DESIGN OF A DISTRICT HEATING SYSTEM FOR THE HAMMAM RIGHA SPA, ALGERIA

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ABSTRACT

Experience has shown that geothermal energy has a large field of applications: electricity production, aquaculture (fish farming), industrial drying, greenhouse heating and district heating. The present project deals with this last application. The main aim of the study is an attempt to master a methodology for the design of district heating system. The district system considered as an example is a group of bungalows situated in a geothermal area in Algeria. The geothermal source gives 67°C hot water. The power demand of the system has been evaluated through the analysis of the weather data of the area and calculation of the heat loss of the buildings. A district heating system has been designed on basis of the results. A cost estimation of the system indicates reasonable heating cost for the spa.
The study also includes a dynamic simulation of the system in order to predict the indoor temperature during a simulated cold spell with a duration of few days.

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INTRODUCTION

The supply of energy can be a heavy financial constraint for some countries. This have been the case in the seventies when the prices of the oil increased. This fact have been leading to a more rational utilization of the oil and opening a large field of research on new forms of energy (solar and wind energy, wave forces etc..). Among the resources is the geothermal energy which has been used in many countries such as Iceland, New Zealand and Italy for example.
The geothermal energy is classified in high temperature or low temperature geothermal field according to the temperature of the fluid harnessed. The difference of these two forms of energy is in their application. Usually the high energy is converted to electricity and low energy is used directly in space heating. So if geothermal energy is available it can well be as efficient as other form of energy.

In Algeria 117 principal geothermal springs have been accounted during a recent preliminary exploration survey (Kedaid, Rezig, Abouriche and Fekraoui, 1988). The temperatures are ranging from 22°C to 98°C. The highest temperature encountered is 98°C.
In the last years a great interest has been shown to this alternative energy source for some local applications, mainly in space heating (greenhouses). And in order to study the possibility of other applications, the present work has its emphasis on district heating systems.
The system we are interested in is to design a heating system for bungalows situated in "Hammam Righa" geothermal area; where a geothermal source is giving out 67°C hot water intended for baths and swimming pools. The main lines of this project are:
- in a first chapter a description of the system is presented,
- the second chapter is concentrated to the study of the power demand. The climate data of the area are analyzed and

the heat requirements based on the type and size of the buildings is carried out.

- based on the results obtained above a design of the system is accomplished and summarized in the third chapter:
- pipe network calculation
- heat exchanger calculation
- radiators design
- chemistry of the water and choice of material
- system control
- economic estimation.
- In the fourth and last chapter a dynamic simulation of the system is accomplished.

1 DESCRIPTION OF THE SYSTEM

The geothermal area under consideration is nowadays used mainly for bathing and hydrotherapy. A lot of people enjoy the different possibilities offered by the spa (bathing, massage, and treatment of different disease). The spa receives an average of 300 persons daily. People who visit the spa stay at the nearby hotel and bungalows.

During the winter the bungalows are heated by gas which actually present some inconveniences, mainly transport. Finding another way to heat the bungalows is desirable. The most convenient and practical mean which can replace the gas is the utilization of the geothermal water owned by the spa.

1.1 Water supply

The geothermal water is delivered by two wells HR1 and HR3, 66 and 35 meters deep respectively.The temperature of the thermal water is 67°C and the flow rate for each well is 4 l/s.

The temperature of the water used in the baths is in the range 40 - 36°C. Using a heat exchanger to cool the thermal water down till a temperature close to the temperature expected in the baths, seem to be an adequate way to produce energy to heat the bungalows (Fig.1.1). Nowadays the geothermal water is cooled by a refrigerating system.

1.2 Capacity of the system to be heated

The total number of bungalows to be heated is 112, separated in 16 groups of 7 bungalows each one. They are situated at approximately 300m from the wells. Location of the houses and the situation of the wells are shown on (Fig.1.2).

2 POWER DEMAND

When calculating a heating system for a part of a town or a group of houses, the first parameter to evaluate is the power demand of the houses.

A power demand Pd of a single house is determined by its size and by its heat loss characteristics and by the effective temperature difference across the building's fabric after adjusting for incidental gain (if neglected, the incidental gain will lead to a slightly overdimensioning of the system).

$$Pd = Vh * G\ (Ti - To) \qquad (2.1)$$

Where G is the total heat loss per unit volume (m^3) and degree centigrade (see chap.2.2).
Since the indoor temperature Ti is constant , Pd is a function of the external air temperature To . Information about To are given by the local climate data.

2.1 Power demand based on climate data analysis

In order to estimate energy consumption, statistic of the daily mean temperature of the area are necessary. These temperatures can be used to provide temperature-time duration curves which by using (2.1) can be converted into thermal power demand-duration curve (Karlsson T., 1982, Harrison R., 1987).

2.1.1 Temperature-time duration curve (Degree days concept)

If Ti is the indoor temperature of a building, and if we assume that the external temperature is constant and equal to To during one hour, the quantity of energy to produce in this space during one hour is:

$$pd = G * Vh * (Ti - To) * 1hour \qquad (2.2)$$

for one day To is replaced by Tom :

$$pd = 24 * G * Vh * (Ti - Tom) \qquad (2.3)$$

for many days:

$$pd = 24 * G * Vh * [\ (Ti-Tom1)+(Ti-Tom2)+....]$$

If N is the total number of days over the year, for which the heating is necessary:

$$Pd = 24 * G * Vh * \sum_{j=1,N} (Ti-Tomj) \qquad (2.4)$$

$$\Sigma(Ti-Tomj) = DD(Ti) = N * (Ti-Tomt)$$

DD(Ti) is the number of degree days; the degree days are defined as the number NT of days out of the year, for which the daily mean temperature is below a given temperature and multiplied by the difference between the given temperature and the mean temperature for these days (Karlsson T., 1982).

$$DD(T) = NT * (T - Tomt) \qquad (2.5)$$

2.1.2 Hammam Righa climate data analysis

Ten years measurement (1975 - 1984) of the daily mean temperature are given in a table which gives the mean temperature of each 5 days during each month of the year. Table 2.1 show the frequency of the daily mean temperature of the region which allow the drawing of the temperature-time duration curve (Fig.2.1). This curve shows that for 143 days the external air temperature is below 12°C. The total number of degree days over the year for a reference temperature of 20°C (inside temperature) is:

$D(20) = 1983.5$ degree days

The total power demand of one house is then :

$$Pdt = 24 * 1983.5 * G * Vh$$

for Nb identical houses heated by the same heating system

$$Pdt = 24 * 1983.5 * G * Vh * Nb$$

The objective of heating systems is to assure in the houses a constant internal temperature whatever is the external temperature. So, it is not only the daily mean temperature which decide the capacity of a district heating system, but mainly the temperature of the particular days for which the temperature is below the daily mean temperature : "cold waves", this theme will be discussed in chap.4.

2.2 Heat requirements based on size and type of building

The heat requirements, which are equivalent to the heat loss of the building depends simultaneously:
- on the atmospheric conditions which are represented by the external temperature,
- on the temperatures of the spaces adjacent to the space to be heated,
- on the temperature wished in the heated rooms,
- on the dimension and the nature of the walls of the space to be heated, on their location and orientation,
- and finally, on the air renewing occurring in each room; this is function of the bad tightness of the windows and doors, of the proportion of glass in the windows and of their situation and orientation. The air renewed is also function of the ventilation opening.

The calculation of all these parameters is described in the

(Document Technique Unifie,Regles Th, French standards, 1975). This document has been used due to the similarity of the building materials used in France and in the houses considered in this project and to the fact that french climate is close to the climate of the area of our interest.

In the following discussion, a description of the different walls of the houses and the calculation of the different heat transfer coefficient are presented.

2.2.1 Description of the different walls

* The houses which must be heated are 112 bungalows separated in 16 identical group of 7 bungalows. So all the calculation will be done for one group of 7 bungalows.

A description of the exterior walls, roofs, floors and their geometry are given in (Figs. 2.2, 2.3, 2.4 and 2.5). These figures show also the dimensions and material of the walls. And (Tab.2.2) summarize the different materials used and their thermal conductivities or thermal resistivities.

* Windows and doors

All the windows and doors of the rooms are facing to the internal terrace.

- Doors (made of wood):
- height: 2.3 m
- with: 0.80 m

* Windows (French windows), single glass, 20 % wood:
- height: 2.3 m
- with: 1- 0.80 m
        2- 1.50 m

2.2.2 Calculation of the heat transfer coefficients for the different walls

The total heat loss of a building to the outside is given by:

$$Q = (\Sigma (U * A + Ul * L) + qi) * (Ti - To) \quad (2.6)$$

* For the exterior walls (outside walls and roofs):

$$1/U = 1/hi + 1/ho + \Sigma t/k \quad (2.7)$$

* For the floors (floor on sanitary space, French Standards)

$$1/U = 0.29 + 1/(1.6 + 5.18 * Lex/Ai) \quad (2.8)$$

* Heat loss through the corners
The linear heat transfer coefficient for the corners is calculated according to (French Standards, DTU, 1975)

* Windows
Mean values of the heat transfer coefficient for house windows are given in (DTU, 1975); these mean values take into account the variation of the thermal conductivity during the day and the night, and the possibility of the curtains and jalousie to be open or closed.
For a single glass window with wood frames:

$U = 4.2 \quad [W/m^2 \cdot C]$

* Correction of the surface heat transfer coefficient
The houses considered in this project are situated in country side and on slightly high hills, the standards used to calculate the heat loss advise in this case to correct the values of the heat transfer coefficients which are greater than 2 $W/m^2 \cdot C$, by subtracting from their thermal resistance 0.02 $m^2 \cdot C/W$. Table 2.3 summarize the over all heat transfer coefficient values for the different walls, U(floor) is calculated according to (2.8) for each room. Detailed calculations of the U values are given in appendix 1.

* Heat loss due to air infiltration
The heat loss due to the fresh air entering the rooms is given by:

$$qi = 0.34 * V \qquad (2.9)$$

In general it is assumed that the total volume of air renewed per hour is equal to the volume of the room.
However for accurate calculation V is given by:

$$V = Vg + \Sigma (P * e) \qquad (2.10)$$

- Σ (P * e): sum of (P*e) for all the exterior walls.
For one exterior wall:

$$P = 0.25 * Ao + \Sigma (m * Am) \qquad (2.11)$$

No information are given about the ventilation orifices, so this term is omitted.

Hence, for each room the total volume renewed is:

$$V = Vg + \Sigma (\Sigma (m * Am) * e)$$

The permeability per m^2 m for the windows and doors considered is given by tables in the (DTU, French Standards):
- windows: m = 4
- doors: m = 6

Tables available in (DTU, French Standards, 1975) give the coefficient of exposure to the wind e. Values of e are given as a function of the class of exposure to the wind and the type of ventilation of the houses. There are four classes of exposure to the wind (Ex), depending on the fact that the houses are situated in cities (protected) or in the country side, on the distance to the sea and the height of the windows compared to the ground level.
In the present case: e = 2.4

2.2.3 Heat loss calculation

On the basis of the parameter described and the formulas given above, a heat loss calculation room per room for one group of 7 bungalows has been done. The results are summarized below:

- Total heat loss:

$$Q = 54.21 \text{ kW}$$

- Total floor area:

$$A = 264.5 \text{ m}^2$$

- Total volume :

$$Vh = 822.65 \text{ m}^3$$

- Heat loss per m^2 of living floor:

$$Q/A = 0.205 \text{ kW/m}^2$$

- Heat loss per m^3 of volume house:

$$Q/Vh = 0.066 \text{ kW/m}^3$$

$$G = Q/(Vh*(Ti-To))$$

$$G = 0.0033 \text{ kW/m}^3 \text{°C}$$

* The total heat loss for the 16 groups of bungalows is:

$$Qt = 867.4 \text{ kW}$$

The heat losses have been estimated for an exterior temperature To = 0°C and an internal temperature Ti = 20°C. then, in order to keep the temperature in the rooms at 20°C when the exterior temperature is 0°C, a heating system which can deliver a total power of 867.4 kW during the heating season is designed; the details of calculation are presented in the next chapter.

3 HEATING SYSTEM DESIGN

The heating system intended is a closed loop pipe system (Fig.1.1) : the hot water coming out of the heat exchanger is carried to the houses through a supply water pipe lines and pumped back from the outlet of the houses to the heat exchanger through the return water lines. the network pipe line is shown on (Fig.3.1).

3.1 Pipe network

From drawing giving the disposition of the bungalows in the area an estimation of the length of all the pipes has been accomplished.

3.1.1 Flow rate in each pipe

The total energy extracted from the water is given by :

$$Q = mf*cp*(Tfs - Tfr) \qquad (3.1)$$

- the temperature of the supply water at the house inlet is $Tfs = 60°C$.
- the temperature of the return water at the house outlet is $Tfr = 39°C$.

The total flow rate is given by:

$$mf = Qt/cp*(Tfs - Tfr)$$

So the flow rate in the first main pipe coming from the heat exchanger is: 9.87 kg/s.
The flow rate in the houses line is the necessary flow rate for each group of 7 bungalows, and is given by:

$$mfh = Q/cp*(Tfs - Tfr)$$

$$mfh = 0.617 \text{ kg/s}$$

When the flow rates in the branches are known, it is possible by back step calculations to determine the flow rate in all the pipes (see appendix 2).

3.1.2 Diameter of the pipes

The diameter of the pipes has been calculated with following assumptions :
- velocity < 1 m/s
- the pressure drop in a straight section of pipe at maximum rate of flow is less than 150 Pa/m.

The flow rate can be written as:

$$mf = Ro*v*a = Ro*v*\pi*D^2/4$$

$$D = \sqrt{(4*mf/(Ro*v*\pi))} \qquad (3.2)$$

All values of required diameters for each pipe are calculated according to (3.2). Then the real pipe diameter which will be used are next standards diameters (German standards DIN 2440) close to the required diameter.

3.1.3 Pressure drop in the pipes

In each pipe the pressure drop is caused by two factors: friction and fitting, and is given by:

$$DP = (Ro*v^2/2)*(f*L/D+Zj) \qquad (3.3)$$

- Zj is the sum of the fittings pressure drop factors. These fittings are : bends, connecting T and valves.

The number of bends, connecting T and valves for each pipe have been evaluated. Figure 3.2 show a schematic drawing of the heating system.
According to (3.3) an estimation of the pressure drop in each

pipe has been carried out.The results obtained are used to calculate the total pressure drop occurring in the supply and return line for each house.With the condition that the pressure drop allowed in each house is 1bar.
Hence, the total pressure needed to bring the water to each house and to return it back to the heat exchanger is:

DP = DP(supply line) + DP(return line) + 1 bar + DP(heat exchanger)

The results show that the biggest pressure drop is the one corresponding to the house H11, DP ~ 2 bars.
So to assure a flow circulation in the system it is necessary to provide it with a 2 bars pressure head pump.

3.1.4 Capacity of the water tank included in the network

In order to avoid any cavitation in the pump a water tank is included upward of it.As the total flow rate in the main pipe is 9.87 l/s,it is estimated that the capacity of the tank, is approximately the pumping capacity during two minutes. The inlet and outlet pipes must be situated at the bottom of the reservoir.

Capacity = 2*60(s)*9.87(l/s) ~ 1000 l

3.2 Heat exchanger calculation

3.2.1 Type of heat exchanger

All the heat exchanger used in the traditional applications may be used in geothermal exploitation, with the condition that the material chosen can stand the nature of the thermal water.

They can be of these different types:

- shell and tubes heat exchanger,
- plate heat exchanger,
- spiral heat exchanger.

The model adopted in this project is: plate heat exchanger. These equipment are constituted by vertical plates assembled together, the spaces between plates are flow channels related alternatively to the primary and secondary fluid. These heat exchanger are very well adapted to geothermal applications: their efficiency may be very high because the heat transfer area might be very big for a small size. They present low pressure drop, this leads to less pumping size. The heat transfer area may be increased by adding the necessary number of plates.

3.2.2 Heat exchanger specifications (present project)

The detail of calculation are given in appendix 3

- Heat duty: 867.4 kW
- Temperatures:

Thi	:	67	°C
Tho	:	42	°C
Tci	:	37	°C
Tco	:	62	°C

- Flow rates:

primary fluid	:	9.87 kg/s
secondary fluid	:	9.87 kg/s

- Heat transfer area: 81.77 m^2
- Number of plates : 132
- Number of passes : 2
- Pressure drop : 40 kPa

3.3 Heating element calculation

This chapter deal with the choice of radiators type and estimation of the heating area (or length) required to heat each typical room.

3.3.1 Heating element choice

The heating elements considered in the bungalows consist of long section of finned steel pipes installed beneath windows or along outside walls. A typical size considered consist of 1 inch (25.4 mm) pipes with circular fins 15 mm height and 1.5 mm thick. The distance between two fins is 5 mm (Fig.3.3).

3.3.2 Calculation of the heating element length

The total area or total length of the pipes are function of the total quantity of heat Q to each room, the geometrical and physical characteristics of the pipes, the temperature of the water flowing in the tube and the overall heat transfer coefficient of the finned pipes.
The quantity of heat delivered in the rooms is given by:

$$Q = U*A*LMTD \qquad (3.4)$$

3.3.2.1 Logarithmic mean temperature difference: LMTD

If we assume that the heating elements are acting counter flow heat exchanger the LMTD can be expressed by:

$$LMTD = ((Tfs- Ti)-(Tfr-Ti))/Ln((Tfi-Ti)/(Tfo-Ti)) \qquad (3.5)$$

3.3.2.2 Radiators overall heat transfer coefficient

From the heating elements, the heat flux is transferred to the ambient by free convection and by radiation.
For a finned pipe the free convective thermal resistance is given by (Hodge B.K, 1985):

$$R = 1/(A*Et*hc) \qquad (3.6)$$

$$Et = 1-(Af/A)*(1-Ef) \qquad (3.7)$$

The quantity of heat transferred by convection is:

$$Qconv = A*Et*hc*(tw-Ti) \qquad (3.8)$$

The thermal radiation is assumed to occur through a fictitious cylindrical surface which envelopes the bare tube and the fins. The thermal radiative heat flux is given by:

$$Qrad = Ae*e*s*(Tw^4-Ti^4) \qquad (3.9)$$

If hr is defined as a radiative heat transfer coefficient the radiative heat quantity can be written as:

$$Ae*e*s*(Tw^4-Ti^4) = A*hr*(Tw-Ti) \qquad (3.10)$$

Which leads to:

$$hr = (Ae/A)*e*s*(Tw+Ti)*(Tw^2+Ti^2) \qquad (3.11)$$

The radiative thermal resistance is given by:

$$1/(hr*A) = 1/(Ae*e*s*(Tw+Ti)*(Tw^2+Ti^2)) \qquad (3.12)$$

The overall heat transfer coefficient for a finned pipe, heat by convection and radiation is:

$$U*A = 1/(1/Ai*hi + Rfi/Ai + Ln(ro/ri)/(2*\pi*b*ks*n) + Rfo/A + 1/A*(Et*hc+hr)) \qquad (3.13)$$

3.3.2.3 Inside convective heat transfer coefficient

Calculations of the velocities inside the pipes (Tab.3.1) show that the velocities are very small, and the Reynolds numbers are in the range of laminar flows.
Heat transfer in laminar tube flow, for constant heat flux, is governed by the following Nusselt number (Holman J.P., 1989):

$$Nu = hi*Di/kw = 4.364 \qquad (3.14)$$

3.3.2.4 Outside convective heat transfer coefficient

For free convection from horizontal cylinder the convective heat transfer coefficient is function of the parameter Gr*Pr:

$$Gr*Pr = g*B*(Tw-Ti)*Do^3*Pr/ny^2 \qquad (3.15)$$

The parameters ny, B, and Pr are evaluated at the film temperature Tf.
The equation (3.15) leads to Nusselt number:

$$Nu = 0.53*(Gr*Pr)^{1/4} \qquad (3.16)$$

$$Nu = hc*Do/ka \qquad (3.17)$$

The convective heat transfer coefficients hi, hc, hr, the temperatures Tw, Tf and the efficiencies Et and Ef are calculated in appendix 4.

3.3.2.5 Evaluation of the necessary pipe length to heat each room

The total length of each radiator can be calculated by the product of the number of fins times the distance between two fins.

$$L = b*n \qquad (3.17)$$

An expression of the number of fins n can be found from the equation

$$n = (Q/LMTD)*((1/hi+Rfi)/0.000352 + Ln(ro/ri)/(2*p*ks*b) + (1/(Et*hc+hr) + Rfo)/0.00248) \qquad (3.18)$$

The results are summarized in (Tab.3.1) which shows for each room of a typical group of 7 bungalows the following quantities:

- total heat demand Q [w]
- necessary flow rate mf [l/mn]
- velocities in the pipes v [m/mn]
- heat transfer area A [m^2]
- necessary length of the fin-pipe L [m]

3.4 Estimation of the steel pipe thermal expansion

The steel pipes used in the hot water distribution system might be subject of expansion due to the effect of the thermal gradient when 62°C hot water start flowing in the pipes .

The ratio of expansion of the steel is in the order of 0.012mm/m°C.

In general it is necessary in all hot water lines of steel to provide expansion compensators to enable the pipes to move in a longitudinal direction.

In (Karlsson T., 1982) an inventory of the different expansion compensators used in district heating systems is given. An estimation of some pipes expansion of this project shows that for pipes varying in length from 173.5 m (first main pipe Fig.3.1) to 3.2 m (smallest house connection) the expansions are from of 87.4 - 1.6 mm.

The values given above show that, the expansion length of the pipes are quite small. However consideration of expansion compensators in the system, must be studied carefully.

3.5 Chemistry of the water and choice of the material

3.5.1 Chemistry of the water (Tab.3.2)

The scaling potential was evaluated by use of the WATCH computer Geochemical Model of ORKUSTOFNUN.
The chemical interpretation of the geothermal water composition shows that:
- the thermal water is saturated with calcite at measured temperature (67°C).At lower temperature the water becomes under saturated, due to the reverse solubility of calcite. thus calcite scaling is not expected during cooling in the heat exchanger.
- the water is also near saturation with fluorite and anhydride, hence scales of these minerals are not expected.
- the silica concentration (~ 48 mg/l), is relatively low and amorphous silica equilibrium is not reached during cooling. Therefore silica scales are not expected.
- The water is under saturated with crysolite and sepiolite and saturated with talc at quartz temperature (89°C). During cooling the water is under saturated and therefore magnesium silicates are not expected to form. Formation of all magnesium silicates are much dependent on pH. If pH is high then it is more likely to have magnesium silicates scales. Therefore, it is not likely that any of these common scales will form in this water during cooling.

In the chemical data available, no measurements of dissolved oxygen are present. The water can be corrosive if dissolved oxygen is present; so it is advisable to make complete analysis about the oxygen contains. Another parameter which can cause corrosion is the low pH of the water (pH = 6.5).

In conclusion it must be pointed out that the water may be corrosive so the heating system materials must be chosen with care. Corrosion and scaling are limited to the supply side of the heat exchanger, as the distribution network contains treated water that is recirculated.

3.5.2 Material choice

The system is constituted by two parts: the secondary side and primary side.

- secondary side: the thermal water is not piped directly to each house. A secondary fluid: water heated up through a heat exchanger, carries the heat through pipelines to the houses. The secondary system is a closed loop.

The material proposed in this secondary system is common steel.

- primary system: it includes the heat exchanger and the pipes carrying the thermal water. For this side, the following materials are proposed:
- fiber glass reinforced (epoxy) tubes might be more convenient than stainless steel pipes; because this last material if it is used under the following three conditions:
- T ~ 70°C,
- presence of chlorite,
- presence of oxygen ,

will lead to cracking in the pipes.

- In the heat exchanger Titanium or stainless steel 316 may be used.

3.6 System control

As it has been introduced in chapter one, the thermal water is intended to baths and swimming pools of the spa. Therefore an important condition must be fulfilled by the system: the temperature of the thermal water at the outlet of the heating system must be in the range of 36 - 40°C. This range of temperature is expected to be respected during the winter time, when the system is operating at full power demand.

However if the weather during the winter gets warmer than the design conditions, the return water temperature from the bungalows T1 (Fig.3.4) will increase. The increase of T1 leads to a higher value of T2 (temperature of the thermal water at the outlet of the heat exchanger supplying the bungalows). In order to control the temperature of the water going to the spa, some equipment have been included in the system:

- a bypass of H.E.1 equipped with valves. The valve V.3 of the bypass must be controlled by a temperature sensor measuring the temperature T1.
- a second heat exchanger H.E.2, which can be in partial or full operation according to the power demand of the bungalows. The flow rate in the secondary side of H.E.2 is given by a flow meter (F.M.2) controlled by the level of the temperature T3 measured by a temperature sensor.

In the summer time H.E.1 is completely closed and then H.E.2 is in full operation; the question which arises is about the cold water system to be used to cool the thermal water. An open system leads to the utilization of a big quantity of cold water which is not available. Hence, an utilization of a closed system seems to be more adequate (if economically possible): the water flowing in the secondary side of H.E.2 can be cooled in a cooling tower or a cooling pond, in this last device the water is cooled in the ponds by spraying the water in fountains from where it is carried to H.E.2 again).

The second heat exchanger may also be replaced by the existing refrigerating system (see chapter 1) if technically possible.

3.7 Economic estimation of the system

3.7.1 Estimation of the capital cost

This chapter deals with the estimation of the total cost of the heating system. Are included in this preliminary estimation:

- the pipe network with all the equipments involved in it: straight pipes, bends, reduction fittings, T-fittings, T-fitting with jump, joints with shells (welding points) and valves.

The type of pipes considered for this study are the habitual distribution pipes used in district heating systems: steel pipe insulated with polyurethane and coated by plastic pipe.

- pumps: two pumps are accounted in the total cost (one reserve pump). The pumps chosen have 2 bars pressure head and 10 l/s maximum flow rate.
- heat exchanger: one plate heat exchanger
- one reservoir which capacity is 1000 l.

The following prices have been obtained:

- pipe network:	5.33 Million	kr
(including the price of labor)		
- valves	399744	kr
- 2 pumps	200000	kr
- heat exchanger	485937	kr
- reservoir	50600	kr

The capital cost of the system is estimated to be:

$$CC = 6.5 \text{ MILLION kr}$$

3.7.2 Annual cost estimation

If the capital cost is a loan from a bank for 30 years with 8% annual interest, the financing annual payment is:

$$AC = CC*K$$

$$K = i/(1-(1+i)**(-n))$$

where:

- $i = 0.08$ is the interest
- $n = 30$ years

$$K = 0.08882$$

$$AC = 577378 \text{ kr}$$

It is also assumed, that the maintenance cost is:

$$MC = 0.01*CC = 65000 \text{ kr}$$

Hence the total annual cost is:

$$TAC = AC+MC = 642378 \text{ KR}$$

3.7.3 Energy cost estimation

According to the results obtained in chapter 2, the annual power demand is:

$$pd = 24*D(20)*Qt/(Ti-To)$$

$$pd = 24*1983.5*867.4/20$$

$$pd = 2065 \text{ kWh}$$

The cost of the energy is:

$$Cp = TAC/pd = 0.31 \text{ kr/kWh}$$

4 DYNAMIC SIMULATION OF THE DISTRICT HEATING SYSTEM

4.1 Design temperature

As it was mentioned in chapter 2, it is not only the daily average temperature which decide the capacity of a district heating system. The cold waves, that mean the most severe weather conditions to which the community is expected to be exposed to must be taken under consideration. Hence, the design temperature of the system must be evaluated so that during these particular cold spells the indoor temperature do not fall under a certain comfort limit. This question on cold spells, both triangular and rectangular shape have been treated by T. Karlsson in his book. A mathematical approach of the problem has led to a differential equation which solutions describe the behavior of the indoor temperature drop as a function of:

- the characteristics of the cold waves
 * profile of the outdoor temperature drop compared to the daily mean temperature.
 * duration of the cold spell
 * minimum outdoor temperature reached during the cold period

- characteristics of the buildings
 * overall heat transfer coefficient
 * heat capacity of the buildings

- the overall heat transfer coefficient of the radiator system.

By defining the maximum value of the indoor temperature drop and following the previous reasoning, the system design temperature can be deduced (for more details see Karlsson T., 1982). However such rigorous estimation of district heating systems capacities are not always followed, because the computation can be very time consuming.

In the present project the system has been designed for 0°C outside temperature. A determination of the design temperature as described above requires climate data about a typical cold period to which the area has been exposed to. These data were not available in this case. However, the data which have been used show that the minimum mean temperature of the coldest month in the area is about 5°C. Hence, 0°C design temperature might lead to a slightly overdimensioning of the system which had required more geothermal water than what is available and generally needed.

4.2 Dynamic simulation

In order to observe how the system can behave during a varying outside temperature, a cold wave have been simulated by using Icelandic weather data for cold spell adjusted to Algerian climate.
The purpose of this dynamic simulation is to find the change in indoor temperature due to changing in outdoor temperature. A computer programme developed at the University of Iceland (SIHEAT) (Valdimarsson P., Jonsson V.K., 1989) which integrates the continuity equation, momentum equation and energy equation for the whole district heating system by using outdoor temperature distribution as shown in(Fig.4.1). The simulation considers only a part of the system consisting of 7 buildings (49 apartments) and the total flow of hot water is restricted in order to get good demonstration the exhaustion of the hot water supply.
The results obtained show that, with a severe simulated cold wave (the temperature reaches -2°C) and with a flow limited to the geothermal water available, the indoor temperature does not fall bellow 17.5 °C. The indoor temperature distribution and the flow rate for one building are plotted versus the outdoor temperature during the cold wave and shown

on (Fig.4.2 and Fig.4.3). It is observed through these figures that for the coldest outdoor temperature (-2°C) the indoor temperature is 17.5°C and the flow rate is 0.48 l/s. This flow rate is in concordance with the maximum flow rate available for each building. And the temperature drop is acceptable and shouldn't induce too much discomfort for the users and it happens very rarely.

Hence, we can conclude by saying that the district heating system might be feasible with the amount of geothermal water available.

CONCLUSION

In this project which constitute the final part of the Geothermal Training Programme (1989), our attention has been concentrated on district heating system design.
A study of the different parameters involved in this kind of design has been carried out taking as example, a group of bungalows located in an Algerian geothermal area.
An analysis of the available weather data and a detailed calculation of the heat loss of the buildings (based on their size and the type of materials used) have led to the estimation of the power demand. Based on this, the determination of the capacity of the heating system and the calculation of all the necessary equipment involved in it has been accomplished.
Through the results obtained it seems that the amount of geothermal water is not sufficient. However, the design condition 0°C are very stringent, while the average temperature over 10 years for the coldest month is 5°C. Therefore the amount of hot water available might be enough with no much inconvenience for the costumers.
The dynamic simulation of the heating system has shown that for a severe cold spell of few days duration where the temperature falls down to -2°C and with limited flow rate of hot water, the indoor temperature in the bungalows has not fallen below 17.5°C which indicates that the supply of geothermal water is sufficient.
The design of the system has included the calculation of the radiator system in each room of the buildings considered and an estimation of the system capital cost has been performed. The annual running cost obtained shows that the price of energy for heating is very reasonable, provided that the hot water will not be evaluated too high; but that is not included in the cost estimation.

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NOMENCLATURE

A : Area of the room walls, total radiator heat transfer area, m^2

Ai : Inside area of the pipes, m^2

Ae : Area of the fictitious cylindrical surface including the fins, m^2

Af : Total exposed fin area, m^2

Am : Area of windows or doors, m^2

An : Profile area of the fins

a : Cross section of the pipes, m^2

B : Coefficient of expansion of the air

b : Distance between two fins, m

cp : Specific heat capacity, kJ/kg°C

D : Diameter of the pipe network and the radiators pipes, m

D(T) : Number of degree days

DP : Pressure drop, bar

Ef : Fin efficiency

Et : Total finned area effectiveness

e : Coefficient of exposure to the wind (chp.2), emissivity of the pipe material (chap.3)

f : Friction factor, dimensionless

G : Heat loss per unit volume and per °C, kW/m^3 °C

Gr : Grashof number, dimensionless

h : Convective heat transfer coefficient, W/m^2°C

k : Thermal conductivity, W/m°C

L : Length of the pipe network and radiators pipe, m

Lex : Exterior perimeter of the rooms, m

m : Permeability per unit area of windows and doors frames,

m : flow rate, l/s

N : Total number of days for which the heating is necessary,

N : Number of houses,

N : Total number of days

Nu : Nusselt number, dimensionless

n : Number of fins,

ny	:	Kinematic viscosity, m^2/s
P	:	Exterior walls permeability to air,
Pd	:	Power demand, kWh
Pdt	:	Power demand over the heating season, k^Wh
Pr	:	Prandtl number, dimensionless
Q	:	Heat loss, kW
qi	:	Air infiltration heat loss, W/°C
R	:	Thermal resistance, m^2°C/W
Rf	:	Fouling resistance, m^2°C/W
Ro	:	Density of the water, kg/m^3
s	:	Stefan-Boltzman constant
T	:	Temperature, °C
Tom	:	Mean outdoor temperature, °C
Tmt	:	Mean temperature for N days, °C
TL	:	Thermal length: heat transfer unit number, °C
t	:	Thickness of the wall
U	:	Overall heat transfer coefficient, W/m^2°C
Ul	:	Linear heat transfer coefficient, W/m°C
Vh	:	Volume of the house, m^3
V	:	Flow rate of fresh air, m^3/h
Vg	:	Volume of principal rooms, m^3
Z	:	Fitting pressure drop factor

Subscript:

a	:	air
c	:	cold
h	:	house and/or hot
i	:	indoor and/or in
o	:	outdoor and/or out
s	:	steel
w	:	water and/or wall

APPENDIX

1 Calculation of the overall heat transfer coefficient for the different walls of the houses

1.1 Exterior walls

According to the formula (1.6) the overall heat transfer coefficient for a vertical exterior wall is given by:

$$1/U = 1/ho + 1/hi + t1/k1 + R2 + R3 + t4/k4 + t5/k5$$

each term of the right side of the formula above is the thermal resistance of the different material layer constituting the wall(see Fig.2.2 and Tab.1.2):

- convection on both sides (1/ho + 1/hi),

- conduction through plaster, brick, air space, concrete, and mortar.

$$1/hi + 1/ho = 0.17 \quad [m^2 \cdot C/W]$$

$$1/U = 0.17 + 0.02/0.46 + 0.27 + 0.16 + 0.1/1.4 + 0.03/1.15$$

$$U = 1.35 \ W/m^2 \cdot C$$

1.2 Roof

The overall heat transfer coefficient for the roof is:

$$1/U = 1/hi + 1/ho + t1/k1 + R2 + t3/k3 + t4/k4$$

the subscript 1, 2, 3, 4 are referring to the different layers constituting the roof (Fig.2.3, Tab.2.2). According to the French Standards (DTU, 1975) for a horizontal exterior wall with upward heat flux:

$$1/ho + 1/hi = 0.14\ m^2{\circ}C/W.$$

$$U = 1.464\ W/m^2{\circ}C$$

1.3 Floor with sanitary space

The equivalent heat transfer coefficient U of a floor with a sanitary space is given by (DTU, 1975):

$$1/U = 1/kp + 1/(al + (Lex/Ai)(Km*H + K)) \quad (m^2{\circ}C/W)$$

- kp = 1/0.29 the over all resistance of the floor is estimated to be 0.29 $m^2{\circ}C/W$

- Km is the heat transfer coefficient of the wall separating the sanitary space and the ambience (28 cm large concrete wall):

$Km = 1.4/0.28 = 5\ (W/m^2{\circ}C)$

- H is the height of the sanitary space: H = 0.75 (m)

- in the sanitary space the ground is coated by a concrete slab 25 cm large; the thermal resistance of this concrete slab is: $r = 0.25/1.4 = 0.18\ (m^2{\circ}C/W)$.
from Fig.28 pp 28 in (DTU, 1975), which give the linear heat transfer coefficient of the ground K versus r:

$K = 1.43\ (W/m{\circ}C)$

- al represents the degree of ventilation of the sanitary

space: for a sanitary space greatly ventilated (DTU, 1975) al = 1.2 ($W/m^2 \cdot C$).
Therefore:

$$1/U = 0.29 + 1/(1.6 + 5.18*Lex/Ai) \qquad (m^2 \cdot C/W)$$

U is function of the exterior perimeter of the room Lex and the internal area of the floor Ai.

1.4 Heat loss through the corner

According to (DTU, 1975), when the angle formed by two exterior walls is constituted by a concrete column, the linear heat transfer coefficient is:

$$Ul1 = 0.45*ew1$$

- ew1 = =0.17 cm, is the arithmetic mean of the two thickness of the two walls (Fig.2.6.).

But in the present case the concrete column do not occupy the total cross section of the wall (Fig.2.6.).
The real corner is formed by two corners in series can be combined by assuming that the real corner is formed by two corners in series:
- the first one with the concrete column,
- the second one is formed by: concrete and air layer.

For this second wall the heat transfer coefficient is given by:

$$Ul2 = 0.2*U*ew2$$

Where:

- ew2 = 16 cm is the thickness of the wall,
- U heat transfer coefficient of the w*all (Fig.2.6.)

(the different parameters and values are shown on Fig.2.2,

Fig.2.6).

$$1/U = 1/ho + Rair + x1/k1 + x2/k2$$
$$= 0.06 + 0.16 + 0.0714 + 0.026$$

$$U = 3.149 \ (W/m^2 \cdot C)$$

Hence the linear heat transfer coefficient is:

$$Ul2 = 0.2*3.149*0.16$$

$$Ul2 = 0.1 \ (W/m^2 \cdot C)$$

$$Ul1 = 0.45*0.15 = 0.068 \ (W/m^2 \cdot C)$$

For the overall corner the linear heat transfer coefficient is given by the calculation of the total thermal resistance:

$$1/Ul = 1/Ul1 + 1/Ul2$$

$$= 1/0.1 + 1/0.068$$

$$UL = 0.04 \ (W/m^{\circ}C)$$

1.5 Doors

The doors are made of wood and all facing to the terrace:

$$1/Udoor = 1/ho + 1/hi + x/kwood$$

- x is the thickness of the door: 0.04 (m)

- kwood is the thermal conductivity of the wood

$$1/Udoor = 0.17 + 0.04/0.23$$

$$Udoor = 2.9\ (W/m^2 \cdot C)$$

Correction due to the effect of the wind (see chapter 1):

$$Udoor = 3.086\ (W/m^2 \cdot C)$$

All these values of the over all heat transfer coefficient are summarized in Tab.2.2.

2 Flow rate in the pipe network

The total flow rate needed to heat the houses is given by:

$$m = Qt/(cp*(Tfs-Tfr))$$

$$m = 867.4/(4.186*(60-39))$$

$$m = 9.87\ (kg/s)$$

The necessary flow rate for each house is:

$$mfh = Q/(cp*(Tfs-Tfr)/16)$$

$$mfh = 54.21/(4.186*(60-39))/16 = m/16$$

$$mfh = 0.617\ (kg/s)$$

since the flow rates for each house are known it is possible to know the flow rate in all the pipes of the network by a back step summation.

3 Heat exchanger: Heat transfer area and pressure drop

3.1 Logarithmic mean temperature difference (L.M.T.D)

Hot fluid: Thi = 67°C Tho = 42°C
Cold fluid: Tci = 37°C Tco = 62°C
For a counterflow heat exchanger:

$$L.M.T.D = ((Thi-Tco)-(Tho-Tci))/Ln((Thi-Tco)/(Tho-Tci))$$
$$L.M.T.D = 5°C$$

3.2 Thermal length

The thermal performance of a heat exchanger may be expressed in terms of the TL value (number of heat transfer units) of which the heat exchanger is capable. For a single phase duty TL is defined as the ratio of temperature change of one fluid to the mean temperature difference between the two fluids (Heat Exchanger Guide, Alfa Laval).

$$TL = (Thi-Tho)/L.M.T.D = 5$$

3.3 Pressure drop and heat transfer area

The pressure drop fixed is 20 kPa per pass. The area per kg/s versus the thermal length TL is given by (Heat exchanger Guide, Alfa Laval).

For TL = 5 and DP = 20 kPa

$$A/kg/s = 7.2\ m^2/kg/s$$

The total flow rate is :

$$m = 9.87\ kg/S$$

Therefore:

$A = 71.1\ m^2$ (clean area)

In order to take into account the effect of fouling, the total area must be increased by 15 %.

Adirty = 81.87 (m^2)

The model of heat exchanger chosen is: Plate Heat Exchanger, P.H.E, model A 15c (Alfa - Laval):

- the area per plate is : 0.63 (m^2)

- the number of plates is : 130 heat exchange plates and two end plates.

The thermal length for this model (A 15c) of plates heat exchanger is TL = 2.5. The thermal length for the heat exchanger needed is TL = 5, then two passes are required.

Therefore:

- number of plates per pass: 65 plates

- total pressure drop in the secondary side:

DP1 = 2 * 20 kPa, DP1 = 40 kPa.

The temperature gradient is the same for both fluids, so the flow rates are the same in both sides of the heat exchanger.

mf1 = mf2 = 9.87 l/s

The pressure drop in the secondary side is given by (Heat

Exchanger Guide, Alfa Laval):

$$DP2 = DP1 * (mf2/mf1)**1.9$$

$$DP1 = DP2 = 40 \text{ kPa}$$

4 Radiator

4.1 Heat transfer area of the radiators pipe

- Total exposed fin area ,Af:

$$Af = 0.0022*n \ (m^2)$$

- Total heat transfer area A, including the fins:

$$A = 0.00248*n \ (m^2)$$

- Internal area of the pipes

$$Ai = 0.000352*n \ (m^2)$$

4.2 Calculation of the radiators wall temperature and film temperature

The mean temperature Tm of the water in the pipes is:

$$Tm = (Tfi + Tfo)/2$$

$$Tm = (39+60)/2 = 49.5 \text{ °C}$$

The pipe wall is assumed to be at the temperature of the water, so its temperature Tw is equal to Tm

$$Tw = 49.5 \text{ °C}$$

The air film surrounding the radiators is at a temperature Tf, which is equal to:

$Tf = (Tw + Ti)/2$; $Tf = 34.7$ °C ~ 300 K

4.3 Convective heat transfer coefficient

4.3.1 Inside convective heat transfer coefficient, hi

$hi = 4.364 * (kw/Di)$

$kw = 0.626$ W/m°C

$Di = 0.0224$ m

$hi = 122$ W/m^2°C

4.3.2 Outside convective heat transfer coefficient

$Gr*Pr = g*B*(Tw-Ti)*Do^3*Pr/ny^2$

At Tf = 300 K, for the air:

- Pr = 0.708
- $B = 1/Tf = 0.0033$ K^{-1}
- $g = 9.8$ m/s^2
- $ny = 15.69*10^{-6}$ m^2/s kinematic viscosity of the air
- $Do = 0.0254$ m
- $kair = 0.02624$ W/m°C

$Gr*Pr = 4.496*10^4$

then according to (Holman J.P., 1989), the outside convective heat transfer coefficient hc is given by:

$$Nu = 0.53*(Gr*Pr)^{1/4} = 7.718$$

hc = (Nu*kair)/Do

hc = 7.97 $W/m^2 {}^\circ C$

4.3.3 Radiative heat transfer coefficient

hr = (Ae/A)*e*s*(Tw2+Ti2)*(Tw+Ti)

hr = (0.087*10**(-2)*n/0.248*10**(2)*n)
*0.8*5.67*10**(-8)(322.5+293)*(322.5 +293)

hr = 1.86 $W/m^2 {}^\circ C$

4.4 Calculation of the efficiencies of the finned pipe Et and Ef

- The efficiency of the fins is given by as a function of the parameter (Holman J.P., 1989):

(Lc**3/2)*(hc/ks*An) = (0.01575**3/2)*
(7.97/54*0.00002363)**1/2

= 0.16

Ef = 0.95

- Total surface effectiveness

Et = 1 - (Af/A)*(1-Ef) ; Et = 0.956

4.5 Calculation of the total length of the pipes

L = b*n

The formula (3.18) give the number of fins as a function of the heat flux Q. Numerical applications leads to:

$$n = Q*2.345$$

$$L = 0.005*2.345*Q$$

$$L = 11.72*10^{-3}*Q$$

TABLE 2.1: Temperature Distribution and Degree Days

To[C]	Nb days	Total Nb Days	20-To	DD(20)	SUM DD20
19.75	5	254	0.25	1.25	1983.5
19.25	10	249	0.75	7.5	
18.75	0	239	1.25	0	
18.25	5	239	1.75	8.75	
17.75	6	234	2.25	13.5	
17.25	5	228	2.75	13.75	
16.75	10	223	3.25	32.5	
16.25	5	213	3.75	18.75	
15.75	5	208	4.25	21.25	
15.25	5	203	4.75	23.75	
14.75	5	198	5.25	26.25	
14.25	16	193	5.75	92	
13.75	15	177	6.25	93.75	
13.25	5	162	6.75	33.75	
12.75	5	157	7.25	36.25	
12.25	10	152	7.75	77.5	
11.75	11	142	8.25	90.75	
11.25	0	131	8.75	0	
10.75	35	131	9.25	323.75	
10.25	15	96	9.75	146.25	
9.75	4	81	10.25	41	
9.25	20	77	10.75	215	
8.75	27	57	11.25	303.75	
8.25	20	30	11.75	235	
7.75	0	10	12.25	0	
7.25	10	10	12.75	127.5	

TABLE 2.2: Thermal conductivity and thermal resistance of the different walls.

Material	Plaster	Mortar	Concrete	Air Brick	Granite
K (W/m°C)	0.46	1.15	1.4		3.5
R (m^2°C/W)				0.27	

TABLE 2.2 (continued)

Materials	Air Space	Hourdi(concrete)	Cork (insulation)
K(W/m°C)			0.1
R(m^2°C/W)	0.16	0.15	

TABLE 2.3 Values of the over all heat transfer coefficient

Walls	outside wall	Roofs	Corner	Door	Windows
U(W/m^2°C)	1.35	1.46		3.09	4.2
Ul(W/m°C)			0.04		

The over all heat transfer coefficient of the floor is calculated according to (formula 2.8)

TABLE 3.1 Characteristics of the Heating Elements

BLOCK N1	Q[W]	m[kg/mn]	V[m/mn]	L[m]	A[m2]
LR 1+2	4306.00	2.94	0.26	50.47	25.04
U1	3271.50	2.23	0.23	38.34	19.03
U2	3271.50	2.23	0.23	38.34	19.03
BLOCK N2					
LR1	4743.00	3.24	0.27	55.59	27.58
LR2	2559.90	1.75	0.20	30.00	14.89
R1	2042.00	1.39	0.18	23.93	11.88
R2	1104.00	0.75	0.13	12.94	6.42
R3	1668.80	1.14	0.16	19.56	9.71
K1	1742.80	1.19	0.17	20.43	10.14
K2	789.00	0.54	0.11	9.25	4.59
BR1	937.60	0.64	0.12	10.99	5.45
BR2	778.00	0.53	0.11	9.12	4.52
WR1	345.00	0.24	0.07	4.04	2.01
BLOCK N3					
LR	3983.80	2.72	0.25	46.69	23.17
R	2339.50	1.60	0.19	27.42	13.61
K	1074.60	0.73	0.13	12.59	6.25
BR	1132.80	0.77	0.13	13.28	6.59
WR	431.40	0.29	0.08	5.06	2.51
BLOCK N4					
LR1	3603.20	2.46	0.24	42.23	20.95
LR2	3883.80	2.65	0.25	45.52	22.59
R1	2398.00	1.64	0.19	28.10	13.95
R2	2398.00	1.64	0.19	28.10	13.95
K1	1074.00	0.73	0.13	12.59	6.25
K2	1074.00	0.73	0.13	12.59	6.25
BR1	1127.00	0.77	0.13	13.21	6.55
BR2	1127.00	0.77	0.13	13.21	6.55
WR1	500.00	0.34	0.09	5.86	2.91
WR2	500.00	0.34	0.09	5.86	2.91

LV: LIVING ROOM R: ROOM K: KITCHEN BR: BATH ROOM WR: WATER ROOM

Table 3.2 Chemical composition of water samples from Hummam Righa.

Sample	°C	l/sec	pH/°C	SiO_2	Li	Na	K	Ca	Mg	CO_2	SO_4	Cl	F	Dis.solid	OHMM.
00-89-1	67.0	4.0	6.50/22.	41.6	.20	210.5	9.8	489.6	35.5	180.4	1095.0	347.0	1.6	2384.0	418.00
00-89-2	67.0	4.0	6.50/22.	45.5	.20	211.3	9.6	453.0	35.5	176.4	1056.0	336.5	1.6	2479.0	385.00
00-89-3	68.0	4.0	6.50/22.	49.0	.21	214.3	9.7	472.0	38.0	183.4	1070.0	351.0	1.5	2370.8	423.00
00-89-4	68.0	4.0	6.50/22.	46.9	.20	208.3	9.7	481.0	36.1	176.4	1067.5	336.0	1.5	2466.0	383.00

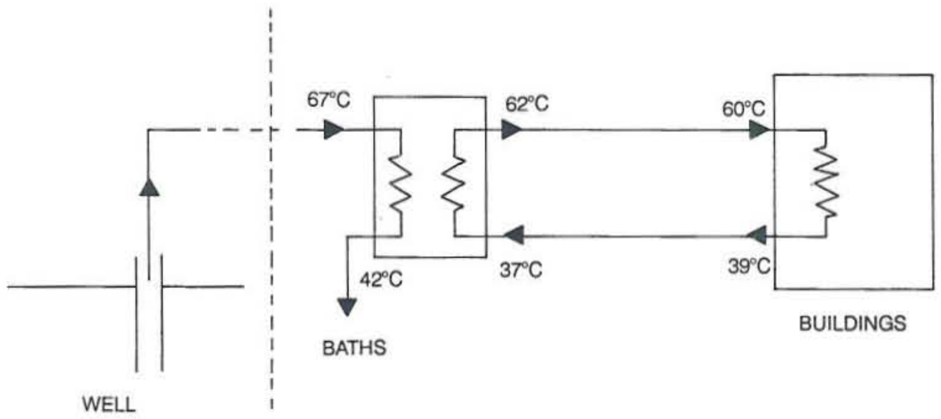

Fig.1.1 Heating System Proposed

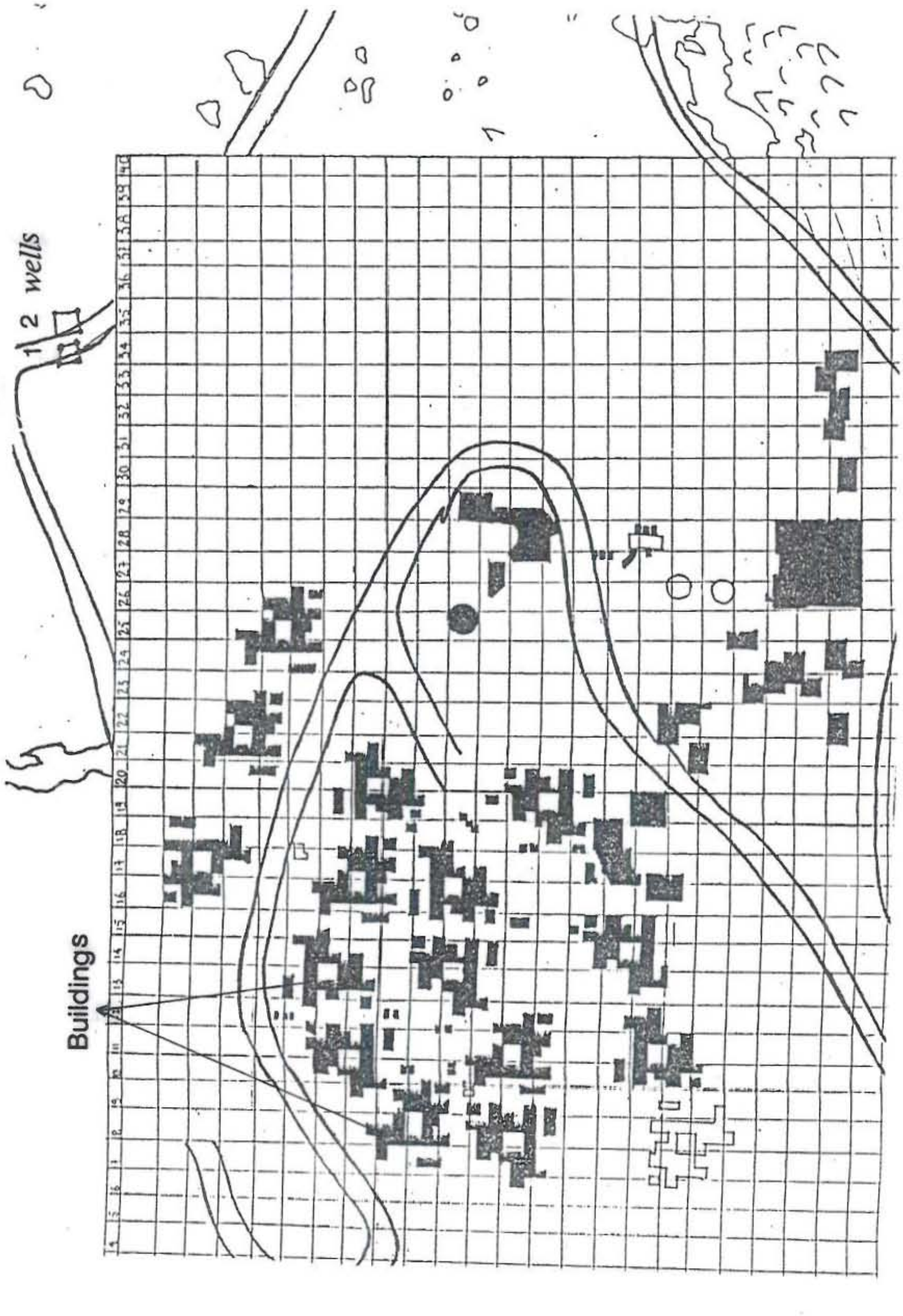

Fig.1.2 Location of the Houses and Wells

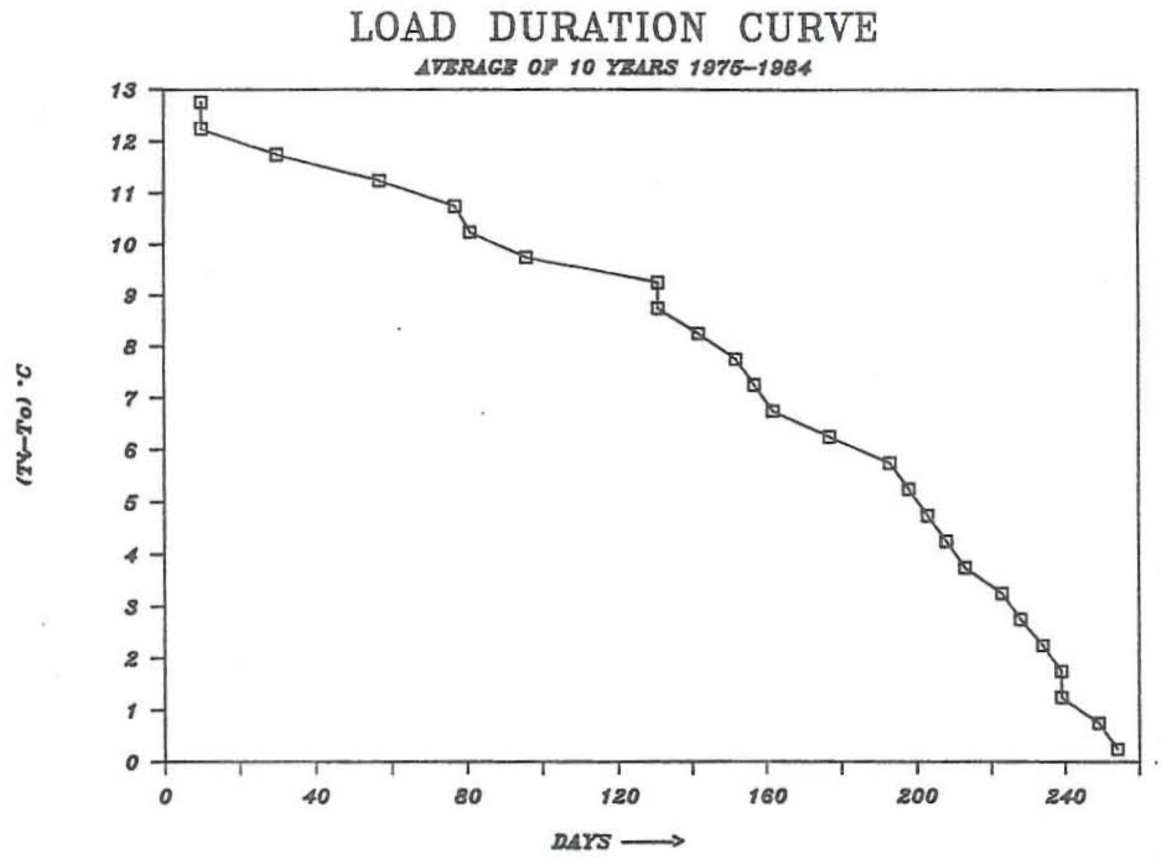

Fig.2.1 Temperature-Time Duration Curve

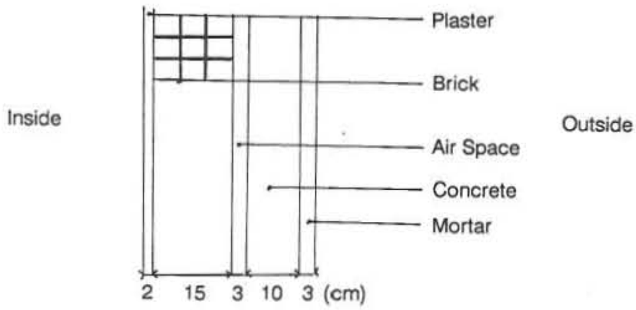

Fig 2.2 Exterior wall

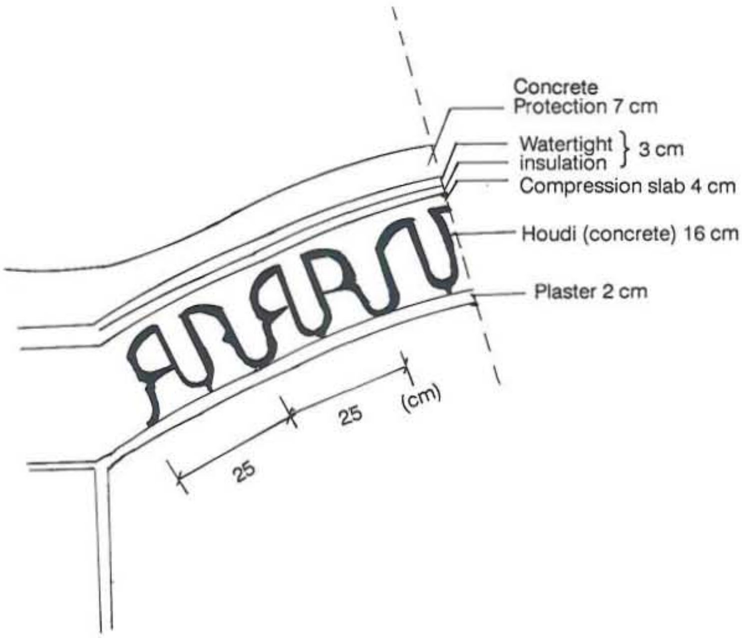

Fig 2.3 Roof cross section

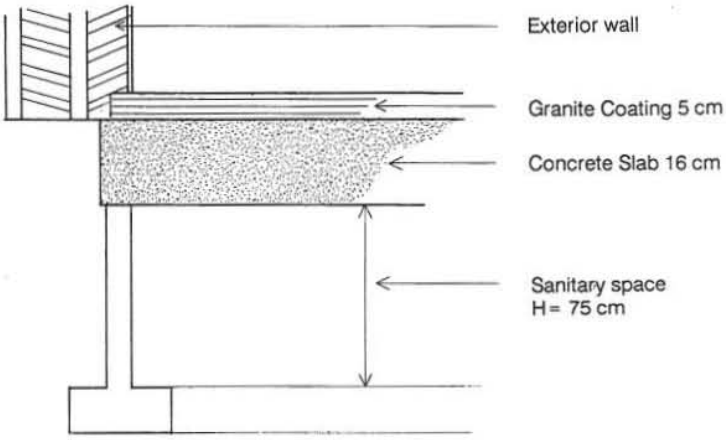

Fig 2.4 Floor Dimensions and Materials

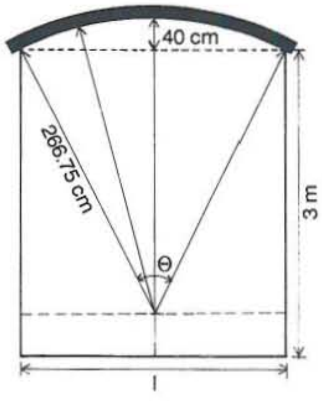

Fig 2.5.1 Geometry of the Exterior wall

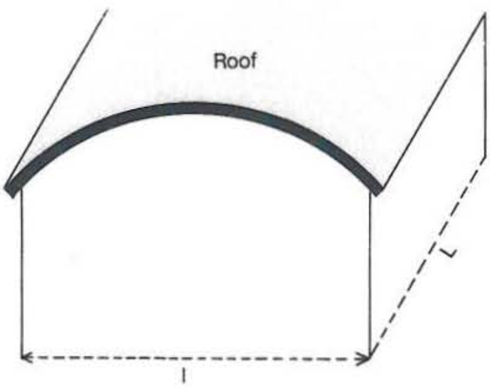

Fig 2.5.2 Geometry of the Roof

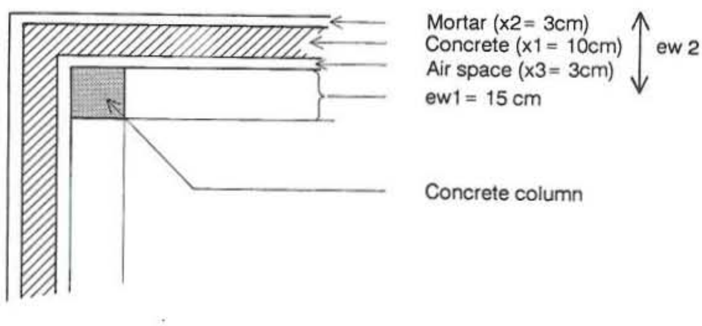

Fig 2.6 Detail of a Corner of a Typical Room

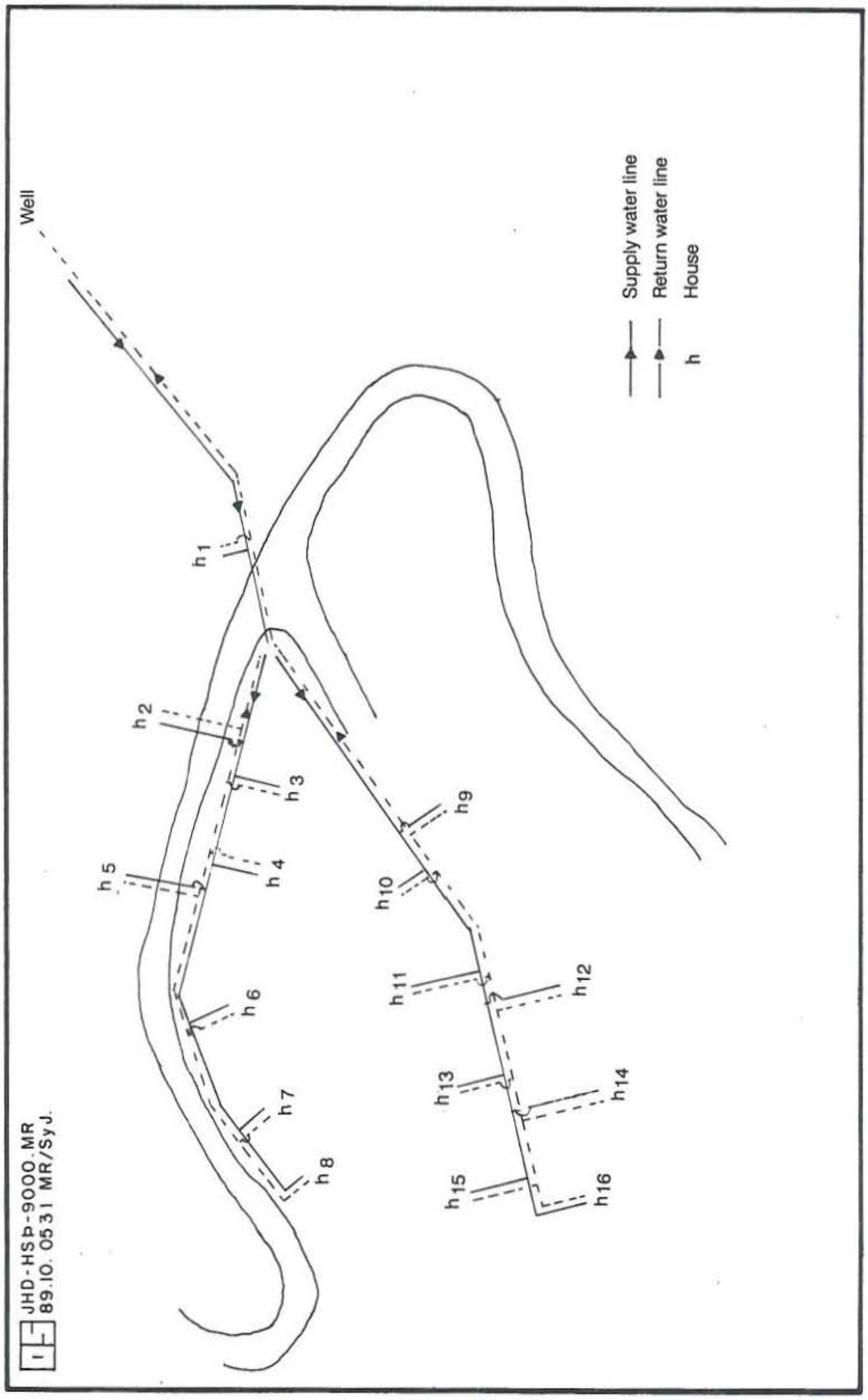

Fig.3.1 Pipeline Network

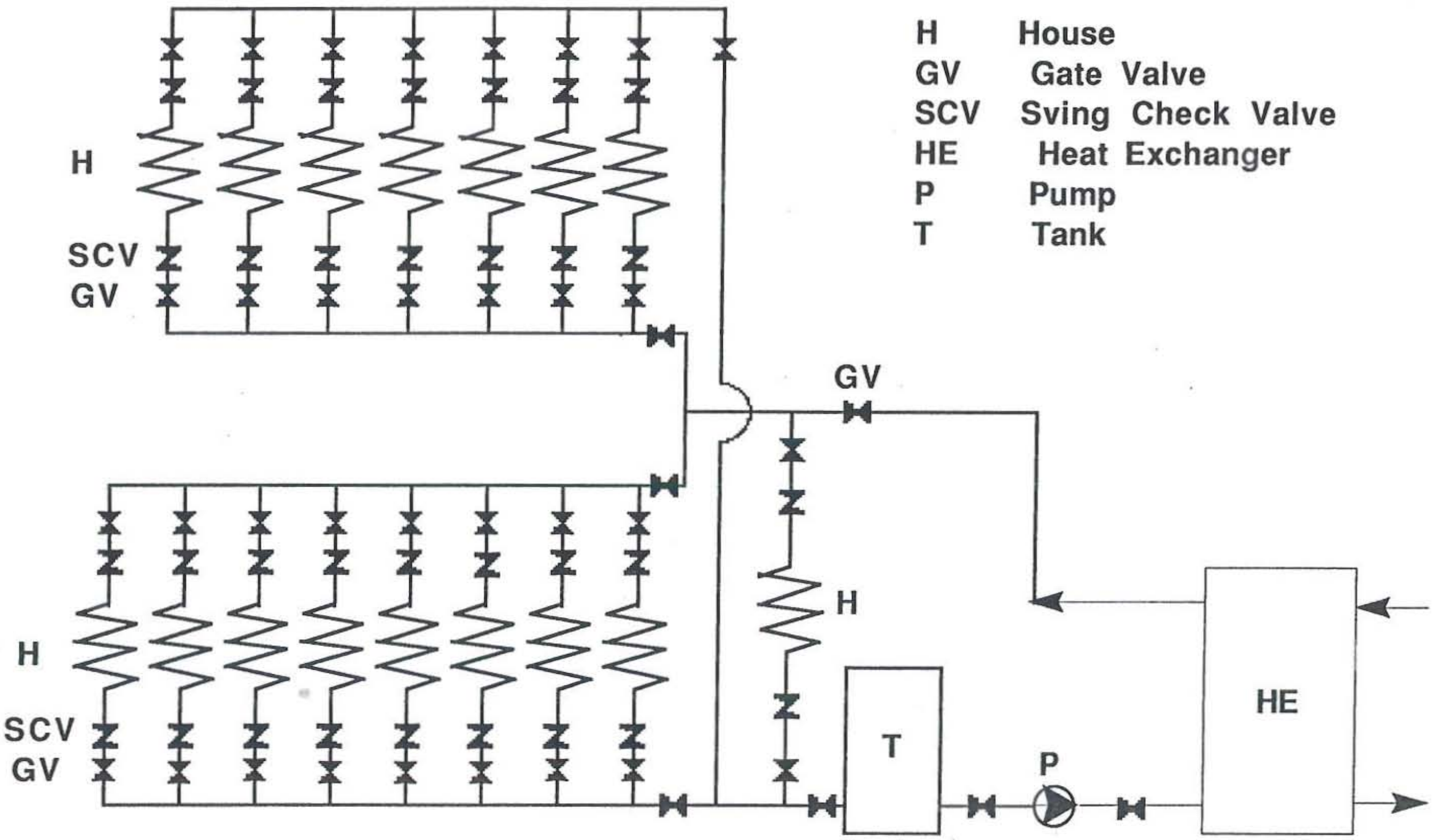

Figure 3.2 Schematic Drawing of the Heating System

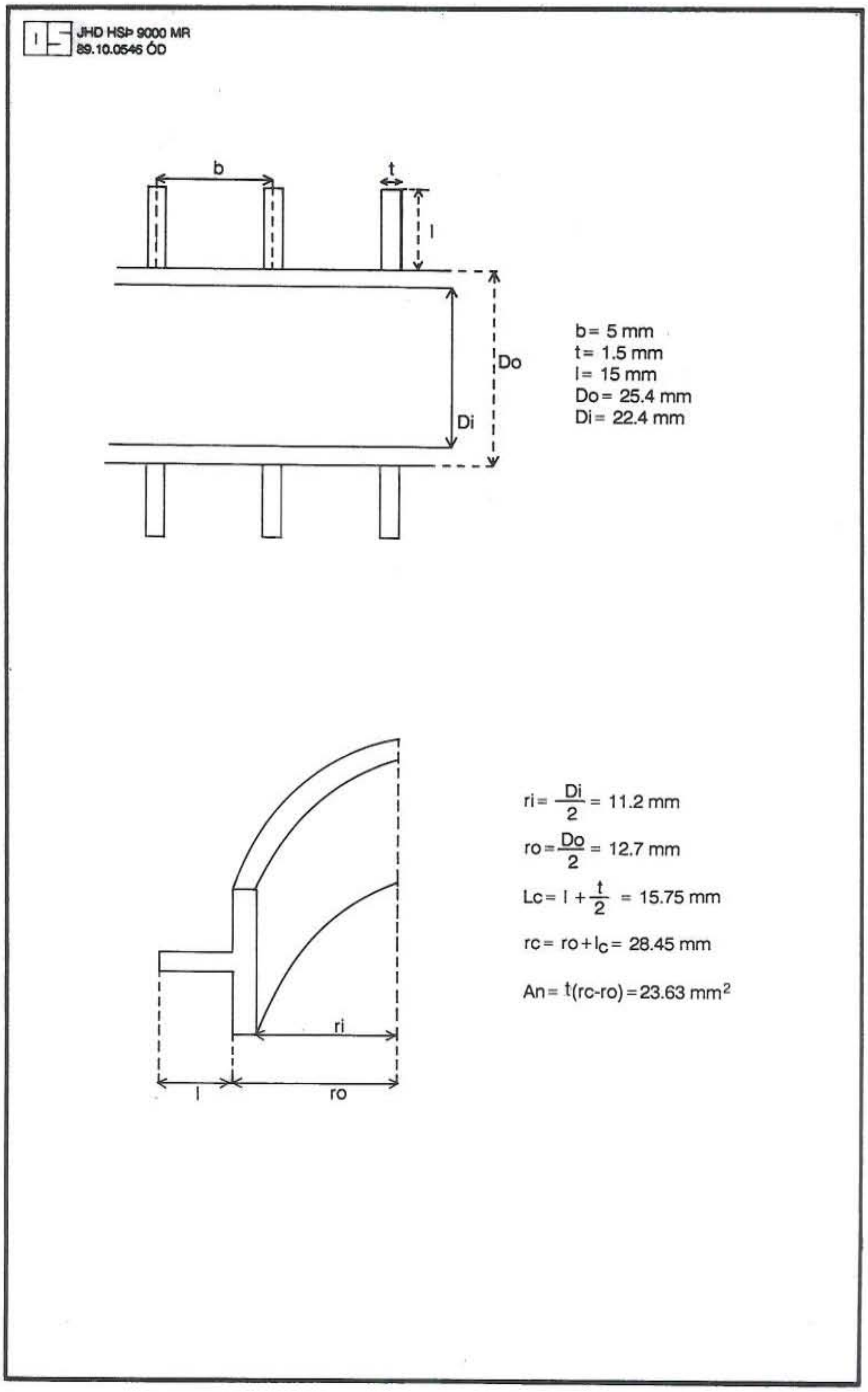

Fig.3.3 Heating Element Section: Finned Pipe

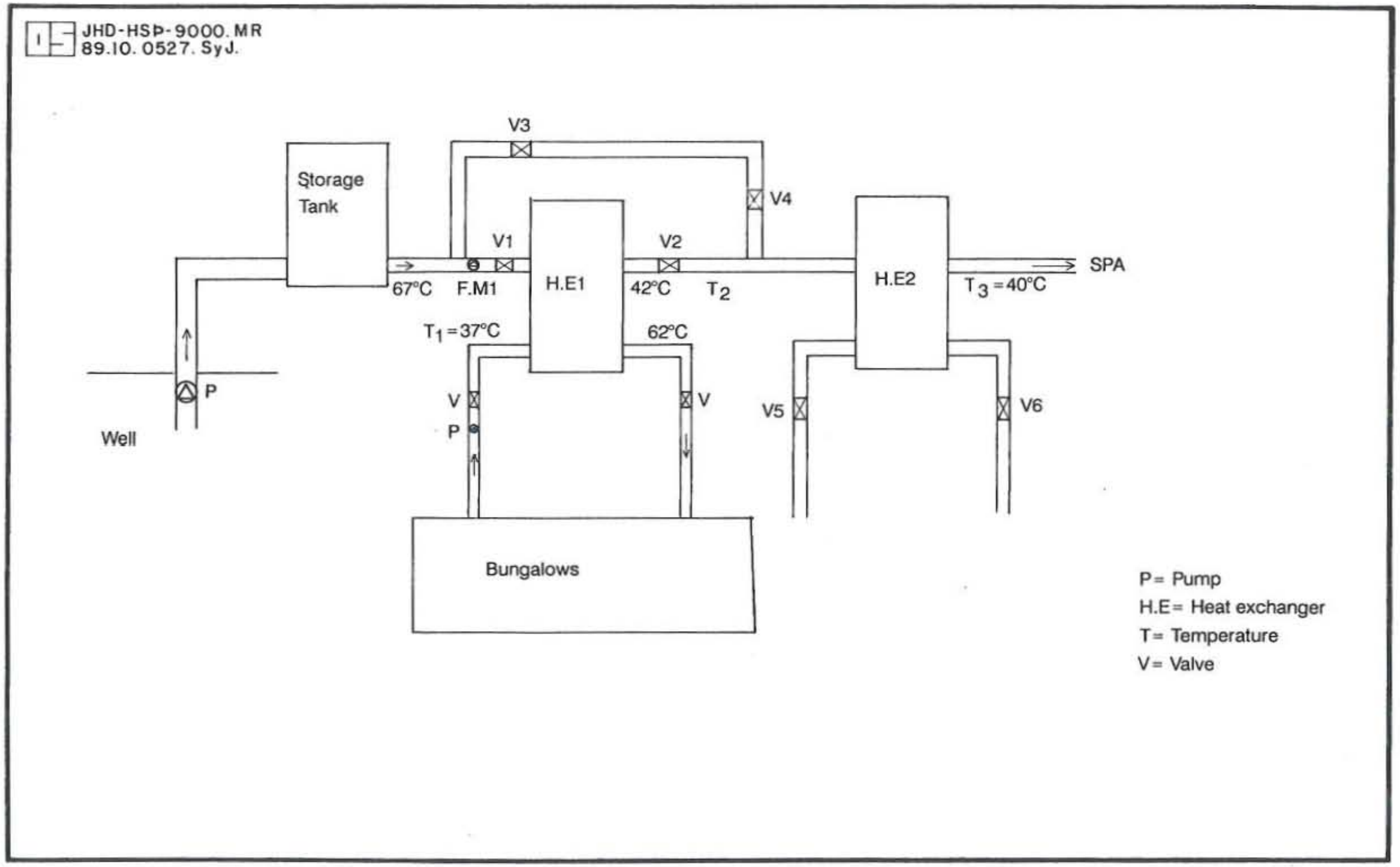

Fig.3.4 General System Control

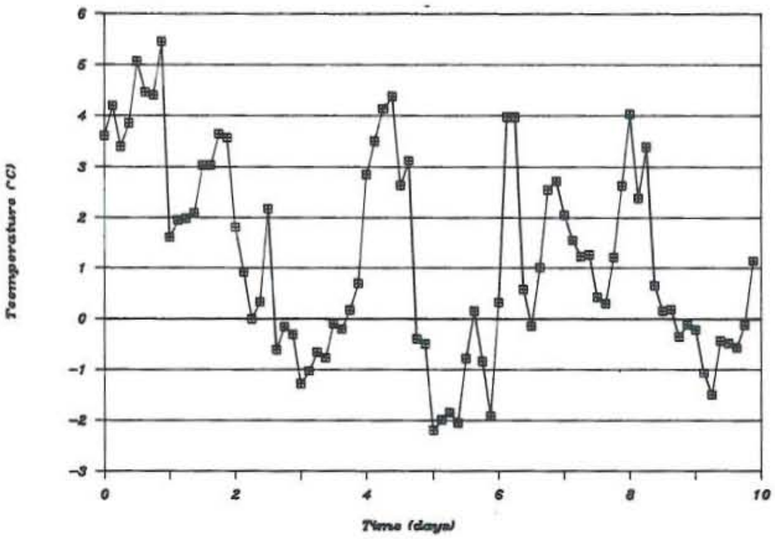

Fig.4.1 Outdoor Temperature
Distribution during a cold wave

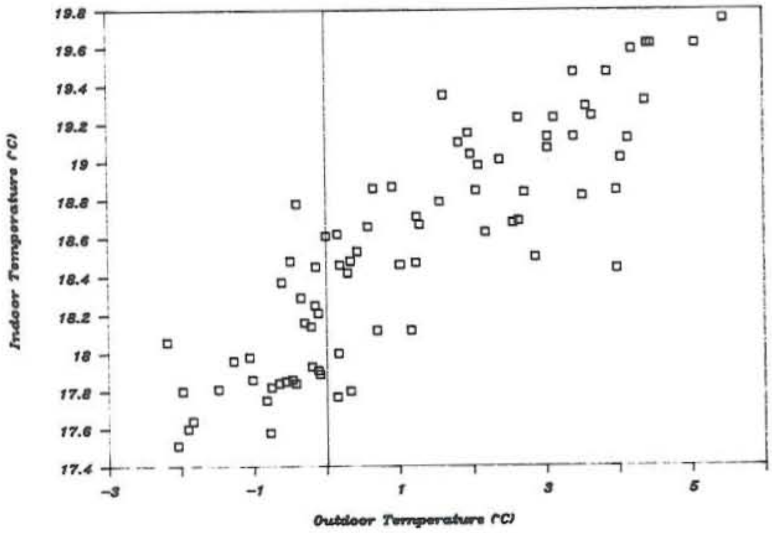

Fig.4.2 Indoor Temperature Distribution
for one building

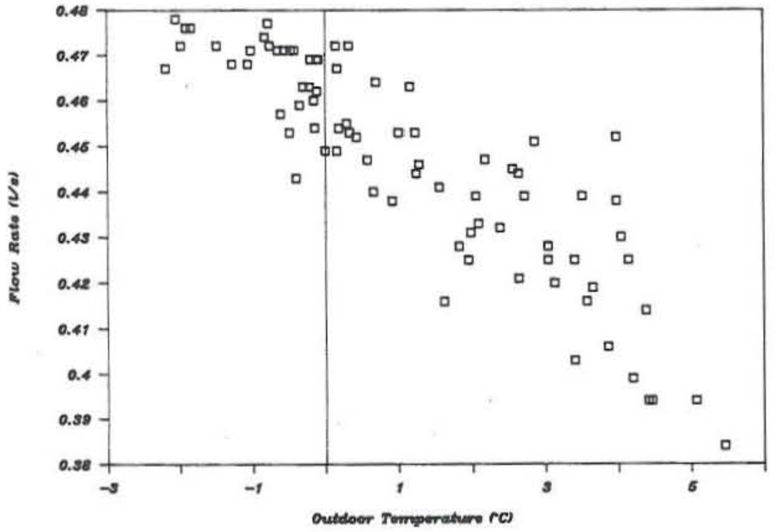

Fig.4.3 Flow Rate Distribution
(for 1 building)