Cranfield Institute of Technology

School of Mechanical Engineering

PhD Thesis

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Theoretical and Experimental Evaluations of the Convective and Conductive Heat Transfers in a Domestic Hot-Water Store

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Summary

The design of a water based thermal store for use in a domestic central heating system has been investigated theoretically, experimentally and numerically. The transient operation of the store during both the space heating and domestic hot-water modes of operation have been investigated separately.

Heat transfer correlations in terms of Nusselt and Rayleigh numbers have been developed in order to predict the natural convection heat transfer coefficient for the outside surface of the horizontal axis finned tube heat exchanger coil located within the store. These heat-transfer correlations can predict the value of the heat transfer coefficient with an accuracy of better than 5 % and are in good agreement with existing heat transfer correlations developed for the same geometry of finned tubes and modes of heat transfer. The effect of the water flow rate in the heat exchanger coil on the internal heat transfer coefficient is also investigated. This flow rate should be above 4 litre/minute to achieve a high rate of heat transfer from the wall of the heat exchanger to the water in the pipe.

A detailed investigation of the use of horizontal and vertical baffles to increase the effectiveness of heat delivery in the domestic hot water mode has been carried out. Some improvements can be achieved by the use of a horizontal flat plate located in the middle of the store. This plate, when correctly sized enhances stratification and hence improves the effectiveness of heat recovery. Vertical plate arrangements and a rectangular duct situated around the upper heat exchanger coil were found to be ineffective. However, due to an increased velocity of the water around the heat exchanger, the external heat transfer coefficient of the heat exchanger was increased by 12%.

The comparison of experimental observations with computer simulations of the development of the thermocline in the store during the space heating mode of operation showed the presence of a jet in the bottom region of the store at the return inlet. The jet induces a significant amount of mixing in the store which reduces the effectiveness of heat recovery. Correlations in terms of Richardson number and effectiveness of heat delivery have been developed to characterize the effect of this jet. An inlet arrangement designed to achieve a Richardson number exceeding 3 significantly reduces the mixing created by the jet and can increase the amount of heat delivered in the space heating mode by approximately 5%.

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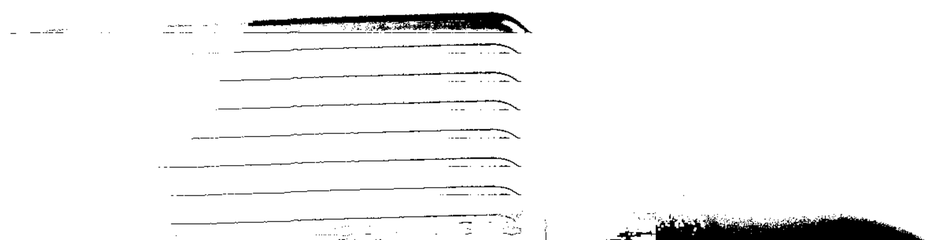

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Notation

A : surface area, m^2
a : constant in the Sleicher-Rouse equation, dimensionless
b : constant in the Sleicher-Rouse equation, dimensionless
b : constant in the natural convection equation, dimensionless
C : constant in the natural convection equation, dimensionless
Cf: correction factor for the thermocouple readings, K
Cm: mixing coefficient in the modified energy equation, dimensionless
Cmi: inlet mixing coefficient, dimensionless
Cmt: mixing coefficient at the top of the store, dimensionless
Cp: specific heat, $Jkg^{-1}K^{-1}$
D : diametre, m
d : ratio of UA values, dimensionless
E : energy content of the insulant, Jkg^{-1}
E : experimental error, dimensionless
e : effective diffusivity factor, dimensionless
e : coefficient of heat loss to the environment, $Wm^{-2}K^{-1}$
ECF: Eddy Conductivity Factor, dimensionless
Eff: effectiveness of heat recovery, dimensionless
Eff: fin efficiency, dimensionless
Err: average error between thermocouples and PRTs, K
F : temperature, ^{o}C
g : acceleration of gravity, ms^{-2}
H : height of the store, m
h : heat-transfer coefficient, $Wm^{-2}K^{-1}$
h : coefficient of heat conduction along the store $Wm^{-1}K^{-1}$
k : thermal conductivity, $Wm^{-1}K^{-1}$
L : length of the duct, m
l : characteristic length for the Rayleigh number, m
l : equivalent fin length, m
M : mass of water in one zone in the tank, kg
m : mass or mass flow rate, kgs^{-1}
m : heat transfer parameter, m^{-1}
n : number of nodes in the finite difference formulation, dimensionless
n : number of thermocouples in the store, dimensionless
NTU: number of transfer units of the heat exchanger, dimensionless
Nu: Nusselt number, dimensionless
P : pressure, Pa
p : perimeter of the fins, m
p : perimeter of the rod, m
Pr: Prandtl number, dimensionless
Q : anount of energy, J

q : rate of heat transfer, W
r : radius, m
r : ratio of thermal capacities, dimensionless
Re: Reynolds number, dimensionless
Ri: Richardson number, dimensionless
S : Cross section of the rod, m^2
s : fin spacing, m
T : temperature, K
TT: temperature at the previous time step, °C
t : time, s
U : overall heat transfer coeficient of the heat exchanger, $Wm^{-2}K^{-1}$
U : velocity in the x direction, ms^{-1}
V : velocity in the y direction, ms^{-1}
W : velocity in the z direction, ms^{-1}
X : optimum thickness of insulant, m
x : coordinate, m
y : coordinate, m
z : coordinate, m

Greek Symbols

ß : coefficient of thermal expansion, m^3K^{-1}
o : density, kgm^{-3}
μ : viscosity, $kgm^{-11}s^{-1}$
α : thermal diffusivity, m^2s^{-1}
δ : boundary layer thickness, m

Subscripts

b : bulk
c : of the coil
c : of the chilled water supply
dhw: at the taps
do : in the lower part of the store
e : external, outside of the pipe
e : of the environment
ex : in the heat exchanger coil
f : delivered to the taps
f : at the end of the experiment
fe : into the heat exchanger coil
i : internal, in the pipe
i : refers to intermediate value
i : refers to node i in finite difference formulation
in : at the inlet of the heat exchanger
max: maximum value

min: minimum value
opt: optimum value
out: at the outlet of the heat exchanger
PRT: refers to the Platinum Resistance Thermometer
up : in the upper part of the store
s : of the store
se : set point of the mixing valve
t : refers to the tank
tc : refers to thermocouple
w : refers to wall or water
1 : refers to the lower zone in the store
2 : refers to the upper zone in the store

Superscript

t : at previous time step
* : dimensionless variable

Abreviations

DHL: Design Heat Loss
DHW: Domestic Hot Water
ITS: Integrated Thermal Store
PRT: Platinum Resistance Thermometer

Chapter 1

The Development of Integrated Thermal Stores

1.1 Heating and Hot Water

Ever since the early ages of human history, mankind has looked for increasing comfort levels. However, it is only with recent advances in technology in the 19th and mainly in the late 20th century that the possibility of providing a reasonable comfort level to a significant proportion of the population of developed countries has become possible.

Additionally, during the 19th and 20th centuries two other phenomena have taken place which are of major importance. The first of these is a large increase in the standard of living. This has led to the growing possibility to fulfill financially the need for comfort. The second is urbanization which has taken place on a relatively large scale.

As a results, there has been significant technological changes in the design of new homes. Most of these changes involve the use of some modern technology and an increase in energy consumption of the dwelling. The most significant of these changes in terms of energy consumption have materialised in the form of space heating and the supply of domestic hot water.

As a consequence of all these changes, there has been a growing need to find sources of energy or of heat for the massive amount of population living in relatively large cities. The prime requirements on these sources of energy are ease of transportation, cleanliness, ease of use and obviously low cost.

At the moment, no sources of energy can fulfill all these requirements. However, natural gas is often regarded as the most suitable source of heat which is partly reflected in the relative success of modern gas central heating systems.

1.2 The Use of Thermal Storage in Dwellings

The heating system of a modern house has to supply enough heat to fulfill two separate demands: space heating and domestic hot water. None of these demand is constant with time, therefore, some fluctuations in the instantaneous heat demand around the average value are inevitable.

For space heating, these fluctuations can be divided in short term fluctuations and long term fluctuations. Long term fluctuations are mainly caused by variations in the outdoor air temperature from one season to the other. Whereas short term fluctuations are created by variations in the ambient conditions during the day such as the incidence of solar radiation or variations in the air temperature between day and night.

The short term fluctuations of the heat demand to fulfill the space heating demand for a typical cold winter day and for an insulated detached house are represented in Fig.1.2(a) [1]. Most of the heat demand is during the daytime period with a peak occurring in the early hours of the morning.

For domestic hot water, the fluctuations in the heat demand are mainly daily fluctuations and are caused by factors such as the temperature of the hot water and the mass flow rate at which it is delivered. The fluctuations of the heat demand to fulfill the hot water requirement for a typical day are represented in Fig.1.2(b) [2]. Again most of the heat demand occurs in the daytime period with peak occurring during the evening.

There are two differences between the space heating and the domestic hot water demand. The first difference comes from the time at which the maximum demand occurs. For space heating , the peak demand is in the morning, whereas for domestic hot water it is mainly in the evening.

The second difference comes from the amplitudes of the fluctuations in the space heating and hot water demand. For space heating, the heat demand tends to vary in the range 1 to 3 during the daytime period. However, for the domestic hot water demand, the fluctuations tend to be very sharp, the heat demand being extremely high for very short periods of time and nil for the rest of day.

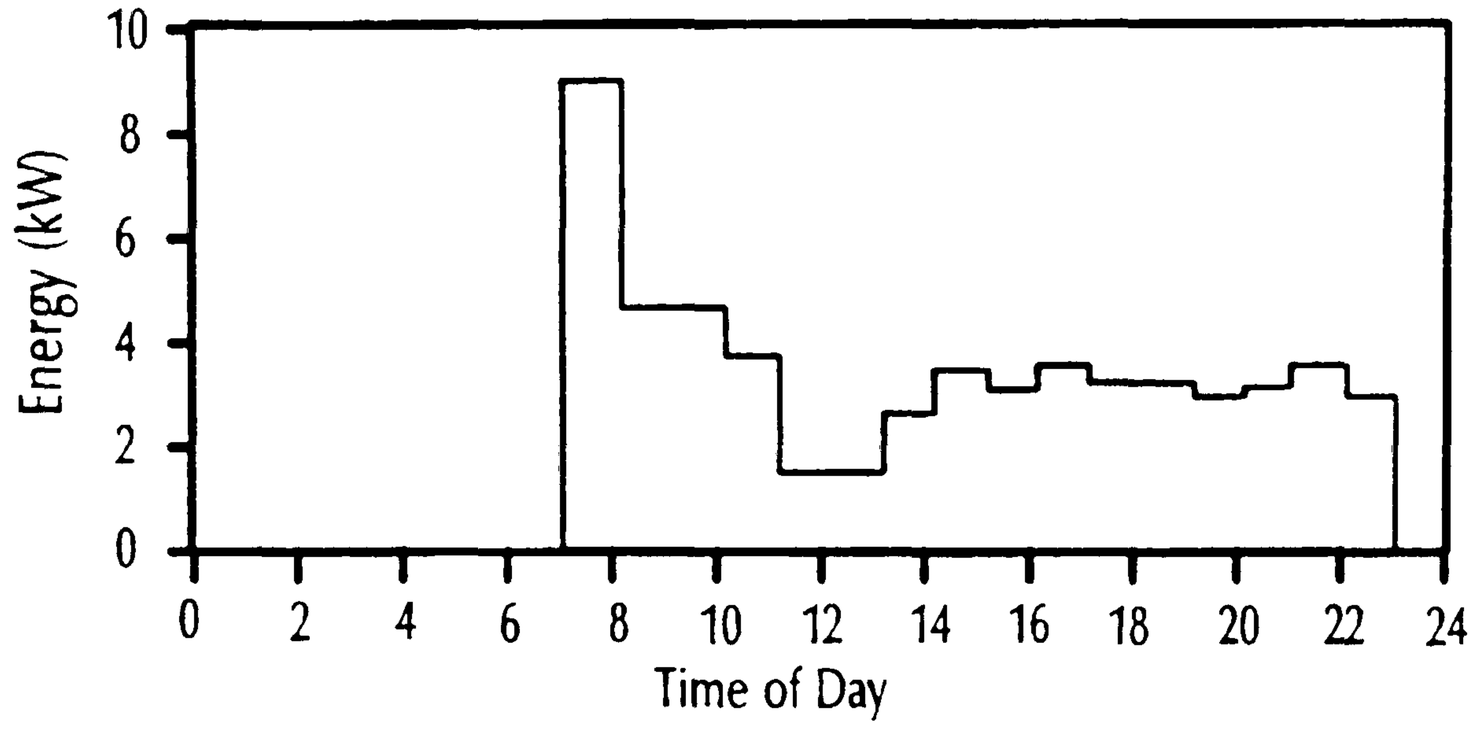


Fig. 1.2(a). Energy Consumption for Space Heating

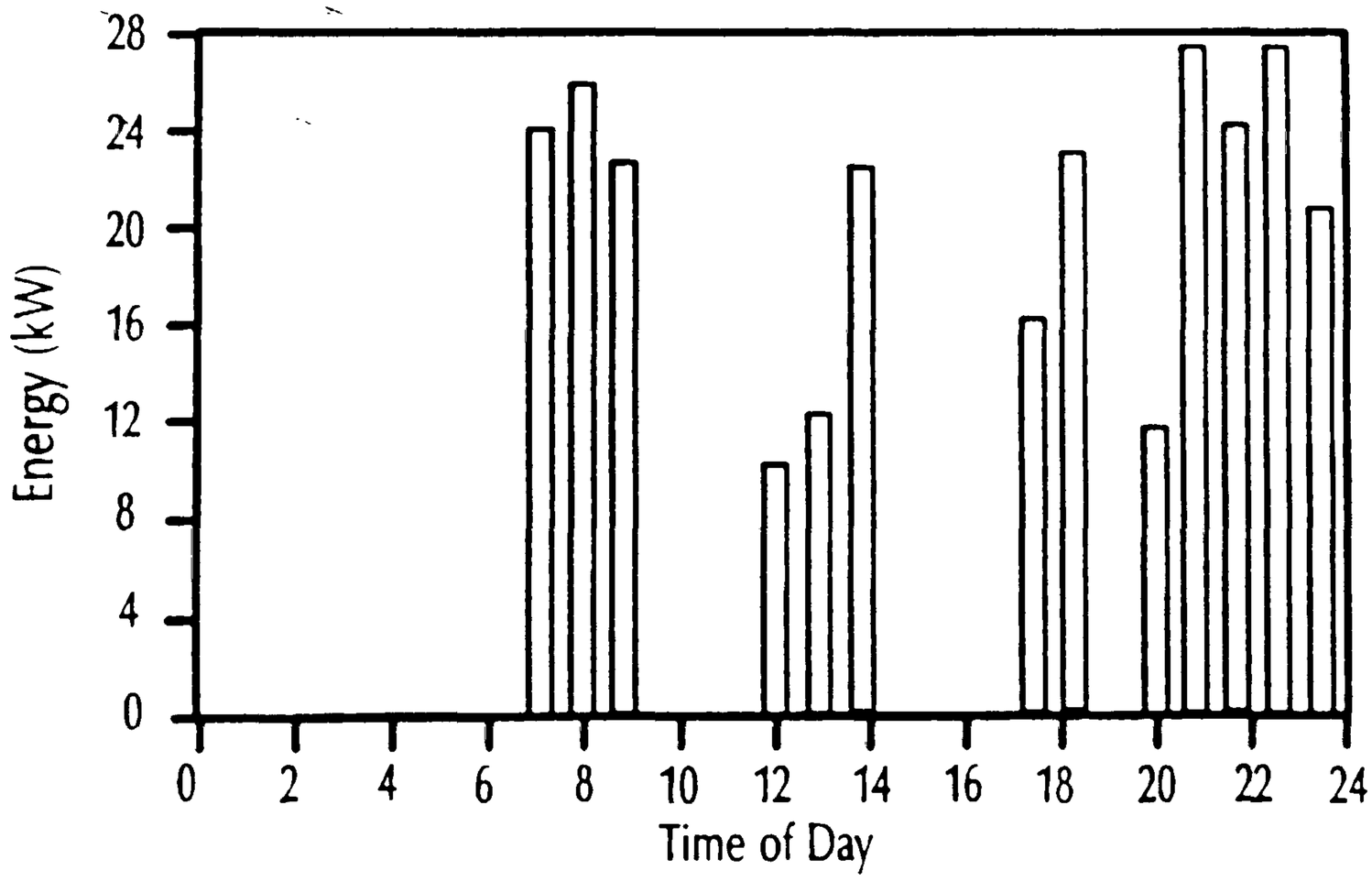


Fig. 1.2(b). Energy Consumption for Domestic Hot Water Production

A careful observation of the domestic hot water demand suggests that a heat source sized to match this demand on an instantaneous basis would need to be around 20 kW. Furthermore, this heat source would have to operate on an on-off cycle very rapidly to match demands of relatively short duration. This type of heat source can be found at the moment in the form of instant gas heater for example. These are characterised by a quick on-off operation capability with a heat output of usually of a few kilowatts.

However, the heat demand profile for the hot water can be matched relatively easily by the use of a thermal storage system. The use of the thermal storage system is justified in several ways. For example the peak demand for the heating profile represented in Fig.1.2(b) is around 20 kW but in fact, when the total of the domestic hot water heat demand is averaged over a 24 hour period, a heat source of around 220 W with a properly sized storage system would in theory be sufficient to provide the same daily amount of heat. This would correspond to a reduction of two orders of magnitude in the output of the heat source. In addition, as the operation of the heat source with a heat storage system would be continuous instead of intermittent and of a constant output, simpler types of heat generators running at higher efficiencies can be used.

1.3 Simultaneous Domestic Hot Water and Space Heating with a Traditional Central Heating Systems

Apart from the storage system issue, the domestic hot water and space heating demands can be provided by two separate heat sources. Each of these is used to satisfy a separate demand. An example of such an application is when using an instant hot water system with an independent space heating system. However, by satisfying these two demands with the same heat source, technologically simpler and hence cheaper and more reliable systems can be used.

Several systems are currently used for supplying at the same time space heating and domestic hot water. Almost all of them use thermal storage to reduce the fluctuations in the domestic hot water demand side. A simplified schematic diagram for such a system is presented in Fig.1.3 [3]. A boiler is used as a heat source and hot water from the boiler is circulated to radiators (for space heating) and to a hot water cylinder which acts as a thermal storage system (for domestic hot water).

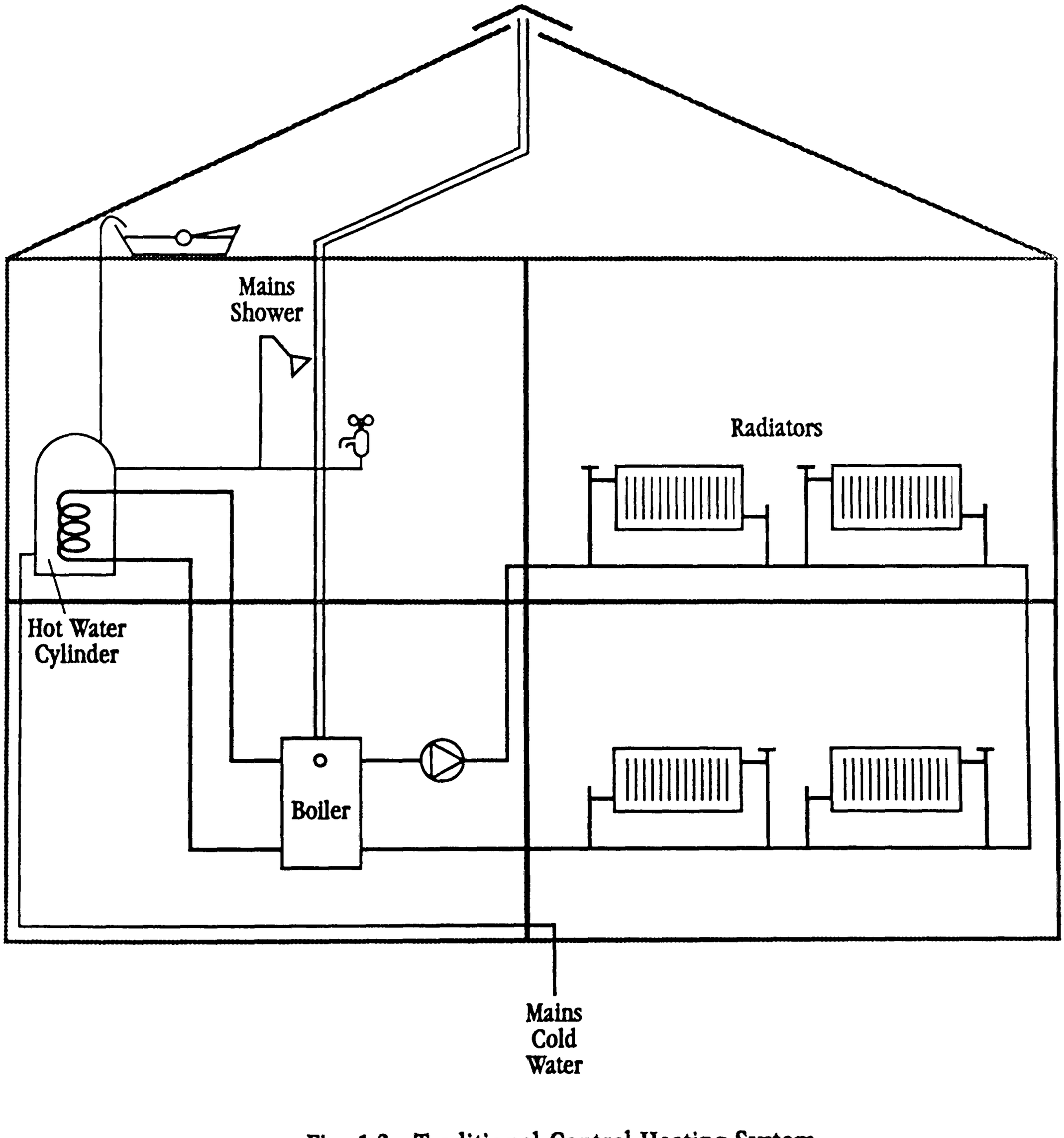


Fig. 1.3. Traditional Central Heating System

1.4 Integrated Thermal Stores

1.4.1 Principle

A further development consists in taking advantage of the thermal storage principle to reduce the fluctuations in both the domestic hot water and space heating demand.

The use of a storage system for space heating can again be justified as the peak space heating demand as represented in Fig.1.2(a) is around 9 kW when in fact when averaged over a 24 hour period, a heat output of only 2.7 kW would be sufficient to provide the same amount of space heating.

Although an independent heat store can be used for satisfying the space heating demand only, again some technologically simpler systems are obtained by using the same heat store to fulfill both the domestic hot water and space heating demands.

The system which uses this principle is known as the Integrated Thermal Store (ITS) and is represented in Fig.1.4.1(a) and Fig.1.4.1(b) [4]. With this system a water based thermal store is interposed between the boiler and the heating circuit.

A finned tube heat exchanger coil, located within the store is used to provide domestic hot water. The use of this heat exchanger presents several advantages. The first advantage is that it eliminates the need for a secondary hot water cylinder and consequently simplifies the installation of the system. which lowers the investment cost. The second advantage is that as the pressure drop along the heat exchanger is relatively small, the mains water supply pressure is available at the point of use. This presents advantages for certain domestic applications such as the use of showers.

The space heating system is connected directly to the thermal store thus taking advantage of the relatively large thermal mass available within the store. The use of the store for space heating purposes is possible in practice as the temperature at which the space heating operates is usually higher than the temperature at which domestic hot water is required.

Also, by the use of this Integrated Thermal Store, the heating load can be almost completely separated from the heat source. The heat source does not have to match the peak heat demand, which could easily be fulfilled by the use of the thermal storage system. Thus further reduction in the output from the heat source can be achieved without detrimental effect on the service.

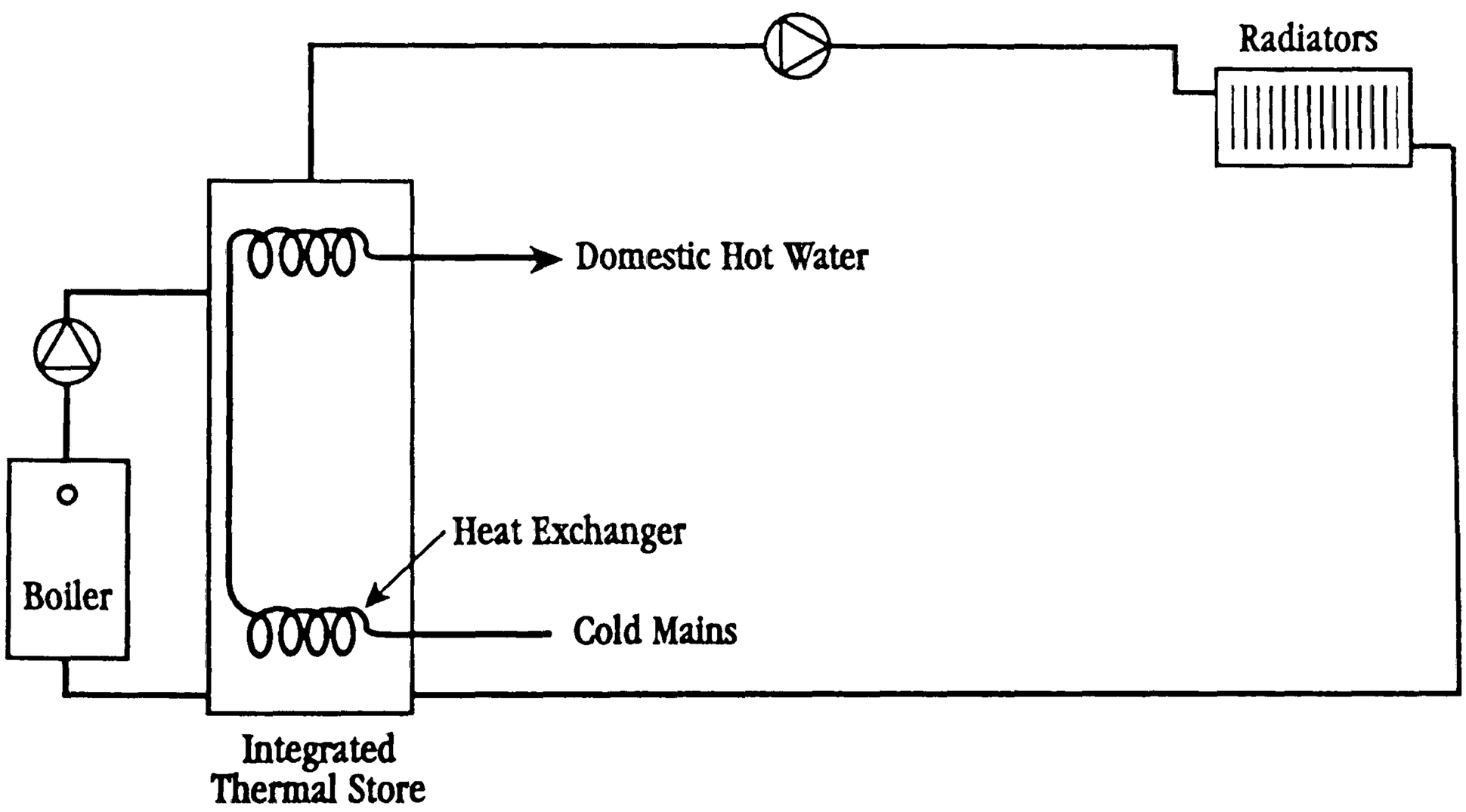


Fig.1.4.1(a): Simplified Schematic of Integrated Thermal Storage Heating System

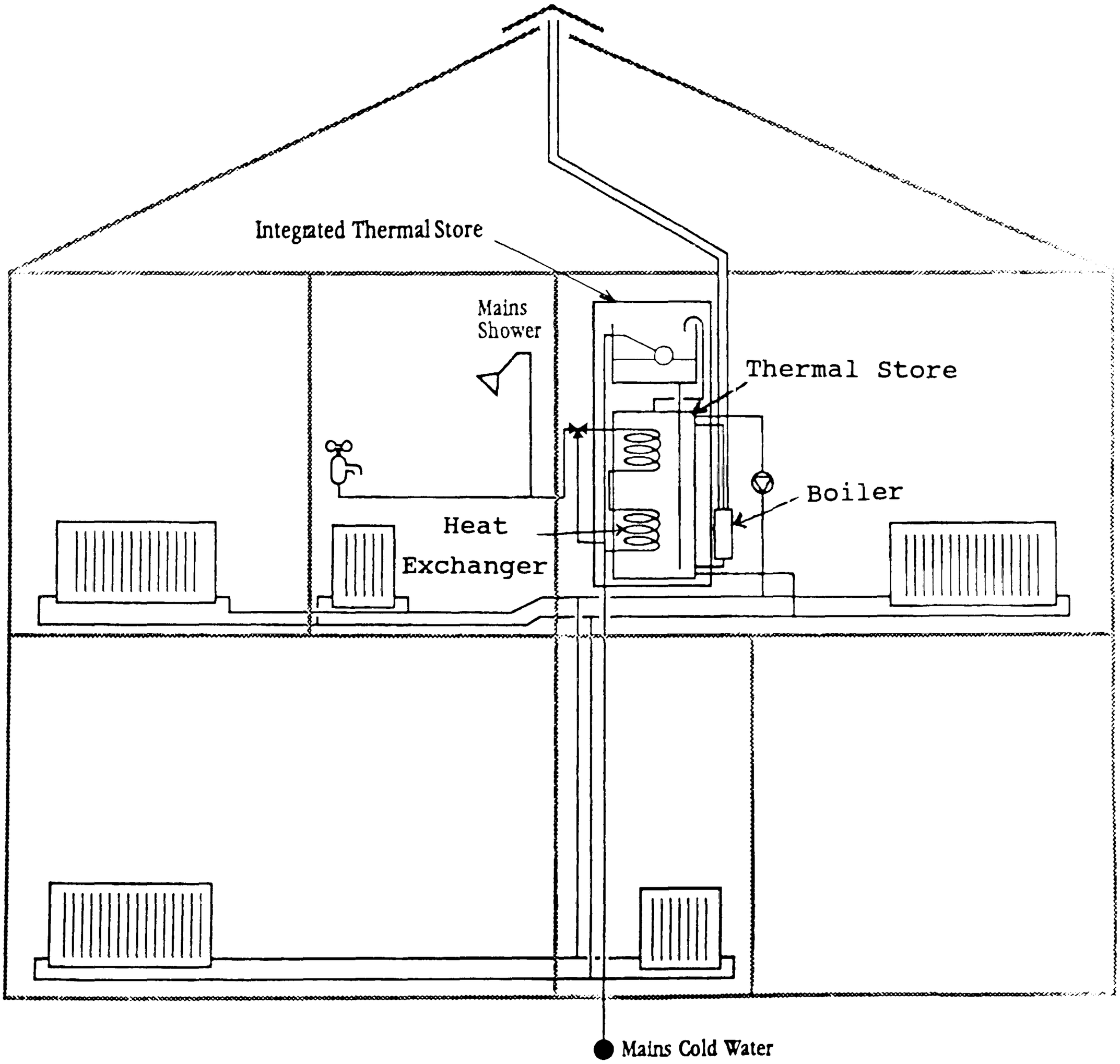


Fig.1.4.1(b): Individual Integrated Thermal Storage Heating System

1.4.2 Early Developments

Although the principle of the Integrated Thermal Store has been known for a long time, it is only in the early 80's that some interest started being shown in this type of system [5]. At that time, early prototypes of appliances for the full provision of domestic hot water and space heating in dwellings using the principle of the Integrated Thermal Storage were developed and extensive field trials were carried out [5],[6] to ensure the reliability and good performance of the system in the field.

During the early stages of development, a relatively detailed investigation of the advantages provided by this type of storage system was carried out by Tanton [5]. Tanton concluded that the main advantages provided by the use of Integrated Thermal Stores over existing conventional systems are:

1) **Reduction in boiler size**
By the use of thermal storage, boilers of a smaller size can be used. These smaller boilers operate at relatively higher loads and consequently higher efficiencies of heat generation can be reached. This contributes to lower the running cost.

2) **Reduction in boiler cycling**
By reducing the frequency of on-off operations of the boiler, the cycling losses of the system can be reduced. This also contributes to lower the running cost.

3) **Lower investment cost**
This lower investment cost than traditional systems is achieved through a reduction of the material and labour required at the installation. This is mainly a consequence that as the ITS is normally not pressurised, there is no need for a secondary hot water cylinder and associated piping for domestic hot water production.

An additional advantages of this type of systems comes from the relatively high pressure available in the domestic hot water network. This relatively high pressure eliminates the need for a boosting pump for the connection of showers to the hot water system. This can further reduce the installation of the system hence lowering the cost.

1.4.3 Recent Developments

Many extra features have been incorporated in new models of Integrated Thermal Stores. These include better controls, heat exchangers and heating mechanisms to improve the performance of the system. Furthermore a wide range of thermal stores and boiler output is now available to accommodate the most common range of heat loads.

Additionally, the principle of the Integrated Thermal Store is now also used in blocks of flats. In this system, one Integrated Thermal Store is used for each flat. The individual heat sources are replaced by a central boiler plant which delivers heat to each Integrated Thermal Store. This again presents several advantages over conventional group heating schemes. A simplified schematic of the system is presented in Fig.1.4.3.

One of the very first prototype of Integrated Thermal Store is presented in Plate I. Plates II and Plate III represent modern and improved versions of the same appliance. These appliances have been marketed relatively recently and can be purchased through specialised suppliers.

1.4.4 New Trends

Several recent trends have changed the ways in which modern central heating systems have to operate. The most important of these trends are:

1) Change in lifestyles
These are the results of an increasing standard of living which has changed the domestic and space heating demand patterns. For example, there has been a general move towards the used of showers (instead of baths) and a higher comfort level. The move towards showers has changed the heat demand pattern for the provision of domestic hot water as they require less heat and a more steady heat output than baths. The higher comfort level are usually linked to higher temperature or longer heating periods in living rooms.

2) Repeated oil crises
These oil crises have been mainly caused by political instabilities in in the middle east and have occurred repeatedly starting in the mid 70s. These oil crisis have triggered prices rises not only for oil but for a wide range of other fuels. Although the extraction cost of oil has not

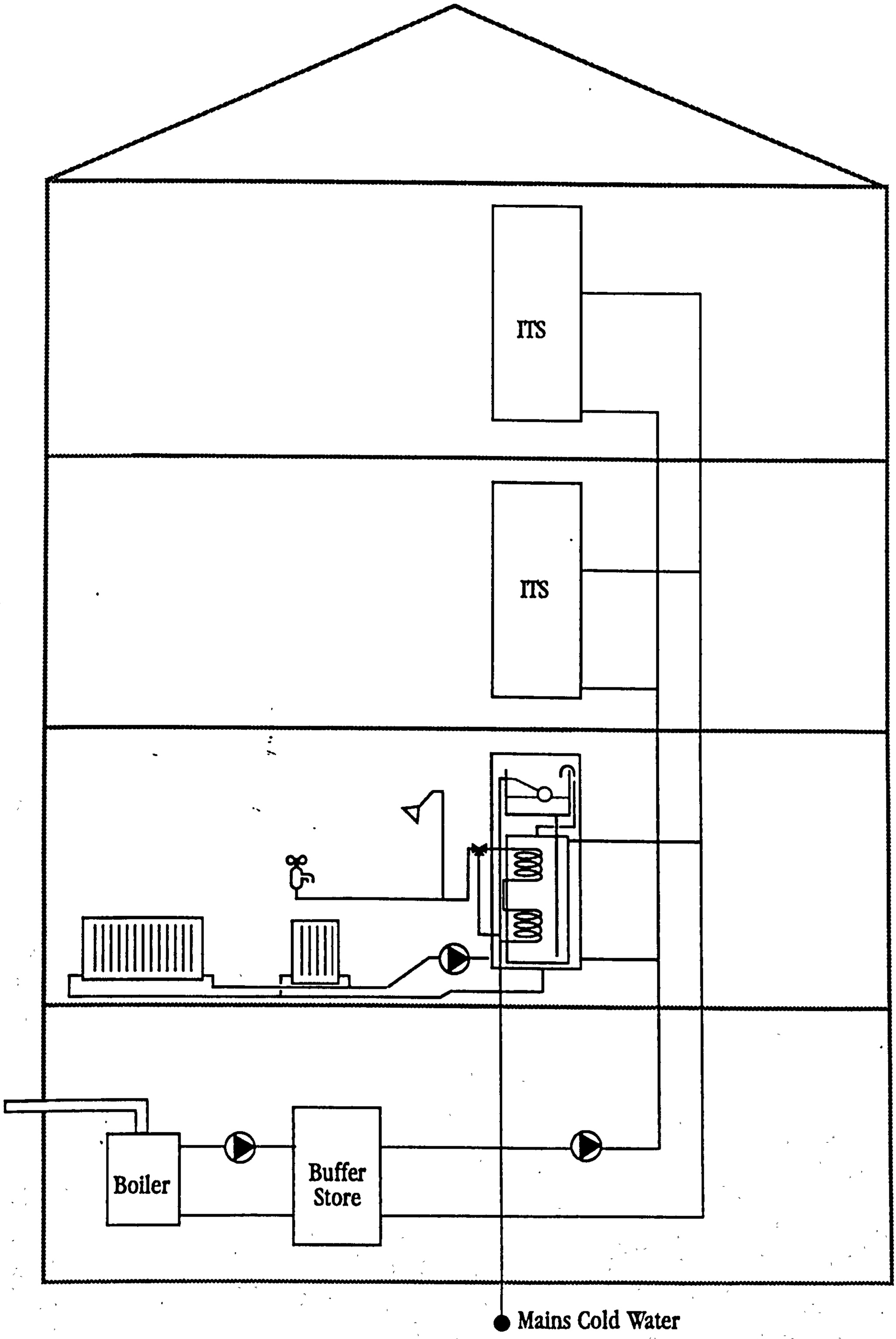


Fig.1.4.3: Integrated Thermal Storage Heating System for Group Heating

Plate I : First Prototype of Integrated Thermal Store

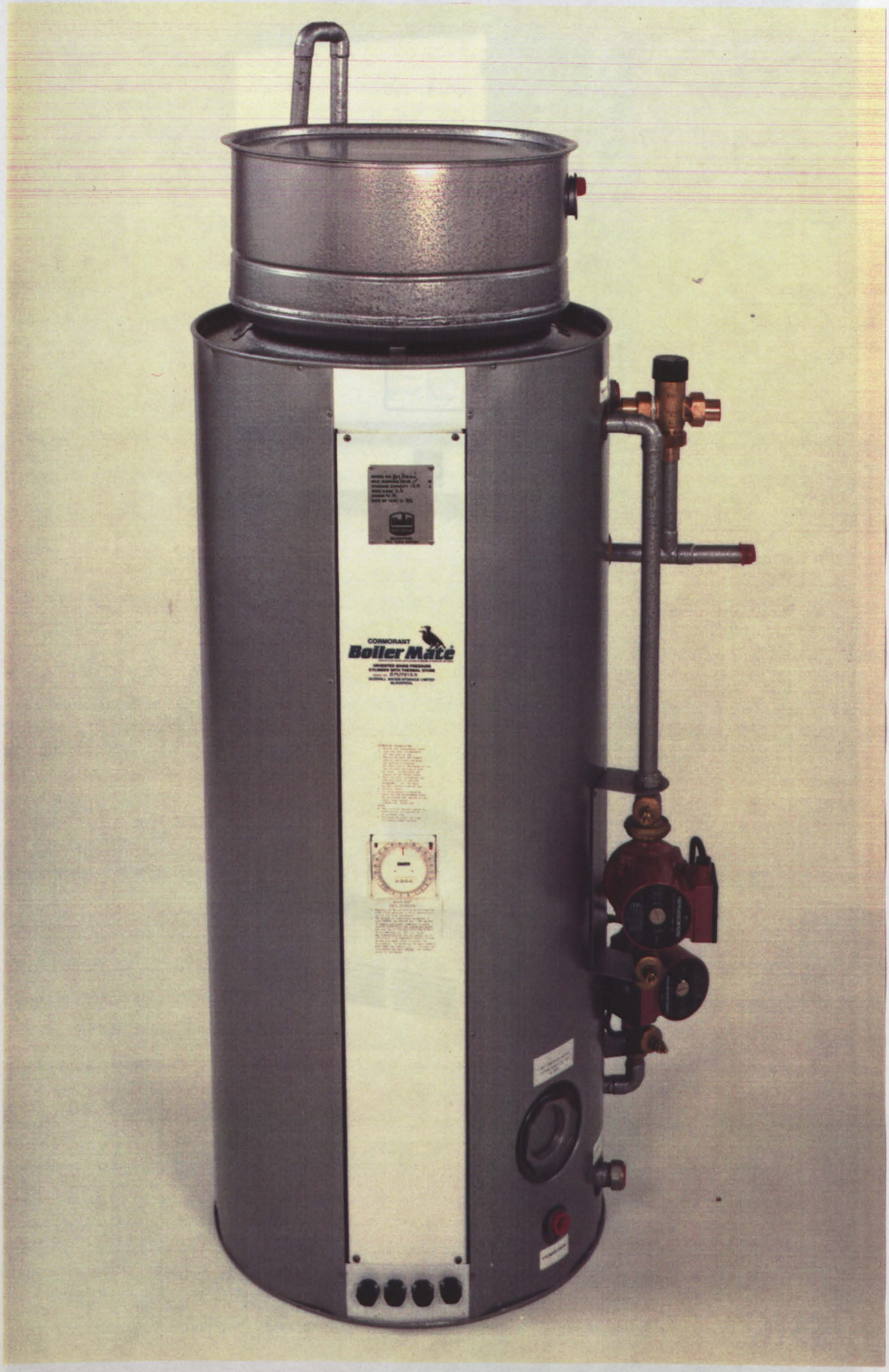


Plate II : Modern Version of Integrated Thermal Store

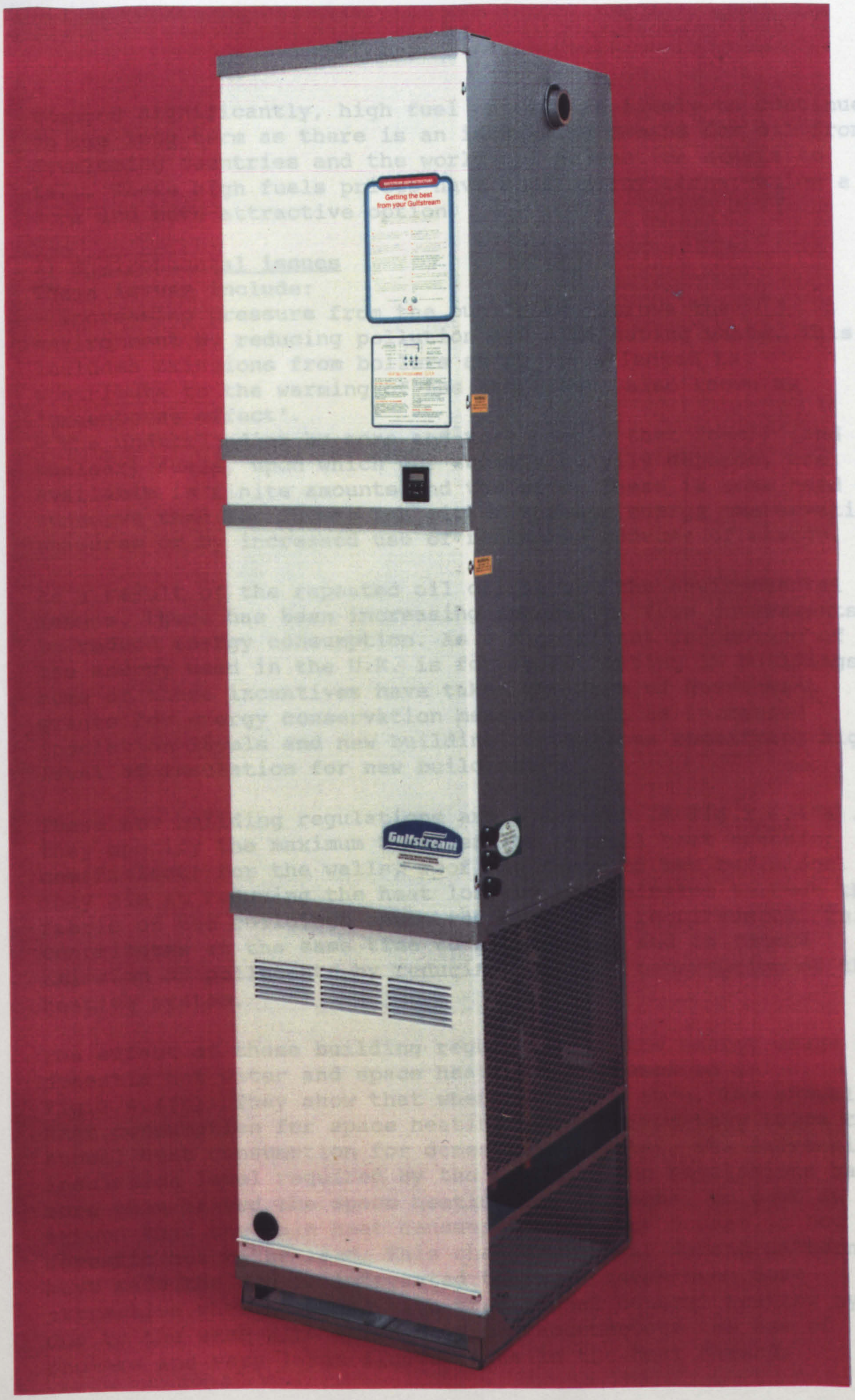


Plate III : Modern Version of Integrated Thermal Store with Gravity Circulation from the Boiler to the Store

changed significantly, high fuel prices are likely to continue on the long term as there is an increasing demand for oil from developing countries and the world oil production starts to fall. These high fuels prices have made energy conservation a more and more attractive option.

3) <u>Environmental issues</u>
These issues include:
- increasing pressure from the public to improve the environment by reducing pollution and eliminating waste. This includes emissions from boilers as CO_2 is alledged to contribute to the warming of the atmosphere also known as 'greenhouse effect'.
- the understanding by more and more people that fossil (and nuclear) fuels, upon which our society heavily depends, are available in finite amounts and therefore there is some need to conserve them for future use either through energy conservation measures or by increased use of renewable sources of energy.

As a result of the repeated oil crisis and the environmental issues, there has been increasing incentives from governments to reduce energy consumption. As a significant proportion of the energy used in the U.K. is for space heating in buildings, some of these incentives have taken the form of government grants for energy conservation measures such as increased insulation levels and new building regulations specifying high level of insulation for new buildings.

These new building regulations are presented in Fig 1.4.4(a). they specify the maximum U-values (or overall heat transfer coefficient) for the walls, roof and floor of new buildings. They aim at reducing the heat loss by transmission through the fabric of the buildings hence reducing heat requirements. This contributes at the same time to save energy and to reduce emission of pollutants by reducing the fuel consumption of the heating system.

The effect of these building regulation on the energy usage for domestic hot water and space heating are presented in Fig.1.4.4(b). They show that whereas in the 60's, the annual heat consumption for space heating was approximately twice the annual heat consumption for domestic hot water, the increasing insulation level required by the new building regulations have more than halved the space heating requirements, to such an extend that the main heat consumption for new houses is now the domestic hot water load. This changes in heat demand patterns have made the use of integrated thermal stores even more attractive when compared with traditional central heating systems due to the ease with which they can accommodate the use of showers and very large fluctuations in the heat demand.

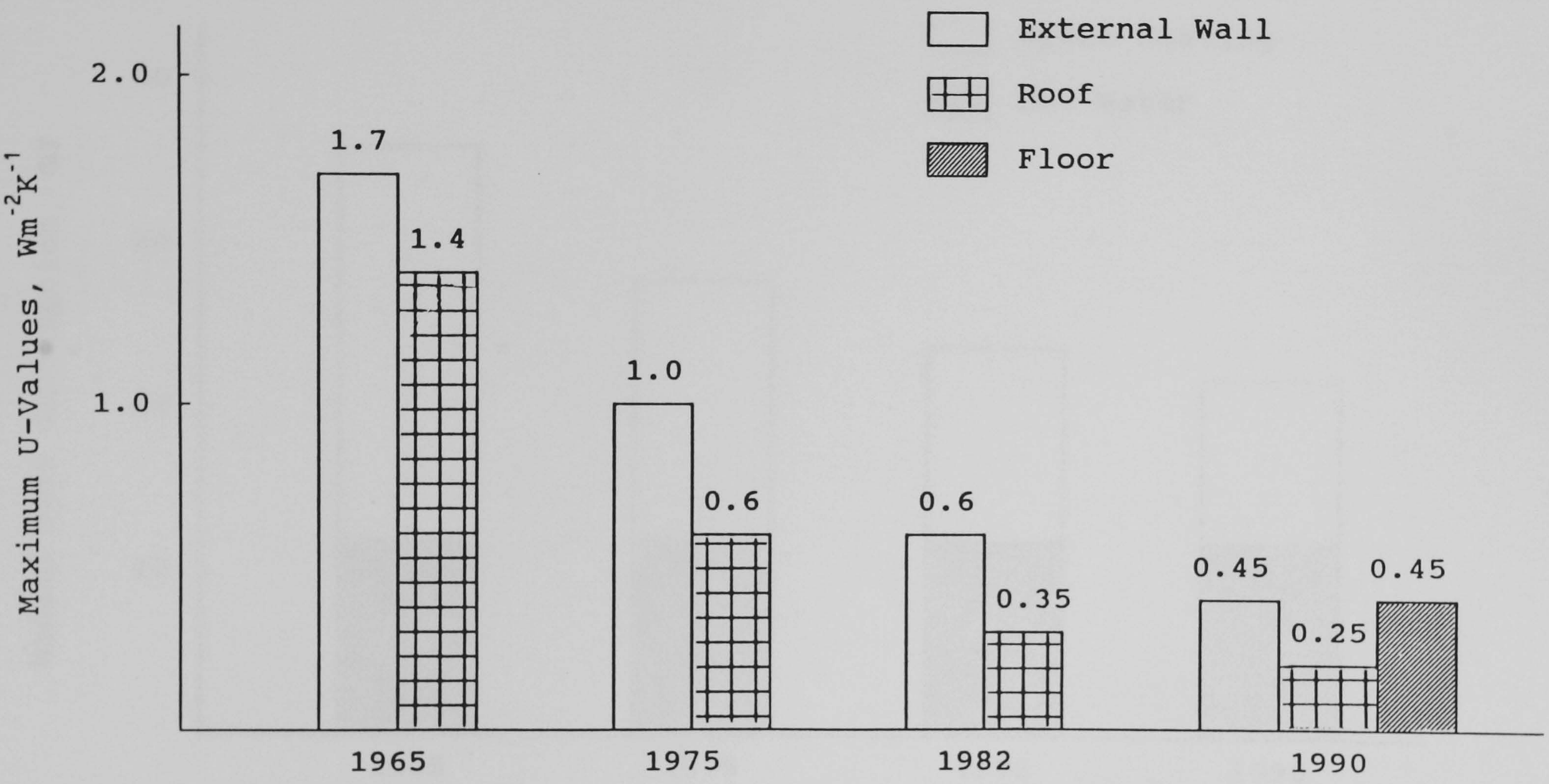


Fig.1.4.4(a): Insulation Requirements in Building Regulations

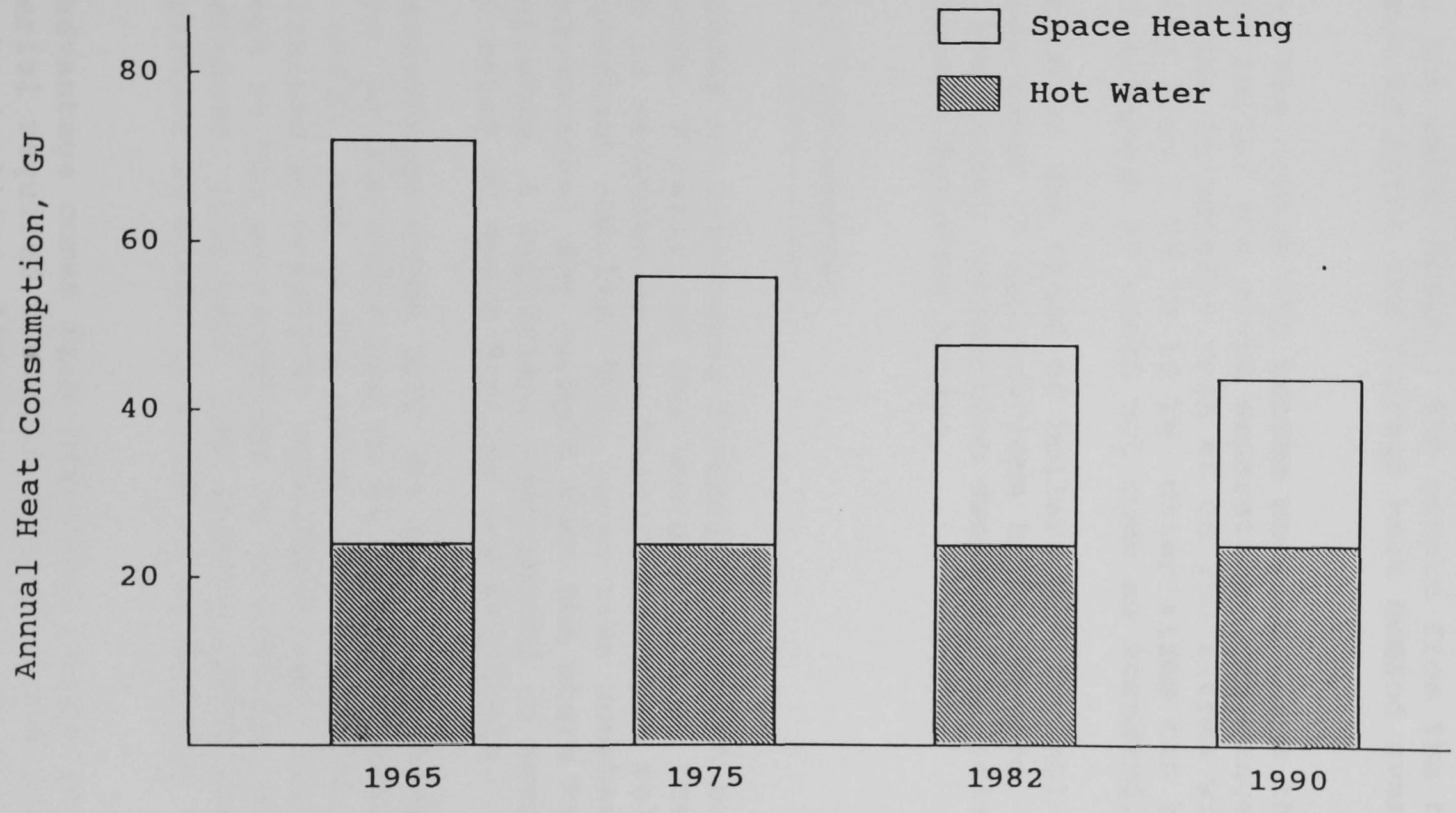


Fig 1.4.4(b): Estimated Annual Energy Consumption
for a Typical Three Bedroom House

1.4.5 System Design

By separating the heat demand and the heat generation, independent methods of sizing can be used for the thermal capacity of the store and the output of the heat source in ITS heating systems. The heat store is mainly sized to accommodate the domestic hot water demand. The output from the heat source is then chosen to match the average heat demand over a 24 hour period.

Consequently, the design can become modular where the thermal store and the boiler are sized separately. The range of store for domestic use is usually from 80 to 200 litres with boilers output ranging from 3 kW to 10 kW. Other sizes can be easily accommodated although it would not come as standard.

The combination of the range of boilers and thermal stores enables a wide range of applications to be covered. These range from modern low energy consumption dwellings to relatively large traditional detached houses.

1.5 Potential Improvements

The performances of Integrated Thermal Stores can be improved in several ways. Firstly, as the thermal mass of the store is usually high in relation to the boiler output, a relatively large time constant results. This large time constant can present disadvantages. For example when the store has been partially depleted, a sufficient time period is needed before domestic hot water or space heating are available.

A second disadvantage comes from the standing losses from the thermal store. As the store has to be kept at relatively high temperature (80^{o}C), and as the amount of insulation which can be used is limited by practical considerations, some heat loss from the store to the surroundings is inevitable. Although in some circumstances, this heat loss might benefit the dwelling were the appliance is sited in terms of background space heating.

Another disadvantage comes from the store itself in terms of the raw material required to build the store and floor area required in the dwelling. Although this is less than a conventional central heating system, this is far higher than say for a direct system with an instant heater.

1.6 Aims of the Work

Several areas can be investigated. However, of particular relevance is the design of the heat exchanger coil used to produce domestic hot water. The arrangement of the heat exchanger inside a rectangular thermal store is presented in Plate IV. This heat exchanger is the centre part of the system. A low effectiveness of heat transfer at the wall of the heat exchanger can only be partially compensated by increasing either the size of the store or the boiler output. Both are detrimental to the system in terms of investment and running cost.

Furthermore as the size of the store is dictated mainly by the domestic hot water demand, an increase in performance of the heat exchanger would lead either to a better service or to a reduced size of store. Both of these would improve the cost effectiveness of these Integrated Thermal Stores. In addition, a reduced size of store would also contribute to reducing the time constant of the system which can be beneficial.

At the moment, the design of the heat exchanger coil has only been partially investigated. The only reference seems to be Tanton [5] work which suggests that a coil with horizontal axis is better than a coil with a vertical axis. No heat transfer correlation is available to predict the performance of this heat exchanger in terms of heat transfer coefficient or effectiveness of heat recovery which presents disadvantages.

One of this disadvantage is at the design stage where the interaction between the space heating and the domestic hot water demand on the thermal store are not clearly understood. This make the sizing of the store and the boiler empirical although some rules of thumb can easily be deduced from past experience. A second disadvantage is that the performance of the heat exchanger in terms of overall heat transfer coefficient and ultimately effectiveness of heat recovery from the Integrated Thermal Store is difficult to evaluate this make the use of any numerical model for the prediction of the performance of Integrated Thermal Store difficult.

Also, the space heating mode of operation of an Integrated Thermal Store has never been investigated in detail to date. Again some improvement in the understanding this mode of operation would present advantages such as helping to improve their performance and maybe by changing their design help to reduce their manufacturing cost.

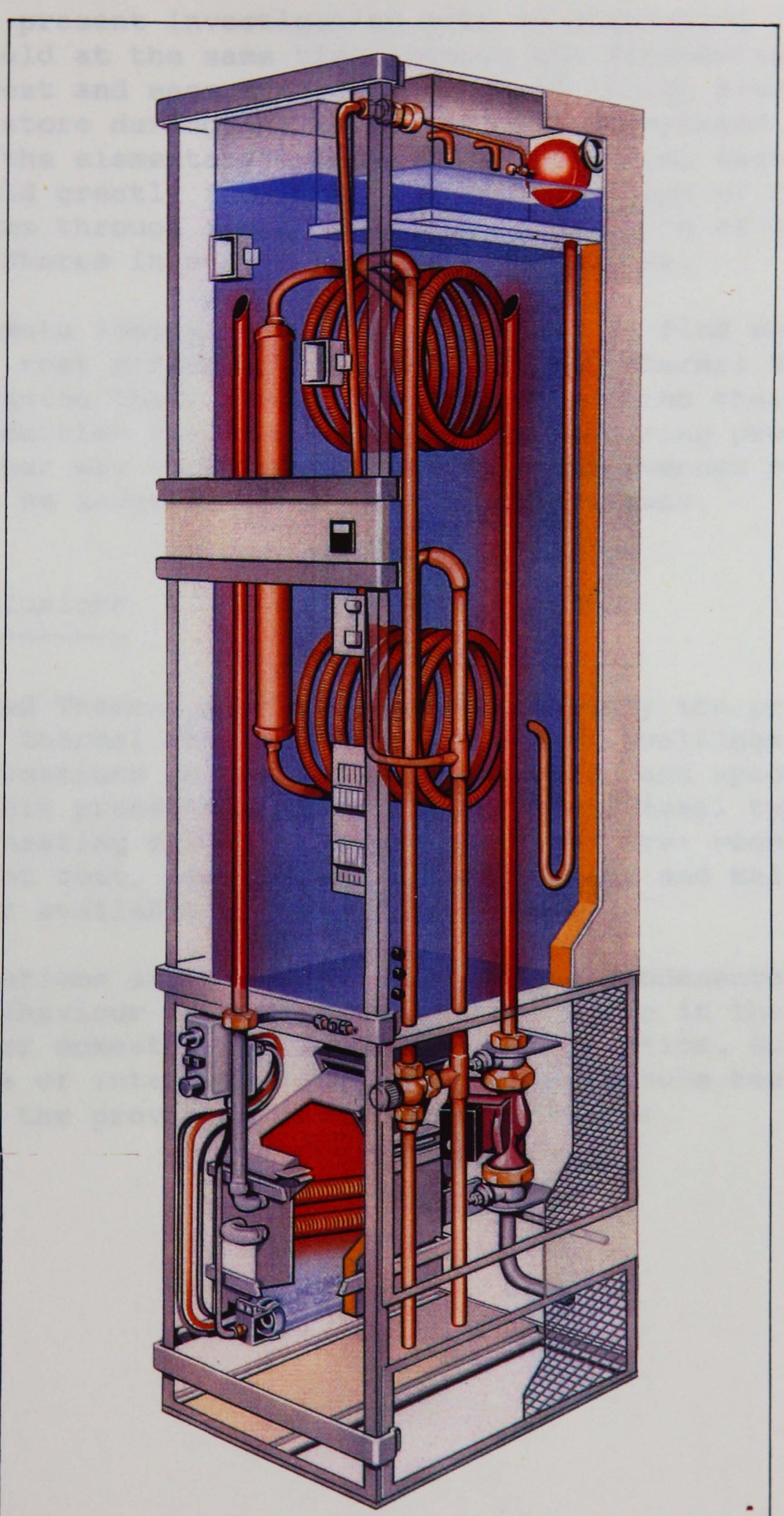


Plate IV : Arrangement of the Heat Exchanger
in the Integrated Thermal Store

Thus the present investigation aims at developing correlations which would at the same time improve the fundamental knowledge of the heat and mass transfer mechanism taking place in the thermal store during operation and help understand the behaviour most of the elementary components constituting thermal store. This could greatly facilitate the design stage of these types of systems through a much better understanding of Integrated Thermal Stores in normal operating conditions.

The ultimate long term aim is to suggest or find ways to achieve a better cost effectiveness of Integrated Thermal Stores either by increasing their performance or by reducing their cost. Costs reduction through change of manufacturing process would be an other way to improve the cost effectiveness ratio but will not be investigated in the present theses.

1.7 Conclusions

Integrated Thermal Stores use advantageously the principle of sensible thermal energy storage in modern dwellings to reduce the fluctuations in the domestic hot water and space heating loads. This presents advantages over traditional types of central heating systems. These advantages are: reduced investment cost, improved boiler efficiency and mains pressure hot water available at the point of use.

Investigations are needed to improve the fundamental knowledge of the behaviour of Integrated Thermal Store in the space heating or domestic hot water mode of operation. One of the main area of interest is the coiled finned tube heat exchanger used for the provision of domestic hot water.

References

[1] Nevrala, D.J.
Heat Services for Future Housing,
Part 1: the Insulated House Design Requirements
Building Services Engineer, 45, October 1977, pp 107-117

[2] Pimbert, S.L. and Peat, B.J.
Hot Water Usage Patterns in Dwellings: Measurements by Non-intrusive Instruments
Build Serv. Eng. Res. Technol., Vol 10, No 4, 155-157, 1989

[3] Central Heating: a Guide to the Design and Operation of Domestic Wet Central Heating Systems
Watson House Bulletin, Vol 47, No 2, March 1983

[4] Nevrala, D.J. and Green, M.B.
Thermal Energy Storage in Domestic and Commercial Heating and Hot Water Systems
International Gas Research Conference, Toronto, Canada, 1986

[5] Tanton, D.M.
Some Aspects of The Use of Water Filled Thermal Stores in Central Heating System
PhD Thesis, Cranfield Institute of Technology, U.K., 1986

[6] Nevrala, D.J. and Pimbert, S.L.
What the Cormorant Has in Store
Watson House Bulletin, Vol 48, No 2, 1984

Chapter 2

Experimental Apparatus

2.1 Introduction

The rig used to obtain data on water based thermal store is presented in Fig.2.1 and in plate V. It is designed to simulate the behaviour of a domestic Integrated Thermal Store when in the domestic hot water or space mode of operation.

The main components of the rig (volume and height to diameter ratio of thermal store and type of heat exchanger) were chosen to correspond to commercially available sizes of Integrated Thermal Stores. This ensured that all the experimental investigations were carried out on the same scale that would be encountered in practical situation and therefore greatly simplified the analysis of the experimental data and the interpretation of the experimental observations.

The domestic hot water demand was simulated by passing chilled water in the heat exchanger coil located within the thermal store as would normally occurs on the field. The space heating demand was simulated by circulating hot water from the experimental storage tank into a small temperature controlled water store. The detail of this arrangement will be explained later but it presented the main advantages of being simple to operate and capable of simulating a very wide range of space heating demands.

Finally, it was possible, using this experimental apparatus to vary widely the type of heat discharge which could be achieved in terms of heat extraction rate from the thermal store, duration of thermal discharge and type of thermal discharge (domestic hot water draw off or space heating demand). This ensured that the most commonly occurring operating conditions to which Integrated Thermal Store were subjected in the field could be simulated easily.

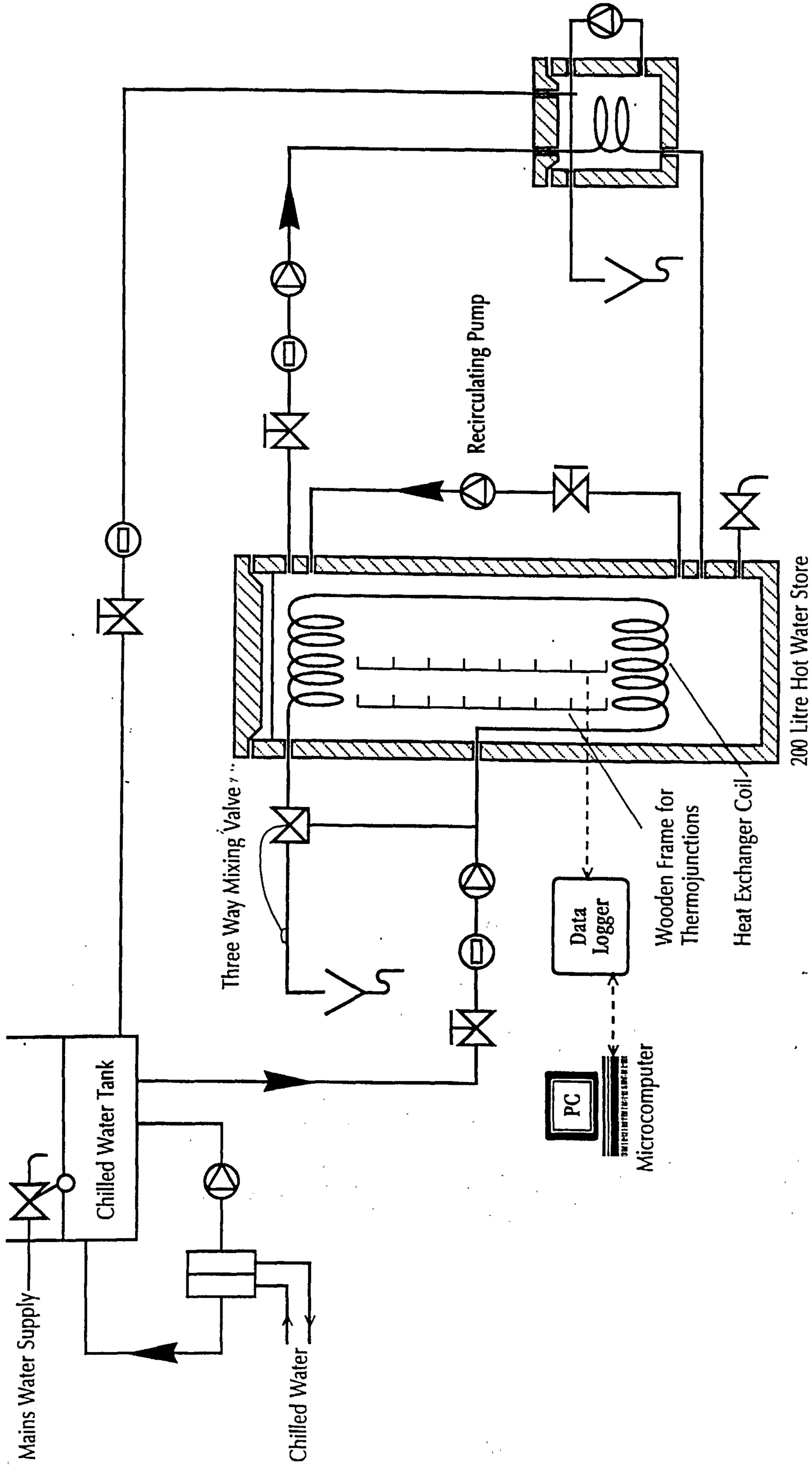


Fig.2.1: Simplified Schematic of Experimental Rig

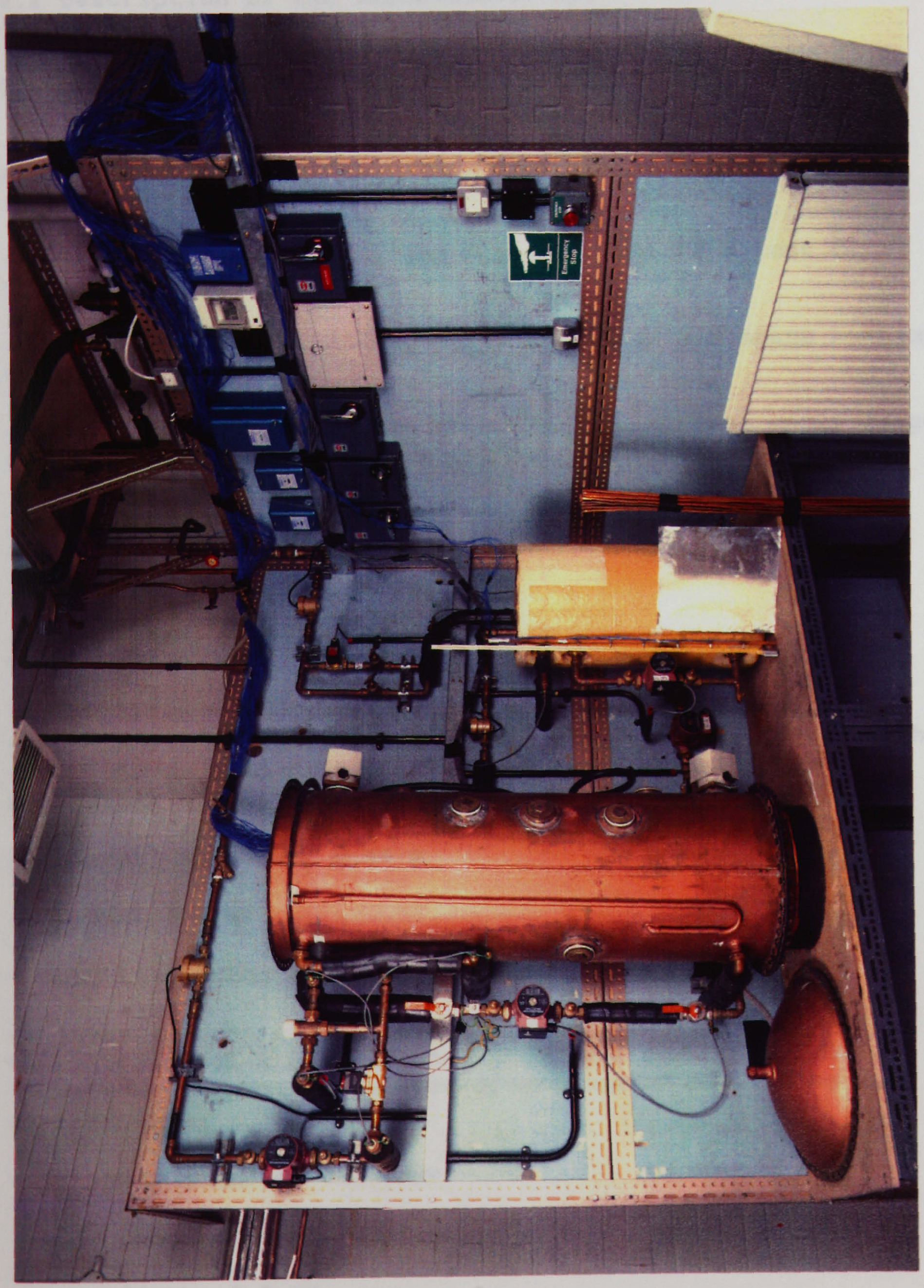


Plate V : General View of the Experimental Rig

2.2 Description of the Experimental Rig

The main components of the experimental rig were:

- a 200 litre hot water store with an internal finned tube heat exchanger coil of the type used in Integrated Thermal Stores.

- a powerful data acquisition system including a fast data logger and microcomputer.

- facilities to simulate domestic hot water and space heating demands.

- a chilled water supply which could deliver reasonably large quantity of constant temperature chilled water to the rig.

- a heating mechanism to heat up the thermal store at the beginning of the experiment.

2.3 Hot Water Storage Tank

The experimental storage tank is represented in Fig.2.3 and is shown in Plate VI. It was made of copper and was fitted with removable ends to facilitate access to the inside. The dimensions of the store were:

- internal diameter: 0.44 m

- height: 1.4 m

- mass (including the heat exchanger coil): 52 kg

An internal heat exchanger coil was fitted into the storage tank. This heat exchanger was a standard component used in Integrated Thermal Store. The dimensions of the tubing are as follows:

- internal diameter: 10.9 mm

- wall thickness: 0.7 mm

- fin height: 3.2 mm

- fin pitch: 2.3 mm

- fin thickness: 0.4 mm

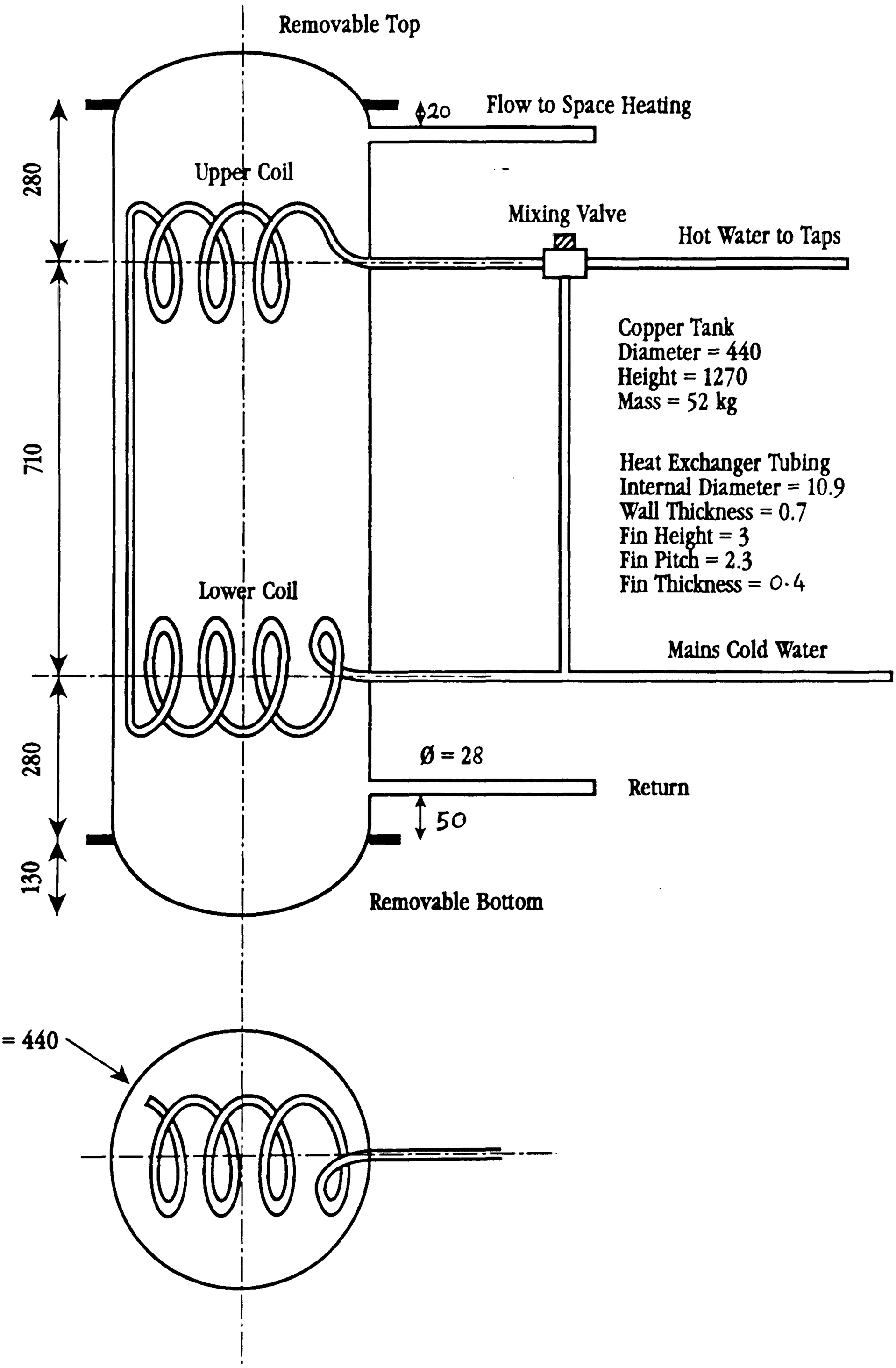


Fig.2.3: Detailed Schematic of Experimental Storage Tank

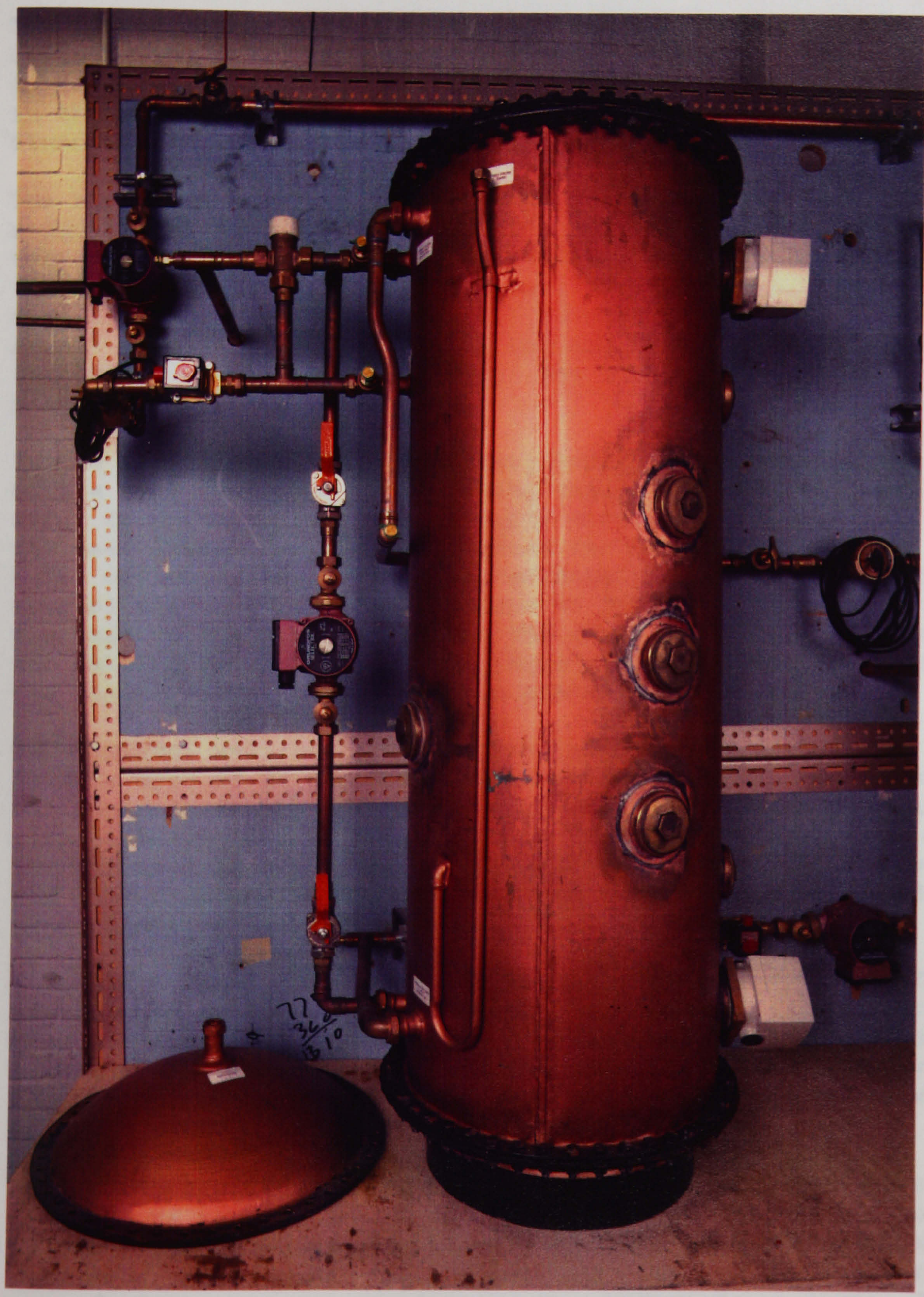


Plate VI : Hot Water Storage Tank Used in the Experiments

The finned copper tubing of the heat exchanger was rolled in 4 coils having a 220 mm diameter. Two of these coils were located in the lower region of the store and the two other were located in the upper region of the store. The total length of the finned tubing was approximately 24 metres.

The heat exchanger was made of two independent channels to separate the fluid flowing into the heat exchanger coil into two streams. Each of these channel forced the fluid flowing in the heat exchanger first into a coil located at the bottom of the store and then at into a coil located at the top of the store. At the top of the store, the two streams were mixed at the outlet of the heat exchanger. The heat exchanger is presented in plate VII.

It was possible to use a three way mixing valve at the outlet of the heat exchanger. This mixing valve by-passed some cold water from the chilled water supply directly to the outlet of the heat exchanger. This maintained the temperature of the water delivered to the taps approximately constant during the thermal discharge. The temperature of the water delivered to the taps could be varied within the range 31-62°C by changing the setting temperature of the three way valve. A close view of the mixing valve is presented in plate VIII.

The vertical wall of the store was insulated with cylinder jackets made of 50 mm of fibrous insulant. The top of the store was covered with approximately 50 mm of polyurethane foam and with approximately 50 mm of fibrous insulant the same type of the one used in the cylinder jackets. The bottom of the store was left uninsulated but was sited on a wooden frame to reduce conduction losses to the floor.

A variable speed pump was fixed to the side of the store. The aim of this pump was to mix the water in the 200 litre store thus avoiding large temperature gradient between the top and bottom of the store. During the heating period, this achieved a lower temperature gradient within the store hence a rapid warm-up.

During the experiments the use of the mixing pump made possible to vary the temperature gradient between the top and the bottom of the store thus reducing stratification. When the pump was running at full speed, the temperature gradient in the store was so small that the store could be assumed fully mixed. The use of a mixed tank was useful for theoretical considerations which will be explained later.

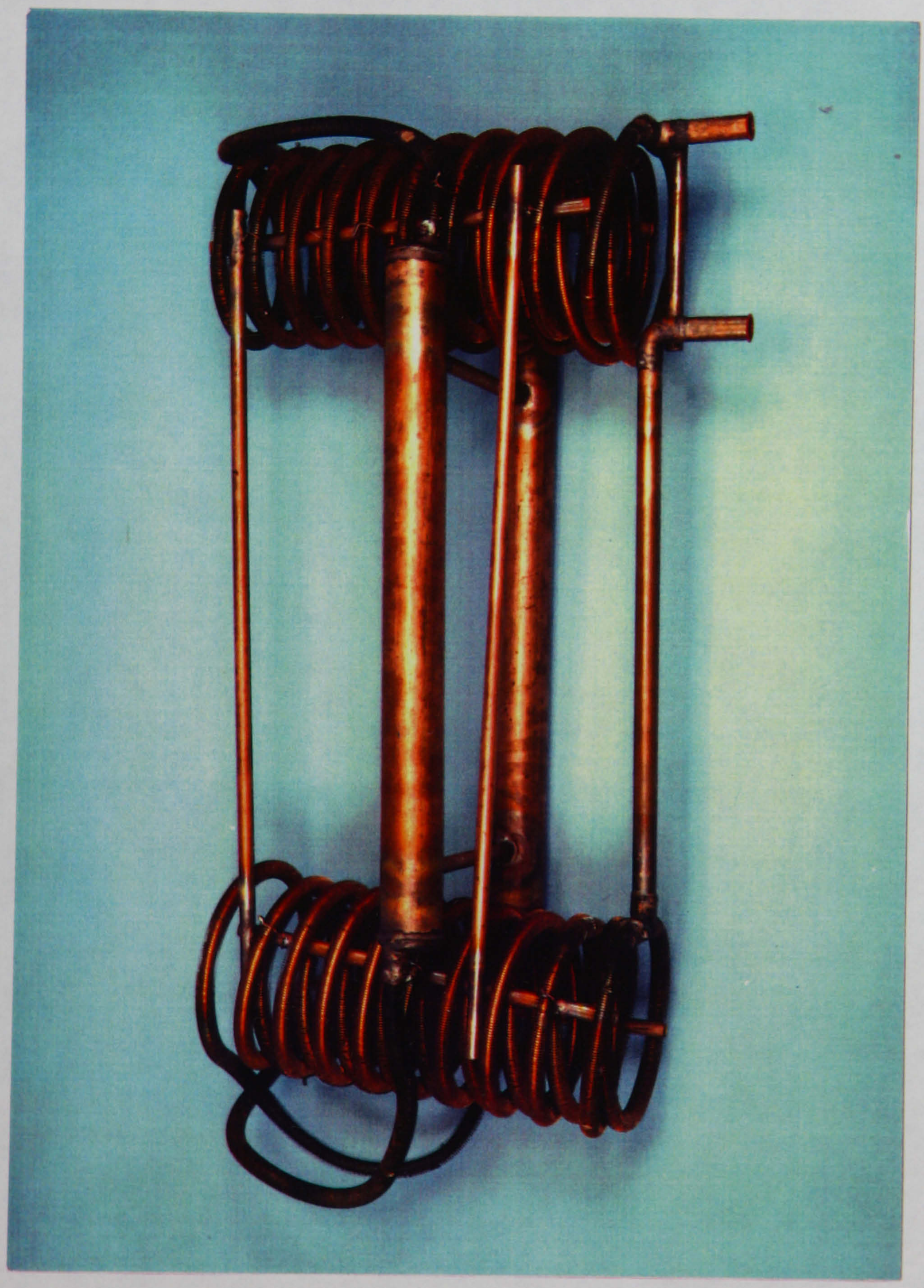


Plate VII : Finned Tube Heat Exchanger Coil
for Domestic Hot Water Production

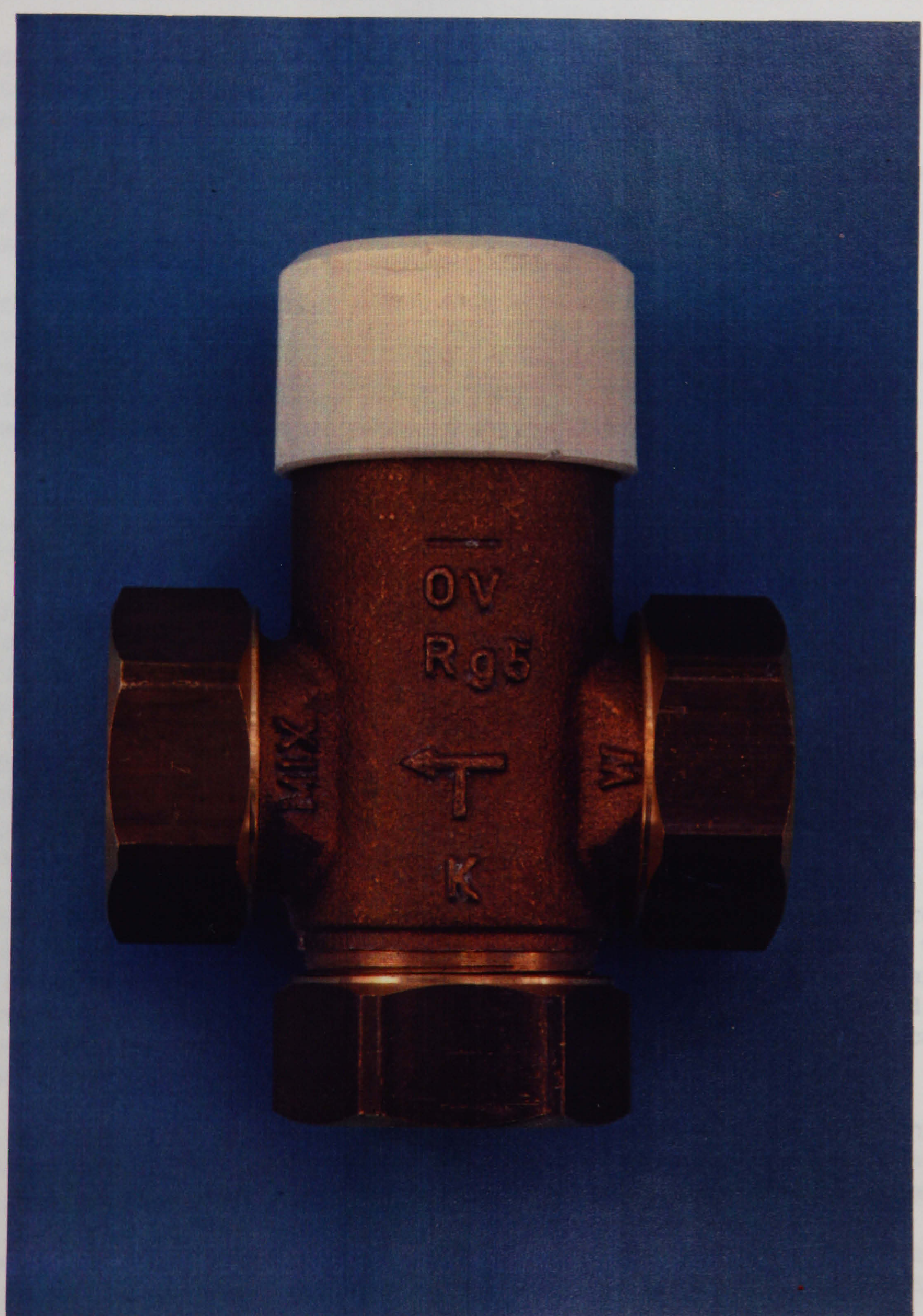


Plate VIII: Close View of the Three Way Mixing Valve

2.4 Domestic Hot Water Facility

The domestic hot water draw-off was simulated by passing cold water from the chilled water supply into the heat exchanger coil located within the 200 litre hot water store.

A variable speed pump was inserted between the chilled water supply and the heat exchanger so that the flow rate passing into the heat exchanger could be varied.

The maximum flow rate which could be passed into the heat exchanger was achieved when the pump was at maximum speed and was around 10 litre/min. Lower flow rates could be achieved either by changing the pump speed or by the use of a throttling valve situated on the water supply pipe.

2.5 Space Heating Facility

The space heating load was simulated by extracting hot water from the top of the 200 litre store, passing it into a 'radiator' where its temperature was decreased and then returning it to the bottom of the 200 litre store.

The 'radiator' was a water/water heat exchanger coil located in the 50 litre store. It was sized to achieve the rate of heat transfer that would normally balance the heat loss from a traditional house during a cold winter day.

This 50 litre store was kept at relatively low temperature by means of the chilled water facility. The cooling rate of the 50 litre store could be varied during the course of the experiment using a secondary cooling system controlled by a solenoid valve located on the chilled water supply. The solenoid valve could be controlled remotely by a computer interface.

A mixing pump was used to create strong convection currents within the 50 litre store. This ensured a high heat-transfer rate between the water in the 50 litre store and the water flowing into the heat exchanger coil. Consequently, the temperature of the water returning to the 200 litre store was always lower than the temperature at which it was extracted.

The maximum flow rate which could be passed into the 'radiator' and the 200 litre hot water store was around 12 litre/min. Again lower flow rates could be achieved by changing the speed of the space heating pump.

The cooling rate of 200 litre store could be varied by changing the cooling rate of the 50 litre store or reducing the speed of the space heating pump or reducing the speed of the mixing pump on the 50 litre store. It could also be changed during the experiment through the possibility to vary the cooling rate of the 50 litre store.

The maximum cooling rate of the 50 litre store was evaluated at approximately 15 kW. As the cooling rate of the 50 litres store could be varied with time, a large range of space heating demands could be simulated.

Finally, some instruments were inserted into the piping. This included a flowmeter to measure the volumetric flow rate of water flowing into the space heating circuit and high accuracy temperature sensors to measure the temperature of the water entering and leaving the 200 litre and 50 the litre stores.

2.6 Chilled Water Supply

All the water used during the experiments was precooled by means of a chilled water facility. This included water used to simulate domestic hot water and space heating demands.

Water from the mains supply was used to continuously refill a 250 litre store so as to keep its water level constant. Water from this store was passed onto a plate water/water heat exchanger connected to a chilled water supply available in the laboratory where its temperature was lowered to around 7°C and was then returned to the 250 litres store.

As the 250 litre store was uninsulated, it could not exactly reach the temperature of 7°C. But the temperature of the water in the 250 litre store could be kept relatively constant to around 8°C for all the experiments.

2.7 Heating Mechanism

The store was heated by two electric immersion heaters. Each heater had an output of 6 kW and could be controlled by a rod thermostat in the temperature range 60-80°C. The differential of the thermostat was fixed to 4°C. The immersed length was 0.41m to ensure a good uniformity of the temperature in the store during heating.

2.8 Instrumentation

Measurements were taken using three type of sensors:

- flowmeters
- high accuracy Platinum Resistance Thermometers (PRTs)
- K-type thermocouples

The flowmeters used to measure the flow of water in the pipes were rated at 70 pulses per litre. This high pulsing rate was required to provide satisfactory accuracy of flow measurements at low flow rate. In the flow range considered (from 3 to 15 litre/min), the maximum error given by the flowmeter was around 2 %.

To this error should be added the error caused by the measurement of pulses from the data logger (±1 pulse) and the errors created by the imperfections of the rig (such as proximity of the bends and tes). It is estimated that the contribution of these was around 1 %.

This gave a maximum error occurring in the flow metering of around ±3 % of the maximum operating range or 0.1 litre/min. This is not thought to be significant except for measurements at very low flow rate.

High accuracy PRTs were used to measure the temperatures of the water flowing in the pipes. These PRTs were 4 wire made and had a A/10 class. The probes used with these PRTs were 3 mm in diameter and were inserted by approximately 100 mm in the pipes so that the conduction of heat along the probes could be neglected. Finally the accuracy of temperature measurements of these PRTs was better than 0.1 ^{o}C in the temperature range considered.

The K-type thermocouples were used to measure the temperatures of the water in all the thermal stores. In the 200 litre store, the thermojunctions were located as represented in Fig.2.8(a). The location of these thermojunctions was chosen to measure the degree of stratification in the 200 litre store and to try to record the effect of a boundary layer developing near the wall of the store.

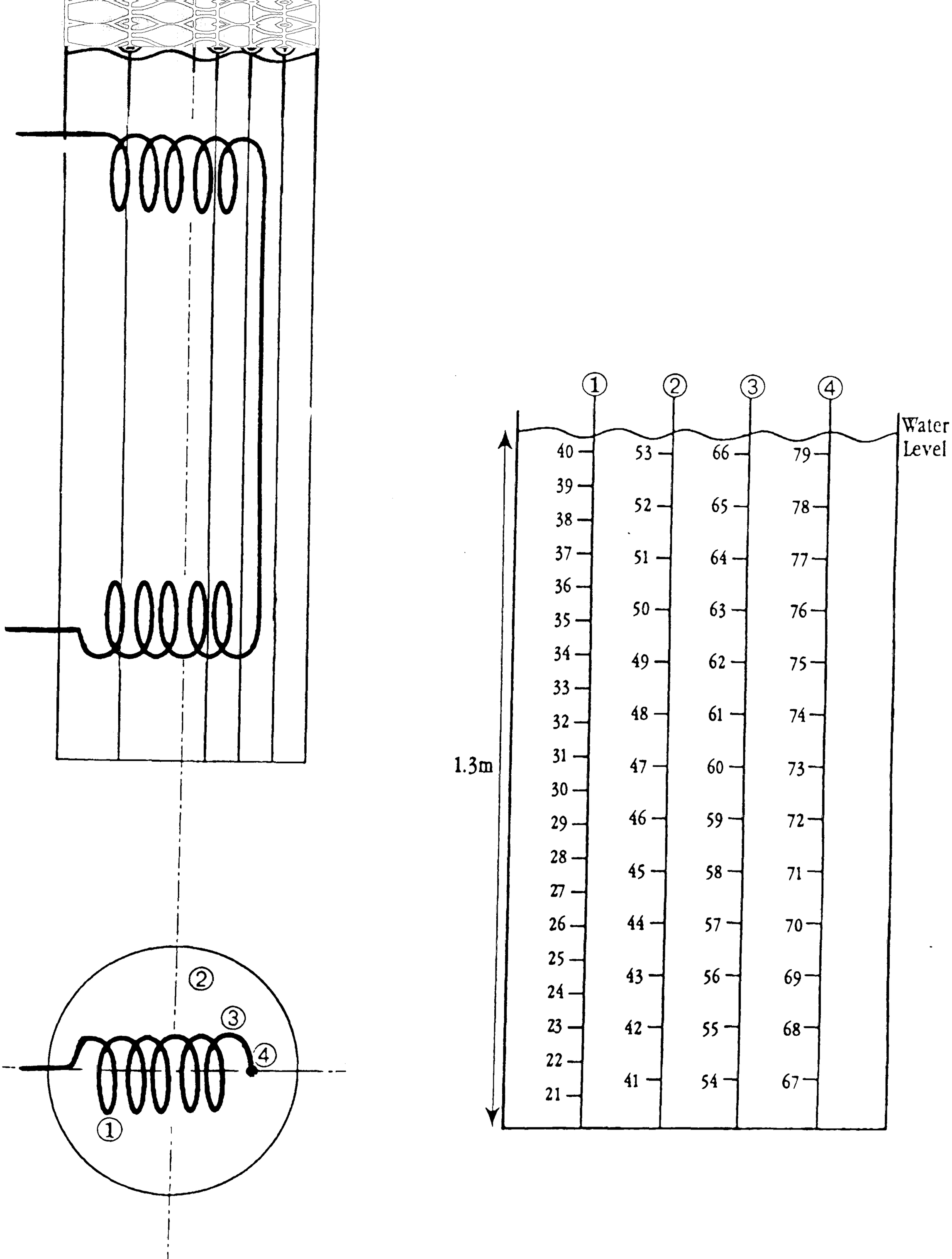


Fig.2.8(a): Location and Numberring of Thermojunctions in the Experimental Storage Tank

Calibration of the Thermocouples

The accuracy of the thermocouple temperature measurements was increased by using correction factors obtained by comparing the temperature measurements of the PRTs with temperature measurements of the thermocouples.

A first experiment was carried out in an isothermal bath to compare the measurements of the PRTs among themselves. Over the temperature range 40-80°C, the maximum deviation around the average value was found to be 0.03°C.

A second experiment was carried out to determine the correction factors to apply to the thermocouple readings. In this experiment, the temperature measurements from the PRTs and thermocouples were compared in the same isothermal bath.

As the number of thermocouple measurement was relatively high, it was found more convenient to analyse the results using simple statistical methods for the calculation of the correction factors to use.

The correction factor to apply to the thermocouple measurements is defined as :

$$C_f = T_{th} - T_{PRT} \qquad (1)$$

where T_{th} is the average temperature as measured by the thermocouples

T_{PRT} is the average temperature as measured by the PRTs

Also relevant is the average error or 'dispersion' of the temperature measurements . The dispersion represents how far is on average a corrected temperature reading using a thermocouple from a the average temperature reading using a PRT. The dispersion is defined by:

$$D_f = \frac{1}{n} \Sigma \left| T_{tc} - T_{PRT} - C_f \right| \qquad (2)$$

where T_{tc} is a temperature measurement using the considered thermocouple

C_f is the correction factor as defined by equation (1).

The correction factor to use for thermocouple measurement and the dispersion as function of temperature are represented in Fig.2.8(b). The correction factor varies approximately in an parabolic fashion with temperature. It is only significant at temperature above 40°C. By using this correction factor, the dispersion is usually around 0.1°C. This means that on average, the difference between the corrected temperature measurements of the thermocouples with PRTs are within 0.1°C.

As we will see later, accurate temperature measurements are required to evaluate the intensity of the buoyancy driven convection current occurring in the thermal store during a thermal discharge. The temperature difference measured are usually of a few degrees and therefore the accuracy of the corrected temperature measurements from the thermocouple is largely sufficient for our experiments.

2.9 Data Processing

The data processing was achieved using a data logger, a microcomputer and an interface relay box. These are shown in plate IX.

The data logger measured the temperatures from the thermocouples, PRTs and counted the pulses from the flowmeter. The sensitivity of the logger with respect to the temperature measurements was 0.01°C.

The fastest scanning rate was given by the speed at which the logger could read the thermocouples and PRTs. This was approximately 8.5 seconds. Scanning faster was possible but only with loss of accuracy and sensitivity in the temperature measurements which was not desirable. The time interval between two scans had to be increased in some experiments to reduce the amount of data to be stored by the computer.

The data from the logger were then sent to the computer via a serial link rated at 9600 BAUDS. The computer was in charge of storing the measurements on a 1.2 MB floppydisc or on a 20 MB hard disc depending on the type of experiment. The computer was also in charge of controlling the logger and displaying the data on a screen. All these duties could be performed in approximately 1 second.

During the experiments, the microcomputer and the data logger were used to process the data at the same time. Consequently, the maximum processing speed was given by the logger which was the slower to perform its tasks.

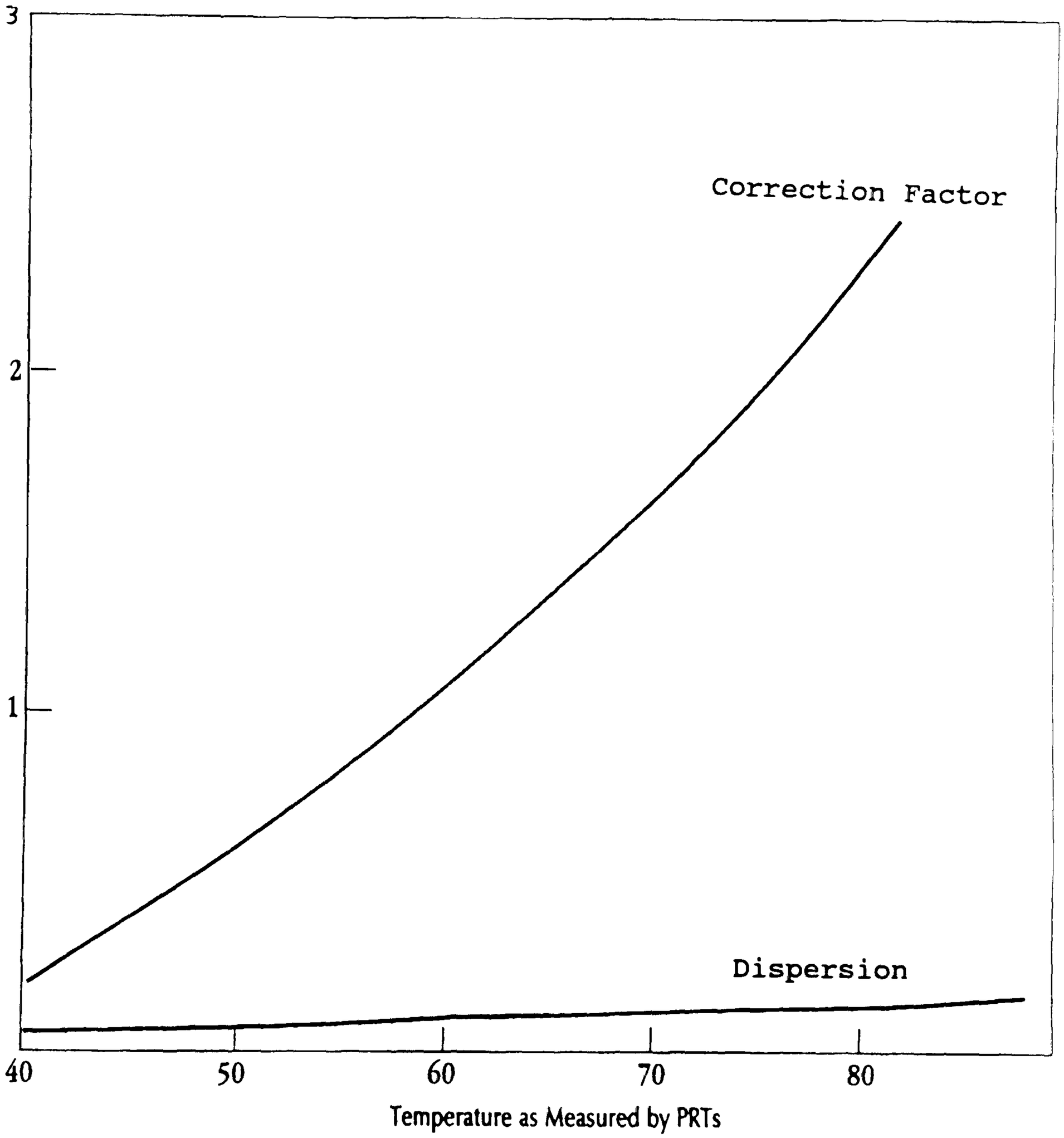


Fig.2.8(b): Average Dispersion and Correction Factors for Thermocouple Measurements

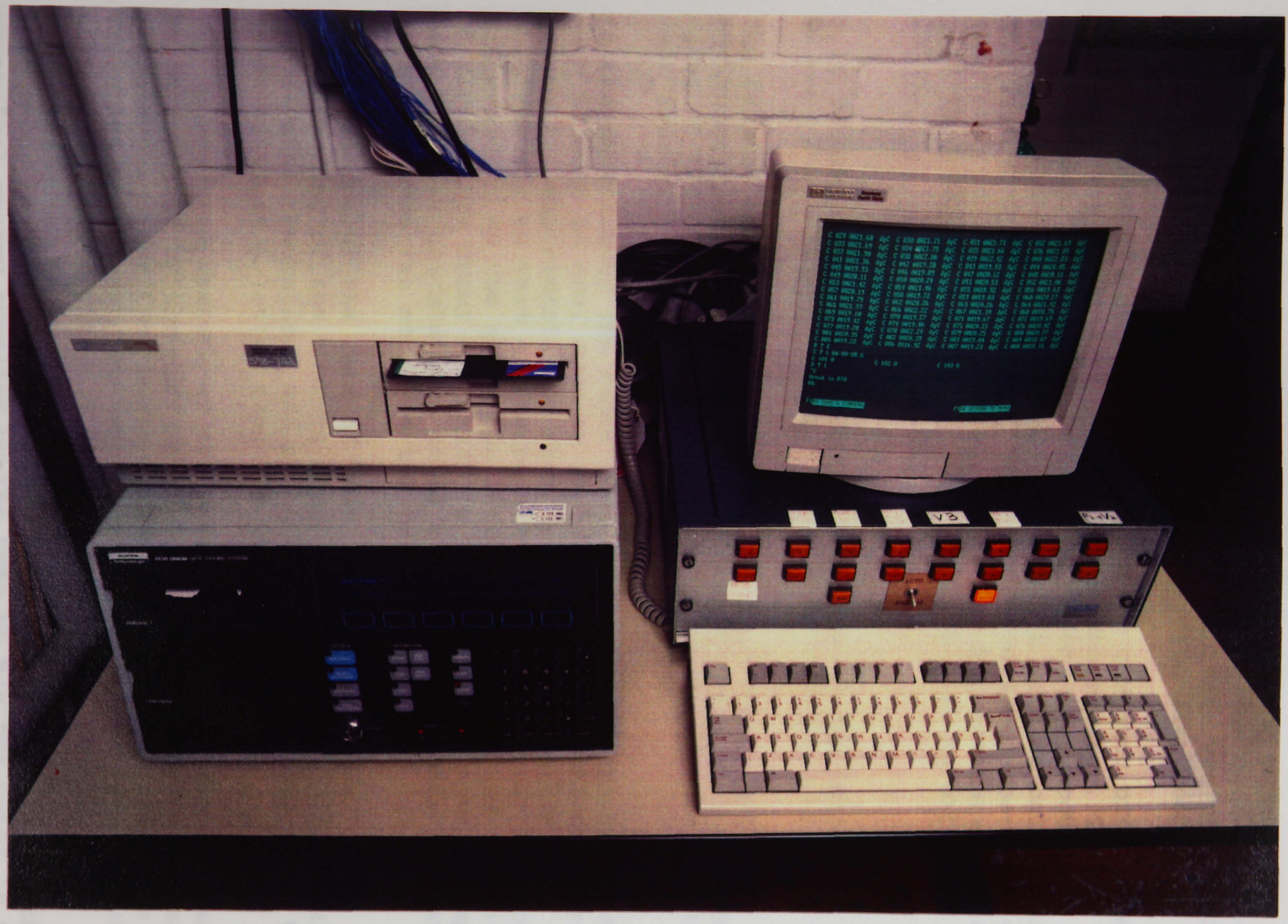


Plate IX : Microcomputer, Logger and Interface Box
for Data Processing and Analysing

The interface relay box was used to control the pumps, the solenoid valves and the heaters used during the operation of the rig. During the experiments, the interface was controlled by the processing facility available in the data logger.

At the end of the experiment, a separate computer program was used to analyse the data on the same microcomputer. This included applying correction factors to the thermocouples readings and carrying simple calculations such as calculating the mass flow rate in the pipes or the average temperature of the water in the tank. Finally, the same microcomputer was also used to develop the computer models which will be exposed later.

2.10 Determination of the Standing Losses of the Store

The rate of heat loss from the 200 litre store to the environment was evaluated by measuring the rate of temperature decay of the hot water store with time in the absence of any heat source.

In the experiment, the store was first heated up to 80°C and then allowed to cool down by the process of natural cooling. During the experiment, the recirculating pump was run continuously so that any temperature gradient within the store was avoided. As the store was nearly isothermal, and the rate of temperature decay was small (a few degree per hour), simple heat transfer calculations could be used to evaluate the rate of heat loss from the store to the environment.

The knowledge of the rate of heat loss improves the understanding of the thermal store. This knowledge is useful for several theoretical and practical considerations which will be detailed at a latter stage in this thesis.

The average temperature of the water in the tank was assumed to be represented by the average of the corrected thermocouples measurements from the thermojunctions numbered 1 to 20. The variation of the average temperature of the water in the tank with time is summarized in Table I.

Some error in this experiment is induced by the fact that the mechanical energy supplied by the pump to mix the water in the store ultimately decays into low grade heat. Fortunately, this is only 40 Watts and can be neglected when the heat loss from the store is high. Consequently, only the values obtained at high store temperature were used to evaluate the heat loss.

Time (s)	Store Temperature (^{o}C)	UA Value (WK^{-1})
0	77.68	--
6000	73.02	11.73
15900	68.40	9.22
29100	63.70	7.97
45600	58.93	7.21
66300	54.17	6.60
93600	49.39	6.02

Table I : Variation of the Store Temperature With Time

The equation governing the natural cooling of an isothermal body in a constant temperature environment when all the heat transfer parameters and thermal properties of the body are assumed constant with temperature is:

$$MCp \frac{dT}{dt} = UA(T-T_e) \qquad (3)$$

By integrating equation (3) between the time interval t=0 and t_i, during which the temperature of the body changes from T_{inti} to T_{fin}, the following expression of the UA value can be found:

$$UA = \frac{MCp}{t} \ln \frac{T_{fin}-T_o}{T_{inti}-T_o} \qquad (4)$$

The UA values obtained from the temperature decay of the thermal store using equation (4) are represented in Table I. It should be noticed that this UA value is not constant but tends to decrease with the store temperature.

In fact, as the temperature of the water in the store decreases, the temperature difference between the wall of the store and the environment also decreases. This reduces the convective and radiative components of the heat transfer coefficient from the wall of the store to the environment and as a result, the UA value decreases.

A simple correlation based on a second order least square interpolation of the results presented in Table I can be used to give the UA value for any store temperature between 20°C and 80°C. This correlation is:

$$UA = -2.5 + 0.11T + 0.001T^2 \qquad (5)$$

In equation (5), T is expressed in °C and UA is obtained in WK^{-1}.

2.11 Conclusions

The rig provides all the facilities to simulate on full scale the behaviour of Integrated Thermal Store when subject either to a space heating or to a hot water demand.

The instrumentation enables to accurately measure all the temperatures and flows relevant to a detailed analysis of the behaviour of the thermal store

The heat loss from the thermal store to the surroundings are small (around 10 WK^{-1} when the store is at 70°C) and decrease with temperature. The rate of heat loss from the store can be approximated reasonably accurately using simple experimental correlations.

Chapter 3

Heat Transfer Correlations for a Finned Heat Exchanger Coil Immersed in an Integrated Thermal Store

3.1 Introduction

At the moment, the effectiveness of heat recovery from a horizontal axis finned tube heat exchanger coil immersed in a hot water store is difficult to evaluate. The main difficulty is that the overall heat-transfer coefficient of the heat exchanger can not be estimated with a sufficient accuracy. This overall heat-transfer coefficient depends mainly on three factors:

- the internal heat-transfer coefficient. This internal heat transfer coefficient corresponds to a forced convection heat transfer process from water flowing in a coiled pipe

- the fin efficiency and the geometry of the heat exchanger

- the external heat-transfer coefficient. This external heat transfer coefficient corresponds to a natural convection heat transfer process for a finned tube immersed in hot water.

Some of these factors can be estimated with an accuracy sufficient to solve some practical engineering problems using existing heat transfer correlations. This is the case for the fin efficiency and the internal heat-transfer coefficient for a fully developed turbulent flow in a coiled pipe. Some of these correlations will be discussed later.

However, due to the complexity of the heat transfer processes involved, no attempts have been made to date to develop simple heat transfer correlations which would describe the transient natural convection heat-transfer process occurring on the outside of the fins of a horizontal axis finned tube heat-exchanger coil immersed in hot water.

Almost all the existing correlations available in the literature which estimate natural convection heat transfers have been obtained at steady state either with experiments using air as convective medium or with heat-mass transfer analogies to evaluate the convective heat-transfer coefficient.

These correlations have been developed for simple geometries of 'heat exchangers' such as vertical plates or horizontal tubes. None of these geometries could be assumed to be identical to a heat exchanger coil with an horizontal axis where the interaction between the coils, the and the inclination of the fins might play a major role in the heat transfer process.

Although these correlations are usually presented in a dimensionless form, as they are based on experimental data from different geometries of heat exchanger and different convective medium their accuracy can be expected to be relatively low when used for our type of heat exchanger coil. Consequently their usefulness is relatively limited.

As the overall heat-transfer coefficient of the heat exchanger might depend heavily on the external heat-transfer coefficient, it was of prime importance to develop a set of heat-transfer correlations to at least approximate this external heat-transfer coefficient for the geometry and mode of heat transfer relevant to our problem.

These correlation will be useful for both theoretical and practical considerations in helping at the same time to improve the understanding of the heat transfer mechanism at the wall of the heat exchanger and evaluate the performance of this type of heat exchanger.

3.2 Theoretical Analysis

Using a detailed theoretical analysis of the heat transfer process involved at the wall of the heat exchanger, it is possible to restrict beforehand the area to investigate in terms of number of parameters and range of variation for these parameters. This will result in significant simplifications in the the mathematical treatment of and the analysis of the experimental data.

The motion of a fluid will be described by a set of equations reflecting the conservation of mass, momentum and energy within the fluid. Additionally an equation of state will have to be included to fully describe the motion.

In the most general sense, when there is natural convection, the temperature gradient within the fluid will be moderate. Thus the thermal properties of the fluid can be assumed constant with temperature. This lead to the following set of equations in square coordinates:

Equation of Continuity:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \qquad (1)$$

Conservation of Momentum:

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = -gc\frac{\partial \rho}{\partial x} - \frac{gc\partial p}{\rho \partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \qquad (2)$$

$$\frac{\partial v}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = -gc\frac{\partial \rho}{\partial y} - \frac{gc\partial P}{\rho \partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) \qquad (3)$$

$$\frac{\partial w}{\partial t} + u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} = -gc\frac{\partial \rho}{\partial z} - \frac{gc\partial P}{\rho \partial y} + \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) \qquad (4)$$

Conservation of Energy:

$$\rho Cp\left(\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) + u\frac{\partial P}{\partial x} + v\frac{\partial P}{\partial y} + w\frac{\partial P}{\partial z} + \phi \qquad (5)$$

In the energy equation, ϕ is the dissipation function which will be neglected for a natural convection heat transfer.

In a natural convection process, the fluid flow is governed by the changes in density of the considered fluid with temperature. This can be approximated in the equations of motion by using the coefficient of thermal expansion ß which is linked to the pressure gradient in the z direction by:

$$dP = \rho g ß (T - T_0) \qquad (6)$$

From this stage, the formulation of the equations of motion has to be changed so that relevant dimensionless numbers appear. For this purpose, the variables x,y,z,t,u,v,w,T and p appearing in the equations of motion are replaced by dimensionless variable defined as follows:

dimensionless length in the x direction: $x=x^{*}L$

dimensionless length in the y direction: $y=y^{*}L$

dimensionless length in the z direction: $z=z^{*}L$

dimensionless velocity in the x direction: $u=V_{o}u^{*}$

dimensionless velocity in the y direction: $v=V_{o}v^{*}$

dimensionless velocity in the z direction: $w=V_{o}w^{*}$

dimensionless temperature: $T = T_{\infty}+\theta(Tw-T_{\infty})$

dimensionless time: $t = \frac{L}{V_{o}}t^{*}$

dimensionless pressure: $P = \frac{r\rho V^{2}P^{*}}{2}$

In the resulting dimensionless equations of motion, two dimensionless parameters appear. These are known as the Rayleigh number (Ra) and the Prandtl number (Pr). The are respectively defined by:

$$Ra = \frac{\rho g\beta(T-T_{0})L^{3}}{\mu\alpha} \qquad (7)$$

$$Pr = \frac{Cp\mu}{k} \qquad (8)$$

As in a natural convection heat transfer process, the flow is mainly buoyancy driven, the equation reflecting the conservation of momentum on the z direction is of particular relevance. The conservation of momentum in the other directions is of less importance and can be neglected for the theoretical considerations which follow.

When the conservation of momentum in the z direction is coupled with the continuity and energy equations, the following system results again in square coordinates:

Equation of Continuity

$$\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} + \frac{\partial w^*}{\partial z^*} = 0 \qquad (9)$$

Conservation of momentum in the vertical direction

$$\frac{Dw^*}{Dt^*} = -\frac{\partial P^*}{\partial z^*} + \Delta^2 w^* \left[\frac{Pr+Pr^2}{Ra}\right]^{\frac{1}{2}} + (1+Pr)\theta \qquad (10)$$

Conservation of energy

$$\frac{D\theta}{Dt^*} = \Delta^2\theta \left[\frac{1+Pr}{RaPr}\right]^{\frac{1}{2}} + \epsilon \qquad (11)$$

Where the term ϵ in the equation conservation of energy is a sum of terms, all of then can be neglected in natural convection problems. These terms correspond to:

- elevation of temperature created by friction

- creation of work with fluid expanding with altitude

- creation of work due to change in dynamic pressure

The dimensionless number particularly relevant to heat transfer processes is defined as the ratio of the rate of heat convected away from the wall of the heat exchanger by the rate of heat conducted away by the surrounding fluid. This is known as the Nusselt number. Using a dimensional analysis, it can easily be shown that it is defined as:

$$Nu = \frac{hL}{k} \qquad (12)$$

It now appears that the heat transfer process will be described by an equation relating the Nusselt number with the parameters appearing in the dimensionless equations of motion. As the only parameters in these equations are the Rayleigh and Prandtl numbers, the heat transfer process will be described by an equation like:

$$Nu = f(Ra,Pr) \quad (13)$$

Equation (13) can be simplified if a steady state heat transfer problem is being studied. In this case, all the partial derivatives with respect to time are nil and can be removed from the equations of motion. These terms can also be removed if the motion of the fluid changes only very slowly with time. In this case, the terms corresponding to time partial derivatives are extremely small and can be neglected in the equations of motion. This lead to the pseudo steady state approximation.

Equation (13) also can be further simplified by taking advantage of the range of Prandtl number involved. From the dimensionless equations of motion, it can be seen that when the Prandtl number is high then its relative contribution in equations (10) and (11) is greatly reduced. Therefore, only the Rayleigh number affects the dimensionless equations.

This approximation is justified and very often used for common heat transfer medium such as gases which have a Prandtl number of approximately 0.71 in normal temperature and pressure conditions. Therefore it is a fortiori valid for most of the liquids including liquid water at relatively low temperature, where the Prandtl Number is in the range 2-4. In this case, as the Prandtl number is high, the Nusselt number depends primarily on the Rayleigh number. This is reduces equation (13) to:

$$Nu = f(Ra) \quad (14)$$

It is now clear from equation (14) that the equation describing the heat transfer process will be restricted to the use of two dimensionless numbers. But, equation (14) is not sufficient to restrict the domain of investigation. Further simplifications are needed.

There are two separate methods in which further simplifications can be achieved. Each of these methods lead to relatively similar results with respect to simplifying the heat transfer equation even further.

The first method involves the use of the boundary layer approximation. The basis of this method is that as in natural convection, the fluid flow is primarily in the vertical direction, terms corresponding to motion in the horizontal direction can be neglected. This assumption is not always valid but might give a reasonable approximation for laminar flow. In this case, it can be shown that the analytical solution of equation (14) is in the form:

$$Nu = CRa^{0.25} \qquad (15)$$

Where C is a constant depending on the geometry of the heat transfer surface considered.

The second method is known as the conduction-layer approximate method. This method was introduced to give a better accuracy of heat transfer predictions than the boundary layer approximate method.

With the conduction layer approximate method, the flow is divided into two limit cases: laminar and turbulent heat transfer. In both case, the heat transfer surface of the body where natural convection occurs is divided into several elementary surfaces. For each of these elementary surfaces the heat flow can be evaluated using simple approximations usually using analogies with heat transfer from flat plates at various orientations. The total final shape of the equation (14) can be obtained by numerically or analytically integrating the local heat flow over the whole of the heat transfer surface. When this is done, the integrated form of equation (14) for <u>laminar</u> flow is found to be:

$$Nu = CRa^{0.25} \qquad (16)$$

Where again C is a constant depending on the geometry of the heat transfer surface area considered.

When the flow is turbulent, the solution of the equations of heat transfer cannot be found analytically using the conduction-layer approximate method. However, experimental measurements suggest that the equation between Nusselt and Rayleigh number is in the form:

$$Nu = CRa^{0.33} \tag{17}$$

Where again the coefficient C is a constant depending on the geometry of the heat transfer surface.

3.3 Existing Natural Convection Heat Transfer Correlations

The values of the constant C for both laminar and turbulent flows have been evaluated for many simple geometries. Saunders [1] and Warner[2] evaluated the heat transfer rate from hot vertical flat plates immersed in cold air over a wide range of Rayleigh number covering both the laminar and turbulent flows. The experimental results from both authors are represented in Fig 3.3(a). The corresponding heat transfer correlations for laminar and turbulent flow are respectively:

$$Nu = 0.57Ra^{0.25} \tag{18}$$

$$Nu = 0.103Ra^{0.33} \tag{19}$$

Experimental results in terms of Nusselt and Rayleigh numbers for a horizontal cylinder at constant temperature again immersed in cold air have been obtained by Hesse [3] and Morgan [4]. These results are represented in Fig.3.3(b) for the laminar and turbulent region of the heat transfer. The corresponding experimental correlations are respectively:

$$Nu = 0.44Ra^{0.25} \tag{20}$$

$$Nu = 0.109Ra^{0.33} \tag{21}$$

Other heat transfer correlations in terms of Nusselt and Rayleigh numbers for other simple geometries of heat exchangers are also available. They include heat transfer from horizontal plates, inclined cylinders, spheres, various types of fin geometries and miscellaneous geometries such as noncircular cylinders.

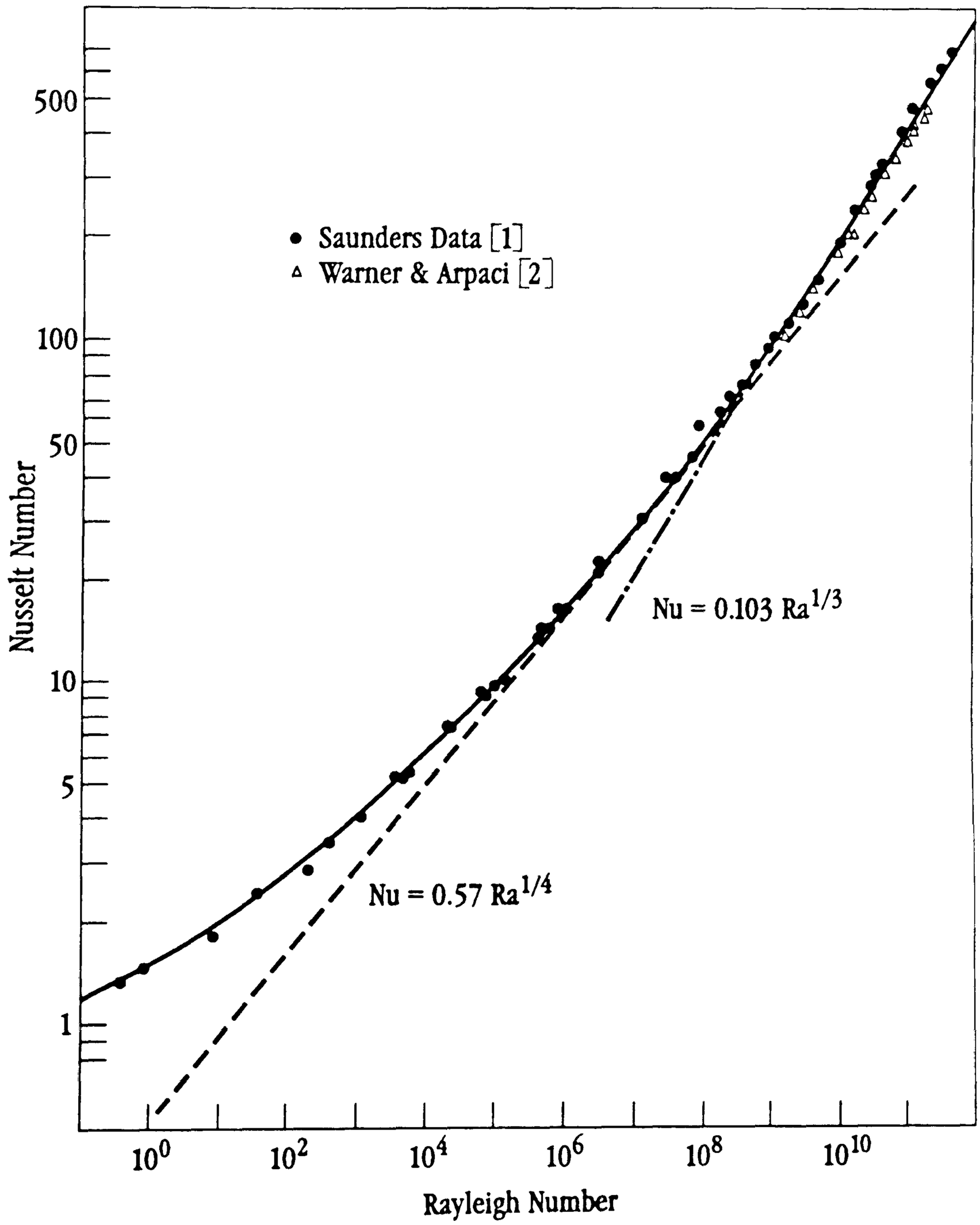


Fig. 3.3 (a). Natural Convection Heat Transfer From a Vertical Plate

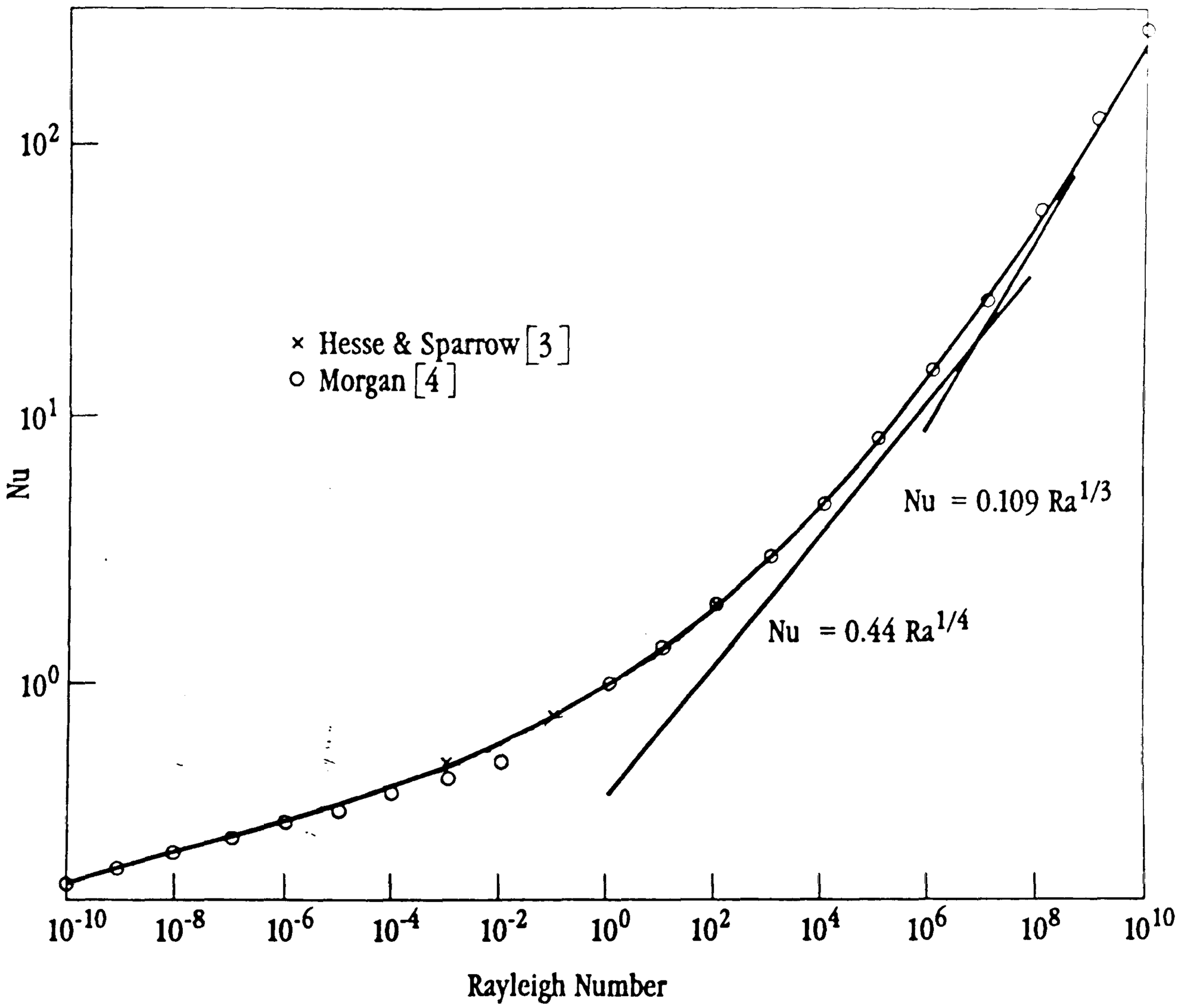


Fig. 3.3 (b). Natural Convection Heat Transfer From a Horizontal Cylinder

3.4 Experimental Apparatus

This is described in detail in Chapter 2 and consisted mainly of a 200 litre cylindrical hot water store fitted with an internal finned tube heat exchanger coil. The hot water store was 1.4 m in height, 0.44 m in diameter and weight 52 kg. It had a capacity of approximately 200 litres. The internal heat exchanger is made of 24 metres of finned copper tubing. This tubing was rolled in four coils. Two of these coils were located at the top of the store while 2 other coils were located in the middle of the store.

A chilled water facility was used to keep the temperature of the water entering the heat exchanger constant at approximately 8°C. This chilled water facility could supply approximately 250 litres of chilled water.

3.5 Methodology

At the beginning of the experiments, the 200 litre store was preheated to 80°C by the use of the heating element. A mixing pump was run continuously during the heating period so that the temperature of the water in the store was uniform at the beginning of the experiments. During the heating period, the chilled water supply was left permanently on so that approximately 250 litres of cold water were available at the beginning of the experiment.

At the end of the heating period, cold water at 8°C from the chilled water supply was allowed to pass into the heat exchanger coil thereby extracting heat from the thermal store. The flow rate of water into the heat exchanger coil was kept constant at around 9.5 litre/min. The flow rate of water passing into the heat exchanger coil was constantly measured during the thermal discharge using an accurate high pulsing rate turbine type flow meter.

Two high accuracy PRTs located at the inlet and the outlet of the heat exchanger respectively were used to measure the temperature of the water entering and leaving the heat exchanger. 59 k-type thermocouples located within the 200 litre store were used to measure the temperature of the water inside the store during the thermal discharge. These allowed to evaluate the thermal properties of the water in the tank accurately and gave some indications of the stratification in the tank and the temperature and velocity fields.

Repeatability of the Experiments

The accuracy of the experimental measurements can be obtained form the detailed description of the experimental apparatus. However, the repeatability of the tests will also depend on the methodology used to carry out the experiments. Although the best methodology is always sought, some preliminary experiments were carried out to try to estimate the whether the repeatability was sufficient.

7 thermal draw-off were carried out using exactly the same methodology to evaluate the repeatability of the experiments. The test consisted in heating up the thermal store up to 80°C and to carry out one 150 litre thermal draw-off. The following parameters were recorded:

- initial tank temperature, (Tt)

- initial chilled water temperature, (Tc)

- average flow rate in the heat exchanger, (m)

- final tank temperature, (Tft)

- final chilled water temperature, (Tfc)

- total cumulative amount of heat extracted from the store as given by:

$$Q = \int_0^t mCp(T_{out} - T_{in})\,dt$$

The results of this investigation are presented in Table II. they show that the repeatability is acceptable as there is very little variation in the experimental parameters and measurements from one experiment to the other.For example, the temperature variations of the chilled water supply and of the store temperature are usually less than 0.5°C and the total cumulative amount of heat extracted from the store varies only by 0.5% between the experiments.

This suggests that the combination of the high accuracy instrumentation used in these experiments and the experimental methodology give a high repeatability and little improvement in the experimental results can be obtained by trying to 'average' the results of several experiments.

Experiment number	Tt (°C)	Tc (°C)	Tft (°C)	Tfc (°C)	m (l/min)	Q (MJ)
1	79.56	8.54	43.82	9.12	9.23	29.26
2	79.82	8.17	44.41	8.89	9.41	29.13
3	79.09	8.68	43.56	9.31	9.82	29.18
4	79.24	8.38	43.67	9.54	9.53	29.27
5	79.62	8.46	43.95	9.36	9.30	29.20
6	79.38	8.31	43.76	9.28	9.63	29.34
7	79.54	8.59	43.89	9.06	9.67	29.15

Table II : Results from the Repeatability Test

3.6 Description of the Heat Transfer Mechanism

At the beginning of the experiment, cold water from the main supply enters the lower heat exchanger coil first. At this coil, the temperature difference between the water in the tank and the water in the coil is the greatest. On the outside surface of the heat exchanger coil, the temperature of the water decreases rapidly which creates strong convection currents in the thermal store. These convection currents make the heat transfer highly efficient. Consequently, the water leaving the lower heat exchanger coil and entering the upper heat exchanger coil is at a relatively high temperature. This high temperature is such that the temperature difference between the water flowing into the upper heat exchanger coil and the water in the upper part of the store is relatively small. This makes the rate of heat transfer at the upper coil relatively low when compared with the heat transfer at the lower coil.

When time increases, as a lot more heat is extracted from the coil at the bottom of the store, the temperature of the water in the bottom of the store decreases rapidly. This water of a higher density tends to remain at the bottom of the store. This creates a significant temperature gradient between the water in the upper region of the store and the water in the bottom region of the store.

As the temperature of the water at the bottom of the store decreases further, the heat transfer rate at the bottom coil decreases. Thus the water entering the upper heat exchanger coil progressively lowers. This colder water increases the temperature difference between the temperature of the water in the upper coil and the temperature of the water in the upper part of the store which progressively increases the heat transfer from the upper heat exchanger coil.

The variation of the temperature of the water in the store with time during the thermal discharge is presented in Fig.3.6. This temperature profile clearly shows the advantage of having a coil at the bottom of the store to create a large temperature difference (or a high degree of stratification) between the top and the bottom of the store.

This temperature profile also divides the thermal store into two temperature zones. The cold zone located at the bottom of the store and the hot zone located at the top of the store. The largest temperature gradient is situated at approximately 500 mm above the bottom of the store.

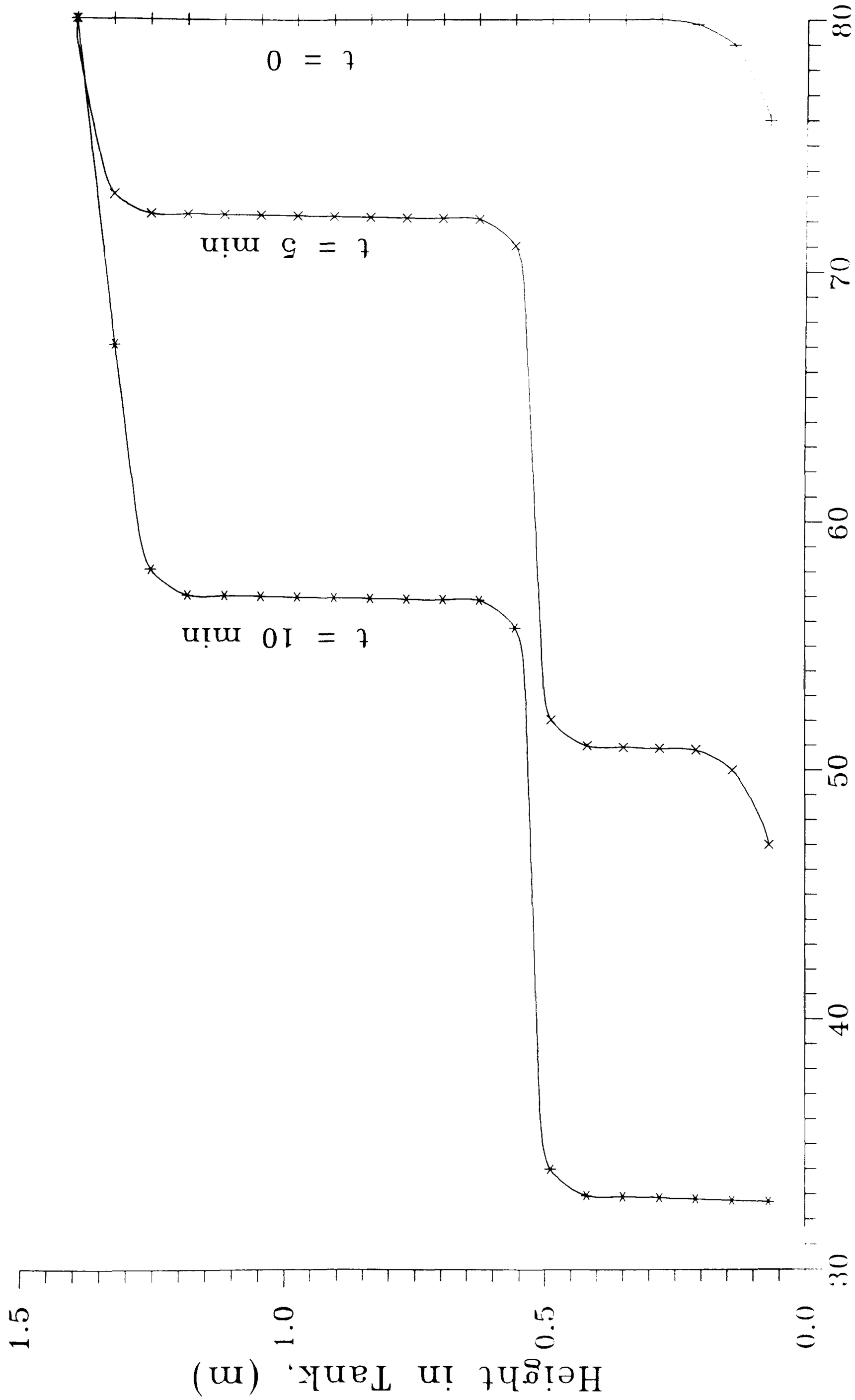


Fig.3.6: Temperature Gradient in the Thermal Store During the Thermal Discharge

If the thermal discharge were pursued for a very long period of time, the temperature of the water in the bottom region of the store would become so low that the bottom heat exchanger would become very nearly ineffective at extracting heat. In this case, most of the heat would be extracted from the top of the store and the temperature of the water in the upper part would fall significantly.

If the thermal discharge were pursued even further, the temperature of the water in the top and bottom regions of the store would become very nearly equal to the temperature of the chilled water entering the heat exchanger coil and the heat transfer process would become very low. In practice, this stage is never be reached during normal operation.

3.7 Data Analysis

The evaluation of a heat transfer correlation between the Nusselt and Rayleigh numbers can not be found by simple methods of calculation. The main reason is that during the thermal discharge, the temperature gradient within the thermal store (stratification) makes the use of any analytical solution impossible. In addition, the complexity of the heat transfer process makes the development of these correlations almost impossible without using some preliminary approximations to simplify the data analysis.

The first approximation comes from the mode of operation of the thermal store. During the normal operation of the heat exchanger in the thermal store, steady state is never reached and transient heat transfer correlations should in theory be developed. However, the notion of transient behaviour in heat transfer processes is only relevant when the time constant of the heat transfer process is the order of a few seconds or less. As the time constant associated with the thermal discharge achieved by means of the heat exchanger coil is in very high (the order 1,000 seconds or more), the transient effect can be neglected.

Although some transient effect might take place for example in the very few seconds of the thermal discharge, this transient effect will quickly disappear. and will be completely negligible after a few minutes. Consequently, the problem can be treated as steady state and heat transfer correlations based on the steady equations of motion can be used as relatively good approximation of the transient heat transfer process.

The second approximation is that the heat transfer by convection is the dominant mode of heat transfer. In theory, the heat transfer from the coil to the surrounding water and from the water flowing in the coil to the wall of the pipe occurs by three processes: conduction, radiation and convection. However, as these heat transfer takes place within liquid water, the convection heat transfer coefficient is always at least two orders of magnitude higher than the sum of the conduction and radiation heat transfer. Therefore, only the convection heat transfer will be considered.

The temperature gradient presented in Fig.3.6, clearly shows two temperature zones in the store during the thermal discharge. This suggests that the store behaves very nearly as to small stores each of which being isothermal.

The UA value of the heat exchanger can therefore be approximated using an analogy with two small independent stores, each of which having an internal heat exchanger as represented in Fig.3.7. Each thermal store is assumed isothermal and each heat exchanger has its own UA value.

Cold water at temperature T_{in} enters the first heat exchanger situated in the first store where it extracts heat. During this stage the temperature of the water flowing into the heat exchanger increases from T_{in} to T_i.

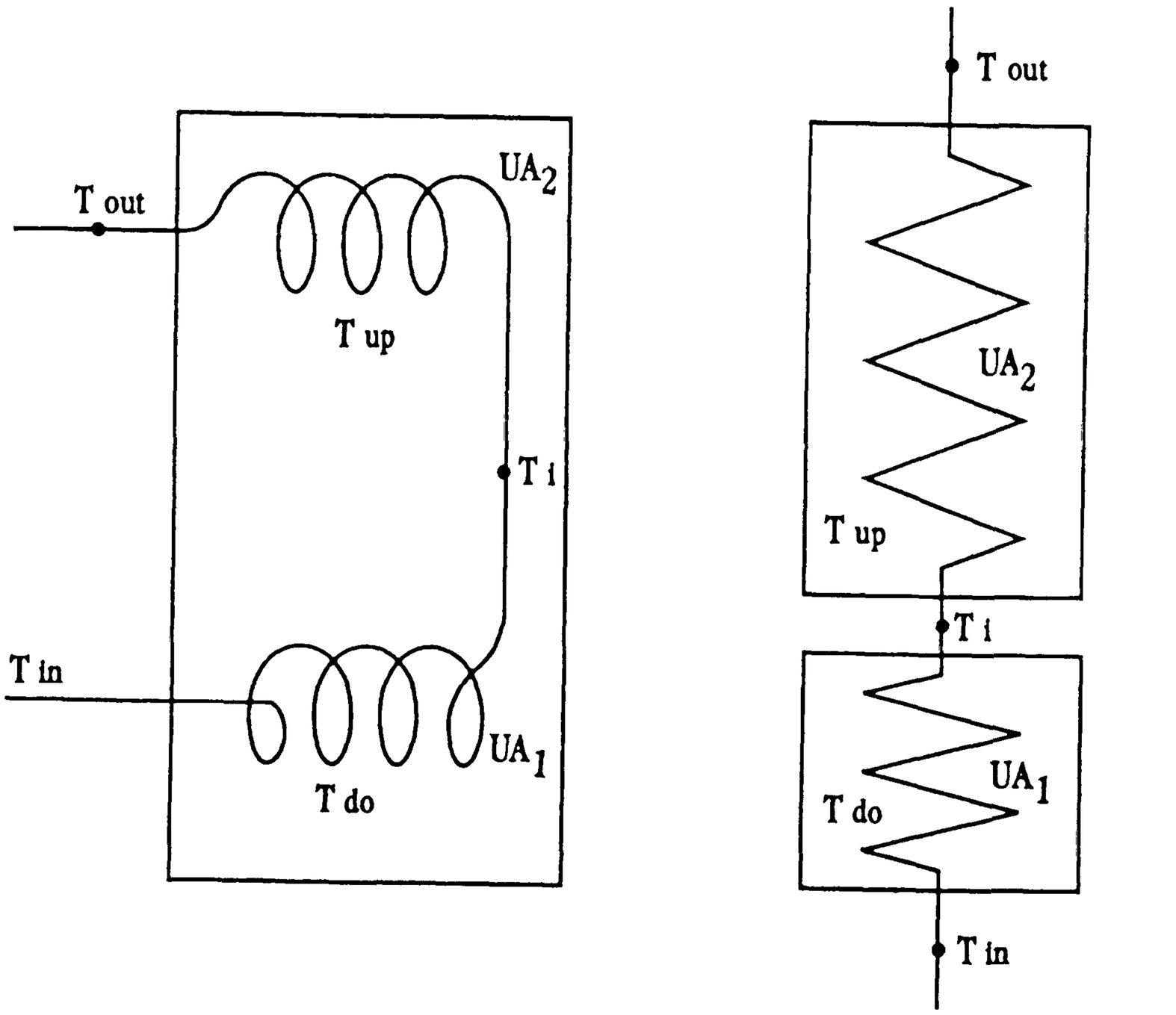
The water at temperature T_i from the first heat exchanger then passes into the second heat exchanger where again it extracts heat from the second store. During this stage, the temperature of the water in the heat exchanger increases from T_i to T_{out}.

The temperatures T_i and T_{out} can be evaluated using simple heat transfer correlations once the parameters governing the heat transfer mechanism are known. Using the assumption that the thermal properties of the fluid are constant, this resulted in equations (21) and (22):

$$T_i = T_{do} + (T_{in} - T_{do}) \exp(-UA_1/mCp) \qquad (22)$$

$$T_{out} = T_{up} + (T_i - T_{up}) \exp(-UA_2/mCp) \qquad (23)$$

Where UA_1 = UA value of the heat exchanger in the bottom zone
UA_2 = UA value of the heat exchanger in the top zone



Integrated Thermal Store Independent Stores

Fig.3.7: Analogy Between an Integrated Thermal Store and two Independent Stores

Unfortunately, the system of equations (21),(22) cannot be solved at this stage as there are three unknowns UA_1, UA_2 and T_i and only two equations (21) and (22). An additional equation is required to solve the system.

This equation will be obtained by a detailed analysis of the heat transfer process taking place at the wall of the heat exchanger coil where the rate of heat transfer across the wall can be calculated once the temperatures of the water in the store and in the pipe are known.

The fin efficiency can be estimated from the internal and external heat-transfer coefficient of the water in and surrounding the pipes, the geometry of the fins and the thermal conductivity of the fin. The methodology is to carry out a heat balance between the heat conducted along the fin and the heat convected away by the surrounding fluid. A convenient correlation based on this heat balance on the fin has been developed by T.Schmit [5]. This correlation is in good agreement with the one developed by Kern [6] although the latter is relatively old. Schmit evaluated the fin efficiency (Eff) with:

$$Eff = \frac{\tanh(ml)}{ml} \qquad (24)$$

where m is a parameters given by:

$$m = \left[\frac{h}{kp} \right]^{0.5} \qquad (25)$$

where h = heat-transfer coefficient on the outside of the fin
k = thermal conductivity of the fin
l = equivalent fin length
p = perimeter of the fin

The external heat transfer coefficient is not known at this stage. However the use of heat transfer correlations for horizontal cylinder in hot water suggests that it ranges from approximately 400 $Wm^{-2}K^{-1}$ to 800 $Wm^{-2}K^{-1}$. Assuming an average value of 600 $Wm^{-2}K^{-1}$, gives ml=0.32. The corresponding fin efficiency is 0.97.

With respect to this result, the assumption of a fin efficiency of unity was used. This assumption slightly reduces the complexity of the system of equations to solve and therefore the data analysis.

Several correlation have been developed for predicting the internal heat transfer coefficient of a fully developed turbulent flow in an helically coiled tube. These correlations relate the Nusselt number to the Dean number (K) which is defined by:

$$K = Re\left[\frac{r_w}{r_c}\right]^{0.5} \qquad (26)$$

where r_w is the internal pipe diameter
r_c is the coil diameter

However, there are several problems in using these correlations. The main problem comes form the small amount of experimental data available. This small amount of data makes it impossible to correlate accurately the effect of the Prandtl number and of the curvature of the coil. Therefore the correlations developed upon this data are relatively inaccurate, particularly at low Dean numbers.

Fortunately, for our heat exchanger coil, the Dean number is relatively high (the order of 1000 or more). In this case, all the correlations tend to converge towards the heat transfer for a fully developed turbulent flow in a straight pipe only slightly modified to take into account the curvature of the coil.

Three type of correlations were compared for this type of heat transfer mechanism. These were developed respectively by Rogers [7], Seban [8] and McAdams [9]. The correlation by Rogers and by Seban are based on experiments carried out for a coiled tube immersed in condensing steam so that the external heat transfer can be assumed infinite. With these assumption, both author developed their own correlation. These correlations are in good agreement with each other and are respectively:

$$Nu = 0.023Re^{0.85}Pr^{0.4}\left[\frac{r_w}{r_c}\right]^{0.1} \qquad (27)$$

$$Nu = 0.021Re^{0.85}Pr^{0.4}\left[\frac{r_w}{r_c}\right]^{0.1} \qquad (28)$$

The heat transfer correlation developed by McAdams gives the heat transfer coefficient into a helically coiled tube from the heat transfer coefficient of a straight tube modified to take into account the curvature of the coil. This correlation is:

$$h_c = (1+3.5\frac{r_w}{r_c})h_s \qquad (29)$$

Where h_c is the internal heat transfer coefficient for the considered coil r_w is the internal diameter of the tube, r_c is the diameter of the coil and h_s is the heat transfer coefficient for a straight pipe operating in the same conditions.

The more recent work carried out by Rogers and by Seban and using independent sets of data, strongly suggests that the coefficient appearing before the ratio of r_w/r_c should be between 5 and 6 and not 3.5 as reported by McAdams. Rogers claims that the error lies in the data analysis carried out by McAdams. To take into account this recent work, equation 29 was modified to:

$$h_c = (1 + 5\frac{r_w}{r_c})h_s \qquad (30)$$

The internal heat transfer coefficient h_s for a turbulent flow inside a straight pipe can be approximated using the formula by Sleicher and Rouse [10]:

$$Nu = 5+0.015Re^aPr^b \qquad \begin{matrix} 0.1 < Pr < 10^4 \\ 10^4 < Re < 10^6 \end{matrix} \qquad (31)$$

The powers a and b can be calculated respectively from:

$$a = 0.88 - \frac{0.24}{(4+Pr)}$$

$$b = 0.333 + 0.5\exp(-0.6Pr)$$

Other formula better known such as the Colburn formula or the Seider and Tate formula, although relatively simpler to use in practice, are found to be less accurate in the operating conditions considered in this heat exchanger coil and currently give value for the internal heat transfer coefficient 15 to 20% lower than the Sleicher-Rouse equation. Such an error would have changed the final results noticeably and therefore was not acceptable.

As we will see later, the limitation to heat transfer comes mainly from the fin side where natural convection limits greatly the rate of heat transfer. As a result, the internal heat-transfer coefficient does not have to be approximated with a great accuracy.

Finally, by combining equations (30) and (31), it is possible to approximate the internal heat transfer coefficient of the water flowing into the heat exchanger coil with an accuracy satisfactory to our problem.

As the heat transfer mechanism occurring at the top and at the bottom heat exchanger coil in the store are similar, it is reasonable to assume that **the same heat-transfer correlation can be used to predict the external heat-transfer coefficient for both coils**.

As explained earlier, this correlation relates the Nusselt and Rayleigh numbers only so that the heat transfer coefficient h could be calculated from the thermal properties of the water and the bulk and the wall temperatures.

The Rayleigh number for a horizontal finned tube can be obtained by analogy with the heat transfer occurring between vertical parallel plates. In the latter case,the characteristic length for the development of heat transfer correlations is the plate spacing as it is between the plates that the thermal and velocity boundary layer will develop. Thus the fin spacing should be used in the calculation of the Rayleigh for a horizontal finned tube with vertical fins.

Additionally, it should be clear that in the case of high fin tube, the heat transfer will be very similar to the heat transfer occurring between parallel plates. Inversely, in the case of low finned tube, the heat transfer process will be very similar to the heat transfer occurring on the surface of unfinned tube.

In the case of circular fins, the difference between high and low finned tube can be obtained by calculating the ratio of the fin diameter by the tube diameter. Depending on the value of this ratio, the value of the Rayleigh number has to be modified either by the ratio of the fin spacing by the fin diameter (for high finned tube) or by the ratio of the fin spacing by the tube diameter (for low finned tube).

The finned tube used in our type of heat exchanger should be regarded as low fined tube and finally, the corresponding definition of the Rayleigh number is:

$$Ra = \frac{\rho g \beta (T_b - T_w) s^4}{\alpha \mu d} \qquad (32)$$

The corresponding Nusselt number should also be calculated using the fin spacing as a characteristic length. This gives:

$$Nu = \frac{hs}{k} \qquad (33)$$

To remain coherent with existing heat transfer the correlations for natural convection described earlier, the heat transfer correlation between the Nusselt and Rayleigh number was sought in the form of a power law:

$$Nu = CRa^b \qquad (34)$$

Where C and b are constants

With the additional equations (24),(25) and (30)-(34), it is possible to calculate the values of the overall heat transfer coefficient UA_1 and UA_2 in equations (22) and (23) when the coefficient C and the power b appearing in equation (34) are known.

Therefore it is equivalent to solve the problem in terms of C and b or in terms of UA_1 and UA_2. Although the number of equations and unknowns is still the same (2 equations and 3 unknowns) it is easier to analyse the problem in terms of C and b. In addition, an extra equation will be obtained from the experimental data which will make it possible to solve the system in terms of C and b.

3.8 The Method of Solution

Many correlations are already presented in the literature about the coefficient C and the power b [11]-[17]. C is usually in the range 0.1 to 0.7 while b varies from 0.25 to 0.33. The respective values of these coefficients depend on the geometry of the heat transfer surface and the nature of the heat transfer (laminar or turbulent).

Edwards and Chaddock [11] reported values for the power b as high as 0.6 for circular fins on horizontal cylindrical tubes. However such a high value for b is restricted to an extremely low range of Rayleigh numbers which is unlikely to occur in the heat transfer process we investigate.

The combination of these wide ranging values for b and C make the use of any of the existing correlation too approximate to be useful. It was therefore necessary to evaluate the values of C and b from experimental data.

The system of equations to solve is highly non linear. The non linearity comes from several sources. Firstly, the thermal properties of water such as the viscosity appearing in the Rayleigh number formulation depend on the temperature of the water in the store. Secondly, the heat transfer correlation which links the coefficient C and the power b is in the form of a power law which by definition is a non linear equation.

Due to the non linearity of the system of equations to solve simple methods of solution, such as a Gauss method, could not be used to solve the system. Therefore an iterative method had to be developed. Although several iterative methods can be used, the one which is presented below has the main advantage of a relatively quick convergence rate. This quick convergence made it possible to approximate the values of the parameters C and b within 99% of their final value in usually less than 4 iterations. This iterative method is as follows:

The first step in the iterative process was to set the power b appearing in equation (34) to a predetermined value so that an extra equation was obtained to solve the system. In addition, setting the value of b to this predetermined value eliminated the main non linear equation in the system and consequently made the resolution much easier.

Equations (22) and (23) were then used to evaluate the UA value of the heat exchanger in the lower and upper zones of the store respectively so that the predicted outlet temperature T_{out} was equal to the temperature experimentally measured at the outlet of the heat exchanger coil. For the first iteration, a simple assumption was used between the values of UA_1 and UA_2. It was in the form $UA_1/UA_2=d$ where d was a constant. In the absence of any data it was assumed that d=1. For other iterations, it was possible from the estimate of the external and internal heat transfer coefficients to evaluate UA_1 and UA_2 separately which gave the value of the constant d.

Once the value of UA_1 and UA_2 were estimated, the temperature of the water flowing inside the heat exchanger between the lower and upper coil T_i was obtained from equation (22).

The thermal properties for equation (31), which gives the internal heat transfer coefficient, were calculated at the bulk temperature for the water flowing in each pipe in each zone. The bulk temperatures were evaluated from:

$$T_{b1} = \frac{T_{in} + T_i}{2} \qquad (35)$$

$$T_{b2} = \frac{T_i + T_{out}}{2} \qquad (36)$$

The value of T_i was also used to calculate the heat transfer rate in the lower and upper zone respectively q_1 and q_2 which are given by:

$$q_1 = mCp(T_i - T_{in}) \qquad (37)$$

$$q_2 = mCp(T_{out} - T_i) \qquad (38)$$

The temperature of the wall of the heat exchanger in each zone was calculated using:

$$T_{w1} = T_{do} - \frac{q_1}{h_1 A_1} \qquad (39)$$

$$T_{w2} = T_{up} - \frac{q_2}{h_2 A_2} \qquad (40)$$

The thermal properties of water for the calculation of the Rayleigh number were evaluated at the film temperature for each zone. The film temperatures were evaluated from:

$$T_{f1} = \frac{(T_{w1}+T_{do})}{2} \qquad (41)$$

$$T_{f2} = \frac{(T_{w2}+T_{up})}{2} \qquad (42)$$

The thermal properties used in the calculation of the Nusselt number were also evaluated at film temperature (A thermodynamic table for the properties of water is presented in appendix 2).

The estimated internal and external heat transfer coefficients were then used to re-evaluate the respective values of UA_1 and UA_2 and the ratio UA_1/UA_2 whose value was used to initiate a second iteration. The iterative process was stopped when the ratio UA_1/UA_2 did not change from one iteration to the next. At this stage, the parameter C was evaluated.

The parameter b was varied from 0.25 to 0.33 by increments of 0.01. The parameter C was then calculated for each value of the parameter b from the experimental data. The values of 0.25 and 0.33 where chosen as they are the respective limits for laminar and turbulent flows.

Due to the experimental background of this work, a sufficient number of values of the coefficient C had to be calculated from the experimental data and statistical methods used to summarize the results.

The average value of the coefficient C for the corresponding values of the power b is represented in Table III. Also shown in Table III are: the dispersion of the experimental results around the corresponding correlation and the maximum experimental errors measured above and below the correlation.

The dispersion of the experimental results is defined by:

$$D = \frac{1}{n} \Sigma \frac{|Nu-Nu_c|}{Nu_c} \qquad (43)$$

Where: Nu is the experimentally measured value of the Nusselt number

Nu_c is the value of the Nusselt number given by the correlation line for the same value of Rayleigh number as the considered experimental point.

n is the number of experimental points.

The maximum experimental error measured above and below the correlation are defined respectively by:

$$E_{max} = Max \left[\frac{Nu-Nu_c}{Nu_c} \right] \qquad (44)$$

$$E_{min} = Max \left[\frac{Nu_c-Nu}{Nu} \right] \qquad (45)$$

Now it is important to note that <u>the best heat transfer correlation is obtained when the dispersion of the experimental data is a minimum</u>.

From the results presented in Table III, a value of b between 0.29 and 0.30 gives the minimum dispersion <u>and</u> the minimum experimental errors. This suggests that the best heat transfer correlation is between these values.

Fig.3.8(a) represents the dispersion of the experimental results with the value of the power b in equation (34). Also represented on the same graph is a second order least square polynomial interpolation of the experimental results. Using this interpolation, it is possible to evaluate the optimal value for b at 0.2929. The corresponding value of C was then computed from the experimental data at and found to be 0.28 exactly. The corresponding dispersion was 2.157 %. Thus the final result:

$$Nu = 0.28Ra^{0.2929} \qquad \text{for } 100 < Ra < 1500 \qquad (46)$$

b	C	Dispersion (%)	E_{max} (%)	E_{min} (%)
0.25	0.3668	3.132	7.306	6.515
0.26	0.3448	2.608	6.526	5.916
0.27	0.3232	2.481	6.064	5.012
0.28	0.3033	2.273	5.440	4.253
0.29	0.2847	2.165	4.847	4.145
0.30	0.2673	2.183	5.612	4.826
0.31	0.2509	2.309	5.580	5.500
0.32	0.2355	2.553	5.605	6.115
0.33	0.2209	2.854	5.614	6.745

Table III : Values of the Coefficient C for Corresponding Values of the Power b

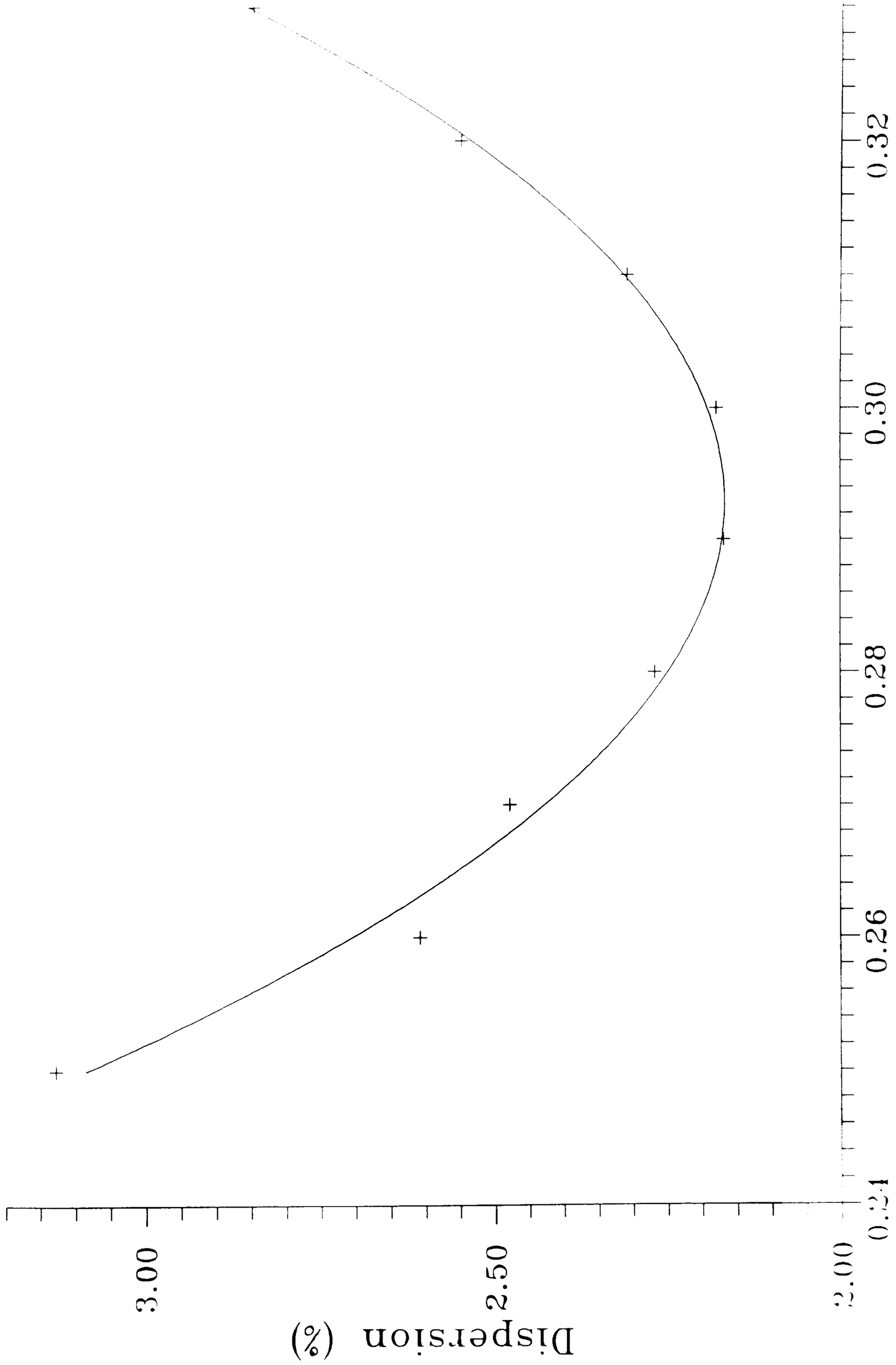


Fig.3.8(a): Dispersion of the Experimental Results
for Various Values of the Power b

Some of the experimental results with the correlation curve corresponding to the values of b=0.2929 and C=0.28 are represented in Fig.3.8(b). A close examination of these experimental results suggests several commentaries:

1) The same correlation curve fits closely the data for the top and the bottom heat exchangers. Which 'a posteriori' justifies the assumption made at the beginning of the calculation that the same heat transfer equation could be used for both the top and bottom heat exchanger coils.

This also suggests that the same heat transfer mechanism occurs at the top and at the bottom coil. Consequently, an investigation aiming at improving the design of the heat exchanger can be first limited to one coil and then the results extrapolated to the second coil.

2) The data presented in Table III shows that the dispersion around the average value is small for values of the power b between 0.28 and 0.3. Thus, although the optimal value of b is 0.2929, in fact, any value of b between 0.28 and 0.30 could have correlated the data relatively well. The only significant increase in dispersion is obtained when b<0.28 or b>0.3.

In many engineering applications, an extremely high accuracy of heat transfer predictions is often not needed. In this case, the optimal correlation can be approximated, with a loss of accuracy but still relatively closely by, simplified formula such as:

$$Nu = 0.3Ra^{0.28} \qquad \text{for } 100 < Ra < 1500 \qquad (47)$$

or

$$Nu = 0.28Ra^{0.29} \qquad \text{for } 100 < Ra < 1500 \qquad (48)$$

3) The optimal value of the coefficient b being 0.2929 suggests that the heat transfer mechanism would most probably not be closely approached by a heat transfer model based either on an entirely laminar or on an entirely turbulent flows. In this case, a new type of model would have to be developed based on other assumptions. For example, the Navier-Stockes equations often used to model laminar flow, would not accurately predict the characteristics of this type of heat transfer mechanism.

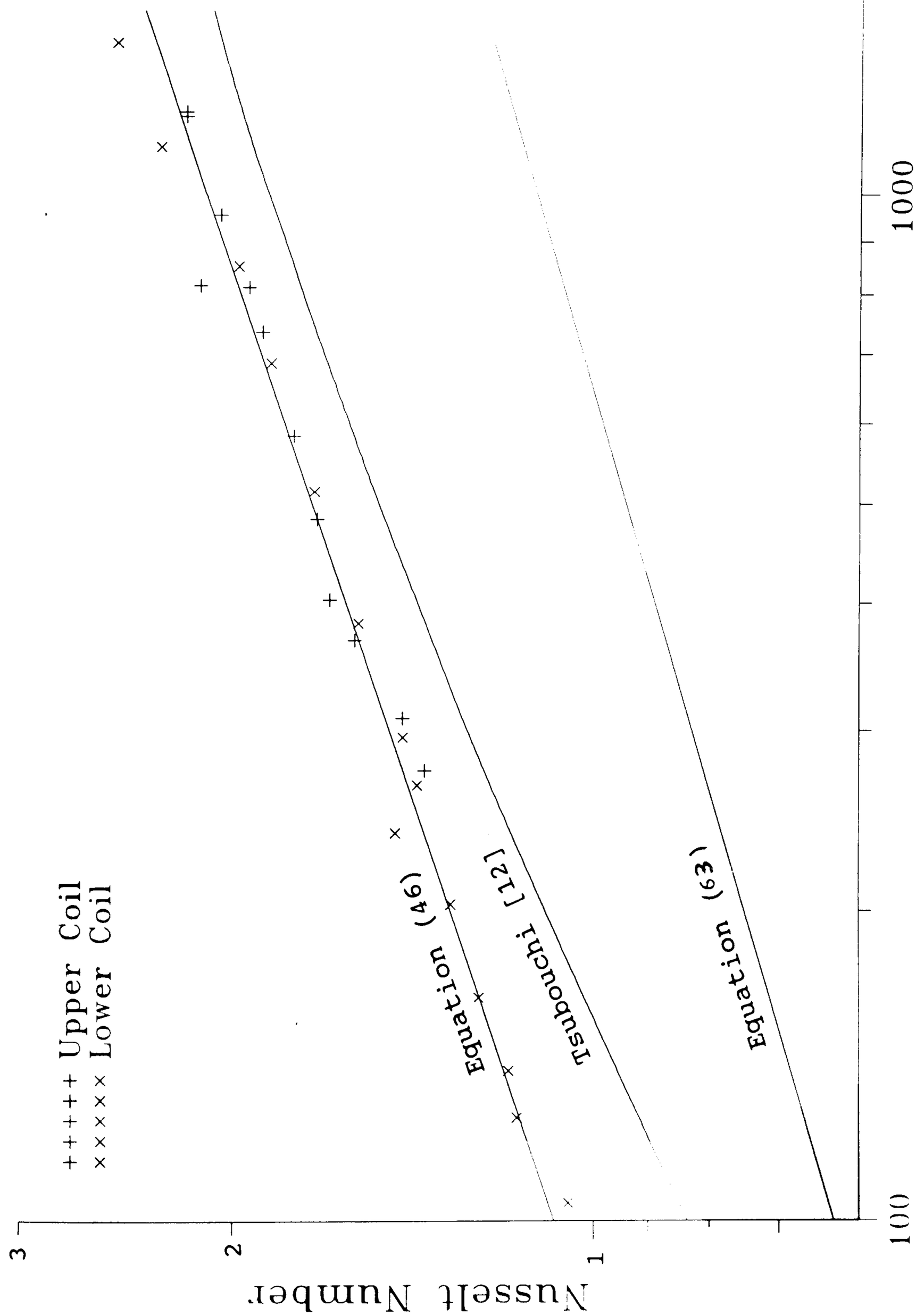


Fig.3.8(b): Best Heat Transfer Correlation and Corresponding Experimental Results for Natural Convection Heat Transfer for the Considered Heat Exchanger coil

4) Tsubouchi and Masuda [12] extensively investigated the heat transfer from horizontal tubes with circular fins. They developed heat transfer correlations based on a wide range of experimental data for natural convection heat transfer in air.

They developed two types of heat transfer correlation depending on the geometry of the finned tube considered (low finned tube or high finned tube). The critical geometric factor they use is the ratio R of the tube diameter by the fin diameter. The limiting case between the two correlations is obtained for a value of this ratio of 0.6.

In addition, they separated the heat transfer area in two regions: the fin tip and the rest of the heat transfer surface area. For the fin tip region, the heat transfer correlation for a low fin tube in natural convection in air is approximately given by:

$$Nu = 0.6Ra^{0.28} \tag{49}$$

For the rest of the heat transfer area, a more complex equation has been developed. This equation is :

$$Nu = C_o Ra^{p}\left[1-\exp\left(-\frac{C_1}{Ra}\right)^{C_2}\right]^{C_3} \tag{50}$$

The constants C_o to C_3 and the power p are given by:

$$C_o = 0.3R - 0.15 \tag{51}$$

$$C_1 = 480R - 180 - 1.4R^{-8} \tag{52}$$

$$C_2 = 0.04 + 0.9R \tag{53}$$

$$C_3 = 1.3 - 1.3R + 0.0017R^{-12} \tag{54}$$

$$p = 0.25 + C_2C_3 \tag{55}$$

The limiting cases obtained for R=0.6 and R=1 for the correlation developed by Tsubouchi and Masuda are represented in Fig.3.8(c). Also presented in Fig.3.8(c) is the straight line obtained as the best correlation of our experimental data (equation (46)). In the range of Rayleigh numbers considered, the predictions obtained with both correlations give relatively coherent results.

Using the value of R=0.658 ,the exact correlation from Tsubouchi and Masuda for the heat transfer area excluding the tip is:

$$Nu = 0.047Ra^{0.694}\left[1-\exp\left(-\frac{96}{Ra}\right)^{0.631}\right]^{0.703} \qquad (56)$$

The exact heat transfer correlation corresponding to equations (49) and (56) for the finned tube used in the heat exchanger coil is represented in Fig 3.8(b). It matches relatively closely the experimental data. However, there are several differences between the correlation which was developed in this analysis and the correlation developed by Tsubouchi and Masuda.

(i) The heat transfer correlation by Tsubouchi was originately developed using an analogy with natural convection heat transfer between vertical parallel plates. Using this analogy, several parameters (coefficients C_0 to C_3 and the power p) can be introduced in the correlation. These parameters allow to match the experimental results and the heat transfer correlation better than equation (46) which only has one parameter.

(ii) The correlation developed by Tsubouchi and Masuda shows a decrease in slope in the Nu-Ra diagram for low Rayleigh numbers. This is caused by interference in the boundary layers which developed between adjacent fins and will be discussed later.

(iii) The correlation developed by Tsubouchi and Masuda gives predictions of Nusselt numbers which are on average approximately 10% lower than equation (46) over the whole range of the Rayleigh number from 100 to 1500. This has important consequences as it shows that the heat transfer coefficient of our finned tube heat exchanger is higher than the heat transfer coefficient which would be achieved for the same tube geometry by a horizontal tube immersed in an infinite medium of cold water.

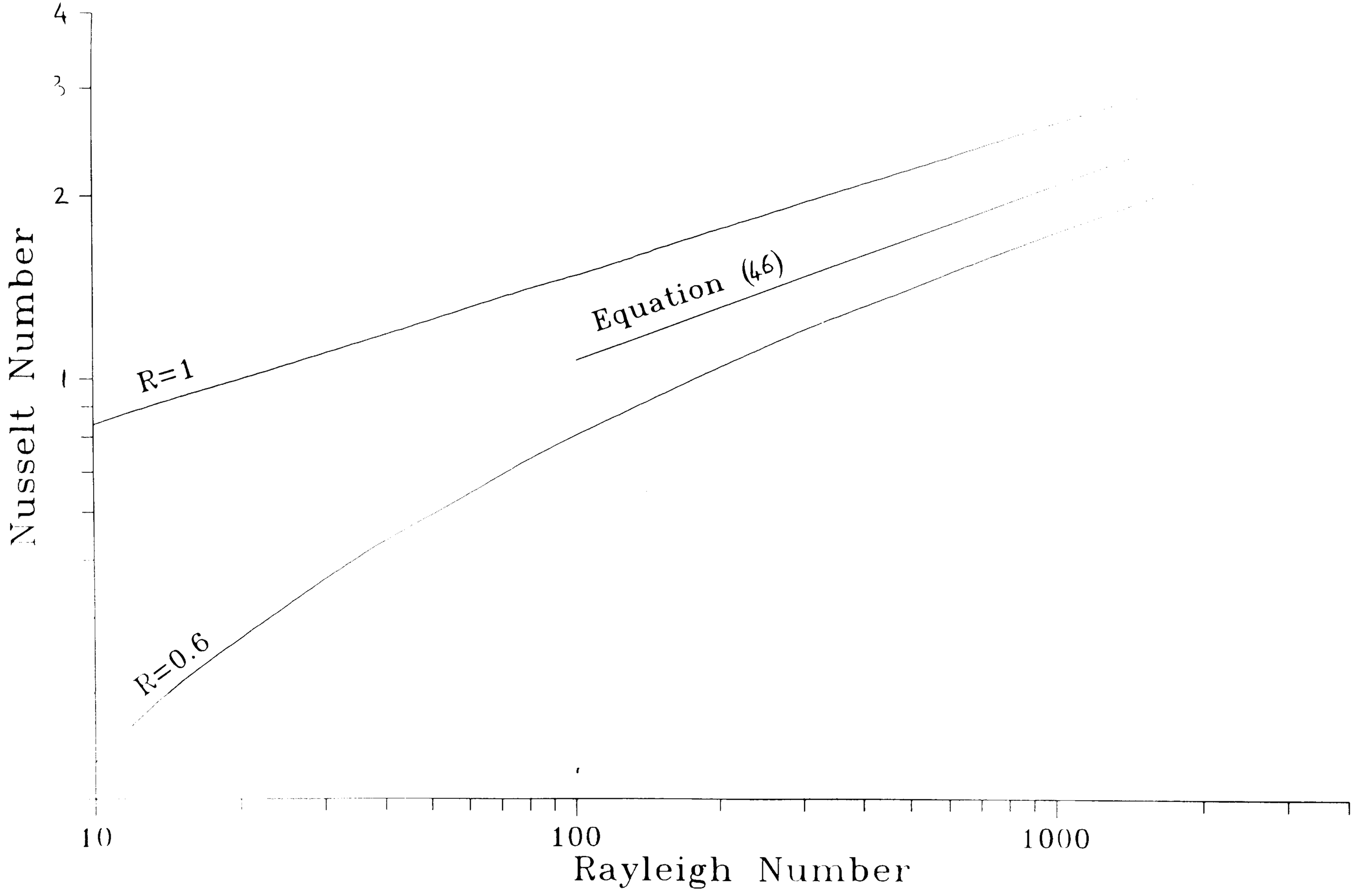


Fig.3.8(C): Experimental Results Compared with Correlations for a Horizontal Tube with Circular Fins

When the combination of (ii) and (iii) is taken into account, with the difference in heat exchanger design in the present series of experiments and in the experiments carried out by Tsubouchi, it can be deducted that a coiled horizontal axis finned tube heat exchanger makes better use of the heat transfer area than a heat exchanger made of entirely horizontal tube with vertical fins as the former leads to a higher heat transfer coefficient and does not allows for the interference between the boundary layer developing in adjacent fins to interfere significantly with the heat transfer process.

5) Edwards and Chaddock [14], investigated the rate of heat transfer from horizontal tubes with vertical circular fins for various fin geometries. Their experiments were carried out in air and for heat leaving pipes.

They defined the Rayleigh number on the fin spacing but corrected it by using the ratio of the fin spacing by the fin diameter. They correlated their data by:

$$Nu = 0.42Ra^{0.25} \qquad (57)$$

As their definition of the Rayleigh number, is slightly different, the correlation we have developed has to be modified in accordance. Assuming a power b of 0.25 the corresponding value for the coefficient C is then 0.3667 from Table III. When this value is modified to use the fin height as characteristic length in the Rayleigh number, the following correlation is found:

$$Nu = 0.4059Ra^{0.25} \qquad (58)$$

The equations are relatively similar. The difference in the coefficient C can easily be explained experimentally by the difference in methodology, experimental procedure, in the geometry being investigated and value of the Prandtl number. No explanation was found for the difference in predictions between the correlations developed by Edwards and Chaddock and Tsubouchi and Masuda.

Both Tsubouchi and Masuda and Edwards and Chaddock reported an increase in the slope of the experimental data at low Rayleigh number which is translated into an increase in the power b appearing in equations (34) and (57) at low Rayleigh numbers.

This change in slope can hardly be observed in the set of experiments being presented. However, a detailed analysis was carried out by splitting the range of Rayleigh number being investigated in two separate ranges. The corresponding heat transfer correlations were then evaluated separately using the methodology previously described. The final correlation corresponding to each range of Rayleigh number is :

$$Nu = 0.2850Ra^{29} \qquad \text{for} \quad 500 < Ra < 1000 \qquad (59)$$

$$Nu = 0.2683Ra^{30} \qquad \text{for} \quad 100 < Ra < 500 \qquad (60)$$

It should be mentioned that very few experimental measurements were available to develop these correlations (approximately 20 measurement for each correlation) therefore some provision should be made in interpreting them as their accuracy might be questioned. However, it can be seen that as in the case of the previously mentionned authors, there is a tendency to an increase in slope of the correlating curve when moving towards low Rayleigh numbers. But this change in slope is much smaller than the one reported by these authors.

6) When heat-transfer occurs between two parallel vertical fins, the equations of momentum can only be used as long as there is no interference between the boundary layers which develop on the opposite side of adjacent fins. The interference between the boundary layers can be avoided when the spacing between the fins is large enough.

The thickness of the boundary layer can be estimated using the value of the <u>local</u> Nusselt number from:

$$Nu_x = \frac{2x}{\delta} \qquad (61)$$

Since the maximum boundary layer thickness is at the lower part of the fin, the value of the fin height should be substituted as the characteristic length in equation (61).

Now, using the free convection correlation developed by Eckert [18], we find that the minimum spacing between the fins to avoid boundary layer interference depends on the Rayleigh number only. It is approximately given by:

$$Nu_x = \frac{2D}{\delta} = 0.424\ Ra^{0.25} \qquad (62)$$

By changing the characteristic length in the Rayleigh number and multiplying by the ratio of the fin spacing to the tube diameter, we obtain approximately:

$$\frac{D}{\delta} = 0.2\ Ra^{0.25} \qquad (63)$$

The straight line corresponding to equation (63) is represented in the Nu-Ra diagram in Fig.3.8(b). It can be seen that, no interference from the boundary layer is observed as the experimental data and the corresponding correlation curve Nu=f(Ra) are always above the straight line obtained for the limiting case for boundary layer interference.

Although a detailed investigation of the design of the fins on the heat transfer performance is not the scope of this investigation, it can be seen that at the moment, no interference between the fins of the heat exchanger is observed. This suggests that the heat transfer coefficient on the external surface of the fins is as high as could be expected in a natural convection process and therefore only marginal improvements in the heat transfer coefficient can be achieved by changing the fin geometry.

However, it also suggests that it is possible without decreasing the external heat transfer coefficient to slightly change the fin geometry to increase the external heat transfer surface area. This would make the overall heat transfer coefficient (UA value) per meter of finned tube slightly higher. Two simple changes in fin geometry can be envisaged: decrease in the fin pitch and increase in the fin height.

The best combinations of fin pitch and fin height can unfortunately only be obtained from practical experimentation as too many parameters would have to be investigated. In addition, further optimisation of the fins could only be carried out knowing the parameters to optimize. This could be either to minimize the amount of material needed to make the fins or the reduction in cost.

3.9 Correlations for Heat Exchanger Design

Although heat transfer correlations in terms of Nusselt and Rayleigh numbers are very useful to describe the heat-transfer mechanism on a theoretical basis, their usefulness tends to have some limitations when predicting heat exchanger performance. These limitations mainly come from the amount of calculation required to approximate the performance of a heat exchanger when operating in a hot water store due to the complexity of these heat transfer mechanism involved.

However, a set of correlations which would be simple to use and accurate enough for engineering purposes would be extremely useful for the prediction of the effectiveness of the heat exchanger to extract heat from the integrated thermal store. Such correlations can be obtained by a detailed analysis of the heat transfer occurring at the wall of the heat exchanger.

3.9.1 Number of Transfer Units

When steady state (or pseudo steady state) heat transfer is achieved, the whole of the heat transferred through the wall of the heat exchanger is gained by the water flowing into the heat exchanger's pipe. For an elementary part of the heat transfer surface area, the temperature of the water flowing in the pipe will increase by an elementary amount.

Assuming an <u>isothermal store</u> and constant thermal properties for the water in the store, the equation of partial derivatives describing the heat transfer process at the wall of the heat exchanger is:

$$UdA(T_t - T) = mCpdT \qquad (64)$$

By dividing the left and the right hand side of equation (64) by the rate of thermal capacity flowing in the heat exchanger coil, this equation becomes:

$$\frac{UdA}{mCp}(T - T_t) = dt \qquad (65)$$

Now, by integrating equation (65) between the inlet and the outlet of the heat exchanger, the ratio UA/mCp appears. This is known as the Number of Transfer Units of the heat exchanger. It represents the ratio of the rate of heat transfer through the wall of the heat exchanger by the rate of heat capacity flowing into the heat exchanger.

$$NTU = \frac{UA}{mCp} \qquad (66)$$

For a given flow rate of water into the heat exchanger, the NTU will be directly proportional to the UA value of the heat exchanger. It is however expected that when the UA value of the heat exchanger is increased, the size and/or cost of the heat exchanger is increased thus the NTU will also indirectly reflect the cost of the heat exchanger to carry out a given duty.

Although, the geometry and mode of heat transfer between our heat exchanger and the thermal store are different than the simplistic model considered previously, a very near definition of the Number of Transfer Units can be used to characterize the design of our heat exchanger.

The new definition of the NTU takes into account the fact that the total heat transfer area of a heat exchanger for the integrated thermal store considered is clearly divided into two areas corresponding respectively to the upper and lower temperature zones in the store. Each of this zone has its own part of the heat exchanger and therefore its corresponding UA value. The total UA value for the heat exchanger is given by:

$$UA = UA_1 + UA_2 \qquad (67)$$

Where UA_1 and UA_2 are the respective value corresponding to the lower and upper heat exchanger coil in the store.

Modifying equation (66) by using equation (67), the Number of Transfer Units relevant to our problem is then defined as:

$$NTU = \frac{UA_1 + UA_2}{mCp} \qquad (68)$$

The values of UA_1 and UA_2 can be estimated relatively easily from the experimental data or from the correlations in terms of Nusselt and Rayleigh numbers which were developed earlier. The rate of heat capacity flowing through the heat exchanger can be also evaluated, by measuring the flow rate of water in the coil. This gives a method for the evaluation of the NTU

During a constant mass flow rate thermal discharge, the rate of heat capacity (mCp) flowing through the heat exchanger tends to be relatively constant. Thus the variations of the NTU are very nearly proportional to the variations in the UA value of the heat exchanger. The correlations in terms of Rayleigh and Nusselt number developed earlier suggest that the UA value decreases with the store temperature. Thus the NTU decreases during the thermal discharge.

3.9.2 Effectiveness of Heat Recovery

The ability of the heat exchanger to extract heat from the thermal store can be evaluated by a dimensionless number known as the Effectiveness of heat recovery.

The Effectiveness of heat recovery is defined as the ratio of the actual heat transfer rate by the maximum achievable heat transfer rate. With the integrated thermal store, the actual heat transfer rate is:

$$q_3 = mCp(T_{out} - T_{in}) \tag{69}$$

The maximum achievable heat transfer rate would be reached with a heat exchanger having an infinite surface area and operating in the same conditions as the considered heat exchanger. In this case, the temperature of the water leaving the heat exchanger and the temperature of the water in the upper part of the store would be equal. This gives:

$$q_4 = mCp(T_{up} - T_{in}) \tag{70}$$

Finally the coiled finned tube heat exchanger considered, the Effectiveness of Heat Recover is by:

$$Eff = \frac{q_3}{q_4} = \frac{(T_{out} - T_{in})}{(T_{up} - T_{in})} \tag{71}$$

This effectiveness can be easily computed from the temperatures of the water entering and leaving the heat exchanger and the temperature of the water in the upper part of the thermal store.

During the course of a thermal discharge, as heat is extracted from the store, both T_{up} and T_{out} decrease. But as T_{out} decreases at a higher rate than T_{up}, the heat exchanger's effectiveness decreases and the heat recovery process becomes less and less efficient.

3.9.3 Correlation of the Experimental Data

The Number of Transfer Units and of the Effectiveness of heat recovery are extensively used for preliminary heat exchanger design. A full detailed investigation of their use has been carried out by Eckert [18].

Many existing correlations between the Effectiveness of heat recovery and the NTU are available for simple types of heat exchangers [18],[19],[20]. It can easily be shown that for a heat exchanger immersed in a fluid of infinite thermal capacity, the corresponding correlation is:

$$Eff = 1-\exp(-NTU) \qquad (72)$$

Other correlations available for parallel flow, cross flow and reverse flow heat exchangers have been developed by Eckert [18]. Analytical solution are not always possible but numerical solution have been developed for all major type of heat exchanger configurations.

However, only traditional types of heat exchanger have been investigated, and not a finned tube horizontal axis heat exchanger coil for an Integrated Thermal Store. Thus it was necessary to develop a new set of correlations to at least approximate the Effectiveness of heat recovery from the Number of Transfer Units.

Fig.3.9.3 gives the results obtained from the experimental data when the NTU and the Effectiveness of the heat exchanger coil are represented on the same graph.

To obtain the experimental background required to cover a wide range of values for the NTU, several experiments were carried out in which the temperature of the water in the store, the temperature of the water flowing in the heat exchanger and the water flow rate in the heat exchanger's pipe were varied.

For practical reasons, due to the limitation in the possibility of the rig, accurate experiments could not be carried out a very low and very high NTU. But it can be seen that although these data were obtained during different experiments, the same correlation curve approximate relatively closely all the experimental data. This suggests that the correlation curve could be used over a wide range of operating conditions either by choosing the NTU to reach a given performance or by predicting the performance when the NTU is known.

In order to simplify the design stage of this type of heat exchanger for integrated thermal store even further, a simple correlation of the experimental data was developed. To remain coherent with existing correlations, this predicted the effectiveness of heat recovery from the value of the Number of Transfer Units. To take into account the logarithmic nature of most of the heat transfer process involved in heat exchangers and by analogy with existing correlations in terms of Effectiveness, best fit curve was sought in the form:

$$Eff = a(1-\exp(-bNTU)) \tag{73}$$

where a and b are constants.

The values of a and b were adjusted to best fit the data using a least square method. This resulted in the following values:

a = 0.989

b = 0.768

This can be closely approximated by:

$$Eff = 1-\exp(-0.768NTU) \tag{74}$$

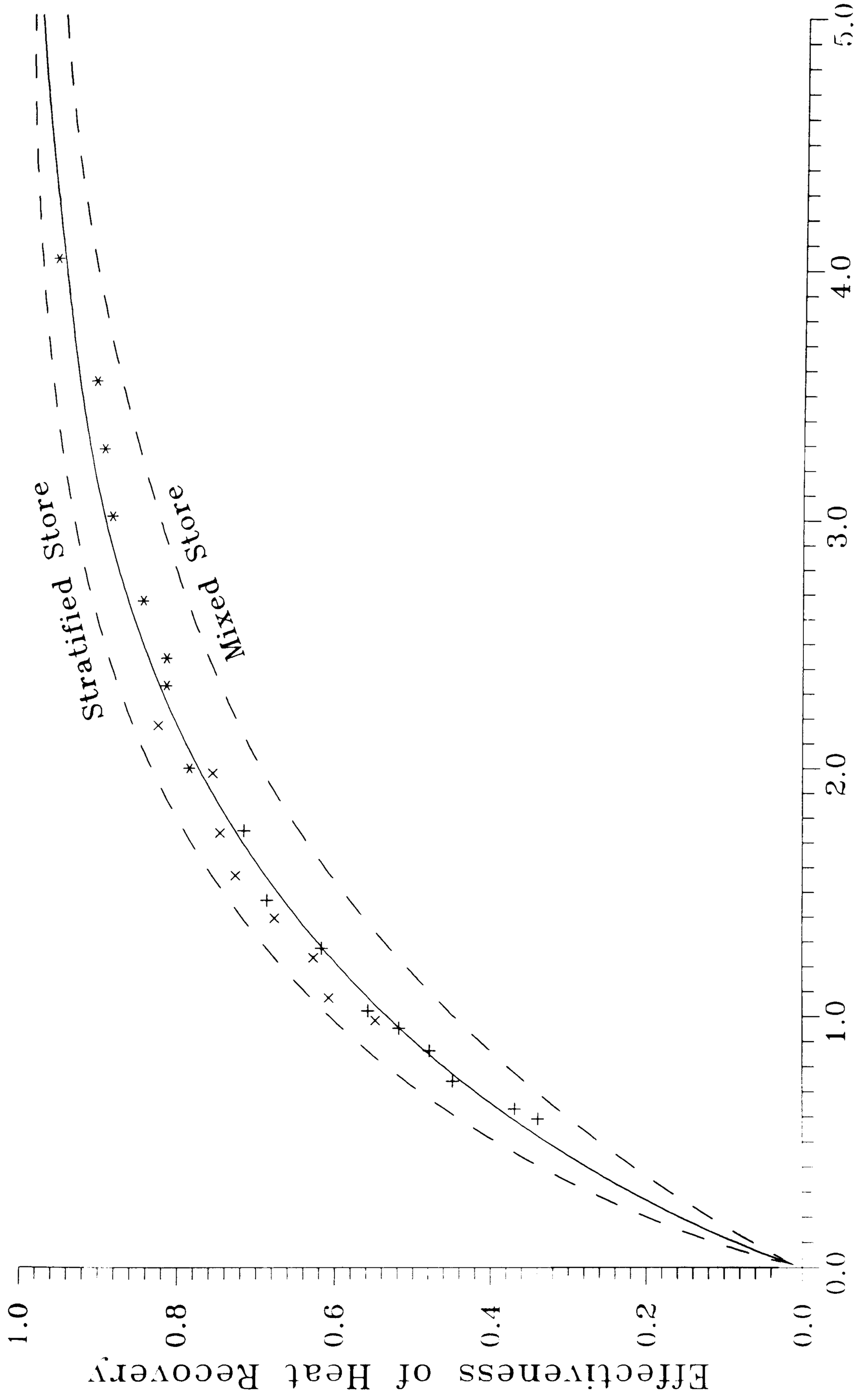


Fig.3.9.3: Effectiveness of Heat Recovery and Number of Transfer Units for the Considered Heat Exchanger Coil

The correlation curve corresponding to equation (74) is shown on Fig.3.9.3. Using this simplified correlation, it can be seen that, providing that the heat transfer area is large enough, the temperature of the water at the outlet of the heat exchanger can be as close as desired of the temperature of the water at the top of the store.

On the same graph are also shown the curve obtained with a thermal discharge achieved by a fully mixed store. This curve was obtained using a finite difference computer model of the thermal store which is described in details in the next chapter.

The results shows that the store with two heat exchangers has a better rate of heat delivery than the store fully mixed. The difference in performance of the heat exchanger is around 16.7% i.e. to match the same performance, mixed store would require a heat exchanger approximately 25% larger than a stratified store. Care should be taken in interpreting these results as fully mixed store do not occur easily in real situations as some form of stratification is almost always achieved.

3.9.4 Discussion

When a thermal discharge with a flow rate of approximately 9.5 litre/min is achieved, the NTU of the heat exchanger is initially around 3 which corresponds to an effectiveness of heat recovery of approximately 0.9.

During the thermal discharge, as the temperature of the water surrounding the heat exchanger decreases and consequently the external heat transfer coefficient of the heat exchanger decreases. This reduces the overall UA value of the heat exchanger, the Number of Transfer Units and finally the Effectiveness of heat recovery.

At the end of the thermal discharge, when 150 litres of water have passed into the heat exchanger coil, the NTU is around 1.5 which is approximately only half its initial value. However, the corresponding effectiveness of heat recovery is still as high as 0.7.

Heat exchangers are usually designed to operate in a range of Number of Transfer Units between 1 and 5. The value of unity is usually limited to low cost and low performance heat exchangers. The values of 5 or more are seldom encountered in practice except for high performance heat exchangers.

The considered heat exchanger coil operates within a range of NTU from 1.5 to 3. This should be regarded as a fair range for the operating conditions of this type of heat exchanger.

To this range of NTU correspond a relatively high effectiveness over the whole of the operating conditions. This suggests that only minor improvements in performance, or Effectiveness of heat delivery can be expected by increasing its NTU.

Consequently further investigations in the design of the heat exchanger should aim at reducing the cost of the heat exchanger instead of increasing its performance.

Inversely, trying to reduce the cost of the heat exchanger by reducing its NTU does not seem to be a viable alternative as any reduction in NTU at this stage would be accompanied by a significant reduction in the effectiveness of heat recovery which is likely to be unacceptable.

3.10 Conclusions

1) A heat transfer correlation for a finned tube heat exchanger coil with a horizontal axis immersed in a hot water store has been developed from experimental data. This is heat transfer correlation is:

$$Nu = 0.28Ra^{0.2929} \qquad \text{for} \quad 100 < Ra < 1500 \qquad (75)$$

This correlation is in good agreement with other experimental and theoretical correlations available in the literature for the same type of heat transfer processes and finned tube geometry. This correlation can be used over the range of operating conditions found in an integrated thermal store when in the heat discharge mode. The correlation approximated the experimental measurements with an accuracy better than 5%. It can be used for the top and the bottom heat exchanger coil.

2) A correlation has been developed to relate the Effectiveness of heat recovery with the Number of Transfer Units of the heat exchanger coil. This correlation is:

$$Eff = 1-\exp(-0.768NTU) \qquad \text{for} \quad 0.5 < NTU < 4 \qquad (76)$$

References

[1] Sanders, O.A.
The Effect of Pressure on Natural Convection in Air
Proc. Royal Soc., ser. A, Vol 157, pp 278-291, 1936

[2] Warner, C.Y. and Arpaci, U.S.
An Experimental Investigation of Turbulent Natural Convection in Air at Low Pressure on a Vertical Heated Flat Plate
Int J Heat Mass Transfer, Vol 11, pp 397-406, 1968

[3] Hesse,G. and Sparrow, E.M.
Low Rayleigh Numbers Natural Convection Heat Transfer from High Temperature Wires to Gases
Int J Heat Mass Transfer, Vol 17, pp 796-798, 1974

[4] Morgan, V.T.
The Overall Convective Heat Transfer from Smooth Circular Cylinders
Advances In Heat Transfer, ed. T.F.Irvine, J.P.Harnett,
Vol 11, pp 199-264, Academic, New-York, 1975

[5] Schmit, T.
Improved Methods for Calculation of Heat Transfer on Finned Surfaces
Kaeltetechnik-klimatisierung, Vol 18, part 4, 1966

[6] Kern, D.Q.
Extended Surface Heat Transfer
McGraw-Hill Book Company, New York, 1972

[7] Rogers, G.F.C. and Mayhew, Y.R.
Heat Transfer and Pressure Loss in Helically Coiled Tubes with Turbulent Flow
Int J Heat Mass Transfer, Vol 7, pp 1207-1216, 1964

[8] Seban, R.A. and McLaughlin, E.F.
Heat Transfer in Tube Coils with Laminar and Turbulent Flow
Int J Heat Mass Transfer, Vol 6, pp 387-395, 1963

[9] McAdans, W.H.
Heat Transmission
McGraw-Hill, New York, 3rd edition,1954

[10] Sleicher, C.A. and Rouse,M.W.
A Convenient Correlation for Heat Transfer to Constant and Variable Property Fluids in Turbulent Pipe Flow
J Heat Mass Transfer, Vol 18, No 5, pp 677-683, 1975

[11] Edwards,J.A. and Chaddock, J.B.
An Experimental Investigation of the Radiation and Free Convection Heat Transfer from a Cylindrical Disk Extended Surface
ASHRAE Transaction, Vol 69, pp 313-322, 1963

[12] Tsubouchi, T. and Masuda, H.
Natural Convection Heat Transfer from Horizontal Cylinders with Circular Fins
Proc 6th Int Heat Transfer Conf, Paris, 1970, paper N.C.10

[13] Knudsen, J.G. and Pan, R.B.
Natural Convection Heat Transfer from Transverse Finned Tubes
Chemical Engineering Progress Symposium Series, N°57, Vol 61, pp 44-49, 1965

[14] Edwards, J.A. and Chaddock, J.B.
Free Convection and Radiation Heat Transfer from Fin-on-tube Heat Exchangers
ASME, paper N°62-wa-205, 1962

[15] Wiebelt, J.A. , Henderson, J.B. and Parker, J.D.
Free Convection Heat Transfer from the Outside of Radial Fin Tubes
Heat Transfer Engineering, Vol 1, N°4, 1980, pp 53-59

[16] Sparrow, E.M. and Bahrami, P.A.
Experiments on Natural Convection Heat Transfer on the Fins of a Finned Horizontal Tube
Int J Heat Mass Transfer, Vol 23, 1980, pp 1555-1560

[17] Feiereisen, T.J. and Klein S.A.
Heat Transfer from Immersed Coils
ASME Trans, paper N°82 wa/sol-18, New York, 1982

[18] Eckert E.R.G. and Drake, R.M.
Heat and Mass Transfer
McGraw-Hill, New York, 2nd Edition, 1959

[19] Kreith, F.
Principles of Heat Transfer
Harper International Edition, New York, 3rd Edition, 1973

[20] Kays, W.M. and London, A.L.
Compact Heat Exchangers
McGraw-Hill, New York, 3rd Edition, 1984

Chapter 4

One Dimensional Computer Model of An Integrated Thermal Store during a Thermal Discharge

4.1 Introduction

Correlations have been developed to evaluate the external heat transfer coefficient of the heat exchanger coil during the thermal discharge. By the use of these correlations, it is possible to build relatively simple mathematical models of the Integrated Thermal Store when subjected to a thermal discharge achieved using the heat-exchanger coil.

Several types of computer models have already been developed. The simplest computer models assume that the thermal store is isothermal (i.e. no temperature gradient exists within the store). The store is then considered as a lump capacity and its temperature is only allowed to vary with time to model the transient operation.

However in an Integrated Thermal Store, due to the geometry of the heat exchanger coil, there is a large temperature variation between the bottom and the top of the store during a thermal discharge. Thus the isothermal store model, although extremely simple to use, would not approximate the complex heat transfer processes taking place in the store with a sufficient accuracy to be useful.

More accurate but relatively more complex models are based on the assumption that the temperature of the water in the thermal store vary with in the vertical direction only. The vertical direction is chosen to simulate the effect of stratification where relatively cold water tend to stay at the bottom of the store whereas relatively warm water is at the top.

Several computer model using the assumption of the one dimensional temperature variation can be found in the literature [1],[2],[3]. They are mainly used for preliminary sizing of the components for use in solar heating systems where the relation between solar radiation, the design of the heat exchanger and the transient operation of the store are not always straightforward.

The model which was developed uses this assumption of the one dimensional temperature variation with the height in the tank. It has several advantages including :

1) Reasonable accuracy of computer predictions
This accuracy is particularly relevant to predict the transient behaviour of the thermal store.

2) Flexibility in use
Ease to accommodate a very wide range of operating conditions and several types of components such as different sizes of heat exchangers and thermal stores. Additionally, extra options can easily be investigated such as different geometries of heat exchangers and the use of a mixing valve.

3) Low complexity of the computer model
This include at the same time low computation power required, low computer memory requirement and above all relatively simple to programme.

4.2 Mathematical Formulation

In this model, the buoyancy driven flow in the store is neglected. The temperature of the water contained within the store is then allowed to vary with the height (to simulate stratification) and with time (to simulate the transient operation of the store). The heat contained in the store can be conducted axially along the store or exchanged with the wall of the heat exchanger and the environment. The one dimensional differential equation describing the conservation of heat at a high x above the bottom of the store is:

$$e(T-To) + UA(T-F) + h\frac{\partial^2 T}{\partial x^2} = mCp\frac{\partial T}{\partial t} \qquad (1)$$

The first term of the left hand side corresponds to the heat which is lost from the thermal store to the environment during the operation of the store. It reflects the fact that as the thermal store is at a higher temperature than its surroundings, some heat loss through the walls of the store are inevitable.

The second term of the left hand side corresponds to the heat transferred between the water in the store and the water flowing in the domestic hot water heat exchanger's pipe.

The last term in the left hand side is introduced to model the heat transfer which takes place within the store by the processes of convection and mixing. It allows to model different temperature profiles in the thermal store during the thermal discharge.

Finally, the term on the right hand side represents the rate of change in internal energy of the water in the store.

There are two unknowns in equation (1) which are respectively the temperature of the water in the store T and the temperature of the water in the heat exchanger F. An additional equation is obtained by a heat balance at the wall of the heat exchanger. This equation is:

$$UA(T-F) = mCp \frac{\partial F}{\partial x} \qquad (2)$$

In addition, a set of boundary conditions is required to solve this system of partial derivative equations. These boundary conditions are :

- at time t=0 the store is assumed isothermal $T=80^{\circ}C$

- the temperature of the water entering the heat exchanger is fixed $T_{in}=8^{\circ}C$

- the temperature of the environment is fixed $T_{e}=20^{\circ}C$

- the flow rate of water delivered to the taps is fixed $m_{dhw}=8$ litre/min

Finally, the mixing valve located at the outlet of the heat exchanger in integrated thermal stores should be modelled. This induces two supplementary unknowns in the system of equations. Fortunately, two extra equations are also obtained. The first equation reflects the energy balance at the three way valve. this is:

$$m_{ex}(T_{out}-T_{in}) = m_{dhw}(T_{dhw}-T_{in}) \qquad (3)$$

The second equation is obtained from the setting temperature of the mixing valve. This equation is:

$T_{se}=60^{\circ}C$

4.3 Finite Difference Formulation

The system of equations (1)-(3) can not be solved analytically and consequently was replaced by an implicit finite difference formulation. The use of explicit formulation might have lead to stability problems and therefore had to be rulled out.

In this implicit formulation, the store is divided in horizontal isothermal slabs. The heat in each slab is exchanged with adjacent slabs, the wall of the heat exchanger and the environment. When the equation describing the conservation of heat in the store are expressed in finite difference terms, the following equations for a slab i in the store are obtained:

$$e_i(T_o - T_i) + UA_i(T_i - F_i) + h_i \frac{T_{i-1} - 2T_i + T_{i+1}}{\Delta x^2} = m_i Cp \frac{T_i - TT_i}{\Delta t} \qquad (4)$$

where e_i = coefficient of heat loss to the environment
UA_i = UA value of the heat exchanger in slab i
h_i = heat transfer coefficient in slab i
m_i = mass of the slab
T_i = temperature of the water in slab i
F_i = temperature of the water in the heat exchanger
TT_i = temperature of the water at previous time step

The energy balance at the wall of the heat exchanger is again for slab i:

$$m_i Cp \frac{F_i - F_{i-1}}{\Delta x} = UA_i(T_i - F_i) \qquad (5)$$

The equation describing the conservation of heat at the three way valve is then :

$$m_{ex}(T_n - T_1) = m_{dhw}(T_{dhw} - T_1) \qquad (6)$$

Note that in equations (4)-(6) the slabs are numberred in increasing number starting from the bottom of the store.

Finally, the other equations required to complete the system can easily be derived from the boundary conditions which are presented earlier.

4.4 Choice of Heat Transfer Parameters

Several parameters depending on the type of store to be modelled appear in the finite difference formulation. The value of these parameters was chosen using existing heat transfer correlations to closely match the behaviour of Integrated Thermal Store when subjected to a thermal discharge achieved by means of the heat exchanger coil.

The e_i coefficient, which reflect the heat loss from the store to the environment, were assumed to vary only with the temperature of the water in the considered slab. They were calculated separately for each slab from the rate of heat loss from the store to the environment. The rate of heat loss from the store was assumed to be given by equation (5) which was developed in Chapter 2 from experimental data.

The h_i coefficients were introduced to simulate different degree of stratification in the store. When the coefficients h_i between the slabs are high (the order of several thousands), the temperature in the corresponding adjacent slabs tend to be equal. This models a mixed store (a store where there is little temperature gradient between the top and the bottom). To the contrary, if the coefficients h_i are small then the temperatures of the adjacent slabs are allowed to vary widely. In this case, large temperature gradients can develop between the top and the bottom of the store and stratification can be modelled.

Thus by varying the coefficients h_i, the temperature profile along the axis of the thermal store can be varied. which allows a lot of flexibility to model different type of thermal discharge.

The h_i coefficients were obtained form the temperature profile in the store during the thermal draw-off represented if Fig.4.4. By a guess-error process, successive values of h_i were tried until the temperature profile in the model matched with a reasonable approximation the experimentally measured temperature profile. The temperature profile obtained after the h_i were chosen is also represented in Fig 4.4.

The UA_i values represent the product of the heat-transfer surface area of the heat exchanger in the considered slab by the overall heat-transfer coefficient of the heat exchanger in the same slab.

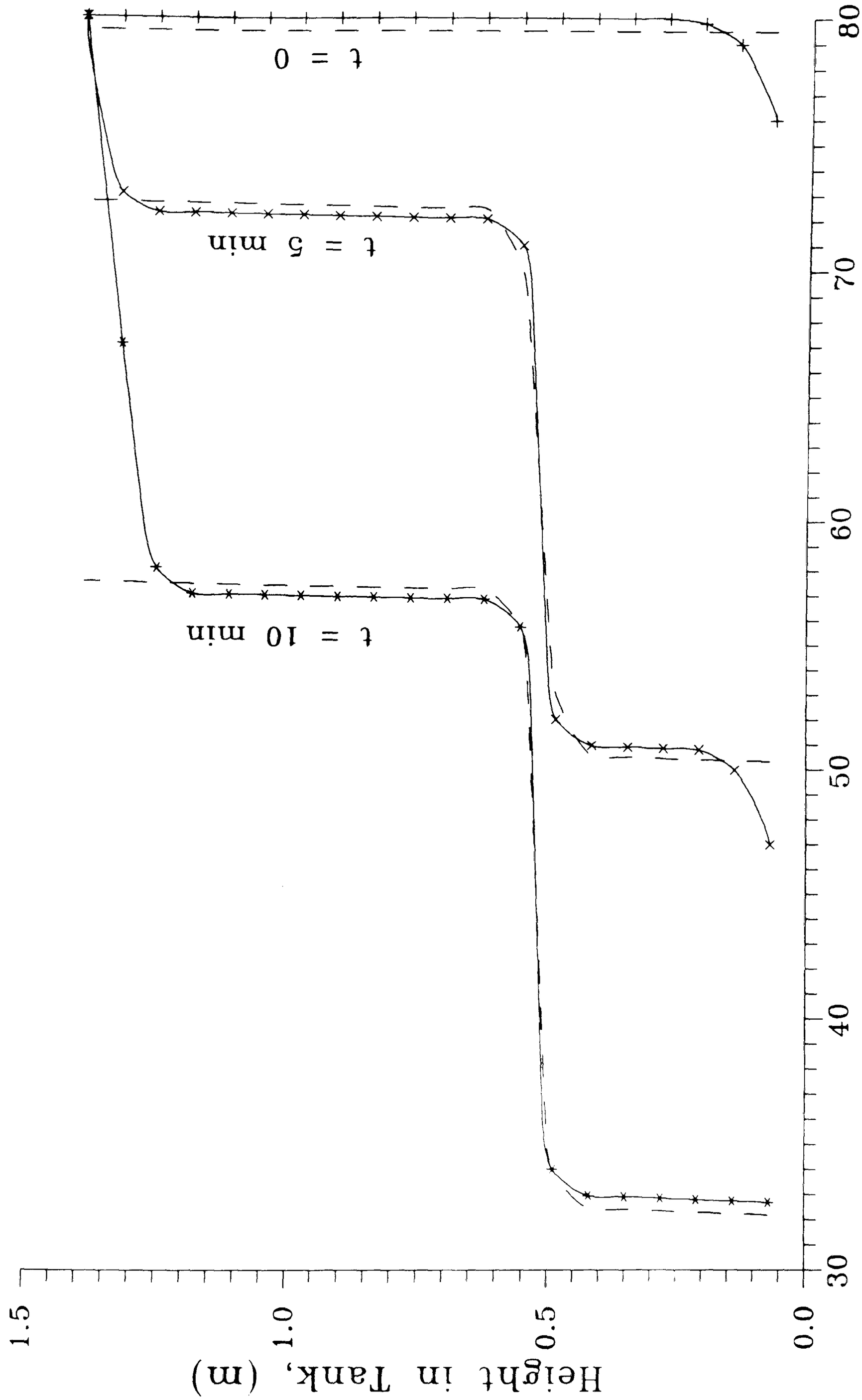


Fig.4.4: Temperature Profile in the Integrated Thermal Store

The heat transfer surface area in each slab was given by the size, location and arrangement of the finned tube heat exchanger in the hot water store.

The overall heat-transfer coefficient was calculated from the internal and external heat-transfer coefficients. The fin efficiency was assumed to be unity. This resulted in the following equation:

$$\frac{1}{UA_e} = \frac{1}{A_e h_e} + \frac{1}{A_i h_i} \qquad (7)$$

where U is the overall heat-transfer coefficient of the heat exchanger
h_e is the external heat-transfer coefficient of the heat exchanger
h_i is the internal heat-transfer coefficient of the heat exchanger
A_e =external heat-transfer surface area
A_i^e=internal heat-transfer surface area

The problem of estimating the internal and external heat-transfer coefficient has already been discussed in Chapter 3 and only a brief summary is presented in the following.

The internal heat-transfer coefficient h_i is given by the formula by Sleicher and Rouse.

$$Nu = 5+0.015Re^{a}Pr^{b} \qquad 0.1<Pr<10000 \qquad (8)$$

$$10000<Re<1000000$$

$$a = 0.88 - \frac{0.24}{(4+Pr)}$$

$$b = 0.333 + 0.5\exp(-0.6Pr)$$

To take into account the curvature of the coil this should be modified according to the modified McAdams's formula:

$$hc = (1 + 5\frac{r_w}{r_c})hs \qquad (9)$$

The external heat-transfer coefficient was evaluated from the natural convection heat transfer correlation for a finned coil with an horizontal axis immersed in a hot water store which was developed in Chapter 3. This correlation is:

$$Nu = 0.28Ra^{0.2929} \qquad (10)$$

In equation (10), all the thermal properties are evaluated at film temperature. The characteristic length for the calculation of the Rayleigh and Nusselt numbers is the fin spacing.

Finally, the remaining parameters which include the size of the store and the mass flow rate of water in the heat exchanger, were fixed to their experimental values. The thermal properties of water were assumed constant.

The number of nodes and the time step for use in this type of finite difference formulation are very important as they will affect at the same time the accuracy of the computer predictions and the computation time.

When less than 6 nodes are used in the finite difference formulation not enough flexibility is allowed in the temperature variations of the water flowing in the heat exchanger and consequently the accuracy of the computer predictions can seriously be questioned. Above 12 nodes, the computer predictions are reasonably accurate (more than 98 % of the value predicted using a very large number of nodes). However, the final number of nodes for use in the model was increased to 22. This allowed the locations of the slabs to correspond exactly to the location of the thermojunctions in the store and made the comparison with experimental data easier.

As an implicit formulation was used, the numerical resolution was stable regardless of the time step. The time step was therefore chosen as a compromise between the accuracy of the solution and the computation time. It was fixed to 30 seconds which is relatively small when compared to the time constant of the thermal discharge of the order of approximately 1000 second. Additionally, a test case was made by running the model with a time step of 60 seconds. The comparison of both run showed no significant difference in the computer predictions.

4.5 Computer Predictions and Experimental Results

An example of computer prediction is presented in Fig.4.5(a). It shows the temperature variations at the outlet of the heat exchanger with time during a thermal discharge of 8 l/min. Experimental results are also shown in solid lines.

From this graph one can notice the imperfections of the three way valve. First the time response which at the beginning leads to some oscillations in T_{dhw}. Then the imperfections in the steadiness of the temperature of the water delivered to the taps. In spite of these imperfections, the accuracy of the computer model predictions seems acceptable.

In addition many parameters can be investigated using the computer model, of particular relevance are parameters which are difficult to evaluate experimentally either because of the cost involved or the experimental problems involved.

Among these is the location of the bottom heat exchanger coil which can not be easily accessed by experimentation. For this reason, no optimisation has ever been carried out so far with respect to its location in the store. It can however be seen that if this heat exchanger coil is situated at the very bottom of the store, it will only extract heat from a very small volume of water and consequently its effectiveness will be greatly reduced. Inversely, if it is situated at the top of the store, no temperature gradient (stratification) can take place and therefore the store will behave like a mixed store which gives a low effectiveness of heat recovery.

Different locations of the heat exchanger in the store can easilly be investigated numerically by changing the values of the h_i coefficients so that the interface between the hot and cold zone is moved up and down in the store.

The computer predictions are presented in Fig.4.5(b). They show that there is a location of the bottom heat exchanger which gives a better effectiveness of heat recovery. This location is obtained when the interface between the hot and cold temperature zones in the store is situated exactly at mid height (between the slabs 10 and 11). It was also noticed that when the interface is at this very location, a better degree of stratification is achieved. Changing the height of the interface in the store is possible by raising the lower heat exchanger by approximately 20 cm in the thermal store (at present, the interface is between the 7th and 8th slabs). The predictions suggest that an improvement in the amount of heat delivered of around 2.5 % can be achieved in the process.

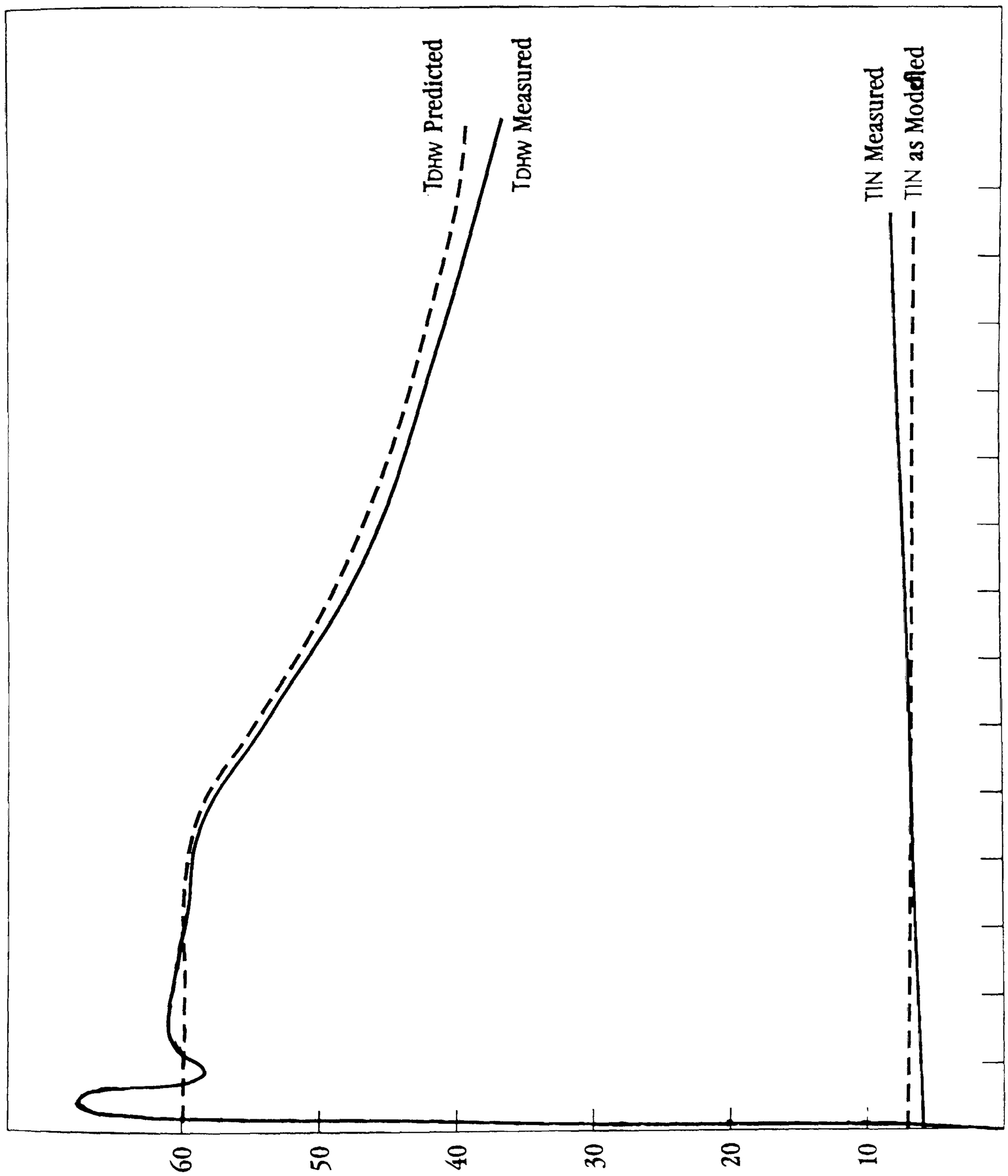


Fig.4.5(a): Variation of the Temperature of the water after the Mixing Valve with Time

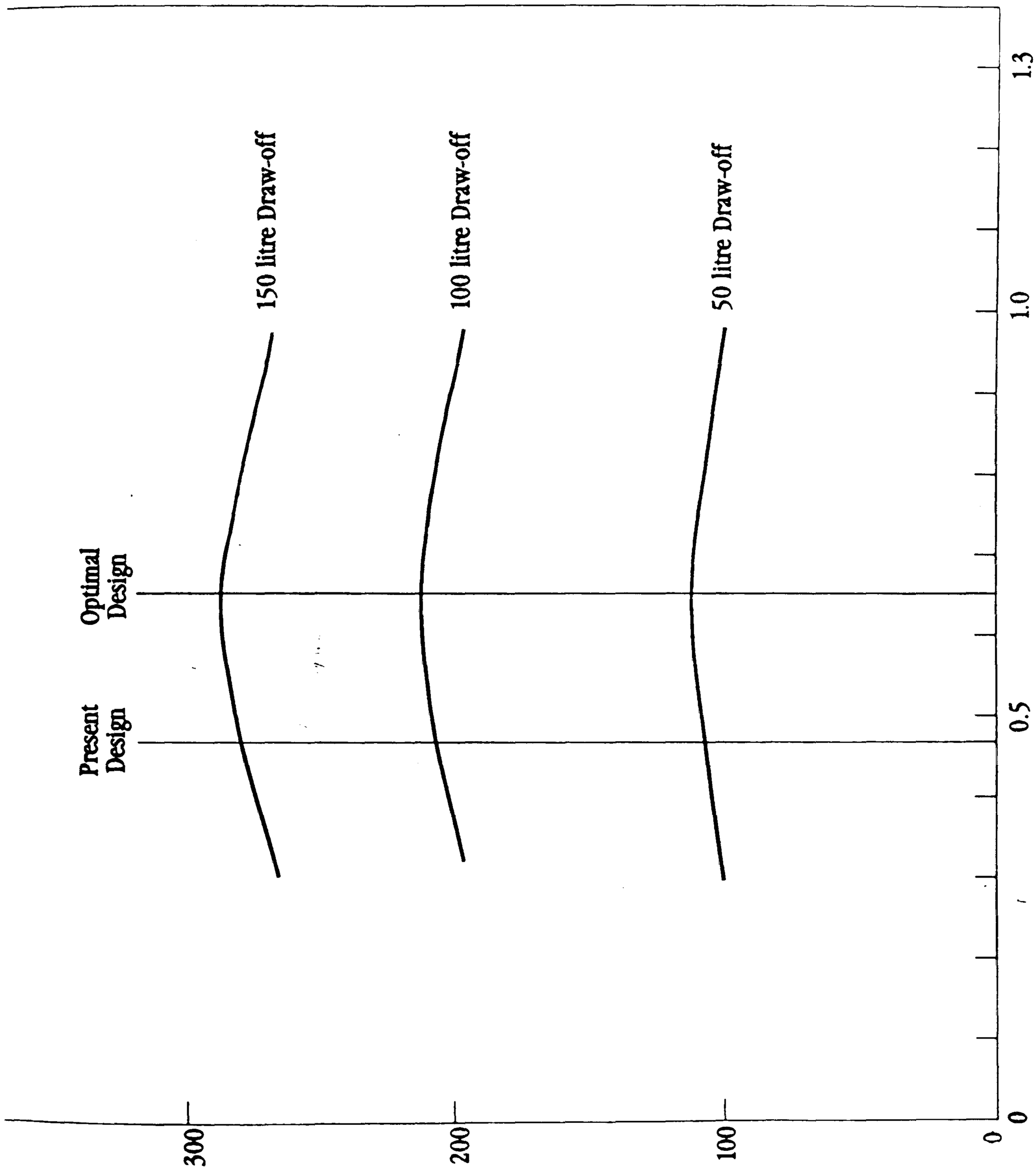


Fig.4.5(b): Heat Recovered for Various Heights of Lower Heat Exchanger Coil in the Hot Water Store

The variation of the UA value of the heat exchanger for both the present heat exchanger and the heat exchanger where the bottom coil is situated between the slabs 10 and 11 are represented in Fig.4.5(c). Also, the curves representing the temperature profile in the tank for both cases are represented in Fig.4.5(d).

The UA value of the store with the optimum location is always slightly higher. The temperature gradient between the top and the bottom of the store however is lower. The balance between the better UA value and the lower temperature gradient is such that only at the middle of the store the performance is maximum.

Again by changing the h_i value along the height of the store, the type of thermal discharge can be varied. For example, when the h_i are very high, the store tends to be completely isothermal. If this is the case, the thermal behaviour of the store and consequently its ability to deliver heat are changed.

A isothermal store with the same geometry as the present store was modelled by setting all the h_i to very high values. The results were used to plot the graph which was presented earlier in the curve representing the variations in the effectiveness of heat recovery with the Number of Transfer Units of the heat exchanger in Chapter 3 (see Fig.3.8)

A fully stratified store would be obtained when the all the coefficient h_i are set to very small values. This can not be modelled without changing the geometry of the heat exchanger and assuming that the UA value of the heat exchanger is evenly distributed in the store.

Assuming a heat exchanger with a UA value evenly distributed, and all the h_i coefficient set to 0, the Eff-NTU curve was evaluated again. This curve is also represented in Fig.3.8.

The computer predictions suggest that the perfectly stratified store is better although some provision must be made in interpreting these results as the heat transfer correlation used in both models corresponds to a heat exchanger coil with a horizontal axis, which might not necessarily be appropriate for the thermal store with an evenly distributed heat transfer surface area.

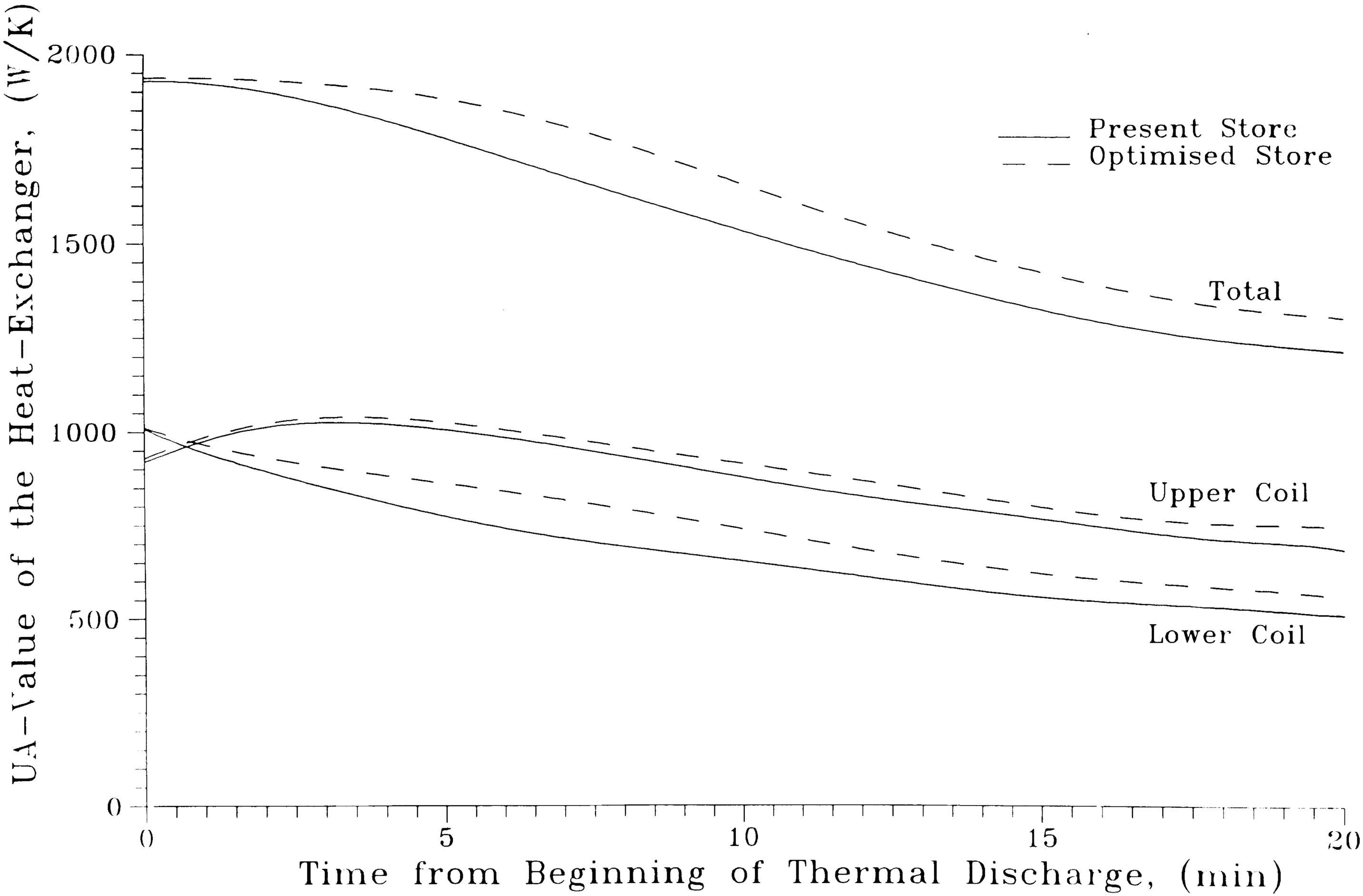


Fig.4.5(c): Difference in heat exchanger's UA Value Between Optimised and Real Store

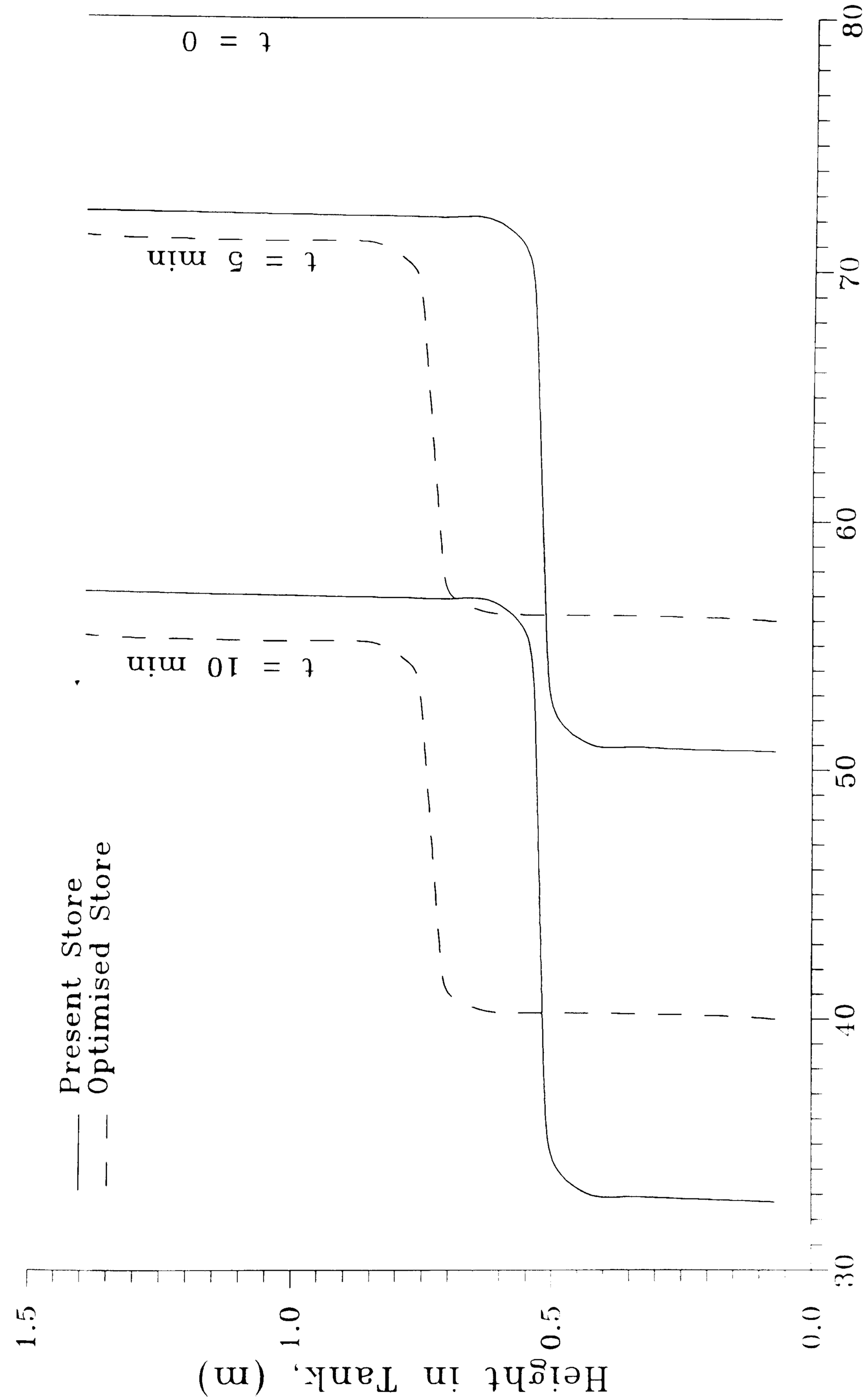


Fig.4.5(d): Difference in Stratification Between Optimised and Real Store

Finally , other parameters can easily be investigated with this type of computer model. Among these are the type of tubing used to make the heat exchanger coil in terms of tube internal and external diameter, fin spacing, fin thickness and fin height.

Moreover, using field data as input, the performance of the store in a real situation can relatively easily be modelled. Further improvement would however be needed such as modelling the effect of the heating mechanism on the thermal store.

4.6 Conclusions

A computer model based on a one dimensional implicit finite difference formulation of a heat balance for the water in the Integrated Thermal Store when subject to a thermal discharge achieved by means of the heat exchanger coil has been developed. The computer predictions are in good agreement with experimental measurements obtained from an Integrated Thermal store operating in similar conditions. This computer model is simple to use and to programme can be used to investigate the effectd of many parameters such as the design arrangement and location of the heat exchanger in the store, the stratification and the effect of the three way valve on the performance of heat delivery.

The computer predictions suggest that the effectiveness of heat delivery from Integrated Thermal Store is maximum when the lower heat exchanger coil is situated in such a way that the interface between the top and bottom temperature zones in the store is situated exactely at mid-height.

References

[1] Chauvet,L., Nevrala, D.J. and Probert, S.D.
Heat Discharge Characteristic of a Finned Heat Exchanger Immersed in a Hot Water Store
Applied Energy, 34,(1989), pp 165-179

[2] Garg, H.P., Mullick,S.C. and Bhargava, H.K.
Solar Thermal Energy Storage
D.Reidel Publishing Company, Dordrecht, pp 82-153

[3] Pissavin, P.
Modélisation du comportement dynamique d'un ballon de stokage solaire à échangeur interne
Revue Générale Thermique, N° 246-247, 1982, pp 521-535

Chapter 5

Effect of the Mass Flow Rate of Water Flowing Into The Heat Exchanger Coil on the Effectiveness of Heat Delivery

5.1 Introduction

The overall heat transfer coefficient of the heat exchanger(UA-value) depends on the external **and** internal heat transfer coefficients. Up to now, the main area of investigation was the natural convection heat transfer coefficient on the outside surface of the finned tube of the heat exchanger.

However, the internal heat transfer coefficient, might play an important role in the heat transfer process particularly when the mass flow rate of water flowing into the heat exchanger's pipe is reduced.

This mass flow rate of water in the heat exchanger coil will have an effect on the rate of heat delivery from an Integrated Thermal Store directly or indirectly through the following parameters:

- the stratification in the store

- the internal heat transfer coefficient of the heat exchanger

- the Number of Transfer Units of the heat exchanger

- the rate of heat delivery (through the term $mCp(T_{out}-T_{in})$)

Furthermore, the mixing valve located at the outlet of the heat exchanger will change the mass flow rate of water passing through the pipe of the coil during the thermal discharge to keep the temperature of the water delivered to the taps relatively constant. This variation in the mass flow rate will induce an extra transient component in the thermal discharge process.

An entirely theoretical evaluation of the effect of the variation of the mass flow rate on the effectiveness of heat delivery would have been possible (although complicated) but would have lacked the experimental basis required to any credible investigation. It was therefore decided to investigate the effect of this flow rate by using a balance between theoretical evaluation and practical experimentation.

A detailed literature survey of the effect of the flow rate flowing inside coiled pipe on the effectiveness of heat recovery from thermal store was found relatively unsuccessful due to several factors. One of this factor is the difficulty to extrapolate experimental results obtained using heat exchanger having different designs than the one located in Integrated Thermal Stores.

Consequently, only basic heat transfer correlations could be used for this investigation. These were relatively useful to describe the heat transfer in qualitative terms but had also limitations due the several factors including their relatively low acuracy for predicting the heat transfer mechanism on inside (and outside) surface of the heat exchanger coil. .

Finally, a numerical evaluation of the effect of mass flow rate on the effectiveness of heat delivery was not possible due to the difficulty to model parameters such as stratification and the buoyancy driven fluid flow in the store.

5.2 Experimental Apparatus

Only part of the experimental rig described in Chapter 2 was used for the experiments. This is presented in Fig.5.2 and corresponds to the use of the domestic hot water facility.

The hot water storage tank was first preheated to 80°C. A time period of 2 minutes was then allowed to pass at the end of the heating period. These 2 minute time period allowed for the water in the store to settle so that a higher degree of repeatability in the experiments was achieved.

Cold water at 8°C from the chilled water supply was then passed into the heat exchanger coil. The experiment were carried out until around 150 litres of cold water has passed into the heat exchanger coil. During the thermal discharge, the heating elements were not used so that the flow in the store was due entirely to the buoyancy effect created by the cold water passing into the heat exchanger.

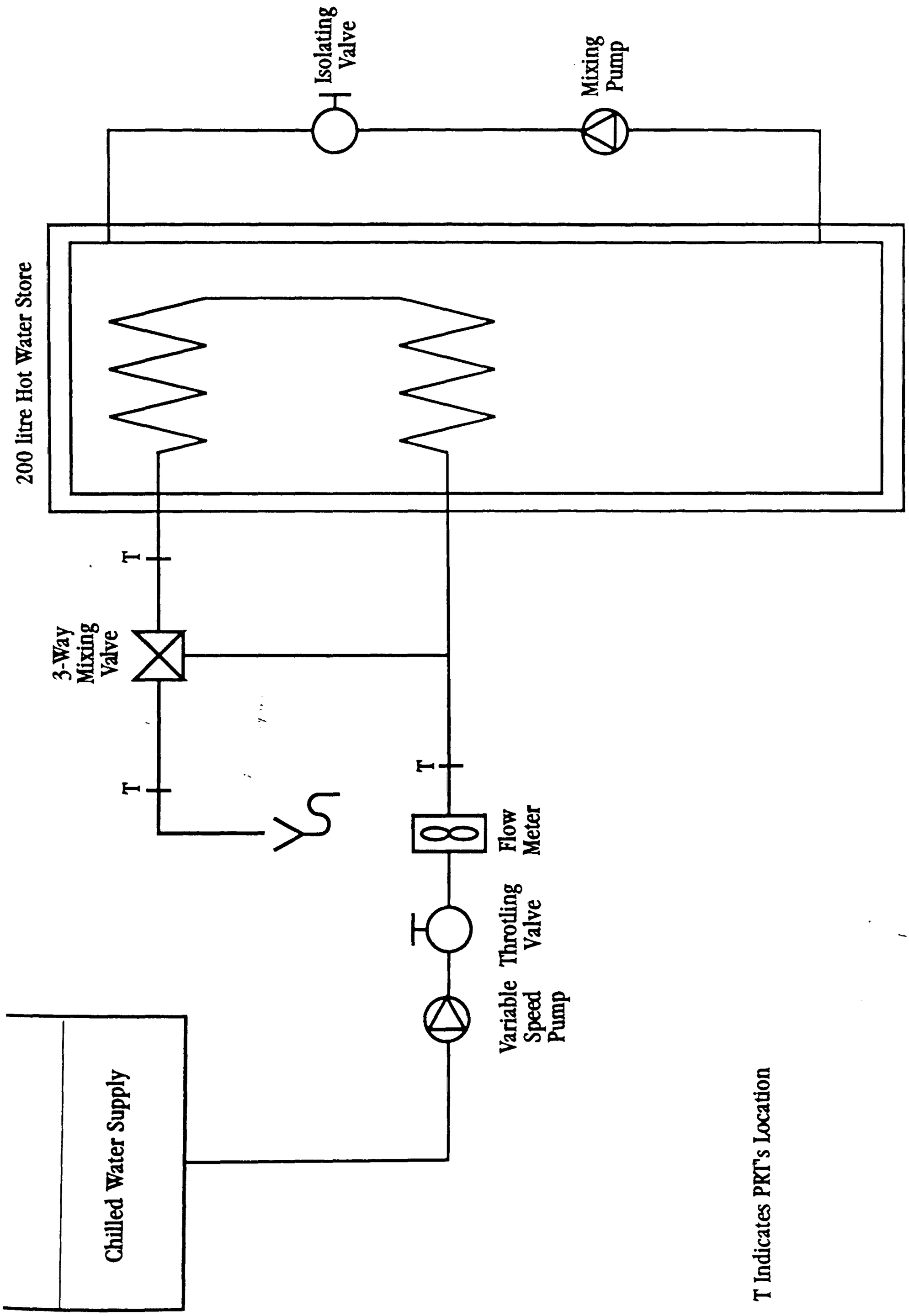


Fig. 5.2. Detailed Schematic of Hot Water Facility

Several series of test were carried out. In the first series of tests the mass flow rate of water flowing through the heat exchanger was kept constant. For this purpose, the three way mixing valve was removed from the heat exchanger outlet. The constant mass flow rate thermal discharge greatly facilitated the experimental observations by removing some of the transient effect caused by the mixing valve.

The water flow rate in the heat exchanger was varied from 2 to 7 litre/min by increments of approximately 0.5 litre/min. Although this flow rate seems low, it covered the whole of the range of flow relevant to our investigation.

In the second series of tests, the effect of the temperature setting of the three way mixing valve on the effectiveness of heat delivery was investigated. Several thermal discharges were achieved with temperature settings varying from 32°C to 62°C.

The instrumentation included high accuracy PRTs, K-type thermocouples and high pulsing rate flowmeters. All of these are described in full details in Chapter 2. The sensors were scanned every 10 seconds which was the minimum time interval which could be used without loss of accuracy in the experimental measurements.

In all the tests, the measurements from the instrumentation provided data about:

- the temperature of the water at the inlet and outlet of the heat exchanger

- the temperature of the water delivered to the taps

- the temperature of the water in the store

- the mass flow rate of water passing into the heat exchanger

From this data, it was possible to evaluate:

- the amount of heat extracted from the store

- the degree of stratification in the store

- the average temperature of the water in the store

- the mass flow rate of water by-passed by the three way valve

The rate of heat delivery from the store is given by:

$$q = m_{dhw} Cp (T_{dhw} - T_{in}) \qquad (1)$$

As the rate at which the PRTs were scanned was relatively high, the variation of the inlet and outlet temperatures of the water in the heat exchanger and the mass flow rate were small between two scans, it is reasonable to assume that the rate of heat recovery is given by:

$$q = \frac{m_f Cp (T_{dhw} - T_{in})}{\Delta t} \qquad (2)$$

Where m_f is the mass of water delivered to the taps between the considered scans.

Equation (2) was used for most of the results which are presented in this analysis. However, in some cases, it was more convenient to calculate the amount of heat recovered by using an equation which is strictly speaking equivalent. This equation is:

$$q = \frac{m_{fe} Cp (T_{out} - T_{in})}{\Delta t} \qquad (3)$$

Where m_{fe} is the mass of water which has passed into the heat exchanger during the time interval dt.

The total amount of heat recovered at the end of the thermal discharge could be obtained by integrating any of the equations (1) to (3) with respect to time. As the scanning rate was relaticely high , this could be approximated relatively accurately by the sum:

$$Q = \sum_{0}^{t} m_{fe} Cp (T_{out} - T_{in})$$

The degree of stratification in the store was defined as the temperature difference between the thermojunctions 1 and 20. No extrapolation was made. The location of the thermojunction in the 200 litre store is shown in Fig.2.8(a) in Chapter 2.

5.3 Variation of the Mass Flow Rate in the Heat Exchanger

For this series of experiments, the three way mixing valve was removed from the rig so that the mass flow rate in the heat exchanger pipe could be kept relatively constant during the whole duration of the thermal discharge. As the time constant of the system was different from one experiment to the other, a comparison was made in the amount of heat recovered when a given amount of water had passed into the heat exchanger.

Table IV shows the degree of stratification and the total amount of heat recovered and at the end of a thermal discharge of 150 litres. The experimental results suggest that there is a flow rate which gives a better rate of heat extraction from the store. This flow rate can be estimated at 4 litre/min. When the thermal store operates at this optimum flow rate, aproximately 4% more heat can be recovered from the store than the amount of heat which would be recovered from a thermal discharge achieved with a flow rate of 6.5 litre/min. It can also be observed that the maximum rate of heat recovery corresponds to the maximum degree of stratification being achieved at the end of the thermal dsicharge.

Fig.5.3 shows the amount of heat recovered during thermal discharge of 50, 100 and 150 litres. It can be seen that there is always a flow rate which gives an maximum rate of heat recovery. The value of this optimum flow rate tends to sligtly increase for longer thermal discharges.

5.3.1 Analysis of the Factors Influencing the Amount of Heat Recovered at the End of the Thermal Discharge

1) Heat loss to the environment

When the mass flow rate in the heat exchanger is reduced, it takes more time to pass the considered amount of water through the heat exchanger and consequently the time constant of the thermal discharge increases. Thus in experiments at very low flow rate, as the thermal discharge lasts longer, more time is given for the heat in the thermal store to escape to the environment by conduction through the walls of the store. This lost heat, which is then not available for useful purposes, can be calculated from:

$$q = \int_0^t UA(T-T_o)dt \qquad (4)$$

Flow Rate In Coil (litre/min)	Heat Recovered (MJ)	Stratification (°C)
2.46	30.685	21.75
3.04	31.174	22.56
3.41	31.542	23.09
3.95	31.659	23.17
4.52	31.327	22.96
5.03	30.944	22.27
5.50	30.436	21.19
6.09	30.402	21.12
6.45	30.375	20.97

Table IV: Heat Recovered and Degree of Stratification for Different Mass Flow Rate in the Heat Exchanger Coil

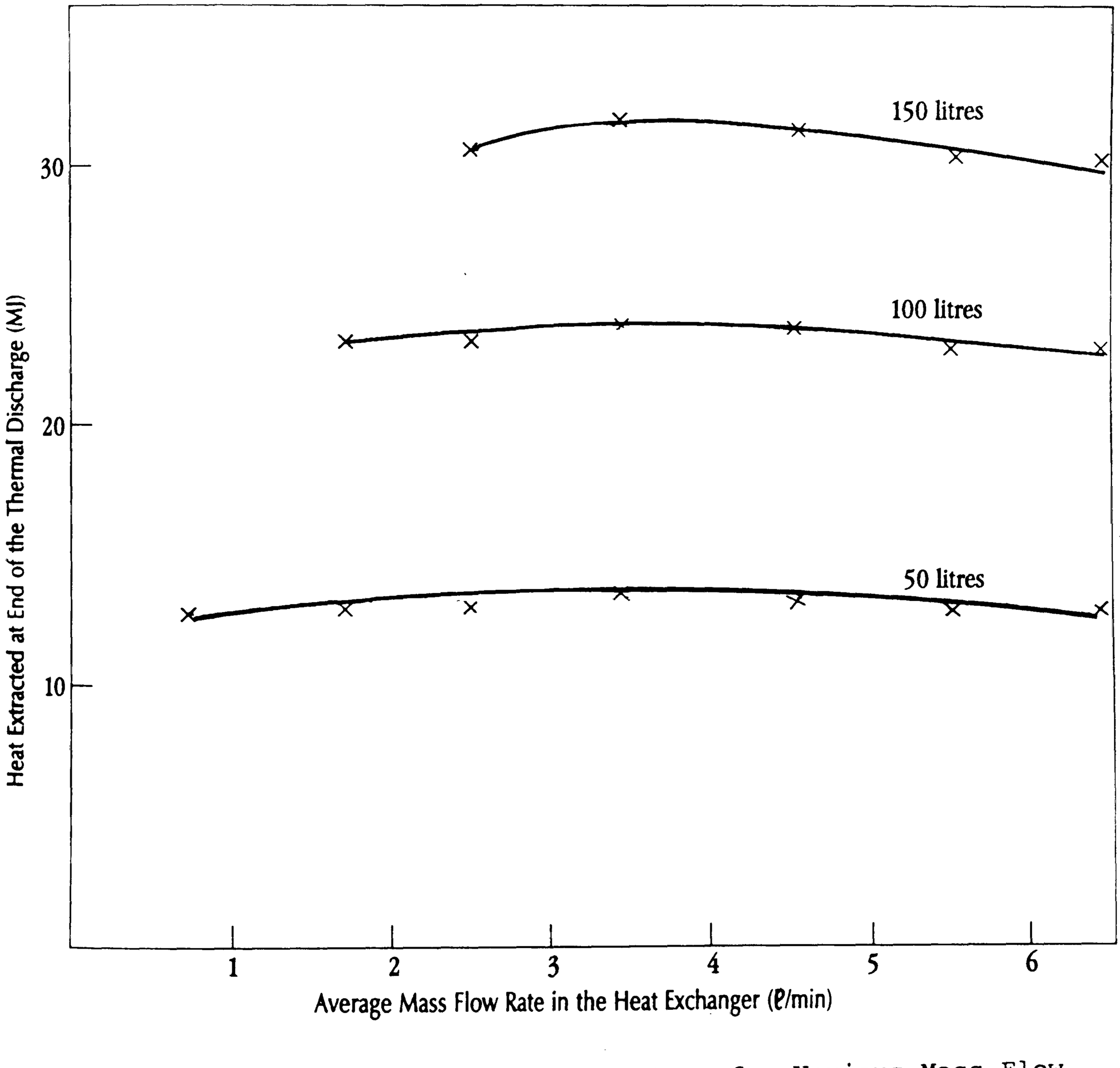


Fig.5.3: Heat Extracted From the Store for Various Mass Flow Rates in the Heat Exchanger

Using the equation giving the UA value of the store to the environment (equation (5), Chapter 2) combined with the temperature variations of the water in the store during the thermal discharge, the total heat loss from the store was evaluated numerically.

During a 2.46 litre/min thermal discharge the heat loss from the thermal store to the environment is around 2.5 MJ. Whereas during the same thermal discharge achieved at a flow rate of 6.5 litre/min, this is only 1.1 MJ. So at low flow rate, around 1.4 MJ less heat is available for the domestic hot water.

It should be remembered that the total heat stored in the tank is around 60 MJ so the 1.4 MJ represent 2.4% extra heat which might be significant as the variations in the amount of heat recovered are measured in a few per cent.

The simplest way to reduce the difference in heat loss between high and low flow rate heat draw-off is to reduce the heat loss from the thermal store to the surrounding environment. This can be achieved either by increasing the thickness of insulation or by using a more efficient insulant. Although it is not always possible for economic or practical considerations, significant energy savings can be achieved by this very simple method.

The optimal thickness of insulation for the store can be calculated using an energy balance at the wall of the thermal store. The idea behind this calculation is that the optimal thickness of insulant corresponds to a minimum energy requirement over the whole of the operating lifetime of the system. The heat loss from the store to the environment is during the lifetime of the system t' is:

$$Q_1 = \int_0^{t'} UA(T-Te)\,dt \qquad (5)$$

Which can be approximated with a reasonable accuracy by:

$$Q_1 = \frac{k}{X} A(T-Te)t' \qquad (6)$$

The energy required to manufacture the insulant is

$$Q_2 = \rho XAE \qquad (7)$$

The total energy requirement is then:

$$Q = Q_1 + Q_2 = \frac{k}{X}A(T-Te)t' + \rho XAE \qquad (8)$$

This total energy requirement is minimum when its partial derivative with respect to the thickness of insulant X is nil. This gives the optimal thickness of insulant as:

$$X_{opt} = \left[\frac{k(T-Te)t'}{\rho E} \right]^{\frac{1}{2}} \qquad (9)$$

Now by using the numerical values corresponding to the operation of an Integrated Thermal Store which are T-Te=60°C, a lifetime of 20 years and the thermal properties of a traditional insulant which are k=0.05 $Wm^{-1}K^{-1}$, o=50 kgm^{-3} and E=4x10^{9} Jkg^{-1}, the optimal thickness of insulant to use is 0.097m.

Provision should be made in interpreting this value of optimal thickness of insulation. Firstly because it disregards economic considerations; secondly because it also disregards practical considerations; and finally because the value of the optimal thickness can vary widely according to the physical properties of the type of insulant used.

2) Stratification

The experimental results suggest that when the rate of heat recovery is high, a high degree of stratification takes place. Although a link between the degree of stratification and the effectiveness of heat recovery would be difficult to quantify.

The degree of stratification in the store depends on factors such as the overall heat-transfer coefficient of the heat exchanger, the rate of heat delivery, the rate at which the buoyancy flow within the store will mix hot and cold water in the store. Most of these depend on the mass flow rate in the pipe.

If the mass flow rate in the heat exchanger's pipe is increased then the internal heat transfer coefficient of the heat exchanger is increased and , due to a lowering of the temperature of the wall of the heat exchanger, stronger buoyancy driven convection currents will be generated within the store. These stronger currents will increase the mixing in the store which will reduce stratification.

Also, if the store is poorly insulated, a cold boundary layer will develop along the walls of the store. This cold boundary layer will mix with the relatively warm water already in the store which will also reduce stratification.

Moreover, in a poorly insulated store, as the heat loss is increased, it is more difficult to store heat at high temperature and therefore to achieve a high degree of stratification.

So again by increasing the insulation level, a slightly better stratification might be achieved which might also help to increase the amount of heat extracted from the store.

3) Heat-Transfer Coefficient

The internal heat transfer coefficient is obtained from forced convection of water flowing inside a coiled pipe. This forced convection mechanism is mainly characterised by the Reynolds number of the water flowing into the coil.

Changing the mass flow rate in the heat exchanger will change the Reynolds number and therefore is very likely to change the internal heat transfer coefficient. This will change the overall heat transfer coefficient (U-value) of the heat exchanger and consequently the amount of heat recovered during a thermal discharge. Although a detailed analysis of how the internal heat transfer coefficient varies with the mass flow rate would be relatively complex, the general guidelines need to be given.

There are only three types of flow: laminar flow, transition flow and turbulent flow. To each of these flows will correspond a different heat transfer mechanism on the internal surface of the heat exchanger and therefore a different heat transfer coefficient. Consequently, each of these flow has to be investigated separately and the final result combined to obtain a global overview of the heat transfer process.

(i) Laminar Flow

In the laminar region, the water remains relatively undisturbed as it flows along the heat exchanger coil. This is characterised by the presence of streamlines.

In this case, the heat transfer from the wall of the heat exchanger to the the water flowing in the inner region of the pipe occurs mainly by diffusion of heat through layers of fluid at different temperatures. As the diffusion process is relatively slow, the heat transfer coefficient of a fluid flowing inside a straight pipe will not be high.

The heat transfer for a laminar flow in a long cylindrical tube has been evaluated theoretically a number of times from the equations of motion [2]. It depends only on the type of boundary condition at the wall of the tube. It does not depend on the Reynolds or Prandlt numbers of the considered fluid. For a uniform wall temperature, the heat transfer coefficient can be calculated from:

$$Nu = 3.66 \qquad (10)$$

For a constant heat flux at the wall of the pipe, the heat-transfer coefficient can be calculated from:

$$Nu = 4.36 \qquad (11)$$

In both equation (10) and (11), the physical properties of the fluid should be calculated at the mean bulk temperature

$$T_b = \frac{(T_{in} + T_{out})}{2} \qquad (12)$$

(ii) Transition Flow

The heat transfer for a transition flow can unfortunately not been approximated as a large number of parameters which cannot be quantified easily will determine when transition occurs.

A detailed analysis of some of the factors which affect the transition from laminar to turbulent flow has been carried out by Bankston [1] for gases flowing in straight pipes. Among the main factors affecting this change of flow are the variation of the fluid properties with temperature, the turbulence created at the entrance region of the heat exchanger and the Reynolds number.

Additionally, it is expected the factors not investigated by Bankston will be important. Among these are the curvature of the coil and the internal roughness of the pipe.

It is however expected that the internal heat-transfer coefficient in the transition region will be between the values obtained for the turbulent and laminar flow.

(iii) Turbulent Flow

In the turbulent region, mixing plays the dominant role in the heat transfer process of a fluid flowing inside a straight pipe. As mixing is usually a relatively quick process, high values of the heat transfer coefficient can be achieved in the turbulent region.

The turbulent heat transfer coefficient will be approximated reasonably accurately for a straight pipe by the formula from Sleicher and Rouse [3]:

$$Nu = 5 + 0.015Re^{a}Pr^{b} \qquad \begin{matrix} 0.1 < Pr < 10000 \\ 10000 < Re < 1000000 \end{matrix} \qquad (13)$$

$$a = 0.88 - \frac{0.24}{(4+Pr)}$$

$$b = 0.333 + 0.5\exp(-0.6Pr)$$

The variations of the Nusselt number with the Reynolds number for water flowing inside a straight pipe are summarized in Fig.5.3.1(a) for the laminar and turbulent regions.

By comparing equations (10), (11) and (13), it can be seen that the the heat transfer achieved in the laminar region is at least one order of magnitude lower than the heat transfer in the turbulent region.

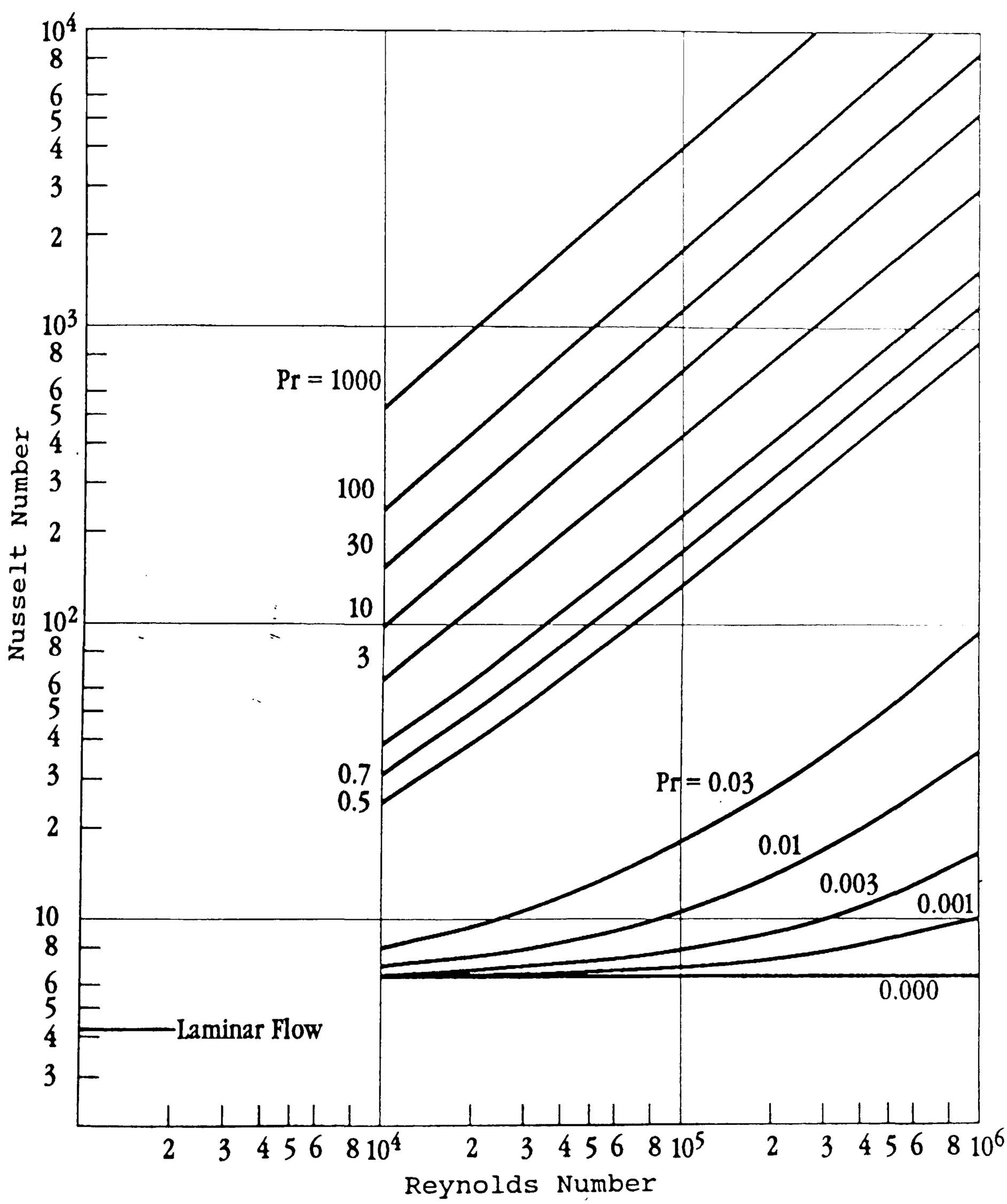


Fig.5.3.1(a): Nusselt Number for Fully Developed Turbulent Flows in Straight Pipes

The overall heat transfer coefficient of the finned tube heat exchanger coil used in Integrated Thermal Store depends both on the internal and external heat transfer coefficients. It can be estimated by:

$$\frac{1}{UA_e} = \frac{1}{A_e h_e} + \frac{1}{A_i h_i} \qquad (14)$$

When the flow in the pipe is turbulent, the internal heat transfer coefficient is high. Therefore the overall heat transfer rate through the wall of the heat exchanger tends to be limited by the external heat transfer coefficient. This external heat transfer coefficient corresponds to a natural convection heat transfer process which has been investigated separately (see Chapter 3).

When the flow in the pipe is reduced to the limit between turbulent and transition flow, then due to difference between the surface area in the inside and the outside of the pipe, both the internal and external heat transfer coefficients have nearly the same contribution to the overall heat transfer coefficient (U value) of the heat exchanger.

If the flow rate in the pipe is reduced further, then the flow becomes a transition flow. Although no correlations are available for transition flow, a sharp decrease in the internal heat transfer coefficient can be expected. This sharp decrease is accompanied by a significant decrease in the overall heat transfer coefficient of the heat exchanger.

The Reynolds number for a straight pipe is defined as

$$Re = \frac{\rho V d}{\mu} \qquad (15)$$

Using this definition of the Reynolds number, and assuming that the heat exchanger is made of straight pipe, the limits between the laminar and transition flows and transition and turbulent flows are obtained respectively for values of 2000 and 10,000. These correspond respectively to flow rates of approximately 1 litre/min and 5 litre/min in the heat exchanger. To achieve a high internal heat transfer coefficient and therefore a high rate of heat transfer through the wall of the heat exchanger, it would then be advisable to have a turbulent flow in the coil. This means a flow rates of 5 litres/min or more in the heat exchanger.

In fact the heat exchanger used in the Integrated Thermal Store is made of coiled tube and not of straight pipe. Due to the increased turbulence induced by the curvature of the coils, the limits between the laminar and transition and transition and turbulent flows will be moved to lower Reynolds numbers than that of a straight pipe.

At the moment, no correlations are available for the prediction of these limits. However, as the curvature of the heat exchanger coil is not extremely high, only a slight reduction in these limits might be achieved.

The limits from transition to turbulent flow being moved to slightly lower flow rates explains why no significant reduction in the effectiveness of heat recovery is observed for the considered coil at exactly 5 litre/min but as suggested by the data presented in Table IV at a slightly lower value of around 4 litre/min.

4) Number of Transfer Units

By definition, the Number of Transfer Units is proportional to the UA value of the heat exchanger and inversely proportional to the mass flow rate of water flowing into the heat exchanger. Thus when the flow rate is decreased, the NTU might increase or decrease according to the respective contribution of these coefficients.

When the flow in the heat exchanger pipe is turbulent, the UA value of the heat exchanger depends mainly on the external heat transfer coefficient. The main contribution to the NTU is then the mass flow rate term which appears at the denominator in the NTU. If the mass flow rate in the heat exchanger is reduced, then the NTU increases and therefore the effectiveness of heat recovery improves.

When the flow in the pipe is reduced so that the flow changes from turbulent to transition there is a sudden and rapid decrease in the internal heat transfer coefficient. This decrease is so sharp that the variations in the UA value of the heat exchanger coil decreases at a much higher rate than the mass flow rate in the pipe. The variations in the NTU are then given primarily by the UA term which appears on the numerator which in the considered case decrease. Finally, the effectiveness of heat recovery decreases.

The variations of the NTU and of the UA value with different mass flow rates in the heat exchanger are presented in Fig.5.3.1(b). They clearly show the two zones corresponding to the turbulent region where the NTU decreases with increasing mass flow rate in the coil and the laminar region where the NTU increases with the mass flow rate in the coil.

The maximum value for the NTU is found at around 4 l/min which is in relatively good agreement with the experimental results from Table IV. This suggests that the effectiveness of heat recovery is maximum with the NTU which is in agreement with the experimental data presented in Chapter 3 (Fig.3.9.3).

Finally, the relatively sharp peak in the NTU is only reflected by a small increase in the amount of heat recovered at the end of the thermal discharge as this peak occurs at a relatively high value of NTU when the effectiveness tends to be relatively independent of the NTU (see Fig 3.9.3 in Chapter 3).

5.3.2 Commentary

1) When water is passed at a flow rate of say 5 litre/min in the heat exchanger coil, the rate of heat output from the thermal store is around 20 kW which is sufficient enough to fulfill most of the domestic hot water demand.

2) During the operation of the store, a gas boiler would try to thermally charge the store during the draw-off. The rate of heat input from the boiler can be as low as 3 kW in a Integrated Thermal Store.

This 3 kW can almost be neglected at high flow rates but might be a significant contribution at low flow rates. Thus experiments at very low flow rate (say less than 2 litre/min) are not very relevant in the absence of the heat source.

3) The previous results, suggests that the effectiveness of heat recovery can be optimised by changing the internal diameter of the heat exchanger. For example, by using a larger or smaller internal diameter pipe, it is possible to design the heat exchanger coil so that the limit between transition and turbulent flows (which corresponds to the highest rate of heat recovery from the store) happens at smaller or higher flow rates than 4 litre/min as required. For example, for the peak in NTU to be at 12 l/min, a 33 mm internal diameter pipe would be required.

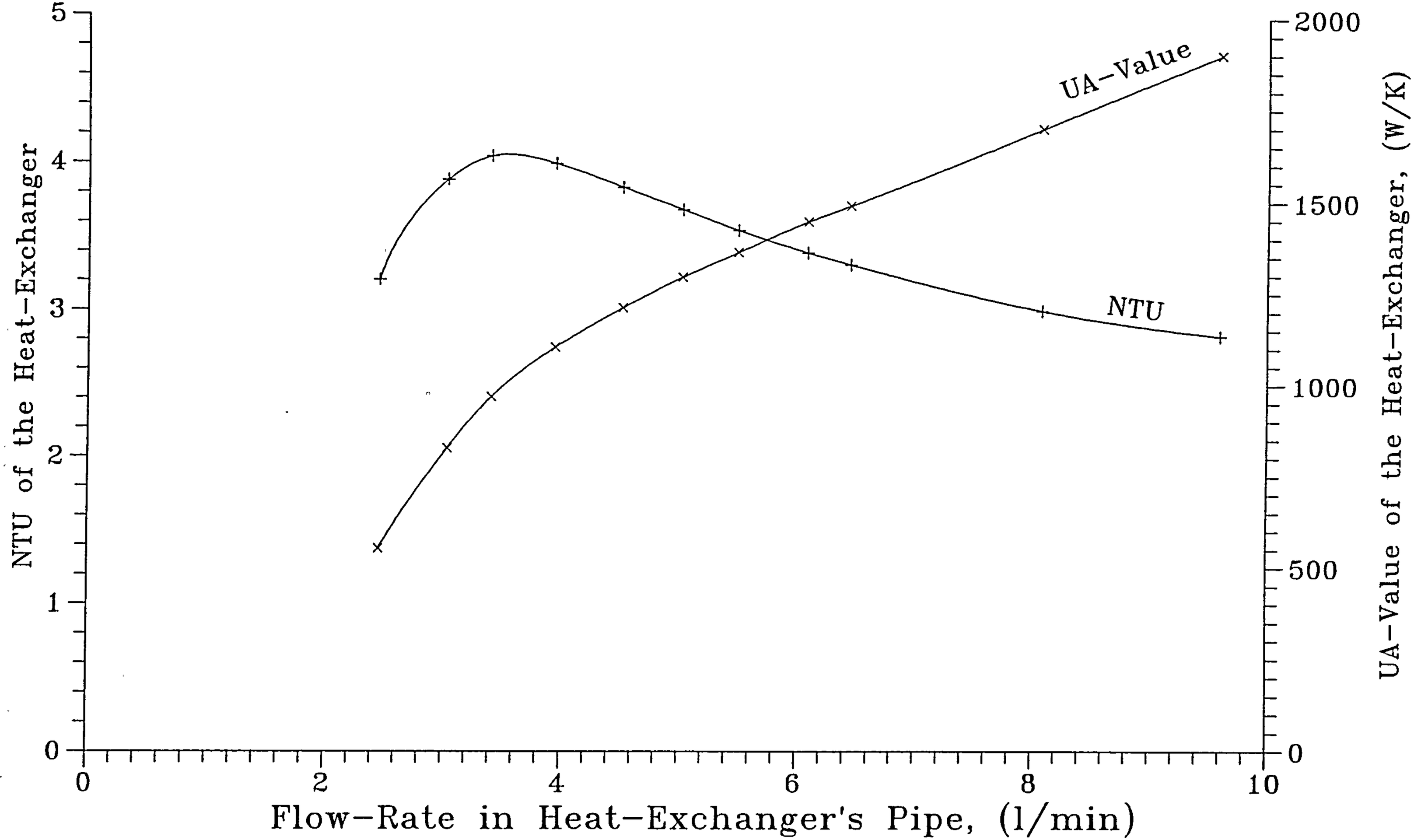


Fig.5.3.1(b): Variation of the Number of Transfer Units and of the UA Value With the Flow Rate in the Heat Exchanger's Pipe

5.3.3 Conclusions

The effectiveness of heat recovery from Integrated Thermal Store is maximum when the Number of Transfer Units of the heat exchanger is maximum.

The maximum value for the NTU (and for the effectiveness of heat recovery) is obtained for the minimum flow rate for which the flow in the heat exchanger's pipe is turbulent. With the present design of heat exchanger, this was evaluated at 4 litres/min. When operating at this optimum flow rate, 4 % more heat can be recovered than when operating at a flow rate of 6.5 litre/min or above.

A simple energy analysis was carried out to evaluate the optimum thickness of insulant to use in Integrated Thermal Stores. This optimum thickness is approximately 0.1 m.

5.4 Variation of the Setting Point of the Three Way Valve

The effect of the temperature setting of the three way valve located at the outlet of the heat exchanger in Fig.5.2 was investigated. For this purpose, a series of experiments was carried out where the temperature setting was changed from 32°C to 62°C by increments of a few °C.

During the Experiment, the flow rate in the heat exchanger was allowed to vary according to the fluctuations dictated by the valve. This varied approximately from 6 to 9 litre/min.

The total volumetric flow rate of water was measured by the flowmeter situated at the inlet of the heat exchanger. The flow rate of water passing through the heat exchanger was calculated using an energy balance at the three way valve. This energy balance is:

$$m_{ex} = m_{dhw} \frac{T_{dhw} - T_{in}}{T_{out} - T_{in}} \qquad (16)$$

As all the temperature measurements used in equation (16) were obtained with high accuracy PRTs, m_{ex} was obtained with nearly the same accuracy than m_{dhw} was measured.

The averaged mass flow rate of water in the heat exchanger coil and at the taps during the thermal discharge of 50 litres were the setting of the three way valve was 62°C is presented in Fig.5.4(a).

The flow rate in the heat exchanger at the beginning of the thermal discharge is around 7 litre/min. Approximately 1 litre/min is by-passed by the three way control valve so that the flow rate of water delivered to the taps is around 8 litre/min.

As time elapses, the store temperature decreases and as a consequence, more and more water is forced through the heat exchanger's pipe. Consequently, the mass flow rate of water delivered to the taps slightly decreases as the presssure drop in the coil is higher than the pressure drop along the by-pass of the three way valve.

It can be noticed that the flow rate of 7 l/min is well above the change from transition to turbulent flow which was identified as the optimum flow rate in the previous series of experiments.

The average flow rate in the heat exchanger's pipe and of the water delivered to the taps for various temperature setting and for a 50 litre thermal draw-off is presented in Fig.5.4(b).

As the setting point of the valve is decreased, two processes take place. Firstly the mass flow rate in the heat exchanger's pipe is reduced. Secondly the mass flow rate of water by-passed by the mixing valve is increased.

When compared with the previous results, it suggests that some improvements in the effectiveness of heat recovery can be achieved by changing the characteristics of the valve so that the flow rate of water in the heat exchanger's pipe is maintained as close as possible of the value of 4 litre/min.

The idealised characteristic of the valve is represented in Fig.5.4(a) in dash line. When the setting temperature of the idealised valve is decreased, this is first accompanied by a reduction in the flow rate of water flowing in the heat exchanger while the flow rate of water delivered at the taps remains constant.

When the setting point of the idealised valve is such that the flow rate in the coil is 4 litre/min, further reduction in the temperature of the water delivered to the taps is achieved by by-passing more and more cold water from the mains.

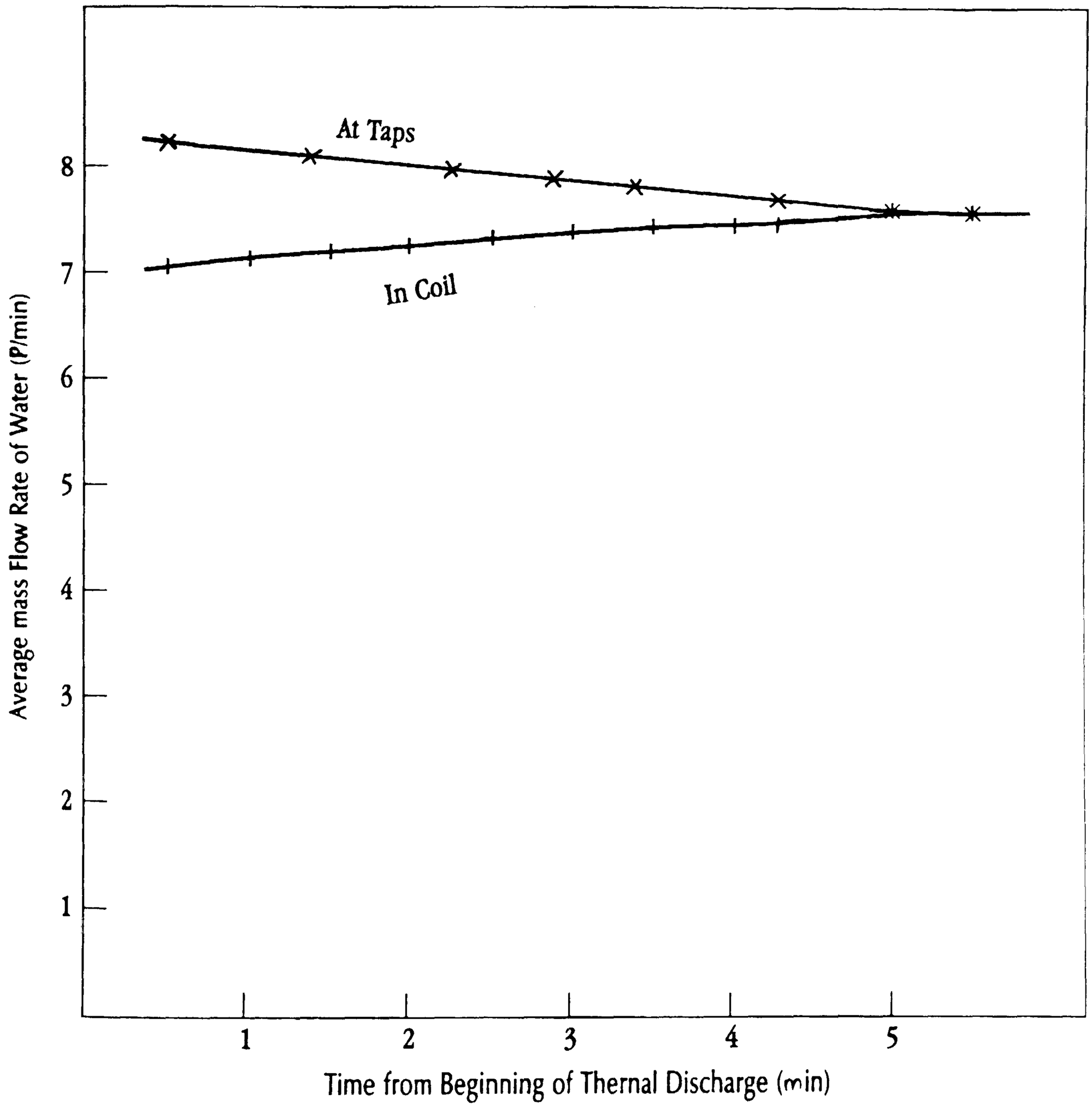


Fig.5.4(a): Variation of the Mass Flow Rate in the Heat exchanger's Pipe During the Thermal Discharge

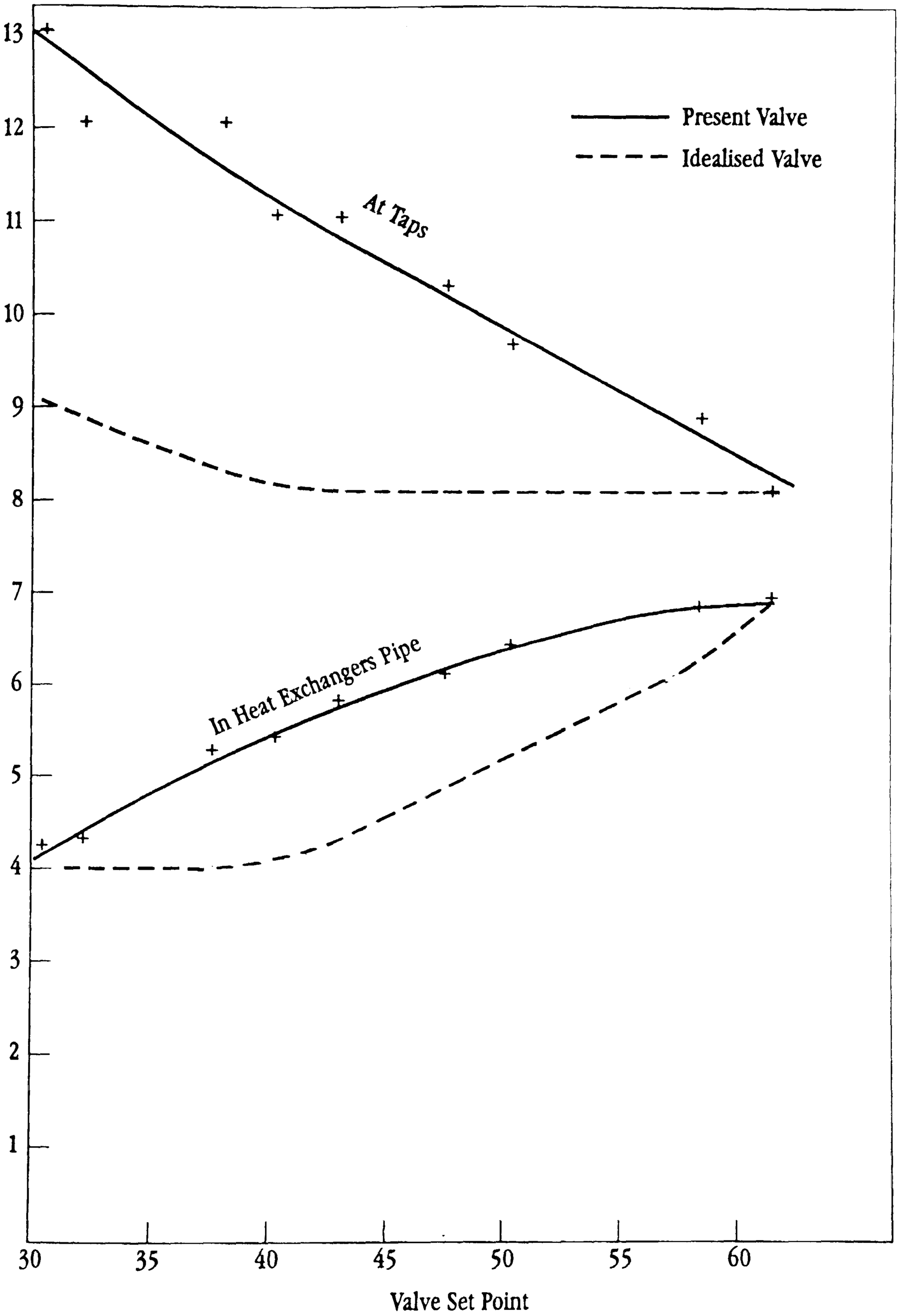


Fig.5.4(b): Effect of the Variation of the Setting temperature of the Mixing Valve on the Flow Rate of Water Passing Through the Heat Exchanger and Delivered at the Taps

The idealised valve would enable the user to achieve always the best possible performance from the thermal energy store when the setting point is reduced. Whereas with the existing valve, in order to achieve the maximum energy recovery from the store, the temperature of the water delivered to the taps has to be say 32°C which is far too low for most applications.

5.5 Heat Recovered at the End of the Thermal Discharge

The amount of heat extracted from the store after a 150 litres draw-off is on average 29.7 MJ. The amount of heat stored initially in the hot water tank at the beginning of the thermal discharge is given by the sum of the heat stored in the water and the heat stored in the mass of the tank between the temperature of 80°C and the temperature of the chilled water supply.

The use of the temperature of the chilled water supply for reference for the evaluation of the heat stored in the ITS is based entirely on theoretical considerations. It assumes that all the heat stored in the tank above the temperature of the water entering the heat exchanger can be reclaimed for useful space heating purpose. This is rarely achieved in practice and consequently the numerical value of the effectiveness of heat recovery when defined on this basis tends to be relatively low.

Finally, the corresponding values are:

water	200x4180x(80-8)	= 60.1 MJ
store	55x385x(80-8)	= 1.5 MJ
	Total	61.6 MJ

By comparing the values of 29.7 MJ and 61.6 MJ, it can be seen that even after a 150 litre draw-off, only 48.2% of the heat initially stored in the tank is recovered for domestic hot water purposes. The 51.7 % of heat remaining in the store is at an average temperature of approximately 50°C. As the temperature difference across the wall of the heat exchanger coil is approximately 15°C, the value of 50° is too low to be useful to produce domestic hot water.

In addition, as the this temperature of 50°C is also too low to be useful for space heating purposes. From this, it can be seen that at the end of the thermal discharge, although the heat content of the store is still very high, this heat at a temperature too low for most of the domestic applications.

5.6 Conclusions

The effectiveness of heat recovery from Integrated Thermal Store is maximum when the flow rate of water passing into the heat exchanger coil corresponds to a maximum in the Number of Transfer Units.

This is obtained for the minimum turbulent flow in the heat-exchanger's pipe which is approximately 4 litre/min. When the flow of water passing through the heat exchanger is set at this optimum, approximately 2.5 % (0.8 MJ) of extra heat can be recovered from the thermal store.

The design of the present mixing valve can be improved to take advantage of this optimum flow rate so that when its temperature setting is decreased, the mass flow rate of water passing into the heat exchanger's pipe is maintained as close as possible of the optimum flow rate.

References

[1] Bankston, C.A.
The Transition From Turbulent to Laminar Flow in Heated Pipes
J of Heat Transfer, Vol 92, pp 569-579, 1970

[2] Sha, R.K. and London, A.L.
Laminar Forced Convection Flow in Ducts
Academic, New York, 1978

[3] Sleicher, C.A. and Rouse, M.W.
A Convenient Correlation for Heat Transfer to Constant and Variable Property Fluids in Turbulent Pipe Flow
J Heat Mass Transfer, Vol 18, No 5, pp 677-683, 1975

Chapter 6

The Use of Baffles to Improve the Effectiveness of Hot Water Delivery

6.1 Introduction

During a thermal discharge achieved by means of the finned tube heat exchanger coil located within the Integrated Thermal Store, the overall heat transfer coefficient of the heat exchanger (U value) is mainly limited by the external heat transfer coefficient. This external heat transfer coefficient is obtained from a natural convection heat transfer process in water. It depends mainly on:

- the geometry of the heat exchanger

- the thermal properties of the convective media (water)

- the mode of operation of the heat exchanger

Many investigations on the effect of the geometry of the heat exchanger on the performance of hot water delivery have been carried out. The geometry of the heat exchanger includes its location, arrangement and orientation in the store. Some of these factors have been investigated by Mote [1]. Moreover, the design of the heat exchanger in terms of type of tubing and geometry of the fins can play an important role in the heat transfer process. The effect of the design of the fins on the heat transfer rate has partially been investigated in Chapter 3.

Neither the thermal properties of water nor the mode of operation of the heat exchanger can be changed as they are dictated by practical considerations. For example the thermal properties of water are given by the temperature at which the store operates which is fixed by several practical considerations. The mode of operation of the heat exchanger is dictated by the domestic hot water demand patterns which also cannot be changed.

There is however one way in which the external heat transfer coefficient of the heat exchanger might be increased: by increasing the velocity of the naturally occurring convection currents on the outside surface of the heat exchanger. This increase in velocity ensures that stronger convection currents take place in the the cold boundary layer developing on the heat transfer surface which facilitates the heat transfer.

This increase in velocity can be obtained by enhancing the existing buoyancy created by the cold water flowing inside the heat exchanger coil. This can be achieved by channelling the flow of cold water away from the heat exchanger by the use of a shroud or duct. This duct would be located around the heat exchanger and create a 'chimney effect'.

Additionally, as the purpose of the duct would be to keep hot and cold water separated in the store, it would avoid unnecessary mixing. It is expected that the lower mixing rate could increase the performance of heat delivery from the thermal store even further by ensuring that the heat exchanger is always surrounded by relatively warm water.

The successful use of a chimney effect to increase the effectiveness of a heat exchanger is reported by Knudsen [2] who used parallel plates to channel a flow of air around a horizontal finned circular tube in a natural convection process and reported an increase in the heat transfer coefficient of the considered finned tube of 10%.

To the contrary, Klein [3] reported a decrease in the UA value of a heat exchanger coil located within a hot water store when a 'shroud' was used to create a chimney effect. However as the methodology used by Klein to measure the UA value of the coil was not accurate, these results are dubious.

As reported earlier, the temperature profile in the Integrated Thermal Store during a thermal discharge clearly shows two separate temperature zones (Fig.3.6). The interface between the zones being approximately the top of the lower heat exchanger coil.

This temperature profile suggests that a horizontal baffle located at the interface between these two temperature zones might be beneficial. This baffle would reduce the heat and mass transfer between these zones during the thermal discharge and therefore increase the temperature difference between the two zones during the thermal discharge (stratification).

Although there is no clear evidence which suggests that increasing stratification would be beneficial, it is strongly suspected that it would have a positive effect on the effectiveness of heat recovery.

This basis for this suspicion comes from Table IV (chapter 5) which was obtained during the experiments when the flow rate of water through the heat exchanger coil was varied and which suggests that when stratification is high, the rate of heat recovery from the store is high.

Preliminary attempts to improve the performance of Integrated Thermal Stores when subjected to thermal discharge achieved by means of a horizontal axis heat exchanger coil by the use horizontal baffles with and without a vertical duct has been made by Collard [4].

Collard investigated several types of baffles arrangement for a finned tube heat exchanger located in a hot water store. Among the arrangement investigated were:

- one horizontal plate located above the bottom heat exchanger coil

- two vertical parallel plates located below the upper heat exchanger coil

- two vertical parallel plates and one horizontal plate

- a rectangular duct located around the upper heat exchanger coil and a horizontal plate.

Although several configurations of baffles were investigated for each of these arrangement, the results of this investigation were not absolutely conclusive. It was however clearly found that the use of parallel vertical plates was not beneficial. Consequently, future investigation will be limited to the use of either a duct arrangement to channel the buoyancy driven flow in the thermal store or the use of horizontal plates.

However, none of these authors [2]-[4] tried to develop simple correlations between the factors affecting the heat transfer at the wall of the heat exchanger and the presence of baffles. Thus the necessity to try to improve the fundamental knowledge of the effect of using baffles to enhance the performance of natural convection heat transfer in water based thermal stores by developing relatively simple methods of evaluating the potential improvements which can be achieved by the use of shrouds or horizontal baffles.

Apart from the claimed benefits of using baffles in enhancing the heat transfer, there are several claimed disadvantages in using long vertical baffles to create a 'chimney effect'.

One of this claimed disadvantage is that baffles made of conductive materials would behave as thermal bridges and rapidly move heat from the hot part of the store to the cold part of the store hence reducing stratification. To date, the effect of baffles in conducting heat along the store has never been quantified. Furthermore, this should be compared with other sources of heat conduction in the store such as the amount of heat conducted along the walls of the store or through the water in the store.

The second disadvantage comes from the boundary layer which develops along the solid walls of the duct. This boundary layer might reduce the velocity of the water flowing in and out of the duct therefore reducing the 'chimney effect'. This disadvantage can be avoided easily by ensuring that the cross section available for the water to flow is large enough so that the boundary layer effect is relatively small.

Finally, the cost of the baffles, do not seem to be an important problem. As the only purpose of the baffles is to channel a buoyancy driven flow, the mechanical stress on the baffles is extremely small. Consequently, baffles could:

- be made with relatively thin metal sheets which would reduce the amount of material required in the baffles arrangement (and therefore the cost of the baffles).

- be fixed in the store by extremely simple means such as currently used techniques of soft welding

In addition of these two factors, the baffles could be designed so that they can be fixed within the existing store relatively easily. Consequently, no special requirement would be needed in the manufacturing of the store and the initial investment at will be minimal.

The cost of a horizontal plate baffle located in the middle of the store was evaluated from the cost of copper, the and the extra manufacturing cost required to fix the baffle in the store. This was estimated (depending on the size of the baffle) from £6 to £12 (1989). This cost is extremely low when compared to the cost of other components of integrated thermal stores such as the heat exchanger or the thermal store (retail price £750).

6.2 Heat Conducted Vertically Along the Store

Very few data presented in the literature relate to the heat conducted along the walls of the store. It is however relatively well established that stores made of massive conductive walls are difficult to stratify.

Cole [5] evaluated the effect of the heat conducted along the walls of water based copper tanks by using the dimensionless ratio of the thermal capacity of the whole of the thermal store (water+walls) by the thermal capacity of the water contained within the store. This ratio is:

$$r = \frac{\text{heat capacity of water + heat capacity of storage tank}}{\text{heat capacity of water}}$$

$$r = \frac{M_w Cp_w + M_t Cp_t}{M_w Cp_w} \qquad (1)$$

Ideally, this ratio should be as close as possible of 1. If less than 1.02, the heat conducted along the store is small and the store is relatively easy to stratify. When this ratio is 1.1 or more the heat capacity of the walls become significant and consequently stratification is difficult to achieve.

Surprisingly, the previous work is based on the specific heat of the thermal store and of the water in the store and not on the thermal conductivity of these respective components. Therefore it is strongly suspected that any predictions based on the use of the ratio r are very empirical.

The Integrated Thermal Store used in the experiments has a ratio r of 1.024 which is relatively low. Furthermore, by adding a few kilogrammes of copper corresponding to the mass of the baffles, this ratio will most probably not move above 1.03 which is also relatively low. Therefore it is likely that stratification will be easy to achieve even when long vertical baffles are fixed in the store.

However, the empirical approach used by Cole [4] does not give any useful information with respect to the amount of heat conducted along the store. It was therefore necessary to at least try to approximate this amount of heat. For this purpose, a simple theory was developed based on a one dimensional analogy with a solid conductive rod.

6.2.1 Experimental Apparatus and Methodology

In the experiment, a heater was positioned at the top of the store so that only a thin layer of water was above it. At time t=0, the store was initially at room temperature and the heater was switched on. As the output from the heater was relatively high, the temperature of the thin layer of water increased quickly up to 80°C. At this stage a rod thermostat sensing the temperature of the upper layer of water switched off the heater.

Heat from the upper layer of hot water is then conducted by the walls of the store and by the water and its temperature starts to decrease. When its temperature falls below 76°C, the rod thermostat sensed the temperature drop and switched on the heater again until the temperature of the water situated above the heater reaches 80°C. At this temperature, the heater was switched off again.

Finally, by cycling on the thermostat of the heater, it is possible to maintain the temperature of the upper layer of water between 76 and 80°C. The variations of the temperature of the water along the axis of the tank with time were then caused entirely by the heat conducted axially from the upper layer of water.

Using the thermojunctions located in the thermal store, it was possible to measure the temperature along the axis of the store with time during the 'experiment'.

It was first observed that the temperature profile at a given time was relatively independent of the radial location of the thermojunctions in the store. This suggested that the heat flow in the store during the experiment was mainly in the axial direction.

Some temperature profiles are represented in Fig 6.2.1 for t=0 (beginning of the experiment), t=24 hours and t=47 hours. They show that as expected, the top of the store always remains at a high temperature. The temperature gradient is more marked at the top and tends to decrease when moving towards the bottom of the store. The temperature at the bottom of the store, tends to be close to the temperature of the environment.

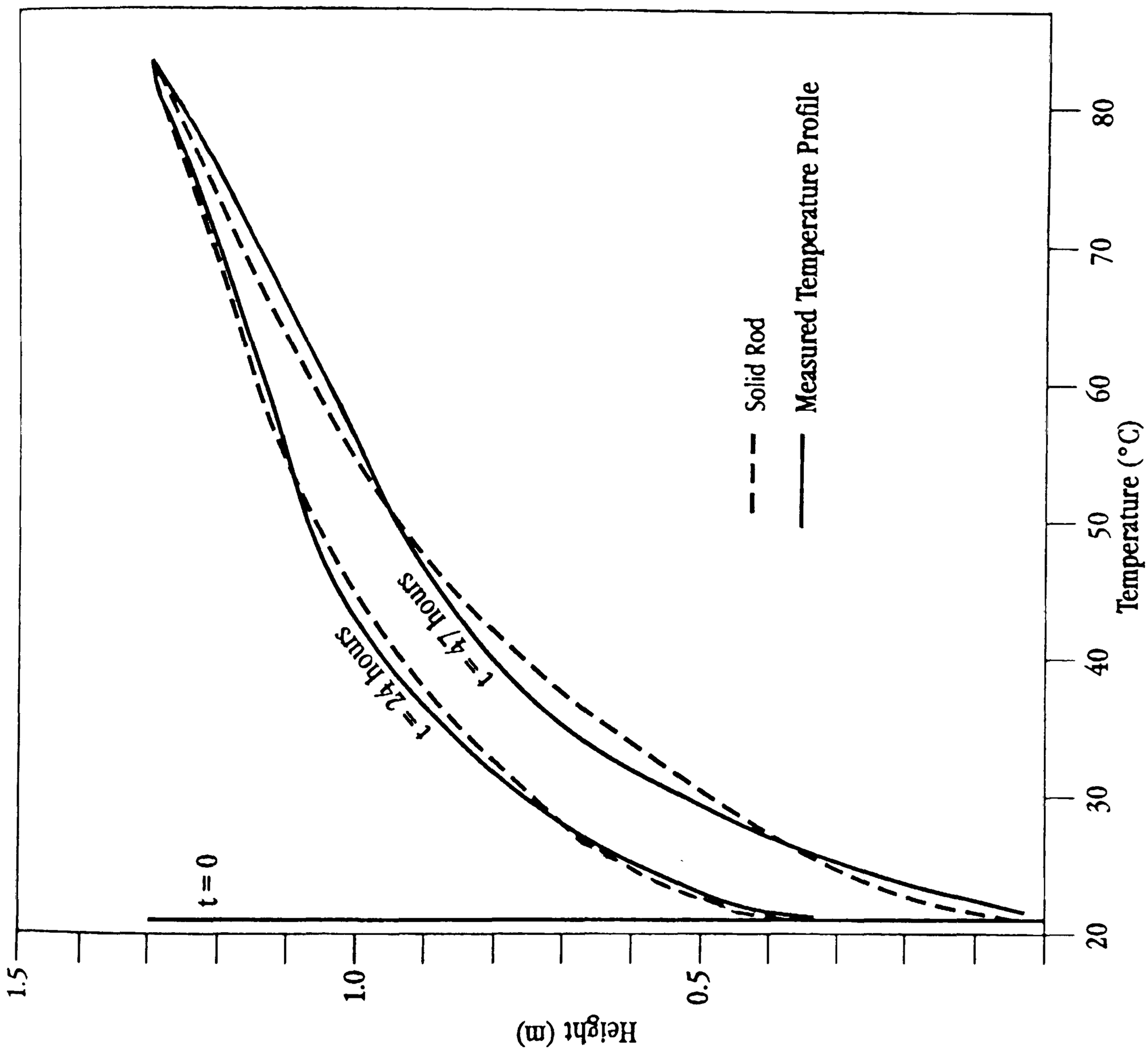


Fig.6.2.1: Measured and Predicted Temperature Profile in Hot Water Store and Solid Rod

6.2.2 Analogy with a Solid Rod

The analogy with a solid rod is represented in Fig.6.2.2. The rod is initially isothermal at a temperature T_i. At time t=0, the upper horizontal surface of the rod is suddenly set to a constant temperature of $T_{hot} > T_i$.

Heat is transferred along the rod by the process of conduction only and the heat loss to the environment is given by a coefficient h which varies with height. The equation describing the conduction of heat along the rod assuming a one dimensional heat flow is:

$$k\frac{\partial^2 T}{\partial x^2} + e\frac{p}{s}(T_e - T) = \rho Cp\frac{\partial T}{\partial t} \qquad (2)$$

The first term of the left hand side of equation (2) corresponds to the variation of temperature along the axis of the rod induced by the heat transfer by conduction within the rod.

The second term of the left hand side corresponds to the heat lost from the wall of the rod to the surrounding environment. The use of this term is necessary to closely match the computer prediction with the behaviour of the thermal store.

The term on the right hand side corresponds to the change in internal energy of the rod due to the transient heat transfer process.

To form a complete system of equations, the boundary conditions corresponding to this type of heat transfer should be considered. These conditions are necessary to define the state of the system at the initial time and at its physical boundaries. These boundary conditions are:

- at t = 0 : the rod is isothermal and at temperature T_{in}

- for t > 0 : top of the rod $T=T_{hot}$
  bottom of the rod adiabatic
  wall of the rod: heat loss to the environment

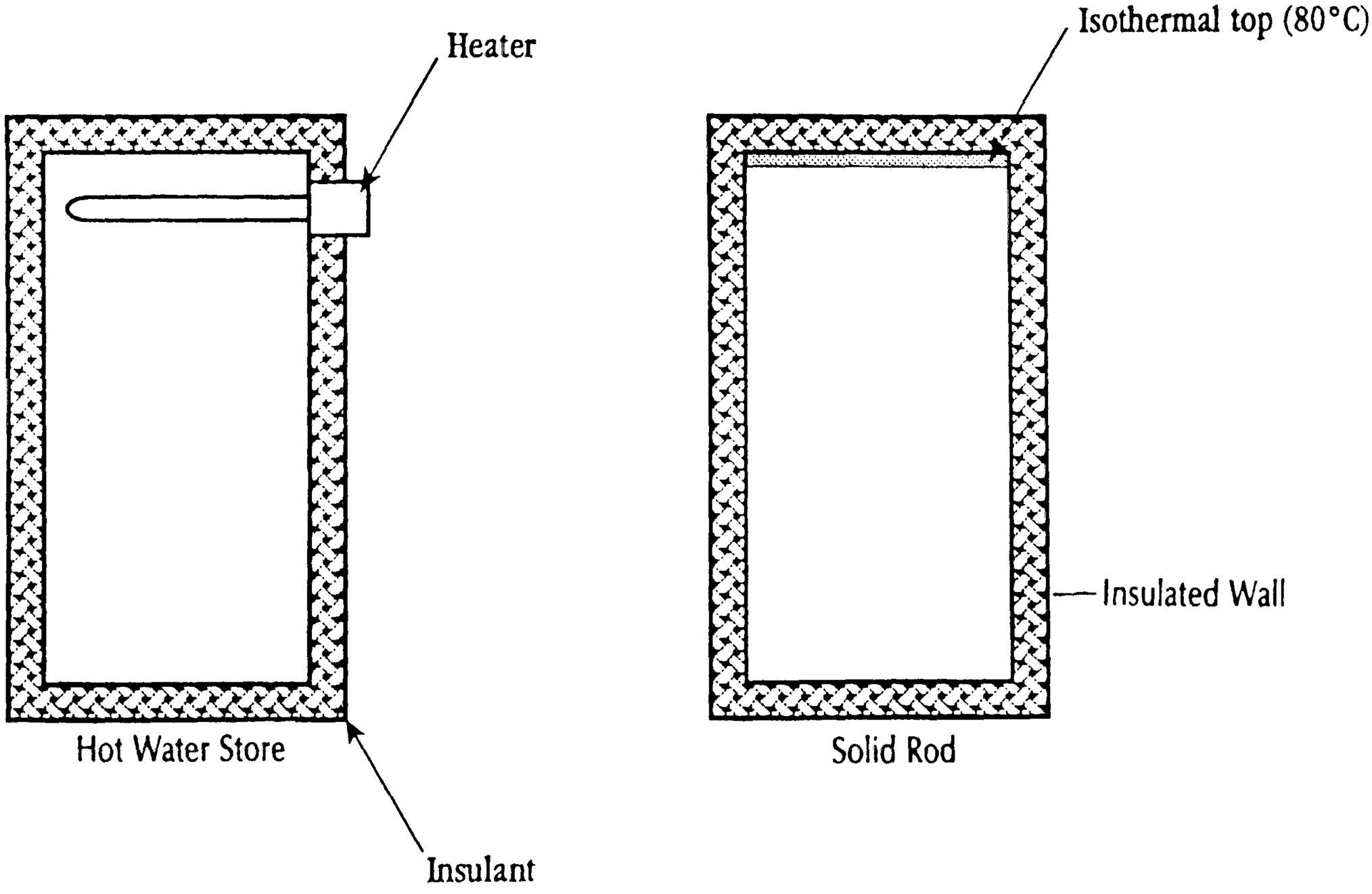


Fig. 6.2.2. Analogy Between Thermal Store and Solid Rod

6.2.3 Method of Solution

Equation (2) was solved numerically using an implicit finite difference formulation. In this formulation, the rod is assumed to be divided in slabs and the partial derivatives appearing in equation (2) are replaced by a system of linear equations in which the temperature in the slabs are the unknowns.

The implicit finite difference formulation leads to a system of n equations and n unknowns which was solved for each time step using a Gauss elimination method. Although this requires more computation than an explicit formulation, it avoids stability problems in the numerical solution.

The nodes were numberred in increasing numbers starting from the bottom of the rod. For a node i in the middle of the rod, equation (2) is finally replaced by:

$$k\frac{T_{i-1}-2T_i+T_{i+1}}{\Delta x^2}+\frac{ep}{S}(T_e-T_i)=mCp\,\frac{T_i-T^t{}_i}{\Delta t} \qquad (3)$$

The boundary conditions required to complete the system of equations are:

- at time $t=0$ $T_i=T_{in}$

- for time $t>0$: top of the rod $T_n=T_{hot}$

  bottom of the rod $T_1=T_2$

The heat transfer coefficient (e) from the rod to the environment was evaluated using equation (5) which was developed in Chapter 2 and which gives the UA value of the walls of the store with temperature.

The rod was divided in 42 nodes. The value of 42 was chosen so that the thermojunctions used in the experiments were situated at the exact locations of the nodes in the finite difference mesh.

The time step was 30 seconds. This was chosen as a balance between the computation time and the accuracy of the solution. Although when the time step was increased to 60 seconds, no difference was observed in the numerical predictions.

By varying the thermal conductivity of the rod in equation (2), different temperature profiles in the solid rod could be obtained. This thermal conductivity was varied numerically until the temperature profile in the rod and the temperature profile experimentally measured in the hot water store exactly coincide.

The best results were obtained for a value of k=2.7 $Wm^{-1}K^{-1}$. The temperature profile for the thermal rod is also represented in Fig.6.2(a).

Although the very complex heat transfer process taking place in the hot water store are more complicated than the simplified solid rod would suggest, the value of k can be regarded as the 'apparent thermal conductivity' of the store. Additionally, the the numerical value of k=2.7 $Wm^{-1}K^{-1}$ can be used as a reference for future measurements of the 'apparent thermal conductivity'.

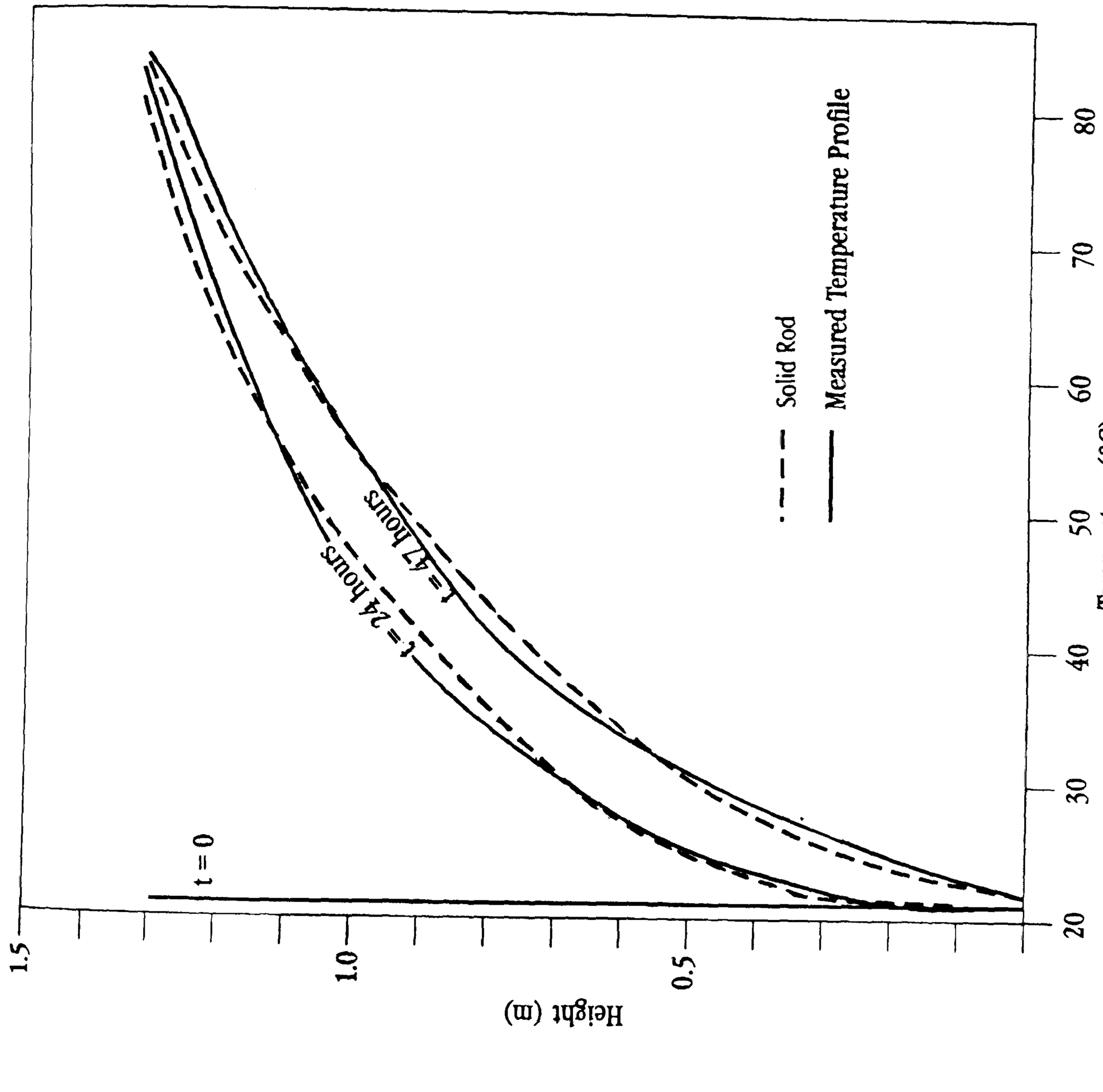
6.3 Heat Conducted Along Baffles

In order to evaluate the effect of baffles on the amount of heat conducted along the tank, a second experiment was carried out. For this second experiment, 23 copper tube 8 mm in diameter and 1.4 m in length were inserted vertically in the tank. The tubes were situated as much as possible in an even distribution in the store. The total mass of the tubes was 3.3 kg.

The purpose of the tubes was to increase the amount of heat conducted along the axis of the store. The number of tubes was chosen to try to simulate the presence of a vertical baffles made of conductive material located in the store.

The same experiment as was previously described to determined the 'apparent thermal conductivity' of the store was repeated. Again the conductivity of the rod was varied numerically by the use of the computer model to match the temperature profile in the thermal store.

The best prediction were obtained for a value of the thermal conductivity of the rod of 3.4 $Wm^{-1}K^{-1}$. The experimental results with the computer predictions are represented in Fig.6.3



Predicted Temperature Profiles With Copper Tubes

Fig.6.3: Measured and P

6.4 Discussion

The difference in 'apparent thermal conductivity' of the store measured with and without the copper tubes is small. Therefore if baffles are used, the amount of heat conducted along these baffles would not significantly change the stratification in the store during a domestic hot water draw-off, even if the baffles are made of a conductive material and are oriented in a vertical manner. Thus any type of baffles arrangement or orientation to enhance the performance of domestic hot water delivery could be investigated.

Correlations in terms of apparent thermal conductivity were not developed as only two experiments were carried. A much larger experimental basis obtained with different size of thermal store and operating in different conditions would be required to develop accurate correlations.

However, the theoretical limiting values for the apparent thermal conductivity can easily be found. If the store were extremely large, the amount of heat conducted along its walls and the piping of the heat exchanger would be negligible and consequently, the apparent thermal conductivity of the store would tend to be close to the thermal conductivity of water (i.e.0.6 $Wm^{-1}K^{-1}$)

Inversely, the maximum value which can be reached for the apparent thermal conductivity of the store would be obtained when the walls of the store and the piping play the major role in conducting heat. In this case, the apparent thermal conductivity of the store would tend to to be close to the thermal conductivity of copper (i.e. 380 $Wm^{-1}K^{-1}$).

At the moment, the apparent thermal conductivity of the store is already one order of magnitude higher than the thermal conductivity of water. This is entirely caused by the heat conducted along the walls of the store and the piping of the heat exchanger and cannot be avoided without significant change in the design of the store.

This might present disadvantages if long term heat storage is considered (the order of a few days or more). In this case, the heat conducted along the store and vertical baffles, if any, might influence significantly the naturally occurring stratification by conducting heat from the bottom part of the store to the top part. In this case, the presence of long vertical baffles made of conductive material would not be beneficial.

6.5 Baffles Arrangement Investigated

The second stage was to investigate several types of baffles arrangement in order to evaluate their effect on the performance of hot water delivery. The baffles arrangement investigated are presented in Fig.6.5.

The first arrangement was made of a horizontal baffle located just above the lower heat exchanger coil. The aim of this baffle was to reduce mixing between the top and bottom zones in the store during the thermal discharge. It was expected that a reduction in the mixing would lead to an increase in stratification thus improving the effectiveness of heat delivery from the store.

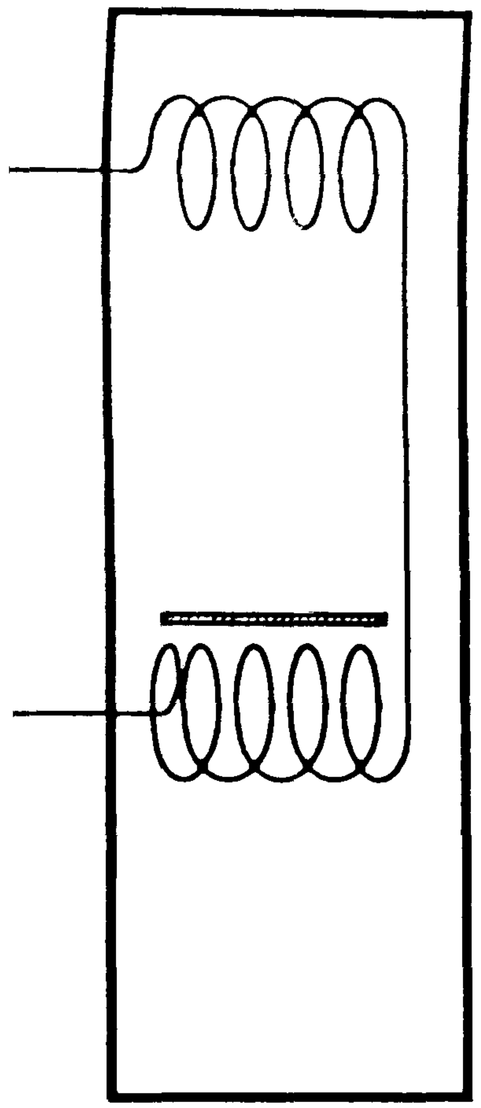
The second and third arrangements were made with small vertical baffles located close to the upper heat exchanger coil. These aimed at improving the flow around the heat exchanger during the thermal discharge by creating a recirculation effect. The recirculation effect would enhance the heat transfer rate at the wall of the heat exchanger, again increasing the rate of heat delivery from the store.

The last arrangement was made with a vertical duct situated around the upper coil. In this arrangement, a horizontal baffle was also used, again situated just above the lower heat exchanger coil. The aim of this arrangement was to increase the recirculation effect even further by carefully channelling the cold water away from the upper heat exchanger coil.

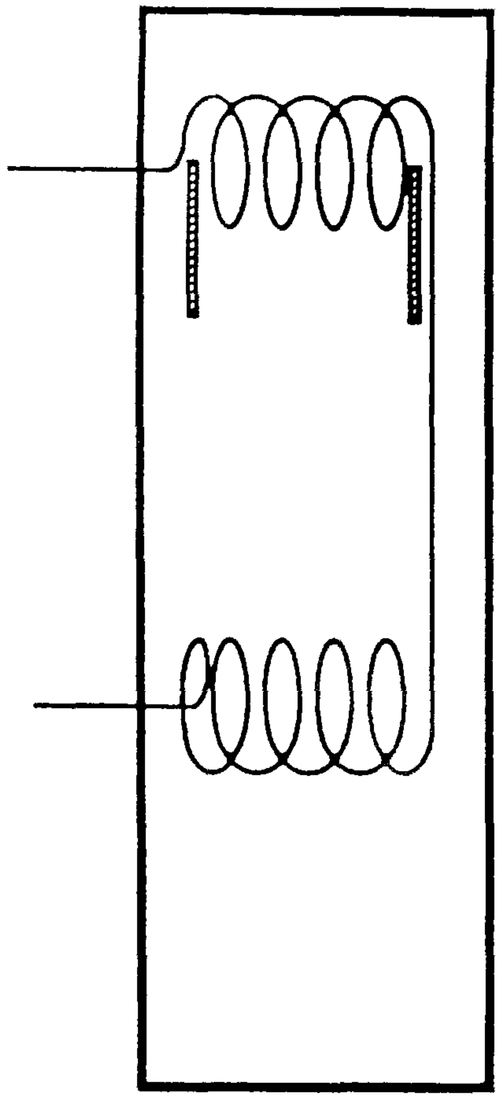
6.6 Experimental Apparatus and Methodology

Although in a real situation the baffles would be made of copper, the use of thin aluminum sheet was more convenient for experimental purposes. The aluminum sheet were 0.6 mm thick. The combination of the use of aluminum with a low thickness reduced the amount of heat conducted along the baffles and the thermal mass of the baffles to negligible values.

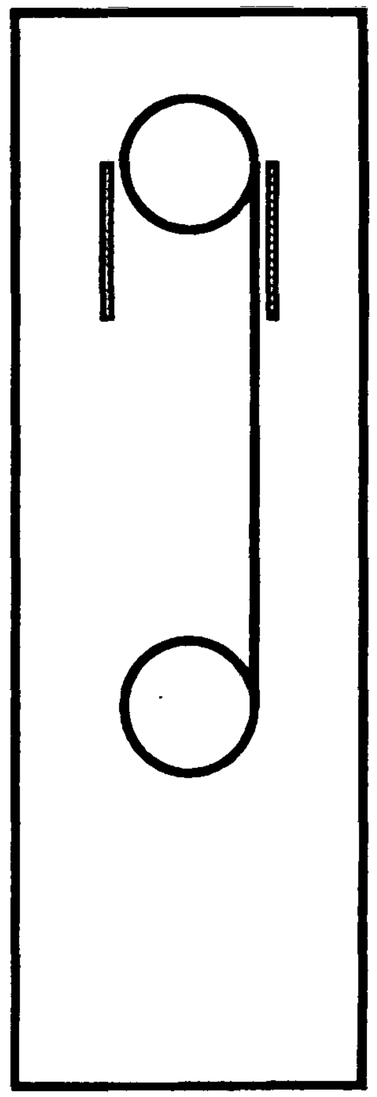
The baffles were usually made of several parts which were first fixed into the thermal store. The parts were then joined together using aluminum adhesive tape so that any leakage at the joints were avoided.



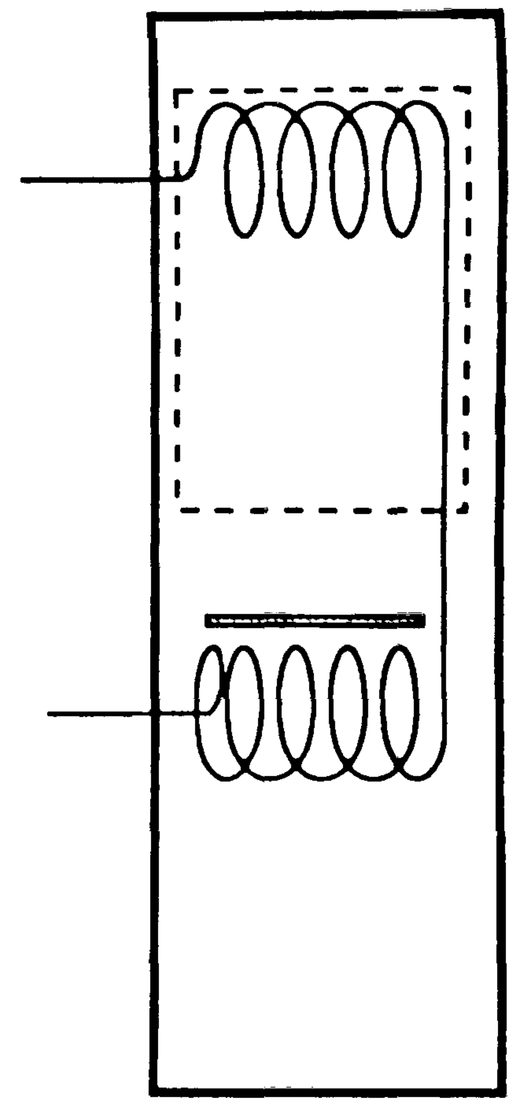
Horizontal Baffle
(0.3 x 0.3m)

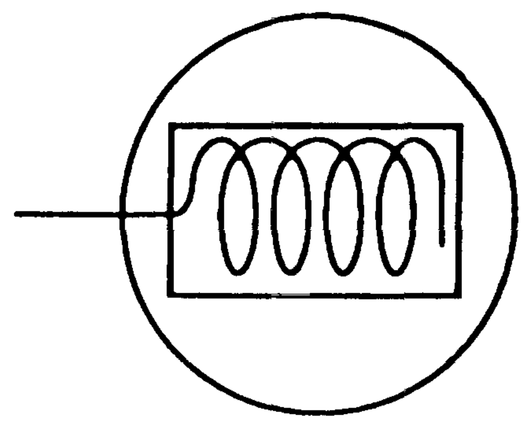


Vertical Baffle
Perpendicular to Upper Coil Axis
(0.2 x 0.3m)



Vertical Baffle
Parallel to Upper Coil Axis
(0.3 x 0.3m)





Rectangular Duct
(0.25 x 0.3 x 0.6m)
+ Horizontal Baffle
(0.3 x 0.3m)

Fig.6.5: Types of Baffles Arrangement Investigated

The store was then heated to 80°C. At the end of the heating period, a constant flow rate of cold water from the chilled water supply was passed into the heat exchanger coil for a duration of approximately 25 minutes. The flow rate of water in the heat exchanger was kept relatively constant at around 8 litre/min for the whole duration of the thermal discharge.

At the end of the test, the baffles were removed from the store and a second thermal discharge achieved in the same operating condition. The role of the second draw-off was to provide data for use as reference.

All the instrumentation used during the test is described in details in Chapter 2. The thermocouples and PRTs were scanned every 10 second during most of the experiments. However, this scanning rate was to be increased up to 1 scan per second in some experiments to evaluate the velocity of the water in the thermal store. This was achieved without loss in accuracy by scanning only the thermocouples which were required to evaluate these velocities.

6.7 Experimental Results

The effectiveness of the baffles could be measured in several ways. The most relevant is however related to the amount of heat extracted from the thermal store by the water flowing into the heat exchanger after a given mass of chilled water has passed into the heat exchanger.

The amount of heat extracted after a 50, 100 and 150 litres of cold water have passed into the heat exchanger coil is summarized in Table V.

It can be seen from these results that no significant improvement in performance was observed. However, the measurements suggest that the presence of baffles is not beneficial and reduces the effectiveness of heat delivery.

Only in the case of the horizontal baffle, some improvement is observed. However, this is improvement is only 0.06 % of the heat extracted from the store. This is well below the experimental error with which the amount of heat is measured and can not be regarded as conclusive.

Experiment Number	50 litres (MJ)	100 litres (MJ)	150 litres (MJ)
1	10.852	20.486	27.962
2	10.541	20.098	27.979
3	10.628	20.231	27.743
4	11.097	20.468	27.876
5	10.919	20.132	27.428

1 - no baffles
2 - horizontal baffle
3 - vertical baffles, perpendicular to coil axis
4 - vertical baffles, parallel to coil axis
5 - rectangular duct, with horizontal baffle

Table V: Energy Recovered With and Without Baffle Arrangements

6.8 Data Analysis

6.8.1 Thermal Discharge Achieved with Vertical and Horizontal Baffles

6.8.1.1 Thermal Discharge Without Baffle

When the chilled water passed inside the heat exchanger's pipe, the velocity of the water inside the pipe was approximately 0.7 ms^{-1}. The corresponding Reynolds number based on the internal pipe diameter and on the mean bulk temperature of the water flowing inside the pipe was approximately 11,000.

This value of Reynolds number corresponded to a turbulent regime in the coil of the pipe which gave a relatively high internal heat transfer coefficient. As a result of this high internal heat transfer coefficient, the main limitation in the overall heat transfer coefficient of the heat exchanger (U value) came from the natural convective heat transfer on the outside surface of the finned heat exchanger coil.

In addition, as the heat exchanger had a relatively high length/internal diameter ratio, the local turbulence created at the entrance region of the heat exchanger had negligible effect on the overall rate of heat transfer.

At the beginning of the thermal discharge, as soon as the chilled water was passed into the heat exchanger, the temperature of the water in the tank surrounding the lower part of the heat exchanger decreased. This difference in temperature induced a difference in density in the water in the store and generated strong buoyancy driven convection flows in the store.

The velocity of these buoyancy driven convection flows could be evaluated by measuring rate at which the initial front of cold of water below the coils moved downwards in the store at the initial time. Below the upper coil, this cold front moved of approximately 0.33 m in 15 seconds giving an average velocity of approximately 0.022 ms^{-1}. In the lower part of the store the cold front moved by approximately 0.2 m in 6 seconds giving a velocity of approximately 0.033 ms^{-1} The higher velocity in the bottom region of the store was attributed to a higher buoyancy force induced by a higher temperature difference between the water flowing in the heat exchanger coil and the water in the thermal store.

Also, at the bottom coils, the temperature of the water flowing in the heat exchanger came directly from the chilled water supply and entered the coils at a lower temperature. Consequently, the rate of heat transfer at the wall of the finned tube was the higher. Hence more heat was extracted from the bottom region of the tank. This created a large temperature difference between the top and the bottom of the store. The temperature profile in the store is represented with time in Fig 3.6.

In the lower part of the store (h<500 mm), the temperature of the water in the store was very nearly uniform and significantly lower than the temperature of the water above the 500 mm height which was also relatively uniform. The temperature difference between the two parts of the store varied during the thermal discharge with time from 0'C at the beginning of the experiment to 16'C after 15 minutes.

At approximately 500 mm above the bottom of the store, there was a sharp temperature gradient which corresponded approximately to 50 mm above the upper part of the lower heat exchanger. This temperature gradient also varied during the thermal discharge from 0 at the beginning of the thermal discharge to approximately 400 $^{\circ}Cm^{-1}$.

6.8.1.2 Vertical Parallel Plates

The vertical parallel plates were located close to the upper heat exchanger. These plates aimed at channelling the flow of cold water away from the heat exchanger and to create a recirculation motion as presented in Fig.6.8.1.2.

experimental observations

a) The vertical plate baffles did not channel all the cold water coming down from the top heat exchanger. Some of this cold water was flowing in the outside region of the plates which at the same time reduced the recirculation effect and created mixing between the hot water flowing upwards in the outer region of the plates and the cold water which was not channelled by the plates flowing downwards in again in the outside of the plates. In addition, some of the cold water flowing in down between the plates tend to escape from the open sides which again created mixing and reduced the recirculation effect.

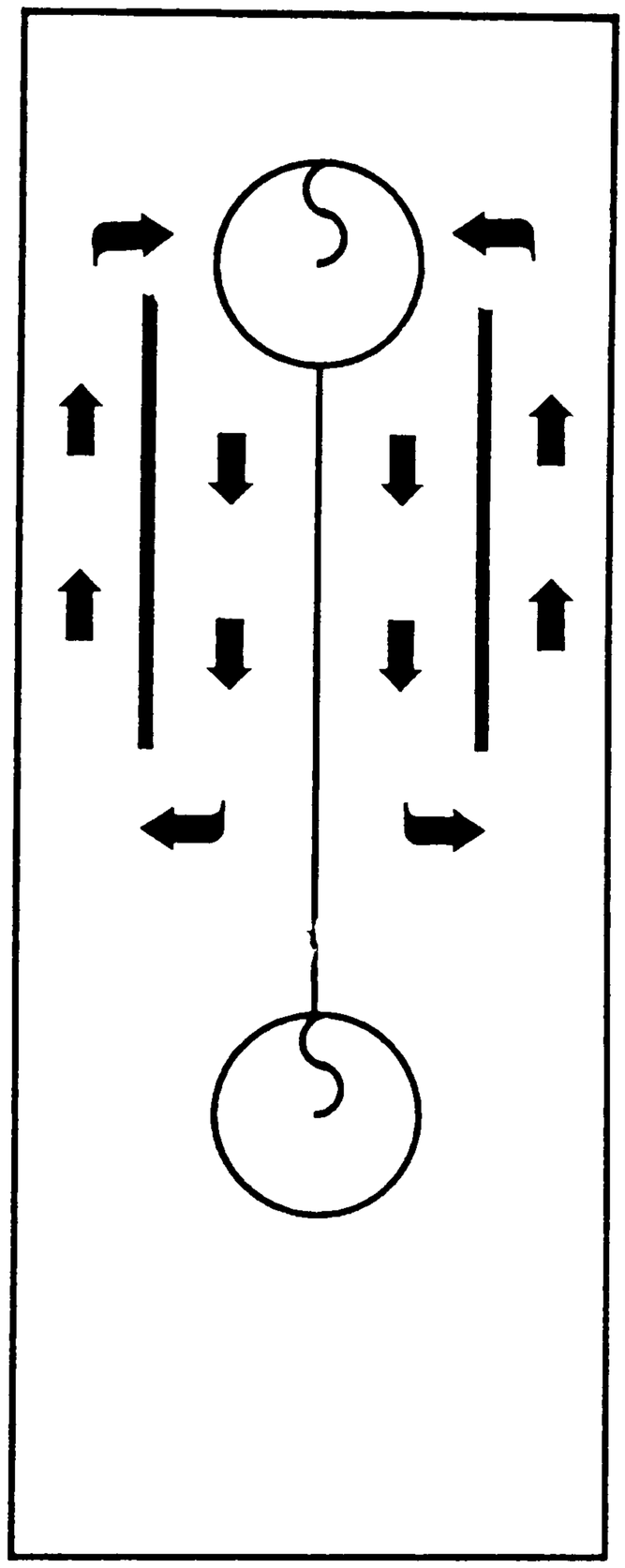


Fig.6.8.1.2: Schematic of the Recirculation Motion Induced by the Vertical Plates

b) The presence of the baffles tends to slow the recirculating process between the cold water moving down from the top of the store and the hot water coming up from the bottom of the store. This reduced the temperature difference between the wall of the heat exchanger and the water in the upper part of the store by keeping the upper heat exchanger surrounded by cold water. This lower temperature difference at the upper heat exchanger reduced the heat transfer rate through the wall of the heat exchanger and therefore the rate of heat delivery from the store.

c) As a result of this mixing taking place in the store and the reduction of the recirculation effect, the velocity of the water flowing onto the surface of the heat exchanger was reduced when compared with the velocity of the water without plates. This lower velocity reduced the convective external heat transfer coefficient at the wall of the upper heat exchanger which made the heat transfer process more difficult.

d) The mean temperature of the water in the tank was higher with the baffles than without the baffles which further suggested that less heat was recovered with the plates than without the plates. This is confirmed by the lower amount of heat recovered at the end of the experiment when compared with the experiment without baffles.

e) Finally, experimental measurements showed that the temperature of the water above the bottom heat exchanger and within the plates decreased before the cold water flowing into the upper heat exchanger had reached the top coils.

This suggested that the turbulence created by the bottom coil went far above the location of this coil and therefore a significant amount of heat and mass transfer was taking place between the top and bottom temperature zones in the store. For this reason, it seemed important to investigate the design of a baffle to reduce the heat and mass transfer between the upper and the lower part of the store.

6.8.1.3 Horizontal Plate

The horizontal plate was located in the middle of the store near to the inflexion point of the temperature gradient. several sizes and location of baffles were investigated. The horizontal plate had two aims:

- reducing the heat and mass transfer between the upper and lower temperature zones in the store to increase stratification.

- creating a recirculation motion in the lower part of the store as represented in Fig.6.1.8.3(a). This recirculation motion would avoid unnecessary mixing thus increasing stratification in the store.

Two parameters were investigated:

- the width of the baffles

- the height of the baffle in the store

Experimental observations

The mean temperature of the water in the store just above the baffle in the horizontal plate experiment and at the same location in the reference experiment is presented in Fig.6.8.1.3(b). The temperature is higher with the horizontal baffle present which shows the effect of the baffle in reducing mixing between the top and the bottom temperature zones in the store.

In addition, no decrease in temperature above the plate was observed at the beginning of the experiment. This suggests that the plate eliminated the upward cold water flow from the bottom of the tank at the very beginning of the experiment which further reduced mixing between the top and bottom temperature zones in the tank.

As a result of this reduced mixing, there was a slightly better stratification when the baffles were present. This slight increase in stratification increased the effectiveness of heat delivery from the store.

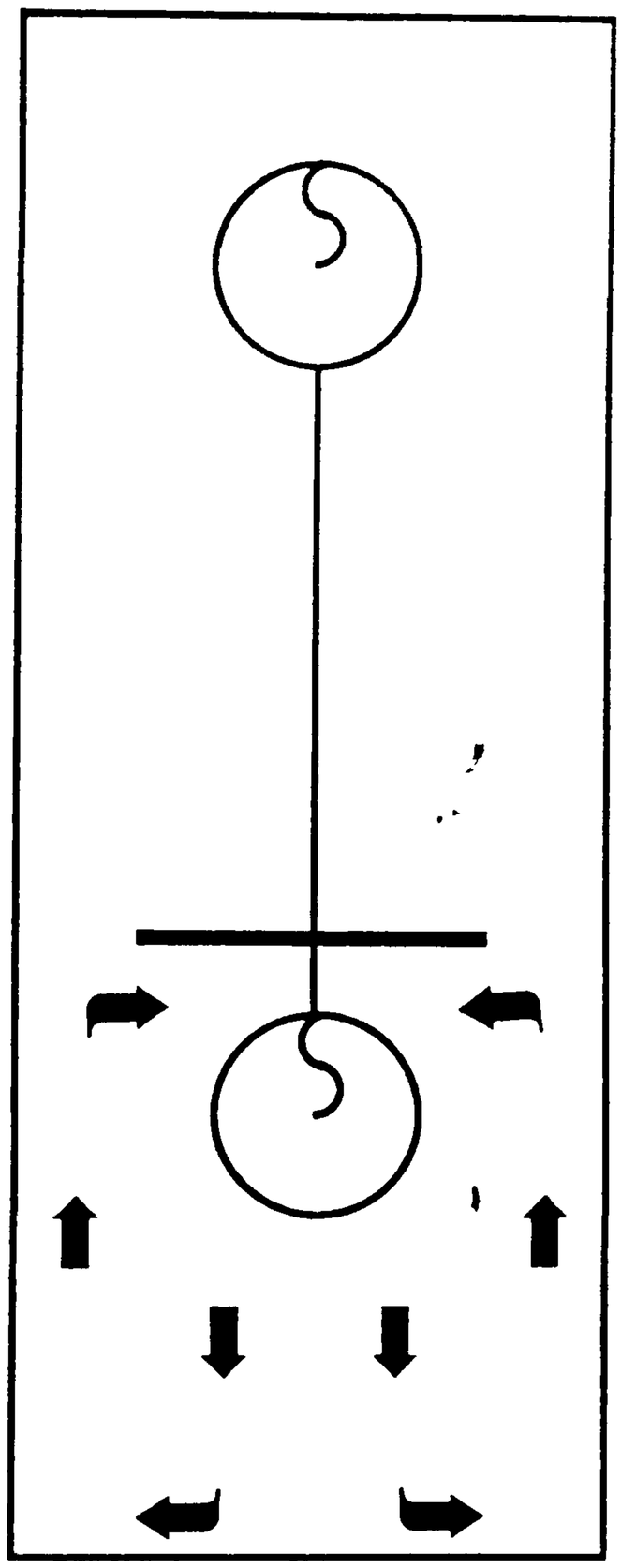


Fig 6.8.1.3(a): Schematic of the Recirculation Motion Induced by the Horizontal Plate

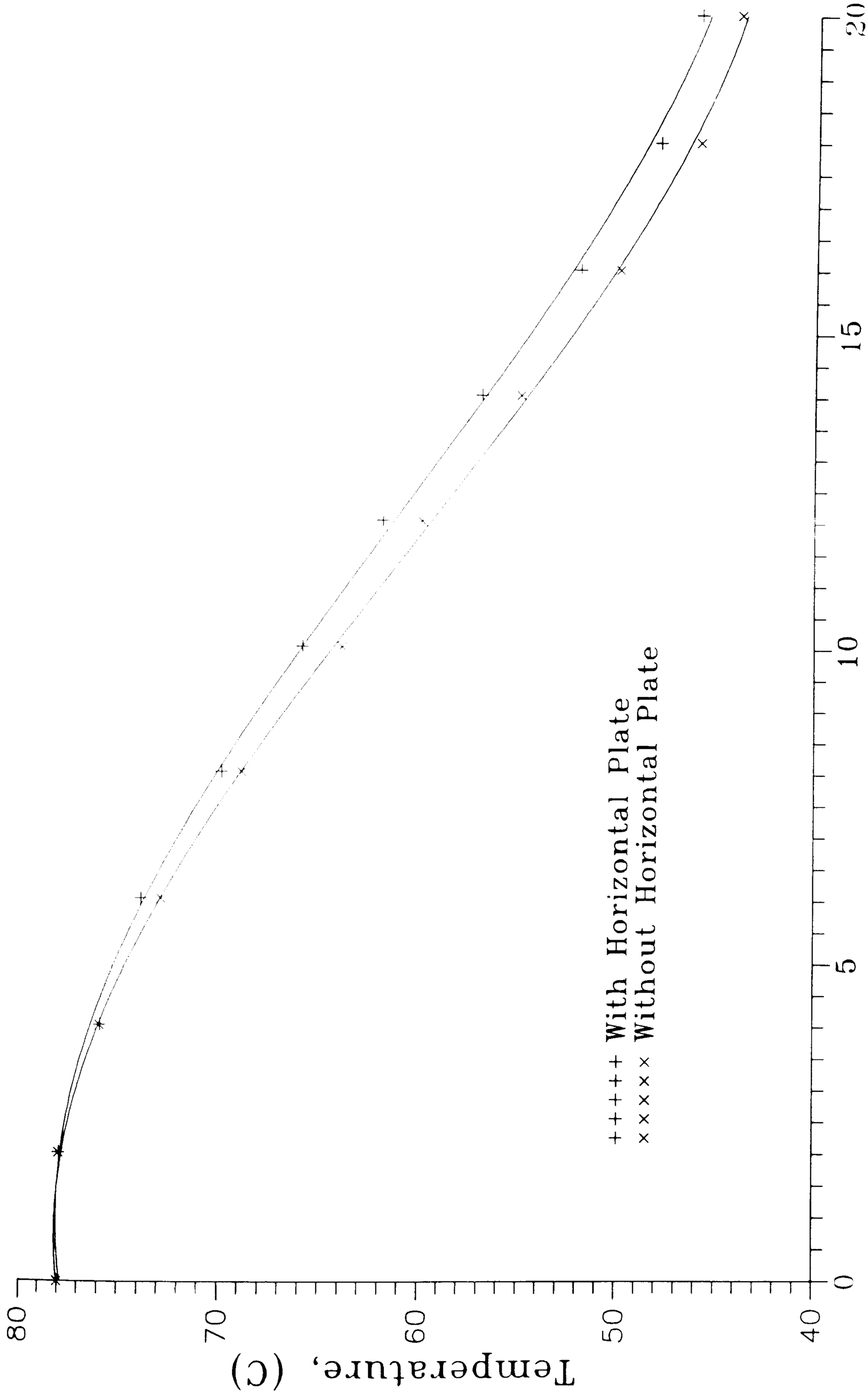


Fig 6.8.1.3(b): Temperature of the water in the store above the Horizontal Plate

The best results were obtained with a horizontal plate 300x220 mm located at 520 mm from the base of the tank. The optimum height of the baffles above the bottom of the store was found to be the 20 mm above the height where the temperature gradient was the sharpest in the absence of baffle.

When the plate was located closer to the bottom of the tank, the volume of cold water could not be confined in the bottom zone of the store and therefore it leaked significantly in the upper zone. This created mixing between the cold water and the hot water above the horizontal plate possible which reduced the effectiveness of the horizontal plate..

Inversely, if the plate was located too high in the store, too much water was confined in the bottom zone. In this case, the mixing was allowed to take place between the cold water coming from the bottom heat exchanger and the hot water trapped below the horizontal plate. This again reduced the effectiveness of the horizontal plate.

Discussion

In the absence of baffles, the steepest temperature gradient is situated just above the lower heat exchanger coil in the store. At this very location, heat is transferred from the upper zone to the lower zone by the process of conduction, convection and mixing.

When the horizontal baffles was inserted in the store, unlike what was expected, only a slight change in the temperature gradient between the top and bottom temperature zones of the store was observed.

However, as suggested by the experimental measurements, the presence of the baffles reduces the heat transfer between the upper and lower zones by the above described processes.

This suggests that the heat transferred between the zones is already small without baffles and is not the prime factor affecting the temperature in the upper and the lower zones during the thermal discharge.

The temperature in both zones is dictated by other factors among which are the volume of the zone, the amount of heat extracted by the heat exchanger, The UA value of the heat exchanger and its dependence on temperature.

Thus stratification can be also improved by other means than using a horizontal baffle. The simplest way of achieving a better stratification is to change the location of the lower

heat exchanger in the store. Other means to change the stratification are more difficult to put into practice. These include changing the design or arrangement of the heat exchanger or of the store.

6.8.2 Rectangular Duct

A lot of attention was paid to the experiment carried out with a rectangular duct. The duct and baffles arrangement is presented in Fig.6.8.2(a) with the location of the thermocouples which were changed for the purpose of this experiment.

The size of the horizontal baffle was 300x220 mm. The size of the rectangular duct was 250x300x600 mm. It was located around the upper heat exchanger coil and so that the upper edge of the duct was exactly at the same height as the upper surface of the heat exchanger.

At this stage, the velocity field generated by the cold water passing into the upper heat exchanger coil in the store is not known. Several possibility can be envisaged for this flow, however, the simplest assumption is as represented in Fig.6.8.2(b)

Cold water flows downwards in the duct till it reaches the bottom of the upper zone. Then due to the presence of the horizontal baffle, the flow deviates and is redirected upwards on the outside of the duct.

At the top of the duct, water is entrained by the buoyancy driven flow created by the cold water passing into the heat exchanger coil. This induces the water first on the top of the duct and then inside the duct where it flows downwards.

A careful examination of the temperature field in the store during the thermal discharge showed two important things:

a) Fig.6.8.2(c) represents the difference in temperature between the inside and the outside of the duct at approximately mid-height of the duct during the thermal discharge. The average temperature difference between the water in the duct and the water out of the duct was around 1.5^{o}C during the experiments. The warmer water was on the outside of the rectangular duct.

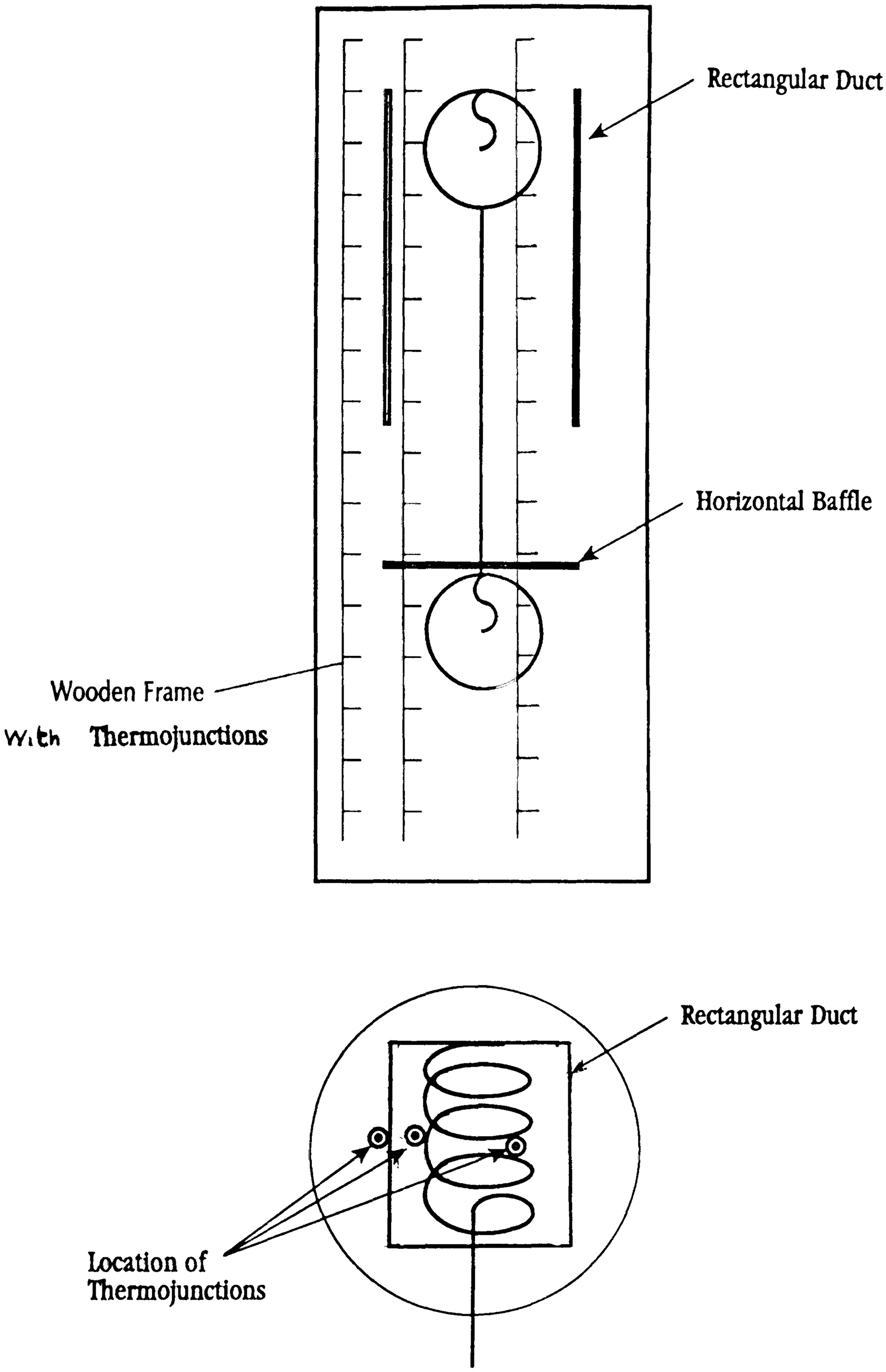


Fig. 6.8.2(a) Rectangular Duct and Thermojunctions Locations

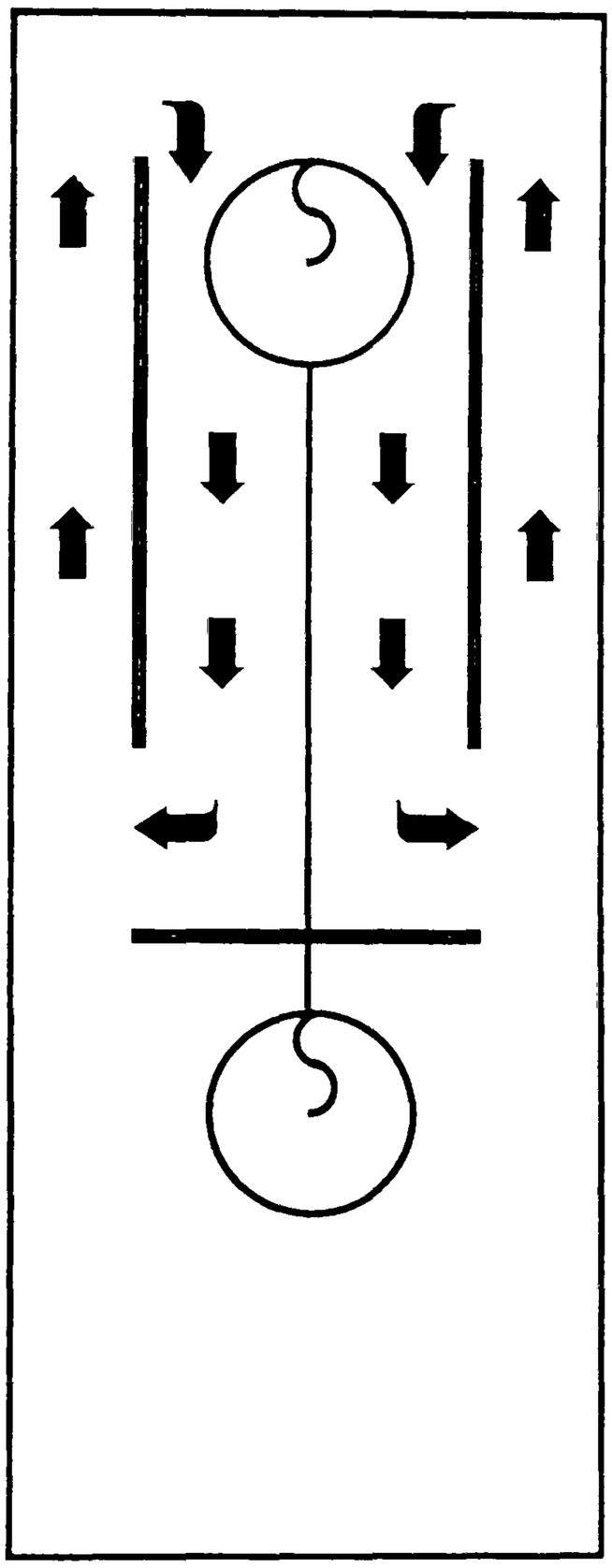


Fig.6.8.2(b) Assumed Buoyancy Driven Flow in the Integrated Thermal Store during the Thermal Discharge

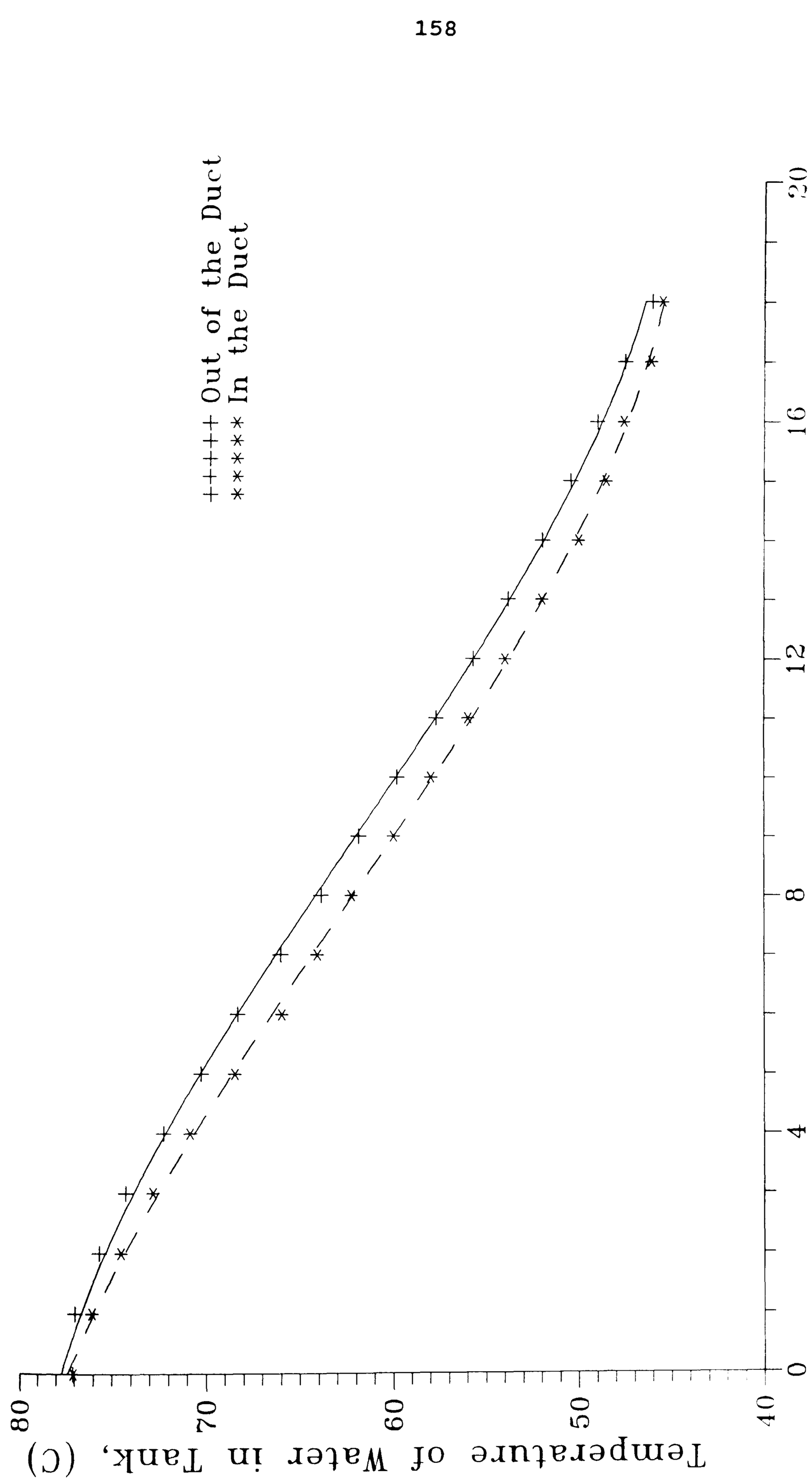


Fig.6.2.8(c): Average Temperature of the Water Flowing inside and outside of the Duct during the Thermal Discharge

b) Fig.6.8.2(d) represents the variation of the average temperature of the water surrounding the upper heat exchanger coil with and without a duct and with time. The temperature of the water surrounding the upper heat exchanger was on average $2^{\circ}C$ lower with a duct than without a duct.

6.8.3 Evaluation of the Velocity of the Water Flowing in the and around the Rectangular Duct

The increase in velocity of the water close to the heat exchanger coil due to the presence of the duct can be estimated by balancing the buoyancy force created by the cold water from the heat exchanger with the friction and inertia forces created by the water flowing in and around the duct.

The buoyancy force induces a pressure lift dictated by the difference in temperature between the cold water in the duct and the relatively warm water outside of the duct. This pressure lift is given by:

$$dP = \rho g \beta dTL \qquad (4)$$

This pressure lift is entirely compensated by the pressure loss due to the friction of the water flowing inside and outside the duct (friction losses), the pressure required to accelerate this water when required (inertia losses) and the pressure drop due to miscellaneous losses. The miscellaneous losses are negligible. They include the pressure loss from the transient operation of the store and provision for the entrance region of the duct.

Starting from just below the upper heat exchanger and moving in the downstream direction, the total pressure drop is:

$$dP = dP_1 + dP_2 + dP_3 + dP_4 \qquad (5)$$

where dP = total pressure drop
dP = pressure drop in the duct
dP^1 = pressure drop at the bottom of the duct
dP^2 = pressure drop of water flowing out of the duct
dP^3_4 = pressure drop at the top of the duct

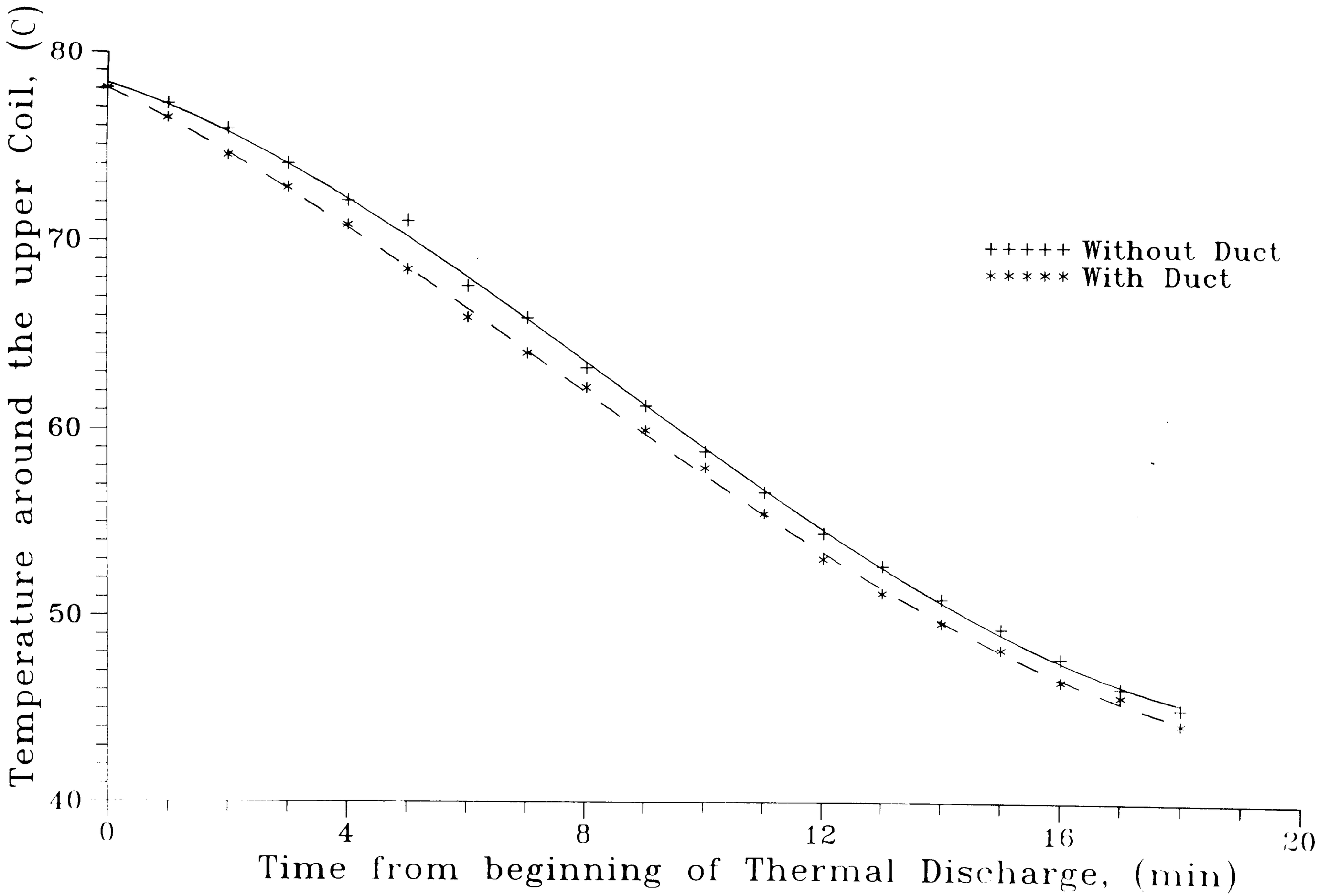


Fig.6.8.2(d): Average Temperature of the Water Surrounding the Upper Heat Exchanger during the Thermal Discharge

The pressure drop caused by the cold water flowing downwards in the duct and by the warm water flowing upwards on the outside of the duct can be calculated from:

$$dP_1 = f_3 \frac{L}{D_1} \frac{\rho V^2}{2} \qquad (6)$$

$$dP_3 = f_3 \frac{L}{D_3} \frac{\rho V^2}{2} \qquad (7)$$

Where f_1 and f_3 are the friction factors, L is the height of the baffle and D is the hydraulic diameter of the enclosed region where water is flowing.

The hydraulic diameter can be estimated from

$$D = \frac{4S}{p} \qquad (8)$$

Where S is the cross section area of the duct and p is the perimeter of the duct corresponding to this cross section.

Although the velocity of the water flowing downwards in the store will be small, the flow will be turbulent in the duct. In this case, the friction factor will be relatively small. An approximate value can be obtained assuming that the roughens of the duct is small. In this case, correlations developed for smooth tubes can be used to approximate the friction factor. One of these correlation presented by Kreith [6] is:

$$f = 0.184 Re^{-0.2} \qquad (9)$$

At the bottom of the duct, the cold water reaches a large zone where the velocity can be neglected, this corresponds to a sudden expansion and the whole of the kinetic energy of the incoming fluid is lost. The pressure drop is then:

$$dP' = \frac{\rho V^2}{2} \qquad (10)$$

Then, still at the bottom of the duct, at the entrance of the outer region of the duct, the fluid needs to be accelerated again to reach the same velocity. This corresponds to a change from pressure into kinetic energy the pressure loss is then:

$$dP'' = \frac{\rho V^2}{2} \qquad (11)$$

This gives

$$dP_2 = dP' + dP'' = \rho V^2 \qquad (12)$$

At the top of the duct, The pressure drop has the same expression that at the bottom of the duct and the pressure loss is:

$$dP_4 = \rho V^2 \qquad (13)$$

Now, by using equations (6)-(13) to evaluate each pressure term and by replacing the corresponding expressions in equation (5), the following expression is obtained for the total pressure drop of the water flowing inside and outside of the duct:

$$dP = 2.244 \rho V^2 \qquad (14)$$

Finally, by eliminating the pressure term between equation (3) and (12), the velocity of the water flowing in the duct can be found as a function of the temperature difference between both sides of the duct:

$$V = \left[\frac{g \beta dT L}{2.244} \right]^{\frac{1}{2}} \qquad (15)$$

By using the 1.5°C as temperature difference (value experimentally measured) between the inside and the outside of the duct, the corresponding velocity is 0.043 ms^{-1}.

This velocity should be compared with the velocity which would normally exists without a duct. At the initial time, temperature measurements give for the velocity of the water surrounding the upper heat exchanger coil of 0.022 ms^{-1}. This suggests that the duct nearly doubles the the velocity of the water surrounding the upper coil.

This increase in velocity should induce a significant increase in the UA value of the heat exchanger. However, only a slight decrease in the effectiveness of heat delivery by the heat exchanger was observed. The explanation of this apparent contradiction can be found only by a detailed analysis of the heat transfer mechanism at the wall of the heat exchanger.

6.8.4 Heat Transfer Correlation for the Heat Exchanger Coil with a Duct

A heat transfer correlation for the prediction of the external heat transfer coefficient for the heat exchanger coil with a duct had to be evaluated from experimental data. For theoretical considerations which have been previously discussed in Chapter 3, this heat transfer correlation was sought in the form:

$$Nu = CRa^{b} \qquad (16)$$

where C and b are constants.

The process by which the coefficient C and the power b can be evaluated has already been discussed in detail and only a brief overview of the method is reported here. The overall UA value of the heat exchanger coil can be estimated by two separate methods.

1) During the thermal discharge, the Integrated Thermal Store is divided in two temperature zones. By using an analogy with two independent thermal stores operating at different temperature, the UA value of the heat exchanger can be estimated using simple heat transfer correlations from the temperature of the top and bottom zones in the store and the temperature of the water entering and leaving the heat exchanger.

2) The UA value of the heat exchanger coil can also be evaluated from the internal and external heat transfer coefficients and the fin efficiency.

- The internal heat transfer coefficient can be obtained by the formula developed by Sleicher and Rouse [7] modified to take into account the curvature of the coil. A correlation based on the results from McAdams[8] is reported in Chapter 3.

- The fin efficiency can be assumed to be unity for this type of heat exchanger and heat transfer mechanism.

- The external heat transfer coefficient of the lower heat exchanger coil can be estimated from

$$Nu = 0.28Ra^{0.2929} \qquad (17)$$

- the external heat transfer coefficient for the upper heat exchanger where the baffles are located can be estimated from:

$$Nu = CRa^{b}$$

where C and b are constants

Finally, by comparing the UA values obtained by both methods, and using the experimental data as an input, it is possible using an iterative routine to evaluate the value of the coefficient C once the power b is known.

To avoid a large amount of data analysis, advantage was taken of previous results obtained for heat transfer correlations in the absence of a duct and by analogy with previous results and the power b was assumed to be 0.2929.

The experimental results are presented in Fig.6.8.4. The corresponding value of the coefficient C is 0.3113. This value for the constant C is 12% above the corresponding value obtained in the absence of a duct.

It can be seen that the correlation curve fits the data relatively closely over the whole range of Nusselt number investigated. The dispersion of the experimental data is 2.173 %. The maximum and minimum errors are respectively 4.571 % and 4.292 %. All these values are in an acceptable range which tends to confirm the validity of the results.

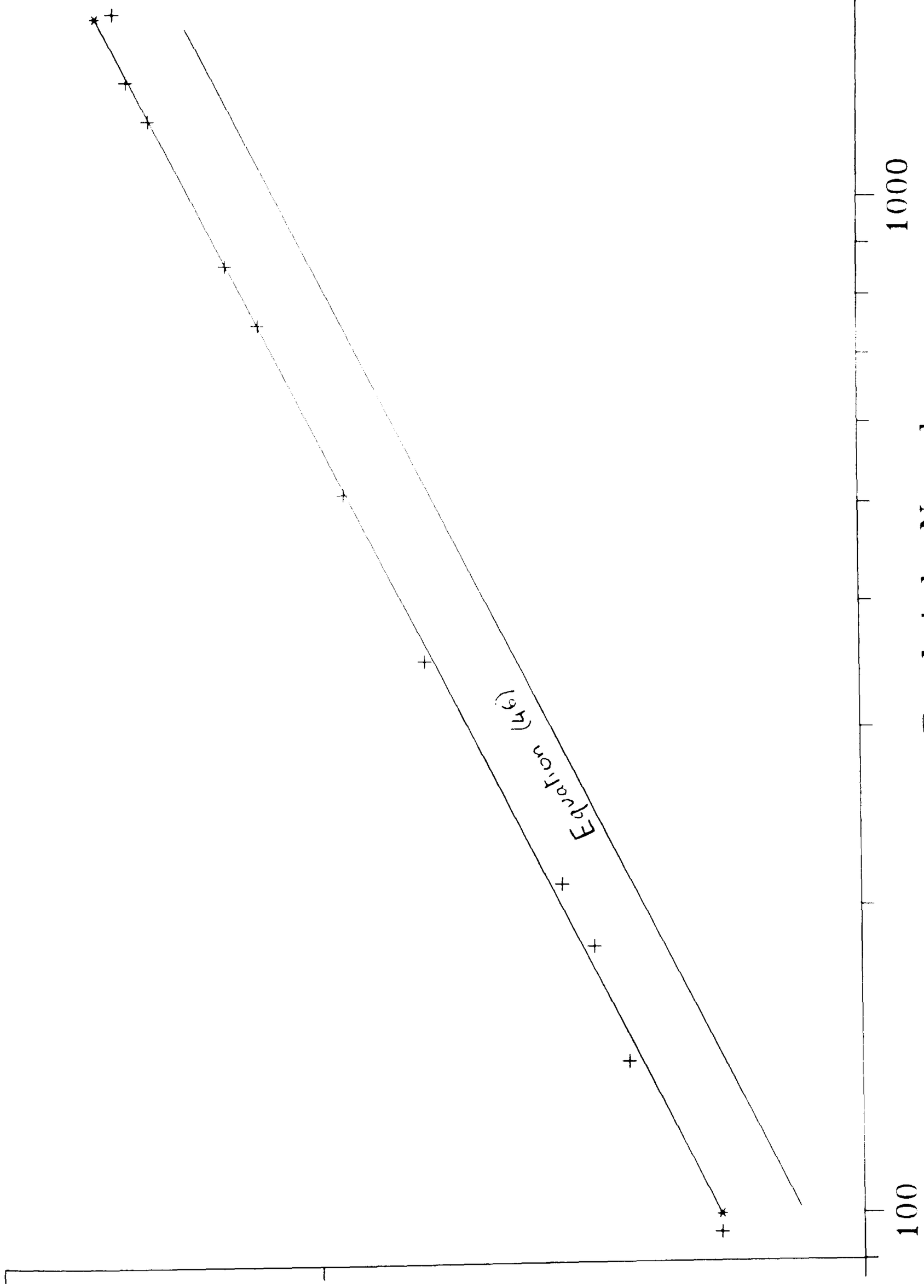


Fig.6.8.4: Heat Transfer Correlation for the Upper Heat Exchanger Coil with a Rectangular Duct

6.8.5 Discussion

1) The duct increases the velocity of the water in the surroundings of the heat exchanger. This increase in velocity is converted in an increase in the coefficient C appearing in the natural convection heat transfer correlation.

But the heat exchanger coil with a duct operates in a slightly lower range of Rayleigh numbers than that of the heat exchanger coil without a duct.

The Nusselt number depends on both the coefficient C and the Rayleigh number. At the moment and with this duct arrangement, the contribution from the coefficient C is more important than the contribution from Rayleigh number and therefore the Nusselt number is increased by the presence of the duct.

As a result of this higher value of the Nusselt number, there is an increase in the external heat transfer coefficient and a corresponding increase in the UA value of the heat exchanger coil.

2) The rate of heat transfer from the water in the thermal store to the water in the pipe is :

$$q = UA(T_w - T_p) \qquad (18)$$

Where T_w is the temperature of the water in the store
T_p is the temperature of the water in the heat exchanger's pipe

The total amount of heat recovered at the end of the thermal discharge can be calculated by integrating the rate of heat delivery from the heat exchanger with time. This gives:

$$Q = \int_0^t UA(T_w - T_p)\,dt \qquad (19)$$

During the thermal discharge, as the temperature of the water surrounding the heat exchanger is lower when there is a duct than when there is no duct, the temperature difference between the water flowing in the heat exchanger and the water surrounding the heat exchanger $(T_w - T_p)$ is reduced significantly (by approximately 11% on average).

This significant reduction in the temperature difference reduces the heat transfer rate and finally less heat is recovered with a duct than without although due to the increase in the external heat transfer coefficient, the UA value of the heat exchanger is higher.

3) The buoyancy driven flow in the store during the thermal discharge is induced by the temperature difference between both sides of the duct. This is created by a lower temperature at the upper coil.

As the temperature of the water surrounding the upper heat exchanger lowers, the buoyancy forces increases and consequently the velocity of the water flowing inside and outside of the duct increases which facilitates the heat transfer on the outside of the heat exchanger. This leads to an increase in the coefficient C appearing in equation (16).

Thus the increase of the external heat-transfer coefficient is a consequence of the lower temperature of the water surrounding the heat exchanger.

In the absence of baffles, mixing between hot and cold water in the store plays a very important role in keeping the upper heat exchanger surrounded by warm water. This warm water creates a high rate of heat transfer at the wall of the heat exchanger and therefore enhance the amount of heat delivered by the store.

This suggests that any type of vertical baffle which aims at reducing the mixing in the surroundings of the heat exchanger in the store is unlikely to improve the effectiveness of heat delivery as it can only operate if the temperature of the water surrounding the heat exchanger is lowered.

The lower temperature decreases the heat transfer rate and consequently the rate of heat delivery from the thermal store with a duct.

4) Equation (15) gives the relation between the velocity of the water and the temperature difference between the inside and the outside of the duct. The velocity is linked to the temperature difference by several coefficients. Only two of these coefficient depend on the duct geometry:

(i) The height of the duct

Although some increase in the velocity of the water flowing around the heat exchanger can be achieved by increasing the height of the duct, there are practical limitations which seriously reduce the improvements which can be achieved in practice. These limitations come mainly from the geometric dimensions of the thermal store in which the duct is located and the location of the heat exchanger coils in the duct.

(ii) The pressure coefficient term 2.244 which is linked to the pressure drop along the duct

The equations giving the pressure drop along the streamline show that the main pressure drop occurs at the top and at the bottom of the duct and not when the water is flowing inside and outside the duct (see equations (7)-(14)).

In view of this, a vertical duct should be designed so that it minimizes the pressure loss along the flow path and particularly at the top and bottom regions of the duct. When this pressure loss is minimized, the velocity of the water flowing around the heat exchanger and the heat transfer coefficient are maximized.

Unfortunately, it is unlikely that simple types of vertical baffles could be designed to create a chimney effect without having a relatively high pressure loss. A reduced pressure drop would most probably need some special arrangement at the top and bottom of the duct to avoid the change from kinetic energy of the flow into pressure and vice-versa.

Even if in order to achieve this reduced pressure loss the present design of the thermal store and/or of the heat exchanger are changed significantly, equation (15), because the pressure drop coefficient appears in a the square root, suggests that it is not likely that large improvements in the performance of hot water delivery will be achieved.
gradient.

6.9 Conclusions

The apparent thermal conductivity of the store has been evaluated experimentally and by the use of a computer model. It is around 2.7 $Wm^{-1}K^{-1}$. When copper tubes to simulate the presence of vertical baffles were inserted in the store this increased to 3.4 $Wm^{-1}K^{-1}$.

The use of vertical and horizontal baffles to improve the effectiveness of heat delivery from the Integrated Thermal Store when subject to a domestic hot water draw-off has been investigated experimentally and theoretically.

Horizontal baffles tend to reduce the mixing between the upper and lower temperature zones in the store. This slightly increase stratification during the thermal discharge and results in a slightly improvement in the performance of heat delivery from the store.

Vertical baffles creating a chimney effect around the upper heat exchanger coil are unlikely to improve the effectiveness of heat recovery. Although they enhance a naturally occurring recirculation effect which increases the overall heat transfer coefficient of the heat exchanger, they can only operate satisfactory if the temperature of the water surrounding the heat exchanger is reduced. This reduction in temperature reduces the rate of heat delivery from the heat exchanger coil.

References

[1] Mote, R ,Probert, D and Nevrala, D.J.
performance of a coiled finned tube heat exchanger submerged in a hot water store: effect of the heat exchanger's orientation
Applied Energy, Vol 34, pp 165-179, 1989

[2] Knudsen, J.G. and Pan, R.B.
Natural Convection Heat Transfer from Transverse Finned Tubes
Chemical Engineering Symposium Progress, N° 57, Vol 61, pp 44-49

[3] Feiereisen, T.J. and Klein, S.A.
Heat Transfer from Immersed Coils
ASME Trans, paper 82/wa/sol-18, 1982

[4] Collard, C.
Effects of baffles on the effectiveness of a heat exchanger located in a hot water tank
MSc Thesis, Cranfield Institute Of Technology, U.K., 1990

[5] Cole, R.L. and Belinger, F.O.
Thermally Stratified Storage Tanks
ASHRAE Trans Vol 88, No2, pp 1005-1016, 1982

[6] Kreith, F.
Principles of Heat Transfer
Third Edition
Harper Internaltiona Edition, New York, 1976

[7] Sleicher, C.A. and Rouse, M.W.
A Convenient Correlation For Heat Transfer To Constant And Variable Property Fluids In Turbulent Pipe Flow
J Heat Mass Transfer, Vol 18, No 5, pp 677-683, 1975

[8] McAdans, W.H.
Heat Transmission
3rd edition, McGraw-Hill, New York, 1954

Chapter 7

Integrated Thermal Store in the Space Heating Mode

7.1 Introduction

In an Integrated Thermal Store, the heat capacity of the store is also used to fulfill space heating requirements. The use of the same store for satisfying both the domestic hot water and space heating demands can be justified as these two demands might not coincide in time.

When there is a space heating demand, hot water is extracted from the top of the Integrated Thermal Store, passed through the radiators where it releases its sensible heat and then returned to the bottom of the store. This type of thermal discharge has completely different characteristics from the thermal discharge achieved by passing cold water through the heat exchanger coil and consequently had to be investigated separately.

A lot of interest has been generated in recent years to this type of thermal discharge for water based thermal store. This interest was caused by the recent advances in solar heating where energy from the Sun is stored during the day in the form of sensible heat in water [1],[2].

Several experimental and numerical investigations [3],[4],[5] have been carried out to improve the design of these thermal stores for these type of thermal discharge. There is general agreement that a high degree of stratification improves the performance of the system [6],[7]. Thus some simple design concepts to enhance stratification in these water based thermal store are immediately evident [8]:

- the cold inlet must be at the bottom of the store and the hot outlet at the top

- the inlet system should distribute water uniformly across the width of the tank

Although these investigations are relatively useful for solar heating systems, the behaviour of an Integrated Thermal Store is expected to be different than that of a solar heating system. The difference ranges in the conditions in which both systems operate. For example, an Integrated Thermal Store will operate at a higher temperature than a solar system, the temperature of the water being passed into the store will be different in terms of flow and return temperature. Additionally, the flow rate of water passing through the Integrated Thermal Store will be higher and as its size will be smaller more turbulence and mixing will take place in an Integrated Thermal Store used for space heating than in a water based store used for solar energy storage.

Therefore it was necessary to investigate the behaviour of the Integrated Thermal Store and not to rely entirely on data available in the literature which would have been obtained and/or developed for different operating conditions. This investigation was carried out experimentally and by the use of a computer model. By comparing the experimental results with the computer predictions it was possible to have a better understanding of the behaviour of Integrated Thermal Stores and develop simple correlations which suggest how to improve its design.

7.2 Computer Modelling

The theoretical background provided by the equations of motion could not be solved analytically or by the use of simple calculations. Therefore the use of a numerical model to simulate the flow in the thermal store was necessary

Several types of formulations for the model can be used. The most complex formulations take into account the fluid flow inside the store and, by means of the equations of conservation, momentum and energy might be able to predict the velocities and temperature field of the water in the store.

These equations, when written in vector notation are:

Equation of continuity

$$\frac{D\rho}{Dt} + \nabla(\rho V) = 0 \qquad (1)$$

Conservation of momentum

$$\rho \frac{Dv}{Dt} = \nabla.\tau - \nabla P + \rho g \qquad (2)$$

Conservation of energy

$$\rho Cp \frac{DT}{Dt} = \nabla.k\nabla T + \mu\phi + q'' \qquad (3)$$

Equations (1) to (3) do not constitute a complete system as there are more unknowns than equations. To obtain a complete system, the equation of state of the fluid considered should be added. This is usually expressed in the form

$$\rho = f(T) \qquad (4)$$

The system of equations (1)-(4) is linked explicitly through the velocity terms appearing simultaneously in all the equations (apart from the equation of state) and implicitly through the physical properties of the considered fluid which depend on temperature and pressure. General methods of solution are not yet available. However, by using a few approximations, these equations can be simplified and their resolution made easier.

The first approximation is that the fluid is incompressible and newtonian. This can be regarded as a relatively good approximation for the liquid water contained in the Integrated Thermal Store as the shear stress is very nearly proportional to the velocity gradient and the coefficient of expansion of water extremely small.

The second approximation is to neglect the variations of the fluid properties with temperature except in the buoyancy term of the momentum equations. This is known as the Boussinesq approximation. This approximation is also justified for the Integrated Thermal Store when in the space heating mode as the temperature variations in the store are usually small. This make the thermal properties of water nearly constant in the temperature range considered. A detailed justification of this approximation can be found in [9].

The third approximation is to neglect some terms in the energy equation. These terms correspond to energy exchange mechanisms within the store which are negligible in water based thermal stores. Examples of these energy exchange mechanisms are the degradation of kinetic energy into heat created by friction or creation of work by fluid expanding with temperature.

This third approximation is fully justified due to the properties of water including low viscosity and coefficient of thermal expansion and the characteristics of the thermal discharge (low velocity of water in the store and small temperature difference).

Finally, when all these approximations are used, the equations of motion when expressed in cylindrical coordinates reduce to:

Equation of continuity

$$-\frac{1}{r}\frac{\partial}{\partial r}(rVr) + \frac{1}{r}\frac{\partial}{\partial \theta}(V\theta) + \frac{\partial}{\partial z}(Vz) = 0 \qquad (5)$$

Conservation of momentum

r-direction

$$\rho\left(\frac{\partial Vr}{\partial t} + Vr\frac{\partial Vr}{\partial r} + \frac{V\theta}{r}\frac{\partial Vr}{\partial \theta} - \frac{V\theta^2}{r} + Vz\frac{\partial Vr}{\partial z}\right) = -\frac{\partial P}{\partial r}$$

$$+ \mu\left[\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(rVr)\right) + \frac{1}{r^2}\frac{\partial^2 Vr}{\partial \theta^2} - \frac{2\partial V\theta^2}{r^2\partial \theta} + \frac{\partial^2 Vr}{\partial z^2}\right] \qquad (6)$$

θ-direction

$$\rho\left(\frac{\partial V\theta}{\partial t} + Vr\frac{\partial V\theta}{\partial r} + \frac{V\theta}{r}\frac{\partial V\theta}{\partial \theta} + \frac{VrV\theta}{r} + Vz\frac{\partial V\theta}{\partial z}\right) = -\frac{1}{r}\frac{\partial P}{\partial \theta}$$

$$+ \mu\left[\frac{1}{\partial r}\left(\frac{\partial}{r\partial r}(rV\theta)\right) + \frac{1}{r^2}\frac{\partial^2 V\theta}{\partial \theta^2} + \frac{2}{r^2}\frac{Vr}{\theta} + \frac{\partial^2 V\theta}{\partial z^2}\right] \qquad (7)$$

z-direction

$$\rho\left(\frac{\partial Vz}{\partial t} + Vr\frac{\partial Vz}{\partial r} + \frac{Vo}{r}\frac{\partial Vz}{\partial \theta} + Vz\frac{\partial Vz}{\partial z}\right) = -\frac{\partial P}{\partial z}$$

$$+ \mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\left(\frac{\partial Vz}{\partial r}\right)\right) + \frac{1}{r^2}\frac{\partial^2 Vz}{\partial \theta^2} + \frac{\partial^2 Vz}{\partial z^2}\right] \qquad (8)$$

Conservation of energy

$$\rho Cp\left(\frac{\partial T}{\partial t} + Vr\frac{\partial T}{\partial r} + \frac{Vo}{r}\frac{\partial T}{\partial \theta} + Vz\frac{\partial T}{\partial z}\right) = k\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2}\right] \qquad (9)$$

+ Equation of state

Although in simplified form, the system of equations (4)-(9) is still extremely difficult to solve. Numerical solution can only be approached by the use of finite difference or finite element methods. Both methods require the use of 3-dimensional nodal mesh which not only are relatively difficult to compute but require large amount of computation time and computer memory to solve.

Very few numerical solutions of these equations are available at the moment. Interesting numerical applications of these equation are presented by Sha [10] and Lin [11] and are used to investigate the use of baffles to enhance stratification in hot water storage tanks.

An additional simplification in the equations of motion can be brought by assuming that the flow within the store is two dimensional (axisymetric flow). This assumption significantly simplifies the formulation of the equations of motion by reducing at the same time the number of equations to solve simultaneously and the number of unknowns.

This simplification can only be used if the geometry of the thermal store is such that the flow within the thermal store can really be approximated by an axisymetric model. This assumption cannot be used for example when the flow in the store is clearly three dimensional.

Some attempts have been made to model 2 dimensional transient axisymetric flows in cylindrical containers. Hyun [12], Lin [13] and Barakat [14] solved this problem using finite difference formulations for various types of boundary conditions corresponding to isothermal and adiabatic walls, free surfaces and heat and mass transfer surfaces. However several major problems remain even with these simplified equations. These problems are:

1- high computation power required to solve the system of equations.

2- high computer storage capacity required to store the variables and the results.

3- complexity to program boundary conditions.

4- low accuracy of computer predictions.

Due to recent improvement in electronics and computer technology, the the first and second problems seem to be at the reach of modern microcomputers. The third problem is linked to the complexity of the boundary conditions. Unfortunately, even in simple practical engineering situations, the boundary conditions corresponding to the fluid flow are difficult to program. Consequently, their programmation requires a lot of engineering time which makes the use of the computer simulation impractical.

Finally, even when all of the previous problems have been overcome, the last problem associated with the low accuracy of the computer predictions seriously limits the usefulness of computer models for fluid flow simulation.

Thus the need to find or develop even simpler models than the axisymetric flow. The ultimate simplification is the one dimensional flow distribution in the store (the justification for this simplistic model will be explained latter).

The assumption of the one dimensional flow in the store leads to further simplifications in the equations of motion by again reducing the number of equations to solve simultaneously. Consequently, computer models can be developed without requiring very high computation time or memory requirement and become accessible to relatively small computers. Such models are simple to program and above all, numerical tricks make it possible to reach a reasonable accuracy of flow prediction.

In these models, the temperature of the fluid within the store, is assumed to vary along the axis of the store and with time to model transient operation. The store is divided into horizontal slabs and the temperature in each slab is calculated at very small time intervals. When the store is subjected to a vertical through-flow, the temperature profile in the vertical direction can be predicted when the temperature in the slabs is known.

Several one dimensional models of hot water storage tanks can be found in the literature. A very comprehensive model was developed by Cabelli [15]. Although the numerical procedure is not greatly detailed.

Cabelli presents all the equations required to solve the energy equation and to extend the results to a two dimensional flow. Several numerical examples are given for hot water storage tanks and the effect of factors such as the Reynolds number of the inlet flow on stratification are investigated.

Opel [16] investigated in detail several numerical problems associated with the computation of the one dimensional flow equation using an explicit finite difference formulation. Among these are stability criteria associated with explicit formulation, pseudo-mixing associated with improper time and spatial increments and modelling of the mixing taking place in the store.

7.2.1 The Energy Equation

When the assumption of the one dimensional flow is made, only the energy equation is used. The energy equation reflects the process of heat transfer within a stratified thermal store. This equation when written in a differential form is reduced to:

$$\frac{\partial T}{\partial t} + Vz\frac{\partial T}{\partial z} = \alpha\frac{\partial^2 T}{\partial z^2} \qquad (10)$$

Where Vz=axial velocity of the water in the store

To this equation should be added equations corresponding to the type of boundary conditions required and the equation of state of the considered fluid.

The equation of state is usually simplified using the assumption that the thermal diffusivity of the fluid is constant. This assumption is relatively valid for small temperature variations in the thermal store. This reduces the equation of state to:

$$\alpha = \text{cte} \qquad (11)$$

The energy equation, although in simplified form, can not be solved analytically. Thus a numerical solution had to be developed. There are two types of methods which can solve this equation with partial derivatives:

- finite element methods: in these methods, the properties of the fluid are averaged over small control volume known as finite elements. These finite elements have usually simple shapes such as triangles (for 2-D problems) or tetraedrons (for 3-D problems). The system of partial differential equation to be solved is then integrated by parts. In this process, the differential equations are replaced by a system of linear equations relating the nodes of the finite elements network. The system of linear equations is then solved using classical techniques such as gaussian elimination or iterative methods.

- finite difference methods: in these methods, the partial derivative terms are replaced by approximations using the development of Taylor formula. Again this formulation leads to a linear system of equations which can also be solved using classical techniques.

Although finite element methods present several advantages such as the possibility to handle complicated geometries and a better accuracy, finite difference methods are easier to programme. It was therefore decided to solve the energy equation with a finite difference formulation.

Several types of finite difference formulation can be used but again the simplest formulations were sought to simplify both the programmation of the problem and the computation of the solution.

This finite difference formulation approximate the partial derivatives by finite difference terms. The approximations for the partial derivatives which will be used in the resolution of the energy equation are as follows:

First order derivative with forward difference:

$$\frac{\partial f}{\partial x} = \frac{f_{x+\Delta x} - f_x}{\Delta x} \tag{12}$$

First order derivatives with backward difference:

$$\frac{\partial f}{\partial x} = \frac{f_x - f_{x-\Delta x}}{\Delta x} \tag{13}$$

Second order derivative with central difference:

$$\frac{\partial^2 f}{\partial x^2} = \frac{f_{x-\Delta x} - 2f_x + f_{x+\Delta x}}{\Delta x^2} \tag{14}$$

Using equations (12)-(14), several formulations of the energy equation can be obtained, not all of them being equivalent. These can mainly be classified into two types: implicit formulations and explicit formulations. The difference between implicit and explicit comes from a different formulation of the terms with respect to the time partial derivative.

The notation used in the finite difference formulation which follows is presented in Fig.7.2.1(a). The nodes are are evenly spaced along the height of the store and numberred from 1 at the bottom of the store to n at the top of the store.

Implicit Formulation

The implicit formulation expresses all the terms appearing in the energy equation at time t+dt. The term in the partial time derivative which is expressed using a first order backward difference approximation with respect to time. This results in the following formulation for a node i in the store:

$$\frac{T_i^t - T_i}{\Delta t} + Vz\frac{T_i^t - T_{i-1}^t}{\Delta z} = \alpha\frac{T_{i-1}^t - 2T_i^t + T_{i+1}^t}{\Delta z^2} \tag{15}$$

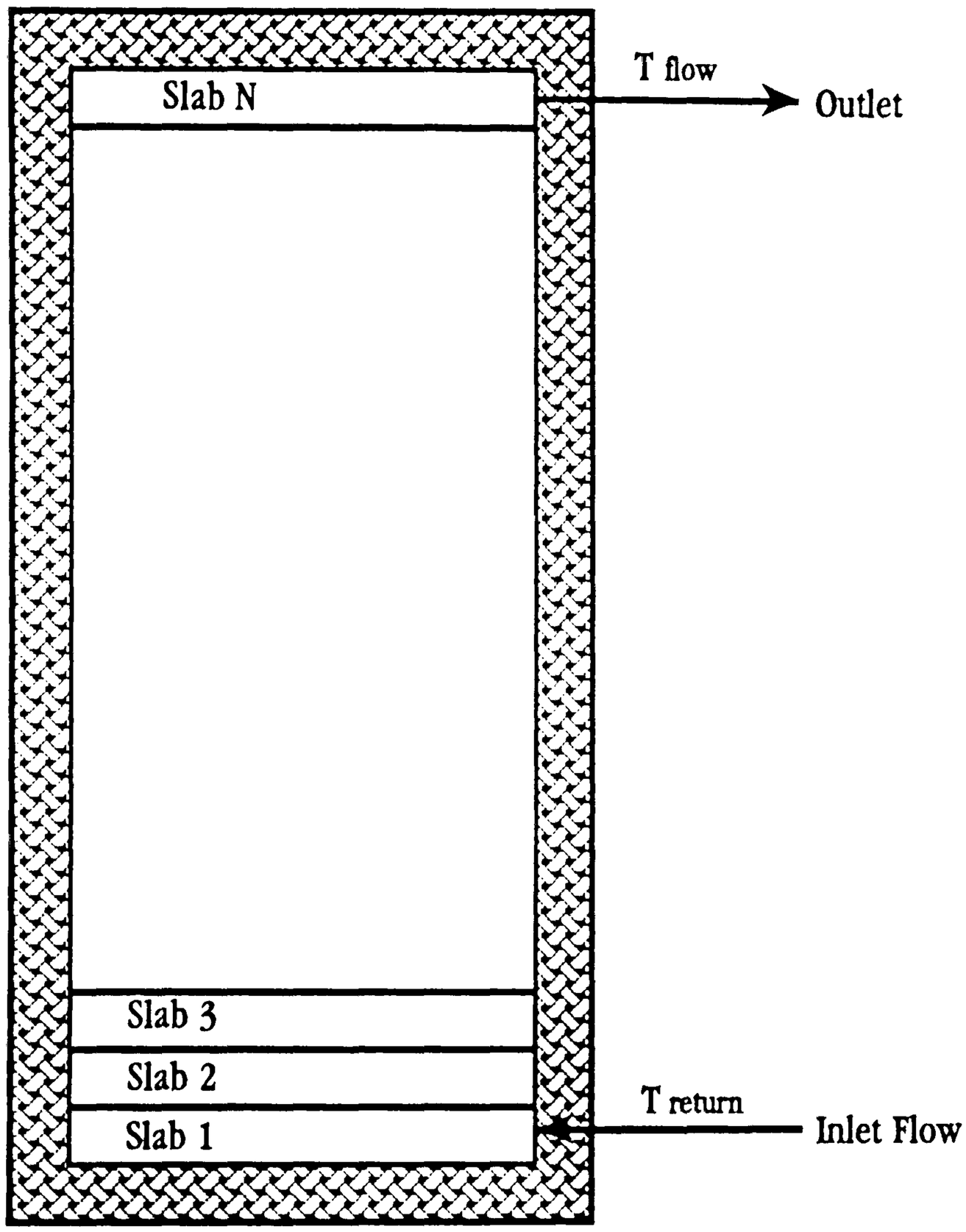


Fig.7.2.1(a) Notation Used in the Finite Difference Formulation

In the implicit formulation, all the nodes are linked to the adjacent nodes through linear equations. As a result, the final formulation is a linear system which has to be solved by relatively complex numerical methods such a gaussian elimination or iterative methods.

Unfortunately, implicit finite difference methods give an extremely poor accuracy of numerical predictions. The poor accuracy comes from a phenomenon called 'pseudo-mixing'. The effect of this pseudo-mixing is equivalent numerically to increase the thermal diffusivity α of the considered fluid thus creating an artificially mixed tank.

The pseudo-mixing induced by this type of formulation is illustrated in Fig.7.2.1(b). Its importance depends on factors such as the time step, the number of nodes used in the formulation and the velocity of the water flowing through the store. As a consequence of this pseudo mixing, although the implicit formulation is stable regardless to the time step and the number of nodes used, it cannot be used to solve the energy equation as the numerical predictions would be physically meaningless.

Explicit Formulation

The accuracy of the computer predictions might be improved using an explicit formulation. In this formulation, all the terms appearing in the energy equation are evaluated at time t. The time partial derivative is approximated using a first order forward difference formulation with respect to time. This results in the following formulation for a node i in the store:

$$\frac{T_i^t - T_i}{\Delta t} + Vz\frac{T_{i+1} - T_{i-1}}{2\Delta z} = \alpha\frac{T_{i-1} - 2T_i + T_{i+1}}{\Delta z^2} \qquad (16)$$

The explicit formulation is stable and converges only if the two following criteria are fulfilled:

$$\frac{V\Delta z}{\alpha} < 2 \qquad (17)$$

$$\frac{2\alpha\Delta t}{\Delta z^2} < 1 \qquad (18)$$

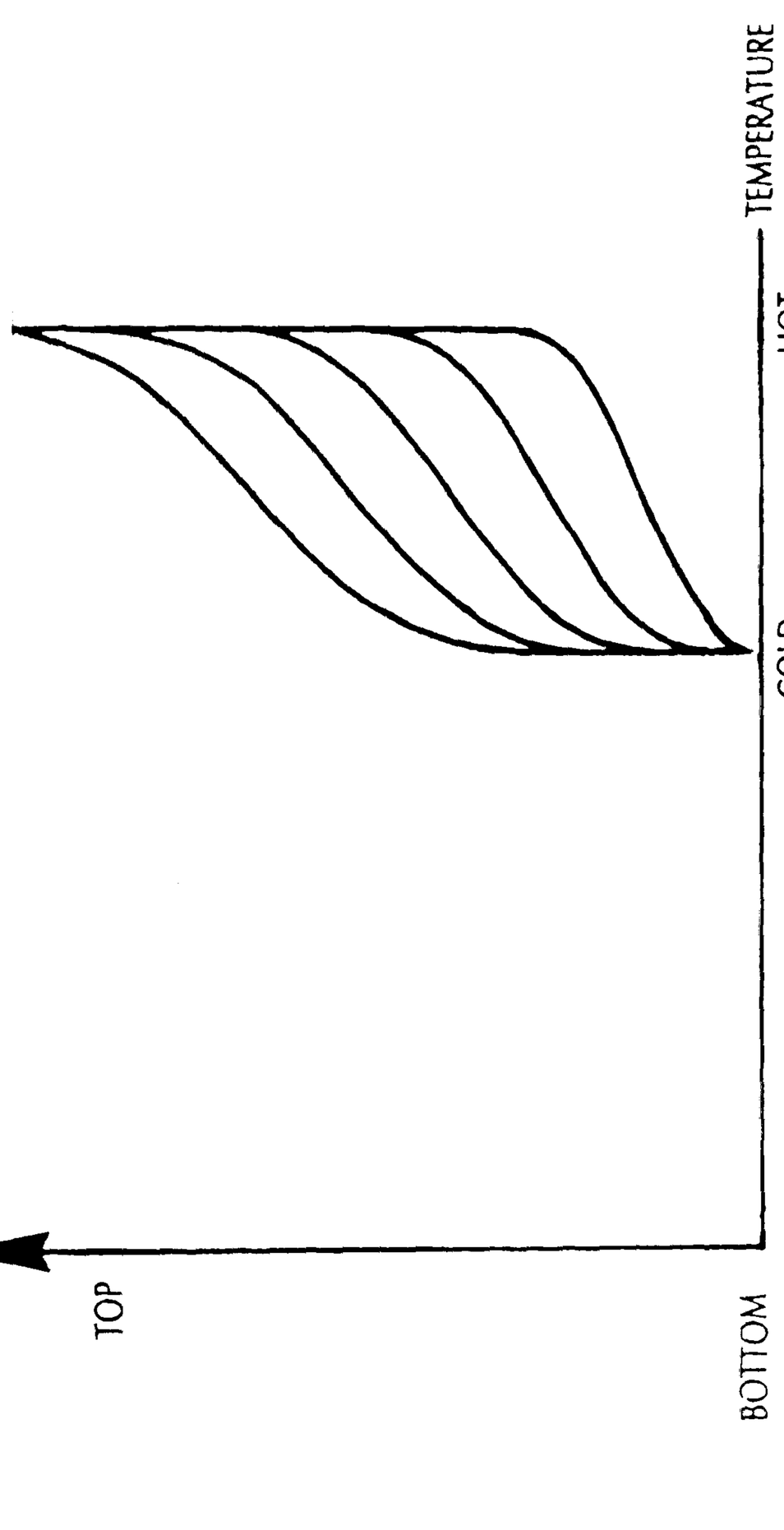


Fig.7.2.1(b): Effect of Pseudo Mixing on the Computer Predictions of the Development of the Thermocline

In the explicit formulation, the unknowns corresponding to the considered node are not linked the unknowns corresponding to adjacent nodes in the final system of equations. Therefore relatively simple numerical methods can be used to solve the corresponding system of equations. Consequently a relatively large number of unknowns can easily be handled.

The explicit formulation is almost always used to model one dimensional flow in thermal stores but there are several disadvantages in using it. These disadvantages come from conditions imposed by the stability criteria.

Equation (17) imposes that the spatial increment dz be reduced when the velocity of the water flowing through the store increases. This leads to increasing the number of nodes in the finite difference formulation and consequently increasing the number of unknowns. As the number of unknowns increases, a larger computation time is required to solve the system of equations and more computer memory is necessary to handle the system. Furthermore, equation (18) imposes that the time increment be reduced as the square of the spatial increment. This imposes that a larger number of time step calculation are carried out to reach the final solution which further increases the computation time. This high computation time and memory requirements might be out of the reach of small microcomputers which limits the usefulness of explicit finite difference formulation of the energy equation.

In addition to the computation problem, when the velocity of the water flowing through the store changes, the stability criteria impose that both the spatial and time step have to be changed. Although this can easily be done numerically, it makes the comparison of the computer predictions with experimental data relatively difficult for practical reasons.

Consequently, an explicit formulations of the energy equation, although possible, would have been impractical to use and therefore a different method of solution had to be developed.

7.2.2 Combined Explicit-Implicit Formulation

A new method of solution, based on a combination of implicit and explicit formulation of the energy equation was developed. This new formulation combines the accuracy of the explicit formulation with the stability of the implicit formulation.

Additional advantages provided by the new formulation include simplicity to programme, low computing power required to reach the solution and flexibility in use. Surprisingly, no reference has been found in the literature about this method of solution.

This new formulation divides the energy equation in two limit cases. Each of these cases correspond to a different mode of heat transfer. These two cases are treated separately and the result combined to give the final formulation. These cases are:

- the conduction case whose simplified equation is:

$$\frac{\delta T}{\delta t} = \alpha \frac{\delta^2 T}{\delta z^2} \qquad (19)$$

- the convection case whose simplified equation is:

$$\frac{\delta T}{\delta t} + Vz \frac{\delta T}{\delta z} = 0 \qquad (20)$$

Conduction case

The temperature gradient corresponding to the conduction case is calculated using an **implicit** finite difference formulation. The implicit formulation removes the need for a stability criteria for the time step. This formulation for a node i in the middle of the store is:

$$\frac{T_i^t - T_i}{\Delta t} = \alpha \frac{T_{i-1}^t - 2T_i^t + T_{i+1}^t}{\Delta z^2} \qquad (21)$$

Convection case

The temperature gradient corresponding to the convection case is calculated using an **explicit** finite difference formulation. This formulation is necessary to avoid large computation errors caused by pseudo-mixing. This formulation is:

$$\frac{T_i^t - T_i}{\Delta t} + Vz\frac{T_i - T_{i-1}}{\Delta z^2} = 0 \qquad (22)$$

The stability criteria associated with the explicit formulation of the convection case is:

$$\frac{Vz\Delta t}{\Delta z} = 1 \qquad (23)$$

By **combining** the equations corresponding to the conduction and the convection case and by taking into account the stability criteria, the following system of equation is obtained:

$$\frac{T_i^t - T_i}{\Delta t} + Vz\frac{T_i - T_{i-1}}{\Delta z} = \alpha\frac{T_{i-1}^t - 2T_i^t + T_{i+1}^t}{\Delta z^2} \qquad (24)$$

$$\frac{Vz\Delta t}{\Delta z} = 1 \qquad (25)$$

The first advantage of this combined implicit-explicit formulation is that, as there is only one stability criteria, either the time step or the number of nodes used in the formulation can be chosen for convenience. This can be a powerful tool either to simplify the comparison between the computer predictions and the experimental results or to match the computation time to a particular computer capability.

The second advantage of this new formulation is that the time step and the spatial increment are linked by the stability criteria. They both vary linearly in the same way but in a linear fashion so that when the spatial increment is decreased moderately, the time increment has to be decreased only moderately.

7.2.3 The Method of Solution

The finite difference formulation of the energy equation (Eq (24)) was used for all the internal nodes in the store.

The top and the bottom of the store were assumed adiabatic. This is obtained when the temperature gradient at the top and bottom of the store are negligible. This results in the following equations for the corresponding nodes:

$$T_1 = T_2 \qquad T_{n-1} = T_n \qquad (26)$$

The number of slabs (or nodes) to use for the numerical resolution had to be chosen carefully as both the accuracy of the computer predictions and the computation time depend heavily on this number.

The computation time required to solved the problem will depend only on the number of nodes. As a Gauss elimination method was used to solve the system of linear equations for each time step, the time required for each time step will be approximately proportional to the number of nodes raised to the power 3. Moreover, as the stability criteria has to be respected, smaller time steps will be required if the number of nodes is increased. As a result of theses two factors, the total computation time to reach a given solution will be approximately proportional to the number of nodes raised to the power 4.

Some computer memory (RAM) will be required to store the matrix of the Gauss system. This will be vary approximately as the number of nodes raised to the power two. Finally, the mass storage capacity of the computer required to store the results will vary approximately as the number of nodes raised to the power two.

Preliminary tests carried out on a 64 K microcomputer showed that the maximum number of nodes which could be handled at any one time by the memory of the computer was around 59. When 59 nodes were used, the computation time was approximately 7 minutes for each time step which was considered as relatively high but could be accommodated. This value of 59 nodes was considerably higher than the lower limit of 20 nodes suggested by some authors [15]-[17] to reach a reasonable accuracy of computer predictions.

However, the use of 59 nodes would have made the comparison with the experimental data difficult. It was therefore decided to reduce this number to 42. The top and bottom nodes were used only to simulate the adiabatic top and bottom parts of the store and therefore did not correspond to any slab of water in the tank. The remaining 40 nodes were then equally spaced along the height of the tank and consequently the nodes corresponding to even numbers were situated exactly at the location of the thermojunctions in the store which greatly facilitated the comparison of the experimental data with the computer predictions.

The choice of a direct Gauss elimination method for the resolution of the system of linear equations was made mainly for reason of simplicity to programme and the assurance that the solution would always be reached after a finite computation time regardless of the values of the parameters used in the programme. The gauss method however, presents the disdvantage that some large computation errors might occur depending on the numerical coefficients appearing in the system of equations to solve. This is true particularly if the pivots have relatively small values.

These computation errors can be reduced using a partial or total pivoting method but these methods are relatively complex to programme. They can also be reduced by increassing the number of significant digits used by the computer during the computation. This was achieved by carrying all the computation in double precision (with 16 significant digits). A small number of computations were however carried out in simple precision and the results compared with double precision results. The difference in the computer predictions were always less than 0.01 % which sugests that the system of equations was relatively stable. However, the computation in double precision was used for all the results presented later in this chapter.

The time step used in the computer simulation was adjusted according to the velocity of the water flowing in the store so that the stability criteria given in equation (25) was always satisfied. The minimum time step was obtained when the velocity of the water was maximum and was approximately 30 seconds.

The overall heat transfer coefficient from the vertical wall of the store to the environment was assumed to vary along the height according to equation (5) (Chapter 2). It was estimated for each slab from the temperature of the considered slab and the temperature of the environment (20°C) with the following equation:

$$U = \frac{1}{A}(-2.5 + 0.11T + 0.001T^2) \qquad (27)$$

The thermal properties of water were assumed constant apart from the thermal diffusivity which was assumed to vary linearly between 40°C and 80°C. The linear relation was obtained using a least square interpolation between the considered temperatures. This equation is given in Appendix 2 with the thermal properties of liquid water at saturation pressure which were used to develop the interpolation.

7.3 Experimental Apparatus and Methodology

The experimental rig is described in full detail in Chapter 2. For this series of experiments, only the space heating facility was used. This is presented in Fig.7.3.

The 200 litre store was first heated at 80°C. During the heating period, the recirculating pump was run continuously to avoid any temperature gradient in the store. At the end of the heating period, the heating mechanism was switched-off. This was left off during the whole of the experiment so that the temperature variations in the 200 litre store could be observed undisturbed by an external source.

A 2 minute time period was allowed to pass before the beginning of the experiment. These 2 minutes allowed the water in the store to settle so that the starting conditions were always identical and a high degree of repeatability in the experiments achieved. After the 2 minutes, the space heating pump was switched on. This extracted a constant mass flow rate of water from the top of the 200 litre store. This hot water was then passed into a heat exchanger coil located within the 50 litre store where heat was transferred from the water flowing in the coil to the water in the 50 litre store. The then relatively cold water from the heat exchanger was returned to the bottom of the 200 litre store.

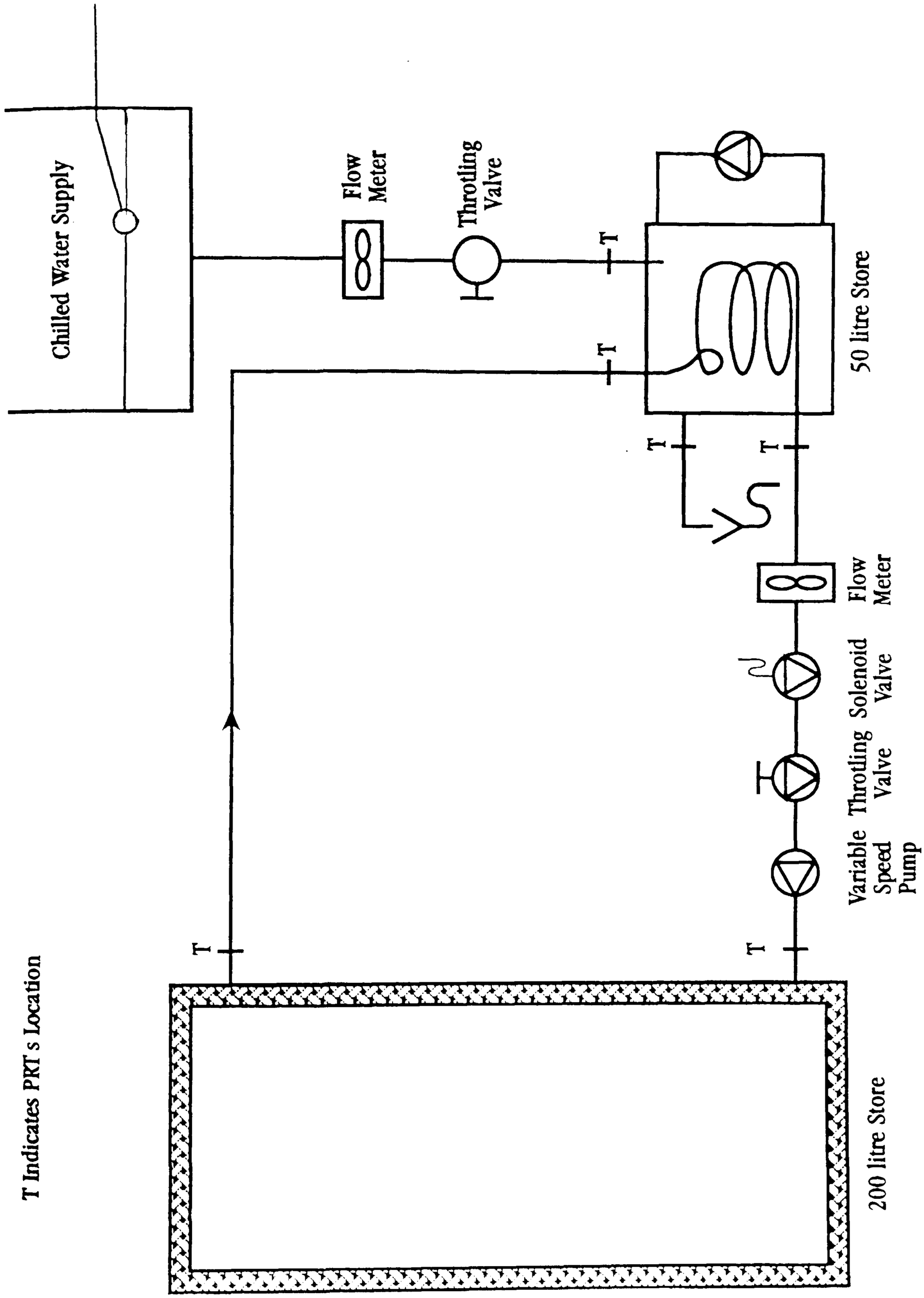


Fig.7.3: Detailed Schematic of the Space Heating Facility

The water in the 50 litre store was kept at relatively low temperature by constantly passing into the store some chilled water coming from the 250 litre chilled water reservoir. The water in the 250 litre reservoir was kept at a constant temperature of 8°C during the whole of the experiment. The mixing pump of the 50 litre store was run continuously to ensure the presence of strong convection currents in the 50 litre store which greatly enhanced the rate of heat transfer through the walls of the heat exchanger.

At the beginning of the experiment, the 50 litre store was at approximately 8°C. Its temperature slowly rose during the experiment to approximately 30°C. Due to this increase in temperature, some variations in the temperature of the water returning to the 200 litre thermal store could not be avoided. These fluctuations were at most a few degree Celcius which was one order of magnitude lower than the temperature difference between the flow and return temperature and could consequently be neglected.

Several experiments were carried out to investigate the wide range of operating conditions which can be accommodated with Integrated Thermal Stores. The mass flow rate through the 200 litre store was varied from 1.5 to 12 litre/min by increments of approximately 1 litre/min. This corresponded to Reynolds numbers based on the store diameter ranging from 110 to 1300.

The experiment's duration varied from 20 minutes to 2 hours depending on the mass flow rate in the 200 litre store. The temperature difference between the flow and return temperature was around 40°C for experiments at low flow rate and 20°C for experiments at high flow rate. This corresponded to heat extraction rate from the Integrated Thermal Store ranging from 4.5 kW to 16.5 kW respectively.

Finally, the instrumentation, which included PRTs, thermocouples and flowmeters were scanned every 10 seconds. This high scanning rate was necessary to obtain a satisfactory accuracy in the flow measurements and in the analysis of the development of the thermocline in the 200 litre thermal store. Due to the large amount of data and the high scanning rate, the data collected during the experiments had to be stored on a hard disc and analyzed separately.

7.4 Comparison of the Computer Predictions and the Experimental Measurements

Fig.7.4(a) and Fig.7.4(b) show the measured variations of the flow and return temperatures during the thermal discharge for flow rates of 6 and 12 litre/min respectively. The computer predictions obtained using the one dimensional finite difference formulation of the energy equation to approximate the flow of water in the thermal store are also shown on the same graph in dash lines.

For a sufficient time period, the temperature of the water delivered by the thermal store (the flow temperature) is relatively constant and high for both the computer predictions and the experimental results. However, the experimental results, show a slight decrease in the flow temperature with time. This slight decrease is most probably caused by a small quantity of heat being rapidly conducted along the walls of the store during the experiment.

After a few minutes, there is a sharp decline in the flow temperature. This decline corresponds to the thermocline reaching the top of the store. The inflexion of both the experimental results and the computer predictions are exactly at the points were the curves cross. However, the slope of the temperature curve obtained from the experimental data at the inflexion point is not as sharp as the slope of the computer predictions at the same point.

Also, when the flow temperature begins to decrease significantly, it can be observed that its rate of change experimentally measured is lower than its rate of change predicted by the computer model. This is particularly noticeable when the maximum curvature of both temperature variation curves are compared.

All these observations suggest that the real store is less sensitive to temperature fluctuations than the modelled store. The reason for this difference in behaviour can be explained only be the fact that some mixing takes place in the real store which tends to make it more isothermal than modelled thermal store. To investigate the cause and the effect of the difference in mixing between the real and the modelled store, a detailed analysis of the temperature field in the store during the thermal discharge was carried out.

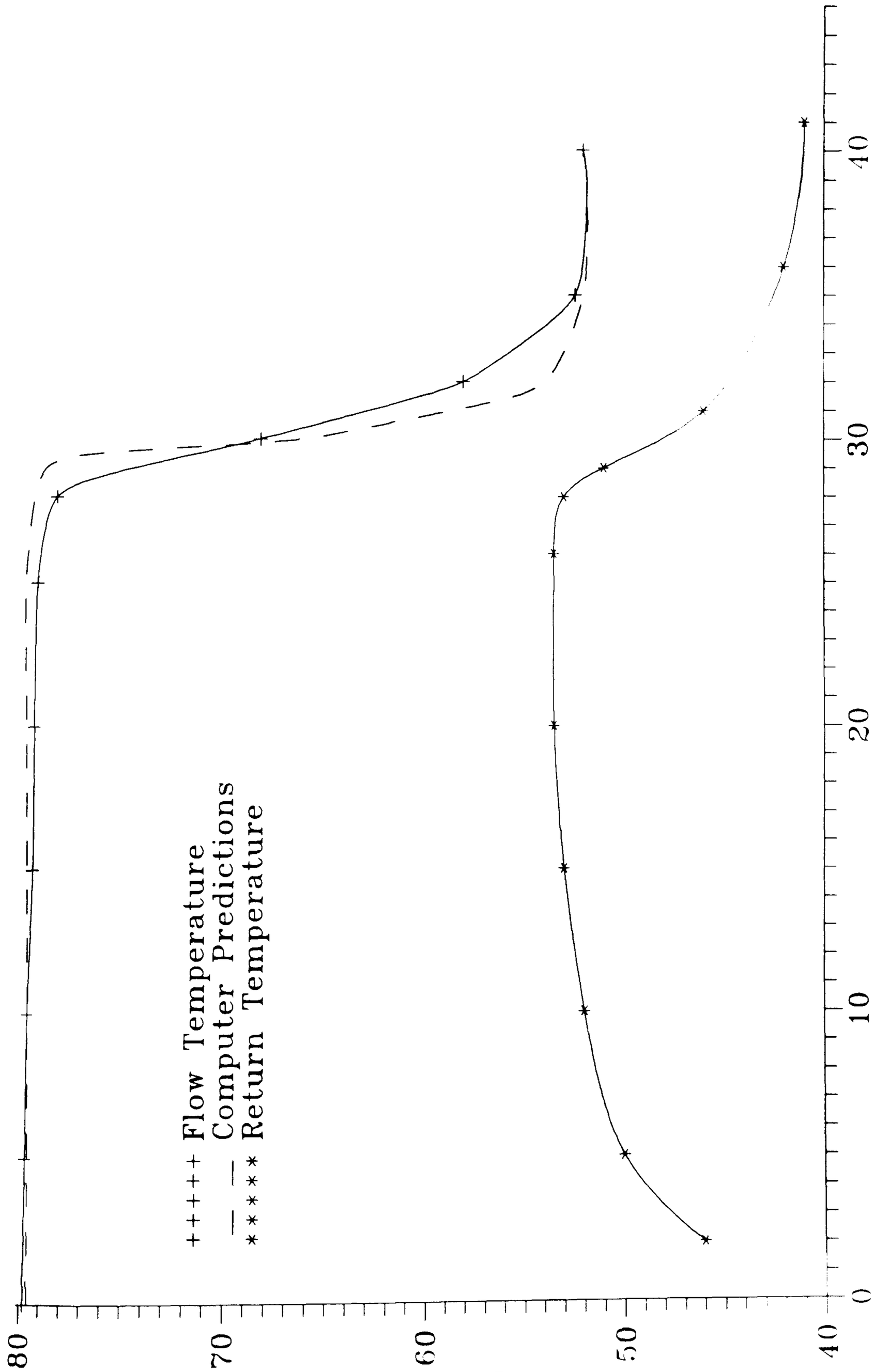


Fig.7.4(a): Flow and Return Temperature Variations During a 6 litre/min Throughflow

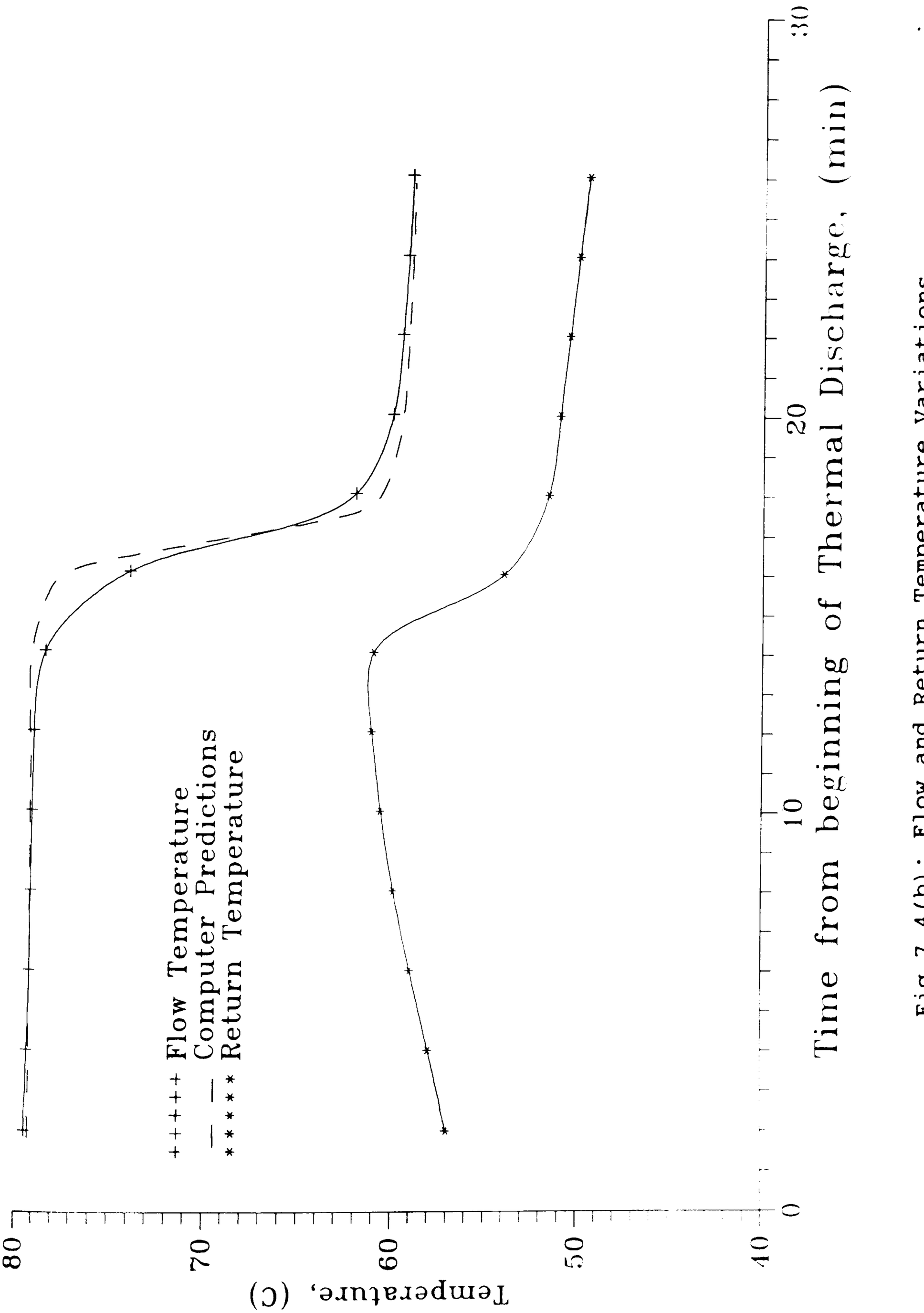


Fig.7.4(b): Flow and Return Temperature Variations During a 12 litre/min Througflow

Fig 7.4(c) to Fig.7.4(f) show the development of the thermocline in the store during the thermal discharge for flow rate ranging from 1.5 to 12 litre/min. The computer predictions are also shown in the corresponding graphs in dash lines.

The predictions of the computer model seem reasonably accurate although they tend to predict a thermocline not as thick as the thermocline observed in the real store. This confirms the previous observations that the computer predictions simulate a store more stratified than the real thermal store. Several factors might affect the thickness of the thermocline in the hot water store. Among these are:

1) the heat conducted along the walls of the store or along the piping of the heat exchanger could if significant modify the thermocline thickness. Fortunately, as the store used for these experiments has a relatively low thermal mass when compared with the thermal mass of the water in the store, it is not likely that a lot of heat is conducted along the walls and the piping.

Moreover, if a significant amount of heat were conducted along the wall of the store and the piping, the thermocline would thicken noticeably when moving upwards in the store. As this does not happens, it can be concluded that the amount of heat conducted along the walls is not high and does not affect the thickness of the thermocline significantly.

2) The cold boundary layer developing along the walls of the store and created by the heat loss from the store to the environment might induce some turbulence in the bottom region of the store. This turbulence mixes the cold water from the return pipe with the warm water already in the store. Although plausible, it is unlikely in these experiments that the momentum of the cold boundary layer could be high enough to create a significant amount of mixing as the thermal store used in these series of experiments is relatively well insulated.

3) The momentum of the inlet flow creates a 'jet' in the bottom region of the store. This jet induces enough mixing and turbulence in the bottom of the store to significantly affect the thermocline. For reasons which will be detailed later, the creation of a jet in the bottom region of the store is the most likely explanation for the difference in mixing observed between the experimental results and the computer predictions. Therefore a detailed analysis of the effect of the jet was carried out.

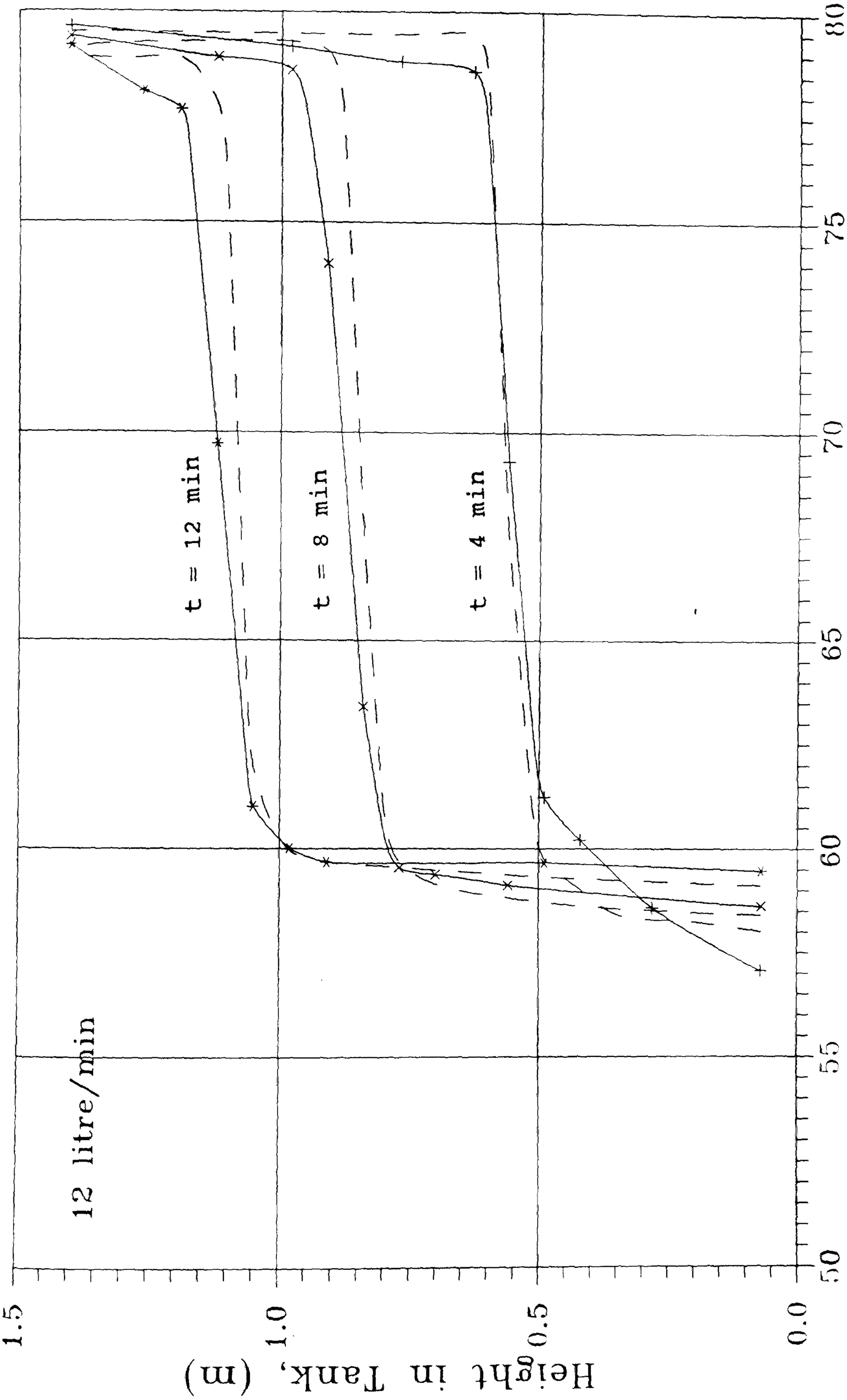


Fig.7.4(c): Development of the Thermocline During a 12 litre/min Throughflow

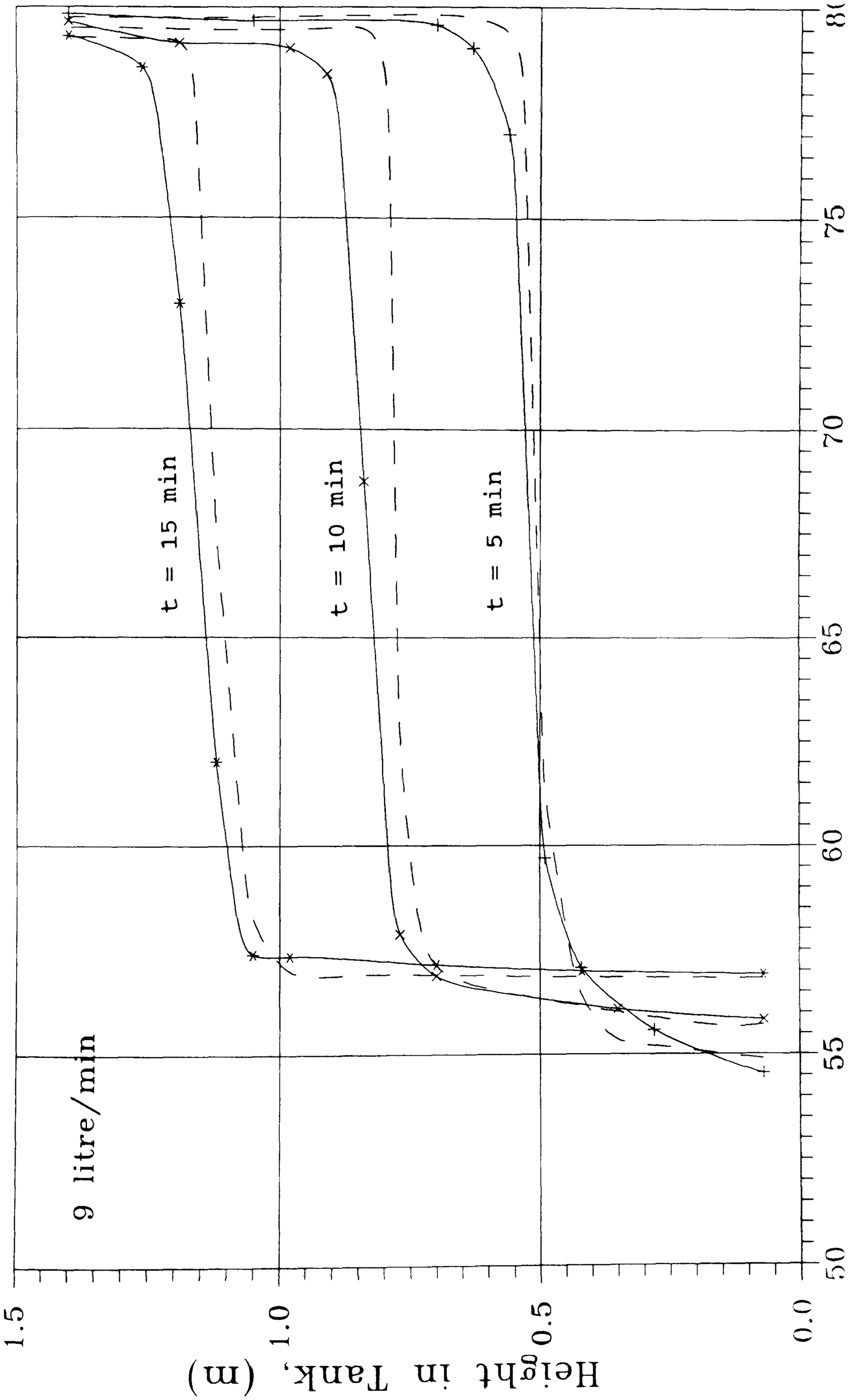


Fig.7.4(d): Development of the Thermocline During a 9 litre/min Throughflow

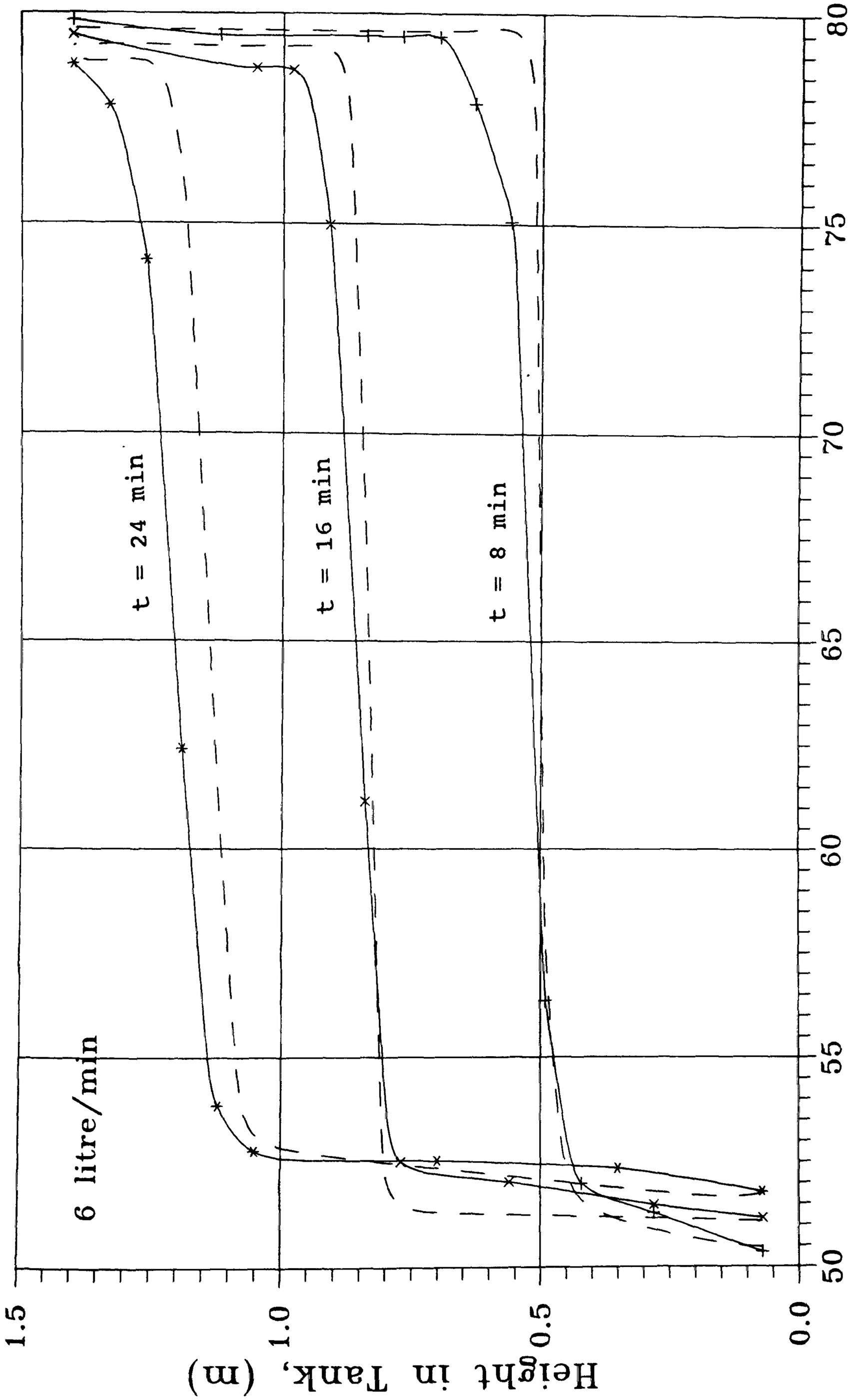


Fig.7.4(e): Development of the Thermocline During a
6 litre/min Throughflow

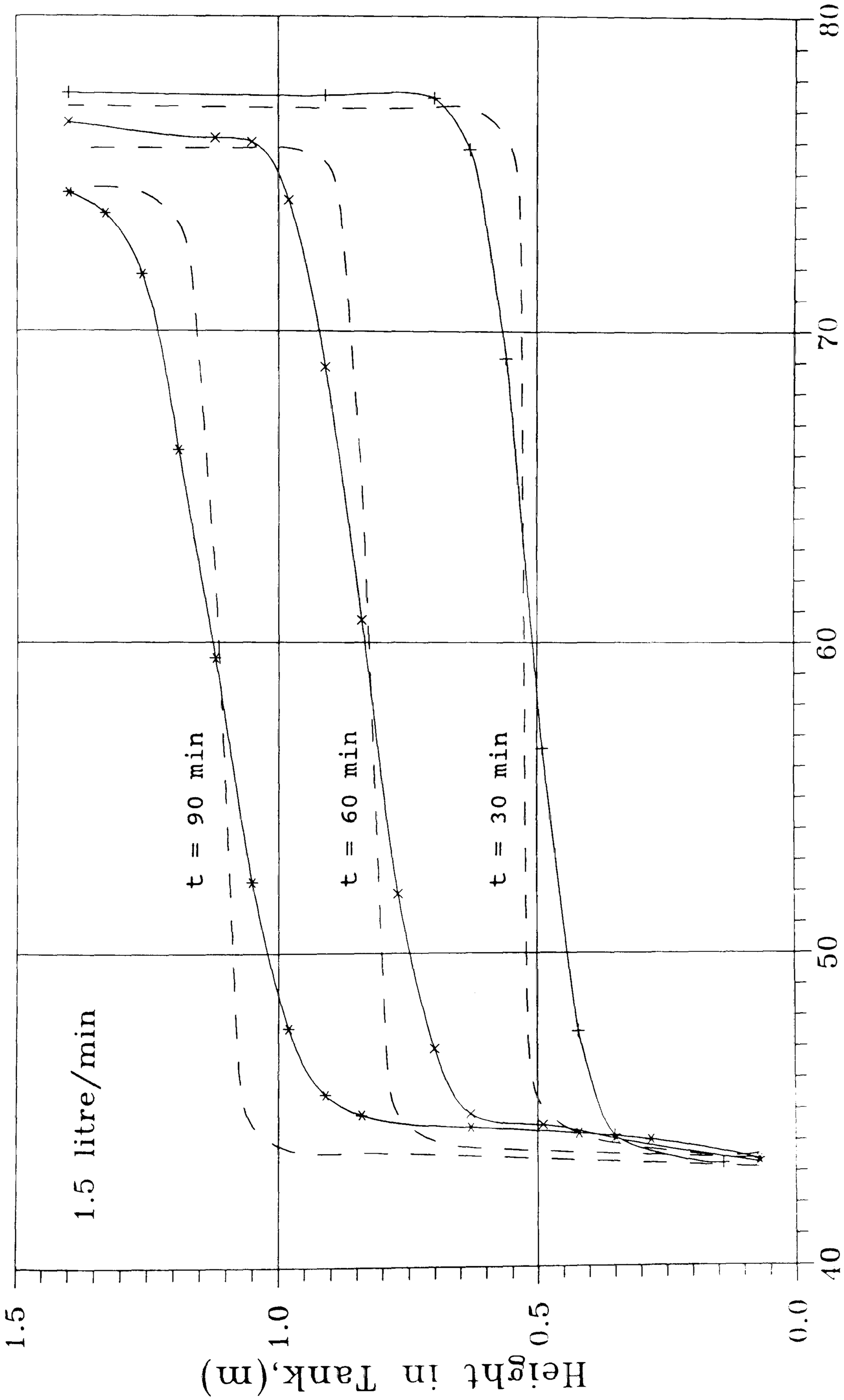


Fig.7.4(f): Development of the Thermocline During a 1.5 litre/min Throughflow

Fig.7.4(g) shows the formation of the thermocline in the bottom region of the store at the beginning of the thermal discharge. for a flow rate of 12 litre/min. At the initial time, The water in the store is at rest and the thermocline is relatively thin (0.02m).

As soon as cold water starts flowing in the store, the thickness of the thermocline increases rapidly as indicated by the change in the slope of the temperature measurements. As time increases, the thickness of the thermocline increases up to 0.2m obtained approximately 40 seconds after the beginning of the thermal discharge. If time increases further, the thermocline moves upwards in the store without thickening significantly.

These observations show that as a result of the water flowing in the store from the return pipe, a significant heat and mass exchange takes place in the bottom of the store. This heat exchange is created by the water flowing into the thermal store from the return pipe which mixes the water in the bottom of the store. This mixing process significantly thicken the thermocline in the bottom of the store. When the thermocline moves upwards, the effect of the mixing induced by the inlet flow on the thermocline is progressively reduced and the thickness of the thermocline remains very nearly constant. From these experimental observations, we can have two important conclusions:

1- <u>it is the mechanism of the formation of the thermocline which plays the major role in determining the thickness of this thermocline</u>.

2- <u>The high momentum of the water entering the store is the dominant factor affecting the formation of the thermocline</u>.

The inlet flow in the bottom region of the store creates a phenomenon which can be assimilated to a 'jet'. This jet induces mixing in two ways: by the mixing induced by the jet itself and by the the mixing indirectly induced by the proximity of the walls of the store.

<u>Mixing induced by the jet</u>:

The formation of jet induced by high momentum incoming flow in water based thermal stores and the mechanism of their formation has been identified by many. When this happens, the incoming fluid entrains significant amount of of fluid initially at rest in the store inducing heat and mass exchanges between the

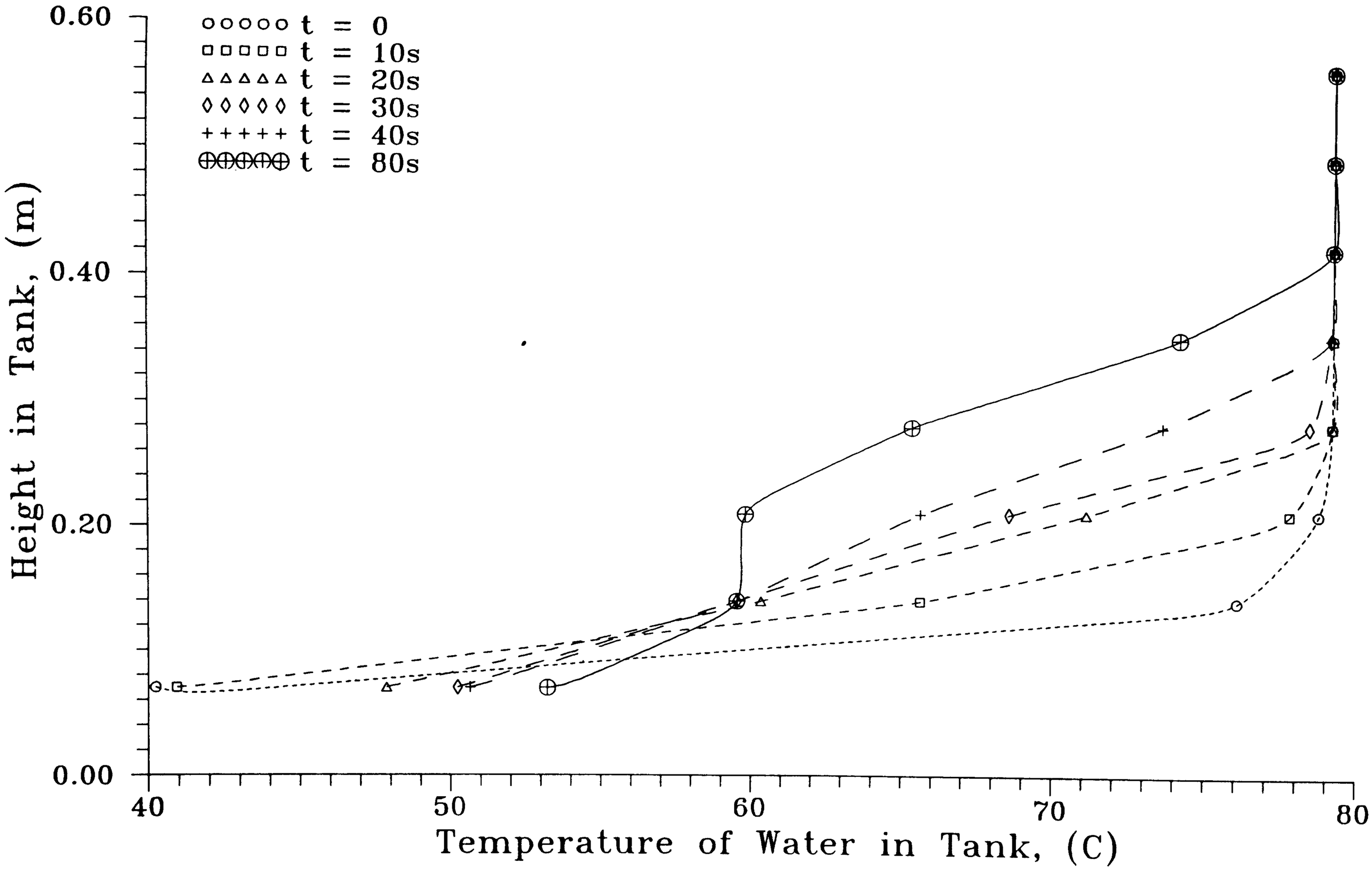


Fig.7.4(g): Formation of the Thermocline in the Bottom Region of the Store

incoming water and the water initially at rest in the store. In the process, the water from the store is entrained in the jet. As the momentum of the jet is relatively constant, this is accompanied by a reduction in the velocity of the jet. This process lasts until the velocity of the jet is low enough to avoid the mixing process.

When this final stage occurs, the nature of the jet changes and in mainly dictated by forces acting in the store other than the momentum force. Among these are the viscous force and the buoyancy force. When there is a significant temperature difference between the incoming flow and the water in the store, the buoyancy forces plays the main role in imposing the type of flow.

Mixing induced by the presence of the walls:

When a jet occurs in an enclosure, the proximity of a wall can affect the development of the jet. This will be particularly relevant when a vertical wall is situated just in front of a high momentum horizontal jet. In this case, the water from the jet has to completely deviate and change its mainly horizontal velocity in horizontal and vertical velocities. This change in direction in the flow is accompanied by a significant amount of mixing. In addition, the vertical velocity component of the deviated jet will tend to considerably disrupt any buoyancy driven flow.

The effect of the proximity of the vertical wall is particularly high in Integrated Thermal Stores which usually have a relatively small diameter and are subject to high velocity inlet flows coming from the space heating system. In addition, the piping of the heat exchanger in the thermal store tends to increase this wall effect.

Finally, the combined effect of the mixing induced by the jet and the mixing induced by the proximity of the walls tend to induce a significant amount of mixing in the bottom region of the store only when the velocity of the jet rapidly decreases when moving upwards in the store. This exactly corresponds to the phenomenon being observed in our present investigation. In addition, visual observations at the outlet of the recirculation pump on the thermal store strongly suggested that the momentum of the jet was significant. Consequently, it was decided to investigate the parameters governing the formation of jets and the effect of their intensity on the performance of the thermal store in the space heating mode.

7.5 Data Analysis

A detailed analysis was carried out to identify the parameters required to predict the likelihood of a jet at the inlet and the effect of this jet on formation of the thermocline and ultimately on the effectiveness of heat recovery from the thermal store when in the space heating mode. This analysis was carried out in three steps:

- identifying the parameters which characterize the formation of a jet at the inlet

- modification of the computer model to take into account the increased mixing created by the jet.

- evaluation of the jet on the Effectiveness of heat recovery from the thermal store.

7.5.1 Richardson Number

The dimensionless parameter indicating the likelihood of a jet at the inlet pipe and its intensity is the Richardson number. The Richardson number is the ratio of the buoyancy force acting within the store by the inertia force created by the momentum of the inlet flow. When written in parameters relevant to our problem (horizontal jet in a stratified medium), the Richardson number can be approximated with a reasonable accuracy by:

$$Ri = \frac{\beta dTgh}{\rho V^2} \qquad (28)$$

When the Richardson number is high, the buoyancy force is predominant in the store and the mixing tends to be avoided. Inversely, if the Richardson number is small, the momentum of the incoming flow cannot be balanced and a jet is likely to occur.

The Richardson number can in theory vary from 0 to infinity. The value of zero is obtained as a limiting case for extremely high momentum and velocity jets in nearly isothermal mediums. To the contrary, extremely high values of the Richardson number can be obtained for extremely low velocity jets in highly stratified mediums.

Extremely high or extremely low values of the Richardson number are only of academic interest. For practical applications in hot water storage tanks, the useful range of Richardson number is approximately from 0.3 to 10 depending on the type of application considered. Therefore the experimental investigation was carried out to cover this range of Richardson numbers.

7.5.2 Mixing Coefficient

In the energy equation, no provision is made for the heat transfer by the process of mixing induced by a jet effect as the only contributions to the temperature gradient are a convection and a diffusion term.

Lavan [21] investigated the mixing created by horizontal jets in cylindrical hot water store of various aspect ratio used in solar heating systems. Their work was based on experimental observations using flow visualization techniques. Their observations suggest that the store can be divided in two zones. A zone at the bottom of the tank extending to a height approximately equal to the diameter of the store where significant mixing occurs and a zone situated above where mixing is not significant. However, their work was mainly qualitative and did not allow to develop useful correlations to characterize the mixing taking place in the store.

A more detailed experimental and numerical analysis of the heat and mass transfer processes in water based thermal stores was carried out by Leyers [19]. This investigation showed that provision can be made for the increased mixing created by jets in water based thermal stores by introducing a correction factor or 'Mixing Coefficient' in the <u>diffusion term</u> of the energy equation. The energy equation when modified with this Mixing Coefficient is then:

$$\frac{\partial T}{\partial t} + Vz\frac{\partial T}{\partial z} = \alpha Cm\frac{\partial^2 T}{\partial z^2} \qquad (29)$$

The Mixing Coefficient, Cm, is a dimensionless number introduced to magnify the effect of heat diffusion. It allows the heat transferred by the process of mixing within the store to be modelled. For laminar flow were the mixing in the store is reduced Cm tends to unity. To the contrary, for a turbulent flow or when a significant amount of mixing takes place in the store Cm tends to infinity.

The effect of the Mixing Coefficient on the thermocline developing in a thermal store is presented in Fig.7.5.2(a). As Cm increases, the thermocline becomes thicker and thicker which approximate the increased mixing in the store created by increased heat and mass transfer processes.

The concept of a Mixing Coefficient to model the heat transfer in thermal stores has been recognized by many authors [16]-[19]. But at the moment, there is no general agreement to the exact formulation for this coefficient. Opel [16] use an 'Eddy Conductivity Factor'(ECF). The Eddy Conductivity Factor can be linked to the mixing coefficient by:

$$ECF = -1 + Cm \qquad (30)$$

It can immediately be seen that for a laminar flow ECF=0 where for a turbulent flow, ECF tends to infinity as the mixing coefficient defined earlier. The use of the Eddy Conductivity Factor instead of the Mixing Coefficient present theoretical advantages in some cases which are out of the subject we investigate and therefore will not be discussed.

Zurigat [18] uses an 'Effective Diffusivity Factor' (EDF) which has the same definition and formulation as the Mixing Coefficient introduced earlier. They investigated in some details the variations of the Mixing Coefficient with the height in the store. They found that the Mixing Coefficient varies in an approximately hyperbolic manner along the height of the store, the maximum value being obtained at the bottom where the jet occurs. From their experimental observations, they developed the following correlation for use in a finite difference formulation of the modified energy equation:

$$Cm = \frac{a}{i} + b \qquad a = \frac{(-1+Cmi)}{(1-1/i)} \qquad b = 5 \qquad (31)$$

Where Cm is the Mixing Coefficient at the considered height in the store, i is the slab number used in the energy equation finite difference formulation and Cmi is the mixing coefficient in the lowest slab in the store. Cmi is a parameter which can be adjusted so that computer predictions correlate accurately experimental data. It reflects the intensity of the mixing created by the jet in the bottom region of the store. The more the mixing in the bottom of the store, the higher the

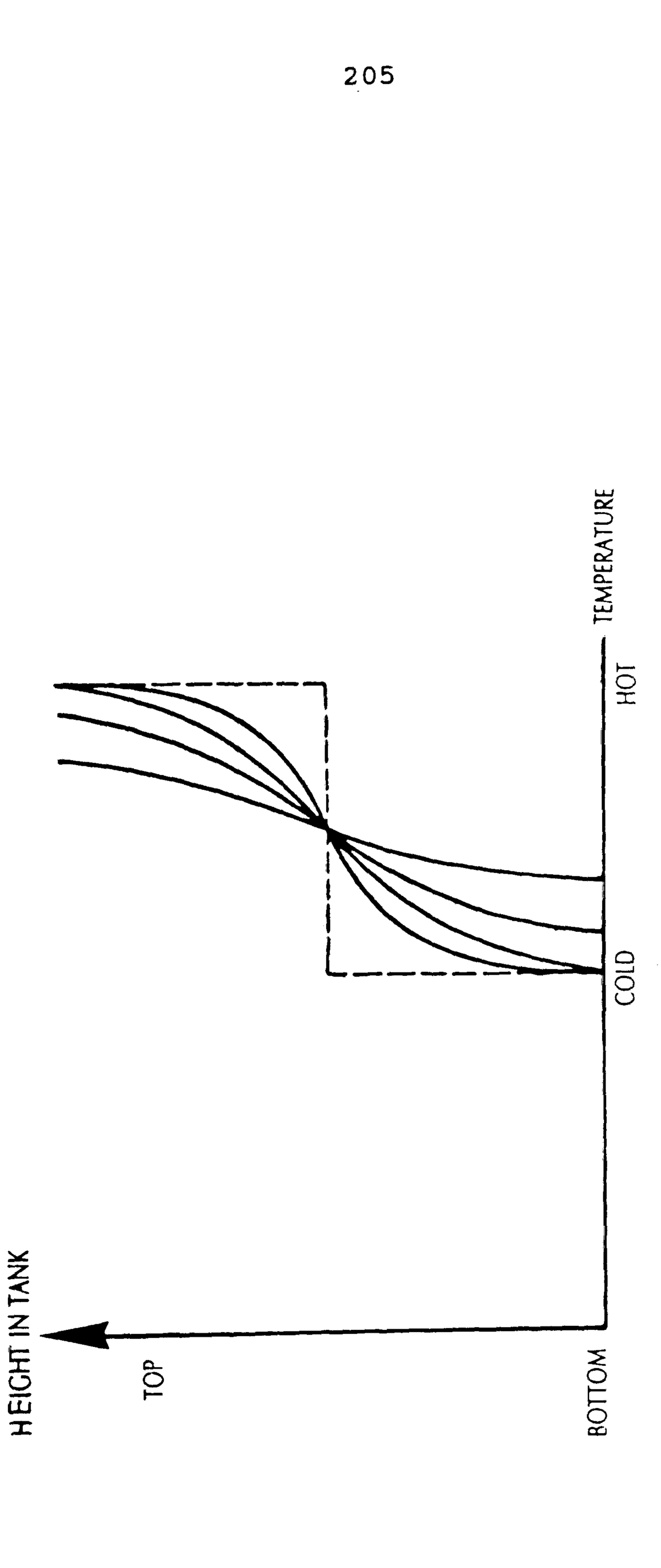


Fig.7.5.2(a): Development of the Thermocline For Increasing Values of the Mixing Coefficient

value of Cmi and consequently of Cm. Additionally, as one moves higher in the store, the effect of the jet is progressively reduced as reflected by equation (31) and Cm decreases.

However, the numerical value of the Mixing Coefficient obtained using equation (31) depends on parameters not reflecting the heat and mass transfer processes in the store and particularly on the number of nodes used in the finite difference formulation. For this reason, this correlation is limited in practical use and a new type of correlation had to be developed. This new correlation for the prediction of the numerical value of the mixing coefficient for use in finite difference or finite element formulations of the energy equations is presented in Fig.7.5.2(b).

The mixing coefficient is maximum at the inlet of the thermal store. It then decreases linearly with the height in the store till it reaches a height equal to the diameter of the store. When moving upwards in the store , the mixing coefficient is then constant and equal to Cmt. This formulation of the mixing coefficient is in agreement with the experimental results of all of the authors previously mentionned and presents the advantage of being independent of the number of nodes used in the finite difference formulation. In addition, as the variations of this mixing coefficient are relatively slow, a wide range of numerical models can easily be accommodated.

The formulation of the mixing coefficient can easily be obtained for a wide range of computer model when the nodes notation is known. For example, for a node i above the bottom of the store when the nodes are equally spaced along the height and numberred in increasing value from the bottom of the store, the following formulation is obtained:

$$Cmi = -\frac{Cmi-Cmt}{H-D}idx + Cmi \qquad \text{for} \quad x<D \qquad (32)$$

$$Cmi = Cmt \qquad \text{for} \quad x>D \qquad (33)$$

By varying the mass flow rate of water in the central heating pipe and the cooling rate of the 50 litre store, the density and velocity of the water returning to the 200 litre thermal store could be varied. This resulted in different types of jet and mixing intensities being achieved in the 200 litre store.

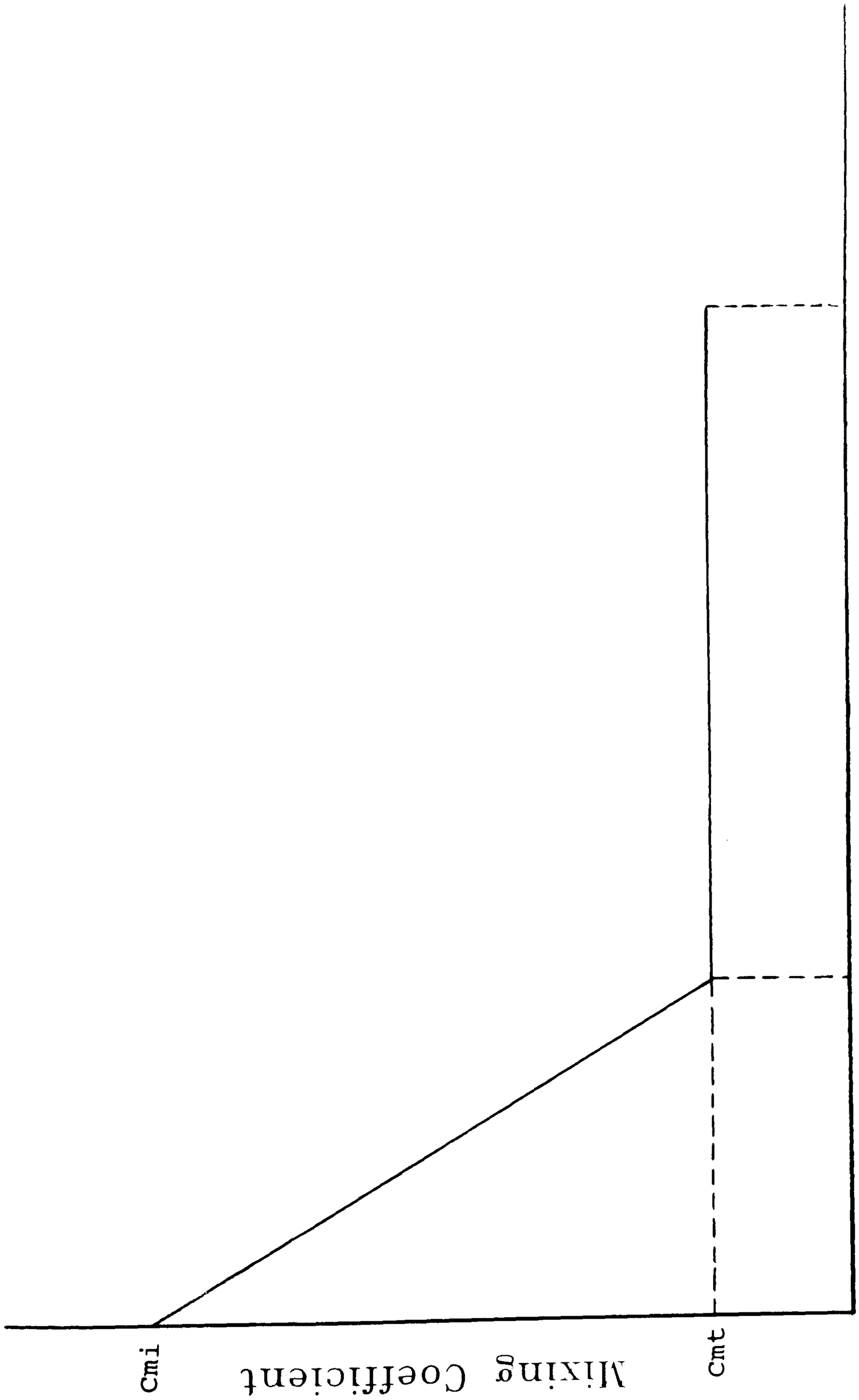


Fig.7.5.2(b): Assumed Variations of the Mixing Coefficient with the Height in the Store

As both the density and the velocity of the jet influence the Richardson number of the inlet flow it was possible without any special arrangement on the rig to vary the Richardson number from 0.3 to 13 approximately. These values covered most of the useful range of Richardson number found during the operation of Integrated Thermal Stores. For this range of Richardson number it was then possible to measure changes in the behaviour of the thermal store in terms of Effectiveness of heat delivery or thickness of the thermocline.

The formulation of the Mixing Coefficient presented in equations (32) and (33) requires the knowledge of two parameters: the mixing coefficient at the inlet pipe Cmi and the mixing coefficient in the upper zone of the store Cmt. Each of these was varied separately in the computer model so that the computer predictions match the experimental results obtained for different jets relatively closely. The final results in terms of Cmi and Cmt are presented in Table VI.

As expected, there is a steady decrease in the mixing coefficients Cmi and Cmt when the Richardson number of the inlet flow increases. This reflects the fact that the buoyancy force acting within the store is predominant for high Richardson number and consequently, the mixing induced by the jet has negligible effects on the flow. There are asymptotic minimum values for Cmt and Cmi which are respectively 11 and 6. These minimum values are obtained for high Richardson numbers achieved with extremely low flow rates (less than 3 litre/min) passing in the 200 litre water store.

From the data represented in Table VI, several type of correlations can be developed between the Richardson number and the Mixing Coefficient. However, of particular relevance are correlations between the Richardson number and the values of Cmi and Cmt which are the parameters already used to model the flow in the store.

These correlations were first sought in a logarithmic form but the corresponding curves did not to match the data closely enough at high and low Richardson number. Therefore correlations curve of an hyperbolic form were developed. These correlations was sought using a desk calculator and therefore are not necessarily extremely accurate. These equations are:

$$Cmt = \frac{2}{Ri} + 5.8 \qquad (34)$$

Richardson Number	Cmi	Cmt	Effectiveness of heat recovery
0.35	170	12	0.21
0.67	100	9	0.32
1.21	80	8	0.50
1.75	50	8	0.69
1.97	30	7	0.75
2.77	25	7	0.79
4.52	17	7	0.92
6.66	14	6	0.97
7.93	13	6	0.98
8.91	12	6	0.99
10.54	11	6	1.00
13.00	11	6	1.00

Table VI : Mixing Coefficient and Effectiveness of Heat Recovery for Various Richardson Numbers of the Inlet Flow

$$Cmi = \frac{60}{(Ri+0.2)^{1.4}} + 9 \qquad (35)$$

These correlations can be very useful when modelling the mixing in thermal stores if the Richardson number of the inlet flow is known. This is very often the case as the Richardson number can usually easily be evaluated from the operating conditions of the thermal store in the space heating mode. However, the numerical procedure used to evaluate numerically the values of the respective mixing coefficient was not extremely accurate. Consequently, care should be taken in using the values of Cmt and Cmi obtained from equations (34) and (35)

7.5.3 Effectiveness of Heat Recovery

Several types of Effectiveness of heat recovery can be defined for an Integrated Thermal Store in the space heating mode. The simplest and most widely used methods are based on measuring the thickness of the thermocline either directly or indirectly. For example by measuring the volume of water undisturbed or the temperature of the water leaving the thermal store or simply the thickness of the thermocline by measuring the temperature of the water in the store.

However, these definitions are usually not entirely satisfactory as the effectiveness can vary according to factors such as the velocity of the water flowing through the store and the temperature difference between the water in the store and the incoming water. Moreover, they are usually difficult to measure experimentally and therefore are not necessarily accurate and easy to use.

However, these definitions of the Effectiveness of heat recovery are not theoretically based on the equations of fluid flow. Therefore, although they can provide useful data on a comparative basis between experiments obtained using similar methodology an equipment, they cannot be used as to compare data obtained from different rigs or using different experimental methodologies. Thus the need to use a definition of the Effectiveness of heat recovery which is independent of experimental parameters, simple to measure and easy to interpret. One way of doing this is to compare the ratio of the slope of the computer prediction at the inflexion point of the thermocline by the slope of the thermocline of the experimental measurements again at the inflexion point.

The computer predictions could have been made using the energy equation only (i.e. with a mixing coefficient of unity). However, in this case, the slope of the computer predictions at the inflexion point of the thermocline would be such that the effectiveness of heat recovery would always be very low. To avoid this problem and obtain an effectiveness of heat recovery varying in the range of 0 to 1, the computer predictions were obtained using a the minimum values for Cmi and Cmt. These were respectively 11 and 6. They correspond to the minimum mixing induced by the jet in the store. With this definition, it was possible to compare the performance of the real store during the thermal discharge with an ideal thermal store operating in the same conditions.

As the ideal thermal store operates in the same conditions as the real store, the effectiveness depends only on how well the real store performs. This presents several advantages over other definitions of the Effectiveness. For example, it is independent of many factors such as the temperature difference between the incoming water and the water in the store.

An additional advantage of this definition of the Effectiveness is that it can easily be measured either by a direct or by an indirect methods of measurements. As the measurements in each of these methods are obtained with different sensors, each of these method is entirely independent from the other. Thus the Effectiveness of heat recovery can be evaluated in two entirely separate ways.

Finally, the definition of the Effectiveness of heat recovery of the thermal discharge process is:

$$Eff = \frac{\text{slope of thermocline of computer predictions}}{\text{slope of thermocline of experimental results}} \qquad (36)$$

The slopes used in the calculation of the Effectiveness of heat recovery can easily been evaluated from the experimental measurements either directly from Fig.7.4(c) and 7.4(d) or indirectly from Fig.7.4(a) and 7.4(b) from the temperature of the water leaving the integrated thermal store.

In the direct method, the respective slopes were numerically evaluated from a first order least square interpolation of the experimental measurements in the surrounding of the inflexion of the thermocline. The slopes were evaluated at approximately 2/3 of the total tank height to avoid interference from both the store inlet and the store outlet.

In the indirect method, a regression analysis from the temperature of the water leaving the store had to be carried out to obtain the temperature profile in the store. Once the profile was known, a method similar to the direct method was used to evaluate the respective slopes.

Both of these methods were used for a the calculation of the effectiveness for 3 experiments selected at random. The purpose of this exercise was to ensure that the analyzing process using both methods gave coherent results. The Effectiveness was calculated when the thermocline reaches the very top part of the store.

The difference in Effectiveness between the two methods was found to be less than 4%, the lower results being always obtained with the indirect method of calculation of the Effectiveness. No explanation was found to this difference in results. However, the results obtained with each methods were relatively close therefore it was decided to use a direct method only to analyse the data as far less computation is required with this method.

The experimental measurements presented in Fig.7.5.3 and in Table V, show that there is a steady increase in the Effectiveness of heat recovery with the Richardson number. When the Richardson number is high, the Effectiveness of heat recovery tends to an asymptotic value of unity.

It is difficult from this data to determine an optimal value for the Richardson number. However, it can be the seen that if the Richardson number is higher than 3, the Effectiveness of heat recovery from the store is higher than 80% which should be sufficient for most engineering purposes. Thus, in the absence of any experimental data, it can be advised at the design stage to size the components of the system so that the Richardson number is greater or equal to three.

Other experimental data available in the literature also suggest that a minimum value for the Richardson number to avoid intense mixing created by incoming jets in hot water store is advisable. Among the recommended values are:

1) Zurigat [18] investigated a large range of Richardson numbers by varying the mass flow rate of water in the inlet pipe and the density of the incoming water using a saline solution. The purpose of the saline solution was to simplify the experimental apparatus required.

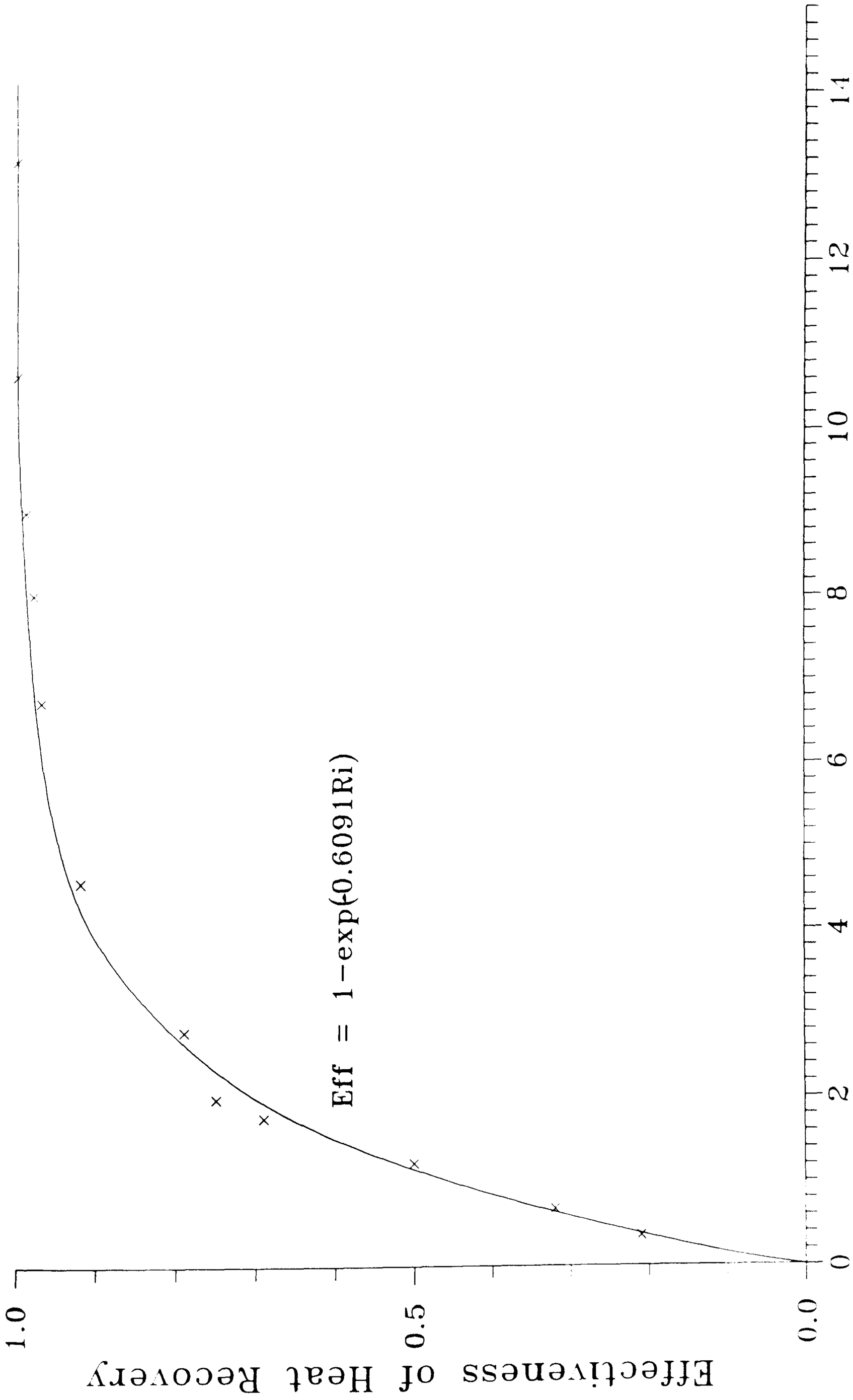


Fig.7.5.3: Effectiveness of Heat Recovery in the Space Heating Mode for Various Richardson Numbers of the Inlet flow

In their experimental work, Zurigat identified the value of the inlet mixing coefficient of 5 as being the minimum attainable. This value was obtained by extrapolating their experimental results to extremely low Richardson numbers. This value of 5 is slightly below the value of 11 measured in the present experimental work. This can easily be explained by the difference in thermal store size used in both experiments (500 litres against 200 litres in the present series of experiments).

Although the purpose of their experimental work was to investigate the effect ot the inlet configuration on the mixing in the store, they failed to develop a correlations between both the inlet mixing coefficient and the Richardson number and the Effectiveness of heat recovery and the Richardson number.

However, they identified a minimum value of the Richardson number of 5 to reach a flow condition above which the 'inlet configuration is unimportant'. This suggests that an Effectiveness close to 100% of the maximum attainable value is achieved. However, detailed experimental measurements in terms of Effectiveness and Richardson number are not given.

2) Abdoly [20], investigated the parameters governing the stratification process in a cylindrical hot water store using a two dimensional axisymetric model. Their computer model was used to simulate hot water storage tanks for solar energy applications.

Their experimental data covered simple types of thermal discharges such as natural cooling and statically and dynamically stratified stores. The height/diameter ratio of their storage tank, was around 6 which would have been sufficient to model the thermal discharge. However, their experimental data were not found to correlate accurately their experimental predictions. The reason for this is thought to be the absence of a mixing coefficient in their computer model which made it impossible to account for the increased mixing in the store.

However, an interesting part of their work is involved in the use of perforated horizontal baffle arrangement situated close to the store inlet to reduce mixing. Using this baffle arrangement the Richardson number of the inlet flow was around 4 and a relatively high degree of stratification was observed in the store which suggests that mixing is significantly reduced.

3) Cole [17] also identified a decrease in the mixing in the tank for increasing Richardson numbers number by comparing experimental results in hot water storage tank for solar heating applications with computer predictions from a one dimensional explicit finite difference model.

Unfortunately, as their definition of the Effectiveness is different and their computer model uses a different definition of the Mixing Coefficient, a direct comparison with their experimental result is difficult. However, they found that a value of the Richardson number of around 5 or above is necessary to avoid significant mixing in the store.

Using a value of the Richardson Number of 5, recommended by most of these authors [17],[18],[20], the corresponding Effectiveness of heat recovery using our experimental measurements and computer predictions can be estimated to be above 95%. This is therefore sufficiently close to the maximum limit to consider that at this stage only minor improvements in the Effectiveness of heat recovery can be achieved by further increases in the Richardson number.

At present, no simple theoretical approach can accurately predict the variation of the Effectiveness with the Richardson number. The only method to approach the theoretical background would be to solve the 3 dimensional equations of motion which would be extremely complicated.

However, from the definition of the Effectiveness and the range of Richardson number investigated, it was reasonable to seek a mathematical correlation between these dimensionless numbers in an exponential form. This was sought in the form:

$$Eff = 1-\exp(bRi) \qquad (37)$$

The value of the coefficient b was estimated from the experimental data using a least square method in logarithmic coordinates. The final results is:

$$b = -0.609$$

The best fit curve is represented in Fig.7.5.3. A dispersion analysis showed that the average dispersion of the experimental data around the correlation curve is 2.381%. This suggests that the correlation matches the experimental points relatively closely in the range of Richardson number from 0.3 to 13.

7.6 Discussion

There are several ways in which the Effectiveness of heat recovery from an Integrated Thermal Stores in the space heating mode can be improved.

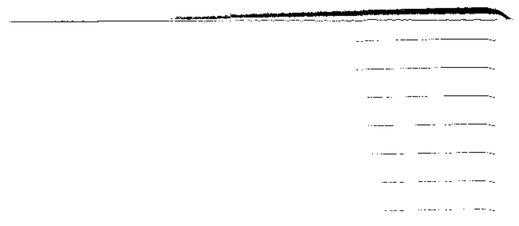
One of this way is to avoid the jet at the bottom of the store which can only be achieved by increasing the Richardson number at the inlet. The definition of the Richardson number given in equation (28) suggests many ways in which this can be done:

1) Increase the height of the store. This increases the buoyancy term appearing in the numerator of the Richardson Number and therefore increases the Richardson number. Although it might sometimes be possible to increase the height of the store, there are both practical and theoretical limitations to the improvements which can be achieved with this type of method.

2) Increase the temperature difference between the incoming flow and the water in the store. This also increases the buoyancy term appearing in the numerator of the Richardson number and therefore increases the Richardson number. Although possible in some circumstances, this would again be difficult to achieve in practice as the temperature difference between the flow and return in a central heating systems is dictated by many practical considerations.

3) Reduce the velocity of the water in the inlet pipe. This reduces the momentum of the inlet flow and therefore increases the Richardson number. This can be achieved either by reducing the flow rate in the inlet pipe or by increasing the inlet pipe diameter. Increasing the pipe diameter seems the easier to achieve and the more effective as the Richardson number depends on this dimension raised to the power 4.

An other way in which the Effectiveness of heat recovery might possibly be increased without changing the Richardson number at the inlet pipe is by the use of a diffuser or a baffle arrangement at the inlet pipe.



The purpose of the diffuser is to avoid that the turbulence created by the momentum of the inlet flow generates a lot of mixing in the bottom of the store. This can be achieved by channelling the jet until its velocity is sufficiently reduced. Alternatively they can ensure an even upward velocity distribution of the water in the store. The use of a diffuser arrangement can also simplify the design of the store itself.

Several types of horizontal and vertical baffles and diffusers arrangement have been investigated experimentally and theoretically [11],[18],[20]. Example of such diffusers can be found in Fig.7.6(a).

Kamandari [3] reported some significant improvements in the mode of operation of water based thermal store by the combined use of a distributor and a mesh as represented in Fig.7.6(b). Unlike the present investigation, their definition of the Effectiveness of heat recovery was based on the thickness of the thermocline which makes any direct comparison of the result difficult. However, they reported an increase in 12% in the Effectiveness of heat recovery from a 200 litre hot water store. The Richardson number of the inlet flow was relatively low in the absence of baffles (in the range 0.3-0.5) which made it possible to successfully use this diffuser.

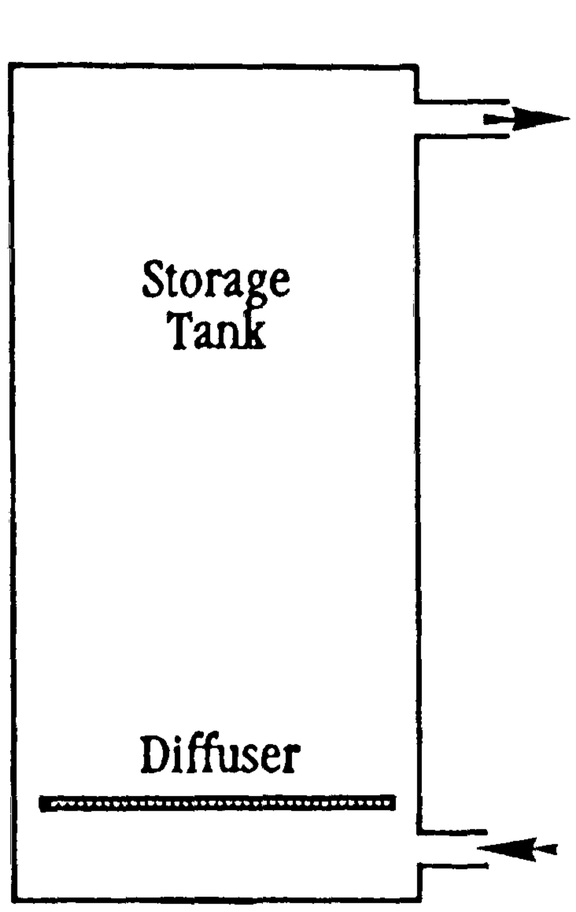
Several types of diffusers have been investigated numerically by Lin [11] using 3-D computer simulation of the equations of motion in cylindrical hot water stores. The main conclusions of this investigation were:

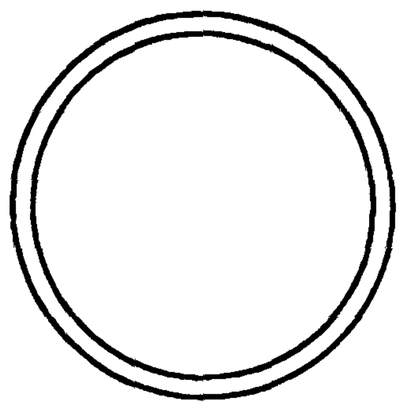
- horizontal baffles arrangement are to be preferred over vertical baffles arrangement.

- non conductive material are to be preferred for these baffles.

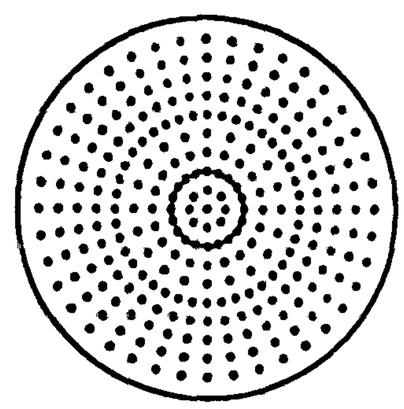
- the extra pumping power required to overcome the pressure drop created by a baffle arrangement can be extremely small if the baffle is designed correctly.

In our case it is likely that improvements achieved by the use of a baffle or by the reduction of the Richardson number to avoid the jet at the inlet pipe would be higher that the observed improvement by the previous authors as it might improve at the same time the space heating and domestic hot water heat delivery.

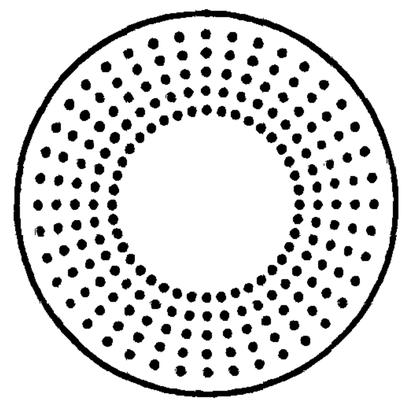




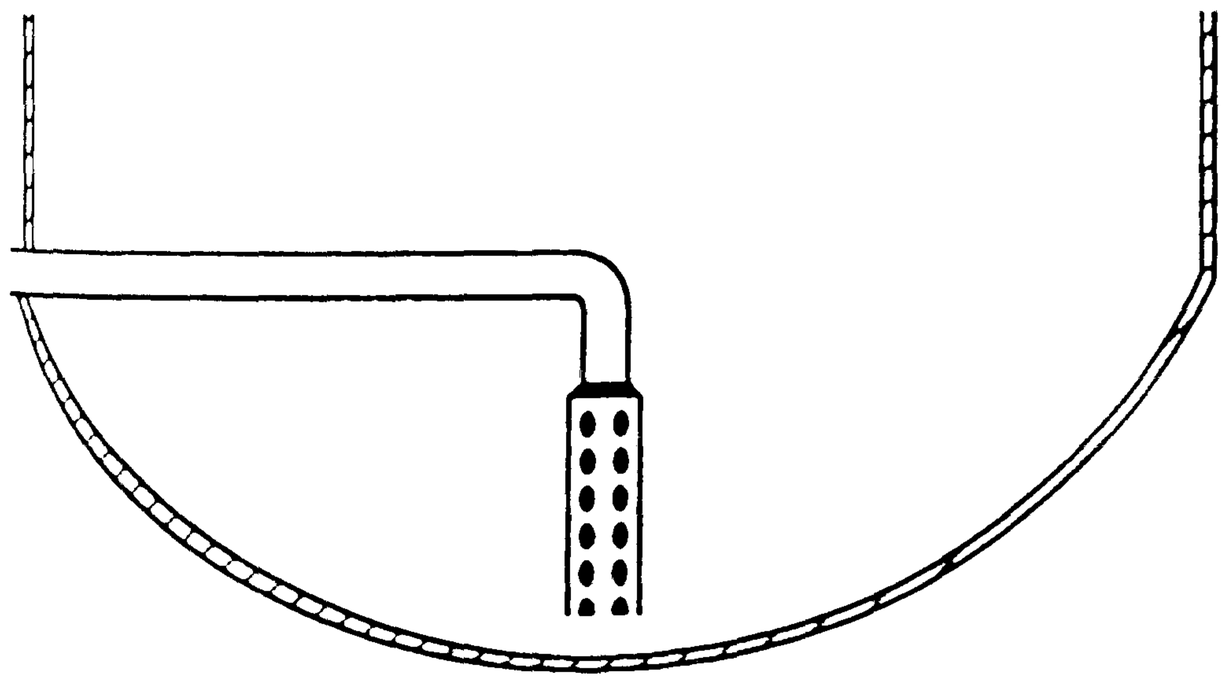
Solid
Diffuser



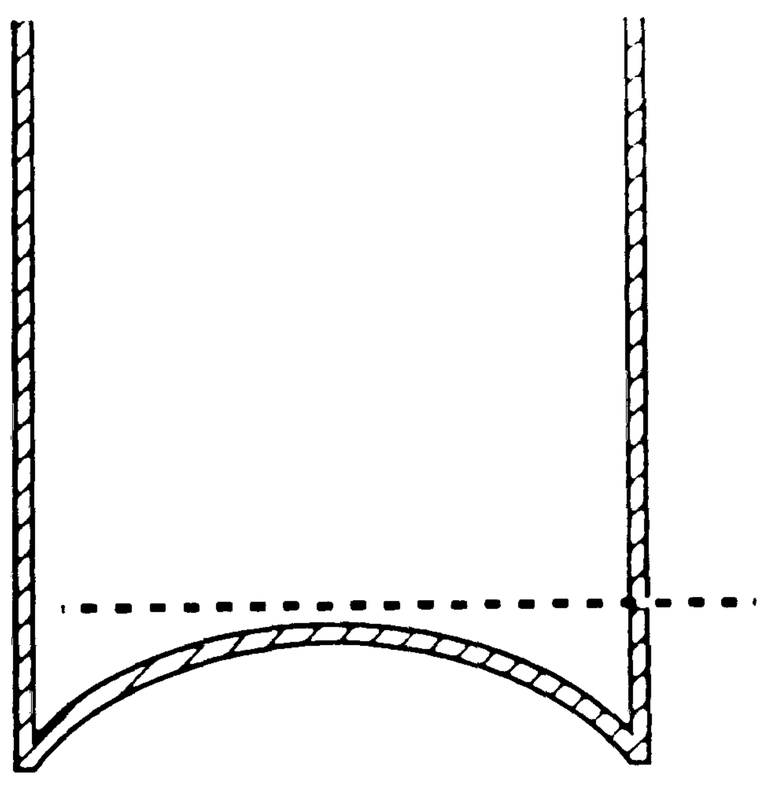
Perforated
Diffuser



Perforated Diffuser
with Solid Centre



Distributor



Perforated Baffle

Fig.7.6(a): Examples of Baffles, Diffusers and Distributors
Arrangement for Hot Water Store

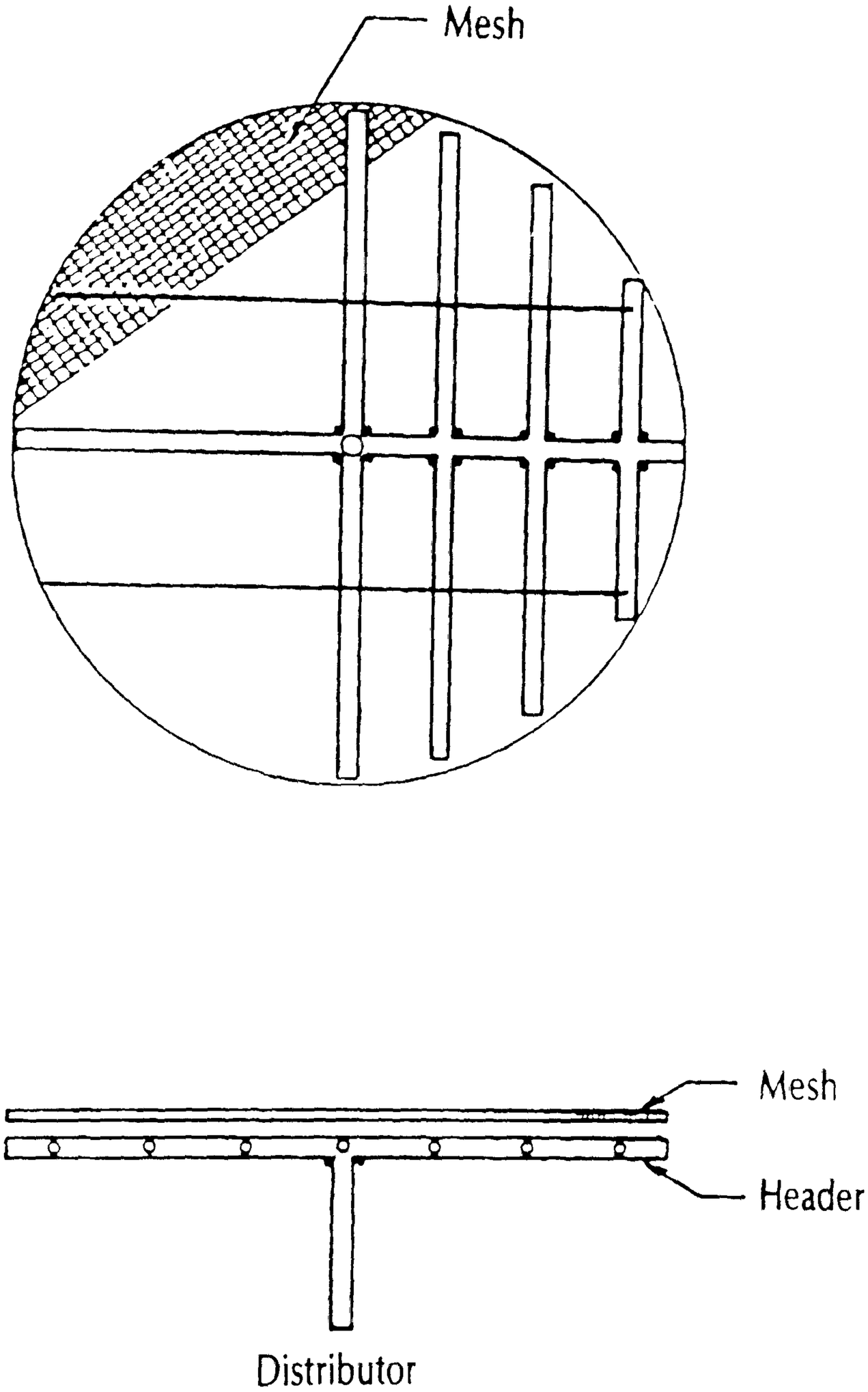


Fig. 7.6(b). Distributor with Mesh [3]

By comparing the computer predictions and the experimental results for a flow rate of 6 l/min and 12 l/min, the potential gain in terms of extra heat delivered by the stratified store can be evaluated.

Assuming that the heat reclaimed below 65°C can not be useful for space heating purposes, the difference in the curves presented in Fig.7.3(a) and Fig.7.3(b) suggest that a stratified store would deliver respectively 6% and 8% more heat for space heating than the present store. These values of 6% and 8% should be regarded as the maximum improvements which can be reached by the use of a diffuser at the return inlet which would avoid mixing.

Finally, when the domestic hot water and space heating demands are simultaneous, the reduction of the mixing created by the jet in the bottom region of the store would improve stratification. Consequently, some improvements in the domestic hot water delivery can be expected. These improvements are difficult to evaluate. They are likely to be small but always worthwhile.

7.7 Other Experimental Results

7.7.1 Heat Recovered at the End of the Thermal Discharge

Using the same experimental data as the one used to plot Fig.7.4(a) which corresponds to a 6 litre/min flow rate of water in the return pipe, the total amount of heat delivered by the central heating system at the end of the thermal discharge was calculated. This was obtained by numerical integration of the area between the curves representing the flow and return temperatures between t=0 and t=30 minutes, time at which the thermocline reaches the top of the store. The final result was 19.7 MJ.

The amount of heat stored by the water in the tank and by the tank itself between the initial store temperature of 80°C and the average return temperature of 52°C is:

water :	200x4180x(80-52) =	23.4 MJ
store :	55x385x(80-52) =	1.3 MJ
	Total	24.7 MJ

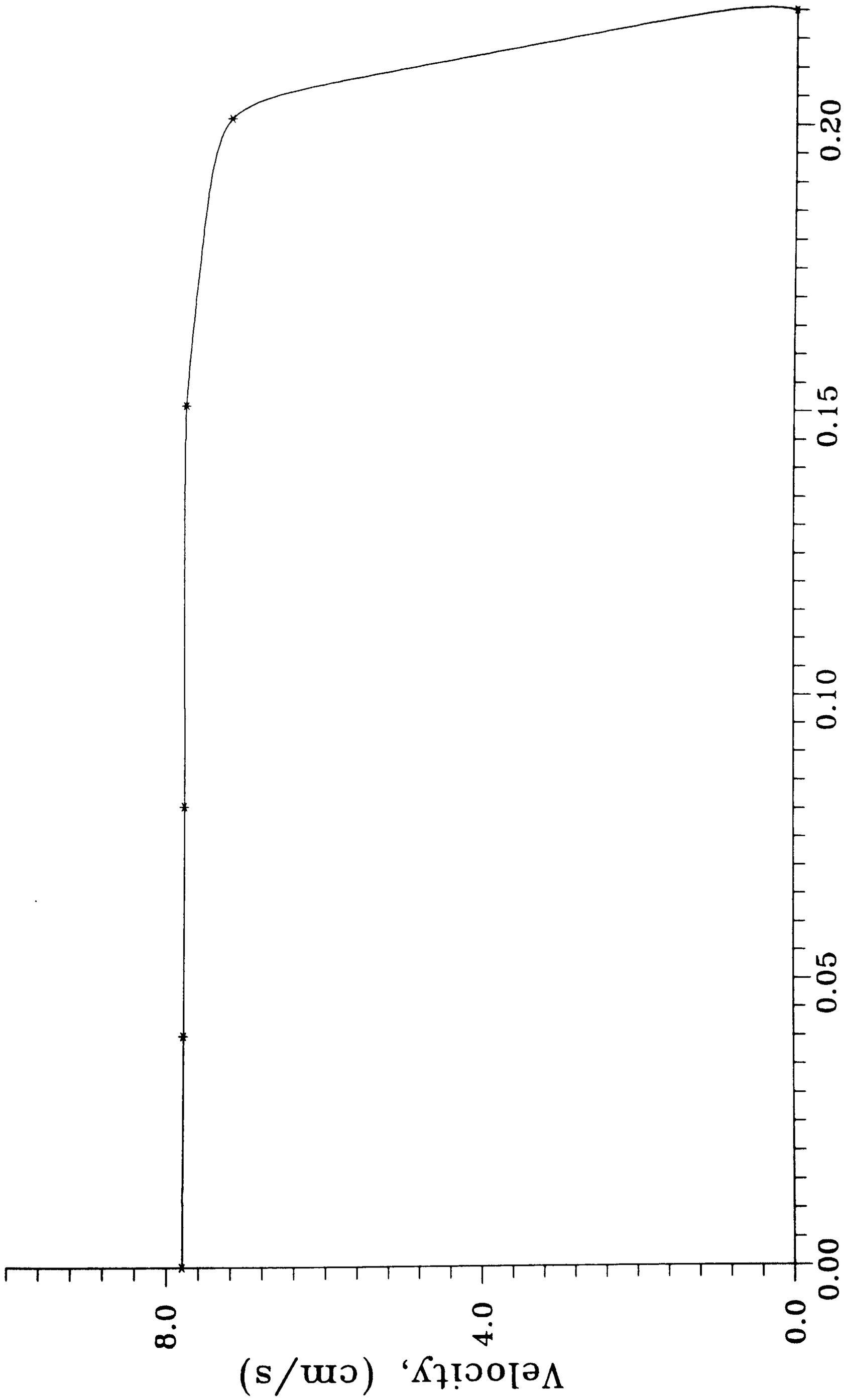


Fig 7.7.2: Measured Velocity Profile in Hot Water Store for a 12 litre/min Throughflow

By comparing the values of 19.7 MJ and 24.7 MJ, it can be seen that 79.7% of the heat initially stored in the tank is recovered for space heating purposes. This value of 79.7% has to be compared with the values of 48.2% which was obtained for a thermal discharge achieved by means of the finned tube heat exchanger coil. It suggests that during a thermal draw-off for space heating, the thermal capacity of the storage system is used in a much more effective way than during a domestic hot water draw-off.

From the domestic hot water and space heating demands pattern presented in Fig 1.2(a) and 1.2(b) it can be clearly seen that the thermal storage is required primarily to compensate for the fluctuations in the domestic hot water production where unfortunately the Effectiveness of heat recovery is the lower.

Consequently the main area to investigate to improve the overall cost/effectiveness or 'performance' of the Integrated thermal store is the domestic hot water draw-off as it is during the domestic hot water draw-off that the thermal store is at the same time the less efficient and the most subjected to a high heat demand.

7.7.2 Velocity profile

By comparing the temperature measurements taken by thermojunctions fixed on different rods in the store, it was possible to estimate the upward velocity of the thermocline at different locations in the store.

This velocity was measured in the upper part of the store to avoid interference by the mixing created by jet in the bottom region of the store. The results obtained from these measurements are summarized in Fig.7.7.2 where the upward velocity in the store during the thermal discharge is presented for different radius.

The measured velocity profile is relatively flat which suggest that although the water flowing in the store corresponds to a laminar flow (The Reynolds number based on the store diameter is less than 2000), the buoyancy force acting within the store plays a dominant role in equalizing the upwards velocities away from the purely theoretical parabolic profile expected in fully developed isothermal laminar flows in circular pipes. In addition, the flat velocity profile justifies 'a posteriori' the assumption of the one dimensional flow in the thermal store which was used to simplify the equations describing the motion of the fluid in the thermal store.

Finally, some provision should be made in interpreting these results as there were only 4 rod in the store therefore the upwards velocity could only be estimated at 4 different radius and consequently very few experimental points are available. More experimental data would be required to develop correlations giving the velocity profile in the thermal store.

7.8 Conclusions

A computer model based on a one dimensional formulation of the energy equation was developed to predict the behaviour of an Integrated Thermal Store in the space heating mode. This computer model uses a new method of solution which combines an implicit and explicit finite difference formulation of the energy equation. This new method of solution has many advantages over existing methods in solving the same type of problems. These advantages are: low computation time and memory requirements, simplicity of programmation and good accuracy of experimental predictions.

Experiments were carried out which consisted in passing relatively cold water in an upwards manner through a 200 litre hot water store. The experimental parameters were set to the simulate the behaviour of an Integrated Thermal Store when in the space heating mode.

The comparison of the experimental results with the computer predictions clearly showed the formation of a jet in the bottom region of the store if the Richardson number of the inlet flow is too low. This jet induces a significant amount of mixing which reduces the Effectiveness of heat recovery from the store. The jet can be avoided either by increasing the Richardson number of the inlet flow or by the use of a baffle arrangement. A value of the Richardson number higher than 3 is desirable to achieve a high Effectiveness of heat recovery from the thermal store.

From the experimental data, several types of correlation were developed. This included modifying the computer model to take into account the mixing created by the jet in the bottom region of the store. In addition, a simple correlations was developed between the Richardson number and the Effectiveness of heat recovery. This correlation is :

$$Eff = 1 - \exp(-0.609Ri)$$

References

[1] Garg,H.P., Mullick, S.C. and Bhargava, A.K.
Solar Thermal Energy Storage
D.Reidel Publishing Company, Dordrecht, pp 82-153, 1979

[2] Abhat, A.
Short Term Thermal Energy Storage
Energy And Buildings, Vol 3, 1981, pp 49-76

[3] Kandari, A.M.
Thermal Stratification in Hot Storage Tanks
Applied Energy, Vol 35, pp 299-315, (1990)

[4] Yoo, J., Wildin, M.W. and Truman, C.R.
Initial Formation of a Thermocline in a Stratified Thermal Storage Tanks
ASHRAE trans, Vol 92(2a), pp 280-292, 1986

[5] Straub, J. and Staudt, A.
Convective and Conductive Heat Transfer in Hot Water Storage Tanks
New Energy Conservation Technologies and their Commercialisation
J.P.Millhone, E.H.Wilis, Berlin, 1981, pp 715-725

[6] Loehrke, R.I.
Stratification Enhancement in Liquid Thermal Storage Tanks
J.Energy, Vol 3, N°3, pp 129-130, 1979

[7] Philips, W.F. and Dave, R.N.
Effects of Stratification on the Performance of Liquid Based Solar Heating Systems
Solar Energy, Vol 29, N°2, pp 111-120, 1982

[8] Baines, W.D., Martin, W. and Sinclair, L.A.
On the Design of Stratified Thermal Storage Tanks
ASHRAE trans, Vol 88(2), 426-439, 1982

[9] Gray, D.D. and Giorgini, A.
The Validity of the Bousinesq Approximation for Liquids and Gases.
Int. J. Heat Mass Transfer, Vol 19, pp 545-551, 1976

[10] Sha, W.T., Lin, E.I.H. and all
COMMIX-SA-1: a Three Dimensional Thermodynamic Computer Programm for Solar Applications
Argone National Laboratory, Report ANL-80-8, US, November 1980

[11] Lin, E.I.H. and Sha, W.T.
Effect of Baffles on Thermal Stratification in Thermocline Storage Tanks
Proc. International Solar Energy Society, Atlanta, 1979, ISBN 0080250742, pp 586-590

[12] Hyun, J.M. and Hyun, J.C.
Adjustment of a Thermally Statified Fluid in a Container with Vertical Throug-Flow
Int J Heat Mass Transfer, Vol 29, N°10, pp 1487-1493, 1986

[13] Lin, Y.S. and Akins, R.G.
Thermal Descritpion of Pseudo Steady State Natural Convection Inside a Vertical Cylinder
Int.J. Heat Mass Transfer, Vol 29, N°2, pp 301-307, 1986

[14] Barakat, H.Z. and Clark, J.A.
Analytical and Experimental Study of the Transient Laminar Natural Convection Flows in Partially Filled Liquid Containers
Proc. 3rd International Heat Transfer Conference, Chicago, 1966

[15] Cabelli, A.
Storage Tank - A Numerical Experiment
Solar Energy, Vol 19, pp 45-54, 1977

[16] Oppel, F.J., Ghajar, A.J. and Moretti, P.M.
Computer Simulation of Stratified Heat Storage
Applied Energy, 23 (1986), 205-224

[17] Cole, R.L. and Belinger, F.O.
Thermally Stratified Tanks
ASHRAE Trans, Vol 88 (2), pp 1005-1016, 1982

[18] Zurigat, Y.H., Ghajar, A.J. and Moretti, P.M.
Stratified Thermal Storage Tank Inlet Mixing Characterisation
Applied Energy, Vol 30, 99-111, 1988

[19] Leyers, H.J., Scholz, F. and Tholen, A.
Analytical and Experimental Determination of The Temperature Distribution in Stratified Hot Water Store.
Energy Conservation in Heating, Cooling and Ventilating Buildings, Hemisphere Publishing Corporation, Washington, 1982

[20] Abdoly, M.A., and Rapp, D.
theoretical and experimental study of stratified thermocline storage of hot water
Energy Convers Mgmt, vol 22, pp 275-285 , 1982

[21] Lavan, Z. and Thompson, J.
Experimental Study of Thermally Stratified Hot Water Storage Tanks
Solar Energy, Vol 19, pp 519-524, 1977

Chapter 8

Discussion and Recommendations

8.1 Integrated Thermal Store

The experimental and theoretical work which were carried out have greatly improved the fundamental knowledge of several very important areas of the behaviour of Integrated Thermal stores either in the space heating or hot water delivery mode of operation.

(i) Correlation for the prediction of the external heat transfer coefficient of the heat exchanger coil and its effectiveness of heat recovery have been developed. Consequently, the performance of the heat exchanger can now be evaluated with a reasonable accuracy using the correlations without requiring expensive practical experimentation. As a consequence, the design of the heat exchanger coil to match a specific duty is now greatly facilitated.

(ii) Heat transfer correlations have also been developed to predict the Effectiveness of heat recovery from the Integrated Thermal Stores when in the space heating mode. Example of simple computer models based on these heat transfer correlations have been used to predict the effectiveness of heat recovery from hot water stores again in the space heating mode of operation.

When (i) and (ii) are combined, it can be seen that there are are now correlations which can be used to predict the overall performance of Integrated Thermal Store in a very wide range of operating conditions. This can be particularly helpful for theoretical considerations when combined with other data at the design stage by avoiding, to a certain extend, the use of expensive practical experimentation.

For example for a given type of dwelling, the space heating load pattern can usually be evaluated once the type of building and the weather data are known. For the same dwelling, the domestic hot water load can be evaluated from the number and lifestyle of the occupants. Both the space heating demand and the domestic hot water demand can be then be used with the set of heat transfer correlations which has been developed to evaluate, on a theoretical basis, the performance of an Integrated Thermal Store allowing for a wide range of investigations to be carried out. For example numerical investigations can be used to optimize the size of the heat exchanger or the thermal store.

8.2 Potential Improvements to Existing Integrated Thermal Stores

Apart from the development of correlations, the experimental work which was carried out suggests that some improvements to the design of the Integrated Thermal Store can be made. Although a detailed cost evaluation would be required to exactly evaluate the cost/benefits provided by implementing some of these changes some simple classification can be made for preliminary conclusions to be drawn. This simple classification compares on a crude basis the different options for improving Integrated thermal store. By order of increasing cost/benefits the suggested changes are:

1) Raise the upper heat exchanger coil in the store.
This would eliminate a dead zone located above the upper heat exchanger. This dead zone act as a heat trap and is extremely unbeneficial as it increases the size of the store and increases the heat loss to the environment unnecessarily. By raisng the upper heat exchanger to the very top of the integrated thermal store, the dead zone can be eliminated. The corresponding improvement in the amount of water delivered during a thermal discharge would be approximately 5%.

2) Raise the lower heat exchanger coil in the store.
Raising the lower heat exchanger in the store would improve the performance of heat delivery. This can be achieved by shortening the pipes between the two coils constituting the heat exchanger thus reducing the amount of copper required for the heat exchanger. This will slightly reduce the manufacturing cost. The combination of a higher heat exchanger performance with a lower manufacturing cost makes this change extremely attractive.

3) <u>Change the design of the three way valve</u>.
Changing the characteristic of the three way valve located at the outlet of the heat exchanger can be achieved without change in the manufacturing cost of the present valve. Only minor components have to be slightly redesigned. The improvements which can be expected are however relatively small as this would be sensible only if a change in the domestic water usage is acceptable (for example lowering the temperature of the water delivered to the taps).

4) <u>Reduce the mixing at the inlet</u> either by increasing the Richardson Number or by using a baffle or diffuser arrangement. This could be achieved at relatively low cost as simple types of baffles or diffusers can be used. A slight improvement in the amount of heat delivered for space heating can be expected. Thus the cost/effectiveness ratio is likely to be favorable.

5) <u>Increase the insulation of the store</u>
This is not exactly a direct conclusion of this analysis but it is suggested by some experimental observations. The optimal thickness of insulation to use can be obtained by an energy balance at the wall of the store. This should be around 0.1 m.

Even by implementing all these, the total increase in performance of the integrated thermal store is expected to be small. However, as these improvements can be achieved at very low cost. This very low cost is likely to make these improvements relatively cost effective thus worthwhile.

Other changes in the design or operation of the store suggested by this work do not seem to be beneficial. These changes include the use baffles to improve the performance of hot water delivery. Horizontal baffles to reduce the mixing in the store during a thermal discharge achieved using the heat exchanger coil does not seem attractive. Although some improvement in the amount of heat delivered for domestic hot water can be expected based on theoretical considerations, no significant improvement was measured experimentally. Thus although the cost of a horizontal baffles would be relatively small, the cost effectiveness ratio makes its use extremely unattractive.

Finally, although only a limited experimental evaluation of the duct arrangement was carried out, the use of a rectangular duct does not seem promising as it would at the same time increase the cost of the integrated thermal store and reduce the Effectiveness of heat delivery.

8.3 Future Developments and other Areas to Investigate

Future developments can take place in many areas and only a brief overview of these areas can be given.

The first development is the implementation of the changes in the design of Integrated Thermal Stores which are suggested by this work. These include particularly raising the upper and lower heat exchangers coils in the store. Most of these changes can be implemented relatively easily, although manufacturers of Integrated Thermal Stores or of other components (such as the mixing valve) might have to be contacted.

Following this change in design of ITS, the evaluation of the performance of the new designs will be necessary. Consequently, the future work will have to include the laboratory and the field evaluation of the new 'improved' prototypes in terms of amount of extra heat recovered from the store or comparison with the old version of Integrated Thermal Store to evaluate accurately the increase in performance of the improved store.

Apart from the entirely practical approach or 'development' described above, there are several areas which are worth being investigated on a scientific basis. Most of these areas require research investigations which can only be implemented on a long term basis. These areas can be investigated in several ways on a theoretical, numerical and experimental basis. The ultimate aim of this research is to find the best cost/benefit Integrated Thermal Store which can be used satisfactorily in a domestic environment.

1) Numerical analysis would be particularly useful at this stage. The equations of momentum can be programmed using models of increasing complexity. At the moment, only a one dimensional analysis has been carried out. However relatively simple two dimensional models could be built for simple types of flows for example corresponding to axisymetric geometries (2-D).

Moreover, recent advances in computer technology have very significantly reduced the computation costs and consequently made the use these models technologically more achievable. Consequently the technical limitations which have for a long time seemed to be the limiting factors in the use of these computer models are disappearing relatively quickly. However, as mentionned earlier, some practical limitations in the use of these computer model still remain. Among these practical limitations are the relatively low accuracy of the computer

predictions and the large amount of engineering time required to build the models.

2) On the experimental side, a lot more work has to be carried out in the flow visualization field. Flow visualization is a more powerful tool for scientific investigation than the present approach which consists in measuring the temperatures of the water in the store and from these temnerature measurements try to estimate the flow of water in the store. Flow visualization would enable to have access to the velocity of the naturally occurring convection current in the store. This would greatly help the investigations in the design or orientation or arrangement of the heat exchanger coil. It would also help to identify dead zones in the store and zones where the mixing occurs which can be relatively useful for example in optimizing the aspect ratio of the store.

This can help in most of the areas which are very relevant to the to design the heat exchanger such as the amount of mixing taking place in the store and the effectiveness of baffles to channel the naturally occurring buoyancy driven flow during the thermal discharge.

In addition, flow visualization can be used in areas not directley related to the store itsel. This include the design of the fins on the outside surface of the heat exchanger.

3) The interference between the charging mechanism (gas boiler) and the store still has to be investigated. The charging process might significantly change the behaviour of the thermal store.

Charging the thermal store is usually achieved using a small boiler and a recirculating pump. During the thermal charge, water is taken at the bottom of the store, then heated up in the boiler and reinjected at the top of the store. The running of the pump induces some turbulence in the store. Mainly to facilitate the experimental side, the interaction between the pump, the boiler and the thermal store has not yet been investigated however, these might affect the performance of the thermal store.

Other charging mechanims which do not use pumps are possible an need investigations too. These include using the buoyancy force to drive the flow of water through the boiler or the use of a heat pipe to transfer heat from the flame of a burner to the thermal store.

4) On the theoretical side, several areas still have to be investigated. Among these are:

- developing heat transfer correlations for finned tube heat exchangers immersed in hot water to predict the natural convection heat transfer coefficient for several types of finned tube. The aim would be to find the best fin geometry for this type of application.

- investigating the design of the heat exchanger in terms of arrangement and orientation in the store. These have major impacts on the buoyancy driven flows occurring during the thermal discharge.

- the possibility of using internal fins to increase the tubeside heat transfer coefficient of the heat exchanger.

- macroscopic considerations such as the aspect ratio of the thermal store or optimisation of the volume of the store and the amount of insulation to use.

5) Finally, some speculative research can be carried out in other areas not directly linked to Integrated Thermal Stores but of more general interest to heat storage. These include for example investigations in the use of Phase Change Materials (PCM's). These are materails which change phase at a temperature which corresponds to the normal operating temperature of Integrated Thermal store. In this case, the relatively large amount of lattent heat given off during the change of phase might be recovered for useful purposes. Using the latent heat of the PCM allows for the same size of storage system to store much more heat than that would be possible with a traditional thermal store using sensible heat storage only. There is therefore some reason to think that the use of PCM could be beneficial.

Chapter 9

Conclusions

A fully detailed investigation of the transient behaviour of an Integrated Thermal Store has been carried out when either in the domestic hot water or space heating delivery modes of operation. This investigation is experimentally based and simple computer models are extensively used to simulate the behaviour of the thermal store. Based on this experimental work, several theoretically based heat transfer correlations have been developed. Consequently, main objective of the thesis which was to improve the fundamental knowledge of the transient behaviour Integrated Thermal Store has been achieved.

Conclusion 1

Heat transfer correlations have been developed to cover all the important aspects of the operation and design of Integrated Thermal Stores when in the domestic hot water mode of operation. For the heat exchanger coil used for domestic hot water production, these correlations are:

(i) A heat transfer correlation for the prediction of the natural convection heat transfer coefficient for the outside surface of a coiled horizontal axis finned tube heat exchanger immersed in hot water has been developed from experimental observations. This heat transfer correlation is:

$$Nu = 0.28Ra^{0.293} \qquad \text{for} \quad 100 < Ra < 1500$$

The accuracy of this correlation is acceptable in the range of Rayleigh number considered (the dispersion of the experimental measurements is less than 2.5%, with a maximum error between the experimental measurements and the predictions from the correlation of less than 5%). The natural convective heat transfer coefficient from the outside of the finned tube to the surrounding water is usually in the range 400 to 800 $Wm^{-2}K^{-1}$. This is approximately 10% above the heat transfer coefficient which would be obtained using a straight horizontal tube with vertical circular fins.

(ii) A correlation between the Effectiveness of heat recovery (Eff) and the Number of Transfer Units (NTU) of the heat exchanger coil has been developed. This correlation is:

$$Eff = 1-\exp(-0.768NTU) \qquad \text{for} \qquad 0.5 < NTU < 4$$

During the operation of the thermal store the Number of Transfer Units of the heat exchanger coil used for domestic hot water production is usually in the range 1.5 to 3. This is sufficient to give a relatively high Effectiveness of heat recovery from the heat exchanger.

(iii) The effect of the mass flow rate of water flowing through the heat exchanger coil on the overall heat transfer coefficient has been evaluated. The critical factor influencing this heat transfer coefficient is the change from transition to turbulent flow in the heat exchanger's pipe. This change occurs at flow rates of approximately 4 litre/min.

When the mass flow rate of water through the heat exchanger is less than this critical value of 4 litre/min, there is a significant decrease in the internal heat transfer coefficient. This reduces the overall heat transfer coefficient and consequently the Effectiveness of heat delivery from the thermal store.

When operating at this optimal flow rate of 4 litres/minute approximately 4% (1.2 MJ for a 150 litre draw-off) extra heat can be recovered from the store when compared with a thermal discharge achieved at a flow rate of 8 litre/min

Conclusion 2

An experimental and theoretical investigation of the use of baffles to improve the Effectiveness of domestic hot water delivery from Integrated Thermal Stores has been carried out.

Horizontal baffle located in the middle of the store and aimed at increasing the stratification during a domestic hot water draw-off can in theory improve the performance of hot water delivery. However, no significant improvement in Effectiveness of heat delivery was observed experimentally with this type of baffle.

Vertical baffles and a rectangular duct aimed at creating a recirculation motion in the store during a domestic hot water draw-off were found to be ineffective. However, due to the increased velocity of the water around the outer surface of the heat exchanger the external heat transfer coefficient can be increased by as much as 12 % when compared with the heat transfer coefficient for the situation without baffles present.

Conclusion 3

When in the space heating mode of operation, the relatively cool system return water entering the store creates a jet in the bottom region of the store. This jet induces a significant amount of mixing between the warm water in the store and the incoming cold water. This reduces the ability of the store to deliver high quality heat for space heating.

A correlation has been developed to quantify the effect of a jet on the Effectiveness of heat delivery. This correlations is based on the Richardson number (Ri) for the inlet flow. This is:

$$Eff = 1-\exp(-0.609Ri)$$

A Richardson number of around 3 is a minimum to avoid the mixing created by the jet and a value of above 5 is ideal. The Richardson number of the inlet flow can be significantly increased by reducing the velocity of the water entering the Integrated thermal store. Simple type of distributor or baffle arrangements can also be used to reduce the mixing created by the jet.

Conclusion 4

Several computer models based on implicit and explicit finite difference formulations of an energy balance for the water in the thermal store have been developed. They can simulate the transient behaviour of the thermal store either in the domestic hot water or in the space heating mode.

These computer models are based on original correlations which were developed and are in good agreement with experimental measurements. They are used to show how to improve the performance of the existing hot water store without significant changes in design.

The first of these change is to raise the upper heat exchanger coil in the store so that its upper heat surface area for heat exchangers reaches the very top of the water in the store. This would eliminate a dead zone located in the top part of the store and inporve the effectiveness of heat delivery from the Integrated Thermal Store in the domestic hot water mode by approximately 5% (1.5 MJ for a 150 litre thermal draw-off).

The second change is to is to raise the lower heat exchanger coil in the store by approximately 0.2 m so that the upper part of the coil is located exactely at mid heigth in the store. This improves stratification and hence increases the amount of heat delivered in the domestic hot water mode.

Other potential changes in the design of Integrated Thermal Stores are to improve the design of the mixing valve located at the outlet of the heat exchanger and to increase the diameter of the pipe returning from the space heating system. These changes in design can also be achieved at no extra manufacturing cost and would improve the performance of Integrated Thermal Stores. In the domestic hot water mode, the improvement in performance would mainly be achieved through a better service over a wider range of operating conditions. In the space heating mode of operation, the improvement in performance is difficult to evaluate but a maximum theoretical value was estimated at 8% (1.6 MJ) extra heat delivered to the radiators.

Appendix 1

Computer Programme for the Solution of the Natural Convection Heat Transfer Equation

A.1.1 Introduction

As detailed in chapter 3, the equation describing the natural convection heat transfer from a finned tube is:

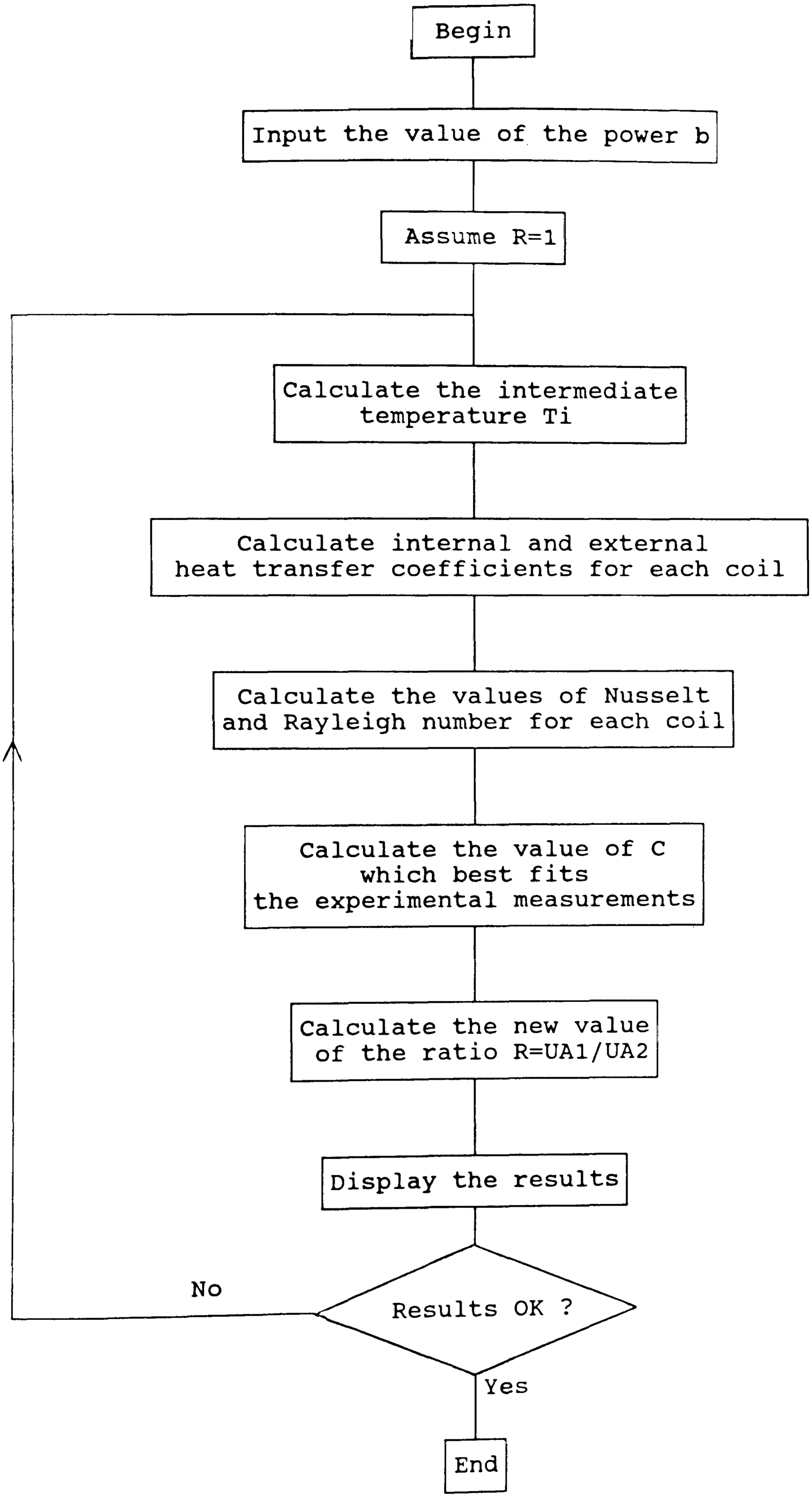
$$Nu = CRa^{b} \qquad (1)$$

where C and b are constants. C depends on the geometry of the heat exchanger and the mode of heat transfer. The power b depends on the mode of heat transfer but can reasonably be expected to be in the range 0.25 to 0.33.

The equations and the methodology used to evaluate the value of the coefficient C from the experimental data and from the value of the power b is described in detail in Chapter 3 of this thesis. A relatively simple computer programme was used to solve the set of equations numerically using a iterative method also described in Chapter 3. However, for reaons of clarity, the computer programme was not detailed. The aim of this Appendix in to describe this programme in detail by presenting a flowchart and a listing for this programme.

A.1.2 Flowchart and Computer Listing

A simplified flowchart for the programme is shown in Fig.A.1.2(a). A computer lising writen in basic is presented in Fig.A.1.2(b) to Fig.A.1.2(e)



the computer programme

```
10 REM ------------------------------------------------------------
20 REM PROGRAM FOR THE CALCULATION OF THE COEFFICIENT C IN THE NATURAL
30 REM HEAT TRANSFER EQUATION FROM THE VALUE OF THE POWER b AND THE
40 REM EXPERIMENTAL DATA
50 REM ------------------------------------------------------------
60 REM NOTATION
70 REM --------
80  REM BE      COEFFICIENT OF THERMAL EXPANSION OF WATER
90  REM COE     POWER b IN THE NATURAL CONVECTION EQUATION
100 REM CF      COEFFICIENT C IN THE NATURAL CONVECTION EQUATION
110 REM DT      TEMPERATURE DIFFERENCE AT THE WALL OF THE HEAT EXCHANGER
120 REM EFF     EFFECTIVENESS OF HEAT RECOVERY OF THE HEAT EXCHANGER
130 REM H       EXTERNAL HEAT TRANSFER COEFFICIENT OF THE HEAT EXCHANGER
140 REM HI      INTERNAL HEAT TRANSFER COEFFICIENT OF THE HEAT EXCHANGER
150 REM K       THERMAL CONDUCTIVITY OF WATER
160 REM M       MASS FLOW RATE IN THE HEAT EXCHANGER COIL
170 REM NTU     NUMBER OF TRANSFER UNITS OF THE HEAT EXCHANGER
180 REM NU      NUSSELT NUMBER
190 REM PR      PRANDTL NUMBER
200 REM Q       HEAT EXTRACTED BY THE HEAT EXCHANGER COIL IN THE STORE
210 REM R       RATIO OF UA1/UA2
220 REM RA      RAYLEIGH NUMBER
230 REM RE      REYNOLDS NUMBER
240 REM TC      BULK TEMPERATURE OF THE WATER IN THE COIL
250 REM TDO     TEMPERATURE OF THE WATER IN THE BOTTOM OF THE STORE
260 REM TIN     TEMPERATURE OF THE WATER ENTERING THE HEAT EXCAHNGER
270 REM TOU     TEMPERATURE OF THE WATER LEAVING THE HEAT EXCHANGER
280 REM TUP     TEMPERATURE OF THE WATER IN THE TOP OF THE STORE
290 REM UA      UA VALUE OF THE HEAT EXCHANGER
300 REM V       VELOCITY OF THE WATER IN THE COIL
310 REM VIS     VISCOSITY OF WATER
320 REM -----------------
330 REM SUBSCRIPTS
340 REM -----------------
350 REM 1       REFERS TO THE LOWER PART OF THE STORE
360 REM 2       REFERS TO THE UPPER PART OF THE STORE
370 REM ------------------------
380 REM OTHER PARAMETERS USED IN THIS PROGRAMME ARE ONLY ARBITRARY VARIABLES
390 REM AND ARE PHYSICALLY MEANINGLESS
400 REM EXAMPLE OF SUCH VARIABLES
410 REM A       LIMIT FOR SEARCH IN DICHOTOMY METHOD
420 REM B       LIMIT FOR SEARCH IN DICHOTOMY METHOD
430 REM FX      NAME OF A FUNCTION USED IN A SUBROUTINE
440 REM FY      NAME OF A FUNCTION USED IN A SUBROUTINE
450 REM ------------------------
460 REM ------------------------
470 REM DATA FOR THE ITERATIONS
480 REM ------------------------
490 M=9.726001/60
500 MT=M*4180
510 V=4*M/(3.1416*.0109*.0109*1000)
520 TIN=10.26001
530 TOU=32.16001
540 TDO=24.43
550 TUP=39.58
560 SI=.34
570 SE=1.74
580 REM ASSUMED VALUE FOR THE POWER b IN THE NATURAL CONVECTION EQUATION
590 COE=.25
600 REM ASSUMED VALUE FOR THE RATIO UA1/UA2
610 R=1
620 REM ASSUMED VALUE OF THE HEAT TRANSFER COEFFICIENT FOR THE 1RST ITERATION
630 H1=800
640 H2=800
```

Fig.A.1.2(b): Listing of the Computer Programme for the Solution of the Natural Convection Equation

```
650 REM
660 REM ------------------------------------
670 REM BEGINNING OF THE ITERATIVE PROCESS
680 REM ------------------------------------
690 REM
700 REM ------------------------------------------------------------
710 REM CALCULATION OF THE TEMPERATURE OF THE WATER Ti IN THE HEAT
720 REM EXCHANGER COIL USING A DICHOTOMY METHOD
730 REM THE LIMITS FOR THE SEARCH ARE UA=100 AND UA=1500W/M2/K
740 REM ------------------------------------------------------------
750 A=100
760 B=1500
770 UA1=A
780 GOSUB 2170
790 IF FX<0 THEN 810
800 C=B : B=A : A=C
810 UA1=B
820 GOSUB 2170
830 IF FX<0 THEN 2420
840 UA1=A
850 UA1=(A+B)/2
860 GOSUB 2170
870 IF SGN(FX)=-1 THEN A=UA1
880 IF SGN(FX)=1 THEN B=UA1
890 REM THE DICHOTOMY METHOD IS STOPPED IF THE OUTLET TEMPERATURE AT THE
900 REM HEAT EXCHANGER COIL IS LESS THAN 1/1000TH OF ITS THEORETICAL VALUE
910 IF ABS(FX)<.001 THEN 940
920 GOTO 840
930 REM HEAT EXTRACTED FROM THE BOTTOM(Q1) AND TOP COIL(Q2) IN THE STORE
940 Q1=MT*(TI-TIN)
950 Q2=MT*(T2-TI)
960 REM TEMPERATURE DIFFERENCE AT THE WALL OF THE BOTTOM AND TOP COILS
970 DT1=Q1/(H1*SE)
980 DT2=Q2/(H2*SE)
990 REM
1000 REM
1010 REM ------------------------------------------------------
1020 REM CALCULATION OF THE INTERNAL HEAT TRANSFER COEFFICIENT
1030 REM FOR THE BOTTOM HEAT EXCHANGER COIL
1040 REM ------------------------------------------------------
1050 REM BULK TEMPERATURE OF THE WATER IN THE BOTTOM COIL
1060 TC1=(TIN+TI)/2
1070 T=TC1
1080 GOSUB 2350
1090 REM FORMULA BY SLEICHER AND ROUSE FOR THE BOTTOM COIL
1100 A=.88-(.24/(4+PR))
1110 B=.333+.5*EXP(-.6*PR)
1120 RE=1000*V*.0109/VIS
1130 NU=5+(.015*RE^A)*(PR^B)
1140 HI1=K*NU/.0109
1150 UI1=1.173*SI*HI1
1160 REM
1170 REM ------------------------------------------------------
1180 REM CALCULATION OF THE EXTERNAL HEAT TRANSFER COEFFICIENT
1190 REM OF THE BOTTOM HEAT EXCHANGER COIL
1200 REM ------------------------------------------------------
1210 T=(TDO+TDO-DT1)/2
1220 GOSUB 2350
1230 RA1=9.810001*BE*DT1*((.0019)^4)*1000*1000*4180
1240 RA1=RA1/(VIS*K*.0189)
1250 KRA1=K*(RA1)^COE
1260 NU1=H1*.0019/K
1270 REM
1280 REM
```

Fig.A.1.2(c): Listing of the Computer Programme for the Solution of the Natural Convection Equation

```
1290 REM ------------------------------------------------
1300 REM CALCULATION OF THE INTERNAL HEAT TRANSFER COEFFICIENT
1310 REM OF THE TOP HEAT EXCHANGER COIL
1320 REM ------------------------------------------------
1330 REM
1340 REM BULK TEMPERATURE OF THE WATER IN THE TOP HEAT EXCHANGER COIL
1350 TC2=(TI+T2)/2
1360 T=TC2
1370 GOSUB 2350
1380 REM FORMULA BY SLEICHER AND ROUSE FOR THE TOP COIL
1390 A=.88-(.24/(4+PR))
1400 B=.333+.5*EXP(-.6*PR)
1410 RE=1000*V*.0109/VIS
1420 NU=5+(.015*RE^A)*(PR^B)
1430 HI2=K*NU/.0109
1440 UI2=1.173*SI*HI2
1450 REM
1460 REM ------------------------------------------------
1470 REM CALCULATION OF THE EXTERNAL HEAT TRANSFER COEFFICIENT
1480 REM OF THE TOP HEAT EXCHANGER COIL
1490 REM ------------------------------------------------
1500 REM
1510 T=(TUP+TUP-DT2)/2
1520 GOSUB 2350
1530 RA2=9.810001*BE*DT2*((.0019)^4)*1000*1000*4180
1540 RA2=RA2/(VIS*K*.0189)
1550 KRA2=K*(RA2)^COE
1560 NU2=H2*.0019/K
1570 REM
1580 REM ------------------------------------------------
1590 REM FIND THE VALUE OF THE COEFFICIENT C WHICH BEST FITS
1600 REM THE ESTIMATES FROM THE PREVIOUS ITERATION
1610 REM THE LIMITS FOR THE SEARCH ARE C=0.1 AND C=1.5
1620 REM ------------------------------------------------
1630 REM
1640 A=50
1650 B=500
1660 D=A
1670 GOSUB 2270
1680 IF FY<0 THEN 1700
1690 C=B : B=A : A=C
1700 D=B
1710 GOSUB 2270
1720 IF FY<0 THEN 2420
1730 D=A
1740 D=(A+B)/2
1750 GOSUB 2270
1760 IF SGN(FY)=-1 THEN A=D
1770 IF SGN(FY)=1 THEN B=D
1780 REM THE DICHOTOMY METHOD IS STOPPED IF THE UA VALUE OF THE
1790 REM HEAT EXCHANGER IS LESS THAN 1/1000TH OF ITS THEORETICAL VALUE
1800 IF ABS(FY)<.001 THEN 1820
1810 GOTO 1730
1820 REM THE VALUE OF D IS PROPORTIONAL TO THE COEFFICIENT C (OR CF)
1830 REM IN THE NATURAL CONVECTION HEAT TRANSFER EQUATION
1840 CF=D*.0019/SE
1850 H1=D*KRA1/SE
1860 H2=D*KRA2/SE
1870 UA1=1/(1/UI1+1/(KRA1*D))
1880 UA2=1/(1/UI2+1/(KRA2*D))
1890 REM
1900 REM
1910 REM
1920 REM
```

Fig.A.1.2(d): Listing of the Computer Programme for the Solution of the Natural Convection Equation

```
1930 REM ------------------------------------------
1940 REM PRINT THE RESULTS AT THE END OF THE ITERATION
1950 REM ------------------------------------------
1960 REM PRINT THE ESTIMATES OF THE RAYLEIGH AND NUSSELT NUMBERS
1970 REM AT THE END OF THE FIRST ITERATION
1980 PRINT "RA1";RA1,"RA2";RA2,"NU1";NU1,"NU2";NU2
1990 REM PRINT "KRA1";KRA1,"KRA2";KRA2
2000 PRINT "UI1";UI1,"UI2";UI2
2010 REM PRINT "D";D
2020 REM CF IS THE COEFFICIENT C IN THE NATURAL CONVECTION EQUATION
2030 PRINT "CF";CF
2040 PRINT "H1";H1,"H2";H2,
2050 PRINT "UA1";UA1,"UA2";UA2
2060 REM REEVALUATE THE RATIO UA1/UA2 FOR THE NEXT ITERATION
2070 R=UA1/UA2
2080 REM ALSO CALCULATE THE NTU AND THE EFFECTIVENESS OF THE HEAT EXCHANGER
2090 NTU=(UA1+UA2)/MT
2100 EFF=(TOU-TIN)/(TUP-TIN)
2110 PRINT "NTU";NTU,"EFF";EFF
2120 PRINT ""
2130 GOTO 670
2140 STOP
2150 REM
2160 REM ------------------------------------------------
2170 REM SUBROUTINE FOR THE EVALUATION OF THE TEMPERATURES
2180 REM OF THE WATER FLOWING IN THE HEAT EXCHANGER COIL
2190 REM ------------------------------------------------
2200 UA2=UA1/R
2210 TI=TDO+(TIN-TDO)*EXP(-UA1/MT)
2220 T2=TUP+(TI-TUP)*EXP(-UA2/MT)
2230 FX=TOU-T2
2240 RETURN
2250 REM
2260 REM ------------------------------------------------------------
2270 REM SUBROUTINE FOR THE CALCULATION OF THE UA VALUE OF THE HEAT EXCHANGER
2280 REM FROM THE INTERNAL AND EXTERNAL HEAT TRANSFER COEFFICIENT
2290 REM ------------------------------------------------------------
2300 FY=1/(1/UI1+1/(KRA1*D))+1/(1/UI2+1/(KRA2*D))
2310 FY=UA1+UA2-FY
2320 RETURN
2330 REM
2340 REM ------------------------------------------------------
2350 REM SUBROUTINE FOR CALCULATION OF THE THERMAL PROPERTIES OF WATER
2360 REM ------------------------------------------------------
2370 PR=11.207-.232244*T+.0015256*T*T
2380 K=(577.6+1.25*T)/1000
2390 VIS=(1551-30*T+.1907*T*T)/1000000!
2400 BE=(-20.38+12*T-.0476*T*T)/1000000!
2410 RETURN
2420 END
```

Fig.A.1.2(e): Listing of the Computer Programme for the Solution of the Natural Convection Equation

The programme begins with a initialization routine. This routine is used to set the values of experimental measurements which include the temperatures of the water in the store and of the mass flow rate in the heat exchanger (lines 460 to 570) and the assumption of the value of the power b in the natural convection equation (line 590). In addition, some assumptions with respect to the UA value of the heat exchanger coil are made to initiate the first loop in the iterative process (lines 600 to 640).

The iterative process begins with the calculation of the temperature of the water in the heat exchanger between the upper and lower zones (TI or Ti). This is calculated from the respectives UA values at the top and bottom heat exchanger coil using a dichotomy iterative process (lines 700 to 920)

The second step in the iterative process is to calculate the internal and external heat transfer coefficient for the top and bottom heat exchanger coil in the store (lines 1010 to 1560). From these values, the coefficeint C of the natural convection equation can be estimated by mathichg the experimentally measured temperatures in the store to the theoretically predicted temperatures obtained using the UA value of the heat exchanger coil which is estimated from the internal and external heat transfer coefficient previously calculated. This is again achieved using a dichotomy iterative method (lines 1580 to 1880).

Finally, other parameters such as the internal and external heat transfer coefficients, the NTU of the heat exchanger and the effectiveness of heat recovery are calculated. The loops ends by displaying the data on the computer screen for visual interpretation (lines 1930 to 2130). This visual interpretation is easilly possible as the computation time is the order of a few seconds for each loop.

Apart from the main programme, three subroutines are used:

<u>Subroutine 1</u>: this subroutine (lines 2160 to 2240) evaluates the temperatures of the water flowing in the heat exchanger coil from the UA value of the heat exchanger, the inlet temperature and the temperature of the water in the tank in the upper and lower zones. This subroutine is used in the dichotomy method for the evaluation of the intermediate temperature Ti.

<u>Subroutine 2</u>: this subroutine (lines 2260 to 2320) calculates the UA value of the heat exchanger from the coefficient C. It is used in the dichotomy method for the evaluation of this coefficient C.

Subroutine 3: this subroutine (lines 2340 to 2410) is used for the evaluation of the thermal properties of liquid water from simple correlations. These thermal properties are used for the calculations of the Rayleigh, Nusselt and Prandtl numbers required during the computation.

A.1.3 Example of Experimental Results

The use of the programme made it possible to evaluate a large number of parameters relatively quickly from experimental data. An typical output for a few experimental data is presented in the following section. The notation is as follows:

- Time : time from beginning of the thermal discharge, (min)

- m : average mass flow rate of water into the heat exchanger coil between two scans, (litre/s)

- Tin : temperature of the water entering heat exchanger, (^{o}C)

- Tout : temperature of the water leaving the heat exchanger, (^{o}C)

- Tdo : temperature of the water in the lower region of the store, (^{o}C)

- Tup : temperature of the water in the upper region of the store, (^{o}C)

- Nu1 : value of the Nusselt number at the lower coil, (dimensionless)

- Ra1 : value of the Rayleigh number at the lower coil, (dimensionless)

- Nu2 : value of the Nusselt number at the upper coil, (dimensionless)

- Ra2 : value of the Rayleigh number at the upper coil, (dimensionless)

- C : value of the coefficient C in the natural convection heat transfer equation, (dimensionless)

- UI1 : product of the internal heat transfer coefficient by the internal surface area for the lower coil, (WK^{-1})

- UE1 : product of the external heat transfer coefficient by the external surface area for the lower coil, (WK^{-1})

- UA1 : overall heat transfer coefficient of the lower heat exchanger coil, (WK^{-1})

- UI2 : product of the internal heat transfer coefficient by the internal surface area for the upper coil, (WK^{-1})

- UE2 : product of the external heat transfer coefficient by the external surface area for the upper coil, (WK^{-1})

- UA2 : overall heat transfer coefficient of the upper heat exchanger coil, (WK^{-1})

- NTU : Number of Transfer Units of the heat exchanger coil, (dimensionless)

- Eff : effectiveness of heat recovery, (dimensionless)

- Exp : experiment number

	Exp 1	Exp 2	Exp 3	Exp 4	Exp 5	Exp 6
Time	5	6	7	8	9	10
m	0.1607	0.1600	0.1600	0.1593	0.1607	0.1614
Tin	9.54	9.63	9.70	9.79	9.87	9.94
Tout	66.27	63.40	60.30	57.58	54.87	52.22
Tdo	52.69	48.40	44.63	41.26	39.25	37.63
Tup	73.15	70.73	68.00	65.35	62.73	60.31
Nu1	1.99	1.87	1.76	1.70	1.64	1.57
Ra1	850	684	557	455	398	360
Nu2	2.28	2.20	2.17	2.14	2.11	2.05
Ra2	1200	1187	1137	1056	971	903
C	0.2759	0.2763	0.2765	0.2837	0.2848	0.2794
UI1	3418	3350	3293	3249	3221	3198
UE1	1152	1074	1006	968	931	884
UA1	861	814	771	746	722	693
UI2	4647	4496	4350	4227	4125	4032
UE2	1332	1323	1301	1299	1266	1209
UA2	1035	1022	1001	994	969	930
NTU	2.84	2.75	2.65	2.57	2.53	2.43
Eff	0.8918	0.8800	0.8679	0.8602	0.8513	0.8394

Appendix 2

Thermal Properties of Liquid Water

The following table gives the thermal properties of liquid water most commonly used in heat transfer calculations. The values correspond to saturation pressure however, they can be used as relativley good approximation for liquid water at atmospheric pressure. The following units used in Table A.2 are:

Temperature (T) : ^{o}C
Density (ρ) : kgm^{-3}
Coefficient of thermal expansion (β) : K^{-1}
Specific heat (Cp) : $Jkg^{-1}K^{-1}$
Viscosity (μ) : $kgm^{-1}s^{-1}$
Thermal conductivity (k) : $Wm^{-1}k^{-1}$
Prandtl number (Pr) : dimensionless

T	ρ	β ($x10^{3}$)	Cp	μ ($x10^{6}$)	k ($x10^{3}$)	Pr
0.01	999.84	-68.05	4210	1752	569	12.96
5	999.96	16.00	4204	1501	578	10.92
10	999.70	87.97	4193	1300	587	9.29
15	999.10	150.87	4186	1136	595	7.99
20	998.20	206.78	4183	1002	603	6.95
25	997.04	257.21	4181	890	611	6.09
30	995.65	303.24	4179	797	618	5.39
35	994.03	345.73	4178	718	625	4.80
40	992.22	385.30	4179	651	632	4.30
45	990.21	422.45	4181	594	638	3.89
50	988.03	457.59	4182	544	643	3.54
55	985.69	491.04	4183	501	648	3.23
60	983.20	523.07	4185	463	653	2.97
65	980.55	553.90	4188	430	658	2.74
70	977.77	583.74	4191	400	662	2.53
75	974.90	612.75	4194	374	666	2.36
80	971.79	641.08	4198	351	670	2.20
85	968.62	668.86	4203	330	673	2.06
90	695.32	696.23	4208	311	676	1.94
95	961.90	723.28	4213	294	678	1.83
100	958.36	750.14	4219	279	681	1.73

Table A.2 : Thermal Properties of Liquid Water at Saturation Pressure

In some engineering problems, the temperatures are required for computation. It is then more useful to approximate the heat transfer properties of water using simple equations. The following correlations have been developed from the data presented in Table I and using a least square interpolation method:

$$\rho = 1007.56 - 0.4359T$$

$$\beta = -13.24 + 11.15T - 4.11\times10^{-2}T^{2}$$

$$Cp = 4190$$

$$\mu = 1522.1 - 27.92T + 0.16434T^{2}$$

$$k = 582.72 + 1.11T$$

$$Pr = 10.966 - 0.215T + 1.307\times10^{-3}T^{2}$$

Where all the thermal properties are expressed in the same units as in Table A.2.

These equations can be used in the temperature interval between $10^{\circ}C$ and $90^{\circ}C$. No garantee can be given with respect to their accuracy.