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SIMULATION OF HEATING SYSTEMS IN JORDANIAN BUILDINGS

M.Sc. Thesis
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by

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INTRODUCTION

The Geothermal Training Programme of the United Nations University (UNU) has operated in Iceland since 1979 with six months annual courses for professionals from developing countries. The aim is to assist developing countries with significant geothermal potential to build up groups of specialists that cover most aspects of geothermal exploration and development. During 1979-1999, 245 scientists and engineers from 36 countries have completed the six months courses. They have come from Asia (44%), Africa (26%), Central America (15%), and Central and Eastern Europe (15%). There is a steady flow of requests from all over the world for the six months training and we can only meet a portion of the requests. Most of the trainees are awarded UNU Fellowships financed by the UNU and the Government of Iceland.

Candidates for the six months specialized training must have at least a B.Sc. degree and a minimum of one year practical experience in geothermal work in their home countries prior to the training. Many of our trainees have already completed their M.Sc. or Ph.D. degrees when they come to Iceland, but several excellent students who have only B.Sc. degrees have made requests to come again to Iceland for a higher academic degree. In 1999, it was decided to start admitting one or two outstanding UNU Fellows per year to continue their studies and study for M.Sc. degrees in geothermal science or engineering in co-operation with the University of Iceland. An agreement to this effect was signed with the University of Iceland. The six months studies at the UNU Geothermal Training Programme form a part of the graduate programme.

It is a pleasure to introduce the first UNU Fellow to complete the M.Sc. studies at the University of Iceland under the co-operation agreement. Mr. Muthafar S. Emeish, research engineer at the Royal Scientific Society in Amman, Jordania, completed the six months specialized training at the UNU Geothermal Training Programme in October 1999. His research report was entitled "Geothermal heating systems for Jordanian greenhouses" (Report 2 in Geothermal Training in Iceland 1999). He enrolled for the M.Sc. studies at the Faculty of Engineering of the University of Iceland in the same fall semester, and has now defended his M.S. thesis entitled "Simulation of heating systems in Jordanian buildings", which is presented here. His studies in Iceland were financed by a fellowship from the Government of Iceland through the UNU Geothermal Training Programme. We congratulate him on his achievements and wish him all the best for the future. We thank the Faculty of Engineering of the University of Iceland for the co-operation.

With warmest wishes from Iceland,

Ingvar B. Fridleifsson, director,
United Nations University
Geothermal Training Programme

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Abstract

Temperature control of buildings is a way to make the indoor climate a pleasant place for humans to live and perform their daily work in an effective and efficient way.

During winter heat is added to the building envelope in order to keep the indoor temperature high enough so people inside feel comfortable. One of the methods of adding heat to a building is to use a central heating system, which is composed of a boiler, circulating pumps and radiators inside the space to emit the heat.

This project is aimed at evaluating the Jordanian codes that deal with the design and installation of the central heating system, by simulating this system. Mathematical models were derived, a program using Matlab was written and run using hourly weather data. The simulation program was run to simulate insulated and non-insulated buildings and different control methods such as using a room thermostat, using thermostatic flow valves and intermittent heating systems were compared.

The simulation results showed that building insulation is very important and financially feasible. The results also showed that using thermostatic flow valves is more expensive than the room thermostat yet it provides more comfort in each room unlike the thermostat which controls the whole system depending on one point measurement. As for the intermittent heating, the simulation results indicated that long heating hours and bigger systems should be used in order to achieve comfortable indoor conditions which needed higher initial cost and consumed more fuel than the continuous heating system. Hence the use of these systems should be avoided.

Finally this work indicates that simulation can be a powerful tool in the hands of the designers to optimize their systems' performance in a relatively short period and at a low cost where there will be no need for building the system to try it.

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Chapter 1

Introduction

Heating is the process that maintains a given space at a constant temperature, usually above the surrounding temperature fluctuations. This process has enabled humans to survive and live in severe conditions, without being highly affected by these environments. Heating may involve the use of fire-places, room stoves, kerosene heaters, liquefied petroleum gas (LPG) heaters, which are not efficient, inconvenient to operate, hard to control and uniform heat distribution is not possible [2]. To overcome these problems, central heating, and district heating systems for colder climate, were introduced. In these systems furnaces or boilers are used to supply the produced heat of combustion to a suitable medium, usually water, to the different parts of the space being heated via radiators. These systems ensure better distribution of heat to the different parts of the heated space, are less dangerous since combustion takes place outside the heated space and are easier to control than the other systems.

Although central heating systems are more efficient and convenient than other heating systems, only 9.5% [8] of the total residences in Jordan have central heating systems. This is mainly due to the high initial cost of these systems, and relatively high running costs of such systems which might be caused by over-designing, lack of use of proper controls or energy efficiency systems.

In this project a review of the Jordanian heating codes [7] and [10] will be carried out, where a simulation program will be constructed using actual data. Then the simulation program will be run with the aim of evaluating these codes and testing the control techniques they suggest. Depending on the results obtained from the simulation recommendations with the attempt of improving the heating system performance and efficiency will be drawn.

Chapter 2

Weather Data

The influences on the load of heating systems are closely related to the influences on the heat dynamics of buildings. Hence the influences can be divided into two groups: The influences from the outdoor climate and influences originating from sources inside the buildings. The latter group consists of influences from light, machines, persons and radiators (including the control system). The group involving the outdoor climate is composed of influences from:

- Air temperature.
- Short wave radiation (the sun).
- Wind speed and direction.
- Precipitation.

The factors are arranged in an assumed decreasing importance for heating systems[5].

Air temperature is affecting the indoor climate through heat conduction in the outer walls and windows, and through free and forced ventilation (infiltration).

Short wave radiation is composed of direct, diffuse and reflected radiation. The direct radiation is the parallel beam radiation from the sun which penetrates through the atmosphere without reflection (scatter) or absorption. The diffuse radiation at the horizontal plane at ground level, which originates from scattering by air molecules, aerosols and cloud particles. Usually meteorologists and climatologists do not distinguish between direct and diffuse radiation, since only the total amount of short wave radiation at horizontal planes at ground level is called the global radiation. However, for the heat dynamics of buildings it is important to distinguish between direct and diffuse radiation, since the angle of incidence is needed to calculate the amount of penetrating radiation at any orientation of the windows.

Wind speed is affecting the buildings through the natural ventilation. Dependent on the speed and direction, the wind introduces a pattern of differences in and around the building, which again implies an air flow through not tight windows, etc. The wind speed is also affecting the temperature of the outer surface of the building, because the wind increases the convective heat transfer.

Long wave radiation is probably of minor importance, but at clear nights the long wave radiation emitted from the outer surface of the buildings to the space may give rise to a lower temperature of the surfaces than of the nearby air-in particular at low wind speeds. Due to a wider optical contract with the space the effect is strongest for the roof.

Precipitation has only a very limited influence on the heat consumption. The slight influence owes mainly to a change in the heat conductivity of the wall or roof if the materials become wet.

The first step in designing and optimizing a heating system of a building is to evaluate the outdoor conditions surrounding this building. Hence, a study of the climatic conditions is important because these climatic conditions control the heat dynamics of the building.

As discussed above, ambient temperature and solar radiation are the most important factors affecting building dynamics. Accordingly, only these two variables will be accounted for in the simulation.

In order to obtain good simulation results, representative climatic data over many years should be used. Unfortunately, in this project, only limited data for some years was available. Firstly hourly data for dry bulb temperature ($°C$), see Figure 2.2. Secondly hourly values of solar radiation on a horizontal surface (W/m^2) for the year 1989 was also available, see Figure 2.4. Finally, only monthly averages of temperature values could be obtained for the years of 1998, and 1999 as well as the monthly average temperature values for the city of Amman were also available.

Before using the temperature profile of 1989 a check was made to determine whether this data is representative of the climate of Amman. In order to conduct this check, a graph representing the average monthly temperature values for the different years was plotted with the results shown in Figure 2.1. The Figure shows that the monthly average values for those years, specially during winter period, are close indicating that the temperature hourly values of 1989 are representative and can be used in the simulation.

Hence the weather conditions of 1989 will be considered representative of the climate of Amman, Jordan and will be used in this project. Hence further

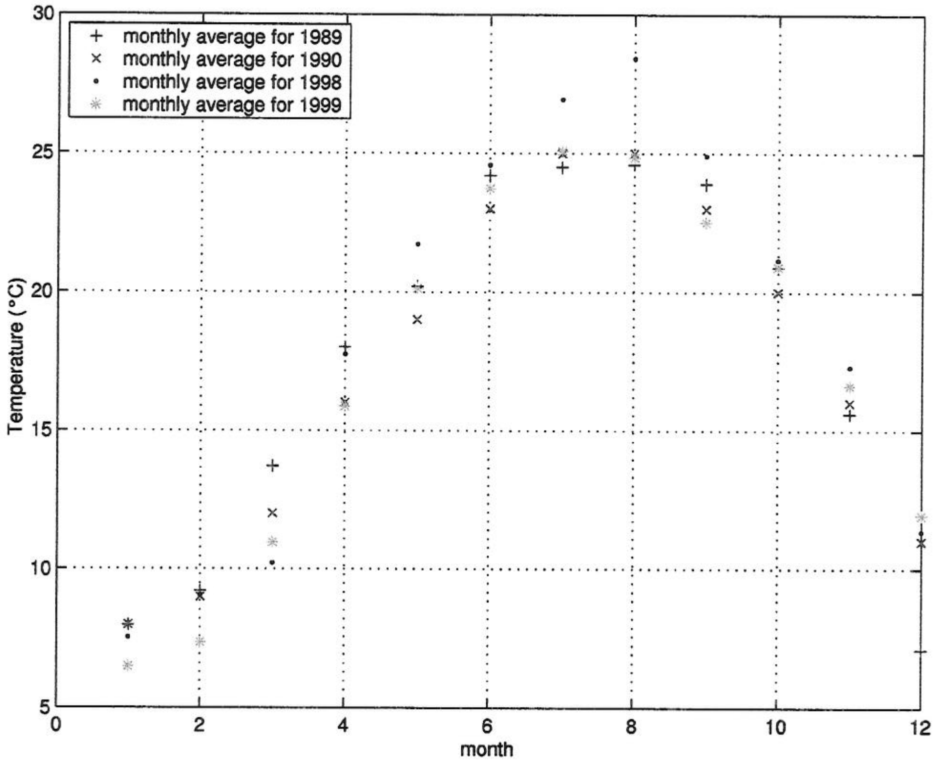


Figure 2.1: Monthly average temperature values at different years for Amman

evaluation of this data will be conducted.

Figure 2.2 shows the temperature profile for the year (1989) on an hourly basis. Figure 2.3, representing the outdoor temperature duration curve, shows that the outdoor temperature is below $20°C$ for 5122 hours, almost half the year. This indicates the necessity for heating and showing that making efforts to try to optimize the system and improve its performance is worthwhile and not a waste of time.

Another reason that should encourage scientists and engineers in Jordan to try to enhance the heating system is the energy cost of the system, where according to the energy survey [8] domestic heating consumes about 10% of the total energy consumption for the country which depends totally on importing its energy.

As for the other variable affecting buildings' dynamics, the solar radiation, Figure 2.4 shows that Jordan is blessed with a huge amount of solar radiation which in turn reduces the heat losses for buildings and decreases its energy demands. This factor is usually neglected when calculating the building heating load using the static method. The effect of solar radiation will be investigated in detail in later chapters.

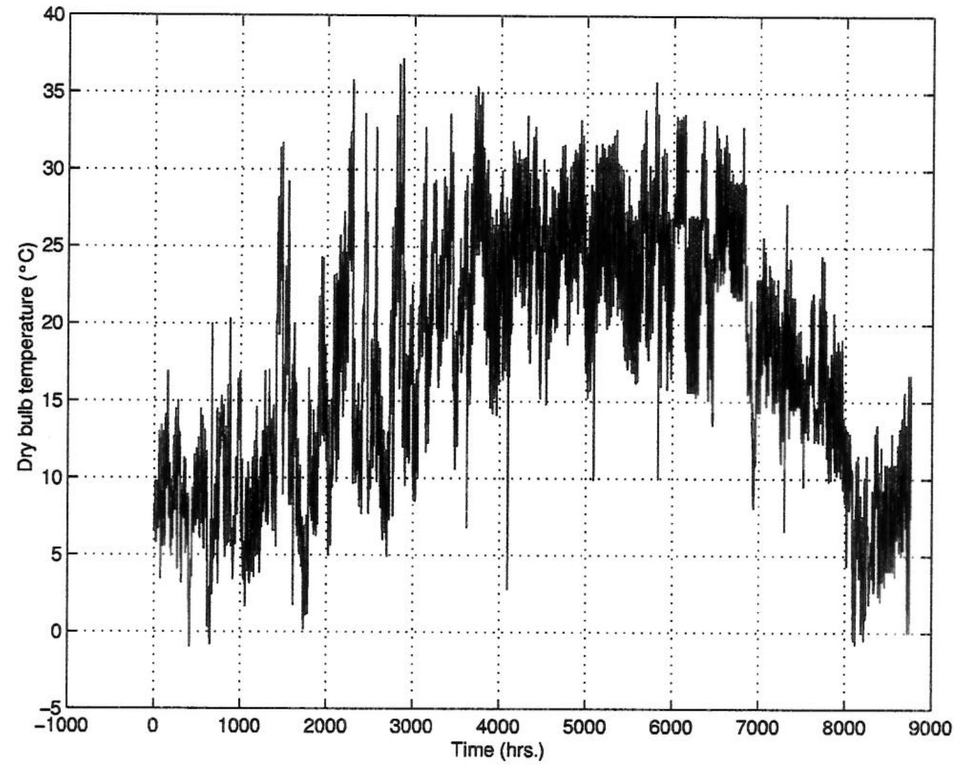


Figure 2.2: 1989 Temperature profile for Amman

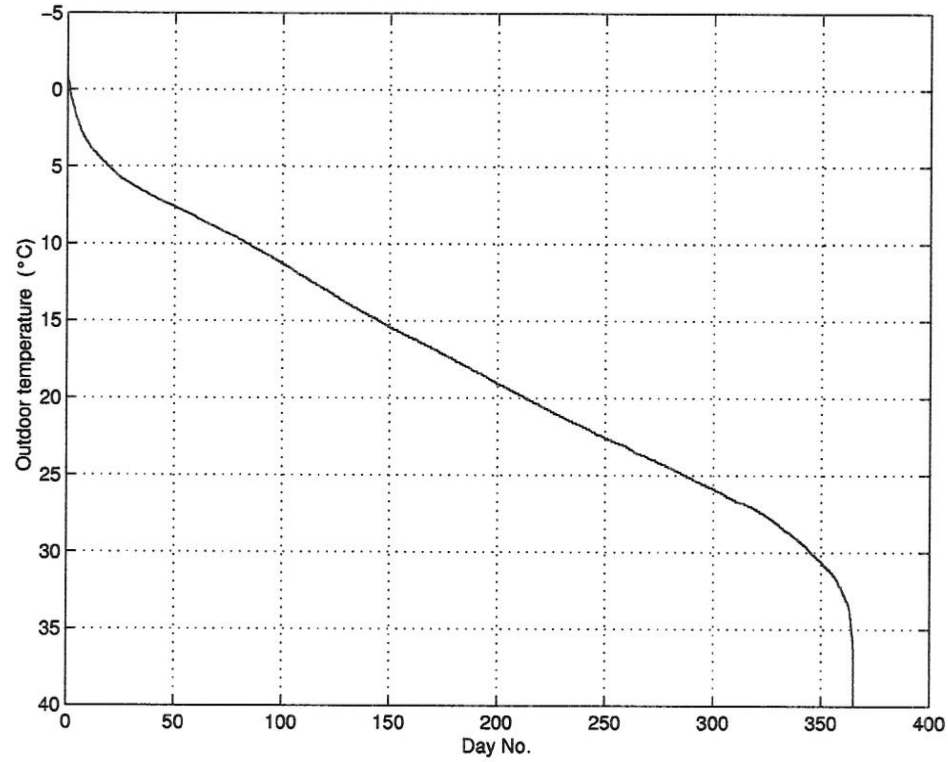


Figure 2.3: Outdoor air temperature duration curve

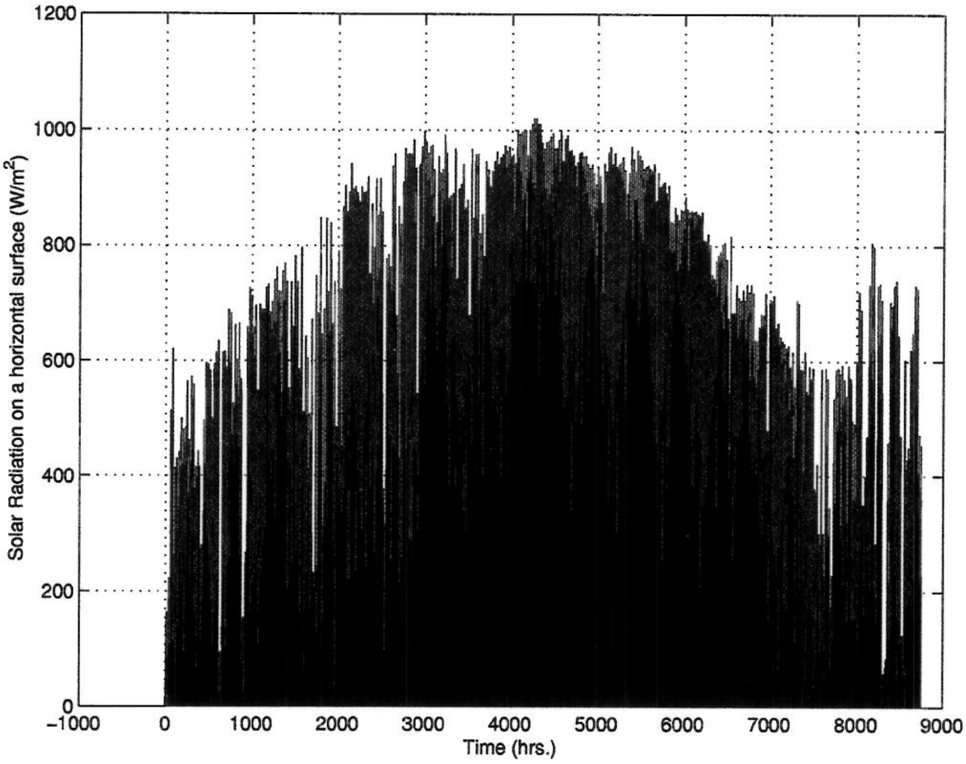


Figure 2.4: Solar Radiation profile for Amman

Chapter 3

Heat Transfer in Buildings

3.1 General

In order to determine the heating load in buildings, it is important to understand buildings dynamics and the interaction between the building's envelope and its environment.

3.2 Modes of Heat Transfer

Heat transfer is a transient flow of thermal energy from one system to another due to temperature difference between the two systems [2]. There are three modes of heat transfer: convection, radiation and conduction. In most real life cases these modes exist simultaneously, but in many cases the effect of one mode dominates the other two.

3.2.1 Convection heat transfer

In convection heat transfer, heat is transferred from one system to another by means of a moving fluid such as air or water. If the fluid is forced to move by a fan or a blower the convection process is called *forced convection.* If the fluid moves as a result of a difference in density of the fluid in the vicinity of the hot system compared with that away from it, the process is named *free convection.* The general equation for heat transfer by convection is given in the following relationship, which is known as Newton's law of cooling.

$$q_h = h \cdot A(T_w - T_o) \tag{3.1}$$

where:

q_h : Rate of heat transfer by convection [W];
h : Convection heat transfer coefficient $[W/m^{2\circ}C]$;
A : Heat transfer surface area $[m^2]$;
T_w : Wall surface temperature $[^\circ C]$;
T_o : Fluid temperature $[^\circ C]$;

The value of h for a given system can be determined analytically or experimentally depending on the complexity of the system.
Equation (3.1) can be rearranged as follows:

$$q_h = \left[\frac{T_w - T_o}{1/hA}\right] = \left[\frac{T_w - T_o}{R_{conv}}\right] \tag{3.2}$$

where $R_{conv} = 1/hA =$ Thermal resistance due to convection heat transfer.

3.2.2 Radiation heat transfer

In contrast to heat transfer by conduction and convection, which require a medium for their existence, heat transfer by radiation does not require a medium and can take place in vacuum. It is an electromagnetic radiation which is of the same nature as solar radiation. The radiation heat exchange between two objects is proportional to the fourth power of their absolute temperatures. The net heat exchange by radiation between two bodies is given by the following equation:

$$q_r = \sigma A_1 F_{1-2} \varepsilon (T_1^4 - T_2^4) \tag{3.3}$$

where:

q_r : Rate of heat transfer by radiation $[W]$;
σ : Stefan-Boltzman constant $[5.669 \times 10^{-8}[W/m^2K^4]$;
A_1 : Surface area of body 1 $[m^2]$;
A_2 : Surface area of body 2 $[m^2]$;
F_{1-2} : View or shape factor which indicates the fraction of energy leaving body 1 and reaching body 2;
ε : Common emissivity of the objects;
T_1 : Body 1 surface temperature $[K]$;
T_2 : Body 2 surface temperature $[K]$.

3.2.3 Conduction heat transfer

In the conduction mode of heat transfer, the systems are in physical contact and heat is transferred from one molecule to the adjacent one. Thus the high agitation of the hotter molecule is transferred to the cooler molecule. For example if one end of a metallic bar is placed in a hot medium such as open fire, the hand holding the other end feels the heat as a result of heat transfer from the first end of the bar to the other end from one molecule to its neighbor. It was observed by Fourier that the heat flux q_c/A in a given direction by conduction is directly proportional to the temperature difference in the direction of heat flow, and inversely proportional to the distance in the same direction. Thus,

$$\frac{q_c}{A} \sim \frac{\Delta T}{\Delta x} \tag{3.4}$$

For very small change in ΔT and Δx and changing the relationship (3.4) to an equality the Fourier's law is as follows.

$$q_c = -kA\frac{dT}{dx} \tag{3.5}$$

where, k is the thermal conductivity of the material through which conduction takes place. The negative sign is necessary because the direction of heat transfer is in the direction of decreasing temperature. k has the units of $W/m \cdot K$, and its values for the different materials can be obtained from tables. The differential Equation (3.5) can be simplified by assuming that the planes across which heat is transferred are homogeneous, the thermal conductivity can be assumed to be constant. Hence by integration the following will result.

$$\frac{q_c}{A} = \frac{k}{\Delta x}(T_1 - T_2) \tag{3.6}$$

where Δx is the thickness separating the surfaces which are at T_1 and T_2. By rearranging Equation (3.6) the following relation is obtained.

$$\frac{q_c}{A} = \left[\frac{(T_1 - T_2)}{\frac{\Delta x}{k}}\right] = \frac{(T_1 - T_2)}{R_c} \tag{3.7}$$

where R_c is the thermal resistance due to conduction, which is defined as follows:

$$R_c = \frac{\Delta x}{k} \tag{3.8}$$

The units of the thermal resistance, R_c is $°C \cdot m^2/W$.

However, a wall could be made of a number of materials in a number of layers, such a wall is called *composite wall*. Figure 3.1 shows that the heat flow is the same in each of the layers, thus.

$$q_c = q_{c1} = q_{c2} = q_{c3} \tag{3.9}$$

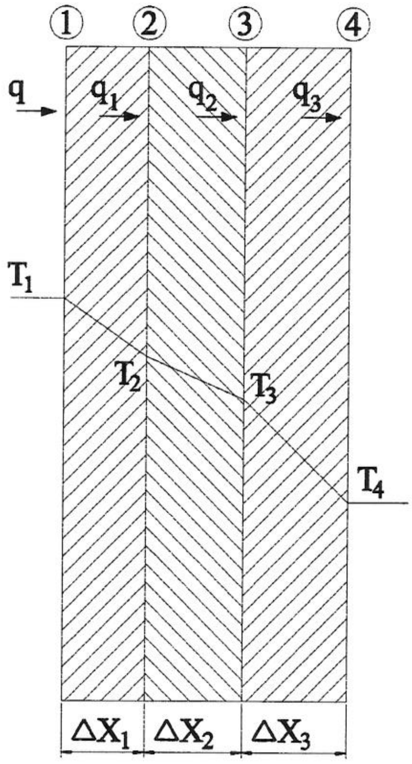


Figure 3.1: Heat transfer through composite walls

Substituting for each of the values in Equation (3.9) in terms of temperatures, the thermal conductivities, and thickness of layers, the following relationship may be obtained:

$$q_c = \left[\frac{(T_1 - T_2)}{\frac{\Delta x_1}{k_1 A}}\right] = \left[\frac{(T_2 - T_3)}{\frac{\Delta x_2}{k_2 A}}\right] = \left[\frac{(T_3 - T_4)}{\frac{\Delta x_3}{k_3 A}}\right] \tag{3.10}$$

where the subscripts are shown in Figure 3.1.

By solving Equation (3.10) the following equation is obtained:

$$\frac{q_c}{A} = \left[\frac{(T_1 - T_4)}{\frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3}}\right] \tag{3.11}$$

or, in terms of thermal resistance, Equation (3.11) becomes:

$$\frac{q_c}{A} = \left[\frac{(T_1 - T_4)}{R_1 + R_2 + R_3}\right] \tag{3.12}$$

where, R_1, R_2 and R_3 are thermal resistances of layers 1, 2 and 3 respectively.

In addition to the thermal resistances of materials out of which the wall is made, the air or fluid in contact with the surface forms a stagnant thin layer adjacent to the surface of the wall. The resistance of this layer, as mentioned before, in association of Newton's law of cooling is given by $1/h$, where h is the fluid film heat transfer coefficient. The thickness of the film and the thermal resistance it generates is a function of the wall geometry, the fluid flow velocity and other convection conditions such as boiling condensation, forced or natural convection.

Thus, if the fluid film resistances are included, Equation (3.12) becomes:

$$\frac{q}{A} = \left[\frac{(T_1 - T_4)}{R_i + R_1 + R_2 + R_3 + R_o}\right] \tag{3.13}$$

where R_i and R_o are the inside and outside film thermal resistances of the fluid films respectively. For buildings' air conditioning purposes, values of R_i and R_o are listed in Tables 3.1 and 3.2 for both construction materials [1] and metal surfaces [2].

Table 3.1: Inside film resistance, R_i

Element	Heat direction	Material type	R_i $m^{2\circ}C/W$
Walls	Horizontal	Construction Materials	0.12
		Metals	0.31
Ceilings and Floors	Upward	Construction Materials	0.10
		Metals	0.21
	Downward	Construction Materials	0.15

Hence, the overall thermal resistance of the heat transfer from the medium on one side of a composite wall to the medium on the other side is given as follows:

$$R_{th} = R_o + \sum_{k=1}^{n} R_k + R_i \tag{3.14}$$

Where n is the total number of homogeneous layers of the composite wall.

[1] Construction materials constitute of stones, concrete, bricks, and plaster.

Table 3.2: Outside film resistance, R_o

Wind speed (m/s)		Less than 0.5	0.5 - 5.0	more than 5.0
Element	Material Type	Outside film resistance, R_o $m^{2\circ}C/W$		
Walls	Construction materials	0.08	0.06	0.03
	Metals	0.10	0.07	0.03
Ceilings	Construction materials	0.07	0.04	0.02
	Metals	0.09	0.05	0.02
Exposed Floors	Construction materials	0.09	-	-

The overall heat transfer coefficient of the wall is defined as follows:

$$U = \left(\frac{1}{R_{th}}\right) \tag{3.15}$$

Therefore, the heat transfer from the building to its environment can be calculated using the following relation:

$$q = UA(T_i - T_o) \tag{3.16}$$

Where T_i is the inside design temperature and T_o is the outside design temperature respectively.

Chapter 4

Heating Load Calculation

The heat load of a building is comprised of the following components:

1. Heat loss through all exposed areas which consist of the walls, roof, windows, doors, ground and walls between the heated space and an unheated spaces.
2. Heat required to warm air infiltrated through cracks of windows and doors, and by opening and closing of doors and windows or to warm mechanical ventilation air to the temperature of the space.

The procedure for the calculation of the heat loss through walls was shown in Chapter 3.

4.1 Heat Loss by Infiltration

Infiltration is the leakage of outside air through cracks and clearances around the windows and doors. The amount of infiltration depends mainly on the tightness of windows and doors and on the outside wind velocity or the pressure difference between the outside and inside. Another factor that may contribute to the amount of infiltration is the *chimney effect* where warm air rises to the top of the building while outside air leaks through to replace it, thus creating a continuous draft. This chimney effect depends on the height of the building and is more pronounced in multistorey buildings.

The heat load due to infiltration consists of two parts. sensible heat load, and latent heat load. Sensible heat is the heat which raises the temperature of air. Latent heat is associated with the production or condensation of vapor at the same temperature.
The sensible heat load, Q_s, is given by the following equation:

$$Q_s = m_{oa} c_p (T_i - T_o) \tag{4.1}$$

where:

Q_s : Sensible load $[kW]$;
m_{oa} : Mass flow rate of infiltrated outside air $[kg/s]$;
c_p : Specific heat of air at constant pressure $[kJ/kg°C]$;
T_i : Indoor temperature $[°C]$;
T_o : Outdoor temperature $[°C]$.

The mass flow of infiltrated air, m_{oa}, can be calculated using the following relation:

$$m_{oa} = \rho V_{oa} \tag{4.2}$$

Where:

V_{oa} : Volume flow rate of infiltrated outside air $[m^3/s]$;
ρ : Density of infiltrated air $[kg/m^3]$.

The latent heat load is given by the following relation.

$$Q_L = m_{oa}(w_i - w_o)h_{fg} = \rho V_{oa}(w_i - w_o)h_{fg} \tag{4.3}$$

Where:

Q_L : Latent load $[kW]$;
w_i : Inside design humidity ratio;
w_o : Outside design humidity ratio;
h_{fg} : Latent heat of evaporation of water at inside design conditions $[kJ/kg]$.

4.2 Estimation of Infiltration

Two methods may be used to estimate the volume of infiltrated air into the heated space.

1. Air change method, which assumes that the air volume in a space is replaced by outside air a certain number of times per hour. The number of air changes depends on the type of space. The Jordanian heating code [10] recommend 1-2 air changes per hour for residential buildings.

2. Crack length method [2], which is based on the perimeter of the window or door, and the square root of the pressure difference across the crack.

Chapter 5

Heating System

At this point, a brief description of the basic heating system will be given. The heating system used in Jordanian buildings consists of (see also Figure 5.1):

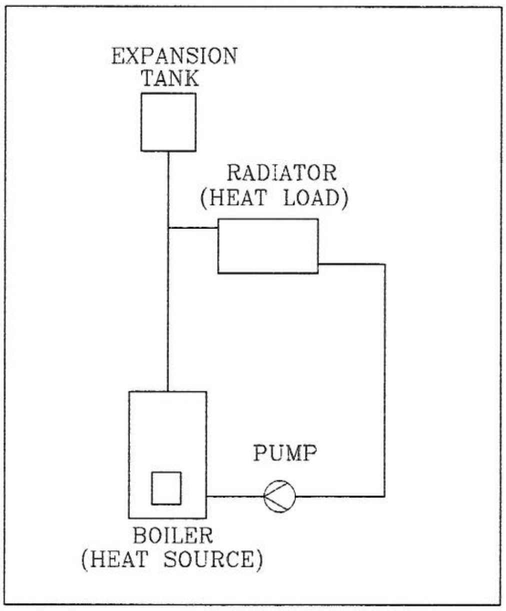


Figure 5.1: Basic heating system in Jordan

- Boilers: A boiler is a pressure vessel designed to transfer heat (produced by combustion) to a fluid. In most boilers the fluid is usually water in the form of liquid or steam.

  The boiler can be cast iron, steel or copper pressure vessel heat exchanger, designed with and for fuel-burning devices and other equipment to (1) burn fossil fuels (or use electric current) and (2) transfer the released heat to water. Boiler heating surface is the area of fluid-backed surface exposed to the products of combustion or the fire-side surface. Various codes and standards define allowable heat transfer rates in terms of heating surface. Boiler design provides for connections to a piping system, which delivers heated fluid to the point of use and returns the cooled fluid to the boiler.

- Radiators: Radiators are heat-distributing devices used in steam and high temperature water heating systems. They supply heat through a combination of radiation and convection and maintain the desired air temperature in a space without the use of fans.

- Expansion tank (chamber): The expansion tank serves both as a thermal function and a hydraulic function. In its thermal function the tank provides a space into which the non-compressible liquid can expand or from which it can contract as the liquid undergoes volumetric changes with changes in temperature. To allow for this expansion or contraction, the expansion tank provides an interface point between the system fluid and a compressible gas. As for its hydraulic function, the expansion tank serves as a reference pressure point in the system, analogous to a ground in an electrical system. Where the tank connects to the piping, the pressure equals the pressure of the air in the tank plus or minus any fluid pressure due to the elevation difference between the tank liquid surface and the pipe. The expansion tank also helps in detecting leaks in the network and it has a vent through which air is vented when it is liberated from the water when it is heated.

- Circulating pump: The pump is used to transfer the heating fluid (water), from the source (boiler) to the load (radiator), through the piping network since the fluid can not flow naturally. The pump is electrically driven and is mostly of centrifugal type.

These are the main components of the heating system used in Jordanian houses, other components such as controls will be discussed in more details in following chapters.

Chapter 6

Review of Codes

In order to be able to evaluate the building heating systems used in Jordan, a review of the related Jordanian codes and standards that control the design and installation of these systems and govern their operation must be conducted.

In this report two of the building codes are of interest. Those are the *insulation code* and the *central heating code.*

6.1 Insulation Code

The insulation code [7] sets the standards for buildings' parameters and determines the minimum requirements for the building insulation and allowable heat losses. It also outlines the heat load calculations where a detailed review of this code is presented below.

First the code divides the buildings into two groups:

1. All buildings that are used for residential purposes and have a central heating and / or cooling system.

2. All buildings that are used for residential purposes and do not have a central heating and / or cooling system.

Then the code sets the overall heat transfer values for the floor and the roof of the buildings which are shown in Table 6.1.

Table 6.1: Maximum overall heat transfer values for floor and roof.

Building type	U-value ($W/m^{2\circ}C$)
1	1.00
2	2.70

After that the code states the heat transfer coefficients for the walls, including doors, windows openings or any other parts, these values are given in Table 6.2.

Table 6.2: Overall heat transfer values for doors and windows.

Window or door material	Door	Single glazing	Double glazing
Wood	3.50	5.00	2.70
Steel[1]	5.8	6.70	3.50
Aluminum [1]	7.00	6.70	3.50
PVC	—	5.00	2.70

In any case the Overall heat transfer values of the walls should not exceed those given in Table 6.3.

Table 6.3: Walls Maximum overall heat transfer values.

Building type	U-value $(W/m^2{}^\circ C)$
1	1.80
2	2.70

The overall heat transfer coefficient, U, for the walls is calculated according to the following relation:

$$U = \frac{\Sigma\left(A_w U_w + A_\circ U_\circ\right)}{\Sigma\left(A_w + A_\circ\right)} \tag{6.1}$$

Where:

U_w : Overall heat transfer coefficient of construction materials $[W/m^2{}^\circ C]$;

$U_\circ$: Overall heat transfer coefficient of openings (windows and doors) $[W/m^2{}^\circ C]$.

The code also discusses thermal intermittence, where it states the although the calculations of the heating load is based on the worst outside conditions, these conditions are not the same all the time, and that the outside temperature changes during the day and from day to day and month to month. Hence, and for the sake of energy conservation, it is preferable to make some modifications

[1]Doors are considered metal if glass openings are $\leq 30\%$ of the total area.

to the heating load calculations on the basis of a percentage of the maximum heating load calculated. In order to determine these percentages, the code classifies the buildings in terms of **thermal inertia** into three categories:

Heavy	:	multistorey Buildings made of stones or concrete and have solid internal partitions.
Medium	:	Buildings made of light construction materials and have solid internal partitions.
Light	:	One storey buildings that have few or no internal partitions.

Then the code gives the thermal energy reduction percentage that the calculated heating load should be multiplied with. This is given in Table 6.4.

Table 6.4: Thermal energy reduction percentage due to building occupancy periods.

		Building type		
		Light	Medium	Heavy
Weekly occupancy period (Days)	(7)	1.00		
	(5)	0.75		0.85
Daily occupancy (hours)	(4)	0.68		0.96
	(8)	1.00		1.00
	(12)	1.25		1.02
	(16)	1.40		1.03

After applying this reduction factor, the code presents another reduction factor depending on the operation time of the heating system, and these values are presented in Table 6.5.

Table 6.5: Thermal energy reduction percentage due to heating system operation periods.

		Building's time lag	
		Small	High
Continuous operation		1.00	
Intermittent daily operation	Light buildings	0.55	0.70
	Medium buildings	0.70	0.85
	Heavy buildings	0.85	0.95

Hence the value of the heating load calculated must be multiplied by these two factors in order to get the final heating load for which the equipment should be sized.

Finally, the winter outside design temperature for Amman is stated as $+5°C$ by the code.

6.2 Heating Code

The second code that is of interest to this work is the heating code [10], which deals with the heating system design, maintenance and installation.

The code starts by suggesting the recommended indoor temperature for residential buildings, which is between 19 and 21 $°C$ and the required ventilation rate of 1–2 (Air changes / hour).

After that, the code assigns safety factors percentages for the boilers capacities that should be added to the calculated heating load, where these values are listed below:

- Boilers
  - The nominal size of the selected boiler should be 20% higher than the maximum load needed.
  - For small installations, or the cases at which the boiler is operated twice a day, the nominal capacity of the boiler should be 25–33% higher than the load.

Also the code dictates using a room–thermostat to control the indoor temperature of the building.

Then the code sets the maximum allowed temperature change at the boilers and radiators which are presented in Table 6.6.

Table 6.6: Maximum allowable temperature changes.

Radiator type	Maximum water temperature entering the network	Maximum allowable temperature change	
		At boiler $^{\circ}C$	At radiator $^{\circ}C$
Panel radiators	82	10	8
Convectors or unit heaters	82	10	7

Chapter 7

Building Under Study

7.1 General

In a survey conducted by the Ministry of Energy and Mineral Resources [8], it was found out that 53.6% of the buildings in Jordan are of single floor house and that 45.2% are apartments. The survey also revealed that almost 62.0% of the houses are built from concrete and building blocks, also 95.2% of the buildings' roofs were made of concrete. As for thermal insulation, the survey indicated that only 5.0% of the residences were insulated of which 42.1% used an air gap for insulation and 22.2% polystyrene panels.

7.2 Sample Building

For the purposes of this study, and in accordance with the survey results shown in section 7.1, the sample building chosen for simulation is of *house* type with concrete walls and roof. As for insulation, the house will first be considered to have no insulation, then the other types mentioned above will be simulated in order to determine their importance and feasibility.

The plan of the house chosen is shown in Figure 7.1 and the construction of the walls and roof is discussed in section 7.2.1. The house is a one story building built of concrete walls, floor and roof and it has single glazing with wooden doors.

7.2.1 Construction materials

In order to evaluate the building parameters, it is important to identify the properties of the construction materials used in the building. In this thesis the building will be divided into external walls, windows, roof, floor and internal walls. The construction of these surfaces are shown in Figure 7.2 and their

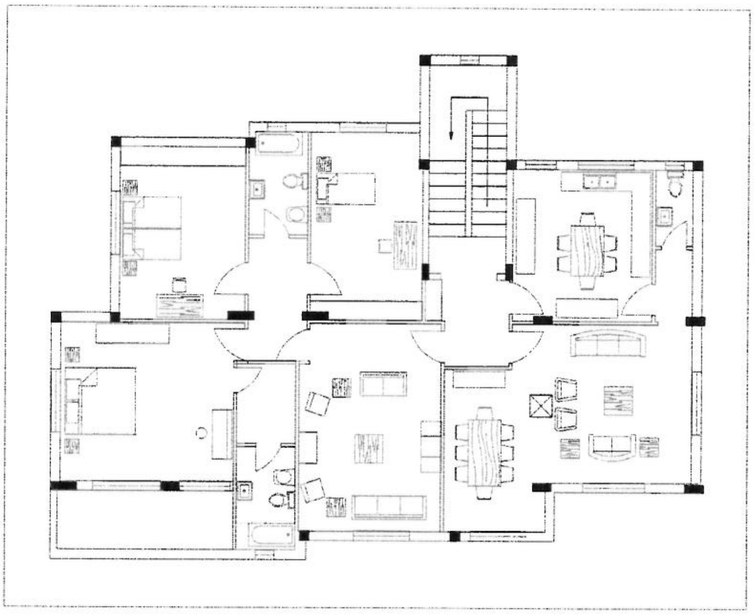
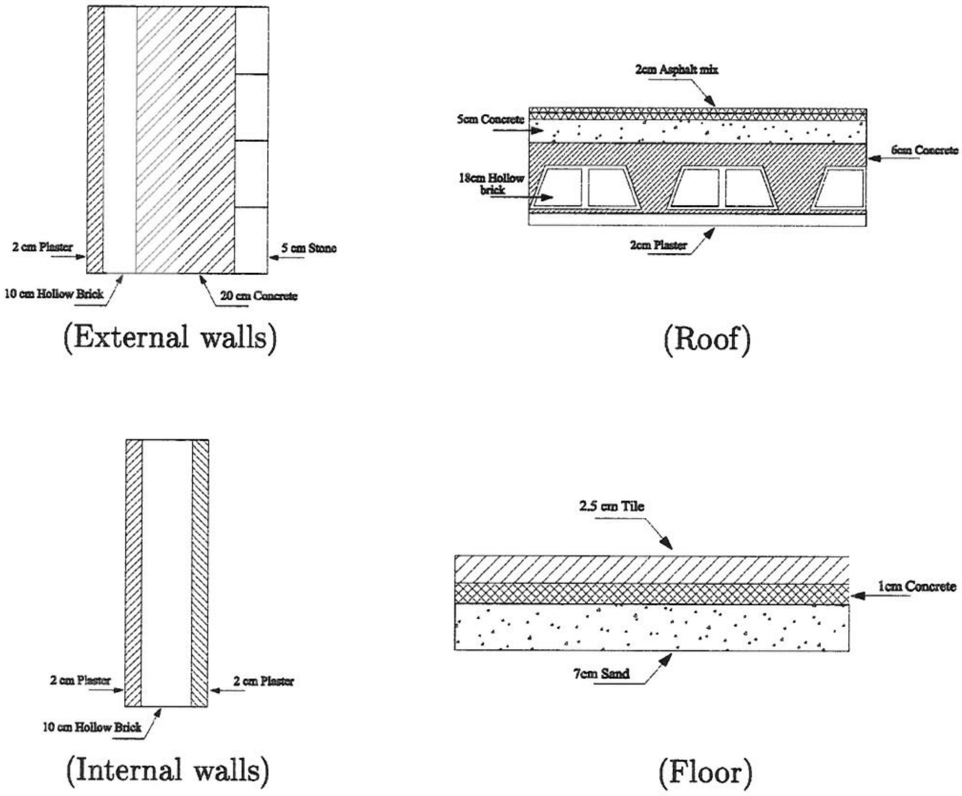


Figure 7.1: Sample house

properties given in Table 7.1.



(External walls) (Roof)

(Internal walls) (Floor)

Figure 7.2: Building surfaces construction layouts

Table 7.1: Construction materials' properties.

Construction material	Density (kg/m^3)	Thermal conductivity ($W/m°C$)	Thermal mass ($J/kg°C$)
Stone	2580	2.27	880
Concrete	2088	1.21	1080
Hollow brick	1380	0.73	1080
Plaster	2000	1.20	1000
Air gap	1.25	0.28	-
Polystyrene boards	30	0.04	-
Asphalt mix	2000	0.70	-
Roof bricks	1400	0.95	1080
Sand	1450	0.38	920
Cement Tiles	2145	1.35	960

7.2.2 Building parameters

In order to obtain realistic results from any simulation program, the parameters used by the program have to be as realistic as possible.
The important parameters that are needed in this project are the building's total heat transfer coefficient and its thermal mass.

1. **Heat transfer coefficient:** The Heat transfer coefficient is a constant describing the heat transfer between the building and its environment due to conduction, convection and radiation heat losses and is calculated according to the relation,

$$U_{total} = \frac{U_1 A_1 + U_2 A_2 + ...}{A_{total}} \tag{7.1}$$

where:

U_n : Heat transfer coefficient of the different components constituting the building's envelope (walls, windows ceilings and floors) $[W/m^{2}°C]$;

A_n : Each component surface area $[m^2]$.

A_{total} : Total surface area of the building $[m^2]$.

The heat transfer coefficient will be calculated for three different cases:

(a) No insulation.

(b) Air gap insulation (for walls only).

(c) Polystyrene insulation (for walls and roof).

As for the floor no insulation will be added.

- **Walls heat transfer coefficient:** By referring to Figure 7.2 the walls' total heat transfer coefficient is given in Table 7.2.

Table 7.2: External Walls heat transfer coefficient

Construction material	x (m)	k $(W/m^\circ C)$	R $(m^{2\circ}C/W)$
Stone	0.05	2.27	0.02
Concrete	0.20	1.21	0.17
Hollow brick	0.10	0.73	0.14
Plaster	0.02	1.20	0.02
R_i			0.12
R_o			0.03
		R_{total}	0.50

Hence $U_{wall} = 2.0\ (W/m^{2\circ}C)$

- **Roof heat transfer coefficient:** As for calculating the heat transfer coefficient for the roof, and by referring to Figure 7.2, it is seen that the construction materials of the roof are not homogeneous throughout the roof's surface, where the hollow bricks cover only 80% of the total roof surface, while the remaining 20% are filled with concrete. Hence two heat transfer coefficients, one for each zone, will be calculated and the roof's total heat transfer coefficient will be obtained using Equation (7.1).

  Tables 7.3 and 7.4 give the heat transfer values for the bricks and concrete zones respectively where the roof's total heat transfer coefficient is.

$$U_{roof} = \frac{U_1 \times A_1 + U_2 \times A_2}{A_{total}}$$
$$U_{roof} = \frac{2.28 \times 0.8 + 2.84 \times 0.2}{1}$$
$$U_{roof} = 2.39\ (W/m^{2\circ}C)$$

- **Floor heat transfer coefficient:** By using Figure 7.2, the floor's total heat transfer coefficient is given in Table 7.5.

- **Windows:** As for windows, the type used in Jordanian buildings is of single glaze, sliding type with a U value of 5.60 $(W/m^{2\circ}C)$.

Table 7.3: Roof's heat transfer coefficient with bricks

Construction material	x (m)	k $(W/m°C)$	R $(m^{2°}C/W)$
Asphalt mix	0.02	0.70	0.03
Concrete	0.05	1.75	0.03
Reinforced concrete	0.06	1.75	0.03
Hollow brick	0.18	0.95	0.19
Plaster	0.02	1.20	0.02
R_i			0.10
R_o			0.04
		R_{total}	0.44

Hence $U_1 = 2.28$ $(W/m^{2°}C)$

Table 7.4: Roof's heat transfer coefficient without bricks

Construction material	x (m)	k $(W/m°C)$	R $(m^{2°}C/W)$
Asphalt mix	0.02	0.70	0.03
Concrete	0.05	1.75	0.03
Reinforced concrete	0.24	1.75	0.14
Plaster	0.02	1.20	0.02
R_i			0.10
R_o			0.04
		R_{total}	0.35

Hence $U_2 = 2.84$ $(W/m^{2°}C)$

Table 7.5: Floor heat transfer coefficient

Construction material	x (m)	k $(W/m°C)$	R $(m^{2°}C/W)$
Sand	0.07	0.38	0.19
Concrete	0.01	1.40	0.01
Tiles	0.025	1.35	0.02
R_i			0.15
R_o			0.09
		R_{total}	0.45

Hence $U_{floor} = 2.21$ $(W/m^{2°}C)$

Building total heat transfer coefficient

After determining each surface's heat transfer coefficient, the buildings total heat transfer coefficient can be easily determined using Equation

(7.1), where Table 7.6 shows the results.

Table 7.6: Building's total heat transfer coefficient

Surface	U $(W/m^{2\circ}C)$	A (m^2)	$U \cdot A$ $(W/^{\circ}C)$
Roof	2.39	200	478.0
Floor	2.21	20	442.6
Walls	2.00	160	320.0
Windows	5.60	27	151.2
Total		**587**	**1391.8**

Where $U_{total} = 2.40\ (W/m^{2\circ}C)$

2. **Building Thermal Mass(C)** Thermal mass is the building's ability to store heat. It is the second parameter needed for the simulation, and is a property of the material itself which can be found in references such as the *The Guide to Buildings Thermal Insulation Materials* [9].

   As shown in Figure 3.1, which shows the heat transfer through composite walls, there is a temperature gradient through the wall's layers, with the lower temperature at the layer adjacent to the outside environment. This is clearly shown in Figure 7.3, which shows the temperature gradient through the different walls, where, for the uninsulated wall, it clearly shows that the surface layer of the outer most layer is $6.2°C$, while the inner most layer has a temperature of $16.9°C$, which is lower than the indoor temperature required, $20.0°C$.

   Hence the concept *effective thermal mass* is defined, which is the part of the wall that can store heat for an indoor temperature of $20.0°C$, and it can be calculated using the relation,

$$\frac{E}{A} = \int_0^S (T_x - T_o)c_p(x)\rho(x)dx = \int_0^S (T_i - T_o)c_p(x)\rho(x)dx \qquad (7.2)$$

where:

E	:	Wall's thermal energy $[kJ]$;
A	:	Wall's surface area $[m^2]$;
S	:	Wall's thickness [m];
T_x	:	Material's mean temperature $[°C]$;
c_p	:	Material's thermal mass $[kJ/kg°C]$;
T_i	:	Indoor temperature $[°C]$;
T_o	:	Outdoor temperature $[°C]$;
ρ	:	Material's density $[kg/m^3]$;

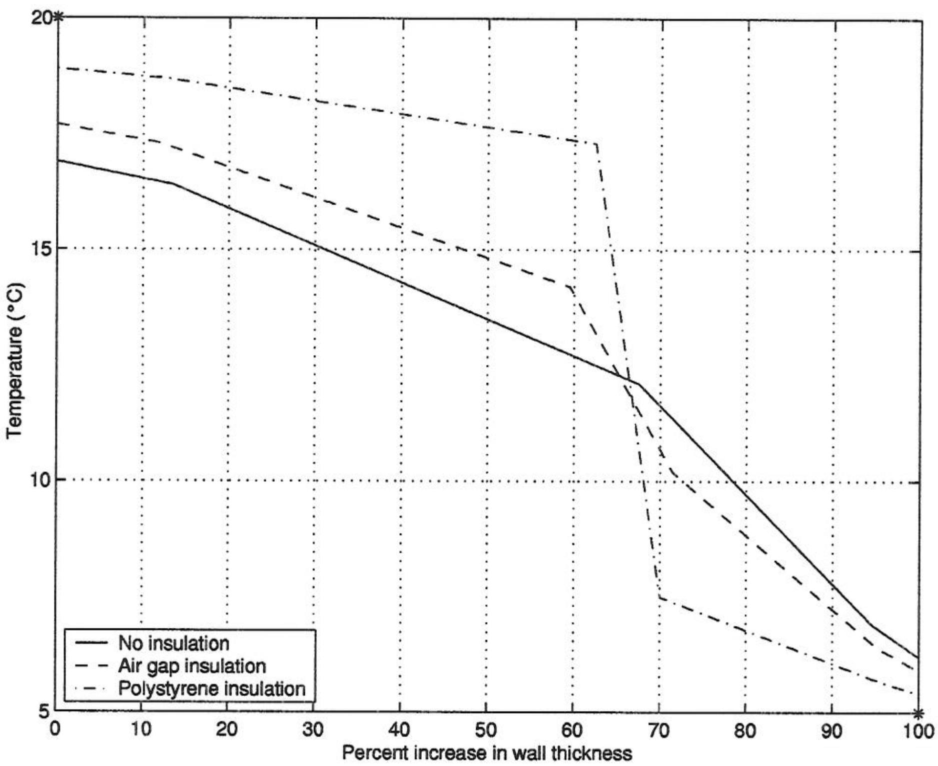


Figure 7.3: Temperature gradient through walls

Since the left hand side of Equation (7.2) is known (the sum of $[(T_x - T_o)c_p(x)\rho(x)]$ for all the different layers), the equivalent thickness (dx) of the wall needed if those layer's temperatures were equal to the indoor temperature can be determined.

Equation (7.2) is better understood using the following example.

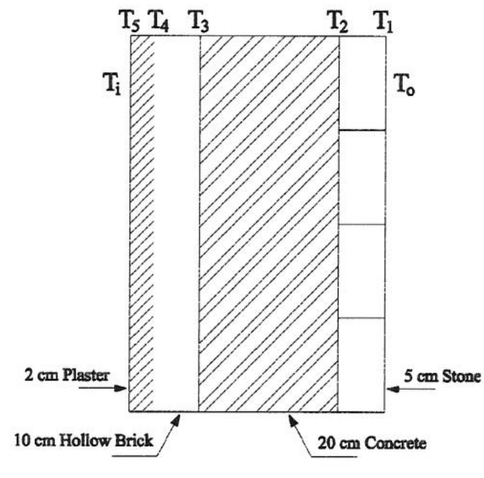


Figure 7.4: Sample wall

Figure 7.4 shows the wall under study, with an outdoor temperature of $5.0°C$, an indoor temperature of $20.0°C$ and a total heat transfer coefficient for this wall of $2.0W/m^2°C$, see Table 7.2. Using Equation (3.16) the total heat loss through this wall is $30W/m^2$. This heat loss is the same through each layer of the wall, as stated in Equation (3.9). Hence by equating Q, for the wall in Figure 7.4 one obtains

$$
\begin{aligned}
Q &= \left[\frac{(T_o - T_1)}{R_o}\right] = \left[\frac{(T_1 - T_2)}{R_1}\right] = \left[\frac{(T_2 - T_3)}{R_2}\right] \\
&= \left[\frac{(T_3 - T_4)}{R_3}\right] = \left[\frac{(T_4 - T_5)}{R_4}\right] = \left[\frac{(T_i - T_5)}{R_i}\right]
\end{aligned}
\tag{7.3}
$$

And by solving Equation (7.3) for the different surfaces temperatures one obtains

$$
\begin{aligned}
T_1 &= 6.2, \\
T_2 &= 6.9, \\
T_3 &= 12.1, \\
T_4 &= 16.4, \\
T_5 &= 16.9,
\end{aligned}
$$

Then the mean temperature of each layer is found subtracted from the outdoor temperature and then multiplied by that material's thermal capacity and density to get the results shown in Table 7.7.

Table 7.7: Wall's thermal capacity

Construction material	T_m $°C$	$(T_m - T_o)$ $°C$	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C (kJ)
Stone	5.46	0.46	0.05	160	2580	0.88	8355
Concrete	6.25	1.25	0.20	160	2300	1.08	99360
Bricks	13.73	8.73	0.10	160	1400	1.08	211196
Plaster	16.08	11.08	0.02	160	2000	1.00	70912
						C_{total}	389823

Table 7.7 represents the left hand side of Equation (7.2) where the right hand side of this equation is solved by replacing the mean surface temperature (T_m) with the indoor temperature (T_i), where only the layers whose sum gives the same as that of the total in Table 7.7 will be used in calculating the wall's thermal mass. Table 7.8 shows the equivalent

thermal mass of this wall, which is only the thermal mass of the plaster, and the bricks layers.

Table 7.8: Wall's equivalent thermal capacity

Construction material	$(T_i - T_o)$ $°C$	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C (kJ)
Bricks	15	0.10	160	1400	1.08	362880
Plaster	15	0.02	160	2000	1.00	96000
					C_{total}	458880

The same procedure applies to all building surfaces with the results given below.

- **External walls thermal mass:** external walls thermal mass as shown in Table 7.8 consist of only the plaster and the brick layers where their thermal mass is given in Table

Table 7.9: External walls thermal mass

Construction material	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C $(kJ/°C)$
Bricks	0.10	160	1400	1.08	24192
Plaster	0.02	160	2000	1.00	6912
				C_{total}	30592

- **Floor thermal mass:** As for the floor, its equivalent thermal mass consists of the concrete and tiles layers, where the results are shown in Table 7.10.

Table 7.10: Floor thermal mass

Construction material	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C $(kJ/°C)$
Concrete	0.01	200	2200	1.08	4752
Tiles	0.025	200	2145	0.96	10296
				C_{total}	15048

- **Roof thermal mass:** As for the roof of the building, its effective thermal mass is shown in Tables 7.11 and 7.12 for both bricks and concrete sections respectively.

Table 7.11: Roof thermal mass(with bricks)

Construction material	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C $(kJ/°C)$
Bricks	0.13	160	1400	1.08	31450
Plaster	0.02	160	2000	1.00	6400
				C_{total}	37850

Table 7.12: Roof thermal mass(without bricks)

Construction material	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C $(kJ/°C)$
Concrete	0.19	40	2300	1.08	18878
Plaster	0.02	40	2000	1.00	1600
				C_{total}	18878

Hence Roof total thermal mass is 37849.6+18878.4 = 56728.0$(kJ/°C)$

- **Internal walls thermal mass:** Although most of the internal walls are constructed as shown in Figure 7.2, some internal walls are constructed of 30 cm of concrete with plaster on both sides of the wall. Hence the thermal mass of both concrete and brick walls are given in Tables 7.13 and 7.14 respectively.

Table 7.13: Internal walls thermal mass(Concrete)

Construction material	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C $(kJ/°C)$
Plaster	0.02	26	2000	1.00	1040
Concrete	0.30	26	2300	1.08	19375
Plaster	0.02	26	2000	1.00	1040
				C_{total}	21455

Table 7.14: Internal walls thermal mass(Brick walls)

Construction material	x (m)	Area (m^2)	Density (kg/m^3)	Specific C $(kJ/kg°C)$	C $(kJ/°C)$
Plaster	0.02	147	2000	1.00	5880
Brick	0.10	147	1380	1.08	21909
Plaster	0.02	147	2000	1.00	5880
				C_{total}	33669

Internal walls total thermal mass = 21455.0+33669.0 = 55124.0$(kJ/°C)$

Building's total thermal mass (uninsulated)

The building's total thermal mass for the uninsulated case can be calculated by summing the thermal masses of the different surfaces of the building which are summarized in Table 7.15.

Table 7.15: Building's thermal mass (uninsulated case)

Surface	C $(kJ/°C)$
Roof	56728
Floor	15048
External Walls	30592
Internal walls	55124
Total	**157492**

Air gap insulation

As for the air gap-insulated case Figure 7.3 shows that the walls inner surface temperature is higher than that of the uninsulated wall. This means higher thermal mass, for the air gap insulated buildings, see Table 7.16.

Table 7.16: Building's thermal mass (air gap-insulated case)

Surface	C $(kJ/°C)$
Roof	56728
Floor	15048
External Walls	31104
Internal walls	55124
Total	**158004**

The table shows that for this case, the external walls' thermal mass value is the only value that differs from the uninsulated case. This is because these are the only surfaces that can be insulated with air which makes the value of the total thermal mass of the air gap insulation very similar to that of the uninsulated one.

Polystyrene insulation

As for the polystyrene insulated wall in Figure 7.3 it is clearly seen that the slope of the gradient significantly decreases behind the polystyrene layer indicating that the transfer rate is highly reduced, hence the importance of insulation. It also shows that the surface temperature of the indoor layer is near the indoor temperature ($18.9°C$), which reduces heat transfer through the wall and means that a bigger part of the wall can store energy leading to an increase in building's thermal mass, where the values of the building's different surfaces thermal mass are given in Table 7.17, showing a higher value than the other two cases.

Table 7.17: Building's thermal mass (polystyrene-insulated case)

Surface	C ($kJ/°C$)
Roof	81354
Floor	15048
External Walls	31104
Internal walls	55124
Total	**182630**

7.2.3 Summary of building parameters

As discussed above, the two building properties of interest are the heat transfer coefficient $U \cdot A$ and thermal mass C which a summarized in Table 7.18 for the three insulation cases.

Table 7.18: Building parameters

Building insulation	$U \cdot A$ ($kW/°C$)	C ($kJ/°C$)
No insulation	1.39	157492
Air gap insulation	1.32	158004
Polystyrene insulation	0.88	182630

Chapter 8

Building Heating System Sizing and Selection

8.1 Building Heating Load

As discussed in Chapter 4, the heating load of the building is divided into three components:

- Heat loss through all exposed areas which consist of the walls, roof, floor, windows, doors and walls between the heated space and an unheated space.
  This load is calculated using Equation (3.16), which is repeated here for convenience:

$$q = UA(T_i - T_o) \tag{8.1}$$

  Where (T_i = 20°C and T_o = 5°C). The heat losses for the three cases are presented in Table 8.1.

Table 8.1: Heat loss by conduction convection and radiation

Building insulation	q (kW)
No insulation	20.9
Air gap insulation	19.8
Polystyrene insulation	13.2

- Infiltration heat loss, and, which is the heating load due to outside air infiltrating to the building through cracks and openings. This heat load was discussed in chapter 4 and will not be calculated for this building due to the difficulty of simulating actual infiltration.

8.2 Equipment Selection and Sizing

After determining the heating load, the second step is system sizing, where the system, as shown in Figure 5.1, consists of boiler, radiator, piping and circulating pump.

8.2.1 Boilers

The boiler's heating capacity should be 20% higher than the required heating load [10].

8.2.2 Radiators

The heating code [10] specifies the entering water temperature by almost 80°C (82°C max temperature entering the network and allowing 2°C drop through piping) and a temperature drop across the radiator by 8°C, yet most radiator catalogues give heat output data at 90°C entering water temperature and 70°C leaving water temperature with a room temperature of 20°C. Hence in order to determine the needed size of the radiator, the following equation is used [3]:

$$Q/Q_o = (LMTD/LMTD_o)^{1.3} \tag{8.2}$$

where:

Q/Q_o	:	The actual heat output from the radiator divided by the heat output at design conditions;
$(LMTD)/LMTD_o)$	:	Ratio of logarithmic mean temperature difference at the actual supply temperature and design conditions respectively.

The Log mean temperature difference is calculated using the following relation:

$$LMTD = \frac{(T_1 - T_i) - (T_2 - T_i)}{\ln((T_1 - T_i)/(T_2 - T_i)} \tag{8.3}$$

with:

T_1 : Supply temperature of water [°C];
T_2 : Return temperature of water [°C];
T_i : Ambient air temperature in room [°C].

Hence taking:

T_{1o} : 80, °C
T_{2o} : 72, °C
T_1 : 90, °C
T_2 : 70, °C and
T_i : 20, °C

$$LMTD = \frac{(90-20)-(70-20)}{\ln((90-20)/(70-20))} = 59.4°C$$

and

$$LMTD_o = \frac{(80-20)-(72-20)}{\ln((80-20)/(72-20))} = 55.9°C$$

Inserting into Equation (8.2) one obtains,

$$Q/Q_o = (59.44/55.90)^{1.3} = 1.08$$

Where Q_o is the design heating load and Q is the actual required radiator output.
The radiator chosen has the following properties.

Height : 600 mm
Heat Output (Q) : 2225 W/m
water content : 5 kg/m

8.2.3 Heating water flow rate

The water flow rate $\dot{m}$ (kg/s) is calculated using

$$Q = \dot{m}c_p\Delta T \tag{8.4}$$

Where:

Q : The radiators heating capacity [kW];
c_p : Specific heat capacity of water [4.186 $kJ/kg°C$];
ΔT : Temperature difference between inlet and outlet of the radiator [8 °C] .

Equation (8.4) can be rearranged to get:

$$\dot{m} = \frac{Q}{c_p\Delta T} \tag{8.5}$$

8.3 Summary

Table 8.2 shows a summary of the system properties for the three cases to be simulated.

Table 8.2: Building heating system properties

Building insulation	Load (kW)	Boiler capacity (kW)	Radiator heat output (kW)	Water flow rate (kg/s)
No insulation	20.9	25	22.6	0.67
Air gap insulation	19.8	23.8	21.4	0.64
Polystyrene insulation	13.2	15.8	14.3	0.43

The radiators' details can be obtained from manufacturers' catalogues and are shown in Table 8.3

Table 8.3: Heating radiators properties

Radiator heat output (kW)	Length (m)	Surface area (m^2)	Water content (kg)
22.6	10.0	6.0	50
21.4	9.6	5.8	48
14.3	6.4	3.8	32

Chapter 9

System Simulation

After determining the heating system's properties and parameters, system simulation can be performed in order to evaluate the system's efficiency in performing its duty for which it was designed. Yet in order to properly evaluate the system and have reasonable results reflecting the actual performance of the system, the simulation program has to be carefully designed and built.

The accuracy of a simulation program depends on many factors, the most important one is finding accurate *mathematical models*, choosing an appropriate *simulation program* and executing the program in the right sequence (*algorithm*).

In this chapter, a detailed discussion of the mathematical models chosen to represent the different system components will be presented along with a brief description of the chosen simulation program and the algorithm will be presented.

9.1 Building Model

A lumped model describing the heat dynamics of the building was chosen [5] with the following assumptions.

- The heat capacity is concentrated in a thin layer inside the wall.
- All surfaces apart from windows, are considered to have the same temperature.
- The heat capacity of the room air is neglected.

After considering these assumptions, the following differential equation regarding the heat dynamics of the building is obtained

$$C \cdot \frac{\partial T_i}{\partial t} = q_{net} = q_{in} - q_{out} \tag{9.1}$$

where:

C	:	Building's thermal mass $[kJ/°C]$;
T_i	:	Indoor temperature $[°C]$;
t	:	Time [seconds];
q_{net}	:	Net heat added to the building [kW];
q_{in}	:	Heat added to the building, heating equipment, solar energy and people $[kW]$;
q_{out}	:	Heat lost from the building to the outside by conduction, convection and radiation $[kW]$;

q_{out} can be expressed by the following relation:

$$q_{out} = U \cdot A(T_i - T_o) \tag{9.2}$$

As for the heat gain, only the gain from the heating system (radiators) and the sun will be considered, while heat gains from the people and lights will not be considered in the mathematical model. Hence the heat gain from the radiators is discussed in section 9.2. As for the solar energy effect, a detailed discussion is presented here.

9.1.1 Solar effect

In calculating the heating load for building using a static method, the maximum load is calculated at the maximum temperature difference between inside and outside the building envelope. Since solar radiation reduces the heat losses from the building, it is not taken into consideration.

On the other hand solar radiation is taken into account when calculating the cooling load of buildings, and ASHRAE (American Society of Heating Refrigerating and Air conditioning Engineers) [4], presents many ways for calculating heat gain due to solar radiation e.g cooling load temperature difference (CLTD) and Sol-Air temperature. Unfortunately these methods include complex calculations and require big changes in the simple lumped model that is being used in this thesis. Hence an attempt to find simple relationship between the Solar radiation and the heat loss of building will be tried and used, if successful, in evaluating the solar effect on heat dynamics of buildings.

In order to determine the solar effect it is assumed that the Solar radiation will affect only the roof of the building. This is because the solar energy effect on the walls is quite complicated and is affected by the wall orientation and

windows and shading effects, also most houses in Jordan have external shutters which are used when heating is on.
In this thesis, the effect of the solar energy will be considered using the Sol-Air temperature method.

Sol-Air Temperature calculation

Definition: Sol-Air temperature is the temperature of the outdoor air that, in the absence of all radiation changes, gives the same rate of heat entry into the surface as would the combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air [4].
Heat flux into exterior sunlit surfaces. The heat balance at a sunlit surface gives the heat flux into the surface $Q_s/$A as:

$$Q_s/A = \alpha I_t + h_\circ(T_o - T_s) - \varepsilon \Delta R \tag{9.3}$$

Where:

Q_s/A	:	Heat flux transferred into the surface $[W/m^2]$;
α	:	*Absorptance* of surface for solar radiation;
I_t	:	Total solar radiation incident on surface $[W/m^2]$;
$h_\circ$	:	Coefficient of heat transfer by long wave radiation and convection at outer surface $[W/m^2]$;
$T_\circ$	:	Ambient temperature $[°C]$;
T_s	:	Surface Temperature $[°C]$;
ε	:	Hemispherical emittance of surface;
ΔR	:	Difference between long wave radiation incident on surface from sky and surroundings and radiation emitted by blackbody at outdoor air temperature $[W/m^2]$.

Assuming the rate of heat transfer can be expressed in terms of sol-air temperature T_e.

$$Q_s/A = h_\circ(T_e - T_s) \tag{9.4}$$

Hence

$$T_e = T_\circ + \frac{\alpha I_t}{h_\circ} - \frac{\varepsilon \Delta R}{h_\circ} \tag{9.5}$$

ASHRAE [4] provides the following values for the different parameters,

α	=	$0.45 - 0.9$ Depending on surface colour where 0.45 for light and 0.9 for dark or unknown surfaces;
$h_\circ$	=	17.034 $[W/m^2]$;
ε	=	1;
ΔR	=	60.31 $[W/m^2]$.

Hence T_e can be calculated as a function of $T_\circ$ and I_t alone and the heat loss through the roof can be calculated in terms of T_e according to

$$Q_s = U_r \cdot A_r(T_i - T_e) \tag{9.6}$$

with:

Q_r	:	Heat loss from the roof $[W]$;
U_r	:	Roof total heat transfer coefficient $[W/m^{2\circ}C]$;
A_r	:	Roof surface area $[m^2]$.
T_i	:	Indoor temperature $[^\circ C]$.

But adding this term to the building model, described in Equation (9.1), will invalidate the assumptions that this equation was built on since the sun–lit surfaces' temperature will be different from the temperatures of the rest of the surfaces. Hence Equation (9.1) will be invalid and can not be used anymore.

From the above discussion, it is clear that a simpler term has to be found to express the effect of the sun on the building heat dynamics and at the same time keep the validity of the simple model of the building represented by Equation (9.1). This is done by trying to find a simple relation between the intensity of the solar radiation (I_t), and the solar heat gain (Q_s) associated with it. This is done by calculating the heat loss through the roof with the outside temperature equal to $T_\circ$ using.

$$Q_r = U_r \cdot A_r(T_i - T_\circ) \tag{9.7}$$

Then the heat loss using the Sol-Air temperature (Q_s) according to Equation (9.6) is calculated, where the value of (T_e) was obtained from Equation (9.5) for actual values of solar radiation (I_t). After that, (Q_s) is subtracted from (Q_r), giving the value of the heat gain due to solar radiation which is plotted against the solar radiation intensity as shown in Figure 9.1. Using Figure 9.1 a simple relationship between the solar heat gain to the building and the solar radiation is then obtained by curve fitting. This relationship is expressed in Equation (9.8) which can then be easily fitted into Equation (9.1), as part of q_{in}, and hence into the simulation program.

$$Q_s = [0.05283 \cdot I_t - 3.7037]/1000 \tag{9.8}$$

where:

Q_s	:	Heat gain due to solar radiation $[kW/m^2]$;
I_t	:	Solar on a horizontal surface $[W/m^2]$.

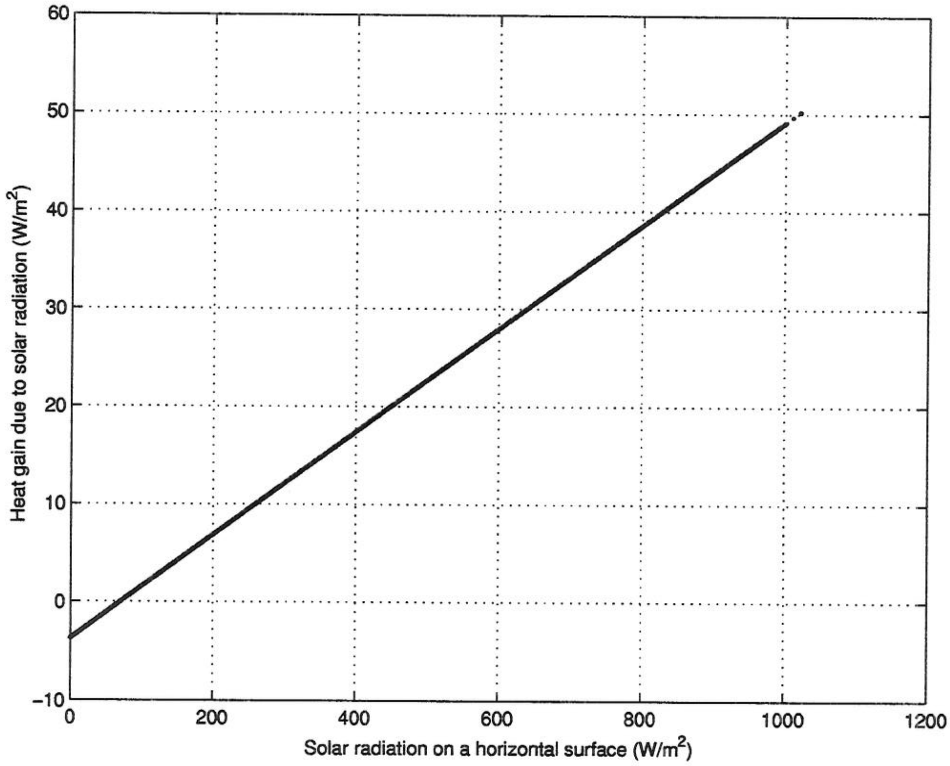


Figure 9.1: Building heat gain due to solar radiation

9.2 Radiator Model

The radiator is considered to be a heat exchanger between the heating water and the room air. Yet there are two cases that must be dealt with in the radiator model, the first is when there is water flow through the radiator and the second is when the water flow is zero, but the water inside the radiator is hot. Mathematical models for these two cases are presented below.

9.2.1 Water flow rate is not zero

In this case the rate of heat transferred from the water side to the ambient air can be expresses in the following, mathematical equation:

$$Q = U \cdot A \cdot LMTD \tag{9.9}$$

where:

Q	:	Rate of heat added to the room air temperature [kW];
U	:	Radiator overall heat transfer coefficient $[kW/m^{2\circ}C]$;
$LMTD$	:	Log mean temperature difference $[^\circ C]$.

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

$$LMTD = \frac{(T_s - T_i) - (T_r - T_i)}{\ln((T_s - T_i)/(T_r - T_i))} \tag{9.10}$$

Where:

T_s	:	Radiator water supply temperature [$°C$];
T_r	:	Radiator water return temperature [$°C$];
T_i	:	Room ambient temperature [$°C$].

The heat transfer from the water can be written as:

$$Q = \dot{m} \cdot c_p (T_s - T_r) \tag{9.11}$$

Where:

$\dot{m}$	:	Mass flow through the radiator [kg/s];
c_p	:	Specific heat capacity of water [$kJ/kg°C$];

Substituting Equations (9.10) and (9.11) and solving for T_r gives

$$T_r = \frac{T_s - T_i}{\exp((A \cdot U)/(m \cdot c_p))} + T_i \tag{9.12}$$

Since the overall heat transfer coefficient U is a function of T_r, iteration is necessary to determine T_r.

Due to the difficulty of calculating the overall heat transfer coefficient at different return temperatures, the heat output from the radiators has traditionally, according to the German standard DIN 4703 [3], been related to the arithmetic mean temperature difference raised to the power 1.3 as given in Equation (8.2) Hence by substituting Equations (9.10) and (9.11) into Equation (8.2), one obtains

$$\frac{m(T_s - T_r)}{m_o(T_{so} - T_{ro})} = \frac{\frac{T_s - T_r}{\ln((T_s - T_i)/(T_r - T_i))}}{\frac{T_{so} - T_{ro}}{\ln((T_{so} - T_{io})/(T_{ro} - T_{io}))}} \tag{9.13}$$

Where the (o) means the states at design conditions which are constants. From Equation (9.13) T_r can be found in three terms, hence one of these T_r's should be chosen to be on the left hand side of the equation in order to perform the iteration. The T_r inside the logarithm of the denominator of the right hand side of Equation (9.13) is chosen yielding

$$T_r = T_i + \frac{T_s - T_i}{\exp\left[\frac{\left(\frac{T_s - T_r}{T_{so} - T_{ro}}\right)^{1/4} \cdot \ln\left(\frac{T_{so} - T_{io}}{T_{ro} - T_{io}}\right)}{\left(\frac{m}{m_o}\right)^{3/4}}\right]} \tag{9.14}$$

Hence Equation (9.14) can be used to find the radiator water return temperature at any condition.

9.2.2 Water flow rate is zero

In this case, a lumped cooling model is used [6], where the water inside the radiator is assumed to cool down in an exponential way with its initial conditions taken from the model when the flow was not zero in section 9.2.1. This can be represented by the following relation.

$$\frac{T_s(t) - T_i}{T_{so} - T_i} = \exp\left[-\frac{U \cdot A}{M_r \cdot c_p} \cdot \Delta t\right] \tag{9.15}$$

Where:

$T_s(t)$	:	Water temperature at the inlet of the radiator at any time $[°C]$;
T_i	:	Ambient air temperature $[°C]$;
T_{so}	:	Water temperature at the inlet of the radiator at $t = 0$ $[°C]$;
U	:	Radiator overall heat transfer coefficient $[kW/m^2 °C]$;
A	:	Radiator surface area $[m^2]$;
M_r	:	Water mass inside the radiator $[kg]$;
Δt	:	Time step $[seconds]$.

As it is difficult to mathematically determine the overall heat transfer coefficient U in Equation (9.15) due to its high dependence on the fluid temperatures, a term can be found to express U. This term can be obtained from Equation (9.9), where Equation (9.15) becomes:

$$\frac{T_s(t) - T_i}{T_{so} - T_i} = \exp\left[-\frac{\frac{Q \cdot A}{A \cdot LMTD}}{M_r \cdot c_p} \cdot \Delta t\right] = \exp\left[-\frac{Q}{LMTD \cdot M_r \cdot c_p} \cdot \Delta t\right] \tag{9.16}$$

The heat output Q can be found from Equation (8.2) and the log mean temperature difference ($LMTD$) can be found in Equation (9.10).
Accordingly the final form of T_s, can be expressed by

$$T_s(t) = T_i + (T_{so} - T_i) \cdot \exp\left[-\frac{Q}{LMTD \cdot M_r \cdot c_p} \cdot \Delta t\right] \tag{9.17}$$

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

$$T_r(t) = T_i + (T_{ro} - T_i) \cdot \exp\left[-\frac{Q}{LMTD \cdot M_r \cdot c_p} \cdot \Delta t\right] \tag{9.18}$$

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

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

1. Calculate new T_s from Equation (9.17).

2. Calculate new T_r from Equation (9.18).

3. If $T_r \leq T_i \Longrightarrow T_r = T_i + 0.1$.

4. If $T_s \leq T_i \Longrightarrow T_s = T_i + 0.1$.

5. If $T_s \leq T_r \Longrightarrow T_s = T_r + 0.01$.

6. Calculate new $LMTD$ from Equation (9.10).

7. Calculate new Q from Equation (8.2).

9.3 Boiler Model

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

$$\frac{\partial T_b}{\partial t} = \frac{q_b - \dot{m} \cdot c_p(T_b - T_r)}{M_b \cdot c_p} \tag{9.19}$$

Where:

T_b	:	Boiler outlet temperature $[°C]$;
t	:	Time $[seconds]$;
q_b	:	Boiler capacity (Heat input) $[kW]$;
$\dot{m}$	:	Water flow rate to the building [kg/s];
c_p	:	Water specific heat capacity $[kJ/kg°C]$;
T_r	:	Radiator return water temperature $[°C]$;
M_b	:	Boiler water content $[kg]$.

integrating equation 9.19, over a time step of Δt yields

$$T_b(t) = \frac{q_b + \dot{m} \cdot c_p \cdot T_r - q_r}{\dot{m} \cdot c_p} \tag{9.20}$$

where:

$$q_r = [q_b + \dot{m} \cdot c_p \cdot (T_r - T_{b(old)})] \cdot \exp\left(-\frac{\Delta t \cdot \dot{m}}{M_b}\right) \tag{9.21}$$

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

9.3.1 Boiler control

The boiler has a control thermostat that is used to control the outlet temperature from the boiler, and this thermostat has a so called "Dead Zone",which is a range of temperatures at which below or above the thermostat temperature and at which the controller does not change the state of the boiler (on or off). This is done to reduce the on-off work and hence elongate the life of the boiler.

The boilers' efficiency is considered to be constant throughout its operation and it was assumed to be 85% which is an average value obtained from catalogues.

9.4 Simulation Period

According to the survey that was conducted by the Ministry of Energy and Mineral Resources [8], the heating season starts in November and ends in April, where the heating system is turned off during the rest of the year.
Hence the simulation program will account only for this time period (November to April) and will ignore the rest of the year, making the simulation time 4341 hours instead of 8761 hours.

9.5 Simulation Program

The first step in simulating a certain system is to understand how this system operates, its control sequence, the different components' parameters and the variables affecting these components. Figure 9.2 shows the different components of the heating system under study along with the controls and variables affecting the operation of these parts.

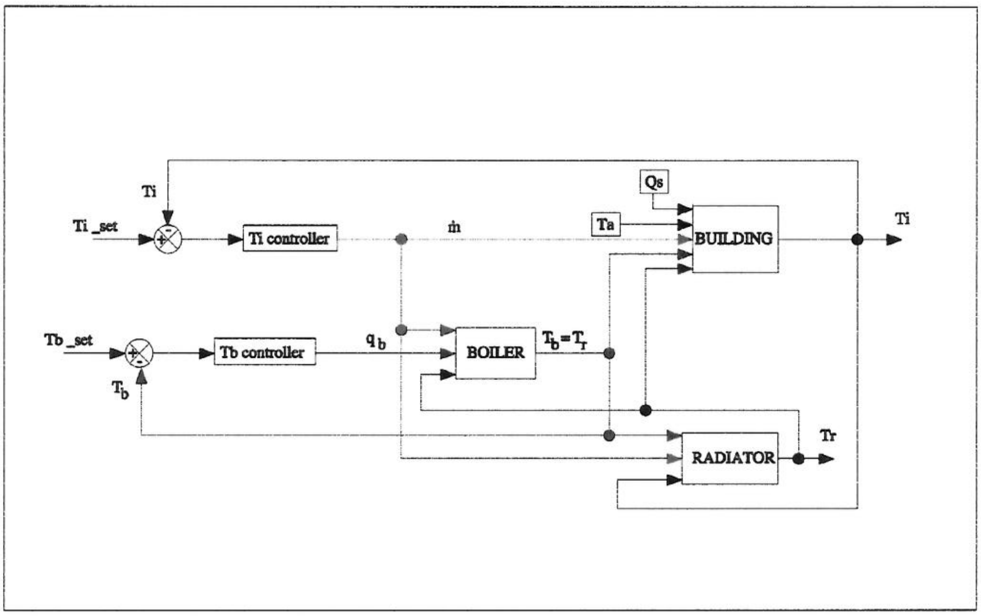


Figure 9.2: Heating system components

The figure shows that the thermostat inside the building senses the indoor temperature and compares it with the preset temperature. If the measured value was equal to or greater than the preset temperature, the thermostat automatically stops the boiler and the circulating pump(i.e. q_b and $\dot{m} = 0$). On the other hand if the measured temperature is less than the preset temperature, the thermostat starts both the boiler and the circulating pump in the heating system ($q_b = q_{max}$) & ($\dot{m} = \dot{m}o$). The figure also shows that the boiler has another controller, (thermostat) which turns the boiler off ($q_b = zero$) whenever the boiler water temperature exceeds a certain value ($80°C$).

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

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

Since most equations representing the different components of the system are differential equations, the program was written in a forward time step mode, that is the properties of the different parameters are given at the beginning and end of the integration period and not on a continuous mode, where the differential equations were first integrated in terms of the time step (dt) and then used in the program. Hence the present value of a certain parameter is obtained from values of the parameters from the previous step, this is shown in Equation (9.22), where the expression for the indoor temperature (T_i) at present time step is obtained from the parameters of the previous time step.

$$
\begin{aligned}
T_{i(i)} &= \frac{q_{r(i-1)} + Q_{s(i-1)} + UA \cdot T_{o(i-1)}}{UA} \\
&- \frac{e^{(-UA \cdot dt/C)}[q_{r(i-1)} + Q_{s(i-1)} + UA(T_{o(i-1)} - T_{i(i-1)})]}{UA}
\end{aligned}
\tag{9.22}
$$

Where the subscript $(i-1)$ means the value from the previous step. Because of the assumptions made above and in order to reduce errors and get as much information as possible about the system, the time step chosen was short (180 seconds).

Program's algorithm

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

The program starts by acquiring the outdoor conditions for the simulation period from another file called (start1.m). Then the program zeros all values to be evaluated later. After that it interpolates the values of outdoor temperature (T_o) and at the required time intervals (180 seconds). It also calculates the solar heat gain (Q_s) in accordance with Equation (9.8). At this point the simulation loop starts by calculating the indoor temperature (T_i) value according to Equation (9.22), then it compares this value with the thermostat set temperature ($T_{i(set)}$). If ($T_i > T_{i(set)}$), the program sets the flow to zero, otherwise it sets the flow to maximum value ($\dot{m}_o$).

If the value of the flow ($\dot{m}$) was not zero, the program first calculates the boiler temperature (T_b) which is also considered to be the supply temperature (T_s) to the radiators, then it calculates the return temperature from the radiators (T_r) according to Equation (9.14) by an iterative technique for which a subroutine called (rett.m) is used. After that the radiator heat output (q_r), and the log mean temperature of the radiator (T_m).

On the other hand, if the flow value was zero, the program skips the above equations and calculates the radiator supply (T_s) and return (T_r) water temperatures using Equations (9.17) and (9.18) respectively. Then the program compares the values of (T_r), (T_s) and (T_i) two at a time, and if any two were equal to each other it increases their values so that (T_s) would be highest and (T_r) is second highest where the amount of increase of (T_r) is ($0.1°C$) above (T_i) and (T_s) is ($0.01°C$) above (T_r). This done so as not to get complex numbers when calculating the log mean temperature of the radiator, and the increase is so small that it does not affect the results. Then the program calculates the log means temperature of the radiator, and sets the boiler water temperature (T_b) equal to the radiator return water temperature (T_r), it also gives the boiler power (q_b) a value of zero. After that the loop starts again. This logic is more clearly shown in Figure 9.3.

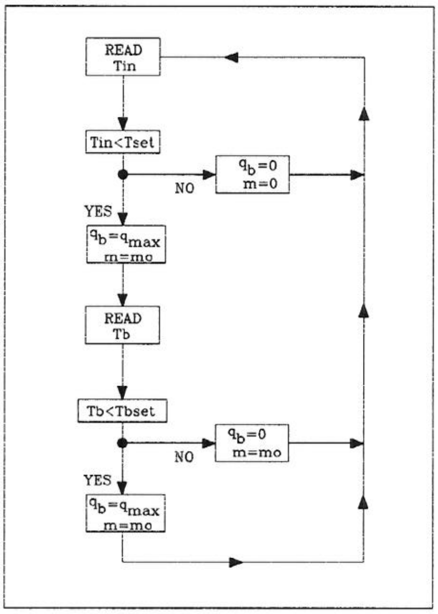


Figure 9.3: Program algorithm

Chapter 10

Results

In this chapter a discussion of the results obtained from the simulation program for the different building types and control systems are presented along with a sensitivity analysis that was carried out on the different systems with the aim of studying the effect of the different parameters on the building under study.

10.1 Time Constant (τ)

The time constant of the building can be defined as the time needed to reach 63.2% of its steady state value.

The time constant of the building can be determined in two ways. The first way is using the simulation program, which is done by adding a step change in the outdoor temperature and measure the time needed for the indoor temperature to reach steady state again. Figure 10.1, shows the response to the step input for the different insulation types, where the input was introduced at time = 100 hours. The other method is to calculate the time constant, assuming lumped heat capacity system [6], using the following relation,

$$\tau = \frac{C}{U \cdot A} \tag{10.1}$$

where:

τ	:	Building time constant[sec.];
C	:	Building thermal mass $[kJ/°C]$;
$U \cdot A$	:	Building's total heat transfer coefficient $[kW/°C]$.

Table 10.1 gives the values for the time constants obtained from both methods. The figure clearly shows that the time constants for the air gap and uninsulated buildings are very close to each other, while it is higher for the polystyrene insulted building.

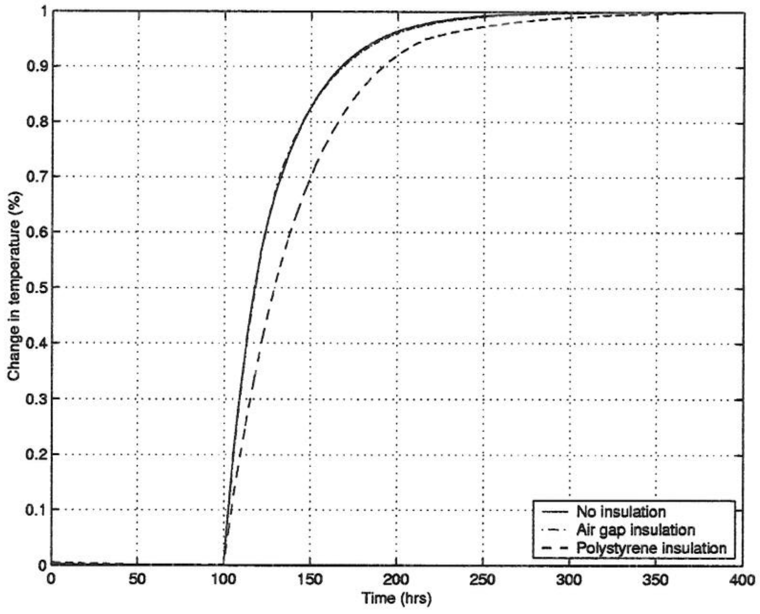


Figure 10.1: Indoor temperature response to a step input

Table 10.1: Building time constant

Insulation type	$U \cdot A$ $(kW/°C)$	C $(kJ/°C)$	(τ) from simulation (hours)	Calculated (τ) (hours)
None	1.39	157492	26.3	31.5
Air gap	1.32	158004	26.4	33.3
Polystyrene	0.88	182630	42.3	57.7

The difference in time constants values between the calculated and simulated values is due to the assumption made in constructing the model where the highest difference in values was for the polystyrene insulated building 27% and lowest for the uninsulated building 16%.

The results obtained above clearly show that the lower the heat transfer coefficient (i.e the better the insulation is), the higher the time constant of the building.

The effect of the time constant on building indoor conditions can be viewed by reviewing the indoor temperature profiles for the uninsulated, air gap insulated and polystyrene insulated buildings, shown in Figures 10.2, 10.3 and 10.4 respectively where it is seen that the values of the indoor temperature at a given time is not the same for the three buildings although they have identical control systems.

This is stressed more clearly in Figures 10.5 and 10.6 which show the indoor

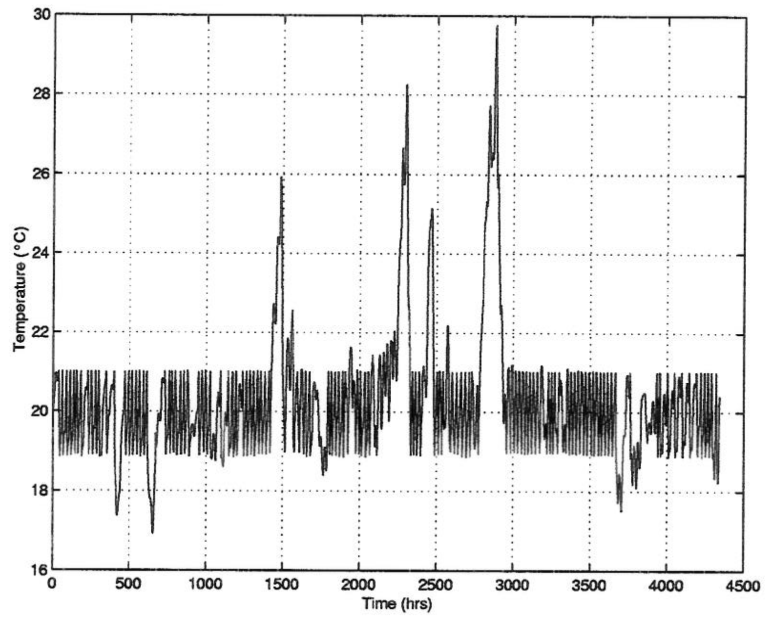


Figure 10.2: Indoor temperature profile (no insulation)

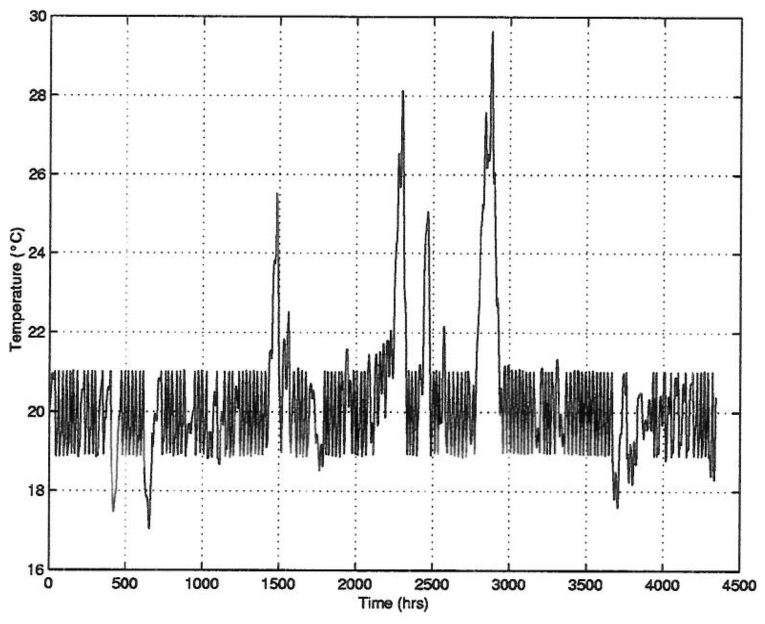


Figure 10.3: Indoor temperature profile (air gap insulation)

temperature profiles for the polystyrene and uninsulated building and air gap insulated and uninsulated building respectively. This difference in the temperature profiles is due to the difference in the time constants of the buildings where it takes more time for the insulated building to cool to the lower limit than the uninsulated building. By comparing Figures 10.5 and 10.6, it is seen that the difference between the air gap insulated building and the uninsulated building is lower than that between the polystyrene insulated building and

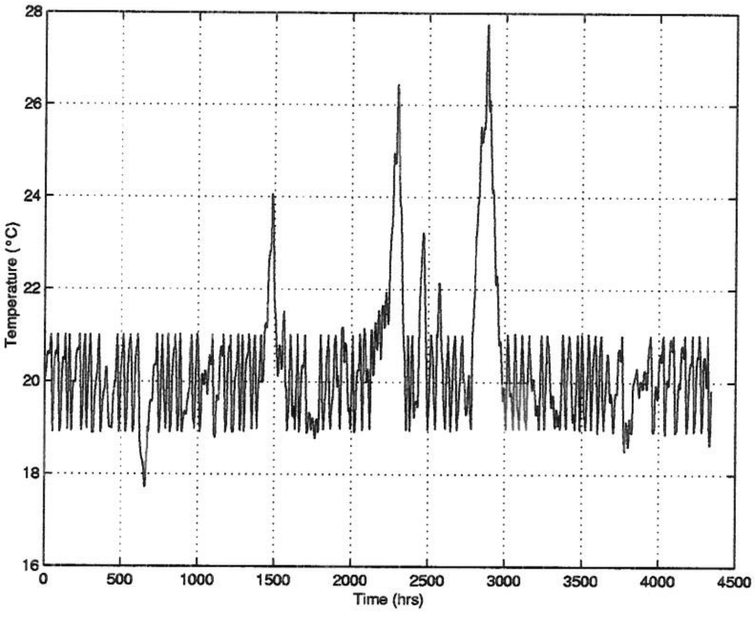


Figure 10.4: Indoor temperature profile (polystyrene insulation)

the uninsulated building. This is due to the fact that the difference in time constant is 1.78 hours while it is 26.2 hours between the polystyrene insulated and the uninsulated buildings.

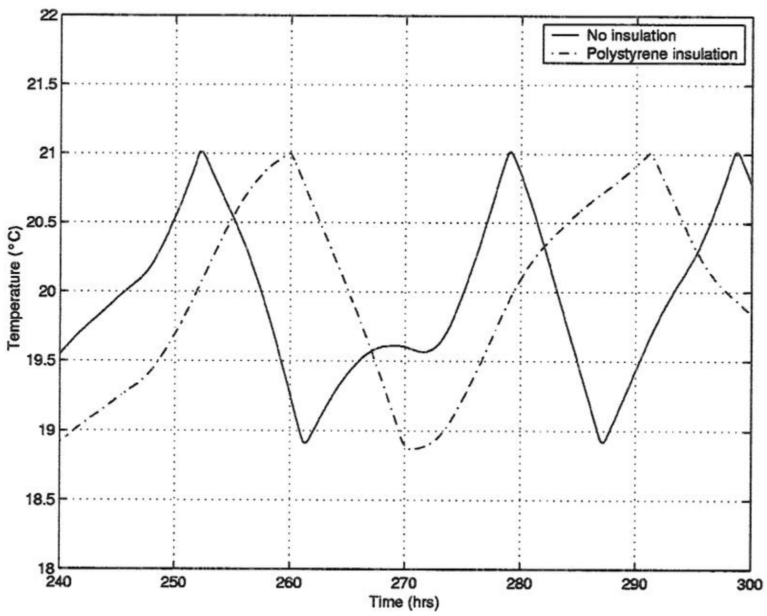


Figure 10.5: Indoor temperature profiles for polystyrene insulated and uninsulated buildings

Finally Figures 10.7 and 10.8 give the difference in indoor temperature values between the polystyrene insulated and uninsulated buildings, and the air gap

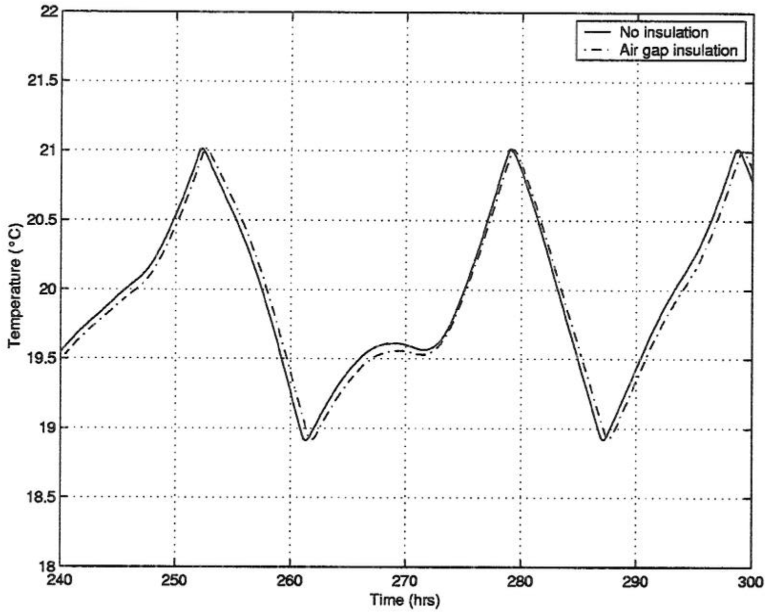


Figure 10.6: Indoor temperature profiles for air gap insulated and uninsulated buildings

insulated and uninsulated buildings for the whole simulation period. The figures clearly show that the difference for the uninsulated and air gap insulated buildings is negligible. On the other hand it is almost 2.5 $^{\circ}C$ between the polystyrene and uninsulated buildings.

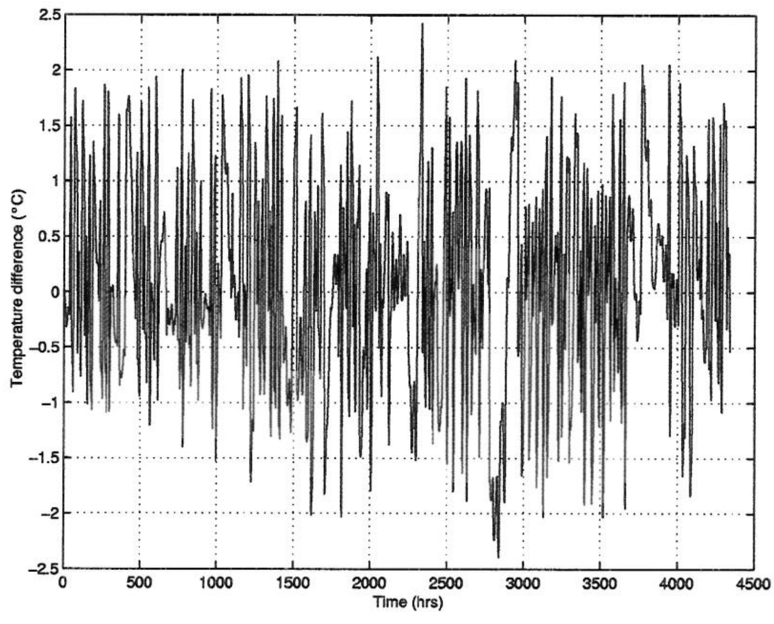


Figure 10.7: Temperature difference between polystyrene and uninsulated buildings

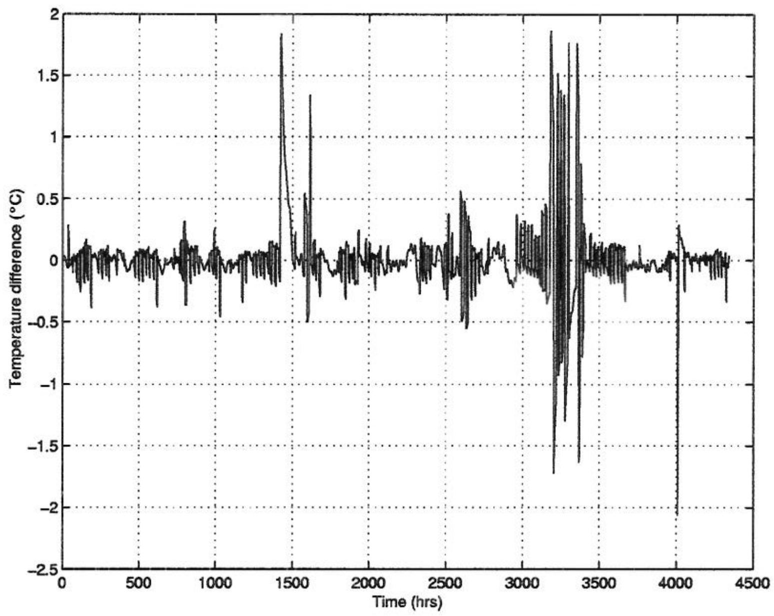


Figure 10.8: Temperature difference between air gap and uninsulated buildings

Determining the time constant of the system helps in designing the control system of the building which will be explained in more details in later sections.

10.2 Building Insulation

As explained in Chapter 7, building insulation tends to decrease the value of the heat transfer coefficient of the building, hence reducing heat leaving the building to its environment. It also increases the building's thermal mass as well as increasing its time constant, as shown in Table 10.1. In addition Figure 7.3 shows that the indoor surface temperatures of external walls and surfaces are higher for the insulated buildings than those of the uninsulated buildings, leading to reduction in radiation heat transfer from occupants to these surfaces, which increases the occupants' feeling of comfort. Yet adding insulation imposes extra cost on the building which, in turn, has to be evaluated. The evaluation of the feasibility of the insulation will be considered in terms of the reduction in fuel consumption that insulation provides while the other benefits of the insulation will be considered as an advantage to insulation.

Before evaluating the financial feasibility of insulation a description of the indoor conditions of the building for the three different cases (no insulation, air gap insulation and polystyrene insulation) will take place.

Comfort conditions

There is no rule dictating the best atmospheric conditions for humans. This is because the feeling is affected by several factors such as health, age, activity, etc. In this project the minimum allowed indoor temperature for the building is $18.0°C$.

Table 10.2 gives the minimum temperature reached within the building, it also gives a measure of the severity of the indoor conditions, where this factor represents the area of the curve under the value of $20.0°C$, which is calculated by multiplying the difference between the $20.0°C$ and the temperatures below it with time. The higher the value of this number the worse the indoor conditions are. This factor can be defined as Degree Hours which is similar to the concept of Degree Days.

Table 10.2: Building indoor conditions

Building insulation	*min.temp.* ($°C$)	Degree Hours
No insulation	17.0	1522
Air gap insulation	17.0	1503
Polystyrene insulation	17.6	1179

By reviewing Table 10.2, it is seen that the minimum indoor temperature for the three cases is below $18.0°C$, where the highest was for the polystyrene insulated case $17.6°C$ and the lowest for the uninsulated and air gap insulated case $17.0°C$. Yet the minimum required indoor temperature was not satisfied in any of the three cases above. The reason for this, is perhaps the cold waves that affect the weather in which the outdoor temperature decreases to values below the design conditions $5.0°C$ as shows Figure 10.9 where the minimum temperature was $-1.0°C$ and that the temperature is below $5.0°C$ for about 453 hours during the heating period.

As for the degree hours, Table 10.2 shows that the lowest value of this criterion was also for the polystyrene insulated building showing that the indoor conditions are more favorable in the insulated building than in the uninsulated building.

10.2.1 Financial evaluation of building insulation

In addition to improved indoor conditions achieved when insulating the building, Table 8.2 shows that smaller equipment are needed to supply the required heat for the building when insulation is used. Thus the size of the system

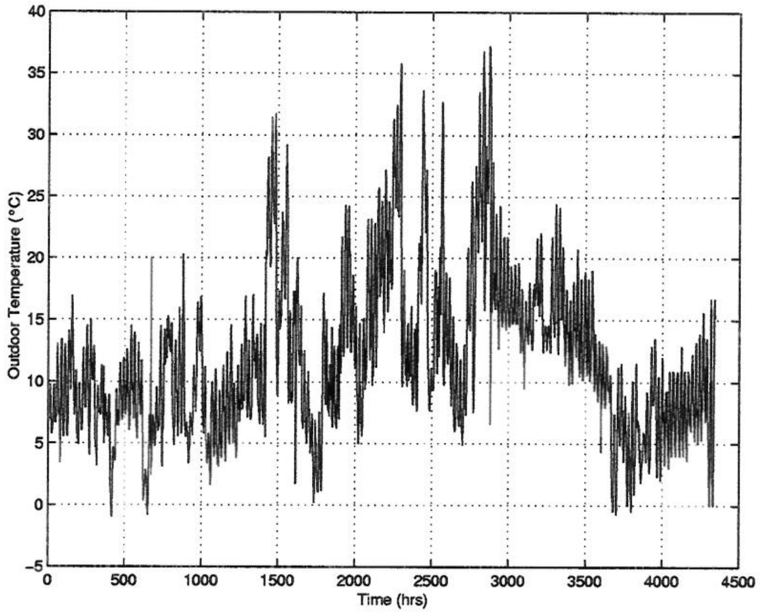


Figure 10.9: Outdoor temperature profile

for the polystyrene insulated building is 40% smaller than for the uninsulated building, hence reducing initial and running costs of the system.

In evaluating the financial profitability of adding insulation to the building only the reduction in fuel consumption will be discussed where reduced initial cost and improved indoor condition will be considered as extra benefits for insulation.

Financial evaluation of projects can be conducted using many techniques such as the Internal Rate of Return (IRR), and the Conventional Payback Period [1].

The internal rate of return for an investment is the interest rate (i) at which the cash flow is discounted, the equivalent present worth of all benefits exactly equals the equivalent present worth of project costs [1] i.e.

$$P = \sum_{j=0}^{N} A_j (1+i)^{-j} \tag{10.2}$$

where:

P	:	Initial investment. Equivalent present value of future cash flows;
N	:	Number of years of the project;
A_j	:	Amount of payment at the end of each period;
i	:	Internal rate of return.

The Conventional Payback [1] is calculated using,

$$P = \sum_{j=0}^{N*} A_j(1+i)^{-j} \tag{10.3}$$

where $N*$ is the number of payback periods. In case of equal payments $N*$ can be expressed as,

$$N* = \frac{P}{A_j} \tag{10.4}$$

In order to calculate the internal rate of return for this project, the project's life is assumed to be 15 years. Polystyrene insulation costs around 3 JOD/m^2, for all external areas, which can be obtained from Table 7.6, hence the insulation total cost would be,

$$(160 + 200) \times 3 = 360 \times 3 = 1080\,\mathrm{JOD}^1 \tag{10.5}$$

As for the annual savings in fuel consumption, Table 10.3 gives the fuel consumption for the three cases along with its annual cost, where the savings are obtained by subtracting the fuel cost for the insulated case from the fuel cost for the uninsulated case, yielding 250 JOD in annual savings. Hence by solving Equation (10.2) for i, an internal rate of return (IRR) value of 22% would be obtained, the project also has payback period of 4.3 years. Hence in addition to improved indoor conditions, insulation is financially feasible.

Table 10.3: Fuel consumption

Building	fuel consumption (L)	Fuel cost (JOD)
No insulation	6024	657
Air gap insulation	5696	621
Polystyrene insulation	3733	407

10.3 Intermittent Heating Systems

So far the control systems under discussion have aimed at keeping the indoor temperature of the whole building at 20.0°C during the heating period with no regard to building occupancy periods or building need for heating (continuous heating). In an attempt to reduce the energy consumption the Jordanian codes [7] & [10], suggest the use of another control system, called Intermittent heating system, which means that the heating system is turned on at certain hours of the day, when occupants are in the building for example, and it is turned off for the rest of the day. The intermittent heating concept aims at

[1] 1.00 JOD = \$1.41

providing the building with the required heat and keep it at the design temperature for certain times only and not for the whole heating period.

There are two methods to implement intermittent heating control systems, the first of which is to totally shut off the heating system during non heating periods with no regard to the indoor temperature during this time. This system is suggested by the Jordanian codes and will be referred to as intermittent heating. The other method is to lower the indoor thermostat temperature during non heating hours, such that the temperature would be $15.0°C$ instead of $20.0°C$ for example. This method is suggested by the American Society for heating Refrigeration and Air conditioning Engineers (ASHRAE) [4] and is usually used in order to prevent the temperature inside the building to go to very low values, resulting in damage to the building or items inside. Figures 10.10 and 10.11 show the heating system operation logic for intermittent and set back systems respectively.

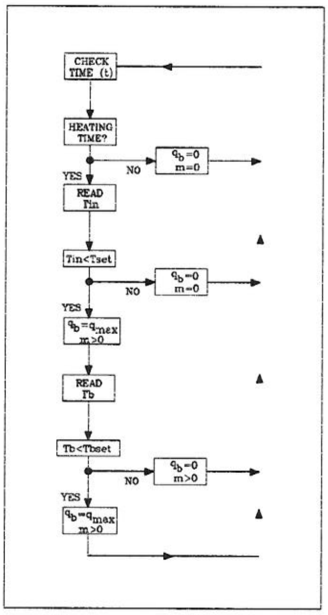


Figure 10.10: Intermittent heating control system

The Jordanian insulation code [7] suggests modification to the heating load calculation procedure when intermittent heating is used, where it suggests multiplying the equipment capacities, obtained from calculating the heating load, by three factors, from Tables 6.4 and 6.5 depending on the type of the building, occupancy period and heating period. In this project, the building is considered to be heavily occupied in the period of 7 days for 16 hours per day with intermittent use of heating equipment.
By using the above mentioned tables one obtains,

$$\text{Factor} = 1.0(7\,\text{days}) \times 1.03(16\,\text{hours}) \times 0.95\,(\text{intermittent use}) = 0.98 \quad (10.6)$$

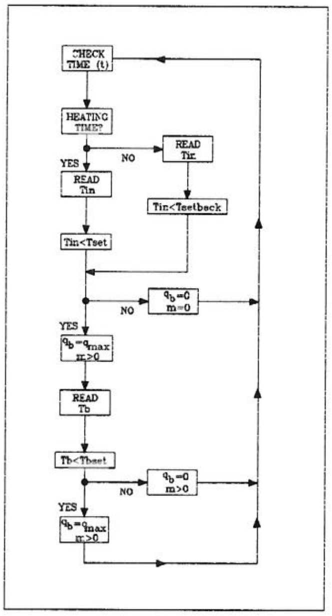


Figure 10.11: Set back heating control system

This means a reduction of 2% in the equipment capacity. Table 10.4 gives the properties of the modified heating system.

Table 10.4: Heating system properties modification (Jordanian code)

	Boiler capacity (kW)	Radiator heat output (kW)	Water flow rate (kg/s)
Continuous heating	15.8	14.3	0.4
Intermittent heating	15.5	14.0	0.4

In Jordan the heating period is considered to be from 15:00-23:00 hours which is time period people finish work and sleep. Hence, in the simulation program, the system was turned on one hour earlier and turned off at the end of this period with the results shown in Figure 10.12. The figure shows that the indoor temperature of the building never reaches 20.0°C during cold weather, on the contrary it reaches very low values 10.7°C during the heating period, making the building uncomfortable and cold for the occupants. Furthermore the building had degree hours value of 4811 during heating periods indicating the poor conditions inside the building, while the fuel consumption was 2163 liters due to the discontinuous use of the heating system, as expected.

By reviewing Figure 10.13 which is a plot of indoor temperature profile along with the set temperature, it is clearly seen that the indoor temperature reaches its maximum value at the end of the heating period i.e when it is not required

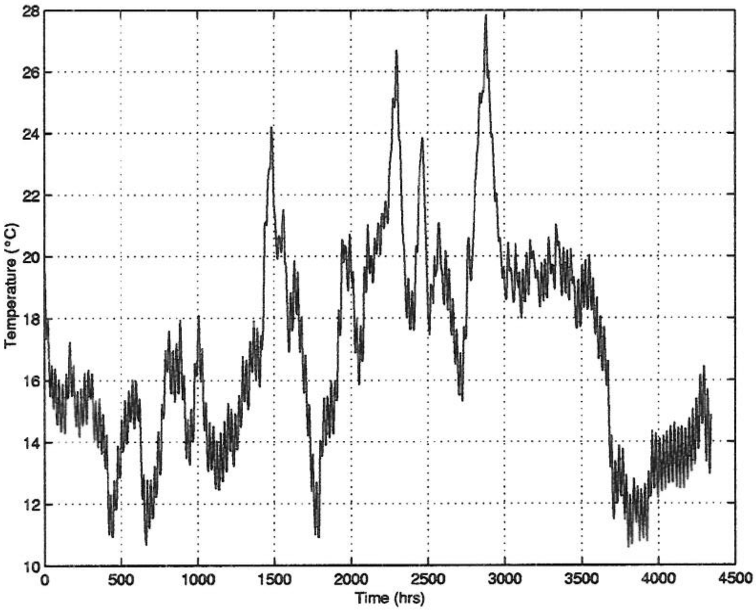


Figure 10.12: Indoor temperature profile for intermittent heating system (Jordanian code). Insulated building

to. This is due to the building's high thermal mass and long time constant resulting in a time shift between adding heat to the building and a change in the building temperature.

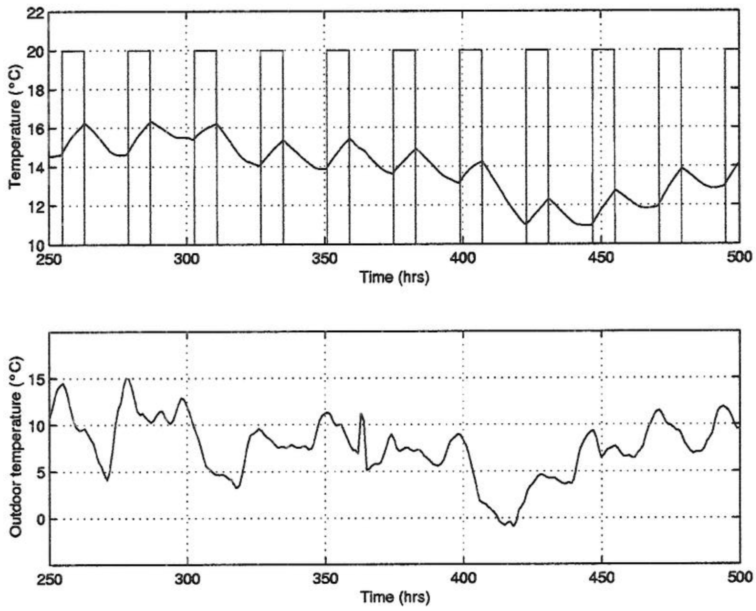


Figure 10.13: Indoor temperature profile against required temperature using intermittent heating system. Insulated building

On the other hand ASHRAE [4] indicates that for a set back of 5°C a 40%

increase in equipment sizes is needed, refer to Table 10.5 for new system parameters. The reason is that the building's temperature has to be raised to the design temperature over a short period of time.

Table 10.5: Heating system properties modification (ASHRAE recommendations)

	Boiler capacity (kW)	Radiator heat output (kW)	Water flow rate (kg/s)
Continuous heating	15.8	14.3	0.4
Set back heating	22.2	20.0	0.6

Figure 10.14 shows the indoor temperature profile where, as in the intermittent case, the indoor temperature does not reach 20°C when it is required to, alternatively minimum indoor temperature during heating time was 14.2°C and the degree hours for the time when the required indoor temperature was 20.0°C was 2584. As for the fuel consumed by this system its value was 2904 liters.

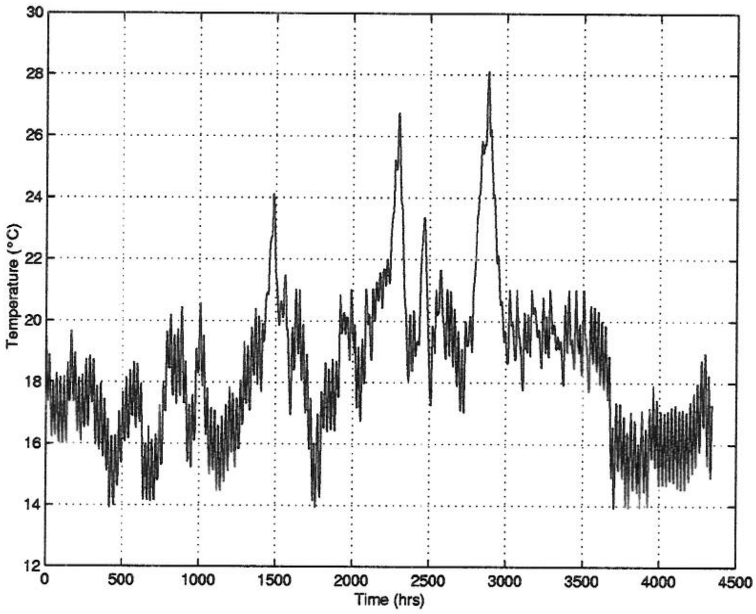


Figure 10.14: Indoor temperature profile for set back heating system (ASHRAE). Insulated building

As the case with the intermittent heating Figure 10.15 shows also that indoor temperature does not reach its maximum when needed, for the same above mentioned reasons. Hence new techniques should be found to implement this control system in order to improve its performance.

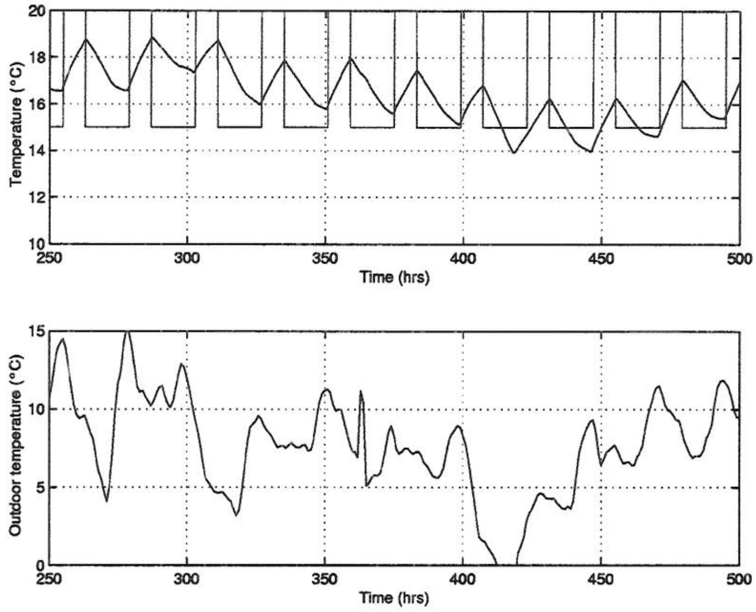


Figure 10.15: Indoor air temperature profile against required temperature, using set back heating system. Insulated building

10.4 Thermostatic Flow Valves

Another device that can be used to control indoor temperature is the *Thermostatic flow valve* which is a valve that can be installed at radiator outlet where it controls the heating water flow proportionally to the indoor temperature, where the flow of heating water would be maximum at the minimum allowable (in this case $19.0°C$) indoor temperature and zero at the maximum allowable temperature ($21.0°C$) while it is in-between these values if the indoor temperature is within those limits. The indoor temperature profile and heating water flow rate are shown in Figures 10.16 and 10.17 respectively.

As for the indoor conditions and fuel consumption that result from using the thermostatic flow valves, the building has a minimum temperature of $18.2°C$, and degree hour of 289 hours and the system consumes 3918 liters of fuel.

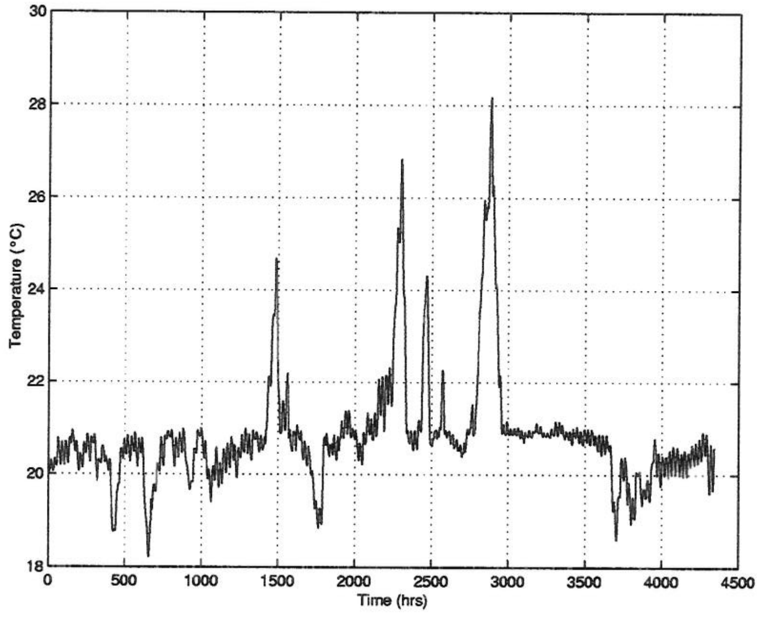


Figure 10.16: Indoor air temperature using thermostatic flow valve control system. Insulated building

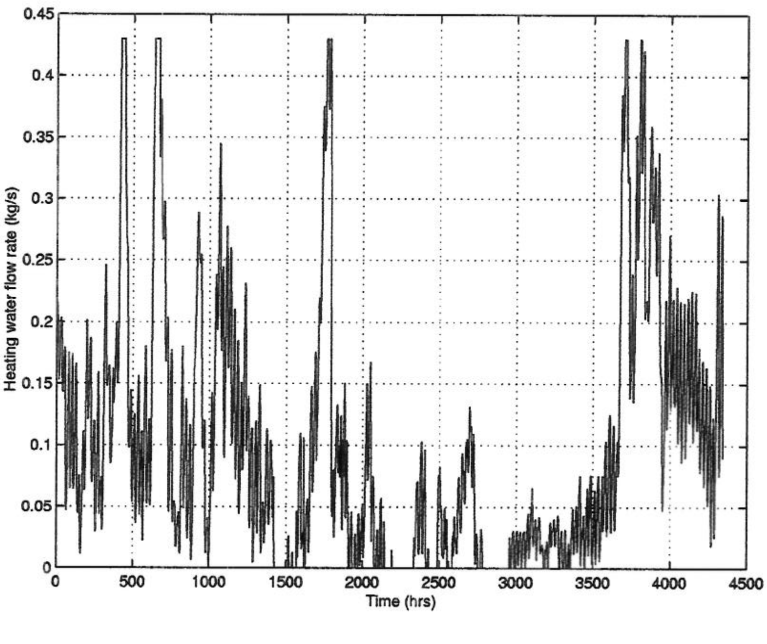


Figure 10.17: Heating water flow rate using thermostatic flow valve control system

10.5 Summary of Results

Table 10.6 gives a summary of the indoor conditions and annual fuel consumption for the different insulation types used in the simulation, and Figure 10.18 shows the indoor temperature histogram for the different insulation used.

Table 10.6: Indoor conditions and fuel consumption for different insulation types

Insulation type	Minimum temp. ($°C$)	Average temp. ($°C$)	Standard deviation ($°C$)	Degree hours	Fuel consumption ($liters$)	Fuel cost (JOD)
None	17.0	20.3	1.6	1522	6024	657
Air gap	17.0	20.3	1.6	1503	5696	621
Polystyrene	17.6	20.3	1.4	1179	3733	407

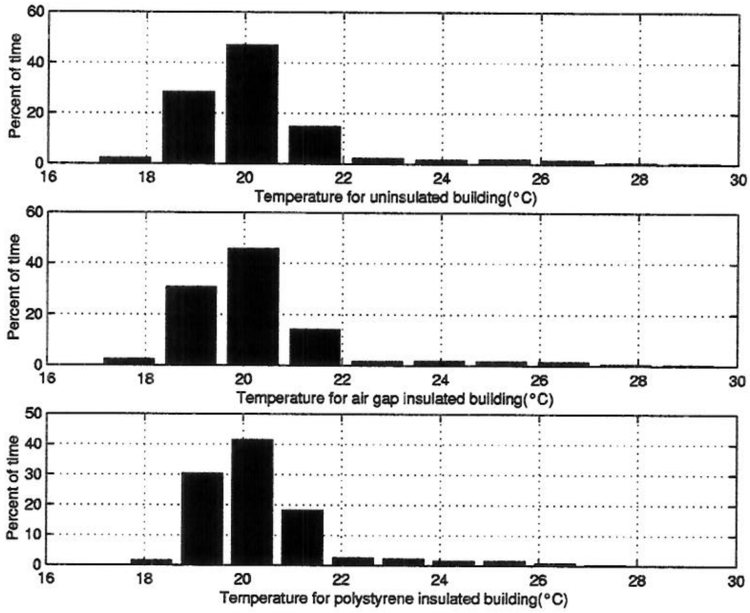


Figure 10.18: Indoor temperature histograms for the different insulation types

Also Table 10.7 provides information regarding indoor conditions for the different heating control systems used, and Figures 10.19 and 10.20 show the indoor temperature histograms for these systems.

It is worth noting that the conditions for the polystyrene insulated building is the same as the thermostat controlled heating system, since all systems were simulated for the insulated building, and different insulation materials were tested using the thermostat control system. This is because a polystyrene insulated building with a thermostat controlled system is recommended by the Jordanian codes.

Table 10.7: Indoor conditions and fuel consumption for the heating systems used

System used	Minimum temp. (°C)	Average temp. (°C)	Standard deviation (°C)	Degree hours	Fuel consumption (*liters*)	Fuel cost (*JOD*)
Thermostat	17.6	20.3	1.4	1179	3733	407
Valves	18.2	20.8	1.2	289	3918	427
Intermittent	10.7	17.0	3.4	4812	2163	236
Set back	14.2	18.7	2.4	2584	2904	316.5

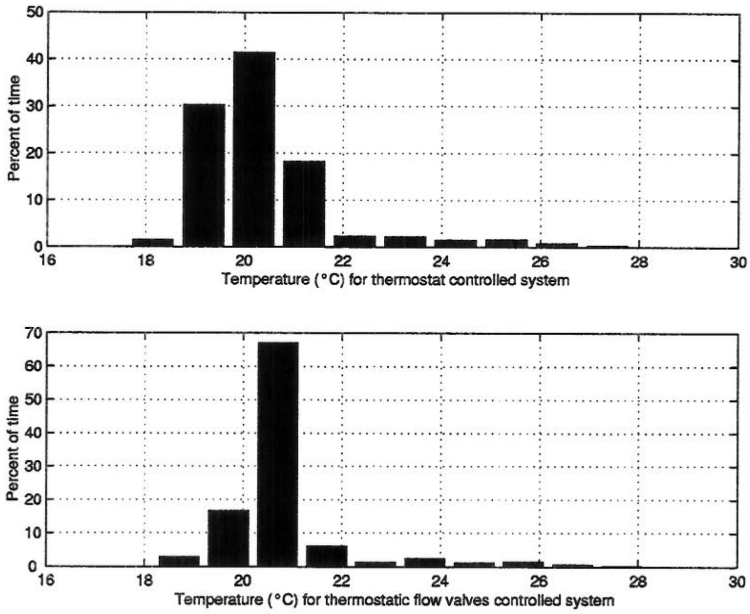
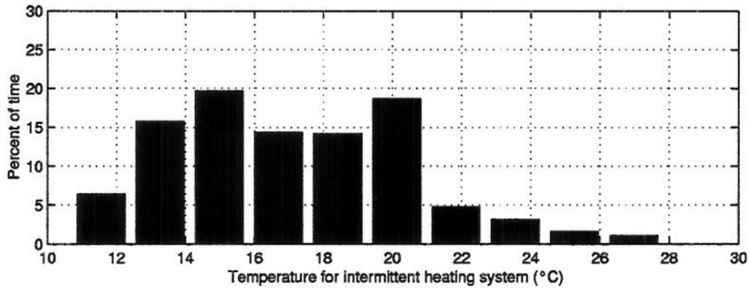


Figure 10.19: Indoor temperature histograms for thermostat and thermostatic flow valve control systems

10.6 Sensitivity Analysis

In order to have a better understanding the heating systems discussed above and in an attempt to improve indoor conditions, a sensitivity analysis will be carried out in this chapter, where the response of the building environment to changes in different parameters of the building and its heating system is evaluated. The sensitivity analysis will extend to evaluate building insulation thickness and heating systems' sizing as well as using different control strategies for the intermittent heating system discussed above.



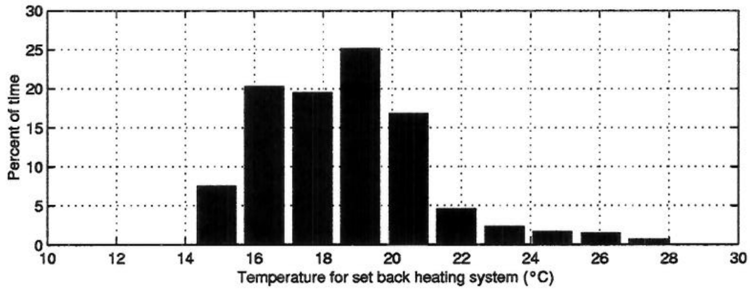


Figure 10.20: Indoor temperature histograms for intermittent and set back control systems

10.6.1 Insulation thickness

In the above discussion the polystyrene insulation that was used in buildings was 3 *cm* thick, as this is the most common thickness used [8]. In this section, the effect of other thicknesses on the indoor conditions will be evaluated.

Table 10.8 gives the minimum indoor temperature, degree hours and fuel consumption for the different insulation thicknesses. Figure 10.21 shows the sensitivity of the indoor conditions and fuel consumption with changes in insulation thickness. It is observed that the degree days are most sensitive to changes in the insulation thickness and minimum indoor temperature has the minimum sensitivity to these changes. Finally Table 10.9 gives information regarding financial feasibility of using higher thicknesses than 3 *cm*.

Table 10.8: Building indoor conditions for different insulation thicknesses

Insulation thickness (cm)	$min.temp.$ ($°C$)	degree hours	Fuel consumption ($Liters$)
1.0	16.6	2235	4518
2.0	17.5	1374.6	4047
3.0	17.6	1179.4	3732.5
4.0	18.7	1065.0	3527.7
5.0	18.5	1111.0	3338.4

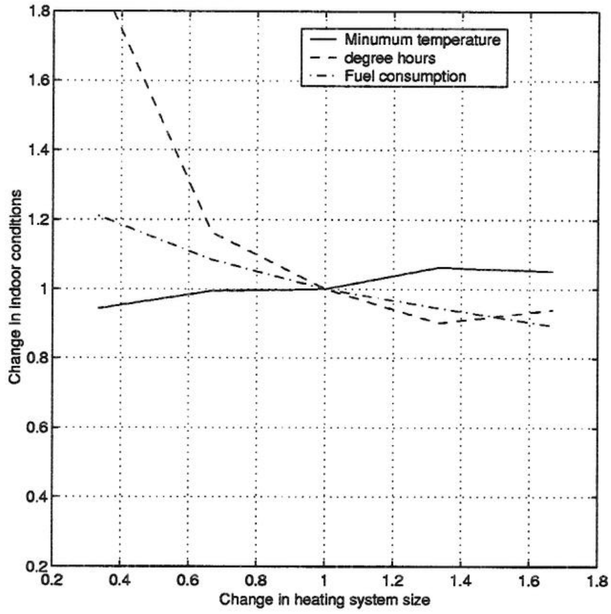


Figure 10.21: Change of indoor temperature with change in insulation thickness

Table 10.9: Insulation thicknesses feasibility

Insulation thickness (cm)	Additional extra cost (JOD)	Annual savings (JOD)	IRR (%)	Payback ($years$)
4.0	180	22.3	9.0	8.1
5.0	360	43.0	8.4	8.4

Although Table 10.8 shows that improved indoor conditions are achieved by increasing insulation thickness, Table 10.9 it is unfeasible to increase the insulation thickness.

10.6.2 Heating system size

As discussed above, one way of improving indoor conditions is to increase insulation thickness. Another way to achieve the same goal is, probably, by increasing the heating system size, which will be discussed here for the uninsulated, air insulated and polystyrene insulated building types.

Uninsulated building

Figure 10.22 shows the response of indoor conditions and fuel consumption to change in system parameters. The Figure clearly shows that degree hours has the highest sensitivity to system change while fuel consumption has the lowest sensitivity, where an increase of 12.0% in heating system parameters is needed to achieve a minimum indoor temperature T_i of $18.0°C$. This increase yields a decrease in degree hours by 20.0% with an increase of only 1.6% in fuel consumption.

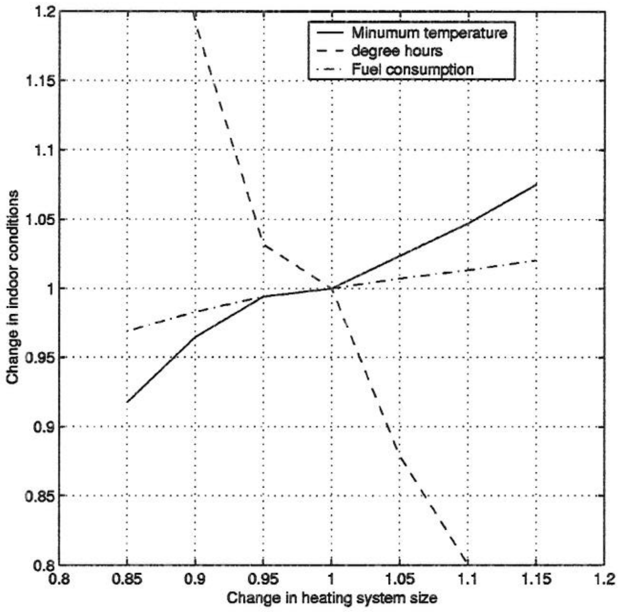


Figure 10.22: Effect of system parameters on indoor conditions and fuel consumption for uninsulated buildings

Air gap insulation

As for the air gap insulation Figure 10.23 shows that an increase of 11.2% in heating system parameters would achieve a minimum indoor temperature T_i of $18.0°C$. This increase would decrease the degree hours of the building by 21.2% and increase the fuel consumption of the building by 1.6%.

Polystyrene insulation

Finally, Figure 10.24 also shows that the degree hours is highest sensitive for changes in system parameters and fuel consumption is lowest sensitive to these changes. In the polystyrene insulated building a system increase 4.0% is needed to obtain a minimum T_i value of $18.0°C$, this size increase is associated with a decrease of 7.4% in degree hours, and an increase by only 0.6% in the annual fuel consumption.

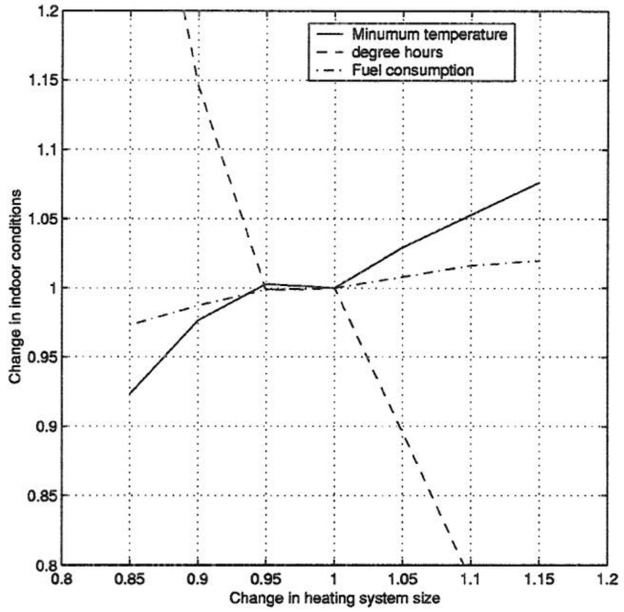


Figure 10.23: Effect of system size on indoor temperature for air insulation

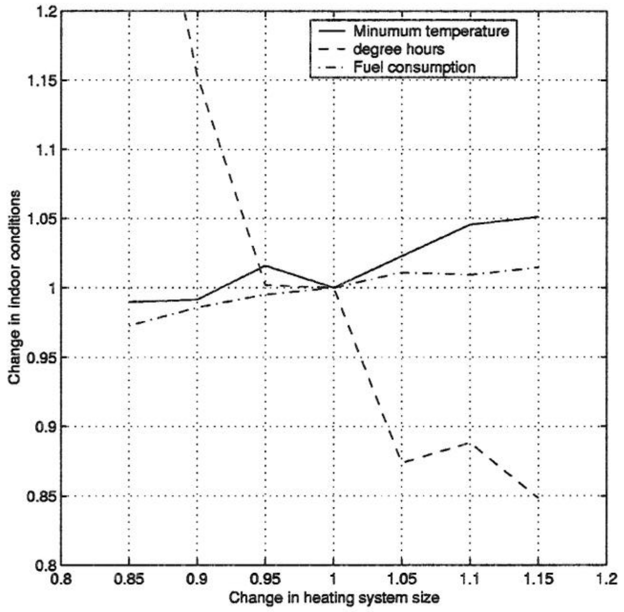


Figure 10.24: Change of indoor temperature with change in system size for polystyrene insulated buildings

10.6.3 Intermittent Heating Systems

Although intermittent heating systems started one hour before the required heating period, the indoor conditions obtained were below the required.

There are two parameters affecting intermittent heating systems' performance. These are, system size and heating start up time. Hence a sensitivity analysis of the effect of each of these parameters in presented in the following discussion.

1. Effect of system start up time on indoor conditions. In this sensitivity analysis only time lead will be considered since starting up the system after the heat is required is not reasonable.

   (a) Intermittent heating system (Jordanian code)
   Figure 10.25 shows that the degree hours are most sensitive to changes in system start up, yet the fuel consumption comes in second place and temperature increase is last. The figure shows that hour lead in system start up an increase of 4.5% in minimum indoor temperature is achieved, associated with an average decrease of 10% in degree hours and an increase of 8% in annual fuel consumption.

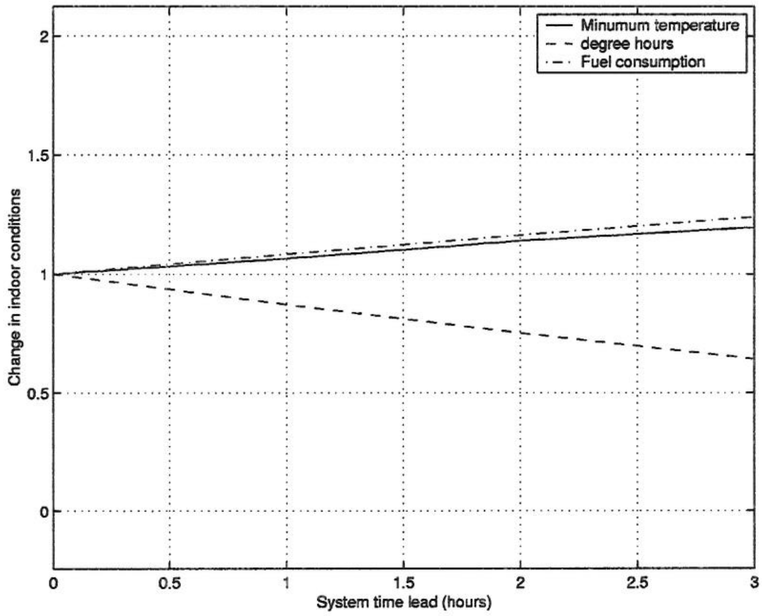


Figure 10.25: Change of indoor conditions using intermittent heating system with change in system start up time

   If extrapolation was used from the figures above to determine the time lead required for the system to achieve the required indoor conditions, 10.3 hours would be needed, resulting in a total of 19 hours of heating and 3951 liters in fuel consumption.

   (b) Set back heating system (ASHRAE).

   The results for the set back heating system are shown in Figure 10.26. This figure also shows that the response in indoor conditions and fuel consumption is highest for degree hours and lowest for minimum temperature. It is clearly seen that for every hour lead in system startup an increase of 3.5% in minimum indoor temperature is acquired with 20% decrease in degree hours and only 5% increase in fuel consumption.

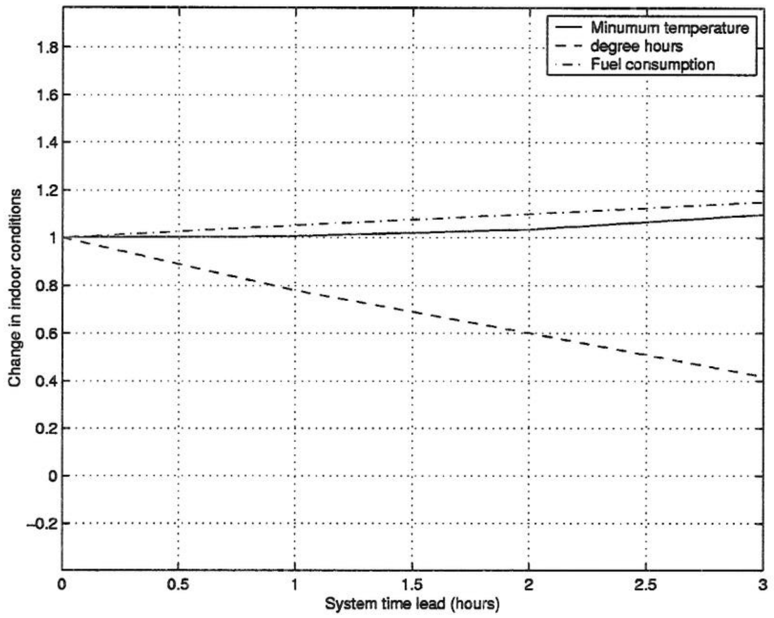


Figure 10.26: Change of indoor conditions using a set back heating system with change in system start up time

The extrapolation of these results yield a time lead of 9 hours with 4065 liters of oil used.

2. Effect of system parameters size on indoor conditions. The second parameter affecting the system performance is its size with the results discussed below.

   (a) Intermittent heating system (Jordanian code)
   As the case for change in start up time, degree hours is highest sensitive to changes in system size followed by fuel consumption and then minimum indoor temperature which is least sensitive to changes in system size. This can be seen by reviewing Figure 10.27. The figure also indicates that for every 10% increase in system size a raise of 4.5% in temperature is obtained which also produced an average decrease of 10% in degree hours and an increase of 7% of fuel consumption.

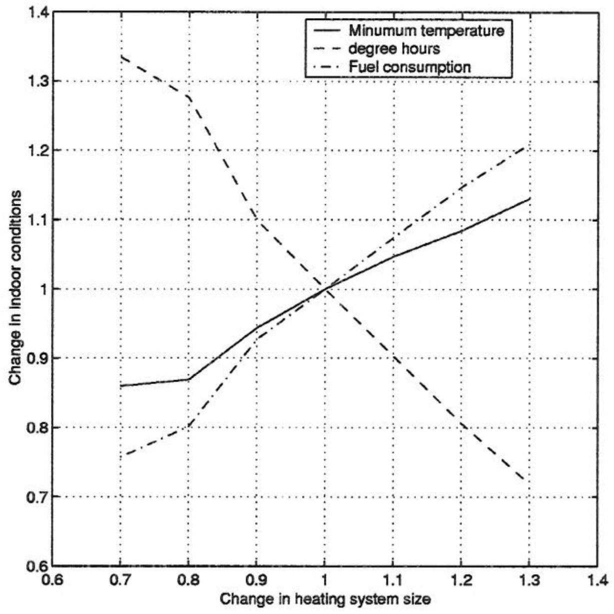


Figure 10.27: Change of indoor conditions using intermittent heating system with change in system parameters

(b) Set back heating system (ASHRAE).

The results for the set back heating system given in Figure 10.28, show again that for every 10% only 0.7% increase in minimum indoor temperature is achieved. While a decrease by almost 14% in degree hours are obtained and only 4% increase in fuel consumption for is obtained.

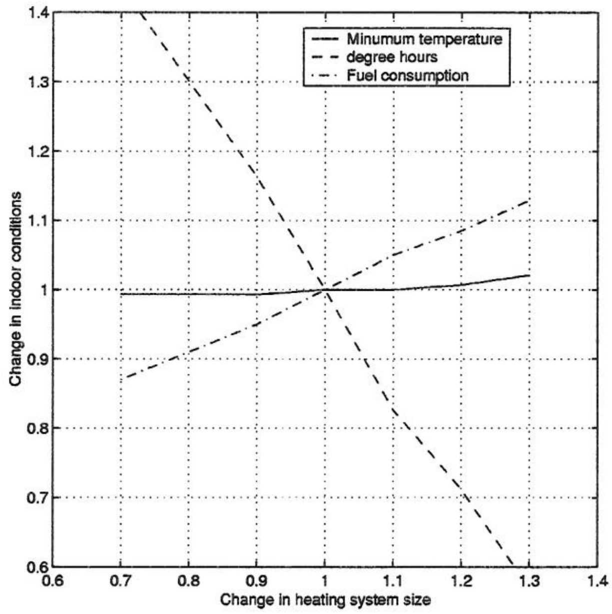


Figure 10.28: Change of indoor conditions using a set back heating system with change in system parameters

Chapter 11

Conclusions and recommendations

By reviewing the results obtained in chapter 10 the following conclusions regarding the heating systems used in Jordanian houses can be drawn.

- A review of the outdoor design temperature value is recommended depending on the fact the the actual outdoor temperature was below this design value ($5.0°C$) for about 453 hours which is a high number. These low outdoor temperature values significantly affect the indoor conditions, where they resemble cold waves leading to significant reduction in indoor temperature as shown in Figures 10.2 through 10.4. These figures show that the indoor temperature dropped to below $18.0°C$ at the time when the outdoor temperature dropped to below $5.0°C$ as shown in Figure10.9.

- The use of polystyrene insulation significantly reduces energy costs and enhances thermal comfort inside the building, as well as being financially feasible. Hence more effort should be undertaken to enforce the building codes in Jordan especially those related to building insulation in order to reduce the energy consumption and improve building performance.

- Although air gap insulation has no actual cost, the indoor conditions that resulted from using this insulation were similar to the uninsulated building and did not improve the building's performance. Hence its use is not recommended.

- Although the use of thermostatic valves is more expensive than using a thermostat,(90 JOD more than a thermostat), higher quality indoor conditions resulted from using those valves. In addition to the fact that these valves control each room in the building, instead of controlling the whole building from one point, as does the thermostat, yielding a more homogeneous temperature distribution inside the building. Hence the extra cost involved in using such equipment is justified by the extra comfort achieved in the building.

- The sensitivity analysis, that was conducted on insulation thickness, showed that small improvement in indoor conditions were achieved where as the decrease in fuel consumption was small which made it clear that higher thicknesses are financially unfeasible.

- Increasing system size significantly improves indoor conditions, while at the same time, the increase in fuel is relatively smaller, as shown in Figures 10.22 to 10.24. This means that the only extra cost involved is the initial cost of the system which when divided by the building life ($>$ 15 years) should be acceptable to owners in order to have favorable indoor conditions.

  Hence increasing system size should be considered as an option when trying to improve indoor conditions.

- Using intermittent heating control systems yielded poor indoor conditions. Also the results obtained from conducting the sensitivity analysis on these systems proved that long heating hours and bigger sized equipment are needed to acquire reasonable indoor conditions. This resulted in high consumption that exceeded fuel consumption in continuous heating systems. Hence intermittent systems should not be used.

- Dynamic simulation can be a powerful tool that helps enhance system's performance and predict problems, thus improving its efficiency before building the real system, resulting in savings of time and money.

Appendix A

Continuous heating system simulation program

```
tic
global a11                  %GET DATA FILE
time=a11(:,1);              %FIRST COLUMN OF DATA FILE (Time)
tout=a11(:,2);              %SECOND COLUMN OF DATA FILE (To)
Sol=a11(:,3);               %THIRD COLUMN OF DATA (Solar Radiation (W/m^2))

nn=input('please enter the iteration period (hrs)')
;%DECIDE THE SIMULATION PERIOD
dt=180;                     %SET TIME STEP (seconds)
dth=dt/3600;                %SET TIME STEP (hours)
dtm=dt/60;                  %SET TIME STEP (minutes)
n=nn/dth;                   %NO. OD ITERATIONS
nnn=nn/dtm;
sf=1.0;                     %safety factor
%BOILER DATA
Mb=90;                  %BOILER MASS OF WATER (kg)
cp=4.186;                   %WATER CP(kJ/kg K)
qmax=15.84*sf;                      %BOILER CAPACITY (kW)
Tb_set=80;                  %BOILER THERMOSTAT SET TEMPERATURE
hyst=3;                     %BOILER THERMOSTAT dT
dies=0;                     %counter for boiler operation
%END OF BOILER DATA
% BUILDING DATA
kl=0.88;                    %HOUSE TOTAL HEAT TRANSFER COEFFICIENT  (kW/C)
C=182629.6;                 %BUILDING THERMAL MASS (kJ/C)
Aroof=200.25;               %BUILDING ROOF AREA (m^2)
%END OF BUILDING DATA

%RADIATOR DATA
Mr = 32*sf;                     %WATER CONTENT IN ALL RADIATORS (kg)
mo=.43*sf;                      %MAX FLOW RATE (kg/s)
%END OF %RADIATOR DATA
```

```
%building control THERMOSTAT+++++++++++++++++++++++++++++++++++
Ti_set=20;              %ROOM THERMOSTAT SET TEMPERATURE

%ZEROING OF MATRICES
Tb=zeros(n,1);          %BOILER OUTLET TEMPERATURE
Ts=zeros(n,1);          %RADIATOR WATER SUPPLY TEMPERATURE
Ti=zeros(n,1);          %INDOOR TEMPERATURE
Tm=zeros(n,1);          %LOG MEAN TEMP DIFFERENCE IN RADIATOR
t=zeros(n,1);           %TIME
To=zeros(n,1);          %OUTDOOR TEMPERATURE
Tr=zeros(n,1);          %RADIATOR OUTLET TEMPERATURE
m=zeros(n,1);           %WATER MASS FLOW RATE
e=zeros(n,1);           %ERROR FOR PI CONTROLLER
qb=zeros(n,1);          %BOILER HEAT INPUT
qr=zeros(n,1);          %RADIATOR HEAT INPUT
s=zeros(n,1);           %SOLAR IRRADIATION
fuelcons=0;
%INITIAL CONDITIONS
qb(1)=0;%qmax;      %INITIAL BOILER STATE (Off)
m(1)=mo;            %INITIAL WATER FLOW (mo)(IT MUST NEVER BE ZERO)

Tb(1)=Tb_set;       %BOILER INITIAL TEMPERATURE
Ti(1)=Ti_set;       %HOUSE INITIAL TEMPERATURE
Ts(1)=Tb(1);
Tr(1)=rettpoly(m(1),Ts(1),Ti(1));   %INITIAL WATER RETURN TEMPERATURE

To=interp1(time,tout,[dth:dth:nn])';
%INTERPOLATE OUTSIDE TEMPERATURE VALUES AT dt VALUES
s=interp1(time,Sol,[dth:dth:nn])';
%INTERPOLATE SOLAR RADIATION VALUES AT dt VALUES

Qs=(0.05283*s-3.7037)*Aroof/1000;       % SOLAR HEAT GAIN (kW)

qr(1)=m(1)*cp*(Ts(1)-Tr(1));
Tm(1)=(Ts(1)-Tr(1))/(log((Ts(1)-Ti(1))/(Tr(1)-Ti(1))));
%CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
qoo=qr(1); Tmoo=Tm(1);

%START SIMULATION LOOP+++++++++++++++++++++++++++++++++++++++++++
for i=2:n

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

   t(i)=t(i-1)+dt;  %TIME INCREMENT
   % BUILDING CONTROLLER +++++++++++++++++++++++++++++++++++++++++++++
```

```
m(i)=m(i-1);
if Ti(i)<Ti_set-1,   m(i)=mo;end;
if Ti(i)>Ti_set+1,   m(i)=0;end;
% END OF BUILDING CONTROLLER +++++++++++++++++++++++++++++++++++++++++++++++++

if m(i-1)>0,          %Ti<Ti_set-2,

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

   %CALCULATING RETURN WATER TEMPERATURE ++++++++++++++++++++++++++++++
   Tr(i)=rettpoly(m(i-1),Ts(i),Ti(i));
   if Tr(i)<=Ti(i); Tr(i)=Ti(i)+0.1; end
   if Ts(i)<=Ti(i); Ts(i)=Ti(i)+0.11; end
   qr(i)=m(i-1)*cp*(Ts(i)-Tr(i));
   Tm(i) = (Ts(i)-Tr(i))/(log((Ts(i)-Ti(i))/(Tr(i)-Ti(i))));
   %CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
   % BOILER CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++
   if Tb(i)>=(Tb_set+hyst)
      qb(i)=0;
   end
   if Tb(i)<=(Tb_set-hyst)
      qb(i)=qmax;
   end

end          %if m>0,

if m(i-1)==0,             %Ti > SET TEMPERATURE

   %CALCULATING RETURN WATER TEMPERATURE ++++++++++++++++++++++++++++++
   Ts(i)=Ti(i-1)+(Ts(i-1)-Ti(i-1))*exp(-qr(i-1)/(Tm(i-1)*Mr*cp)*dt);
   Tr(i)=Ti(i-1)+(Tr(i-1)-Ti(i-1))*exp(-qr(i-1)/(Tm(i-1)*Mr*cp)*dt);
   if Tr(i)<=Ti(i); Tr(i)=Ti(i)+0.1; end
   if Ts(i)<=Ti(i); Ts(i)=Ti(i)+0.11; end
   if Ts(i)==Tr(i); Ts(i)=Tr(i)+0.01; end
   Tm(i)=(Ts(i)-Tr(i))/(log((Ts(i)-Ti(i))/(Tr(i)-Ti(i))));
   %CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
   qr(i)=qoo*(Tm(i)/Tmoo)^(4/3);
   Tb(i)=Tr(i);
   qb(i)=0;
end      %if m==0,


% BOILER CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++
```

```
   %qb(i)=qb(i-1);

   %if Tb(i)>=(Tb_set+hyst),qb(i)=0;end
   %if Tb(i)<=(Tb_set-hyst),qb(i)=qmax;end
   if qb(i)>0
      dies=dies+1;
      %beff(i)=0.85;%74.5+0.025*Tr(i-1)/100;
     %fuelconspoly(i)=(qb(i)*dt)/(42000*0.85);
    end

   % END OF Boiler model ++++++++++++++++++++++++++++++++++++++++++++++++
end     %i=2:n
%END OF THE SIMULATION LOOP
fuel=qmax*dies*dt/42000/.85   %Fuel consumption in kg
X=min(Ti) uu=find(Ti<=20); Y=sum((20-Ti(uu))*dt) toc

figure(1) plot(t/3600,Ti),grid
%title('Indoor Air Temperature(Ti) (Polystyrene insulation)')
xlabel('Time (hrs)') ylabel('Temperature (C)')
%hold
%plot(t/3600,To,'r--'),grid
%axis([540 700 -2 22])

figure(2) plot(t/3600,Tr),grid
%title('Radiator Return Temperature (Polystyrene insulation)')
xlabel('Time (hrs)') ylabel('Radiator Return Temperature (C)')

figure(3) plot(t/3600,Ts),grid
%title('Radiator Supply Temperature (Polystyrene insulation)')
xlabel('Time (hrs)') ylabel('Radiator Supply Temperature (C)')

figure(4) plot(t/3600,To),grid
%title('Outdoor Temperature')
xlabel('Time (hrs)') ylabel('Outdoor Temperature (C)')
```

Appendix B

Intermittent heating system simulation program

```
tic
global a11              %GET DATA FILE
time=a11(:,1);          %FIRST COLUMN OF DATA FILE (Time)
tout=a11(:,2);          %SECOND COLUMN OF DATA FILE (To)
Sol=a11(:,3);           %THIRD COLUMN OF DATA (Solar Radiation (W/m^2))

nn=input('please enter the iteration period (hrs)');
%DECIDE THE SIMULATION PERIOD
dt=180;                 %SET TIME STEP (seconds)
dth=dt/3600;            %SET TIME STEP (hours)
dtm=dt/60;              %SET TIME STEP (minutes)
n=nn/dth;               %NO. OD ITERATIONS
nnn=nn/dtm;
sf=.90;                 %safety factor
%BOILER DATA
Mb=90;              %BOILER MASS OF WATER (kg)
cp=4.186;               %WATER CP(kJ/kg K)
qmax=15.84*.98*sf;                  %BOILER CAPACITY (kW)
Tb_set=80;              %BOILER THERMOSTAT SET TEMPERATURE
hyst=4;                 %BOILER THERMOSTAT dT
dies=0;                 %counter for boiler operation
j=1;
%END OF BOILER DATA
% BUILDING DATA
k1=.88;                 %HOUSE TOTAL HEAT TRANSFER COEFFICIENT  (kW/C)
C=182629.6;             %BUILDING THERMAL MASS (kJ/C)
Aroof=200.25;           %BUILDING ROOF AREA (m^2)
%END OF BUILDING DATA

%RADIATOR DATA
Mr = 32*.98*sf;                 %WATER CONTENT IN ALL RADIATORS (kg)
mo=.43*.98*sf;                  %MAX FLOW RATE (kg/s)
```

```
%END OF %RADIATOR DATA

%building control THERMOSTAT++++++++++++++++++++++++++++++++++++
Ti_set=20;              %ROOM THERMOSTAT SET TEMPERATURE

%ZEROING OF MATRICES
Tset=zeros(n,1);
Tb=zeros(n,1);          %BOILER OUTLET TEMPERATURE
Ts=zeros(n,1);          %RADIATOR WATER SUPPLY TEMPERATURE
Ti=zeros(n,1);          %INDOOR TEMPERATURE
Tm=zeros(n,1);          %LOG MEAN TEMP DIFFERENCE IN RADIATOR
t=zeros(n,1);           %TIME
To=zeros(n,1);          %OUTDOOR TEMPERATURE
Tr=zeros(n,1);          %RADIATOR OUTLET TEMPERATURE
m=zeros(n,1);           %WATER MASS FLOW RATE
e=zeros(n,1);           %ERROR FOR PI CONTROLLER
qb=zeros(n,1);          %BOILER HEAT INPUT
qr=zeros(n,1);          %RADIATOR HEAT INPUT
s=zeros(n,1);           %SOLAR IRRADIATION

%INITIAL CONDITIONS
qb(1)=0;%qmax;  %INITIAL BOILER STATE (Off)
m(1)=mo;        %INITIAL WATER FLOW (mo)(IT MUST NEVER BE ZERO)

Tb(1)=Tb_set;               %BOILER INITIAL TEMPERATURE
Ti(1)=Ti_set;           %HOUSE INITIAL TEMPERATURE
Ts(1)=Tb(1);
Tr(1)=rettpoly(m(1),Ts(1),Ti(1));   %INITIAL WATER RETURN TEMPERATURE
Ton(1)=Ti(1); Tset(1)=15;
To=interp1(time,tout,[dth:dth:nn])';
%INTERPOLATE OUTSIDE TEMPERATURE VALUES AT dt VALUES
s=interp1(time,Sol,[dth:dth:nn])';
%INTERPOLATE SOLAR RADIATION VALUES AT dt VALUES
tt(1)=dt;

Qs=(0.05283*s-3.7037)*Aroof/1000;      % SOLAR HEAT GAIN (kW)

qr(1)=m(1)*cp*(Ts(1)-Tr(1));
Tm(1)=(Ts(1)-Tr(1))/(log((Ts(1)-Ti(1))/(Tr(1)-Ti(1))));
%CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
qoo=qr(1); Tmoo=Tm(1);

%START SIMULATION LOOP++++++++++++++++++++++++++++++++++++++++++++
for i=2:n
   ii=i*dt/3600;
   time_of_day=ii-24*floor(ii/24);
```

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

t(i)=t(i-1)+dt;  %TIME INCREMENT
% BUILDING CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++
m(i)=m(i-1);

 if time_of_day>=15 & time_of_day<=23
     Tset(i)=20;
 end

 if time_of_day>=14 & time_of_day<=23
     m(i)=m(i-1);

   if Ti(i)<Ti_set-1
      m(i)=mo;
   end
   if Ti(i)>Ti_set+1
      m(i)=0;
   end
else
  m(i)=0;
end
% END OF BUILDING CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++

if m(i-1)>0,          %Ti<Ti_set-2,

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

   %CALCULATING RETURN WATER TEMPERATURE ++++++++++++++++++++++++++++++

   Tr(i)=rettpoly(m(i-1),Ts(i),Ti(i));
   if Tr(i)<=Ti(i); Tr(i)=Ti(i)+0.1; end
   if Ts(i)<=Ti(i); Ts(i)=Ti(i)+0.11; end
   if Ts(i)<=Tr(i); Ts(i)=Tr(i)+0.01; end
   qr(i)=m(i)*cp*(Ts(i)-Tr(i));
   Tm(i) = (Ts(i)-Tr(i))/(log((Ts(i)-Ti(i))/(Tr(i)-Ti(i))));
   %CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
   % BOILER CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++
   if Tb(i)>=(Tb_set+hyst)
      qb(i)=0;
   end
   if Tb(i)<=(Tb_set-hyst)
```

```
          qb(i)=qmax;
        end

    end           %if m>0,

    if m(i-1)==0,            %Ti > SET TEMPERATURE

       %CALCULATING RETURN WATER TEMPERATURE +++++++++++++++++++++++++++++
       Ts(i)=Ti(i-1)+(Ts(i-1)-Ti(i-1))*exp(-qr(i-1)/(Tm(i-1)*Mr*cp)*dt);
       Tr(i)=Ti(i-1)+(Tr(i-1)-Ti(i-1))*exp(-qr(i-1)/(Tm(i-1)*Mr*cp)*dt);
       if Tr(i)<=Ti(i); Tr(i)=Ti(i)+0.1; end
       if Ts(i)<=Ti(i); Ts(i)=Ti(i)+0.11; end
       if Ts(i)<=Tr(i); Ts(i)=Tr(i)+0.01; end
       Tm(i)=(Ts(i)-Tr(i))/(log((Ts(i)-Ti(i))/(Tr(i)-Ti(i))));
       %CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
       qr(i)=qoo*(Tm(i)/Tmoo)^(4/3);
       Tb(i)=Tr(i);
       qb(i)=0;
    end      %if m==0,


    % BOILER CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++
    %qb(i)=qb(i-1);

    %if Tb(i)>=(Tb_set+hyst),qb(i)=0;end
    %if Tb(i)<=(Tb_set-hyst),qb(i)=qmax;end
    if qb(i)>0
       dies=dies+1;
    end

    % END OF Boiler model ++++++++++++++++++++++++++++++++++++++++++++++++
end     %i=2:n
%END OF THE SIMULATION LOOP
fuel=qmax*dies*dt/42000/.85   %Fuel consumption in kg
X=min(Ti) uu=find(Ton<=20); Y=sum((20-Ton(uu))*dt)
hh=find(Tset==20);
TTi_20=Ti(hh);                         %Ti at Tset = 20
ht_20=find(TTi_20<=20);

Yh=dt*sum(20-TTi_20(ht_20))     %DD of low set temperature
min_high=min(Ti(hh))

toc

figure(1) plot(t/3600,Ti),grid
%title('Indoor Air Temperature(Ti) (Polystyrene insulation)')
%hold on
```

```
%plot(t/3600,Tset),grid
xlabel('Time (hrs)') ylabel('Temperature (C)') hold off figure(2)
plot(t/3600,m),grid
%title('Radiator Return Temperature (Polystyrene insulation)')
xlabel('Time (hrs)') ylabel('Heating water flow rate (kg/s)')

figure(3) plot(t/3600,Ts),grid
%title('Radiator Supply Temperature (Polystyrene insulation)')
xlabel('Time (hrs)') ylabel('Radiator Supply Temperature (C)')

figure(4) plot(t/3600,To),grid
%title('Outdoor Temperature')
xlabel('Time (hrs)') ylabel('Outdoor Temperature (C)')

figure(5) plot(t/3600,Ti)
%title('Indoor Air Temperature(Ti) (Polystyrene insulation)')
hold on plot(t/3600,Tset,'r'),grid xlabel('Time (hrs)')
ylabel('Temperature (C)') hold off
```

Appendix C

Set back heating system simulation program

```
%Night set back ASHRAE Design
tic
global a11               %GET DATA FILE
time=a11(:,1);           %FIRST COLUMN OF DATA FILE (Time)
tout=a11(:,2);           %SECOND COLUMN OF DATA FILE (To)
Sol=a11(:,3);            %THIRD COLUMN OF DATA (Solar Radiation (W/m^2))

nn=input('please enter the iteration period (hrs)');

%DECIDE THE SIMULATION PERIOD

dt=180;                  %SET TIME STEP (seconds)
dth=dt/3600;             %SET TIME STEP (hours)
dtm=dt/60;               %SET TIME STEP (minutes)
n=nn/dth;                %NO. OD ITERATIONS
nnn=nn/dtm;
sf=.7;                   %safety factor
%BOILER DATA
Mb=90;               %BOILER MASS OF WATER (kg)
cp=4.186;                %WATER CP(kJ/kg K)
qmax=15.84*1.4*sf;                   %BOILER CAPACITY (kW)
Tb_set=80;               %BOILER THERMOSTAT SET TEMPERATURE
hyst=3;                  %BOILER THERMOSTAT dT
dies=0;                  %counter for boiler operation
j=1;
%END OF BOILER DATA
% BUILDING DATA
kl=.88;                  %HOUSE TOTAL HEAT TRANSFER COEFFICIENT  (kW/C)
C=182629.6;              %BUILDING THERMAL MASS (kJ/C)
Aroof=200.25;            %BUILDING ROOF AREA (m^2)
%END OF BUILDING DATA
```

```
%RADIATOR DATA
Mr = 32*1.4*sf;                 %WATER CONTENT IN ALL RADIATORS (kg)
mo=.43*1.4*sf;                  %MAX FLOW RATE (kg/s)
%END OF %RADIATOR DATA

%building control THERMOSTAT++++++++++++++++++++++++++++++++++++
Ti_set=20;              %ROOM THERMOSTAT SET TEMPERATURE

%ZEROING OF MATRICES
Ton=zeros(n,1); tt=zeros(n,1); Tset=zeros(n,1);
Tb=zeros(n,1);          %BOILER OUTLET TEMPERATURE
Ts=zeros(n,1);          %RADIATOR WATER SUPPLY TEMPERATURE
Ti=zeros(n,1);          %INDOOR TEMPERATURE
Tm=zeros(n,1);          %LOG MEAN TEMP DIFFERENCE IN RADIATOR
t=zeros(n,1);           %TIME
To=zeros(n,1);          %OUTDOOR TEMPERATURE
Tr=zeros(n,1);          %RADIATOR OUTLET TEMPERATURE
m=zeros(n,1);           %WATER MASS FLOW RATE
e=zeros(n,1);           %ERROR FOR PI CONTROLLER
qb=zeros(n,1);          %BOILER HEAT INPUT
qr=zeros(n,1);          %RADIATOR HEAT INPUT
s=zeros(n,1);           %SOLAR IRRADIATION

%INITIAL CONDITIONS
qb(1)=0;%qmax;     %INITIAL BOILER STATE (Off)
m(1)=mo;           %INITIAL WATER FLOW (mo)(IT MUST NEVER BE ZERO)

Tb(1)=Tb_set;      %BOILER INITIAL TEMPERATURE
Ti(1)=Ti_set;      %HOUSE INITIAL TEMPERATURE
Ts(1)=Tb(1);
Tr(1)=rettpoly(m(1),Ts(1),Ti(1));
%INITIAL WATER RETURN TEMPERATURE
Ton(1)=Ti(1); Tset(1)=15;
To=interp1(time,tout,[dth:dth:nn])';
%INTERPOLATE OUTSIDE TEMPERATURE VALUES AT dt VALUES
s=interp1(time,Sol,[dth:dth:nn])';
%INTERPOLATE SOLAR RADIATION VALUES AT dt VALUES
tt(1)=dt;

%mc=interp1(tim,mc1,[dtm:dtm:nnn]);
%INTERPOLATE COLD WATER FLOW RATE AT dt VALUES
%mc=mc';
%mod(([dth:dth:nn]*60),24*60);

Qs=(0.05283*s-3.7037)*Aroof/1000;
% SOLAR HEAT GAIN (kW)
```

```
qr(1)=m(1)*cp*(Ts(1)-Tr(1));
Tm(1)=(Ts(1)-Tr(1))/(log((Ts(1)-Ti(1))/(Tr(1)-Ti(1))));
%CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
qoo=qr(1); Tmoo=Tm(1);

%START SIMULATION LOOP+++++++++++++++++++++++++++++++++++++++++++++
for i=2:n
   ii=i*dt/3600;
   time_of_day=ii-24*floor(ii/24);

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

   t(i)=t(i-1)+dt;  %TIME INCREMENT
   % BUILDING CONTROLLER +++++++++++++++++++++++++++++++++++++++++++++++
   m(i)=m(i-1);
   Tset(i)=17;
   if Ti(i)<Ti_set-6,
     m(i)=mo;
   end
   if Ti(i)>Ti_set-4,
      m(i)=0;
    end
    if time_of_day>=15 & time_of_day<=23
       Tset(i)=20;
   end
    if time_of_day>=14 & time_of_day<=23
       m(i)=m(i-1);
       %j=j+1;
       Ton(i)=Ti(i);
       %tt(j)=tt(j-1)+dt;

      if Ti(i)<Ti_set-1
         m(i)=mo;
      end
      if Ti(i)>Ti_set+1
         m(i)=0;
      end
   end
   if Ton(i)==0; Ton(i)=nan; end
   % END OF BUILDING CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++

   if m(i-1)>0,          %Ti<Ti_set-2,

      % Boiler model +++++++++++++++++++++++++++++++++++++++++++++++
      Tb(i)=(qb(i-1)+m(i-1)*cp*Tr(i-1)-exp(-m(i-1)*dt/Mb)*(qb(i-1)+m(i-1)...
         *cp*Tr(i-1)-Tb(i-1)*m(i-1)*cp))/(m(i-1)*cp);
```

```
   Ts(i)=Tb(i);  % only valid for m>0
   % END OF Boiler model ++++++++++++++++++++++++++++++++++++++++++++++++

   %CALCULATING RETURN WATER TEMPERATURE +++++++++++++++++++++++++++++++

   Tr(i)=rettpoly(m(i-1),Ts(i),Ti(i));
   if Tr(i)<=Ti(i); Tr(i)=Ti(i)+0.1; end
   if Ts(i)<=Ti(i); Ts(i)=Ti(i)+0.11; end
   if Ts(i)<=Tr(i); Ts(i)=Tr(i)+0.01; end
   qr(i)=m(i)*cp*(Ts(i)-Tr(i));
   Tm(i) = (Ts(i)-Tr(i))/(log((Ts(i)-Ti(i))/(Tr(i)-Ti(i))));
   %CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
   % BOILER CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++
   if Tb(i)>=(Tb_set+hyst)
      qb(i)=0;
   end
   if Tb(i)<=(Tb_set-hyst)
      qb(i)=qmax;
   end

end          %if m>0,

if m(i-1)==0,             %Ti > SET TEMPERATURE

   %CALCULATING RETURN WATER TEMPERATURE +++++++++++++++++++++++++++++++
   Ts(i)=Ti(i-1)+(Ts(i-1)-Ti(i-1))*exp(-qr(i-1)/(Tm(i-1)*Mr*cp)*dt);
   Tr(i)=Ti(i-1)+(Tr(i-1)-Ti(i-1))*exp(-qr(i-1)/(Tm(i-1)*Mr*cp)*dt);
   if Tr(i)<=Ti(i); Tr(i)=Ti(i)+0.1; end
   if Ts(i)<=Ti(i); Ts(i)=Ti(i)+0.11; end
   if Ts(i)<=Tr(i); Ts(i)=Tr(i)+0.01; end
   Tm(i)=(Ts(i)-Tr(i))/(log((Ts(i)-Ti(i))/(Tr(i)-Ti(i))));
   %CALCULATE NEW LOG MEAN TEMPERATURE DIFFERENCE
   qr(i)=qoo*(Tm(i)/Tmoo)^(4/3);
   Tb(i)=Tr(i);
   qb(i)=0;
end      %if m==0,


% BOILER CONTROLLER ++++++++++++++++++++++++++++++++++++++++++++++++
%qb(i)=qb(i-1);

%if Tb(i)>=(Tb_set+hyst),qb(i)=0;end
%if Tb(i)<=(Tb_set-hyst),qb(i)=qmax;end
if qb(i)>0
   dies=dies+1;
end
```

```
   % END OF Boiler model +++++++++++++++++++++++++++++++++++++++++++++++++
end     %i=2:n
%END OF THE SIMULATION LOOP
fuel=qmax*dies*dt/42000/.85   %Fuel consumption in kg
X=min(Ti) uu=find(Ton<=20); Y=sum((20-Ton(uu))*dt)
ll=find(Tset==15);  %find the time where the set temperature was 15
TTi_15=Ti(ll);                          %Ti at Tset = 15
lt_15=find(TTi_15<=17);
Yl=dt*sum(17-TTi_15(lt_15))                 %DD of low set temperature
min_low=min(Ti(ll)) hh=find(Tset==20);
TTi_20=Ti(hh);                          %Ti at Tset = 20
lt_20=find(TTi_20<=20);

Yh=dt*sum(20-TTi_20(lt_20))                 %DD of low set temperature
min_high=min(Ti(hh))

toc

figure(1) plot(t/3600,Ti),grid
%title('Indoor Air Temperature(Ti) (Polystyrene insulation)')
%hold on
%plot(t/3600,Tset),grid
xlabel('Time (hrs)') ylabel('Temperature (C)') hold off figure(2)
plot(t/3600,m),grid
%title('Radiator Return Temperature (Polystyrene insulation)')
xlabel('Time (hrs)') ylabel('Heating water flow rate (kg/s)')

figure(3) plot(t/3600,Ts),grid
%title('Radiator Supply Temperature (Polystyrene insulation)')
xlabel('Time (hrs)') ylabel('Radiator Supply Temperature (C)')

figure(4) plot(t/3600,To),grid
%title('Outdoor Temperature')
xlabel('Time (hrs)') ylabel('Outdoor Temperature (C)')

figure(5) plot(t/3600,Ti)
%title('Indoor Air Temperature(Ti) (Polystyrene insulation)')
hold on plot(t/3600,Tset,'r'),grid xlabel('Time (hrs)')
ylabel('Temperature (C)') hold off
```

Bibliography

[1] Fleischer G. A. *Introduction to Engineering Economy.* PWS Publishing Company, 20 Park Plaza, Boston, MA 02116-4324, 1994.

[2] Hammad M.A. Alsaad M.A. *Heating and Air Conditioning.* University of Jordan Amman-Jordan, 1994.

[3] Anon. Din 4703 teil 3, wrmeleistung von raumheizkrpern beuth verlag, 1977.

[4] American Society for Heating Refrigeration and Air Conditioning Engineers (ASHRAE). *ASHRAE HANDBOOK Fundamentals.* American Society for Heating Refrigeration and Air Conditioning Engineers Inc., 1993.

[5] Ken Sejling Henning Tangen Sogaard Henrik Madsen, Olafur Petur Palsson. *Models and Methods for Optimization of District Heating Systems Part I: Models and Identification Methods.* IMSOR,DtH, December 1990.

[6] Holman J.P. *Heat Transfer.* McGraw-Hill, Inc., USA, 8^{th} edition, 1997.

[7] Akawi Kh. *Code of Thermal Insulation.* Ministry of Public Works and Housing, 1990.

[8] Ministry of Energy and Mineral Resources-Jordan. Household energy survey. Technical report, Amman-Jordan, May 1997.

[9] Alaish A. S. *A Guide to Buildings Thermal Insulation Materials, in Arabic.* Royal Scientific Society, Amman Jordan, August 1990.

[10] Al shishany M. *Code of Central Heating.* Ministry of Public Works and Housing, 1990.