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DISTRICT HEATING SYSTEM MODELLING FOR IZTECH CAMPUS, TURKEY

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ABSTRACT

Izmir Institute of Technology (IZTECH), founded in 1992, is the third university of Izmir, the third largest city in Turkey. IZTECH campus has individual fuel boiler heating systems for each faculty building since its foundation and the campus is still under development. In this project three different district heating systems were studied and compared, using two different heating strategies, intermittent and semi-intermittent heating. The performance of these alternatives was studied, regarding efficiency and estimated indoor temperature, and recommendations are made.

1. INTRODUCTION

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 (Harrison et al., 1990). IZTECH Campus has a geothermal resource and its surface temperature is nearly 35°C. Studies indicate that the geothermal water temperature will be low. For this reason, heat pump heating systems were considered in this project. Also a fuel boiler heating system was designed to replace the existing heating system and for comparison with new heating systems. The campus is mostly used during working hours and the existing heating system is run only during this period. So two heating strategies have been made for these planned heating systems, intermittent and semi-intermittent heating. For intermittent heating simulation, the heating period is considered to be from 09-17 hours. In the simulation the system was turned on one hour earlier and turned off at the end of it. As for the semi-intermittent heating simulation, during working hours the system was run in the same way. The difference between the intermittent and semi-intermittent heating simulations is that heating is continued after working hours with only geothermal energy in the semi-intermittent heating. For these heating simulations, the main control parameter is the indoor temperature of the buildings. Mathematical models were derived with programs using Matlab (The MathWorks, 2002) and EES (F-Chart Software, 2002) written and run using hourly weather data. In all, 50 heating alternatives with different parameters were studied to determine the best heating system. Besides heating system simulations, a network simulation was made using the software Pipelab, which uses the Matlab program as a basis.

2. ABOUT IZTECH CAMPUS

Izmir Institute of Technology (IZTECH) was founded in 1992 as the third State University in Izmir, W-Turkey in order to use, develop and produce advanced technologies. The campus is located in Urla, about 40 km west of downtown Izmir with a highway connection. Facing the blue waters of the Aegean Seaat the coast , it has a total area of 3,500 ha (Figure 1).

FIGURE 1: Location of IZTECH Campus in Turkey

IZTECH University is comprised of three Faculties, Engineering, Architecture, and Science, which carry out advanced level research, education/training, production, publishing and consultancy. At the Institute of Engineering and Science, there are currently 18 M.Sc. and 3 Ph.D. programmes. IZTECH started offering 4 year undergraduate programmes in the academic year 1998-1999.

Construction of campus buildings started in November 1994, (Iztech, 2002). The campus includes now 13 buildings with 41,830 m^2 floor area. In the fall of 2001, the number of students was 854 and the number of staff 538. At present, 8 buildings and 40 staff houses are under construction. The total number of campus buildings is planned to become approximately 110.

3. WEATHER ANALYSIS

The main input signal for a district heating simulation model is the weather. When a district heating system is designed, the characteristic climatic conditions are the main design criteria, along with consumer behaviour. 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 outdoor temperature. The primary interest here is, therefore, in the dynamics of the outdoor temperature, but other weather factors are taken into account as well for the sake of completeness (Valdimarsson, 1993).

As can be expected from its location, Izmir has a Mediterranean climate. The following analysis is based on weather data for the City of Izmir for the years 1976-1996, as obtained from the Meteorological Office of Turkey. Observations were made at one-hour intervals. They were used to determine the probability distribution properties of various weather indicators for each day of the year, assuming Gaussian distribution. Gathering these daily values, one can establish an average year for each of these indicators based on 20 years of weather data. The year 1993 is accepted as a typical year for the city of Izmir, with –1°C the coldest temperature according to 1993's weather data. About 99.9% of the hours were above 0°C. The minimum outdoor design temperature is 0°C for Izmir. Figure 2 shows the outdoor air temperature duration curve. Maximum, minimum and average outdoor air temperatures of this typical year are shown in Figure 3.

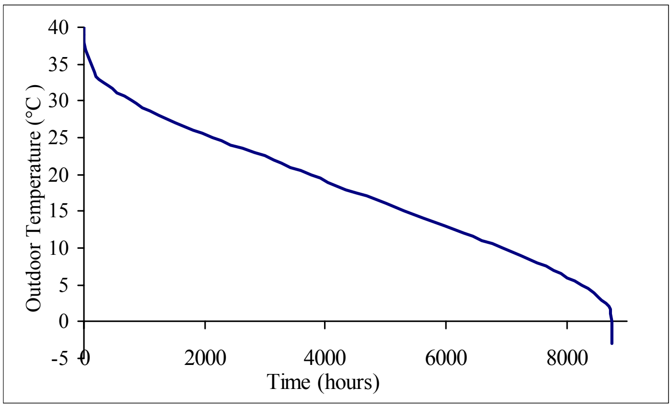

FIGURE 2: Outdoor air temperature duration curve

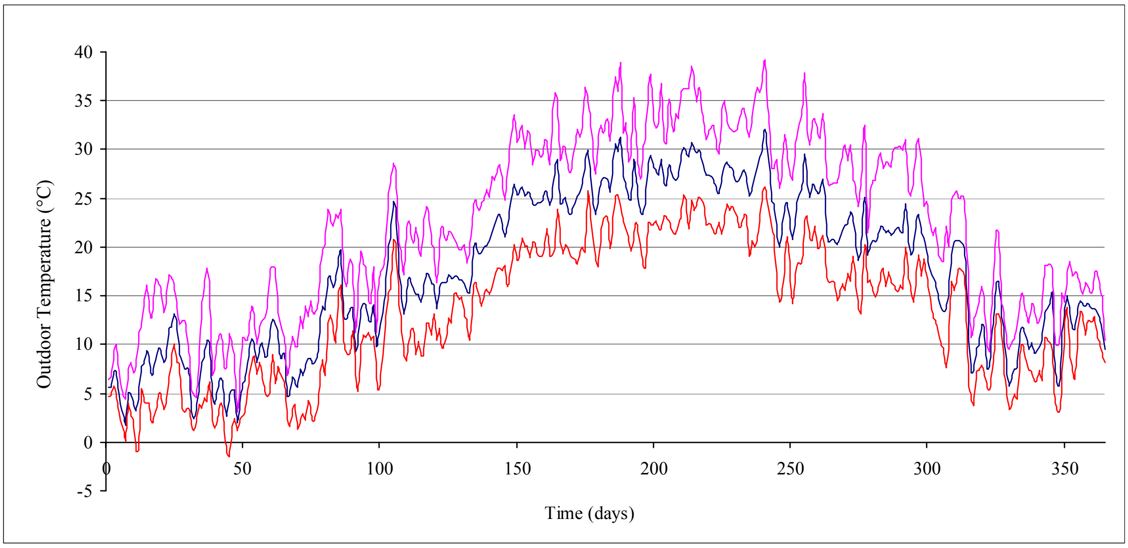

FIGURE 3: Daily maximum, minimum and average outdoor air temperatures of the year 1993

4. GEOTHERMAL DISTRICT HEATING SYSTEMS

Geothermal district heating 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.

- *Heat production*, which includes the geothermal production and discharge fields, a conventionally fuelled peaking station, and a wellhead heat exchanger.
- *Transmission/distribution system*, which delivers the geothermal fluid or geothermally heated water to the consumers.
- The third part includes *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 (Lund, 2001).

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 load on the environment (Valdimarsson, 1993).

The methods by which heat is extracted from geothermal fluid depend strongly on the temperature of the fluid and 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 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.

Two basic methods of heat extraction are used in heating applications.

Direct heat exchange. 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. The amount of heat which can be extracted across the heat exchangers is

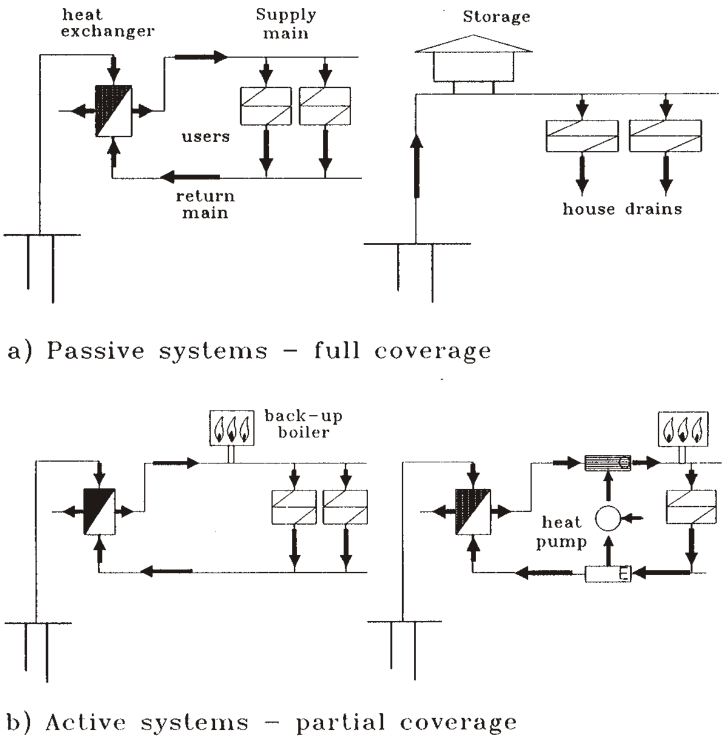

FIGURE 4: Surface systems - basic arrangements (Harrison et al., 1990)

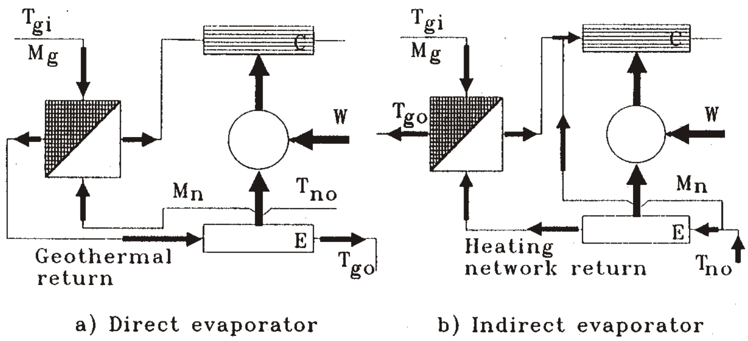

FIGURE 5: Heat pump assisted (HPA) heat transfer schematic layouts (Harrison et al., 1990)

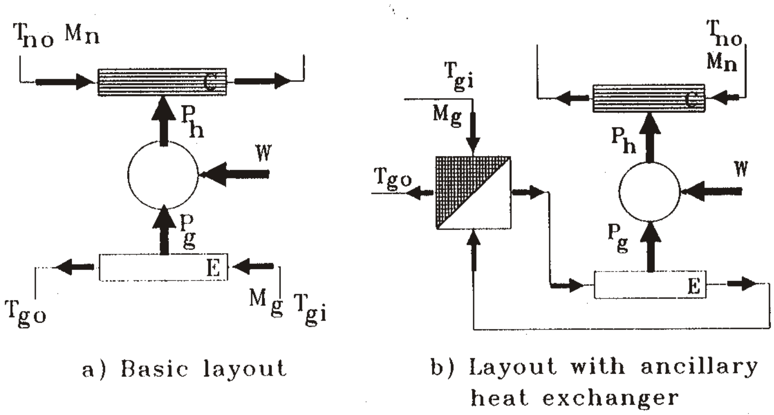

FIGURE 6: Heat pump only (HPO) heat transfer schematic layouts (Harrison et al., 1990)

limited by the temperatures of the fluids returned from the heating application and also by the smallest flow. Usually compatibility of geothermal fluid temperature with the supply temperature of the heating application is less important than obtaining low return temperatures. This is because any shortfalls can be made up for by supplementary heating (see Figure 4).

Heat pumps. The use of heat pumps is often considered when the fluid temperature is too low for necessary heat transfer to occur by direct heat exchange. Or, alternatively, they can be used to reduce return temperatures and improve heat recovery. Whichever way, the economic viability is usually marginal and careful optimisation is required (see Figure 4).

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

- The heat pump assists the primary heat exchanger, supplying additional heat from the geothermal fluid. This is called the heat pump assisted (HPA) approach (see Figure 5).
- 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 (see Figure 6).

5. DISTRICT HEATING SYSTEM MODELLING

The treatment on the system modelling in this chapter is mostly based on Valdimarsson (1993 and 1995). Models of district heating systems can be classified by

Type:	Microscopic or macroscopic
Method:	Dynamic or steady-state
Approach:	Physical or black box
Usage:	Design or operation

The concepts "microscopic" and "macroscopic" refer to whether 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 relationships 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.

5.1 Macroscopic models

The district heating system is spread out over the city to be heated. In a macroscopic model, the whole system is lumped into one model block, relating the output signals to relevant inputs. In each macroscopic model, the entire distribution system and all of the consumers are considered as seen from the district heating water supply station. In each model, use is made of either black box methods to relate the input signal to the output signal, or physical knowledge of the process involved. Parameter estimation techniques are used to obtain estimates of the unknown values of the various parameters. The macroscopic models treated in the present work are physical models, with their parameters estimated by statistical methods.

5.1.1 Physical models

The basis for physical district heating models is thermodynamic knowledge of the heat balance and thermal behaviour of buildings. The logical choice of influence factors is to take weather as the input signal, and system water supply temperature as a control signal. The model output signals are then the water mass flow to the district heating system, and the return water temperature.

It is logical for a physical approach to macroscopic modelling to base the model on a building model. The whole system is then lumped into a single "equivalent consumer". The system is then treated in the same manner, as a single building would be. The indoor temperature plays a central role. The building internal energy (stored in the building itself) is directly proportional to this temperature. The indoor temperature could be treated as an output signal, but is more akin to an internal state variable, as there is no way of measuring it directly. As a refinement, the model of one equivalent building can be serially coupled to a pipeline-cooling model, to account for the heat loss in the distribution system.

Basically, a building model is composed of four components, a building energy storage component, a heat loss component, a radiator 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 indoor temperature. The radiator element describes how heat is transferred from the district heating water to the building as a function of water mass flow, building water supply

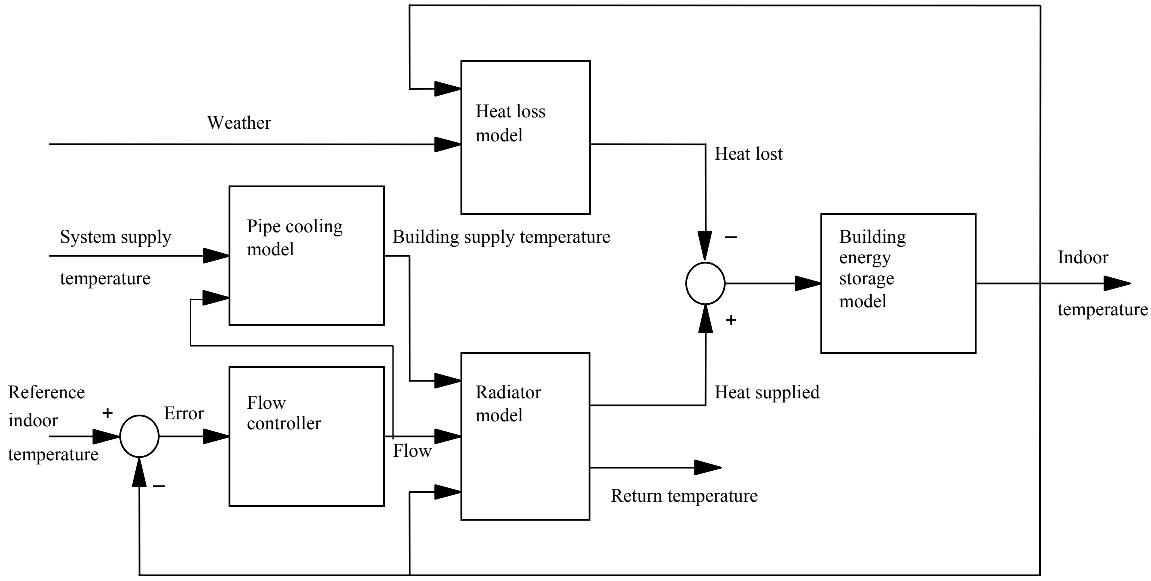

FIGURE 7: Block diagram of a lumped district heating model (Valdimarsson 1993)

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 radiator control system, and cannot be determined theoretically. The radiator control systems have known characteristics, but are different types for each building. The residents have almost unpredictable behaviour. Tolerance to variations in the indoor temperature is very individualistic.

A block diagram for a physical district heating model is shown in Figure 7.

Heat loss model of a building
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's 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:

$$Q_{loss} = k_l\,(T_i - T_o) \tag{1}$$

where Q_{loss} = Building heat loss (W);
k_l = Building heat loss factor (W/°C);
T_i = Indoor temperature (°C);
T_o = Outdoor air temperature (°C).

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

$$Q_{supp} = m \times c_p \times (T_s - T_r) \tag{2}$$

where Q_{supp} = Heat supplied (W);
m = Water flow (kg/s);
c_p = Water heat capacity (J/(kg°C));
T_s = Building supply water temperature (°C);
T_r = Building return water temperature (°C).

The cooling of the district heating water is a non-linear function of the operational and design parameters, and can be written as

$$T_r = f\,(T_s,\ T_i,\ m,\ T_{so},\ T_{io},\ m_o,\ T_{ro}) \tag{3}$$

where T_{so} = Supply water temperature at reference condition (°C);
T_{ro} = Return water temperature at reference condition (°C);
T_{io} = Indoor temperature at reference condition (°C);
m_o = Water flow at reference condition (kg/s).

Energy storage model for a building
By assuming that all heated parts of the building are heated at a uniform indoor temperature at all times, the building can be modelled as a single heat capacity element. A differential equation is then written relating the net heat flow to the building to time derivative of the indoor temperature and the building heat capacity. Then the building energy storage becomes as described in Equation 4:

$$\frac{dT_i}{dt} = \frac{1}{C} Q_{net} = \frac{1}{C}(Q_{supp} - Q_{loss}) \tag{4}$$

where C = Building heat capacity (J/°C);
Q_{net} = Net heat transferred to the building (W).

The heat supplied is an output signal from the radiator model, and the lost heat is calculated in the heat loss model.

Pipe heat loss model
Some heat is lost in the pipes connecting the pumping station and the buildings to be heated. The amount of the loss can be calculated by using the district heating pipe transmission effectiveness parameter. According to Valdimarsson (1993) the transmission effectiveness τ is defined as

$$\tau = \frac{T_2 - T_g}{T_1 - T_g} = e^{-\frac{U_p}{mc_p}} \tag{5}$$

where T_1 = Pipe inlet temperature (°C);
T_2 = Pipe outlet temperature (°C);
T_g = Ground temperature (°C);
U_p = Pipe heat loss factor (W/°C).

The reference value of τ can be calculated from reference flow conditions

$$\tau_o = \frac{T_{2o} - T_g}{T_{1o} - T_g} = e^{-\frac{U_p}{m_o c_p}} \tag{6}$$

where T_{1o} = Pipe inlet temperature at reference condition (°C);
T_{2o} = Pipe outlet temperature at reference condition (°C).

The parameters U_p and c_p are assumed to be constant all over the system. Combining Equations 5 and 6, the transmission effectiveness can be obtained

$$\tau = \tau_o^{\frac{m_o}{m}} \tag{7}$$

Combining Equations 5 and 7, the supply temperature to the house can then be calculated as

$$T_2 = T_g + (T_1 - T_g)\tau = T_g + (T_1 - T_g)\tau_o^{\frac{m_o}{m}} \tag{8}$$

The transmission effectiveness is normally well known at the design reference condition, so the steady state outgoing temperature can easily be calculated for any set of operational values of T_1 and m. This is used in the macroscopic model by calculating the signal T_S from the temperature of the water supplied to the system T_1 (which is a control signal) and the flow m, which is the flow controller output signal.

Flow controller model

The flow controller is unknown, there being no direct physical relationship between the indoor temperature and the water flow. Buildings have different regulating systems, and its inhabitants have different tolerances to changes in the indoor temperature. The relationship used between the indoor temperature and the water flow has to represent some average of all consumers in the system. One way to treat this problem is to study typical controller characteristics. A P-controller (Proportional controller) is defined by Equation 9:

$$m = k_p(T_{iref} - T_i) + \overline{m} \tag{9}$$

where k_p = Flow controller gain ((kg/s)/°C);
T_{iref} = Indoor temperature set point (°C);
$\overline{m}$ = Average flow (kg/s).

The proportional controller adjusts the water flow as a linear function of the deviation of the indoor temperature from the desired value. Here the water flow is assumed to be at its average value when the indoor temperature is at the desired level. The indoor temperature will not be at the desired value for any other situation.

A PI-controller (Proportional and Integral) uses, in addition to this, the integral of temperature error, which is defined by Equation 10:

$$m = k_p(T_{iref} - T_i) + k_i \int_0^t (T_{iref} - T_i)dt \tag{10}$$

where k_i = Flow controller integration factor ((kg/s)/(°Cs));
t = Time (s).

Macroscopic steady-state model

A steady-state model assumes that the system is memoryless, i.e. that any previous history can be discarded. Thus, the output signals are only dependent on the input and the control signals at the same point in time. This implies that there are no state variables; the input temperature is a constant. That also means that the flow controller is ideal, i.e. it will be able to adjust the flow such that the heat lost will be supplied to the building exactly. Recalling Equations 1 and 2, this implies that

$$Q_{supp} = Q_{loss} \quad \Rightarrow \quad m\, c_p\, (T_s - T_r) = k_l\, (T_i - T_o) \tag{11}$$

The steady state model for the flow can then be obtained directly as:

$$m = \frac{k_l(T_i - T_o)}{c_p.(T_s - T_r)} \qquad \text{or} \qquad m = -\frac{k_l}{c_p.(T_s - T_r)}T_o + \frac{k_l}{c_p.(T_s - T_r)}T_i \tag{12}$$

This is the steady-state model, relating the input signal T_o to the flow m for constant indoor temperature T_i.

Further information on factor k_l can be extracted from the reference condition by equating the reference condition heat supplied and lost:

$$Q_o = m_o c_p\, (T_{so} - T_{ro}) = k_l\, (T_{io} - T_{ao}) \qquad \text{or} \qquad k_l = \frac{m_o c_p (T_{so} - T_{ro})}{T_{io} - T_{ao}} \tag{13}$$

Macroscopic dynamic P-controller model

A model with a P-controller can be written as a first order differential equation by differentiating Equation 9:

$$\frac{dm}{dt} = -k_p \frac{dT_i}{dt} \tag{14}$$

Additionally, Equation 9 can be solved for T_i to obtain

$$T_i = T_{iref} - \frac{m - \overline{m}}{k_p} \tag{15}$$

and the model for a building with a P-controller is then obtained by combining Equations 1, 2, 4, 14 and 15 as

$$\frac{dm}{dt} = -\frac{k_p}{C}\left(c_p(T_s - T_r) + \frac{k_l}{k_p}\right)m - \frac{k_p k_l}{C}T_a + \frac{k_p k_l}{C}\left(T_{iref} + \frac{\overline{m}}{k_p}\right) \tag{16}$$

This is the final form of the P-controller building model.

Macroscopic dynamic PI-controller model

The model of a building with a PI-controller is a second order model, and is obtained by first differentiating Equation 10:

$$\frac{dm}{dt} = -k_p \frac{dT_i}{dt} + k_i(T_{iref} - Ti) \tag{17}$$

By combining Equations 1, 2 and 4, the first state equation for the model becomes

$$\frac{dT_i}{dt} = -\frac{k_l}{C}T_i + \frac{c_p}{C}(T_s - T_r)m + \frac{k_l}{C}T_a \tag{18}$$

By inserting Equations 15 and 18 into Equation 17, the second state equation becomes

$$\frac{dm}{dt} = \left(k_p \frac{k_l}{C} - k_i\right)T_i - k_p \frac{c_p}{C}(T_s - T_r)m - k_p \frac{k_l}{C}T_a + k_i T_{iref} \tag{19}$$

No measurements are available for the indoor temperature, and it is also only a measure of the stored energy in the hot mass of the buildings. A measurement equation relates the two states to the available output signal. The classical state form for Equations 18 and 19, together with the measurement equation is

$$\begin{bmatrix} \dfrac{dT_i}{dt} \\ \dfrac{dm}{dt} \end{bmatrix} = \begin{bmatrix} -\dfrac{k_l}{C} & \dfrac{c_p(T_s - T_r)}{C} \\ \dfrac{k_p k_l}{C} - k_i & -\dfrac{k_p c_p (T_s - T_r)}{C} \end{bmatrix} \begin{bmatrix} T_i \\ m \end{bmatrix} + \begin{bmatrix} \dfrac{k_l}{C} & 0 \\ -\dfrac{k_p k_l}{C} & k_i T_{iref} \end{bmatrix} \begin{bmatrix} T_a \\ 1 \end{bmatrix}$$
$$m = \begin{bmatrix} 0 & 1 \end{bmatrix} \begin{bmatrix} T_i \\ m \end{bmatrix} \tag{20}$$

This is the final form of the PI-controller building model.

Identifiability
It is not certain that all parameters needed in a model can be identified. Some of the parameters may be related through the model structure, or they may not be differentiated in the world. The former case can be tested for by calculation, but the latter case only by studying the model behaviour.

Calculation of identifiability of a certain model structure involves calculating the transfer functions from the input signals to the output signals, counting the number of independent parameters, and ensuring there are not any common factors cancelling out. This number is the maximum number of parameters available for identification for a given model structure. It is possible to show that the number of identifiable parameters is three times the number of states. A maximum of six parameters can, thus, be estimated in the PI-controller model. Since the second input in Equation 20 is constant, one of these parameters cannot be estimated, as there is no time dependency in the signal. Therefore, only five parameters in Equation 20 can be estimated for this model structure.

Equation 20 contains eight physical parameters, C, c_p, k_i, k_l, k_p, T_{iref}, T_s and T_r. One is given, in that the heat capacity of water c_p is known. The difference between the building supply and return temperatures goes into the model, so only their difference T_s-T_r can be estimated. The indoor temperature set point can be estimated as close to 20°C, and to remain constant at that value. This brings the number of individual parameters down to five, which allows them all to be estimated. The underlying physics of the system, however, shows a coupling between parameters k_l and T_s-T_r, so only the ratio between them can be estimated (Valdimarsson, 1995).

5.1.2 Statistical models

Statistical models are used to calculate the model output signal as a function of the input and control signals by using some mathematically convenient structure. Such models do not need to use *a priori* knowledge of the underlying physics of the real system. They are often called "black box" systems, since they treat the real system as a black box with a completely unknown interior.

The model structure defines which input and control signals are to be taken into account in the modelling, and the degree of the model defines the "time horizon" for all the signals. The parameter estimation is made by minimising a score function to find a set of suitable values for the model parameters. The score function is a quality measure for the model, stating how well the model performs in modelling some measured output signal. When a good model has been found, the difference between the simulated and measured output signal is "white", that is, all traces of information have been removed from the error signal and incorporated into the model. The remaining error is then white noise, that is, the value of the error at any time is not related to the previous error values.

Modelling a system with statistical models is a four step process:

1. Choice of a structure;
2. Choice of a model degree;
3. Parameter estimation:
4. Model validation.

This process is iterative, as model validation can demand a new structure or a new degree for the model. The more parameters a model has, the better it can model the measurements. At the same time the model can be a bad description of the underlying process. If the model has as many parameters as there are measurements, the model will fit exactly, but will most certainly perform badly when used to model a new set of data. In order to get around this difficulty, the model has to be used on a different data set from the one used for model parameter identification when the model is validated.

The macroscopic models in the present work can be used to derive the structure and the model degree from the physical knowledge about the system.

5.2 Microscopic models

Microscopic models can be used to describe the spatially distributed district heating system behaviour. The goal of developing such models is to be able to calculate the water flow, pressure and temperature in all pipes of the system as a function of time. Network theory provides convenient ways of determining the flow in a given network. The thermal state of the network can then be calculated from the flow solution (Valdimarsson, 1993).

6. DESIGN OF A GEOTHERMAL DISTRICT HEATING SYSTEM FOR IZTECH CAMPUS

6.1 Determination of the total heat load of the campus

IZTECH Campus has individual HVAC (heating, ventilation and air conditioning) systems for each building and the campus is still under development. For this reason, characteristics of existing buildings will be determined, and characteristics of buildings, planned for construction in the near future, will be estimated. These values will determine the total heating load of the campus. After consideration, 0.073 kW/m^2 as the unit-heating load per m^2 and 0.021 kW/m^3 as the per unit volume are acceptable values for 20°C internal temperature. These values are given in Table 1. Figures 8 and 9 show unit heating loads of the buildings per m^2 and per m^3.

TABLE 1: Unit heating loads

	Building name	**Heat loss (kW)**	**Total building usage area (m^2)**	**Total building usage volume (m^3)**	**Unit heating load (kW/m^2)**	**Unit heating load (kW/m^3)**
Faculty of engineering	Main building	162	2852	9412	0.057	0.017
	Classrooms building	127	1870	6171	0.068	0.021
	Laboratories	135	1948	6428	0.069	0.021
	Laboratories	160	2913	9613	0.055	0.017
	Mec. eng. lab.	193	2141	7065	0.09	0.027
	Mec. eng. lab.	228	1805	5957	0.126	0.038
Faculty of architecture	Studio building	310	4800	17280	0.065	0.018
	Main building	272	4897	17629	0.056	0.015
Faculty of science	Main building	213	3538	11675	0.06	0.018
	Laboratories	203	3276	10811	0.062	0.019
	Laboratories	186	3606	11900	0.052	0.016
Building of president	Main building	353	2994	9880	0.118	0.036
	Pre. of dep. building	442	5190	17127	0.085	0.026
Cafeteria	Cafeteria	412	4700	23500	0.088	0.018
CAMPUS TOTAL		**3395**	**46530**	**164448**	**0.073**	**0.021**

In this project, the campus buildings are considered in two groups: existing buildings and new buildings, which are under construction at present or that will be built in the future. These new buildings are shown in Table 2 with their assumed heat loads to determine campus heat load. Total heat load for existing buildings is 3988 kW and the total heat load for new buildings was assumed to be 4750 kW. Campus total heat load was determined as 8738 kW in this project.

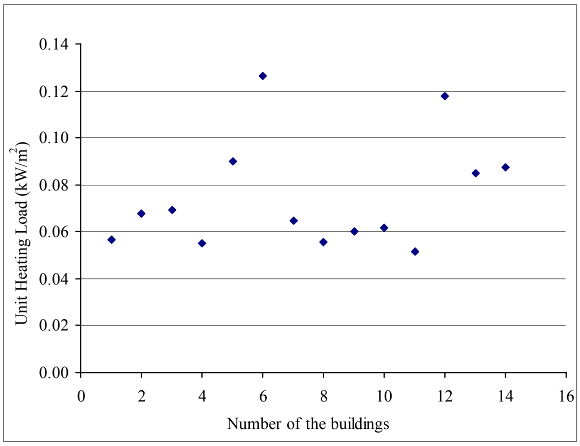

FIGURE 8: Unit heating load per area (kW/m^2)

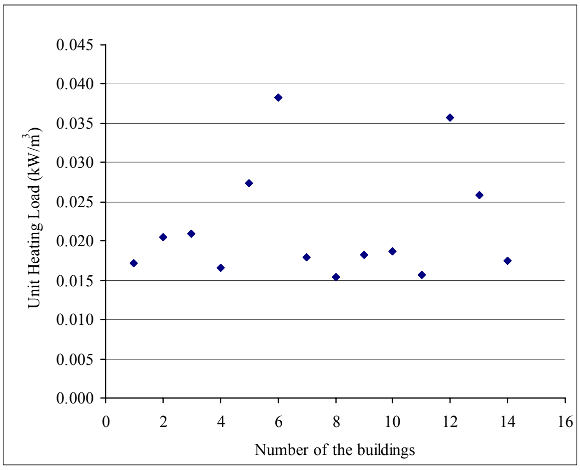

FIGURE 9: Unit heating load per volume (kW/m^3)

TABLE 2: Total heating load of the campus

Buildings	**Total heating load (kW)**
Faculty of architecture	5.823004e+69
Staff houses*	
Building A*	
Building B*	
Building C*	
Building D*	
Faculty of science	
Cafeteria	
Building E*	
Building F*	
Res.& dev. centre	
Sport centre*	
Medical centre*	
Library*	
Building G*	
Chemical engineering	
Faculty of engineering	
Mechanical engineering	
Mecatronic buildings*	
Incubator building	
Rectorship building	
Pre.of dep. building	
Campus TOTAL	

* These values are assumed

6.2 Heating system design

IZTECH Campus has a geothermal resource. However, the temperature of a production well and the flowrate of the geothermal fluid are not known because no geothermal wells have yet been drilled. The surface temperature in the spring in the campus is about 35°C. The results of studies indicate that the geothermal water will not be suitable to be used directly for heating purposes.

In this project, three types of heating systems are considered, direct and indirect evaporator types of heat pumps, and a fuel boiler heating system without geothermal energy. Figures 10, 11 and 12 show block diagrams of the different district heating systems. The existing heating systems with boilers were designed for obtaining peak loads for the buildings. As can be seen from Figures 11 and 12, the systems are designed with two loops: a geothermal loop and a city loop to reduce the investment costs due to the heat exchangers needed for the buildings.

The direct evaporator 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.

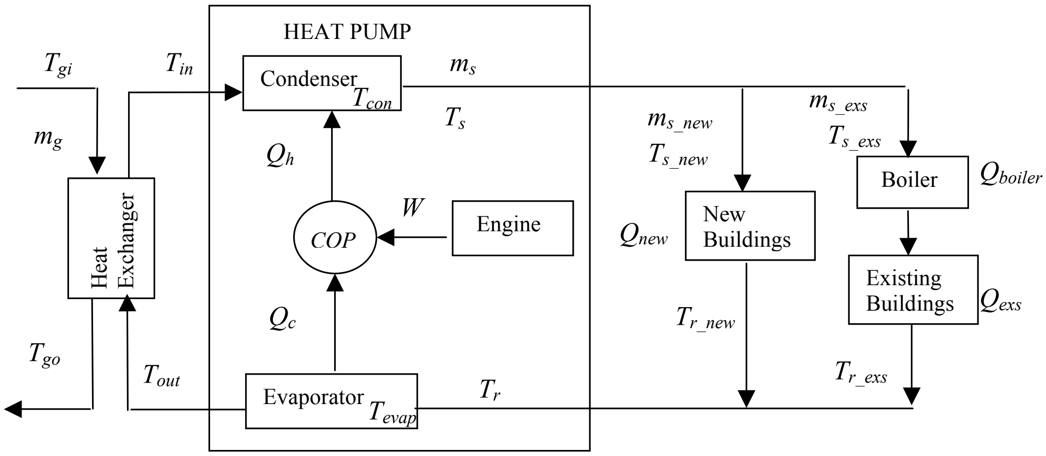

FIGURE 10: A block diagram of a planned district heating system with an indirect evaporator heat pump

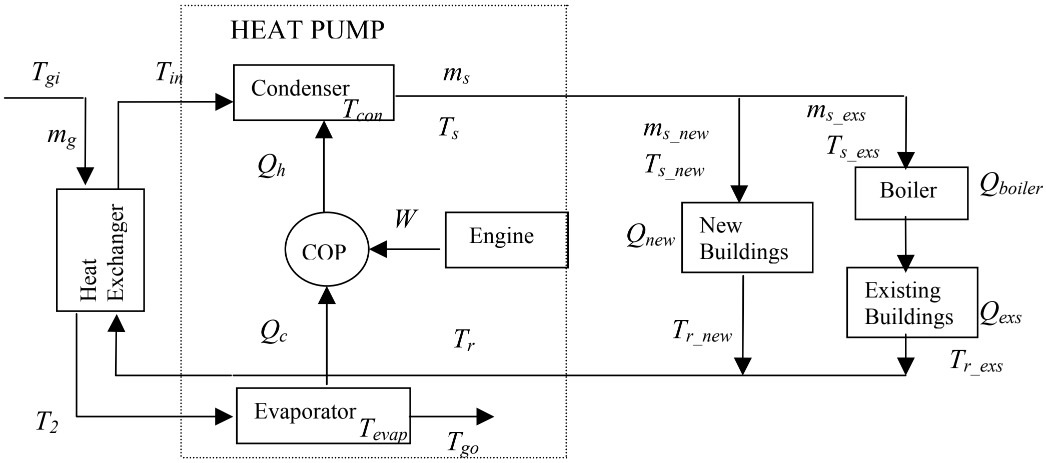

FIGURE 11: A block diagram of a planned district heating system with a direct evaporator heat pump

Variables used in Figures 10-12:

T_{gi} = Geothermal water temperature at heat exchanger inlet (°C);
T_{go} = Geothermal water discharge temperature (°C);
T_{in} = Condenser water inlet temperature (°C);
T_{out} = Evaporator outlet temperature (°C);
T_2 = Evaporator inlet temperature for direct evaporator heat pump (°C);
T_s = District heating water supply temperature at condenser outlet (°C);
T_{s_exs} = District heating water supply temperature to the existing building radiators (°C);
T_{s_new} = District heating water supply temperature to the new building radiators (°C);
T_{r_exs} = District heating water return temperature from the existing building radiators (°C);
T_{r_new} = District heating water return temperature from the new building radiators (°C);
T_r = District heating water return temperature at evaporator inlet (°C);
W = Net heat pump inlet power (kW);
Q_h = Heating power of heat pump condenser (kW);
Q_c = Cooling power of heat pump evaporator (kW);
m_{s_exs} = The existing buildings flowrate (kg/s);
m_{s_new} = The new buildings flowrate (kg/s);
m_s = District heating system secondary flowrate (kg/s);
m_g = Geothermal water flow (kg/s);
T_{evap} = Heat pump evaporating absolute temperature (K);
T_{cond} = Heat pump condensing absolute temperature (K).

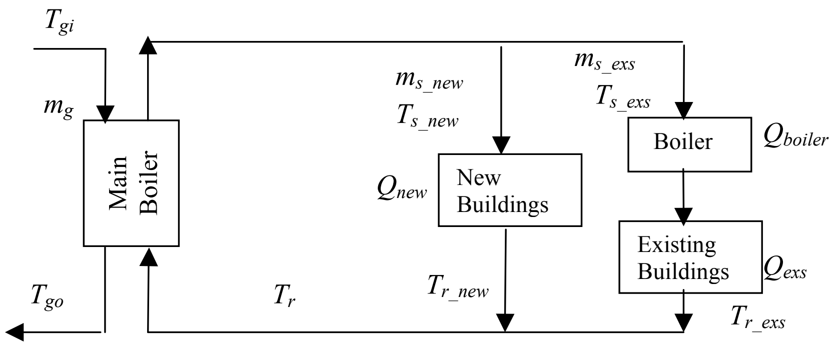

FIGURE 12: A block diagram of a planned district heating system using a main fuel boiler

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

The indirect evaporator arrangement is a little more complex. The evaporators are located on the heating system return main. Heat is extracted from the return fluids; this reduces the return temperature to the heat exchanger, increases the geothermal heat exchange and reduces the brine outlet temperature. The effect of the heat pump is the same as in the direct evaporator arrangement, in that residual heat is extracted from the brine, but in this case the action is indirect. All of the geothermal heat is transferred by a single route across the primary heat exchanger, which is being assisted by the heat pump. In the indirect evaporator arrangement, the heat pump and the heat exchanger influence each other. The advantage of the indirect evaporator arrangement over the direct evaporator is that corrosion in the evaporator is avoided without resorting to the expense of an additional heat exchanger (Harrison et al., 1990).

6.3 Simulation of the heating systems with dynamic approach

The campus is mostly used during working hours, from 9-17, and the existing building heating systems of the campus have been run during this time. For this reason, two main simulations have been considered, intermittent heating system and semi-intermittent heating system.

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 is turned off the rest of the time. The intermittent heating concept aims at providing the building with the required heat, keeping it at the design temperature for certain times, but not for the whole heating period. The heating period is considered to be from 9-17, which is the working time period for the campus. Hence, in the simulation program, the system was turned on one hour earlier and turned off at the end of this period.

For the semi-intermittent heating system simulation, the heating system with the heat pump and the boilers is run during working hours . In between, only the geothermal water without heat pump or boiler is used for heating. The reason is the need for the buildings' temperature to be raised to design temperature over a short period of time.

For these simulations, a control system with constant flowrate and variable return water temperature is used. The calculations were made for the existing buildings and new buildings, individually. In the calculations, each group was considered as a single building. Temperature drop along the pipeline is omitted. According to Emeish (2001), the time constant for uninsulated buildings is 26.3 hours. So,in the simulations, building thermal mass was assumed with building time constants of 24 hours.

6.3.1 Performance of the radiators

The logarithmic mean temperature difference for a radiator is defined as

$$\Delta T_{lm\_rad} = \frac{(T_s - T_i) - (T_r - T_i)}{\ln\left(\frac{(T_s - T_i)}{(T_r - T_i)}\right)} \tag{21}$$

The performance of the building radiator system depends on the supply and return water temperatures (Nappa, 2000), as given in Equation 22:

$$Performance = \frac{Q}{Q_o} = \left(\frac{(T_s - T_i)}{\ln\left(\frac{(T_s - T_i)}{(T_r - T_i)}\right)} \cdot \frac{\ln\left(\frac{(T_{so} - T_{io})}{(T_{ro} - T_{io})}\right)}{(T_{so} - T_{ro})} \right)^{(4/3)} \tag{22}$$

The index zero refers to the reference conditions. The reference condition of the existing space heating equipment (radiators, fan coil etc.) is 90-70°C.

Adding an extra radiator may be necessary for various supply or return temperature designs for a heating system. Table 3 and Figure 13 show the relationship between radiator design temperatures and the necessary increase ratio of the radiator size.

TABLE 3: Relationship between radiator design temperatures and the increase ratio of radiator size

T_s-T_r (°C)	Performance of existing radiators (%)	Necessary increase ratio of existing radiator size
90-70	100	1
85-65	88.8	1.126
85-60	82.5	1.211
80-60	78	1.282
80-55	71.8	1.392
80-50	66.5	1.527
80-40	52	1.922
80-35	44.6	2.24
80-30	36.5	2.74

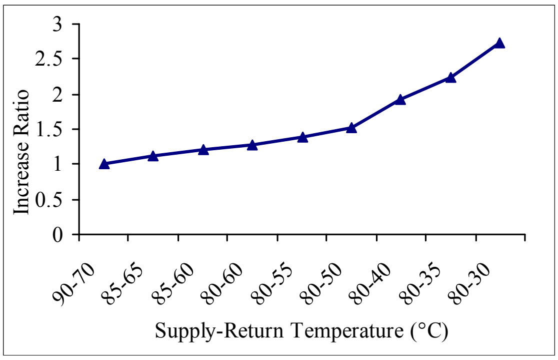

FIGURE 15: Radiator design temperatures vs. the increase ratio of radiator size

6.3.2 Calculating return water temperature

When a system is dynamic, T_r is calculated from Equation 23 (Nappa, 2000):

$$\frac{Q}{Q_o} = \frac{m}{m_o}\left(\frac{T_s - T_r}{T_{so} - T_{ro}}\right) = \left(\frac{(T_s - T_r)}{\ln\left(\frac{(T_s - T_i)}{(T_r - T_i)}\right)} \cdot \frac{\ln\left(\frac{(T_{so} - T_{io})}{(T_{ro} - T_{io})}\right)}{(T_{so} - T_{ro})} \right)^{(4/3)} \tag{23}$$

An iteration loop is made to find out T_r, as in the steady-state model.

$$T_{r,n+1} = (T_s - T_i).e^{-y} + T_i \qquad \text{where} \qquad y = \frac{\ln\left(\frac{T_{so} - T_{io}}{T_{ro} - T_{io}}\right)}{\left(\frac{(T_s - T_{r,n})}{(T_{so} - T_{ro})}\right)^{\left(\frac{3}{4}-1\right)} \left(\frac{m}{m_o}\right)^{\left(\frac{3}{4}\right)}} \tag{24}$$

The total secondary flowrate for the heating system is

$$m_s = m_{s,exs} + m_{s,new} \tag{25}$$

and the system return temperature can be calculated as

$$T_r = (m_{s,exs}T_{r,exs} + m_{s,new}T_{r,new}) / m_s \tag{26}$$

6.3.3 Relationships between mass flow and indoor temperature

For these simulations a control system with constant flowrate and variable return water temperature is used. The calculations were made for existing and new buildings, individually, and each group was considered as a single building.

For calculation of indoor temperature, Equation 20 can be written in different matrix formation as:

$$\begin{bmatrix} \dfrac{dT_i}{dt} \\ \dfrac{dm}{dt} \end{bmatrix} = \begin{bmatrix} -\dfrac{k_l}{C} & \dfrac{c_p(T_s - T_r)}{C} \\ \dfrac{k_p k_l}{C} - k_i & -\dfrac{k_p c_p (T_s - T_r)}{C} \end{bmatrix} \begin{bmatrix} T_i \\ m \end{bmatrix} + \begin{bmatrix} \dfrac{k_l}{C} & 0 \\ -\dfrac{k_p k_l}{C} & k_i T_{iref} \end{bmatrix} \begin{bmatrix} T_a \\ 1 \end{bmatrix}$$

$$m = \begin{bmatrix} 0 & 1 \end{bmatrix} \begin{bmatrix} T_i \\ m \end{bmatrix} \tag{27}$$

Equation 27 can easily be solved by the discrete method. A common expression of a differential matrix equation is

$$\frac{d\mathbf{x}}{dt} = \mathbf{Ax} + \mathbf{Bu} \tag{28}$$

The discrete version of the equation can be written as:

$$x_{t+\Delta t} = \mathbf{\Phi}_t x_t + \mathbf{\Gamma}_t \mathrm{u}_t \tag{29}$$

where

$$\mathbf{\Phi} = e^{\mathbf{A}\Delta t} \tag{30}$$

By using the Taylor equation, this can be written as

$$\mathbf{\Phi} = e^{\mathbf{A}\Delta t} = \mathbf{I} + \Delta t\mathbf{A} + \frac{(\Delta t\mathbf{A})^2}{2} + \cdots = \sum_{j=0}^{\infty} \frac{(\Delta t\mathbf{A})^j}{j!} \tag{31}$$

And when the inverse of **A** exists **Γ** can be solved as

$$\mathbf{\Gamma} = \mathbf{A}^{-1}(\mathbf{\Phi} - \mathbf{I})\mathbf{B} \tag{32}$$

If x is known at a time t, it can be calculated in the next step ($t+\Delta t$) by using Equation 31 (Nappa, 2000).

6.3.4 The heat pump

It is possible to use the heat pump technique to extract thermal energy from a heat source with low temperature to make thermal energy available at higher temperature. Heat pumps need external energy input to work. The most common are electric motor driven heat pumps but there are also gas motor or chemical absorption systems. A heat pump works like a refrigerator, where the working fluid is circulated in a closed circuit, removing heat from inside the freezer and discharging it to the surroundings. In the heat pump, the working fluid extracts heat from the heat source through evaporation and discharges it by condensation to the district heating water. To do this work an external energy input is required and the most commonly used type is a compressor driven by an electric motor, but chemical absorption, gas compression and other methods are available.

The ratio of the output energy to operating energy input is the basic measure of the effectiveness of a heat pump and very important to the economics of the heat pump operation, as previously referred to. This ratio is known as the "Coefficient of Performance" *COP*, and is very attractive for heat sources with temperature in the range 20-40°C. For example, if the geothermal resource is 30°C and is cooled down to 20°C, and the hot water to space heating is 55°C, then the *COP* factor could be around 4. This means that the heat output for space heating is about four times the energy input to the compressor motor. The typical performance limits of geothermal heat pumps are:

- Geothermal source temperature range: 18-65°C;
- Geothermal water flow: 50-300 m^3/h;
- Heating water temperature range: 50-90°C;
- Heat capacity: 0.5-30 MW.

In a heat pump plant the following equipment is installed (Geothermal Energy Network, 2002):

In the *evaporator,* geothermal water transmits its heat to the working fluid and brings it to a boiling point at low pressure, causing its evaporation.

Turbine compressor, an electrical motor-driven one increases the pressure and therefore the temperature of the gas. Different heat capacity of the heat pump plant can be achieved by combining the various compressor frame sizes with various commercially available working fluids.

In the *condenser* the heated working fluid (gas) transfers its heat to the circuit of the heated water and is brought back to liquid phase.

Pressure control valve. After the condenser, the pressure of the working fluid is decreased by a reduction valve and the working fluid (after the flash box) is returned to the evaporator in order to complete the cycle, and can then be re-used.

Figure 14 shows the schematic operation of vapour compressor heat pumps and Figure 15 a typical temperature vs. enthalpy diagram.

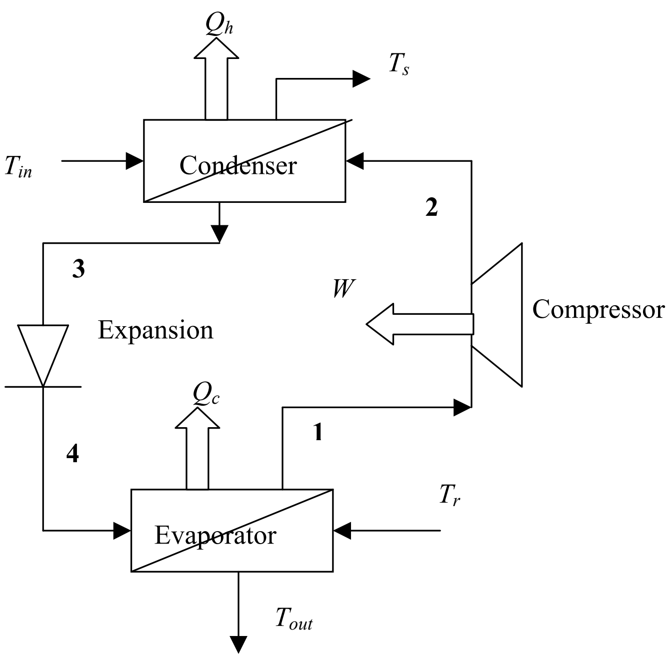

FIGURE 14: Schematic operation of vapour compression heat pumps

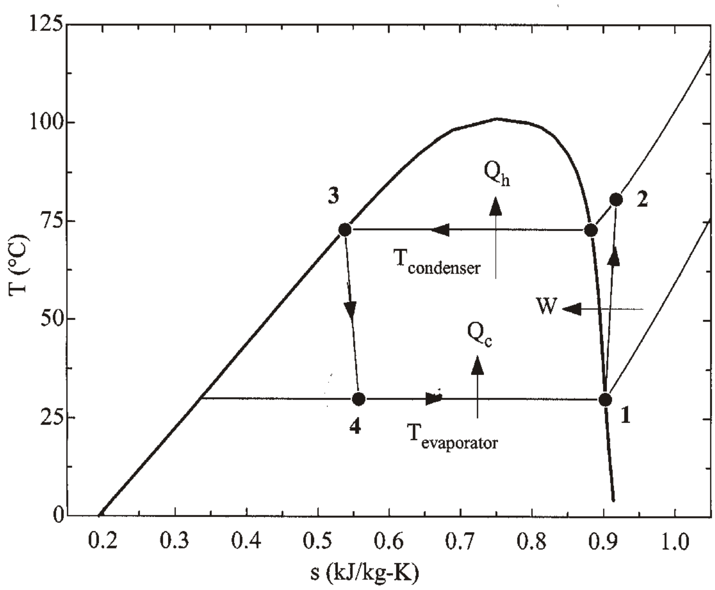

FIGURE 15: T-s diagram for vapour compression heat pumps

Simplifying heat pump capacity calculations

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

$$COP_{car} = \frac{T_{cond}}{T_{cond} - T_{evap}} \tag{33}$$

or

$$COP = \frac{Q_h}{W} = \frac{Q_h}{Q_h - Q_c} \tag{34}$$

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. Hence, the *COP* becomes (Harrison et al., 1990)

$$COP = 0.5 \times COP_{car} \tag{35}$$

Here, the theoretical value was first used for the 50 heating alternatives, and the heat pump capacity calculated by simplification. Thereafter, the real *COP* was calculated for the best heating systems.

According to Figure 10, the heat pump heat flows can be written as:

$$Q_h = m_s\, c_p \left(T_s - T_{in}\right) \tag{36}$$

$$Q_c = m_s\, c_p \left(T_r - T_{out}\right) \tag{37}$$

To calculate heat pump capacity, the evaporator outlet temperature (T_{out}) and condenser water inlet temperature (T_{in}) must first be calculated. Some assumptions are necessary, in this study they are:

- The exact geothermal water temperature is not known, hence calculations were done for the temperature 50°C at heat exchanger inlet;
- The condenser water inlet temperature is 3°C lower than geothermal water temperature at heat exchanger inlet, i.e. $T_{in} = T_{gi}$ - 3;
- Heat pump condensing absolute temperature is 3°C higher than the district heating water supply temperature at the condenser outlet, or $T_{cond} = T_s$ + 3;
- Heat pump evaporating absolute temperature is 3°C lower than evaporator outlet temperature, i.e. $T_{eva} = T_{out}$ - 3 for indirect evaporator heat pump, and $T_{eva} = T_{go}$ - 3 for direct evaporator heat pump.

Using these assumptions and Equations 33, 34, 35, 36 and 37, the evaporator outlet temperature can be calculated by iteration. After that the heat pump capacity is calculated.

Real calculation
*COP*s are inversely proportional to the temperature difference of the working fluid when releasing and absorbing heat. These temperatures of the working fluid determine the performance of the heat pump. This has important implications for the modelling of a 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 (Harrison et al., 1990).

In this project, real Carnot efficiency is calculated using R134a as a working fluid and compressor efficiency is assumed 0.8.

$$Q_h = m_{R134a}\,(h_2 - h_3) \tag{38}$$

$$Q_c = m_{R134a}\,(h_1 - h_4) \tag{39}$$

$$W = m_{R134a}\,(h_2 - h_1) \tag{40}$$

where m_{R134a} = Flowrate of the working fluid, R134a (kg/s);
h_1 = Enthalpy of working fluid at evaporator outlet and compressor inlet (kJ/kgK);
h_2 = Enthalpy of working fluid at compressor outlet and condenser inlet (kJ/kgK);
h_3 = Enthalpy of working fluid at condenser outlet and expansion valve inlet (kJ/kgK);
h_4 = Enthalpy of working fluid at evaporator inlet and expansion valve outlet (kJ/kgK).

In Figures 16 and 17, real Carnot efficiency and *COP* profiles are shown depending on evaporator temperature with 73°C condenser temperature and 45 kg/s secondary flow rate. Carnot efficiency changes between 0.565 and 0.61. As can be seen from the figures, *COP* and Carnot efficiency decrease if the evaporator temperature drops to lower values. Because of heat demands, the evaporator temperature should be limited.

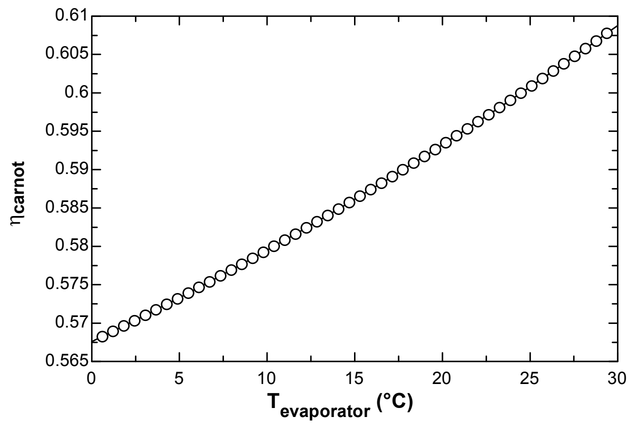

FIGURE 16: Real Carnot efficiency as a function of evaporator temperature

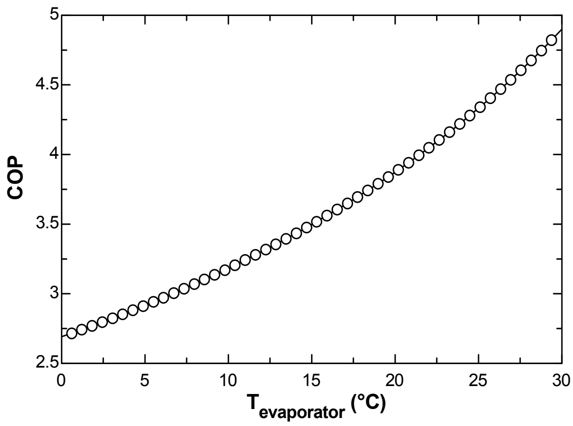

FIGURE 17: *COP* as a function of evaporator temperature

6.3.5 Heat exchangers

Two types of heat exchangers have proven most satisfactory in geothermal service, tube and shell-straight through with geothermal energy on the tube side; and the plate type. Tube and shell exchangers have some disadvantages such as a lack of flexibility to accommodate changes in temperature and flow conditions to meet load changes, greater floor space required, and they are considered less efficient. 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 (Lienau, 1981).

Normally the geothermal temperature and the flow remains fixed while the return temperatures and flows from the heating network fluctuate as the heat demands of the users change.

The geothermal heat transfer is given by

$$P_g = M_s E_x (T_{gi} - T_{out}) \tag{41}$$

where P_g = Actual rate of heat transfer (kW);
E_x = Heat exchanger effectiveness;
M_s = The smaller of the thermal capacities of the flows (kW/°C).

For a counter flow plate heat exchanger, effectiveness is given by (Harrison, 1997)

$$E_x = \frac{[1 - \exp(-N(1-R))]}{[1 - R\exp(-N(1-R))]} \tag{42}$$

where R = Ratio of capacity rates;
N = Number of transfer units.

and N is given by (Kevin et al., 1991)

$$N = \frac{UA}{M_s} = \frac{\Delta T_m}{\Delta T_{lm\,HEX}} \tag{43}$$

where U = Overall heat transfer coefficient of the heat exchanger (kW/°Cm^2);
A = Surface area of the heat exchanger (m^2);
ΔT_m = Larger temperature change (°C);
$\Delta T_{lm\,HEX}$ = The logarithmic mean temperature difference of the heat exchanger (°C).

The two basic rules are that the schemes be designed and operated so that
- secondary flow through the primary heat exchanger is always greater than the geothermal flow; and
- network return temperature must be as low as possible at all times.

Thus, the network must be operated with variable temperatures and flows (Harrison, 1997).

6.3.6 Approximate operation cost of the heating system

The cost of yearly energy consumption is calculated approximately by

$$C_{tot} = C_h + C_c + C_w \tag{44}$$

where C_{tot} = Total cost of the yearly energy consumption (USD);
C_h = Cost of the heating system main energy consumption (USD);
C_c = Cost of the electricity consumption of the circulation pump (USD);
C_w = Cost of the electricity consumption of the well pump (USD).

The cost of energy consumption for the heat pumps is

$$C_h = W_{el} \times P_{el} = \frac{W_{annual}}{\eta_{el}} P_{el} \tag{45}$$

where W_{el} = Total electric power input of the heat pump motor (kWh);
W_{annual} = Total heat pump energy (kWh);
P_{el} = Electrical energy price (USD/kWh);
η_{el} = Electric motor efficiency of the heat pump.

The electrical energy price was 0.11 USD/kWh in August 2002 (TEDAS, 2002). Electric motor efficiency is assumed 0.8 in the calculations

The cost for the fuel boiler is

$$C_h = m_{fuel} \times P_{fuel} = \frac{Q_{Boiler}}{Hu.\eta_{boiler}} P_{fuel} \tag{46}$$

where m_{fuel} = Total fuel oil consumption (kg);
P_{fuel} = Fuel oil price (USD/kg);
Hu = Specific heat capacity of the fuel oil (kWh/kg);
η_{boiler} = Efficiency of the boiler.

The specific heat capacity of fuel oil is 11.27 kWh/kg and the fuel price was 0.42 USD/kg in August 2002 (Petrol Ofisi, 2002). In the calculations the boiler efficiency is assumed 0.8.

The cost for the circulation pumps is

$$C_c = \frac{m\, g\, h_p}{1000\, \eta_{motor}\, \eta_{pump}} P_{el} \tag{47}$$

where m = Secondary flowrate (kg/s);
g = Gravitational acceleration constant (9.81 m/s^2);
h_p = Total dynamic head (TDH) of pump (m);
η_{motor} = Motor efficiency;
η_{pump} = Pump efficiency.

The total dynamic head of pump has been estimated from the heating system pressure drop with different flow rates with the help of the Pipelab program. Motor efficiency is assumed 0.85 and pump efficiency is assumed 0.7. Equation 47 can also be used for the well pump, but in this case, the total dynamic head is not known. Hence, this value is assumed 100 m.

6.4 Heating alternatives

In this project, heating simulation alternatives were made for heating systems based on indirect evaporator heat pump, direct evaporator heat pump and a fuel boiler. These heating alternatives were split mainly into two groups, intermittent heating and semi-intermittent heating. In these alternatives, different supply temperatures, different radiator design temperatures, and different flow rates were used for the heating systems, and their degree hours and approximate operation costs were calculated. Degree hours were calculated from the sum of the differences between design inside temperature and building inside temperature. With heating, a degree hour value of 0 is desired. The value of the degree hour shows how much the building can be heated. If the degree hours are calculated for the weather data without heating, during working hours the value is 7624.6 and for the whole year it is 42,370, i.e. if indoor temperature is assumed to be the same as outdoor temperature. Calculation of the approximate operation cost is discussed in Section 6.3.6. The best heating system model was determined from these alternatives.

In all, 50 heating alternatives were studied. Some of these alternatives are given in Table 4. Figure 18 indicates the results for the alternatives, depending on their degree hours and operation costs.

TABLE 4: Some of the studied alternatives for heating systems

Alternative	Explanations
1	Intermittent heating, fuel boiler Flowrate 45-40 kg/s, supply temperature 90-90˚C, radiator size 1 (90-70)
2	Intermittent heating, indirect evaporator heat pump Flow rate 45-40 kg/s, supply temperature 70-70˚C without boiler, radiator size 2.25 (70-40)
3	Semi-intermittent heating, indirect evaporator heat pump Flowrate (25+20)-(25+15) kg/s, supply temperature 70-70˚C without boiler, radiator size 1 (90-70)
4	Semi-intermittent heating, indirect evaporator heat pump Flowrate (25+20)-(25+15) kg/s, supply temperature 70-70˚C without boiler, radiator size 2.25 (80-35)
5	Semi-intermittent heating, Indirect evaporator heat pump Flowrate: (40-15)-(40-20) kg/s, supply temperature 60-60˚C without boiler, radiator size 2.25 (80-35)
6	Semi-intermittent heating, only geothermal. Flow rate (40)-(40) kg/s, supply temperature 47-47˚C without boiler, radiator size 2.25 (80-35)

As can be seen from Figure 18, semi-intermittent heating generally gives better results than intermittent heating, with low degree hours and operation costs. There was one exception, semi-intermittent heating had high operation cost and degree hours for a fuel boiler heating system.

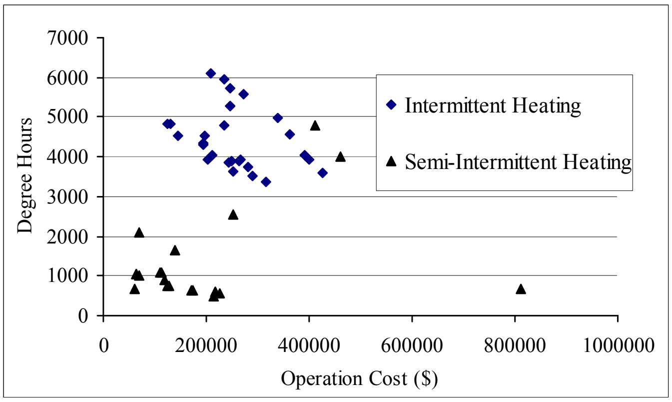

FIGURE 18: Relationship between degree hours and the operation cost of the heating alternatives

6.4.1 Alternative 1

At present, the campus has individual fuel boiler heating systems and these heating systems are run during working hours between 9 and 17 all week. Outside that time period the systems are turned off. The radiators were designed for 90-70°C inlet and outlet water temperature in the existing buildings.

Figure 19 shows the indoor temperature profile of the existing buildings for the intermittent fuel boiler heating system. In this alternative the supply temperature is 90°C for existing and new buildings. Radiator design temperatures are kept at 90-70°C; the flow rate is 45 kg/s for new buildings and 40 kg/s for existing buildings.

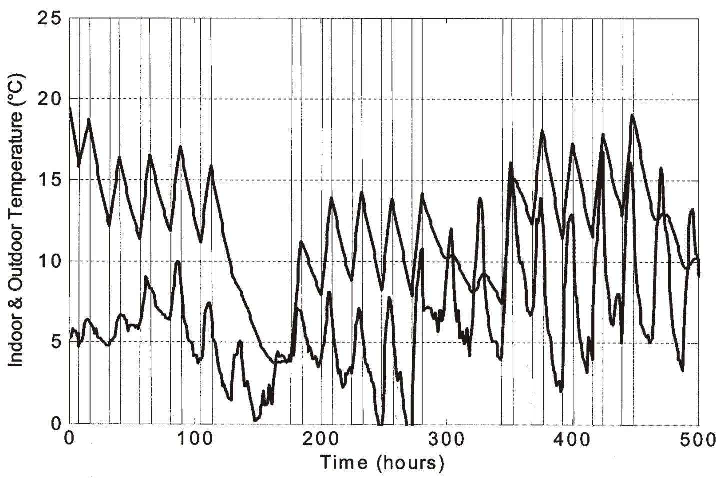

FIGURE 19: Indoor temperature profile for the intermittent fuel boiler heating system in Alternative 1

As can be seen from Figure 19, the indoor temperature of the building mostly cannot reach 20°C during cold weather. On the contrary, it can become very low during the heating period (Table 5), and the indoor temperature reaches its maximum value at the end of the heating period i.e when it is not required. Furthermore, the buildings have a degree hours value of 4090 for new buildings and 4028 for the existing buildings during the heating periods, indicating poor conditions inside the building. As expected and due to the discontinuous use of the heating system, the fuel consumption is high, or 921,050 kg.

6.4.2 Alternative 2

The second alternative covers an indirect evaporator heat pump heating system with the same flowrate, 45 kg/s for new buildings and 40 kg/s for existing buildings. In this alternative, supply water temperature is 70°C for new and existing buildings. Radiator design temperature was changed to 70-40°C for each group of buildings. Hence, the radiator size must be 2.25 times the existing radiator size. The extra investment cost for the existing buildings is nearly USD 88,000 (Demirdokum, 2002).

The indoor temperature profile of the existing buildings for the intermittent indirect evaporator heat pump heating system is given in Figure 20. The buildings have a degree hours value of 3381 for new buildings and 3282 for existing buildings. The total operation cost is USD 270,974.1, and poor conditions still exist inside the buildings. As can be seen from Figures 21 and 22, the heat exchanger and the heat pump must have big capacities. The heating system is run nearly 894 hours during the working period in a year.

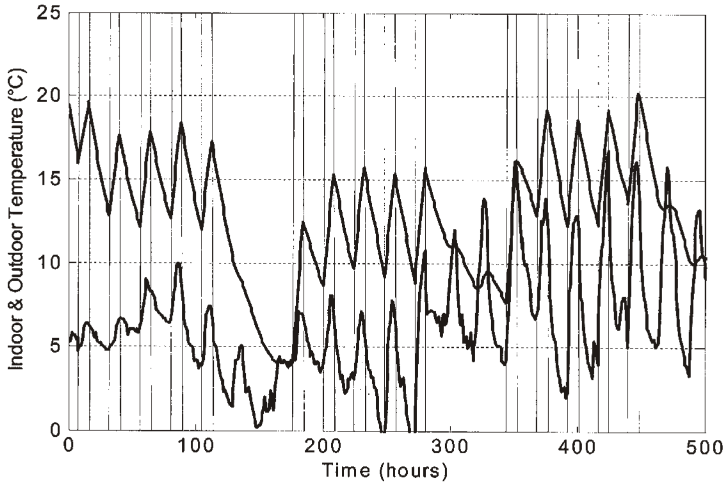

FIGURE 20: Indoor temperature profile for the intermittent indirect evaporator heat pump heating system in Alternative 2

Using the same values for supply temperature, flow rate and radiator design temperature, calculations were done for a heating alternative based on a direct evaporator heat pump heating system. The results are similar to the ones based on an indirect evaporator heat pump.

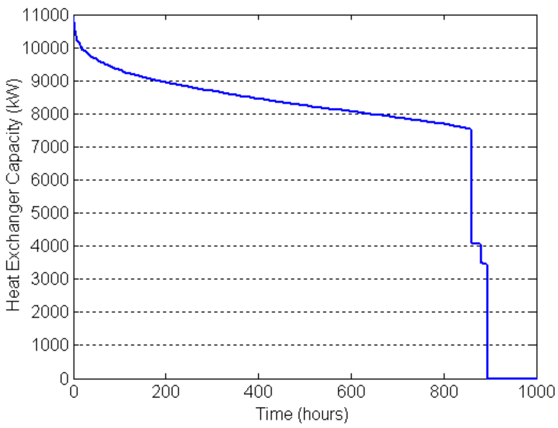

FIGURE 21: Heat exchanger capacity duration curve for the intermittent indirect evaporator heat pump heating system in Alternative 2

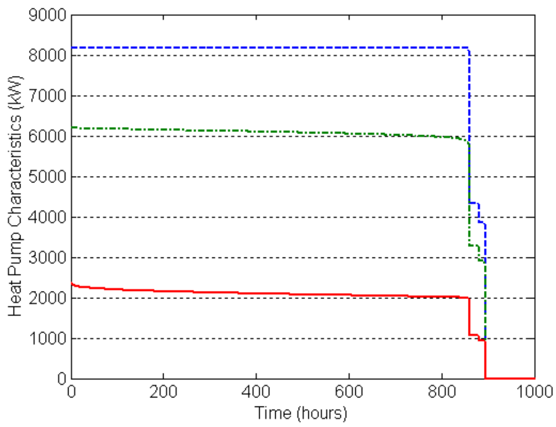

FIGURE 22: Indirect evaporator heat pump characteristics duration curve for the intermittent heating system in Alternative 2

6.4.3 Alternative 3

This alternative deals with a semi-intermittent indirect evaporator heat pump heating system. Heating with the heat pump and boilers was run during working hours, after that heating continued with only geothermal energy. Supply water temperature was 70°C during working hours, and 47°C after working hours for each group of buildings. Radiator design temperature was kept the same at 90-70°C. Flowrate was 45 kg/s for new buildings and 40 kg/s for existing buildings, during working hours. After working hours at night and on weekends, the flowrate was 25 kg/s for each group of buildings.

As can be seen from Figure 23, in this semi-intermittent heating alternative, indoor conditions are still poor while total operation cost is USD 243,111, degree hours are 2560 for new buildings and 2462 for existing buildings. In spite of the improvement in indoor temperature during non-working hours, during working hours indoor temperature cannot be changed so much. That means supply heat is less and the buildings do not receive the necessary heat during this period.

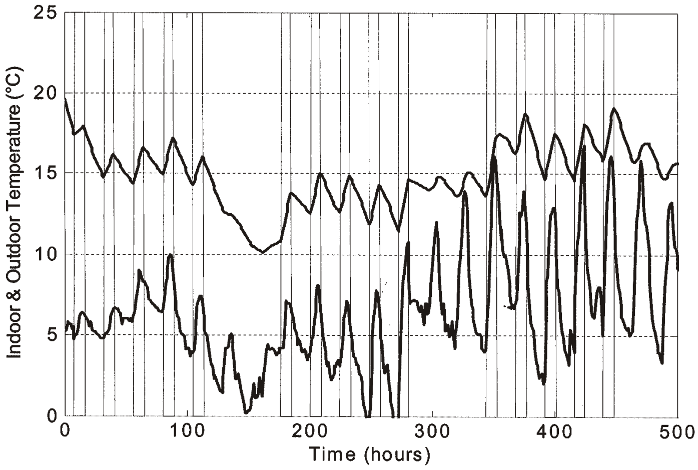

FIGURE 23: Indoor temperature profile for the semi-intermittent indirect evaporator heat pump heating system in Alternative 3

6.4.4 Alternative 4

This is the same as Alternative 3, with only one difference; the alternative radiator design temperature was changed to 80-35°C. This means that the radiator size must be increased 2.25 times. The initial cost is nearly USD 88,000 for existing buildings.

By looking at Figure 24, which is a plot of the indoor temperature profile, it is clear that the indoor temperatures usually reach the set temperature 20°C, even during cold weather. Total operation cost is USD 199,062, degree hours are 624 for new buildings and 484 for existing buildings. It is obvious that the indoor temperatures of the buildings can be improved with increased radiator size. This also reduces operation costs.

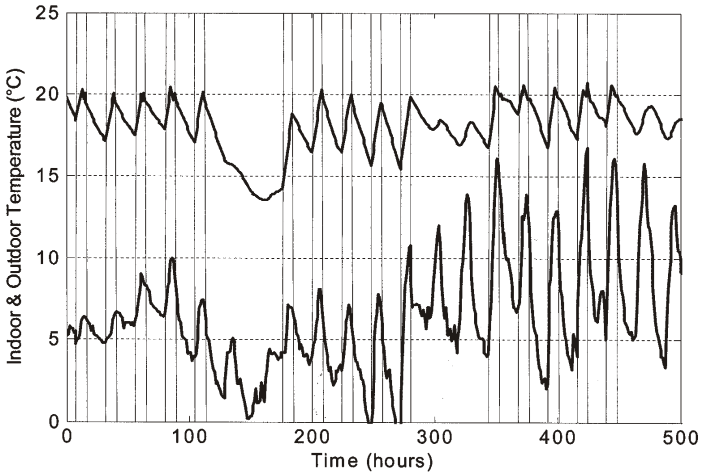

FIGURE 24: Indoor temperature profile for the semi-intermittent indirect evaporator heat pump heating system in Alternative 4

6.4.5 Alternative 5

Here a semi-intermittent indirect evaporator heat pump heating system is used to reduce total operation cost. Supply temperature is reduced to 60°C. Flowrate is decreased during working hours, and increased after working hours. For new buildings it is 25 kg/s and for existing buildings 20 kg/s during working hours, after that 40 kg/s for each group of buildings. Figure 25 shows the indoor temperature profile.

For this alternative, the degree hours value is 937 for new buildings and 838 for existing buildings during working hours, while the total approximate operation cost is USD 111,564. The total operation cost of the heating system can be reduced significantly with this heating alternative.

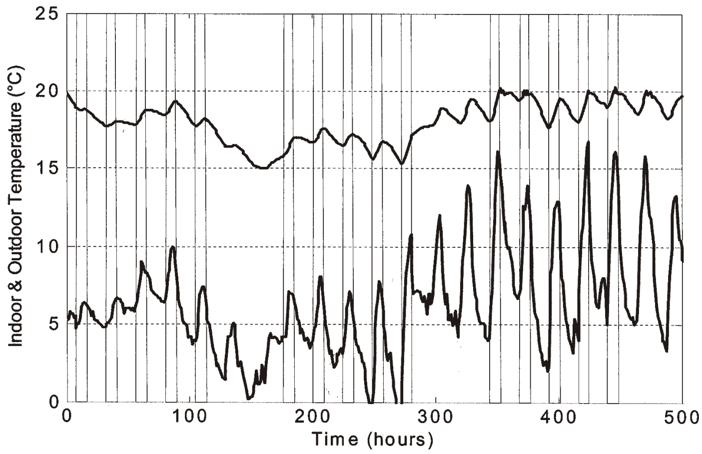

FIGURE 25: Indoor temperature profile for the semi-intermittent indirect evaporator heat pump heating system in Alternative 5

Figures 26 and 27 show heat pump characteristics and heat exchanger capacity duration curves. In this alternative their value is lower than in the previous alternatives. Maximum heat pump inlet power is nearly 600 kW until outdoor temperature reaches 13°C . Above that, the capacity decreases to half the maximum capacity. The heating system return water temperature profile is given in Figure 28 with an indoor temperature profile. They have similar characteristics, as can be seen.

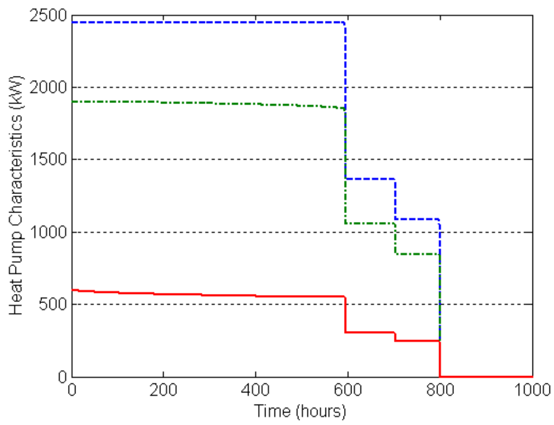

FIGURE 26: Indirect evaporator heat pump characteristics duration curve for the semi-intermittent heating system in Alternative 5

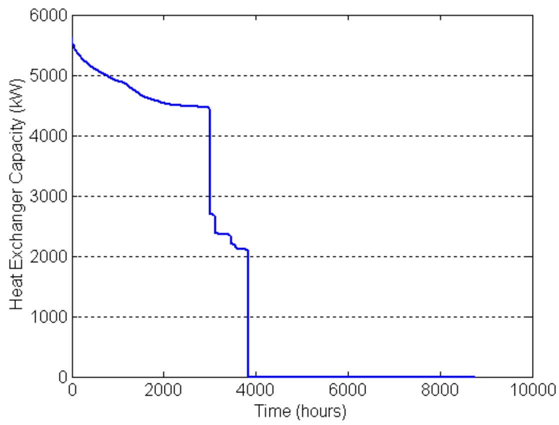

FIGURE 27: Heat exchanger capacity duration curve for the semi-intermittent heating system in Alternative 5

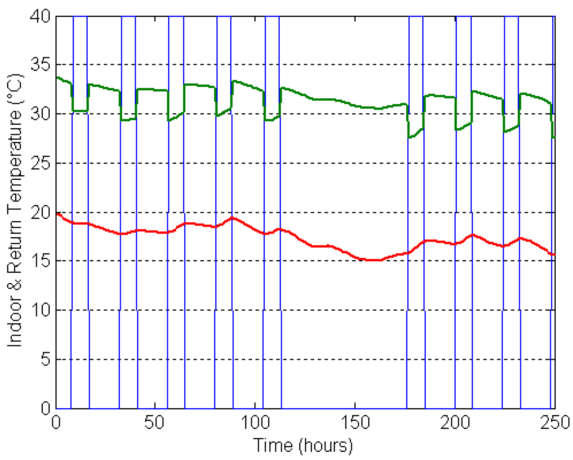

FIGURE 28: Return and indoor temperature profiles for semi-intermittent indirect evaporator heat pump heating system in Alternative 5

6.4.6 Alternative 6

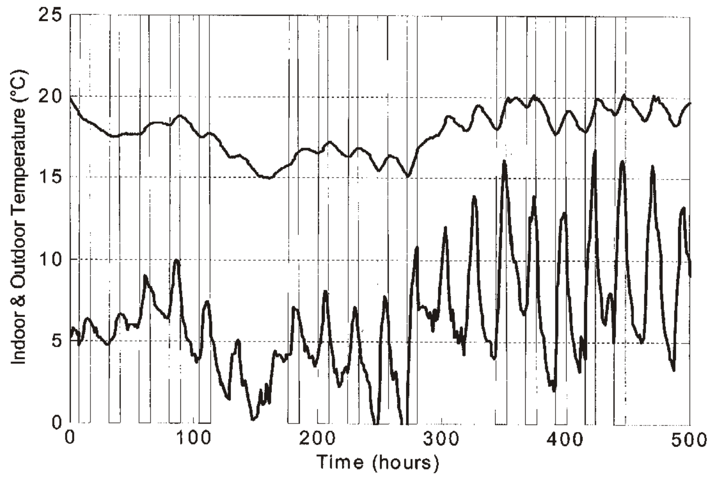

FIGURE 29: Indoor temperature profile for continuous heating with only geothermal energy as in Alternative 6

In this alternative only geothermal energy is used for heating all day. Supply temperature is 47°C and flow rate is 40 kg/s for each group of buildings. Radiator design temperature is 80-35°C and the increased ratio for radiator size is 2.25. The degree hour value is 1197 for new buildings and 966 for existing buildings during working hours, while the total approximate operation cost is USD 64,022. The total operation cost of this heating system is the lowest for this case, including only circulation pumps and well pump electricity consumption costs. On the other hand, the indoor temperature is generally near the set inside temperature except during cold weather (Figure 29).

In Figure 30, the heat exchanger duration curve is given. When the heat exchanger is not the only heat source of the system, its capacity is not so high. As can be seen from Figure 31, the return water temperature is between 30 and 35°C.

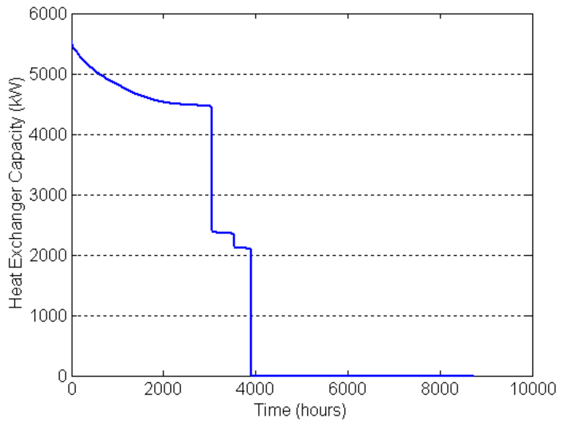

FIGURE 30: Heat exchanger capacity duration curve for continuous heating with only geothermal energy as in Alternative 6

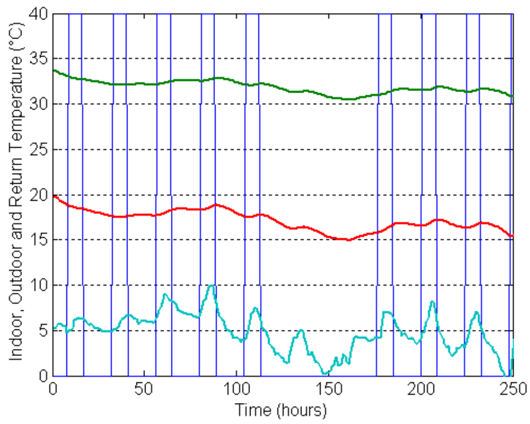

FIGURE 31: Return, indoor and outdoor temperature profiles for continuous heating with only geothermal energy as in Alternative 6

6.5 Results and comparison of the alternatives

The main results of these 6 heating alternatives are given in Tables 5, 6 and 7. They show that the intermittent heating system gives poor indoor conditions, indoor temperatures generally cannot reach the set temperature. On the contrary, it can go down to very low values 4.9°C during working hours. The detailed results on indoor temperatures are given in Appendix I. Similarly, operation costs are high. For this type of heating system long heating hours and bigger systems should be used in order to achieve comfortable indoor conditions, which require higher initial cost and more consumption than the semi-intermittent heating system. Hence the use of this system is not suitable.

Table 5 shows that with semi-intermittent heating, comfortable indoor conditions can be obtained. System capacity and operation cost are less than for intermittent heating systems. In Table 5, average deviation shows the average temperature difference between indoor temperature in buildings and design indoor temperature during the period. Indoor temperature is very close to comfortable temperature , especially for Alternatives 4, 5 and 6, and not only during working hours but all the time. Thus, buildings can be heated throughout the year with small capacity and low operation cost with semi-intermittent heating.

TABLE 5: Indoor conditions of the existing buildings for different alternatives

Type of heating system	Working hours			All year		
	Degree hours	Min. temp. (˚C)	Av. deviation (˚C)	Degree hours	Min. temp. (˚C)	Av. deviation (˚C)
1	4028.2	4.90	3.97	22478	3.72	4.39
2	3282.1	5.24	3.23	20311	3.9	3.97
3	2416.5	11.16	2.38	11565	10.15	2.26
4	483.6	14.85	0.48	3901.3	13.54	0.76
5	838.2	15.06	0.83	3648.1	14.99	0.71
6	966.3	14.89	0.95	3965.9	14.89	0.78

TABLE 6: Energy consumption of the heating systems and degree hours for intermittent heating simulation

Type of heating system	Heat exchanger (MWh)	Heat pump (MWh)	Boiler (MWh)	Degree hours for new buildings		Degree hours for old buildings	
				Working hours	All year	Working hours	All year
1	-	-	8306.8	4089.7	22658	4028.2	22478
2	7426.9	1840.4	-	3381.2	20589	3282.1	20311
3	11690	1398.8	-	2559.7	12232	2416.5	11565
4	15386	1142.2	-	623.74	4705.5	483.63	3901.3
5	16254	398.25	-	937.09	4210.9	838.22	3648.1
6	16421	-	-	1196.7	4860	966.29	3965.9

TABLE 7: Cost of energy consumption of the heating systems for intermittent heating simulation

Type of heating system	Heat pump electricity (USD)	Boiler fuel (USD)	Circulation pump electricity (USD)	Well pump electricity (USD)	Total cost (USD)
1	-	386840	5168.2	-	392008.2
2	253050	-	4957.1	12967	270974.1
3	192340	-	11813	38958	243111
4	157050	-	9122.9	32889	199061.9
5	54759	-	14448	42357	111564
6	-	-	16805	47217	64022

Although the campus is mostly used during conventional working hours, heating is necessary after that for staff houses, the medical centre and a few other buildings. Heating these buildings after working hours with the district heating system is not suitable because the necessary secondary flowrate is so low and these buildings are located far from each other and the heating centre. So, a semi-intermittent heating system is definitely one of the solutions for heating these buildings after working hours. On the other hand, making new heating systems for these buildings is necessary for peak loads during the weekends and after working hours. On campus, each existing building heating system generally has more than one boiler. These boilers can be separated according to building heat loads. The rest of the boilers can be used during peak loads.

According to operation costs, as seen in Table 7, the fuel boiler heating systems in Alternative 1 have high operation costs because of high fuel-oil costs. Therefore, using the existing fuel-boiler heating systems only during peak loads in cold weather is suggested.

The results indicate that radiator design temperature is very important for heating systems. It can easily be seen comparing Alternatives 3 and 4. Increasing radiator sizes results in better indoor conditions, reducing operation costs and system capacity. In this project 80-35°C inlet and outlet temperature is suggested as a radiator design temperature for each group of buildings. Hence, for new buildings radiators must be selected according to these temperatures, and the radiator sizes of the existing buildings must be increased 2.25 times.

In this project the best heating alternative is the continuous geothermal heating system. It has the lowest operation cost and acceptable indoor conditions. The results of this heating system are given in Tables 5, 6 and 7 as Alternative 6. In the calculations, the temperature drop along the pipeline is omitted and geothermal resource temperature is assumed 50°C. Compared to Alternative 1, the fuel fired boiler operating only during working hours, the saving is estimated as USD328,000, or almost four times the cost of added radiator surface for existing buildings. It is very clear and obvious that a design based on Alternative 6 is the most economical solution. This calls for a closer feasibility study, where a realistic estimate is made of the total refurbishment cost for the older buildings, and including both thermal and hydraulic calculations of the distribution network. Use of this geothermal heating system depends also on exact geothermal resource characteristics.

If the geothermal resource is not suitable for direct-use, a semi-intermittent indirect evaporator heat pump system with low-supply temperature as for example in Alternative 4 can be suggested. Direct evaporator heat pump heating systems generally had the same results as the indirect evaporator heat pump, but due to corrosion and operation effects, they are not optimal.

6.6 Modelling of the network for IZTECH Campus

The design of a 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 (Lienau, 1981). The map of the campus and the considered district heating system is shown in Figure 32. The district heating system was designed according to the location of the buildings, especially those planned in the future. The network consists of a two-pipe system and the pipes are buried underground.

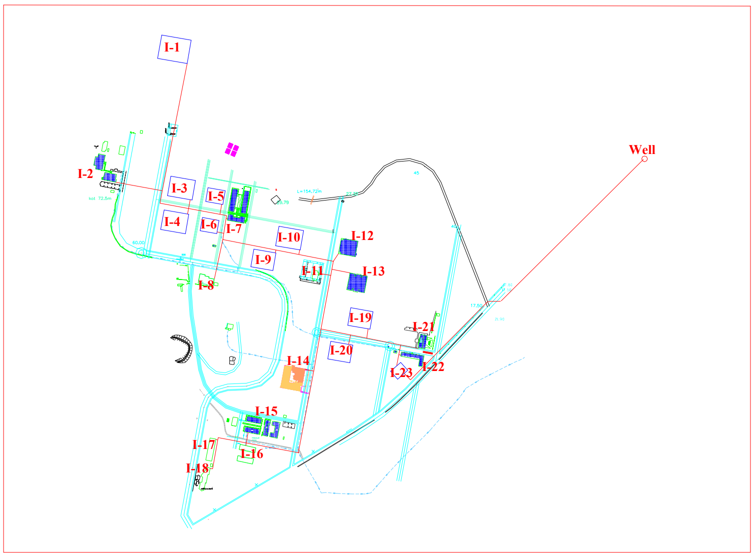

FIGURE 32: The map of the IZTECH Campus and the considered district heating system

I-1: Staff houses,
I-2: Faculty of Architecture,
I-3: Buildings A,
I-4: Buildings B,
I-5: Buildings C,
I-6: Buildings D,
I-7: Faculty of Science,
I-8: Cafeteria,
I-9: Buildings E,
I-10: Buildings F,
I-11: Res. & Dev. Centre,
I-12: Sport Centre,
I-13: Medical Centre,
I-14: Chemical Engineering,
I-15: Faculty of Engineering,
I-16: Mechanical Engineering,
I-17: Mecatronic Building,
I-18: Incubator Building,
I-19: Library,
I-20: Buildings G,
I-21: Rectorship Building,
I-22: Pre. of Dep. Building,
I-23: Heating Centre

The software "Pipelab", using the Matlab algorithms, was used to simulate the district heating system (Figure 33). Necessary input files include the number of nodes in the system, their xyz coordinates, connectivity relative to the nodes of the pipes with their length, diameter, roughness, heat loss, and boundary conditions, as necessary flow rates and the pressure head at the starting point. Here, the diameter of the pipes is assumed and then the software was used to select the optimum diameter for each pipe. The results are given in Appendix II. Necessary flow rates were calculated according to design temperatures 80-35°C, and 60°C supply temperature for new and existing buildings. Pressure head for the system was selected at 45 m.

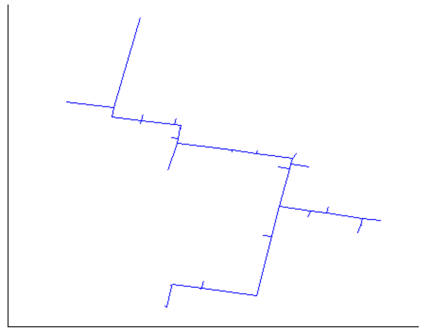

FIGURE 33: Sketch of the distribution network in Pipelab software

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 utilised, but the operating cost is high. On the other hand, if the pressure loss is low, the investment is badly utilised, but the pumping cost is low. The heat loss in a district heating pipe is higher for badly utilised pipes. The pressure loss per unit length is, thus a good indicator of optimality, but not a real cost function. The district heating practise is to design for say 0.5-2 bar/km pressure loss (Valdimarsson, 2001). The *h/L* diagram of the network, after finding the nodal heads according to the least squares solution, is shown in Figure 34. The target pressure loss has been selected 0.065 bar/km.

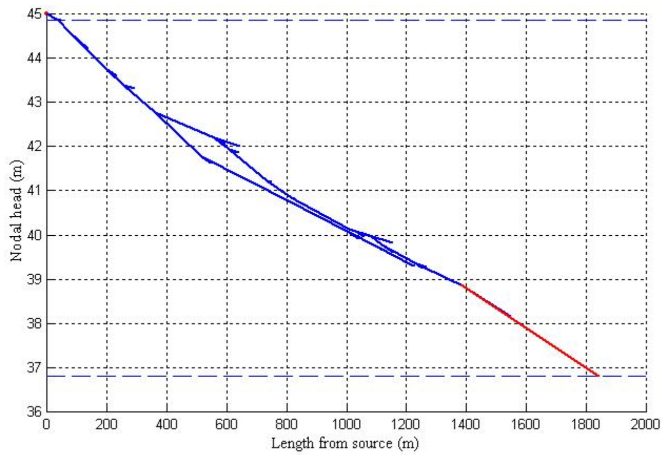

FIGURE 34: h/L diagram for the district heating system

Using the Pipelab software, many district heating parameters can be obtained, such as nodal heads, water flow rates and velocities in the pipes, length and diameter of the pipes, and pipe heat losses. Figure 35 shows the diameter and length of the pipes in the supply network. The biggest diameter in the network is 0.3 m and the smallest diameter is 0.065 m. The number of pipes in the supply network is 47 and their total length is 3525 m. Pressure drop was calculated as 8.18 m for the supply network. The results are the same for the return network. Figure 36 shows water velocity in the pipes, with a range of 0.44-1.51 m/s, while Figure 37 shows the relationship between the pipe diameter and water velocity in the pipe.

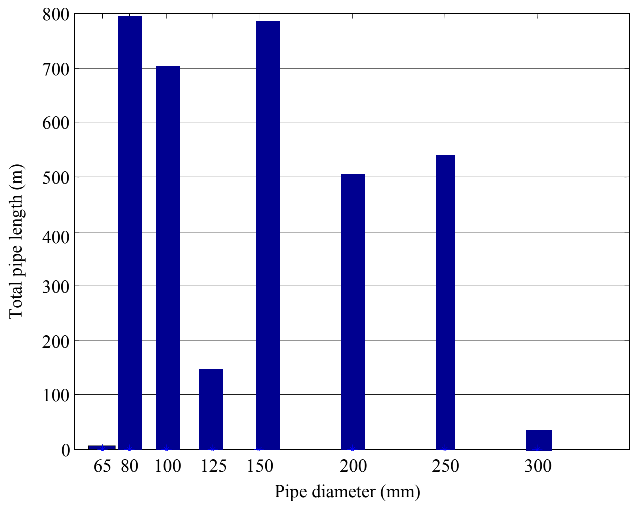

FIGURE 35: Pipe diameter histogram of the network

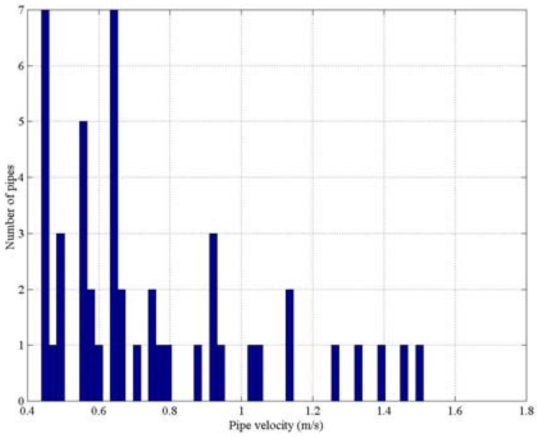

FIGURE 36: Pipe velocity histogram of the network

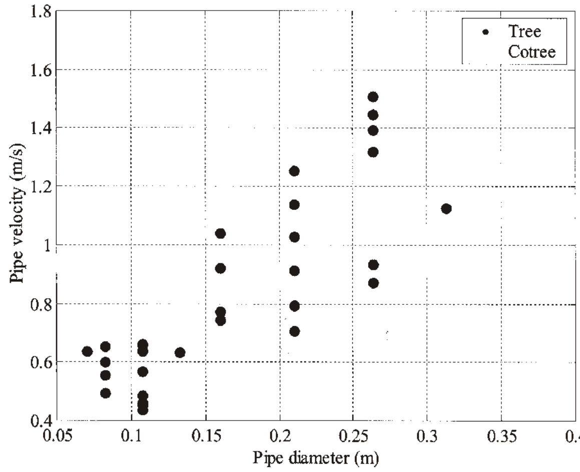

FIGURE 37: Relationship between pipe diameter and velocity of water

7. CONCLUSIONS

In this project many district heating alternatives were looked at for different kinds of heating systems. Comparing the buildings indoor conditions and the system operation costs of these alternatives, the best heating alternatives were determined and heating recommendations given to improve system performance and efficiency.

- According to the results, intermittent heating systems yield poor indoor conditions. Required heat of buildings cannot be obtained by running a heating system only during working hours. To achieve comfortable indoor conditions, heating should be started much earlier than start of working hours. That means big system capacity is needed, which also means more consumption in this kind of a heating system.
- Semi-intermittent heating systems with geothermal energy give better results than intermittent heating systems. They have less operation cost and provide better indoor conditions not only for the working period but year round compared to the intermittent heating. The system capacity, is also smaller because of the good indoor conditions.
- Using larger radiators improves the indoor conditions of buildings and reduces operation cost and the needed capacity of the system. Radiator design temperature is very important for heating systems. For this reason, radiators should be selected for low supply and return temperatures for new buildings, and the radiator sizes in existing buildings should be increased.
- Continuous heating with only geothermal energy is the best heating system with has the lowest operation cost and acceptable indoor conditions. In this project, the geothermal resource temperature was assumed at 50°C with necessary flowrate, and temperature drop along the pipeline was omitted. Exact values for the geothermal source will reveal whether it is possible to use this heating system.
- Finally, a semi-intermittent indirect evaporator heat pump heating system is found to be one of the best heating systems. To reduce operation cost and the capacity of the system, low supply temperature and low flowrate during working hours should be used for this heating system.

In this project, the simulations were based on assumptions. Further study is recommended with extensive simulations for each building with calculated real values. The results of this can help determine the best heating system for the buildings, dependent on capital cost and system efficiency. Suitable building construction characteristics, insulation type and thickness, and radiator design temperatures should be determined for future buildings for improved indoor conditions and reduced system operation costs.

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REFERENCES

Demirdokum, 2002: *Panel radiators*. Demirdokum internet website, http://www.demirdokum.com.

Emeish, M.E., 2001: *Simulation of heating systems in Jordanian buildings*. University of Iceland, M.Sc. thesis, UNU Geothermal Training Programme, Iceland, report 1, 91 pp.

F-Chart Software, 2002: *EES, Engineering equation solver*. F-Chart Software internet website, http://www.fchart.com/ees/ees.shtml.

Geothermal Energy Network, 2002: *Direct uses & electricity production: Collection of technologies.* Internet website, http://www.geotermie.de/egec-geothernet/geo_tech/geothermal_technologies_annex.htm/

Harrison, R., 1997: *Design and performance of direct heat exchange geothermal district heating schemes.* Geothermal District Heating Schemes, 12, 14 pp.

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

Iztech, 2002: *General information*. Izmir Institute of Technology internet website, http://www.iyte.edu.tr/geninfo.htm.

Kevin, D., Rafferty, P.E., and Gene, C., 1991: Heat exchangers. In: Lienau, P.J., and Lunis, B.C. (editors), *Geothermal Direct Use Engineering and Design Guidebook*. Geo-Heat Centre, Institute of Technology, Klamath Falls Or, 247-261.

Lienau, J., 1981: *Geothermal district heating analysis guide*. Oregon Institute of Technology, Geo-Heat Center, Klamath Falls, Or, 75 pp.

Lund, J.W., 2001: Geothermal district heating systems in the United States. In: *European Summer School on Geothermal Energy Applications*, The International Geothermal Training Centre of the University of Oradea, Romania, 253-259.

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

Petrol Ofisi, 2002: *Petrol Ofisi,* internet website, http://www.petrolofisi.com.tr

TEDAS, 2002: *Turkish Electricity Distribution Corporation,* internet website, http://www.tedas.gov.tr.

The MathWorks, 2002: *Matlab program*. The MathWorks internet website, http://www.mathworks.com/products/.

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

Valdimarsson, P., 1995: Flow controller assumptions in physical models of geothermal district heating systems. *Proceedings of World Geothermal Congress 1995, Florence, Italy, 3,* 2187-2192.

Valdimarsson, P., 2001: Pipe network diameter optimisation by graph theory. *Water Software Systems: Theory and applications, 1,* 13 pp.

APPENDIX I: INDOOR TEMPERATURE ESTIMATES AND DURATION CURVES

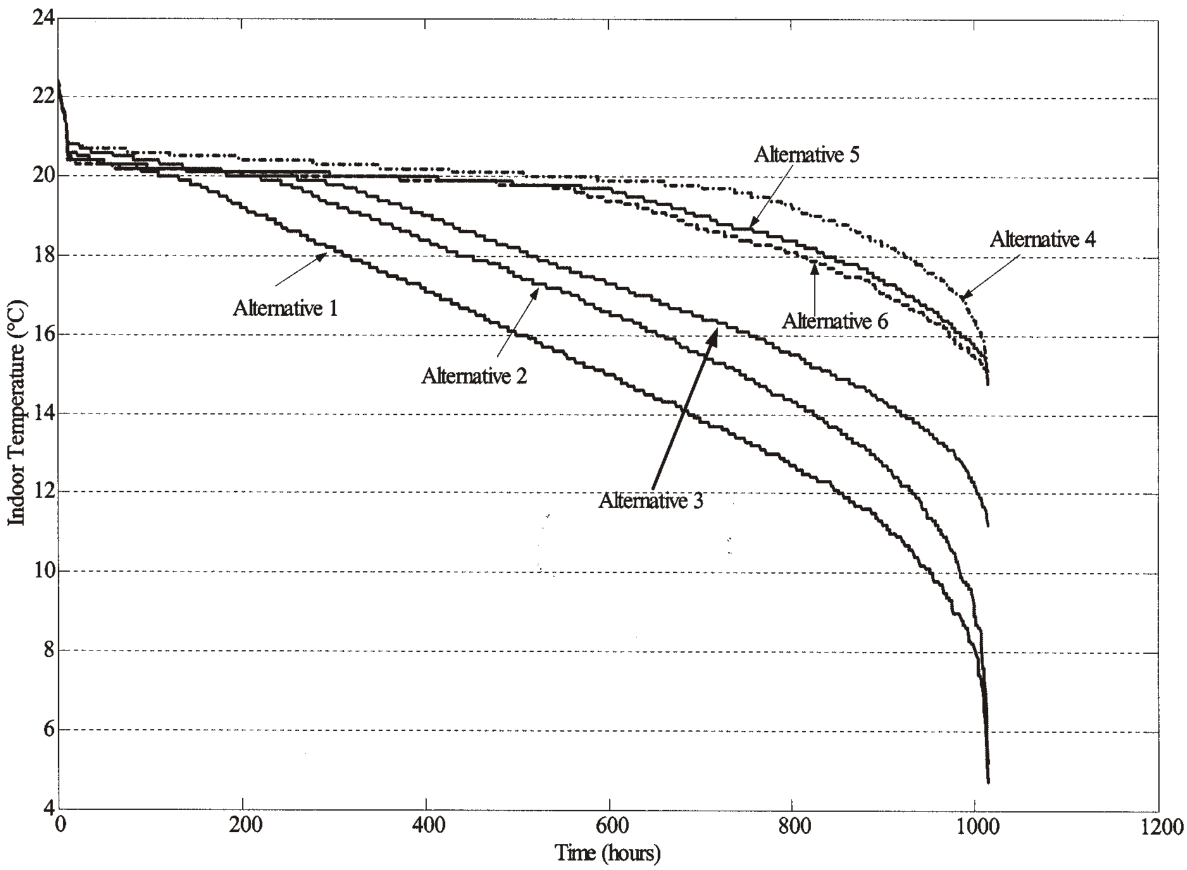

FIGURE 1: Indoor temperature duration curves during the working hours for the heating alternatives

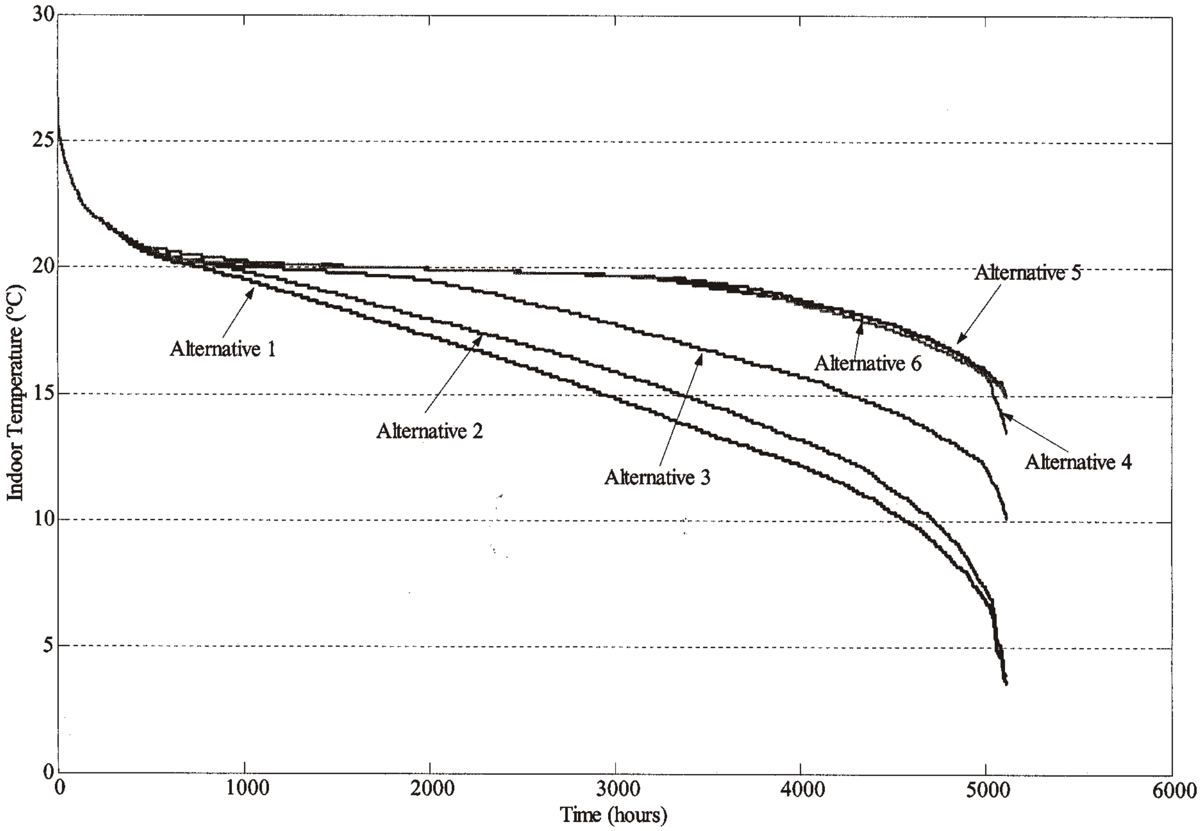

FIGURE 2: Indoor temperature duration curves for all year for the heating alternatives

TABLE 1: Estimated indoor temperature results for the different alternatives

Indoor temp. (˚C)	**Working hours**						**All year**					
	Alt.1	**Alt.2**	**Alt.3**	**Alt.4**	**Alt.5**	**Alt.6**	**Alt.1**	**Alt.2**	**Alt.3**	**Alt.4**	**Alt.5**	**Alt.6**
22	2	2	3	3	3	3	192	192	195	195	195	195
21	7	7	6	6	6	6	162	163	181	187	187	185
20	99	172	190	497	285	244	337	465	530	1102	1052	986
19	83	108	136	233	280	292	429	510	1082	1714	1869	1779
18	82	94	104	89	124	131	386	454	499	616	462	465
17	91	107	107	50	76	84	407	458	444	322	311	339
16	82	84	107	21	50	64	382	412	440	183	215	244
15	83	83	81	6	25	39	339	367	357	62	106	141
14	80	74	70	1		1	327	328	309	51		2
13	78	58	52				332	315	225	17		
12	64	40	23				324	258	149			
11	51	28	10				265	164	61			
10	37	18					191	157	37			
9	22	15					145	123				
8	17	9					126	81				
7	7	3					86	75				
6	3	1					48	24				
5	2	2					14	27				
4	1						34	15				
3							19	7				

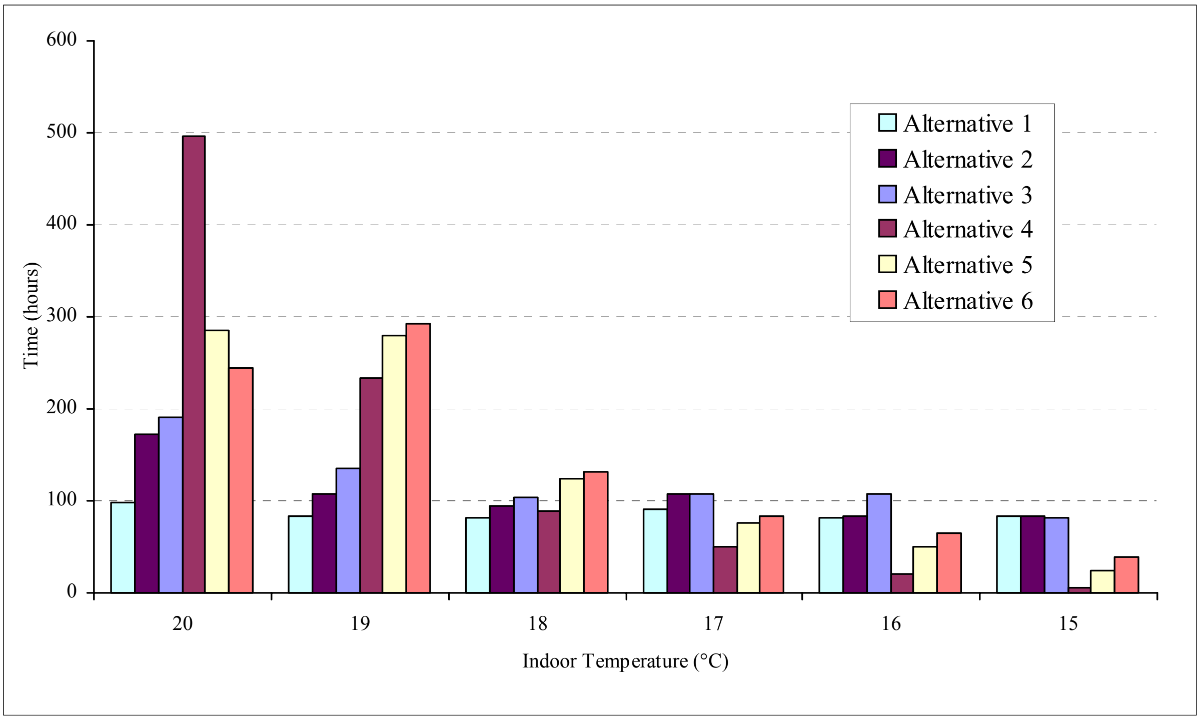

FIGURE 3: Results of estimated indoor temperature during working hours for the different alternatives

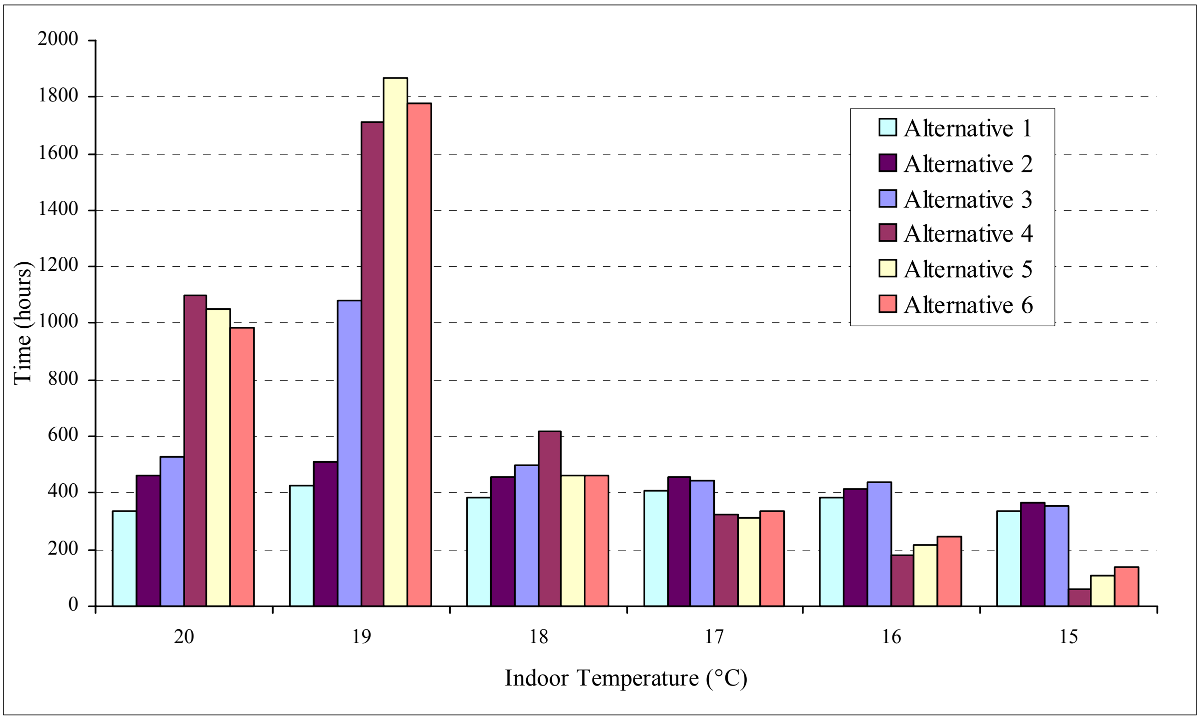

FIGURE 4: Estimated indoor temperature results for all year for the different alternatives

APPENDIX II: NETWORK MODELLING RESULTS

Pipe	Node		L	Diam	Type	Flow	Velocity	Head loss		Node	Head	Load
	1	2	(m)	(m)		(kg/s)	(m/s)	(Pa)	(Pa/m)		(m)	(kg/s)
1	1	2	39.3	0.3127	DN300	83.49	1.125	1348.2	34.31	1	45	
2	2	3	7.2	0.1071	DN100	4.22	0.485	180.85	25.12	2	44.87	
3	2	4	35.4	0.263	DN250	79.27	1.511	2645.64	74.65	3	44.85	-4.22
4	4	5	63.6	0.0825	DN80	3.37	0.653	3863.92	60.72	4	44.6	
5	4	7	123	0.263	DN250	75.9	1.446	8439.3	68.61	5	44.21	
6	5	6	2	0.0825	DN80	3.37	0.653	122.64	60.72	6	44.2	-3.37
7	7	8	33.2	0.0825	DN80	2.87	0.556	1487.91	44.78	7	43.76	
8	7	9	58.2	0.263	DN250	73.03	1.392	3707.38	63.67	8	43.61	-2.87
9	9	10	34.6	0.1071	DN100	3.82	0.439	719.81	20.82	9	43.39	
10	9	11	107.6	0.263	DN250	69.21	1.319	6173.66	57.37	10	43.31	-3.82
11	11	38	151.6	0.1603	DN150	20.27	1.04	9978.33	65.82	11	42.77	
12	11	12	193.9	0.263	DN250	48.94	0.933	5698.26	29.39	12	42.2	
13	12	14	22	0.263	DN250	45.83	0.873	569.38	25.9	13	41.98	-3.11
14	38	39	31.8	0.1071	DN100	5.73	0.658	1427.15	44.84	14	42.14	
15	12	13	41.4	0.0825	DN80	3.11	0.602	2158.79	52.13	15	42.01	-3.82
16	38	40	295.8	0.1603	DN150	14.54	0.746	10294.64	34.8	16	41.94	
17	14	16	29.1	0.2101	DN200	42.01	1.254	1984.15	68.21	17	41.88	-3.82
18	14	15	64.9	0.1071	DN100	3.82	0.439	1351.76	20.82	18	41.24	
19	40	41	190.7	0.1603	DN150	14.54	0.746	6636.32	34.8	19	41.2	-3.82
20	41	43	9.9	0.1071	DN100	4.02	0.462	228.06	22.92	20	40.85	
21	16	17	32.4	0.1071	DN100	3.82	0.439	674	20.82	21	40.83	-3.82
22	41	42	38	0.1071	DN100	5.58	0.641	1621.67	42.63	22	40.14	
23	41	44	101.8	0.1071	DN100	4.94	0.568	3442.27	33.83	23	39.83	-3.94
24	16	18	124.8	0.2101	DN200	38.19	1.14	7080.26	56.73	24	40.08	
25	44	45	7.9	0.1071	DN100	4.94	0.568	267.94	33.83	25	39.97	-2.87
26	18	19	16.6	0.1071	DN100	3.82	0.439	346.27	20.82	26	39.96	
27	18	20	82.7	0.2101	DN200	34.37	1.026	3830.34	46.29	27	39.93	-5.75
28	45	46	7	0.0703	DN65	2.39	0.637	497.81	70.71	28	39.9	
29	20	21	10.7	0.1071	DN100	3.82	0.439	222.17	20.82	29	39.77	
30	45	47	104.9	0.0825	DN80	2.55	0.494	3756.13	35.82	30	39.61	-2.87
31	20	22	192.6	0.2101	DN200	30.55	0.912	7108.18	36.9	31	39.35	
32	47	48	9	0.0825	DN80	2.55	0.494	320.92	35.82	32	39.26	-2.87
33	22	24	22.9	0.2101	DN200	26.61	0.795	647.91	28.32	33	39.28	-3.82
34	22	23	143.1	0.1071	DN100	3.94	0.453	3157.06	22.07	34	39.02	
35	24	25	24.6	0.0825	DN80	2.87	0.556	1103.73	44.78	35	38.87	
36	24	26	52.7	0.2101	DN200	23.74	0.709	1200.66	22.77	36	38.17	-5.56
37	26	27	5.3	0.1071	DN100	5.75	0.661	240.57	45.13	37	36.81	-2.87
38	26	28	11.2	0.1603	DN150	17.99	0.923	586.03	52.32	38	41.77	
39	28	29	23.9	0.1603	DN150	17.99	0.923	1251.08	52.32	39	41.63	-5.73
40	29	30	36.1	0.0825	DN80	2.87	0.556	1618.66	44.78	40	40.74	
41	29	31	113	0.1603	DN150	15.12	0.776	4238.6	37.5	41	40.08	
42	31	32	20.6	0.0825	DN80	2.87	0.556	921.05	44.78	42	39.92	-5.58
43	31	33	31.7	0.1071	DN100	3.82	0.439	660.47	20.82	43	40.05	-4.02
44	31	34	103.3	0.1325	DN125	8.43	0.633	3312.3	32.07	44	39.73	
45	34	35	45.1	0.1325	DN125	8.43	0.633	1447.83	32.07	45	39.71	
46	35	36	167.1	0.1071	DN100	5.56	0.639	7076.4	42.34	46	39.66	-2.39
47	35	37	460.5	0.0825	DN80	2.87	0.556	20619.43	44.78	47	39.33	
										48	39.3	-2.55