more production fields as sources of heat to supply thermal energy to a group of buildings.

A geothermal district heating system comprises three major components;

1) *Heat production*. Which includes the geothermal production and recharge fields, conventional fuelled peaking station, and wellhead heat exchanger.

2) *Transmission/distribution system*. Which delivers the geothermal fluid or geothermally heated water to the consumers.

3) *Central pumping stations and in-building equipment.* Geothermal fluids may be pumped to a central pumping station/heat exchanger or heat exchangers in each building. Thermal storage tanks may be used to meet variations in demand [27].

The sole purpose of a district heating system is to supply adequate heat to its consumers. The consumer uses the heat to maintain indoor temperature at a reasonably constant level and counter for building heat loss to the surroundings, and for preparation of domestic hot tap water. The benefits of this method of energy distribution are possibilities of centralised heat generation with an associated economy and a low emission to the environment [20].

The initial step of a GDHS feasibility study is to provide the thermal load inventory for determining whether or not district heating system could be technically and economically viable, and for developing a strategy for improving the energy efficiency and district heating system favourability through land use planning.

Thermal load density (load per unit of land area), that can be determined from the thermal load inventory, is critical to the feasibility of district heating system because it is one of the major determinants of the distribution network capital and operating cost. In a district heating system the cost of the distribution network constitutes the largest portion (up to 70% percent of the total capital investment) [1]. Thus, thermal load density should be sufficient to support such a system.

Table 4.1 gives the thermal load density and the fovarurability relation for various type of land use.

Table 4.1: Favourability Based on Thermal Load Density [1].

Type of Land-Use	Thermal Load Density (MW/ha)	Desirability for District Energy
Downtown; high rises	Greater than 0.70	Very favourable
Downtown; multi-storied	0.51 - 0.70	Favourable
City core; commercial buildings & multi-family apartments	0.20 - 0.51	Possible
Two-family residential	0.12 - 0.20	Questionable
Single-family residential	Less than 0.12	Not possible

Once it is decided that to install a GDHS, is possible or favourable, the next step is to decide the heat extraction method.

4.1 Heat Extraction Methods

The methods by which heat is extracted from geothermal fluid depend strongly on the temperature of the fluid and on the nature of the heating application. High temperature fluids are highly versatile and can be used for direct heating or for electricity production. With lower fluid temperatures it becomes increasingly difficult to extract the heat. In these cases it is important to match the fluid with heating applications, which require temperatures that are lower than the fluid temperature. With very low fluid temperatures heat pumps are required to match supply and demand temperatures.

Based on the fluid temperature there are two basic methods of heat extraction; direct heat exchange and heat pumps.

4.1.1 Direct Heat Exchange

Direct heat exchange is a conventional heat recovery technology, which is shown in Figure 4.1. The heat is transferred by passive conduction processes from the higher temperature geothermal fluid either directly to room heaters or through primary heat exchangers to the lower temperature fluids supplying the heating system. The supply temperature of the geothermal fluid is fixed by the reservoir conditions and the amount of heat, which can be extracted across the heat exchangers, is limited by the temperatures of the fluids returned from the heating application and also by the smallest flow.

4.1.2 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. Or, alternatively, they can be used to reduce return temperatures and improve heat recovery. Whichever way they are used the economic viability is usually marginal and careful optimisation is required. (Figure 4.1).

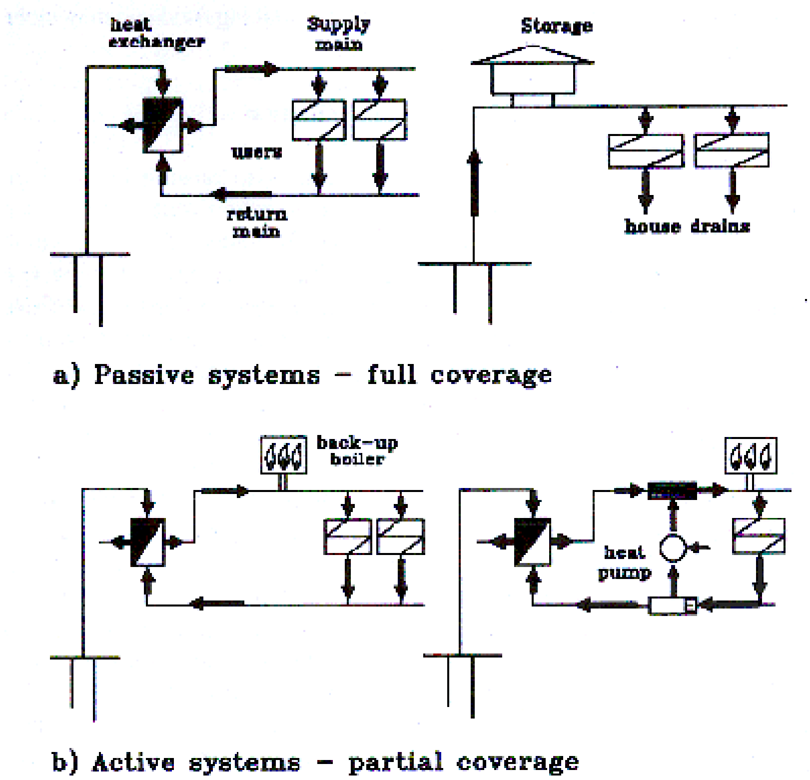


Figure 4.1: Surface systems; basic arrangements [15].

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.

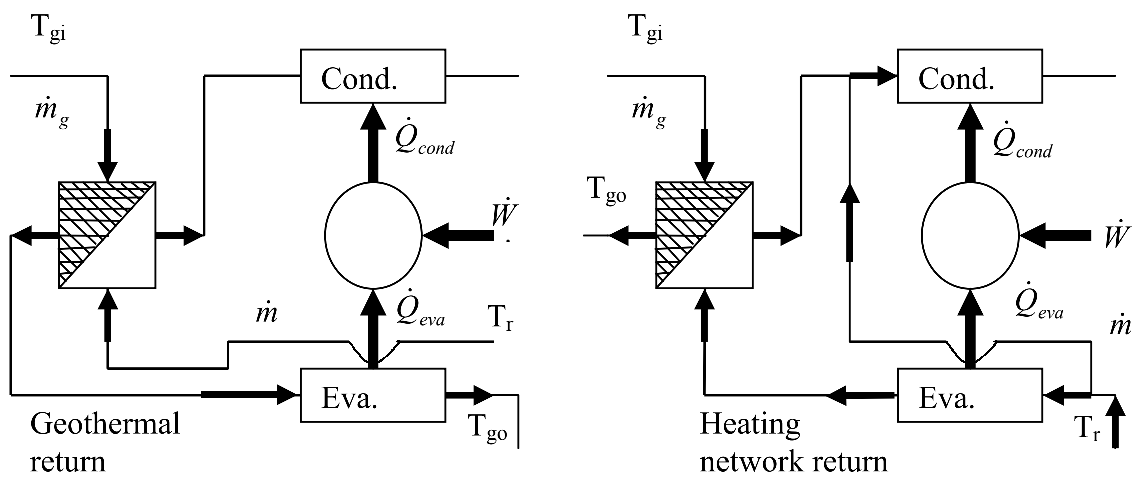
In large heat pump schemes it is common to use a number of separate heat pump units grouped together so that they operate effectively as one heat pump. The advantages of using a group of smaller heat pumps as opposed to one large one include

- Improved performance because each unit works between closer evaporator and condenser temperatures.
- Improved reliability and easier control because individual units can shut down independently of the group.

4.1.2.1 Heat Pump Types And Performance

Depending on the type of heat supply, 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 (Figure 4.2).



a) Direct evaporator

a) Indirect evaporator

Figure 4.2: Heat Pump Assisted (HPA) heat transfer schematic layouts [15].

- 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. (Figure 4.3).

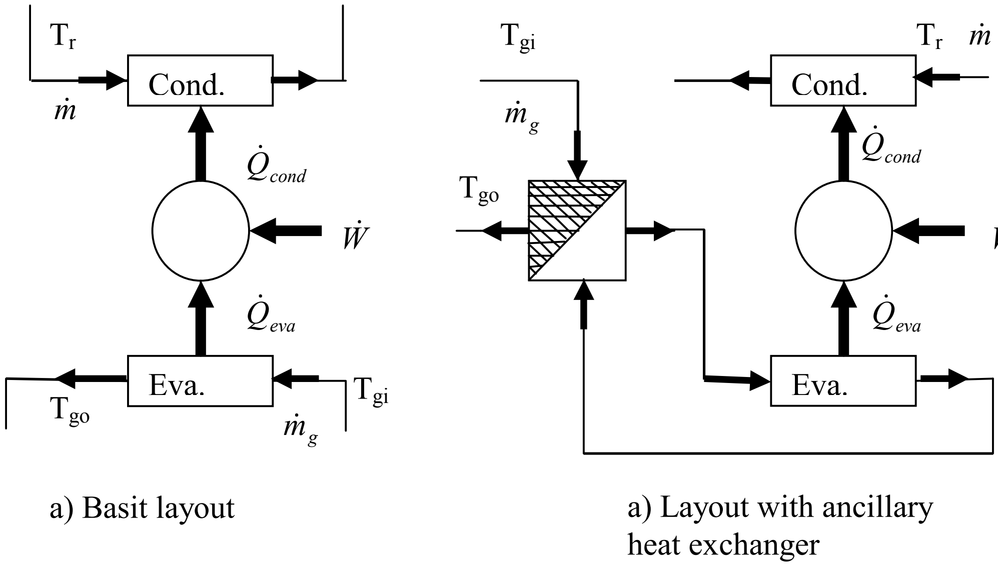


Figure 4.3: Heat Pump Only (HPO) heat transfer schematic layouts [15].

4.1.2.1.1 Heat Pump Assisted Heat Transfer (HPA)

For heat pump applications, the direct evaporator type is the simplest; the evaporators are located on the geothermal return main and extract residual heat directly from the brine leaving the primary heat exchanger. The action of the heat pump does not affect the operating conditions of the primary heat exchanger but the reverse is not true. With this arrangement there are clear and distinct heat transfer paths

- Heat is transferred by simple heat exchange and the heat flows are unaffected by the action of the heat pump.
- The residual heat extracted from the brine is transferred by the heat pump to the heating system supply.

Although this arrangement has the advantage of simplicity there can be problems of the corrosion if saline fluids pass through the evaporators [15].

4.1.2.1.2 Heat Pump Only Heat Transfer (HPO)

This arrangement tends to be used when the temperature of the supply fluids, aquifer, brines or ground waters, are 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. In this arrangement, the heat pump operates in a way which is closest the simple arrangement and, indeed, if the flows on the geothermal side are large enough, this will act as a infinite constant temperature reservoir. The heat pump upgrades the heat extracted so that the condenser outdoor temperature is higher than the geothermal supply temperature.

Bypass connections may also be used in this arrangement. Thus, in those situations where the temperature stretch is too low for the heat pump to operate, a condenser bypass can be used to raise the condenser output temperature and restore normal operating conditions.

A coefficient of energy performance can be defined as follows

$$CEP = \frac{Q_{gh} - Q_g}{Q_c} \tag{4.1}$$

In the Figure 4.4 overall energy performance of heat pump assisted and heat pump only layouts at different fluid temperatures is shown. As it can be seen from the figure, clearly the dividing line is at about 37°C. As a general rule it may be that;

- above T_{gi}=40°C 'Heat Pump Assisted' layouts give better performance
- below T_{gi}=40°C 'Heat Pump Only' layouts are better [15].

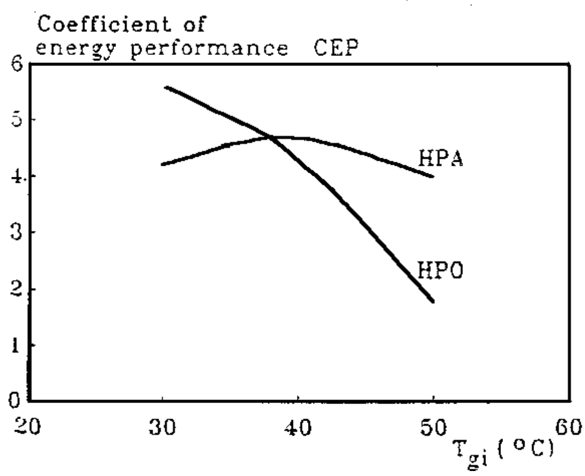


Figure 4.4: Comparison of overall energy performance of heat pump assisted and heat pump only layouts at different fluid temperatures T_{gi} [15].

In the following sections, equipments of GDHS will be mentioned.

4.2 Geothermal District Heating System Basic Equipments

Here brief information about these equipment will be given as background.

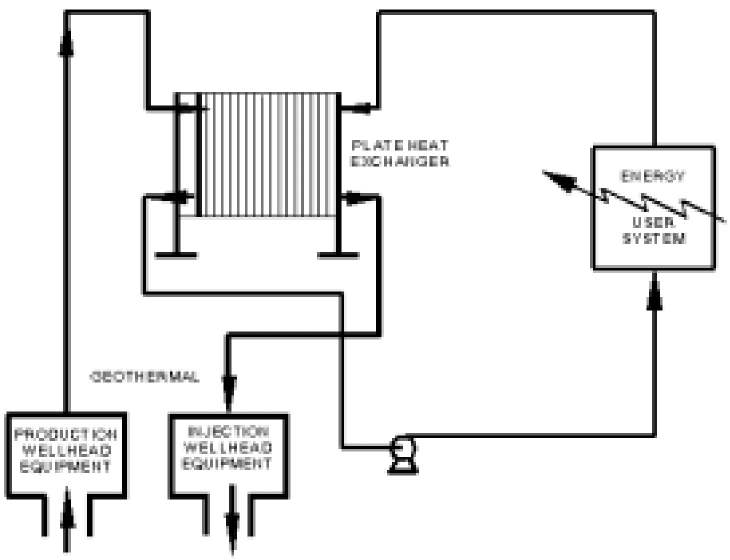


Figure 4.5: Geothermal direct utilization system using a heat exchanger.

The primary components of most low-temperature direct-use systems are downhole and circulation pumps, transmission and distribution pipelines, peaking or back-up plants, and various forms of heat extraction equipment (Figure 4.5). Fluid disposal is either surface or subsurface (injection). A peaking system may be necessary to meet maximum load. Thus can be done by increasing the water temperature or by providing tank storage. Both options mean that fewer wells need to be drilled. When the geothermal fluid temperature is warm (below 50°C), heat pumps are often used. The equipment used in direct-use projects represent several units of operations. The major units will now be described in the same order as seen by geothermal fluids produced for district heating.

4.2.1 Downhole Pumps

Unless the well is artesian, downhole pumps are needed, especially in large-scale direct utilization systems. Downhole pumps may be installed not only to lift fluid to the surface, but also to prevent the release of gas and the resultant scale formation [28].

There are primarily two types of production well pumps; (a) lineshaft turbine pumps and (b) submersible pumps - the difference being the location of the driver. The lineshaft pump system (Figure 4.6a) consists of a multi-stage downhole centrifugal pump, a surface mounted motor and a long drive shaft assembly extending from the motor to the pump. Most are enclosed, with the shaft rotating within a lubrication column that is centreed in the production tubing. This assembly allows the bearings to be lubricated by oil, as hot water may not provide adequate lubrication. A variable-speed drive set just below the motor on the surface can be used to regulate flow instead of just turning the pump on and off.

The electrical submersible pump system (Figure 4.6b) consists of a multi-stage downhole centrifugal pump, a downhole motor, a seal section (also called a protector) between the pump and motor, and electric cable extending from the motor to the surface electricity supply.

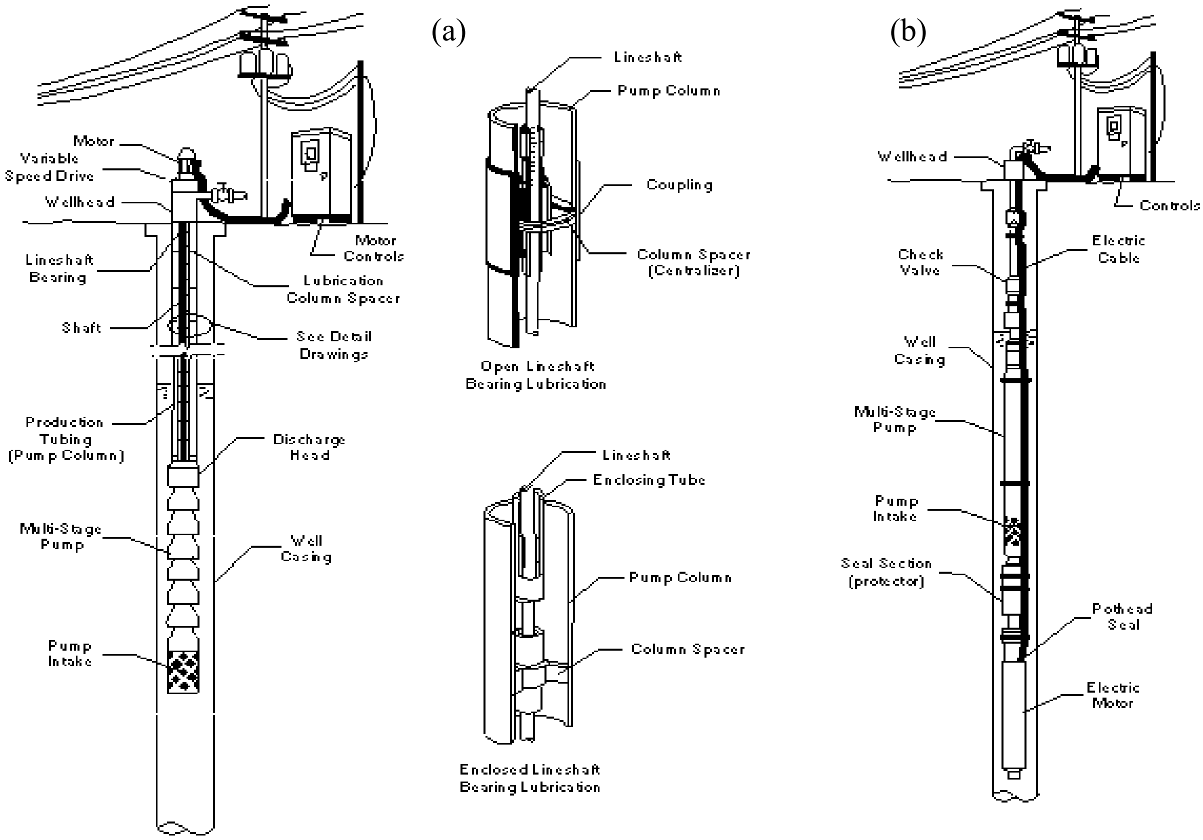


Figure 4.6: Downhole pumps: (a) lineshaft pump details, and (b) submersible pump details.

Both types of downhole pumps have been used for many years for cold water pumping and more recently in geothermal wells [28]. A general comparison of lineshaft and submersible pumps appears in Table 4.2.

Lineshaft pumps have two definite limitations: (a) they must be installed in relatively straight wells and (b) they are economically limited to settings of less than or equal to 600 m. For direct heat applications, the economic setting depth limit is probably closer to 250 m [29].

In some installations, selection of a pump type will be dictated by setting depth, well size, well deviation, or temperature. If not restricted by these, the engineer or developer should select a pump based on lowest life cycle costs, including important factors such as expected life, repair costs, availability of parts, and downtime costs. Power consumption costs and wire-to-water efficiency, although certainly worth evaluating, may not be nearly as important as others factors, such as those above. For most direct heat applications, the lineshaft pump has been the preferred selection.

Table 4.2: Comparison of lineshaft and submersible pumps [29].

Lineshaft	Submersible
Pump stage efficiencies of 68 to 78%. Lower head/stage and flow/unit diameter. Higher motor efficiency. Little loss in power cable. Mechanical losses in shaft bearings.	Pump stage efficiencies of 68 to 78%. Generally, higher flow/unit diameter. Lower motor efficiency--operates in oil at elevated temperature. Higher losses in power cable. Cable at least partially submerged and attached to hot tubing.
Motor, thrust bearing and seal accessible at surface.	Motor, thrust bearings, seal, and power cable in well--less accessible.
Usually lower speed (1,750 rpm or less). Usually lower wear rate.	Usually higher speeds (3,600 rpm). Usually higher wear rate.
Higher temperature capability, up to 200 °C.	Lower temperature capability but sufficient for most direct heat and some binary power applications, assuming the use of special high-temperature motors.
Shallower settings, 600 m maximum.	Deeper settings. Up to 3,600 m in oil wells.
Longer installation and pump pull time.	Less installation and pump pull time.
Well must be relatively straight or oversized to accommodate stiff pump and column.	Can be installed in crooked wells up to 4 degrees deviation per 30 m. Up to 75 degrees off vertical. If it can be cased, it can be pumped.
Impeller position must be adjusted at initial startup.	Impeller position set.
Generally lower purchase price at direct use temperatures and depths	Generally higher purchase price at direct use temperatures and depths.

4.2.2 Heat Exchangers

There are two types of heat exchangers that have proven most satisfactory in the geothermal service: 1 shell and tube-straight through with geothermal in the tube side; and 2) plate type. Tube and shell exchangers have some disadvantages such as lack of flexibility to accommodate changes in temperature and flow conditions to meet load changes, greater space floor required, less efficient.

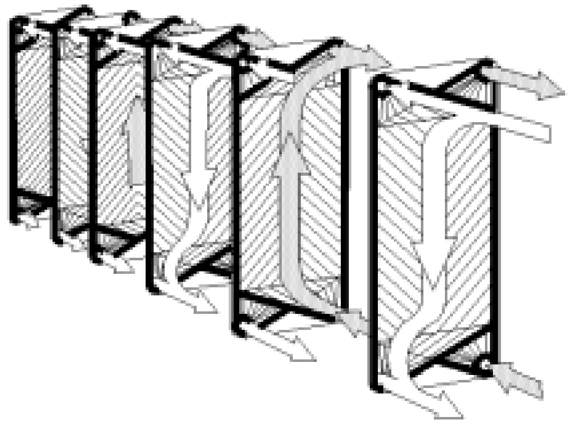


Figure 4.7: Nature of fluid flow through the plate heat exchanger.

The plate-type heat exchanger is generally considered superior in applications for geothermal liquid-to-liquid heat transfer where close approach temperatures are desirable and plate materials other than mild steel are required for corrosion resistance. They require little floor space, are easily cleaned, and are much more efficient [30].

Normally the geothermal temperature and the flow will remain fixed while the return temperatures and flows from the heating network fluctuate as the heat demands of the user changes.

The two basic rules are that the schemes must be designed and operated so that

- the secondary flow through the primary heat exchanger is always greater than the geothermal flow.
- the network return temperature must be as low possible at all times;

Thus the network must be operated with variable temperatures and flows [31].

4.2.3 Piping

The source of geothermal fluid for a direct use application is often located some distance away from the user. This requires a transmission pipeline to transport the geothermal fluid. Even in the absence of transmission line requirements, it is frequently advisable to employ other than standard piping materials for in-building or aboveground piping. Geothermal fluid for direct use applications is usually transported in the liquid phase and has some of the same design considerations as water distribution systems. Several factors including pipe material, dissolved chemical components, size, installation method, head loss and pumping requirements, temperature, insulation, pipe expansion and service taps should be considered before final specification [32].

Thermal expansion of pipelines heated rapidly from ambient to geothermal fluid temperatures (which could vary from 50 to 200°C) causes stress that must be accommodated by careful engineering design [28].

Piping materials for geothermal heating systems have been of numerous types with great variation in cost and durability. Some of the materials which can be used in geothermal applications include: asbestos cement (AC), ductile iron (DI), slip-joint steel (STL-S), welded steel (STL-W), gasketed polyvinyl chloride (PVC-G), solvent welded PVC (PVC-S), chlorinated polyvinyl chloride (CPVC), polyethylene (PE), cross-linked polyethylene (PEX), mechanical joint fiberglass reinforced plastic (FRP-M), FRP epoxy adhesive joint-military (FRP-EM), FRP epoxy adhesive joint (FRP-E), FRP gasketed joint (FRP-S), and threaded joint FRP (FRP-T). The temperature and chemical quality of the geothermal fluids, in addition to cost, usually determines the type of pipeline material used [32].

Carbon steel is now the most widely used material for geothermal transmission lines and distribution networks, especially if the fluid temperature is over 100°C. Other common types of piping material are fiberglass reinforced plastic (FRP) and asbestos cement (AC). The latter material, used widely in the past, cannot be used in many systems today due to environmental concerns; thus, it is no longer available in many locations. Polyvinyl chloride (PVC) piping is often used for the distribution network, and for uninsulated waste disposal lines where temperatures are well below 100°C.

In Figure 4.8 above and below ground pipeline examples are shown.

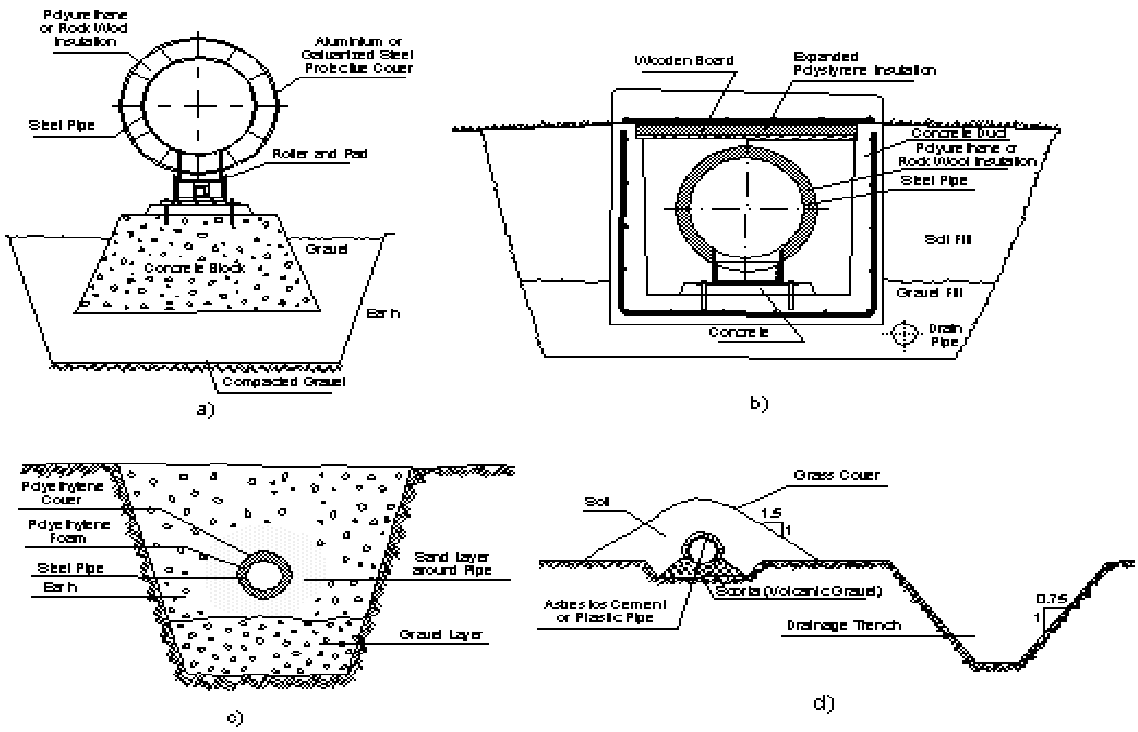


Figure 4.8: Examples of above and below ground pipelines: a) aboveground pipeline with sheet metal cover, b) steel pipe in concrete tunnel, c) steel pipe with polyurethane insulation and polyethylene cover, and d) asbestos cement pipe with earth and grass cover [28].

Conventional steel piping requires expansion provisions, either bellows arrangements or by loops. A typical piping installation would have fixed points and expansion points about every 100 m. In addition, the piping would have to be placed on rollers or slip plates between points. When hot water pipelines are buried, they can be subjected to external corrosion from groundwater and electrolysis. They must be protected by coatings and wrappings. Concrete tunnels or trenches have been used to protect steel pipes in many geothermal district heating systems. Although expensive, tunnels and trenches have the advantage of easing future expansion, providing access for maintenance, and a corridor for other utilities such as domestic water, waste water, electrical cables, phone lines, etc.

4.2.3.1 Selection of the Pipe Diameters

All pipes in the network are numbered from end of the pipeline to heat centre. Then for each pipe flowrate ($\dot{\mathbf{m}}$) is calculated with the help of the equations below.

The total energy extracted from the water is given by

$$\dot{\mathbf{Q}} = \dot{\mathbf{m}} \cdot \mathbf{C_p} \cdot (\mathbf{T_s} - \mathbf{T_r}) \tag{4.2}$$

Flowrate $\dot{\mathbf{m}}$, can be calculated from Equation (4.2);

$$\dot{\mathbf{m}} = \frac{\dot{\mathbf{Q}}}{\mathbf{C_p}.(\mathbf{T_s} - \mathbf{T_r})} \tag{4.3}$$

Flowrate $\dot{\mathbf{m}}$, also equals to;

$$\dot{\mathbf{m}} = \rho \cdot \mathbf{A} \cdot \mathbf{w} = \rho \cdot \pi \cdot \frac{\mathbf{D}^2}{4} \cdot \mathbf{w} \tag{4.4}$$

From Equation (4.4), an equation for pipe diameter can be found as;

$$\mathbf{D} = \sqrt{\frac{4 \cdot \dot{\mathbf{m}}}{\rho \cdot \pi \cdot \mathbf{w}}} \tag{4.5}$$

Pressure drop in the pipe can be explained as;

$$\Delta P_{pipe} = f \cdot \frac{L}{D} \cdot \rho \cdot \frac{w^2}{2} \tag{4.6}$$

For geothermal system convenient fluid velocity is limited between 1-3 m/s for liquid and the pressure loss per unit length is also limited. The pressure loss per unitary length is a common design parameter. If the pressure loss is high, then the investment in the pipe is well utilized, but the operating cost is high. On the other hand, if the pressure loss is low, the investment is badly utilized, but the pumping cost is low. The heat loss in a district-heating pipe is higher for badly utilized pipes. The pressure loss per unit length is thus a good indicator of optimality, but not a real cost function. The district heating practice is to design for 50-200 Pa/m pressure loss [33].

Steps below explain procedure of selection of the pipe diameters

1. Necessary flowrate for each pipe is calculated from Equation (4.3)
2. Pipe diameter is calculated from Equation (4.5). Here the velocity is assumed as 2.5 m/s.
3. Pipe diameter is selected at least one size bigger than calculated pipe diameter.
4. New velocity is calculated with the selected pipe diameter from Equation (4.4).
5. Pressure drop of the pipe is calculated with new velocity and selected pipe diameter values from Equation (4.6).
6. But generally pipe diameter is selected much more bigger than calculated pipe diameter to reduce pressure drop of the pipeline. Thus cost of pumps is reduced significantly. Thus, if pipe diameter is selected again, steps 4 and 5 are repeated.

Chapter 5

MODELLING OF DISTRICT HEATING SYSTEM FOR IZTECH CAMPUS

Models of district heating systems can be classified as follows:

by type:	microscopic or macroscopic
by method:	dynamic or steady state
by approach:	physical or black box
by usage:	design or operation

The concepts "microscopic" and "macroscopic" refer to if the state of the district heating system is to be studied in detail both in time and space, or if the district heating system is lumped into a few model blocks, ignoring spatial variance of the system state.

Dynamic models depend on previous state history, whereas steady state models are time-independent and assume steady state conditions.

Physical models are based on *a priori* knowledge of the nature of the district heating system, whereas black box models are based on relations determined from measured data.

The design usage of a model refers to when the model is used primarily to study the design of a system, mainly by predicting system performance under various extreme conditions.

Operational usage of a model aims at fine-tuning the operation of an existing system in order to improve its economy or performance.

According to Valdimarsson [20], basically, a building model is composed of four components, a building energy storage component, a heat loss component, a heating equipment component and a flow controller component. The building energy storage element describes the building thermal storage effect, that is the reaction of the indoor temperature to the net heat flow into the building. The heat loss element describes the heat lost to the surroundings as a function of the weather and of the indoor temperature. The heating equipment element describes how heat is transferred from the district heating water to the building as a function of water mass flow, building water supply temperature and indoor temperature. As an additional output signal the water return temperature is calculated. The flow controller element describes how the indoor temperature controls the district heating water flow.

The flow controller is a combination of the behaviour of the people living in the building, and of the heating equipment control system, and cannot be determined theoretically. The heating equipment control systems have known characteristics, but they are of different types from one building to another. The residents have almost an unpredictable behaviour. Their tolerance to variations in the indoor temperature is very individual.

A block diagram for a physical district heating model is shown in Figure 5.1.

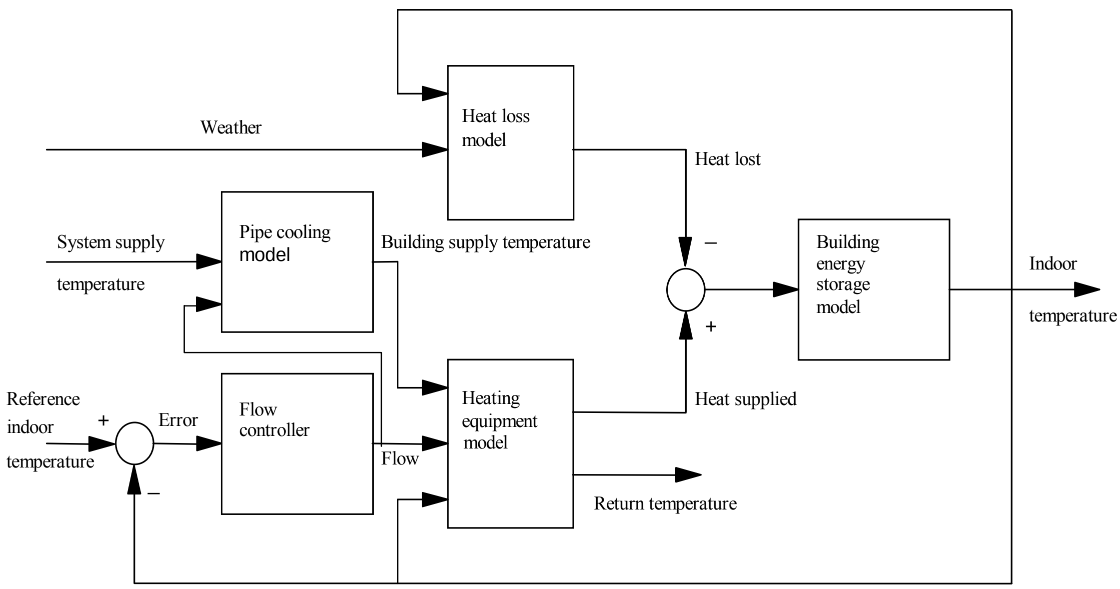


Figure 5.1: Block diagram of a lumped district heating model [20].

In the Thesis, district heating system is modelled according to macroscopic, dynamic model depending on black box approach. For modelling, heating equipment, heat loss and building energy storage models are used. Temperature drop in the pipes is omitted. Thus, pipe-cooling model does not considered. Heating system is simulated according to constant flowrate. Therefore flow controller model is not used in the Thesis.

Heating system model is added to these models because of two heating system types, heat pump district heating and conventional boiler system are considered in the

Thesis.

Each model is explained and procedure of the heating system simulation is given below.

5.1 Building Heat Loss Model

The heat loss is a function of the outdoor weather conditions and the indoor temperature. The heat is lost by heat transfer through the building surfaces, and by exchange of air between the heated space and the building surroundings. 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:

$$\dot{Q}_{loss} = U_{building} \cdot A_{building} \cdot (T_i - T_o) \tag{5.1}$$

5.2 Heating Equipment Model

The radiator or fan coil is considered to be a heat exchanger between the heating water and the room air. The schematic of heating equipment model is shown in Figure 5.2.

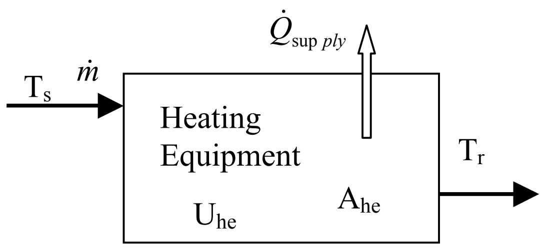


Figure 5.2: Schematic of heating equipment model

The input signals to the heating equipment model are:

- Indoor temperature
- Water flow
- Building supply temperature.

The output signals are:

- Heat supplied
- Return temperature.

The district heating water return temperature from a building is determined by the performance of the heating equipment.

Performance of Heating Equipments

The heating equipment transfers heat from the district heating water to the indoor air. The heat transferred from the water is written as:

$$\dot{Q}_{\sup ply} = \dot{m} \cdot C_p \cdot (T_s - T_r) \tag{5.2}$$

Addition to Equation (5.2), in this case the rate of heat transferred from the waterside to the ambient air can be expressed in the following, equation:

$$\dot{Q}_{\sup ply} = U_{he} \cdot A_{he} \cdot LMTD_{he} = \dot{Q}_{he} \tag{5.3}$$

with

$$LMTD_{he} = \frac{(T_s - T_i) - T_r - T_i)}{\ln((T_s - T_i)/(T_r - T_i))} \tag{5.4}$$

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

$$Performance = \frac{\dot{Q}_{he}}{\dot{Q}_{he0}} = \left(\frac{LMTD_{he}}{LMTD_0} \right)^{(n)} \tag{5.5}$$

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 given in the manufacturer's catalogs for radiators. In this Thesis, it is taken as 1.35 for radiator. And for fan coil, the heating capacities for different heating regime are given in the catalogs. One example is given in Table 5.1 and the performance of the fan coil is calculated according to these catalog capacities depending on reference capacity at 90/70 °C.

Table 5.1: Heating capacities and performance of unit size 2 fan-coil with 2 pipe [34].

Heating Regime (°C)	Heating Capacity (kW)	Fan Coil Performance*100 (%)
90-85	8.4	121
90-80	7.92	114
90-75	7.64	110
90-70	6.93	100
80-75	7.1	102
80-70	6.7	97
80-65	6.15	89
80-60	5.84	84
70-65	5.85	84
70-60	5.5	79
70-55	4.88	70
70-50	4.53	65
60-55	4.59	66
60-50	4.09	59
60-45	3.73	54
60-40	3.3	48
50-45	3.35	48
50-40	2.92	42
50-35	2.47	36
50-30	2.07	30
40-35	2.05	30
40-30	1.65	24
40-25	1.23	18

On the other hand, using the manufacturer catalogue heating capacities of fan coils, n is determined for fan-coils with Equation (5.5).

From Equation (5.5), n is found as nearly 1. The performance, which is calculated from Equation (5.5), is called here theoretical performance. And actual performance is calculated from fan coil heating capacities at the catalogues. In Figure 5.3, the theoretical and actual performances are shown. As it can be seen from the Figure 5.3 the theoretical and actual performance values are very close to each other if n is taken 1.

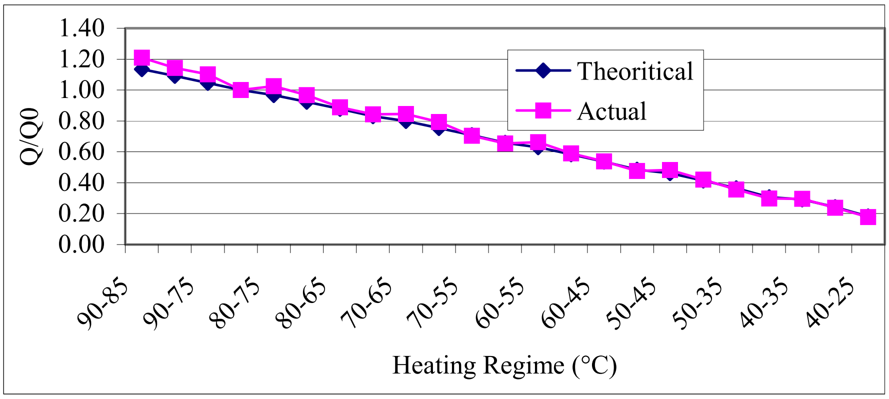


Figure 5.3: Performance of unit size 2 fan-coil with 2 pipe

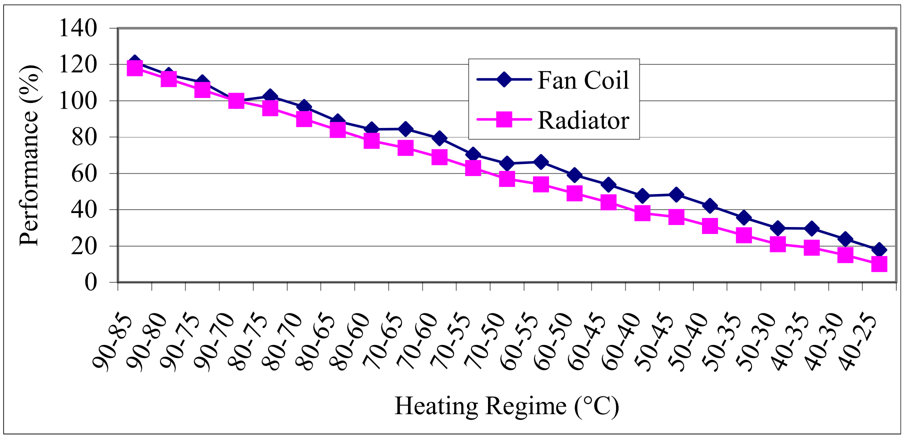


Figure 5.4: Comparison of performance of radiator and fan-coil for different temperature regime.

In the Figure 5.4, radiator and fan coil performances are shown for the same heating regime. As it can be seen from this figure fan coil has nearly 5-10% better performance than radiator.

Depending on performance of the heating equipment, it can be necessary to add extra heating equipment for various supply or return temperature design for heating system.

5.2.1 Calculate Return Water Temperature

Yet there are two cases that must be dealt with in the heating equipment model model. The first is when there is water flow through the heating equipments and pipes and the second is when the water flow is zero, but the water inside the heating equipments and pipes is hot. The water mass inside the pipes in the building is assumed as 50% of the water mass inside the heating equipments. Mathematical models for these two cases are presented below.

5.2.1.1 Water Flowrate is not Zero

Substituting Equations (5.2), (5.3), (5.4) and solving for T_r gives

$$T_r = \frac{T_s - T_i}{\exp((A_{he} \cdot U_{he})/(\dot{m} \cdot C_p))} + T_i \tag{5.6}$$

Since the overall heat transfer coefficient U_{he} is a function T_r, iteration is necessary to determine T_r.

Due to the difficulty of calculating the overall heat transfer coefficient at different return temperatures, the heat output from the radiators has traditionally, been related to the arithmetic mean temperature difference as given in (5.5). Hence by substituting Equations (5.2) and (5.4) into (5.5), one obtains

$$\frac{\dot{m}\cdot(T_s - T_r)}{\dot{m}_0\cdot(T_{so} - T_{ro})} = \left(\frac{\dfrac{T_s - T_r}{\ln((T_s - T_i)/(T_r - T_i))}}{\dfrac{T_{so} - T_{ro}}{\ln((T_{so} - T_{io})/(T_{ro} - T_{io}))}} \right)^n \tag{5.7}$$

Where the (0) means the states at design conditions which are constants. From Equation 5.7 T_r can be found in three terms, hence one of these T_r's should be chosen to be on the left hand side of the equation in order to perform the iteration. The T_r inside the logarithm of the denominator of the right hand side of Equation 5.7 is chosen yielding.

$$T_r = T_i + \frac{T_s - T_i}{\exp\left[\dfrac{\left(\dfrac{T_s - T_r}{T_{so} - T_{ro}} \right)^{(1-1/n)} \cdot \ln\left(\dfrac{T_{so} - T_{io}}{T_{ro} - T_{io}} \right)}{\left(\dfrac{\dot{m}}{\dot{m}_0} \right)^{-1/n}} \right]} \tag{5.8}$$

Hence Equation (5.8) can be used to find the radiator or fan coil water return temperature at any condition.

5.2.1.2 Water Flowrate is Zero

In this case, a lumped cooling model is used, where the water inside the radiator or fan coil is assumed to cool down in an exponential way with its initial conditions taken from the model when the flow is not zero in section 5.2.1.1.

$$\frac{T_s(t) - T_i}{T_{so} - T_i} = \exp\left[-\frac{U_{he}\cdot A_{he}}{M_{he}\cdot C_p}\cdot \Delta t \right] \tag{5.9}$$

As it is difficult to mathematically determine the overall heat transfer coefficient U_{he} in Equation (5.9) due to its high dependence on the fluid temperatures, a term can be found to express U_{he}. This term can be obtained from Equation (5.3), where Equation (5.9) becomes:

$$\frac{T_s(t)-T_i}{T_{so}-T_i} = \exp\left[-\frac{\frac{\dot{Q}_{he}\cdot A_{he}}{A_{he}\cdot LMTD_{he}}}{M_{he}\cdot C_p}\cdot\Delta t\right] = \exp\left[-\frac{\dot{Q}_{he}}{LMTD_{he}\cdot M_{he}\cdot C_p}\cdot\Delta t\right] \tag{5.10}$$

The heat output $\dot{Q}_{he}$ can be found from (5.5) and the logarithmic mean temperature difference ($LMTD_{he}$) can be found in Equation (5.4). Accordingly the final form of T_s, can be expressed by

$$T_s(t) = T_i + (T_{so}-T_i)\cdot = \exp\left[-\frac{\dot{Q}_{he}}{LMTD_{he}\cdot M_{he}\cdot C_p}\cdot\Delta t\right] \tag{5.11}$$

where the (0) indicates the conditions at the moment the flow became zero. The same analysis applies to the heating equipment return water temperature, which is written as

$$T_r(t) = T_i + (T_{ro}-T_i)\cdot = \exp\left[-\frac{\dot{Q}_{he}}{LMTD_{he}\cdot M_{he}\cdot C_p}\cdot\Delta t\right] \tag{5.12}$$

The only complication that would results from the assumptions given above is in the case when the heating equipment supply temperature (T_s) or return temperature (T_r) are either equal to each other, or any of them is equal to the indoor temperature (T_i). This results in a value of zero for ($LMTD_{he}$) in the first case (T_s=T_r) and a value of infinity is the other two cases (T_s=T_i) or (T_r=T_i). In order to avoid such cases the value of both (T_s) and (T_r) are increased by a small number, where such an increase does not affect the results. This procedure is shown in the calculation algorithm below.

From the above discussion, it is clear that an algorithm for calculating the heat output from the heating equipment to the building is needed in the case of no flow of water into the heating equipment. This algorithm is shown below [35]:

1. Calculate new T_s from Equation (5.11).

2. Calculate new T_r from Equation (5.12).
3. If $T_r \leq T_i \Rightarrow T_r = T_i + 0.1$.
4. If $T_s \leq T_i \Rightarrow T_s = T_i + 0.1$.
5. If $T_s \leq T_r \Rightarrow T_s = T_r + 0.01$.
6. Calculate new $LMTD_{he}$ from Equation (5.4).
7. Calculate new $\dot{Q}_{he}$ from Equation (5.5)

5.3 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. Calculating building heat capacity is very complex because it depends each construction elements of the buildings. In this Thesis, the air in the building is considered as a system and the 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 energy storage becomes as described in Equation (5.13).

$$C_{air} \cdot \frac{dTi}{dt} = \dot{Q}_{net} = \dot{Q}_{\sup ply} - \dot{Q}_{loss} \tag{5.13}$$

where:

$$C_{air} = m_{air} \cdot C_v = \rho_{air} \cdot V_{building} . C_v \tag{5.14}$$

As for the heat gain, only the gain from the heating system (radiators or fan coils) will be considered, while heat gain from the people and lights will not be considered in the mathematical model. Schematic of building energy storage model is shown in Figure 5.5.

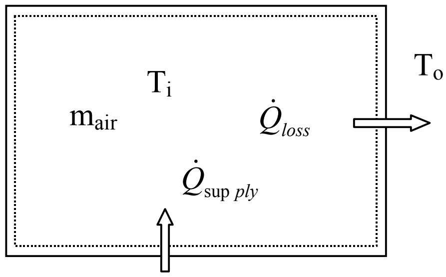


Figure 5.5: Schematic of building energy storage model.

5.4 Heating System Model

In this Thesis, mainly two heating system types are considered; heat pump district heating system and conventional boiler heating system.

Because of the low geothermal fluid temperature at present, 33°C, heat pump district heating system is considered for the reason that heat pump is very attractive for heat sources with a temperature in the range 20-40°C. And heat pump only layout is selected as heat pump type, which are explained in Chapter 4. Because generally heat pump only layouts have better performance than heat pump assisted if geothermal fluid is below 40°C [15].

A fuel boiler district heating system is also simulated to replace the existing heating system and for comparison with new heat pump systems.

5.4.1 Heat Pump Model

Because of the corrosion effects of geothermal fluid, a heat exchanger is considered. Geothermal fluid passes from heat exchanger rather than evaporator.

The considered heat pump heating system is shown by Figure 5.6.

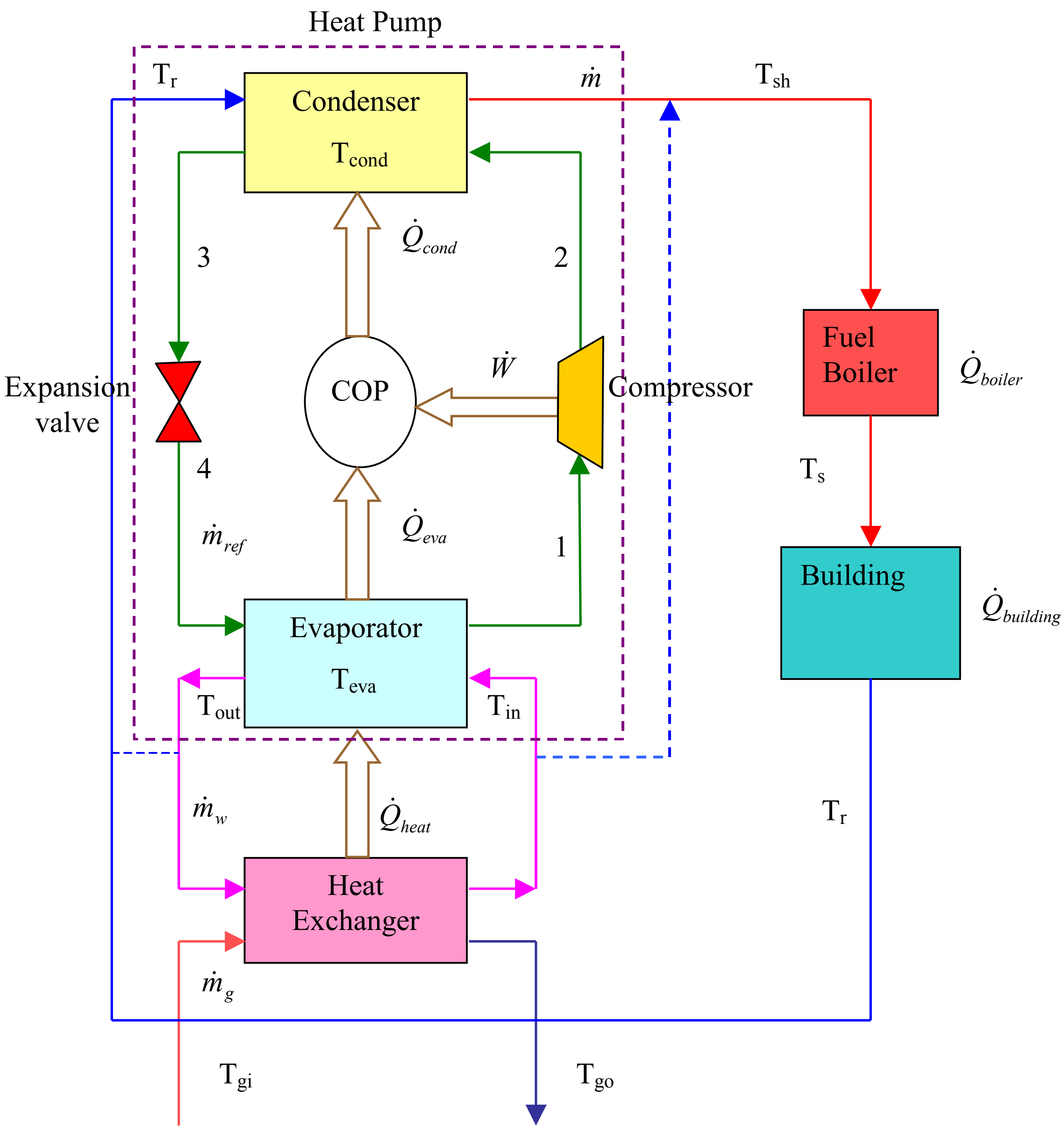


Figure 5.6: Considered district heating system with heat pump.

- **Simplify Heat Pump Capacity Calculation**

The Carnot efficiency of the heat pump can be defined as the ratio of the heat released to work input.

$$COP_{car} = \frac{T_{cond}}{T_{cond} - T_{eva}} \quad (5.15)$$

$$COP = \frac{\dot{Q}_{cond}}{\dot{W}} = \frac{\dot{Q}_{cond}}{\dot{Q}_{cond} - \dot{Q}_{eva}} \quad (5.16)$$

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 [15]

$$COP = 0.5 \cdot COP_{car} \quad (5.17)$$

According to Figure 5.6 the heat pump heat flows can be written as:

$$\dot{Q}_{con} = \dot{m} \cdot C_p (T_{sh} - T_r) \quad (5.18)$$

$$\dot{Q}_{eva} = \dot{m}_w \cdot C_p \cdot (T_{in} - T_{out}) \quad (5.19)$$

- **Actual Cycle**

Vapour compression heat pumps used in geothermal schemes work by evaporation and condensation. Heat is absorbed by the working fluid converting it from a liquid to a vapour at a low temperature. This cycle consists of the following stages.

- **Evaporation;** heat is transferred to the working fluid by conduction from the cold reservoir and the liquid, which is at a low pressure, evaporates.
- **Compression;** the vapour is compressed adiabatically, its temperature rises, and it passes to the condenser as a high pressure, high temperature, saturated or superheated vapour.
- **Condenser**; in the condenser the liquid condenses at this higher temperature with the latent heat being conducted away to the high temperature reservoir.
- **Throttling;** the liquid is returned to the low pressure part of the cycle passing through an expansion valve. Here the pressure is reduced and there is partial

evaporation accompanied by cooling.

Finally the cooled liquid passes back to the evaporator.

COPs are inversely proportional to the temperature difference of the working fluid when releasing and absorbing heat. It is these temperatures of the working fluid, which determine the performance of the heat pump, and this has important implications for the modelling of the heat pump operation. The theoretical levels of heat pump performance indicated by the Carnot cycle are never achieved in practical heat pumps for a variety of reasons [15].

Equations used in the real calculations are given below

$$\dot{Q}_{cond} = \dot{m}_{ref} \cdot (h_2 - h_3) \tag{5.20}$$

$$\dot{Q}_{eva} = \dot{m}_{ref} \cdot (h_1 - h_4) \tag{5.21}$$

$$\dot{W} = \dot{m}_{ref} \cdot (h_2 - h_1) \tag{5.22}$$

T-s diagram of vapour compression heat pumps is shown in Figure 5.7.

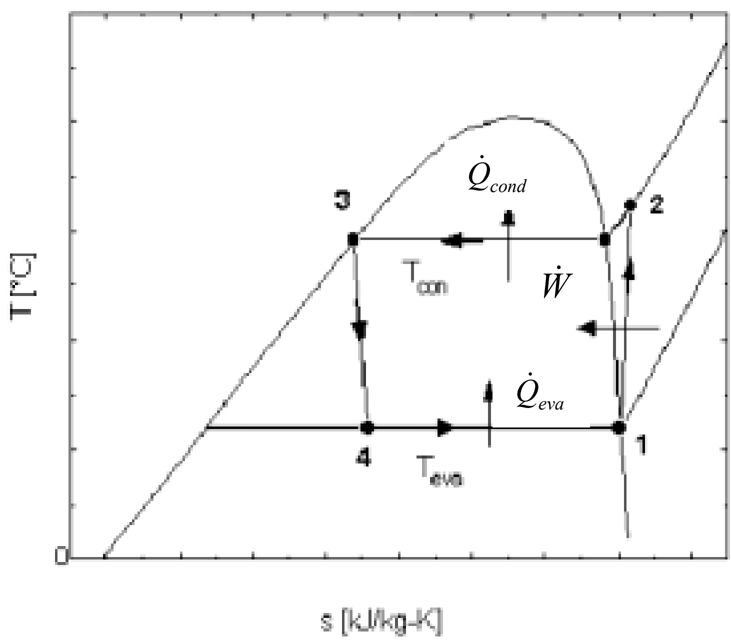


Figure 5.7: T-s diagram of vapour compression heat pumps.

5.4.2 Boiler Model

The boiler is considered as a source with constant heat added to the water flowing into it. The outlet temperature from the boiler T_b is calculated according to the following relation, which is based on the principle of energy conservation of the boiler, i.e.

$$\frac{dT_b}{dt} = \frac{\dot{Q}_{boiler} - \dot{m} \cdot C_p \cdot (T_b - T_r)}{M_{boiler} \cdot C_p} \tag{5.23}$$

In this work, supply water temperature is assumed equal to boiler outlet temperature.

Integrating Equation (5.23), over a time step of Δt yields

$$T_b(t) = \frac{\dot{Q}_{boiler} - \dot{m} \cdot C_p . T_r - \dot{Q}_r}{\dot{m} \cdot C_p} \tag{5.24}$$

where:

$$\dot{Q}_r = \left[\dot{Q}_{boiler} + \dot{m} \cdot C_p \cdot (T_r - T_{b(old)})\right] \cdot \exp\left(- \frac{\Delta t \cdot \dot{m}}{M_{boiler}} \right) \tag{5.25}$$

$T_{b(old)}$ is the boiler water temperature at the previous time step.

Schematic of boiler model is given in Figure 5.8.

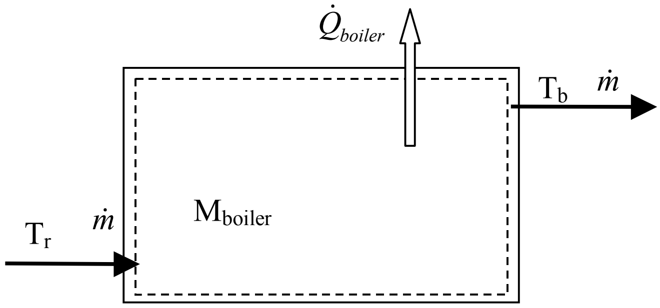


Figure 5.8: Schematic of boiler model.

5.5 Simulation Period

According to weather data, heating season starts in October and ends in April, where the heating system is turned off during the rest of the year. Hence, simulation program will account only for this time period (October to April) and will ignore the rest of the year, making the simulation time 3874 hours instead of 8760 hours.

5.6 Simulation Type

The Campus mostly is used during the working hours, which is between 9.00 a.m. and 17.00 p.m. and the existing building heating systems of the Campus have been run during this period. Thus, considered district heating systems are simulated according to intermittent heating regime.

For intermittent heating system simulation, the heating system is turned on at certain hours of the day, when people are in the building for example, and it is turn off for the rest of the day. The intermittent heating concept aims at providing the building with the required heat and keeps it at the design temperature for certain times only not for the whole heating period. The heating period is considered as between 9.00 a.m. and 17.00 p.m., which is working time period for the Campus in the week. Hence, in the simulation program, the system is turned on one hour earlier and turned off at the end of this period.

For the simulations a control system with constant flowrate and variable return water temperature is used.

5.7 Simulation Program

The program is written in Matlab, for its relative flexibility and ease of programming.

Since most equations representing the different components of the system are differential equations, the program is written in a forward time step mode, that is the properties of the different parameters are given at the beginning and end of the integration period and not on a continuous mode, where the differential equations are first integrated in terms of the time step (dt) and then used in the program. Hence the present value of a certain parameter is obtained from values of the parameters from the previous step, this is shown in Equation (5.26), where the expression for the indoor temperature (T_i) at present time step is obtained from the parameters of the previous time step [35].

$$T_{i(i)} = \frac{\dot{Q}_{he(i-1)} + U_{building} \cdot A_{building} \cdot T_{o(i-1)}}{U_{building} \cdot A_{building}} - \frac{e^{(-U_{building} \cdot A_{building} \cdot dt / C_{air})}\left[\dot{Q}_{he(i-1)} + U_{building} \cdot A_{building} \cdot (T_{o(i-1)} - T_{i(i-1)})\right]}{U_{building} \cdot A_{building}} \quad (5.26)$$

Where the subscript (i-1) means the value from the previous step. Because of the assumptions made above and in order to reduce errors and get as much information as possible about the system, the time step chosen is short (360 seconds).

5.7.1 Heat Pump Heating System Simulation Program's Algorithm

Firstly, heat pump condenser outlet water temperature (T_{sh}) and supply water temperature (T_s) are decided. If the value of the flow ($\dot{m}$) is not zero, heat pump heating system is run. Then return temperature from the heating equipments (T_r) is calculated according to Equation (5.8) by an iterative technique. After that the heating equipment heat output ($\dot{Q}_{he}$), and the logarithmic mean temperature of the heating equipment $LMTD_{he}$.

According to supply water temperature to calculate heat pump capacity firstly evaporator outlet temperature (T_{out}) must be calculated. Because of this some assumptions are necessary. These assumptions are:

- Geothermal fluid temperature at heat exchanger inlet is 33°C.
- Heat exchanger outlet (evaporator inlet temperature) is 2°C lower than geothermal fluid temperature at heat exchanger inlet.

$$T_{in} = T_{gi} - 2 \quad (5.27)$$

- Evaporator outlet temperature (heat exchanger inlet) is 3°C lower than geothermal fluid temperature at heat exchanger outlet.

$$T_{out} = T_{go} - 3 \tag{5.28}$$

- Heat pump condensing absolute temperature is 3°C higher than district heating water supply temperature at condenser outlet.

$$T_{cond} = T_{sh} + 3 \tag{5.29}$$

- Heat pump evaporating absolute temperature is 2°C lower than evaporator outlet temperature.

$$T_{eva} = T_{out} - 2 \tag{5.30}$$

- Heat exchanger efficiency is 0.95.

Firstly, evaporator outlet temperature is assumed. Then using assumptions (5.27), (5.28), (5.29) and (5.30) and equations (5.15), (5.16), (5.17), (5.18) and (5.19) exact evaporator outlet temperature can be calculated by iteration. Then heat pump capacity and geothermal outlet temperature are calculated.

If the flow value is zero, T_s and T_r are calculated according to section 5.2.1.2.

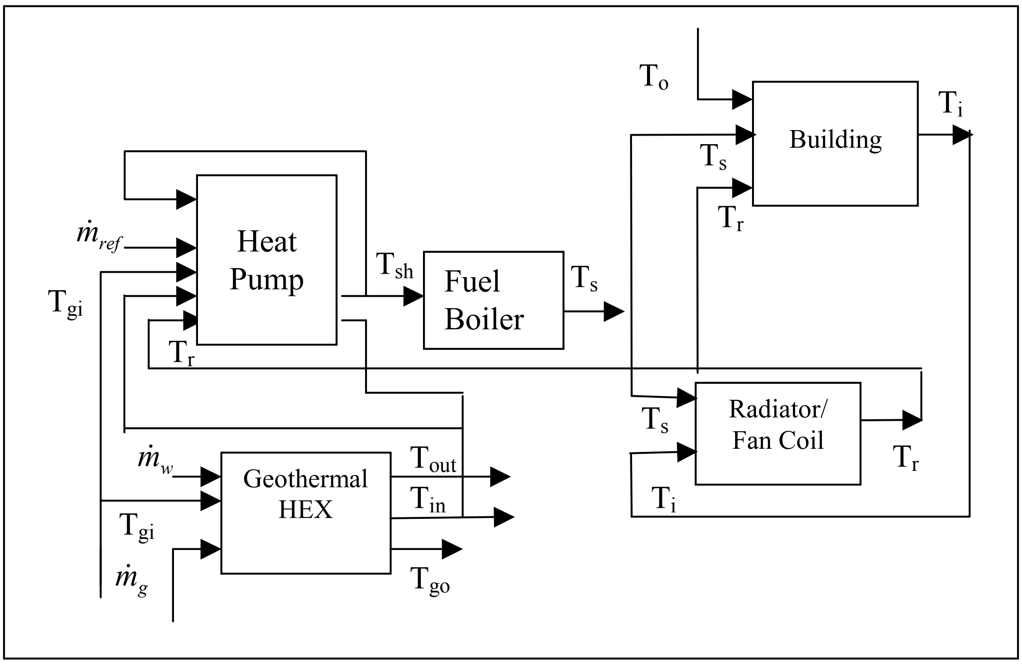


Figure 5.9: Schematic of heat pump heating system simulation.

5.7.2 Fuel Boiler Simulation Program's Algorithm

The first step in simulating a certain system is to understand how this system operates, its control sequence, the different components' parameters and the variables affecting these components. As it can be seen from Figure 5.10, the boiler has thermostat controller, which turns the boiler off ($\dot{Q}_{boiler}$ =zero) whenever the boiler water temperature exceeds a certain value (T_{b_set}).

The figure also shows the different parameters affecting each component of the system and the output values from these components explaining the equations used to represent these components.

After describing the heating system and the type of the simulation used, a brief description of the algorithm used by the program to perform the simulation is given here.

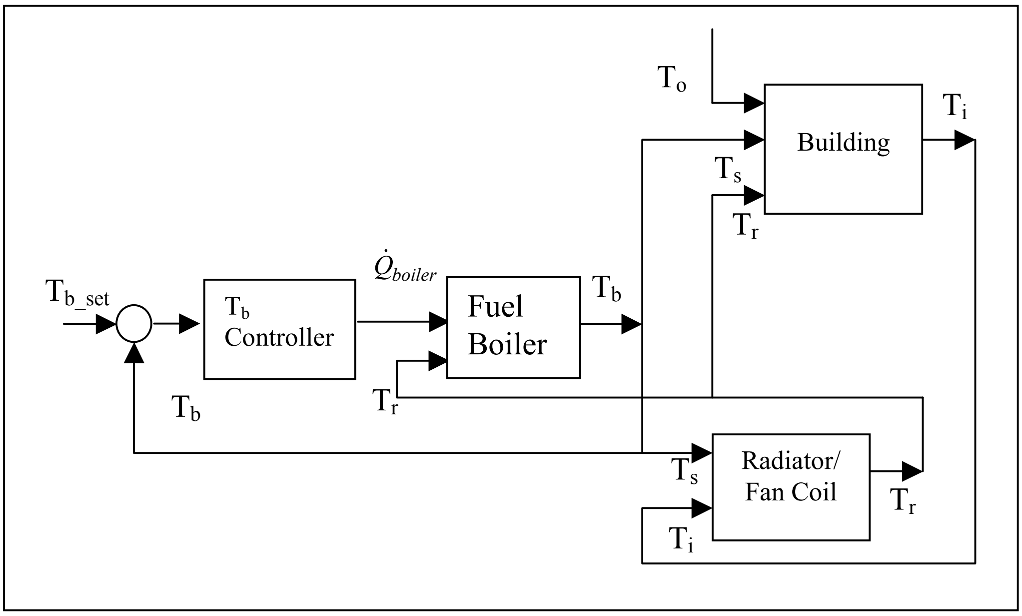


Figure 5.10: Fuel boiler heating system components

The program starts by acquiring the outdoor conditions for the simulation period from another file called (temp.dat). Then the program zeros all values to be evaluated later. After that it interpolates the values of outdoor temperature (T_o) and at the required time intervals (360 seconds). At this point the simulation loop starts by calculating the indoor temperature (T_i) value according to Equation (5.26). Then the time is checked. If the time is not in the working period, the program sets the flow to zero, otherwise it sets the flow to maximum value ($\dot{m}_0$).

If the value of the flow ($\dot{m}$) is not zero, the program first calculates the boiler temperature (T_b), which is also considered to be the supply temperature (T_s) to the heating equipments, then it calculates the return temperature from the heating equipments (T_r) according to Equation (5.8) by an iterative technique for which a subroutine called (T_ret.m) is used. After that the heating equipment heat output ($\dot{Q}_{he}$), and the log mean temperature of the heating equipment $LMTD_{he}$.

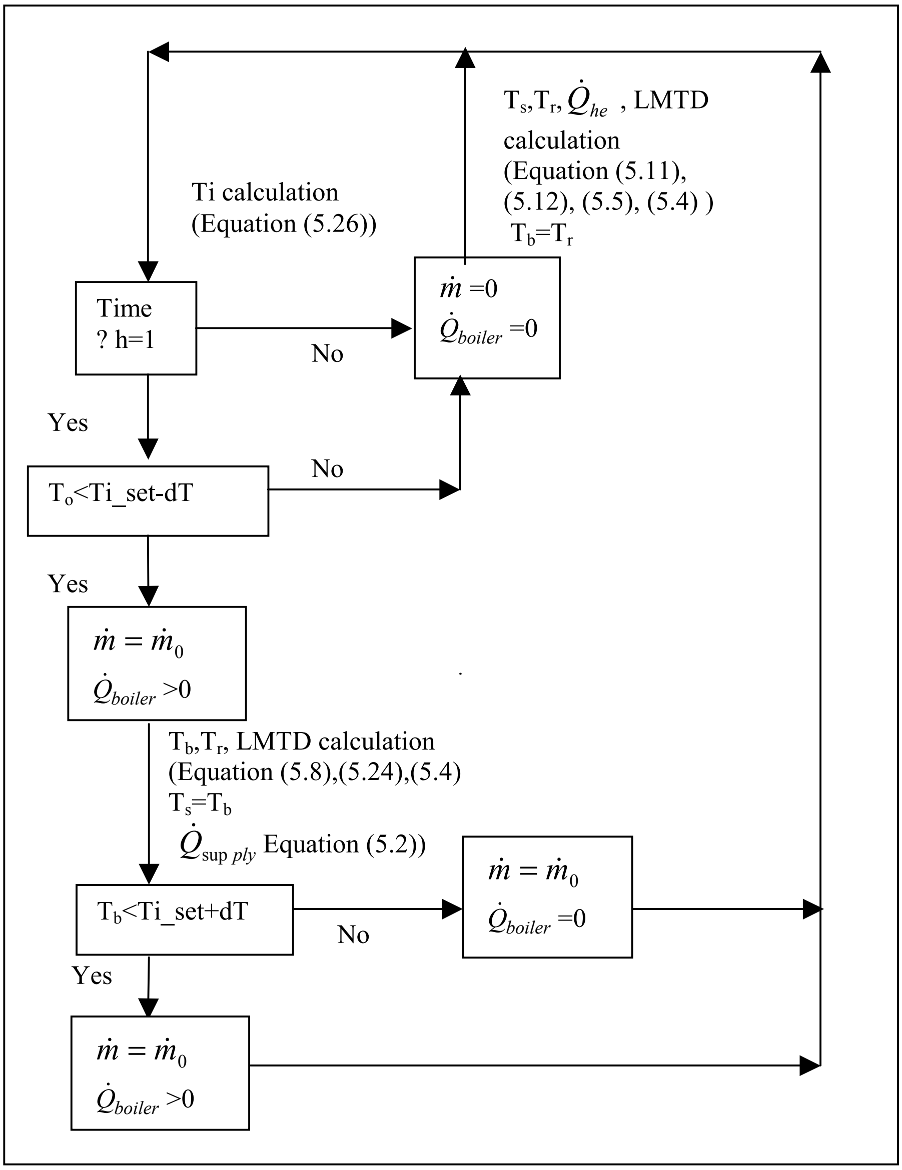


Figure 5.11: Boiler heating system program algorithm.

On the other hand, if the flow value is zero, the program skips the above equations and calculates the heating equipment supply (T_s) and return (T_r) water temperatures using Equations (5.11) and (5.12) respectively. Then the program compares the values of the (T_r), (T_s) and (T_i) two at a time, and if any of two are equal to each other it increases their values so that (T_s) would be highest and (T_r) is the second highest where the amount of increase of (T_r) is (0.1°C) above (T_i) and (T_s) is (0.01°C) above (T_r). This done so as not get complex numbers when calculating the log mean temperature of the heating equipment, and the increase is so small that it does not affect the results.

Then the program calculates the log means temperature of the heating equipment, and sets the boiler water temperature (T_b) equal to the heating equipment return water temperature (T_r), it also gives the boiler power ($\dot{Q}_{boiler}$) a value of zero. After that the loop starts again. The logic is more clearly shown in Figure 5.11.

5.8 Economic Analysis

5.8.1 Approximate Energy Consumption Cost of the Heating System

The cost of the annual energy consumption is calculated approximately using following equations:

For heat pumps:

$$\mathrm{Cost}_{el} = W_{el} . P_{el} = \frac{W_{annual}}{\eta_{elc}} \cdot P_{el} = \frac{Q_y}{COP \cdot \eta_{elc}} \cdot P_{el} \tag{5.31}$$

For fuel boiler:

$$\mathrm{Cost}_{fuel} = m_{fuel} . P_{fuel} = \frac{Q_{annual\_boiler}}{H_u . \eta_{boiler}} \cdot P_{fuel} \tag{5.32}$$

For circulation and well pumps:

$$Cost_{cir} = \frac{\dot{m} \cdot g \cdot h_p}{1000 \cdot \eta_{motor} \cdot \eta_{pump}} \cdot P_{el} \tag{5.33}$$

Total dynamic head of pump (h_p) has been nearly calculated from heating system pressure drop with different flow rates with the help of Pipelab program.

5.8.2 Cost Comparison of Investment Alternatives

For the most part, cost analysis involves selection of the minimum cost or maximum profit alternatives. There are basically four accepted methods of evaluating the alternatives.

1. *Present Worth Method*: all the project cash flows are converted to an equivalent single sum at the appropriate interest rate at time zero.
2. *Annual Worth Method*: all the projects cash flows are converted to an equivalent uniform annual series of cash flows spread over the planning horizon.
3. *Future Worth Method*: all the projects cash flows are converted to an equivalent single sum at the end of the planning horizon or at some other future time.
4. *Internal Rate of Return*: this method determines the interest rate that yields a zero present worth for the project [36].

Differences between investment, operational and amortization costs of the alternatives are used for the IRR calculations. The amortization life is considered as 20 years and amortization coefficient (ac) is calculated as 0.1 by dividing 200% to the amortization year. 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.

Book value (BV) is calculated using the formula below.

$$BV_{n_y} = BV_{(n_y-1)} \cdot (1-ac)^{n_y} \tag{5.34}$$

In Equation (5.34), n is the number of the year and the time zero year BV equals to total investment cost of the system [37].

Annual amortization cost is the difference between two book values, which are consecutive. Cash flow is the difference between annual profit and amortization cost of the systems.

Chapter 6

RESULTS AND DISCUSSION

6.1 Design of a District Heating System for IZTECH Campus

Design of a district heating system, initially requires a thermal load inventory consisting of total heat load and total annual heat requirement of the system. Design flow diagram of design procedure is given in Figure 6.1.

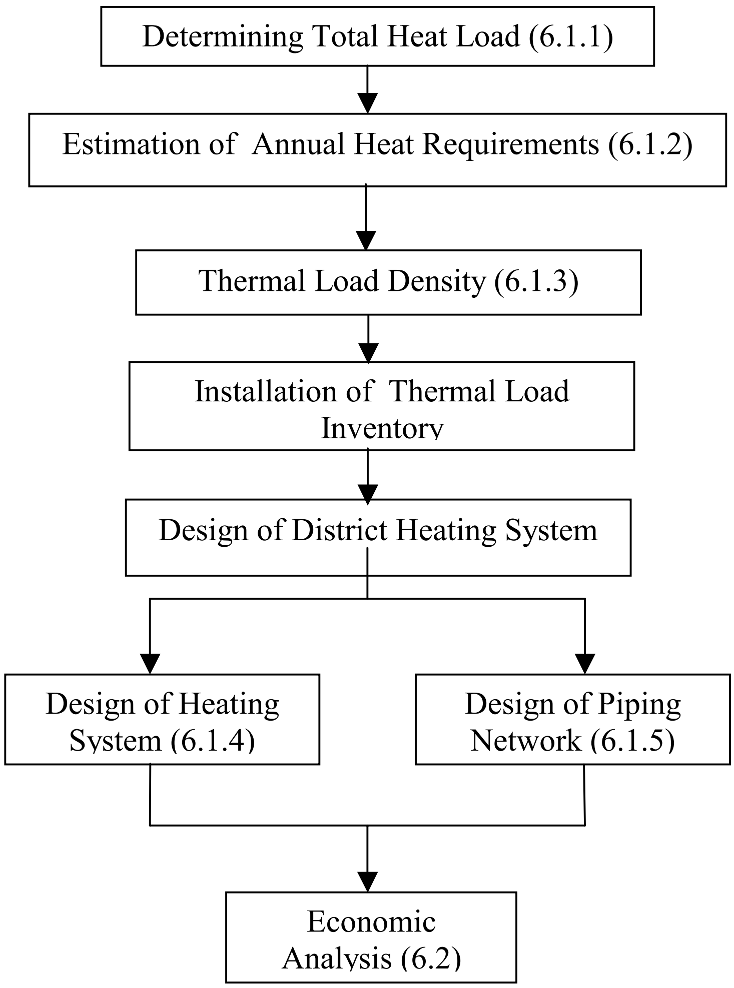


Figure 6.1: Design flow diagram.

Estimating the annual heat requirements; degree-day, degree-hour and heat load factor methods are used. Mainly two heating system types, heat pump and fuel boiler, are considered for designing the heating system.

6.1.1 Total Heat Load of the Campus

Campus heat load is evaluated into two parts; 1) *existing buildings*, which are introduced in Chapter 2, and 2) *new buildings*, which are under construction, or planned whose mechanical projects do not exist at present. As it is given in Chapter 2, total heat load of the existing buildings is 3,662 kW. To be able to determine the heat load of the new buildings, unit heat load (building heat load/ building usage area and volume) is calculated based on the existing building data. Then, total heat load of the Campus can be determined. Calculated unit heat loads of the existing buildings are given in Table 6.1.

Table 6.1: Unit heat loads of the existing buildings.

	Building Name	Heat Load (kW)	Total Building Usage Area (m^2)	Total Building Usage Volume (m^3)	Unit Heat Load (kW/m^2)	Unit Heat Load (kW/m^3)
Engineering Faculty	Main Building	162	2,852	9,412	0.057	0.017
	Classrooms Building	127	1,870	6,171	0.068	0.021
	Laboratories	135	1,948	6,428	0.069	0.021
	Laboratories	160	2,913	9,613	0.055	0.017
	Mec. Eng. Lab.	193	2,141	7,065	0.090	0.027
	Mec. Eng. Lab.	228	1,805	5,957	0.126	0.038
Architecture Faculty	Studio Building	310	4,800	17,280	0.065	0.018
	Main Building	272	4,897	17,629	0.056	0.015
Science Faculty	Main Building	213	3,538	11,675	0.060	0.018
	Laboratories	203	3,276	10,811	0.062	0.019
	Laboratories	186	3,606	11,900	0.052	0.016
Rectorship Buildings	Main Building	353	2,994	9,880	0.118	0.036
	Dep. Pre. Building	442	5,190	17,127	0.085	0.026
	Cafeteria	412	4,700	23,500	0.088	0.018
	Incubator Building	267	4,200	14,700	0.064	0.018
CAMPUS		**3,662**	**50,730**	**164,448**	**0.072**	**0.022**

Average unit heat load per unit area and volume is calculated as 0.072 kW and 0.022 kW respectively for 20°C indoor temperature. The unit heat loads of the new buildings are assumed to be the same with Campus average.

Unit heat load distribution for the existing buildings in the Campus is shown in Figure 6.2. The Figure points out some peak points. Building construction properties; such as material, total window area, location of the buildings, type of the HVAC system equipment (boiler, air conditioning, etc.) are the parameters that effect unit heat load.

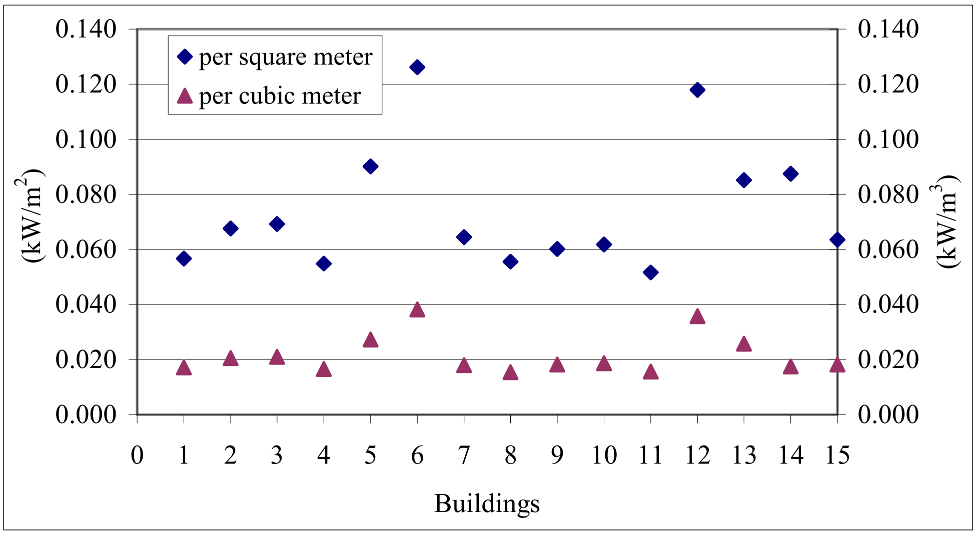


Figure 6.2: Unit heating load of the buildings depending on their usage area and volume.

Estimation of heat loads for the new buildings are given in Table 6.2 and totals as 7,545 kW. The usage area for planned buildings is assumed as 3,500 m^2 each. Consequently, the Campus total peak load reaches to 11,207 kW.

Table 6.2: Estimation of heat load for the new buildings.

	Building Name	Building Usage Area (m^2)	Building Heat Load (kW)
Under Construction	Library	12,500	900
	Chemical Engineering	13,500	1,750
	Mechatronic Building	3,500	252
	Research&Development Centre	2,250	162
	Medical Centre	9,000	648
	Sport Centre	10,830	780
	Staff Houses	3,600	259
	Dormitories	14,300	1,030
Planned	Building A	3,500	252
	Building B	3,500	252
	Building C	3,500	252
	Building D	3,500	252
	Building E	3,500	252
	Building F	3,500	252
	Building G	3,500	252
	TOTAL	**93,980**	**7,545**

6.1.2 Estimation of Annual Heat Requirement of the Campus

Energy estimating methods, which are given in Chapter 3, are used to determine annual heat requirement of the Campus. Calculations are based on the weather data of typical year which is determined as 1993. The weather data constitutes hourly outdoor temperatures, which was measured by Güzelyalı Meteorological Station. Calculations are performed for various balance temperatures such as 16, 18, 18.3, 20, 22°C [26] and 0°C reference outdoor temperature for İzmir.

- **Degree-Day**

The degree-day values of İzmir City are calculated using Equation (3.6) and the results are given in Table 6.3. For 20°C balance temperature, which is the design temperature of heating systems, the degree-day value is 1,738 for İzmir.

Table 6.3: Degree-day values of İzmir for different balance temperature.

MONTH	Balance Temperature (°C)				
	16	18	18.3	**20**	22
January	268.9	330.9	340.2	392.9	454.9
February	263.3	319.3	327.7	375.3	431.3
March	158.8	214.4	223.1	274.3	336.3
April	49.7	96.1	103.8	148.5	203.1
May	3.7	23.8	28.0	53.1	93.2
June	0	0	0	0	0
July	0	0	0	0	0
August	0	0	0	0	0
September	0.0	0.0	0.0	0.1	11.0
October	1.3	6.5	7.7	18.1	50.5
November	126.5	174.1	181.6	225.0	282.8
December	126.6	188.6	197.9	250.6	312.6
TOTAL	998.8	1,353.6	1,410.0	**1,737.8**	2,175.6

Table 6.3 indicates that heating season for İzmir is between November and April and the coldest month is January. The number of heating days, which is the total number of the days that daily average temperature is below balance temperature, is determined as 202 for 20°C indoor temperature. The results for the other balance temperatures are given in Table 6.4.

Table 6.4: Number of heating days for various balance temperatures.

Balance Temperature (°C)	16	18	18.3	**20**	22
Total of Heating Days	163	187	188	**202**	237

- **Degree-Hour**

Since degree-hour uses hourly data unlike degree-day, which is calculated daily outdoor temperature, gives more sensitive results for energy estimation of the system. Therefore, degree-hour method is preferred for determining annual heat requirement of the Campus. Table 6.5 gives degree-hour values, which are calculated using Equation (3.7). For 20°C balance temperature, the degree-hour value is calculated as 41,162.

MONTH	Balance Temperature (°C)				
	16.0	18.0	18.3	**20.0**	22.0
January	6,452.4	7,940.4	8,163.6	9,428.4	10,916.4
February	6,319.5	7,663.5	7,865.1	9,007.5	10,351.5
March	3,756.0	5,016.9	5,239.7	6,469.9	7,963.6
April	1,123.4	2,068.1	2,271.7	3,389.0	4,755.0
May	74.4	442.2	546.4	1,171.4	2,105.5
June	0	0	0	0	0
July	0	0	0	0	0
August	0	0	0	0	0
September	0.0	0.0	0.0	0.7	162.7
October	20.7	115.8	138.6	353.9	1,002.9
November	2,983.3	4,082.4	4,270.8	5,327.5	6,713.8
December	3,008.4	4,499.3	4,739.5	6,013.4	7,501.4
TOTAL	23,738.2	31,828.7	33,235.4	**41,161.6**	51,472.8

Table 6.5: Degree hour values for various balance temperatures.

- **Heat Load Factor**

Heat load factor for İzmir City is calculated using Equation (3.1) for 0°C outdoor design temperature. The results are given in Table 6.6 and exhibit a variation between 0.157 and 0.670 for 20°C balance temperature. Thus, for Campus design case to choose the maximum heat load factor value, 0.67, will be conservative. In geothermal district heating system design it is common practice to use a heat load factor around 0.6 [38].

Table 6.6: Heat load factors for various balance temperatures.

MONTH	Balance Temperature (°C)				
	16	18	18.3	**20**	22
January	0.542	0.593	0.600	0.634	0.667
February	0.588	0.634	0.640	**0.670**	0.700
March	0.320	0.384	0.393	0.442	0.493
April	0.209	0.295	0.306	0.365	0.423
May	0.072	0.119	0.125	0.157	0.188
June	0	0	0	0	0
July	0	0	0	0	0
August	0	0	0	0	0
September	0	0	0	0	0
October	0.040	0.092	0.098	0.167	0.243
November	0.220	0.307	0.318	0.376	0.433
December	0.369	0.439	0.448	0.495	0.541
TOTAL	0.295	0.358	0.366	0.413	0.461

Table 6.7 summarizes the previous calculations made for 20°C balance temperature. Table indicates that annual average outdoor temperature is 23.3°C, total heating hours is 5,057 for 202 heating days. Annual degree-day and degree-hour values are 1,738 and 41,162 respectively.

Table 6.7: Summary of the calculations for 20°C balance temperature.

MONTH	Average Outdoor Temperature (°C)	Number of Heating Day	Degree-Day Value	Number of Heating Hour	Degree-Hour Value	Heat Load Factor
January	7.3	31	392.9	744	9,428.4	0.634
February	6.6	28	375.3	672	9,007.5	0.670
March	11.2	31	274.3	700	6,469.9	0.442
April	15.4	27	148.5	600	3,389.0	0.365
May	19.6	17	53.1	442	1,171.4	0.157
June	25.6	0	0	0	0.0	0
July	27.5	0	0	0	0.0	0
August	27.9	0	0	0	0.0	0
September	23.6	1	0.1	161	0.7	0
October	20.6	10	18.1	352	353.9	0.167
November	12.6	26	225.0	642	5,327.5	0.376
December	11.9	31	250.6	744	6,013.4	0.495
TOTAL	**23.3**	**202**	**1,737.8**	**5,057**	**41,161.6**	**0.413**

IZTECH Campus is active mostly during the office hours, which is between 9.00 a.m. and 17.00 p.m. during the week. Thus, annual heat requirement is calculated from Equation (3.7) for hourly outdoor temperature data. Degree-hour value of the Campus is calculated as 9,155 for working hours for 20°C balance and 0°C design outdoor temperature. Consequently, annual heat requirement of the Campus is determined as 5,129,892 kWh from Equation (3.9) for working hours using Campus degree-hour value. The results are given in Table 6.8.

Table 6.8: Total peak load and annual heat requirement of the Campus.

	Total Peak Load (kW)	Total Annual Heat Requirement (kWh)
Existing Buildings	3,662	1,676,244
New Buildings	7,545	3,453,648
Total	**11,207**	**5,129,892**

6.1.3 Thermal Load Density of the Campus

Thermal load density is an important indication on the decision whether if the district heating system should be installed or not.

The buildings in the Campus are distributed in an area as large as 71.3 ha as it is shown in Figure 6.3. Using the area and total heat load, thermal load density of the Campus is calculated as 0.16 W/ha. According to the criteria, which are given in Table 4.1, favourability of district heating system for IZTECH Campus is questionable, because of the widespread distribution of the buildings. On the other hand, only heating requirements are considered in this study. If cooling is also aimed, the favourability ratio increases depending on the total heat load.

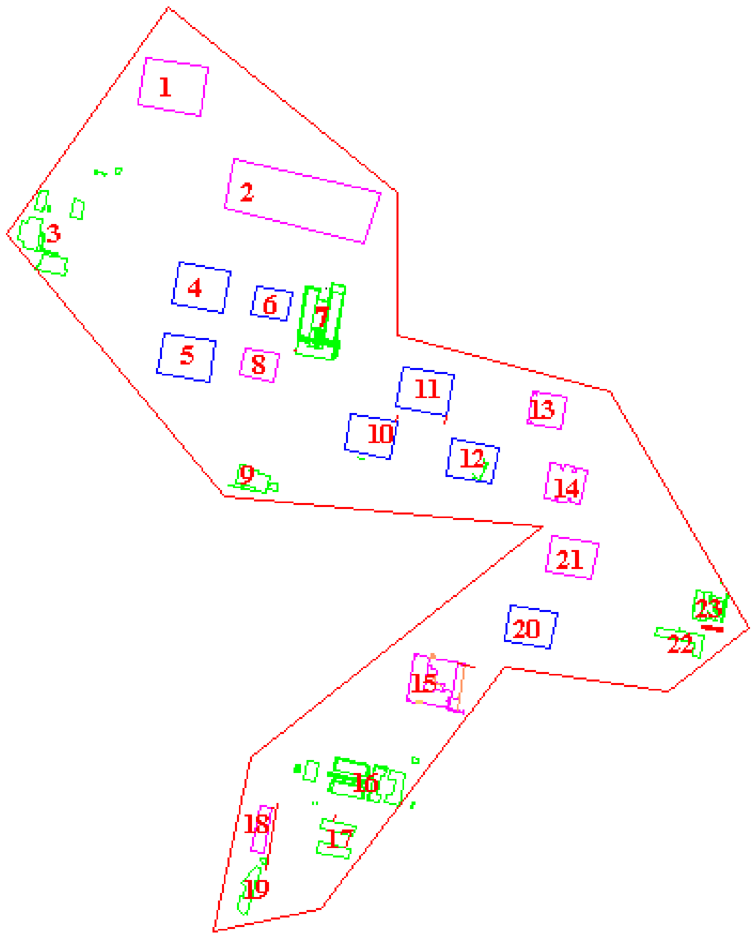


Figure 6.3: Location of the buildings in the Campus

Considering the future development and cooling requirements, Campus DHS is appeared to be possible.

Once the heat load and annual heating requirements are determined and favourability of DHS is proven, it can be moved to the next step which is to develop scenarios to able to determine the best heating system technically and economically.

District heating system design consists of two parts; heating system design and piping network design.

6.1.4 Design of Heating System

In this Thesis, mainly two heating system types are considered; GDHS and a conventional heating system. The type of geothermal heating system considered is "heat pump heating system" given the low geothermal fluid temperature, which is 33°C. Fuel boiler heating system is chosen as conventional heating system since it is the system since it is the system that currently used.

In this section, considered heating systems are simulated using building heat loss, heating equipment, building energy storage and heating system model, which are explained in Chapter 5 and results are given.

Assumptions:

- Intermittent heating system, which is turned on at certain hours of the day, is adapted to all heating system types because the Campus is used mostly during the office hours, which is between 09.00 a.m. and 17.00 p.m. In the simulations, the system is turned on one hour earlier and turned off at the end of this period.
- Geothermal flowrate of the existing production well is not yet known, but between 10 and 40 kg/s is expected. Therefore, each well is assumed with 30 kg/s flowrate.
- As a control system constant flowrate and variable return water temperature is used.
- All of the Campus is considered as a single building.
- Temperature drop along the pipeline is omitted.
- The piping network of the Campus is considered as two loops, geothermal and Campus loops. And each loop is considered with two-pipe system as supply and return mains (Figure 6.4).

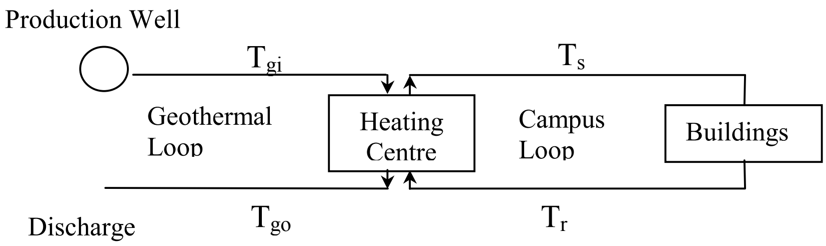


Figure 6.4: Schematic of the district heating system.

Design of both alternatives will be given in detailed in the following sections.

6.1.4.1 Heat Pump District Heating System

As it was mentioned in Chapter 5, there are two heat pump options; heat pump only (HPO) and heat pump assisted (HPA). Between two options, heat pump only layout is selected because it exhibits better performance than heat pump assisted at geothermal fluid temperatures below 40°C. Considered heat pump district heating system (HPDHS) is shown in Figure 5.6. Because of the corrosive effects of geothermal fluid, a heat exchanger (HEX) is employed prior to the heat pump unit.

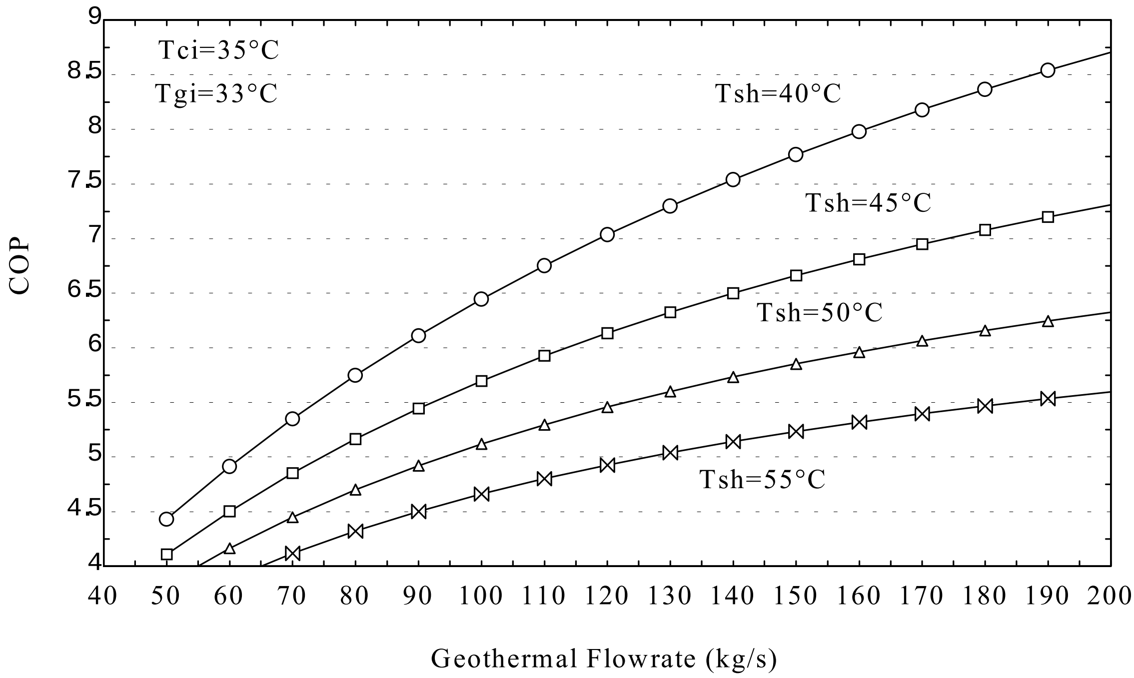


Figure 6.5: Relationship between geothermal flowrate and COP.

The parameters to be determined for the heat pump heating system (Figure 5.6) 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.

- **Determining 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 6.5 for various condenser outlet temperatures (40°C-55°C) at 35°C condenser inlet temperature and 33°C geothermal fluid temperature (T_{gi}). 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, increase in 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) temperature. 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 results.

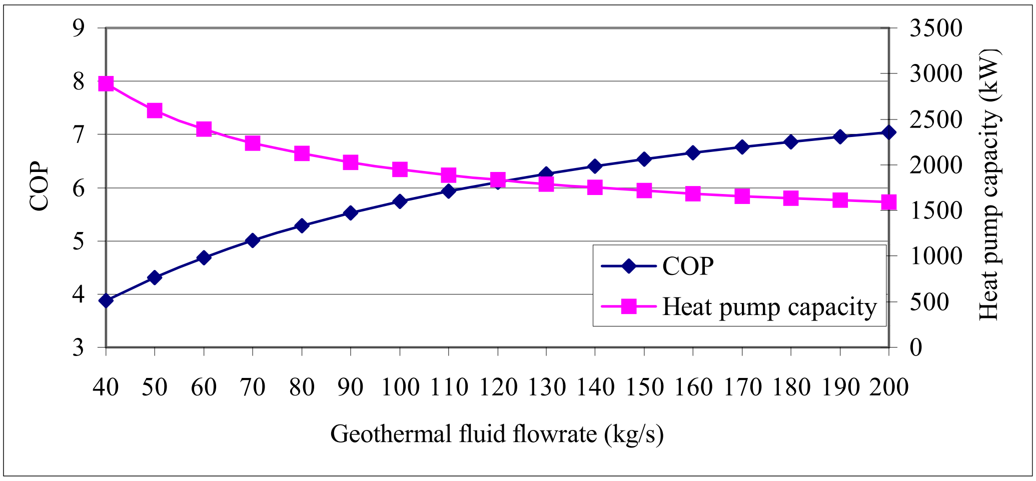


Figure 6.6: Relationship between geothermal fluid flowrate, COP and compressor work for 45° C condenser outlet temperature.

- **Geothermal Fluid Flowrate**

For determined condenser outlet temperature, relationship among geothermal fluid flowrate, COP and heat pump capacity is shown in Figure 6.6. With chosen COP of 6.2, Figure 6.6 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.

Depending on 45°C condenser outlet temperature, heating system is selected to have 45°C supply/35°C return temperature. Consequently, flowrate in the Campus loop, is calculated as 179.4 kg/s with Equation (5.2) based on heat load factor of 0.67.

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 Chapter 5. Simulation program, which is written in Matlab, is given in Appendix B.

For IZTECH Campus DHS, 4 separate heat pump units of same capacity are considered because of the improved performance, reliability and operational flexibility [15]. 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°C and 5°C, all heat pumps,

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

Simulation Results

- Heating system runs 1055 hours for determined heating period.
- Variations of the heat pump and heat exchanger characteristics for heating period are shown in Figure 6.7.

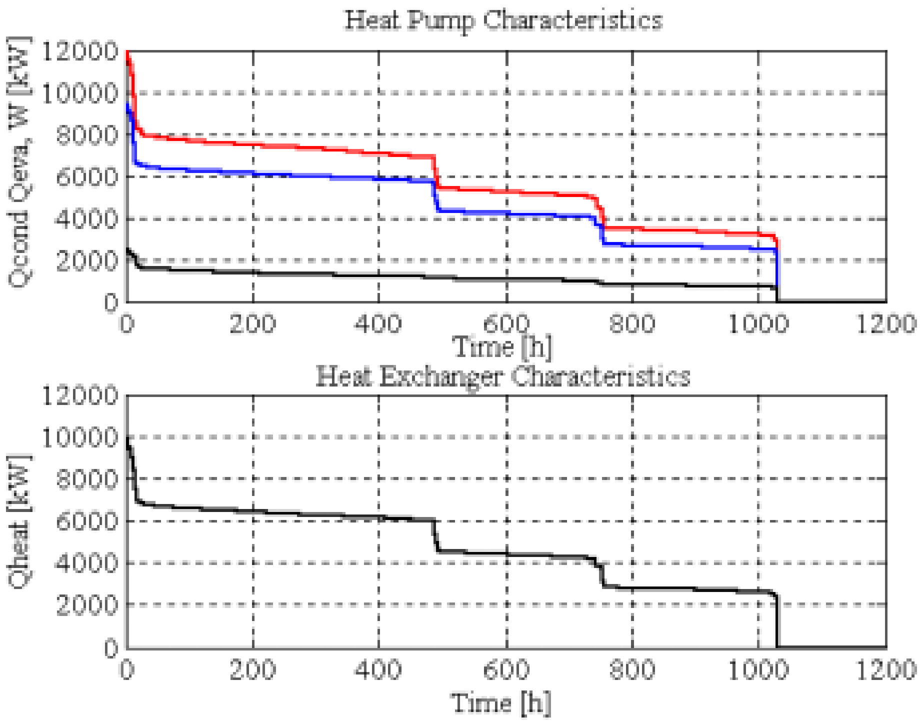


Figure 6.7: Variations of heat pump and heat exchanger characteristics according to simulation program.

The heat pump units are considered to be installed in parallel. Thus, there is a relation between geothermal flowrate and the number of heat pump units in operation. If total number of heat pump units is N_{hp}, the number of heat pump units in operation at any time is n_{hp} and maximum flowrate of the geothermal fluid is m_{g0}, geothermal flowrate at any time can be calculated as;

$$m_g = \frac{n_{hp}}{N_{hp}} \cdot m_{g0} \tag{6.1}$$

- Variations of the geothermal flowrate requirements for heating period are calculated using Equation (6.1) and shown in Figure 6.8. Variable flowrate of the geothermal fluid can be managed using storage tank or variable speed circulation pumps.
- The peak flowrate is 120 kg/s and annual flowrate requirement is 281,124 tones, which is the area under the line in the Figure 6.8. The average necessary flowrate is

calculated as 74 kg/s.

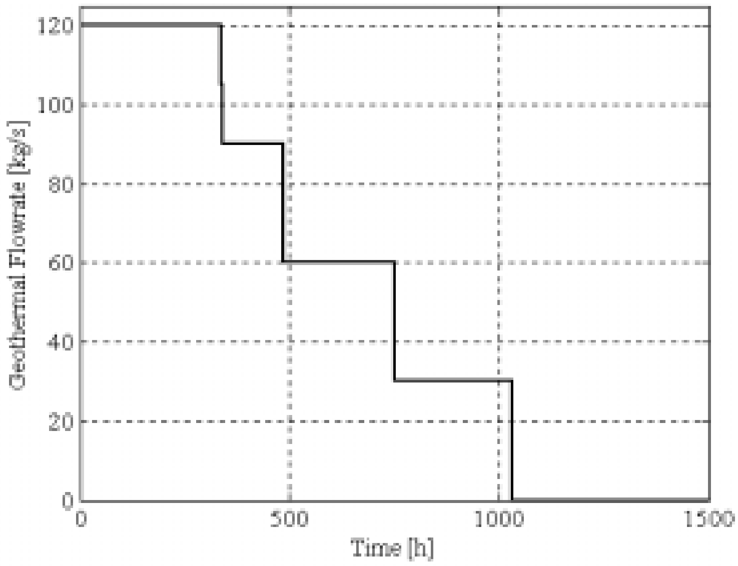


Figure 6.8: Variation of geothermal flowrates during the heating period.

- Geothermal fluid return temperature varies between 10 and 21°C and is displayed in Figure 6.9.

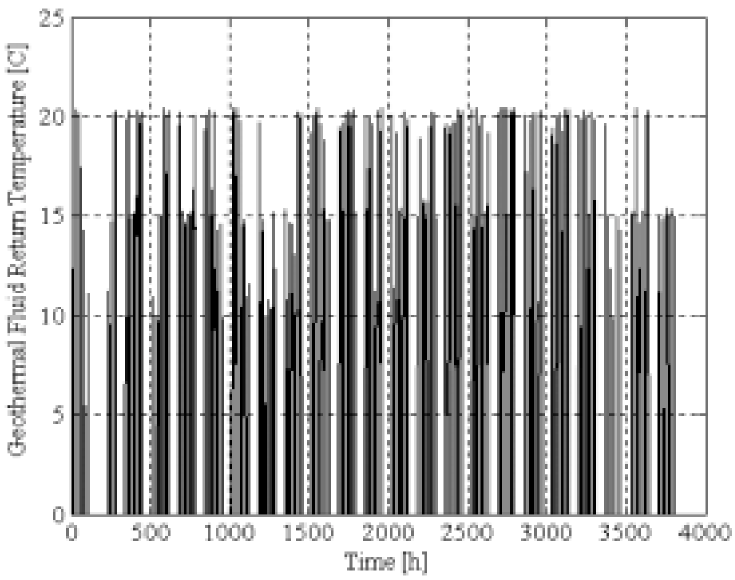


Figure 6.9: Variation of return temperatures of geothermal fluid

Campus loop is operated for constant flowrate, variable supply temperature principle. Supply temperature changes with the number of the heat pump units in operation. If heat pump is completely shut down, condenser outlet temperature equals to return temperature at previous step for that heat pump. Thus, supply temperature of the Campus loop system can be calculated as;

$$T_s(i) = \frac{T_{sh}(i) \cdot n_{hp} + T_r(i-1) \cdot (N_{hp} - n_{hp})}{N_{hp}} \tag{6.2}$$

Here, T_{sh} represents condenser outlet temperature, when heat pump is operating and (i-1) represents previous step.

- Variations of the supply and return water temperatures during the heating season can be viewed in Figure 6.10.

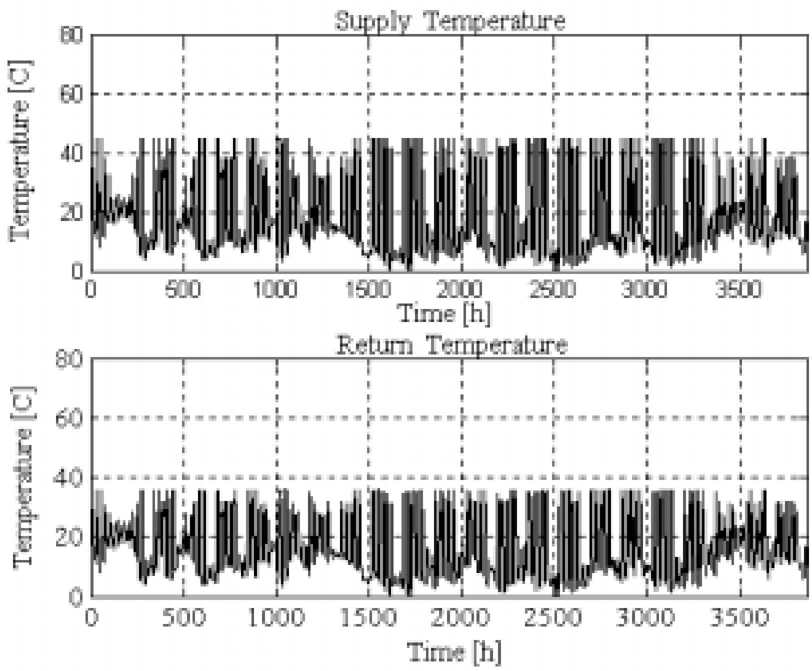


Figure 6.10: Variations of supply and return water temperatures.

- For steady state approach indoor temperature is assumed constant at balance temperature, 20°C. But for dynamic approach indoor temperature is calculated using

Equation (5.26). Figure 6.11 exhibits the indoor and outdoor temperature variations are shown during the heating season.

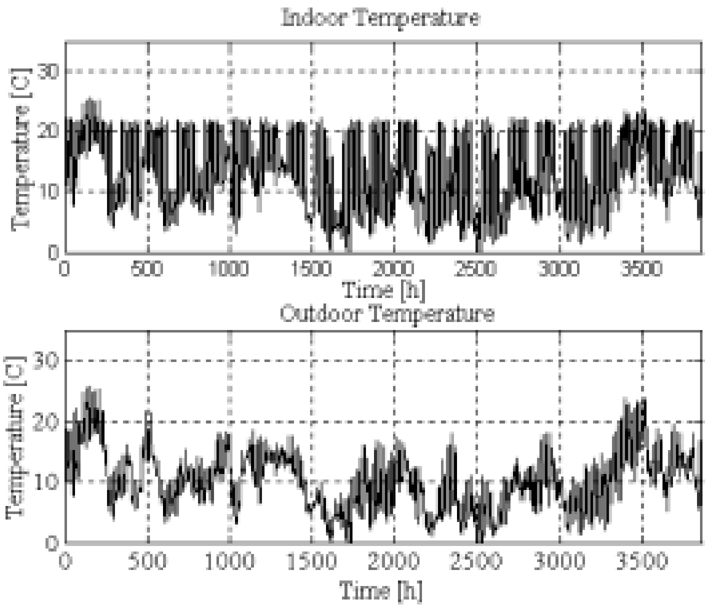


Figure 6.11: Variation of indoor and outdoor temperatures during the heating season.

- Heating equipment heat supply and building heat loss duration curves are shown in Figure 6.12. As it is clearly seen that fan coils meet the required duty most of the time except some points, which are shown in the circles in the Figure.

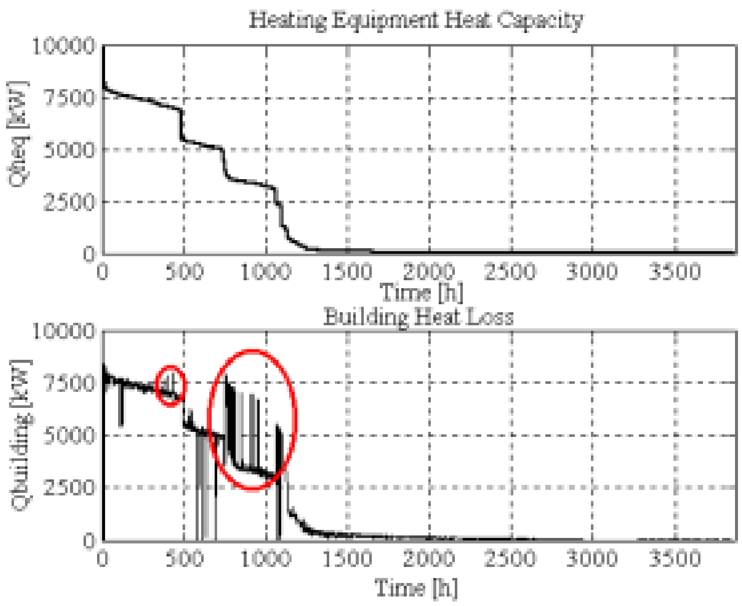


Figure 6.12: Duration curves of heating equipment heat supply and building heat loss.

* Variations of indoor temperature and outdoor temperature during the operating period are shown in Figure 6.13.

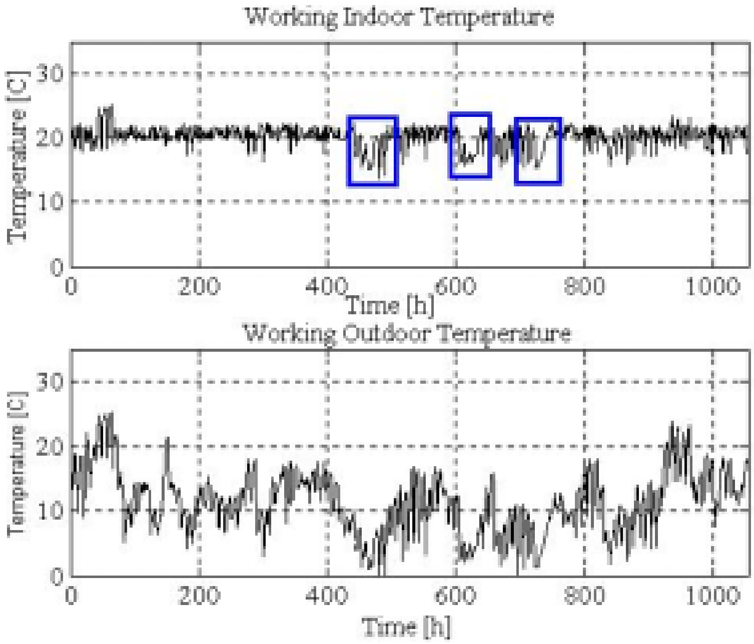


Figure 6.13: Variations of indoor and outdoor temperature during the working period.

As it can be seen from the figure, the indoor temperature is mostly over 20°C, but some times, which are shown in the rectangular, it is below 20°C. At that times, outdoor temperature drops below 5°C, and since the system is designed for 67% of the peak load, heat supply from the heating equipments are not enough to maintain 20°C indoor temperature.

6.1.4.2 Boiler Heating System

The current heating system of the Campus is individual boilers employed at each building group. The existing boiler capacities are given in Table 2.2.

Boiler heating system (BHS) is simulated according to Section 5.7.2 and the results will be compared with the heat pump district heating system. The simulation program, which is written in Matlab, is given in Appendix C.

- **Determining the best heating regime**

Individual heating systems in the Campus are operated manually by technicians. Each technician turns on or 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 each alternative, simulation results are given in Table 6.9.

Table 6.9: Results of the simulations of FBDHS.

Alternative No	Boiler set temperature (°C)	Average indoor temperature during the working hours (°C)	Fuel-oil consumption of the boiler heating system (kg)	Cost of the fuel consumption (US$)
1	Tb_set=80	24.3	899,540	476,756
2	Tb_set=60	21.7	684,048	362,545
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	351,330
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	327,780

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.

Despite the existing individual boiler system, to be able to compare the boiler system with HPDHS, a fuel boiler district heating system (FBDHS) will be designed here. Piping network and location of the heat centre are chosen the same with heat pump district heating system. 5 boiler with 2326 kW/each are considered for fuel boiler district heating system. Individual fuel boiler heating system (IFBHS) is also considered as an alternative in economical analysis Section 6.2.

In the simulations, the temperature difference for boiler set temperature is taken as 4°C. That means, when boiler outlet temperature exceeds boiler set temperature more than 4°C, boiler turns off automatically. On the other hand, when boiler temperature decreases more than 4°C from boiler set temperature, boiler turns on automatically.

Indoor temperature variations of the alternatives are plotted in Figure 6.14-6.17.

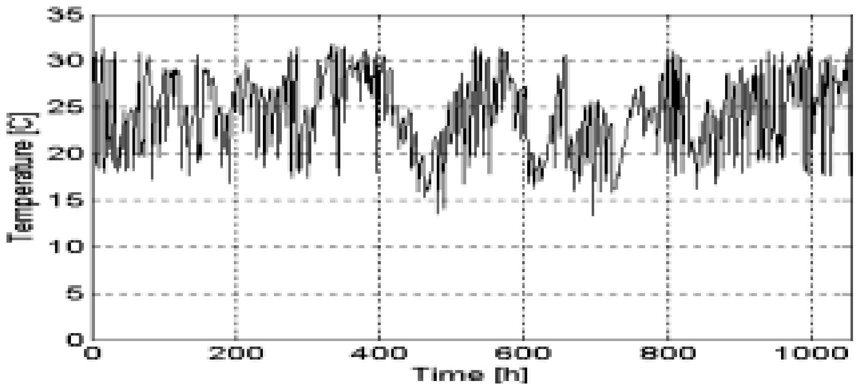


Figure 6.14: Variation of indoor temperatures of Alternative 1 of FBDHS.

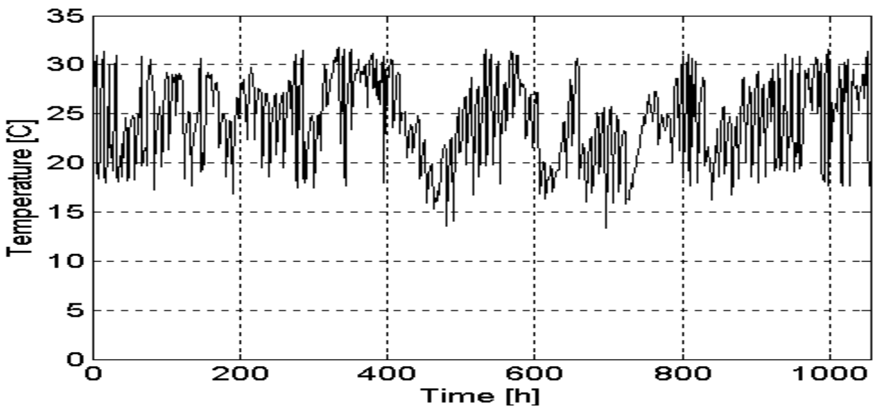


Figure 6.15: Variation of indoor temperatures of Alternative 2 of FBDHS.

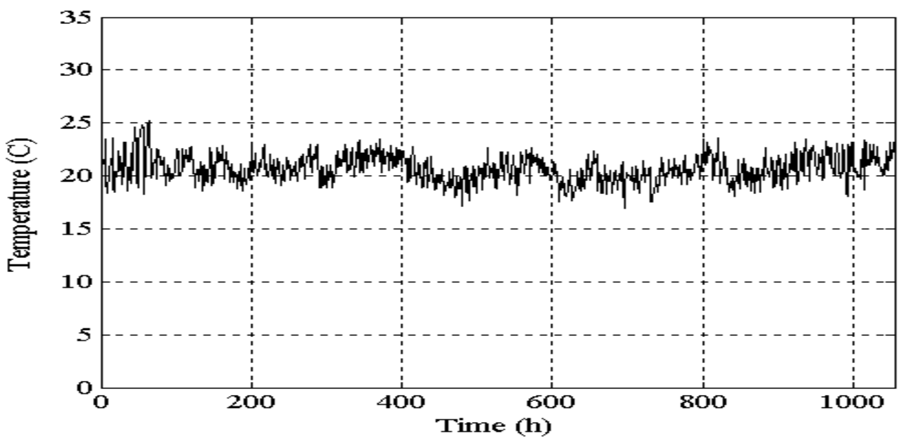


Figure 6.16: Variation of indoor temperatures of Alternative 3 of FBDHS.

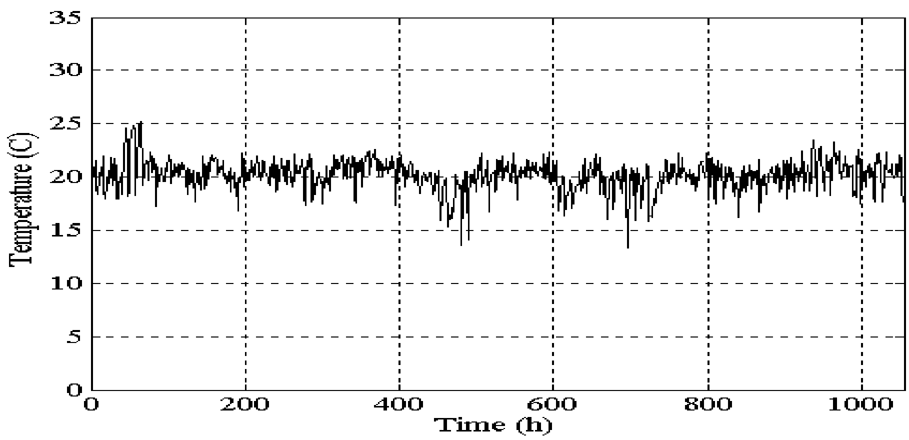


Figure 6.17: Variation of indoor temperatures of Alternative 4 of FBDHS.

Table 6.9 and Figure 6.14-6.17 indicates that Alternative 4, which uses a boiler set temperature recommended by Demirdokum [39], is the best one with least fuel consumption and best indoor temperature around 20°C.

6.1.5 Design of Piping Network for the Campus

The design of the district heating piping network is of vital importance to the economics of the system. There is a trade-off between economics and reliability depending upon the pipe material, insulation and placement method selected [30].

Schematic of the district heating system is shown in Figure 6.4. As it can be seen from the figure, the piping network of the Campus is considered as two loops, geothermal and Campus loops. And each loop is considered with two-pipe system as supply and return mains.

District heating network is designed considering not only existing buildings but also future development.

- **Heat centre Location Alternatives**

The location of the heat centre is very important depending on the economy of the district heating system for designing a district heating piping network. Therefore, several alternatives have been studied and 3 of which are given in the Thesis.

Alternative 1: Heat centre is close to the Campus entrance (Figure 6.18);

Alternative 2: Heat centre is close the production well (Figure 6.19);

Alternative 3: Heat centre is almost in the middle of the Campus (Figure 6.20).

Diameters of pipes in the network are selected according to Section 4.2.3.1. Pipelab software, which has been created under Matlab program by Valdimarsson [18], is used. Necessary input file includes number of nodes in the system, their xyz coordinates, connectivity relation to the nodes of the pipes with their length, diameter, roughness and heat loss, boundary conditions, necessary flowrate and the pressure head at the starting point. Initially, pipe diameters are assumed and using Pipelab software optimum diameter for each pipe is calculated. Pressure drop of the critical line and pipe diameters are calculated for the Campus loop (45/35°C) and the results are given in Table 6.10 for each heat centre location alternative. Table 6.10 gives total pressure drops and pipe lengths in various diameters for each alternative. Alternative 2 has the highest pressure drop and requires the longest pipeline. Alternative 1 and 3 are close to eachother in pipe length but Alternative 3 requires shorter piping for larger diameters. This makes it cost efficient. Because of the lowest piping cost and pressure drop, Alternative 3 seems to be the best option.

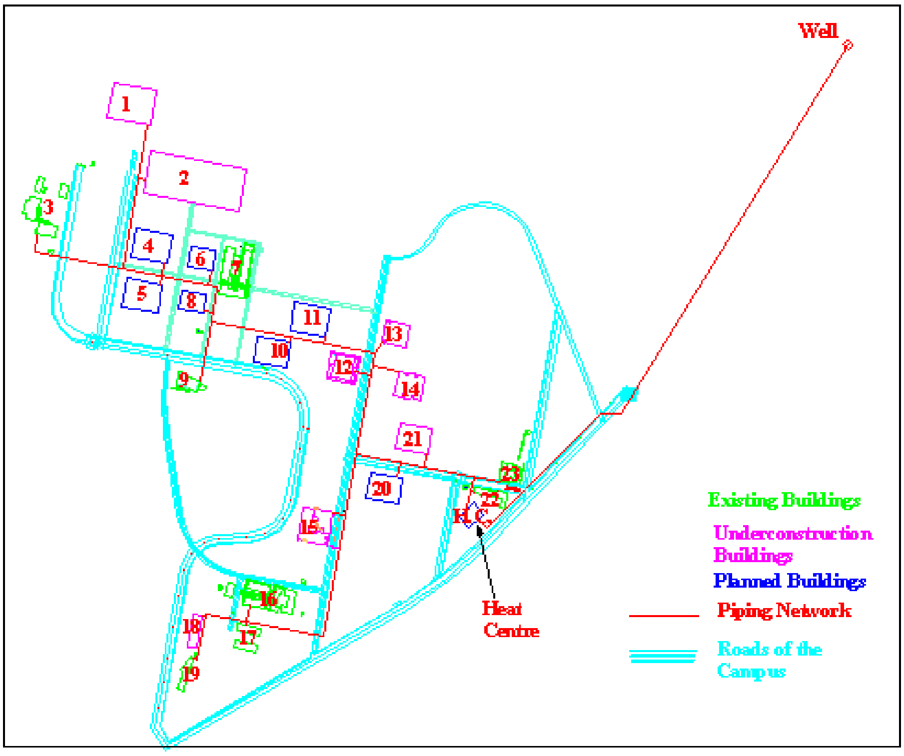


Figure 6.18: Alternative 1 for district heating piping network.

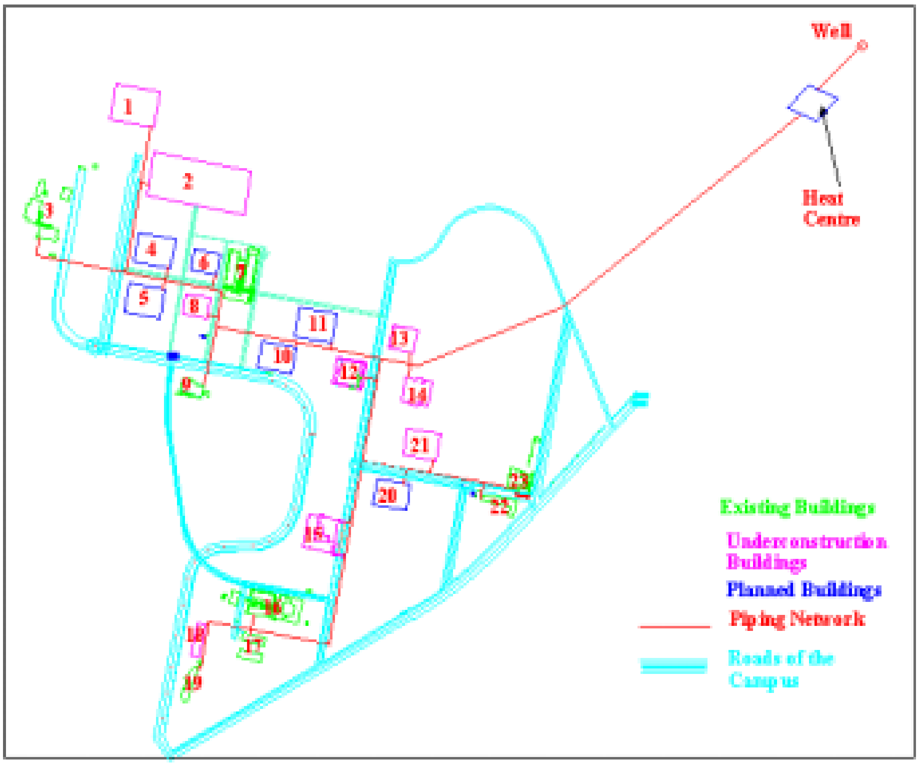
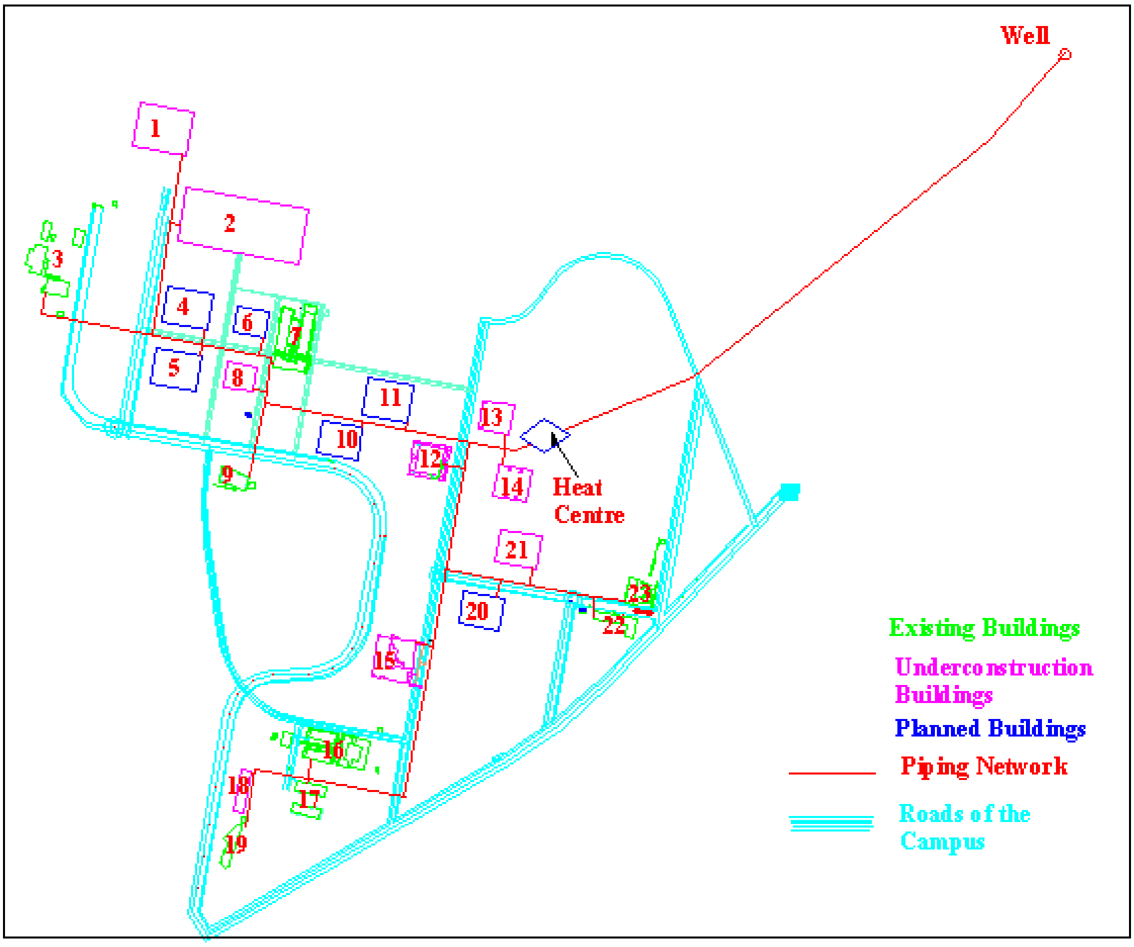


Figure 6.19: Alternative 2 for district heating piping network.



1: Staff houses
2: Dormitories
3: Architecture Faculty
4: Buildings A
5: Buildings B
6: Buildings C
7: Science Faculty
8: Buildings D
9: Cafeteria
10: Buildings E
11: Buildings F
12: Research & Development. Centre
13: Sport Centre
14: Medical Centre
15: Chemical Engineering
16: Engineering Faculty
17: Mechanical Engineering
18: Mechatronic Building
19: Incubator Building
20: Buildings G
21: Library
22: Presidency of Depart. Building
23: Rectorship Building

Figure 6.20: Alternative 2 for district heating piping network.

Table 6.10: Design alternatives for various heat centre location (45/35°C, Campus loop).

Alternative of the piping network	Total pressure drop (m)	Pipe Length (m)											Cost (US$)
		DN80	DN100	DN125	DN150	DN200	DN250	DN300	DN350	DN400	DN450	Total	
Alternative 1	14.8	41.36	639.28	494.64	304.58	728.56	537.62	175.86	193.94	324.26	39.27	3,479.37	713,774
Alternative 2	17.1	41.36	632.17	494.97	423.68	894.28	534.88	374.47	0	70.44	1120	4,586.25	687,981
Alternative 3	11.2	41.36	632.17	494.97	423.68	894.28	534.88	374.47	0	70.44	52.92	3,519.17	532,117

- **Materials and Installation Type**

Unit cost of carbon steel and composite pipes are given in Table 6.11. As it can be seen in Table 6.11, carbon steel pipes, which are commonly used in geothermal applications, are nearly 13-35% cheaper than composite pipes depending on the pipe diameter. But carbon steel pipes are not suitable for geothermal loop, because of the corrosive effects of the geothermal fluid. Thus, composite (FRP) pipes are used for geothermal loop while Campus loop is installed with carbon steel pipes.

Table 6.11: Unit cost of the carbon steel and composite pipes [40].

	Diameter (DN)	65	80	100	125	150	200	250	300	350	400	450
Unit cost ($/m)	Carbon Steel Pipe (Steel+PU+PE)	13	15	25	26	31	47	57	72	95	128	165
	Composite Pipe (CTP+PU+CTP)	20	23	31	32	42	54	76	87	115	155	199

The total length of the pipes in the supply main of Campus loop and geothermal loop are 3,520 m and 1,223 m, respectively. Geothermal fluid disposal can be managed in two ways depending on chemical properties of the geothermal fluid and reservoir properties. One is reject the fluid to the waterways or the sea, the other one is to drill an injection well and inject the fluid to this well. First alternative is chosen since the chemical properties of the fluid is very close to the seawater [14] and there is no study on an injection well location in the field so far. Geothermal return main is the same as the supply main in length and diameter. Thus, total length of considered network of the district heating system is nearly 9,486 m.

6.2 Economical Analysis

Initially the investment and operational costs of the heating system alternatives, which are HPDHS and fuel boiler heating systems (FBDHS and IFBHS), are calculated. Then, the investment costs are analysed according to internal rate of return (IRR) method.

In economical analysis 1 US$ is 1,500,000 TL. (March 2003).

6.2.1 Investment Cost of the Heating System

Main equipments are selected and approximate investment cost is calculated for each alternative heating system types.

6.2.1.1 Heat Pump District Heating System

A. Selection and Cost of the Equipment

Considered main equipment of HPDHS are

- Piping network,
- Well pumps,

- Geothermal fluid circulation pumps,
- Campus loop circulation pumps,
- Heat pumps,
- Heat exchangers,
- Heating equipment.

- **Piping Network**

Optimum pipe diameters are calculated based on the target pressure loss, water velocity, head loss in the network and economy of the system. Piping network has an important share of the total investment cost. Thus, optimisation of the pipe diameter is very important for economy of the system. Pipe diameter selection also has a crucial impact on pumping cost (operational cost).

As it was explained in Section 6.1.4.1, for heat pump heating system, Campus loop is designed for 45/35°C. Based on this temperature regime, pipe diameters are calculated by Pipelab software.

The pressure loss per unit length is a common design parameter. Various target pressure losses, which are 62.5, 100 and 150 Pa/m, are tested for diameter selection using Pipelab software. The results are given in Table 6.12, where length of each pipe diameter and total piping cost can be viewed for different pressure losses.

Table 6.12: Pipe diameters in the supply main of the Campus loop for various target pressure loss.

Target Pressure Loss (Pa/m)	Pipe Length (m)											Pipe Cost (US$)
	DN65	DN80	DN100	DN125	DN150	DN200	DN250	DN300	DN350	DN400	DN450	
62.5		41	632	495	424	894	535	374	0	70	53	159,657
100	41	429	362	521	726	443	821	53	70	53		136,963
150	41	489	638	205	1000	565	457		123			124,495

If the pressure loss is high, investment cost of the pipe is low, but the operating cost is high. On the other hand, if the pressure loss is low, the investment (pipe diameter are larger) is badly utilized, but the pumping cost is low. Heat loss in a district heating pipe is higher for badly utilized pipes. The pressure loss per unit length is thus a good indicator of optimality, but not a real cost function.

Pumping cost, which will be explained in detailed in the following sections, and piping costs for various target pressure losses are given in Figure 6.21. The piping cost in Figure 6.21 is calculated using unit pipe cost of each pipe diameter [40]. The Figure indicates that, while piping cost decreases drastically with increasing target pressure loss, operational pumping cost is nearly constant. Therefore, the highest acceptable target pressure loss, 150 Pa/m, is selected for the district heating piping network.

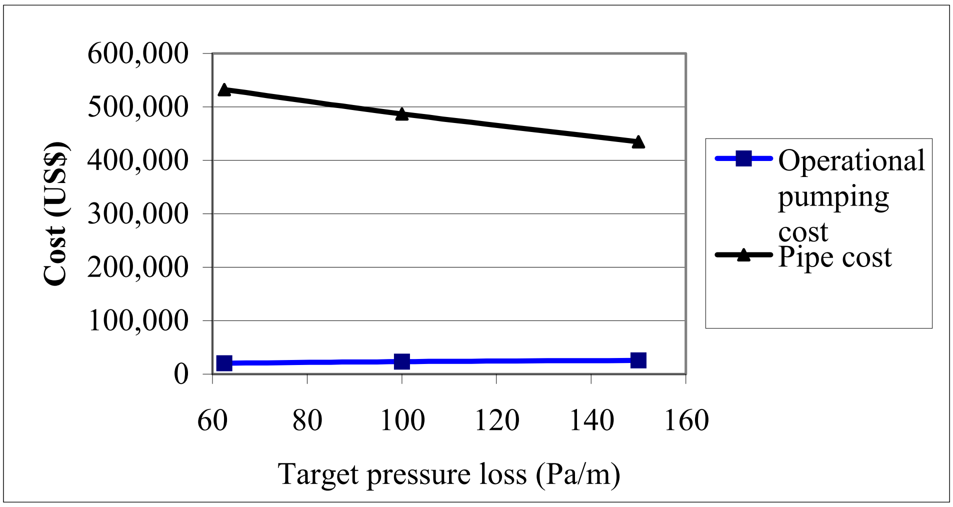


Figure 6.21: Operational pumping and piping cost of the district heating network of the HPDHS for IZTECH Campus.

Length of each pipe diameter for the supply main can be seen in Table 6.12 and Figure 6.22. The largest nominal diameter in the network is 350 and the smallest diameter is 65. Number of the nodes in the supply main is 46 and total pipe length is 3,520 m. Return main is assumed to have the same pipe diameter and length with supply main.

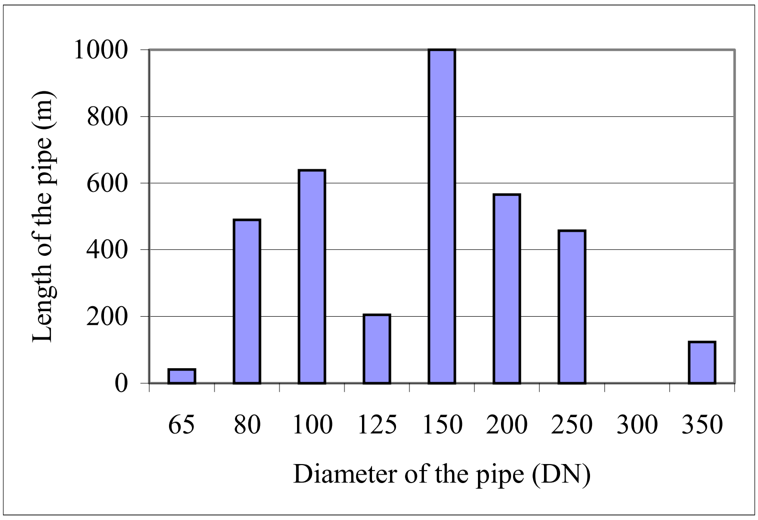


Figure 6.22: Length of each diameter used in the supply main of the Campus loop.

Pipelab software also exhibits the head loss distribution on the network. The *h/L* diagram of the supply main of the Campus loop is shown on Figure 6.23. Pressure drop is calculated as 24.6 m for the supply and return mains. Heat centre pressure drop is assumed as 25 m. Thus, pressure head for the system is 80 m.

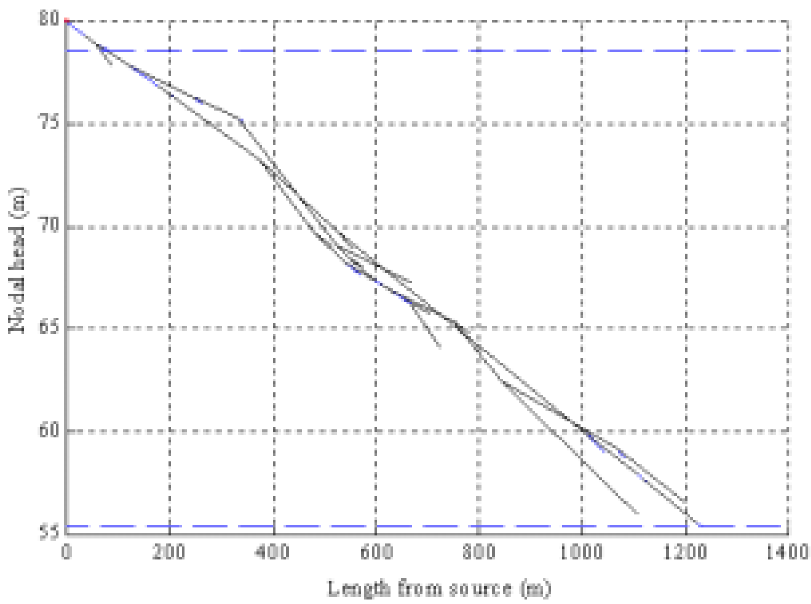


Figure 6.23: h/L diagram for Campus loop supply main of HPDHS.

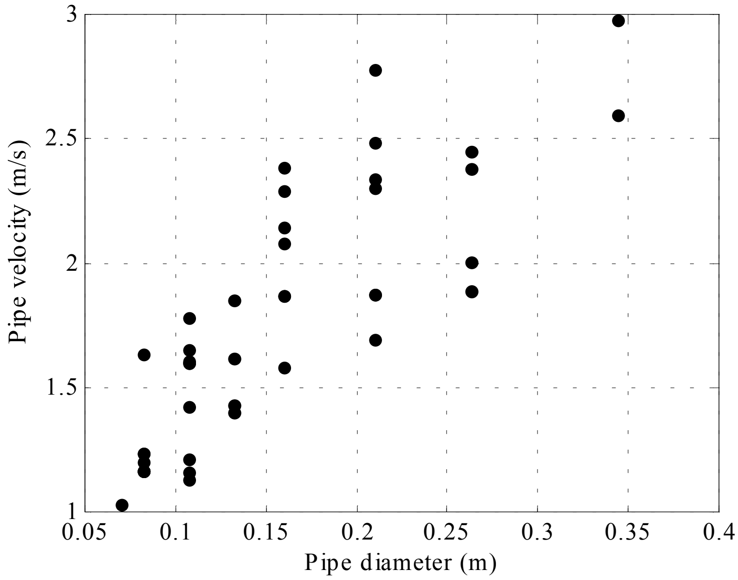


Figure 6.24: Relationship between pipe diameter and velocity of water.

Water velocities in each pipe diameter are calculated by the help of Pipelab software and the relationship between the pipe diameter and the water velocity is displayed in Figure 6.24. Velocity range is calculated as 1.03-2.98 m/s. The results are also the same for the Campus return main.

Same calculations and simulation are conducted for the geothermal loop. In the supply main total length of the pipes is 1,223 m, total pressure drop is 25.3 m for DN250 pipe diameter. The results are also the same for the return main of the geothermal loop.

Total piping cost of the Campus and geothermal loops is 248,991 US$ and 185,896 US$, respectively using the values in Table 6.11 and Table 6.12. Thus, the total piping cost of the district heating system amounts to 434,887 US$ approximately. Cost of fittings and wages for the workers are assumed to be 30% of total cost of pipeline.

Installation type

Pipelines are installed either aboveground or underground. Underground installation is preferred here to avoid heat losses and esthetic concerns. Underground installation could be directly buried into the soil or in concrete tunnels. Unit construction cost for underground (buried) installation is 33.4 US$/m and for underground (concrete tunnel) is about 200 US$/m [41].

Total piping cost including construction and fittings and wages is given in Table 6.13.

Table 6.13: Total piping cost for HPDHS for underground installation.

Cost (US$)	Buried	Concrete Tunnel
Total pipe cost	434,887	434,887
Fittings and wages	130,466	130,466
Construction	158,132	948,600
TOTAL	**723,425**	**1,672,085**

Table 6.13 clearly indicates that piping cost is 2.3 times more expensive for concrete tunnel. Therefore, for IZTECH Campus DHS underground (buried) pipeline installation is selected. Thus, all pipes are isolated in the piping network.

- **Well Pumps**

Selection of the well pump depends on various factors, such as the dimension of the well, characteristics of the geothermal fluid, flowrate, temperature, etc.

As it was mentioned in Section 6.1.4.1, for considered heat pump district heating system, 4 wells are required to obtain sufficient geothermal fluid. The temperature of the existing well fluid is 33°C and the depth is 260 m. It is assumed that the following drilled wells will have the same temperature, depth, a 30kg/s flowrate and 100 m pressure drop.

Unit drilling cost is about 350 US$/m. Thus, the drilling cost of one well (260 m) is 91,000 US$ and the total drilling cost for the system is 364,000 US$.

For geothermal production well of the IZTECH Campus, Vansan 103-1032 Deep Well Pump with 8 stages was selected. Cost of each well pump is about 30,000 US$ [42] and total well pump cost is 120,000 US$.

- **Geothermal Fluid Circulation Pumps**

As it was explained in Part A of Section 6.2.1.1, total pressure drop of the pipeline in the geothermal loop is about 25.3 m for both supply and return mains. Heat centre pressure drop is assumed as 25 m. Thus, total pressure drop of the geothermal loop is taken as 80 m.

To meet the required duty, 5 pumps (4 permanent and 1 standby) with 108 m^3/h and 80 m should be employed. Each pump costs 12,000 US$ [41] and total geothermal circulation pump cost is 60,000 US$ for 8 pumps.

- **Campus Loop Circulation Pumps**

In Part A of Section 6.2.1.1, the pressure drop of the supply main was calculated as 24.6 m. Total pressure drop of the pipeline in the Campus loop is about 49.2 m for supply and return mains. Heat centre pressure drop is assumed as 25 m. Thus, total pressure drop of the Campus loop is taken as 80 m.

According to Section 6.1.4.1, flowrate in the campus loop is 179.4 kg/s (646 m^3/h). To meet the required duty, 5 pumps (4 permanent, 1 standby) with 161.5 m^3/h and 80 m should be employed. Its cost is about 16,340 US$/pump [41] and total circulation pump cost is 81,700 US$ for 5 pumps.

- **Heat Pumps**

4 main heat pumps (fluid to fluid) with a capacity of 1,836 kW/each will be employed to the system as it was mentioned in Section 6.1.4.1. The cost of each unit is 250,000 US$ [41] and total heat pump cost is 1,000,000 US$ for 4 heat pumps units.

- **Heat Exchangers**

A plate-type heat exchanger is considered to serve as the central heat exchanger, for liquid-to-liquid heat transfer. This choice is primarily based on the following factors: low temperature losses, good corrosion resistance, less space required, favourable economics and the ease of accommodation of the changes in flow and temperature conditions by adding or removing plates.

APV-Q80 127, 170 plates, 4 titanium heat exchangers are selected for the heat centre with a 2,845 kW/each capacity. Each HEX cost is 29,000 US$ [43] and total heat exchanger cost is 116,000 US$.

- **Heating Equipment**

Fan coils are used as heating equipment in the existing system. Performance of fan coil for 45/35°C heating regime of HPDHS, which was determined in Section 6.1.4.1, is calculated as 33.3% from Equation (5.5). That means size of the fan coil should be increased about 3 times according to 90/70°C heating regime. But in the Campus each building is required cooling system. Thus, fan coils are selected according to cooling loads for cooling regime, which is 7/12°C. As an example, the office, Z11, in the Engineering Faculty is examined based on its heating and cooling loads and selected fan-coil.

Table 6.14: Heating, cooling loads and selected fan-coil unit and its capacity for the office, Z11, in the Engineering Faculty [34, 44].

Space	Project Value			Selected Fan-Coil			
	Cooling Load (7/12°C)		Heating Load (18°C)	Unit	Cooling Capacity	Heating Capacity	
	Sensible (kW)	Total (kW)	(kW)		(kW)	(90/70°C) (kW)	(45/35°C) (kW)
Engineering Faculty. 27.1-Z11	1.79	2.24	1.29	1	1.97	7.17	2.39

As it can be seen in Table 6.14, in the mechanical project, required cooling and heating loads were calculated as 1.79 kW and 1.29 kW, respectively and one fan-coil unit with cooling capacity of 1.97 kW was selected for the office. The heating capacity of the selected fan-coil unit is calculated as 2.39 kW for 45/35°C heating regime. The results indicate that existing fan-coil capacity is greater than the required heating capacity and there is no extra fan-coil required for 45/35°C heating regime.

B. Total Investment Cost

Total cost of each equipment in the heat pump district heating system is summarized and total approximate cost is calculated as 3,040,125 US$ in Table 6.15. Heat pump cost constitutes about 33% of the total cost.

Table 6.15: Summary of the investment cost of the HPDHS.

Part of the System	Investment Cost	
	(US$)	(%)
Piping network	723,425	23.8
Drilling of the wells	364,000	12
Well pumps	120,000	3.9
Geothermal fluid circulation pumps	60,000	2.0
Secondary water circulation pumps	81,700	2.7
Heat pumps	1,000,000	32.9
Control system of the heat pumps	575.000	18.9
Heat exchangers	116,000	3.8
Heating equipments (fan coils)	0	0.0
TOTAL	**3,040,125**	**100.0**

6.2.1.2 Fuel Boiler Heating System

Fuel boiler heating system has two sub-alternatives, namely FBDHS and IFBHS. Heating regime is 90/70°C for each alternatives.

6.2.1.2.1 Fuel Boiler District Heating System

A. Selection and Cost of the Equipment

Considered main equipment of FBDHS are

- Boilers,
- Circulation pumps,
- Piping network,
- Heating equipment.

- **Boilers**

Piping network and location of the heat centre are considered the same as heat pump DHS. Conventional heating systems are designed according to peak load. Thus, design load of the fuel boiler district heating system is 11,207 kW and 6 boilers (5 permanent, 1 stand by) with 2,326 kW capacity/each are considered for the fuel boiler district heating system for the Campus. The cost of each fuel boiler is 10,080 US$ [45] and total fuel boiler cost is 60,480 US$.

- **Circulation Pumps**

Each fuel boiler is employed with a circulation pump, which costs 58,000 US$ in total [41].

- **Piping Network**

Pipe diameters are selected based on 90/70°C heating regime with the help of Pipelab program. Length of each diameter used in the supply main is given in Table 6.16 and Figure 6.25. The return main is the same as the supply main.

Table 6.16: Length of the each diameter of the pipes in the supply main of FBDHS.

Supply main	Campus Loop								
Diameter (DN)	DN50	DN65	DN80	DN100	DN125	DN150	DN200	DN250	**Total**
Length (m)	41	489	638	424	781	565	457	123	**3,520**

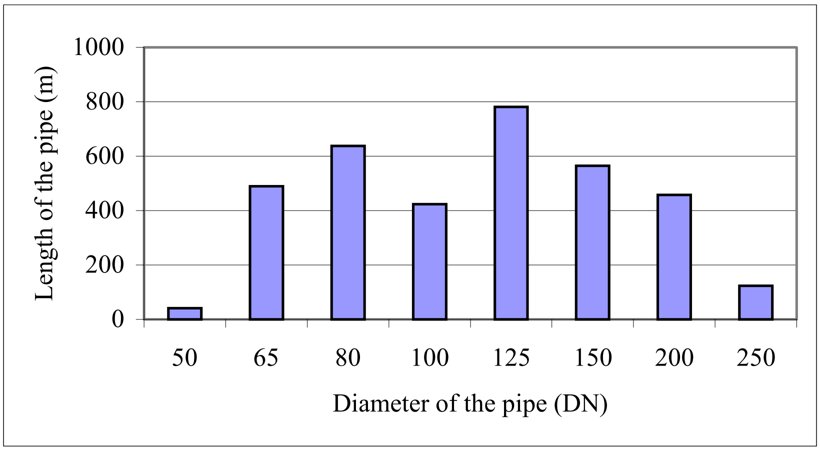


Figure 6.25: Length of each diameter used in the supply main.

The largest nominal diameter in the network is 250 and the smallest diameter is 50. Number of the nodes in the supply main is 46 and total pipe length is 3,520 m.

The head loss (*h/L)* diagram of the network after finding the nodal heads according to the least squares methods is shown on Figure 6.26. The target pressure loss has been selected same as heat pump piping network, 150 Pa/m, and pressure drop is calculated as 25.3 m for the supply main. Pressure head for the system is calculated as 80 m.

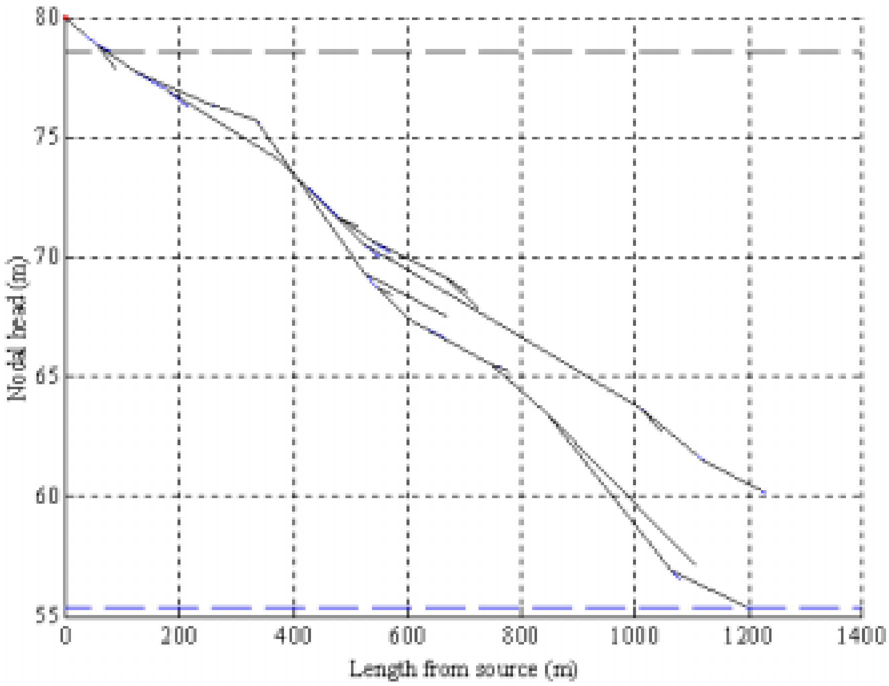


Figure 6.26: h/L diagram for the district heating system.

Relationship between the pipe diameter and the water velocity in the pipes is shown on Figure 6.27. Range of the velocity is 0.8-2.55 m/s. The results are also the same for the campus return main.

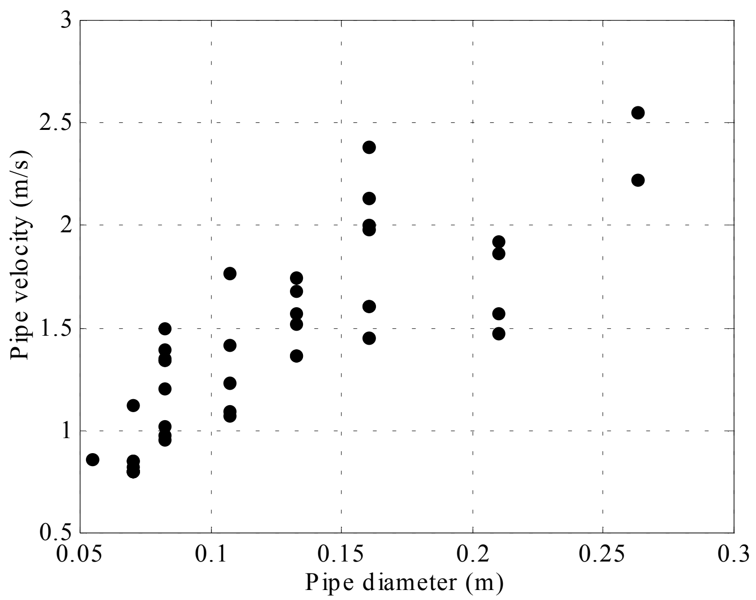


Figure 6.27: Relationship between pipe diameter and velocity of water.

Total cost of the piping network of the fuel boiler district heating system is given in Table 6.17. As it can be seen from the Table, total piping cost of the fuel boiler DHS is 374,821 US$.

Table 6.17: Total cost of the underground piping network for FBDHS.

Components	Cost (US$)
Total pipe cost	197,887
Fittings and wages	59,366
Construction	117,568
TOTAL	**374,821**

- **Heating Equipment**

Since fuel boiler system is designed for 90/70°C reference condition there is no need to increase fan coil units in the existing heating systems. For new buildings, fan coils are selected according to the reference condition as well.

B. Total Investment Cost

Total cost of each equipment in the fuel boiler district heating system is summarized in Table 6.18.

Table 6.18: Summary of the investment cost of the FBDHS.

System Components	Investment Cost	
	(US$)	(%)
Piping network	374,821	35.1
Secondary water circulation pumps	58,000	5.4
Fuel boilers	60,480	5,7
Control system of the boilers	575.000	53.8
Heating equipments (fan coils)	0	0.0
TOTAL	**1,068,301**	**100.0**

As it can be seen from the Table, total approximate cost of the fuel boiler district heating system of IZTECH Campus is 1,068,301 US$.

6.2.1.2.2 Individual Fuel Boiler Heating System

As it was mentioned before, the existing heating system in the Campus consists of individual fuel boilers. Thus, in this section IFBHS is studied for whole Campus to represent existing heating system. Heat centre construction for the new buildings is also considered.

A. Selection and Cost of the Equipment

Considered main equipment of IFBHS are

- Boilers,
- Circulation pumps,
- Heating equipment.

- **Boilers**

In Table 6.19, selected boilers and their costs are given for the new buildings. The boilers are selected one unit extra as standby. As it can be seen from Table, total boiler cost is 97,331 US$ for the new buildings [45].

Table 6.19: Selected boilers and their costs for the new buildings.

Heating Center	Building Heat Load (kW)	Number of the Boiler Unit	Unit Boiler Capacity (kW)	Total Capacity (kW)	Unit Boiler Cost (US$)	Total Boiler Cost (US$)
Library	900	4	349	1,395	2,543	10,170
Chemical Eng.	1,750	4	698	2,791	3,537	14,148
Mechatronic	252	2	291	581	2,318	4,635
Res &Dev.	162	2	174	349	2,070	4,140
Medical Centre	648	3	349	1,047	2,543	7,628
Sport Centre	780	4	291	1,163	2,318	9,270
Staff Houses	259	2	291	581	2,318	4,635
Dormitories	1,030	4	407	1,628	2,565	10,260
Build. A	252	2	291	581	2,318	4,635
Build. B	252	2	291	581	2,318	4,635
Build. C	252	2	291	581	2,318	4,635
Build. D	252	2	291	581	2,318	4,635
Build. E	252	2	291	581	2,318	4,635
Build F	252	2	291	581	2,318	4,635
Build G	252	2	291	581	2,318	4,635
TOTAL	**7,545**			**13,605**		**97,331**

- **Circulation Pumps**

Circulation pumps are selected in the same way with boilers and number of the circulation pump units is considered same as fuel boiler units for the each heat centre. Table 6.20 gives selected circulation pumps and their costs for the new buildings. Total circulation pumps cost is about 27,000 US$ [41].

Table 6.20: Selected circulation pumps and their cost for the new buildings.

Heat centre	Number of the Boiler Unit	Unit Pump Cost (US$)	Total Pump Cost (US$)
Library	4	780	3,122
Chemical Eng.	4	1,293	5,171
Mechatronic	2	549	1,098
Res &Dev.	2	427	854
Medical Centre	3	780	2,341
Sport Centre	4	549	2,195
Staff Houses	2	549	1,098
Dormitories	4	854	3,415
Build. A	2	549	1,098
Build. B	2	549	1,098
Build. C	2	549	1,098
Build. D	2	549	1,098
Build. E	2	549	1,098
Build F	2	549	1,098
Build G	2	549	1,098
TOTAL			**26,977**

- **Heating Equipment**

Same as FBDHS, there is no extra heating equipment required.

B. Total Investment Cost

Total cost of each equipment in the IFBHS is summarized in Table 6.21.

Table 6.21: Summary of the investment cost of the IFBHS.

System Components	Investment Cost	
	(US$)	(%)
Fuel boilers	97,331	30.0
Campus loop circulation pumps	26,977	8.3
Control system of the boilers	200,000	61.7
Heating equipments (fan coil)	0	0
TOTAL	**324,308**	**100.0**

Total approximate cost of the IFBHS of IZTECH Campus is 324,308 US$.

6.2.2 Operational Cost of the Heating System

In this section, operational cost of the heating system is calculated depending on the main heat engine energy consumption (heat pump or fuel boiler), pumping electricity, personnel, water, inhibitor, other chemicals and maintenance costs.

As it was calculated in Section 6.1.2, annual heat requirement of the Campus is 5,129,892 kWh for between 8.00 a.m. and 17.00 p.m. during the week. Thus, the operational cost of the system is calculated for this period.

6.2.2.1 Heat Pump District Heating System

Operational cost items of the heat pump district heating system;

- Electricity consumption of the heat pumps,
- Electricity consumption of the circulation and well pumps,
- Personnel,
- Water,
- Inhibitor,
- Other Chemicals (nitrogen and oxygen gases, nitric acid, rock salt, NaOH),
- Maintenance costs.

- **Electricity Consumption of the Heat Pumps**

Hourly electricity consumption of the heat pumps is calculated using Equation (5.31) and sum of these gives annual electricity consumption. Annual electricity consumption of the heat pump heating system for 4 heat pump units is nearly 953,740 kWh with electric motor efficiency of 0.8. The unit selling cost of electricity is 0.09 US$/kWh (March 2003) [46]. Thus, cost of this consumption is calculated as 85,837 US$ using Equation (5.31).

- **Electricity Consumption of the Circulation and Well Pumps**

Electricity consumption of the circulation pumps is calculated with Equation (5.33). In the equation total dynamic head of the pump (h_p) has been calculated from heating system pressure drop with the help of Pipelab program depending on the flowrate. Figure 6.28 displays the variation of the total dynamic head of the Campus loop water circulation pump versus Campus loop flowrate. As it can be seen from the Figure, the equation, $h_p = 0.0007.(\dot{\mathbf{m}}_{\mathbf{sec}})^2 + 0.0097.(\dot{\mathbf{m}}_{\mathbf{sec}}) + 16.865$, is obtained for total dynamic head of the Campus loop circulation pumps. The equation is used in the heat pump district heating system simulation program, which is written in Matlab software, for determining annual electricity consumption of the Campus loop circulation pumps.

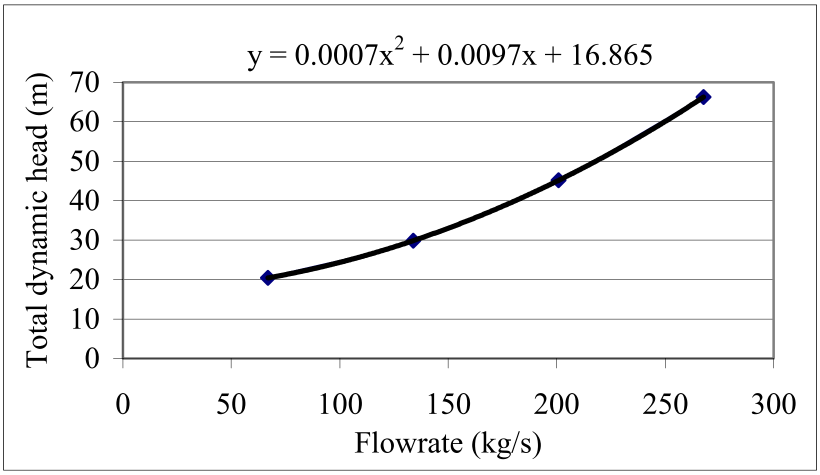


Figure 6.28: Variation of the total dynamic head of the secondary water circulation pump versus secondary water flowrate.

Annual electricity consumption of the Campus loop circulation pumps is calculated as 106,270 kWh and it costs 9,564 US$ for 0.95 motor efficiency and 0.75 pump efficiency.

Annual electricity consumption of the geothermal loop circulation pumps is determined in the same way with Campus loop circulation pumps. Variation of the total dynamic head of the geothermal loop circulation pump versus geothermal fluid flowrate is shown in Figure 6.29 and the equation obtained from the Figure is $h_p = 0.0027.(\dot{\mathbf{m}}_g)^2 + 0.0162.(\dot{\mathbf{m}}_g) + 16.895$.

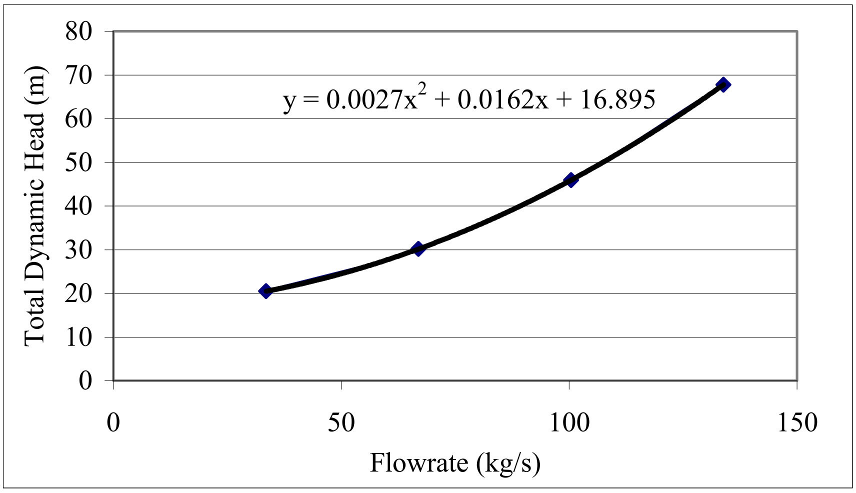


Figure 6.29: Variation of the total dynamic head of the geothermal loop circulation pump depending on the geothermal fluid flowrate.

Using the equation, annual electricity consumption of the geothermal loop circulation pumps is determined as 38,739 kWh and it costs 3,487 US$ with 0.95 motor efficiency and 0.75 pump efficiency.

Equation (5.33) can also be used for electricity consumption of the well pumps. But total dynamic head has not been known. Hence, h_p is assumed as 100 m. Using simulation program, annual electricity consumption of the well pumps is calculated as 8,915 US$. Total pumping cost of the heat pump district heating system is 21,966 US$.

- **Personnel Cost**

All buildings in the campus are heated between 8.00 a.m. and 17.00 p.m. during the week, 2 technicians and 1 operator are in charge. Salary of the each staff is 275 US$. Total annual personnel cost is 9,900 US$.

- **Water Cost**

Water in the Campus loop should evacuated during the maintenance, which is determined as twice a year. Campus loop contains 174.9 m^3 water in supply and return mains. The unit selling cost of water is 3.2 US$/$m^3$ [47] and annual water cost is calculated as 1,120 US$.

- **Inhibitor Cost**

Because of the corrosive effects and scaling potential of geothermal fluid and secondary water, inhibitor is used to protect the system. Referring the inhibitor cost of Balçova GDHS, which is 0.05 US$/kW [47], annual inhibitor cost is calculated as 573 US$ for 11,207 kW for the Campus.

- **Cost of Other Chemicals**

The cost of the other chemicals is calculated in the same way as inhibitor cost. It is calculated as 292 US$ using Balçova data of 0.026 US$/kW [47].

- **Maintenance Cost**

Maintenance cost is determined using Balçova DHS data as well, which recorded as 0.73 US$/kW [47] for 2002. Thus, the maintenance cost is about 8,155 US$ for the Campus DHS.

Total Operational Cost

Annual operational cost of the heat pump district heating system is summarized in Table 6.22 and total cost amounts as 127,843 US$.

Table 6.22: Summary of the operational cost of the HPDHS.

Operational Cost Components		Operational Cost (US$)	Operational Cost (%)
Electricity	Heat pumps	85,837	67.1
	Circulation and well pumps	21,966	17.2
Personel		9,900	7.7
Water		1,120	0.9
Inhibitor		573	0.4
Other Chemicals		292	0.2
Maintanence		8,155	6.4
TOTAL		**127,843**	**100.0**

Electricity consumption of the heat pumps itself has 67% share whereas total electricity cost constitutes 84% (circulation and well pumps) in total operational cost of the system.

6.2.2.2 Fuel Boiler Heating System

6.2.2.2.1 Fuel Boiler District Heating System

Operational cost items of the FBDHS;

- Fuel oil consumption of the boilers
- Electricity consumption of the circulation pumps
- Personnel
- Water
- Maintenance costs

- **Fuel Oil Consumption of the Boilers**

Using Equation (5.32) fuel oil consumption of the fuel boilers is calculated as 618.5 tones with specific heat capacity of the fuel oil of 11.27 kWh/kg. In the calculations boiler efficiency is assumed 0.8. The unit selling cost of fuel oil of 0.53 US$/kg (March 2003) [48]. The cost of fuel oil consumption is calculated as 327,780 US$/year.

- **Electricity Consumption of the Circulation Pumps**

Electricity consumption of the circulation pumps is calculated in the same way with heat pump district heating system. Figure 6.30 displays the variation of the total dynamic head of the Campus loop water circulation pump versus Campus loop flowrate. As it can be seen from the Figure the equation, $h_p = 0.0026.(\dot{\mathbf{m}}_{\mathbf{sec}})^2 + 0.0208.(\dot{\mathbf{m}}_{\mathbf{sec}}) + 16.835$, is obtained for total dynamic head of the Campus loop circulation pumps.

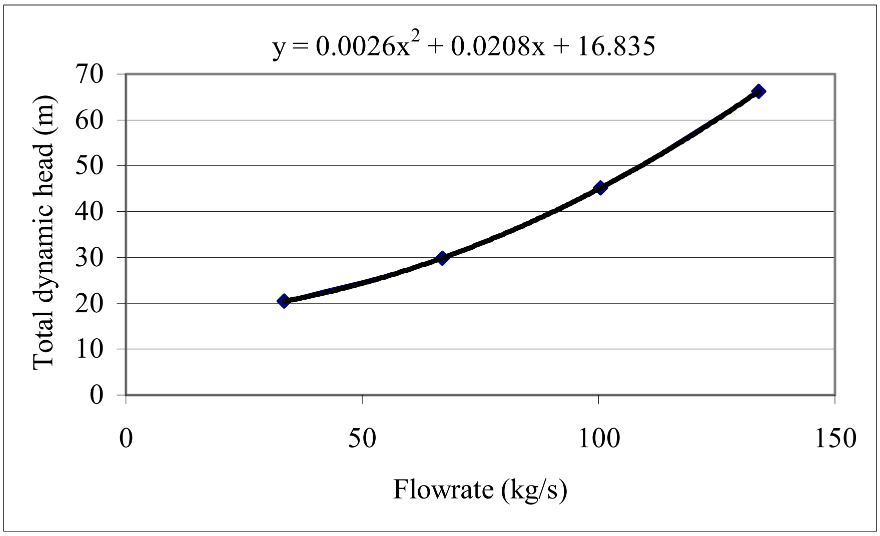


Figure 6.30: Variation of the total dynamic head of the water circulation pump depending on the water flowrate in the Campus loop.

Annual electricity consumption of the Campus loop circulation pumps is determined as 130,110 kWh and it costs 11,710 US$ for 0.95 motor efficiency and 0.75 pump efficiency by simulation program.

- **Personnel, Water and Maintenance Cost**

Personnel, water and maintenance costs are considered same with heat pump district heating system and their costs are 9,900 US$, 1,120 US$ and 8,155 US$ respectively.

Total Operational Cost

Annual operational cost of the FBDHS is given in Table 6.23. As it can be seen from the Table, approximate operational cost of the fuel boiler DHS is determined nearly as 358,664 US$.

Table 6.23: Summary of the operational cost of the FBDHS.

Operational Cost Components	Operational Cost	
	(US$)	(%)
Fuel Oil	327,780	91.4
Electricity	11,710	3.3
Personnel	9,900	2.8
Water	1,119	0.3
Maintanence	8,155	2.3
TOTAL	**358,664**	**100.0**

Fuel oil has a 91.4 % share in total operational cost. Electricity cost is not significant, constitutes only 3.3%.

6.2.2.2.2 Individual Fuel Boiler Heating System

- **Fuel Oil Consumption of the Boilers**

Using Equation (5.32) fuel oil consumption of the fuel boilers is calculated as 669 tones and it costs 354,670 US$/year. Fuel consumption increases by 8% according to fuel oil consumption of the FBDHS.

- **Electricity Consumption of the Circulation Pumps**

Electricity consumption of the circulation pumps is calculated with the help of the increase ratio, 8%, of the fuel oil consumption between individual and district fuel oil heating system. Since, to make calculation for each building is very complex and takes long time to be able to calculate circulation pump electricity consumption. Consequently, total cost of the circulation pumps electricity consumption is determined as 12,647 US$.

- **Personnel Cost**

The number of the heat centre of the new building is 15. Total number of the heat centres reaches 20 with number of 5 existing heat centre in the Campus. 10 technicians are in charge with assumption each person is responsible 2 heat centres. Salary of the each staff is 275 US$. Total annual personnel cost is 33,000 US$.

- **Water Cost**

Water cost is considered 50% of the district heating system. Thus, water cost is 560 US$ for IFBHS.

- **Maintenance Cost**

As it was mentioned previous section, total number of the boiler in FBDHS is 6. But, for individual heating system this number reaches to 50 with the existing boilers. Thus, maintenance cost is determined with the help of the ratio, 8 times, between number of the boilers for individual and district fuel boiler heating system. Consequently, the maintenance cost of the IFBHS is considered about 65,240 US$.

Total Operational Cost

Annual operational cost of the system is given in Table 6.24. As it can be seen from the Table, approximate operational cost of the IFBHS is determined nearly as 466,117 US$.

Table 6.24: Summary of the operational cost of the IFBHS.

Part of the Operational Cost	Operational Cost	
	(US$)	(%)
Fuel Oil	354,670	76.1
Electricity	12,647	2.7
Personel	33,000	7.1
Water	560	0.1
Maintanence	65,240	14.0
TOTAL	**466,117**	**100.0**

Fuel oil has a 76.1 % share in total operational cost. The second biggest portion is the maintenance cost with 14%.

6.2.3 Cost Comparison of Investment Alternatives

Total investment and operational cost of the heating system alternatives are given in Table 6.25 and the alternatives are evaluated according to internal rate of return (IRR) method.

Table 6.25: Total investment and operational cost of the heating system alternatives.

Alternatives	Heat Engine	Heating System Type	Total Investment Cost	Total Operational Cost
			(US$)	(US$)
Alternative 1	Heat Pump	District	3,040,125	127,843
Alternative 2	Boiler	District	1,068,301	358,664
Alternative 3	Boiler	Individual	324,308	466,117

The amortization cost of Alternative 1 and 2 and cash flow are given in Table 6.26.

Table 6.26: Amortization costs for Alternative 1 and 2 and cash flow during the 20-year period.

Year	Alternative 1		Alternative 2		Amortization Cost Difference (US$)	Cash Flow (US$)
	Book Value (US$)	Amortization Cost (US$)	Book Value (US$)	Amortization Cost (US$)		
0	3,040,125		1,068,301			-1,971,824
1	2,736,113	304,013	961,471	106,830	-197,182	33,639
2	2,462,501	273,611	865,324	96,147	-177,464	53,357
3	2,216,251	246,250	778,791	86,532	-159,718	71,103
4	1,994,626	221,625	700,912	77,879	-143,746	87,075
5	1,795,163	199,463	630,821	70,091	-129,371	101,450
6	1,615,647	179,516	567,739	63,082	-116,434	114,387
7	1,454,082	161,565	510,965	56,774	-104,791	126,030
8	1,308,674	145,408	459,869	51,097	-94,312	136,509
9	1,177,807	130,867	413,882	45,987	-84,881	145,940
10	1,060,026	117,781	372,494	41,388	-76,393	154,428
11	954,023	106,003	335,244	37,249	-68,753	162,068
12	858,621	95,402	301,720	33,524	-61,878	168,943
13	772,759	85,862	271,548	30,172	-55,690	175,131
14	695,483	77,276	244,393	27,155	-50,121	180,700
15	625,935	69,548	219,954	24,439	-45,109	185,712
16	563,341	62,593	197,958	21,995	-40,598	190,223
17	507,007	56,334	178,162	19,796	-36,538	194,283
18	456,306	50,701	160,346	17,816	-32,884	197,937
19	410,676	45,631	144,312	16,035	-29,596	201,225
20		410,676		144,312	-266,364	-35,543
		3,040,125		1,068,301	-1,971,824	672,772

As it can be seen from Table 6.26, sum of the cash flows during 20 years is

672,772 US$ and it indicates that, Alternative 1 (HPDHS) is more attractive than Alternative 2 (FBDHS). Using the cash flow values, IRRs, which shows the profit of the investment, is calculated as 3.02% for Alternative 1.

Alternative 1 is compared with Alternative 3 in the same way. Total cash flows and IRR are calculated as 1,333,846 US$ and 4.07%, respectively.

6.2.4 Various Heating Scenarios

Up to here, intermittent heating regime where the system is operated between 8.00 a.m. and 17.00 p.m was adapted (Scenario 1). In this section, besides Scenario 1, two more scenarios, which are given in Table 6.27, will be evaluated for operational costs. Annual heating requirements of the buildings for each scenario are given in Table 6.28.

Table 6.27: Heating scenarios for the buildings in the Campus.

Scenario No	Explanation
1	All buildings in the campus are heated between 8.00 a.m. and 17.00 p.m. during the week.
2	All buildings in the campus are heated between 8.00 a.m. and 20.00 p.m. during the week.
3	Office buildings in the campus are heated between 8.00 a.m. and 20.00 p.m. during the week.
	Sport Centre, Medical Centre and Library are heated between 8.00 a.m. and 22.00 p.m. during all week.
	Sport Centre, Medical Centre and Library are heated between 9.00 a.m. and 15.00 p.m. during weekend.
	Staff houses and dormitories are heated 24 hours for everyday

Table 6.28: Annual heating requirements of the buildings for various heating scenarios.

Scenarios No	Annual Heating Requirements (kWh)		
	Existing Buildings	New Buildings	Campus Total
Scenario 1	1,676,244	3,453,648	**5,129,892**
Scenario 2	2,253,760	4,643,533	**6,897,293**
Scenario 3	2,253,760	7,358,796	**9,612,556**

For Scenario 2 and 3, annual operational costs are determined depending on the annual heating requirements of the buildings. The ratio (Operational cost/Annual heating energy requirements) of the Scenario 1 is used to determine annual operational cost of the Scenario 2 and 3. Annual operational cost of each scenario is given in Table 6.29.

Table 6.29: Total investment and operational cost of the heating system alternatives for each scenario.

Alternative No	Heat Engine	Heating System Type	Total Investment Cost (US$)	Total Operational Cost (US$)		
				Scenario 1	Scenario 2	Scenario 3
Alternative 1	Heat Pump	District	3,040,125	127,843	171,889	239,556
Alternative 2	Fuel Boiler	District	1,068,301	358,664	482,234	672,076
Alternative 3	Fuel Boiler	Individual	324,308	466,117	626,708	873,425

The cash flow and IRR are calculated for each scenario and the results are given in Table 6.30.

Table 6.30: Cash flows and IRRs of the heating system alternatives for each scenario.

Alternative No	Scenario 1		Scenario 2		Scenario 3	
	Cash Flow (US$)	IRR (%)	Cash Flow (US$)	IRR (%)	Cash Flow (US$)	IRR (%)
Alternative 1-2	672,772	3.02	2,263,252	8.73	4,706,752	17.10
Alternative 1-3	1,333,846	4.07	3,664,746	10.05	7,245,746	19.04
Alternative 2-3	661,074	6.95	1,401,494	13.67	2,538,994	24.34

The Table indicates that Alternative 1 (HPDHS) is more attractive than Alternative 2 (FBDHS) and Alternative 3 (IFBHS).

The same analyses are conducted for fuel boiler heating systems (FBDHS and IFBHS). Sum of the cash flows during 20 years of Alternative 2 is 661,074 US$ depending on Alternative 3. Using the cash flow values, IRRs, is calculated as 6.95% for Alternative 2-3. The results indicate that Alternative 2 is more attractive than Alternative 3.

It is clearly seen that in Table 6.30 cash flows and IRRs increase with increasing operational period of the heating systems.

Chapter 7

CONCLUSIONS AND RECOMMENDATIONS

Individual HVAC (Heating, Ventilation and Air Conditioning) systems are employed at each department of IZTECH Campus since the Campus is under development. But the Campus has a geothermal resource. The main purpose of this Thesis is to investigate the possibility of a district heating system for IZTECH Campus using this geothermal source. Several exploration studies have been made about the geothermal source to investigate whether the campus can be supplied with geothermal heat. In 2002, 5 gradient wells were drilled. Of these, one well has a geothermal fluid of 33°C is obtained but the actual flowrate of the geothermal fluid has not been measured yet.

To provide information about district heating favourability of the Campus, a simple thermal load inventory is made. To determine thermal load density of the Campus, total heat load and annual heat requirement of the Campus have been determined. Unit heat load is used to determine total heat load of the Campus. Unit heat load of the Campus is determined as an average of the unit heat loads of the existing buildings. For 20°C indoor temperature, the average unit heat load per square meter and volume is calculated as 0.072 kW and 0.022 kW, respectively. The unit heat loads of new buildings are assumed as the same with unit heat load of the Campus. Thus, total heat load of the Campus has been determined as 11,207 kW.

To determine annual heat requirement of the Campus, energy estimating methods such as degree-day, degree-hour and heat load factor have been used. The results of energy estimating methods indicate that annual average outdoor temperature is 23.3°C, the number of the heating days and hours are 202 and 5,057, respectively. Annual degree-day and degree-hour for İzmir are 1,738 and 41,162, respectively. Heat load factor changes between 0.157 and 0.670 for 20°C. For heating system design purpose, the maximum heat load factor value of 0.67 is chosen to maintain the peak load.

In the Thesis, mainly two heating system types have been considered;

1. Heat pump heating system (HPHS) (using a renewable energy source,

geothermal energy),

2. Fuel boiler heating system (FBHS) (using a conventional energy source, fuel-oil).

While HPHS is considered only as district, FBHS is considered as district and individual. Each heating system is simulated using hourly outdoor temperature data. For these heating simulations, the main control parameter is the indoor temperature of the buildings. Mathematical models are derived using Matlab [16] and EES [17] programs.

HPHS has two options depending on resource temperature, heat pump only (HPO) and heat pump assisted (HPA). HPO layout is selected for the Campus heating system because it exhibits better performance than HPA at geothermal fluid temperatures below 40°C. Various heating regime alternatives have been studied for HPHS for the various condenser outlet temperature and geothermal fluid flowrate. Consequently, the heating regime with 35°C condenser inlet and 45°C condenser outlet temperature with 120 kg/s geothermal fluid flowrate considered to be the best option. For IZTECH Campus DHS, 4 separate heat pump units of the same capacity are considered because of the improved performance, reliability and operational flexibility. Each heat pump is fed by each production well and heat pumps are operated depending on outdoor temperature. For each heat pump unit, one heat exchanger is employed.

In the Campus, the existing boiler heating systems are operated manually by technicians. Technicians decide the boiler set temperature according to their experiences and each heat centre of the Campus is operated in a different way. In the Thesis, FBHS was run for various boiler set temperatures and the results show that fuel consumption of the boiler changes drastically with changing boiler set temperature. Boiler set temperatures have been recommended by Demirdokum [39], is the best alternative with least fuel consumption and best indoor temperature around 20°C. Therefore, in the Campus, energy management system should be developed for the Campus heating systems. General operational regulations should be formed to improve efficiency and achieve energy cost savings. Each heat centre should be operated according to these regulations simultaneously. The combustion products of the existing heating system should be analysed periodically. Energy consumption and energy saving measures of the Campus should be measured regularly.

Besides heating system simulations, piping network simulation is made using the software Pipelab [18], which uses the Matlab program as a basis. The piping network of the Campus has been considered with two loops as geothermal and Campus. Each loop contains supply and return main. The location of the heat centre is very important for economy of the system. Therefore, several alternatives have been studied and 3 of which are given in the Thesis. In the first alternative, heat centre is close to the Campus entrance. For the second alternative, heat centre is close the production well and heat centre is almost in the middle of the Campus in the third alternative. With the help of Pipelab software, pipe diameters and total pressure drops of the piping network are calculated for each alternative. The results indicate that, Alternative 2 has the highest pressure drop and requires the longest pipeline. Alternative 1 and 3 are close to eachother in pipe length but Alternative 3 requires shorter piping for larger diameters. Because of the lowest piping cost and pressure drop, Alternative 3 is considered to be the best option.

The pressure loss per unit length is a common design parameter. Various target pressure losses, which are 62.5, 100 and 150 Pa/m, are tested for diameter selection using Pipelab software. While piping cost decreases drastically with increasing target pressure loss, operational pumping cost is nearly constant. Therefore, the highest acceptable target pressure loss, 150 Pa/m, is selected for the district heating piping network. Main results of piping network design for supply mains of HPDHS are given in Table 7.1. As it can be seen in the Table, total length of the pipes in the supply main of Campus loop and geothermal loop are 3,520 m and 1,223 m, respectively and because of the corrosive effects of the geothermal fluid, composite (FRP) pipes are used for geothermal loop while Campus loop is installed with carbon steel pipes. Geothermal return main is the same as the supply main in length and diameter. Thus, total length of considered network of the district heating system is nearly 9,486 m.

Table 7.1: Main results of piping network design for supply mains of HPDHS.

Part of the Piping Network	Total Pipe Length (m)	Pipe Material	Piping Cost (US$)	Pressure Drop (m)
Campus Loop	3,520	Carbon Steel	248,991	24.6
Geothermal Loop	1,223	Composite	185,896	25.3

Table 7.1 indicates that while total pipe length of Campus loop is nearly 3 times of geothermal loop, the piping cost is only 1.34 times. Since composite pipes 13-35% more expensive than carbon steel pipes depending on the pipe diameter.

Pipelines are installed either aboveground or underground. Underground installation is preferred here to avoid heat losses and esthetic concerns. For installation type of piping network two options are considered, directly buried into the soil or in concrete tunnels. Piping cost is 2.3 times more expensive for concrete tunnel. Therefore, for IZTECH Campus DHS underground (buried) pipeline installation is preferred.

Economic analysis has also been done for 3 heating alternatives.

Investment costs of the alternatives are given in Table 7.2. The Table indicates that HPDHS has maximum investment cost and it is nearly 3 times of FBDHS investment cost and 10 times of IFBHS investment cost. While heat pump cost has the biggest portion with 33% of total investment cost of HPDHS, for boiler heating system control systems have the biggest portion of the total investment costs.

For operational cost, 3 heating scenarios, which are given in Table 6.27, 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, various buildings, Medical Centre, Sport Centre, Library are heated longer than office buildings and 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 and given in Table 7.2.

Table 7.2: Investment and operational costs of the heating system alternatives.

Alternative No	Total Investment Cost (US$)	Total Operational Cost (US$)		
		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
Alternative 3 (IFBHS)	324,308	466,117	626,708	873,425

The Table exhibits that HPDHS has minimum operational cost and the electricity consumption cost of the heat pumps, circulation and well pumps constitutes nearly 84% in total operational cost of the system. For boiler heating systems, fuel oil cost has the biggest portion of the total operational costs.

The alternatives are evaluated according to internal rate of return (IRR) method, which shows the profit of the investment. The results of cost comparison of the investment are summarized in Table 7.3.

Table 7.3: Summary of cost comparison of the investment of the heating system alternatives.

Scenario No	Alternative 1-2		Alternative 1-3		Alternative 2-3	
	Cash Flow (US$)	IRR (%)	Cash Flow (US$)	IRR (%)	Cash Flow (US$)	IRR (%)
Scenario 1	672,772	3.02	1,333,846	4.07	661,074	6.95
Scenario 2	2,263,252	8.73	3,664,746	10.05	1,401,494	13.67
Scenario 3	4,706,752	17.10	7,245,746	19.04	2,538,994	24.34

According to the results HPDHS is more attractive than FBHSs. For the first heating scenario, the cash flow of the HPDHS (Alternative 1) is 672,772 US$ depending on the FBDHS (Alternative 2) for 20-year period. IRR is calculated as 3.02% for Scenario 1. That means, the HPDHS has 3.02% profit at the end of the 20-year period comparing with FBDHS. The same analyses are conducted for fuel boiler heating systems (FBDHS and IFBHS). Sum of the cash flows during 20 years of FBDHS is 661,074 US$ depending on IFBHS. Using the cash flow values, IRRs, is calculated as 6.95% for Alternative 2-3 for Scenario 1. The results indicate that FBDHS is more attractive than IFBHS. The Table 7.3 indicates that cash flow and IRR increases with increasing operating period of the heating systems.

In this thesis, only heating system design is considered and economical analyses have been conducted for heating requirements. But each building is also equipped with cooling system. While considered HPDHS can be used for cooling requirements as well, for boiler heating systems, chillers should be installed to the system. Thus, the investment costs of the boiler heating systems increase. That means the HPDHS could be more attractive than FBHSs if cooling requirements of the buildings are considered.

District heating systems are designed according to 0.67 heat load factor and heating system is not enough only when the outdoor temperature is below 5°C. Thus, existing boiler heating system of the Campus can be used as peaking plant.

It is likely to reach higher temperatures at higher depths. Since the geothermometry gave a temperature of 60-100°C. In this case, the considered heating alternatives should be evaluated again and direct use heating system should be considered as an alternative.

Design and economical analysis are conducted under the assumption of a geothermal flowrate of 30 kg/s since the flowrate of the existing well is not yet known. Temperature and flowrate of the geothermal resource are the vital parameters determining the capacity of the well and size of the heating system, which will be fed from that well. For further study, it is recommended to repeat the study extensively after obtaining the actual flowrate and temperature.

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APPENDIX A

Table A 1: Energy consumption of IZTECH Campus

YEAR	MONTH	FUEL CONSUMPTION (ton)					ELECTRICITY CONSUMPTION (kwh)					WATER CONSUMPTION (m^3)				
		Rector.	Fac. of Eng.	Fac. of Science	Fac. of Arch.	Campus Total	Rector.	Fac. of Eng.	Fac. of Science	Fac. of Arch.	Campus Total	Rector.	Fac. of Eng.	Fac. of Science	Fac. of Arch.	Campus Total
2000	January	5,6	5,1	7,3	4,9	22,9	6479	9070	7775	5701	29025					
	February	2,5	2,6	4,0	3,5	12,6	13301	12806	16604	10208	52919					
	March	2,0	2,5	2,7	3,7	10,9	7920	8840	9050	6220	32030					
	April						7200	8268	8150	4820	28438	280	420	400	435	1535
	May						10240	7612	7954	4873	30679	228	250	360	320	1158
	June						17920	10288	10272	7436	45916	420	310	390	360	1480
	July						21008	15112	14232	10504	60856	490	500	440	470	1900
	August						21712	19668	14556	10320	66256	510	590	330	470	1900
	September						10000	10316	8256	6776	35348	370	590	410	310	1680
	October						7048	8096	6744	6432	28320	270	460	510	258	1498
	November	1,0	1,0	1,0	1,0	4,0	10032	10524	10284	8764	39604	370	490	540	290	1690
	December	6,5	5,3	4,0	3,9	19,7	11000	12498	10308	9164	42970	230	220	360	270	1080
	TOTAL	***17,6***	***16,4***	***19,0***	***17,0***	***70,0***	***143860***	***133098***	***124185***	***91218***	***492361***	***3168***	***3830***	***3740***	***3183***	***13921***
2001	January	5,5	6,3	5,0	6,7	23,5	13084	15338	12324	11280	52026	240	220	240	360	1060
	February	4,0	7,5	6,0	4,1	21,6	10872	19753	11352	10752	52729	210	230	380	300	1120
	March	0,0	2,1	1,0	0,0	3,1	7372	13140	8724	7660	36896	230	290	270	260	1050
	April						7608	12536	9492	7196	36832	210	490	650	440	1790
	May						8338	14698	9396	6412	38844	326	964	670	340	2300
	June						7291	11546	7003	5055	30895	754	1468	2216	459	4897
	July						12801	20191	11024	6847	50863	654	1353	954	373	3333
	August						23730	38239	17343	13873	93185	728	1676	1027	394	3825
	September						14214	21845	11489	8678	56226	977	2121	1669	809	5576
	October						10878	18806	11303	9838	50825	701	1341	963	697	3702
	November	4,0	4,8	2,5	3,0	14,3	14068	24973	13797	11865	64702	500	647	622	569	2338
	December	5,4	6,8	5,1	4,4	21,7	18756	31274	17279	14730	82038	309	585	749	499	2141
	TOTAL	***18,9***	***27,4***	***19,6***	***18,2***	***84,1***	***149011***	***242339***	***140525***	***114187***	***646061***	***5839***	***11384***	***10410***	***5499***	***33131***

Table A 1: Continue

YEAR	MONTH	FUEL CONSUMPTION (ton)					ELECTRICITY CONSUMPTION (kwh)					WATER CONSUMPTION (m^3)				
		Rector.	Fac. of Eng.	Fac. of Science	Fac. of Arch.	Campus Total	Rector.	Fac. of Eng.	Fac. of Science	Fac. of Arch.	Campus Total	Rector.	Fac. of Eng.	Fac. of Science	Fac. of Arch.	Campus Total
2002	January	7,5	16,8	6,5	6,5	37,4	19673	33198	17256	13379	83506	347	733	804	700	2583
	February	2,5	2,7	4,4	2,0	11,7	11446	25818	11555	9262	58080	239	606	365	643	1852
	March	1,8	3,2	1,5	2,8	9,3	15013	34271	14832	12381	76495	434	1063	575	578	2649
	April	0,2	1,3	0,0	0,9	2,4	14168	31705	13714	10927	70514	615	657	446	D	1717
	May						13209	27604	12086	9793	62692	430	991	765	D	2186
	June						14479	24975	13800	8330	61583	593	1245	349	536	2723
	July						36994	51395	24590	15672	128652	555	1062	D	422	2039
	August						34634	53338	23794	13160	124926	413	767	D	348	1529
	September						15857	26039	13136	8193	63224	453	980	D	466	1899
	October						13654	26189	14880	11433	66156	273	747	D	438	1458
	November						17537	32160	14516	13174	77387	324	1029	D	518	1871
	December						20757	37132	17987	14987	90863	293	1233	D	674	2200
	TOTAL	***12,0***	***24,1***	***12,4***	***12,3***	***60,8***	***227422***	***403823***	***192145***	***140692***	***964082***	***4969***	***11110***	***3304***	***5323***	***24707***
2003	January						20573	38840	19124	15842	94379	607	2725	D	421	3754
	February						18341	34824	19559	14869	87593	451	1110	D	484	2045
	March															
	TOTAL						***38914***	***73665***	***38683***	***30711***	***181973***	***1058***	***3835***	***0***	***906***	***5799***

Table A 2: Relationship between electricity consumption and monthly average outdoor temperature for 2002.

Month	Average Outdoor Temperature (°C)	Electricity Consumption (kWh)
January	7,3	83506
February	6,6	58080
March	11,2	76495
April	15,4	70514
May	19,6	62692
June	25,6	61583
July	27,5	128652
August	27,9	124926
September	23,6	63224
October	20,6	66156
November	12,6	77387
December	11,9	90863

Table A 3: Cumulative hours of outdoor temperature of the year 1993 for İzmir.

Outside Temperature	Section of the day						Total Hour	Cumulative Hour
	1	**2**	**3**	**4**	**5**	**6**		
40	0	0	0	0	0	0	0	8760
39	0	0	0	4	0	0	4	8756
38	0	0	1	4	0	0	5	8751
37	0	0	0	14	4	0	18	8733
36	0	0	3	29	4	0	36	8697
35	0	0	11	35	9	0	55	8642
34	0	0	15	23	13	0	51	8591
33	0	0	20	55	13	0	88	8503
32	0	0	34	63	30	1	128	8375
31	0	1	49	72	36	5	163	8212
30	0	2	70	89	58	14	233	7979
29	0	4	63	48	55	22	192	7787
28	6	9	77	62	74	39	267	7520
27	22	27	67	59	52	54	281	7239
26	34	29	67	56	62	43	291	6948
25	40	61	56	36	51	80	324	6624
24	55	60	37	39	41	73	305	6319
23	82	85	55	24	56	75	377	5942
22	81	67	28	37	59	55	327	5615
21	64	60	35	35	50	38	282	5333
20	71	69	47	35	51	69	342	4991
19	53	51	36	44	32	52	268	4723
18	54	57	43	53	41	60	308	4415
17	80	53	43	44	57	55	332	4083
16	73	66	46	59	44	40	328	3755
15	42	55	47	30	61	57	292	3463
14	60	41	56	62	53	64	336	3127
13	59	66	60	66	37	53	341	2786
12	65	51	48	50	52	65	331	2455
11	54	71	40	44	53	37	299	2156
10	54	53	56	57	61	45	326	1830
9	50	55	55	43	54	56	313	1517
8	49	40	41	20	54	63	267	1250
7	56	55	46	24	45	55	281	969
6	54	42	27	15	31	58	227	742
5	54	61	33	13	24	44	229	513
4	54	60	21	9	19	35	198	315
3	40	45	13	2	16	28	144	171
2	35	33	5	4	6	22	105	66
1	9	13	8	2	2	3	37	29
0	10	6	1	0	0	0	17	12
-1	0	12	0	0	0	0	12	0
-2	0	0	0	0	0	0	0	0
-3	0	0	0	0	0	0	0	0
TOTAL	1460	1460	1460	1460	1460	1460	8760	

Table A 4: Degree-day value for İzmir [22]

Year	Degree-Day Value
1976	1785.8
1977	1576.4
1978	1570.1
1979	1503.2
1980	1697.3
1981	1599.2
1982	1820.2
1983	1747.3
1984	1704.1
1985	1614.1
1986	1614.6
1987	1792.2
1988	1736.8
1989	1632.7
1990	1499.3
1991	1746.5
1992	1857.5
1993	*1737.8*
1994	1537.8
1995	1624.8
1996	1685.4
Average	1670.6

APPENDIX B

HEAT PUMP DISTRICT HEATING SYSTEM SIMULATION PROGRAM

```
clear all;
load temp1.dat;,              %GET Data file
Tout=temp1(:,4);              % outdoor temperature
le=length(Tout);              % Length of the data (hours)
time=(1:le)';
dt=360;                       %Set time step (seconds)
dth=dt/3600;                  %Set time step (hours)

n_day=ceil(le/24);            %number of days of temperature data
n_week=ceil(n_day/7);         %number of weeks of temperature data
nn=le;                        % Calculation hours
n=nn/dth;                     % number of iterations
t_time=(1:n)';

To=zeros(n,1);
To=interp1(time, Tout, [dth:dth:nn])';
s=1/dth;

deltaT=2;
cp=4.186;  %kJ/kgK

a=7;     % Turn on time (hour)
b=17;    % Turn off time (hour)
hhi=heatingtime1(time,n_day,n_week,dt,nn,a,b); % heating hours matrix

c=8;     % Time, when office hours start (hour)
d=17;    % Time, when office hours finish(hour)

hhiw=workingtime1(time,n_day,n_week,dt,nn,c,d); % office hours matrix

Ti_set=20;                    %Indoor set temperature (C)
Ti0=Ti_set;                   % Indoor design temperature (C)
To0=0;                        % Outdoor design temperature(C)
kl=560.4;                     %Building heat transfer coefficient (kW/C)
q_building=kl*(Ti_set-To0);   %Building peak heat load (kW)
nr=3/3;                       %heating equipment coefficient
ro_air=1.188;                 % Density of air at 100 kPa and 20 C
cv=0.718;                     % Specific heat of air (kJ/kg.K)
C=heatcapacity(ro_air,cv);    % heat capacity of the air (kJ/C)

heating_load=0.67;            %heat load factor
Ts0=45;%Tb_set;               %design supply water temperature(C)
Tr0=35;%72;                   % design return water temperature (C)

%Heating equipment heat capacity (kW)
q_radiator=radiator(Ts0, Tr0, Ti_set, q_building, nr);

sf=0.9; %safety factor

Mr=systemwater(q_radiator,sf);          % Water capacity of the heating equipments and pipes
m_r=q_radiator/((Ts0-Tr0)*cp)*sf;       %Max. flowrate (kg/s)
m_d=q_building*heating_load/(cp*(Ts0-Tr0));

ms0=m_d*1.0;                            % Secondary flowrate (kg/s)

%Geothermal Inputs
Tgi0=33;                      % geothermal inlet temperature(C)
```

```
mg0=120*1.0;                % total geothermal flowrate (kg/s)

n_hp_total=4;               % number of the heat pump units

qoo=ms0*cp*(Ts0-Tr0);       % Heating equipment heat at reference condition (kW)
Tmoo=(Ts0-Tr0)/(log((Ts0-Ti0)/(Tr0-Ti0)));  %Log. Mean temperature at reference condition

****************************************************************************************
%zeros matrix;
    T_hexi=zeros(n,1);      % heat exchanger inlet temperature (C)
    Q_c=zeros(n,1);         % Condenser heat (kW)
    Q_e=zeros(n,1);         % Evaporator heat (kW)
    T_e=zeros(n,1);         % Evaporator temperature (kW)
    COP=zeros(n,1);         % Heat pump performance coefficient
    Q_heat=zeros(n,1);      % Heat exchanger heat (kW)
    W=zeros(n,1);           % Heat pump capacity (kW)
    T_go=zeros(n,1);        % Geothermal outlet temperature (C)
    Q_g=zeros(n,1);         % Geothermal fluid heat (kW)
    Ti=zeros(n,1);          % Indoor temperature (C)
    Tr=zeros(n,1);          % Return water temperature ( C)
    m=zeros(n,1);           % Secondary flowrate (kg/s)
    Ts=zeros(n,1);          % Supply water temperature ( C)
    qr=zeros(n,1);          % Fan coil/radiator heat (kW)
    Tm=zeros(n,1);          % Log. mean temperature
    Tsh=zeros(n,1);         % Heat pump condenser outlet temperature (C )
    m_g=zeros(n,1);         % Geothermal fluid flowrate (kg/s)
'......................................................................

% Initial conditions

initial=min(find(hhi(:,1)==1));
m(initial,1)=ms0;
Ti(initial,1)=Ti_set;
Ts(initial,1)=Ts0;
Tr(initial,1)=T_ret(initial,m(initial,1),m_d,Ts(initial,1),Ti(initial,1),Ts0,Tr0,Ti0,nr);
qr(initial,1)=m(initial,1)*cp*(Ts(initial,1)-Tr(initial,1));
Tm(initial,1)=(Ts(initial,1)-Tr(initial,1))/(log((Ts(initial,1)-Ti(initial,1))/(Tr(initial,1)-Ti(initial,1))));

******************************************************************************************
**

%    START SIMULATION LOOP
for i=initial+1:n;
        To(1:s,1)=Tout(1,1);
        T_gin=Tgi0;

        Ti(i,1)=(qr(i-1,1)+kl*To(i-1,1)-exp(-kl*dt/C)*(qr(i-1,1)+kl*(To(i-1,1)-Ti(i-1,1))))/kl;

        %heat pump operation criteria

        if hhi(i,1)==1;

                if To(i,1)<=5; n_hp=n_hp_total; end;
                if ((To(i,1)>5)&(To(i,1)<=10)); n_hp=n_hp_total-0; end;
                if ((To(i,1)>10)&(To(i,1)<=13)); n_hp=n_hp_total-1; end;
                if ((To(i,1)>13)&(To(i,1)<18));n_hp=n_hp_total-2;end;

        m_g(i,1)=mg0*n_hp/n_hp_total;
        m(i,1)=m_d;

        else
                m(i,1)=0;
        end

        if To(i,1)>=(Ti_set-deltaT);
                m(i,1)=0;
                m_g(i,1)=0;
```

```
end

q_build(i,1)=kl*(Ti(i,1)-To(i,1));

if (m(i,1)>0)&(hhi(i,1)==1);

        Tsh(i,1)=45;
        Ts(i,1)=(Tsh(i,1)*n_hp+Tr(i-1,1)*(n_hp_total-n_hp))/n_hp_total;
        Tr(i,1) = T_ret(i, m(i,1), m_d, Ts(i,1), Ti(i,1), Ts0, Tr0, Ti0,nr);

*************************************************************************************************
%Heat Pump Calculation

        T_pinch=3;
        Tc=Tsh(i,1)+T_pinch;
        mg=m_g(i,1);
        eta_c=0.58;
        T_pinch1=2;
        T_hexo=T_gin-T_pinch1;
        eta_hex=0.95;

        T_hexi1=20;
        error_pump=1;
                while (error_pump>0.001),
                        Q_co=m(i,1)*(n_hp/n_hp_total)*cp*(Tsh(i,1)-Tr(i,1));
                        T_eva=T_hexi1-T_pinch1;
                        COP_s=eta_c*(Tc+273)/(Tc-T_eva);
                        Q_eva=Q_co*(COP_s-1)/(COP_s);
                        W_s=Q_co-Q_eva;
                        Q_hex=Q_eva/eta_hex;
                        Qg=Q_hex/eta_hex;
                        T_gout=T_gin-Qg/(mg*cp);
                        T_hexi2=T_gout-T_pinch;
                        error_pump=abs(T_hexi2-T_hexi1);
                        T_hexi1=T_hexi2;
                end

        T_hexi(i,1)=T_hexi2;
        Q_c(i,1)=Q_co;
        Q_e(i,1)=Q_eva;
        T_e(i,1)=T_hexi(i,1)-T_pinch1;
        COP(i,1)=COP_s;
        Q_heat(i,1)=Q_hex;
        W(i,1)=W_s;
        m_g(i,1)=mg;
        T_go(i,1)=T_gout;
        Q_g(i,1)=Qg;
        Tm(i,1)=(Ts(i,1)-Tr(i,1))/(log((Ts(i,1)-Ti(i,1))/(Tr(i,1)-Ti(i,1))));
         qr(i,1)=m(i,1)*cp*(Ts(i,1)-Tr(i,1));


else

        Ts(i,1)=Ti(i-1,1)+(Ts(i-1,1)-Ti(i-1,1))*exp(-qr(i-1,1)/(Tm(i-1,1)*Mr*cp)*dt);
        Tr(i,1)=Ti(i-1,1)+(Tr(i-1,1)-Ti(i-1,1))*exp(-qr(i-1,1)/(Tm(i-1,1)*Mr*cp)*dt);

        if Tr(i,1)<=Ti(i,1); Tr(i,1)=Ti(i,1)+0.1; end
        if Ts(i,1)<=Ti(i,1); Ts(i,1)=Ti(i,1)+0.11; end
        if Ts(i,1)<=Tr(i,1); Ts(i,1)=Tr(i,1)+0.01; end

        Tm(i,1)=(Ts(i,1)-Tr(i,1))/(log((Ts(i,1)-Ti(i,1))/(Tr(i,1)-Ti(i,1))));
         qr(i,1)=qoo*((Tm(i,1)/Tmoo)^(4/3));
        Q_c(i,1)=0;
        T_e(i,1)=0;
        COP(i,1)=0;
        W(i,1)=0;
        COP(i,1)=0;
```

```
                    Q_g(i,1)=0;
                    Q_heat(i,1)=0;

            end  % end of  if (m(i,1)>0)&(hhi(i,1)==1);

        g=9.81;
        H_geo=100;
        ef_pump=0.75;
        ef_motor=0.95;

        %DP=150m
        H_sec(i,1) =(0.0007*(m(i,1)^2) + 0.0097*m(i,1) +16.865); % m
        H_pri(i,1) =(0.0027*(m_g(i,1)^2) + 0.0162*m_g(i,1) +16.895); % m

        P_cir_sec(i,1)=(m(i,1))*g*H_sec(i,1)/1000/(ef_motor*ef_pump); %kW
        P_cir_pri(i,1)=(m_g(i,1)*g*H_pri(i,1)/(1000*ef_pump*ef_motor));
        P_cir_geo(i,1)=(m_g(i,1)*g*H_geo/(1000*ef_pump*ef_motor));


end  % end of  for i=initial+1:n;
*****************************************************************************************

% COST of the Consumption

   % Heat Pump Electricty Consumption

            eff_heatpump=0.8; %heat pump compressor efficiency
            elect_price=0.09; % US$/kwh
            W_annual=(sum(W))*dt/3600;
            W_elc_heatpump=W_annual/ eff_heatpump; %Electricity consumption in kWh
            Cost_heatpumpelec=W_elc_heatpump*elect_price;

*********************************************************************************************
** % Circulation pump cost

            P_cir_sec_annual= sum(P_cir_sec)*dt/3600;
            P_cir_pri_annual= sum(P_cir_pri)*dt/3600;
            P_cir_geo_annual= sum(P_cir_geo)*dt/3600;
            Cost_cir_sec=P_cir_sec_annual*elect_price;
            Cost_cir_pri=P_cir_pri_annual*elect_price;
            Cost_cir_geo=P_cir_geo_annual*elect_price;

            Cost_cir=Cost_cir_sec+Cost_cir_pri+Cost_cir_geo;
            Cost_total= Cost_heatpumpelec+Cost_cir;


            % Interpolations

 To=interp1(time, Tout, [dth:dth:nn])';
 T_indoor=interp1(t_time, Ti, [1/dth:1/dth:n])';
 T_s=interp1(t_time, Ts, [1/dth:1/dth:n])';
 T_r=interp1(t_time, Tr, [1/dth:1/dth:n])';
 Q_con=interp1(t_time, Q_c, [1/dth:1/dth:n])';
 Q_ev=interp1(t_time, Q_e, [1/dth:1/dth:n])';
 W_com=interp1(t_time, W, [1/dth:1/dth:n])';
 Q_heat_ex=interp1(t_time, Q_heat, [1/dth:1/dth:n])';
 T_gout=interp1(t_time, T_go, [1/dth:1/dth:n])';
 COP_h=interp1(t_time, COP, [1/dth:1/dth:n])';
 Q_r=interp1(t_time, qr, [1/dth:1/dth:n])';
 Q_loss=interp1(t_time, q_build, [1/dth:1/dth:n])';
 m_sec=interp1(t_time, m, [1/dth:1/dth:n])';
 m_geo=interp1(t_time, m_g, [1/dth:1/dth:n])';
 hhi_wor=interp1(t_time, hhiw, [1/dth:1/dth:n])';

            Ti_w=T_indoor.*hhi_wor;
            To_w=Tout(1:nn,1).*hhi_wor;
```

```
        i=find(Ti_w(:,1)>0);
        Ti_working=Ti_w(i,1);
        To_working=To_w(i,1);
```

```
          % drawing of graphs

 figure(1)
 subplot(2,1,1)
 plot(T_indoor)
 axis([-2 le 0 35])
 title('Indoor Temperature'),grid
 xlabel('Time [h]'),ylabel('Temperature [C]')

 subplot(2,1,2)
 plot(Tout)
 axis([-2 le 0 35])
 title('Outdoor Temperature'),grid
 xlabel('Time [h]'),ylabel('Temperature [C]')


figure(2)

 [dummy,ii]=sort(-Q_con(1:nn));
 [dummy,iii]=sort(-Q_ev(1:nn));
 [dummy,iv]=sort(-W_com(1:nn));

 subplot(2,1,1)
 plot([Q_con(ii) Q_ev(iii) W_com(iv)]);
 %axis([0 1250 0 8000])
 title('Heat Pump Characteristics'),grid
 xlabel('Time [h]'),ylabel('Q_c Q_e W [kW]')

 [dummy,v]=sort(-Q_heat_ex(1:nn));
 subplot(2,1,2)
 plot([Q_heat_ex(v)]);
% axis([0 1250 0 8000])
 title('Heat Exchanger Characteristics'),grid
 xlabel('Time [h]'),ylabel('Q_heat [kW]')

 figure(3);

 subplot(2,1,1)
 plot(T_s);
  axis([0 le 0 80])
 title('Supply  Temperature'),grid
 xlabel('Time [h]'),ylabel('Temperature [C]')

 subplot(2,1,2)
 plot(T_r);
  axis([0 le 0 80])
 title('Return  Temperature'),grid
 xlabel('Time [h]'),ylabel('Temperature [C]')

 figure(4)

 subplot(2,1,1)
 plot(Ti_working)
 t_working=length(Ti_working);

 axis([0 t_working 0 35])
 title('Working Indoor Temperature'),grid
 xlabel('Time [h]'),ylabel('Temperature [C]')

 subplot(2,1,2)
 plot(To_working)
```

```
    axis([0 t_working 0 35])
  title('Working Outdoor Temperature'),grid
  xlabel('Time [h]'),ylabel('Temperature [C]')

  figure(5)

  plot([ T_gout])
  title('Geothermal Outdoor Temperature'),grid
  xlabel('Time [h]'),ylabel('Temperature [C]')

   figure(6)

  plot([ COP_h])
  title('COP'),grid
  xlabel('Time [h]'),ylabel('COP [-]')

figure(7)

  plot([ W_com])
  title('Heat Pump Capacity'),grid
  xlabel('Time [h]'),ylabel('W [kW]')


  figure(8)

    subplot(2,1,1)
   [dummy,vii]=sort(-Q_r(1:nn));
  plot([ Q_r(vii)])
  axis([0 le 0 8000])
  title('Heating equipment Heat Capacity'),grid
  xlabel('Time [h]'),ylabel('Qheating equipment [kW]')

    subplot(2,1,2)
    [dummy,viii]=sort(-Q_loss(1:nn));
  plot([ Q_loss(vii)])
  axis([0 le 0 8000])
  title('Building Heat Loss'),grid
  xlabel('Time [h]'),ylabel('Qbuilding [kW]')


    figure(9)

  [dummy,ix]=sort(-m_sec(1:nn));
  plot([ m_sec(ix)])
  title('Secondary Flowrate'),grid
  xlabel('Time [h]'),ylabel('m sec [kg/s]')


    figure(10)

   [dummy,x]=sort(-m_geo(1:nn));
  plot([ m_geo(x)])
  axis([0 1500 0 125])
  title('Secondary Flowrate'),grid
  xlabel('Time [h]'),ylabel('Geothermal Flowrate [kg/s]')
```

APPENDIX C

BOILER HEATING SYSTEM SIMULATION PROGRAM

```
clear all;
load temp1.dat;,            %GET Data file
Tout=temp1(:,4);            % outdoor temperature
le=length(Tout);            % Length of the data (hours)
time=(1:le)';
dt=360;                     %Set time step (seconds)
dth=dt/3600;                %Set time step (hours)

n_day=ceil(le/24);                  %number of days of temperature data
n_week=ceil(n_day/7);       %number of weeks of temperature data
nn=le;                      % Calculation hours
n=nn/dth;                   % number of iterations
t_time=(1:n)';

To=zeros(n,1);
To=interp1(time, Tout, [dth:dth:nn])';
s=1/dth;

deltaT=2;
cp=4.186;  %kJ/kgK

a=7;     % Turn on time (hour)
b=17;    % Turn off time (hour)
hhi=heatingtime1(time,n_day,n_week,dt,nn,a,b); % heating hours matrix

c=8;     % Time, when office hours start (hour)
d=17;    % Time, when office hours finish(hour)

hhiw=workingtime1(time,n_day,n_week,dt,nn,c,d); % office hours matrix

Ti_set=20;                  %Indoor set temperature (C)
Ti0=Ti_set;                 % Indoor design temperature (C)
To0=0;                      % Outdoor design temperature(C)
kl=560.4;                   %Building heat transfer coefficient (kW/C)
q_building=kl*(Ti_set-To0);  %Building peak heat load (kW)
nr=3/3;                     %heating equipment coefficient
ro_air=1.188;               % Density of air at 100 kPa and 20 C
cv=0.718;                           % Specific heat of air (kJ/kg.K)
C=heatcapacity(ro_air,cv);          % heat capacity of the air (kJ/C)

% BOILER DATA

n_boiler_total=5;                   %number of the boiler
Mb=5*2310;                          %boiler mass of water (kg)
sf=0.9; %safety factor
qmax=(5*2000000)/860*sf; %boiler capacity (kW)

  %Boiler thermostat set temperature

        hyst=4;             %Boiler thermostat dT
        dies=0;             %counter for boiler operationTs0=90;%Tb_set;

Ts0=90;%Tb_set;             %design supply water temperature(C)
Tr0=70;%72;                         % design return water temperature (C)
```

```
%Heating equipment heat capacity (kW)
q_radiator=radiator(Ts0, Tr0, Ti_set, q_building, nr);


Mr=systemwater(q_radiator,sf);      % Water capacity of the heating equipments and pipes
m_r=q_radiator/((Ts0-Tr0)*cp)*sf;      %Max. flowrate (kg/s)
m_d=q_building*heating_load/(cp*(Ts0-Tr0));
ms0=m_d*1.0;                        % Secondary flowrate (kg/s)

*************************************************************************************

%ZEROING OF MATRICES

        Tb=zeros(n,1);              %Boiler outlet temperature
        Ts=zeros(n,1);              %Heating equipment water supply temperature
        Ti=zeros(n,1);              %Indoor temperature
        Tm=zeros(n,1);              %Log. mean temp. difference in heating equipment
        Tr=zeros(n,1);              %Heating equipment outlet temperature
        m=zeros(n,1);               %Water mass flowrate
        qb=zeros(n,1);              %Boiler heat input
        qr=zeros(n,1);              %Heating equipment heat input
        Tb_set=zeros(n,1);          % Boiler set temperature
        q_build=zeros(n,1);         % Building heat load
%*********************************************************************************
***

%INITIAL CONDITIONS

initial=min(find(hhi(:,1)==1));
qb(initial,1)=qmax;                 %INITIAL BOILER STATE
m(initial,1)=mo;            %INITIAL WATER FLOW (mo)
Tb_set_initial=80;
Tb(initial,1)=Tb_set_initial;   %Boiler initial temperature
Ti(initial,1)=Ti_set;           %Initial indoor temperature
Ts(initial,1)=Tb(initial,1); %Initial water rsupply temperature
Tr(initial,1)=T_ret(initial,m(initial,1),mo,Ts(initial,1),Ti(initial,1),Ts0,Tr0,Ti0,nr);  %Initial water return
temperature

t(initial,1)=dt;
qr(initial,1)=m(initial,1)*cp*(Ts(initial,1)-Tr(initial,1));
Tm(initial,1)=(Ts(initial,1)-Tr(initial,1))/(log((Ts(initial,1)-Ti(initial,1))/(Tr(initial,1)-Ti(initial,1))));
qoo=qr(initial,1); Tmoo=Tm(initial,1);


%START SIMULATION LOOP

for i=initial+1:n;

        Ti(i,1)=(qr(i-1,1)+kl*To(i-1,1)-exp(-kl*dt/C)*(qr(i-1,1)+kl*(To(i-1,1)-Ti(i-1,1))))/kl;

                        if hhi(i,1)==1
                        m(i,1)=mo;
                else
                        m(i,1)=0;
                end

  %BUILDING CONTROLLER***************************

        if (hhi(i,1)==1);
```

```
 if (hhi(i-1,1) == 0) & (To(i, 1) <= (Ti_set-deltaT));
                         qb(i,1)=qmax;
              m(i,1)=mo;
            end

            if (To(i,1)<=(Ti_set-deltaT));
                        qb(i,1)=qmax;
            else
                        qb(i,1)=0;
            end

    else
            qb(i,1)=0;
    end     %end of (hhi(i,1)==1);


        if (m(i-1,1)>0);

            if (To(i,1)>-3)&(To(i,1)<=0);Tb_set(i,1)=90;end;
            if (To(i,1)>0)&(To(i,1)<=3);Tb_set(i,1)=81.6;end;
            if (To(i,1)>3)&(To(i,1)<=6);Tb_set(i,1)=72.9;end;
             if (To(i,1)>6)&(To(i,1)<=9);Tb_set(i,1)=63.8;end;
             if (To(i,1)>9)&(To(i,1)<=12);Tb_set(i,1)=54.2;end;
            if (To(i,1)>12)&(To(i,1)<=15);Tb_set(i,1)=43.7;end;
             if (To(i,1)>15)&(To(i,1)<18);Tb_set(i,1)=31.7;end;


         % Boiler model*********************************************

            Tb(i,1)=(qb(i-1,1)+m(i-1,1)*cp*Tr(i-1,1)-(exp(-m(i-1,1)*dt/Mb)*(qb(i-1,1)+m(i-
1,1)*cp*(Tr(i-1,1)-Tb(i-1,1)))))/(m(i-1,1)*cp);
            Ts(i,1)=Tb(i,1);    %only valid for m>0

        %End of Boiler model****************************************

        %CALCULATING RETURN WATER TEMPERATURE**********************

            Tr(i,1)=T_ret(i,m(i-1,1),mo,Ts(i,1),Ti(i,1),Ts0,Tr0,Ti0,nr);
                        if Tr(i,1)<=Ti(i,1); Tr(i,1)=Ti(i,1)+0.1; end
                        if Ts(i,1)<=Ti(i,1); Ts(i,1)=Ti(i,1)+0.11; end
                        if Ts(i,1)<=Tr(i,1); Ts(i,1)=Tr(i,1)+0.01; end

            Tm(i,1)=(Ts(i,1)-Tr(i,1))/(log((Ts(i,1)-Ti(i,1))/(Tr(i,1)-Ti(i,1))));
            qr(i,1)=m(i-1,1)*cp*(Ts(i,1)-Tr(i,1));


             if (hhi(i,1)==1)&(To(i,1)<(Ti_set-deltaT));

                        if Tb(i,1)>=(Tb_set(i,1)+hyst);qb(i,1)=0;end

                                    if Tb(i,1)<=(Tb_set(i,1)-hyst);qb(i,1)=qmax;end
             else
                        qb(i,1)=0;

            end

        else

            Tb_set(i,1)=0;
```

```
                    Ts(i,1)=Ti(i-1,1)+(Ts(i-1,1)-Ti(i-1,1))*exp(-qr(i-1,1)/(Tm(i-1,1)*Mr*cp)*dt);
                    Tr(i,1)=Ti(i-1,1)+(Tr(i-1,1)-Ti(i-1,1))*exp(-qr(i-1,1)/(Tm(i-1,1)*Mr*cp)*dt);
                    Tb(i,1)=Tr(i,1);

                                    if Tr(i,1)<=Ti(i,1); Tr(i,1)=Ti(i,1)+0.1; end
                            if Ts(i,1)<=Ti(i,1); Ts(i,1)=Ti(i,1)+0.11; end
                                    if Ts(i,1)<=Tr(i,1); Ts(i,1)=Tr(i,1)+0.01; end

                    Tm(i,1)=(Ts(i,1)-Tr(i,1))/(log((Ts(i,1)-Ti(i,1))/(Tr(i,1)-Ti(i,1))));
                    qr(i,1)=qoo*((Tm(i,1)/Tmoo)^(nr));

            end  %end of (m(i-1,1)>0);


            if qb(i,1)>0
                    dies=dies+1;
            end

        q_build(i,1)=kl*(Ti(i,1)-To(i,1));

g=9.81;
H_geo=100;
ef_pump=0.75;
ef_motor=0.95;


  %DP=150m
H_sec(i,1) =(0.0026*(m(i,1)^2) + 0.0208*m(i,1) +16.835); % m
P_cir_sec(i,1)=(m(i,1))*g*H_sec(i,1)/1000/(ef_motor*ef_pump); %kw



   end %end of for i=initial+1:n;

% COST of the Consumption

        % Boiler Fuel Cost

                    fuel_price=0.53; % US$/kg
                    eff_boiler=0.8;
                    Hu=9700;
                    m_fuel=qmax*dies*dt/((3600*Hu/860)*eff_boiler);
        Cost_fuel=fuel_price*m_fuel;

                    elect_price=0.09;
        P_cir_sec_annual= sum(P_cir_sec)*dt/3600;
        Cost_cir_sec=P_cir_sec_annual*elect_price;

          Cost_total= Cost_fuel+Cost_cir_sec;




*****************************************************************************************
*****

           % Interpolations
```

```
T_indoor=interp1(t_time, Ti, [1/dth:1/dth:n])';
T_s=interp1(t_time, Ts, [1/dth:1/dth:n])';
T_r=interp1(t_time, Tr, [1/dth:1/dth:n])';
Q_r=interp1(t_time, qr, [1/dth:1/dth:n])';
Q_loss=interp1(t_time, q_build, [1/dth:1/dth:n])';
m_sec=interp1(t_time, m, [1/dth:1/dth:n])';
hhi_wor=interp1(t_time, hhiw, [1/dth:1/dth:n])';
        Ti_w=T_indoor.*hhi_wor;
        To_w=Tout(1:nn,1).*hhi_wor;
        i=find(Ti_w(:,1)>0);
         Ti_working=Ti_w(i,1);
        To_working=To_w(i,1);
```

```
        % drawing of graphs

    figure(1);

        subplot(2,1,1)
        plot(T_indoor)
        axis([-2 le 0 35])
        title('Indoor Temperature'),grid
        xlabel('Time [h]'),ylabel('Temperature [C]')

        subplot(2,1,2)
        plot(Tout)
        axis([-2 le 0 35])
        title('Outdoor Temperature'),grid
        xlabel('Time [h]'),ylabel('Temperature [C]')


    figure(2);

        subplot(2,1,1)
        plot(T_s);
        axis([0 le 0 100])
         title('Supply  Temperature'),grid
        xlabel('Time [h]'),ylabel('Temperature [C]')

        subplot(2,1,2)
        plot(T_r);
        axis([0 le 0 100])
        title('Return  Temperature'),grid
        xlabel('Time [h]'),ylabel('Temperature [C]')


    figure(3);

        subplot(2,1,1)
         plot(Ti_working)
         t_working=length(Ti_working);
         axis([0 t_working 0 35])
        title('Working Indoor Temperature'),grid
        xlabel('Time [h]'),ylabel('Temperature [C]')

        subplot(2,1,2)
        plot(To_working)
        axis([0 t_working 0 35])
        title('Working Outdoor Temperature'),grid
```

```
    xlabel('Time [h]'),ylabel('Temperature [C]')


figure(4);

    subplot(2,1,1)
    [dummy,vii]=sort(-Q_r(1:nn));
    plot([ Q_r(vii)])
    axis([0 le 0 10000])
     title('Heating equipment Heat Capacity'),grid
    xlabel('Time [h]'),ylabel('Qheating equipment [kW]')

    subplot(2,1,2)
    [dummy,viii]=sort(-Q_loss(1:nn));
    plot([ Q_loss(vii)])
    axis([0 le 0 10000])
    title('Building Heat Loss'),grid
    xlabel('Time [h]'),ylabel('Qbuilding [kW]')
```