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R. R. COHEN

Thermal Energy Accumulation in
Stratified Hot Water Stores

Supervisor: P. W. O'Callaghan

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SUMMARY

Hot water thermal energy stores have the potential to improve and extend the performance of many kinds of energy system. Temperature stratification in the store is likely to affect the system's efficiency. A basic but accurate computer model of the hot water store under various inlet flow conditions is a requisite means of assessing promising applications of hot water storage by system computer simulation techniques.

A microprocessor-controlled test facility has been constructed to evaluate the performance of a 3m^3 hot water store under a wide range of inlet flow conditions, using a temperature step input approach. Three types of inlet/outlet ports have been examined: horizontal, vertical and distributors. The results show that two distinct regions evolve within the store: a fully-mixed region adjacent to the inlet port and a region of smooth 'plug-flow' in the remaining volume of the store. The performance of the store is shown to be defined by the initial depth of the fully-mixed region which in turn is seen to be closely related to the buoyancy and momentum fluxes of the inlet flow.

The behaviour of the store and the evident correlations have enabled a one-dimensional computer model of the store to be developed, taking into account the turbulent mixing, vertical heat conduction and heat losses to the surrounding areas. The model has been successfully validated against the results from the step input experiments.

The model has been integrated into a computer simulated central heating system which incorporates a hot water store. Predictions have been made, using the simulation, of the energy savings which may be achieved with the use of storage in comparison to a conventional system, and an assessment has been made of the economic viability of the application.

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Finally I should admit that this work might never have been completed, but for the heroic forbearance of my long-suffering family, for whom the concept of "the thesis" will forever be evocative of the experience of single parenthood.

Robert Cohen

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NOTATION

<u>Symbol</u>	<u>Meaning</u>	<u>Units</u>
A	Cross-sectional area	m^2
B	Buoyancy flux	$m^4 s^{-3}$
C	Constant	—
C_p	Specific heat of water	$J kg^{-1} K^{-1}$
D	Store diameter	m
d	Inlet port diameter	m
E	Energy stored	J
g	Gravitational acceleration constant	$m s^{-2}$
Gr	Grashof number	—
H	Height of store	m
H_D	Height of deadspace	m
k	Thermal conductivity	$W m^{-1} K^{-1}$
L	Distance between inlet and outlet ports	m
L^*	Distance from inlet port to start of stratified region	m
[L]	Length dimension	m
M	Mass of store	kg
M_O	Momentum flux	$m^4 s^{-2}$
$\dot{m}$	Mass flow rate	$kg s^{-1}$
N	Number of zones in store	—
N_m	Number of zones in fully-mixed region	—
P	Lavan and Thompson parameter	—
$\dot{Q}$	Rate of energy flow	W
Q_{LOSS}	Heat loss energy	J
Re	Reynolds number	—
Ri	Richardson number (length = r)	—
Ri^*	Richardson number (length = L)	—
r	Inlet port radius	m
T	Temperature	$^{\circ}C$
TACT	Actual temperature (i.e. measured)	$^{\circ}C$
[T]	Time dimension	s
t	Time	s

t^*	Lavan and Thompson time of store 'depletion'	s
u	Velocity of water at inlet to store	$m\ s^{-1}$
U	Overall heat transfer coefficient	$W\ m^{-2}\ K^{-1}$
'UA'	Store heat loss rate per degree	$W\ K^{-1}$
$(UA)_{s,a}$	Store heat loss rate per degree	$W\ K^{-1}$
V	Volume of store	m^3
$\dot{V}$	Volume flow rate	$m^3\ s^{-1}$
v	Plug flow velocity in store	$m\ s^{-1}$
x	Index for momentum flux	-
y	Index for buoyancy flux	-
z	Depth of penetration of flow into store	m

ABBREVIATIONS

ABS	Absolute value
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BHRA	British Hydrodynamic Research Association
CEGB	Central Electricity Generating Board
ETSU	Energy Technology Support Unit (Dept. of Energy)
GDP	Gross Domestic Product
INT	Integer value
PCM	Phase Change Material
PRT	Platinum Resistance Thermometer
TES	Thermal Energy Storage
TSC	Thermal Storage Capacity

SUBSCRIPTS

a	Ambient
amb	Ambient
c	Charge
comp	Computer
D	Deadspace
e	Effective

ex	Exit
in	Inlet
initial	Initial
m	Fully-mixed region
max	Maximum
mout	Out of the fully-mixed region
R	Reaction
S	Store
s	Salt
t	Tank or Time
w	Water
z	Zone of store
0	At time equals zero

GREEK SYMBOLS

α	Thermal diffusivity	$\text{m}^2 \text{s}^{-1}$
β	Volume expansion coefficient	K^{-1}
ΔT	Temperature difference between inlet flow and water in tank	K or °C
Δt	Step in time	s
Δx	Step in distance	m
δV	Small volume	m^3
η	Performance coefficient for store	-
η^*	Lavan and Thompson performance coefficient	-
ν	Kinematic viscosity	$\text{m}^2 \text{s}^{-1}$
ρ	Density	kg m^{-3}
τ	Charge or discharge period for store	s
τ_c	Time constant for store heat loss	s
τ_m	Fill time of fully-mixed region	s

CHAPTER ONE

INTRODUCTION

1.1. WORLD ENERGY PERSPECTIVE

1.1.1 Background

Over the last 15 years, the world energy situation has become a crucial issue. The publication in 1972 of "The Limits to Growth" [Ref.1.1], marked an unprecedented though long overdue awakening in the developed world to the global consequences of the exponential growth of population. It was at last recognised that finite natural resources were being progressively squandered for rabid consumerism. The world seemed to be travelling inexorably towards doomsday.

A different crisis broke in 1973 however, when the Middle East conflict and the formation of OPEC (the Organisation of Petroleum Exporting Countries) caused oil prices to quadruple, with the consequent sharp decline in world economic growth. The industrialised nations were forced to introduce measures to conserve oil, for economic and political reasons.

A further steep rise in oil prices occurred in 1979-80, triggered by the revolution in Iran and the start of the Gulf War. The World's energy demand had been growing at nearly 6% per annum before 1973, but has followed less than a 2% growth rate in the last decade.

Today, we are experiencing the unexpected trend of an oil glut, created mainly by non-OPEC producers, particularly the U.K. The current fall in demand for oil has also contributed to the gradual reduction in prices since 1982 (see Figure 1.1a). OPEC is in decline, embarrassed by the combination of a diminishing world market share (see Figure 1.1b) and the falling prices.

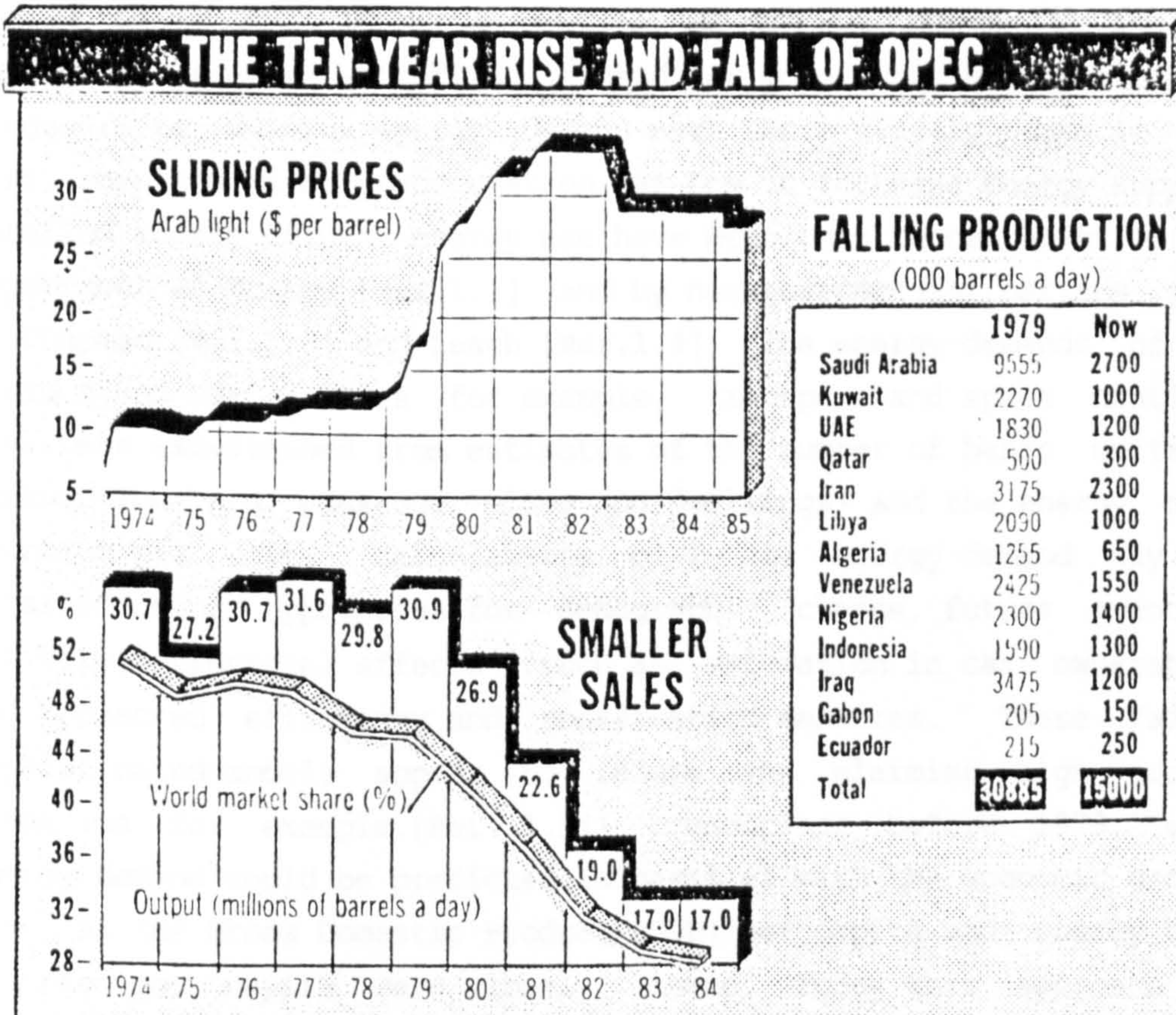


FIGURE 1.1 a. (above) THE PRICE OF OIL SINCE 1974

b) (below) OPEC SALES AND MARKET SHARE SINCE 1974

(FROM THE SUNDAY TIMES 7 JULY 1985).

1.1.2 Future Trends in Energy Demand

(a) The United Kingdom

The oil crises have established a need for government and the energy producers to examine energy demand much more closely than in the past, when the prime preoccupation was how to increase energy supply. Detailed models of U.K. energy use have been constructed both by the Department of Energy [Ref.1.2] and by non-governmental parties such as Chapman [Ref.1.3] and Leach [Ref.1.4]. The energy demands of the various end use sectors (for example, transport and space heating) have been ascertained from estimates of the number of basic units of consumption (e.g. passenger miles and dwellings) and the energy consumption per unit. Calculations of future energy demand may be obtained using hypotheses for demographic change, future economic activity, structural effects (such as saturation in car ownership) and projected efficiency and conservation measures. These fairly sophisticated models appear to refute the alarmist high growth scenarios (for example, [Ref 1.5]) commonplace before 1973, when energy demand could be predicted in parallel with the economic growth rate, as the Gross Domestic Product (GDP) per capita and energy consumption per capita were closely linked. Today this approach is unreliable, with a definite trend towards a less energy-intensive economy (see Figure 1.2).

Figure 1.3 illustrates some current projections of U.K. primary energy demand up to 2025 by the Government and, for comparison, by proponents of 'low energy futures'. A modest steady increase in GDP is assumed in all cases, but differing assumptions have been made on the potential for energy conservation. Other factors will also effect energy demand, including the primary fuel mix, the decline of heavy industry and any changes in the real price of energy.*

* FOOTNOTE

Despite the current uncertainty (see Figure 1.4), the long term outlook suggests costs will increase as primary hydrocarbon fuels become more expensive to extract, necessitating a shift towards more highly processed secondary fuels such as substitute natural gas, synthetic liquid fuels and electricity, for use in applications met previously by oil and natural gas.

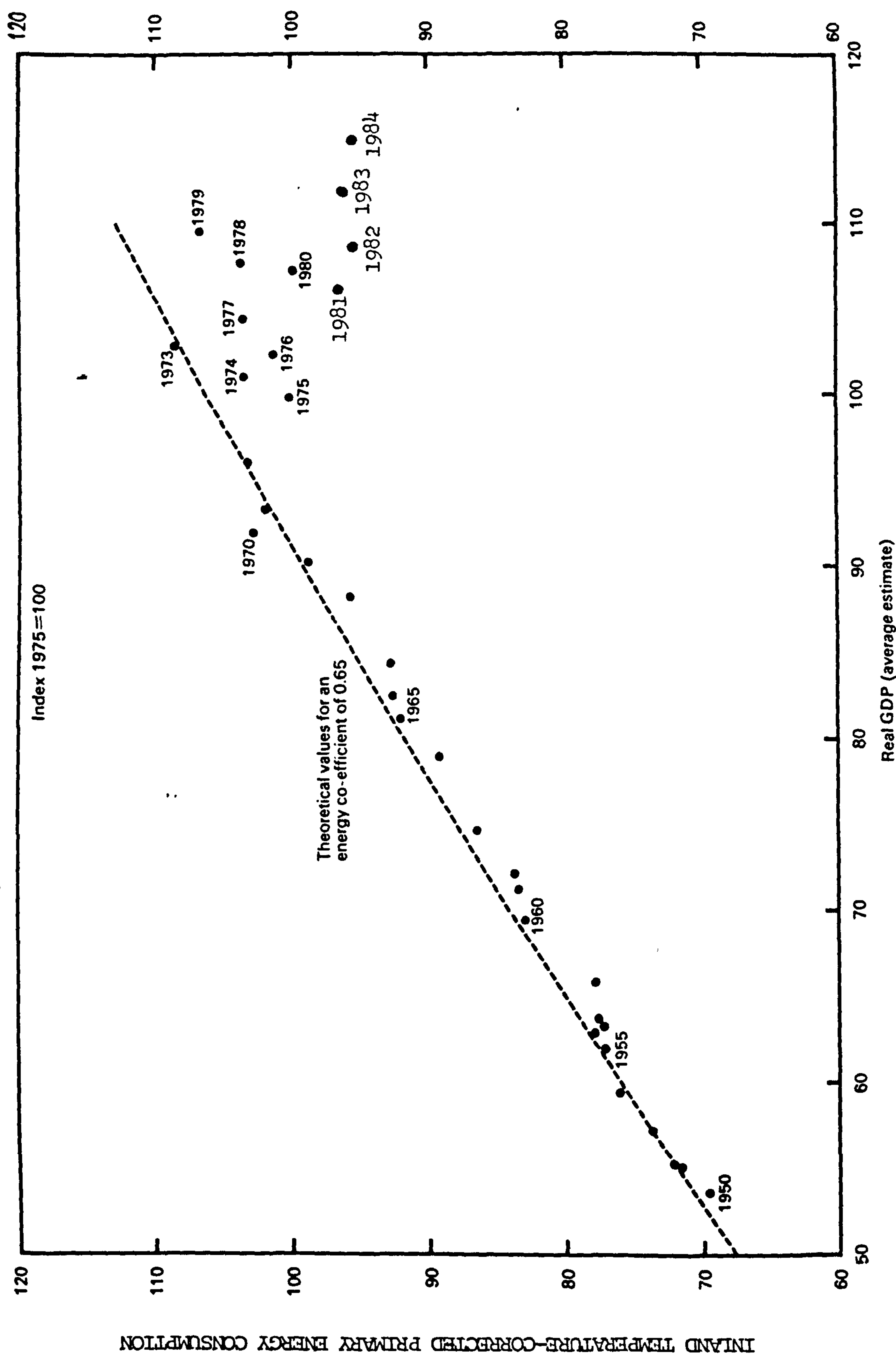


FIGURE 1.2 PRIMARY ENERGY CONSUMPTION AND REAL GDP OF THE U.K. SINCE 1950 (FROM REF. 1.6).

ETSU = ENERGY TECHNOLOGY SUPPORT
UNIT, DEPT. OF ENERGY
IIED = INTERNATIONAL INSTITUTE FOR
ENVIRONMENT AND DEVELOPMENT
FoE = FRIENDS OF THE EARTH

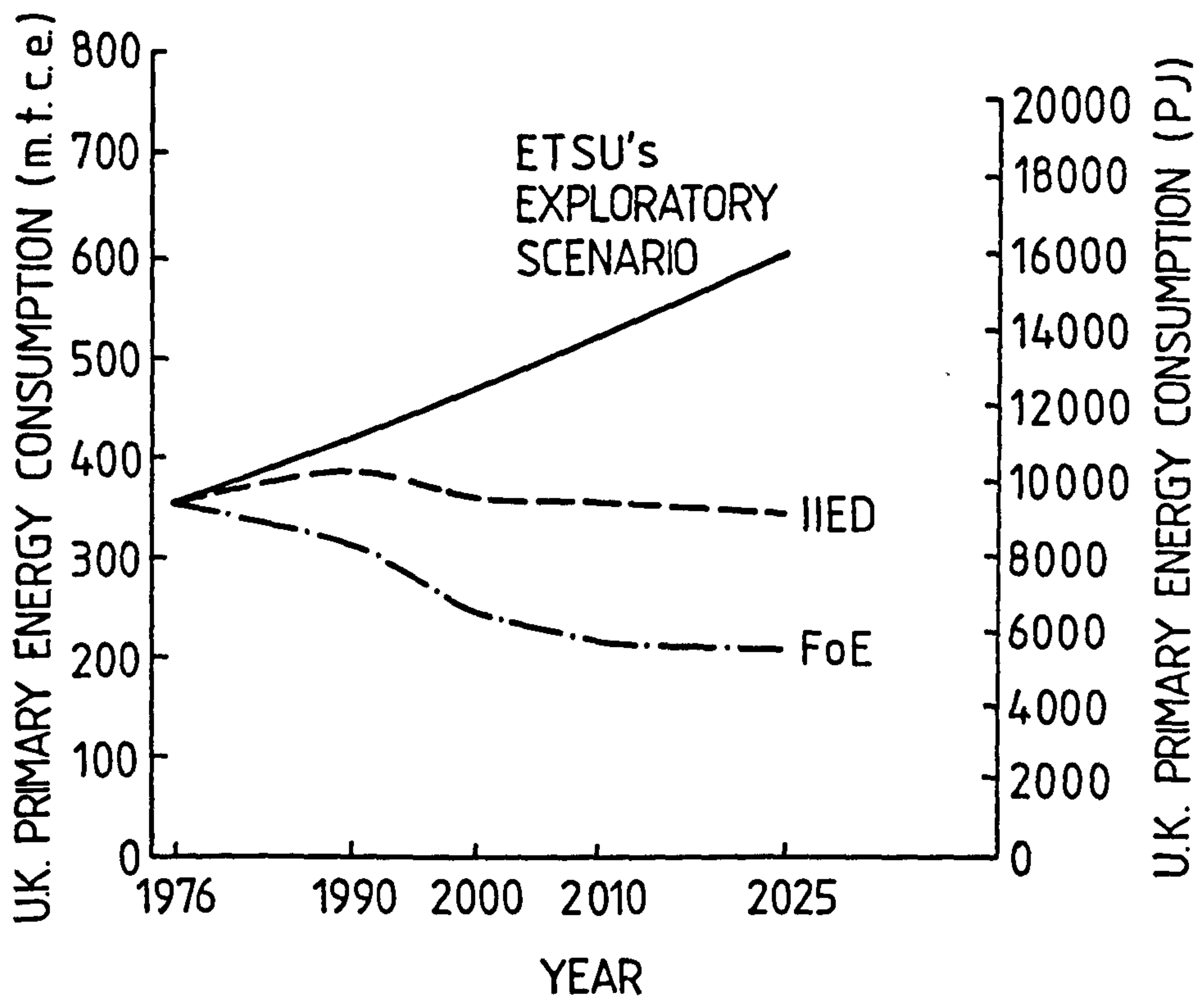


FIGURE 1.3 PREDICTED U.K. PRIMARY ENERGY CONSUMPTION
UP TO 2025 (REF.1.7).

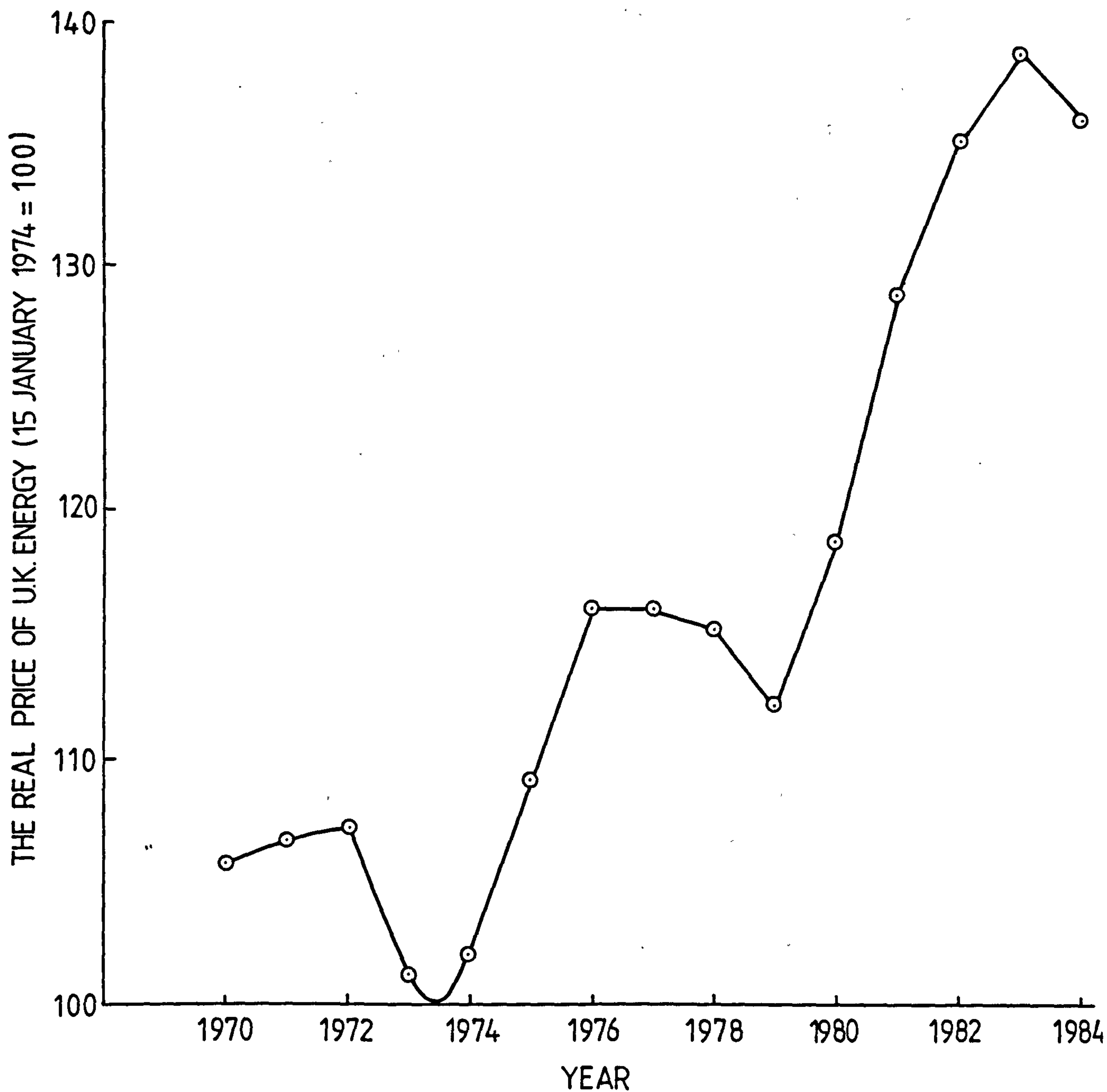


FIGURE 1.4 THE PRICE (IN REAL TERMS) OF ENERGY IN THE U.K. FROM 1970 TO 1984 (DERIVED FROM REF. 1.6).

(b) Other Industrialised Countries

Many Western industrialised societies have relatively stable populations so that it might be conjectured that their energy demands could be pegged to current levels if rising living standards were accompanied by the implementation of effective conservation measures. However, many of those nations, now plagued by unemployment, have adopted a policy of maximum economic growth to stimulate employment. Consequently, energy demand might conceivably continue to grow at around its modest post-1974 rate of 1.8% per annum.

(c) Developing Countries

The Developing Countries, with 75% of the World's population, account for only 16% of the World's commercial energy. It is clear that the governments of those nations who aspire politically towards a Western style economic growth ethic will instigate policies which are likely to cause a continued rise in the World energy demand. Despite the paralysing effect on their economies of war and natural disasters - factors which cannot reliably be predicted - commercial growth, and the intensification of agriculture* and indigenous industries represent the only available option, if these countries are to support adequately their populations, whose growth may not stabilise for many generations (see Figure 1.5). It seems inevitable that as their GDP per capita rises so will their energy demand per capita, following a general trend (see Figure 1.6).

Conservation and education in the lessons learned by the West are crucial to halt and reverse the ecological bankruptcy faced by these nations who are struggling to achieve growth at the cost of their precious but fast vanishing natural resources.

* FOOTNOTE

The fuel requirements of tractors, fertilisers, pesticides, etc, can reduce the ratio of food energy reaped to human or fuel energy expended from up to 30:1 in 'primitive' agricultural systems to around breakeven in a modern intensive farm [Ref.1.8].

Another typical example of the dilemma is the depletion of local sources of firewood leading to a growing demand for imported alternative fuels, such as kerosine.

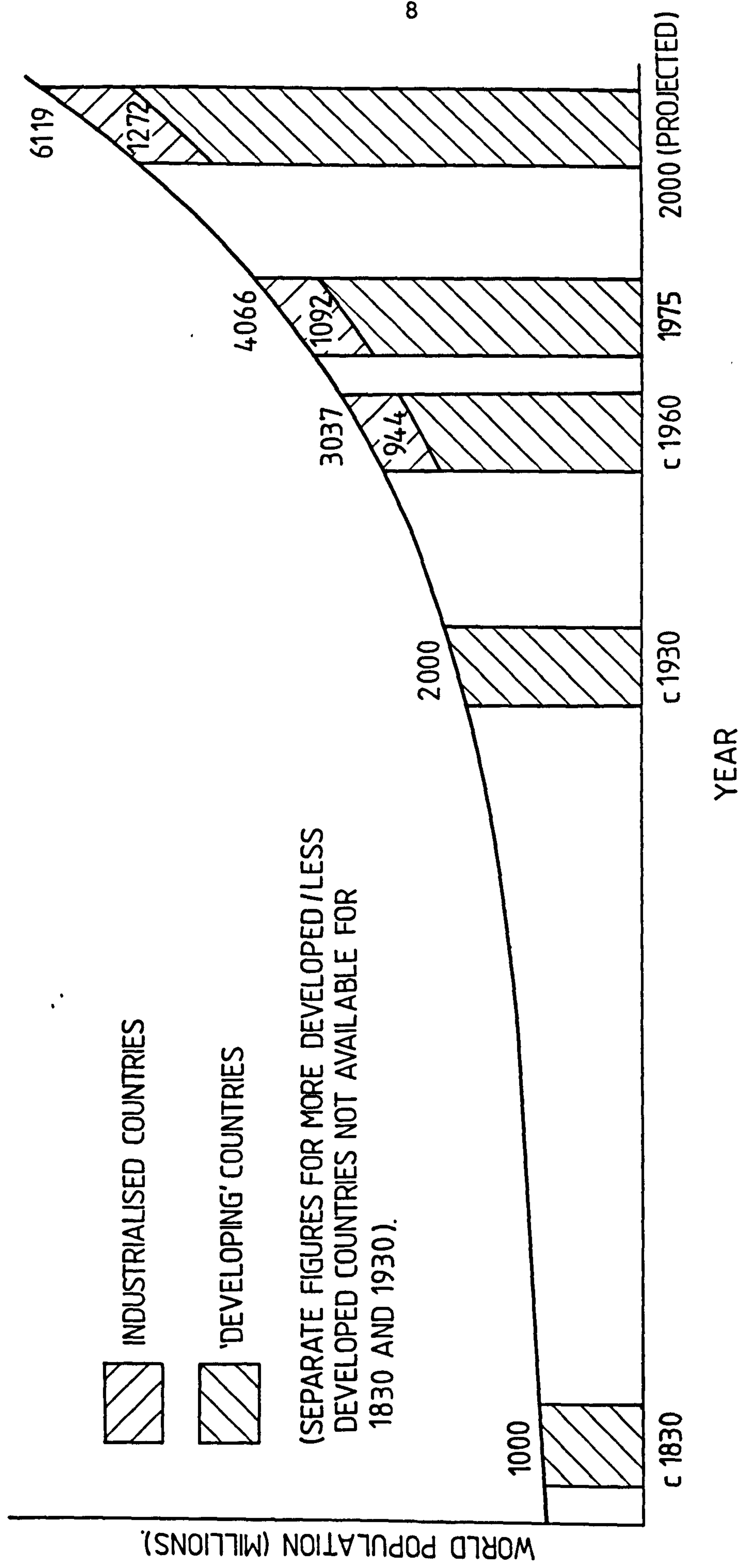


FIGURE 1.5 PAST AND PREDICTED WORLD POPULATION GROWTH (SOURCE: U.N. POPULATION DIVISION, 1980).

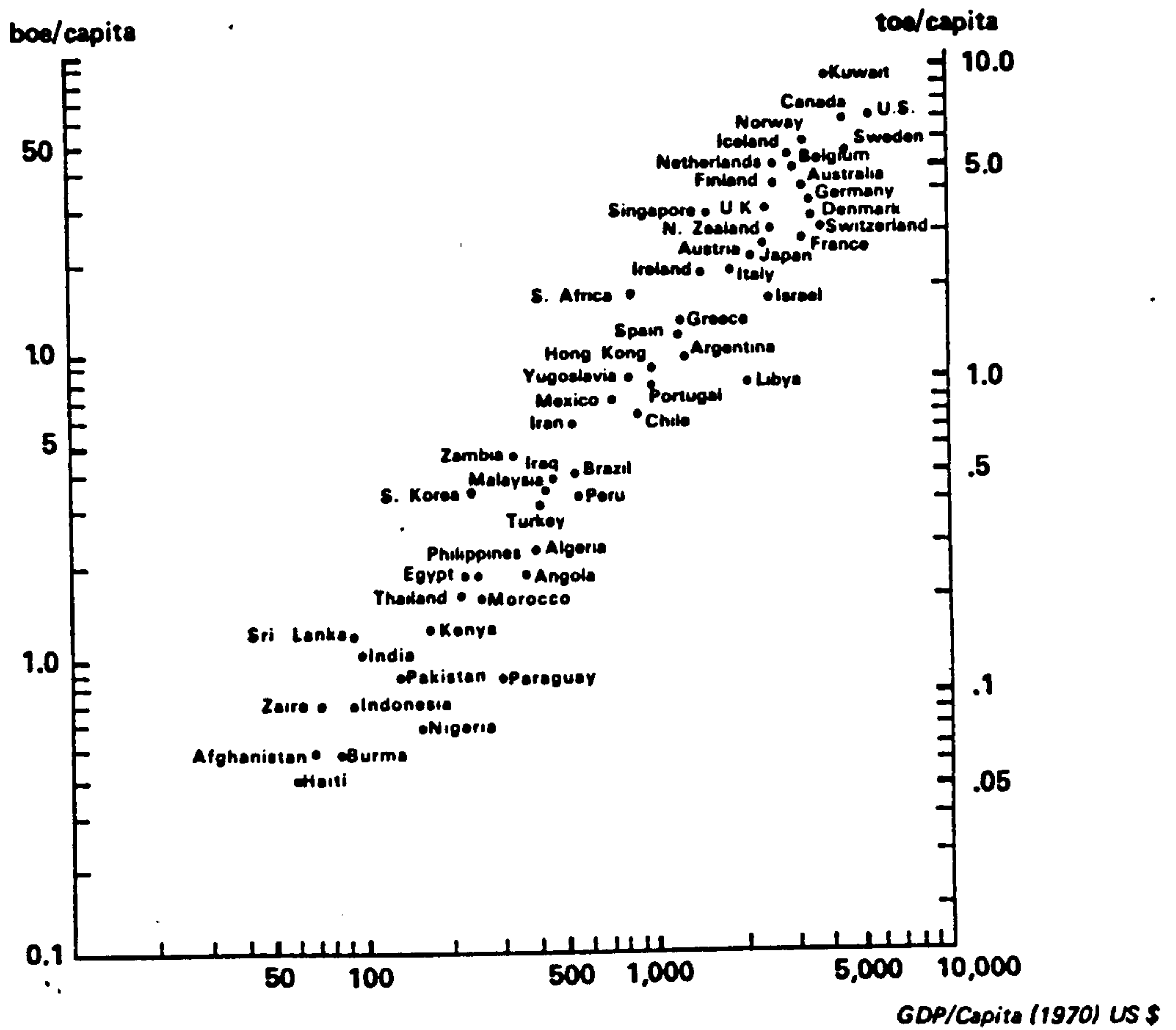


FIGURE 1.6 PRIMARY ENERGY/CAPITA VERSUS GDP/CAPITA IN 1973 FOR VARIOUS COUNTRIES (FROM REF. 1.9).

(d) The World

Forecasts for World energy demand ought to be a summation of the forecast demands for individual countries. However, given the fallibility in predicting the energy demand in the U.K., which has a well-monitored economy, political stability, large indigenous energy reserves and a stable population, it is not surprising that World demand projections tend to be merely extrapolations of past trends combined with a prediction of an overall growth rate. The U.K. Department of Energy forecast shown in Figure 1.7 assumes energy demand will grow at 3.2% per annum until 2000 and 1.3% thereafter, an average of 2% until 2050.

The "energy gap" implied by this figure cannot, of course, ever exist, but it shows that, unless a new source of energy is harnessed or conventional energy is supplied in greater quantities than is at present thought possible, energy demand will be greater than supply and consumption.

1.1.3 Physical Limits to Global Energy Consumption(a) The Heat Limit

The most fundamental limit to World energy consumption is postulated to be the point where the rate of heat release disrupts the climatological equilibrium which regulates surface temperatures on Earth. It is widely believed that major perturbations of this equilibrium led to the Ice Ages. It is conjectured that surface temperatures fell as a result of vast quantities of dust in the atmosphere generated by volcanic activity or impact from meteors. Indeed it has recently been postulated that nuclear war would be followed by a prolonged winter for the same reason. It could be equally catastrophic to raise the surface temperature as this might melt the polar ice caps, thereby raising the global sea level by up to 300 feet (100m). Chapman [Ref.1.3] proposed that for these reasons an upper limit to the rate of global energy consumption should be set at 1% of the level of solar radiation on Earth, i.e. $1.75 \cdot 10^{12}$ kW.

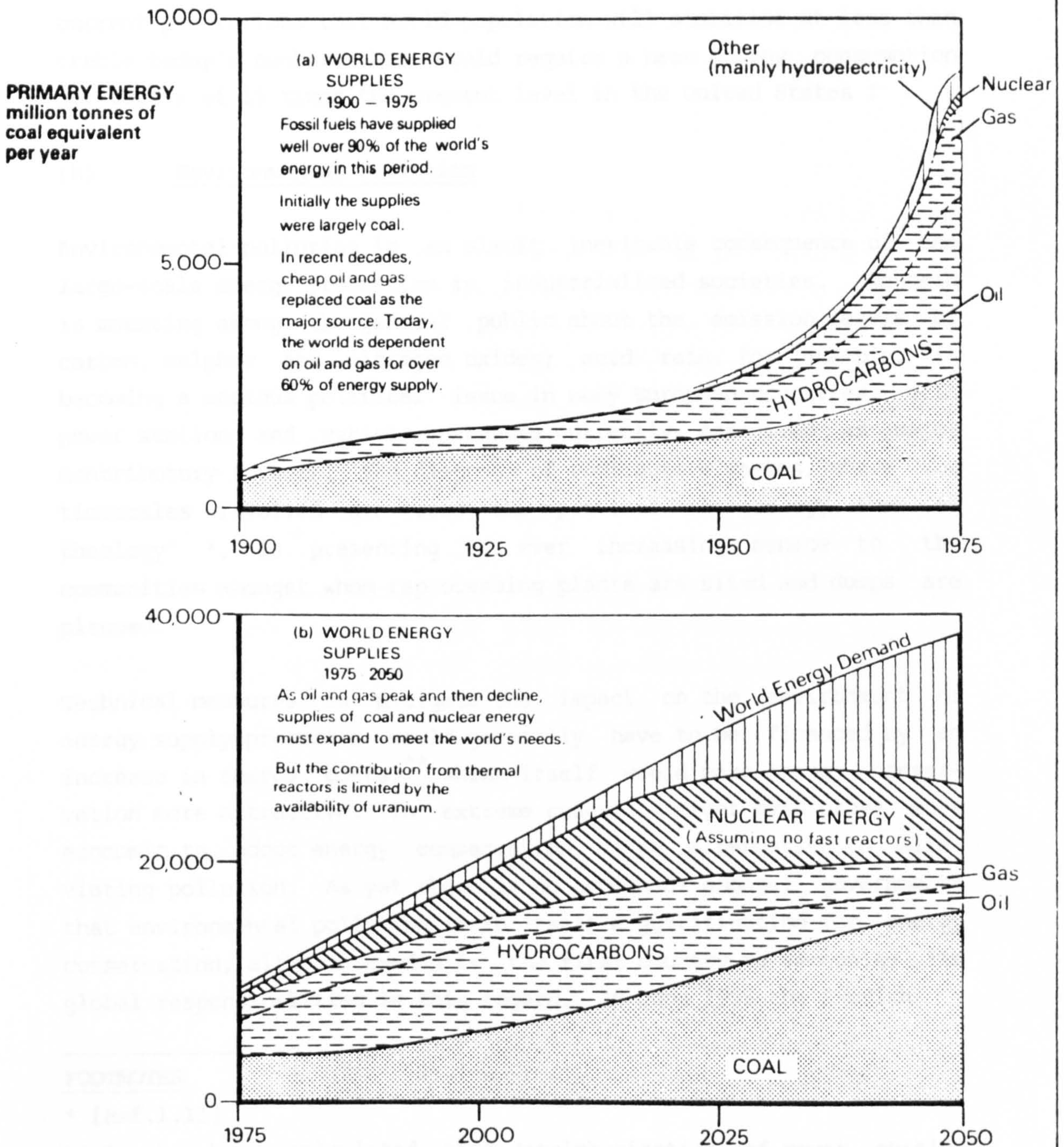


FIGURE 1.7 WORLD ENERGY SUPPLIES - PAST AND FUTURE (FROM REF. 1.2).

The mean rate of World energy consumption in 1982 was $9.1 \cdot 10^9$ kW [Ref.1.10], 190 times less than this limit. If however, World energy consumption were to grow at its pre-1973 rate of 6% per annum, then demand would reach the critical level in just 90 years. But, with current predictions that World population will stabilise at less than treble today's number, this would require a mean energy consumption per capita of 15 times the present level in the United States !

(b) Environmental Pollution

Environmental pollution is an almost inevitable consequence of the large-scale energy production in industrialised societies. Concern is mounting among the general public about the emission levels of carbon, sulphur and nitrogen oxides; acid rain, for example, is becoming a serious political issue in many European countries, with power station and vehicle exhausts currently perceived as major contributory factors. The disposal of radioactive wastes, where "the timescales involved are less the province of geology than of theology" *, is presenting an ever increasing menace to the communities amongst whom reprocessing plants are sited and dumps are planned.

Technical measures to mitigate the impact on the environment of energy supply processes would generally have to be financed by an increase in energy costs,** which itself would make energy conservation more attractive. In extreme cases, it might even prove more economic to adopt energy conservation as an alternative to alleviating pollution. As yet there is no consensus amongst governments that environmental pollution is in itself a justification for energy conservation, although Sweden is the first country to recognise its global responsibilities in this regard. In June 1985 the Swedish

FOOTNOTES

* [Ref.1.11]

** It has been calculated that desulphurisation of power station emissions might add about 15% to electricity costs, whilst decarbonisation (more hypothetically) might cause them to treble. [Ref.1.12].

Government approved a Bill on Guidelines for Energy Policy [Ref.1.13] which emphasised that "all countries have a duty to ensure that resources are conserved and utilized in the best possible way. An active policy of energy conservation and oil substitution will enable us to pre-empt a smaller proportion of the World's total oil resources and to reduce our share of environmental impact".

(c) Finite Resources

Coal, oil, gas and uranium are all finite resources. It is therefore possible to combine estimates of present reserves with predictions of future demand and obtain an estimate of the lifetime of the reserves. This analysis has been regularly performed in the last 20 years, but the predictions have mostly been excessively alarmist both because they fail to allow for the fact that total reserves keep increasing as new discoveries are made, and also because demand has not grown nearly as fast as anticipated. Another common misconception is that demand will grow exponentially for ever.

A brief analysis here is based on a comprehensive assessment of World energy resources made by the World Energy Conference in 1980 [Ref.1.14]. It estimated the proved reserves, recoverable under current technical and economic conditions, and the potential resources, recoverable allowing for improvements in technology of extraction and price rises, as follows :

Source	Proved Reserves 10^{21} J	Potential Resources 10^{21} J
Coal	20.3	296.0
Hydrocarbon	11.1	50.8
Uranium	1.9*	156.0 ⁺
Total	33.3	502.8

* if used in thermal reactors

+ if used in breeder reactors.

TABLE 1.1 Estimates of World Energy Resources

For the analysis, three scenarios are considered : (i) that World energy demand stays at its current level of $0.3 \cdot 10^{21}$ J p.a., (ii) that it grows at 2% p.a. as conjectured by the U.K. Department of Energy in Figure 1.7, and (iii) that it grows at a higher rate of 3% p.a. In the case of growth, demand is taken to stabilise when it reaches 10 times its current level (i.e. $3 \cdot 10^{21}$ J p.a.), which occurs after 116 years at 2% p.a. growth and after 78 years at 3% p.a. growth. If the proved and potential resources in Table 1.1 are taken as the lowest and highest estimates of reserves, and the different fuels are amalgamated to form one resource, then the following lifetimes are calculated :

Growth Rate % p.a.	Shortest Lifetime Years	Longest Lifetime Years
0	111	1676
2	59	239
3	50	215

TABLE 1.2 Estimated Lifetimes of Finite Energy Resources

These figures indicate that shortfalls should not occur before the middle of the next century, rather than the end of this century as was being prophesied during the 1970's, and may indeed be deferred until well into the 22nd Century. However, it is vital to remember that the hopefully realistic energy consumption growth rates conjectured here are conditional upon a concerted and continuing energy conservation policy, so as to sever the link between growth of energy demand and economic growth, as the latter might well increase more quickly.

1.1.4 Conclusions

The following conclusions can be abstracted from this brief foray into some of the central issues of the World energy situation :

1. Market forces have, at least temporarily, halted the high exponential growth in World energy demand that precipitated the alarmist prophecies of pre-1973. Although drastic shortages are unlikely in the medium-term future, it is essential to prevent a new acceleration in energy consumption due to the growth of GDP.
2. Energy production and consumption is an increasingly significant cause of environmental pollution.
3. Beyond the short term, energy is likely to become more expensive as coal becomes harder to extract and oil and natural gas are replaced by secondary fuels.
4. The uneven global distribution of world energy consumption should be recognised to be as unethical as the pattern of glut and famine which currently exists in World food supplies.
5. Energy conservation must be an essential component of any strategy aiming for a stable energy future. It is the only option which tackles headlong all the problems summarised above.

1.2 ENERGY STORAGE

The storage of energy is an essential feature of energy utilisation. As energy is transformed from one form to another, storage enables a reserve of energy to be established for subsequent use at a release rate that is not dependent upon the initial rate of transformation. This two-fold decoupling in time of the energy transformation processes is the means by which temporal and spatial gaps in the demand for energy are bridged. Efficient energy storage can make very significant contributions to energy conservation by preserving the quality of excess energy for use at a later time or different place when otherwise it would be rejected to the ambient heat sink. At the same time it acts as an economically attractive method of load management, reducing the size of capital equipment required, increasing its load, or utilisation, factor and thereby improving the efficiency of energy usage.

1.2.1 Techniques for Energy Storage

Energy is the capacity for doing work. There are a number of fundamental forms of energy, which are inter-convertible by suitable means. Techniques exist for storing energy in all of its forms, as can be seen from the following examples :

<u>ENERGY FORM</u>	<u>TECHNIQUES FOR STORING ENERGY</u>
Nuclear	Natural or processed fissile material, e.g. Radium, Uranium, Plutonium.
Potential	Natural lakes and rivers, pumped storage.
Kinetic	Flywheels, compressed air.
Electrical	Batteries
Chemical	(1) Hydrocarbons, e.g. fossil-fuels, biomass, food. (2) Other chemicals which can produce exothermic reactions.

Thermal	Sensible, latent or reversible thermo-chemical heat.
---------	--

An energy store will be charged and discharged during its lifetime a number of times, which may be once in the case, for example, of a coal mine, or several thousand in such stores as a car petrol tank. The charge and discharge may be separated by millions of years in the case of fossil fuel deposits, sequential, in for example petrol tanks, or even simultaneous as happens sometimes with domestic hot water tanks. The means for creating or replenishing a store can be natural, e.g. photosynthesis or rain, mechanical work, e.g. pumping water, compressing air, injecting heat, e.g. via heat exchangers in hot water storage vessels or via solar radiation onto a Trombe wall, or simply refilling a tank, bunker or cavern with oil, coal or gas.

The diversity of the forms and techniques of energy storage which are in common use is illustrated by Figure 1.8, in which the density of energy storage achievable by a particular technique is plotted against the volume of the energy stored. The diagonal lines represent the total capacity of the store.

The ten prime examples of energy stores shown in Figure 1.8 are listed in Table 1.3, along with information which indicates the technique's significance on a national scale. Following the methodology developed by Russell [Ref.1.15], the total installed energy storage capacity in the U.K. of the ten examples is shown on the diagonal lines of Figure 1.9, in which the capacity of one installation is plotted as a function of the number of such installations. The centralised and decentralised storage of fossil-fuels can be seen to dominate the overall picture, which is only to be expected as they form 95% of primary fuel supplies. The common thermal energy storage techniques of hot water tanks and night storage heaters have a similar and considerable total installed capacity, which is nevertheless much lower than that of the fossil-fuel stores. Interestingly, the single pumped-hydro storage facility of the Central Electricity Generating Board at Dinorwic has the same capacity as one third of all the country's night storage heaters.

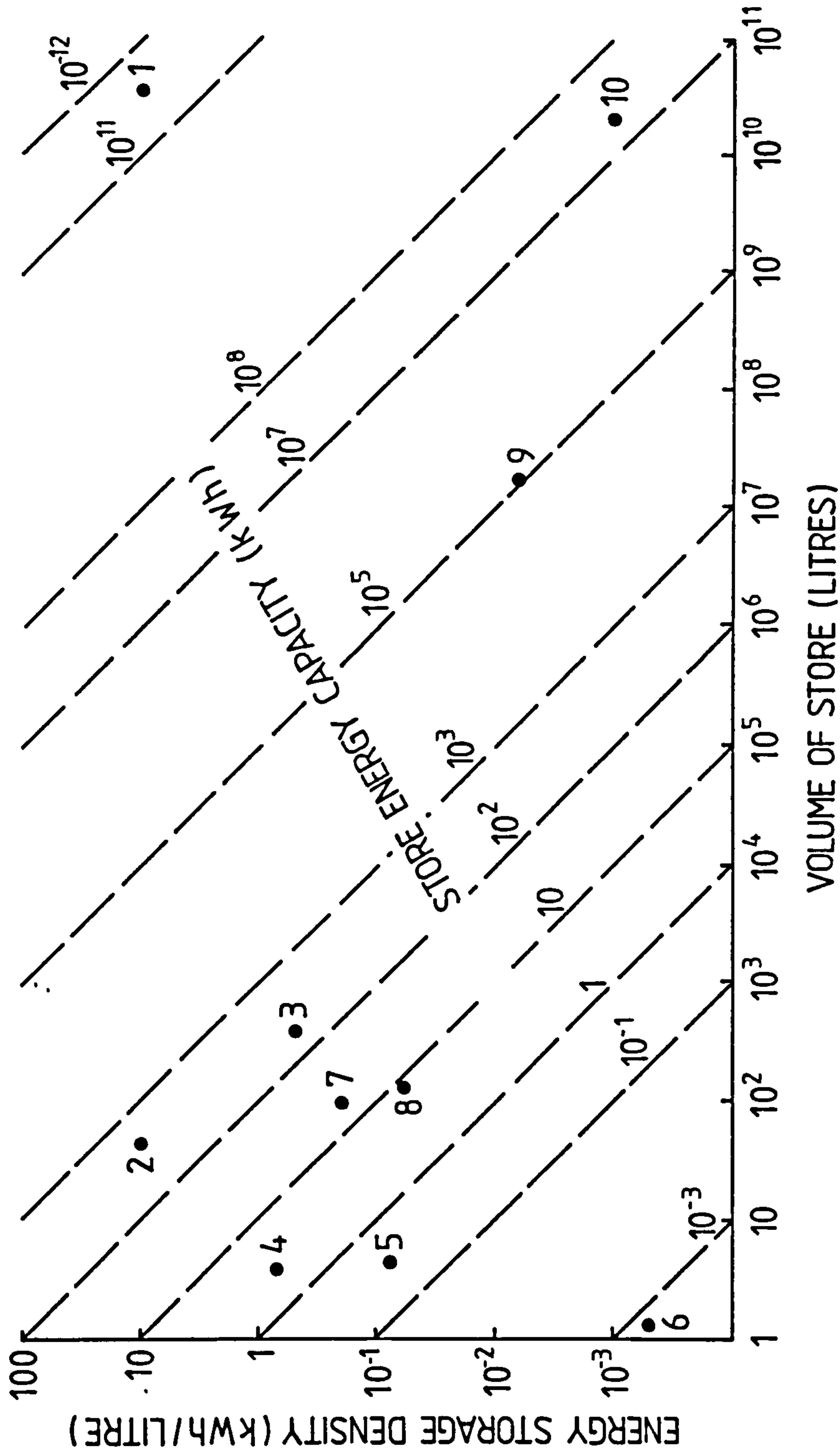


FIGURE 1·8 THE STORAGE DENSITY AND TYPICAL VOLUME OF TEN PRIME STORAGE TECHNIQUES.

No. on Graphs	Energy Form	Storage Technique	Size in Usual Units	Volume		Energy Storage Capacity		Time to Charge Store	Time to Discharge Store	Standstill Time (i.e. Between Charge and Discharge)	No. Units in U.K.	No. Cycles/ Year	Total Stored Energy
				litres	kWh	kWh	kWh/litre						kWh/year
1.	Chemical	Coal mine	50 million tons of coal	35.10 ⁹	390.10 ⁹	11		Millions of years	100 years	Millions of years	200	10 ⁻²	780.10 ⁹
2.	Chemical	Car petrol tank	10 gallons	45	450	10		Minutes	Days	0	16.10 ⁶	26	190.10 ⁹
3.	Chemical	Bale of straw	1 bale	400	200	0.5		4 months	Hours	Months	160.10 ⁶	1	32.10 ⁹
4.	Chemical	Food for 1 per- son for 24 hrs	2500 calories	4.1	2.9	0.7		Months	1 day	Days- Months	56.10 ⁶	365	59.10 ⁹
5.	Electrical	Car battery	30 amp hrs (12 volt)	4.5	0.36	8.10 ⁻²		1-10 hrs	3-6 hrs	0	16.10 ⁶	40	230.10 ⁶
6.	Kinetic	Car flywheel	10 kg	1.3	0.7.10 ⁻³	5.10 ⁻⁴		Milli- seconds	Milli- seconds	Milli- seconds	16.10 ⁶	26.10 ³	290.10 ⁶
7.	Thermal	Night storage heater	3.4 kW	108	24	0.22		7 hours	14 hours	0	2.5.10 ⁶	200	12.10 ⁹
8.	Thermal	Domestic hot water cylinder	130 litres	130	8	6.10 ⁻²		1 hour	Minutes- hours	0-12 hrs	20.10 ⁶	350	56.10 ⁹
9.	Thermal	Aquifer	17000 m ³	17.10 ⁶	100.10 ³	6.10 ⁻³		6 months	6 months	0	100	1	10.10 ⁶
10.	Potential	Dinorwic pumped storage	1728 MW	20.10 ⁹	20.10 ⁶	1.10 ⁻³		Hours	Hours	Hours	1	365	7.10 ⁹

TABLE 1.3 The magnitude and cycling rate of ten prime examples of Energy Stores

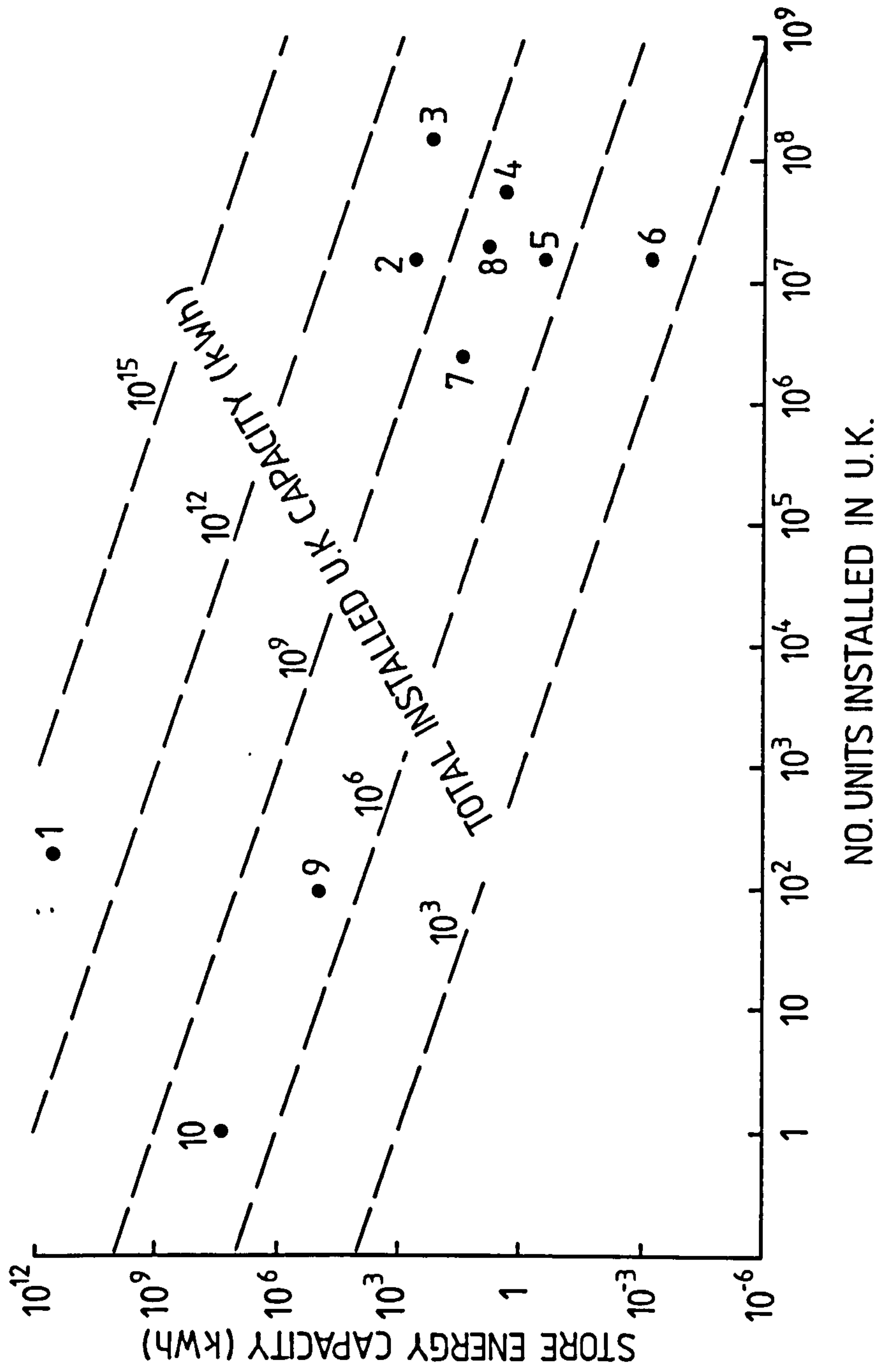


FIGURE. 1.9 THE ENERGY CAPACITY AND NUMBER OF U.K. UNITS OF TEN PRIME STORAGE TECHNIQUES.

The overall significance of each particular storage technique is determined by the number of times the total installed capacity is charged and discharged over, say, a year. This produces the total energy stored during a year of operation, and is shown in Figure 1.10, on diagonal lines, for the ten chosen examples. Coal mines and car petrol tanks can be seen to be leading the field comfortably, followed by a closely packed group of unlikely companions : human food, domestic hot water cylinders and bales of straw.

1.3 THERMAL ENERGY STORAGE

Thermal energy storage (TES) was shown in the last section to play a very significant role in meeting energy demand in the U.K. It is of course fundamentally different from the chemical energy storage in fossil-fuel or biomass, the other dominant technique, in the way that energy is accumulated into the store. TES can be created by applying heat from any available source to any medium. Once formed (charged), a heat store is similar to a chemical energy store in that it can be maintained for use at a later time and that it releases (discharges) its energy in the form of heat. TES does however degrade with time and usually must be insulated to delay this process. TES is also very expensive to transport owing to its relatively low energy density (about two orders of magnitude lower than that of hydrocarbons). For these reasons, TES is always employed close to the end use of energy and is therefore highly decentralised.

This section serves to categorise and describe all the common techniques for TES and then highlights the main areas of the energy scene where the need for TES is, or might become, most pressing.

1.3.1 Categorisation and discussion of TES techniques

The primary categorisation (see Figure 1.11) of TES indicates the reaction of the storage medium when it is heated. Thus, a steady rise in the material's temperature denotes sensible heat storage, a change in its phase denotes latent heat storage and the separating of some of its chemical constituents specifies reversible thermochemical storage. A secondary categorisation defines the phase or phases of the storage medium; for example, sensible heat can be

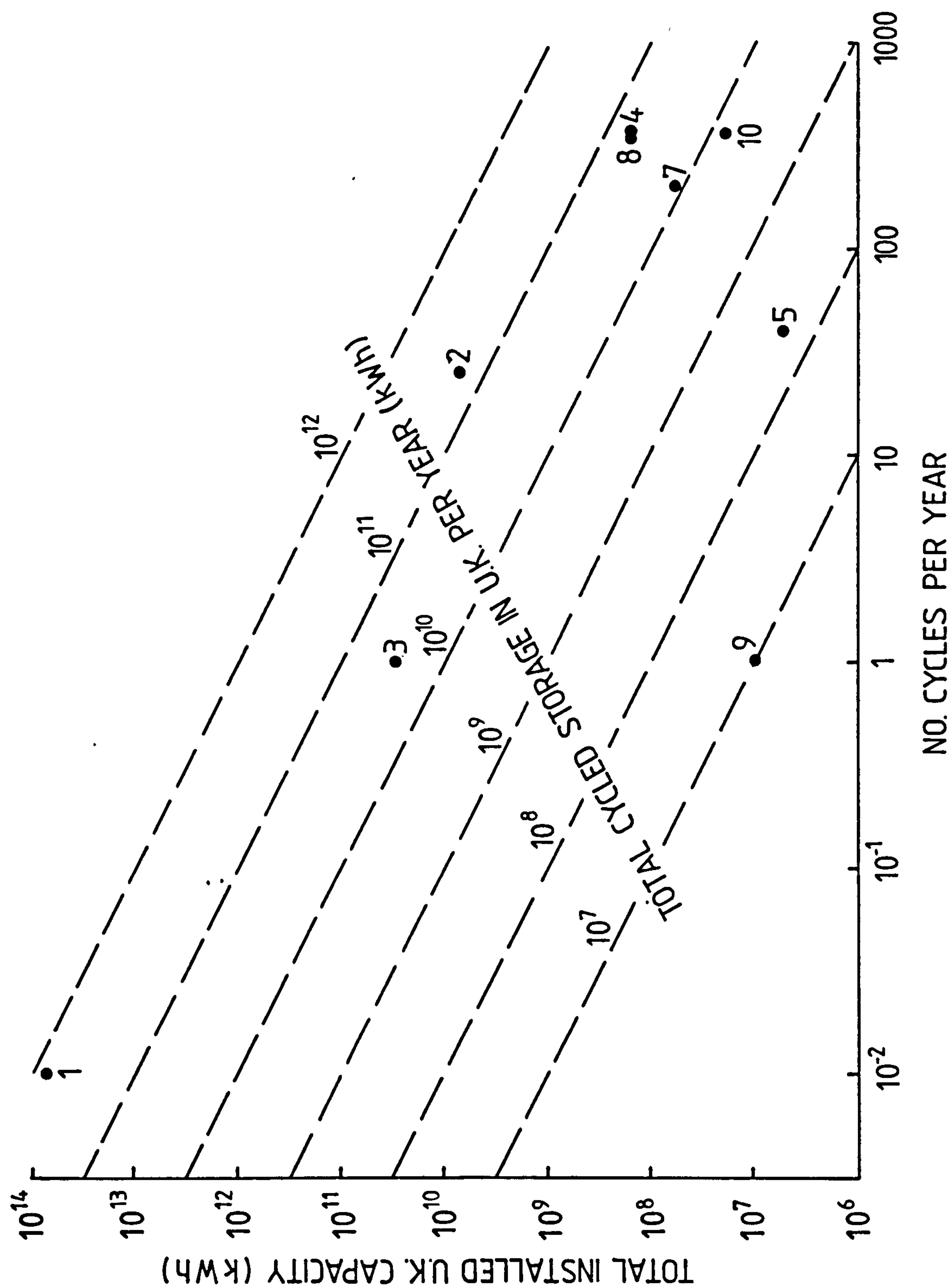


FIGURE 1-10 THE U.K. CAPACITY AND NUMBER OF CYCLES PER YEAR OF TEN PRIME STORAGE TECHNOLOGIES.

PRIMARY CATEGORISATION	TEMPERATURE CATEGORISATION	SECONDARY CATEGORISATION			CYCLE DURATION CATEGORISATION
		SOLID	LIQUID	GAS	
SENSIBLE	LOW 0 ↔ 100 °C	THE GROUND *** ROCK BED *	WATER ***		
	MEDIUM 100 ↔ 250 °C		OIL *	STEAM *	HOURLY *
	HIGH > 250 °C	CERAMIC REGENERATOR *			
LATENT					DIURNAL **
	LOW 0 ↔ 100 °C	SPECIAL HYDROCARBONS *	ICE ↔ WATER *** WAX *** SALT HYDRATES **		
	MEDIUM 100 ↔ 250 °C		SALT EUTECTICS *	WATER ↔ STEAM *	INTER-SEASONAL ***
THERMO-CHEMICAL					
	LOW 0 ↔ 100 °C		SODIUM SULPHIDE	WATER VAPOUR ***	
	MEDIUM 100 ↔ 250 °C			SULPHURIC ACID WATER VAPOUR **	
	HIGH > 250 °C				

FIGURE 1.11 CATEGORISATION OF TES TECHNIQUES BY : REACTION TO HEAT, PHASE, TEMPERATURE AND CYCLE DURATION, WITH EXAMPLES.

stored in solids, liquids or gases and latent heat can involve solid to liquid, solid to solid or liquid to vapour phase changes. All heat storage media can be slotted into one of these categories, and some common examples are shown in Figure 1.11.

Two further, but less rigorous categorisations can be made which are basically determined by the particular application for the TES. The first of these relates to the duration of the complete storage cycle (i.e. charge, standstill and discharge); cycles of up to a few hours are defined as short term or hourly storage, cycles of up to a few days are defined as diurnal storage and cycles of weeks or months are called long term or inter-seasonal heat storage. The second categorisation is used to indicate the maximum temperature level in the store, and here the definitions vary quite widely. On a general level, low temperature means up to 100°C, medium temperature denotes 100 to 250°C and high temperature is anything above 250°C. However, in certain application-oriented fields, narrower definitions are often employed so that, for example, in the heating of buildings, low, medium and high temperature can mean respectively up to 40°C, 40 to 70°C and 70 to 100°C.

A brief discussion of the various TES genres follows, and it will be seen that the most attractive or promising media have spawned further, more detailed, classification networks all to themselves.

1.3.1.1 Sensible Heat Storage

(a) Solids

Heat storage in solids has three main advantages :

- suitable materials are cheap and readily available;
- the materials are generally self-supporting so that containment of, and heat exchange to and from, the material are relatively easily achieved;
- the materials can be subjected to large temperature swings (i.e. many hundred °C) without changing state.

The main disadvantages of solids are :

- a low specific heat capacity, which can make stores bulky and heavy if very large temperature swings are not available.
- air must often be used as the heat transfer fluid; this may cause a high parasitic consumption by fans overcoming the pressure drop across the store and makes it expensive to use a hydronic heat distribution system;
- heat transfer rates to and from the storage media can be low due to the low conductivity of most suitable media, the reliance on conduction for heat transfer within the media and, in the case of air, the low thermal capacity of the heat transfer fluid.

Common examples of solid sensible heat stores are night storage heaters using off-peak electricity to heat feolite blocks, rock beds in air-based solar energy space heating systems, ceramic regenerators for high temperature industrial waste-heat recovery and the ground, i.e. earth, rock or clay, for the inter-seasonal storage of low grade solar heat. In the latter case, a heat-exchanger, e.g. plastic pipe, must be dug or vibrated into the ground. The fabric (e.g. walls) of buildings can also be considered in this category. A comparison of the pertinent physical properties of the storage materials concerned in these five examples is given in Table 1.4. It highlights the fact that low energy densities are inevitable unless high temperature swings are possible.

Material	Density kg m ⁻³	Specific Heat J kg ⁻¹ K ⁻¹	Application	Temperature Swing °C	Energy density over given ΔT kWh/litre
Feolite	4000	920	Night storage heaters	100 → 800	0.72
Ceramic	2000	850	Industrial regenerator	20 → 1500	0.7
Rock	2785	790	Solar rock beds	20 → 50	0.018
Brick	1670	840	"passive" solar	10 → 20	3.9 . 10 ⁻³
Clay	1270	790	Inter-seasonal solar	10 → 15	1.4 . 10 ⁻³

TABLE 1.4 Five Common Materials for Sensible Heat Storage in Solids

(b) Liquids - Water

Water is by far the most common TES medium, and for good reasons. It has the highest specific heat capacity of all natural materials. It is benign, ubiquitously available and almost free. Its physical properties and heat transfer characteristics are extremely well-known. In many applications, stored hot water can be used directly (i.e. without heat exchange) in other parts of the system, e.g. district heating, domestic hot water and water-filled radiators.

The only major drawback to the use of water is its relatively narrow operating temperature range (0 - 100°C). At the low end anti-freeze is often mixed with water to prevent freezing. Two other potential problems with water are mineralisation and biological activity, e.g. algae or bacteria growth; both problems can usually be controlled by appropriate precautions. Similarly care must be taken to avoid corrosion of storage vessels and any other metal components in the system, e.g. valves and pipework, with which the stored water comes into contact.

Hot water storage can be split into a number of categories as shown in Figure 1.12. The main distinction is whether the store is above or below the ground. Subsurface hot water stores include ground water (aquifers), natural or man-made caverns and disused mine-shafts. In these cases, particularly, mineralisation must be carefully monitored and steps taken to prevent the clogging of any pipework or heat exchangers in the circuit. Above ground, hot water can be stored in metal, plastic or concrete tanks, lined pits or natural or man-made reservoirs. It is common to classify above ground water stores according to their volume, with small, medium and large referring roughly to up to 1m^3 , 1 to 100m^3 and 100m^3 to $100,000\text{m}^3$ respectively.

Two common examples of hot water stores, domestic hot water cylinders and aquifers, were considered in Section 1.2. Another and rather special example is the solar pond, which manages to combine into one heat storage and solar energy collection. Hot water storage in tanks is the main subject of the other chapters of this thesis.

(c) Liquids - Non-water

Heat transfer oils and other hydrocarbons are the main members of this category. They are employed (in preference to water) when temperatures greater than 100°C must be stored, or corrosion precludes the use of water. Examples of such stores are found in the Prometheus Project [Ref.1.16], involving oil fed medium-temperature (120°C) solar collectors, and a novel high temperature (300°C) wax store proposed by Hawker Siddeley Power Engineering [Ref.1.17].

(d) Gases

Sensible heat storage in gases almost inevitably means storing the gas at a high pressure as well as temperature in order to keep the storage volume reasonable. There is one outstanding example of this technology and that is in steam accumulation. There are many of these installed in industrial steam plant systems, and, apparently, a crying need for many more [Ref.1.18]. The main barriers to their more widespread adoption are high capital cost and a lack of awareness of their potential benefits.

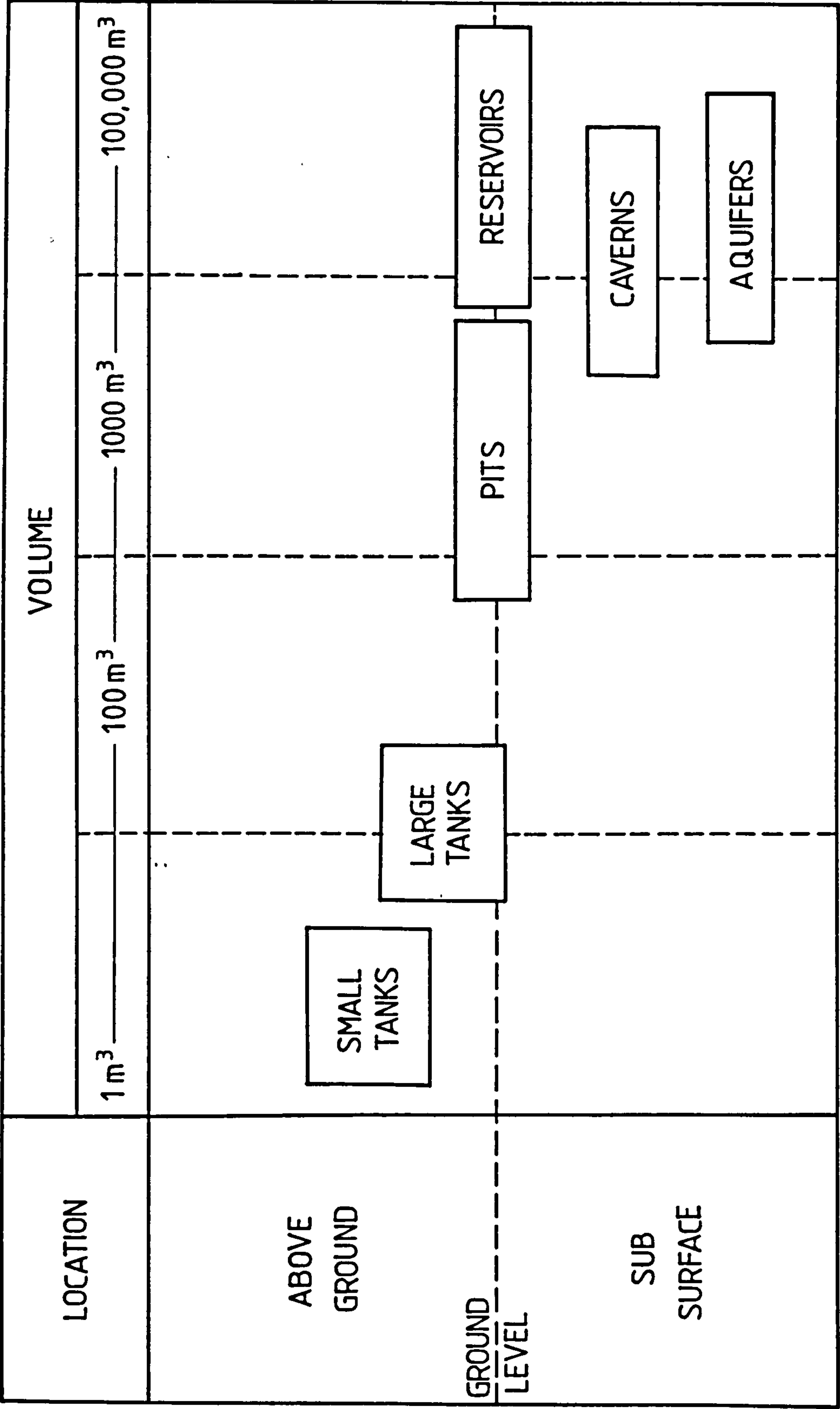


FIGURE 1-12 CATEGORISATION OF HOT WATER STORAGE TECHNIQUES BY LOCATION AND VOLUME.

1.3.1.2 Latent Heat Storage

When materials change state or phase they absorb or release large quantities of heat relative to their sensible heat capacity and yet they remain at an almost constant temperature. For the purposes of heat storage these properties promise two extremely important benefits :

- a much higher energy density than sensible heat systems, and
- a near isothermal discharge temperature.

However, intense research efforts around the World during the last fifteen years is only now beginning to yield commercially viable products or systems.

(a) Solid-liquid Phase Change

This is the most convenient change of phase to exploit as volume alterations during the change of state are relatively small (up to about 10%), and latent heats of fusion are high. There are many candidate phase-change materials (PCM's) for the purposes of heat storage, both organic and inorganic. However, a suitable PCM should satisfy most of the following criteria :

A. THERMODYNAMIC

- Appropriate melting point or melting range
- High latent heat of fusion per unit volume
- High specific heat
- High thermal conductivity
- Congruent melting
- Small volume change on melting.

B. KINETIC

- High rate of crystallisation
- Little or no subcooling

C. CHEMICAL

- Chemically stable
- Non-corrosive to containers
- Non-toxic
- Non-flammable.

D. ECONOMIC

- Reasonably cheap
- Available in large quantities.

Organic materials, such as paraffin waxes, score highly on this list, except on two key points: they are generally expensive and have low densities and therefore low latent heats per unit volume. Inorganic materials, conversely, are generally cheaper and have high volumetric latent heats but many are incongruent, most subcool significantly and some are corrosive or toxic. Incongruency leads to gravitational phase segregation and a progressive degeneration of the material's heat storage capabilities. To prevent segregation, researchers have developed additives to act as suspension media or thickeners [Refs.1.19 and 1.20]. Neither approach has been satisfactorily validated over long-term tests. To reduce subcooling, three techniques are possible : (i) the addition of a 'seed' material, with a similar crystal structure to that of the PCM, (ii) the addition of microscopically 'rough' materials, e.g. activated charcoal, to provide potential nucleating sites and (iii) if the first two fail, the use of a 'cold finger', which maintains a small quantity of the PCM permanently in its frozen (crystalline) state.

Salt hydrate PCM's are commercially available in the U.K. from two companies : Calor Ltd., and Dow Chemicals. The Calor materials are all thickened incongruent hydrates, whilst the Dow materials are all congruent hydrates. The latent heats and melting points of these materials and some organic PCM's are plotted in Figure 1.13. The datum for the ice-water transition is also shown on this figure. This phase change is highly expedient in applications where 0°C is a suitable storage temperature. Ice storage in air conditioning systems is now commercially viable both on a diurnal

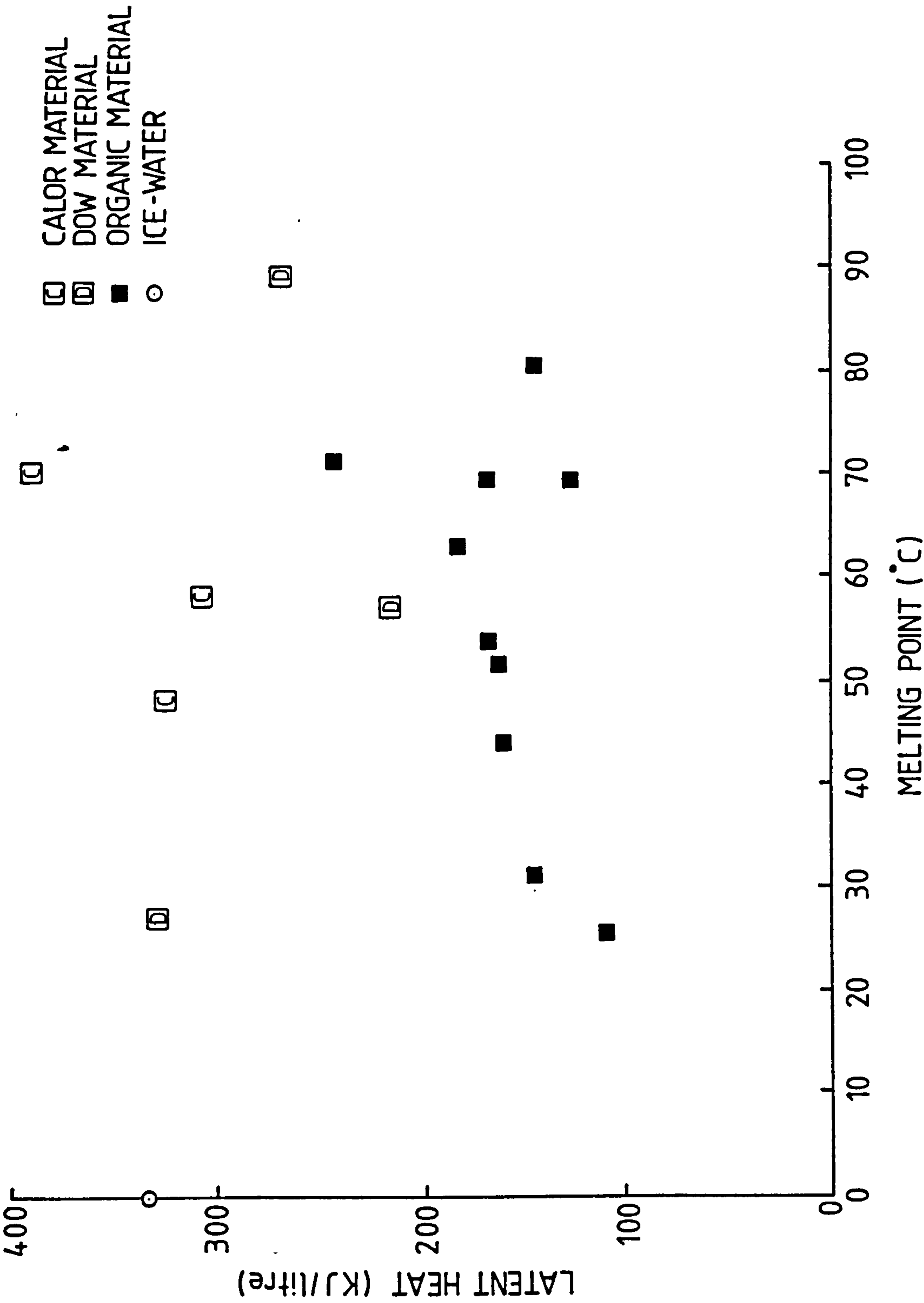


FIGURE 1·13 THE LATENT HEATS OF SOME COMMERCIALY AVAILABLE PCM'S.

cycle where the ice is created at night using off-peak electricity [Ref.1.21] and on a seasonal cycle in climates having cold winters and hot summers (e.g. Northern U.S.A. and Canada) where ice is produced using ambient winter temperatures [Ref.1.22].

Heat-exchangers for Solid-liquid PCM's

Effective heat transfer to a heat store is imperative for it to be a worthwhile component of a system. Most PCM's have relatively low thermal conductivities in their solid phase and also shrink on solidifying. Both these factors lead to a high thermal resistance being created around the heat exchange surface during a store discharge. As a result, the heat exchange surface area per unit of storage medium volume must be very large. This is still the case even in applications where the critical store power density occurs during the charge process.

Potential heat exchange techniques fall into two basic categories : one in which the PCM is encapsulated in, for example, a large number of cans, tubes or trays and the other in which the PCM is all in one bulk and heat exchange is effected by, for example, immersed tubes. For most applications, encapsulation is cheaper; however it requires specialised automatic production plant which may or may not exist. The containers must be compatible with the PCM and capable of withstanding the mechanical stresses imposed during repeated thermal cycling through the phase transition. They should not inhibit heat transfer and, particularly with salt hydrates, must be impervious to water. The size and shape of the containers should be determined by the power density requirements of a particular application and the most economical configuration. Table 1.5 gives a general comparison of the merits and disadvantages of the most commonly proposed heat exchanger designs.

(b) Solid-solid Phase Change

Some materials undergo reversible transformations involving high latent heats whilst remaining in a solid state. In particular, certain hydrocarbons exhibit this behaviour during transformations of their crystal structure [Ref.1.23]. Such compounds have the potential

GENERIC TYPE	PRINCIPLE	INSTALLATION FACTORS	HEAT EXCHANGER VOLUME	POWER DENSITY	NUCLEATION (For salt hydrates)	SEGREGATION (For salt hydrates)	COST
ENCAPSULATED PCM	CAPSULES	PCM	v.low	v.good	poor	fair	v.low
	TRAYS	HANDLED	fair	good	fair	good	high
	TUBES	IN FACTORY	fair	good	poor	poor	low
BULK PCM	COILED TUBE BANK	PCM	v.high	poor	fair	poor	v.high
	DIRECT CONTACT	HANDLED	v.low	good	v.good	good	fair
	HEAT PIPE	ON	low	good	fair	fair	high
	ROTATING CYLINDER	SITE	v.low	v.poor	v.good	v.good	v.high

TABLE 1.5. Heat Exchanger Configurations for PCM Based Stores

for providing a practical method to incorporate latent heat storage into building materials such as wood, gypsum board and lightweight concrete, which could have significant implications in the areas of passive solar design and room temperature control. However their cost effectiveness and long term reliability have yet to be demonstrated.

(c) Liquid-vapour Phase Change

This category is dominated by the water to water vapour transition, either in conventional steam to condensate systems, or in association with hygroscopic materials such as silica gel. The latent heat of vaporisation of water is very high, about 2.5 MJ/kg, but storage of the vapour phase can be expensive, or bulky. Hygroscopic materials, like silica gel, can absorb water in quantities up to 40% of their weight, releasing the latent heat. Recharge of the store is achieved by evaporating off the water which is usually released to the environment. Practical energy densities of about 0.2 kWh/litre have been achieved, but too much dehumidification of the ventilation air during the discharge (water absorption) stage can reduce significantly the system's attractions.

1.3.1.3 Thermo-chemical Heat Storage

The basic components of these systems are an absorber and a working fluid. When the working fluid is allowed to react with the absorber, heat is released, generally the heat of absorption in addition to the heat of reaction. The store is regenerated by the application of heat to the absorber-working fluid mixture, which is thereby separated into its original components. Energy storage densities of around 0.3 kWh/litre have been achieved. Unfortunately, useful thermochemical storage must involve at least one highly reactive substance which, almost by definition, is likely to be toxic, flammable or corrosive. There is therefore doubt about the acceptability of such systems, at least in the domestic environment.

(a) Solid-vapour Systems

This category comprises a solid absorber and usually water as the working fluid. Probably the most advanced system is being developed

in Sweden, and is known as "TEPIDUS" [Ref.1.24]. It uses sodium sulphide and water. Other research has concentrated on the use of zeolites as absorbers [Ref.1.25].

(b) Liquid-vapour Systems

These systems work on the same principle as the solid-vapour ones. Examples of components that have been the subject of detailed research are : sulphuric acid - water, sodium hydroxide - water and ammonia - water.

1.3.1.4 Summary of TES Techniques

- (i) Sensible heat stores are generally simple, reliable, use cheap materials, but require large volumes.
- (ii) Latent heat stores require elaborate means of heat exchange, use expensive materials which are prone to degradation, but require maybe two to three times less volume than sensible heat stores.
- (iii) Thermochemical stores promise further significant volume reductions, but are still at the research stage and must be proved to be viable and safe in practice.

1.3.2 The Need for TES

The purpose of this section is to focus attention briefly on four important areas of the energy scene with perhaps the greatest market potential for existing and new TES technologies. The main advantages of utilising heat storage can be summarised as :

- Capital cost reduction, by reducing the maximum system load and hence the size of plant required to supply a given amount of energy;
- Fuel conservation, by improving the efficiency of energy supply or by enabling greater use of recovered industrial process heat or by creating further applications of renewable energy sources, and

- Simpler and cheaper system regulation and control, since heat storage provides an inherently flexible means of coping with short-term fluctuations in energy (heat) demand.

The following examples serve to highlight how these benefits accrue in practice.

1.3.2.1 The C.E.G.B. Trough

There are considerable daily (and seasonal) variations in the demand on the U.K. Central Electricity Generating Board (C.E.G.B.) national grid as shown in Figure 1.14. The night time (off-peak) trough amounts to about 70 GWh or of the order of 10% of a typical 24 hour demand. Energy storage can perform two important functions: correction of fluctuations in frequency caused by short-term variations in demand and reduction of day time (i.e. peak) generating load by meeting part of the peak demand with energy stored during the previous night.

Generally, both centralised bulk storage (i.e. pumped hydro) and decentralised demand management storage techniques can be applied. In the case of short-term variations, frequency control is currently accommodated by pumped hydro, but the very cause of the problem could be greatly mitigated by demand management in the form of tele-control of water heaters, freezers, etc. At the same time, stringent peak demand tariffs coerce heavier consumers to a smoother demand profile, a practice that will become increasingly effective as more and more electronic energy management systems are installed.

Transferring some electricity production from day to night produces three valuable results :

- The C.E.G.B. supply capacity can be reduced. This means that older, less efficient, power stations, now only brought on stream to meet peak demand, can be completely shut down, and there is a less pressing need to construct new power stations. The highest peak demand (46219 MW) ever supplied by the C.E.G.B. occurred on January 17th 1985 [Ref.1.26], incidentally during the miners' strike. Examination of the

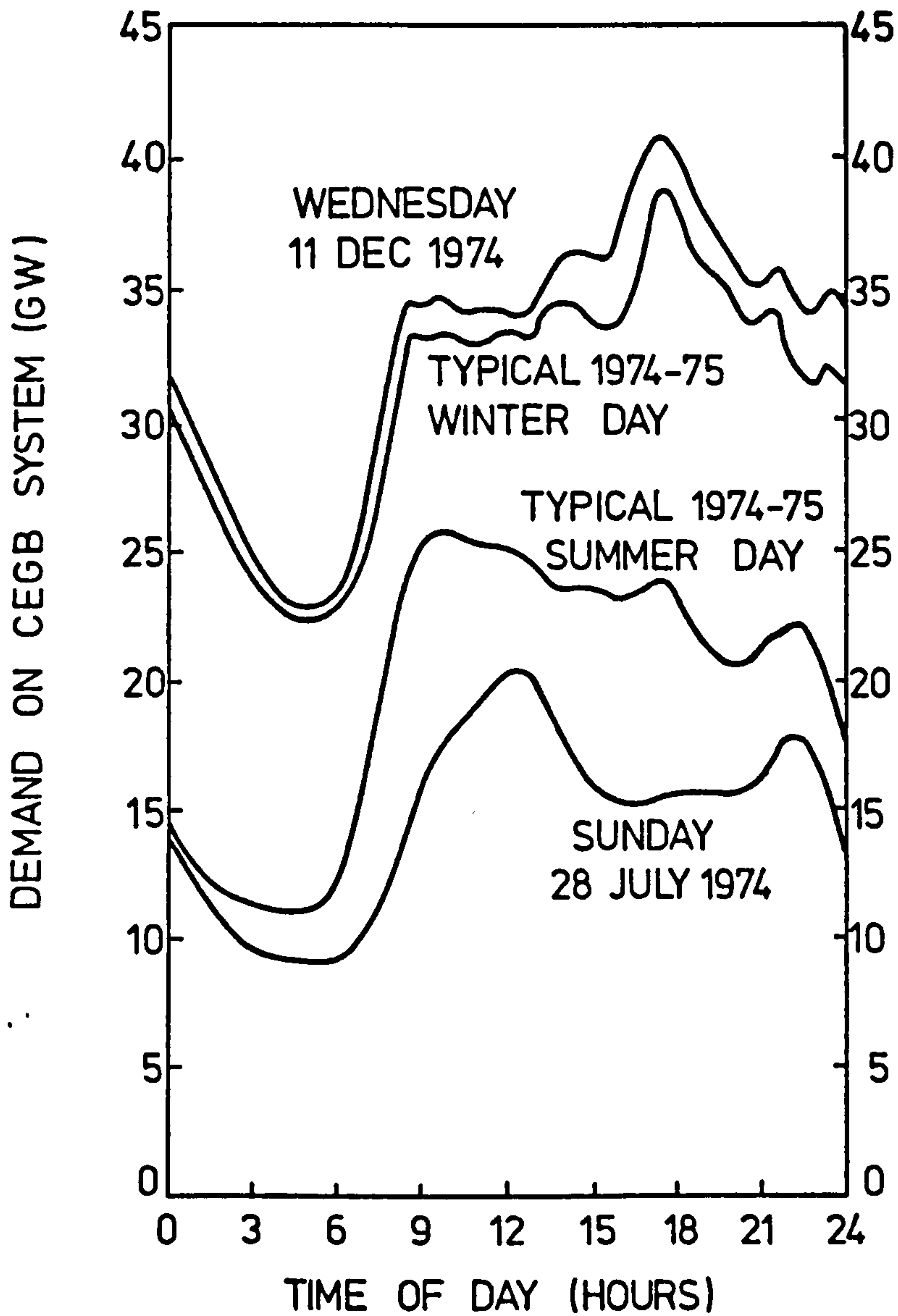


FIGURE 1.14 SUMMER AND WINTER DEMANDS, ON CEGB SYSTEM IN 1974-75, INCLUDING DAYS OF MAXIMUM AND MINIMUM DEMANDS

demand profile on a similar day, shown in Figure 1.15, reveals the sharpness of the peak and therefore the high marginal benefit in this respect for each additional unit of storage.

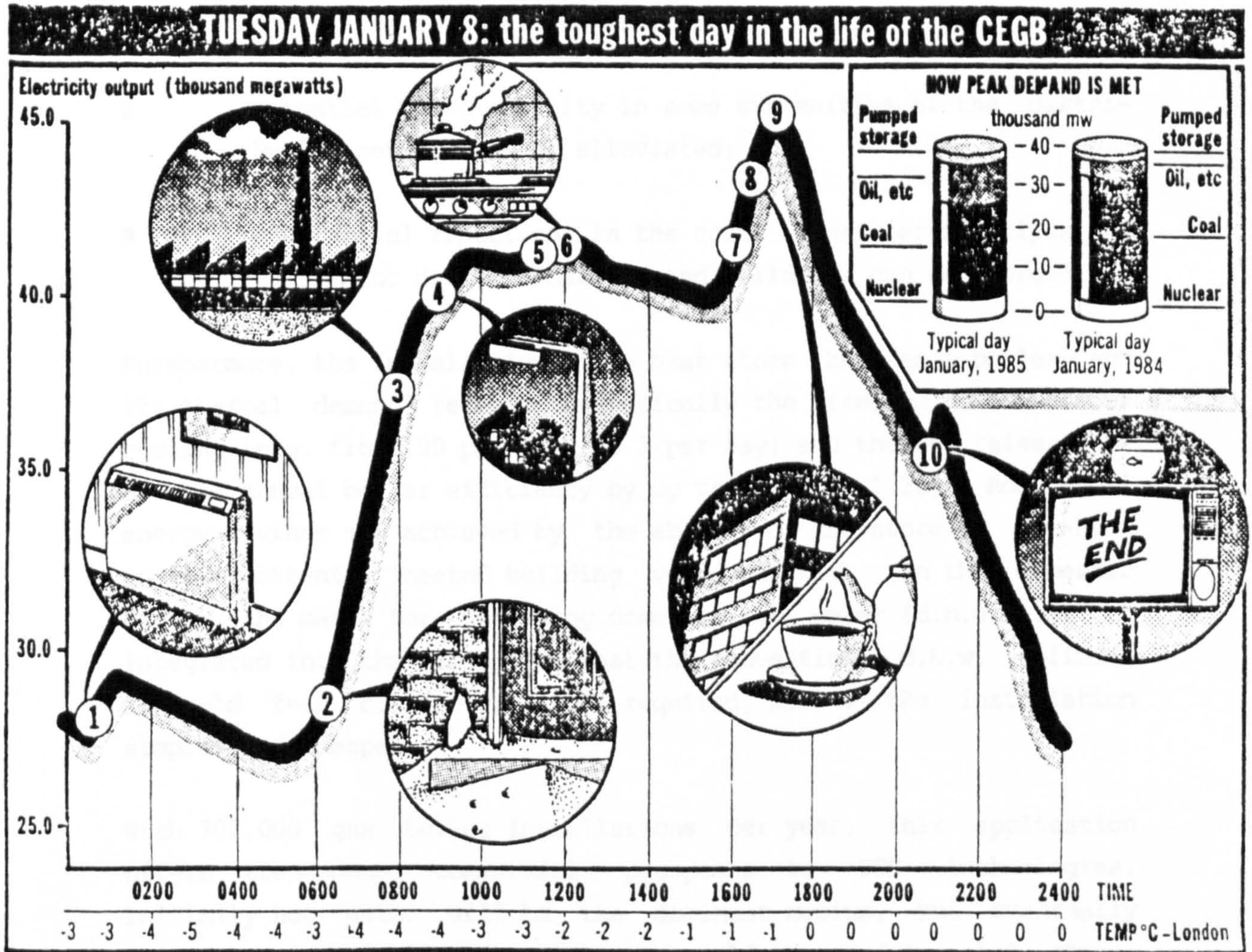
- There will be less part load operation of thermal plant, which will increase its generating efficiency and reduce its failure rate. This benefit would become almost an obligation if nuclear capacity started to exceed the summer night demand, since it is impracticable as well as uneconomic to vary the output of nuclear power stations.
- There will be an overall increase in generating efficiency due to a greater utilisation of higher merit-order plant.

There are geographical and environmental limits to the extent to which day to night transfer can be effected by pumped hydro, and therefore demand management particularly in the form of overnight water heating and off-peak storage heaters is highly profitable. This fact is strongly reflected in the cost of off-peak electricity, which is now approaching one third the price of day time rates.

In 1980, the total energy consumption by domestic off-peak space heating appliances was 11,450 GWh [Ref.1.27], or about 60 GWh per day. To fill the off-peak trough would therefore require at least a doubling of current consumption. Last year (1984) about 300,000 off-peak storage heaters were installed representing a consumption of maybe 4 GWh per day. Given that a large proportion of these installations were replacements for existing storage heaters, the growth in off-peak consumption is probably only around 3% per annum, or a doubling time of 23 years. There is therefore a huge market potential for TES products, especially those which can be connected to hydronic distribution systems (radiators) to give 'full' central-heating.

1.3.2.2 Diurnal TES in Gas-fired Central Heating Systems

The national gas distribution network must satisfy similar demand profiles as the electricity grid, but bulk gas storage techniques,



1: Big switch-on of storage heaters using Economy Seven off-peak tariffs. 2: Britain begins to wake up to one of the coldest mornings of winter. 3: Breakfast time overlaps with switch-on in factories and offices. 4: Overcast day means street lights stay on, postponing usual morning decline in demand. 5: Highest-ever morning demand - 43,000 megawatts. 6: Mid-day peak as Britain cooks its lunch. 7: Lighting-up time 4.31 (London). 8: Late afternoon surge. Factories and offices still working: people arriving home switch on kettles, heating, etc. 9: Peak 'spot' demand at 5.01 pm is 45,046 megawatts - the highest ever recorded. 10: The Italian Job ends on BBC-1 and Fresh Fields on ITV, several million people make a hot drink or go to the lavatory, creating a surge in demand from water pumping stations.

FIGURE 1.15 THE PEAKS AND TROUGHS IN THE DEMAND ON THE CEB ON A VERY COLD DAY (FROM THE SUNDAY TIMES 13 JANUARY 1985).

from the familiar low pressure holders to line packing and salt cavities, are relatively easy and cheap (20p/kWh of storage capacity compared to about £20/kWh of storage capacity at Dinorwic). However, heat storage at the point of use smooths the heat demand, removes the need for boiler over-capacity and therefore lowers the peak demand on the gas supply, providing two major benefits :

- potential under capacity in some extremities of the distribution network can be alleviated; and
- substantial reductions in the costs of gas service pipes and meters for new or rehabilitated buildings can be accrued.

Furthermore, the installation of a heat store between a boiler and its thermal demand reduces dramatically the frequency of boiler cycling (e.g. from 100 per day to 3 per day) and thereby raises the overall annual boiler efficiency by up to 15% [Ref.1.28]. Additional energy savings are achieved by the ability of the store to pre-heat an intermittently heated building very quickly. In the domestic sector, the means for providing domestic hot water (d.h.w.) can be integrated into the store, so that the conventional d.h.w. cylinder and cold feed cistern are not required, making the installation simpler and cheaper.

With 700,000 gas boiler installations per year, this application offers tremendous commercial prospects to TES technologies. Initially hot water will be the dominant medium, but eventually encapsulated solid-liquid PCM stores could command a market share, particularly in combination with condensing gas boilers.

1.3.2.3. Industrial Waste Heat Utilisation

In industry, particularly during batch operations, the process heat often cannot be reused simultaneously and so is wasted. TES allows the energy to be held for use later in other processes or for space heating. Existing applications include the use of regenerators in the iron and steel industry and in glass making. Surveys in a number of countries indicate a large potential for TES : in Sweden , for example, it is estimated that 13% of all the energy used by

industry could be recycled via short term heat storage [Ref.1.29]. The main barrier to a more widespread introduction of such systems is the reluctance of many industries, particularly in the U.K., to invest in non-productive capacity.

1.3.2.4 Inter-seasonal Storage of Solar Energy

The main difficulty in exploiting solar energy for the space heating of buildings is the very great seasonal mismatch between supply and demand, as is made apparent by figure 1.16. Inter-seasonal heat storage is the most cost-effective means of providing the majority of a building's space heating needs via solar energy, although it is not yet financially competitive with conventional energy. However, whether owing to market, environmental or political forces, this technology might well see its time come. This prospect has generated intensive research and development in many countries, with Sweden, France, Canada and the U.S.A. leading the way. Sadly, the U.K.'s efforts in this area are conspicuous by their absence.

At a recent Conference, dedicated to heat storage [Ref.1.30], the latest results of this research were reported: in a number of countries, a wide variety of seasonal heat (and cold) storage technologies have been operated successfully at full scale. They have been proven technically, and are becoming increasingly attractive, due to the success of the research in reducing capital costs.

Two Swedish projects demonstrate the remarkable progress that has been achieved : "Suncourt" [Ref.1.31] comprising seventy-one apartments built around a glazed-in courtyard of 650m^2 , below which drill holes in the bedrock provide seasonal heat storage; and "Sunclay" [Ref.1.32] involving a school with solar collectors linked via plastic pipes to a ground heat store made up of $87,000\text{ m}^3$ of clay. The capital costs of these stores are only about 40p/kWh of storage capacity and payback periods below 10 years are being predicted, so even stores that are only cycled once per year can be made to pay.

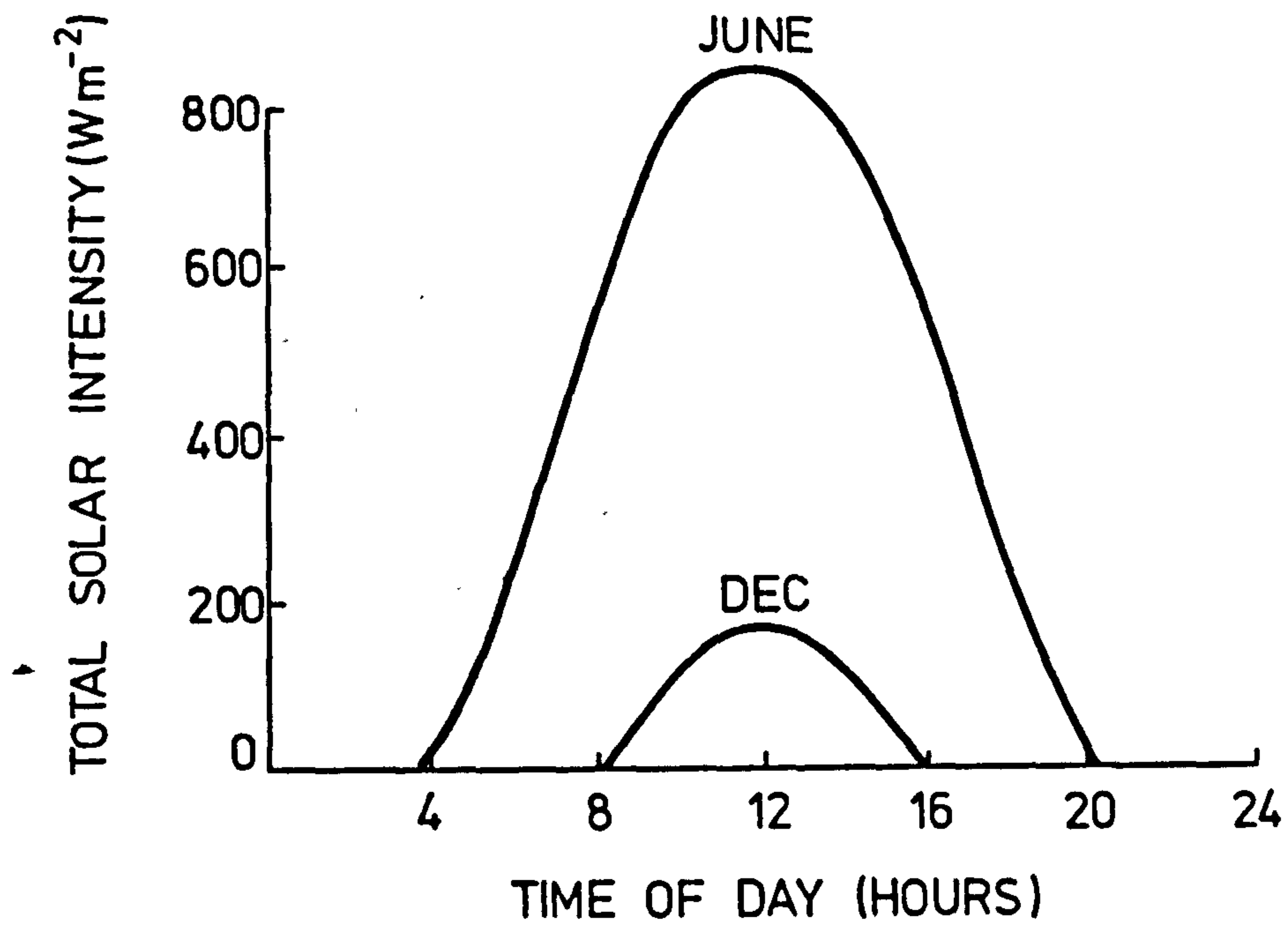
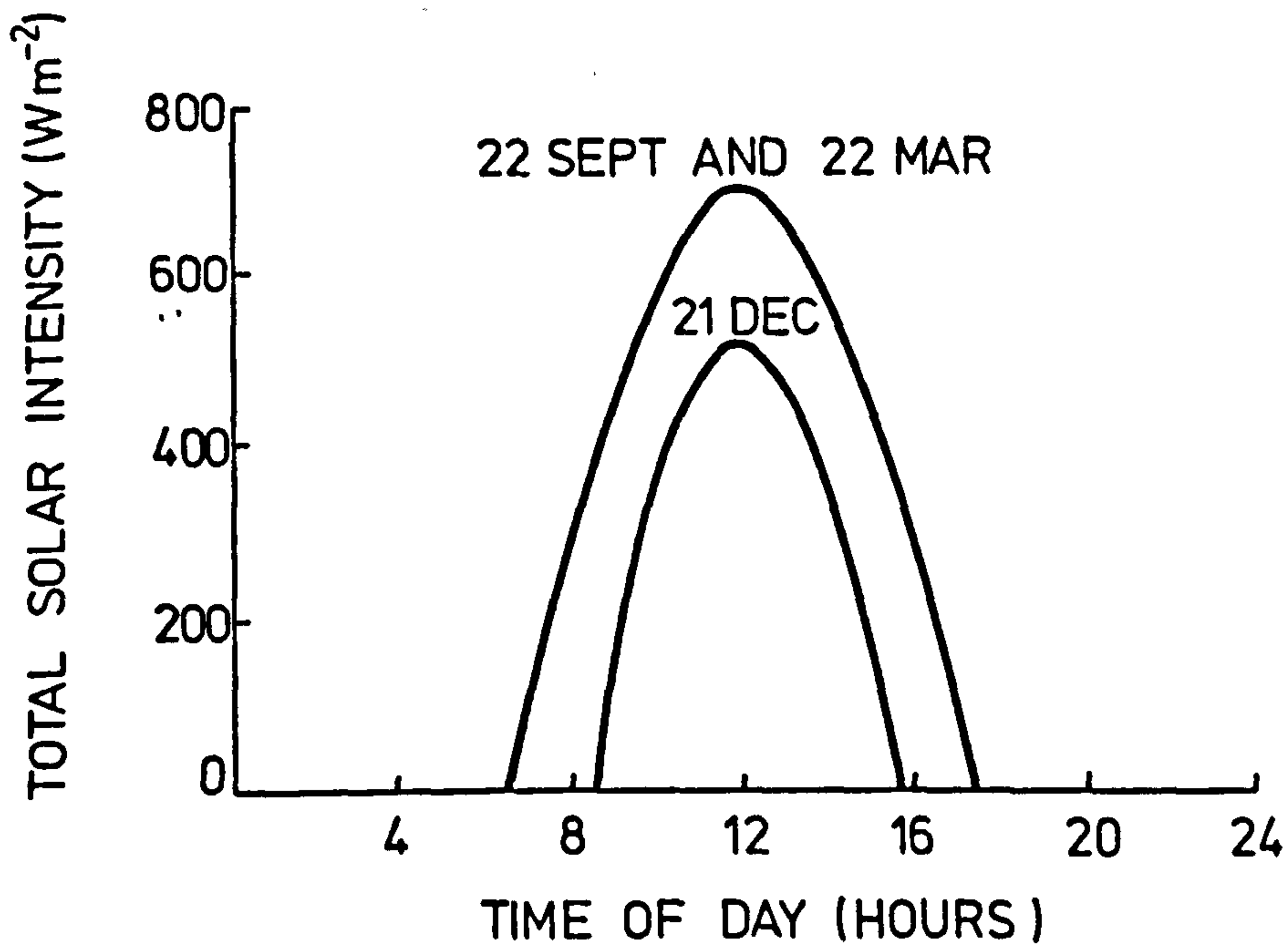


FIGURE 1.16 TOTAL SOLAR RADIANT INTENSITIES (DIRECT PLUS DIFFUSE) STRIKING HORIZONTAL SURFACE (TOP) AND SOUTH FACING WALL (BOTTOM)

1.4 THESIS STRUCTURE

Following this introduction into the field of energy storage and heat storage in particular, the rest of the thesis concentrates on the main subject of the research that has been conducted: hot water storage. In the next chapter there is a thorough review of other experimental work on this topic and of computer modelling techniques that other researchers have developed to simulate the behaviour of hot water stores.

Chapter Three describes the design and performance of the test facility that was constructed to perform experiments on a water store. It also gives details of a water store built for this purpose, the various porting arrangements and the experimental procedure. The results of the experiments are discussed and analysed in Chapter Four.

Chapter Five describes a computer program that was specially developed to model the observed behaviour of the water store, and gives comparisons between the experimental and predicted results. The integration of the hot water store model into a computer model of a complete heating system incorporating such a store is described in Chapter Six, and predictions of the annual performance of the system are given which include design parameter sensitivity studies. Finally, the conclusions to be drawn from the work are stated in Chapter Seven.

CHAPTER TWO

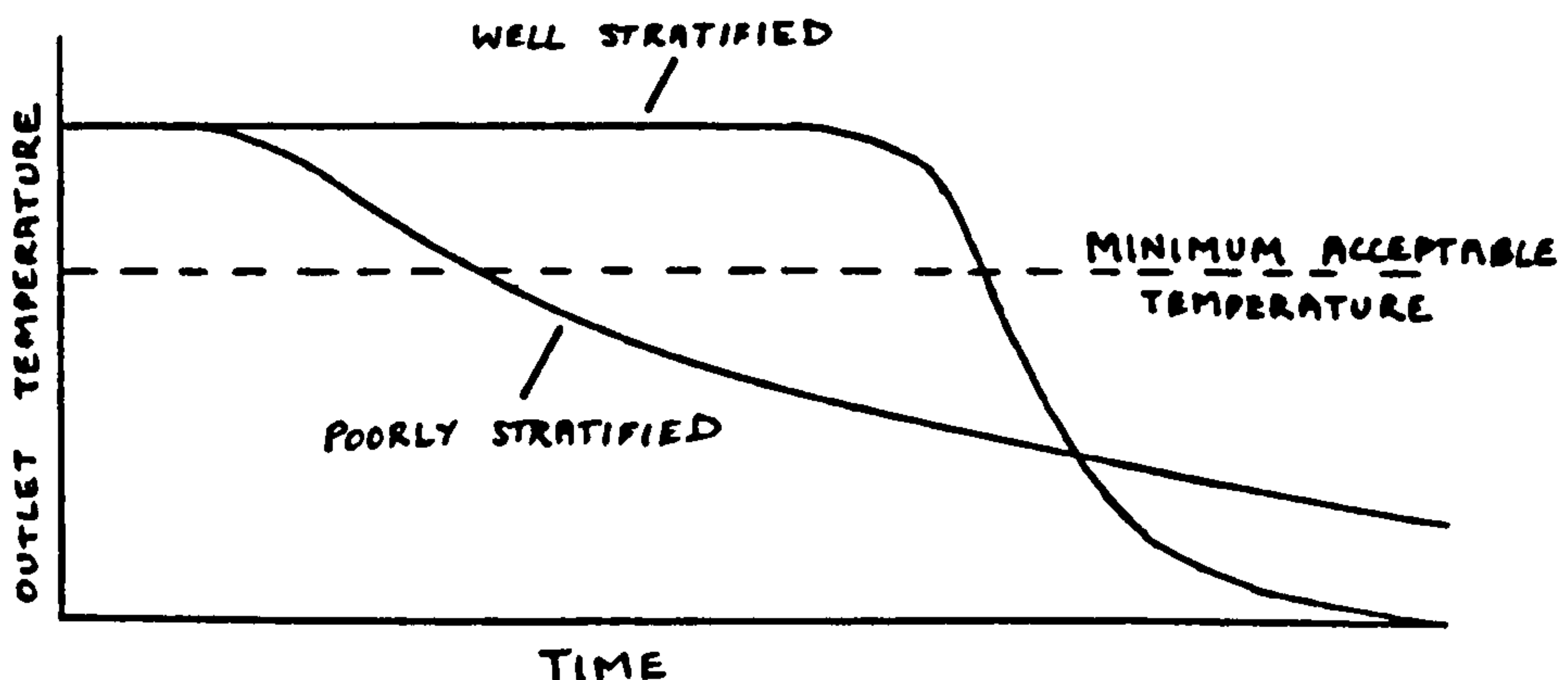
STRATIFIED HOT WATER STORAGE - REVIEW

2.1 INTRODUCTION

The thermal behaviour and performance of a hot water store under given conditions are determined by just two factors: the degree of temperature stratification that is induced in the water and the rate of heat loss from the vessel. The latter is simply a function of the thickness of insulant applied to the vessel together with the potential for any thermosyphon-driven flows up pipes connected to the store. Thermal stratification, on the other hand, is a complicated phenomenon that can be created or destroyed to a greater or lesser degree in a variety of ways.

Thermal stratification in a water store describes the situation where there is a vertical temperature gradient. The gradient can be created because, above 4°C, the density of water decreases with an increase in temperature, and therefore, warmer water is more buoyant and therefore floats on colder, heavier water. Stratification is significant in practice in a number of ways, for example:

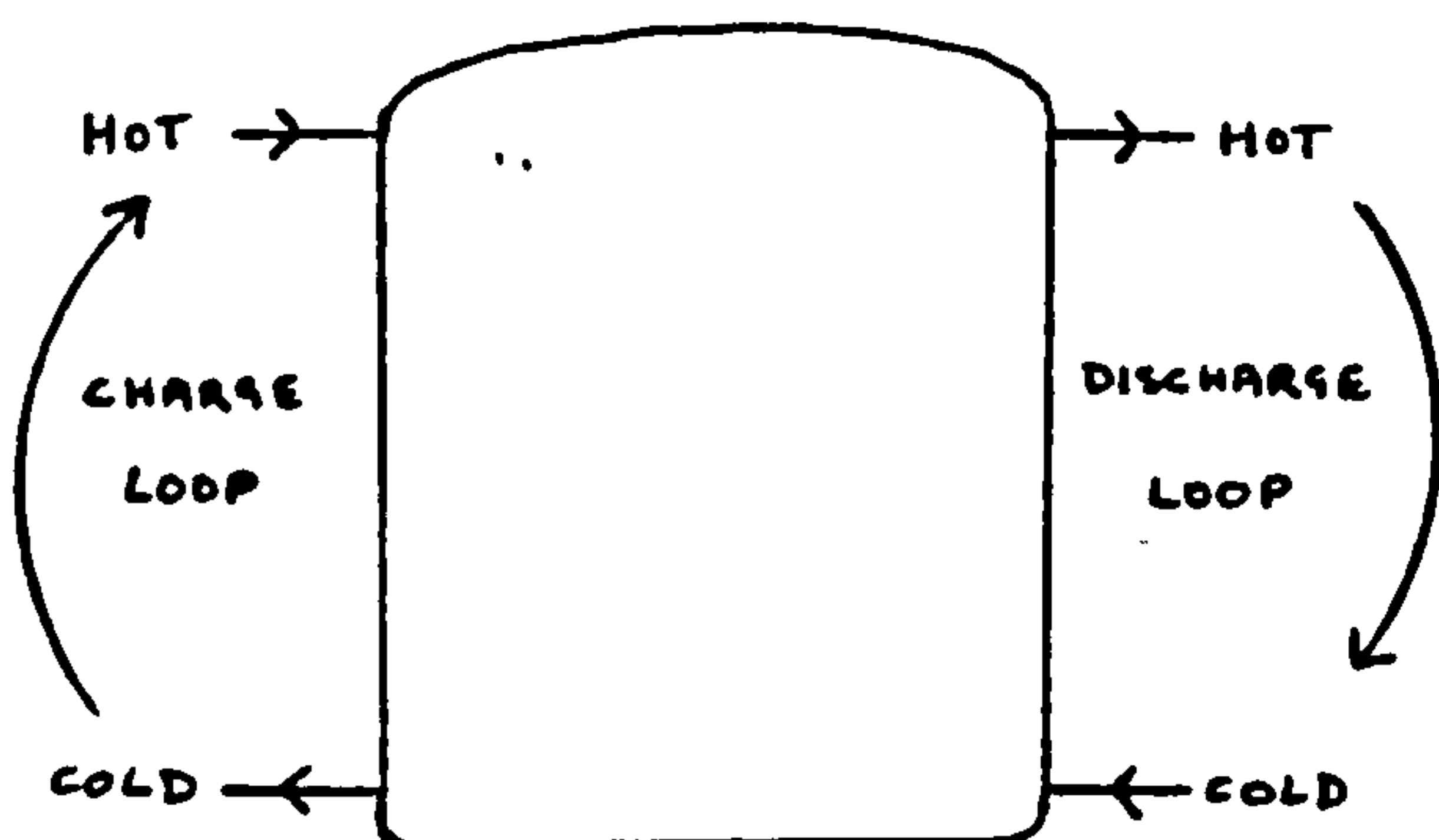
- (i) During the discharge of a water store, the outlet temperature depends on the degree of stratification being created. As illustrated below, if good stratification is maintained, the outlet temperature remains roughly steady at its initial (maximum) level until nearly all the stored heat has been withdrawn. In contrast, poor stratification leads to a degrading of the outlet temperature.



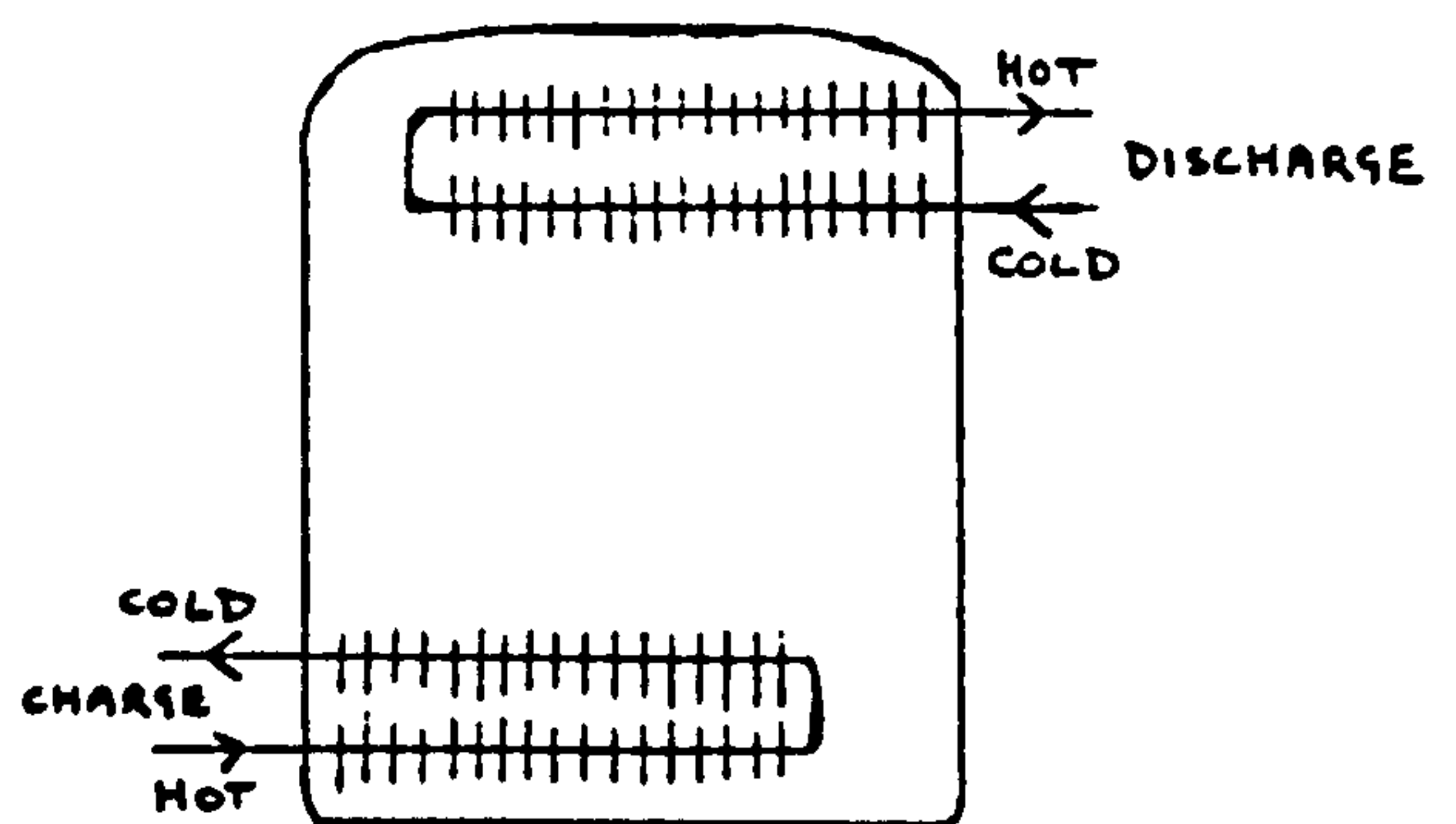
This effect can be crucial in certain applications, such as domestic hot water service or the pre-heating of buildings (see Chapter 6).

- (ii) During the charge of a water store, the outlet temperature (i.e. that of the water entering the heat source) similarly depends on stratification. With good stratification, the outlet temperature can be maintained at its initial, minimum level. In some devices, for instance solar collectors and fossil-fuel fired boilers, thermal efficiency is enhanced if the inlet temperature can be kept low.
- (iii) Stratification can occur during the discharge of waste heat from power stations etc. into lakes and rivers; stratification can prevent unnecessary thermal pollution of lower depths [ref.2.1].

The degree of stratification produced in a water store is determined primarily by how much turbulent mixing in direct systems ((a) below) or convective motion in indirect systems ((b) below) is created during the charge and discharge processes.



a) DIRECT SYSTEM



b) INDIRECT SYSTEM

Once a temperature gradient exists, there are three further mechanisms which act to reduce it, although all are comparatively slow processes. The first is temperature equalisation by molecular conduction, i.e. heat conduction from a hot layer to a colder layer below. Because the thermal conductivity of water is relatively low, this form of heat transfer does not result in a rapid deterioration of stratification.

The second mechanism is convective motion caused by conduction in the vessel wall; this naturally becomes more important the greater the thickness and conductivity of the wall material. The third mechanism is convective circulation driven by heat loss through the walls which is obviously largely determined by the thickness of insulant on the vessel.

Many of the issues raised in this introduction have been tackled by other researchers, either experimentally, theoretically or using computer simulation techniques. The rest of this chapter reviews the work that has been reported in the literature.

2.2 REVIEW OF EXPERIMENTAL WORK

2.2.1 Early Work

The most pertinent research into stratified hot water stores has been carried out in the last ten years, the impetus often being research into solar energy systems. Before this, however, some highly relevant work was done on other thermally stratified fluids. In particular, between 1962 and 1966, Turner [Ref.2.2], published four papers concerned with stratification in the atmosphere, the final one of which considered jets with negative or reversing buoyancy. He showed by dimensional analysis and confirmed by experiment (see figure 2.1) that the depth of penetration of a vertical turbulent jet with negative buoyancy was dependent on the jet volume flux, the initial density difference and the inlet port diameter. Although his experiments used salt water injected vertically upwards into fresh water, this situation closely resembles cold water entering vertically the bottom of a tank of hot water, or the identically performing converse of hot water entering vertically the top of a tank of cold water. Turner's methodology has therefore formed the foundation of the analysis in this work (see Chapter 4).

2.2.2 Stratification Decay

Another field which generated early relevant research was the storage of cryogenic liquids, often for aerospace applications. The main issue here was the destratification caused by conduction in the

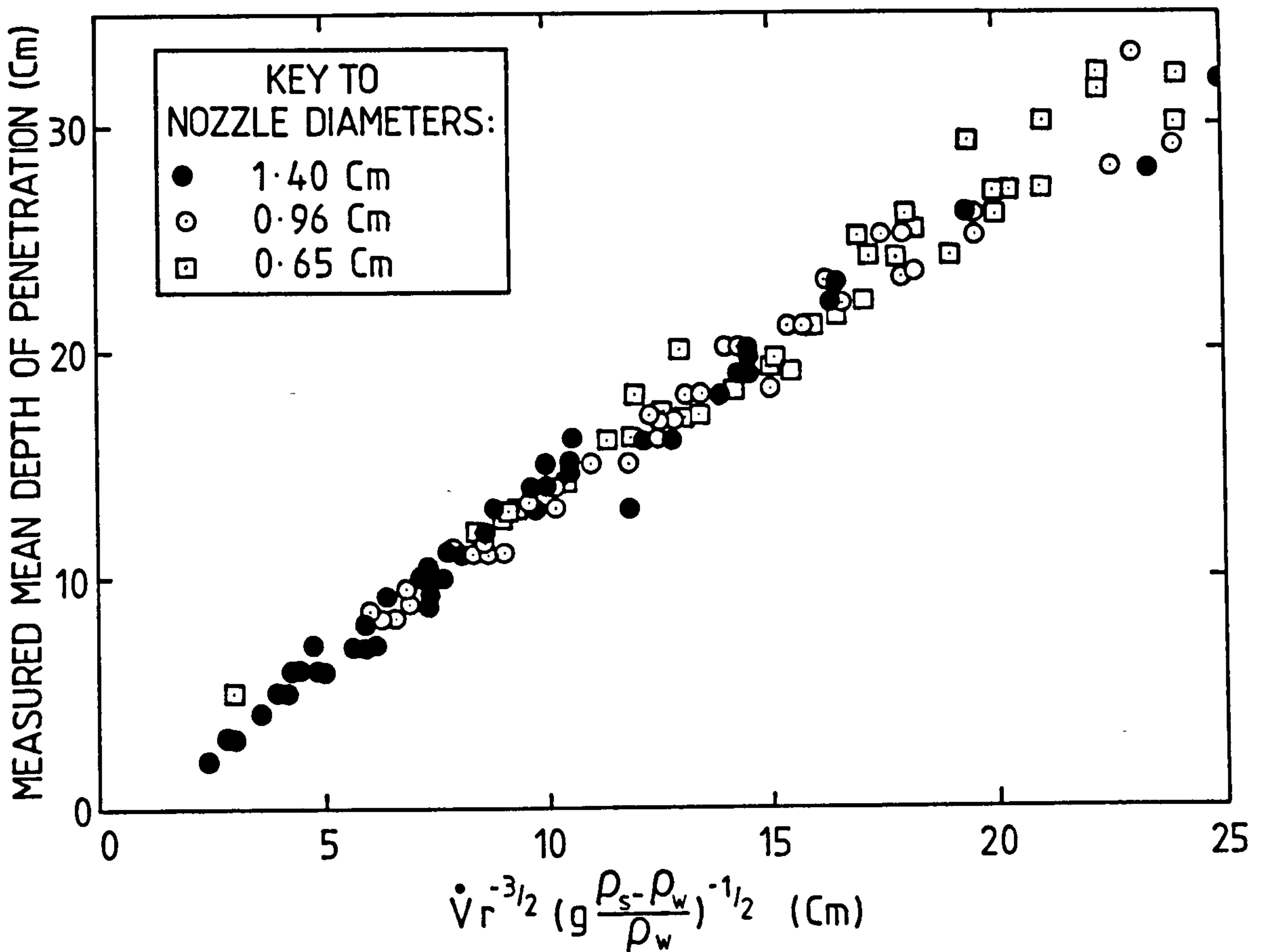


FIGURE 2.1 MEASURED MEAN DEPTH OF PENETRATION OF SALT JETS PROJECTED UPWARDS COMPARED WITH FORM SUGGESTED BY DIMENSIONAL ARGUMENTS (FROM REF. 2.2).

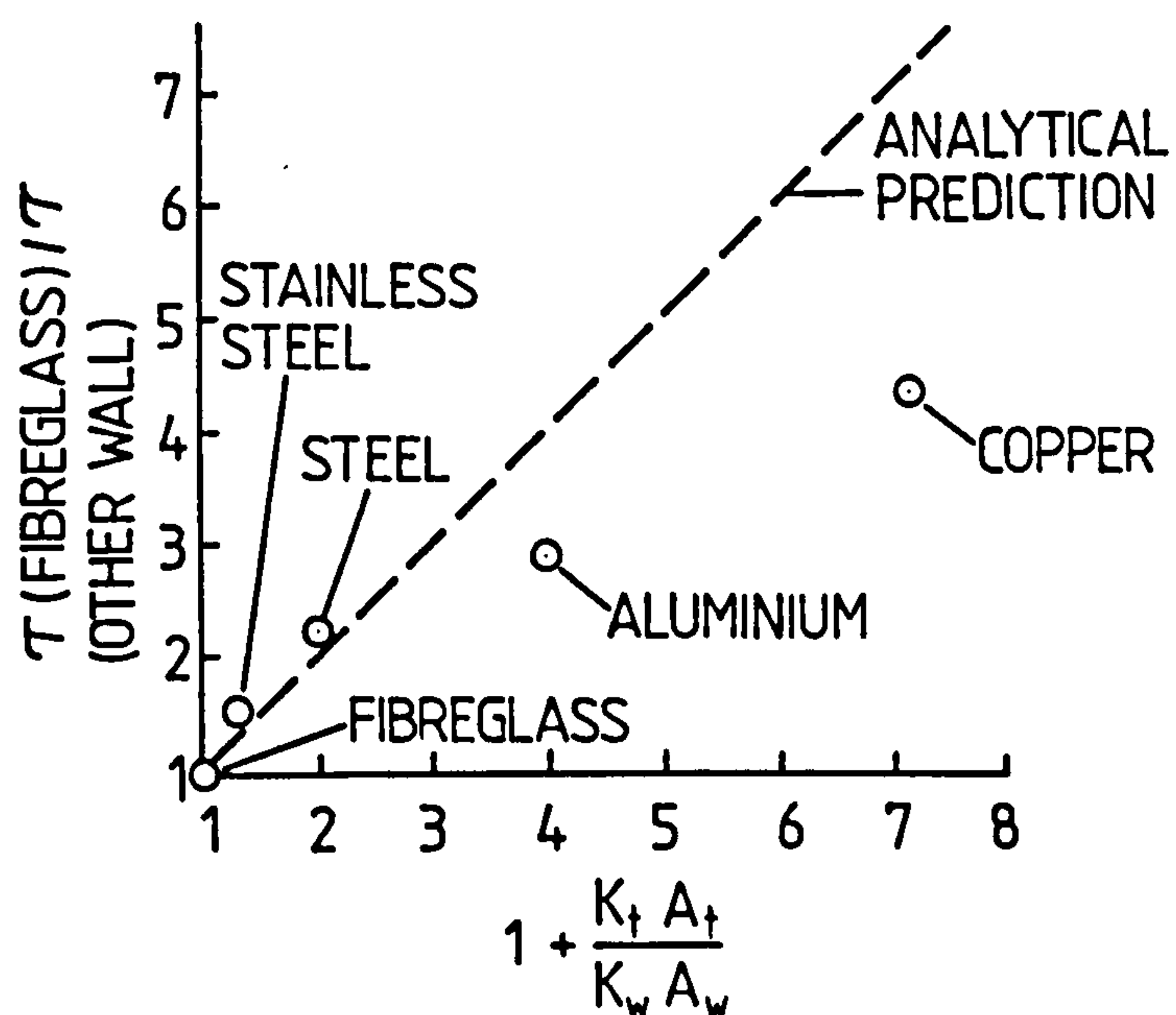


FIGURE 2.2 COMPARISON OF ANALYTICAL AND EXPERIMENTAL EQUALISATION TIMES (τ) AS A FUNCTION OF THE WALL (t) AND WATER (w) CONDUCTANCES (FROM REF. 2.5, SEE ALSO SECTION 5.2.1).

vessel walls. In 1964, Schwind and Vliet [Ref.2.3] published detailed results of experiments on small scale tanks of water and air with heated side walls. Ten years later hot water storage applications generated their own research into this phenomenon; in 1977 Miller [Ref.2.4] published results for aluminium and glass cylinders and in 1979 Sherman [Ref.2.5] gave results for copper, aluminium, steel, stainless steel and fibreglass cylinders, which indicated that a simple linear approximation of the effect of the wall can be used in cases where the wall conductance is at most 3 times that of the liquid (see figure 2.2).

Miller's theoretical treatment of wall conduction, which assumed that there was no heat loss to the surroundings, (i.e. perfect insulation) predicted that the convection currents, generated at the fluid/wall interface, fill the entire tank. The same outcome appears to be generated by heat loss through the walls, as was shown in a numerical analysis by Viskanta and Hale [ref.2.6], which conversely neglected heat conduction in the walls.

The other 'slow' mechanism for destratifying a quiescent hot water store, heat conduction from hot to cold layers, has also attracted attention. Clearly the smaller the horizontal cross-sectional area of the vessel relative to the total storage volume, the less significant this vertical heat transfer becomes. Leyers et al [Ref.2.7] carried out experiments on a full-scale (42 m^3) store, which demonstrated that wall conduction enhances the heat transfer between water layers to a degree consistent with Sherman's formula. Experiments by Saluria and Gupta [Ref.2.8] on small scale tanks indicated that, as the vertical temperature gradient gradually decays, the temperature distribution in the water remains essentially one-dimensional.

2.2.3 Stratification during Charge and Discharge

As mentioned earlier the dominant influence on stratification occurs during the charge and discharge of a store, and this fact is reflected in the quantity of literature on the subject. A dilemma recognised early on by researchers was that there are many different applications for hot water stores and that therefore there are no

general rules about the rates at which a store should be charged or discharged. It was therefore difficult to decide whether to experiment on hot water stores using realistic cycles (for example, typical solar collector output and hot water demand data) or to subject the stores to rigid but artificial test sequences. The former has the great advantage that the store is tested under conditions it would meet in practice; however, the results of the tests are difficult to analyse or generalise, comparing the performance of different storage devices is complicated and there is the by no means small problem of specifying a typical realistic cycle. The latter overcomes all these disadvantages but is unsatisfactory because there is no guarantee that the best store in an artificial test will retain its superiority in the field. Not surprisingly therefore, there has been considerable work on each of these approaches, and even with this experience, the latest consensus on a general test procedure for heat storage devices recommends that both approaches should be employed [Ref.2.9].

2.2.3.1 Step Input Testing

Early theoretical studies by Nakajima [ref.2.10] and Matsudaira and Sakakura [ref.2.11] were based on a step input test in which the water entering the store undergoes a step change in temperature. If the store is described by linear equations, the theory shows that once the response to a step input is determined, the response of the outlet temperature to an arbitrary inlet temperature can be predicted. The work thereby hinted that realistic cycle testing might not be necessary. However the theory assumed that a water store could be described by one-dimensional and linear equations when in fact it is three-dimensional and not completely linear.

The theory was nevertheless a persuasive argument in favour of specifying a step input procedure for an artificial test as opposed to, for example, a constant heat flux between the outlet and inlet or a sinusoidal inlet temperature. In 1976, the U.S. National Bureau of Standards (N.B.S.) [Ref.2.12] proposed a standard test procedure for thermal storage devices centred around step input tests, later published as a paper [Ref.2.13]. They cited three main reasons for selecting this method :

- (i) It permits the determination of an effective storage capacity and thus allows an easy comparison of different types of storage units.
- (ii) It appears to be the most fundamental approach since linear theory shows that the outlet temperature response to a constant or variable heat flux can be predicted if one knows how the outlet temperature changes with a step change in inlet temperature.
- (iii) The relative performance of stores using this method is likely to be the same if other procedures were used.

The test method proposed by N.B.S. was soon adapted into an ASHRAE Standard [Ref.2.14], and a little later N.B.S. [Ref.2.15] released a paper describing the testing of a 1.9m^3 cylindrical hot water store according to the Standard [ref.2.15]. This study concluded that the procedure was satisfactory, but proposed a dimensionless format for the results which generated a meaningful overall performance coefficient. However, it also hinted that a large variation in rating of water tanks according to this coefficient did not necessarily mean a large variation in performance of a system of which the tanks were part. Since then, the Standard has received more specific criticism from Marshall [Ref.2.16] concerning the heat loss expression involved in the analysis of charge tests. He states confusingly that the expression is fallacious beyond the recommended test period (for which it was not intended anyway), but correctly asserts that a similar expression should be incorporated into the analysis of discharge tests.

At the same time as the ASHRAE Standard was being developed, Lavan and Thompson [Ref.2.17] were already conducting fundamental experimental studies of stratification in hot water stores using a step input procedure. Their pioneering work, which has since been frequently and reverentially quoted, assessed the effect on stratification, during discharge and with horizontal ports, of tank length to diameter ratio, inlet-initial temperature difference, mass flow rate, inlet port location and inlet port diameter. Their experimental results (see figure 2.3) were given in terms of an extraction efficiency defined as the percentage of the volume of

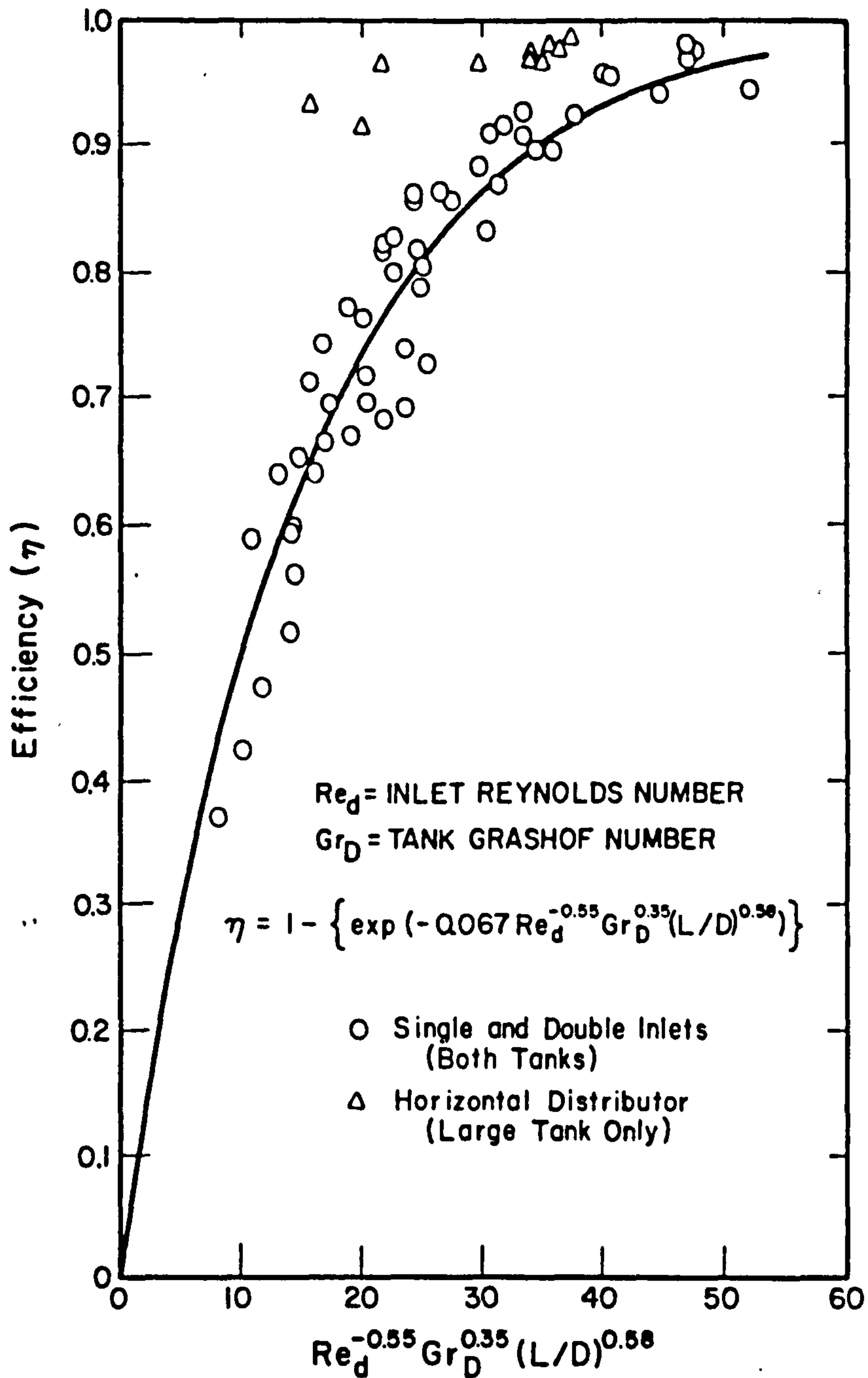


FIGURE 2.3 STEP INPUT EXTRACTION EFFICIENCY CORRELATED STATISTICALLY AGAINST DIMENSIONLESS PARAMETERS (FROM REF. 2.17).

water in the tank discharged before the inlet-exit temperature difference fell to 90% of its initial value. In certain conditions this parameter could be significantly different from the performance coefficient proposed in reference 2.15, although it can be closely related to Turner's jet penetration depth as is described in Chapter 4 (Section 4.2.3.2). They were able to correlate all their results in terms of three dimensionless parameters : the length-to-diameter ratio, the inlet Reynolds number and the tank Grashof number (based on the tank diameter). The studies were performed on two unpressurised (open-topped) vertical plexiglass cylinders with volumes of about 4 and 50 litres. Clearly the correlating parameters were selected to enable full scale designs to be assessed, although, given the dependence on actual size of vertical conduction and tank heat losses, this might be optimistic when storage devices two or three orders of magnitude larger are being considered.

Another detailed experimental study of stratification, this time on a full scale (300 litres) unpressurised cylindrical tank, was carried out by Sliwinski, and reported in two papers, published in 1978 [ref.2.18] and 1980 [ref.2.19]. His experimental procedure was again based on a step input test and horizontal ports were used. Both charge and discharge experiments were conducted for a range of inlet-initial temperature differences and mass flow rates and for two sizes of inlet port diameter. The novel aspect of the work consisted of determining the degree of stratification by measuring the vertical temperature profiles along the centre-line of the tank during an experiment. A depth was observed at which a nearly constant temperature gradient first occurred. The depth varied with the experimental conditions and was found to correlate well with a dimensionless parameter (a modified Richardson number) incorporating all the experimental variables (see figure 2.4). The depth is similar to Turner's penetration depth, and the results are assessed in more detail in Chapter 4 (section 4.2.3.2).

Preliminary work on stratification was also being conducted at Cranfield [Ref.2.20] at this time, in the form of experiments on a small scale (100 litre) cubical tank. Charge and discharge experi

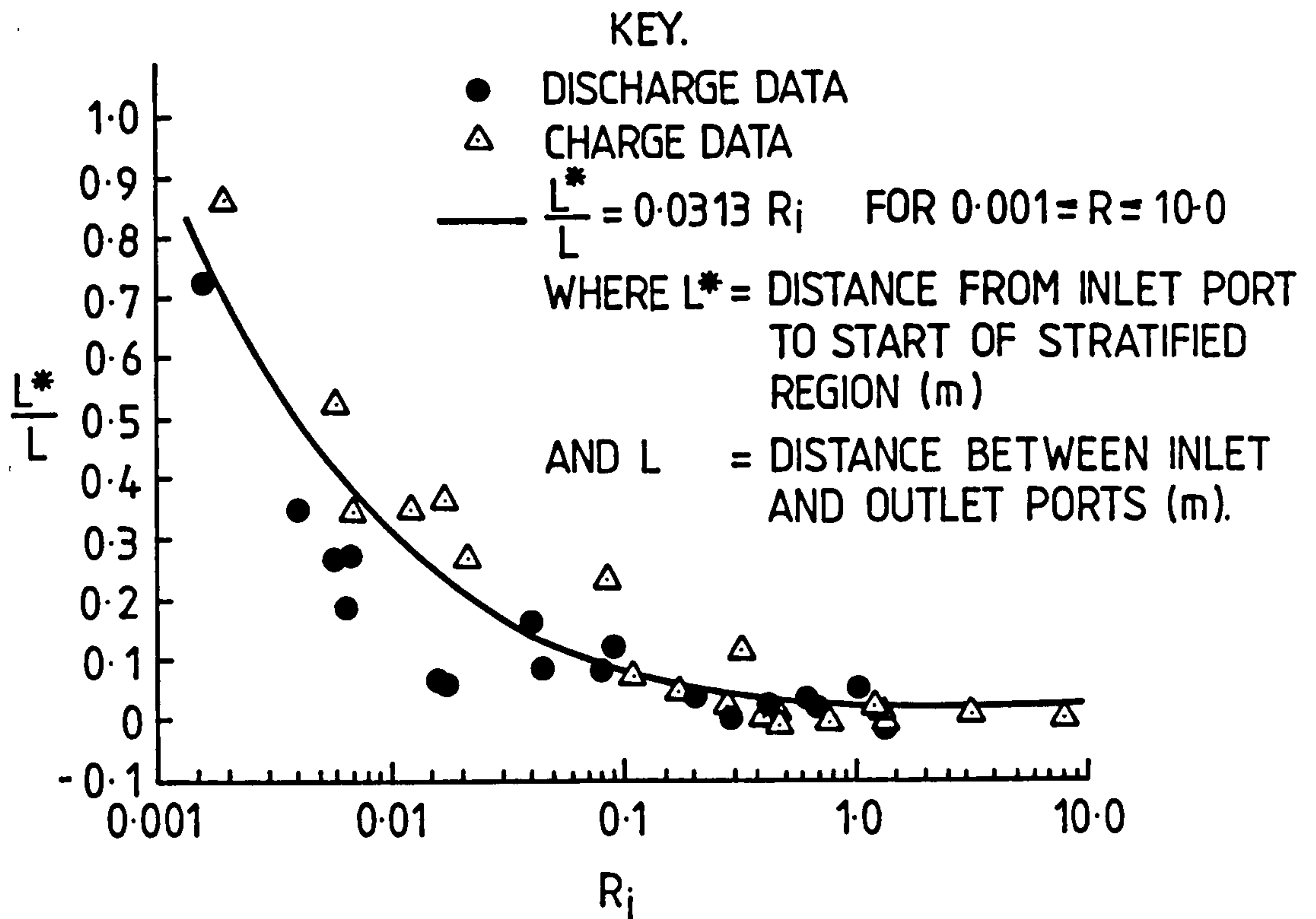


FIGURE 2.4 DIMENSIONLESS DISTANCE TO START OF STRATIFIED REGION CORRELATED AGAINST RICHARDSON NUMBER (R_i) (FROM REF. 2.19).

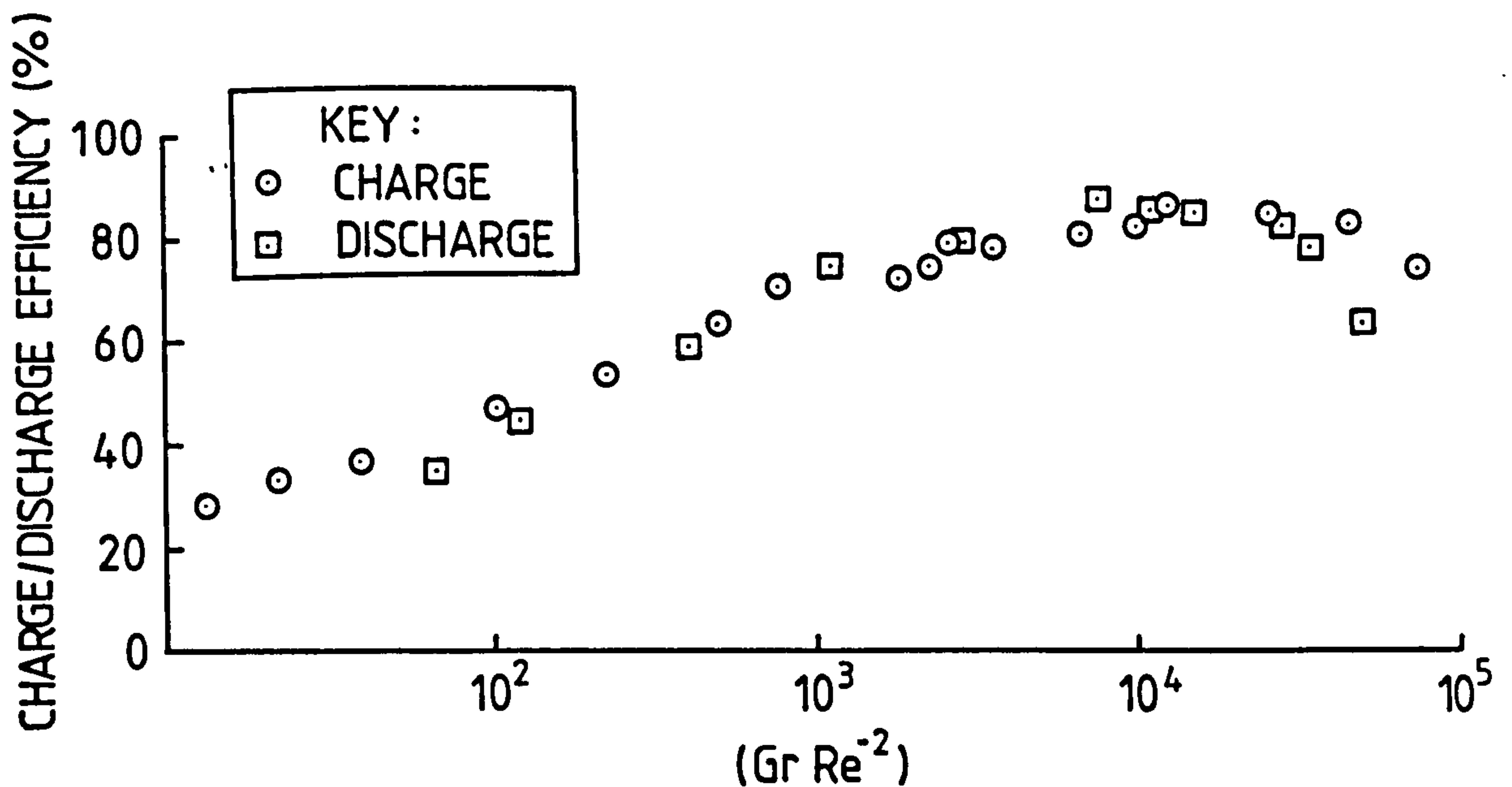


FIGURE 2.5 STEP INPUT EFFICIENCY PLOTTED AGAINST ARCHIMEDES NUMBER ($Gr Re^{-2}$) (FROM REF. 2.20).

ments using vertical ports were carried out, for a wide range of inlet-initial temperature differences and mass flow rates, according to the ASHRAE Standard procedure. The performance coefficient proposed in reference 2.15 was calculated for each experiment and correlated against a dimensionless parameter referred to as the Archimedes Number and defined as $Gr Re^{-2}$, where Re is the inlet Reynolds number and Gr is the Grashof number based on the water depth (i.e. the distance between the inlet and outlet ports). This parameter resembles Sliwinski's modified Richardson number, but in contrast to Turner's approach, both appear to be flawed in the light of dimensional analysis, and inappropriate for the purpose of scaling results (see section 4.2.1.2).

These early Cranfield results (shown in figure 2.5) were however interesting in these respects : (i) they demonstrated the equivalence of the charge and discharge situations, (ii) relatively low efficiencies were obtained owing to inevitable deadspace in a cubical tank, (iii) calculated efficiencies decreased at low inlet velocities because of errors in allowing for heat losses during very long tests and (iv) at high inlet velocities short-circuiting between the inlet and outlet ports was very much in evidence.

2.2.3.2 'Real' Cycle Testing

Real cycle testing has been carried out under two general circumstances : (i) full-scale field trials where the aim was to gain insight into the effects of stratification on system performance, or (ii) laboratory based experiments where the primary purpose was the validation of computer models of stores or the development of components which enhanced stratification.

Davis and Bartera [Ref.2.21] in 1975 were one of the first to publish results on stratification from a field trial. They estimated it improved the performance of their solar water heating system by about 10%. They also reported that some baffle plates incorporated into their store to promote stratification had no perceivable effect. In contrast, Van Koppen [Ref.2.22], discussing a solar space heating system at the Solar House of Eindhoven University, claimed excellent performance from a floating inlet device (a thin-walled, flexible plastic hose) for charging the store and a multiport control valve

for discharge. The enhanced stratification led to an increase in the output of the solar system of at least 15%.

An alternative means for introducing liquid of arbitrary temperature into a tank without destroying pre-existing stratification was developed over a number of years at Colorado State University [refs. 2.23 - 2.26]. It comprised a vertical, rigid porous distribution manifold, located up the whole height of the store. The incoming water enters a chamber at the top, and, if its density is lower than the surrounding water, flows downwards. The porous section inhibits turbulent mixing between the manifold and tank fluids until the level is reached at which the densities of the two fluids match. One advantage claimed for the vertical manifold over the floating inlet is that it is self-purging of entrapped air, which can seriously degrade the performance of a flexible-hose distributor. The manifold was tested in a full scale (2270 litre) tank under extreme variable inlet temperature conditions : the tank was initially half charged with hot water, then the cold water from the bottom of the tank was recycled directly into the inlet manifold. The results indicated that most of the stratification was preserved.

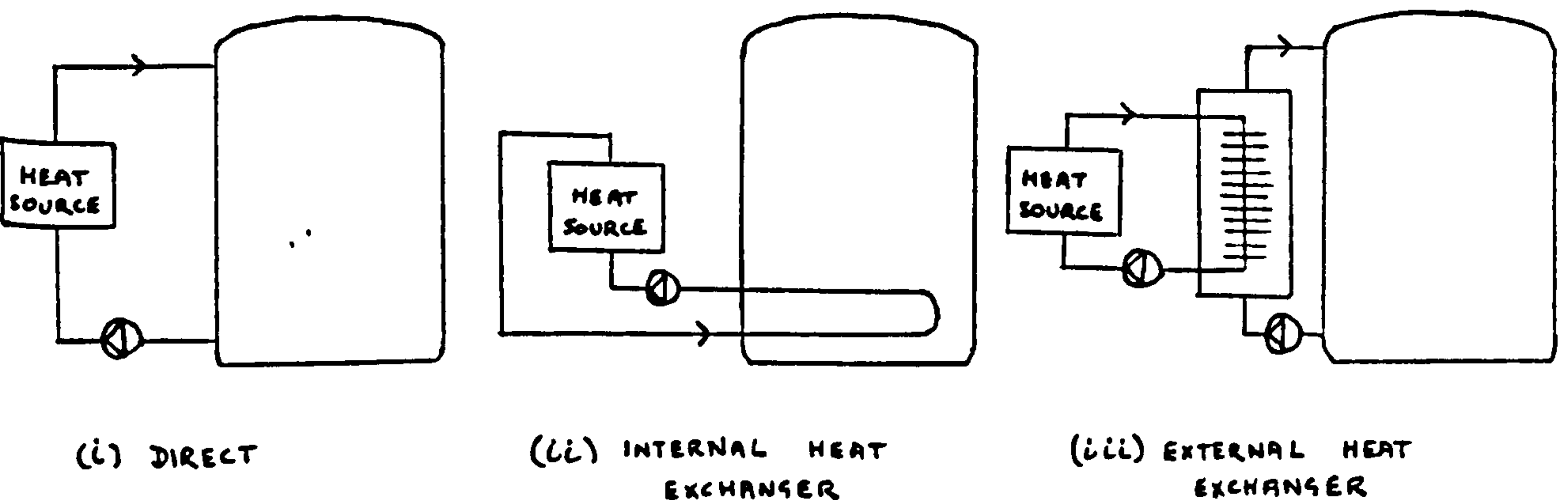
Experimental tests to validate a computer model were conducted by Wu [refs. 2.27 - 2.28] on a large scale (6800 litre) storage tank which was part of a solar heating system. The tests showed that this particular design of tank produced far from ideal stratification. However, the temperature distribution was effectively one-dimensional and could be closely predicted by a one-dimensional computer model which incorporated a "viscous entrainment" term, i.e. a term which artificially increased the rate of heat conduction between layers to cater for turbulent mixing during charge and discharge. Although Wu specifies this term as a constant, this is clearly not the case as was demonstrated by Turner's results in figure 2.1. Wu concluded that better stratification (and thereby improved system performance) could be achieved by a more innovative storage tank design, but that consequently it might be necessary to use more complex store models (e.g. three-dimensional) to assess these designs both as individual components and in system simulation studies.

An analytical model for predicting the effect of stratification on

overall system performance was proposed by Phillips and Dave [ref. 2.29]. It was validated against both a detailed one-dimensional numerical solution and also experimental data obtained with solar collectors. Another approach to model validation using real cycle tests is to replace all parts of the system except the store by computer controlled simulators. This permits exactly the same 'real' cycles to be performed on different store designs or alternatively the same store to be subjected to slightly different cycles, for example to assess the influence of system flow rate on store stratification. Bowman et al [Ref.2.30] reported preliminary results using this technique.

2.2.4 Stratification with Heat Exchangers

There are three basic ways of exchanging heat between a water store and its heat source or heat demand : (i) directly, as discussed so far, (ii) via internal heat exchangers and (iii) via external heat exchangers. The techniques are illustrated below for the case of charging a store :



Direct systems are simplest, promote better stratification and avoid the inevitable temperature loss across heat exchangers. However, there are two circumstances where a heat exchanger may be preferred : (i) where protection against freezing requires that the water flowing through the heat source includes about 20% by volume antifreeze. In direct systems this would mean 20% of the store volume must be antifreeze which, at about £1/litre, can add significantly to the

system cost, especially for larger stores. (ii) where the heat in the store is to generate domestic hot water, which must obviously not be contaminated by antifreeze, corrosion inhibitors, etc., in other parts of the system.

External heat exchangers (e.g. shell and tube) are far more expensive than internal ones and require an extra pump, which adds to capital costs and parasitic energy losses. However internal heat exchangers rely on heat transfer by natural convection and can impair stratification, both of which can reduce overall system performance. A recent experimental study at Cranfield [ref.2.31] has determined performance coefficients for five commercially available heat exchange components when immersed in a hot water store. The influence of the mass flow rate (i.e. Reynolds number) of the water inside the heat exchanger was assessed. All the experiments were conducted with the same inlet temperature to the heat exchanger and the same initial temperature inside the tank. A summary of the results is shown in Figure 2.6 and an economic comparison of the five designs is given in Figure 2.7. A complementary study was performed by Hughes and Thring [Ref.2.32] on one of the same components ('Integron' tube), only in this case they assessed the influence of the temperature difference (i.e. Grashof number) between the inlet water and tank water (see Figure 2.8).

2.3 REVIEW OF THEORETICAL WORK AND COMPUTER MODELLING

2.3.1 Introduction

The ultimate aims of theoretical studies are generally (a) to discover or assess better designs of components and (b) to assess the effect of changes in component design on system performance. Theory consists of describing components or systems by sets of self-consistent, scientifically-sound equations, which nevertheless are based on assumptions to a greater or lesser degree. The method for solving these equations depends on their complexity, which in general lessens as the assumptions become more sweeping. If they are simple enough, analytical solutions are possible, otherwise numerical solutions, based on, for example, finite difference approximations to partial differential equations, must be employed. Computer models are usually just an electronic means of carrying out numerical solutions. The

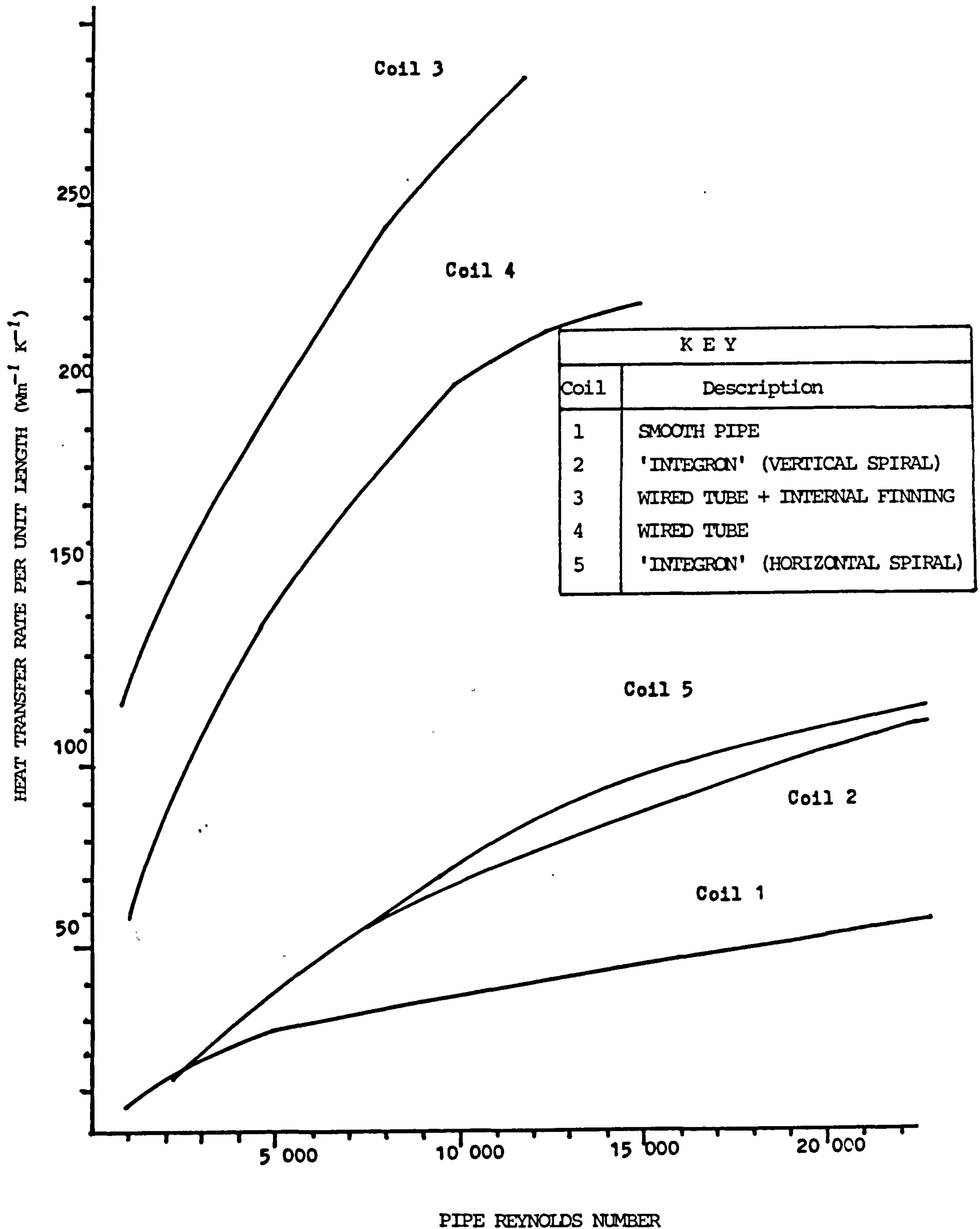


FIGURE 2.6 THE HEAT TRANSFER PERFORMANCE OF VARIOUS COILS IMMERSSED IN WATER AS A FUNCTION OF REYNOLDS NUMBER (FROM REF. 2.31).

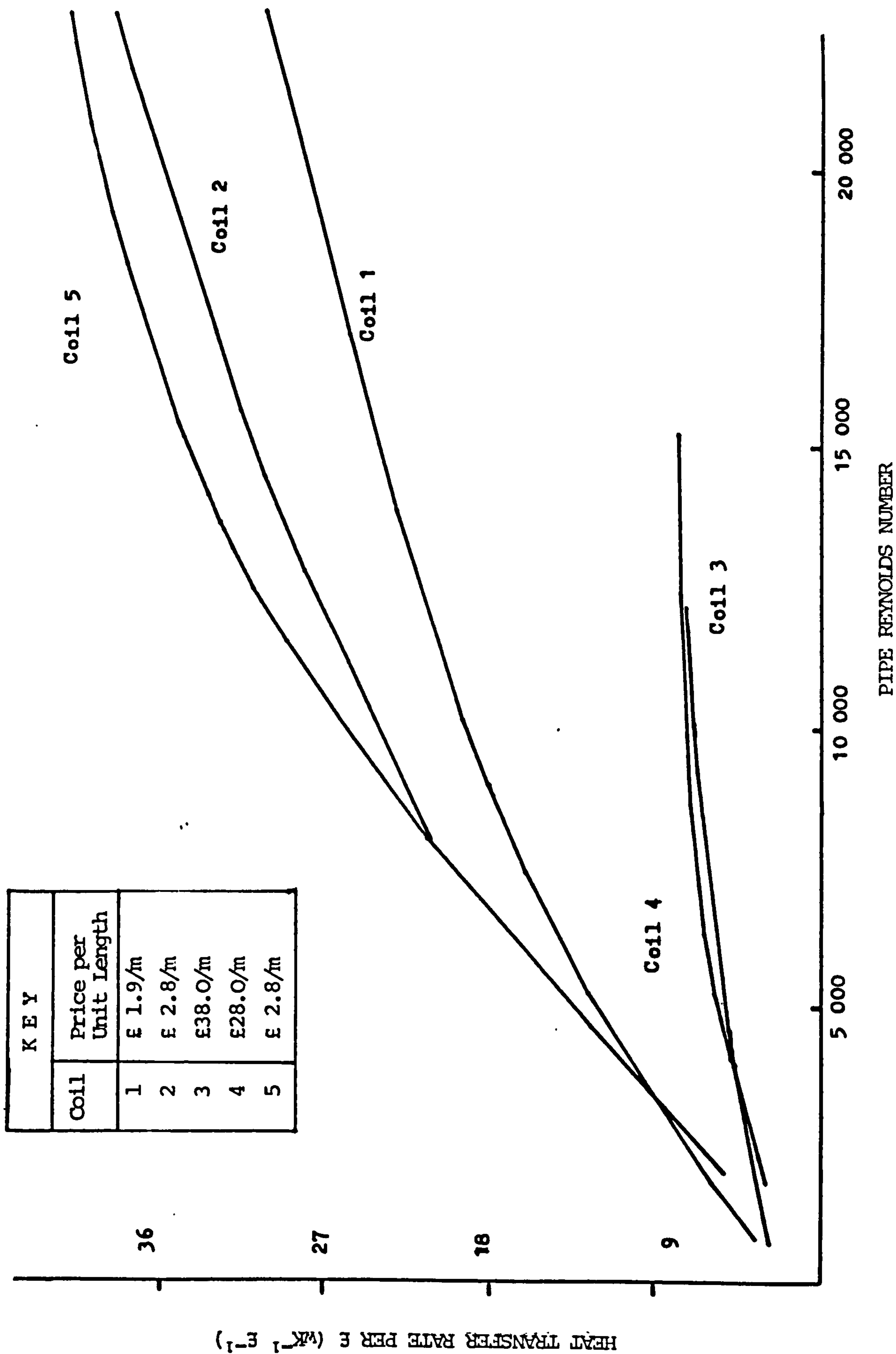


FIGURE 2.7 THE HEAT TRANSFER PERFORMANCE PER E OF THE COILS IN FIGURE 2.6 AS A FUNCTION OF REYNOLDS NUMBER
(FROM REF. 2.31).

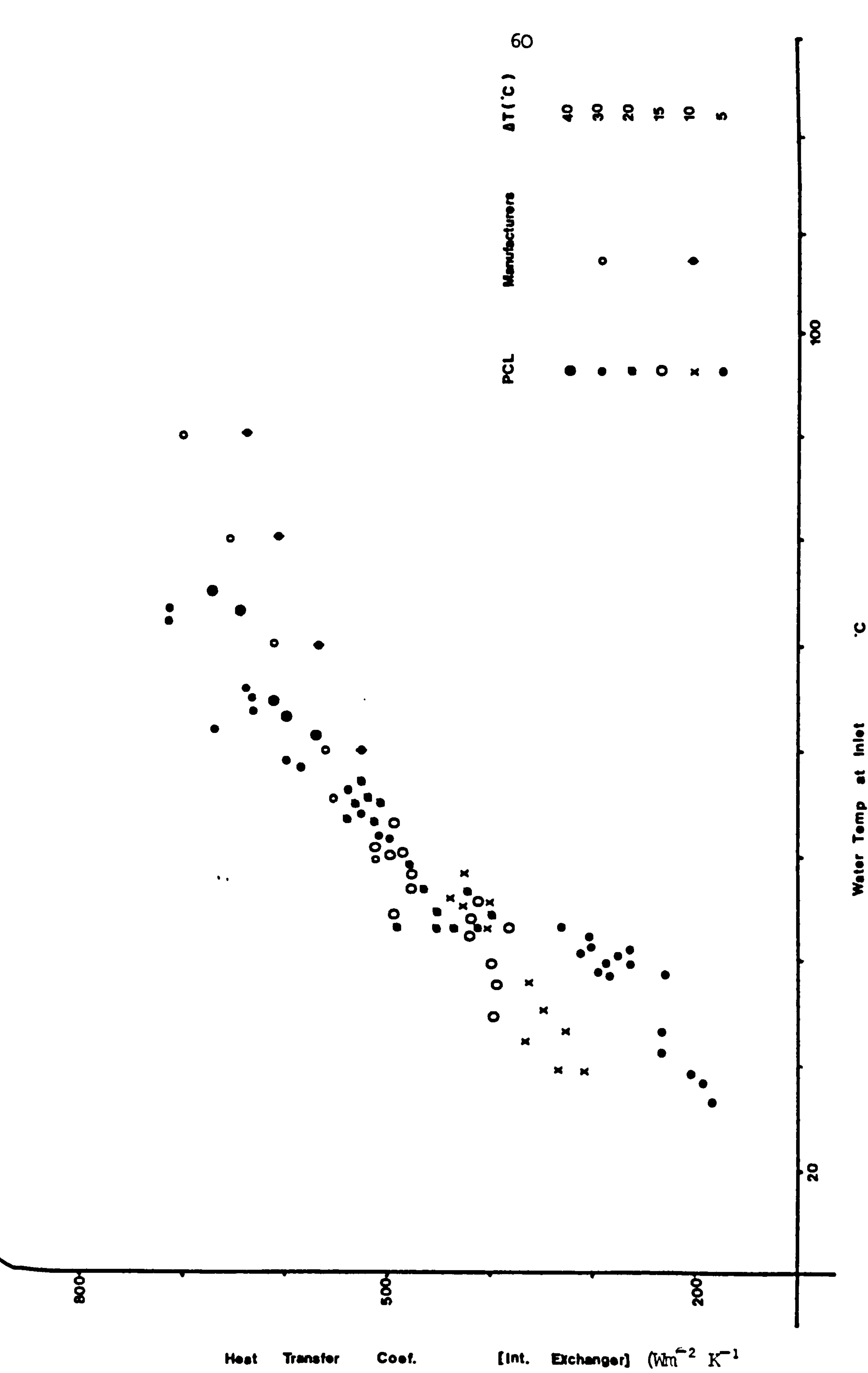


FIGURE 2.8. Variation in heat transfer coefficient with inlet temperature to internal heat exchanger (FROM REF. 2.32).

tremendous computing power available to most researchers now, compared even to just ten years ago, has made numerical solutions increasingly viable and popular. However computer time and costs still impose limits on what can be done, especially in the simulation of multi-component systems over long periods (e.g. years).

Validation of the predictions of analytical or numerical solutions must in the end come from experiment. However this can entail models of systems being validated by experiments on just the individual components which make up the system; or simplified models being validated by complex models of the same components or systems, as long as the complex model has itself been validated at some time by experiment.

2.3.2 Three and Two-dimensional Models

A theoretical treatment of stratified hot water storage is difficult because (a) the thermodynamic and hydraulic processes which influence the distribution of thermal energy within the stored water are complex in nature and (b) the possible storage tank configurations, including the shape of the tank and the locations and forms of the inlets and outlets, are very diverse. The time-dependent, three-dimensional conservation equations of mass, momentum and energy governing the fluid-flow and heat transfer processes in a cylindrical tank have been solved numerically using a computer model christened COMMIX-SA [Ref.2.33]. The model has been used to assess the effects on stratification of baffle plates, distributors, tank height to diameter ratio and inlet flow rate [Ref.2.34]. Although the predictions of the model appear to be accurate, no experimental validation has been presented. One possible flaw in the model is its neglect of turbulent mixing effects.

COMMIX-SA is an invaluable tool for assessing new storage tank designs but is far too complex to be used as part of a system simulation model. A numerical solution of a two-dimensional model given by Cabelli [Ref.2.35] suffers from the same failing and is probably not even reliable as a design tool. Cabelli did however make use of his model by comparing its predictions with those of an analytical solution of a one-dimensional formulation of the temperature distribution in the storage tank. The discrepancies

between the two models were less than 15% of the maximum range of temperature, when they were applied to a step input situation. Variable inlet temperatures were not considered but it can be assumed that they would have caused much greater disparities.

2.3.3 One-dimensional Models

It is generally recognised that only one-dimensional models of stratified hot water stores are acceptable in terms of computer time for the purposes of system simulations. In such models, heat transfer within the liquid is considered only in the vertical direction. As mentioned in Section 2.2, there is much experimental evidence to justify this extremely helpful simplification.

There are four basic heat or mass transfer processes which take place in a hot water store: (i) flow during charge and discharge, (ii) turbulent mixing, (iii) heat conduction between layers and (iv) heat loss to the environment. The complexity and validity of one-dimensional models depend on how carefully these are treated. Matsudaira and Tanaka [Ref.2.36] took into account only the first of the above factors, and assumed the store to be composed of a fully mixed region and a piston flow region. These fairly drastic assumptions allowed them to use a very simple analytical solution which in fact gave reasonable agreement with experimentally measured values. However in many circumstances, such as moderately insulated stores or diurnal stores, satisfactory agreement would not occur. Cabelli's one-dimensional model [Ref.2.35] included all the above factors except turbulent mixing, but the analytical solution was too complex for inclusion in system simulations.

The first authors to describe a numerical solution to a one-dimensional model were Duffie and Beckman [Ref.2.37] in their famous text book on solar energy. The model neglected turbulent mixing and heat conduction, which were considered, justifiably, to be second order effects. The degree of stratification was nominally fixed by the choice of number of segments for the store, ranging from 1 for fully-mixed up to about 10 for good stratification. The solution is flawed by not specifying the time step, Δt , for the finite difference approximation. The key period in such models is the fill-time of one

segment, τ , Only if $\Delta t \sim \tau$ does the number of segments truly determine the degree of stratification. If $\Delta t < \tau$ the store response is considerably blurred. $\Delta t > \tau$ causes the solution to be unstable. These points were recognised by Marshall [Ref.2.38] in 1981. Refinements to the basic Duffie and Beckman model have been made, notably by Sharp [ref.2.39] who incorporated heat conduction between layers and Wu [ref.2.27] who added a form of turbulent mixing.

The Duffie and Beckman model was also characterised by the inlet flows to the tank being introduced at the location where the difference between the tank temperature and the incoming water temperature was a minimum. The degree to which the numerical model approximated this ideal (floating-inlet) case improved as the number of segments increased. Sharp and Wu both assumed fixed inlet locations in their models and dealt with unstable thermoclines, i.e. cold water on top of hot water or vice versa, by downward or upward mixing respectively. Comprehensive validation of Sharp's model was provided by Kuhn et al [Ref.2.40] based on both step input and variable inlet temperature experiments. They showed that there was good agreement between model and experiment if diffusers were employed on the inlets (in which case a 20 segment tank was most appropriate). For inlets without diffusers only an approximate matching was obtained (using a 5 segment tank).

Another interesting variation of the Duffie and Beckman model was developed by Wuestling [Ref.2.41] with the purpose of simulating a store which achieved the maximum possible stratification. The idea was to compare system performance using this store and a fully mixed store, and thus to define absolute upper and lower limits for the performance. The principle of the model was to use a variable number of variable size segments of water. Its operation is illustrated in figure 2.9 for both charge and discharge situations. The model had facility for variable position (floating) inlets and, naturally, neglected turbulent mixing and heat conduction. In effect the model would predict the same response as a standard Duffie and Beckman model which set $\Delta t = \tau$, but gained by having no restriction on the time step.

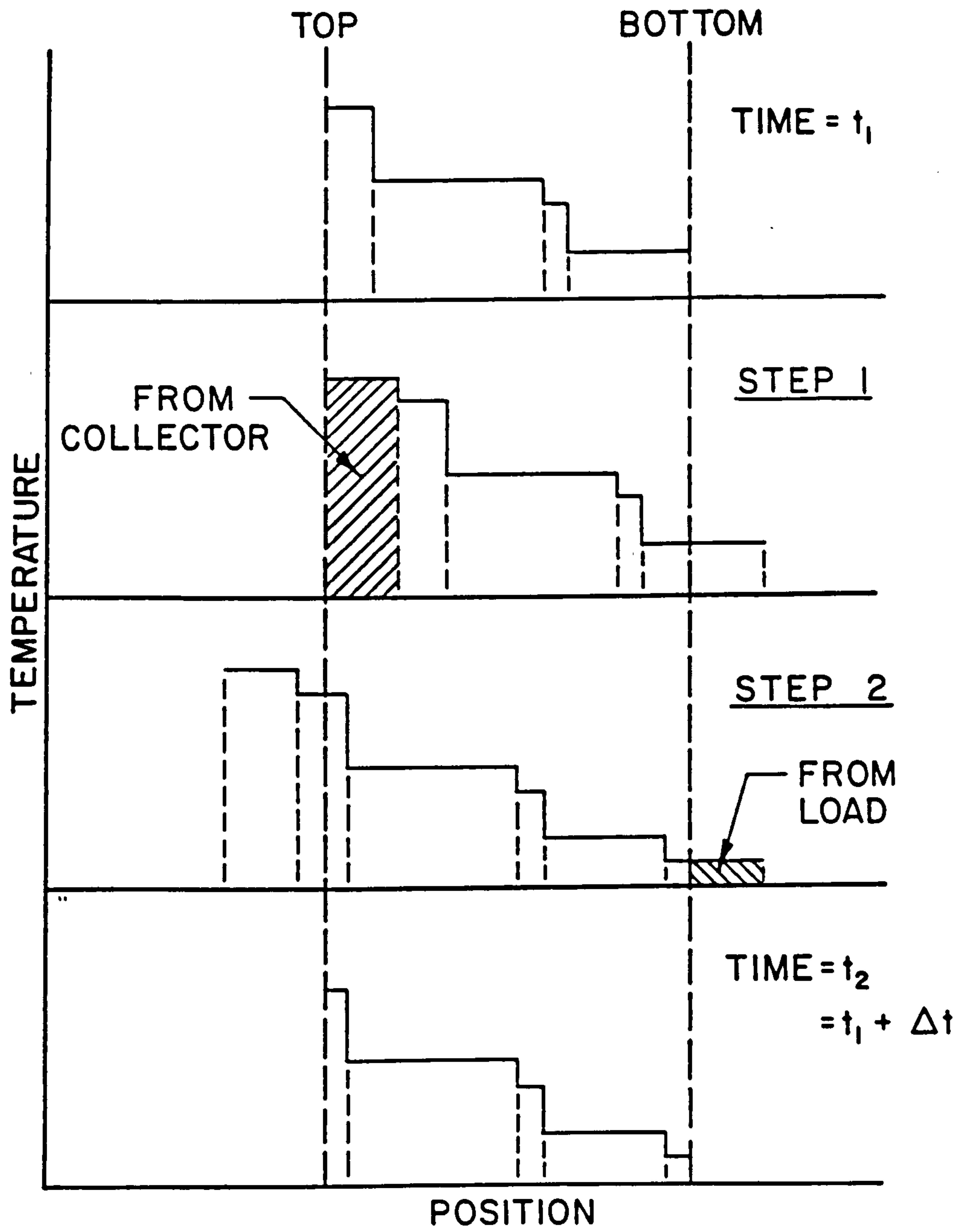


FIGURE 2.9 SCHEMATIC DESCRIPTION OF THE OPERATION OF THE PLUG-FLOW STORAGE TANK MODEL IN REF. 2.41.

2.3.4 Model Predictions of the Benefits of Stratification

A large number of computer-based studies have been undertaken to predict the potential benefit of stratification on system performance. Because of the necessity, yet inherent fallibility, of using a one-dimensional model, great care must be exercised when interpreting the results. Some studies are based simply on comparing the two theoretical, extreme cases of fully-mixed and ideally-stratified storage systems; when they generate widely differing results one is left wondering how much of the potential gain is realisable in practice. Other studies are based on models, producing partially stratified storage, which are claimed to be realistic for certain circumstances.

Two examples of simulations of solar energy systems illustrate this : Sharp and Loehrke [Ref.2.42] used Sharp's relatively sophisticated model (Ref.2.39, discussed above) to predict a 5 to 15% improvement in solar fraction if a store were fitted with inlet diffusers. Whilst Veltkamp [ref.2.43] used a Duffie and Beckman model to predict a 10% improvement in solar fraction if a store had floating inlets, as long as collector flow rates were about three times lower than usual. One of the main purposes of this thesis is to provide a one-dimensional model which reasonably represents the measured stratification achieved by a variety of inlet designs subjected to a range of inlet flow rates and temperature differences between the inlet water and the tank water.

CHAPTER THREE

THERMAL ENERGY STORAGE TEST FACILITY

3.1 INTRODUCTION

A test facility was constructed in order to perform temperature step input experiments on a full scale (3m^3) hot water storage vessel. The experiments were carried out over a wide range of flowrates into the store and for a variety of inlet temperatures and temperature differences between the inlet water and the vessel water. A microprocessor-based system was employed to monitor and control the test facility and for data acquisition. This chapter describes the design of the test facility, the precision it achieved in the step input experiments and the design of the hot water storage vessel.

3.2 DESIGN OF THE TEST FACILITY

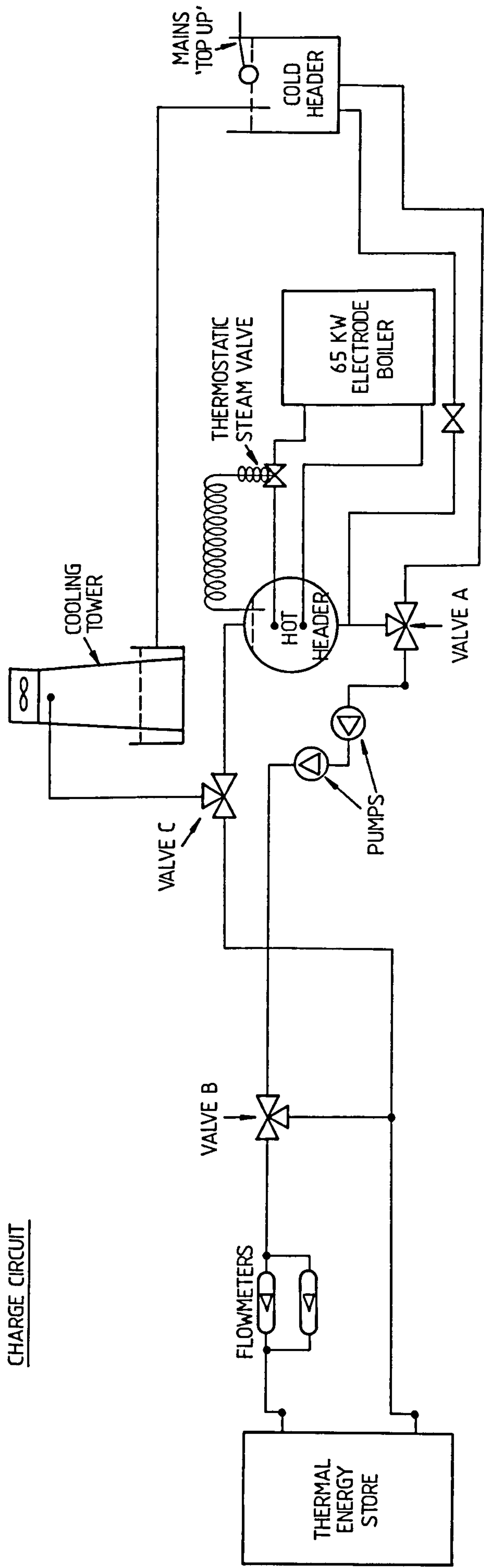
3.2.1 The Pipework

The pipework of the test facility is shown schematically in figure 3.1 and in a photograph in figure 3.2. Its main objective was to pass water into a storage vessel at a temperature and flowrate as constant as possible. For each experiment it also had to achieve a step change in the inlet water temperature which had to be as sharp as possible. These requirements were met by the use of 3-way motorised valves controlled by the microprocessor.

The test facility contained a 450 litre cold header tank, maintained at between 10 and 20°C, depending upon environmental conditions, and a 450 litre hot header tank which was heated by steam from a 65 kW electrode boiler and maintained at about 90°C. The steam flow was controlled by a thermostatic valve, operated by a highly expansive material contained in the thermostat tube which was immersed in the top of the hot header and connected to the valve by a capillary tube.

The hot and cold header tanks supplied two almost identical circuits of predominantly one inch pipework, one being for experiments involving a step rise in inlet temperature (i.e. charging the store) and the other for a step drop in inlet temperature (i.e. discharging

CHARGE CIRCUIT



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DISCHARGE CIRCUIT

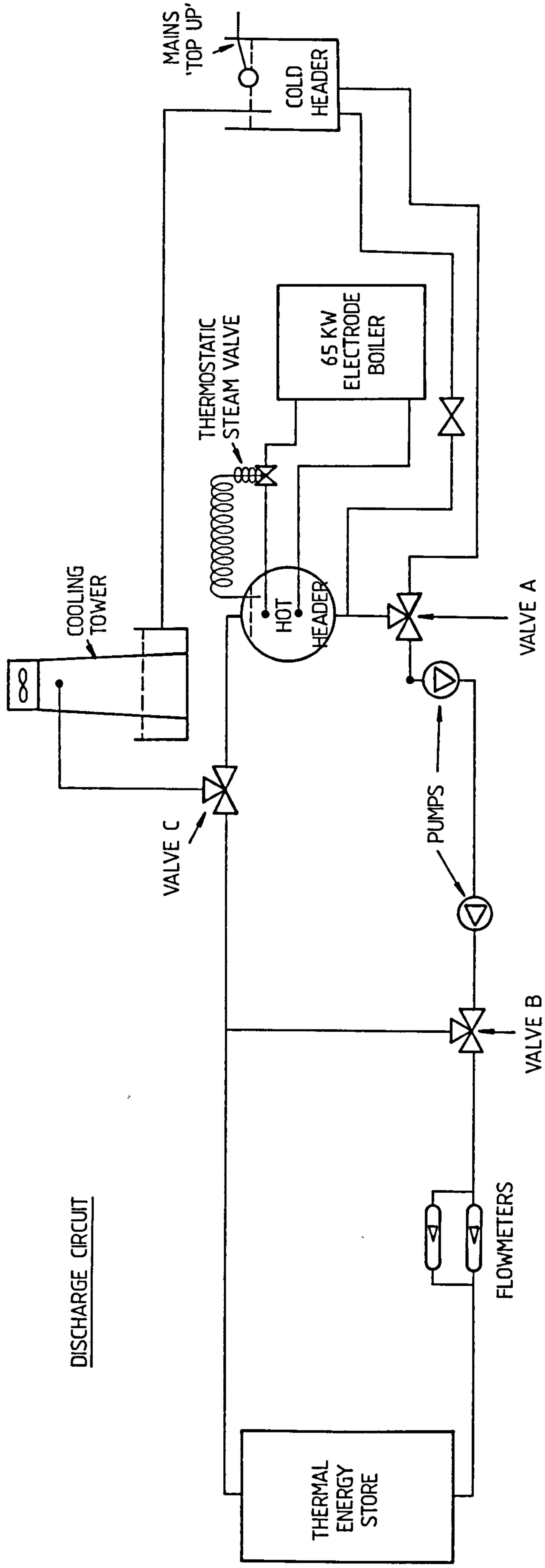


FIGURE 3-1 SIMPLIFIED DIAGRAM OF TEST FACILITY PIPEWORK.



FIGURE 3.2 VIEW OF THE THERMAL ENERGY STORAGE TEST FACILITY SHOWING THE INSTRUMENTATION (FOREGROUND) , THE STORE AND PART OF THE PIPEWORK (BACKGROUND)

the store). In each circuit, water supplies from the hot and cold header tanks were mixed to the required temperature at a 3-way motorised valve (A). A step change in water temperature was obtained by driving the valve in the appropriate direction.

The water exiting from the mixing valve at the correct temperature then passed into a 3-way motorised diverting valve (B) which diverted part of the flow in order to leave the correct flowrate for passing into the storage vessel. The water leaving the storage vessel was then recombined with the diverted portion of the original flow and then passed into another 3-way motorised diverting valve (C). This valve was electrically connected to the temperature control valve (A) so that it split the return flow so as to return the same proportions of hot and cold water as were initially removed back to the hot and cold header tanks. The portion of the flow destined to return to the cold header tank was first passed through a cooling tower so that it was cold when it entered the header tank. To prevent stratification in the hot header tank, which could have caused highly transient and therefore troublesome temperatures at the inlet to the temperature control valve (A), the return water, which was likely to be cooler than the water in the hot header tank, entered at the top of the tank. In addition an electrically driven mechanical stirrer was employed to keep the contents of the hot header tank fully mixed.

3.2.2 Instrumentation

It was essential to achieve a high degree of accuracy in the measurement of temperatures and flowrates in order to produce meaningful experimental results. Furthermore, the large number of experiments to be carried out, each requiring many sets of readings to be taken, dictated that an automatic data acquisition system be employed, incorporating magnetic storage of the data for subsequent processing by a computer. The accuracy of each reading had to be maintained therefore from the point of measurement, through a signal conditioning process and past an interface which converted the signal into a digital form which could be handled by computer software. The layout of the complete monitoring and control arrangement is shown in figure 3.3. The measurement of temperatures and flowrates is described in detail in the rest of this section.

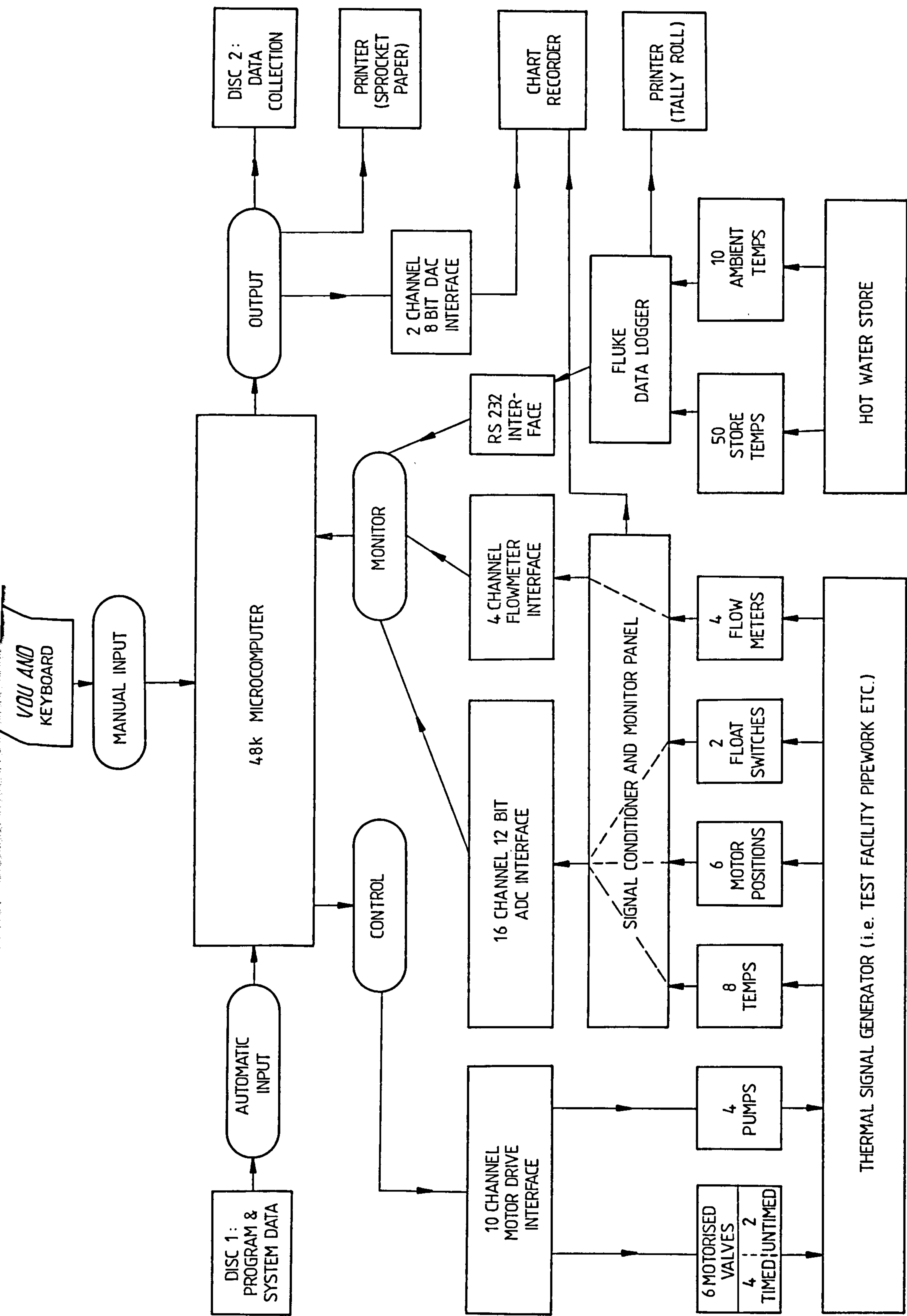


FIGURE 3.3 LOGIC FOR MONITORING AND CONTROL OF TEST FACILITY.

3.2.2.1 Temperature Measurement

(a) Flow temperatures

Flow temperatures (i.e. the temperatures of the water after the temperature controlling 3-way mixing valve and of the water entering and leaving the store) were measured by Platinum Resistance Thermometers (P.R.T.'s). Each P.R.T. was connected to a signal conditioner containing a transistor and zener diode which aimed to produce a constant current through the P.R.T. In this way the voltage across the P.R.T. was directly related to its resistance and therefore its temperature. This voltage, which was set to lie in the range $0 \rightarrow 10$ volts to correspond to temperatures in the range $0 \rightarrow 100^{\circ}\text{C}$, was directly output to an Analog to Digital Converter (A.D.C.). A 12 bit A.D.C. was employed so that the voltage was converted to a number between $0 \rightarrow 4095$, giving a resolution of 0.024°C per digit. This was more than sufficient given the accuracy of the P.R.T. itself at about $\pm 0.1^{\circ}\text{C}$. The number from the A.D.C. was read by the software program running in the microcomputer and converted directly back into a temperature.

Two factors affected the accuracy of this method of reading a P.R.T. : stability and linearity.

Stability

This determined the degree of drift in a reading caused by fluctuations in ambient temperature. The drift occurs because the base-emitter characteristic of the transistor in the signal conditioner is slightly dependent on ambient temperature. The diode acts to compensate for this; the transistor and diode are mounted on top of each other on the circuit board and are covered in silicone rubber to try to ensure they are at the same temperature. However a slight drift still occurs so that for these experiments a proprietary conditioner was incorporated into the measuring circuit to reduce the drift. It had a nominal stability rating of 0.03°C change in reading per 1°C change in ambient temperature. Therefore each temperature reading was accurate to $\pm 0.2^{\circ}\text{C}$ assuming that the maximum potential swing in

the laboratory temperature was about $\pm 7^{\circ}\text{C}$ (i.e. the laboratory temperature was always in the range $11 \rightarrow 25^{\circ}\text{C}$). N.B. More expensive and sophisticated "chips" are available to reduce this drift even further.

Linearity

The signal conditioner for each P.R.T. incorporated "zero" and "gain" adjusters which could be set to give the correct readings at the bottom and top of the scale (i.e. in this case at 0°C and 100°C). Between these limits one could assume that the resistance of the P.R.T. was linear with temperature. However, this is not quite the case and would create significant inaccuracies ($\pm 0.4^{\circ}\text{C}$). The conditioner incorporated to improve stability also enhances the linearity and reduces the maximum error to $\pm 0.2^{\circ}\text{C}$ over the 100°C range, but even this measure is not sufficient. Therefore, each measuring circuit was calibrated using standard resistances to imitate the P.R.T. at 10°C temperature intervals across its working range. The calibration data, given in Appendix C.1, was held as a table in the software program and, by means of interpolation, could be used to calibrate each temperature reading. In this way errors due to a linearity were diminished to a maximum of $\pm 0.02^{\circ}\text{C}$, over the complete working range of the P.R.T.

In conclusion, the accuracy of each P.R.T. reading was determined by stability errors and these were up to $\pm 0.2^{\circ}\text{C}$.

(b) Temperature difference measurement

One of the main parameters to be determined in these experiments was the temperature difference of the water between the inlet and outlet of the store. This value was required to calculate the energy being charged into or discharged from the store. A value derived by calculating the difference between the measured inlet and outlet temperatures would have an associated inaccuracy of $\pm 0.4^{\circ}\text{C}$ due to stability errors. Although this might be just about tolerable for large temperature differences (e.g. 20°C or more, so that the error is less than $\pm 2\%$), it becomes totally unacceptable for small temperature differences. For example, on 5°C it represents $\pm 8\%$ and on 2°C it

represents $\pm 20\%$, and such values of temperature difference were common, particularly towards the end of an experiment, as the store became nearly fully charged or discharged.

Therefore it was decided to link the inlet and outlet P.R.T.'s to a differential temperature conditioner ("chip"), so that the same current was passed through both sensors. By measuring the temperature difference directly the associated error was immediately halved. In fact, the "chip" employed (Ancom I5RP-5) had a stability rating of 0.01°C per $^{\circ}\text{C}$ change in ambient temperature, three times better than for the single P.R.T. conditioners. Thus the stability error in the differential temperature was only $\pm 0.07^{\circ}\text{C}$, representing $\pm 0.3\%$ at 25°C , $\pm 1.4\%$ at 5°C and only $\pm 3.5\%$ at 2°C .

The maximum range of the temperature difference was set at 0 to 25°C , corresponding to 0 to 10 volts, and, after the A.D.C., 0 to 4095. This gave a resolution of 0.006°C per digit, again representing a negligible error in comparison to the stability error.

(c) Store and ambient temperatures

The temperatures of the water inside the store and of the air surrounding the store were measured by copper/constantan thermocouples. These were connected via an isothermal input connector to a Fluke 2240B Data Logger which incorporates automatic reference junction compensation. The nominal accuracy of each temperature measurement by this method was $\pm 0.4^{\circ}\text{C}$. A special interface was fitted in the data logger to enable the temperature measurements to be read by the software program in the microcomputer. This transfer was accomplished in ASCII character format which was translated by the software, giving an identical reading in the computer as was output by the data logger.

3.2.2.2 Flowrate Measurement

The volumetric flowrate of the water entering (and therefore leaving) the store was measured by in-line turbine flowmeters.

Two different flowmeters were necessary in order to cover the range of flow rates used in the experiments, i.e. 0.036 to 0.7 litres/sec. Both meters were manufactured by Rhodes & Son and operated on the same principle but with different designs. For the lower flowrates, i.e. 0.03 to 0.2 litres/sec, their 1/4" bore model R.T. meter was used (specially modified to have 1/2" fittings instead of the usual 1/4" fittings, in order to reduce the pressure drop). For the higher flowrates, their 1" bore model P.T. meter was used.

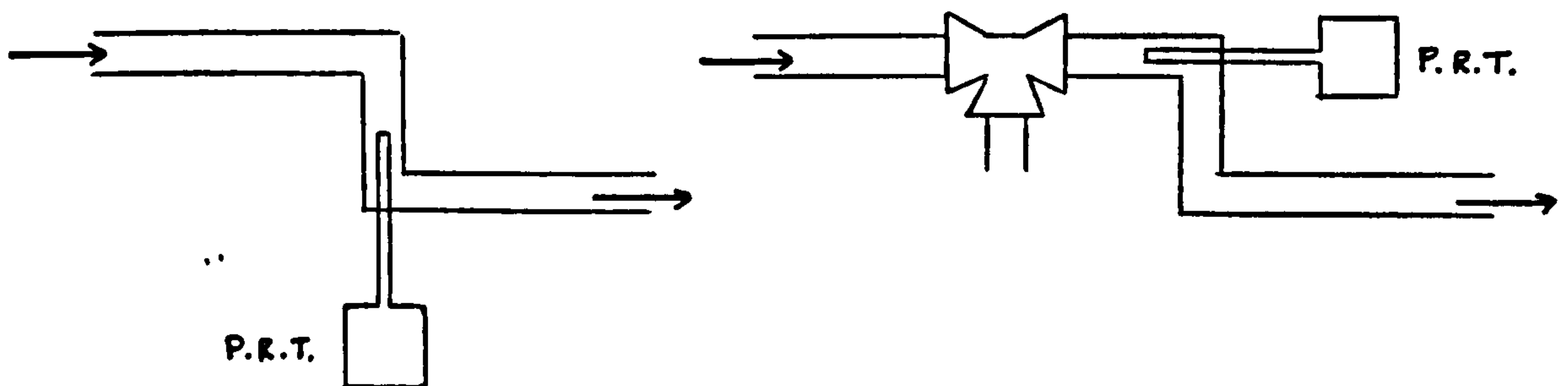
Both meters contain magnetic reed switches which are actuated by the water rotating a rotor or a propeller for the R.T. and P.T. models respectively. For each revolution of the rotor or propeller the reed switch contacts are closed twice, giving two pulses. Each meter is supplied with a calibration certificate giving a direct conversion from the pulse rate to the flowrate; this data is given in Appendix C.2. The nominal accuracy of the calibration is $\pm 0.1\%$, with a repeatability better than $\pm 1\%$.

A special interface was developed for the microcomputer, to enable the pulse rate to be determined by the software program, which then used the calibration data to calculate the flowrate. The interface worked by measuring the time for it to receive a certain number of pulses. The number was specified by the software program so that the program was never 'tied up' in the timing process for more than about 0.3 seconds. The time was measured digitally in units of 0.1 milliseconds and was therefore extremely accurate for all flowrates considered. For example, at the bottom of the range of each flowmeter, the pulse rate is about 7 pulses/sec. Assuming that only 1 pulse were timed this would take 1430 units of time (0.1 ms) so that an error of ± 1 digit would be $\pm 0.07\%$. At the top of the range, the pulse rate of each meter is about 34 pulses/sec and say 10 pulses might be timed taking 3000 units of time, so that ± 1 unit entails an error of $\pm 0.03\%$. This technique contrasts sharply in accuracy terms with the more conventional method of counting the number of pulses received in a fixed period of time ('gating'). Even if the software program were tied up on this counting process for 1 second (which is a relatively long time to leave the test facility without control) an error of ± 1 pulse in the pulse count total would represent $\pm 3\%$ at the top of the flowrate range and over $\pm 14\%$ at the bottom of the range. Such errors would have been unacceptable.

In conclusion, the software program was able to determine the meter's pulse rate with a negligible error and the accuracy of each flowrate reading was determined by the possible repeatability error in the calibration data of less than $\pm 1\%$.

3.2.2.3 Position of the Instrumentation

The location of the instrumentation on the test facility is shown schematically in figure 3.4. The charge and discharge circuits each contained 3 P.R.T.'s and 2 flowmeters. The P.R.T.'s were positioned after the temperature controlling 3-way mixing valve and at the inlet and exit of the store. At each point, the P.R.T. was inserted through an elbow in the pipework pointing into the flow. At the store ports it was a little downstream from another bend in the pipework (see the sketch below) whose purpose was to encourage a complete mixing of the water before it reached the P.R.T. After the temperature-controlling mixing valve it was positioned in the first elbow, to minimise the distance-velocity lag.



(a) At store ports

(b) after temperature controlling mixing valve

In fact for most of the experiments conducted, the flow in the pipe would have been highly turbulent. The pipe Reynolds number is plotted against flowrate for various flow temperatures in figure 3.5. Although it has been found that the critical Reynolds number for the

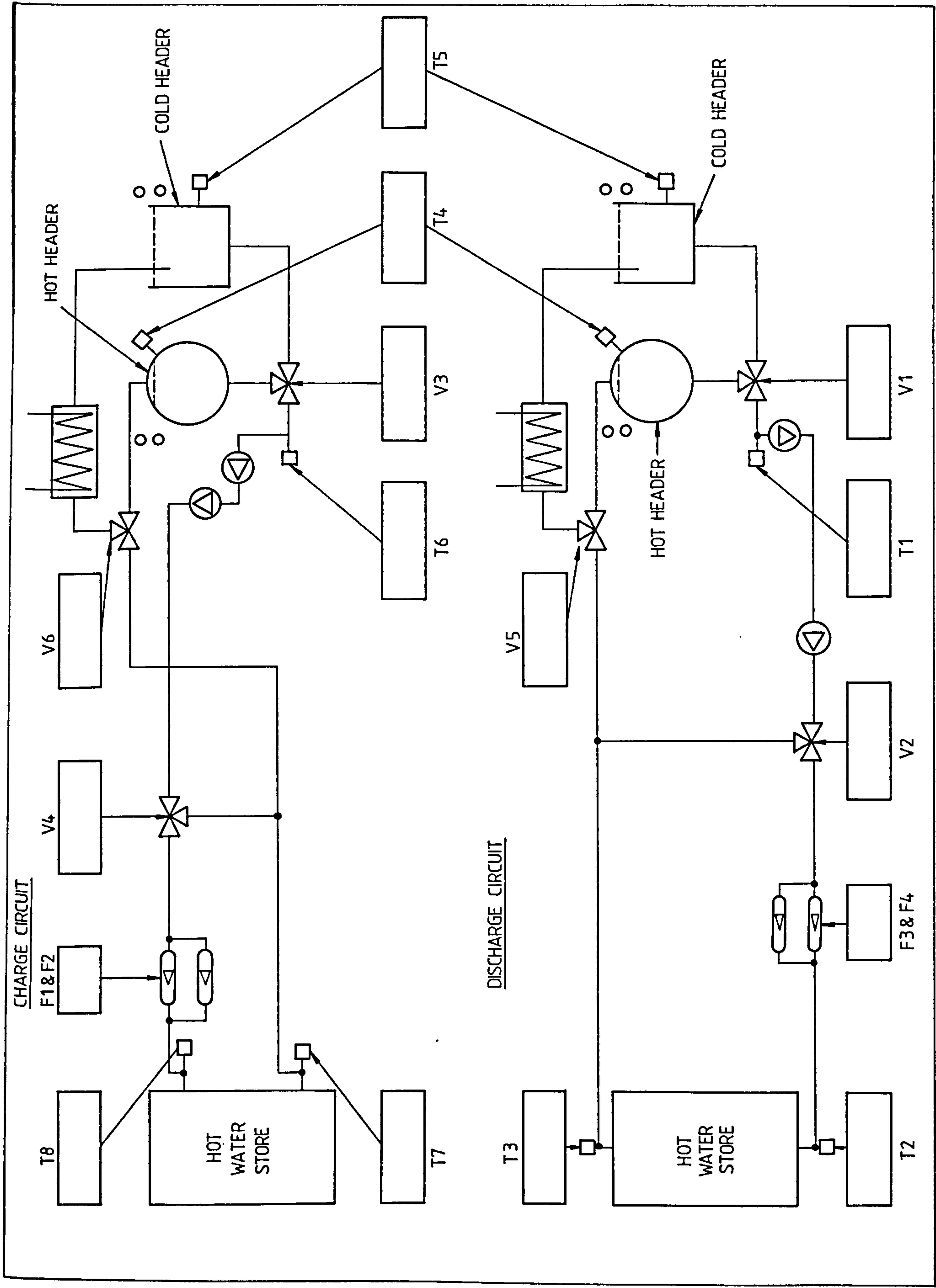


FIGURE 34 LOCATION OF TEST FACILITY INSTRUMENTATION.

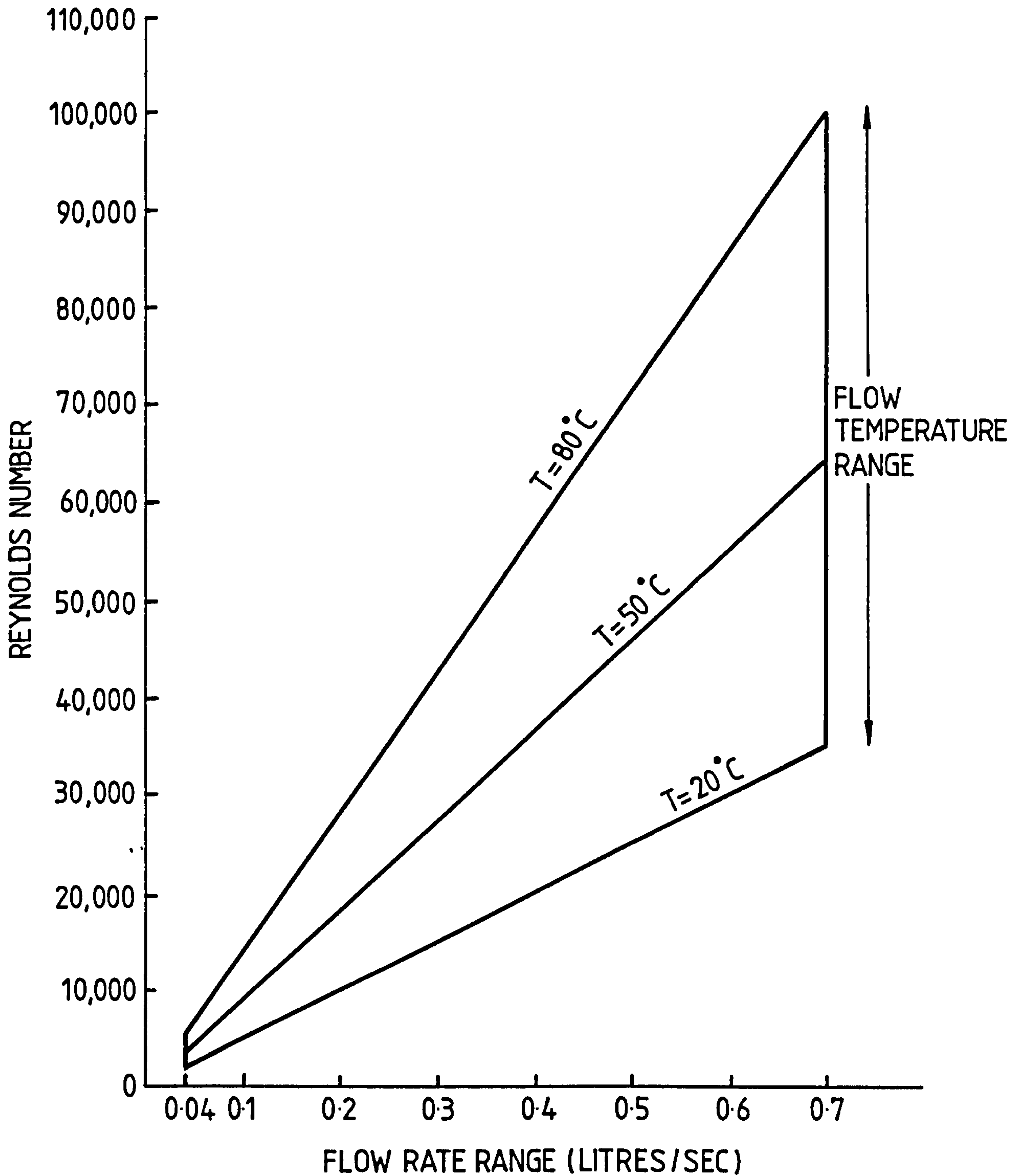


FIGURE 3.5 THE INLET PIPE REYNOLDS NUMBER OVER THE RANGE OF CONDITIONS AVAILABLE FROM THE TEST FACILITY.

onset of turbulence can vary between 2000 and 40000 [ref.3.1], in this case, given all the bends and obstructions in the pipework circuit, only experiments at the lowest flowrates and at low flow temperatures might possibly have entailed laminar pipe flow.

The 2 flowmeters, one for high and one for low flows, were placed in parallel, with the low flowmeter being joined to 1/2" pipework. The flowmeters were connected to straight lengths of pipe, longer than the recommended ten times the diameter upstream and five times the diameter downstream.

3.2.3 Computer Control and Monitoring

The step input experiments were driven by a software program running on a North Star Horizon microcomputer. At the start of the program, the operator was asked interactively to input the precise conditions required for the experiment; the computer then took over completely. It brought about the required initial conditions, carried out the step input experiment for the required period of time and then performed a summary analysis of the experiment. Precise details of this procedure and the program are described in Section 3.5. The prime functions of the computer were monitoring and control.

3.2.3.1 Monitoring

Monitoring was carried out for two purposes : experimental analysis and control of the experimental conditions. For analysis, the following data were recorded at constant time intervals throughout an experiment :

50 temperatures inside the store, arranged in a 2-dimensional lattice, as shown in figure 3.6;

- * the temperature at store inlet;
- * the temperature at store exit;
- * the temperature difference between store inlet and exit;

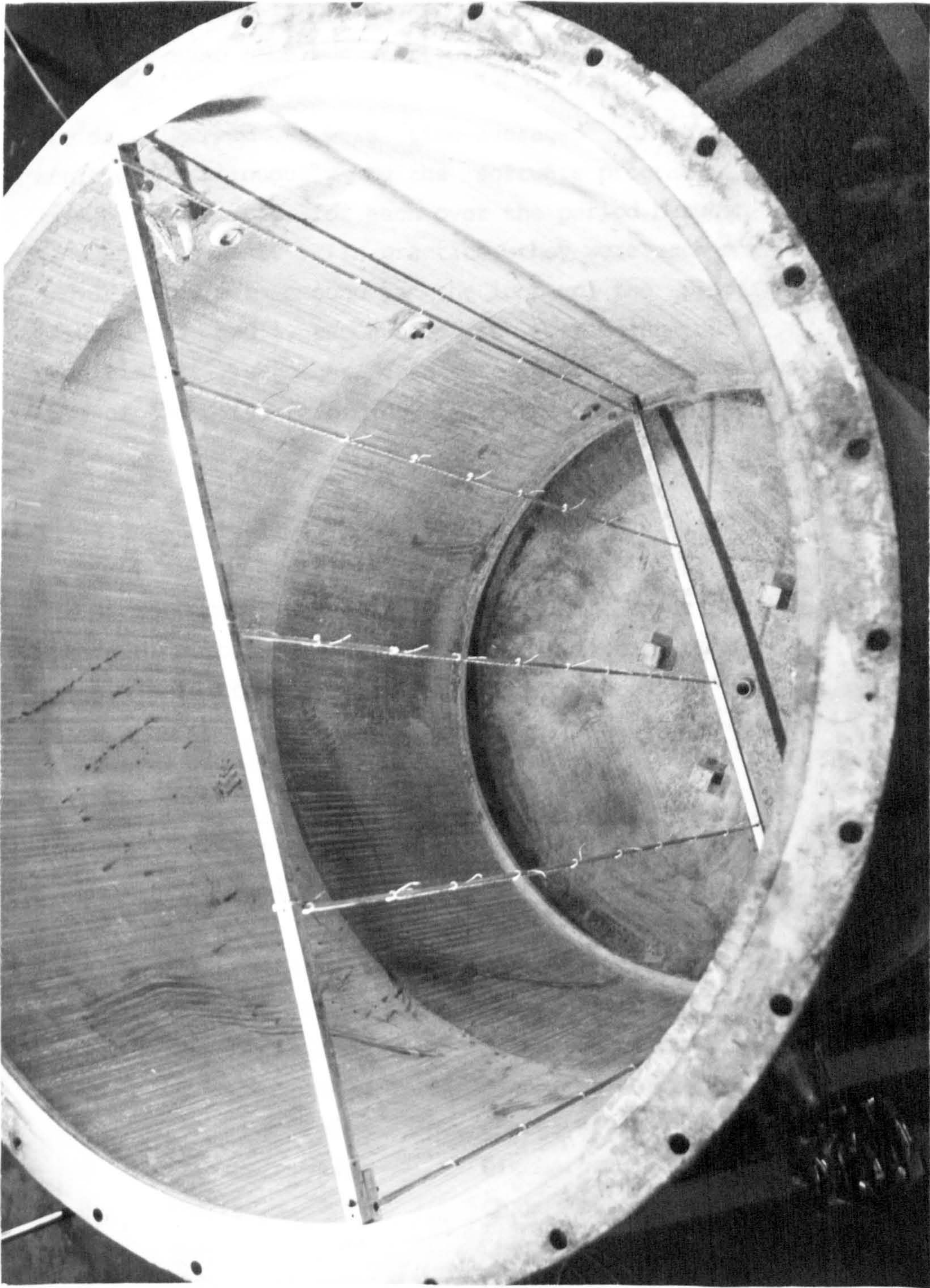


FIGURE 3.6 THE 2-DIMENSIONAL LATTICE OF THERMOCOUPLES INSIDE THE HOT WATER STORE.

the temperature of ambient (taken as the mean of 10 temperatures from all around the store);

* the flowrate into the store;

the time from the start of the experiment.

The data marked * were time-averaged values : they were being monitored continuously by the software program, which calculated a time-averaged value for each over the period between one recording of data and the next. (In practice they were each monitored between 1 and 3 times per second as the loop of the program concerned took between 0.3 and 1 seconds to run, depending upon the experimental conditions). The other data were instantaneous values taken at the time of the data recording. All the data were recorded onto floppy discs.

3.2.3.2 Control

The microcomputer was able to control the experiments by driving the motors of motorised 3-way valves and switching on and off circulating pumps. A 10 channel input/output interface was used, configured as below :

Channel	Controlled device	Purpose
1	Timed motorised valve	Charge circuit temperature control
2	Timed motorised valve	Charge circuit flowrate control
3	Timed motorised valve	Discharge circuit temperature control
4	Timed motorised valve	Discharge circuit flowrate control
5	Motorised valve	Charge circuit return flow divider
6	Motorised valve	Discharge circuit return flow divider
7	Pump	Charge circuit flow
8	Pump	Charge circuit flow
9	Pump	Discharge circuit flow
10	Pump	Discharge circuit flow

Software instructions could drive the valves forward or reverse and switch the pumps on and off. In addition the timed valves could be driven for a set period of time, up to 2.5 seconds, in steps of 10 msec. The drive time could be reset on each loop of the program. This meant that the valve would be driven continuously if the drive time was set to be greater than the loop time and, alternatively, the valve could be driven in a modulating fashion by setting the drive time to be less than the loop time. The valve could therefore be driven at almost any speed between zero and maximum speed. This was invaluable for achieving fast control whilst avoiding overshoot.

3.3 PERFORMANCE OF THE TEST FACILITY

Specific guidelines for the performance of a test facility were laid down in the NBS test procedure [ref.2.12] and the subsequent ASHRAE Standard 94-77 [ref.2.14]. The most pertinent criteria were :

1. Measurement of temperature should be accurate to $\pm 0.2^{\circ}\text{C}$, of temperature difference to $\pm 0.1^{\circ}\text{C}$ and of flowrate to $\pm 1\%$.
2. Ambient room temperature should not vary by more than $\pm 2^{\circ}\text{C}$ during a test.
3. The temperature of the water entering the store during a test should be maintained within $\pm 1^{\circ}\text{C}$ of the desired test value at all times, and the flowrate should vary by less than $\pm 1\%$.
4. During a step change experiment, the time taken to reach 90% of the step change in inlet water temperature should not exceed 2% of the test duration.

Condition 1, concerning instrumentation accuracy, was satisfied in this work, the achieved accuracies being temperature, $\pm 0.2^{\circ}\text{C}$ (see Section 3.2.2.1 (a)); temperature difference, $\pm 0.07^{\circ}\text{C}$ (see Section 3.2.2.1 (b)); and flowrate, $\pm 1\%$ (see Section 3.2.2.2).

Due to the size of the test facility, it was not possible to control ambient room temperature in order to satisfy condition 2. To make up for this, it was monitored at ten locations around the storage vessel

and the measurements were taken into account in the analysis.

Condition 3 and 4, concerning control of the inlet flow and the sharpness of a step change, were satisfied using special control algorithms described in the following section.

3.3.1 Development of Control Algorithms

(i) Flow temperature control

The controlling motorised valve had to mix water from a hot header tank at about 90°C and a cold header tank at about 15°C to a set flow temperature. The valve had a stroke time (i.e. the time to reverse its position) of 4 minutes, so that on average, driving the valve would change the flow temperature at a rate of 0.3°C/s. In practice, maximum changes in temperature with time ranged from 0.27°C/s to 0.48°C/s (see for example, figure 3.9) depending upon whether the valve was near the end or middle of its stroke and the precise temperatures in the header tanks. In general, the boundary conditions on the valve, i.e. the temperatures of the hot and cold header tanks, were continuously changing so that dynamic control was required. A simple feedback control algorithm, i.e. driving or reversing the valve continuously according to whether the flow temperature was above or below its set point, produced an oscillating flow temperature. This was the case even if some damping was introduced in the form of a 'deadband' of amplitude 0.3°C around the set temperature, in which no control action was taken (i.e. the valve was stopped). The result of such a control algorithm is shown in figure 3.7. The amplitude of the oscillations was well over 1°C and is therefore unacceptable according to the ASHRAE standard condition 3. A wider deadband would produce greater dampening and perhaps ultimately a stable, non-oscillating temperature response, but with the possibility of an unacceptably large offset.

The oscillatory behaviour was caused by the various lags in the system, the most important of which were :

- (a) Deadtime caused by the gap or dead space between the cogs on the gears of the motorised valve. Experiment showed that

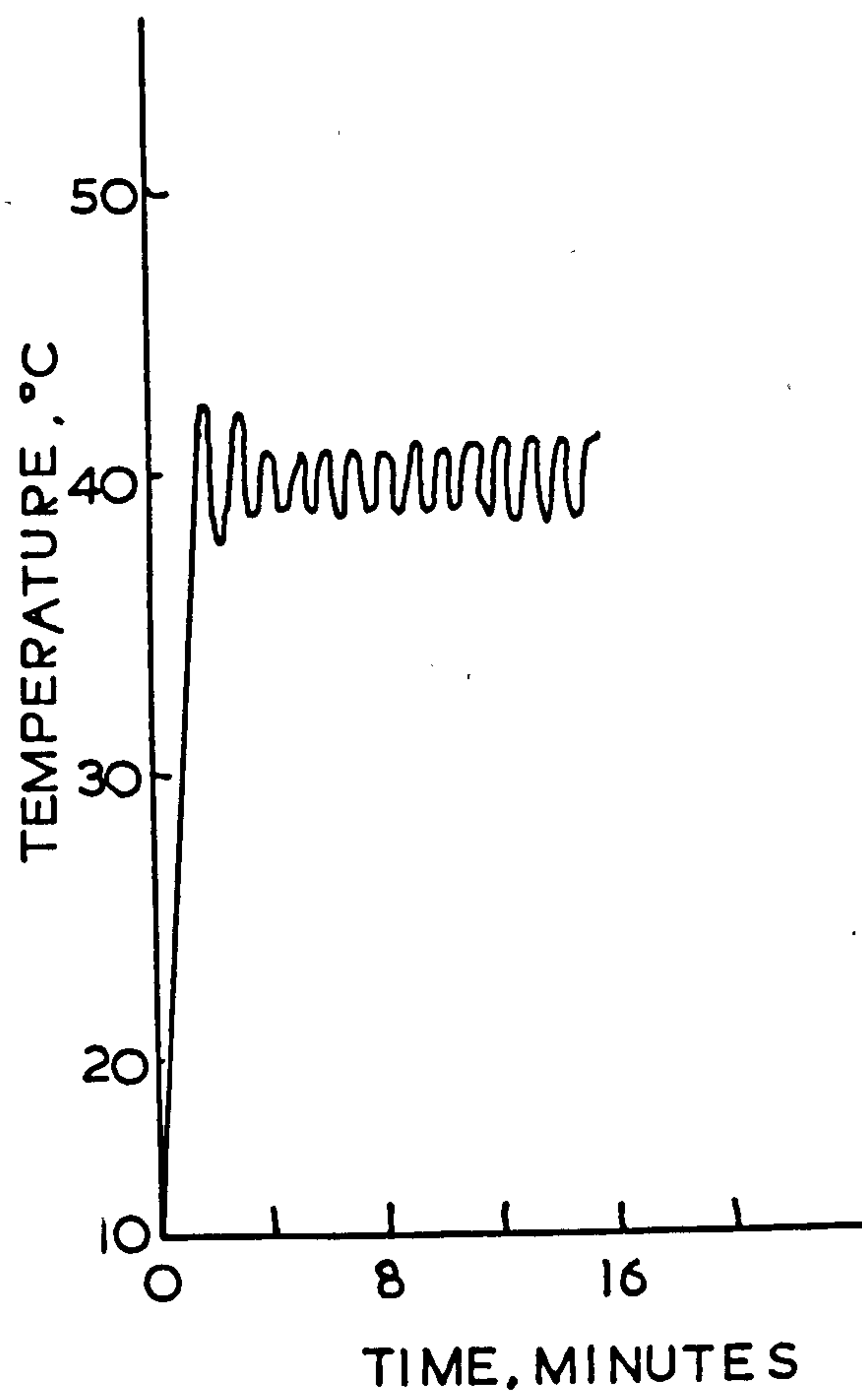


FIGURE 3.7 FLOW TEMPERATURE RESPONSE TO SIMPLE
FEEDBACK CONTROL AND A DEADBAND OF $\pm 0.3^{\circ}\text{C}$.

this deadtime delayed the reversal of the driving direction of the valve by about 5 seconds. In general this deadtime was a direct cause of the temperature overshooting its set value. However, when a steady condition was to be maintained, it could be helpful, if, for instance, it allowed a fluctuation to pass before the valve had time to move;

- (b) The heat transfer lag caused by the thermal resistance and capacity of the stainless steel casing on the P.R.T. probe. The time constant of the P.R.T. was about 10s. and although this prevented wild fluctuations in the measured temperature, it was another direct cause of the flow temperature overshooting its set value;
- (c) The distance-velocity lag caused by the distance between the valve and the P.R.T. probe. The distance was made as small as practicable at about 0.1m. The temperature-controlling valve came before the flow diverting valve, and so it was always passing full flow which, at 0.5 kg s^{-1} , had a velocity of about 1 ms^{-1} . The lag was therefore only about 0.1s and was normally swamped by the lags in (a) and (b) above.

To prevent overshoot and an oscillatory flow temperature a predictive algorithm was developed. This involved determining a trend for the flow temperature as well as its current value. The principle of the control method is illustrated in figure 3.8. Two bands were defined : an inner band within 0.3°C of the set temperature and an outer band within 10°C of the set temperature. If the current value of the flow temperature was inside the inner band, no control action was taken, i.e. the valve was stopped. If the current value was outside the outer band, the valve was driven continuously in the appropriate direction. If the current value lay between the two bands then the trend was also calculated by comparing the current value with the previous value. The trend would indicate that the flow temperature was either overshooting the set temperature or moving towards it. If it was overshoot, then the valve was driven continuously to try to reverse the trend. If it was progress towards the set point then the

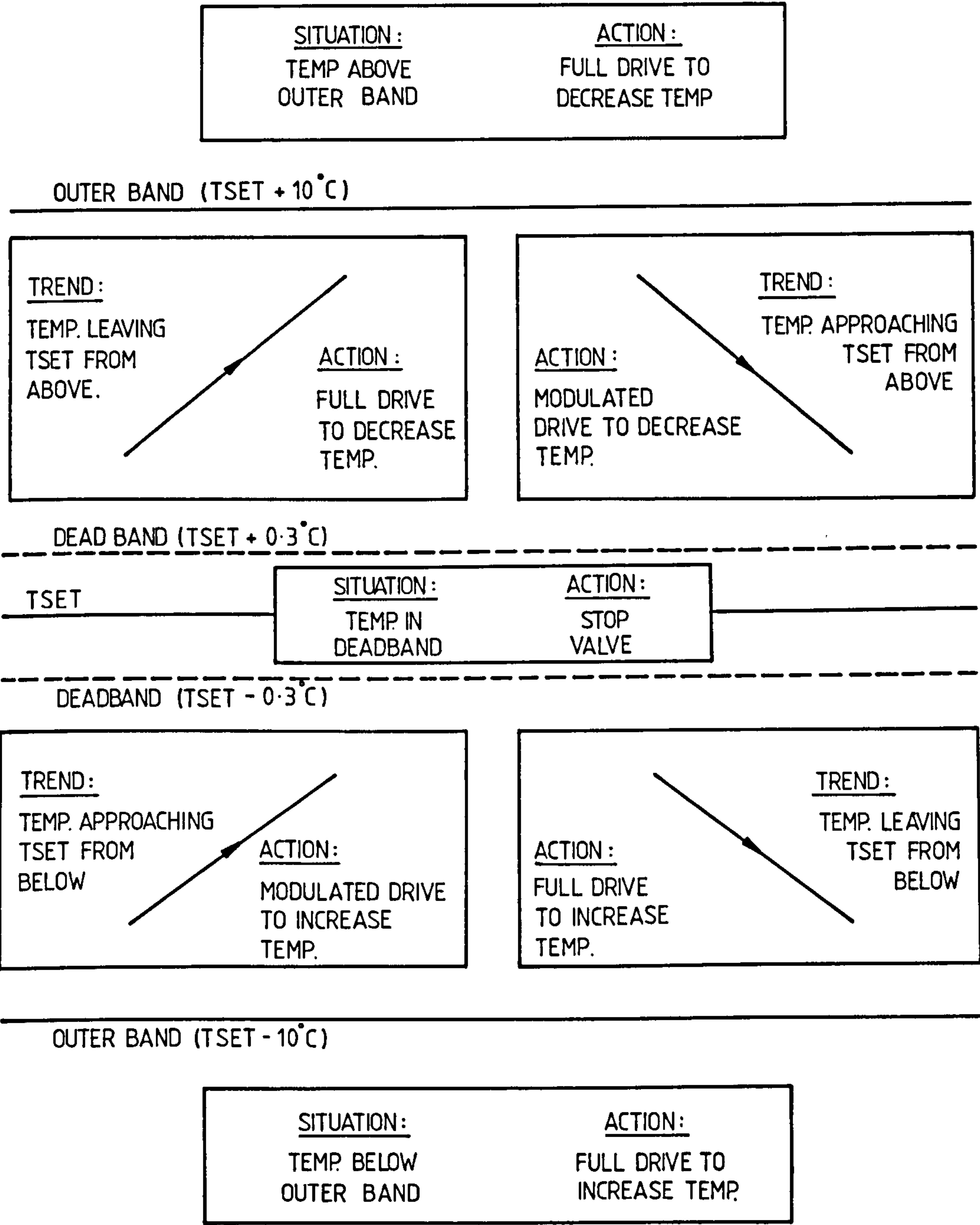


FIGURE 3·8 MOTORISED VALVE ACTION TO CONTROL FLOW TEMPERATURE
ACCORDING TO PAST TREND.

valve was driven in a modulating fashion (see Section 3.2.3.2), so that the temperature 'edged' towards the set point. Experimentation revealed the optimum modulating speed, being a compromise between wishing to reach the set point as fast as possible and not wanting to overshoot.

A characteristic step change in flow temperature, produced using the finally adopted control algorithm, is shown in figure 3.9. It can be seen that the initial overshoot is only of the order of 0.5°C and after that the flow temperature remains within 0.2°C of the set point. This is comfortably within the third criterion prescribed by the ASHRAE Standard.

The other point of interest in figure 3.9 is the speed of the step changes. The step rise of 20°C takes 80 seconds and the step drop 100 seconds. These are far less than 2% of the 2 and 4 hour test durations recommended by the ASHRAE Standard.

(ii) Flow rate control

The controlling motorised valve had to divert part of the incoming flow to produce the set flowrate. In general, the boundary condition on the valve, i.e. the incoming flowrate, was changing with time. This was basically due to the fact that the flow circuits leading from and returning to the hot and cold header tanks were different, had different resistances and, since they used common pumps, gave different flowrates. Thus the incoming flowrate varied according to the proportions of water being drawn from the hot and cold header tanks, which were themselves continuously changing with the position of the temperature-controlling mixing valve. Once again, therefore, dynamic control was required.

As with the temperature control, a simple feedback control algorithm produced an oscillating response because of the lags in the system. In this case these were :

- (a) The deadspace in the gears of the motorised valve as described in the section on flow temperature control;

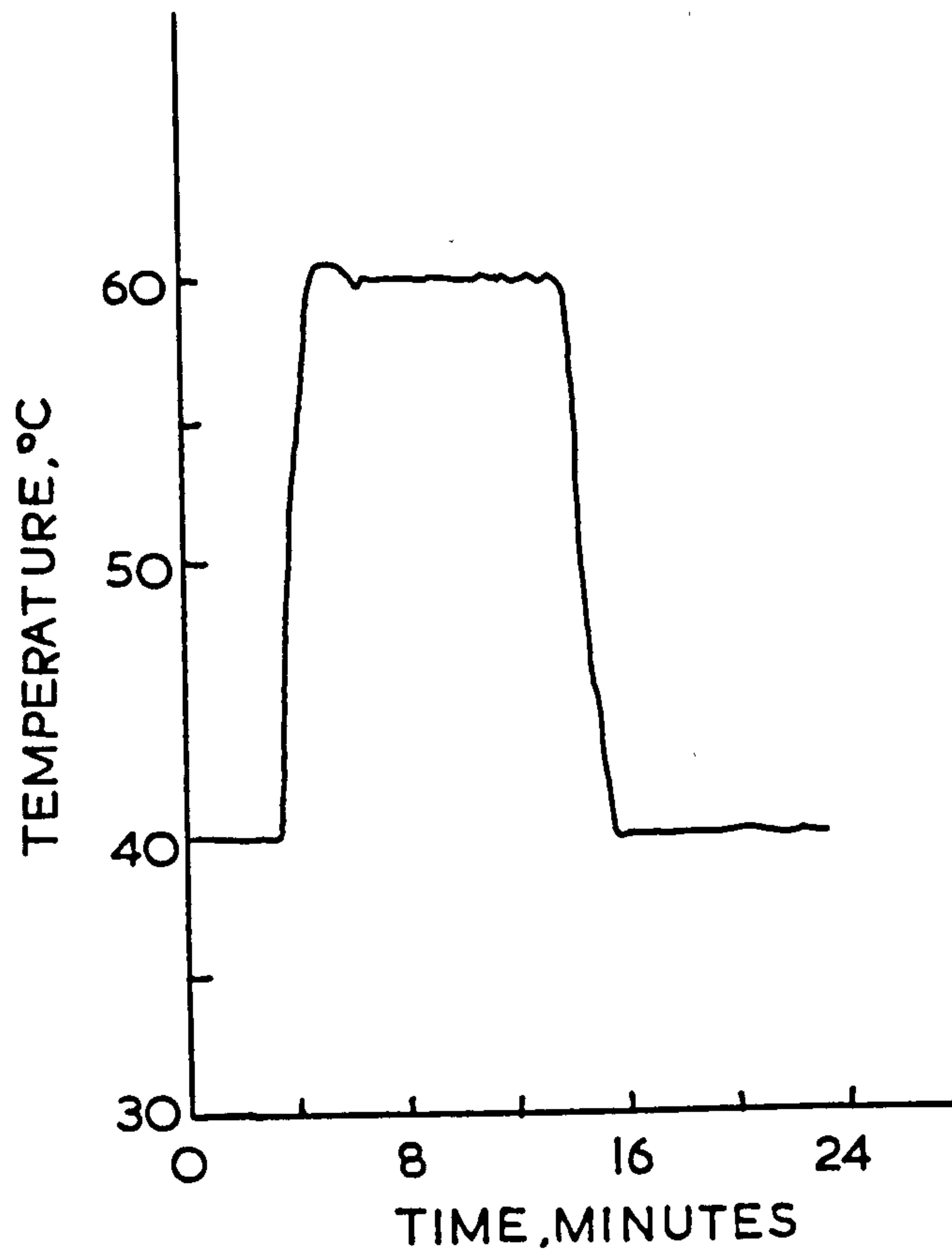


FIGURE 3.9 STEP CHANGE IN FLOW TEMPERATURE USING FINAL CONTROL ALGORITHM.

- (b) The mechanical lag created by the finite acceleration of the turbine or rotor in the flowmeter as it responded to changes in flowrate. This lag should have been relatively small compared to the heat transfer lag incurred with the P.R.T. probe.

There should have been no distance-velocity lag due to the incompressibility of the flow. Overall the lags were thus less severe than for flow temperature control. Nevertheless, to prevent the oscillatory behaviour, a predictive control algorithm, based on the principles used for flow temperature control, was employed. The inner 'dead' band was set at 0.004 litres/s or 2% of the set flowrate, whichever was lower, and the outer band was set at 0.01 litres/s.

For the purposes of this work, it was only required to maintain a constant flowrate, while a step change in flow temperature was carried out. The precision of the control of flowrate achieved by the predictive control algorithm is best illustrated by a typical step input experiment and an example is the chart recording reproduced in figure 3.10. The control can be seen to be almost perfect, with only occasional blips off the set flowrate of magnitude 0.004 litres/s. For the same experiment, examination of the data recorded by the computer for the time averaged flowrate reveals that no datum deviated from the set flowrate by more than 0.001 litres/s. This precision is better than that shown on the chart recording for 2 reasons : (i) the obvious smoothing effect of time-averaging and (ii) due to the way the chart recording was generated. This was done as follows : the computer converted the flowrate into an '8 bit' integer (i.e. an integer between 0 and 255) which was output through a digital to analog converter (DAC) to produce a corresponding signal in the range 0 to 10 volts. The calibration used was to equate 1 litre/s to an integer value of 255. Thus 1 digit on the DAC represented 0.004 litres/s and the blips on the chart recording depict the integer input to the DAC 'flipping' by 1. They are the smallest changes in flowrate that would show up on the chart recording.

EXPERIMENT NUMBER 21-090883

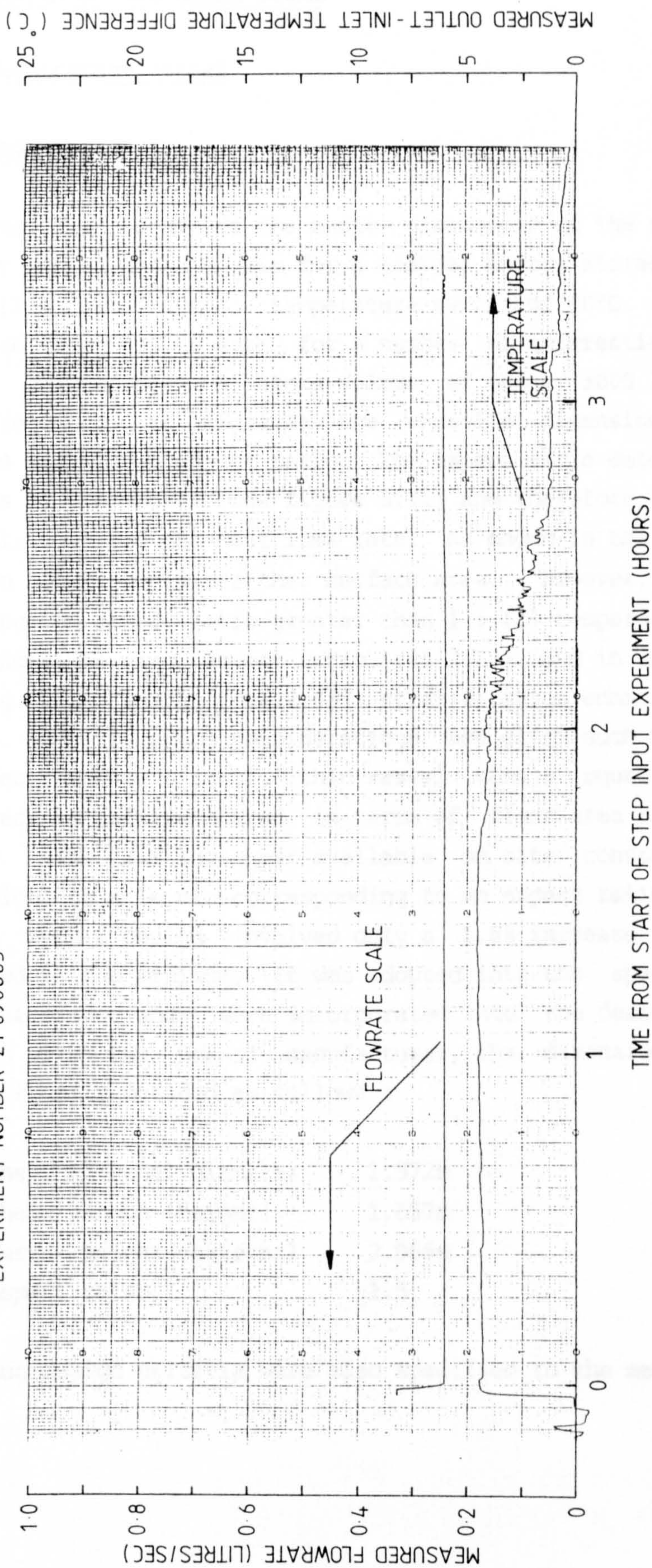


FIGURE 3-10 CHART RECORDING OF A TYPICAL EXPERIMENT DEMONSTRATING THE QUALITY OF FLOWRATE CONTROL.

3.4 DESIGN OF THE WATER STORE

3.4.1 The storage vessel

3.4.1.1 Dimensions

The first step was to decide the basic dimensions of the store. To this end, it was proposed to aim for a nominal energy storage capacity of 0.5 GJ (139 kWh) over a temperature swing of 40°C. This was considered an appropriate size for a number of interesting applications (see Chapter 6). A store volume of about 3000 litres was therefore required. To finalise the vessel's dimensions it was necessary to specify an aspect ratio. The aspect ratio determines the surface area of the vessel (see figure 3.11) and therefore influences the materials cost and the heat loss rate. As shown in the figure, an aspect ratio of 1 minimises the surface area. However, two other factors favour an aspect ratio greater than 1 : (i) temperature stratification increases with aspect ratio [ref.2.17] and in some applications (e.g. solar energy [ref.2.43]) stratification enhances system performance, and (ii) space considerations are often such that it is easier to accommodate a tall, thin vessel than a squat one, viz. building space is always costed in terms of floor area rather than room volume. The lifting gear available on site constrained the vessel's height to a level corresponding to an aspect ratio of about 1.5. Since such a design involved only a 1.8% increase in surface area compared to the minimum, it was adopted into the specification. Standard size dished ends were incorporated into the design and, in consultation with the vessel manufacturer, the dimensions of the vessel were finally decided as follows :

Shell external diameter	1.372m
Shell length (height)	1.657m
Vessel height (external	2.066m
Aspect ratio	1.5

The following design criteria were also specified to the manufacturer:

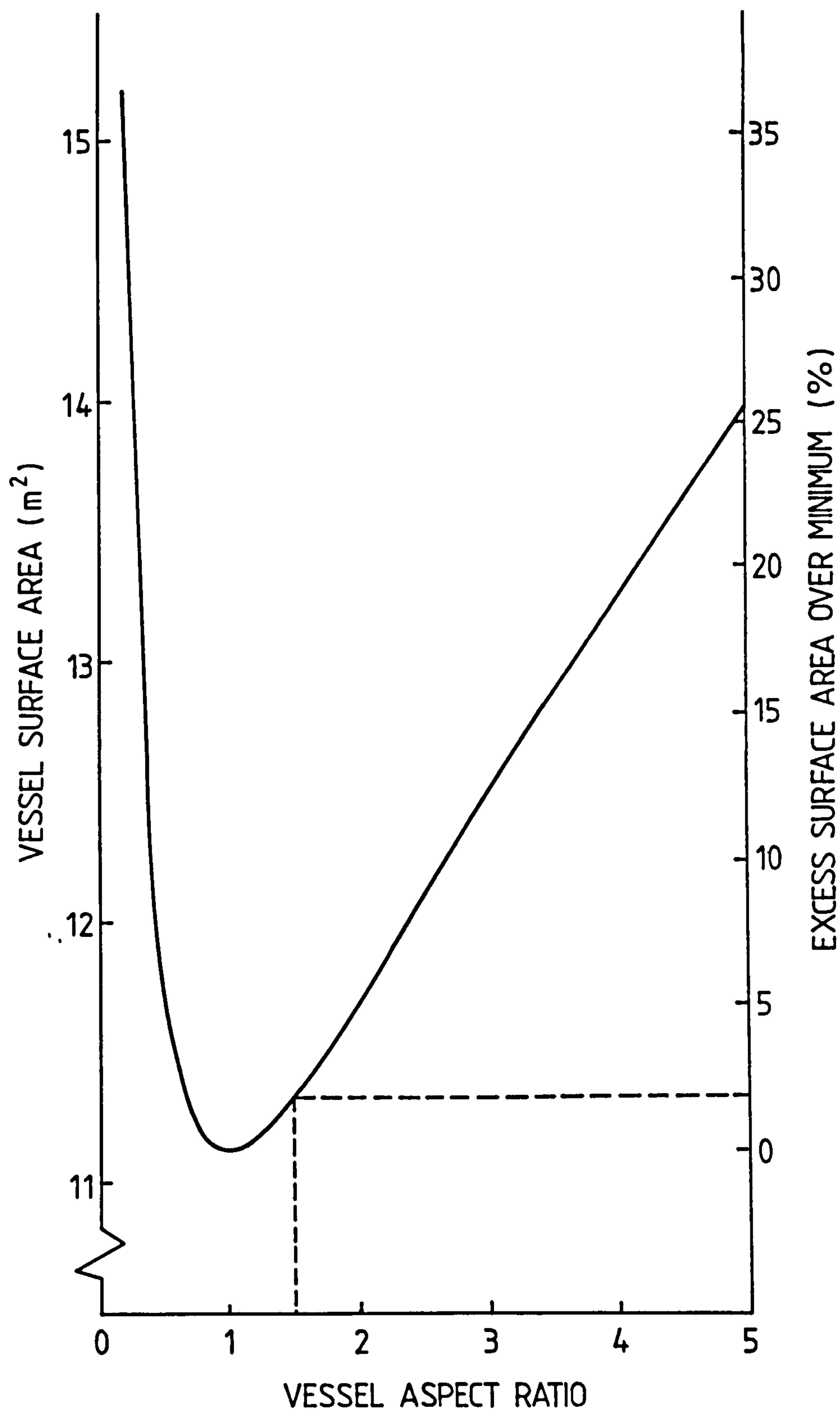


FIGURE 3-11 THE VARIATION OF SURFACE AREA WITH ASPECT RATIO FOR A CYLINDER OF VOLUME 2851 LITRES.

1. Vessel to contain water, maximum working temperature 95°C, maximum working pressure 5 p.s.i.
2. Identical dished ends to be removable from shell
3. Complete vessel to be galvanised.

Drawings of the final design are shown in figures 3.12 and 3.13. The vessel was manufactured from 6mm thick steel boiler plate.

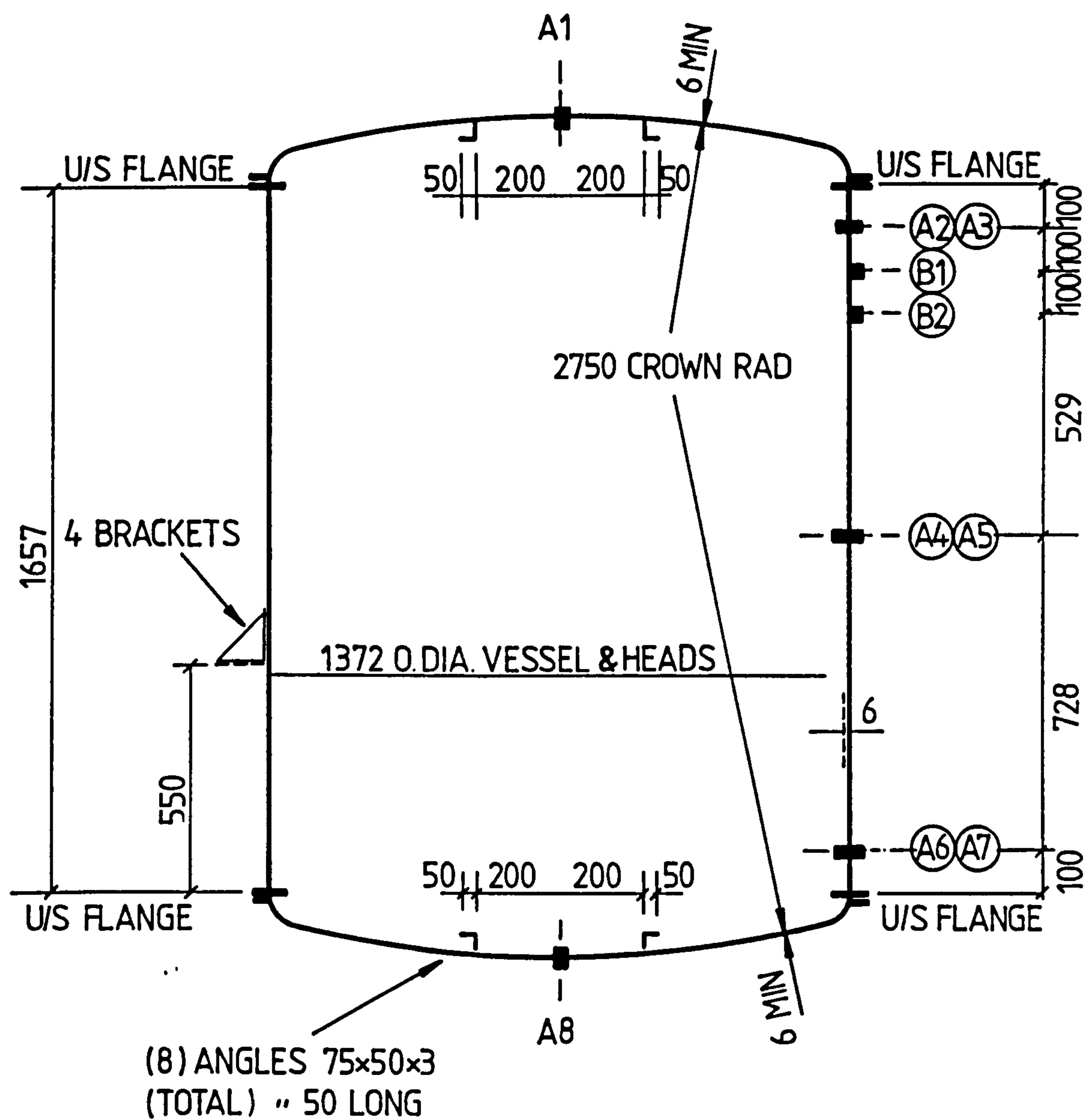
3.4.1.2 Tank Support frame

A rigid, free-standing frame was constructed to support the vessel. It permitted sufficient ground clearance to enable pipework to enter the vessel from below and also to allow the dished end on the bottom of the vessel to be removed, when free access to the vessel from below was required. (It was easier to gain access to the inside of the vessel by 'dropping down the bottom' than lifting off the top using an overhead lifting gantry). Part of the framework was made demountable to permit the initial installation of the vessel by the overhead lifting gear.

The design of the framework is shown in figures 3.14 and 3.15. It can be seen that sufficient clearance was left between the frame and the vessel sides for at least 100mm of insulation to be applied. The contact points on the frame that actually supported the vessel represented another potential path for heat losses from the store, and so 10mm pads of low thermal conductivity 'Paxolin' was sandwiched between the frame and the supporting brackets on the vessel.

3.4.1.3 Vessel connections

The vessel was made with a number of 1" BSP fittings, located as shown in figure 3.12. In the centre of each dished end there was a connection allowing 'vertical' inlet and outlet ports (A1 and A8). Adjacent to these were brackets which could support distributors connected to the inlet and outlet ports (see Section 3.4.2). On the shell of the vessel there were connections for 'horizontal' inlet and outlet ports (A2 and A6). These were located 1.457m apart and 0.298m vertically



CONNECTIONS MK'D A: FULL SCR D COUPLINGS BSP THREAD
" " B: HALF " " " "

FIGURE 3-12 ELEVATION OF EXPERIMENTAL HOT WATER STORAGE VESSEL.

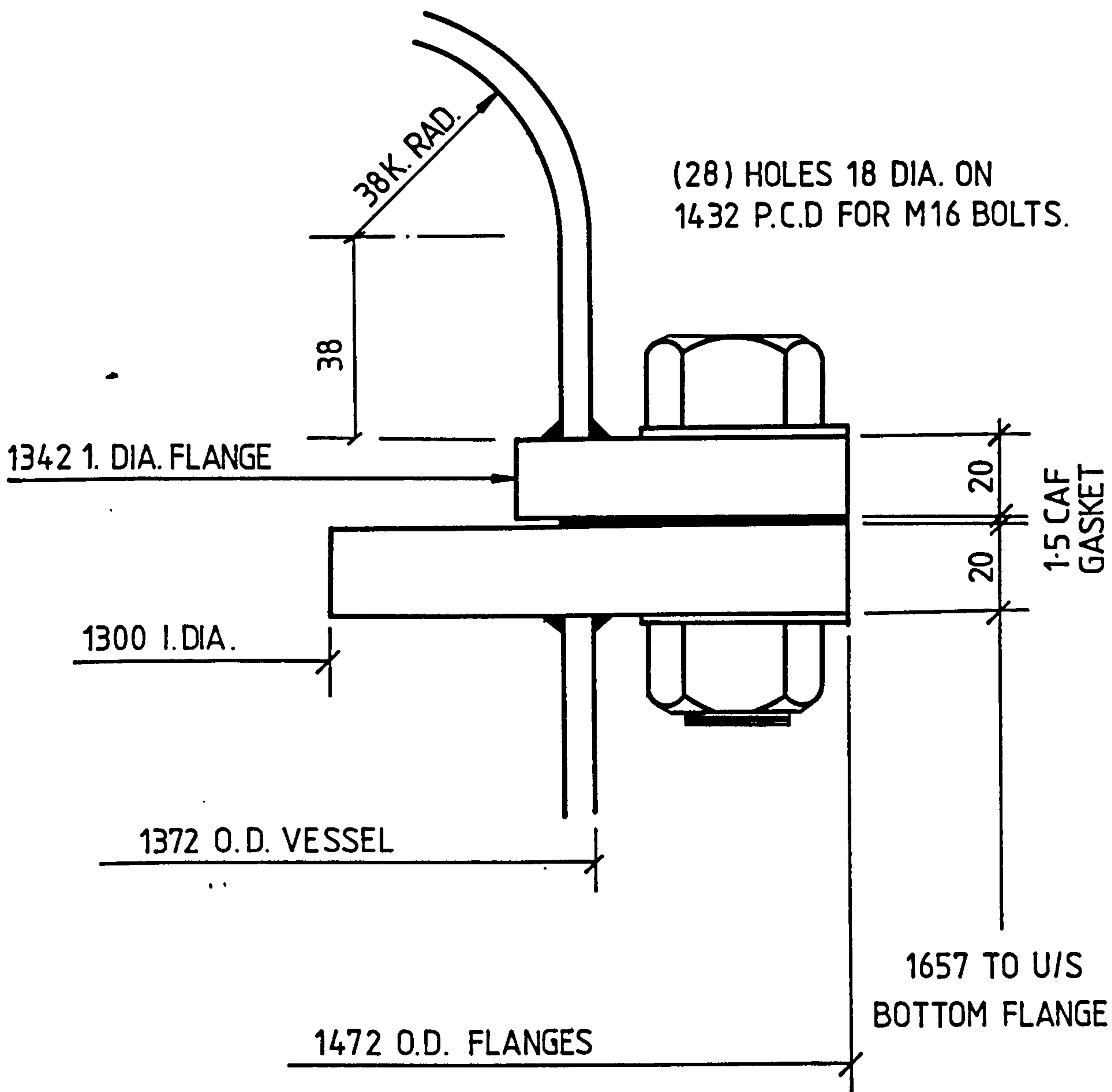


FIGURE 3-13 ENLARGED DETAIL OF FLANGE ARRANGEMENT FOR STORAGE VESSEL.

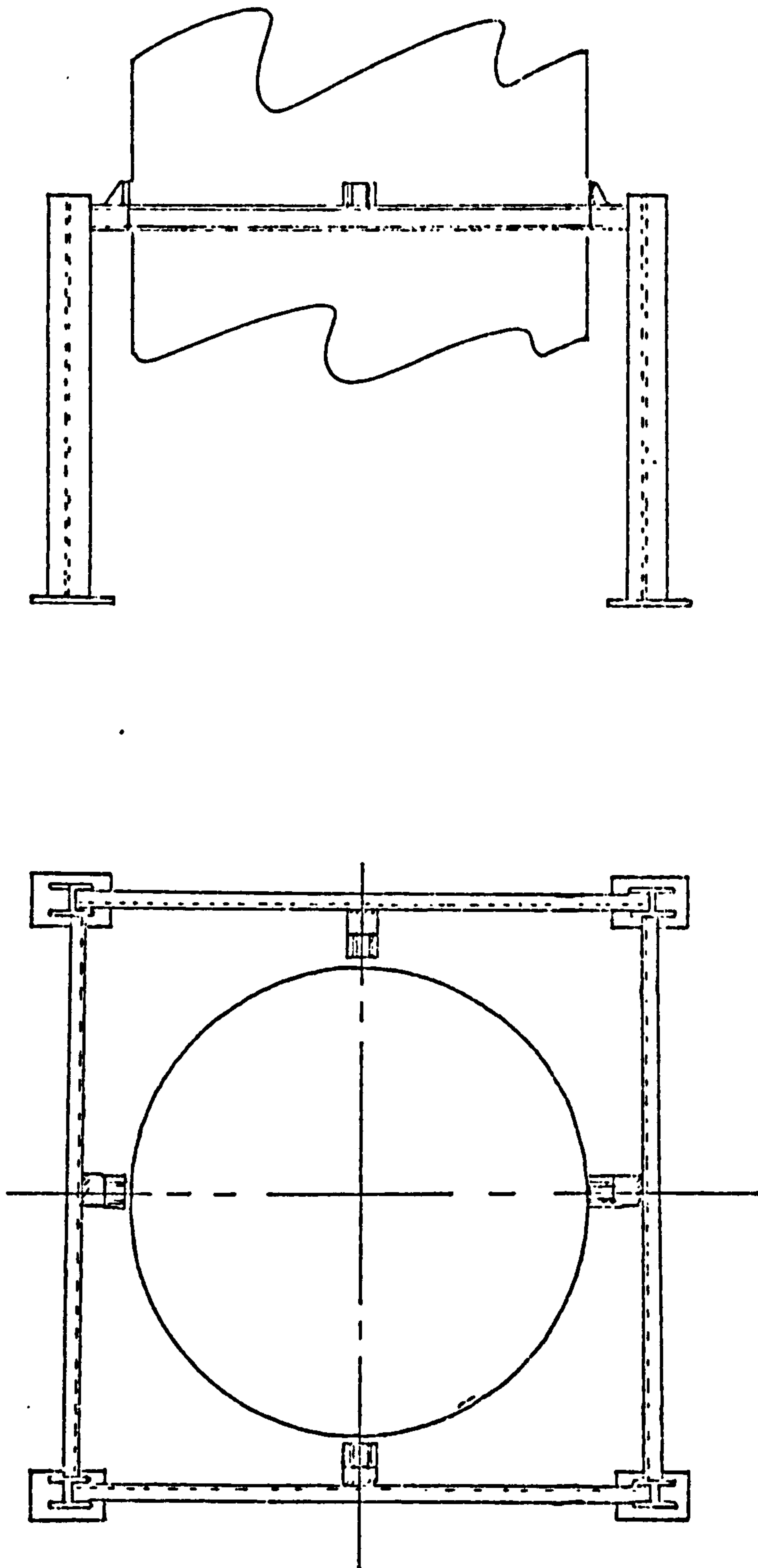
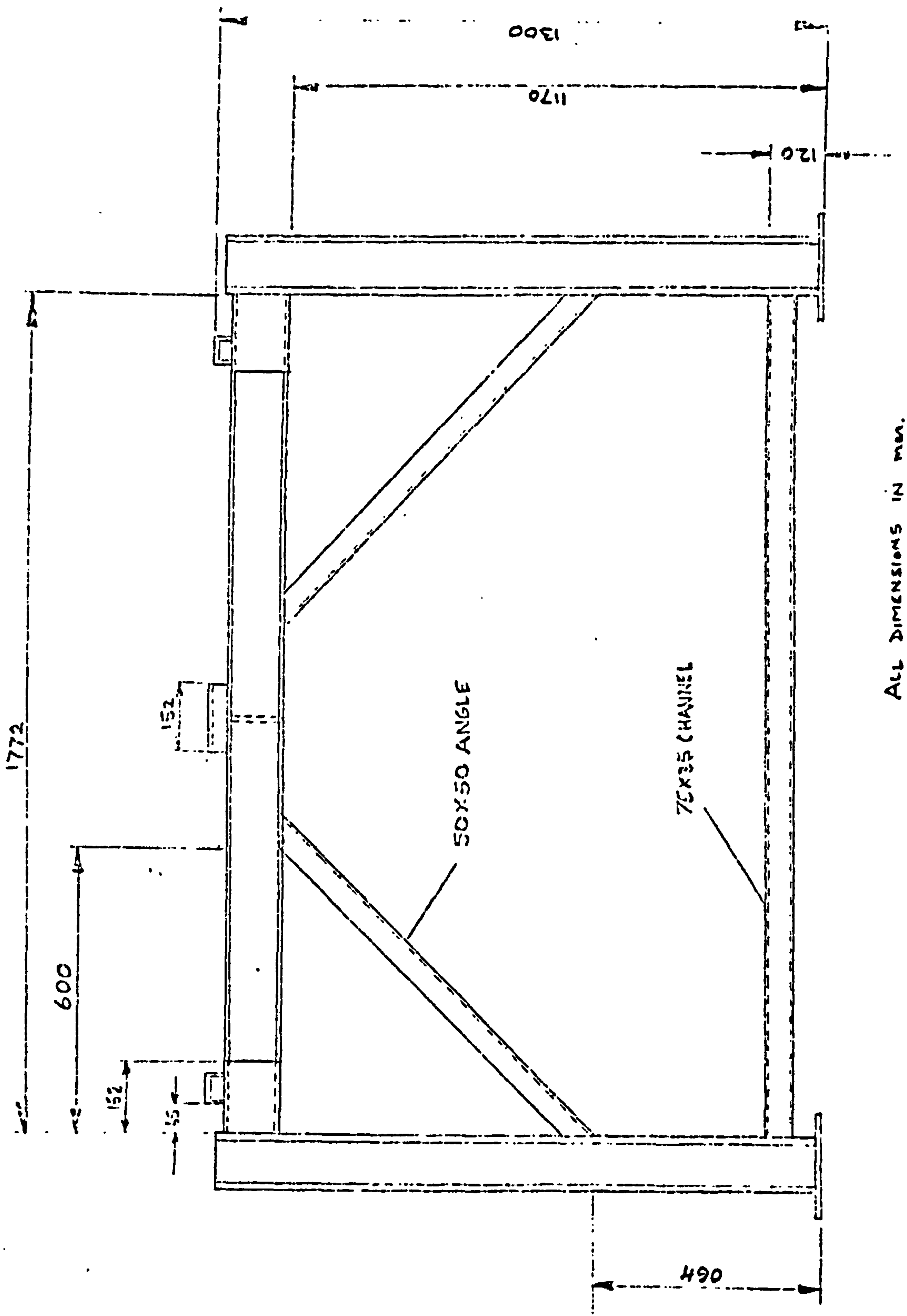


FIGURE 3.14 TANK SUPPORT FRAME - ELEVATION AND PLAN.



ALL DIMENSIONS IN MM.

FIGURE 3.15 TANK SUPPORT FRAME - SIDE.

from the centre of each dished end. The fitting marked B1 was for the 50 thermocouple leads being used to measure the water temperature inside the vessel. The leads passed straight through a conical section tube fixed to the connection and a watertight seal was achieved by filling the conical section with silicone rubber.

3.4.1.4 Vessel volume

The internal volume of the vessel was calculated to be 2853 litres. The calculation was based on the dimensions on the design drawing (figure 3.12) which were approximately verified by measuring the actual vessel with a tape measure. Details of the calculation are given in Appendix C.3.

The capacity of the vessel was also measured directly by weighing. The vessel was filled and then the water was syphoned out into a tank on some scales. The scales were accurate to $\pm 0.1\text{kg}$ and a number of tank loads were necessary. The final result gave a capacity of $2844.8 \pm 2.8 \text{ kg}$. Since the temperature of the water was about 20°C , its density was 998 kg/m^3 . This means that the vessel volume was 2851 litres, which is gratifyingly close to the calculated value.

3.4.1.5 Vessel theoretical storage capacity

The volume of steel and insulant used for the vessel were calculated, as shown in Appendix C.4. The volume, mass and heat capacity of all the components of the store are given in Table 3.1. This shows that the vessel's theoretical storage capacity (TSC) per degree of temperature rise was 12.3 MJ K^{-1} , 97% of which was contributed by the water. As will be shown in Chapter 4, it was important to make a meticulous estimate of TSC.

	Volume m ³	% Volume	Density @ 20°C kg/m ³	Mass kg	% Mass	Specific Heat Jkg ⁻¹ K ⁻¹	Heat Capacity MJK ⁻¹	% Heat Capacity
Water	2.851	67	998	2845	79	4182	11.9	97
Steel	0.0908	2	7800	708	20	500	0.35	3
Insulant	1.344	31	40	54	1	1000	0.05	0
TOTAL	4.286			3607			12.3	

Table 3.1 Breakdown of contributions to theoretical storage capacity of the water store

Over a 40°C temperature swing the TSC is 492 MJ (137 kWh), very close to the design aim.

3.4.1.6. Store insulation and heat loss rate

Analysis of the results of a step input experiment on a store must take thorough account of the heat losses from the store during the experiment, otherwise the quality of the heat and/or mass transfers inside the store can be obscured. There is an inherent difficulty in estimating the rate of heat loss from the store throughout an experiment, due to the rapidly changing store temperature distribution, variations in ambient temperature and the temperature and flowrate dependence of the heat loss rate factor, 'UA'. It was therefore decided to insulate the store to a level where the heat losses during an experiment were very small compared to the energy input or output. Thus even large errors in the estimates of heat losses would be relatively insignificant.

The vessel was insulated with polyisocyanurate foam, sprayed to a minimum thickness of 100mm (see figure 3.16). The thermal conductivity of the foam was quoted by its manufacturer as 0.029 Wm⁻¹ K⁻¹ at a mean temperature of 50°C, so under these conditions the theoretical 'U'



FIGURE 3.16 THE HOT WATER STORAGE VESSEL INSULATED WITH FOAM

value was $0.29 \text{ Wm}^{-2} \text{ K}^{-1}$. The surface area of the vessel was calculated to be 11.9 m^2 , assuming it to be a perfect cylinder of diameter 1.372 m and height 2.066 m . Thus the theoretical 'UA' value was 3.4 WK^{-1} . ('UA' is the store heat loss rate per degree temperature excess over ambient, also denoted $(UA)_{s,a}$.)

A standing heat loss test was performed on the store over a period of 13 days, with the initial temperature of the store being 51.1°C . The mean store temperature and the mean ambient temperature during the experiment are plotted in figure 3.17. The 'UA' value can be shown (see Appendix C.5) to be given by :

$$\ln \frac{T_o - T_{\text{amb}}}{T - T_{\text{amb}}} = \frac{(UA)_{s,a}}{M C_p} \cdot t$$

The experimental data has been plotted in the form suggested by this equation in figure 3.18. Calculating the gradient of the graph yields, 'UA' = 3.9 WK^{-1} assuming a mean ambient temperature of 18°C and 'UA' = 4.5 WK^{-1} assuming an ambient of 21°C . It was decided to use a value of 'UA' = 4 WK^{-1} , which is not much above the theoretical value. The store's heat loss time constant (also derived in Appendix C.5) is given by :

$$\tau_c = \frac{M C_p}{(UA)_{s,a}} = \frac{12.3 \cdot 10^6}{4} = 854 \text{ hours } (\sim 36 \text{ days})$$

The charge or discharge period of the store (defined in Section 4.2.1.5), which, for the range of flowrates considered, varied between 1 and 20 hours, was therefore between only 0.1% and 2% of the heat loss time constant. This meant that heat losses during an experiment were indeed only a small fraction of the energy input or output.

3.4.2. The Distributors

As mentioned in Section 3.4.1.3, three categories of inlet and outlet ports were assessed in these experiments :

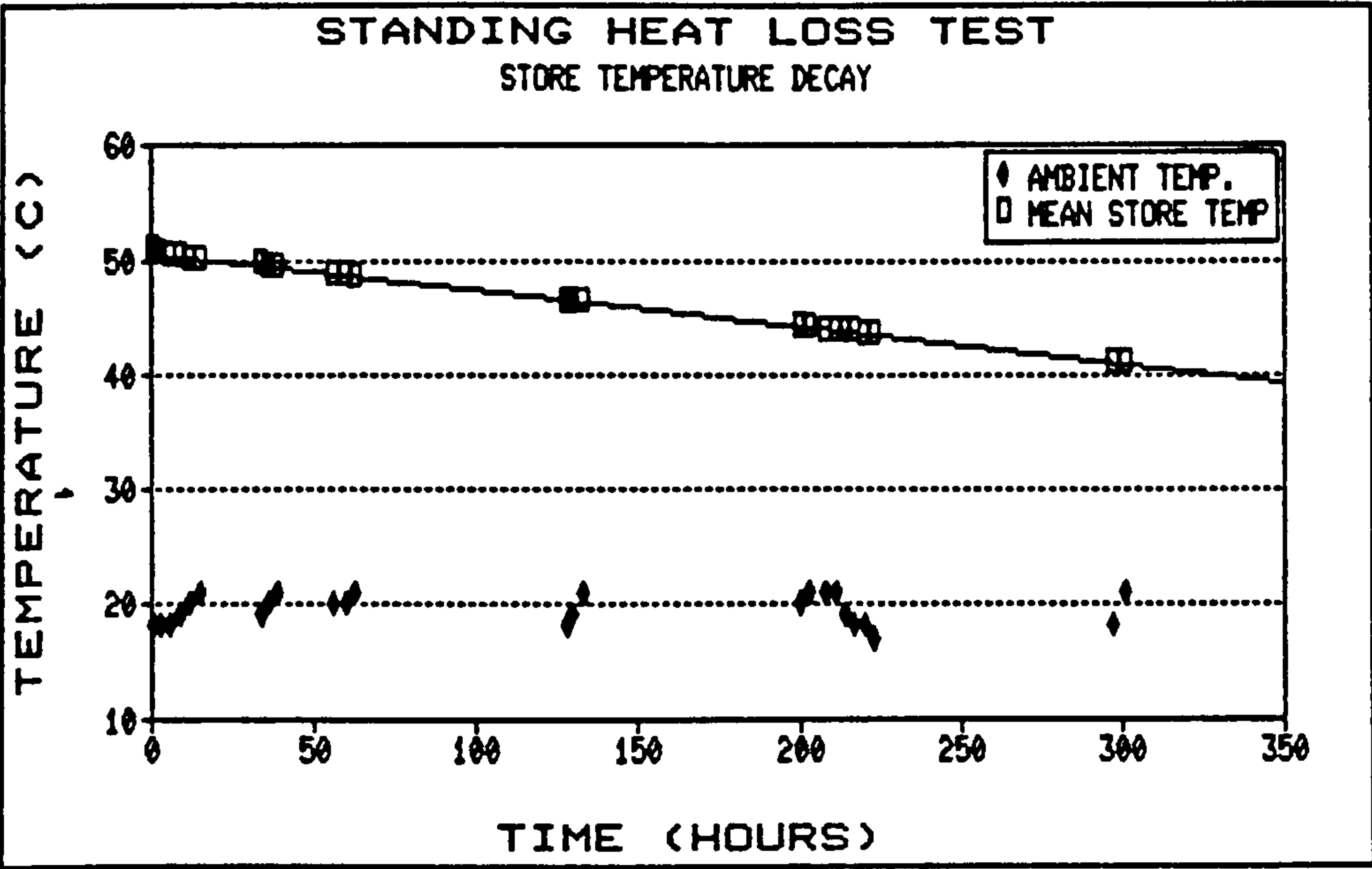


FIGURE 3.17 THE MEAN STORE AND AMBIENT TEMPERATURES DURING THE STANDING HEAT LOSS TEST .

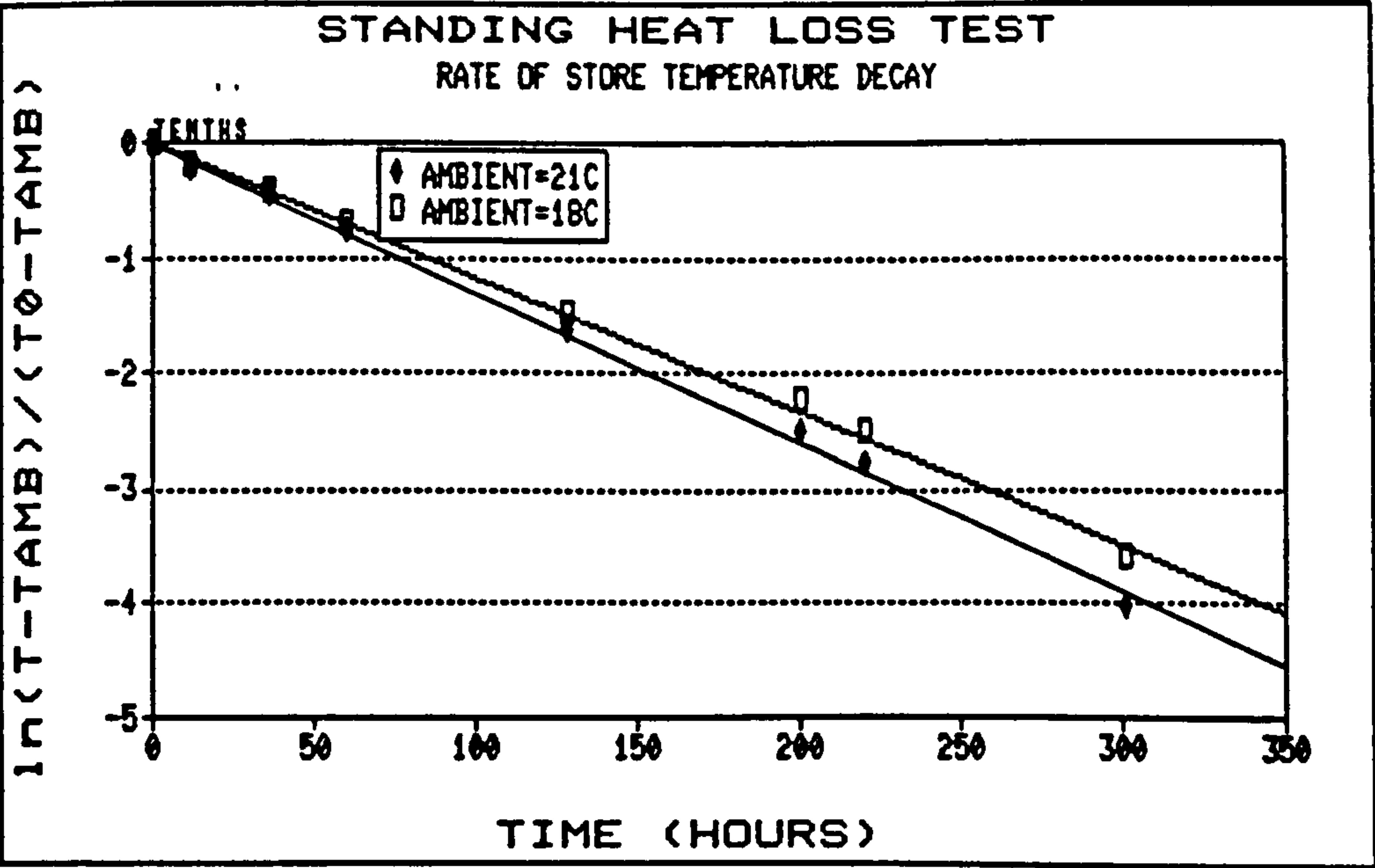


FIGURE 3.18 THE STANDING HEAT LOSS TEST DATA PLOTTED FOR THE CALCULATION OF 'UA' .

- (i) plain horizontal ports
- (ii) plain vertical ports
- (iii) vertical ports connected to horizontal distributors.

Both sets of plain ports were thought liable to suffer from tank dead-space or highly turbulent (unstratified) conditions in the tank. Therefore, distributors were designed to fit onto the vertical ports, absorb the vertical momentum of the inlet flow and encourage the flow to enter the store in a horizontal plane, aiming to produce perfect plug flow.

The distributors consisted of 4 horizontal arms, each containing 64 1/16" (1.5mm) diameter holes. These were spaced so that each hole "served" a horizontal annulus of the same area. The method used to calculate the position of each hole is described in Appendix C.6. Initially it was thought that, in order to obtain an equal flowrate through each hole, it would be necessary to steadily increase the diameter of the holes the further down each arm they were. This would, in theory, have compensated for the increasing head loss of the water as it travelled further down the arm. A computer program was written to calculate the required diameter for each hole allowing for the various head losses, and this is described in Appendix C.6. The results of the program in fact showed that, since the flowrate is decreasing along each arm (i.e. after each hole), the head loss is decreasing sharply as well. The net effect is that even the last hole of the arm need only be about 1% larger in diameter than the first hole in order to have the same flow. This was considered negligible, and, as stated above, all the holes were drilled to the same diameter (1/16"). A photograph of one of the distributors attached to a dished end is shown in figure 3.19.

3.5 EXPERIMENTAL PROCEDURE

3.5.1 The basic test procedure

As discussed initially in Chapter 2, the basic test procedure used in this work was that prescribed in ASHRAE Standard 94-77 [ref.2]. The procedure can be summarised as follows :

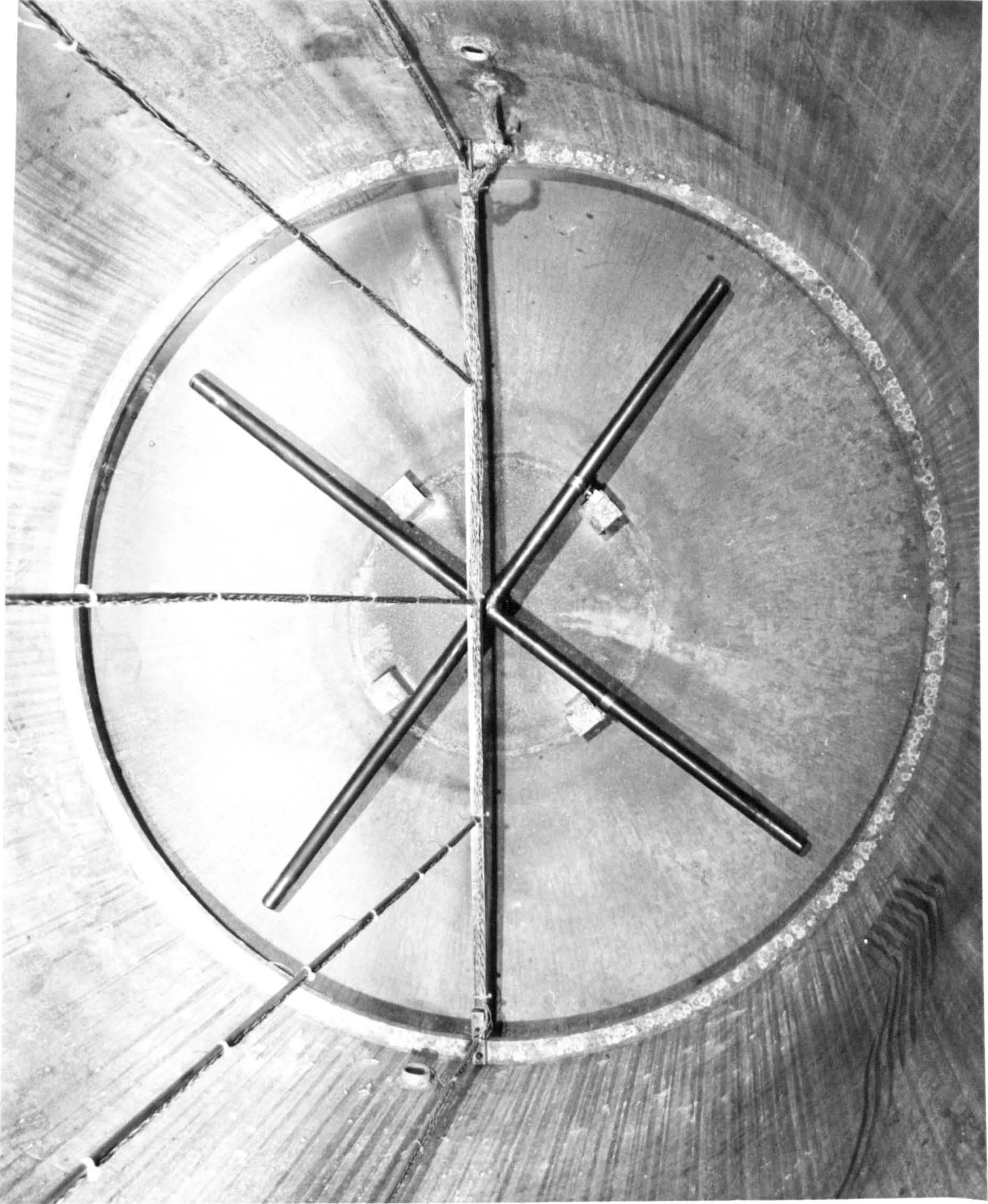


FIGURE 3.19 A DISTRIBUTOR IN POSITION ON ONE OF THE DISHED ENDS OF THE HOT WATER STORE

1. Water at a constant temperature was passed into the store until the store attained a uniform temperature at the desired initial value. (In this work, a nominal value for the initial temperature was set, and the experiment did not proceed until all the temperature readings inside the store were within $\pm 1^{\circ}\text{C}$ of this value. This ensured that the initial store temperature was approximately at its required value. The actual store temperature readings were always much more uniform than might be suggested by this relatively wide tolerance: the maximum and minimum readings were unlikely to be more than 0.3°C apart. In the analysis of the experiment, the actual initial store temperature rather than the nominal value was, of course, employed.)
2. The flowrate of the water entering the store was then adjusted to its required value for the experiment.
3. The inlet temperature and flowrate were maintained at their required initial values during a stabilisation period lasting 5 minutes.
4. The temperature of the water entering the store was increased (charge) or decreased (discharge) in as step-like a manner as possible to its required value.
5. The inlet temperature and flowrate were then maintained constant until a steady state was reached, i.e. the temperature of the water leaving the store was no longer changing with time.

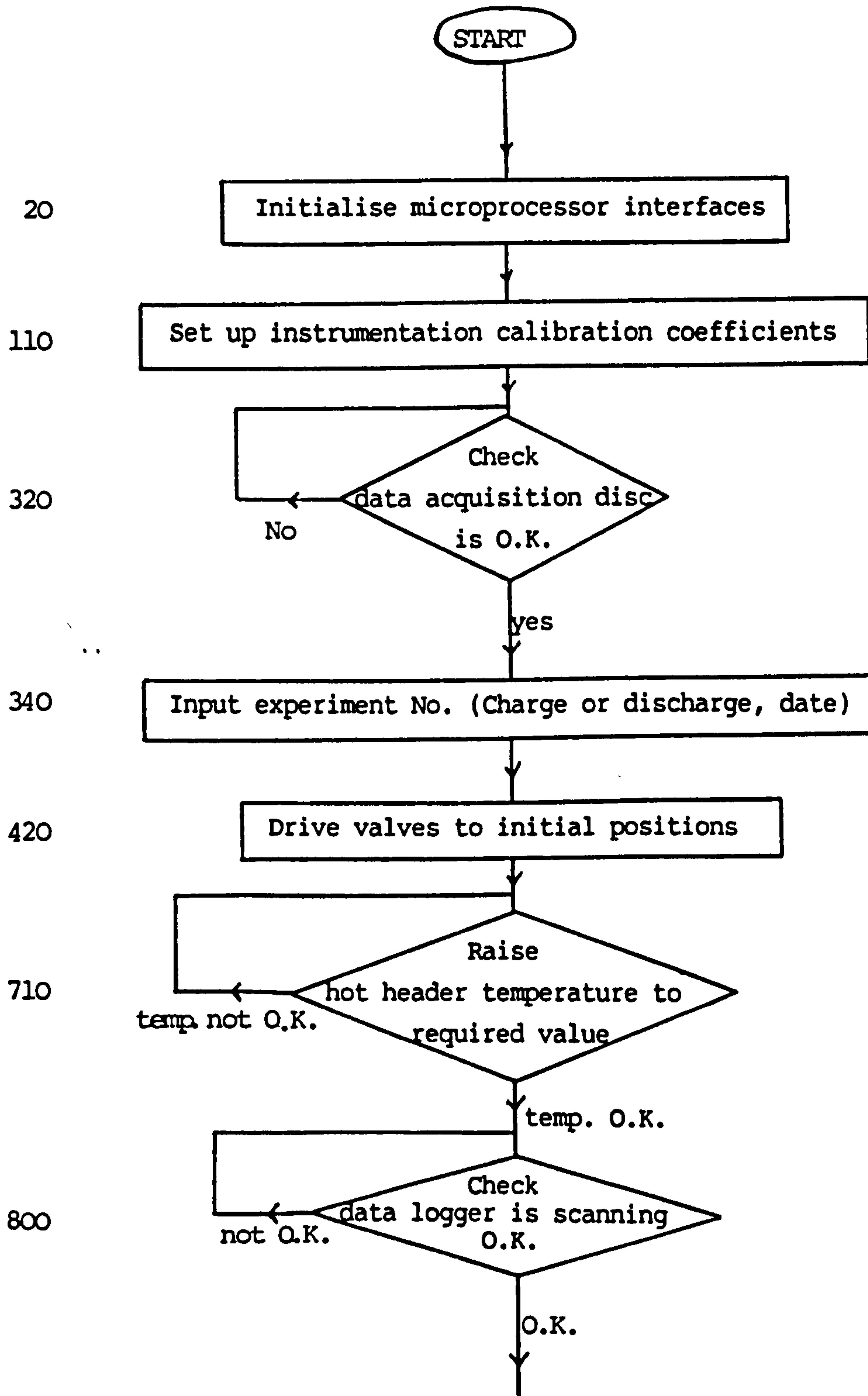
Illustrative inlet and outlet temperatures and the inlet flowrate during the 5 stages of the procedure are shown in figure 3.20.

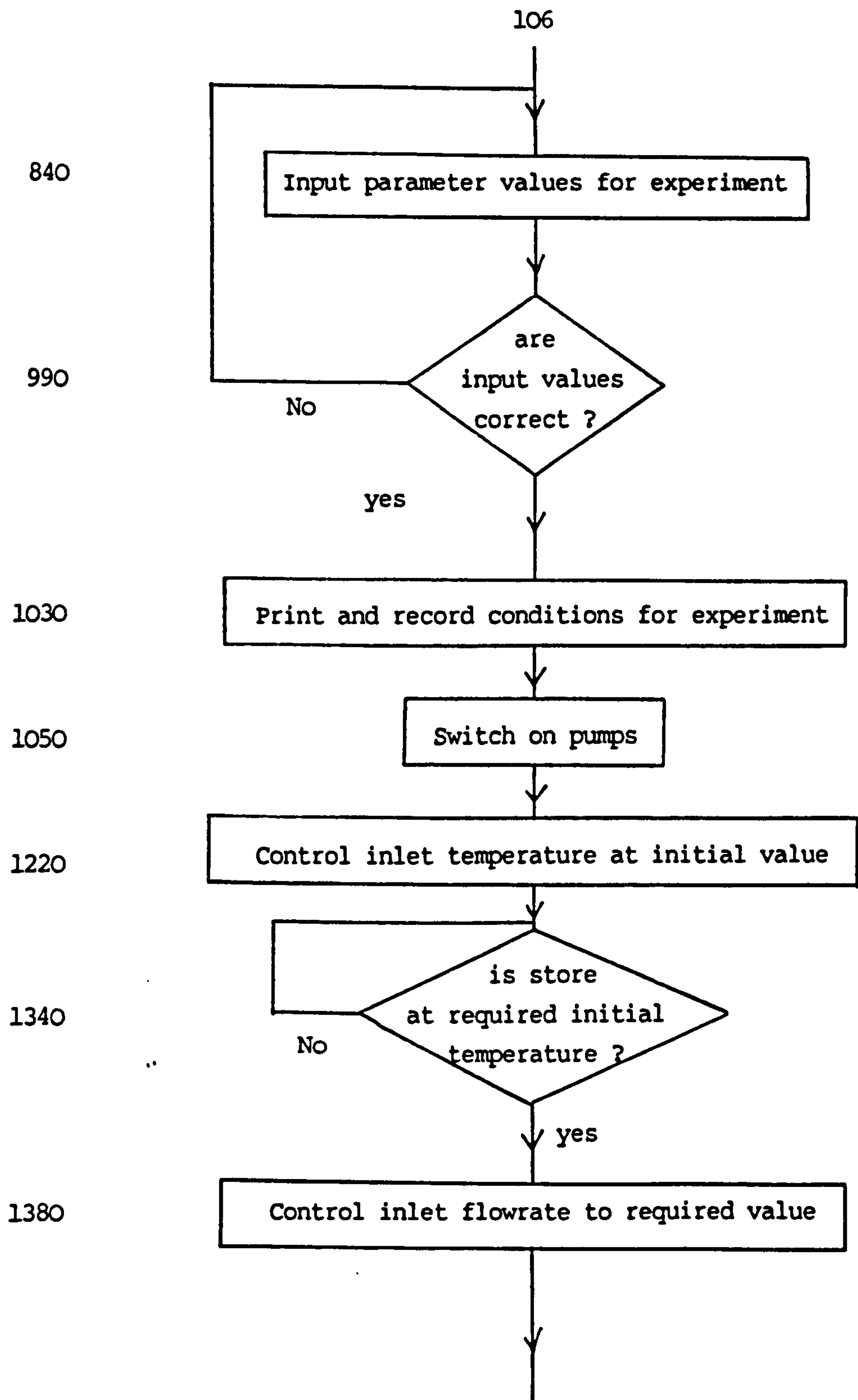
3.5.2 The monitoring and control computer program

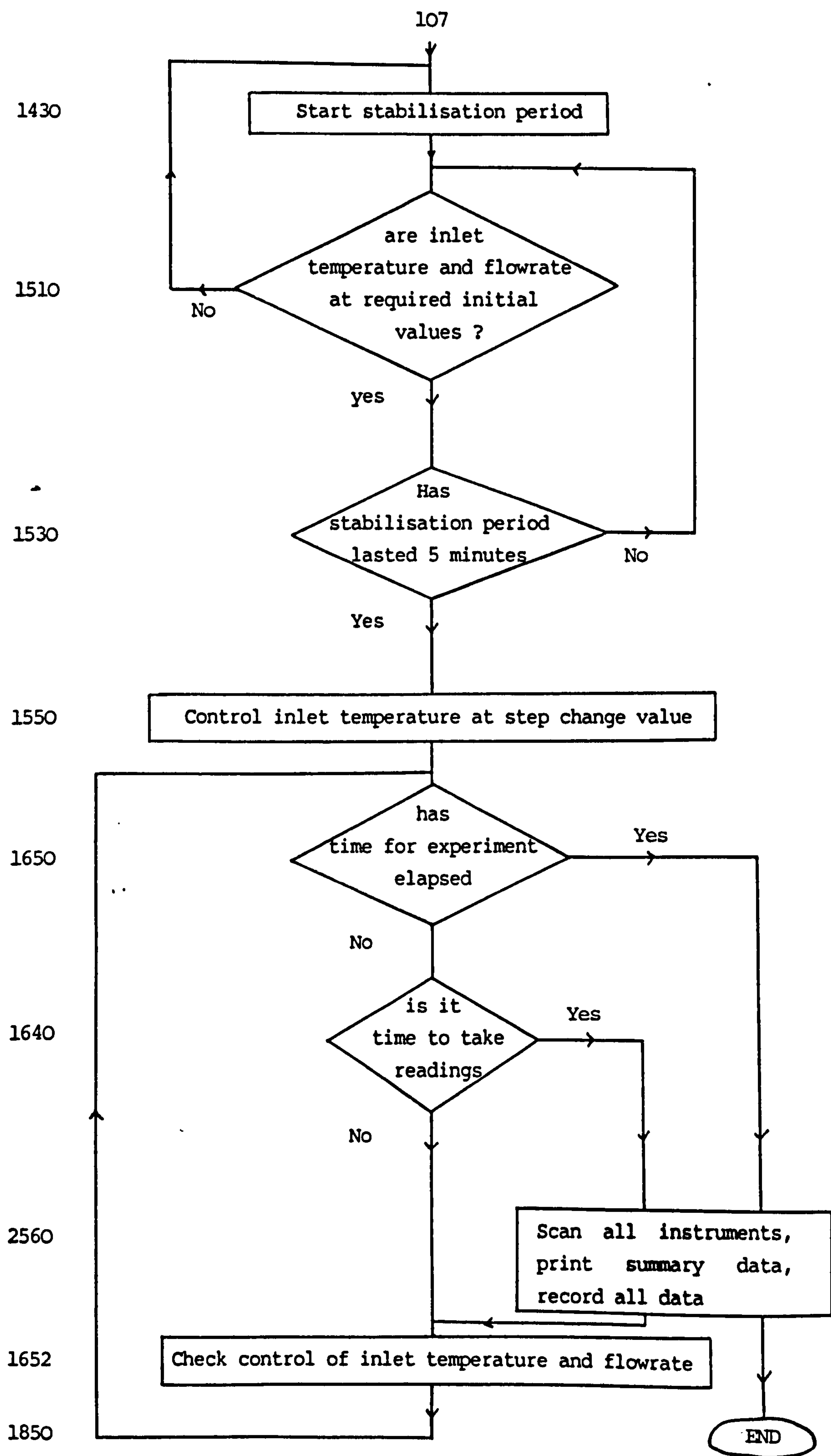
The key elements of the program which monitored and controlled the test facility during a step-input experiment are shown below in a flow diagram. A complete print-out of the program is given in Appendix C.7. Execution of the program was found to be too slow when it was

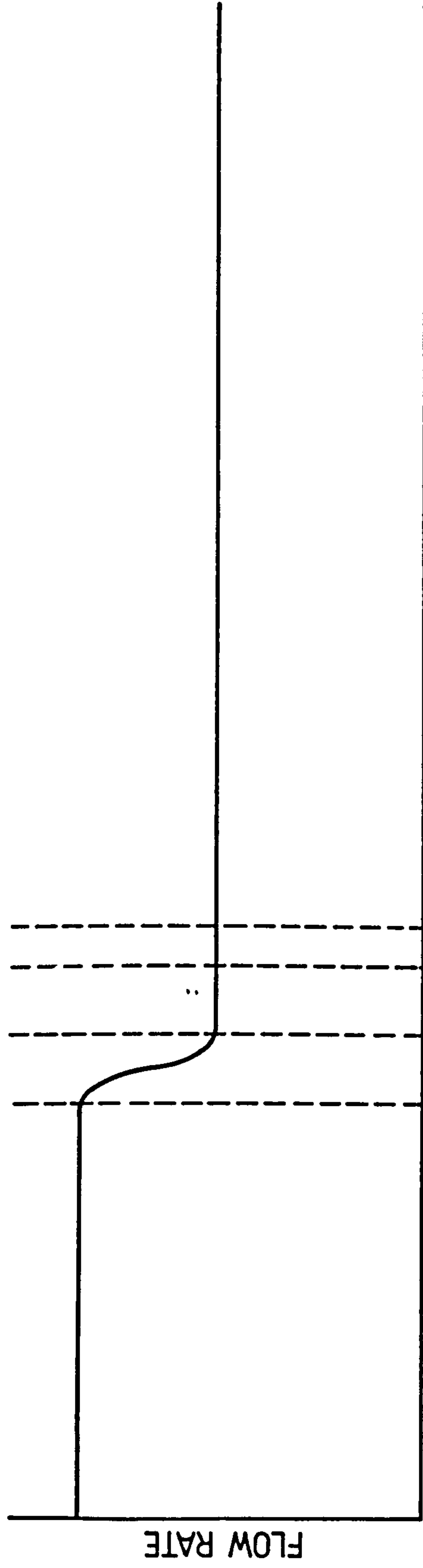
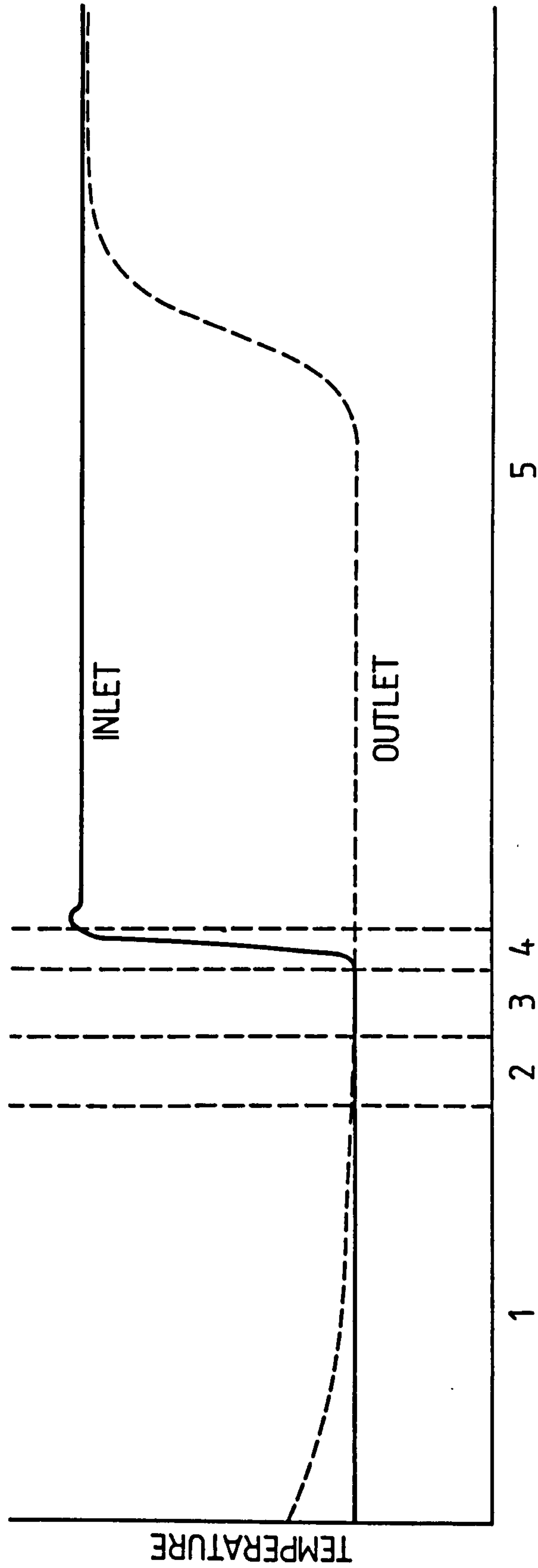
run in ordinary BASIC: the main monitoring and control loop of the program took up to 3 seconds to complete, causing a marked deterioration in the quality of control. The program was therefore run in a compiled form which was over three times faster.

line number
in program









EXPERIMENTAL STAGE (SEE TEXT)

FIGURE 3-20 SCHEMATIC REPRESENTATION OF A STEP INPUT CHARGE TEST PROCEDURE.

CHAPTER FOUR

EXPERIMENTAL RESULTS

4.1 INTRODUCTION

A large number of step input charge and discharge experiments were carried out for each of the 3 inlet port configurations considered, i.e. vertical ports, distributors and horizontal ports. Also, an experiment was performed under standstill conditions, where a temperature gradient was created in the store and then the temperatures inside the store were monitored over a prolonged period whilst the gradient decayed. This was in addition to the 13 day heat loss test described at the end of Chapter 3, which started with the whole store at the same temperature.

The potential range of step input experiments was set by the limits of operation of the test facility. These constrained the inlet flowrate to a value between 0.035 and 0.6 l/sec, the initial store temperature to between 10 and 90°C and the inlet temperature to between 10 and 90°C. The degree of stratification in the store and thus the store performance was clearly going to be largely determined by the relative magnitude of the momentum of the inlet flow into the tank and the counteracting buoyancy of the flow. The wide variety of experimental conditions available from the test facility ensured that almost all interesting combinations of these two properties of the flow could be assessed.

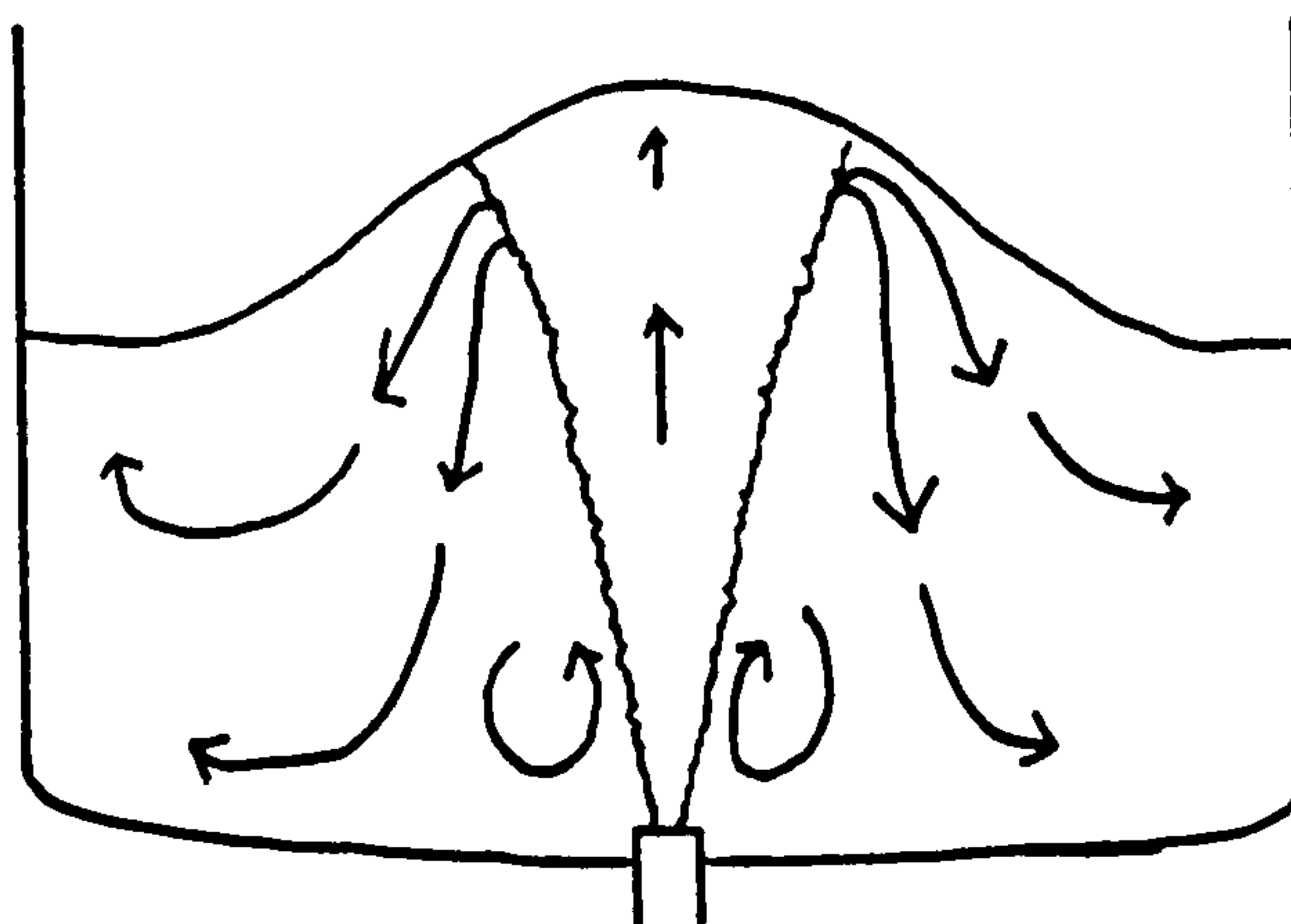
4.2 CHARGE AND DISCHARGE TEST RESULTS

4.2.1 Vertical Ports

4.2.1.1 A Qualitative Theory of Jet Penetration

The single vertical ports provided the perfect configuration for investigating the penetration of momentum driven jets with reversing buoyancy. Qualitatively the jet might be expected to behave as follows (taking the discharge case of a cold jet entering the bottom

of a tank of hot water as an example) : the momentum of the jet will be at its maximum at the port exit. As it enters the hot water, mixing will occur around the jet's periphery; the mixing will be enhanced by turbulence in the jet. The mixing reduces the upward momentum of the jet in accordance with momentum conservation law and eventually a position will be attained where the upward velocity is zero. The cold inlet water which has reached this point is heavier than its surroundings and therefore falls back, inducing further mixing with the incoming water and its own surroundings (see below).



It might be expected therefore that the jet would attain a greater height initially when its edges were interacting with a stationary environment, than subsequently when the upflow was mixing with heavier descending water.

It is an accepted practice (see for example Ref 3.1 or Ref 4.1) in the study of turbulent flow in particular geometries or circumstances, to employ dimensional analysis to derive relationships describing the flow, which can then be validated by experiment. In the case of the turbulent jet with reversing buoyancy, the flow from the small (almost point) source appears to be defined by just two parameters : the momentum flux, M_0 , and the negative buoyancy flux, B , (both divided by ρ to remove the dependance on mass). The dependence on molecular viscosity has been neglected since it does not usually effect the larger-scale eddies which are chiefly responsible for turbulent mixing (Ref. 4.2). The theory is therefore likely to be restricted to jets which have a Reynolds number at the nozzle greater than a few thousand. Under these conditions, it should be possible to express the penetration of the jet, z , as a function of these two terms, i.e.

$$z = C M_O^x B^y \quad \text{..(1)}$$

where M_O is defined by $M_O = \frac{\dot{m} u}{\rho} = \dot{V} u \quad \text{..(2)}$

and B is defined by $B = \frac{(\rho - \rho_1) \dot{V} g}{\rho} \quad \text{..(3)}$

Since the downward 'buoyancy' force on volume V of inlet water is

$$(\rho - \rho_1) V g$$

where ρ is the inlet water density and ρ_1 the tank water density.

Now $\frac{\rho - \rho_1}{\rho} = \frac{V_1 - V}{V} = \frac{\delta V}{V} \quad \text{..(4)}$

where δV is the volume change of a volume of water, V , changing density from ρ to ρ_1 .

By definition, the volume expansion coefficient, β , is given by

$$\beta(T) = \frac{1}{V} \frac{\partial V}{\partial T} \quad \text{..(5)}$$

So combining (4) and (5) gives $\frac{\rho - \rho_1}{\rho} = \beta \delta T$

So that (3) becomes $B = \beta \Delta T \dot{V} g. \quad \text{..(6)}$

Now, the volume flux, $\dot{V} = \pi r^2 u \quad \text{..(7)}$

where r is the exit port radius and u the exit velocity.

Thus substituting (7) into (2) and (6) gives, respectively,

$$M_O = \pi r^2 u^2 \quad \text{and} \quad B = \pi g \beta \Delta T r^2 u \quad \text{..(8)}$$

The dimensions of M_O are thus $[L]^4 [T]^{-2}$

and those of B are $[L]^4 [T]^{-3}.$

Applying dimensional analysis to (1) gives, therefore,

$$1 = 4x + 4y \quad \text{and} \quad 0 = -2x - 3y$$

$$\text{so that } y = -\frac{1}{2} \quad \text{and} \quad x = \frac{3}{4}$$

$$\text{and} \quad z = C M_o^{3/4} B^{-1/2} \quad \dots(9)$$

Inserting (8) in (9) gives

$$z = C \pi^{1/4} (g \beta \Delta T)^{-1/2} r^{1/2} u \quad \dots(10)$$

and using (7) gives

$$z = C \pi^{-3/4} \frac{\dot{m}}{\rho} r^{-3/2} (g \beta \Delta T)^{-1/2} \quad \dots(11)$$

4.2.1.2 Selection of a Correlating Parameter

A similar expression to (11) was tested experimentally by Turner (ref. 2.2), as described in Chapter 2, using heavy salt jets.

Equation (10) on the other hand can be rearranged to give

$$\frac{z}{r} = C \pi^{1/4} (g \beta \Delta T r)^{-1/2} u$$

so that,

$$\frac{z}{r} = C \pi^{1/4} (Ri)^{-1/2} \quad \dots(12)$$

$$\text{where} \quad Ri = \frac{g \beta \Delta T r}{u^2} \quad \dots(13)$$

Ri is the 'internal' Richardson number of the flow based on the initial properties (i.e. at the nozzle exit) of the negatively buoyant jet.

The use of the Richardson number as a correlating parameter under these circumstances is supported by the theory of similarity (ref.3.1) which generates it as a fundamental dimensionless group representing the ratio of buoyancy forces to inertial forces. It

can also be expressed in terms of the Grashof and Reynolds numbers of the flow,

$$\text{i.e.} \quad Ri = Gr Re^{-2} \quad \dots(14)$$

as long as the physical properties and characteristic lengths for both Gr and Re are taken to be those pertaining at the nozzle, i.e. the local conditions as opposed to the conditions of the bulk fluid in the tank. Thus

$$Gr = \frac{g \beta \Delta T r^3}{\nu^2} \quad \text{and} \quad Re = \frac{ur}{\nu} \quad \dots(15)$$

The Richardson number is also referred to as the Archimedes number (see, for example, ref. 4.3), particularly in the study of mechanically-ventilated rooms, although in this case the characteristic length is normally taken as a dimension of the room.

The purpose of a correlating parameter is to allow the scaling of experimental results as well as to provide a suitable way of validating a theoretical hypothesis such as equation (11). The parameter should therefore be capable of being used for the comparison of results from different experimental facilities and for the design of new equipment under untested conditions. In the present work, results will be tested in the form suggested by equation (11) which provides a means of plotting essentially raw experimental data. However on the same graphs, the dimensionless form of equation (11) given in equation (12) will also be shown.

The above analysis shows the dimensionless quantity (z/r) to be the correct one for scaling purposes although it lacks the physical meaning of say the ratio of z to the tank height, H . The analysis implies that the dimensions of the tank, i.e. the height and diameter, D , do not directly influence the jet penetration. In the present work the ratios of the tank dimensions to the nozzle diameter are high (i.e. $D/d = 54$ and $H/d = 77$) as should be the case generally for hot water storage applications. Under these circumstances the above assumption should hold. The limitations of the approach occur either when the ratio D/d is much smaller so that the flow regime

tends more to like that in a pipe, or when the ratio H/d is small enough for z to be of the same order or greater than H , so that short-circuiting between the inlet and outlet ports is possible.

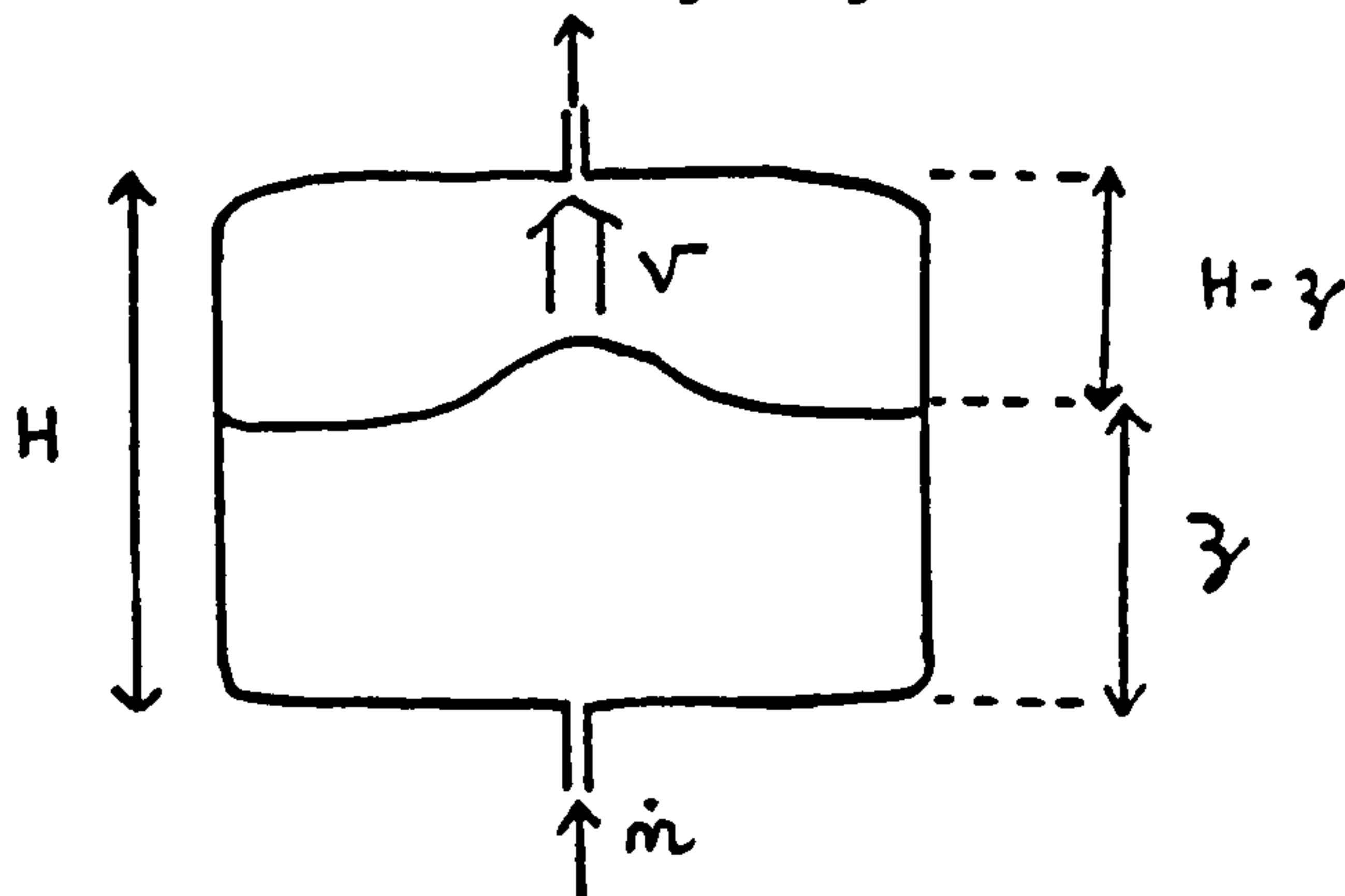
A major aim of this work is to approximate the behaviour of a water store by a one-dimensional numerical model, since this can be comfortably accommodated in system computer simulation models. The analysis of the step input experiments is based on the approximation that the temperature distribution is one-dimensional. The size of the error in making this approximation will become apparent in Chapter 5 when the temperature distributions predicted by a one-dimensional model are compared with the measured values. An important implication of this approximation is that the volume of the store up to the depth of penetration is assumed to be fully-mixed to the same temperature. However, the penetration of a jet into a store clearly creates a two-dimensional temperature distribution, although natural buoyancy will always be acting to correct this unstable situation. In general the extent, in time and space, of this two-dimensionality will depend on scale factors such as the tank-to-nozzle diameter ratio (D/d) and the fill time of the store.

[Incidentally, in the opinion of the author, some other research in this field is flawed by the use of an inappropriate correlating parameter. Lavan and Thompsen (ref. 2.17) used Re and Gr , with Gr based on the tank diameter. This definition means that equation (13) is no longer consistent with equations (14) and (15), so that equation (12) cannot be expected to hold. The analysis of Wood et al (ref. 2.20) suffers from the same shortcoming. Sliwinski (ref. 2.19) presented a correlation in the form of equation (12) using the definition of Ri given in equation (13) except that he used (inadvisably) the distance between the inlet and outlet ports as the characteristic length instead of the nozzle radius, r . The results of the present work are compared with other published results in section 4.2.3.2.]

4.2.1.3 Measurement of the Depth of Jet Penetration

Turner carried out his work on a small (45cm square by 140cm deep) transparent Plexiglass tank, and was therefore able to observe the

inlet jets and measure by visual means their depth of penetration. In the present work, flow visualisation was highly impracticable, given the size and construction of the storage vessel. Inferring the depth of penetration from the temperature distribution inside the vessel was also unsatisfactory. First of all, a much finer mesh of thermocouples would have been required to produce reasonable resolution; secondly, the phenomenon of the jet initially penetrating further than its 'steady-state' value would have made interpretation of the temperature distributions very difficult. It was therefore decided to determine the penetration in a slightly roundabout way : by measuring the time taken from the start of an experiment for there to be a significant change in the temperature at the store outlet. This is illustrated in the following diagram :



To simplify the slightly non-cylindrical geometry of the store, the overall store height is defined by

$$H = \frac{V}{A} \quad \text{..(16)}$$

where V is the store volume and A is the cross-sectional area of the main body of the store.

Also, the plug flow velocity is defined as

$$v = \frac{\dot{m}}{\rho_{in} A} \quad \text{..(17)}$$

The time for the outlet temperature to change, defined here as the reaction time, t_R , is given by :

$$t_R = \frac{H - z}{v}$$

$$\text{so that } z = H - v t_R = H - \frac{\dot{m} t_R}{\rho_{in} A} \quad \dots(18)$$

Care had to be exercised in specifying t_R from an examination of the sequence of outlet temperatures, so as to avoid the possibility of confusing a heat loss or conduction induced change with a jet penetration induced change. Changes caused by heat losses or conduction occurred very gradually and so it was possible to adopt the following criterion for determining t_R : it was the time at which the outlet temperature first changed from its previous value, with the additional proviso that on the next scan its value changed again. (The outlet temperature was recorded to one decimal place, so the minimum observable change was of 0.1°C .)

4.2.1.4 Vertical Ports : depth of penetration results

A total of twenty-two good quality step input experiments were carried out using the vertical ports. Many other experiments had to be rejected owing to problems with control of the inlet temperature or flowrate, with the monitoring system or computer, with the electrode boiler or with myriad other gremlins which beset all such laboratory work.

The initial store temperature and the inlet temperature and flowrate of each experiment are listed in Table 4.1. The 'reaction time' and performance coefficient (see section 4.2.1.6) for each experiment are also given. The expression from equation (11)

$$\frac{\dot{m}}{\rho} r^{-3/2} (g \beta \Delta T)^{-1/2}$$

was christened the Turner parameter, in acknowledgement of its origins. It was calculated for each experiment, along with the depth of penetration using (18), and these data are plotted in the graph in Figure 4.1. The graph also has axes graded according to the dimensionless parameters z/r and $Ri^{-1/2}$ in equation (12).

PORT	EXPT NO.	INLET TEMP. C	INITIAL TEMP. C	FLOW RATE Kg/S	PERF. COEFF. %	REACTION TIME HOURS

VERTICAL	21.160781	27.7	58.4	0.399	91.1	1.609
VERTICAL	11.170781	66.0	28.0	0.196	96.8	3.390
VERTICAL	11.220781	31.1	27.1	0.448	93.4	1.468
VERTICAL	21.210781	26.7	63.6	0.249	93.8	2.595
VERTICAL	22.220781	23.6	31.2	0.449	71.1	0.775
VERTICAL	21.240781	21.8	73.4	0.160	93.3	4.120
VERTICAL	21.290783	21.7	26.5	0.600	61.7	0.000
VERTICAL	21.280783	25.8	37.8	0.499	75.5	0.810
VERTICAL	21.090883	25.1	29.5	0.241	79.2	2.226
VERTICAL	21.040883	23.0	30.0	0.240	81.8	2.327
VERTICAL	21.280783	38.6	59.7	0.596	83.9	0.911
VERTICAL	21.020883	70.2	90.7	0.588	87.5	0.972
VERTICAL	21.270783	60.3	80.7	0.491	86.5	1.132
VERTICAL	21.170883	22.8	30.4	0.139	86.6	4.263
VERTICAL	21.020883	54.0	69.6	0.237	90.7	2.587
VERTICAL	21.160883	45.3	66.1	0.219	91.7	2.883
VERTICAL	21.080883	40.8	62.3	0.159	91.4	4.078
VERTICAL	21.050883	66.7	87.7	0.137	93.3	4.671
VERTICAL	21.160883	30.3	45.5	0.070	88.3	8.420
VERTICAL	21.080883	29.3	41.9	0.050	84.0	11.511
VERTICAL	21.150883	66.7	88.3	0.059	91.1	9.865
VERTICAL	21.030883	30.0	53.7	0.050	87.2	11.460
VERTICAL	21.100883	63.6	85.3	0.039	84.3	12.783

Table 4.1 The step input conditions for the experiments with vertical ports.

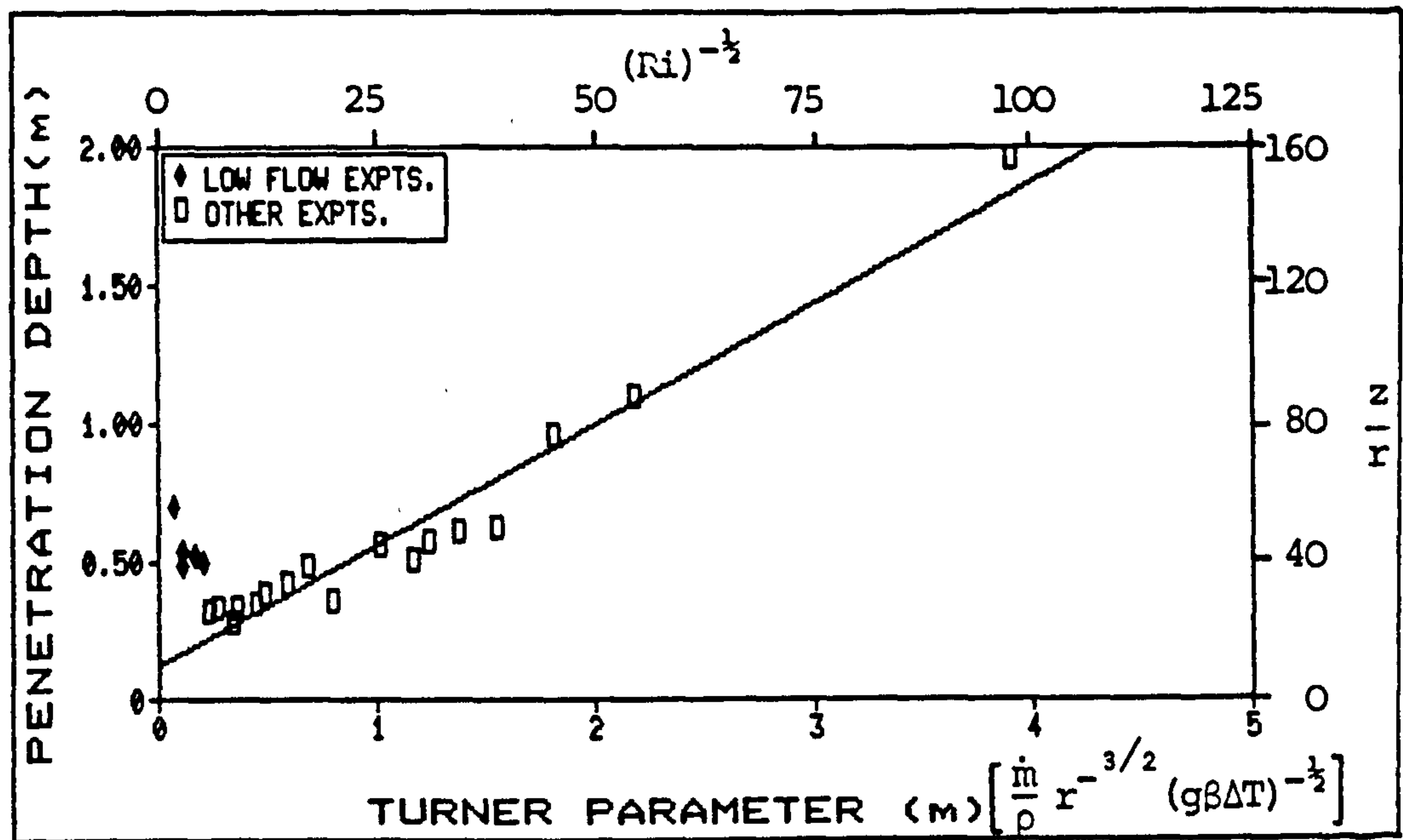


FIGURE 4.1 THE CORRELATION BETWEEN PENETRATION DEPTH (z) AND TURNER PARAMETER FOR THE EXPERIMENTS WITH VERTICAL PORTS.

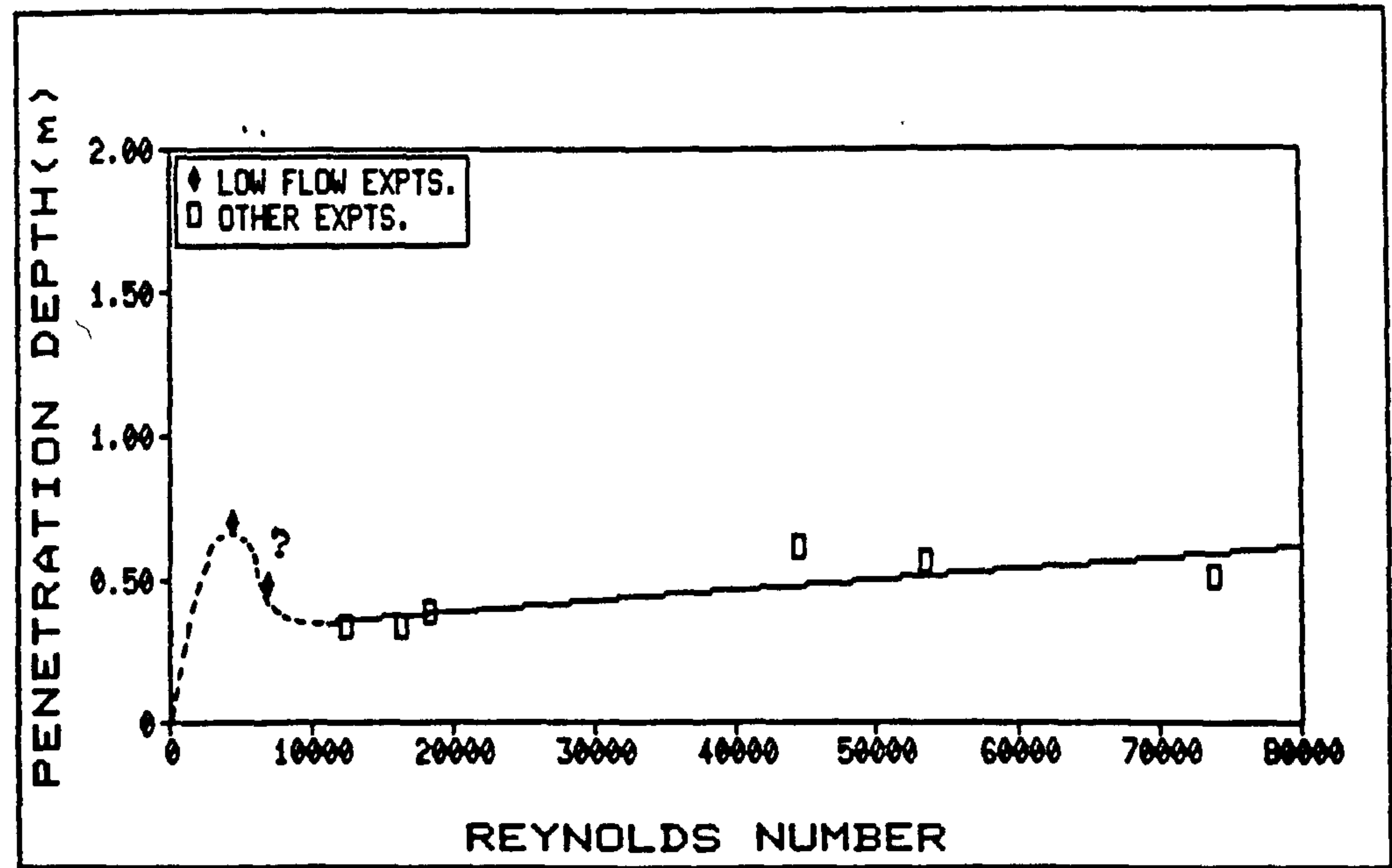


FIGURE 4.2 THE PENETRATION DEPTH PLOTTED AGAINST INLET REYNOLDS NUMBER FOR VERTICAL PORT EXPERIMENTS WITH $\Delta T \sim 21^{\circ}\text{C}$.

The graph strongly vindicates the validity of equations (11) and (12) for all experimental conditions except those where the Turner parameter is very small (i.e. $< 0.2m$ or $Ri > 0.04$). These represent experiments with very low flowrates (i.e. < 0.07 kg/sec) and are denoted by a different symbol. The straight line best-fitting the other points is given by :

$$z = 0.128 + 0.442 \frac{\dot{m}}{\rho} r^{-3/2} (g \beta \Delta T)^{-1/2}$$

Logically one might expect the line to pass through the origin, however the vertical ports intruded into the vessel by 50mm (0.05m), and equation (18) was reached by assuming the vessel to be a perfect cylinder.

The gradient of the line is more significant; it yields a value for the constant C in equation (11) of 1.04. This is reasonably close to the value of 1.85* quoted by Turner for his results and that of 1.33 determined by the researchers at Colorado State University (CSU) for their results (Ref.2.41). Closer agreement is not to be expected since Turner used salt jets in a square cross-section tank and the CSU experiments were conducted on an unpressurised (i.e. free water surface) tank.

4.2.1.5 Penetration at Low Inlet Flowrates

The results in figure 4.1. clearly demonstrate that the depth of penetration is much greater at low inlet flowrates than that predicted by equation (11). The qualitative explanation for this is that the flow at these low flowrates is no longer fully turbulent as it leaves the inlet. This would mean that the rate of mixing between the inlet flow and the water in the tank would be reduced, so that the jet's momentum was dissipated more slowly and the jet could

*FOOTNOTE

Examination of figure 2.1 from which Turner is believed to have derived this value, suggests that this value is in error and that, in fact, a value of C~3.1 is correct for Turner's results.

penetrate further into the vessel. The critical Reynolds number for fully turbulent flow in a pipe can vary between 2,000 and 40,000, the latter case applying to quiescent inlet conditions, and depends on a number of factors - for example, system geometry, surface roughness and free stream turbulence (Ref.3.1). A sharp cut-off point is therefore unlikely. A clearer insight into the effect of turbulence on jet penetration can be gained by removing one of the key variables in equation (11), that is ΔT . Eight of the experiments had inlet-initial temperature differences within 0.6°C of 21.1°C . Apart from a small influence of the value of β , the volume expansion coefficient, the penetration depth for these experiments should be directly proportional to the volume flow rate $\dot{V}$ and thus the inlet flow velocity at the port exit. This is closely related to the pipe Reynolds number, again neglecting the variations in the values for density and viscosity at the inlet temperatures occurring in these eight experiments (i.e. $38.6^{\circ}\text{C} \rightarrow 70.2^{\circ}\text{C}$). The penetration depth for these eight experiments is plotted against Reynolds number in Figure 4.2. In general terms, it confirms the above hypothesis : for $\text{Re} > 10,000$ the results sit convincingly close to a straight line; for $\text{Re} < 10,000$ there is a large jump in the depth of penetration.

With hindsight, it would have been attractive, from an academic standpoint, to have conducted a series of experiments (maybe up to 20) in which the initial store temperature and inlet temperature were always the same and only the inlet flowrate was varied. This would remove any influence of the values of the physical properties of the water. Such tests would generate a more complete picture than is provided by Figure 4.2 but it is unlikely the above general conclusions would be altered.

At the limit of $\text{Re} \rightarrow 0$, common sense dictates that the penetration of a laminar jet with reversing buoyancy will also tend to zero. This hypothesis could not be confirmed in the present study as it was not possible to run experiments at low enough flowrates on the test facility. Despite an extensive search, no reference to this situation was found in the literature. The implications of such a trend are indicated on figure 4.2 but it must be emphasized that this is purely supposition.

4.2.1.6 Definition of a Thermal Performance Coefficient

It is convenient to be able to quantify the thermal performance of a water store subjected to a step input test. The obvious figure of merit is the amount of energy accepted by the store in a certain period of time, in the case of a charge experiment, and the amount of energy delivered by the store in a certain period of time, in the case of a discharge experiment. Taking, as an example, a charge experiment, and ignoring all heat losses for the time being, the energy stored up to time t is given by :

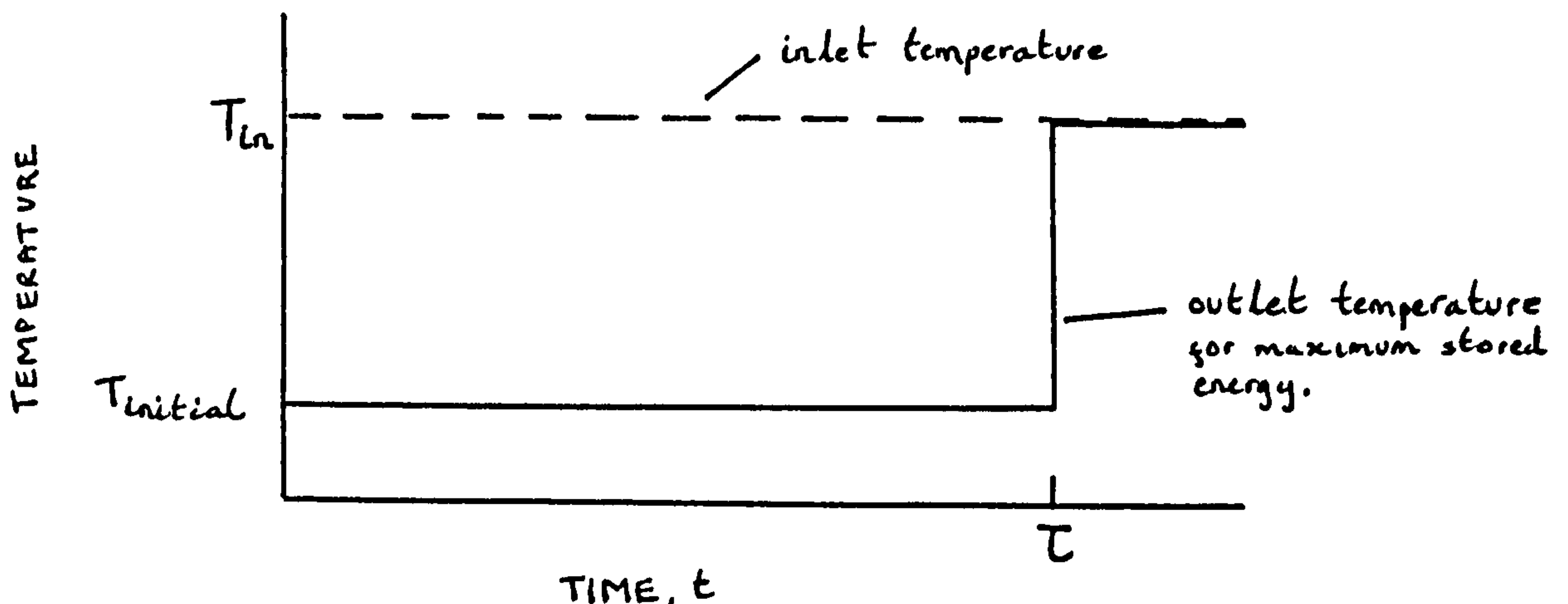
$$E_c = \int_0^t \dot{m} C_p (T_{in} - T_{ex}) dt \quad \dots(19)$$

To produce a useful performance coefficient, $\eta(t)$ (i.e. one that ranges from 0 to 100%), it is appropriate to divide the actual energy stored by the theoretical maximum energy stored, E_{max} .

$$\text{Thus } \eta(t) = \frac{E_c}{E_{max}} \quad \dots(20)$$

$$\text{where } E_{max} = \int_0^t \dot{m} C_p (T_{in} - T_{initial}) dt \quad (t \leq \tau) \quad \dots(21)$$

Equation (21) only applies until the time τ , defined as the charge period, when the store is fully charged. It basically states that maximum energy is stored if the outlet temperature remains at its initial value right up until time τ , as illustrated below.



The theoretical storage capacity (TSC) of the store when its temperature is raised from T_{initial} to T_{in} is given by

$$\text{TSC} = \sum M C_p (T_{\text{in}} - T_{\text{initial}}) \quad \text{..(22)}$$

By definition of E_{max} , one can also state

$$\begin{aligned} \text{TSC} = E_{\text{max}}(\tau) &= \int_0^{\tau} \dot{m} C_p (T_{\text{in}} - T_{\text{initial}}) dt \\ &= \dot{m} C_p (T_{\text{in}} - T_{\text{initial}}) \tau \quad \text{..(23)} \end{aligned}$$

Inserting (22) in (23) gives

$$\tau = \frac{\sum M C_p}{\dot{m} C_p} \quad \text{..(24)}$$

Since the thermal capacity of most water stores is nearly all made up by the water itself, τ is simply a little greater than the fill time of the store ($M_w/\dot{m}$). The extra allows for the storage vessel itself and its insulation, etc. to be fully charged.

It is most appropriate to compare η values at time τ , when, from (19), (20) and (23),

$$\eta(\tau) = \frac{\int_0^{\tau} \dot{m} C_p (T_{\text{in}} - T_{\text{ex}}) dt}{\text{TSC}} \quad \text{..(25)}$$

For a perfectly stratified store,

$$T_{\text{ex}}(t) = T_{\text{initial}}(t) \quad (0 \leq t \leq \tau), \text{ and } \eta(\tau) = 1.$$

If complete short-circuiting occurs,

$$T_{\text{ex}}(t) = T_{\text{in}}(t), \text{ and } \eta(\tau) = 0.$$

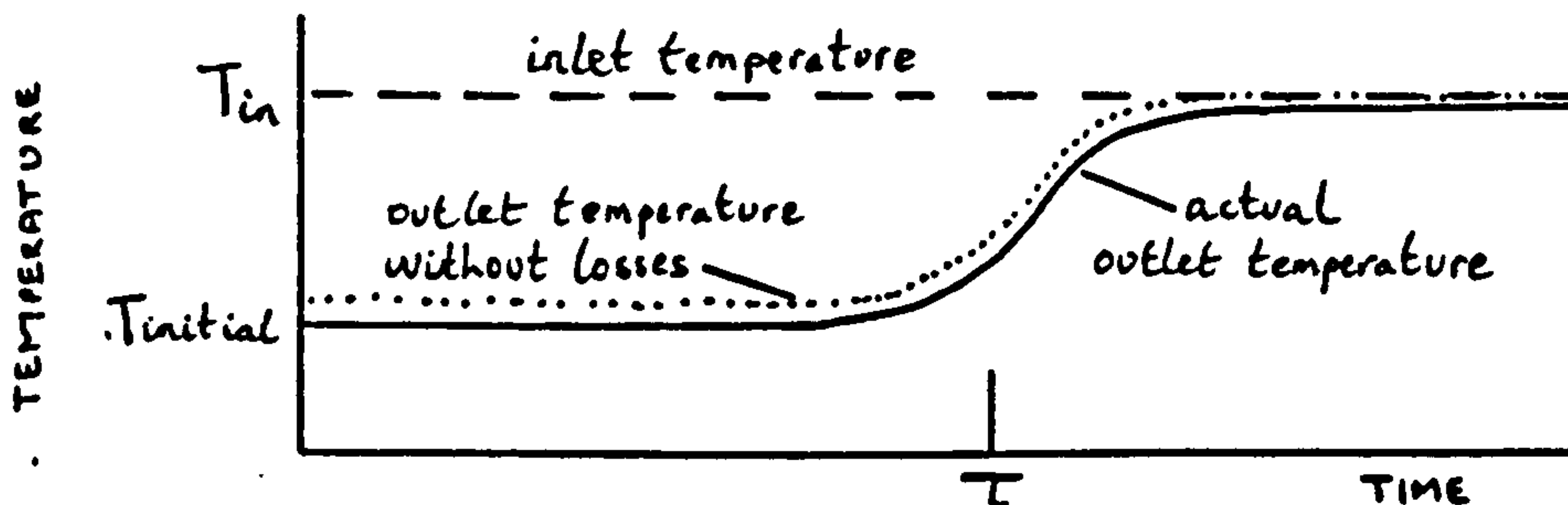
A third limit of operation occurs when the store is fully-mixed at all times, but no short-circuiting occurs. This circumstance is analysed in Appendix D.1, where it is shown that

$$\eta(\tau) = 0.632$$

These limits of operation, which apply equally to discharge experiments, are associated with extreme values for the depth of jet penetration. Thus a perfectly stratified store should have $z = 0$, corresponding to $\eta(\tau) = 1$, a fully mixed store should have $z = H$, corresponding to $\eta(\tau) = 0.632$, and a store with short circuiting should have $z > H$, corresponding to $\eta(\tau) < 0.632$.

4.2.1.7 Allowing for Heat Losses

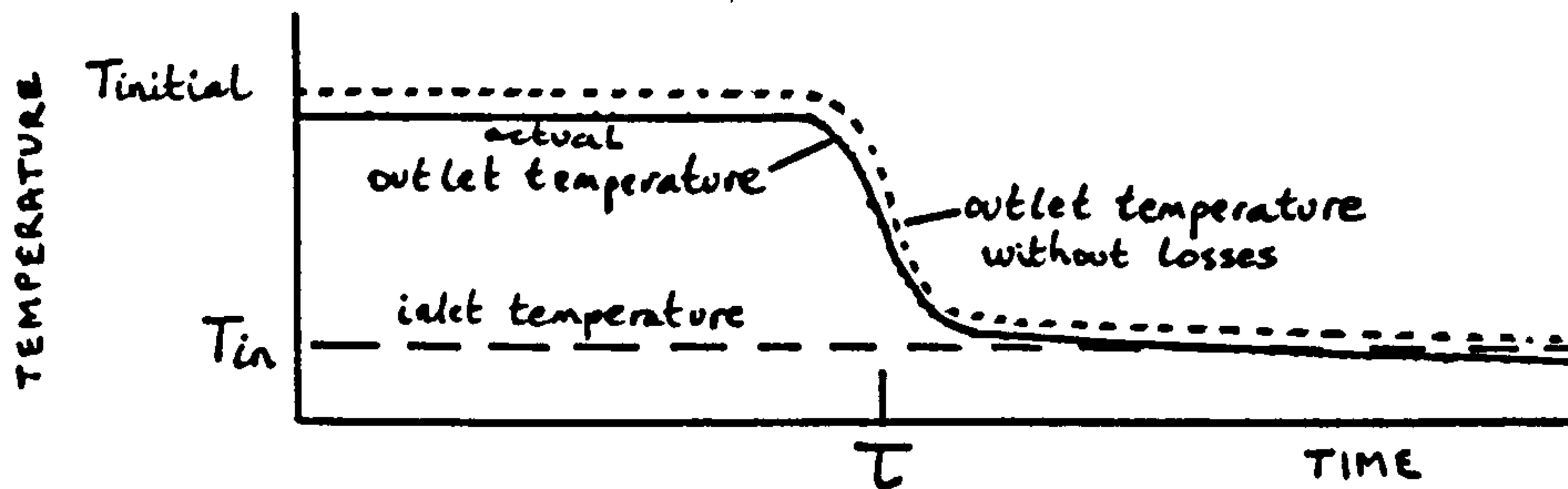
Heat losses mean that the temperature inside a store and hence the outlet temperature are always lower than they would be without heat losses, during both charge and discharge experiments. Thus during a charge experiment, the difference between the inlet and outlet temperature is greater due to heat losses, as shown below:



It is apparent from equation (25) that, if this were not taken into account, $\eta(\tau)$ would be higher the greater the heat losses. Since heat losses during charge and discharge situations are essentially simply determined by the thickness of insulant on the store, it is most interesting to try to discount their influence and determine only the effect of the port configuration, the flow conditions, etc., on the stratification in the store. To this end therefore, the heat losses are measured and subtracted from the apparent energy stored when calculating η . This is thus an attempt to quote η values which would occur if there were zero heat losses. Thus equation (25) becomes (for charge experiments) :

$$\eta(\tau) = \frac{\int_0^\tau \dot{m} C_p (T_{in} - T_{ex}) dt - \int_0^\tau (UA)_{s,a} (T_s - T_a) dt}{TSC} \quad ..(26)$$

During a discharge experiment, the outlet temperature is again lower than it would be without heat losses, but, as shown below, this decreases the difference between the inlet and outlet temperatures.

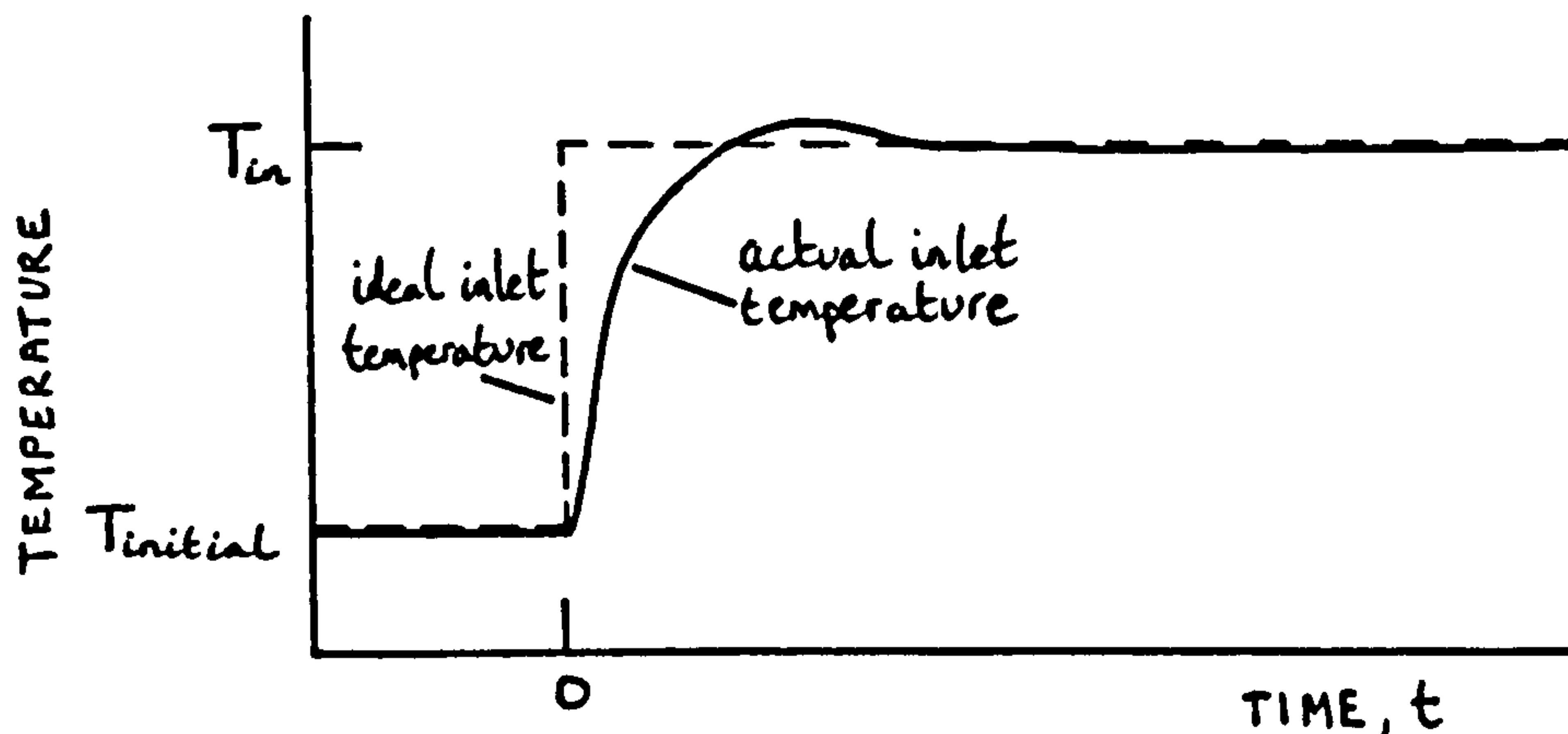


The heat losses in the discharge case are therefore added to the apparent energy delivered when calculating η , and for discharge experiments equation (25) becomes

$$\eta(\tau) = \frac{\int_0^\tau \dot{m} C_p (T_{in} - T_{ex}) dt + \int_0^\tau (UA)_{s,a} (T_s - T_a) dt}{TSC} \quad ..(27)$$

4.2.1.8 Allowing for Imperfect Step Inputs

Equations (26) and (27) assume that at time $t = 0$ the inlet temperature to the store changes instantaneously from $T_{initial}$ to T_{in} . This is impossible to achieve in practice and allowance has to be made for imperfect step inputs, of which a typical charge experiment example is illustrated below.



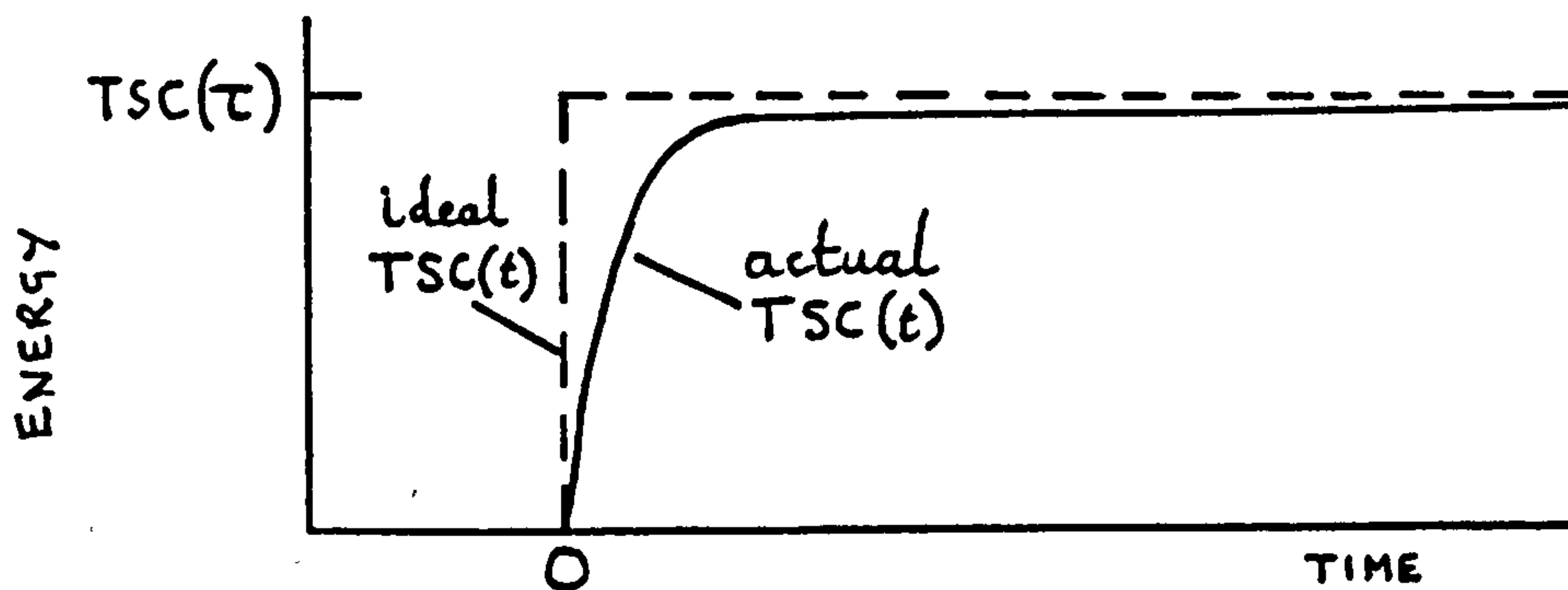
Two factors are affected by an imperfect step :

(i) Theoretical Storage Capacity (TSC)

The calculation of TSC in equation (22) assumes $T_{in}(t) = T_{in}$ for $t \geq 0$. To account for an imperfect step it is necessary to use :

$$TSC(t) = \sum M C_p \frac{\int_0^t (T_{in} - T_{initial}) dt}{t} \quad \dots(28)$$

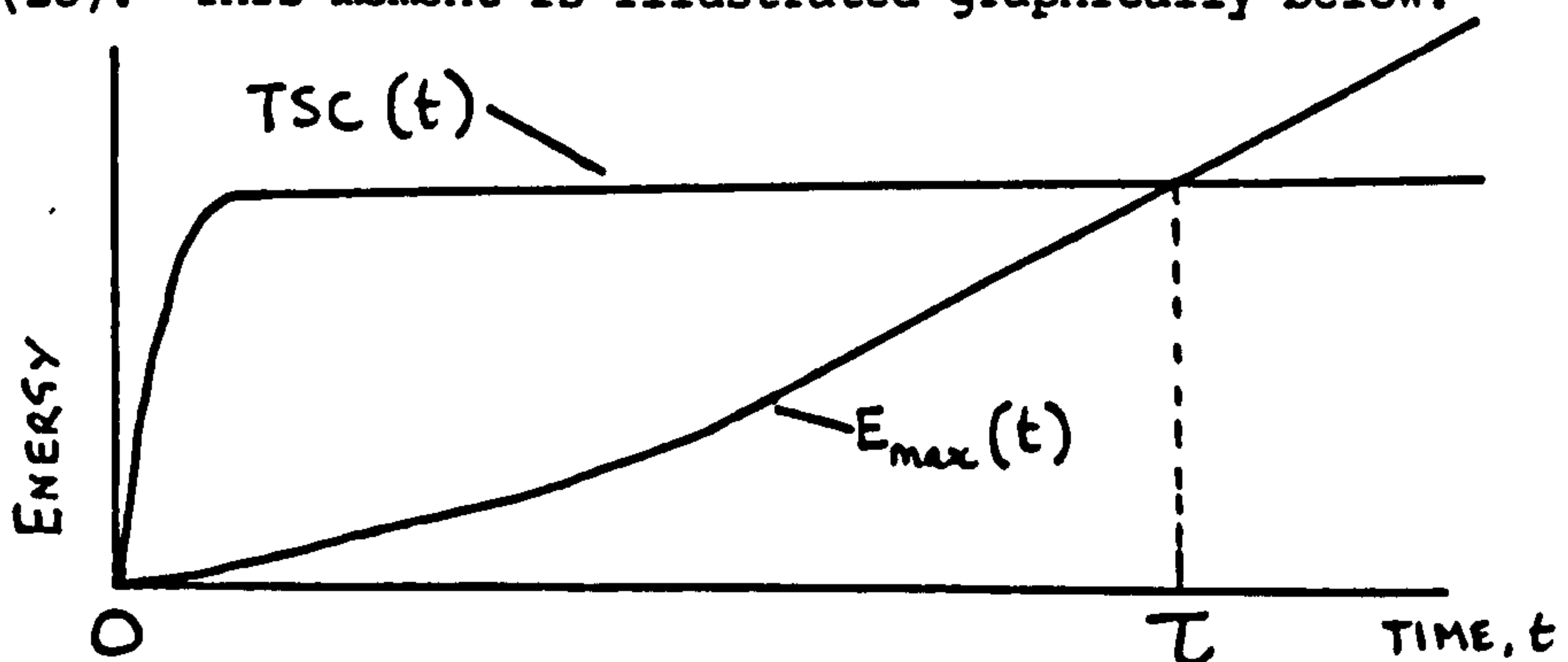
This means that $TSC(t)$ is always increasing during an experiment asymptotically towards its ideal value, as shown here:



The further into the experiment, the less the significance of the initial imperfection.

(ii) Charge (or discharge) Period (τ)

Equation (24) cannot be used in practice as it only applies for an ideal step. A practical definition of τ is the time when the maximum possible stored energy, E_{max} , defined in (21), equals the actual TSC(t), defined in (28). This moment is illustrated graphically below:



Note that with an ideal step $E_{\max}(t)$ would be a straight line which rises faster initially than its actual value does. This definition of τ ensures that the true maximum value of $\eta(\tau)$ is indeed 1.

4.2.1.9 Vertical Ports : Performance Coefficient Results

Values of $\eta(\tau)$ were calculated for each experiment using equations (26) and (27), and the above definitions of TSC and τ , and they are shown in Table 4.1. The data for $\eta(\tau)$ is plotted against the Turner parameter and the non-dimensional $(\text{Ri})^{-1/2}$ in Figure 4.3. A close correlation is evident for the data not at low flowrates. At high values of the Turner parameter ($> 1\text{m}$ or $\text{Ri} < 0.0016$ i.e. where the momentum forces dominate buoyancy forces) the performance coefficient falls off almost linearly as the parameter (and dominance of momentum) increases. At lower values of the parameter, where buoyancy is more important, the performance coefficient attains its peak value (about 96%). However at very low values ($< 0.2\text{ m}$ or $\text{Ri} > 0.04$) the performance coefficient drops sharply. This region comprises very low flow rate experiments, where the inlet flow is no longer fully turbulent and so penetrates the tank further. The experiments also take a long time to complete, enabling conduction to degrade somewhat initially strong stratification in the tank.

The correlation between performance coefficient and depth of penetration is shown in Figure 4.4. An axis is provided on the figure showing the ratio of the penetration depth to the tank height. Two theoretical points are plotted on the graph, representing the perfectly stratified tank ($z = 0$, $\eta(\tau) = 100\%$) and the fully-mixed tank ($z = H$, $\eta(\tau) = 63.2\%$). The experimental points lie between these two extremes, with very little scatter. This graph confirms that once the depth of the mixed region at the inlet port is defined, then so is the performance coefficient, and also therefore, by implication, the store outlet temperature during the experiment. It is noteworthy that this maxim holds even for the experiments with very low flow rates - for a given depth of penetration there is no difference between the performance coefficients of experiments with low flow rates and those of experiments with higher flow rates. This suggests that the influence of conduction mentioned above, is not enough to depress the performance coefficients significantly over the

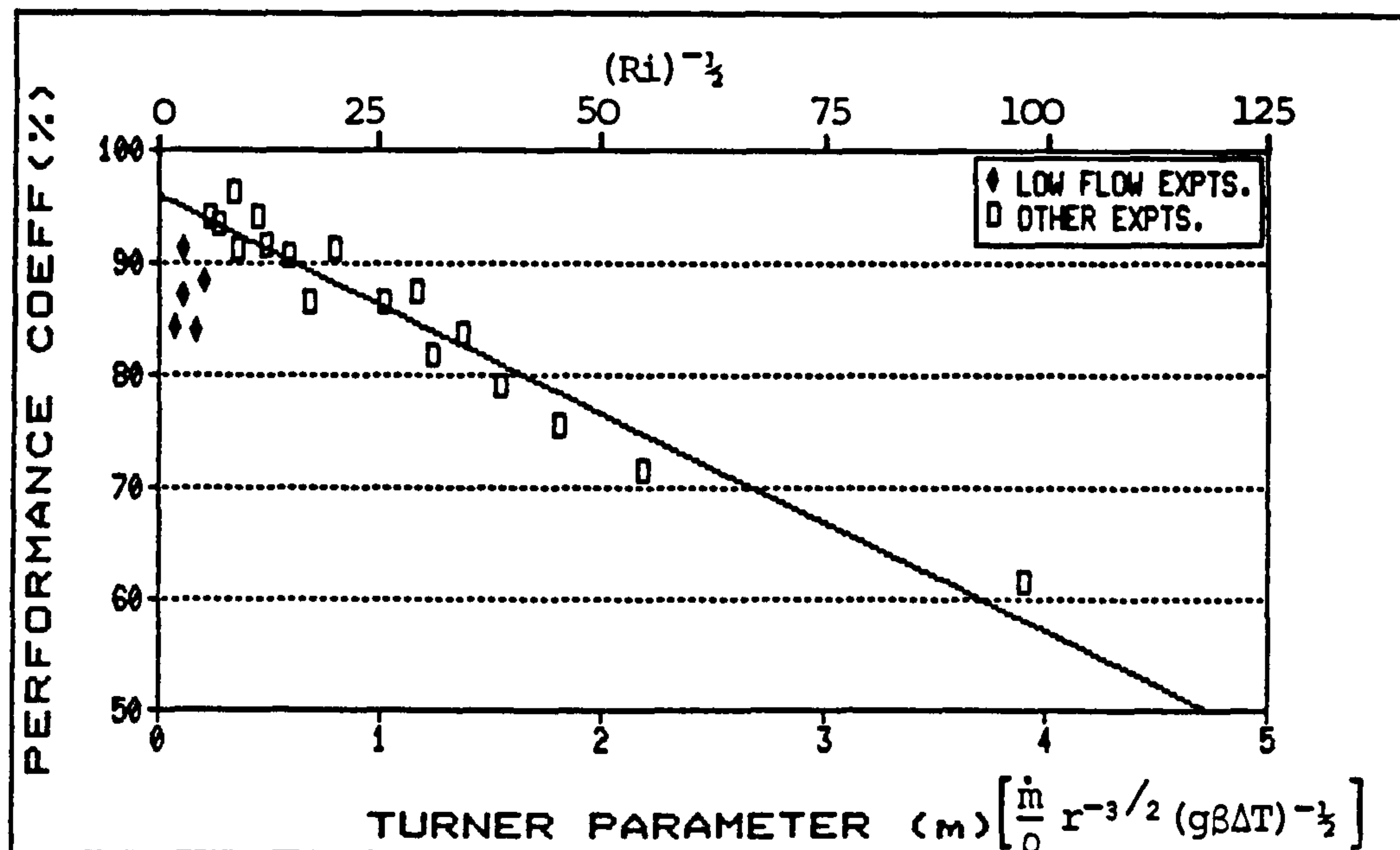


FIGURE 4.3 THE CORRELATION BETWEEN PERFORMANCE COEFFICIENT AND TURNER PARAMETER FOR THE EXPERIMENTS WITH VERTICAL PORTS.

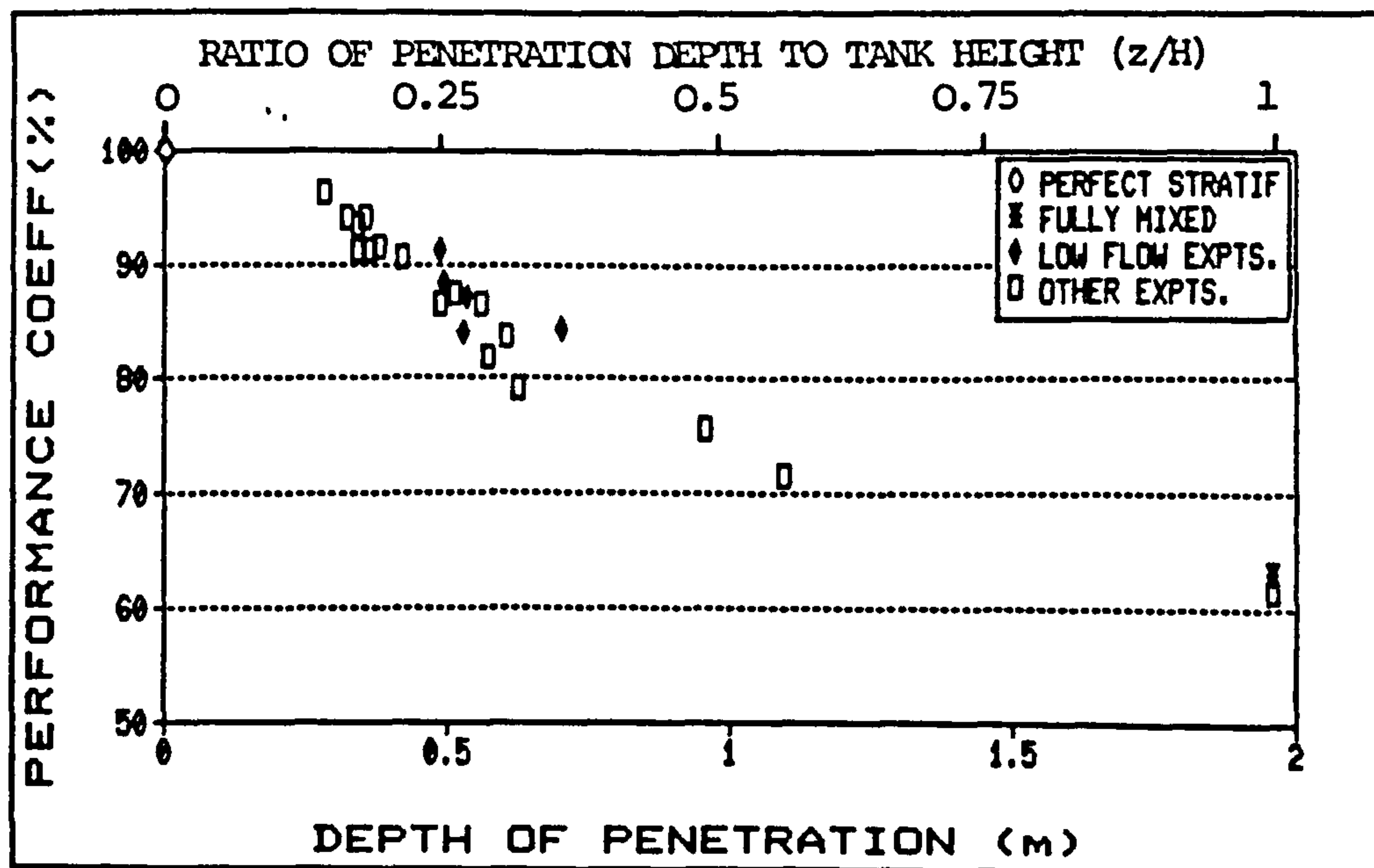


FIGURE 4.4 THE CORRELATION BETWEEN PERFORMANCE COEFFICIENT AND DEPTH OF PENETRATION FOR THE EXPERIMENTS WITH VERTICAL PORTS.

12 to 21 hour charge periods of the low flowrate experiments. This hypothesis is examined in detail with the aid of a numerical computer model of the store in Chapter 5 (see section 5.5) and in the light of the results of the temperature equalisation test (see sections 4.3 and 5.2).

4.2.1.10 Outlet Temperature Response

The outlet temperature during a typical discharge experiment with vertical ports is shown in Figure 4.5. For comparison the outlet temperature responses of a perfectly stratified and a fully-mixed store with no heat losses are also shown. For most of the test, the outlet temperature almost follows that of the perfectly stratified store; there is just a very slight fall off reflecting the heat losses from the store. The point at which the outlet temperature breaks totally from the ideal case marks the reaction time described in Section 4.2.1.3. After this point the outlet temperature falls off sharply in the characteristic inverted 'S' curve until the store is fully discharged not long after (in this experiment) the discharge period, τ .

It is convenient, for the purposes of comparing the store performance in different step input experiments, to non-dimensionalise the axes in Figure 4.5. This is done by dividing the real time by the discharge period, τ , so that the x-axis represents t/τ , and setting the y-axis to the parameter

$$\frac{T_{\text{out}} - T_{\text{in}}}{T_{\text{initial}} - T_{\text{in}}} .$$

The resultant graph is shown in Figure 4.6.

4.2.1.11 Store Thermoclines

The ultimate criterion for assessing the validity of a particular representation of a store, whether in the form of an analytical or numerical model, is the accuracy of the predicted internal temperature distribution of the store. Clearly if the internal

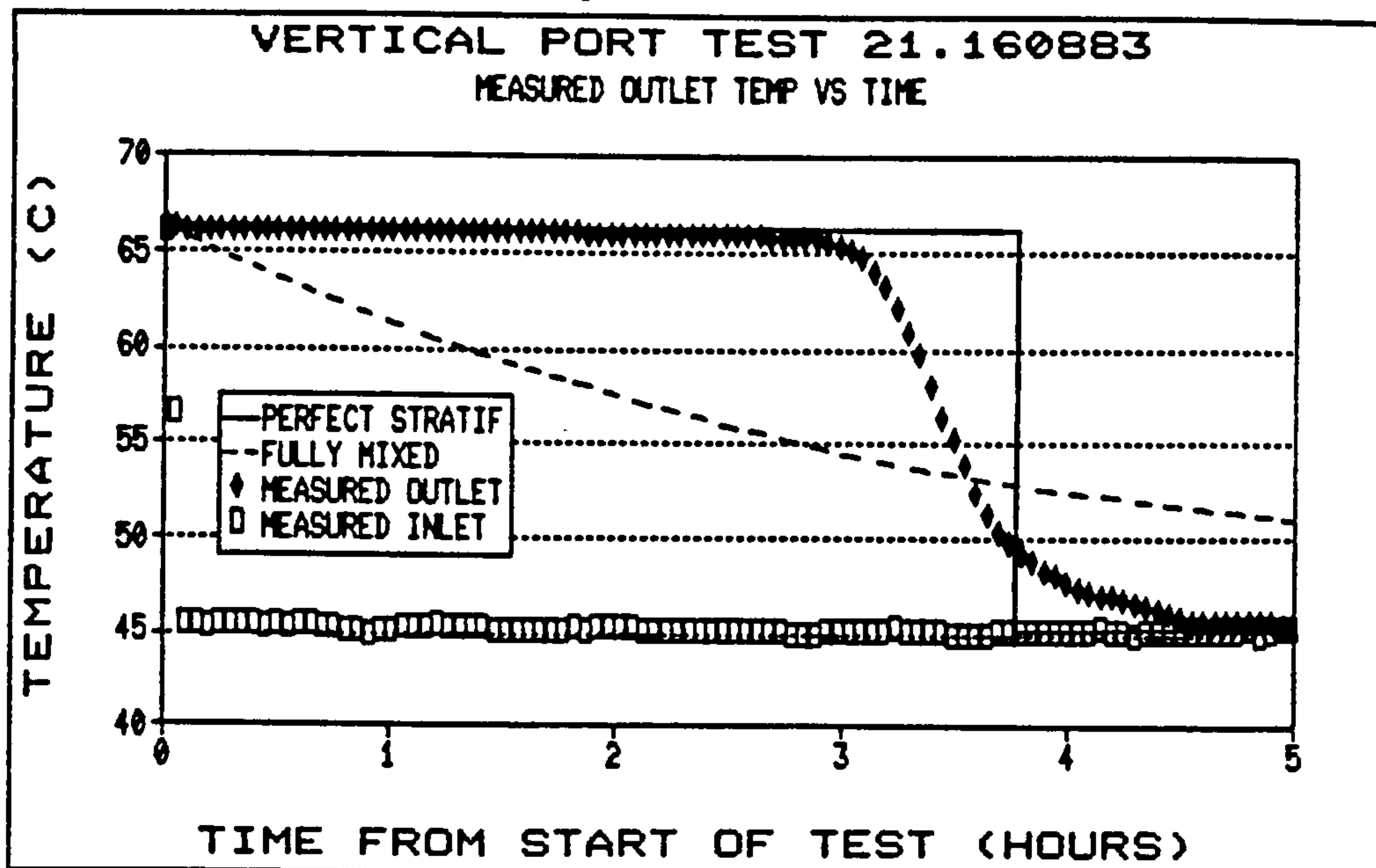


FIGURE 4.5 THE MEASURED OUTLET TEMPERATURE DURING A TYPICAL DISCHARGE EXPERIMENT WITH VERTICAL PORTS COMPARED WITH THEORETICAL PERFECTLY-STRATIFIED AND FULLY-MIXED RESPONSES.

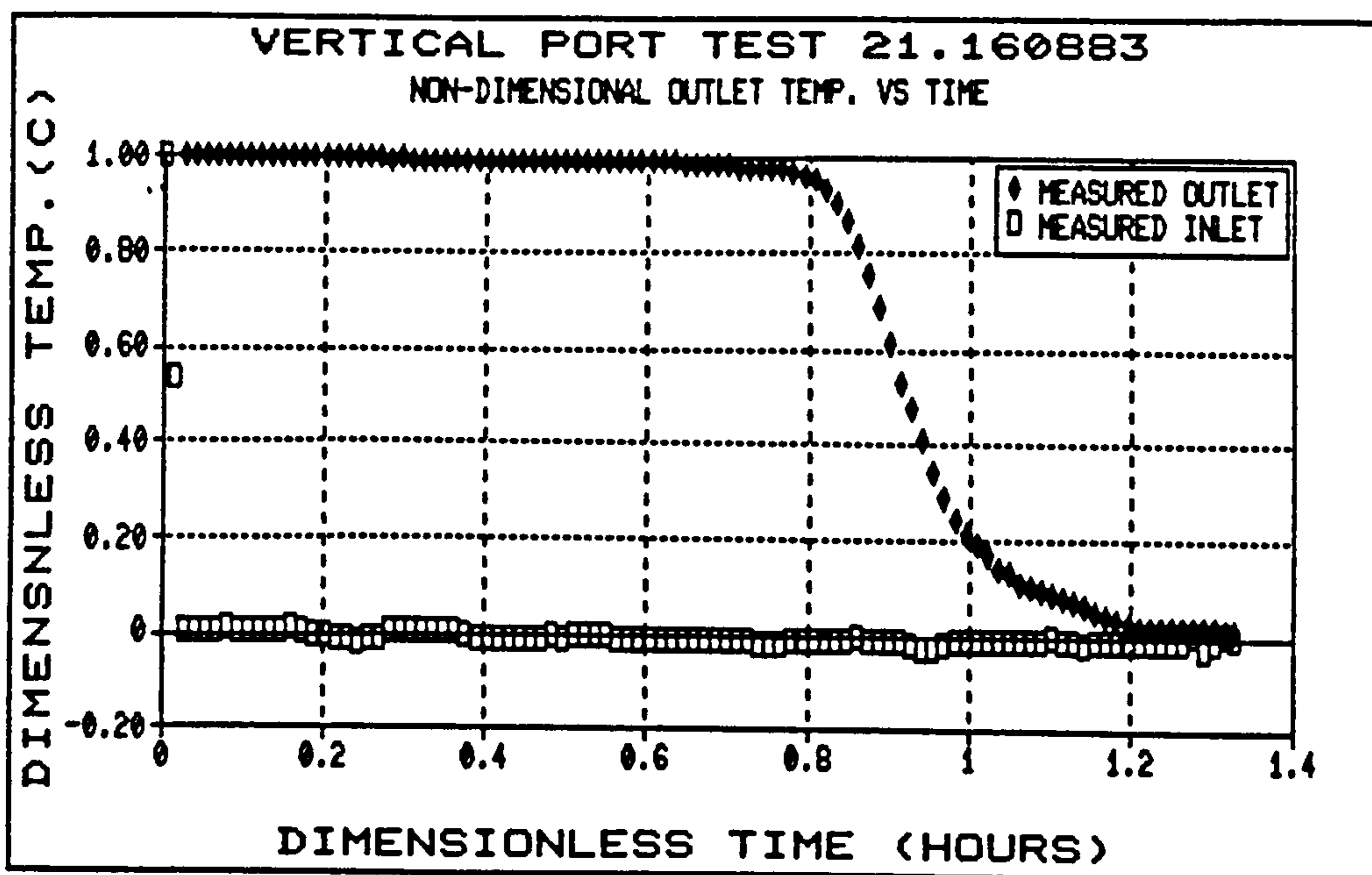
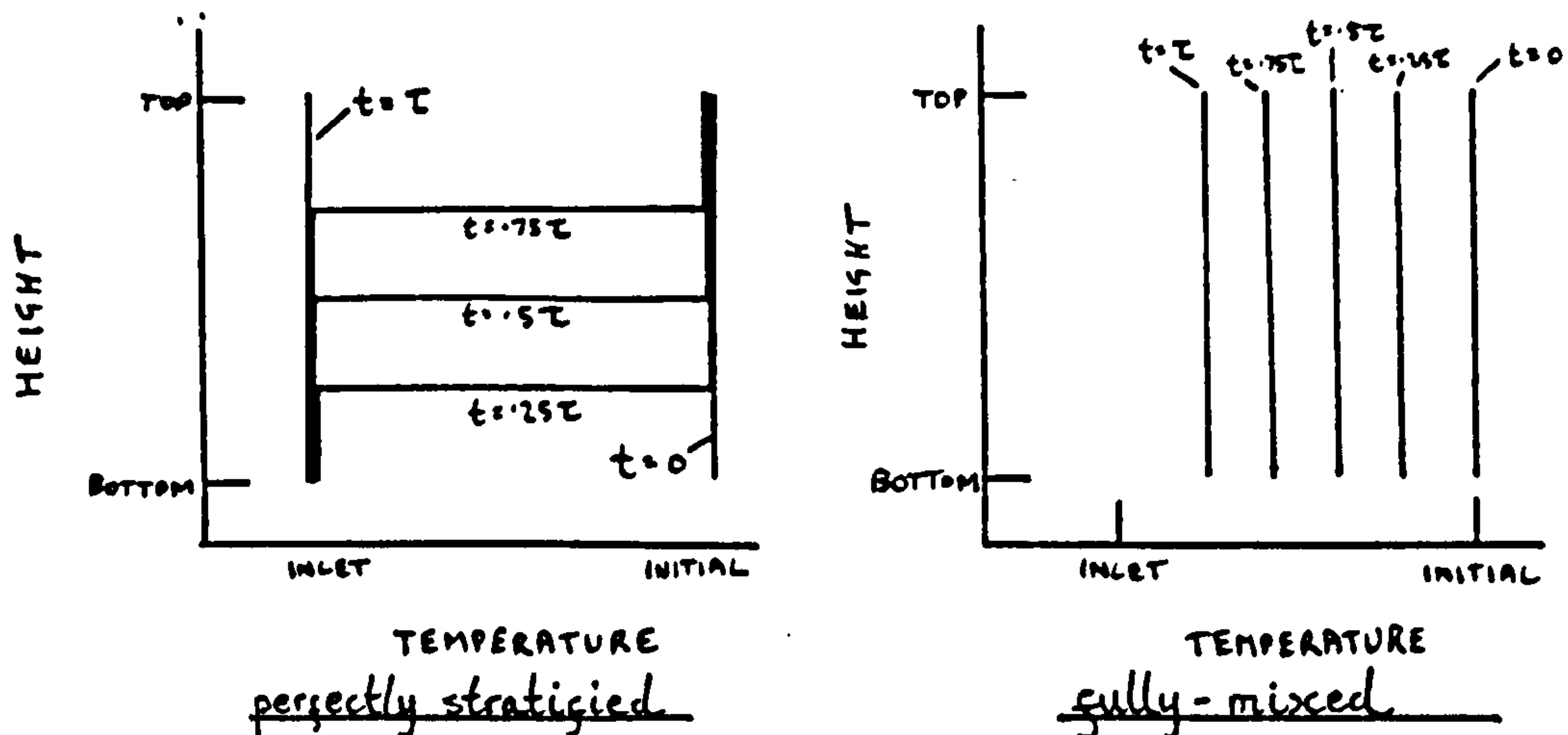


FIGURE 4.6 A NON-DIMENSIONAL REPRESENTATION OF FIGURE 4.5.

temperatures are closely predicted then so will be the store outlet temperature and heat losses, the factors which determine store performance. Of course, the store has a continuous three-dimensional temperature distribution which can only be measured at discrete points. Similarly representations of the store temperature distribution can range from a single average temperature to three-dimensional networks. In this work, a two-dimensional measuring grid has been used to assess a one-dimensional representation of the store. To this end, the mean temperature was calculated (from the 5 temperatures measured along a diameter of the store) at each of the 10 heights where measurements were taken (see Figure 4.7). The temperature distribution up the height of the store, defined as the store thermocline, can then be shown graphically by plotting the height against the temperature. The thermocline can be plotted at regular intervals to give a fairly full picture of the internal behaviour of the store through a step input test.

The store thermoclines for a discharge test with vertical ports are shown in Figure 4.8. Relatively flat thermoclines, as occurred in this particular test, indicate strong stratification. Indeed, the thermoclines of a perfectly stratified store would be horizontal lines on such a figure, whilst those of a fully-mixed store would be vertical lines, 'marching' down in time from the initial store temperature towards the inlet temperature (see below).



In figure 4.8, the general features of the flow regimes inside the tank are evident. The development of a fully-mixed region at the inlet can be seen, whilst the unperturbed temperatures in the rest of the store indicate that in this region there is a smooth plug-flow.

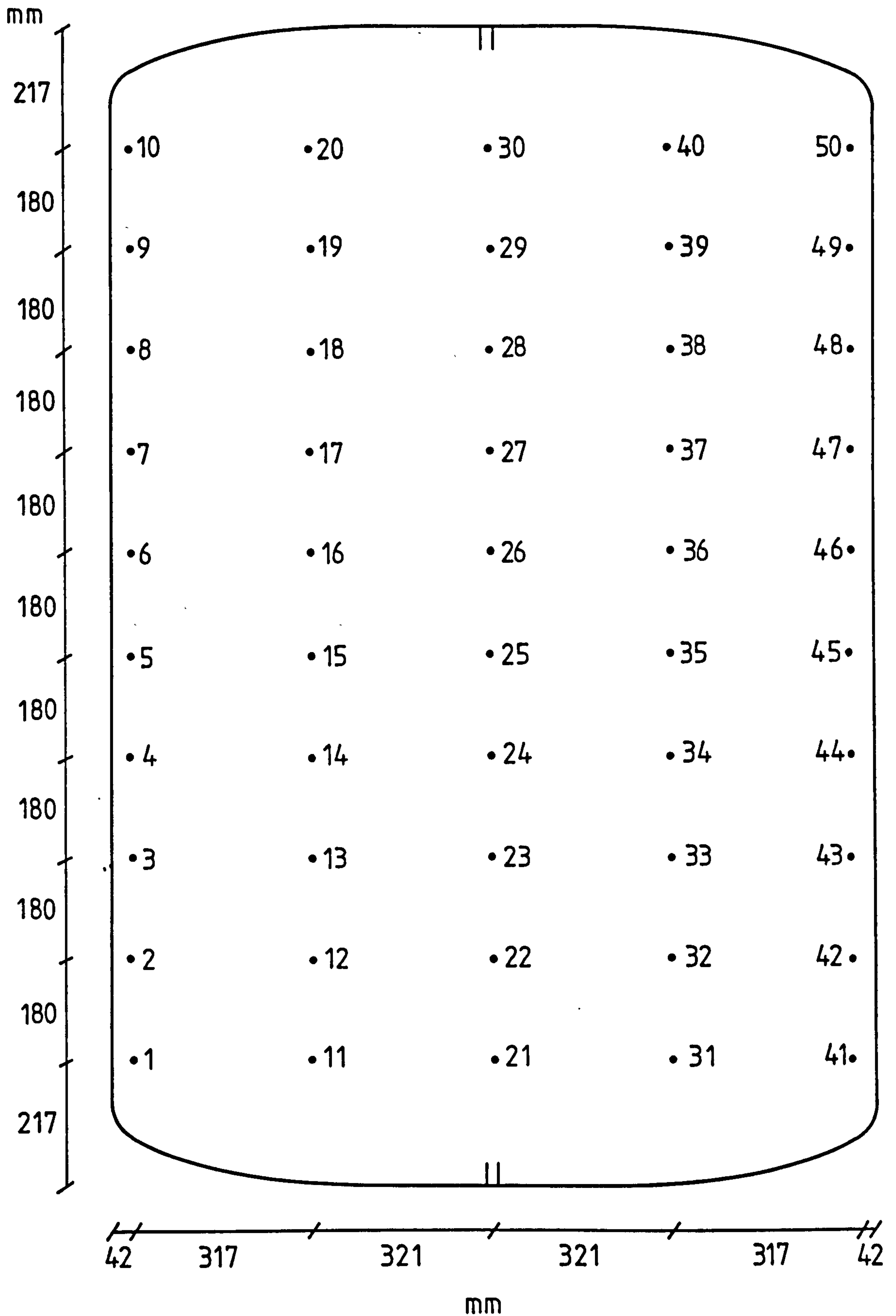


FIGURE 4.7 PRECISE LOCATION OF THE THERMOCOUPLES INSIDE THE WATER STORE.

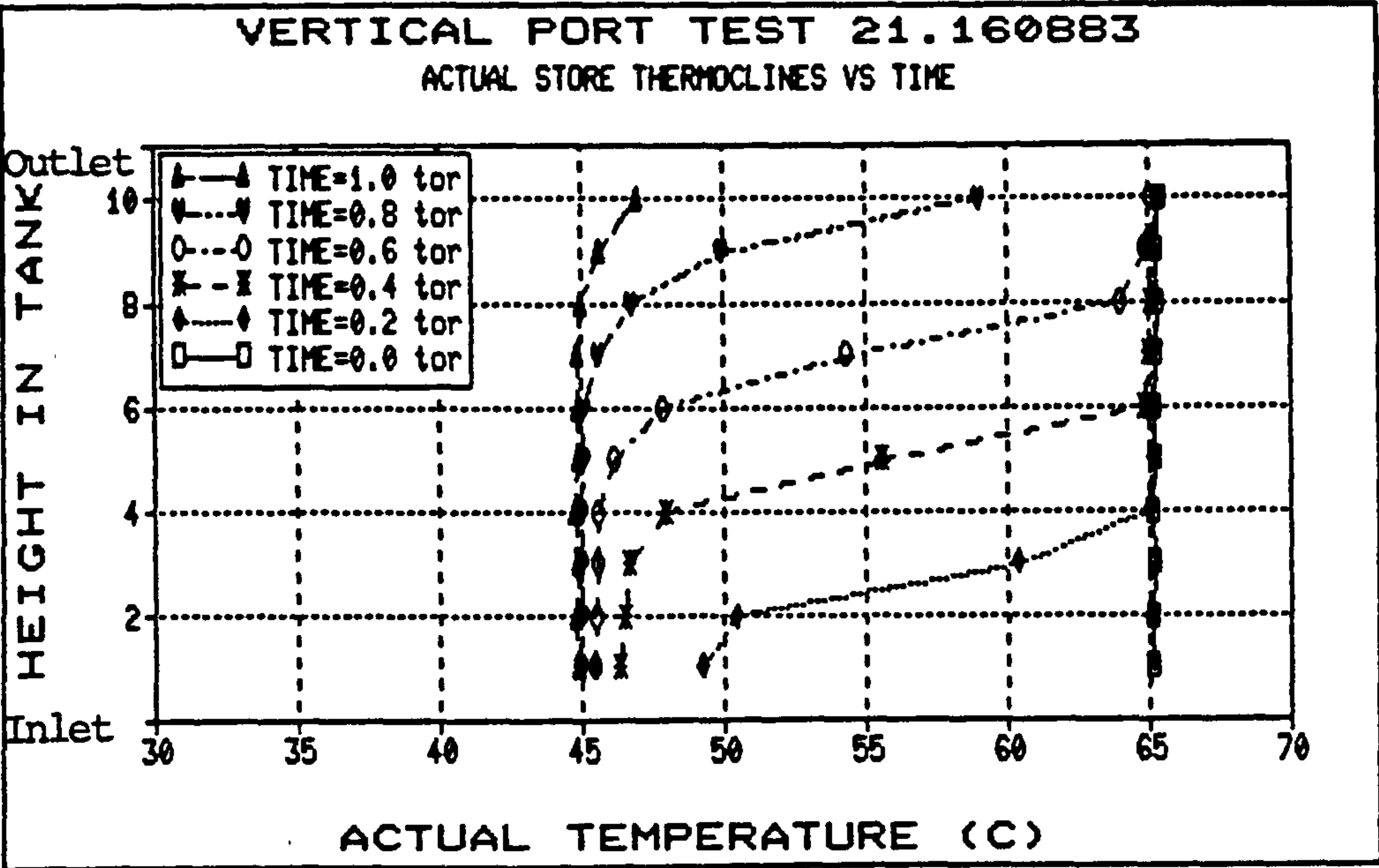


FIGURE 4.8 THE MEASURED STORE THERMOCLINES DURING A TYPICAL DISCHARGE EXPERIMENT WITH VERTICAL PORTS.

The depth of the fully-mixed region increases steadily throughout the test and its temperature gradually approaches the temperature of the inlet flow.

4.2.2 Distributors

4.2.2.1 Penetration Depth with the Distributors

When distributors are employed, the inlet flow enters the store through many small holes in a horizontal direction and is clearly not going to behave like a vertical jet. However, observation of the temperature distribution in the store during many step input tests, led to the conclusion that the net effect was the same. That is, the flow created around the inlet distributor, a fully-mixed region whose depth was determined by the relative magnitude of the momentum and buoyancy fluxes of the inlet flow.

Twenty-six good quality step input tests were carried out using the distributors, and they are listed in Table 4.2. The depth of the mixed region, i.e. a quasi depth of penetration, was determined for each experiment, using the same method as for the vertical ports, i.e. by measuring the 'reaction-time'. This depth was plotted against the Turner parameter to produce Figure 4.9.

In general features, this graph is very similar to the one for the vertical ports (Figure 4.1) : for all but the low flow rate experiments the points are clustered around a straight line given by :

$$z = 0.1 + 1.358 \frac{\dot{m}}{\rho} r^{-3/2} (g \beta \Delta T)^{-1/2}$$

Those experiments at low flow rates produced much larger depths of penetration than predicted by this correlation.

The Turner parameter in Figure 4.9 was calculated taking the assumption that each hole of the distributor was independent of the others and acted as a single port to the tank. Thus the radius, r , was taken as 0.794mm (1/32"), i.e. 1/16 of the vertical port radius, and the inlet flowrate was taken as 1/256 of the total flow into the

PORT	EXPT. NO.	INLET TEMP. C	INITIAL TEMP. C	FLOW RATE Kg/S	PERF. COEFF. %	REACTION TIME HOURS
DISTRIBUTOR	11.261081	56.1	17.2	0.197	96.4	3.492
DISTRIBUTOR	21.271081	18.8	56.0	0.200	95.4	3.390
DISTRIBUTOR	11.281081	21.5	17.8	0.449	76.6	0.684
DISTRIBUTOR	21.021181	28.6	48.0	0.299	95.3	2.310
DISTRIBUTOR	22.021181	25.4	29.8	0.449	72.1	0.734
DISTRIBUTOR	11.260182	18.4	8.4	0.549	72.5	0.383
DISTRIBUTOR	11.270182	50.7	16.0	0.267	97.1	2.55
DISTRIBUTOR	21.290182	16.2	22.9	0.599	72.7	0.406
DISTRIBUTOR	21.280182	21.2	49.0	0.259	96.1	2.712
DISTRIBUTOR	12.290182	37.5	17.0	0.397	90.3	1.417
DISTRIBUTOR	21.010282	25.3	34.7	0.329	90.9	1.973
DISTRIBUTOR	11.100282	45.1	19.4	0.396	92.7	1.502
DISTRIBUTOR	11.160282	58.0	39.0	0.295	95.1	2.306
DISTRIBUTOR	21.170382	15.2	31.2	0.400	87.6	1.525
DISTRIBUTOR	11.060482	80.1	30.9	0.156	95.6	4.437
DISTRIBUTOR	21.070482	18.5	76.7	0.160	95.4	4.418
DISTRIBUTOR	11.200482	51.9	16.5	0.049	91.1	13.1
DISTRIBUTOR	21.040582	26.3	66.0	0.050	87.6	12.48
DISTRIBUTOR	21.260482	15.2	45.0	0.048	85.9	12.921
DISTRIBUTOR	11.270482	78.1	14.8	0.038	90.1	15.508
DISTRIBUTOR	11.260582	61.8	17.4	0.078	91.7	8.065
DISTRIBUTOR	21.031281	24.6	40.8	0.643	81.9	0.467
DISTRIBUTOR	21.021281	17.6	26.9	0.644	76.9	0.507
DISTRIBUTOR	11.011281	28.4	13.9	0.628	79.4	0.527
DISTRIBUTOR	11.301181	20.2	10.6	0.629	73.9	0.405
DISTRIBUTOR	22.031281	21.1	25.4	0.644	73.8	0.385

Table 4.2 The step input conditions for the experiments with distributors.

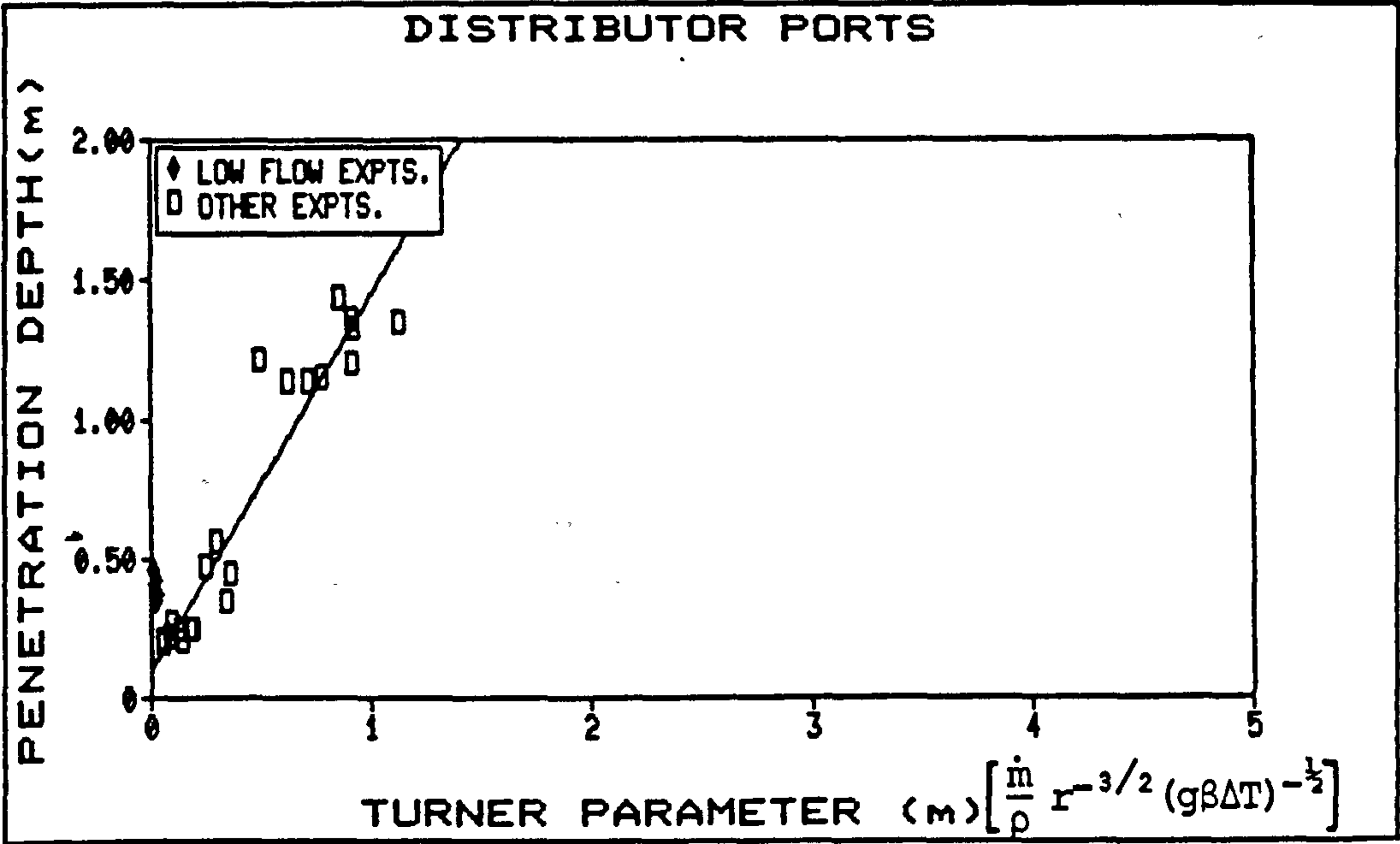


FIGURE 4.9 THE CORRELATION BETWEEN A QUASI DEPTH OF PENETRATION AND THE TURNER PARAMETER FOR THE EXPERIMENTS WITH DISTRIBUTORS .

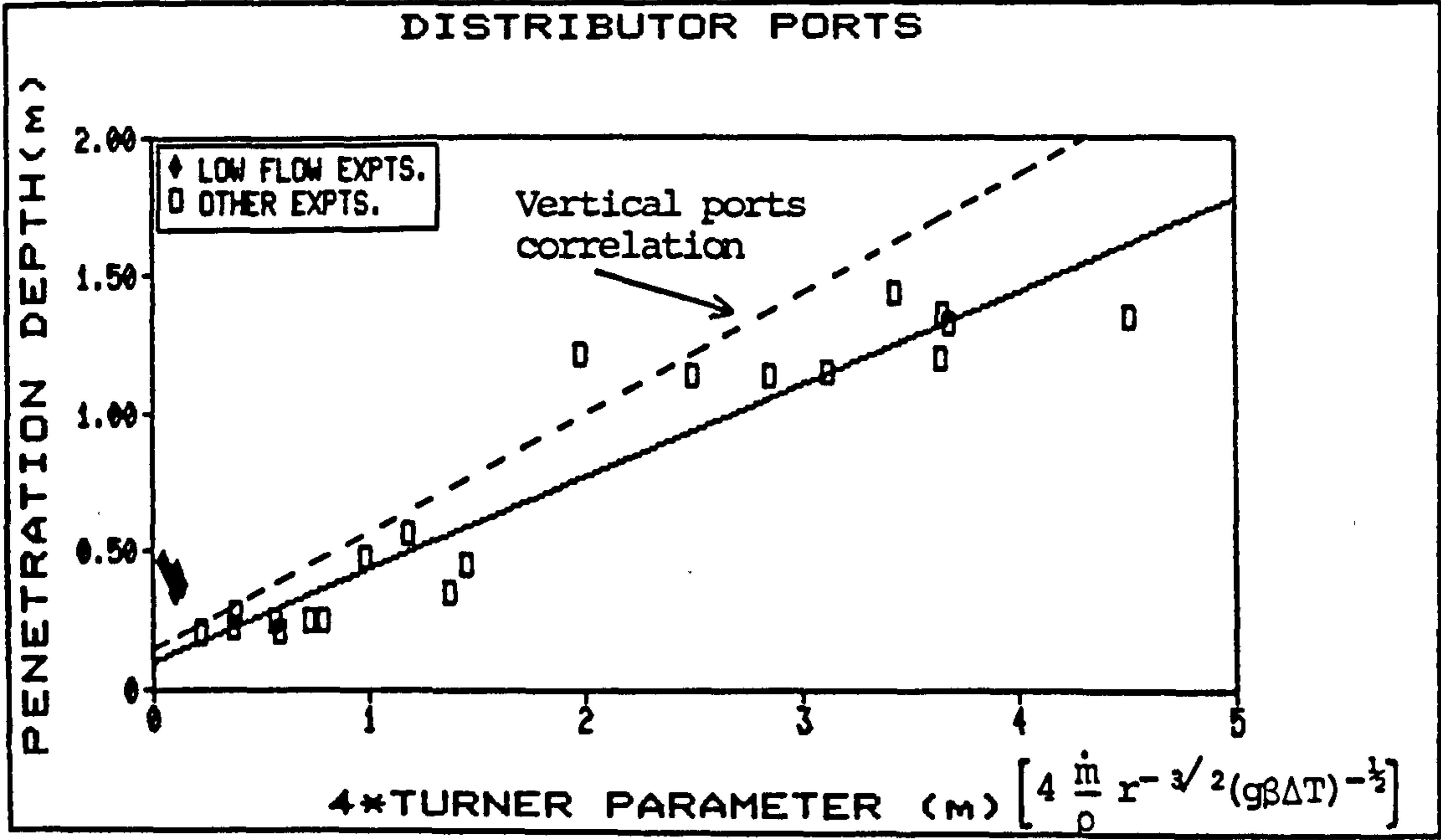


FIGURE 4.10 THE CORRELATION BETWEEN A QUASI DEPTH OF PENETRATION AND FOUR TIMES THE TURNER PARAMETER FOR THE EXPERIMENTS WITH DISTRIBUTORS .

vessel. Now equation (11) suggests that if the radius of a single vertical port is reduced by a factor of 16 and, at the same time, the inlet flowrate is reduced by a factor of 256, then the net effect should be a reduction in the depth of penetration by a factor of 4. The gradient of the best-fit line for the distributors is 1.358 or about 3 times higher than that for the vertical ports. Thus the distributors produced a mixed region about 12 times greater than a single hole of the distributor, acting as a vertical port, would have produced according to equation (11). For academic completeness, it would have been interesting to have tested the robustness of equation (11) to changes in the vertical port radius, although this was in fact successfully done by Turner (see Figure 2.1). However, it is unlikely such a study could have investigated nozzle diameters of less than 1/4 in, owing to the high pressure drop that would have resulted.

The discussion in the previous paragraph should not lead the reader to believe that the distributors performed worse than the single vertical ports. For the same total inlet flowrate and initial-inlet temperature difference, the Turner parameter calculated for the distributors is 1/4 of that for the vertical ports. It is therefore necessary to plot the penetration depth with the distributors against 4 times the Turner parameter in order to make a direct comparison with the vertical ports. This has been done in Figure 4.10, on which the best-fit line for the vertical ports has also been plotted. The figure shows that the distributors succeeded in reducing the depth of penetration, and were increasingly effective as the Turner parameter increased, i.e. as the inlet flow momentum rose.

4.2.2.2 Distributors : Performance Coefficient Results

The performance coefficient, $\eta(\tau)$, for each distributor experiment, is plotted against the Turner parameter in Figure 4.11. Again, for direct comparison with the vertical ports, the distributor results have also been plotted against 4 times the Turner parameter in Figure 4.12. The results show very similar trends to those for the vertical ports (see Section 4.2.1.9). A best-fit line for each set of data (excluding the low flowrate experiments) has been superimposed on Figure 4.12 to illustrate the improvement in performance achieved by

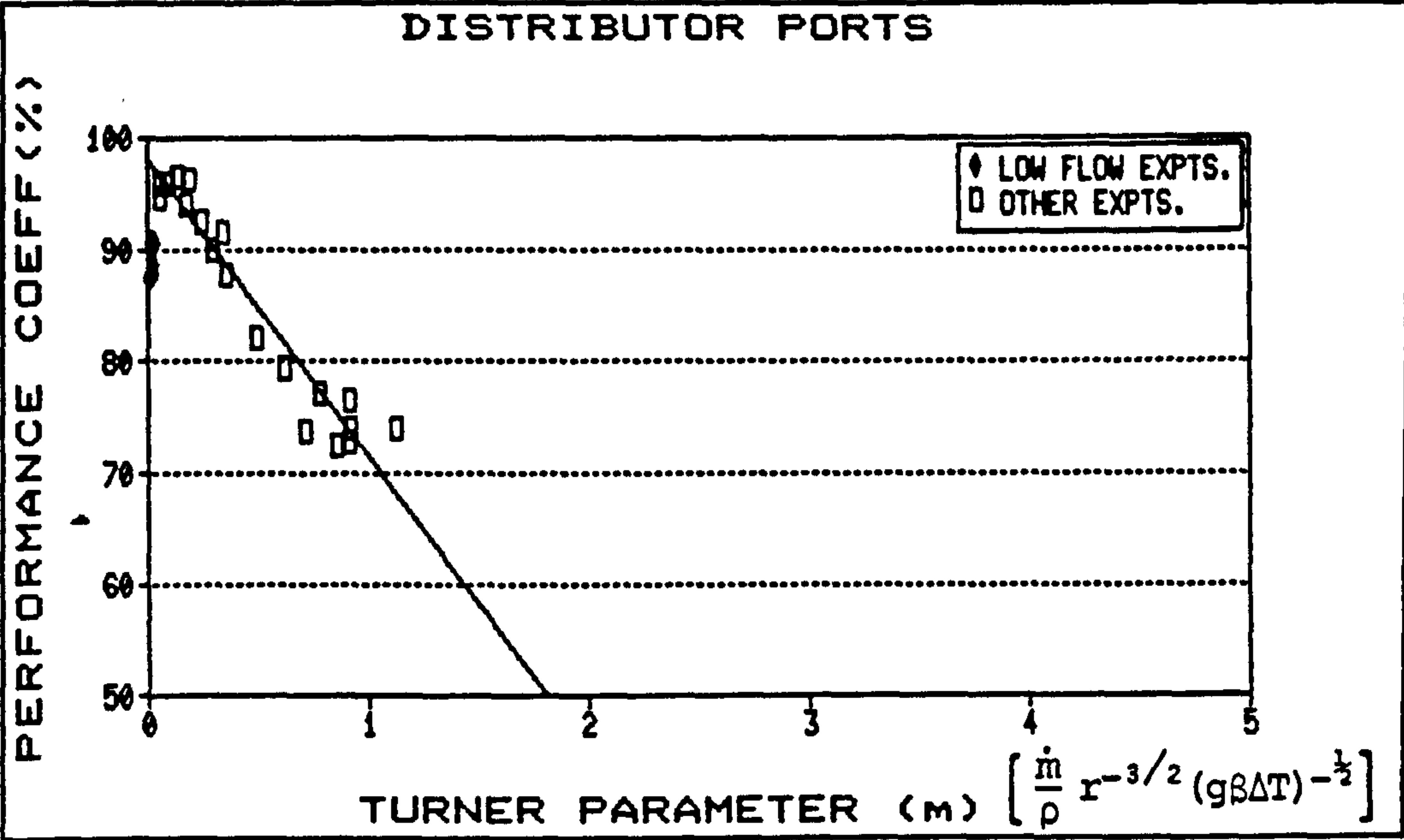


FIGURE 4.11 THE CORRELATION BETWEEN PERFORMANCE COEFFICIENT AND TURNER PARAMETER FOR THE EXPERIMENTS WITH DISTRIBUTORS .

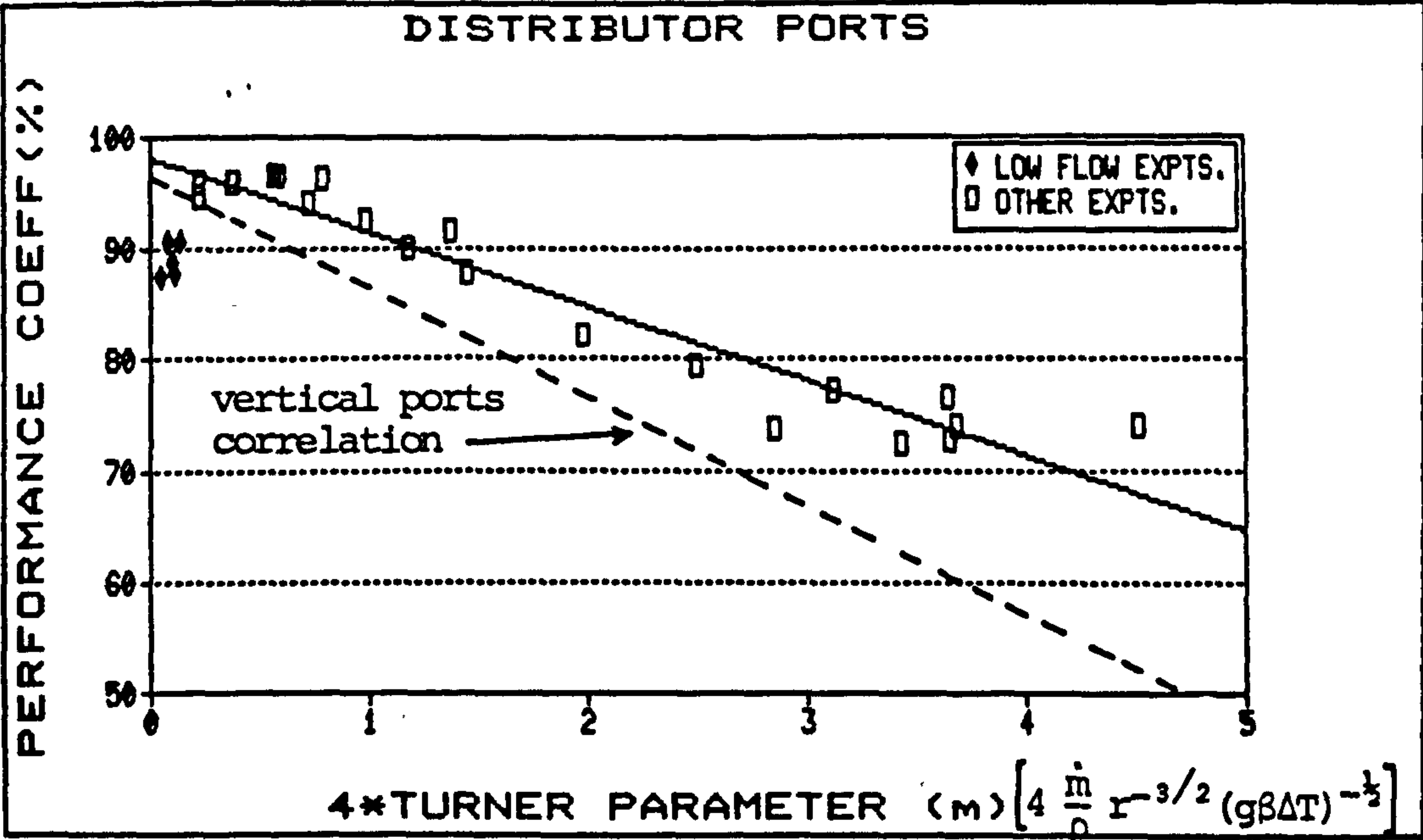


FIGURE 4.12 THE CORRELATION BETWEEN PERFORMANCE COEFFICIENT AND FOUR TIMES THE TURNER PARAMETER FOR THE EXPERIMENTS WITH DISTRIBUTORS .

use of the distributors. As the momentum of the inlet flow increases (i.e. as the Turner parameter increases or Ri decreases), the performance of the distributors is progressively superior. When the Turner parameter equals $1m$ ($Ri = 0.0016$) the distributor performance is better by about 7%, whilst with a Turner parameter of $4m$ ($Ri = 0.0001$) the distributor out-performs the vertical port by nearly 30%.

The performance coefficient has been plotted against the depth of penetration in Figure 4.13. As with the vertical ports, the experimental points lie near to a line between the two theoretical points pertaining to a perfectly stratified and a fully-mixed store. The results are very close to those for the vertical ports, for a given depth of penetration.

4.2.2.3 Distributors : Outlet Temperature and Thermoclines During a Typical Experiment

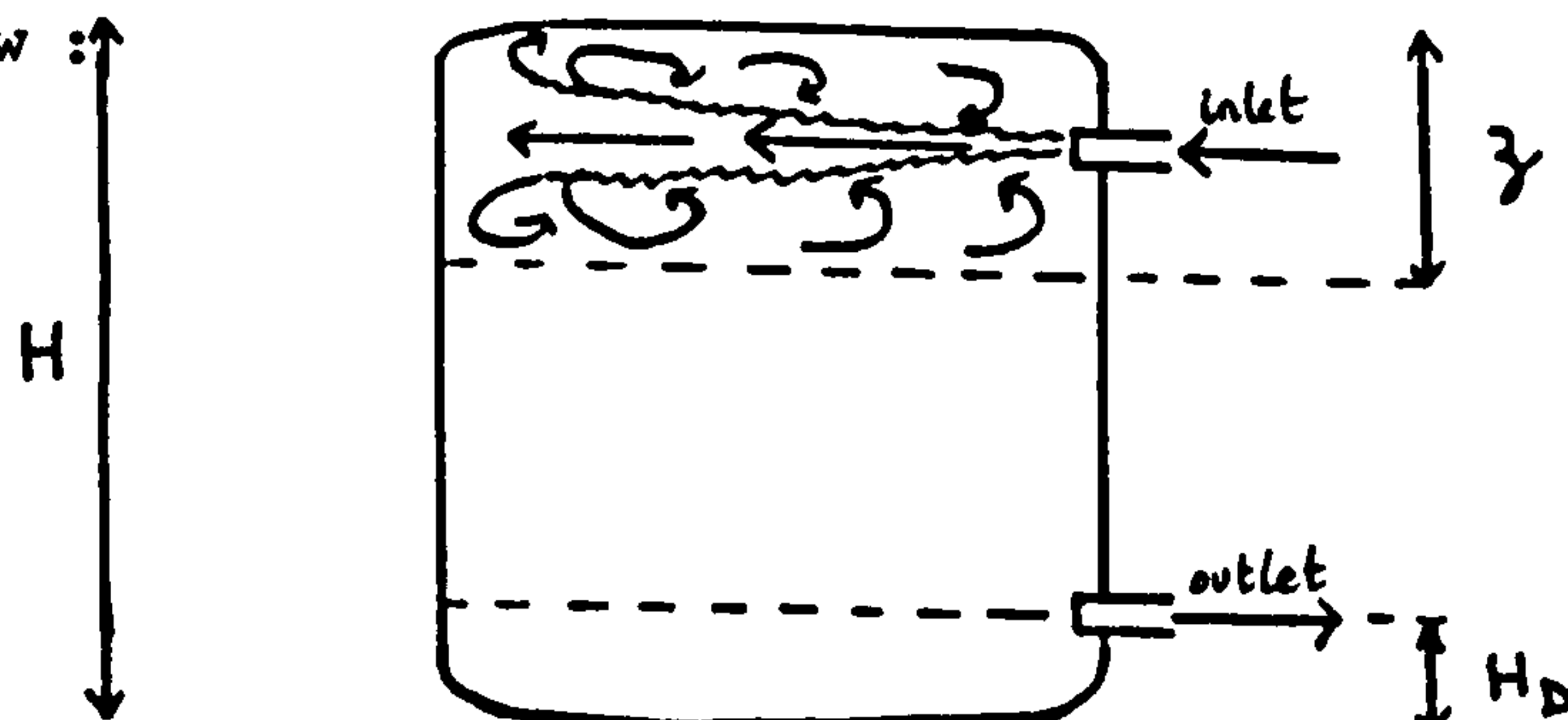
A plot of non-dimensional outlet temperature against time for a typical distributor discharge experiment is shown in Figure 4.14. It displays the same characteristics as the vertical port experiment plotted in Figure 4.6; comparing these two particular examples, the fall off in the outlet temperature comes slightly later and is a little steeper with the distributors indicating marginally better stratification and performance.

The thermoclines during this distributor experiment are shown in Figure 4.15. In general terms they have similar features to those for the vertical port test given in Figure 4.8. However, they are appreciably flatter, confirming that better stratification occurred.

4.2.3 Horizontal Ports

4.2.3.1 Penetration Depth with Horizontal Ports

With the single horizontal ports, the inlet flow enters from one side of the tank and creates a flow pattern approximately like that illustrated below :



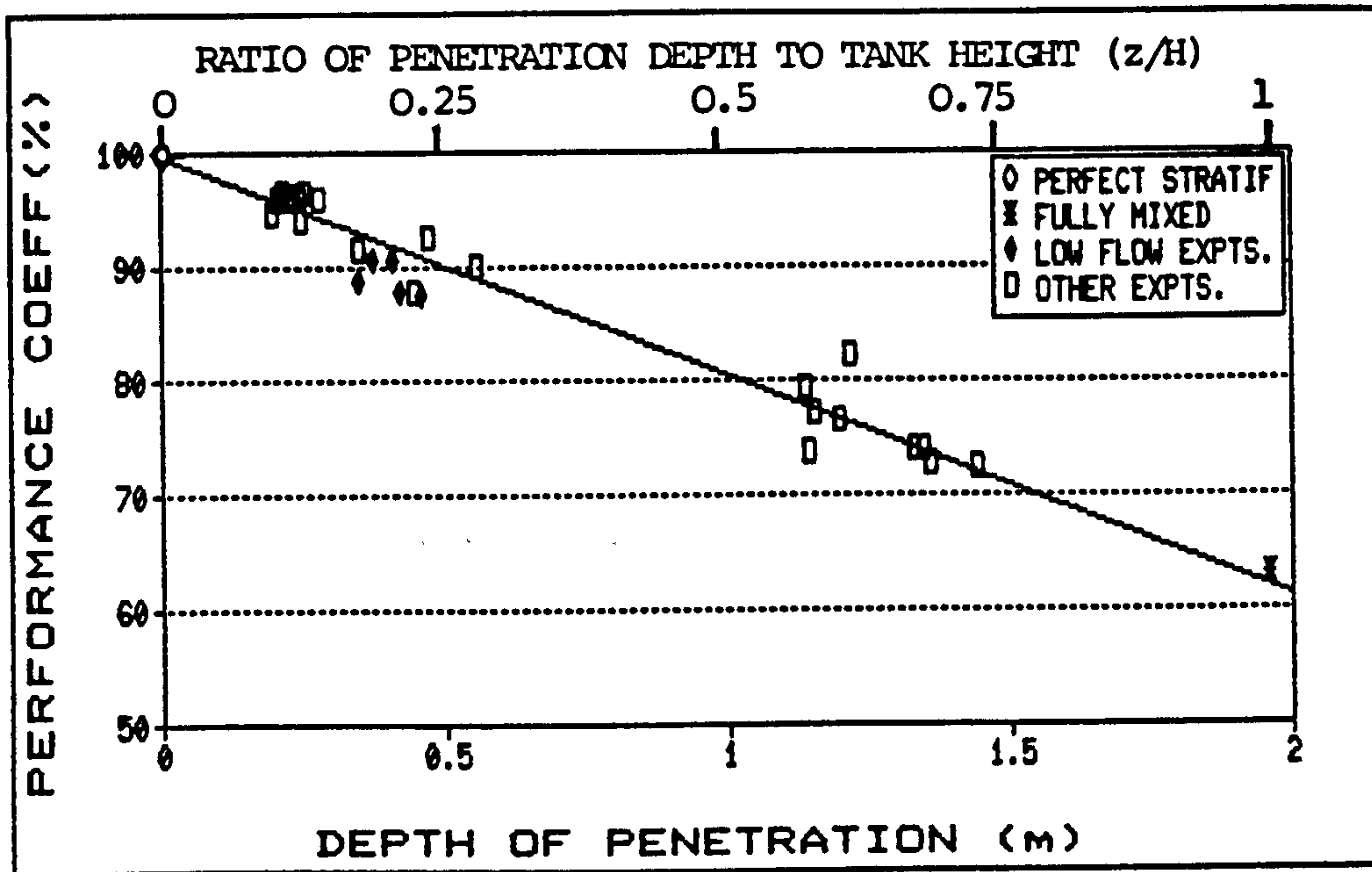


FIGURE 4.13 THE CORRELATION BETWEEN PERFORMANCE COEFFICIENT AND DEPTH OF PENETRATION FOR THE EXPERIMENTS WITH DISTRIBUTORS .

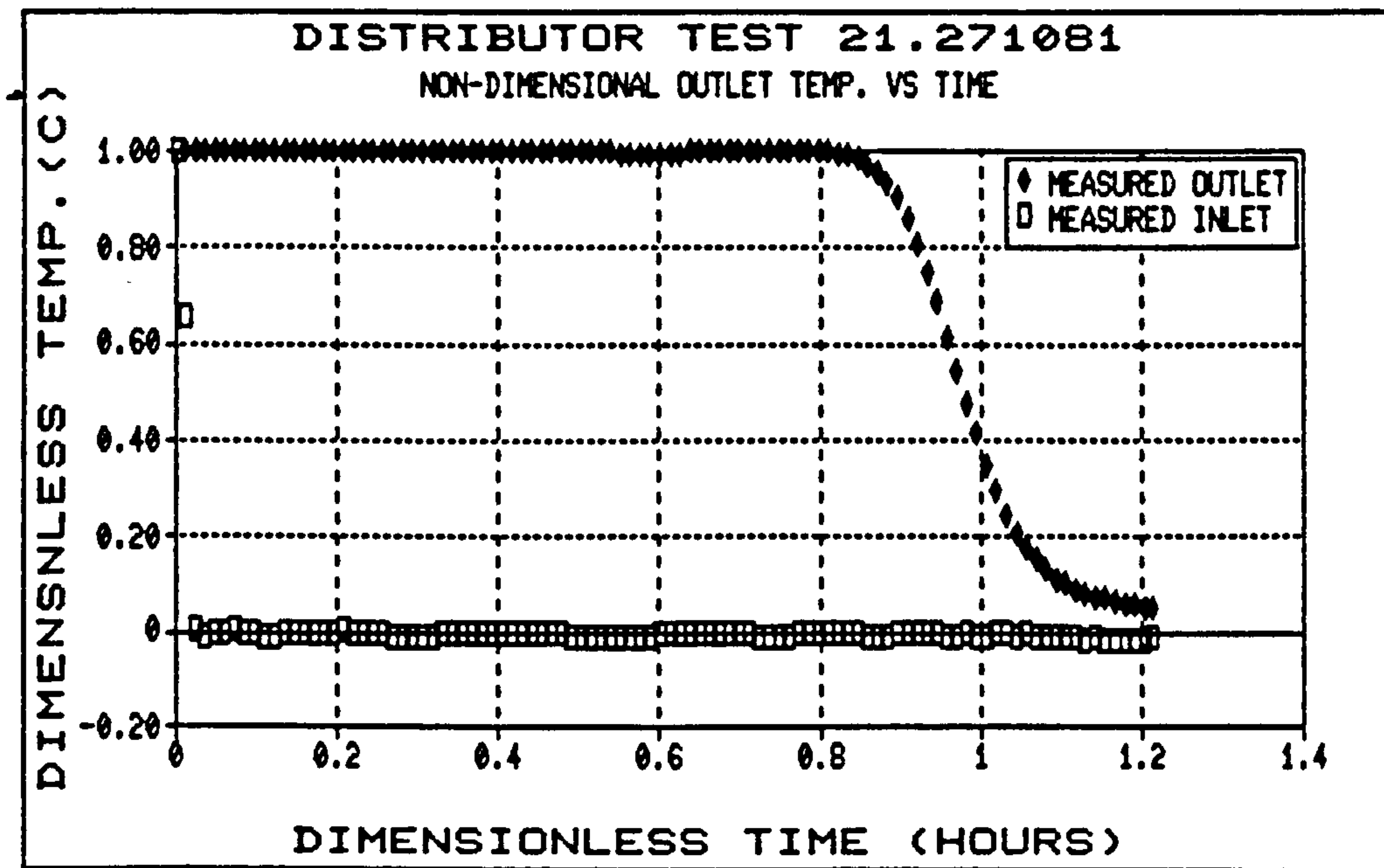


FIGURE 4.14 A NON-DIMENSIONAL REPRESENTATION OF THE MEASURED OUTLET TEMPERATURE DURING A TYPICAL DISCHARGE EXPERIMENT WITH DISTRIBUTORS.

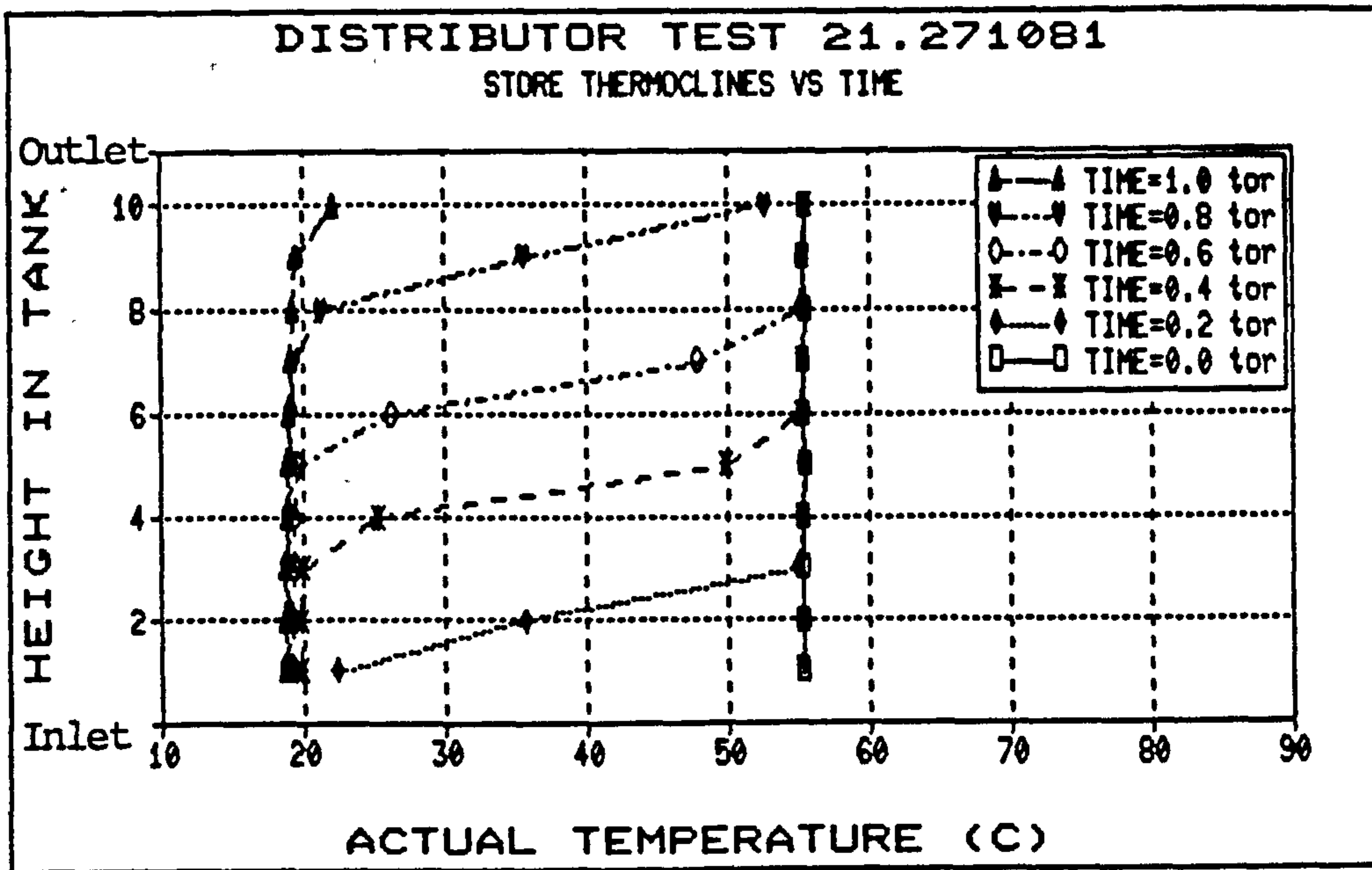


FIGURE 4.15 THE MEASURED STORE THERMOCLINES DURING A
TYPICAL DISCHARGE EXPERIMENT WITH DISTRIBUTORS.

Observation of the temperature distribution in the store again showed that a fully-mixed region was created around the inlet port, with a depth dependent on the momentum and buoyancy of the inlet flow. The observations also indicated that the region between the level of the outlet port and the end of the tank (i.e. the bottom in the case of a charge test, as illustrated) was undisturbed by the flow and could thus be considered dead space. The formula for calculating the reaction time had to be modified, therefore, to take this into account.

$$\text{i.e. } t_R = \frac{H - H_D - z}{v}$$

where H_D is the height of the dead space between the outlet port and the tank end, and is defined (again to simplify the geometry) by

$$H_D = \frac{V_D}{A}$$

where V_D is the deadspace volume.

$$\text{Hence, } z = H - H_D - vt_R = H - H_D - \frac{\dot{m} t_R}{\rho_{in} A} \quad \dots(29)$$

Nineteen good quality step input tests were carried out using the single horizontal ports and they are listed in Table 4.3. The modified quasi penetration depth, z , defined above, was plotted against the Turner parameter to give the graph in Figure 4.16. A good correlation, in the form of equation (11), again exists for all experiments, apart from those at very low flow rates. It is given by:

$$z = 0.15 + 0.238 \frac{\dot{m}}{\rho} r^{-3/2} (g \beta \Delta T)^{-1/2} \quad \dots(30)$$

The not-fully turbulent, low flow rate tests again produced higher penetration depths than given by this expression.

4.2.3.2 Comparison with Other Research

(a) Lavan and Thompson

Whilst reporting the results of the single horizontal ports it is

PORT	EXPT NO.	INLET TEMP. C	INITIAL TEMP. C	FLOW RATE Kg/S	PERF. COEFF. %	REACTION TIME HOURS

HORIZONTAL	11.280283	12.9	7.8	0.450	68.4	0.405
HORIZONTAL	11.010383	32.2	23.4	0.475	74.3	0.809
HORIZONTAL	11.161182	28.3	12.3	0.450	76.9	0.938
HORIZONTAL	11.250283	76.2	69.2	0.400	79.7	1.344
HORIZONTAL	11.140283	48.2	21.2	0.450	82.1	1.044
HORIZONTAL	11.100283	50.3	14.6	0.450	84.5	1.237
HORIZONTAL	11.280283	24.5	11.1	0.200	85.7	2.749
HORIZONTAL	11.020383	68.2	47.1	0.350	80.0	1.581
HORIZONTAL	11.161182	52.0	26.7	0.250	83.9	2.281
HORIZONTAL	11.230283	62.4	11.9	0.250	83.6	2.153
HORIZONTAL	11.191182	60.6	50.2	0.180	85.4	3.379
HORIZONTAL	11.010383	46.1	30.8	0.160	82.8	3.361
HORIZONTAL	11.110283	69.6	49.0	0.190	91.7	3.060
HORIZONTAL	11.070383	59.5	14.6	0.161	86.4	3.480
HORIZONTAL	11.240283	70.6	60.4	0.100	82.5	5.821
HORIZONTAL	11.020383	76.0	66.9	0.050	77.8	11.825
HORIZONTAL	11.070383	67.3	58.1	0.040	74.6	13.704
HORIZONTAL	11.080383	80.4	65.9	0.044	79.1	11.811
HORIZONTAL	11.030383	79.7	11.3	0.040	81.6	11.390

Table 4.3 The step input conditions for the experiments with horizontal ports.

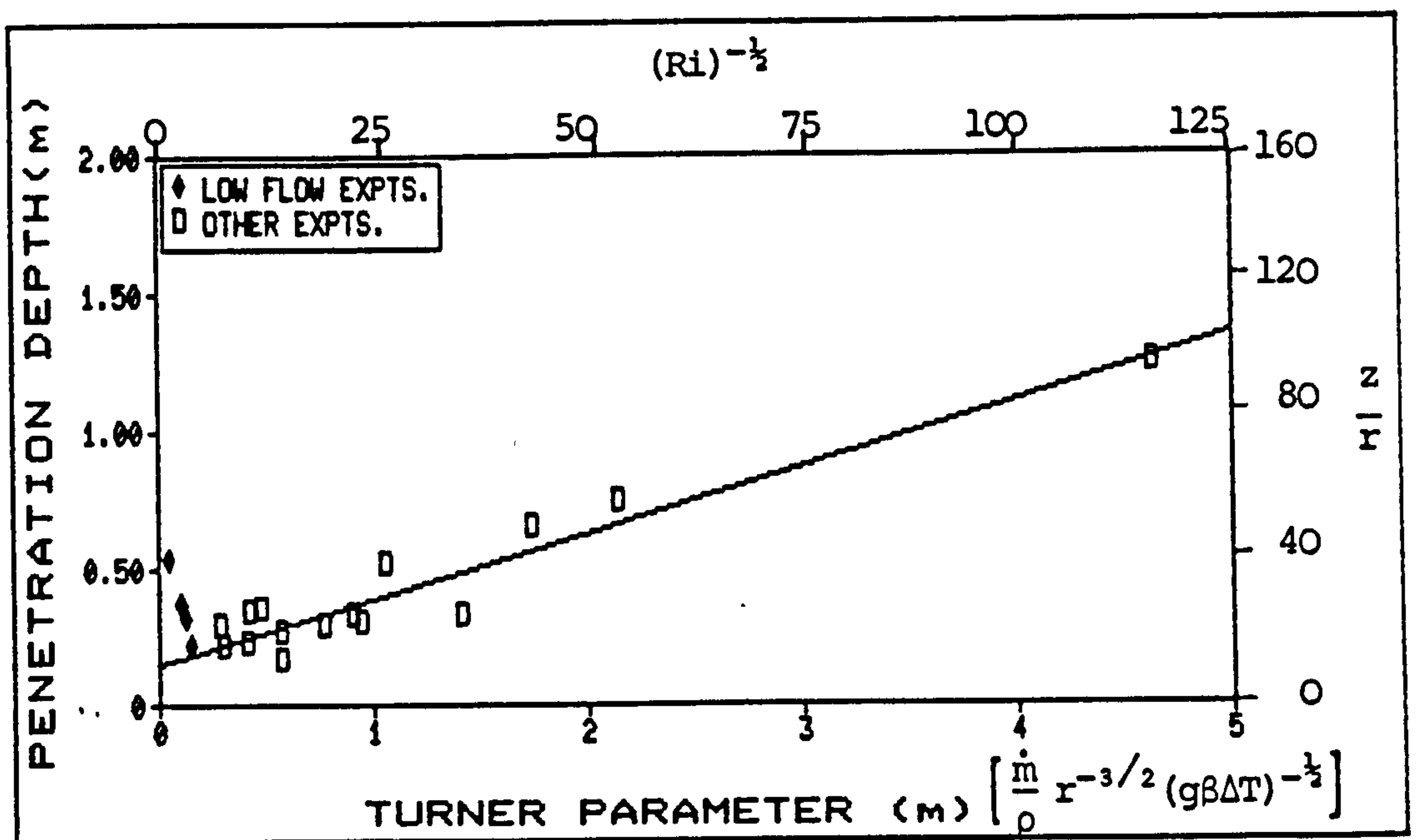


FIGURE 4.16 THE CORRELATION BETWEEN PENETRATION DEPTH (z) AND TURNER PARAMETER FOR THE EXPERIMENTS WITH HORIZONTAL PORTS.

interesting to compare them to those of Lavan and Thompson (Ref.2.17), mentioned in Chapter 2 and section 4.2.1.2. They did a comprehensive series of tests with this port configuration and an identical step input procedure. However they did use very much smaller tanks, at 4 and 50 litres about 700 and 60 times smaller respectively than the vessel used in this work. Nevertheless they presented their results using a dimensionless correlating parameter, which aimed to reflect the buoyancy and momentum fluxes of the inlet flow, as shown in Figure 2.3.

They adopted a rather arbitrary figure of merit, which they called the extraction efficiency, η^* , defined as

$$\eta^* = \frac{\dot{V} t^*}{V} \quad \dots(31)$$

where t^* is the time at which the inlet-exit temperature difference has dropped to 90% of its initial value. Clearly t^* is very similar to the reaction time, t_R , which is simply the time when the inlet-exit temperature difference first starts to drop from its initial value. Since the outlet temperature tends to drop fairly sharply, the difference between t^* and t_R is fairly negligible, a comment also inferred by Lavan and Thompson.

Now the volume flow rate, $\dot{V}$, can be expressed in terms of the plug flow velocity, since (as in equation (16))

$$\dot{V} = v A$$

also from equation (17) $V = H A$

$$\text{So that (31) becomes } \eta^* = \frac{v t^*}{H} \quad \dots(32)$$

Non-dimensionalising z in equation (29), by dividing through by $(H - H_D)$, gives :

$$\frac{z}{H-H_D} = 1 - \frac{v t_R}{H-H_D}$$

$$\text{whence } 1 - \frac{z}{H - H_D} = \frac{v t_R}{H - H_D} \quad \dots(33)$$

where the left hand side is a parameter, ranging from 0 to 1 as z ranges from its highest (worst) value to zero (best), which will now be known as the penetration efficiency.

Examination of equations (32) and (33) shows that the penetration efficiency, $1 - z/(H-H_D)$, is identical to η^* apart from the small difference between t_R and t^* , and the allowance for deadspace, which did not occur with the apparatus of Lavan and Thompson. It was therefore decided to calculate the penetration efficiency for each horizontal port experiment and plot it against the correlating parameter, P , derived by Lavan and Thompson as

$$P = Re^{-0.55} Gr^{0.35} (L/D)^{0.58}$$

The resulting graph is shown in Figure 4.17. The indices in the expression for P were calculated by Lavan and Thompson so as to yield a curve given by the equation :

$$\eta^* = 1 - e^{-0.067P} \quad \dots(34)$$

which produced the best least-square fit to all their data. Equation (34) is also plotted on Figure 4.17. The following comments arise from a comparison of the two sets of results :

- (i) Lavan and Thompson only gave results for $p < 50$ (see Figure 2.3). They did not mention what happened for $P > 50$ but the results in the present work indicate strongly that η^* reaches a fairly flat peak at $60 < P < 120$ and then falls off steadily in the range $120 < P < 200$, the region of low inlet flow rates. The implication is therefore that their correlation (34) is not valid outside the range $0 < P < 60$. This is in common with the corresponding correlation given in equation (11), based on the Turner parameter, which has also been shown to be invalid for low inlet flowrates.
- (ii) The results of the present work do follow the shape of their correlation in the range $0 < P < 60$, but values for the penetration efficiency are consistently lower than those for η^* . Only a small part of this difference can be explained

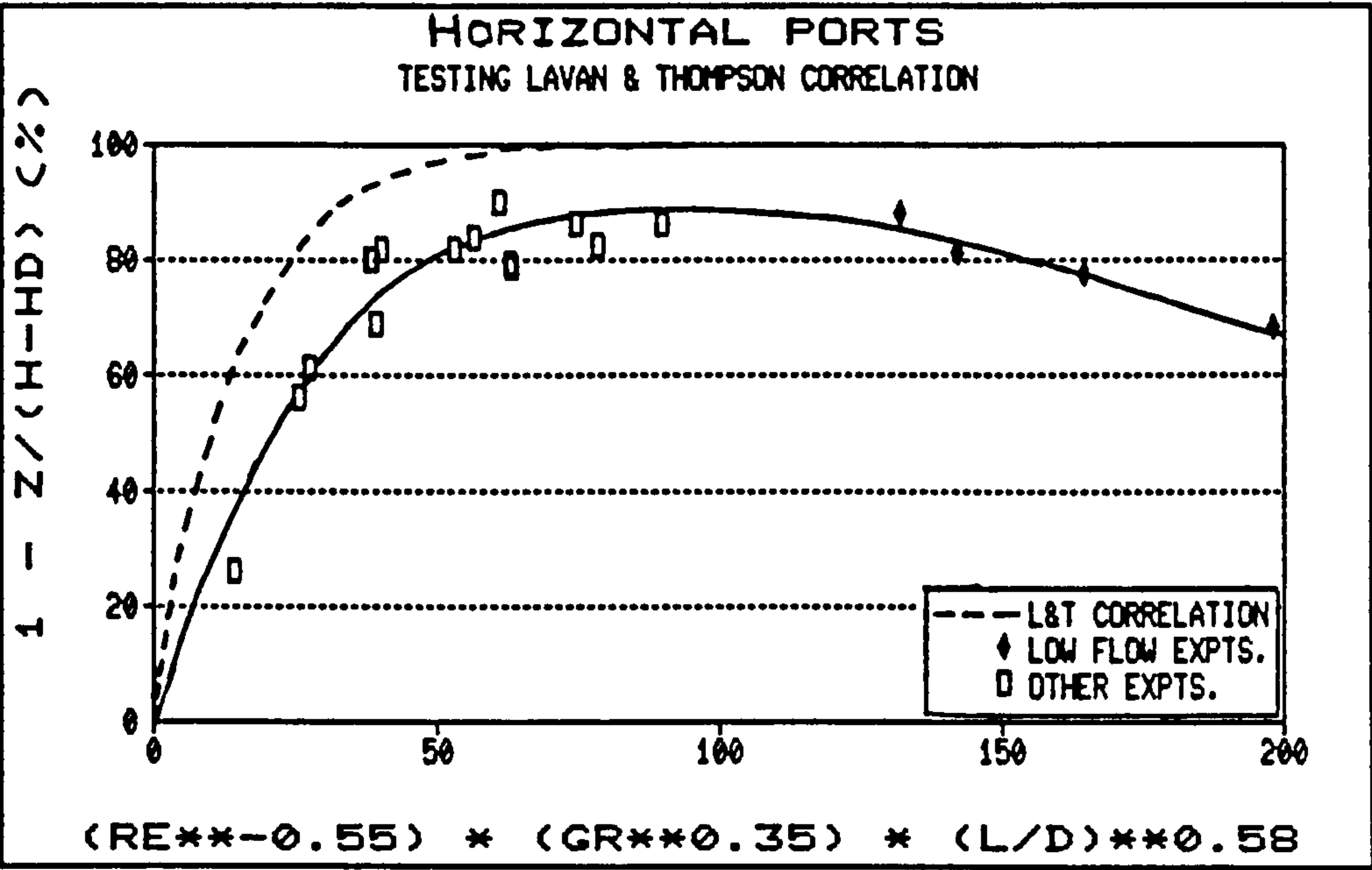


FIGURE 4.17 DATA FOR EXPERIMENTS WITH HORIZONTAL PORTS
COMPARED TO LAVAN AND THOMPSON CORRELATION .

by the fact that $t^* > t_R$. The rest may be due to this work using a pressurised tank (cf. an open-topped vessel) and to scale effects.

- (iii) Their correlation lacks the robustness of being derived from theory (cf. the Turner parameter obtained by a dimensional analysis), and its applicability to full scale tanks has essentially had doubt cast upon it.

(b) Sliwinski

The work of Sliwinski (ref. 2.19) also used horizontal ports; his results are shown in Figure 2.4. As mentioned in section 4.2.1.2, he used a modified Richardson number given by

$$Ri^* = \frac{g \beta \Delta T L}{v^2} \quad \text{..(35)}$$

where L is the separation of the inlet and outlet ports (m).

He calculated the following correlation for his experimental results:

$$\frac{L^*}{L} = 0.0313 (Ri^*)^{-1/2} \quad (0.001 \leq Ri^* \leq 10) \quad \text{..(36)}$$

where L^* is a length similar to the depth of the mixed region at the inlet port (m).

Using equation (13) it can be seen that

$$\frac{L^*}{r} = \frac{L}{r} \cdot \frac{L^*}{L} = \frac{L}{r} \cdot 0.0313 \left(\frac{L}{r} Ri \right)^{-1/2} \quad \text{..(37)}$$

Now, Sliwinski had $L = 0.88\text{m}$ and he used two different port diameters, 6.35 mm (1/4 in.) and 12.7 mm (1/2 in.).

Substituting these values into equation (37) gives

$$\begin{aligned} \frac{L^*}{r} &= 0.521 (Ri)^{-1/2} && \text{for the 1/4 in port} \\ \text{and } \frac{L^*}{r} &= 0.368 (Ri)^{-1/2} && \text{for the 1/2 in port} \end{aligned}$$

The correlation given by equation (30) means that the constant C in equation (11) is given by

$$C\pi^{-3/4} = 0.238$$

So that equation (12) applied to the data for the horizontal ports becomes

$$\frac{z}{r} = 0.748 (Ri)^{-1/2}$$

This expression is similar to the relationships for L^*/r above, especially given the difference in definition between z and L^* , and the scatter in Sliwinski's data, which might have yielded a different correlation constant to that in equation (36) had he used Ri instead of Ri^* .

Another interesting point of comparison concerns the limit for the correlation in equation (30) at very low flow rates. This occurs at a Turner Parameter of 0.2 m which corresponds to $Ri = 0.04$. Using Sliwinski's definition for Ri^* (equation (36)), this corresponds to $Ri^* = 11$ for the 1/4 in port and $Ri^* = 5.5$ for the 1/2 in port. Sliwinski gave his limitation as $Ri^* = 10$ (in fact, he shows only two data points with $Ri^* > 1.1$) which is in reasonable agreement.

4.2.3.3 . Horizontal Ports : Performance Coefficient Results

The performance coefficient, $\eta(\tau)$, for each horizontal port test, is plotted against the Turner parameter in Figure 4.18 and against the depth of penetration in Figure 4.19. In both graphs, the most noticeable features are the low values of $\eta(\tau)$, in spite of the relatively low penetration depths achieved. The cause of this is the deadspace in the vessel, which comprises about 13% of the volume and therefore effectively limits the maximum value of $\eta(\tau)$ to 87%. It is gratifying to see that the best-line fits to the data (excluding low flow rate experiments) on both graphs hit the $\eta(\tau)$ axis at around this value.

The deadspace volume could be reduced, by locating the horizontal ports closer to the ends of the vessel. However, it is possible that

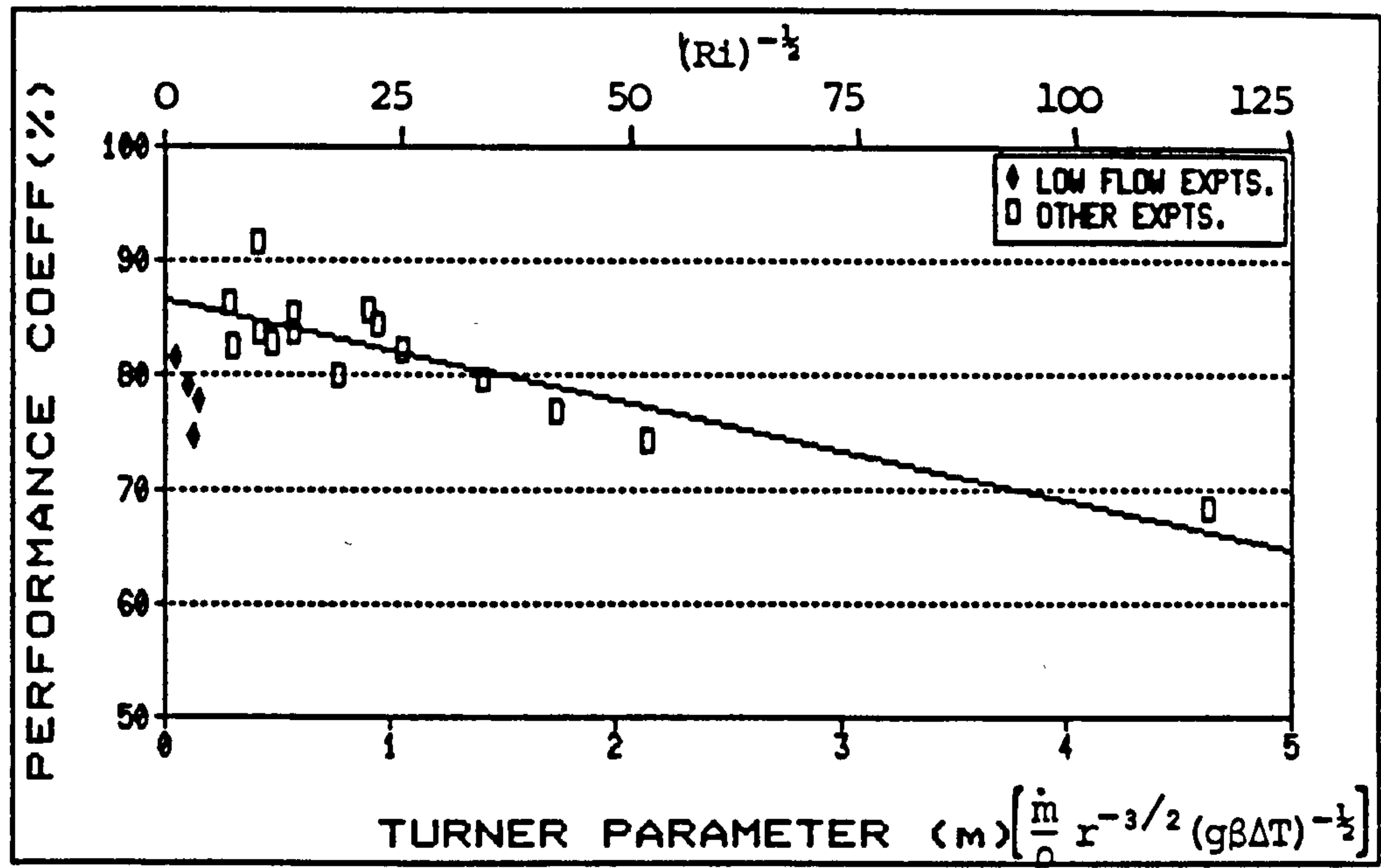


FIGURE 4.18 THE CORRELATION BETWEEN PERFORMANCE COEFFICIENT AND TURNER PARAMETER FOR THE EXPERIMENTS WITH HORIZONTAL PORTS.

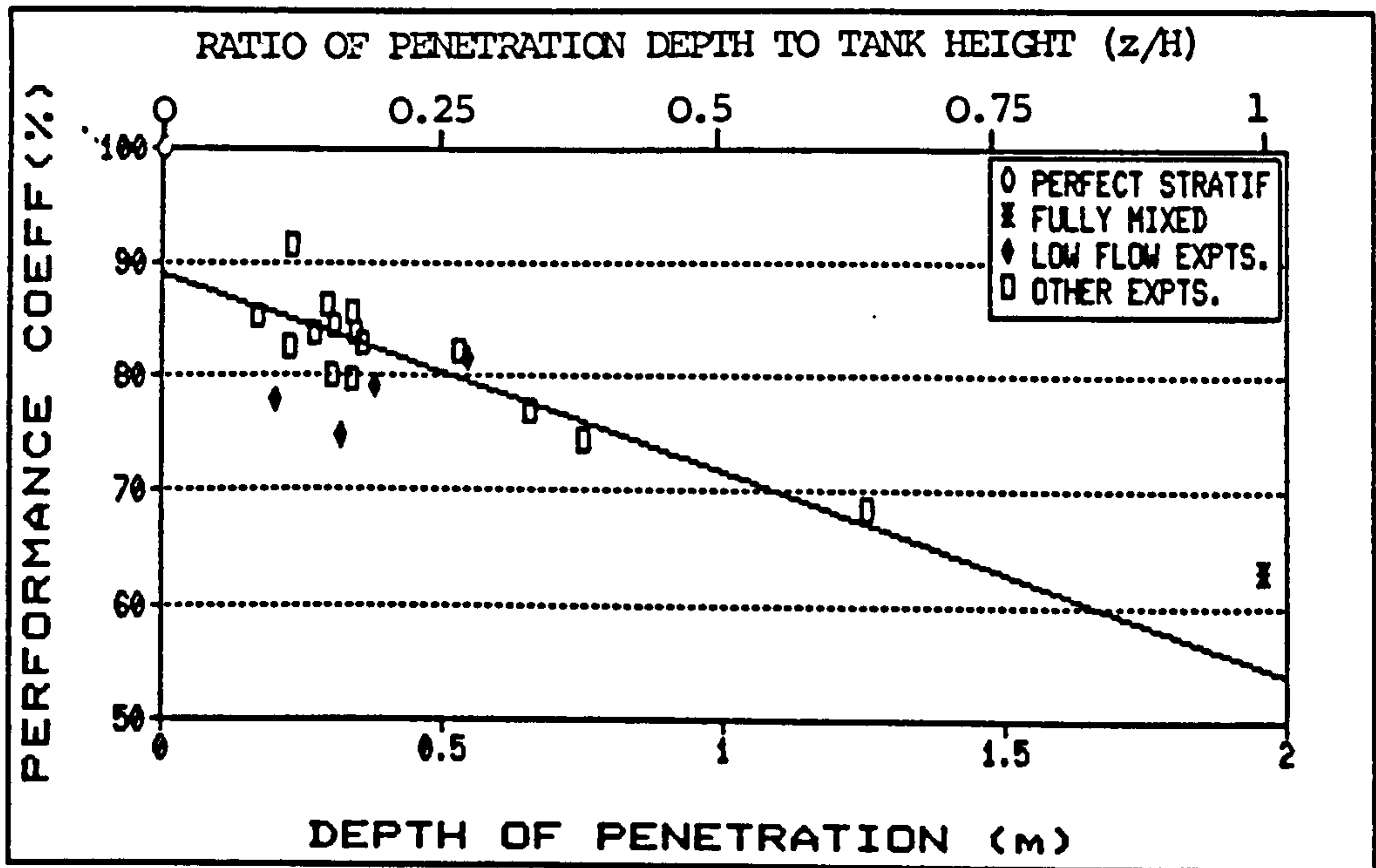


FIGURE 4.19 THE CORRELATION BETWEEN PERFORMANCE COEFFICIENT AND DEPTH OF PENETRATION FOR THE EXPERIMENTS WITH HORIZONTAL PORTS.

a decrease in the volume of the space between the inlet port and the tank end might reduce this region's capability to absorb the momentum of the inlet flow, and result in an increased depth of penetration. Therefore the performance coefficient would not necessarily increase proportionately.

4.2.4 Comparison of the Three Port Configurations

It is useful to compare the results of all three port configurations and Figure 4.20 shows the best-fit straight lines for each plot of penetration depth against Turner parameter. The reduced depth of penetration achieved by the distributors compared to the vertical ports has already been discussed. Rather unexpectedly, the horizontal ports show a further improvement compared to the distributors, and one which increases with the momentum of the inlet flow. It is assumed that this is due to the capability of the fairly large region between the horizontal inlet port and the tank end to absorb momentum.

The performance coefficient results for the three port configurations are given in Figures 4.21 and 4.22, as a function of the Turner parameter and penetration depth respectively.

The distributors show a clear improvement over the other two configurations, although it seems possible that their increasing advantage over the horizontal ports as the Turner parameter decreases could be narrowed if the deadspace which occurred with the latter were reduced as discussed above (section 4.2.3.3).

The same measure might also shift vertically upwards the horizontal ports line on Figure 4.22, making it roughly coincide with the other 2 lines, confirming the hypothesis that, for any configuration avoiding deadspace, $\eta(\tau)$ is fixed by the depth of penetration.

In summary,

- vertical ports should be avoided;

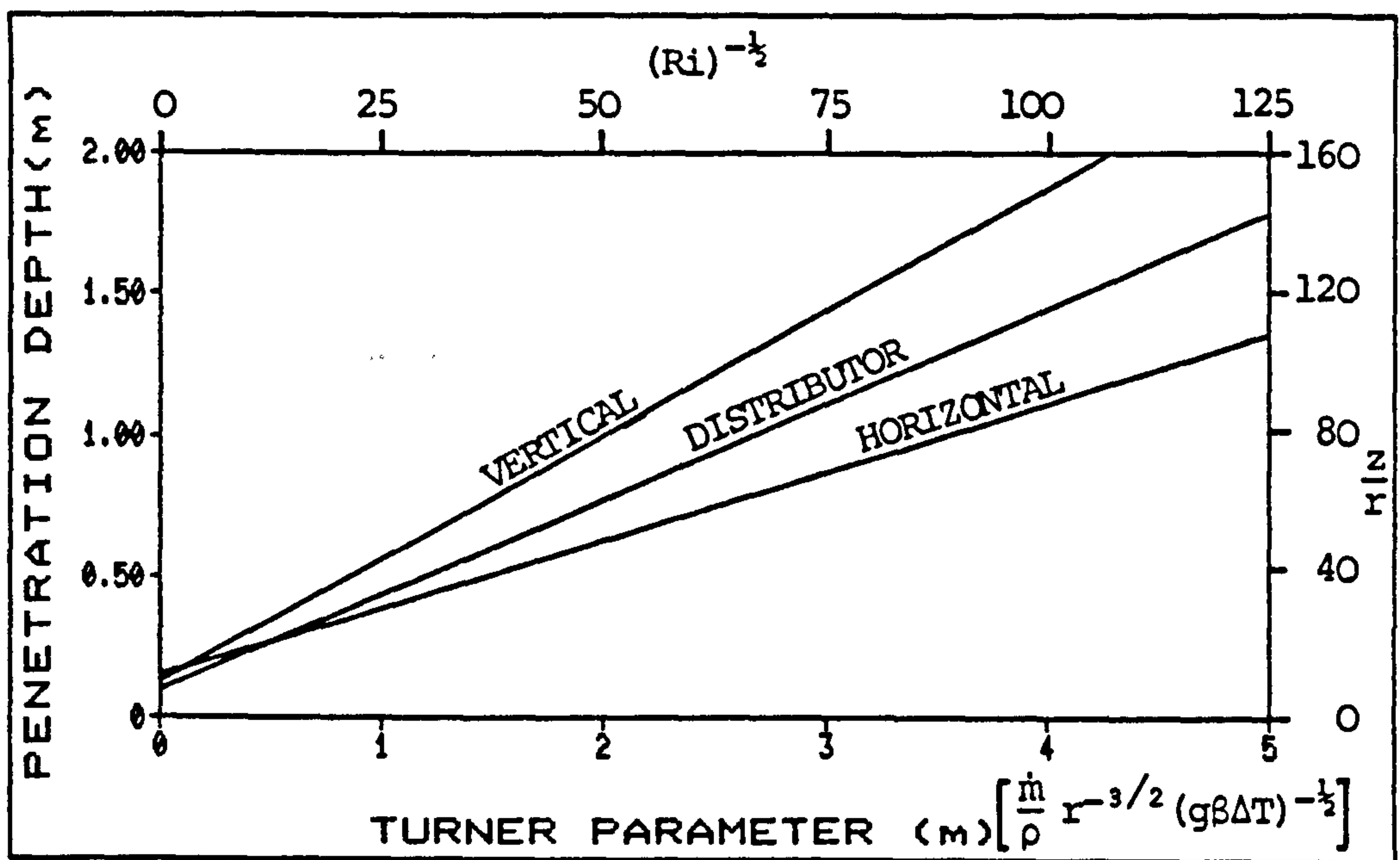


FIGURE 4.20 COMPARISON OF THE PENETRATION DEPTH VS. TURNER PARAMETER CORRELATIONS FOR THE 3 PORTS.

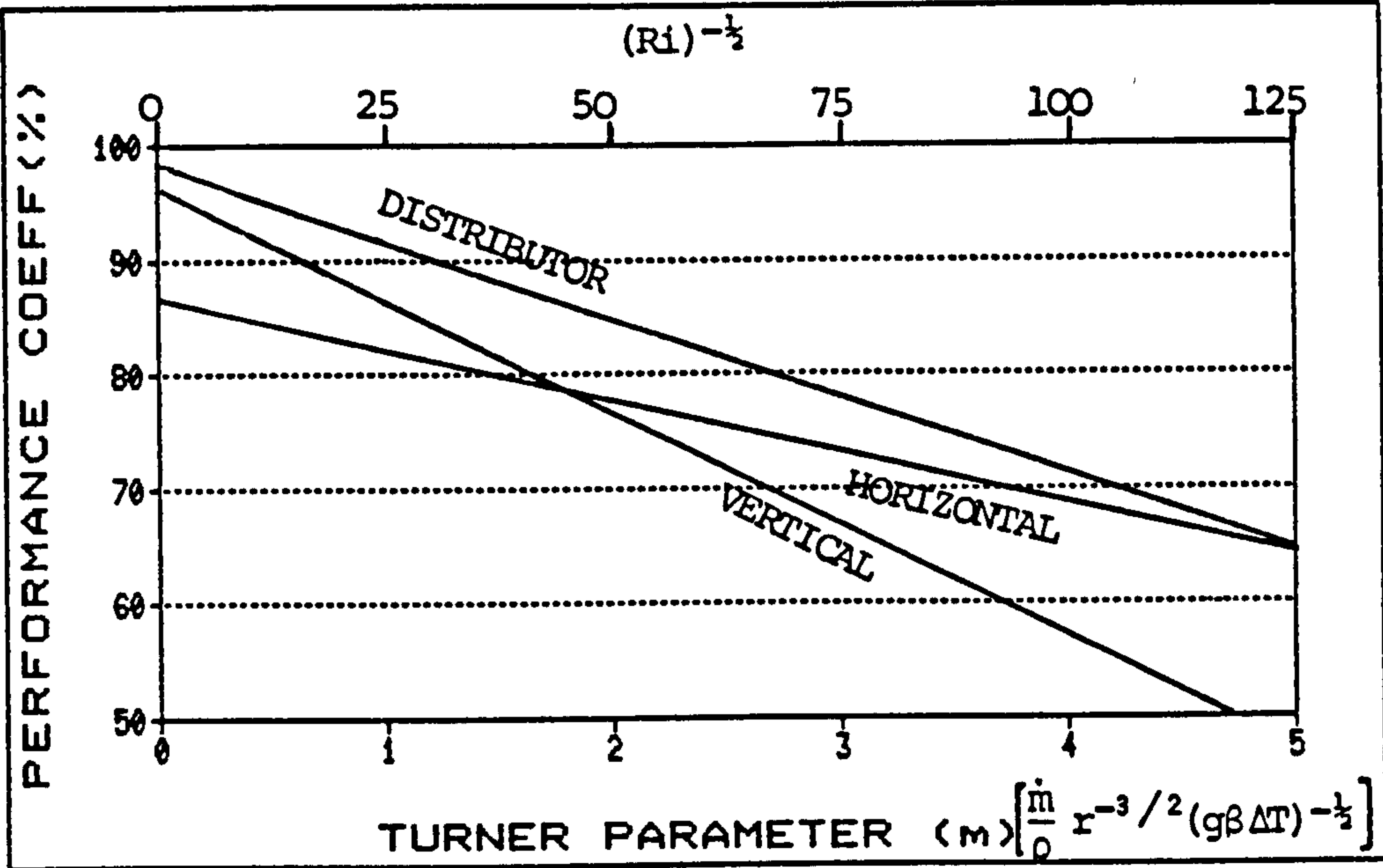


FIGURE 4.21 COMPARISON OF THE PERFORMANCE COEFFICIENT VS. TURNER PARAMETER CORRELATIONS FOR THE 3 PORTS .

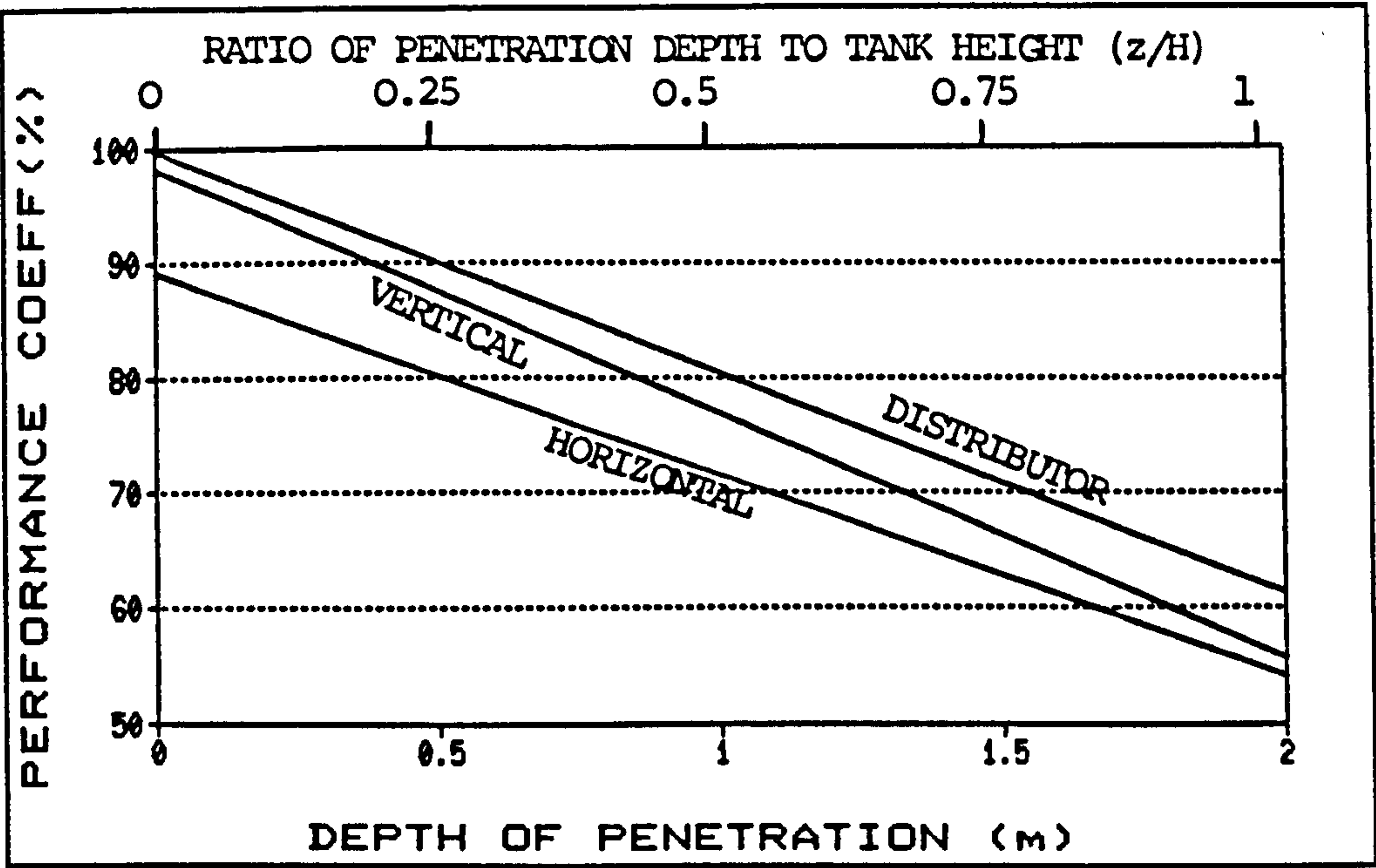


FIGURE 4.22 COMPARISON OF THE PERFORMANCE COEFFICIENT VS. PENETRATION DEPTH CORRELATIONS FOR THE 3 PORTS .

- horizontal ports are a better option if excessive deadspace can be avoided;
- distributors work satisfactorily, though maybe not as well as had been expected; their undoubted extra cost would have to be carefully justified.
- the performances of all three port configurations under different turbulent flow conditions follow closely the trend predicted by equation (11).

Afterword

One option that was not assessed in this work was that of using 2 different port geometries on the same vessel. One such apparently attractive hybrid would comprise a horizontal port for the inlet, thereby gaining its superior performance in terms of stratification compared to a vertical port, and a vertical port for the outlet, thereby avoiding the deadspace problem associated with horizontal outlet ports. This combination would undoubtedly achieve a superior performance compared with 2 horizontal or 2 vertical ports. However it has a major practical drawback: a store is usually connected to both a heat supply and a heat demand, and the use of different port geometries for inlets and outlets would necessitate the use of 4 ports instead of 2. The advantages gained by employing these extra connections would therefore have to be balanced against their cost and the cost of the considerable extra associated pipework that would be required.

4.3 TEMPERATURE EQUALISATION TEST RESULT

It was important to determine accurately the magnitude of the heat conduction effects within the store. The best way to do this was to perform an experiment on the store in which heat conduction had a dominant influence, and then use the result to validate a theoretical representation of the experiment, such as a numerical model. The model could then be used to assess conduction under other circumstances, or with different store sizes.

It was decided therefore to carry out a test under standstill conditions, to remove the effect of flow through the store, and at temperatures close to the ambient surroundings, to minimise the influence of heat losses. To generate significant heat conduction effects, a sharp thermocline was set up in the store, so that the experiment started with the top 60% of the store at about 25°C, the bottom 30% at about 16°C and a sharp temperature gradient joining these two regions.

The resulting temperature distributions in the store over the following 144 hours are shown in Figure 4.23. They are represented by the tank centreline thermocline, plotted at 24 hour intervals. The tendency for temperature equalisation can be seen, with the temperature of the upper region falling and that of the lower region rising, as heat is transferred from one to the other. The pivot point marking the boundary between these two regions is shown to sit at about 35% up the height of the tank. At this boundary the temperature remains roughly static throughout the experiment. The warmer upper region naturally loses more heat to the environment than the lower region, and so its temperature is seen to fall more rapidly than the lower region's rises. A full theoretical analysis of this test is included in the next chapter.

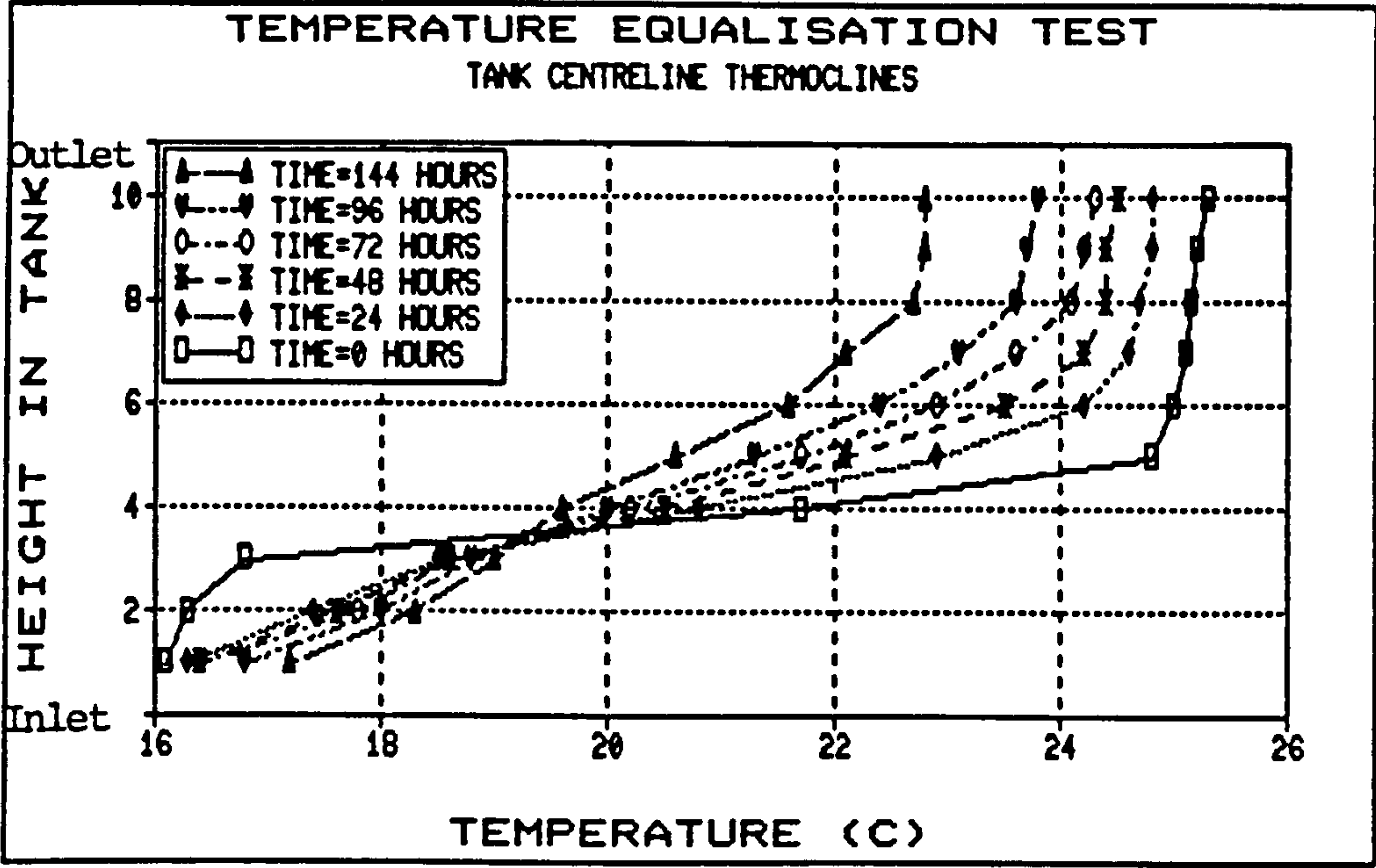


FIGURE 4.23 THE MEASURED STORE THERMOCLINES DURING THE TEMPERATURE EQUALISATION TEST.

CHAPTER FIVEDEVELOPMENT AND VALIDATION OF A NUMERICAL MODEL5.1 DESCRIPTION OF THE MODEL5.1.1 Introduction

A numerical model of a stratified hot water store was primarily required to assess via computer simulations the annual performance of systems incorporating such stores. It was necessary therefore to develop a model which represented realistically stratification in the store, and yet which was not too expensive in terms of computer time. The most suitable modelling method for both the store and all the other components in the system was one using an explicit, finite difference technique, with time steps of between about 15 seconds and 1 hour depending upon system flow rates and which components were in operation (see Chapter 6). These constraints dictated that a reasonably sophisticated, yet one-dimensional store model was most appropriate.

A promising approach suggested by the experimental results would be to divide the store into a large number (i.e. $\gg 10$) of isothermal equal volume horizontal segments, or zones. When there was a flow up or down the store, some of these zones, adjacent to the inlet of the store, would be assumed to be fully-mixed to a uniform temperature. The size of this mixed region would be determined by applying the experimental correlations for the depth of penetration to the flow conditions being simulated at the time. Perfect plug-flow would be assumed for the remainder of the store. The principle of such a store algorithm is illustrated in Figure 5.1. The model would also take account of heat losses from each zone to the ambient surroundings, and heat conduction between zones, including an allowance for the effect of the tank walls.

This chapter describes the development of such a model, applied specifically to the store used in the experimental work. The model is then used to simulate the temperature equalisation test and step input tests, and the predictions of temperature distributions, outlet

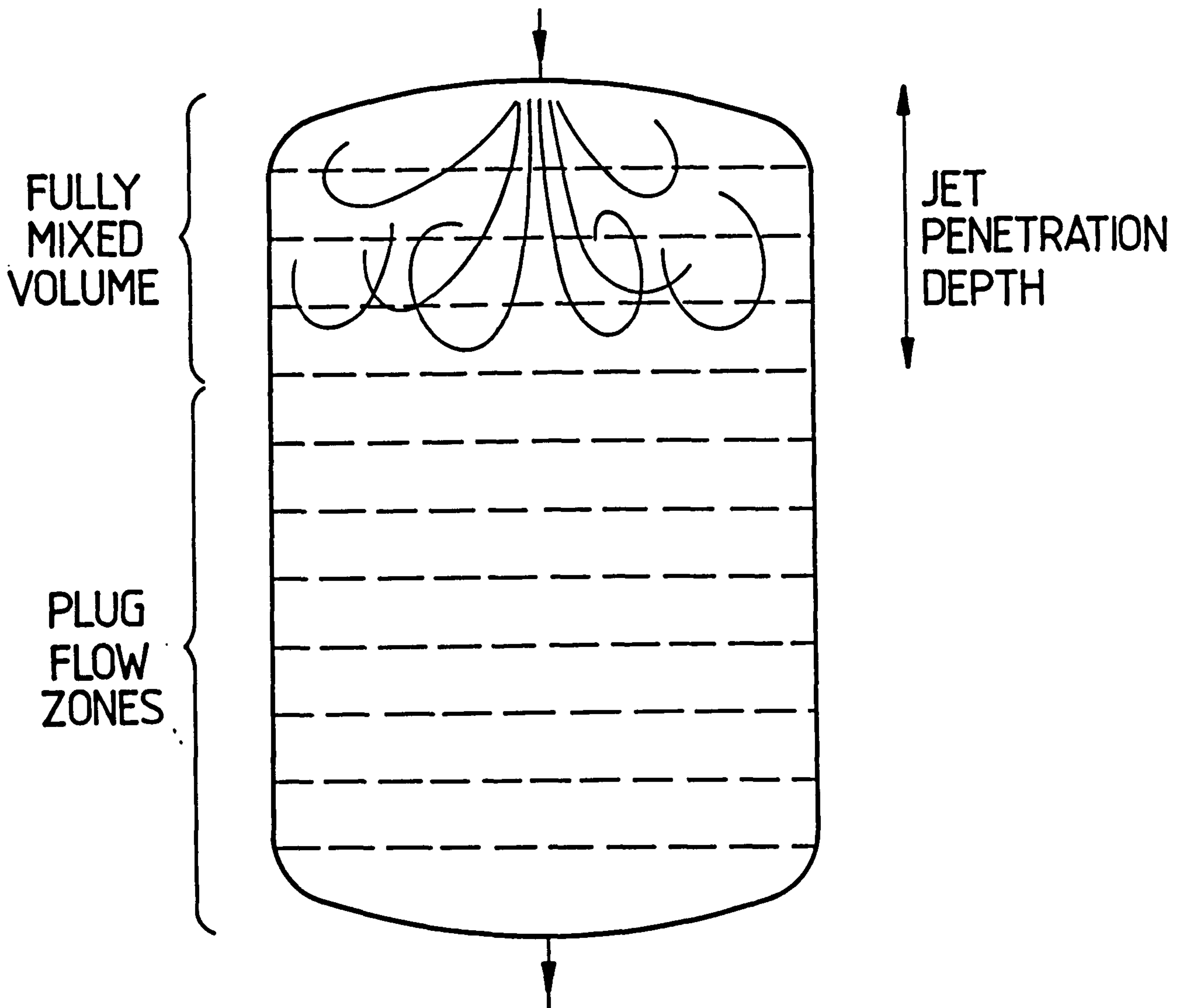


FIGURE 5.1 THE BASIC CONCEPT FOR A ONE-DIMENSIONAL NUMERICAL MODEL OF A STRATIFIED HOT WATER STORE (ILLUSTRATED FOR THE STORE BEING CHARGED).

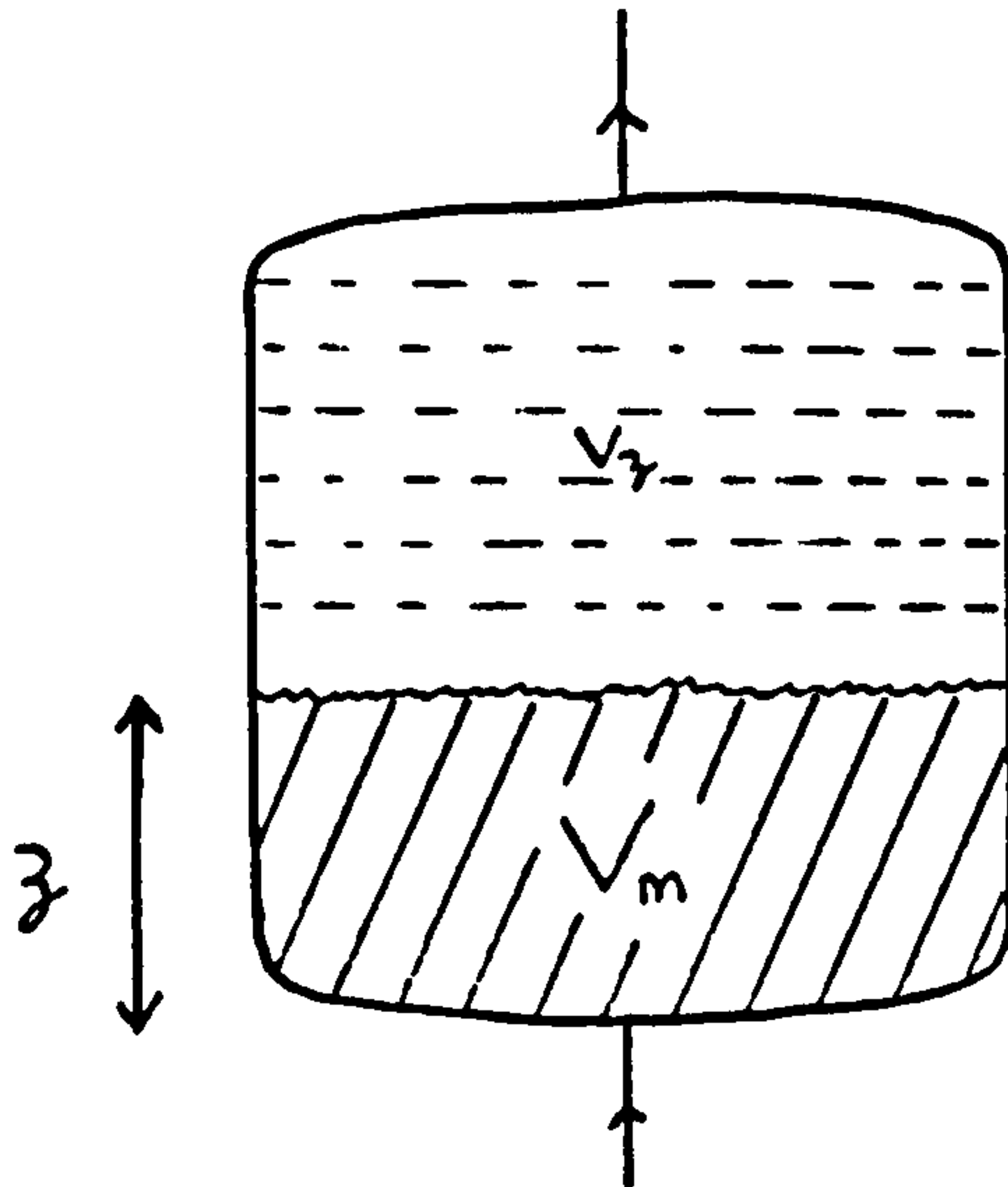
temperatures and performance coefficients are compared to the measured values. In the next chapter, the model is integrated into an annual simulation of a complete heating system incorporating a hot water store.

5.1.2 Modelling the Fully-mixed Region

The penetration depth, z , into the store of the inlet flow at any time is defined by the experimental correlations given in Chapter 4 for each port configuration. The volume of the fully-mixed region, V_m , (see sketch below), is therefore defined by :

$$V_m = z A \quad \dots(1)$$

where $A = \frac{V}{H}$ = mean cross-sectional area of the store (m^2)



Hence, the number of zones, N_m , in this region is given by

$$N_m = \text{INT} \left(\frac{V_m}{V_z} \right) \quad \dots(2)$$

where INT signifies integer value of

and V_z is the volume of a single zone, defined by

$$V_z = \frac{V}{N} \quad \dots(3)$$

where V = the total store volume (m^3)
 and N = the total number of zones

Inserting (1) and (3) in (2) gives

$$N_m = \text{INT} \left(z \frac{V}{H} \frac{N}{V} \right) = \text{INT} \left(N \frac{z}{H} \right) \quad \dots(4)$$

After the rounding that occurs in (4) it is necessary to redefine V_m

$$\text{as } V_m = N_m \cdot V_z \quad \dots(5)$$

Having determined the extent of the fully-mixed region, it is necessary to define its temperature, T_m , as the mean temperature of all the zones in the region. Thus

$$T_m = \frac{\sum_{i=1}^{N_m} T_i}{N_m} \quad \dots(6)$$

where T_i is the temperature of the i^{th} zone ($^{\circ}\text{C}$).

(The description of the model given in this chapter assumes, simply for example, that the store is being discharged, so that the flow enters at the bottom of the store and goes upwards. The zones are numbered from 1 at the inlet (the bottom) to N at the outlet (the top). The model would apply equally to a store charge where of course the inlet would be at the top and the outlet at the bottom).

After calculating T_m , the temperatures of the fully-mixed zones must be reset to this value

$$\text{i.e. } T_i = T_m \quad (1 \leq i \leq N_m)$$

Neglecting, for the moment, heat loss to the surroundings (see Section 5.1.6) and heat conduction (see Section 5.1.5), the energy balance for the mixed region during flow is given by

$$-\rho V_m C_p \frac{\partial T}{\partial t} = \rho \dot{V} C_p (T - T_{in}) \quad \text{..(9)}$$

This equation must be solved analytically rather than numerically to ensure that this region is truly fully-mixed for any value of the time step. As shown in Appendix D.1, therefore, one finds

$$T - T_{in} = (T_o - T_{in}) e^{-t/\tau_m} \quad \text{..(10)}$$

where $\tau_m = \frac{V_m}{\dot{V}}$ = the fill time of the mixed zone and T_o is the initial temperature of the mixed zone.

Rearranging (10), one gets

$$T = T_o - \left(1 - e^{-t/\tau_m}\right) (T_o - T_{in})$$

which can be rewritten in a finite-difference form as

$$T_{t+1} = T_t - \left(1 - e^{-\Delta t/\tau_m}\right) (T_t - T_{in}) \quad \text{..(11)}$$

where ΔT is the time step between time t and time $t+1$.

Equation (11) thus defines the new temperature of the mixed region. It is also necessary to calculate the temperature, T_{mout} , of the flow leaving the mixed region during the time step, and entering the plug flow region. To ensure conservation of energy, a time-averaged value for T_{mout} must be taken, using equation (9). Thus

$$-\rho V_m C_p \left(\frac{T_{t+1} - T_t}{\Delta t} \right) = \rho \dot{V} C_p (\bar{T}_{mout} - T_{in})$$

Using (11) and rearranging :

$$\bar{T}_{mout} = T_{in} + \frac{\tau_m}{\Delta t} \left(1 - e^{-\Delta t/\tau_m}\right) (T_t - T_{in}) \quad \text{.. (12)}$$

5.1.3 Modelling the Plug Flow Region

The energy balance for a plug flow zone during flow is of the same form as equation (9), i.e.

$$- \rho V_z C_p \frac{\partial T}{\partial t} = \rho \dot{V} C_p (T - T_{in})$$

In this case the equation is solved numerically by taking a finite difference approximation for the partial derivative. Hence for the i^{th} zone, using a forward difference formula :

$$-V_z \left(\frac{T_i^{t+1} - T_i^t}{\Delta t} \right) = \dot{V} (T_i^t - T_{i-1}^t)$$

where T_i^t = the temperature of the i^{th} zone at time t ,

$$\text{and } T_i^{t+1} = \frac{\dot{V} \Delta t}{V_z} T_{i-1}^t + \left(1 - \frac{\dot{V} \Delta t}{V_z} \right) T_i^t \quad \dots(13)$$

The factor $\frac{\dot{V} \Delta t}{V_z}$, (known as the Courant number [Ref.5.1], determines the degree of mixing between adjacent zones and must not be greater than 1 to ensure the stability of equation (13). To achieve perfect plug flow, no mixing is wanted and the Courant number must be set equal to 1, that is the time step must be set as

$$\Delta t = \frac{V_z}{\dot{V}} \quad \dots(14)$$

which is simply the fill time of a zone.

The numerical solution of equation (13) then becomes plainly

$$T_i^{t+1} = T_{i-1}^t \quad \dots(15)$$

This equation applies to all the plug flow zones (i.e. for $N_m + 1 \leq i \leq N$) although in the case of the bottom plug flow zone ($i = N_m + 1$) the incoming flow is from the mixed region and (15) becomes

$$T_{N_m+1}^{t+1} = \bar{T}_{mout} \quad (N_m < N)$$

where $\bar{T}_{mout}$ is as given in equation (12).

The outlet temperature from the store, T_{out} , is given by the temperature of the top plug flow zone or, in the case of the whole store being fully-mixed, by the temperature of the flow leaving the mixed region. Thus

$$T_{out} = T_{N,t} \quad N_m < N$$

$$T_{out} = \bar{T}_{mout} \quad N_m = N$$

5.1.4 Selecting the Number of Zones

From the description of the model so far, the choice of into how many zones to split the store is clearly important. From equation (4) it can be seen that the more zones there are, the closer will be the match between the depth of the fully-mixed region and the calculated value, z . It should be emphasised here that the size of the fully-mixed region tends to increase throughout a step input test, as the temperature difference between the inlet flow and the mixed region reduces. Thus if there are a large number of zones, the rounding in equation (4) will allow the size of the mixed region to increase in small jumps.

A similar situation applies to the outlet temperature response : the greater the number of zones, the smoother the response will be. This is not to say that the outlet temperature is effected ~~per se~~ by how

many zones there are, because it is an important feature of this model, consequent on the strict definition of the time step, that the outlet temperature at any given time is independent of the choice of number of zones (except in so far as it determines the rounding error in the size of the mixed region) and of the time step. Rather, it can be seen by combining (3) and (14) that

$$N = \frac{V}{\dot{V}\Delta t} = \frac{\tau}{\Delta t} \quad \dots(16)$$

where τ is the fill time of the store defined in the model to be also the charge or discharge period. Thus the number of zones is the same as the number of time steps there are up to time τ , in a step input test. Hence the more zones there are, the more points there will be on the outlet temperature curve up to a given time, and the more reliable will be the calculation of a performance coefficient (see Section 5.2).

A large number of zones is therefore clearly desirable, but there is a restraint imposed by the time and expense of running the model on a computer. In fact, the computer time, t_{comp} , is proportional to the number of zones, since an individual calculation is required for each zone, and also is inversely proportional to the time step, since a complete set of calculations is necessary at each time step.

Hence, $t_{\text{comp}} \propto \frac{N}{\Delta t}$

and inserting (16) : $t_{\text{comp}} \propto N^2$.

For the purposes of modelling the step input tests in this work, it was felt that dividing the store into about 20 zones was a fair compromise. The final choice was actually made so as to facilitate the validation of the model, by making the position of the zones correspond to the location of the thermocouples in the store. The horizontal lines of thermocouples were equally spaced up the height of the store (see Figure 4.7) and each effectively represented a volume of 261.5 litres. At the top and bottom of the store there were further volumes of 118.5 litres each (see Figure 5.2), to make up the total store volume of 2852 litres.

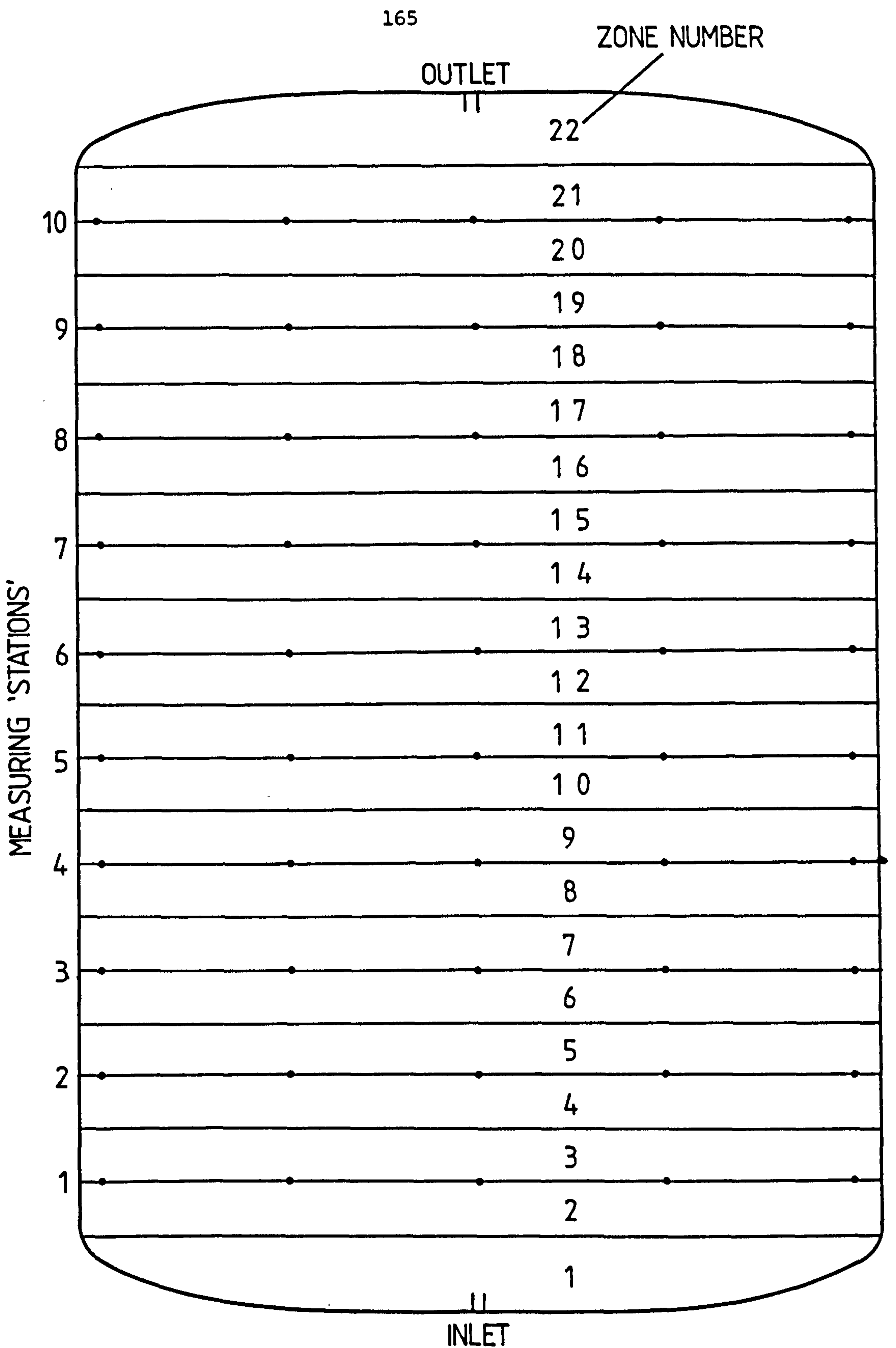


FIGURE 5.2 THE CORRESPONDENCE OF THE MEASURING POINTS WITH THE ASSUMED ZONES OF THE STORE.

It was decided to divide the store into 22 zones, each with a volume of about 130 litres. This meant that each measuring point lay almost on the boundary between two zones, the mean temperature of which could be taken as the predicted temperature for that point. The bottom (1st) and top (22nd) zones corresponded closely to the bottom and top volumes respectively in the store, as shown in Figure 5.2.

If the measurement points are numbered from 1 at the bottom to 10 at the top, it can be seen that the predicted temperature, $TEMP(I)$, for the I^{th} measuring point is given by

$$TEMP(I) = (T_{2I} + T_{2I+1})/2$$

where T_{2I} is the predicted temperature for the $2I^{th}$ zone.

5.1.5 Modelling Heat Conduction Between Zones

Heat conduction takes place in a store wherever there is a temperature gradient. The one-dimensional model only considers vertical gradients, and since temperature inversions are destroyed as soon as they occur (see Section 5.1.7), the heat flow is always downwards. The net conductive heat flow into a zone is the difference between the heat gained from the zone above and the heat lost to the zone below. The rate of heat flow between two zones is given by Fourier's Law :

$$\dot{Q} = -kA \frac{dT}{dx} \quad \dots(17)$$

where x represents distance in the vertical direction.

The rate of change of temperature of a zone due to heat conduction is given by Fourier's equation, derived from (17), and applied to one-dimensional conduction :

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t}$$

where $\alpha = \frac{k}{\rho C_p}$ = thermal diffusivity of water.

This equation can be solved by applying finite difference approximations to the partial derivatives, so that for the i^{th} zone :

$$\frac{T_{i+1}^t - 2T_i^t + T_{i-1}^t}{\Delta x^2} = \frac{1}{\alpha} \frac{T_i^{t+1} - T_i^t}{\Delta t} \quad \dots(18)$$

where $\Delta x = \frac{V_z}{A} =$ the height of one zone,

and again a forward difference formula has been adopted for the derivative with respect to time. Rearranging to get the new temperature of zone i in terms of current temperatures :

$$T_i^{t+1} = (1 - 2C) T_i^t + C(T_{i+1}^t + T_{i-1}^t) \quad \dots(19)$$

where $C = \frac{\alpha \Delta t}{\Delta x^2}$, a constant, and $2 \leq i \leq 21$

To ensure the stability of (19), C must not be greater than 0.5.

Now, $\alpha \sim 3 \cdot 10^{-7} \text{ m}^2 \text{ s}^{-1}$ (taking the "effective" conductivity for water given in equation (24)),

and with 22 zones, $\Delta x = 0.089\text{m}$.

So for stability, $\Delta t \leq 13200 \text{ s}$ or 3.7 hours.

Now when there is flow in the store Δt is determined by (14), so the

stability criterion applies to the flow rate $\dot{V}$

$$\text{i.e. } \dot{V} \geq \frac{V_z}{\Delta t}$$

Now $V_z = 0.13 \text{ m}^3$ so $\dot{V} \geq 0.01 \text{ litres/s}$.

In fact the lowest flowrate considered in the experimental work was about four times this minimum value, and so equation (19) can be used freely in the validation process.

Now, as stated, (19) only applies to internal zones ($2 \leq i \leq 21$), where heat is flowing in from above and out from below. For the top zone ($i = 22$), (18) becomes :

$$-\left(\frac{T_{22}^t - T_{21}^t}{\Delta x^2}\right) = \frac{1}{\alpha} \frac{T_{22}^{t+1} - T_{22}^t}{\Delta t}$$

So that $T_{22}^{t+1} = (1 - C) T_{22}^t + C T_{21}^t$

Similarly for the bottom zone ($i = 1$), (18) becomes :

$$\frac{T_2^t - T_1^t}{\Delta x^2} = \frac{1}{\alpha} \frac{T_1^{t+1} - T_1^t}{\Delta t}$$

So that $T_1^{t+1} = (1 - C) T_1^t + C T_2^t$

5.1.6 Modelling Store Heat Losses

The store loses heat to the ambient surroundings via conduction through the tank wall and insulation. The energy balance for one zone losing heat in this way is given by :

$$\rho V_z C_p \frac{\partial T}{\partial t} = U A_z (T - T_{amb}) \quad \dots (20)$$

where A_z is the surface area of the zone wall (m^2)
 U is the overall heat loss coefficient through the wall and insulation ($Wm^{-2} K^{-1}$)
 and T_{amb} is the temperature of the ambient surroundings ($^{\circ}C$)

Applying a finite difference approximation to the partial derivative in (20) gives for the i^{th} zone :

$$\rho V_z C_p \frac{T_i^{t+1} - T_i^t}{\Delta t} = -U A_z (T_i^t - T_{amb})$$

which can be rearranged to give

$$T_i^{t+1} = T_i^t - \frac{U A_z \Delta t}{\rho V_z C_p} (T_i^t - T_{amb}) \quad \dots (21)$$

Equation (21) applies to all zones as long as the appropriate value for A_z is taken. For simplicity the model assumes the store to be a perfect cylinder for the purposes of heat loss. Therefore, for the 'internal' zones ($2 \leq i \leq 21$),

$$A_z = \pi D \frac{V_z}{A}$$

and for the 'extreme' zones ($i = 1, i = 22$),

$$A_z = \pi D \frac{V_z}{A} + A.$$

For the model to produce the same overall rate of heat loss from the store as had been measured in the standing heat loss test described in Section 3.4.1.6, it was necessary to set the value of U so that

the product $U \sum_{i=1}^{22} A_z$ equalled the measured value of 4 W K^{-1} . This means

$$U = \frac{4}{\sum A_z}$$

or

$$U = \frac{4}{20 \pi D \frac{V_z}{A} + 2 \left(\pi D \frac{V_z}{A} + A \right)}$$

5.1.7 Modelling Temperature Inversions

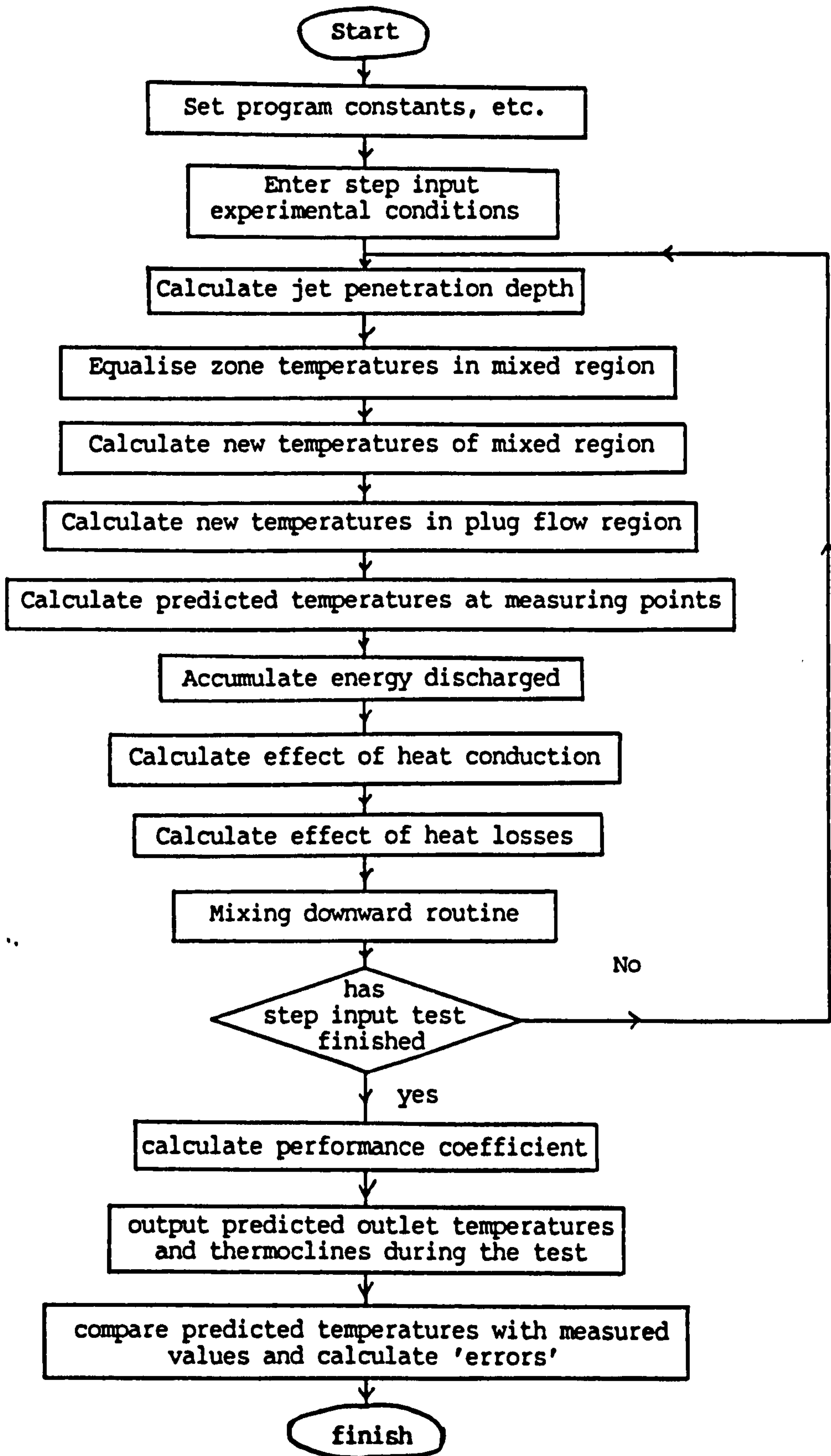
Temperature inversions - when cooler water finds itself above warmer water - can occur temporarily in practice, for example, if cool water is injected into the top of the store, or when heat losses through the top surface of the store lead to the water at the top becoming

cooler than the water below. Natural convection generated by buoyancy forces acts effectively to dispel this unstable situation, with the general result that regions of the store where inversions exist become well-mixed. In both step input and temperature equalisation tests, inversions could occur owing to heat losses from the top surface of the store, and so it was necessary to cater for them in the model, by incorporating a 'mixing downward routine'.

In the model, the temperature distribution in the store at the end of each time step is tested for inversions. The process starts at the top of the store : if the temperature of the top zone is less than that of the zone below, then the temperatures of the top two zones are set to the mean temperature of both zones, simulating the two zones being mixed. This new mean temperature is then compared to the temperature of the next zone down, and if it is again lower, the top 3 zones are mixed. This routine progresses down the store until the mixed region becomes warmer than the zone below and the inversion is destroyed.

5.1.8 Implementation of the Model on a Computer

The algorithms for simulating the store temperature distribution, as described in this Chapter, were written into a FORTRAN computer program called MIXPLUG, which was run on the VAX computer at Cranfield. The routines for flow through the store, heat conduction, heat losses and downward mixing were applied sequentially at each time step, to obtain the combined effect of all of them. A summary flow diagram of the program is shown on the following page:



5.2 RESULTS OF MODELLING THE TEMPERATURE EQUALISATION TEST

5.2.1 Validation of the Model

A modified version of MIXPLUG, which left out the routines modelling flow through the store, was implemented to simulate the temperature equalisation test described in Section 4.3.

The first task of this modelling exercise was to determine an optimum expression for the thermal conductivity k , first mentioned in (17). The thermal conductivity of water k_w , increases with temperature as shown in Figure 5.3. A quadratic expression which fits this curve extremely well was found to be

$$k_w(T) = 0.56915 + 1.846 \cdot 10^{-3} T - 7.4 \cdot 10^{-6} T^2 \quad \text{..(22)}$$

As mentioned in Chapter 2 (see figure 2.2), other research [Ref.2.5] has determined that the effect of heat conduction down the tank wall can be taken into account approximately by using an enhanced composite conductivity k_e given by

$$k_e = \frac{k_w A_w + k_t A_t}{A_w + A_t} \text{ as long as } \frac{k_t A_t}{k_w A_w} < 3 \quad \text{..(23)}$$

where the subscript "t" designates the tank wall.

For this tank* $\frac{k_t A_t}{k_w A_w} = 0.885$, so the expression is applicable.

Also $A_w \gg A_t$ so that (23) can be rewritten

$$\frac{k_e}{k_w} = 1 + \frac{k_t A_t}{k_w A_w}$$

and the enhanced conductivity is given by $k_e = 1.885 k_w(T)$. ..(24)

*FOOTNOTE

Assuming the following values :

$$k_w = 0.64 \text{ Wm}^{-1} \text{ K}^{-1}$$

$$k_t = 32 \text{ Wm}^{-1} \text{ K}^{-1}$$

and

$$\frac{A_t}{A_w} = \left(\frac{1372}{1360} \right)^2 - 1 = 0.0177$$

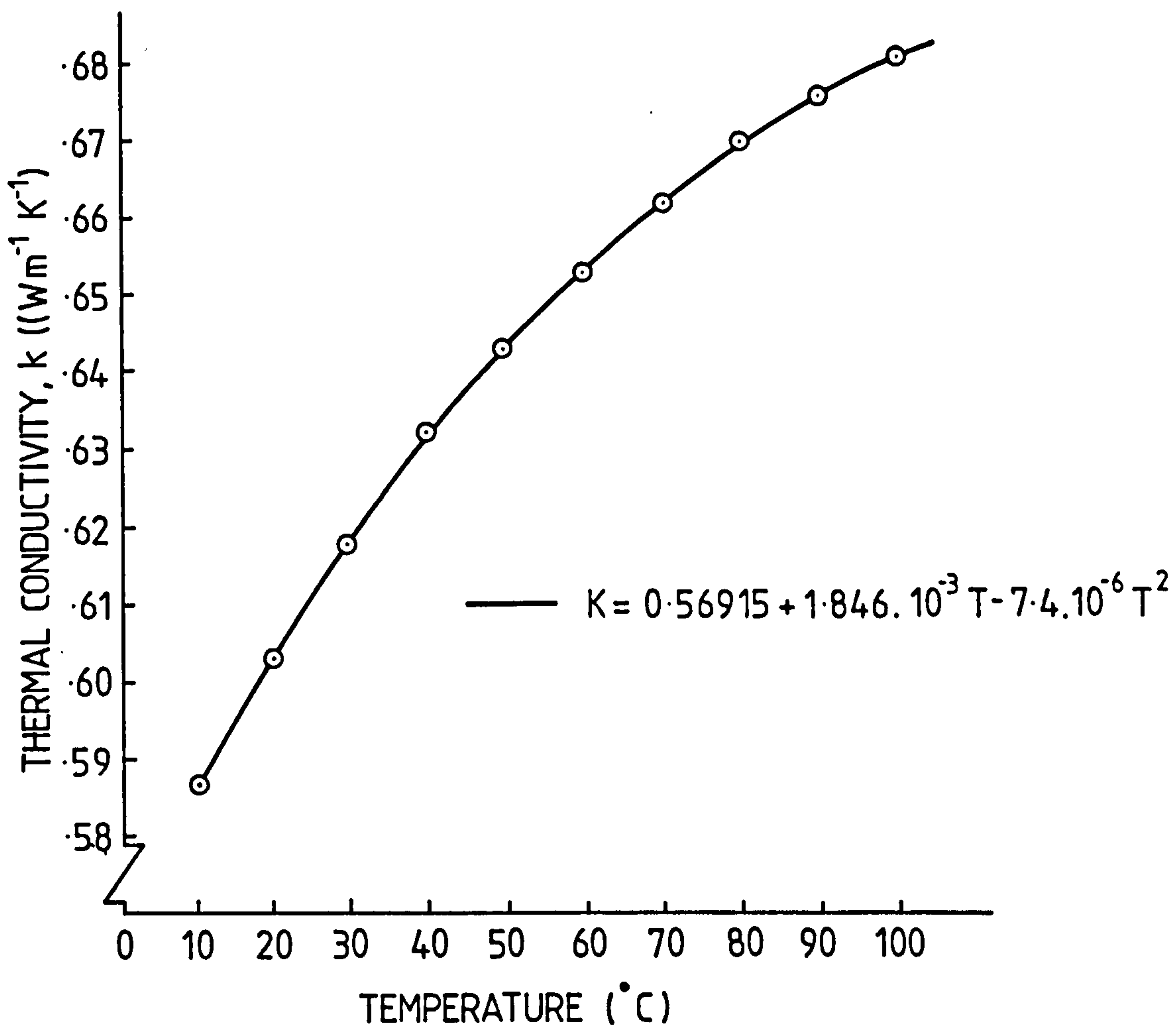


FIGURE 5.3 THE THERMAL CONDUCTIVITY OF WATER AS A FUNCTION OF TEMPERATURE (DATA FROM REF. 3.1).

The program was run using 4 expressions for k :

$$\begin{aligned} k &= k_w \\ k &= k_e \\ k &= 2 k_e \\ \text{and } k &= 0. \end{aligned}$$

The predicted temperature distributions after 144 hours were compared with the measured values, and the error of the simulation was defined as :

$$\text{Error} = \frac{\sum_{I=1}^{10} \text{ABS} \left[\text{TEMP}_I^{144} - \text{TACT}_I^{144} \right]}{\sum_{I=1}^{10} \text{TACT}_I^{144}} \cdot 100\%$$

where TEMP_I^{144} is the predicted temperature at the I^{th} measuring point after 144 hours and TACT_I^{144} is the corresponding measured value.

The values of this error obtained with the above 4 expressions for conductivity are plotted on the graph in Figure 5.4. The minimum error, of only 0.7%, occurs with $k = k_e$, corroborating the simple theory of equation (23). This expression for k was therefore adopted for all other simulations.

[Note :At each time step of the simulation, the mean store temperature was calculated, and used to evaluate $k_w(T)$ using (22); this value was then adopted for the calculation of k_e in (24) for each zone rather than evaluating (22) according to each individual zone's temperature. This measure was taken to speed up the program and was justified by the fact that $k_w(20^\circ\text{C})$ and $k_w(90^\circ)$ differ by only 10%.]

The predicted temperature distributions during the test are shown in Figure 5.5, and, as can be expected from the minimal error, they match very closely the measured values shown in Figure 4.23.

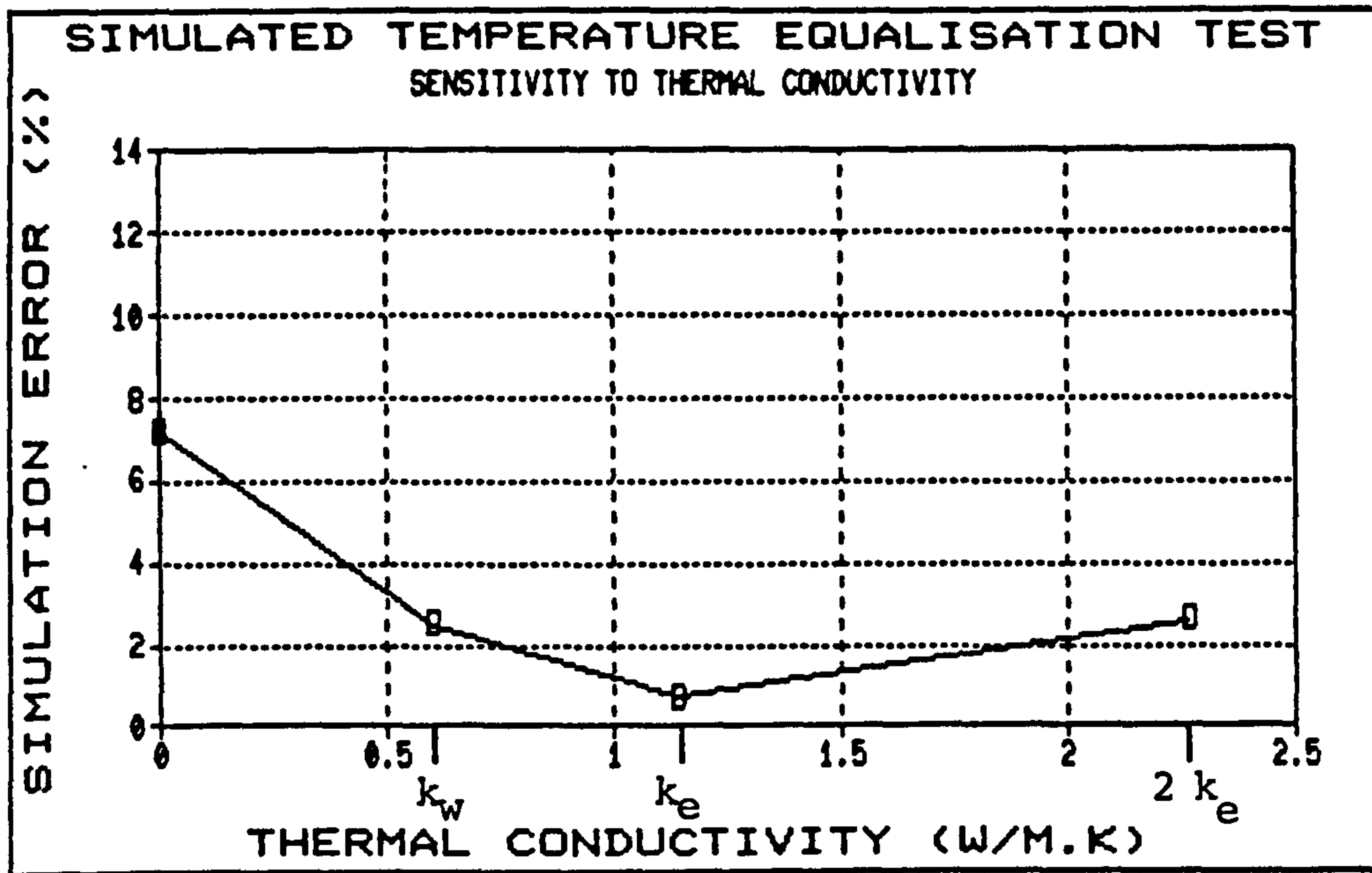


FIGURE 5.4 THE ERROR BETWEEN THE PREDICTED AND MEASURED STORE TEMPERATURE DISTRIBUTION AT THE END OF THE TEMPERATURE EQUALISATION TEST AS A FUNCTION OF THE ASSUMED VALUE FOR THE THERMAL CONDUCTIVITY

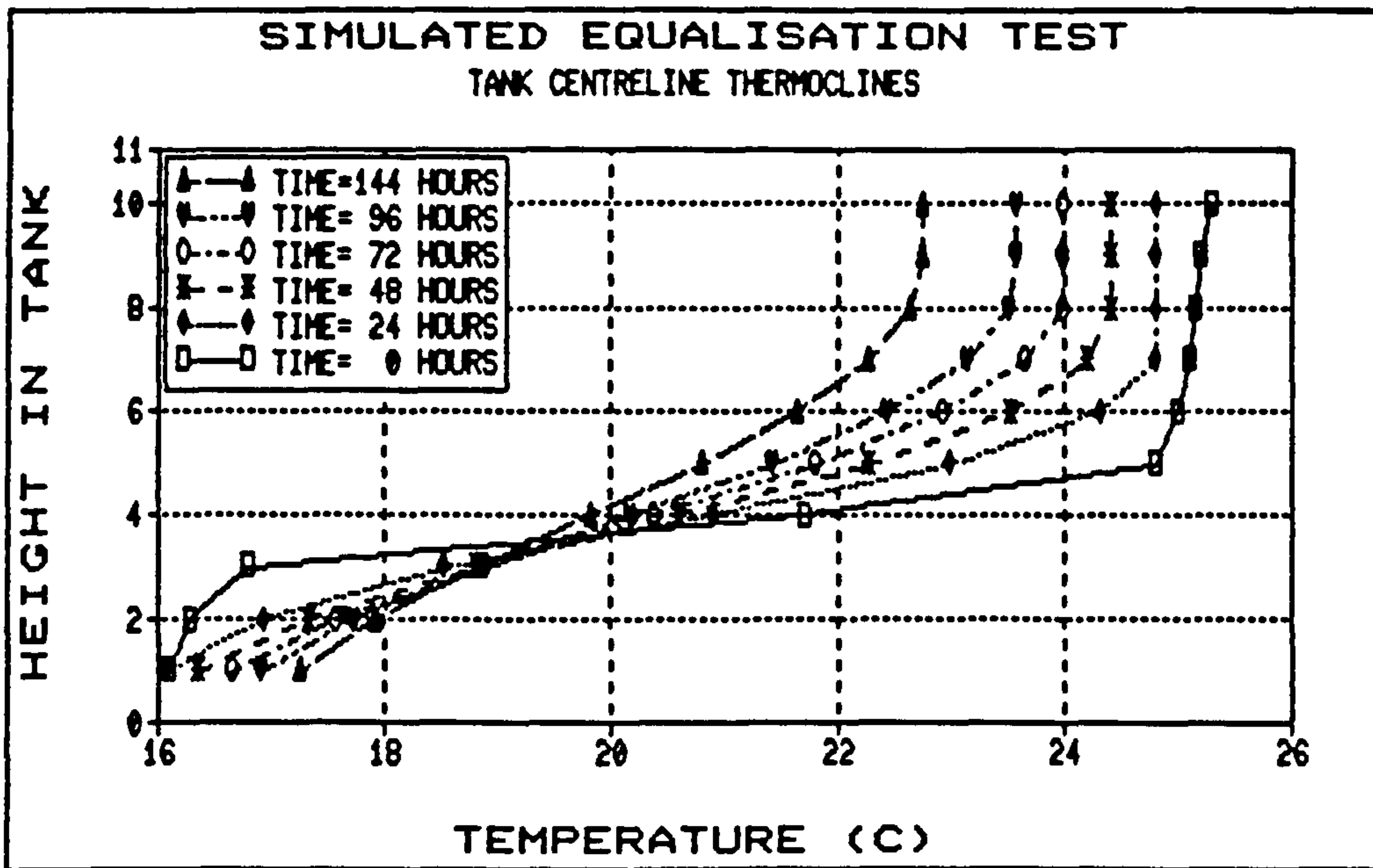


FIGURE 5.5 THE PREDICTED STORE THERMOCLINES DURING THE TEMPERATURE EQUALISATION TEST.

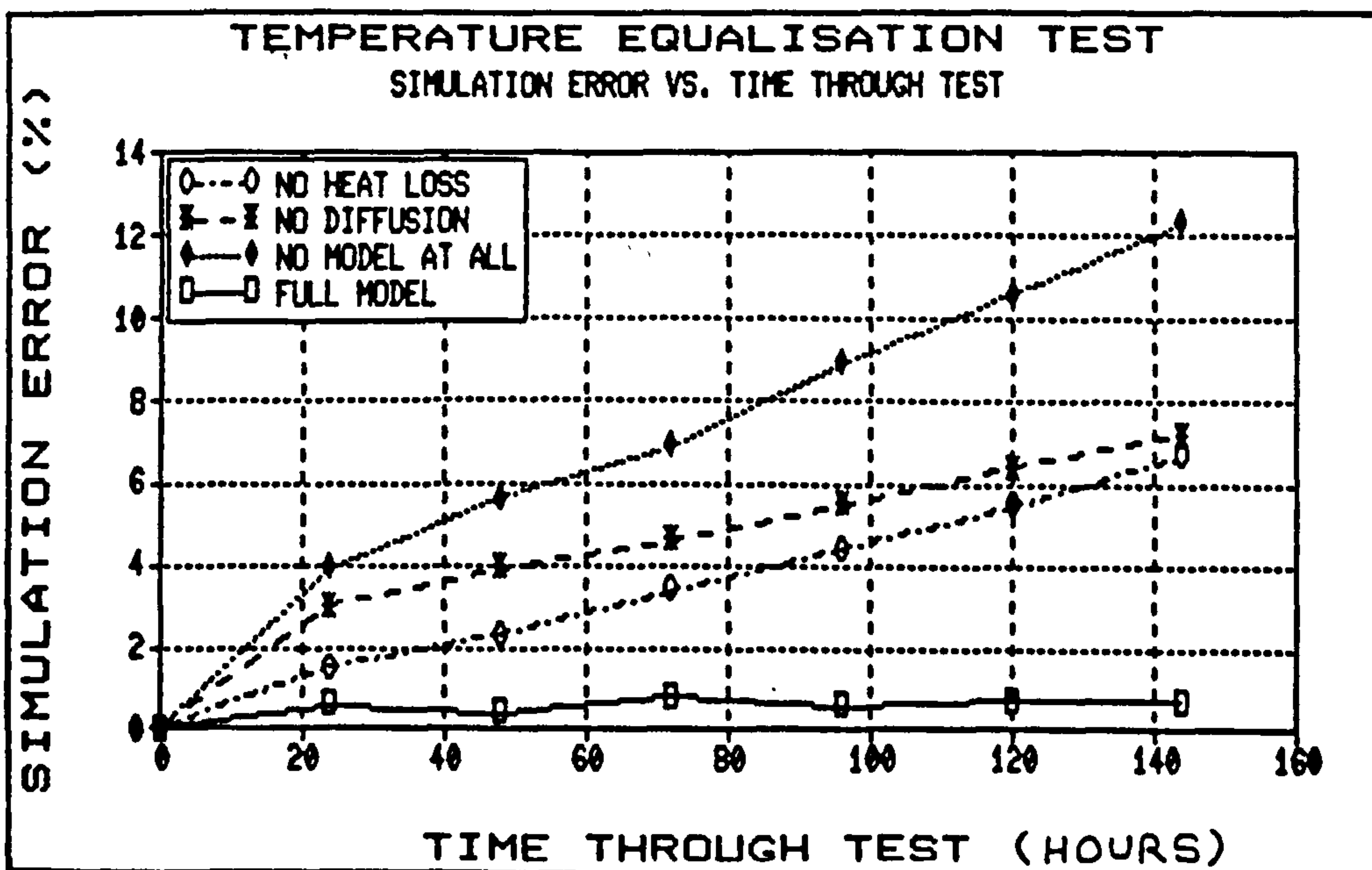


FIGURE 5.6 THE ERROR BETWEEN THE PREDICTED AND MEASURED STORE THERMOCLINES DURING THE TEMPERATURE EQUALISATION TEST.

Since conduction and heat loss are time dependent, it is interesting to calculate the simulation error throughout the test, and this is shown in Figure 5.6. The accuracy of the model, at least for this test, is confirmed by the fact that the error shows no sign of increasing with time.

Also shown on the figure is the error that would result if the model neglected completely both conduction and heat loss (denoted "no model at all"), which in this test would mean that the predicted temperatures did not change with time. This should be considered as the maximum tolerable error of any model and can be seen to increase steadily with time, reaching 13% after 144 hours.

Similarly the effect of neglecting just conduction (i.e. $k = 0$) is shown. It is interesting to note that the error in this case remains below 2% up to 18 hours. After 144 hours it has reached 7%. The effect of ignoring heat loss (i.e. $U = 0$) is also plotted and is less serious than neglecting conduction. The error remains below 2% up to 36 hours, although it does climb to nearly 7% by 144 hours.

Finally, with respect to this test, the model was run without the downward mixing routine, and the resulting temperature distributions are plotted in Figure 5.7. The temperature inversions which were expected to develop at the top of the store are clear to see. (In this store, the surface area for heat loss of the top zone is nearly 5 times greater than that of an internal zone). However they have a negligible effect on the error of the simulation which is constant at about 0.8% throughout the test.

5.2.2 Simulation of a High Temperature Equalisation Test

The equalisation experiment was carried out at a relatively low temperature to accentuate heat conduction effects in relation to heat losses. Having validated a computer model of heat conduction and heat loss processes, it is interesting to use it to predict the result of an equalisation test carried out at a high temperature and with a higher temperature difference between the top and bottom of the store. A test was simulated therefore with an initial temperature distribution in which the top 60% of the store was at 85°C and

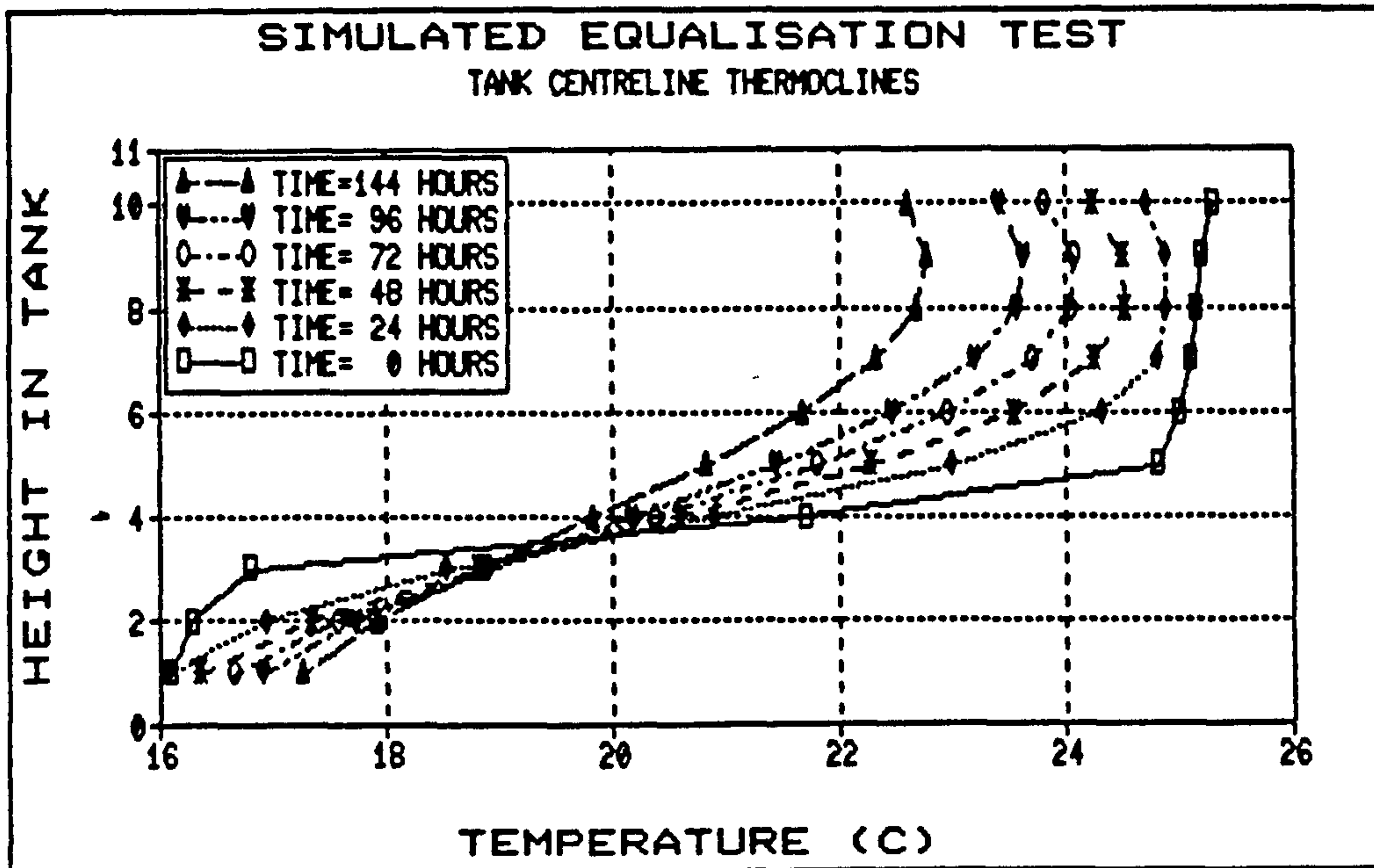


FIGURE 5.7 THE PREDICTED STORE THERMOCLINES DURING THE TEMPERATURE EQUALISATION TEST ASSUMING NO 'DOWNWARD MIXING'.

the bottom 30% at 16°C. The predicted store thermoclines are shown in Figure 5.8 at 24 hour intervals up to 144 hours.

The importance of heat loss and conduction in such conducive circumstances was examined by calculating the effect of neglecting one or both of them. The simulation was repeated therefore with no heat loss and no conduction, and the differences between the results obtained and the predictions of the full model were defined as the errors of ignoring heat loss and conduction. The development of the errors for these two cases and of the error when both processes were ignored ("no model at all") is shown in Figure 5.9. Conduction is again shown to be more significant than heat loss, which emphasises the high quality of the store insulation. However the errors are not that much higher than for the low temperature test - the error of leaving the store temperatures at their initial values is only 20% after 144 hours.

5.3 PREDICTED PERFORMANCE COEFFICIENTS

The analysis of the experiments in Chapter 4 presented a convincing argument for believing that the performance coefficient of a step input test is largely determined by the initial depth of penetration of the inlet flow. As the first stage of validating the model under step input conditions, its ability to reproduce this result was examined. Over a hundred step input tests were simulated using the model. A full range of experimental conditions were tried, with the flowrate ranging from 0.01 to 1 litre/second, the initial store temperature ranging from 15 to 90°C and the inlet-initial temperature difference being varied between 5 and 80°C. The initial depth of penetration in each simulation was calculated using the correlation derived from the experimental results for the vertical ports, i.e.

$$z = 0.128 + 0.442 \frac{\dot{m}}{\rho} r^{-3/2} (g \beta \Delta T)^{-1/2} \quad \dots(25)$$

The performance coefficient (for a discharge simulation) was calculated by the following formula :

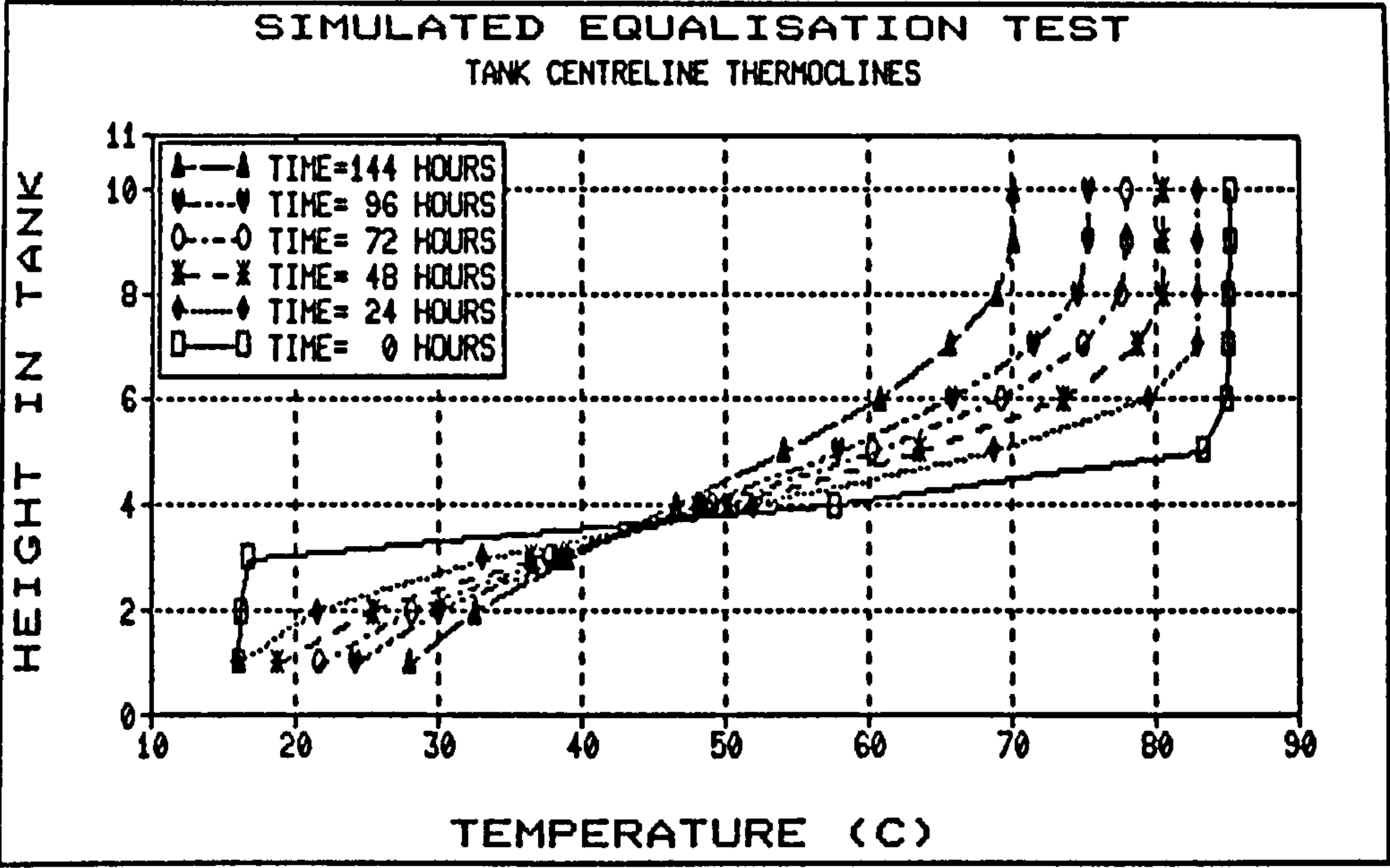


FIGURE 5.8 THE PREDICTED STORE THERMOCLINES DURING THE SIMULATED 'HIGH TEMPERATURE' EQUALISATION TEST.

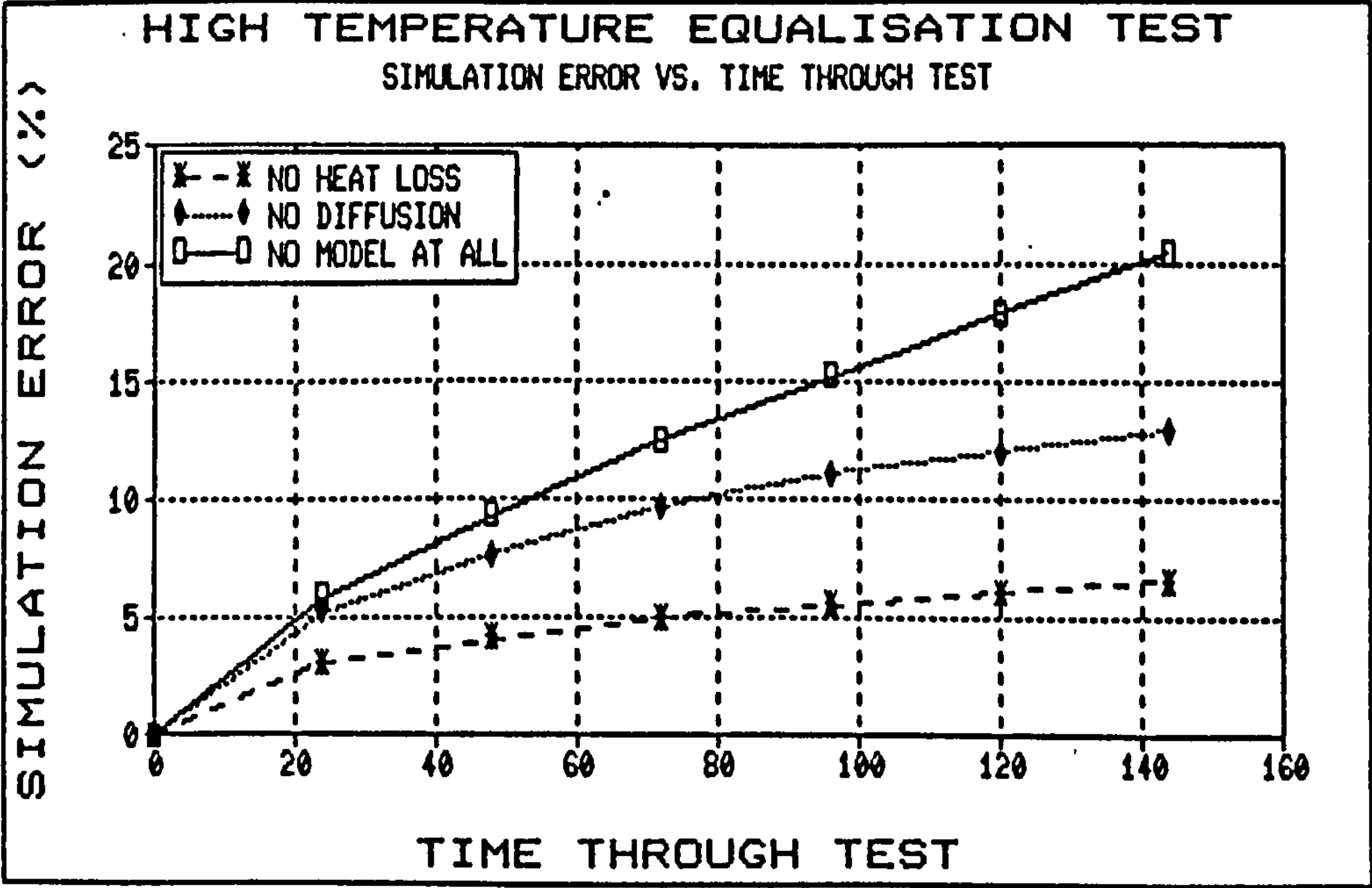


FIGURE 5.9 THE ERROR IN NEGLECTING HEAT LOSS, DIFFUSION OR BOTH DURING THE SIMULATED 'HIGH TEMPERATURE' EQUALISATION TEST.

$$\eta(\tau) = \frac{\sum_{t=1}^{22} \left[\rho \dot{V} C_p (T_{out}(t) - T_{in}) \Delta t + Q_{LOSS}(t) \right]}{\rho V_s C_p (T_{init} - T_{in})} \quad ..(26)$$

$$\text{where } Q_{LOSS}(t) = \sum_{z=1}^{22} U A(z) (T(z) - T_{amb}) \Delta t$$

The formula (26) applies since the simulation assumed that ideal step inlet conditions were achieved, and because the outlet temperature at the end of each time step, $T_{out}(t)$, is really the mean outlet temperature over the whole time step. (This follows from the definition of $\bar{T}_{mout}$ in equation (12)). It is therefore correct to sum the discrete 'lumps' of energy discharged during each time step, rather than taking the outlet temperatures to be points on a curve and using something like Simpson's Rule to evaluate the area enclosed by the curve. It should also be noted that the thermal capacity of the tank is not considered by the model.

The results of the simulated step input tests are plotted in Figure 5.10. The points are clustered at distinct values of the depth of penetration, because z was rounded to the depth of an integer number of zones. A best fit curve has been drawn through the points. This curve has also been superimposed on the corresponding experimental results for the vertical ports (Figure 4.4) in Figure 5.11. This shows that in general the results of the model match very closely the experimental results.

The accuracy of the model in predicting performance coefficients for specific experiments is now examined. The typical vertical port and distributor experiments described in Chapter 4 are taken as examples. The vertical port experiment (Reference No. 21.160883) had operating conditions which meant that the Turner parameter for the inlet flow was 0.485m. The experimental correlation for the vertical ports (25) predicts that this should produce a penetration depth of 0.343m.

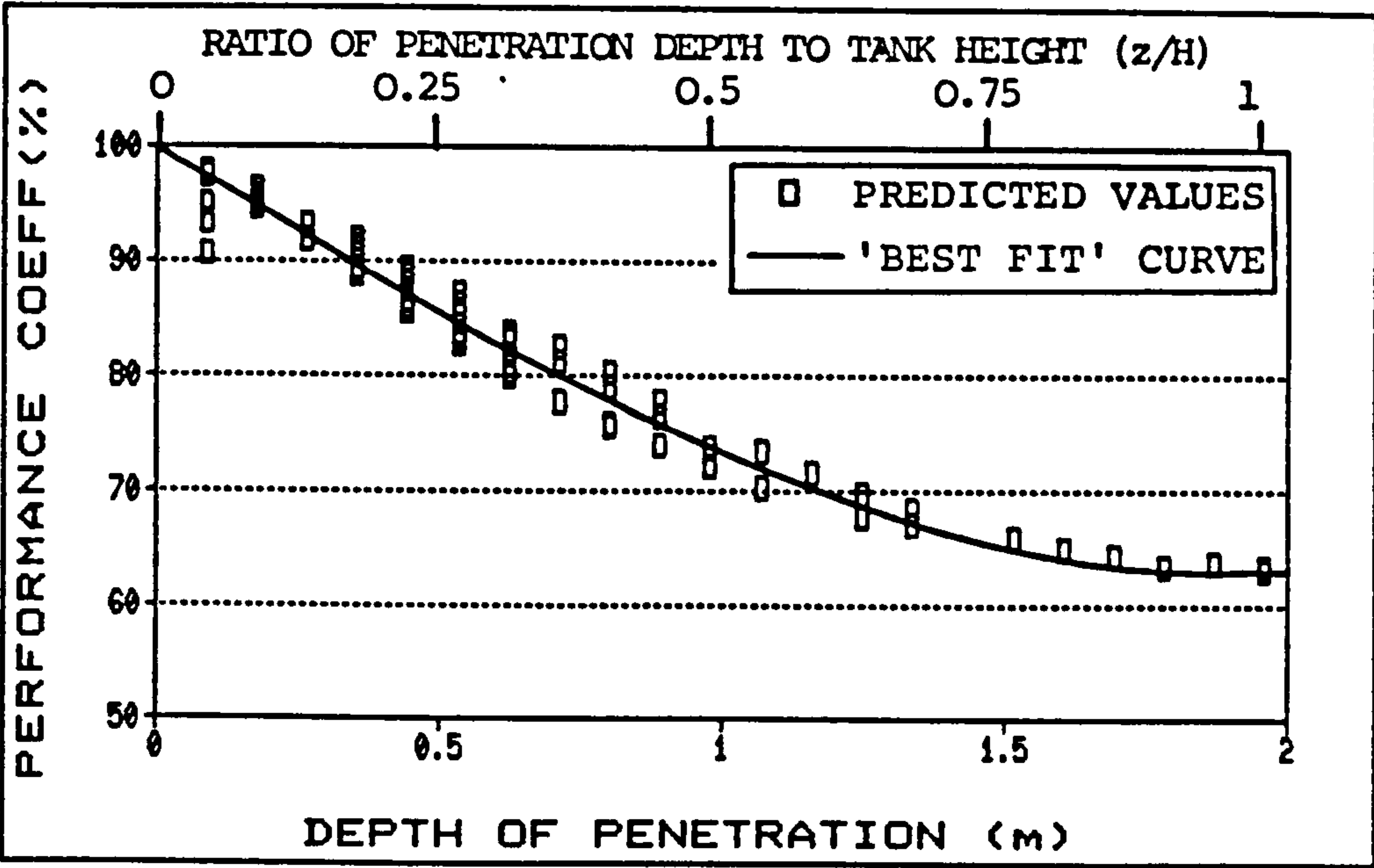


FIGURE 5.10 THE PREDICTED PERFORMANCE COEFFICIENTS FOR SIMULATED STEP INPUT TESTS, UNDER A WIDE RANGE OF CONDITIONS, PLOTTED AGAINST THE PENETRATION DEPTH CALCULATED USING AN EXPERIMENTAL CORRELATION.

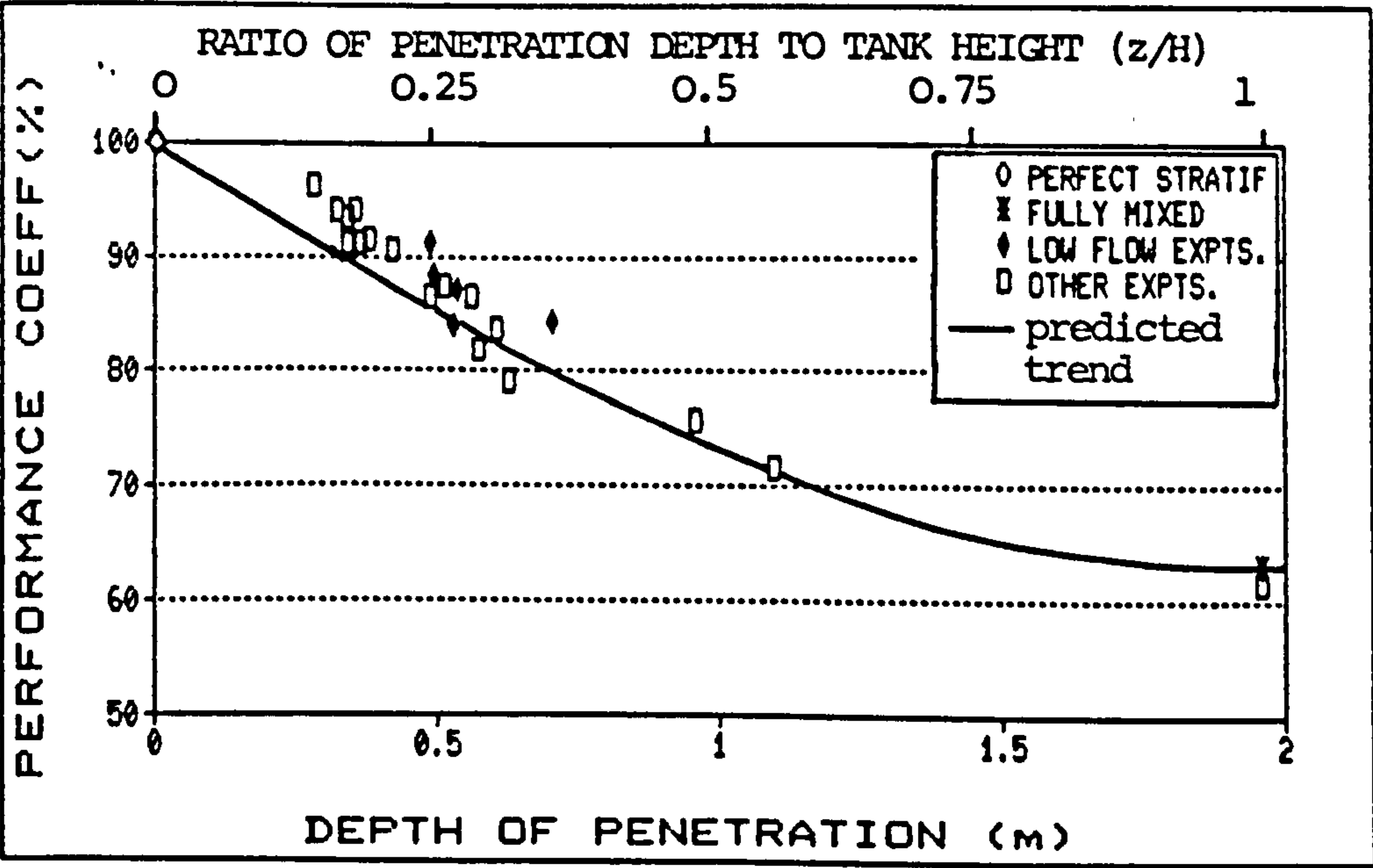


FIGURE 5.11 THE PREDICTED TREND FOR PERFORMANCE COEFFICIENT AS A FUNCTION OF PENETRATION DEPTH COMPARED TO MEASURED VALUES FOR THE EXPERIMENTS WITH VERTICAL PORTS.

With the store divided into 22 zones, each of depth 0.089m, this is equivalent to a mixed region of 4 zones. The model was run under the test conditions and predicted a performance coefficient of 91.28%, which was only 0.3% below the measured value of 91.6%.

The distributor experiment (Reference No.21.271081) had a Turner parameter value of 0.0958m which, inserted into the experimental correlation for the distributors, gives a mixed region depth of 0.236m that, when rounded up, is equivalent to 3 zones. For this experiment, the model predicted a performance coefficient of 92.76% which was 3.3% below the measured value of 95.9%. (It was found that if the number of zones in the mixed region had been rounded down to 2, then the model would have predicted a performance coefficient of 95.01%, which would be only 0.9% below the measured value. Although this is a little beside the point, since the whole purpose of the model is to be predictive, it does illustrate that the results of the model are a little sensitive to the process of rounding the number of zones in the mixed region.) Nonetheless, an error of 3.3% is still small, and these two examples are a further validation of the model.

5.4 PREDICTED TEMPERATURES

Conclusive validation of the model requires the ability to predict the store outlet temperatures and internal temperature distributions during a step input experiment.

5.4.1. Predicted Outlet Temperatures

The predicted and measured outlet temperatures for the vertical port experiment are shown in Figure 5.12. The match of the two sets of data is very good. The difference between them, i.e. the error of the model, is directly represented by the error in the performance coefficient, which in this case was only 0.3%.

The corresponding graph for the distributor experiment is given in Figure 5.13. Here there is a noticeable discrepancy in the simulated temperature towards the end of the discharge period, although it does only amount to an error of 3.3% over the whole period. (If the model had, hypothetically, been constrained to round down the size of the

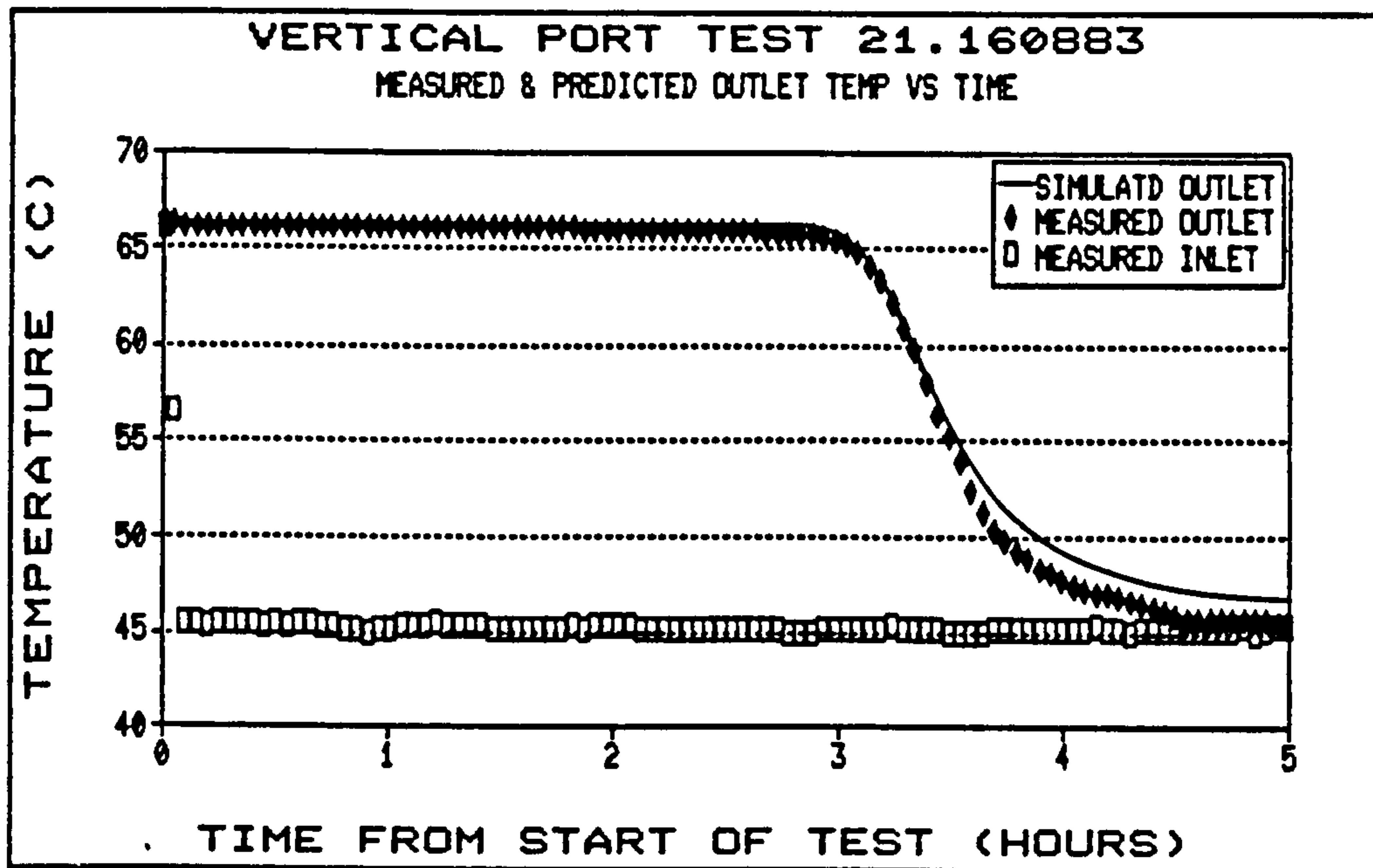


FIGURE 5.12 THE MEASURED AND PREDICTED OUTLET TEMPERATURES FOR AN EXPERIMENT WITH VERTICAL PORTS.

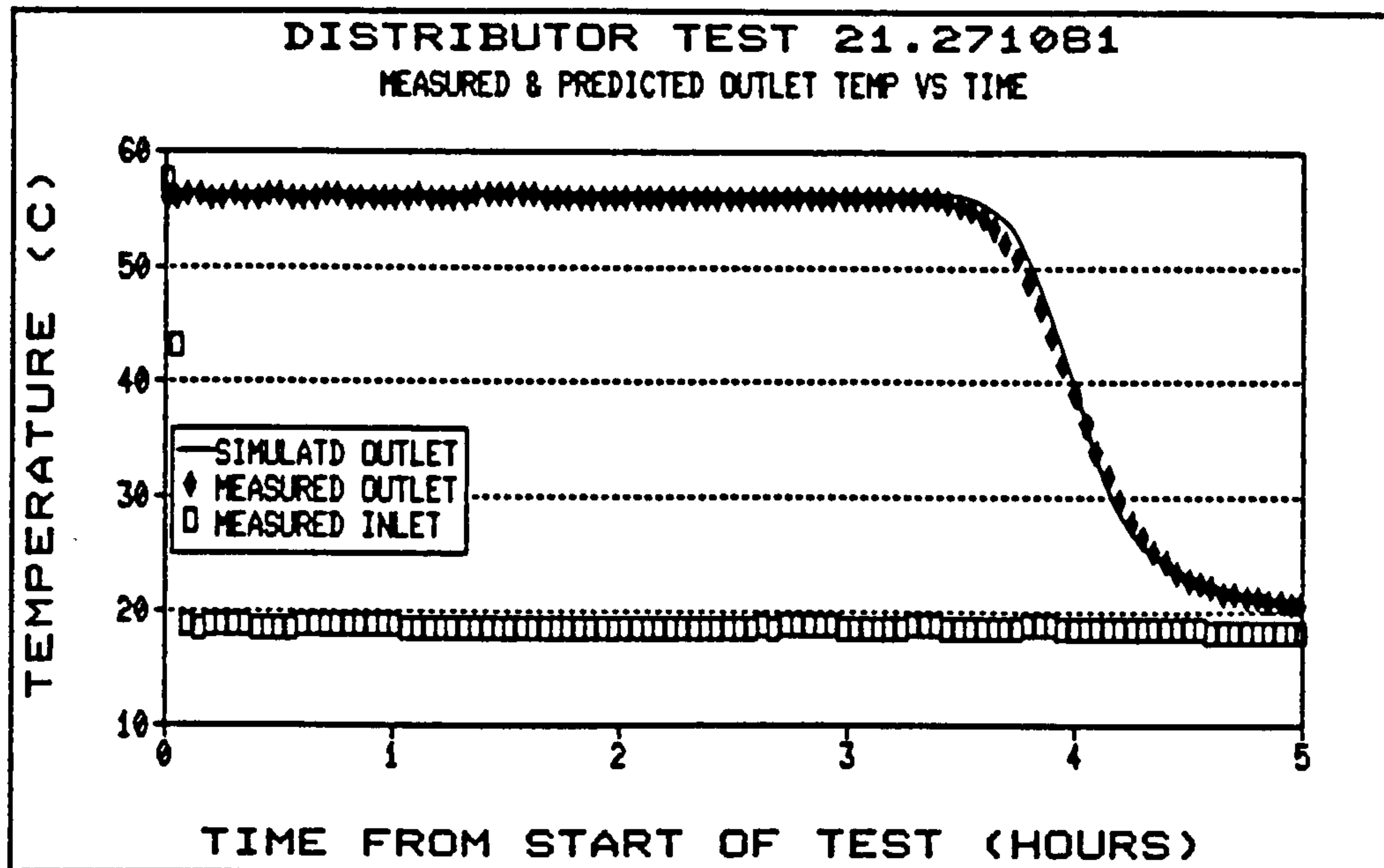


FIGURE 5.14 THE PREDICTED OUTLET TEMPERATURE FOR THE EXPERIMENT IN FIGURE 5.13 WITH THE INITIAL MIXED REGION DEPTH SET AT 2 ZONES (I.E. 1 LESS THAN THE VALUE CALCULATED FROM THE EXPERIMENTAL CORRELATION).

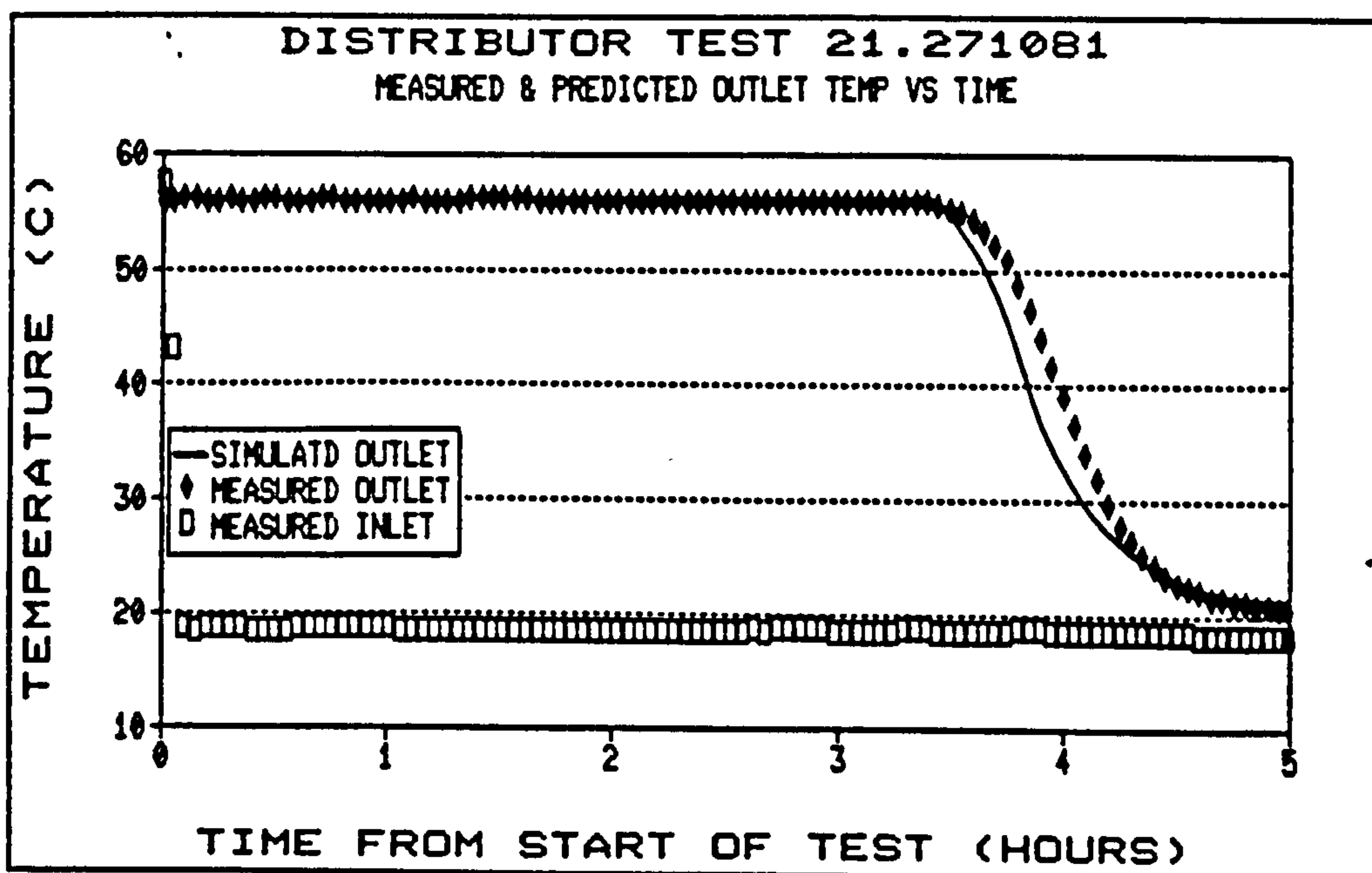


FIGURE 5.13 THE MEASURED AND PREDICTED OUTLET TEMPERATURES FOR AN EXPERIMENT WITH DISTRIBUTORS.

mixed region to 2 zones, the result, as might be expected from the performance coefficient, would be even closer, as shown in Figure 5.14.)

5.4.2 Predicted Temperature Distributions

The predicted store thermoclines, for the vertical port test, at intervals of 20% of the discharge period, are shown in Figure 5.15, plotted on the same graph as the measured thermoclines, which were given on their own in Figure 4.8. The measured and predicted thermoclines can be compared in a number of ways : the most direct is to calculate the mean difference in temperature at a particular time between the measured and simulated results, so that the simulation error at time t , is defined as :

$$\text{Error (t)} = \frac{\sum_{I=1}^{10} \text{ABS} [\text{TEMP}_I^t - \text{TACT}_I^t]}{10} \quad ^\circ\text{C}$$

The magnitude of this error through the test is plotted in Figure 5.16, where it can be seen to be steady at about 2°C . The error when the initial mixed region is taken to be 1 zone larger or smaller than that calculated using the experimental correlation is also shown, to illustrate the sensitivity of the model.

It is more meaningful to calculate the error as a percentage, by dividing the error by the correct mean temperature. The definition of the error then becomes :

$$\text{Error (t)} = \frac{\sum_{I=1}^{10} \text{ABS} [\text{TEMP}_I^t - \text{TACT}_I^t]}{\frac{10}{\sum_{I=1}^{10} \text{TACT}_I^t}} \cdot 100\% \quad \dots(27)$$

The error in the vertical port experiments based on the above definition is plotted in Figure 5.17. Its magnitude remains steady for most of the test at around 4%. A potential drawback of this technique of presenting the error is that it makes the errors in

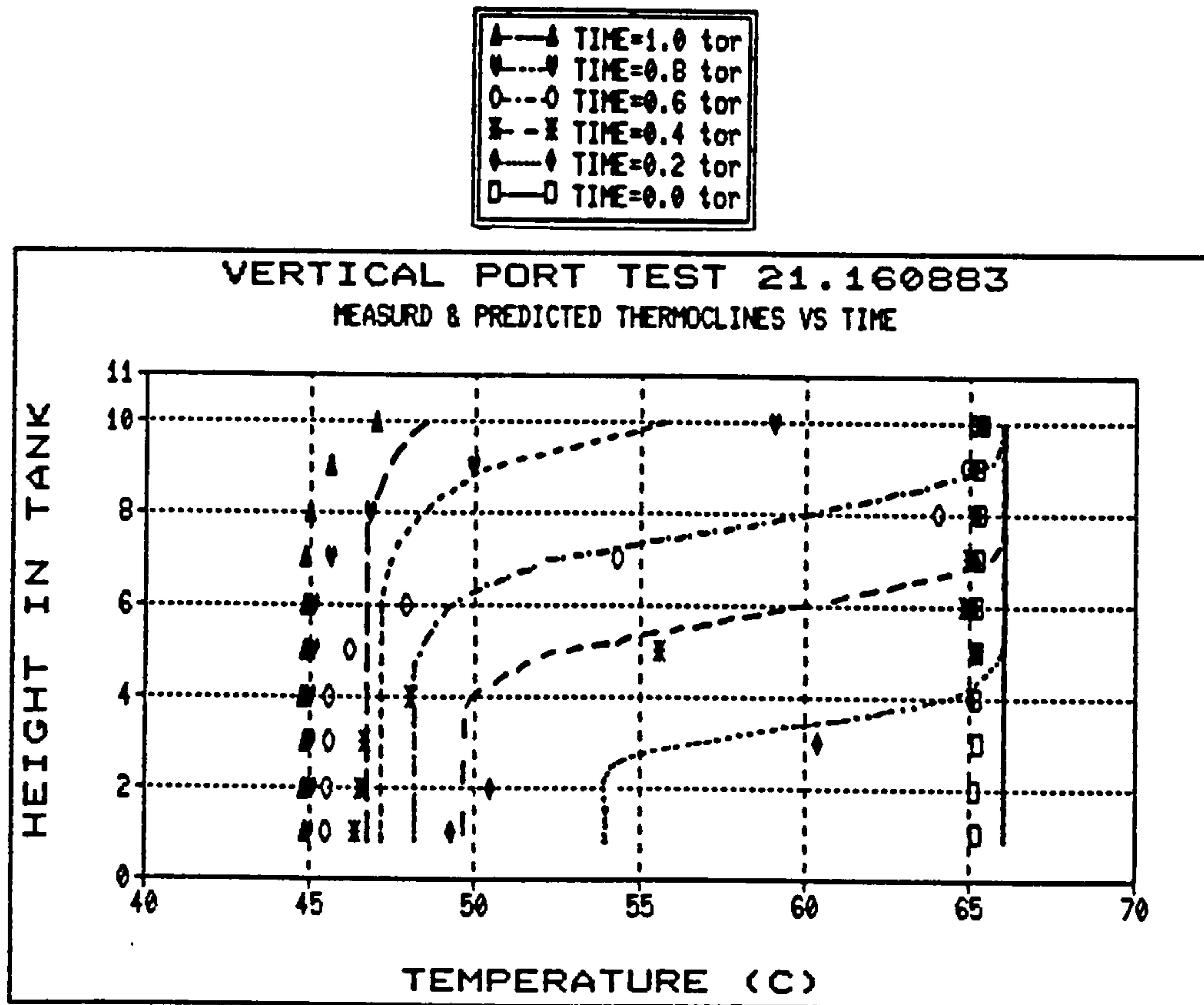


FIGURE 5.15 THE MEASURED AND PREDICTED STORE THERMOCLINES FOR AN EXPERIMENT WITH VERTICAL PORTS.

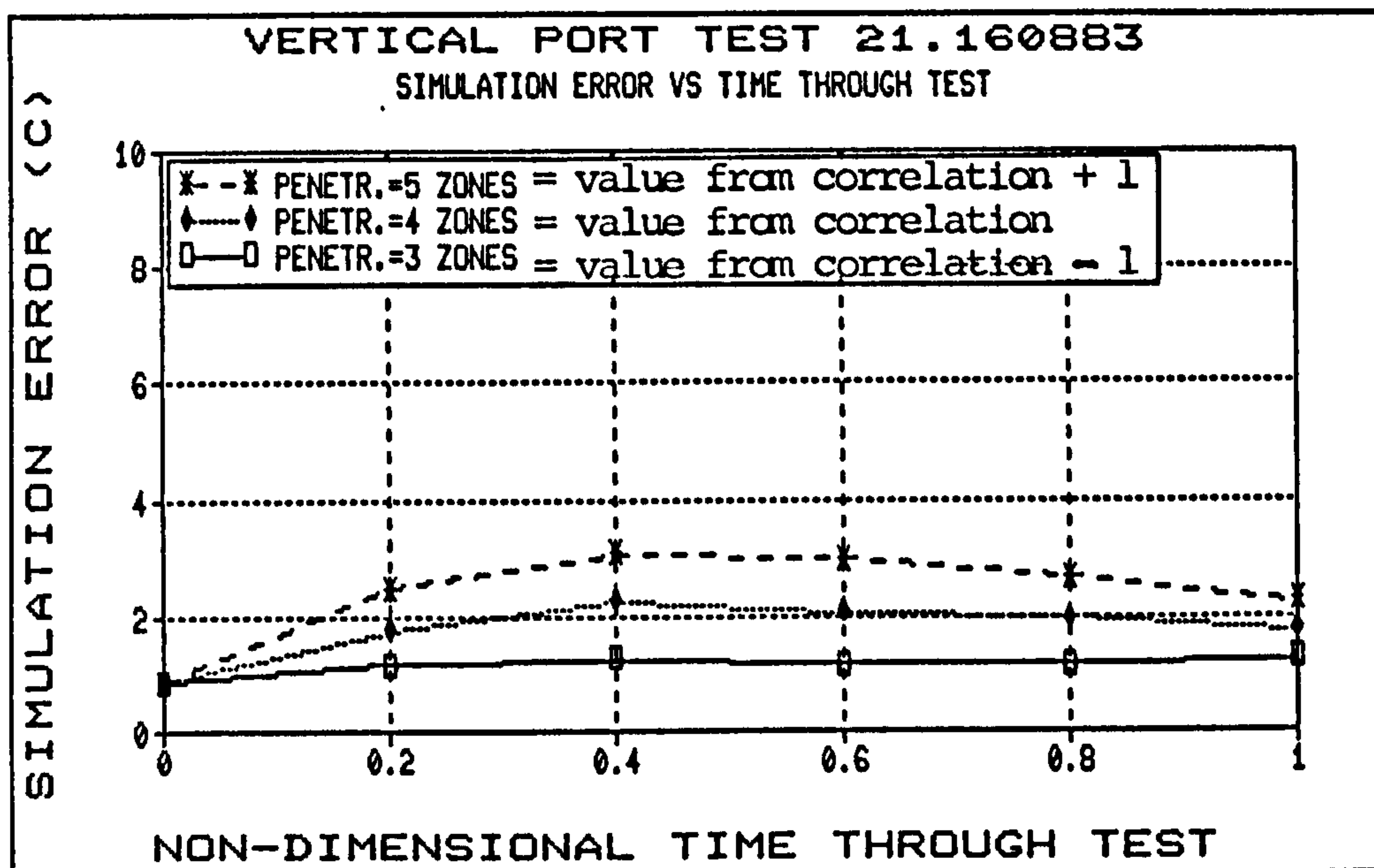


FIGURE 5.16 THE MEAN ERROR IN °C BETWEEN THE PREDICTED AND MEASURED STORE THERMOCLINES FOR A VERTICAL PORT TEST.

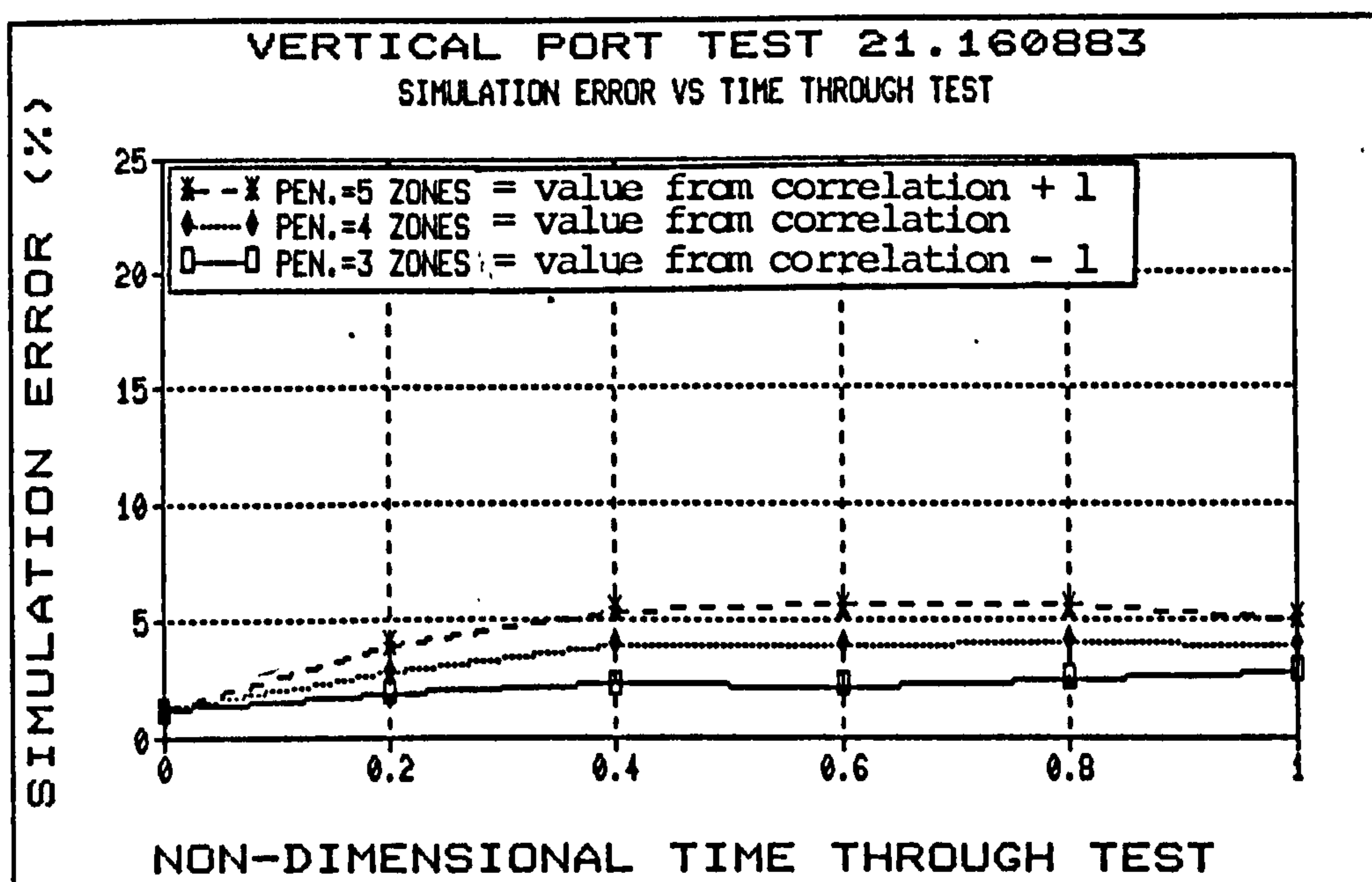


FIGURE 5.17 THE MEAN ERROR IN % BETWEEN THE PREDICTED AND MEASURED STORE THERMOCLINES FOR A VERTICAL PORT TEST.

experiments at high temperatures appear low. Another method is to give the error as a percentage of the measured mean change in store temperature since the start of the test. The definition is then given by :

$$\text{Error (t)} = \frac{\sum_{I=1}^{10} \text{ABS} [\text{TEMP}_I^t - \text{TACT}_I^t]}{\sum_{I=1}^{10} [\text{TACT}_I^t - \text{TACT}_I^0]} \cdot 100\%$$

The result of this approach is shown in Figure 5.18. As can be seen, this has the great disadvantage of producing a large error initially because there has been little change in the actual temperatures. It is therefore considered preferable to present the error as defined previously in equation (27).

This method has been adopted for the distributor port experiment, for which the predicted thermoclines are compared with the measured temperatures in Figure 5.19. The error between the predicted and measured thermoclines (see also Figure 4.15) is shown in Figure 5.20. The error appears greater in this case, but this is largely due to the experiment being at a lower temperature level (i.e. a discharge from 56 to 19°C, cf. 66 to 45°C). The error at the end of the experiment is about 10%, which is equivalent to an error of under 2°C in the temperature of the final thermocline.

In summary, the relatively straightforward one-dimensional model developed in the present work shows a fair degree of success in predicting the thermal behaviour of the hot water store, under step input conditions with a fully-turbulent inlet flow. The error in predicting the performance coefficient and outlet temperatures during an experiment could be less than 3%, whilst the internal temperature distribution might be estimated to an accuracy of 10% or less.

5.5 THE SIGNIFICANCE OF HEAT CONDUCTION

The simulated high temperature equalisation test revealed that heat

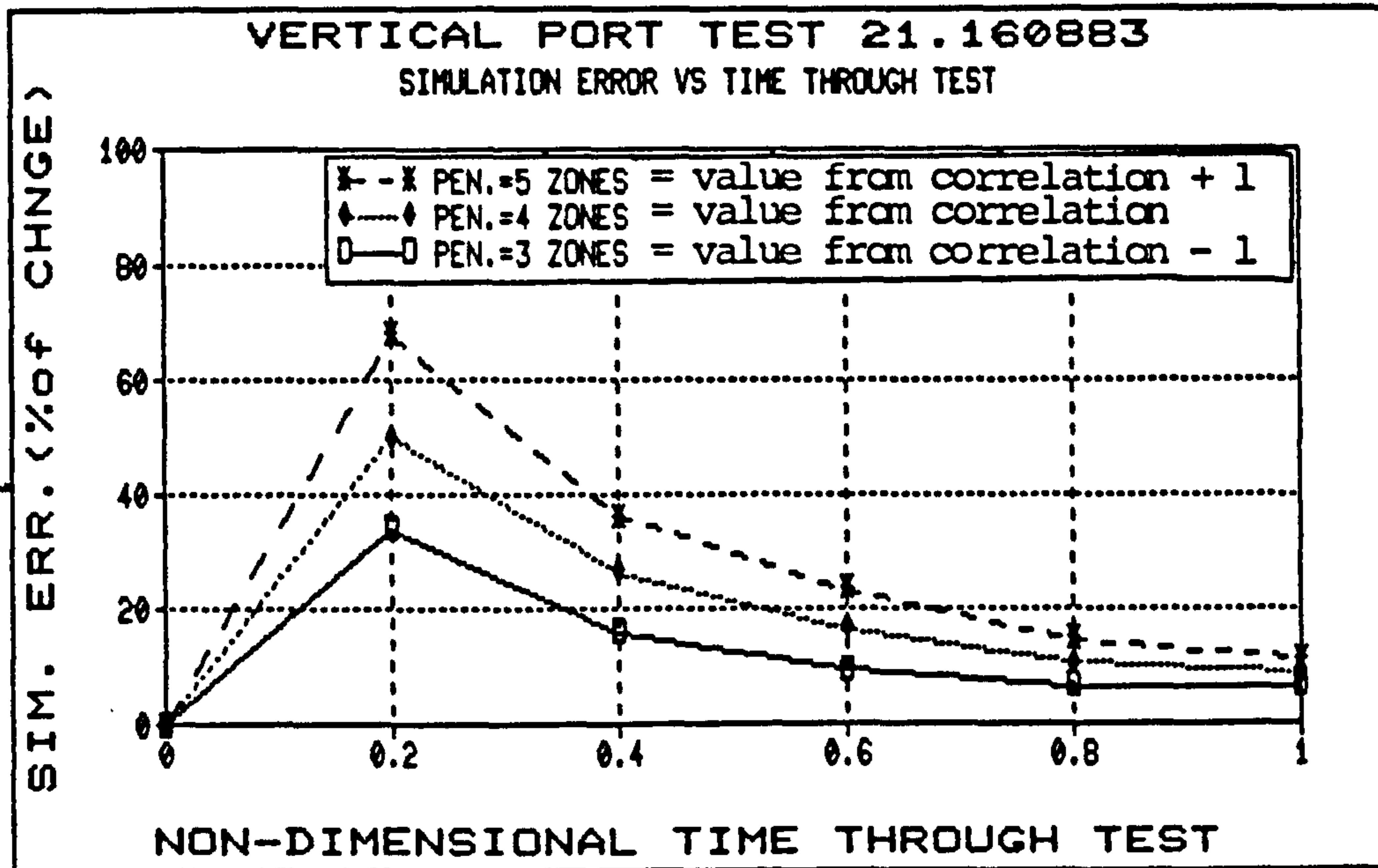


FIGURE 5.18 THE MEAN ERROR BETWEEN THE MEASURED AND PREDICTED STORE THERMOCLINES FOR A VERTICAL PORT TEST CALCULATED AS A % OF THE MEASURED MEAN CHANGE IN STORE TEMPERATURE .

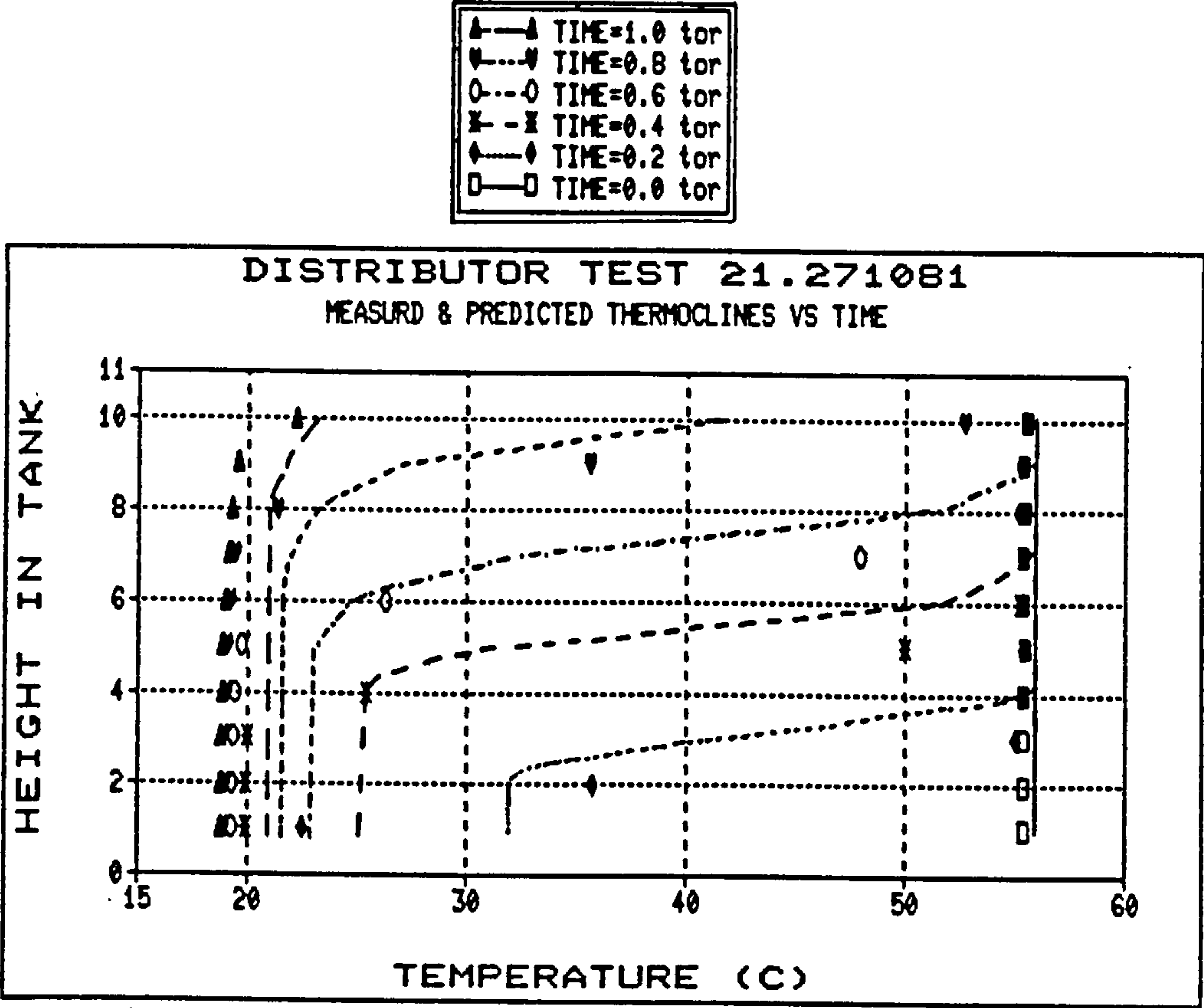


FIGURE 5.19 THE MEASURED AND PREDICTED STORE THERMOCLINES FOR AN EXPERIMENT WITH DISTRIBUTORS.

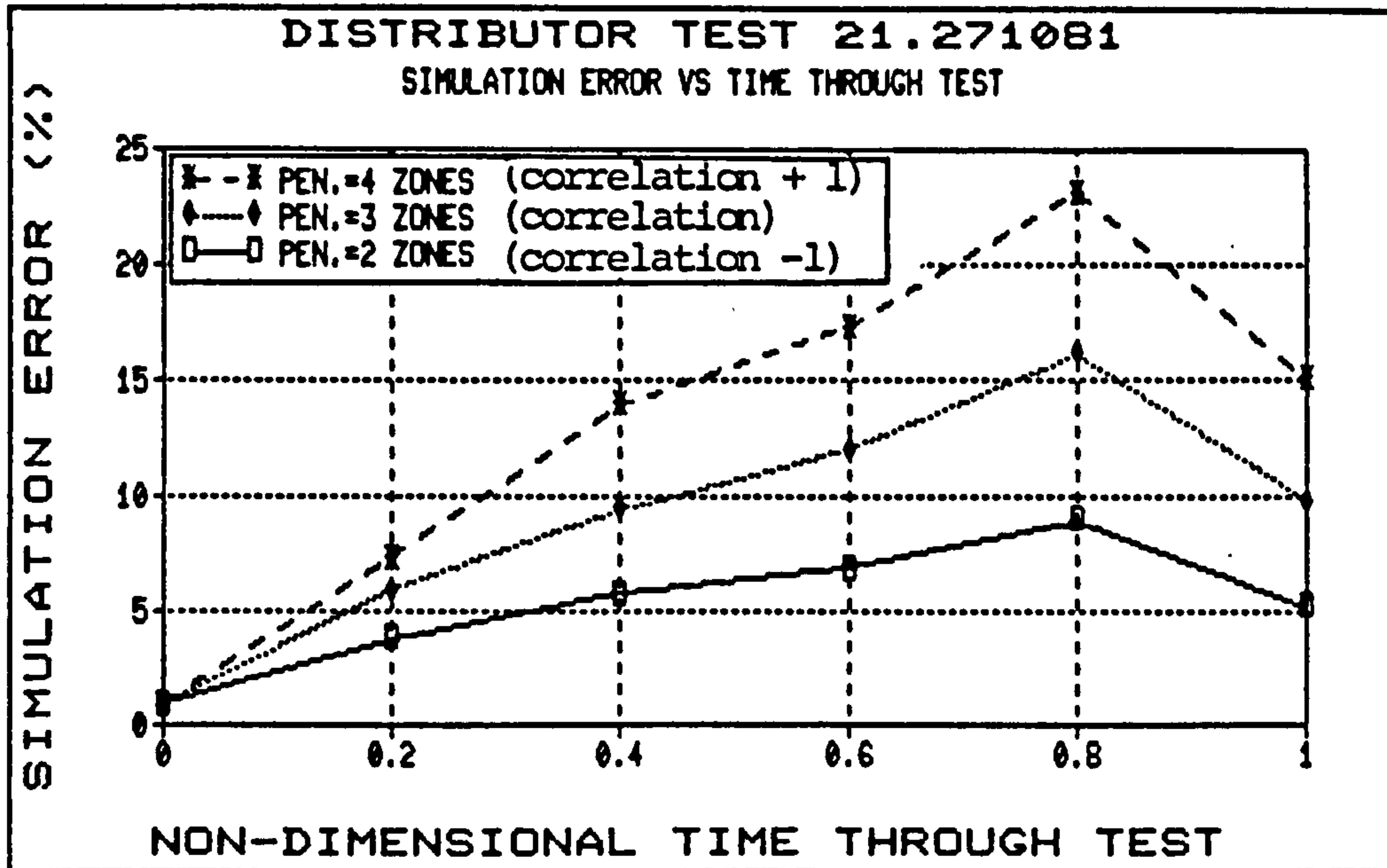


FIGURE 5.20 THE MEAN ERROR IN % BETWEEN THE PREDICTED AND MEASURED STORE THERMOCLINES FOR AN EXPERIMENT WITH DISTRIBUTORS.

conduction is a relatively slow process. Even under these conducive conditions, it modified the store temperatures by only 6% after 24 hours and 13% after 144 hours. Since conduction requires a time-consuming routine in the computer model, it was decided to assess the errors involved in neglecting it under step input conditions.

The model was run with the same flow conditions, first with the conductivity of water at the value given by equation (24) and then with it set to zero. This procedure was carried out with the initial-inlet temperature difference at a high value (i.e. a discharge from 90 to 20°C) and a low value (i.e. a discharge from 30 to 20°C). The simulation was also done for a full range of flow rates, so that the fill time of the store varied from 5 to 100 hours. The predicted performance coefficient for each simulation is plotted against the store fill time in Figure 5.21.

The influence of conduction is negligible for both the low and high ΔT tests lasting 5 hours. It amounts to only 1% for the high ΔT test lasting 10 hours and for the low ΔT test lasting 20 hours. For both tests lasting 40 hours, conduction reduces the performance coefficient by 4%. Its significance continues to increase slowly as the tests take longer and remains similar for both the high and low ΔT cases; by the 100 hour test it reaches 9%.

This exercise shows that conduction can be neglected with little error in those applications where there is a similarly sized store which cycles more than once per day. Where longer cycles, or standstill periods of about 24 hours or more occur, simulations would incur errors of the order of 5 to 10% in the store internal temperature distribution, if they neglected conduction. In most applications, this inaccuracy would be insignificant in terms of overall system performance. Finally it should be remembered that the influence of heat conduction is dependent on scale, and particularly the cross-sectional area to volume ratio of the store, as well as on time. Also, errors in the store internal temperature distribution need not and usually do not imply an error in the amount of energy in the store, merely its apportionment.

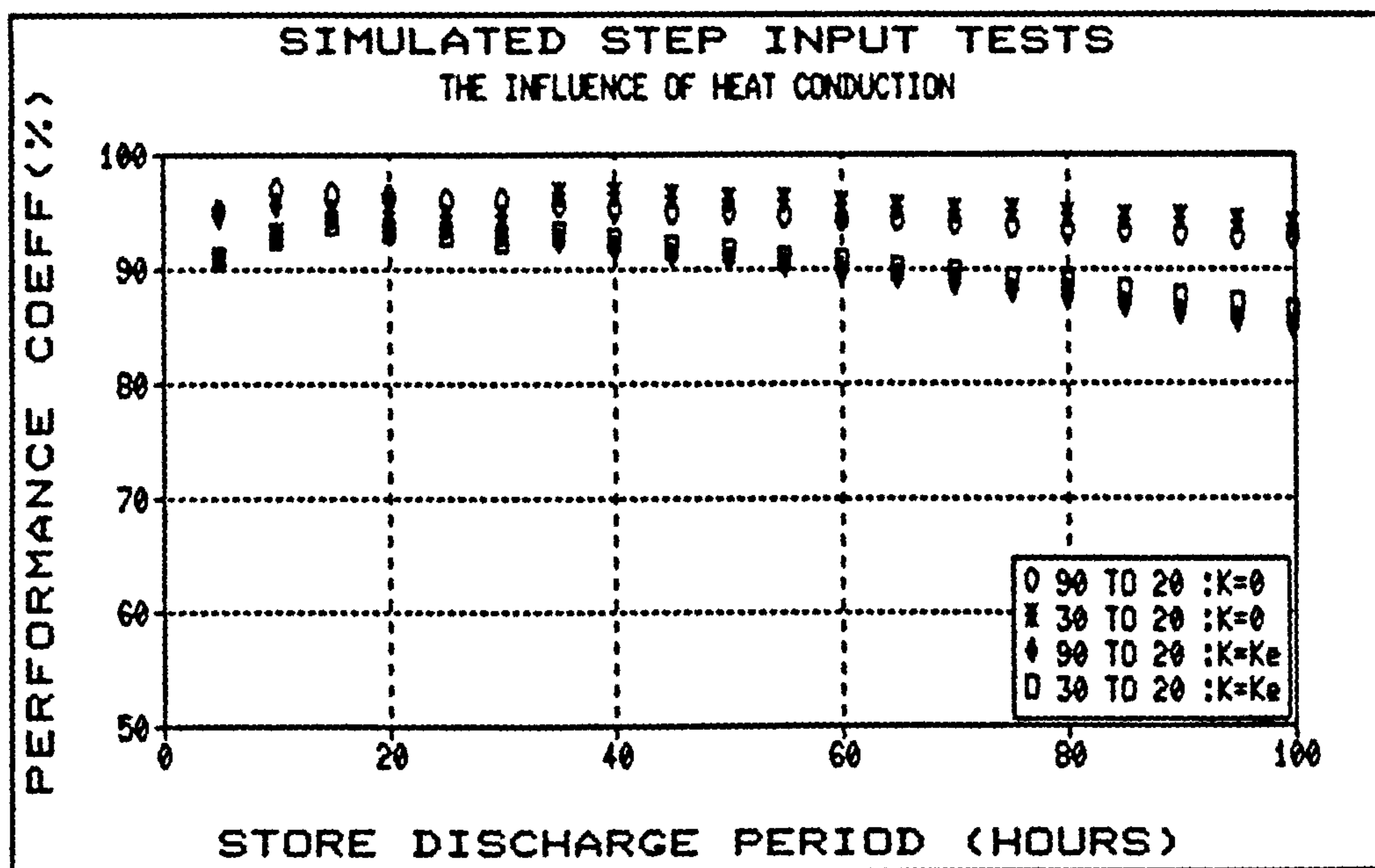
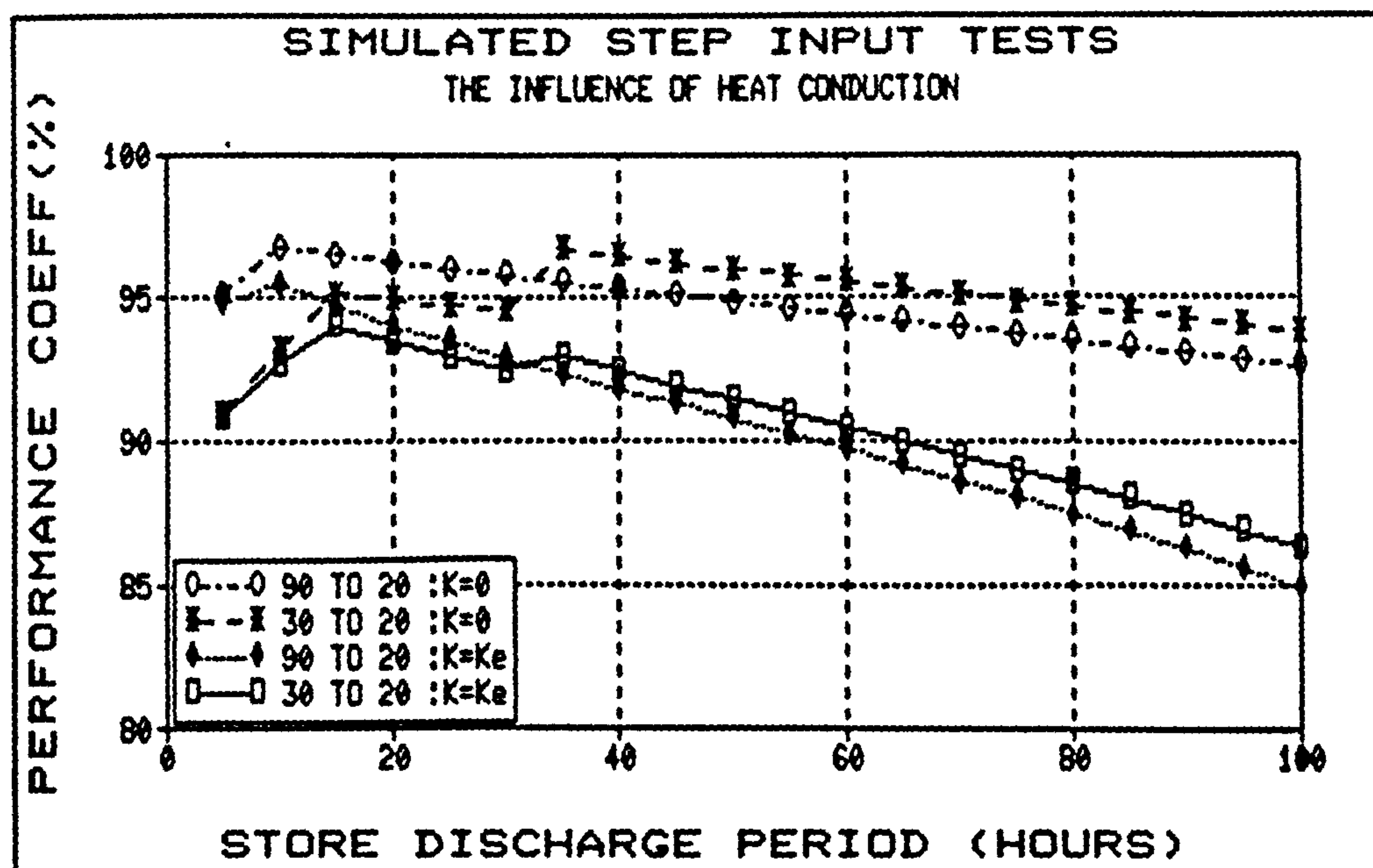


FIGURE 5.21 THE PREDICTED INFLUENCE OF HEAT CONDUCTION ON THE PERFORMANCE COEFFICIENT AS A FUNCTION OF THE DISCHARGE PERIOD (i.e. INLET FLOWRATE) FOR HIGH AND LOW ΔT STEP INPUT TESTS:

(ABOVE) ON THE SCALE USED IN CORRESPONDING GRAPHS

(BELOW) ON AN ENLARGED SCALE.

CHAPTER 6

ANALYSIS OF AN APPLICATION OF HOT WATER STORAGE

6.1 INTRODUCTION

This chapter examines in some detail an interesting application for a hot water store of the scale of the one used in the experiments. The concept is to interpose a thermally-insulated hot water store in the circuit between a central-heating boiler and a hydronic heat-distribution system. The potential benefits are first discussed qualitatively; these include energy savings due to the less frequent switching, on and off, of the boiler, and the prospect of using a smaller boiler to satisfy the heating needs of a building. More precise predictions are then presented, based on computer simulations which incorporate the model for a stratified hot water store described in the previous chapter. The annual energy savings achievable and their sensitivities to changes in heating system and building parameters are estimated. Finally there is an assessment of the economic viability of this application of heat storage.

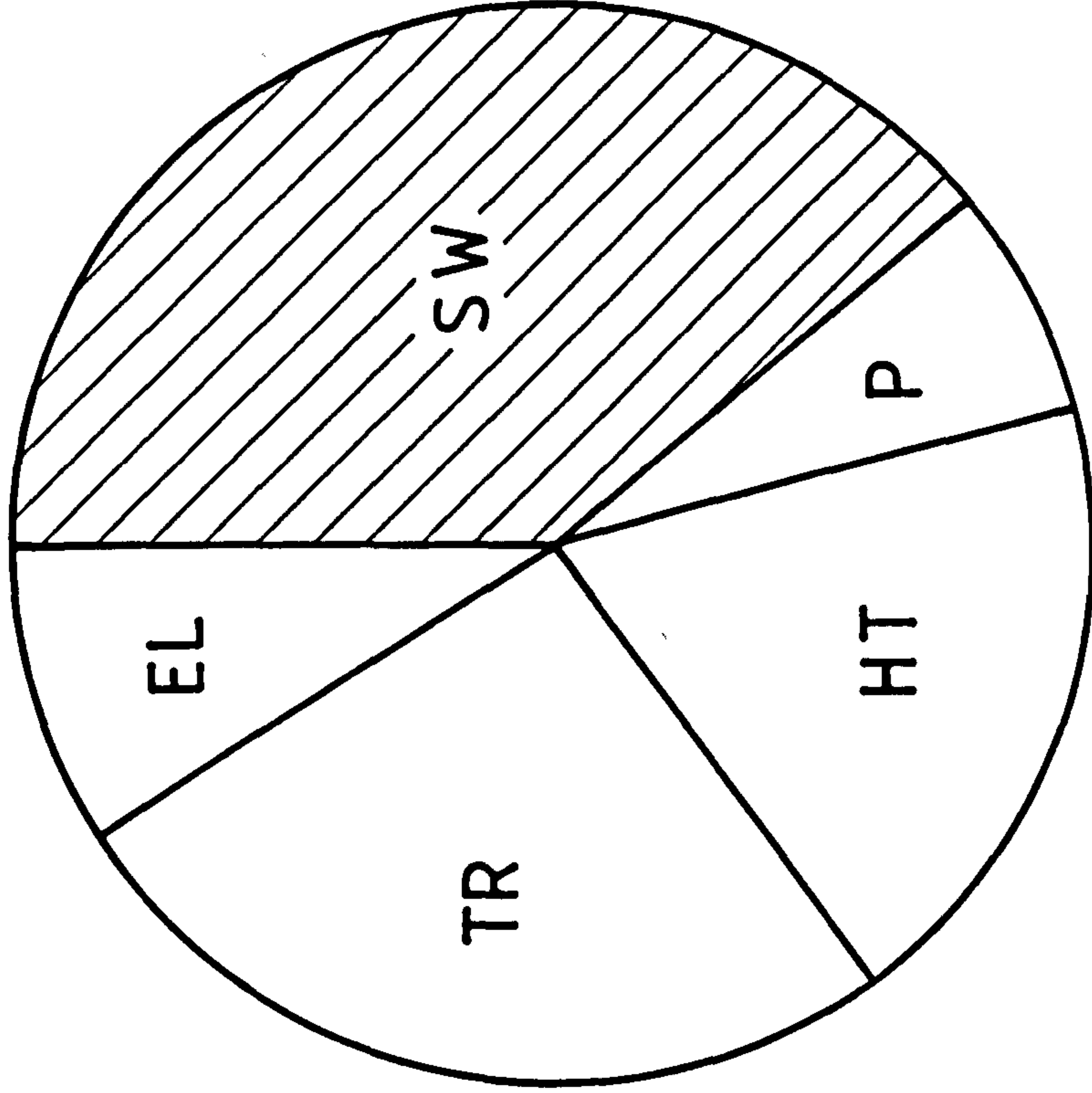
6.2 THE APPLICATION OF HEAT STORAGE TO CENTRAL-HEATING SYSTEMS

6.2.1 The Importance of Energy Use in Boilers

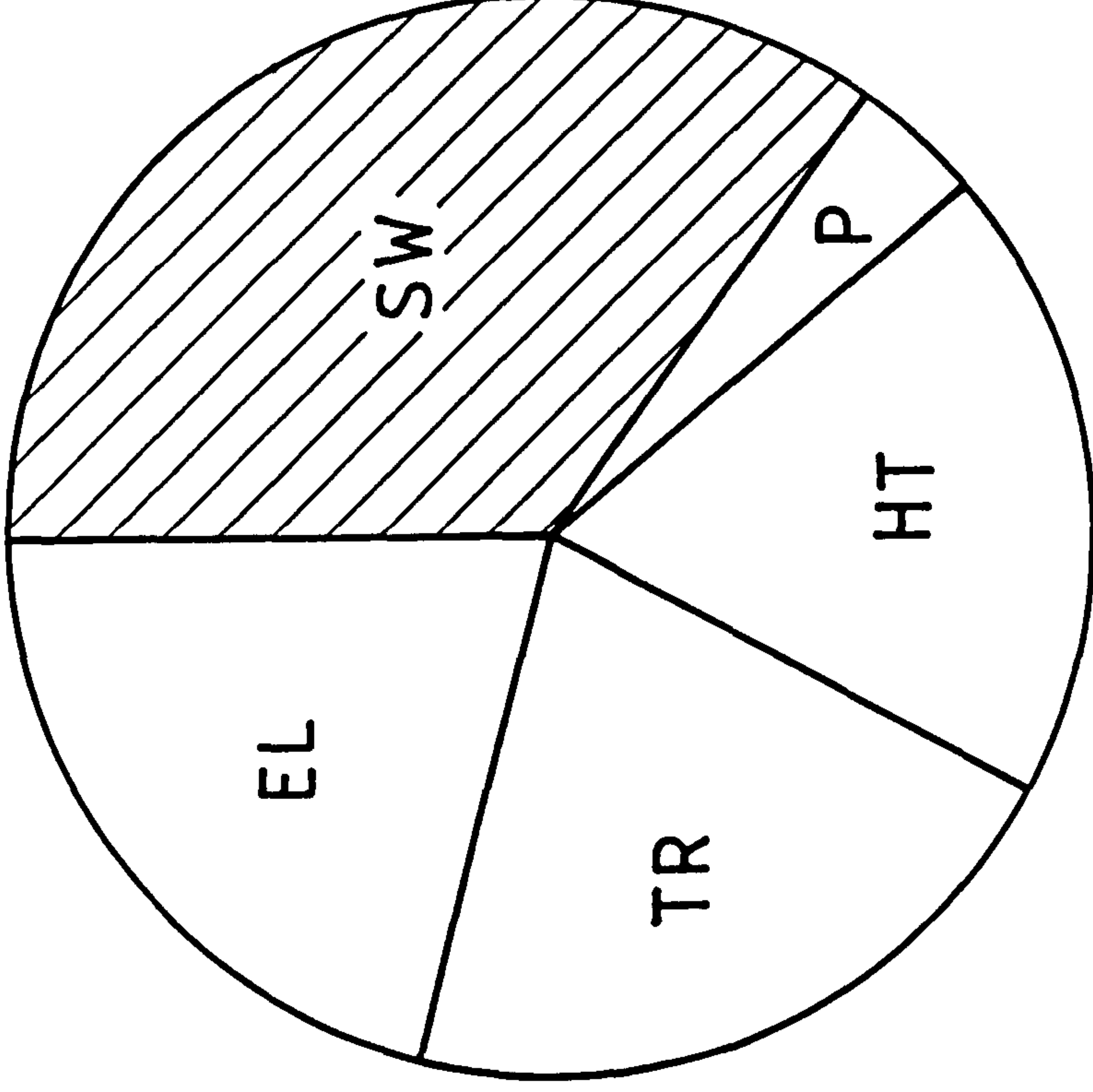
Space and water heating is the major consumer of fuel in the U.K. (see Figure 6.1). Whilst the scope for reducing these energy requirements by measures such as properly-installed thermal insulation is recognised, it is essential that the subsequent residual demand is satisfied with the least energy consumption.

Hydronic central-heating boilers are used to meet nearly all of this demand in the commercial and institutional sector, as well as in a considerable portion of domestic buildings (see Figure 6.2). There are now about 10 million homes in the U.K. with central-heating boilers and up to approximately 1 million commercial and institutional buildings.

EL NON-HEAT USES OF ELECTRICITY
SW = SPACE & WATER HEATING
P = LOW TEMPERATURE PROCESS HEAT
HT = HIGH TEMPERATURE HEAT
TR = TRANSPORT



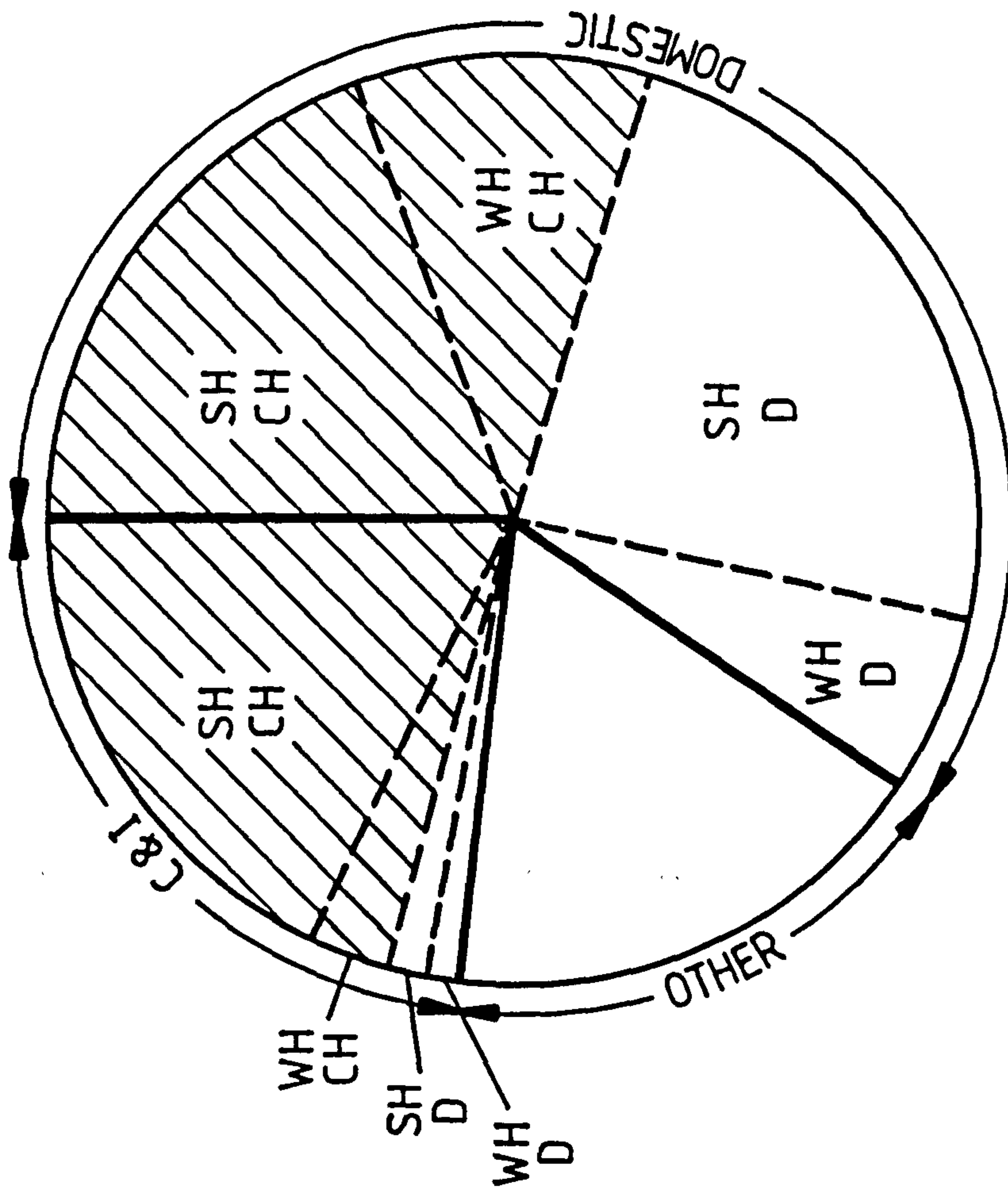
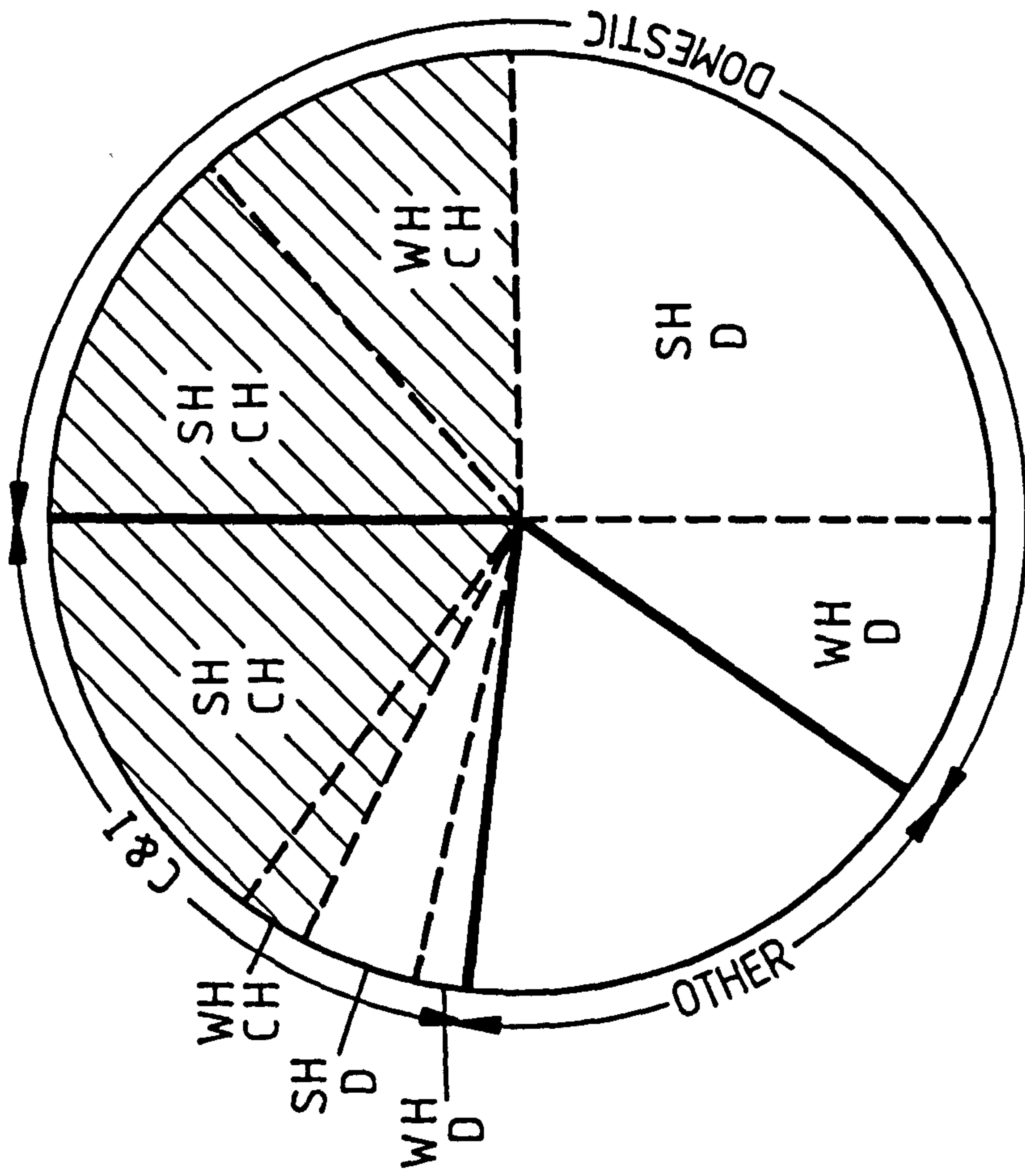
a) DELIVERED ENERGY



b) PRIMARY ENERGY

FIGURE 6.1 THE END-USES OF ENERGY IN THE UK, 1980, (FROM REF. 6.1).

SH = SPACE HEATING
WH = WATER HEATING
CH = VIA CENTRAL HE
D = DIRECT



a) DELIVERED ENERGY

b) PRIMARY ENERGY

FIGURE 6.2 THE SPACE AND WATER HEATING END-USE ENERGY SPLIT ACCORDING TO SECTOR AND SYSTEM TYPE, (FROM REF. 6.1).

6.2.2 Conventional Heating Systems

Today's boilers are cheaper in real terms, more efficient and less bulky than their predecessors of twenty years ago. The recent introduction into the market-place of condensing gas-boilers could be a dramatic step forward in these trends. Simultaneously, system controls are becoming far more versatile and effective, benefitting from the prolonged experience gained with heating systems, and the ready availability of cheap micro-electronics.

However, with most buildings being heated only intermittently, boilers are sized to satisfy the large and rapid pre-heat demands on the coldest days of the year, and are therefore grossly oversized at most other times. In all buildings, incidental gains from lighting, occupants and solar radiation result in further large perturbations of the heat load imposed on the boiler. To match the boiler output to the heat demand, boiler on/off cycling occurs. The number of cycles depends on the building, the radiator size, the boiler capacity and the boiler and system controls, but will always be of the order of tens of thousands of cycles per annum. Inevitable heat losses from the flue and boiler case ensue during the 'off' portion of each cycle, reducing fuel efficiency. (It should be noted that boiler cycling losses are far smaller for forced-draught burners, which are commonly employed in oil and condensing-gas boilers.)

6.2.3 The Heat-Storage Solution

The installation (shown schematically in Figure 6.3) of a heat store between a boiler and its thermal demand, as originally proposed [Ref.6.2], removes the need for boiler over-capacity and reduces the frequency of boiler cycling typically to less than 5 per day, i.e. a few hundred cycles per annum. Thus an annual boiler efficiency close to the peak full-load boiler efficiency is achievable, and annual energy savings of between 5 and 15% could be realised.

In its simplest form, the daily operation would be as follows. Before the pre-heat period, the boiler charges the store: when the pre-heat commences, the store is discharged. After the store's heat is depleted, the boiler fires continuously again, serving the

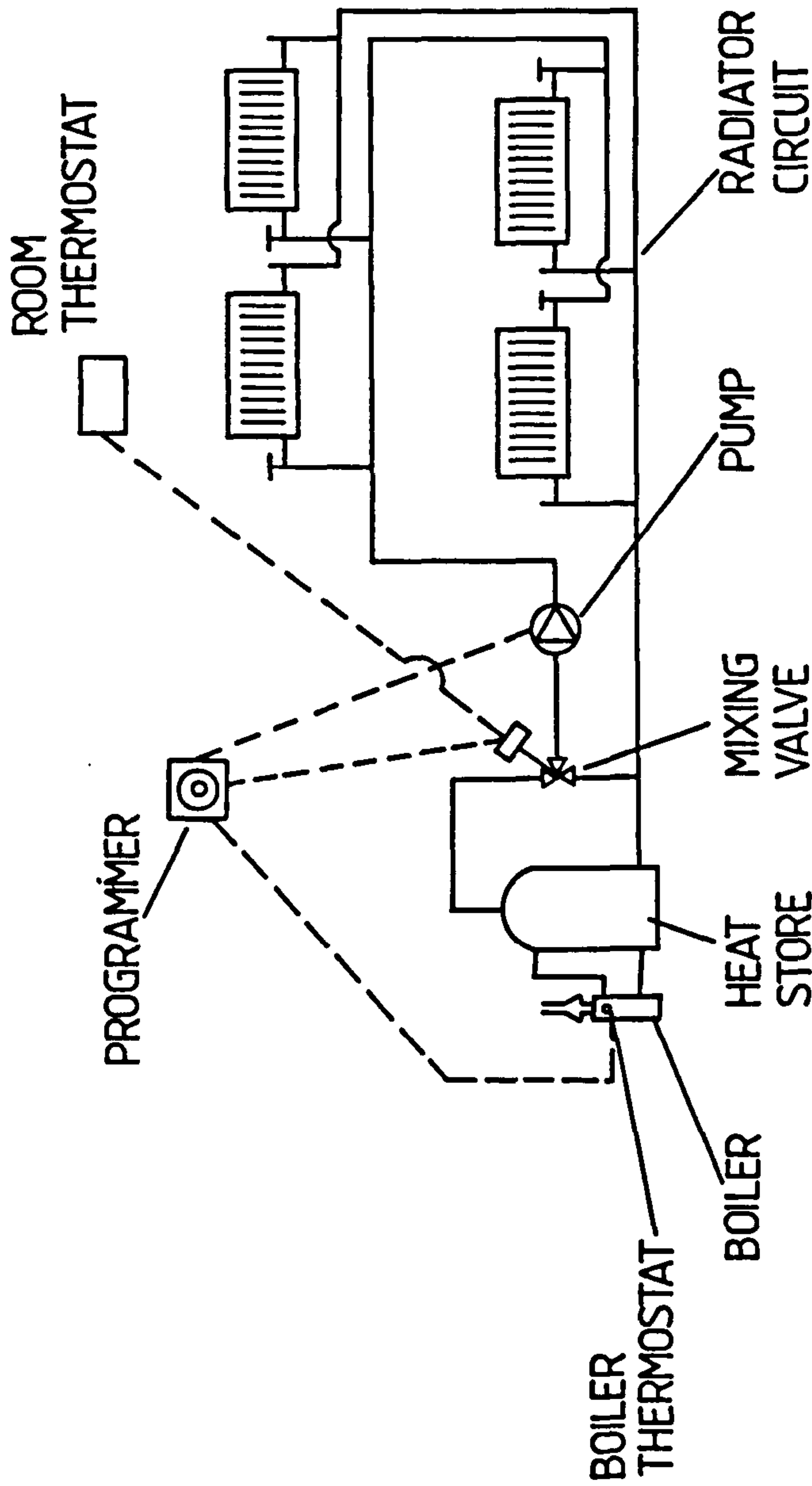


FIGURE 6.3 SCHEMATIC REPRESENTATION OF A DOMESTIC SPACE HEATING SYSTEM INCORPORATING A HEAT STORE.

building's heating needs and recharging the store with any surplus heat. When eventually the store is fully charged, the boiler is switched off and the building is supplied solely from the store.

This routine is repeated until the end of the heating period. In mild weather, the initial store discharge should be adequate to satisfy the demands during the whole day, whereas, in extremely cold conditions, the first recharge might continue all day. For days 'in-between' these extremes, up to about 4 discharge - recharge cycles might occur.

6.2.4 Fast Pre-heat

A prime function of modern heating systems is to bring intermittently-heated buildings to a comfortable temperature as quickly as possible. Storage by means of hot water offers a unique advantage in this respect: it can replace the cold water contained in the heat distribution system by very hot (e.g. 85°C) water almost instantaneously (see Figure 6.4). Assuming stratification can be maintained, a store might deliver to radiators of the order of seven times the power available from the corresponding conventional boiler during this initial period.* The building heats up faster, and therefore the pre-heating process can be started later (see Figure 6.5). As the building loses less energy while it is being heated up, energy savings are realised. This advantage might be further exploited if the comfort benefits of radiant heat from the very hot radiators are sufficient to postpone the start of pre-heat beyond the point indicated in Figure 6.5. A total delay of up to one hour may be possible, which, based on experience with a start-up controller [Ref.6.3], might produce energy savings of up to 5%. This estimated saving is dependent on the building characteristics. It is additional to that discussed in Section 6.2.3 which arises from an increase in a system's annual efficiency.

*FOOTNOTE

The factor of seven has been calculated by assuming that the temperature rise across a conventional boiler is 10°C whilst the temperature rise across a store, during the initial discharge, might be 70°C.

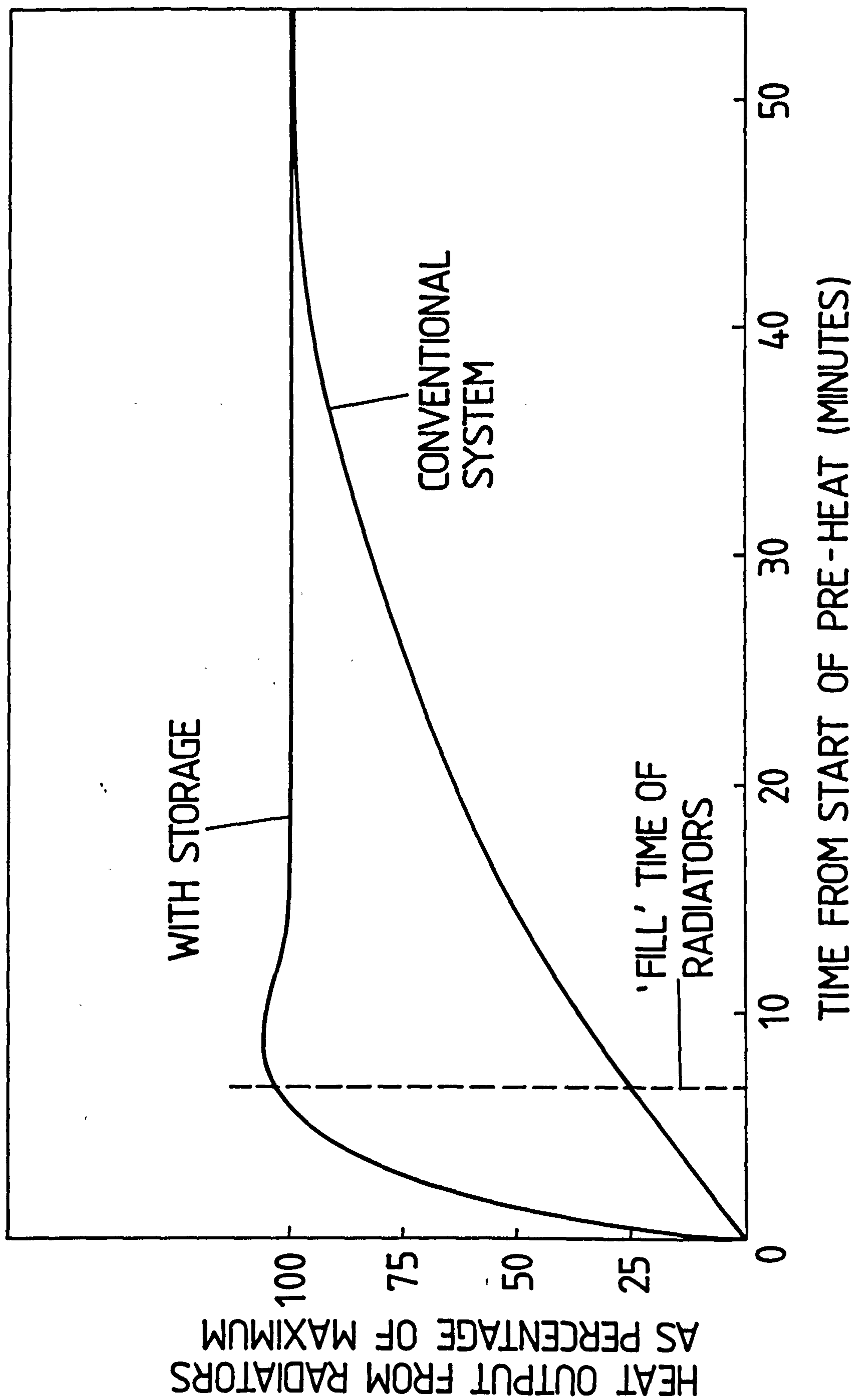


FIGURE 6.4 THE ADVANTAGE OF HEAT STORAGE DURING THE PRE-HEATING OF A BUILDING.

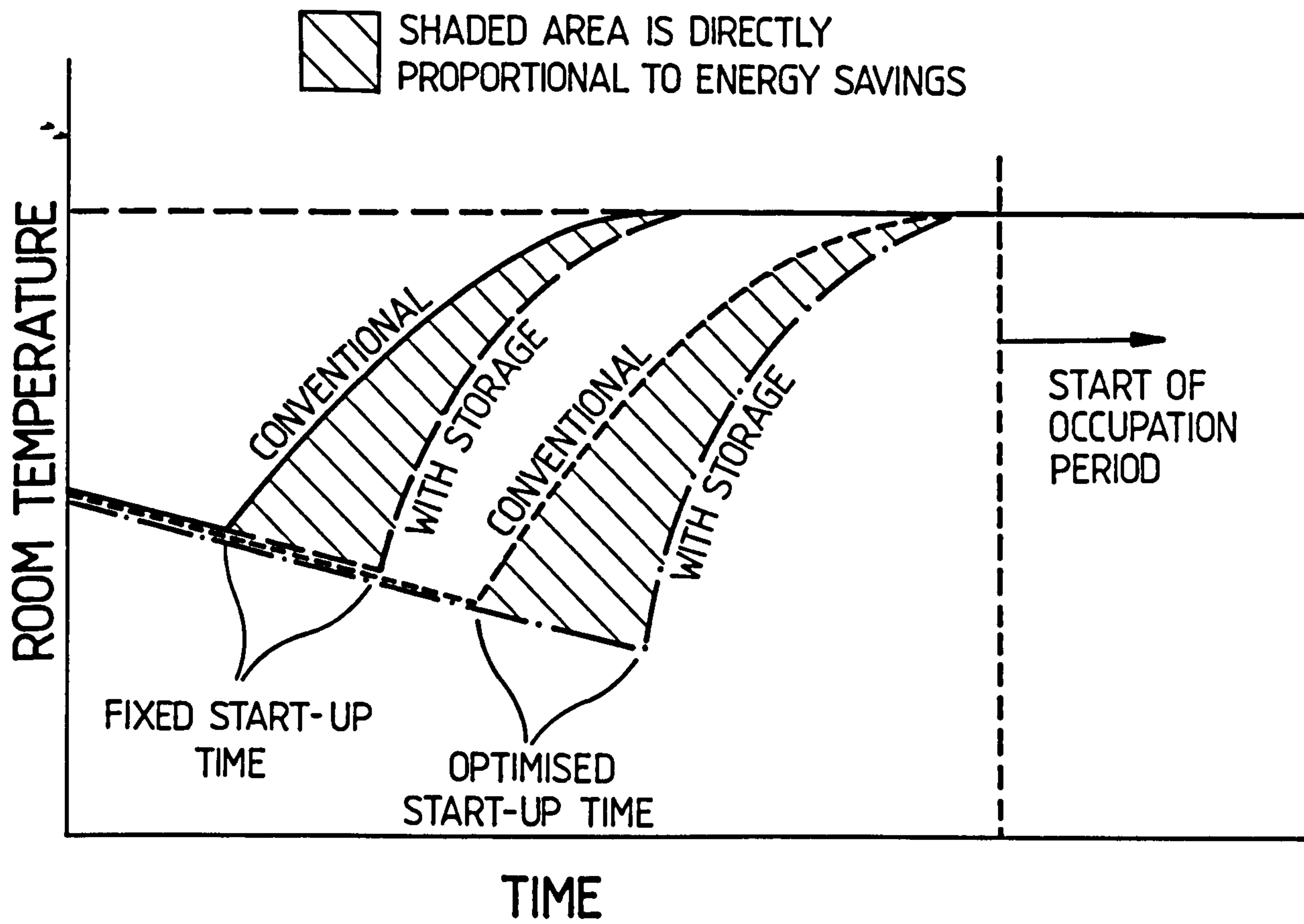


FIGURE 6.5 THE POTENTIAL ENERGY SAVINGS WITH HEAT STORAGE DUE TO MORE RAPID PRE-HEATING.

6.2.5 Sizing of a Boiler and Store Combination

In the U.K., systems are usually sized to satisfy the required performance criteria on a 'design day' (i.e. at an external temperature of -1°C). To achieve this, conventional systems have to be sized to cope with the peak demand, whereas, in systems incorporating thermal storage, it is the time-averaged demand which determines the boiler capacity.

This means that there is a wide choice available for the boiler size and store volume, ranging from a near-conventionally sized boiler with a small store to a much smaller boiler with a very large store. The possible combinations will have various annual efficiencies because, as the store volume to boiler output ratio increases, the boiler cycling frequency reduces whilst the store-jacket heat losses increase.

It is desirable that the store has sufficient capacity to achieve a rapid pre-heating of the building, under 'design day' conditions. A store of 2 to 3 times the volume of the heat distribution system would probably enable the radiators to attain and maintain the required temperature during the initial discharge: this makes due allowance for the thermal capacity of the radiator circuit and for the heat flowing into the building from the radiators as they heat up.

The final selection will be determined by the capital costs of suitable capacity boiler-store combinations and the financial decision criteria of the particular individuals or organisations involved, as well as by any such practical considerations as the space available within, and the means of access to, the 'boiler room'.

6.2.6 Integrated Domestic Hot Water

In the domestic sector, a heat store, as described, can be exploited to provide hot water. A simple coil immersed in the store, and fed by a standard cold mains pipe can be connected to all the required hot water outlets (see Figure 6.6). This arrangement :

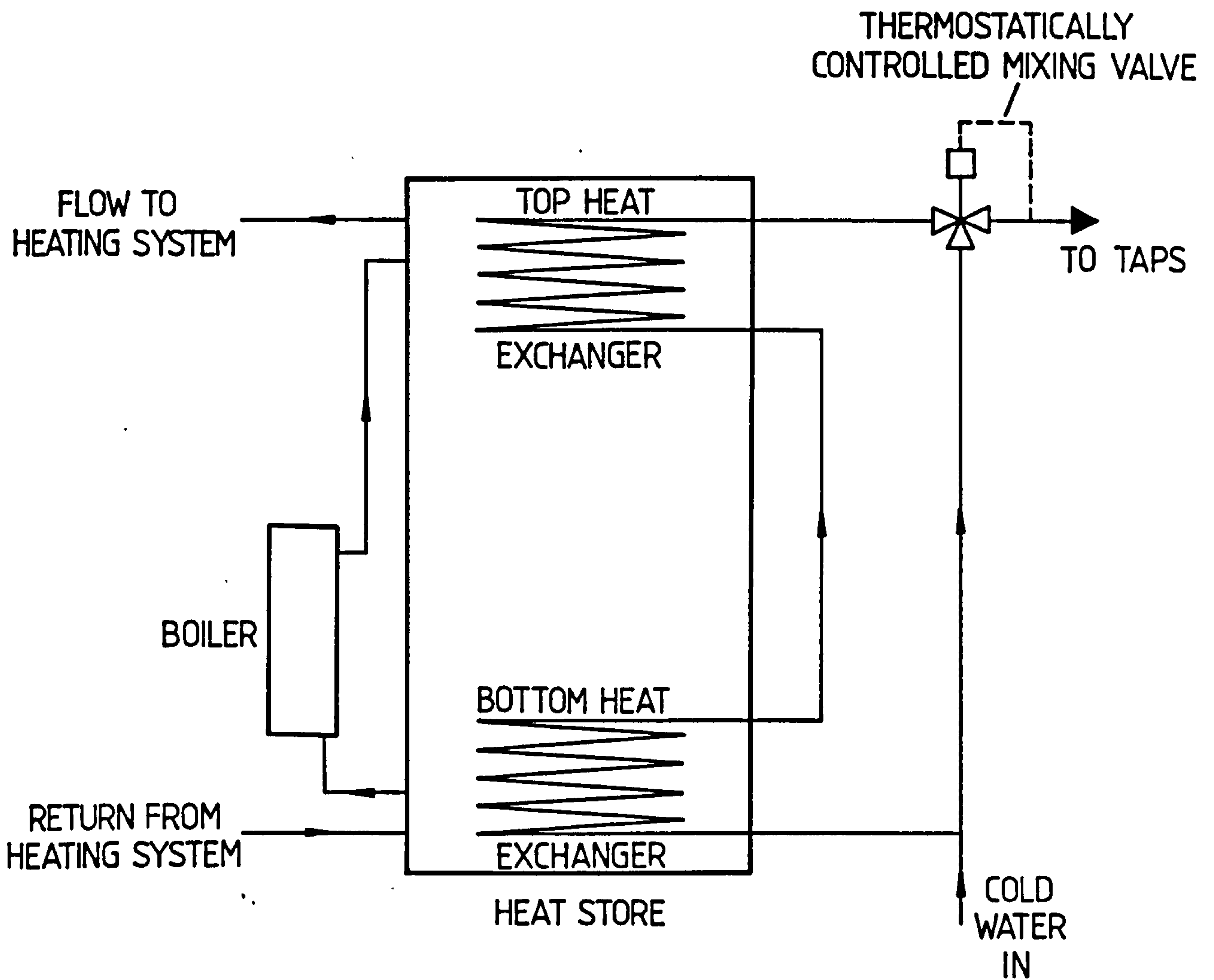


FIGURE 6.6 THE USE OF A HEAT STORE TO PROVIDE DOMESTIC HOT WATER AS WELL AS SPACE HEATING.

- 0 removes the need for a cold feed cistern;
- 0 is very quiet in operation (as it involves no replenishment of the feed cistern water);
- 0 avoids the need for plumbing in attic/roof spaces (thereby reducing capital costs and reducing the likelihood of loft condensation); and
- 0 facilitates the provision of adequate water pressures for showers.

6.2.7 Benefits for the Gas Supply Industry

Heat storage is particularly attractive in the case of gas-fired systems, due to the lower installed capacities needed and, therefore, the lower peak demands on the gas supply. This has two major benefits :-

1. Because the times at which the peak demands on the natural-gas supply network ensue are those when maximum space-heating occurs, decentralised heat storage rather than a centralised gas storage facility, can alleviate potential under-capacity in some extremities of the distribution network.
2. Substantial reductions in the costs of gas service pipes and meters for new or rehabilitated buildings can accrue.

6.3 DESCRIPTION OF THE SYSTEMS TO BE MODELLED

6.3.1 Selection of an Example System

Heat storage can be incorporated into the heating system of a wide range of domestic, commercial and institutional buildings. These include schools, offices, hospitals and hotels, and involve a variety of boiler types, control methods and distribution systems. It was not possible to deal with them all as each building is almost a one-off. Instead it was thought better to deal thoroughly with one which is typical (in so far as that has any meaning) and for which reliable heating demand data were available.

A suitable subject for a modelling exercise was considered to be the typical Institutional sector heating system at Spa School in London. The conventional system there had been monitored by the British Gas Corporation [Ref.6.4] and the School had been selected to be the site of the first comprehensive field trial of heat storage in the commercial and institutional sector [Ref.6.5]. It was therefore appropriate to choose the existing conventional and proposed storage systems at Spa School as the base cases for the simulation studies. Service hot water at Spa School is provided separately, and is therefore not considered by the models. (It was desirable that the provision of service hot-water should not be allowed to confuse the issue. Although there is a trend towards separate heaters for hot water, there are many installations where one appliance has to provide both. This is particularly true where the hot-water load is large, e.g. in hotels.)

6.3.2 The Conventional System at Spa School

This comprised five Hamworthy R320 modular boilers (see Figure 6.7) with a heat input of 97kW each. These were controlled by a Hamworthy sequence controller, set to operate in a compensating mode through an external thermostat. A time-switch brought the system into operation at approximately 3.30 am Tuesday to Friday and 1.00 am on Mondays. The heating was turned off at 3.0 pm each day and did not operate at weekends.. (unless severe weather conditions caused a condensation-prevention thermostat to override the time-switch).

In such a modular boiler system, the previously described cycling losses still take place in the marginally-firing boilers, and, in addition, losses occur from un-fired boilers, through which the return warm water continues to circulate.

6.3.3 The Storage System at Spa School

The system is illustrated in Figure 6.8. Three of the original boilers are used with a 3300 litre water-store. (This sizing was decided after analysis of the behaviour of the conventional system). A mixing valve is positioned between the store and the radiators to produce the same externally-compensated flow temperature as would

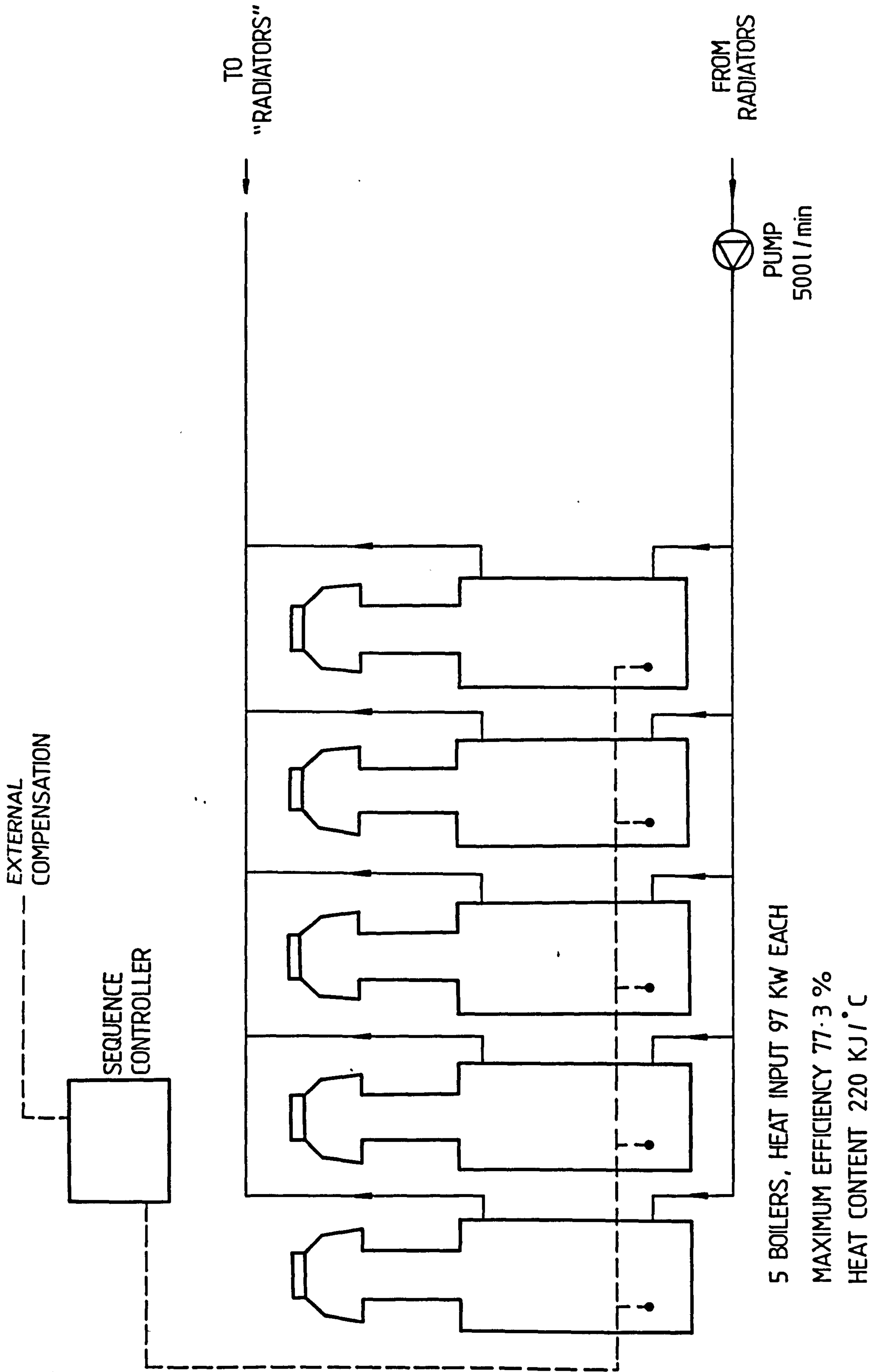


FIGURE 6.7 THE CONVENTIONAL HEATING SYSTEM USED AS THE 'BASE CASE' IN THE COMPUTER MODEL.

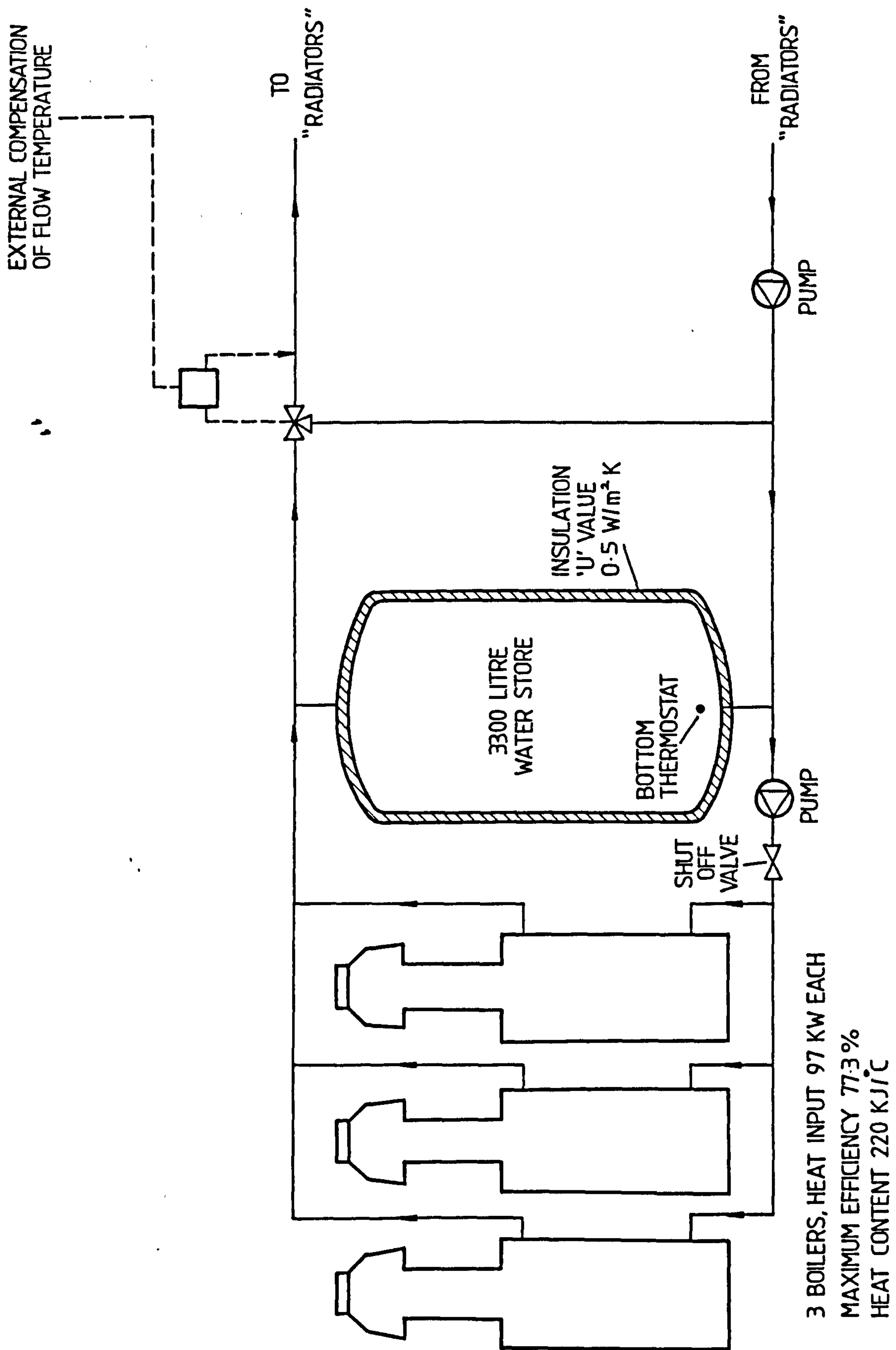


FIGURE 6.8 THE HEAT STORAGE SYSTEM USED AS THE 'BASE CASE' IN THE COMPUTER MODEL.

have been obtained with the conventional system. The boilers do not operate sequentially: they are either all-on or all-off depending on whether the store is being charged or discharged, which is determined by the thermostats in the store.

6.4 DESCRIPTION OF COMPONENT MODELS AND LOAD DATA

6.4.1 Conventional System Model

The program assumes each boiler to have one water and one metal component. It simulates heat transfers from the combustion products to the metal section with an efficiency related to the return-water temperature and from the metal section to the boiler-water at a rate indicated by previous experimental work [Ref.6.6]. When a boiler is off, heat is lost up the flue at a rate discussed in Section 6.4.5. Whether on or off, the heat transfer is described by a set of linked partial-differential equations, which are solved numerically in the simulation. The full equations and their finite-difference formulations are shown in Appendix F.1.

6.4.2 Storage System Model

The boiler model is similar to that for the conventional system except that the thermal capacity of the metal is 'lumped' with that of the boiler water to produce a single component. This simplification is justified by the very low cycling frequency (i.e. about 5 cycles/day) of the boiler. The equations for this boiler model are also given in Appendix F.1.

The store is simulated using the model developed in Chapter 5, except that heat conduction is neglected, since its small influence does not justify the significant increase in computer time that it would cause. The store is divided into 100 litre zones; the number of zones in a mixed region around the inlet (i.e. the depth of penetration of the inlet flow) is calculated using the correlation for the distributors found in Chapter 4. The flow rate through the store is decided by the control algorithm (see section 6.4.4.(b)) before each time step to suit the heat demand. Normally the temperature of the

inlet flow is such that it has reversing buoyancy. When this is not the case, a 'mixing inward' routine is invoked to prevent any inversions occurring. With this precaution, the transient conditions are treated as if they were step input conditions as far as the performance of the store is concerned.

6.4.3 Load Data

The conventional heating system at Spa School was monitored comprehensively between January and May 1982 [Ref.6.4]. Data for forty-five complete days were obtained, (the other days in this period being incomplete due to problems with the monitoring equipment). These data were converted to represent a reduced heating season of 10 weeks duration, which should be reliably representative of a full season. The modified data are in the form of average hourly values of the flow and return temperatures to and from the School's distribution system. In the simulations, these data were combined with a measured value for the distribution-system flow rate to arrive at hourly values for the heat demand on the boilers. The difficult task of simulating the radiator circuit was thereby avoided.

6.4.4 Control Method

(a) Conventional system

The main task of the control algorithm in the program was to make the temperature of the water coming from the boilers as close as possible to the target flow-temperature, which was given as an input datum (see above). This meant, in effect, simulating the operation of the sequence controller. The algorithm worked by changing the number of boilers firing to achieve the right flow-temperature, with the proviso that the number of boilers firing was not altered more than once per minute and that no change was made if the actual temperature was within 0.2°C of the target. This 'dead-band' was selected on a trial-and-error basis, so that the cycling frequency approximated to that observed in the monitored data, as shown in Figure 6.9.

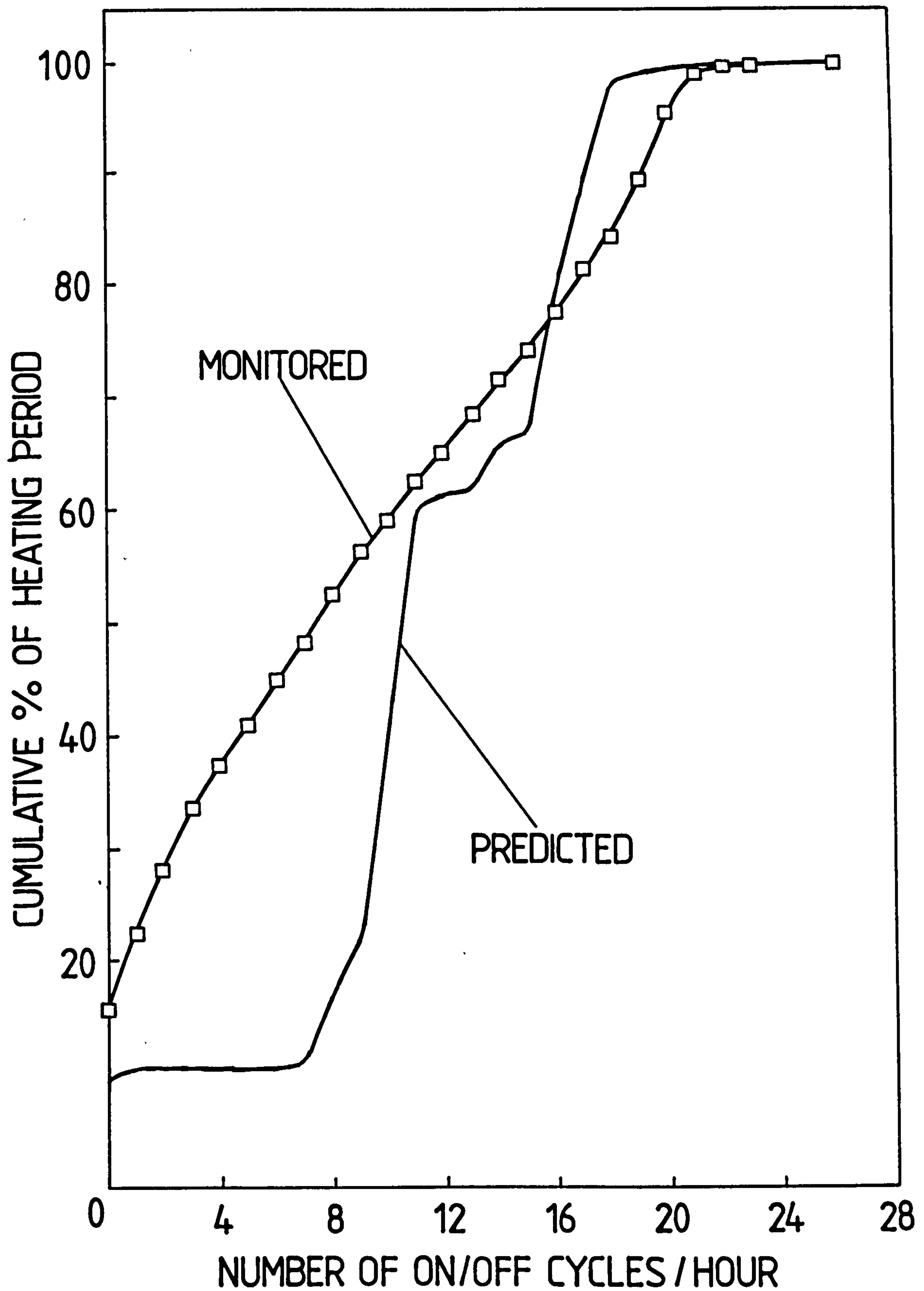


FIGURE 6.9 COMPARISON OF THE MEASURED CYCLING FREQUENCY OF THE CONVENTIONAL SYSTEM BOILERS WITH THE FREQUENCY PREDICTED BY THE COMPUTER MODEL.

(b) Storage System

The system operated basically in the way described in section 6.2.3. The store was taken to be fully charged when the temperature of the bottom zone attained a certain value (i.e. the setting of the bottom thermostat). In the discharge mode, the store was considered to be fully depleted when the temperature of the top zone fell below the required flow temperature for the radiators (an input datum for each hour).

The flow temperature to the distribution system was controlled by the 3-way mixing valve. It was found that the best algorithm involved proportional control of the valve with a small dead-band, i.e. the flow-rate through the store was changed at the end of each time-step by an amount proportional to the off-set between the actual temperature and the target. If the available temperature from the boilers was less than the target temperature, the valve was set so that all the flow went through the boilers.

For both the conventional and storage systems, the final control algorithms succeeded to within about 10% in delivering the required amount of heat.

6.4.5 Validity of the Models

The models for the conventional and storage systems are internally consistent in so far as they both produce precise energy balances over the complete heating season. As regards their realism, they contain as many common elements as feasible, and use identical heat load data, so that, whilst an unavoidable uncertainty in the validity of parts of each model make the absolute seasonal simulation results questionable, an evaluation by comparison of the benefits of incorporating storage should be dependable.

The only truly uncertain part of the models concerned the cooling-down of the boiler. The key parameter in the algorithm is the heat loss rate from the boiler to the ambient air, (i.e. the flue-loss factor) irrespective of whether the loss is up the flue or through its casing. This has a considerable effect on the seasonal

efficiency of the conventional system, as is shown in Figure 6.10, but only a very small effect on the storage system. (For the latter, the boiler is off for so long, that it loses most of its heat even at a relatively slow rate).

There are three different, independent and plausible means of nominating a value for this parameter :

(a) Calibration of the model against monitored data

This method, illustrated in Figure 6.10, fixes the value of the parameter so that the model predicts the same seasonal efficiency as was found for the monitored data. The only snag occurs if this matching has ensued for the wrong reasons.

This method predicts a flue-loss factor of 116 W/K.

(b) Using published data for ventilation heat loss rates from boilers.

A recent paper [Ref.6.7] has presented the results of experimental measurements of ventilation heat losses. Those for one boiler are reproduced in Figure 6.11. Unfortunately, the make of the boiler was not revealed, so its efficiency and thermal capacity cannot be estimated. However, the boiler algorithm was run on the computer under the same circumstances as the reported experiment, assuming a boiler with the thermal properties of the Hamworthy R320. The results for various values of the flue-loss factor are superimposed on Figure 6.11. The experimental results can then be matched to a certain heat-loss rate.

This method predicts a flue-loss factor of 58 W/K.

(c) Using a measured steady-state value.

Measurements of the steady-state heat loss rate from a Hamworthy R320 have been made by British Gas [Ref.6.8]. These were based on the heat loss rate from an un-fired boiler through which warm return water flowed in steady-state conditions. This method is bound to

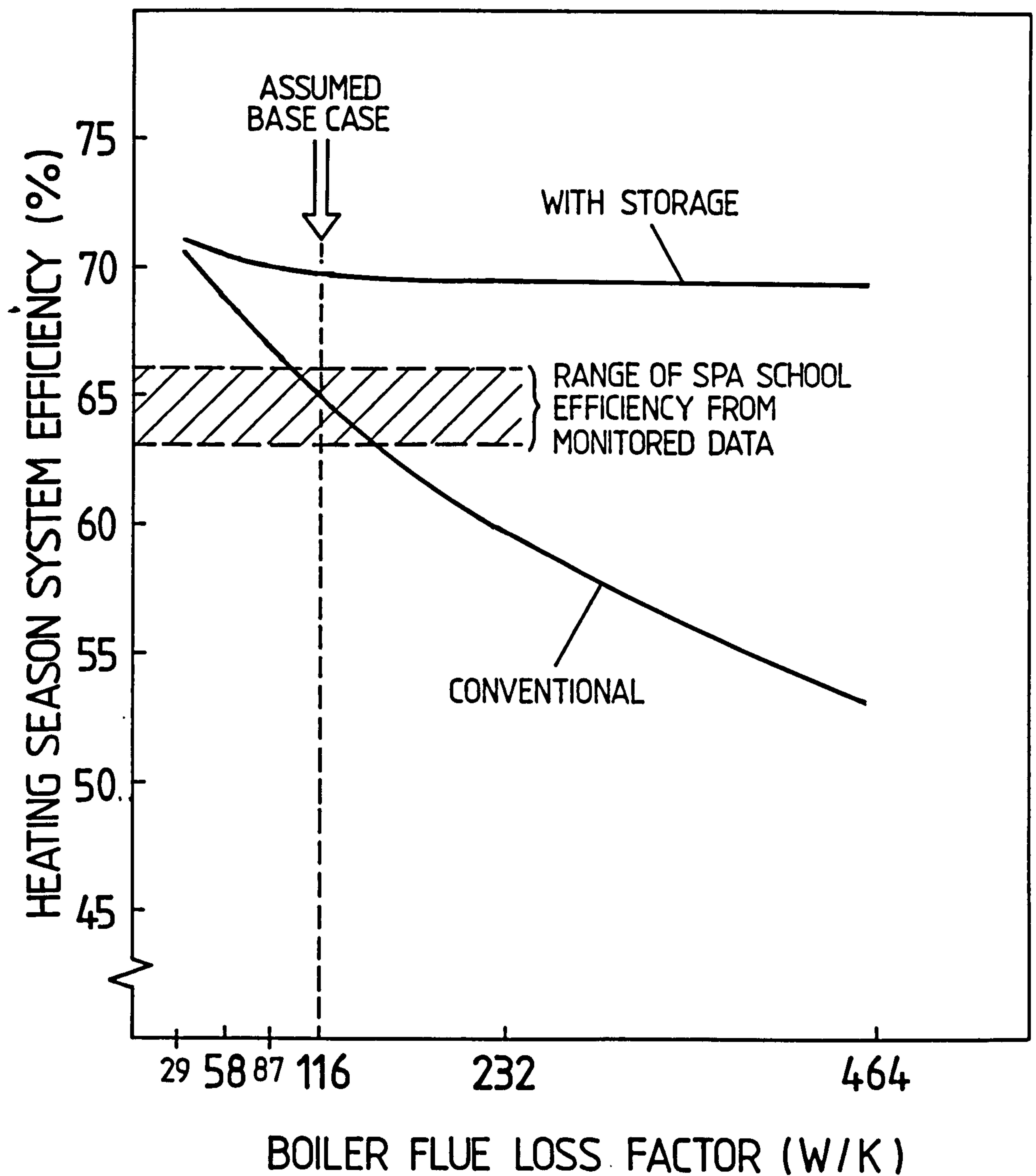


FIGURE 6.10 THE SENSITIVITY OF THE PREDICTED SEASONAL EFFICIENCY TO THE 'FLUE-LOSS' FACTOR FOR THE CONVENTIONAL AND STORAGE HEATING SYSTEMS.

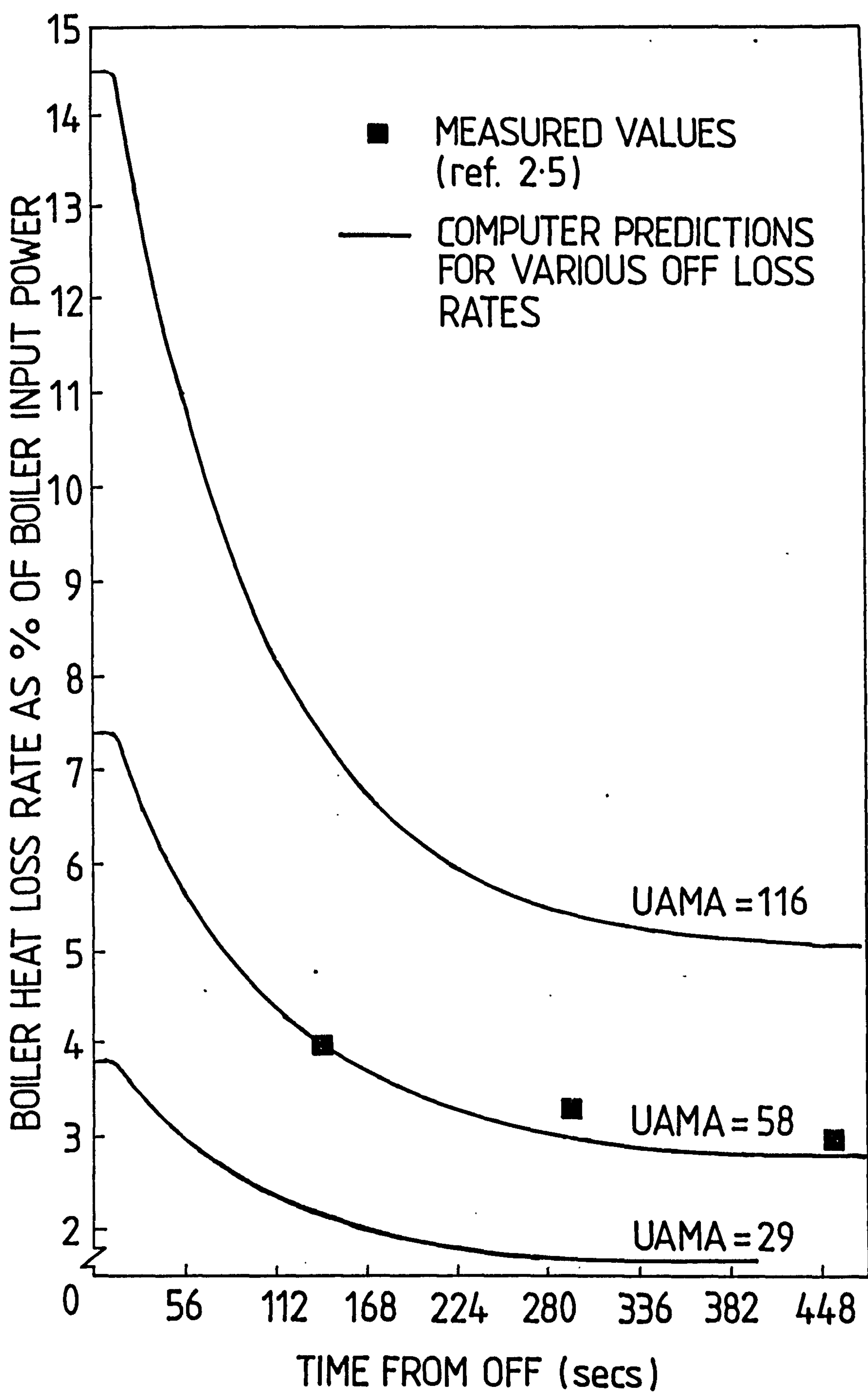


FIGURE 6.11 THE PREDICTED HEAT LOSS RATES FROM A HAMWORTHY R320 BOILER FOR 3 VALUES OF 'FLUE-LOSS' FACTOR COMPARED WITH PUBLISHED MEASURED VALUES FOR AN UNAMED BOILER.

underestimate the transient heat loss rate just after a boiler is switched off. Although it should indicate the asymptote for a true curve plotted in Figure 6.11.

This method predicts a flue-loss factor of 29 W/K.

Agreement of the predicted results for the seasonal efficiency from the conventional model with its measured value was considered to be paramount, and therefore a flue-loss factor of 116 W/K has been used for the Hamworthy R320 boiler throughout this study.

6.5 SIMULATION RESULTS

6.5.1 Base-case Results

The main results for the base-case runs are given below; the heat flows are illustrated in the bar chart in Figure 6.12.

Quantity		Conventional System	Storage System
Gas input	GJ	348	331
Useful heat	GJ	226	231
Target heat demand	GJ	220	220
% of target achieved	%	103	105
Combustion losses	GJ	84	82
Boiler off losses	GJ	33	12
Pilot losses	GJ	6	4
Store losses	GJ	-	2
Average number of boiler cycles/day*	-	142	5
Seasonal efficiency	%	64.8	69.8
Maximum efficiency	%	74.2	74.0
Relative efficiency (= seasonal/maximum)	%	87.3	94.3

* N.B. For the conventional system this represents the sum of all the individual boiler cycles; for the storage system this is the number of times all the boilers cycle.

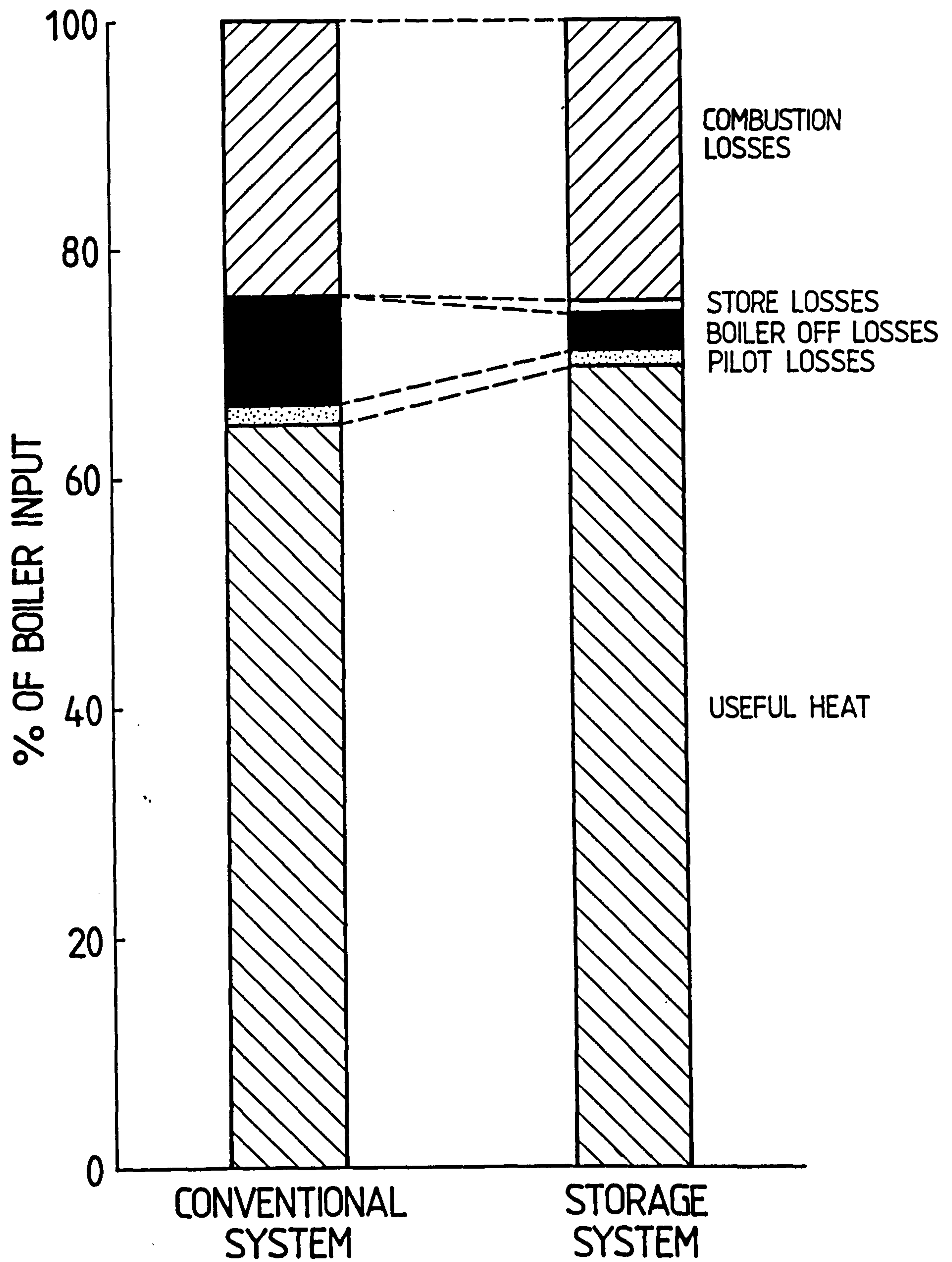


FIGURE 6.12 ENERGY AUDIT OF THE BASE-CASE CONVENTIONAL AND STORAGE HEATING SYSTEMS SIMULATED OVER A COMPLETE SEASON.

The results predict an 8% improvement in seasonal performance through the use of storage. This is nearly all achieved through reduced boiler cycling. Heat losses from the store are very small, amounting to only 0.7% of the gas input. It is assumed throughout the study that store losses are not useful.

6.5.2. Sensitivity Studies

This section gives an assessment of the sensitivity of the base case results to changes in key heating system and building parameters. The method employed was to keep all of the variables at their base-case values except the one being studied, which was varied through a realistic range.

(a) Building type

In order to assess the influence of building type, hourly heat demand data for a large office block (SEGAS House) were obtained from the British Gas Corporation. Using their THERM program [Ref 6.9], the offices were simulated for the full 244 day heating season of 1959-60, assuming a 17-hour heating period. The average daily heat demand of the office block was about twice that of the school. The heat demand data was made compatible with the Spa School computer model by converting it to radiator flow and return temperatures using a fictitious external compensator, the external temperature for each hour being known.

Approximately the same percentage improvement in seasonal system efficiency is predicted for the storage systems at Spa School and at the SEGAS offices (see Figure 6.13).

(b) Heating period

The effect of heating period was most conveniently determined for the SEGAS office block. Heat demand data for the same heating season (1959-60) was attained using the THERM program, assuming a continuous (24-hour) heating period. The performances of the conventional and storage heating systems were then compared under the regimes of continuous and intermittent heating. If the conventional system

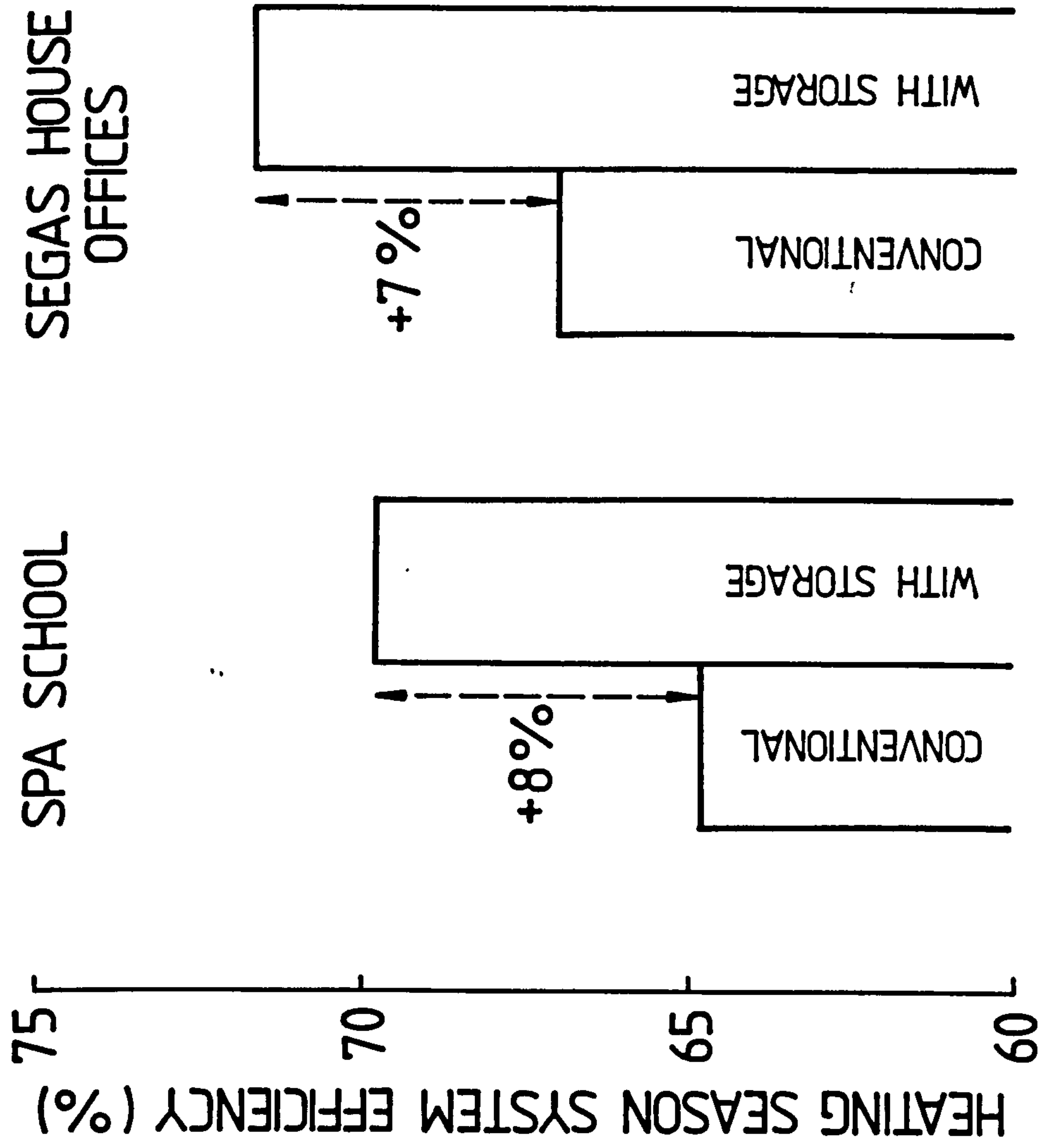


FIGURE 6.13 A STORAGE SYSTEM IS PREDICTED TO ACHIEVE SIMILAR % ENERGY SAVINGS IN DIFFERENT BUILDING TYPES.

boiler plant is specifically sized for continuous heating, it need be no larger than the storage system plant. In this situation, the efficiency of the storage system is only slightly superior (1%), as shown in Figure 6.14. This compares to the 7% improvement predicted with intermittent heating, when the conventional system plant needs to be 40% larger. However, if it were felt necessary to size the conventional system for intermittent heating, then its efficiency under a continuous heating regime is predicted to plummet to a level 9% below the storage system's.

The rest of the sensitivity studies are done using the heat demand data for the Spa School reduced heating season of 10 weeks, described in section 6.4.3.

(c) Number of boilers

The performances of both systems are predicted to drop markedly as more boilers are employed (see Figure 6.15). This is a consequence of the lower load factor producing more cycling, the extra boiler thermal capacity available to be lost and the extra pilot-light consumption. The figure shows that one of the main benefits of storage is being able to manage with fewer boilers than a conventional system.

(d) Boiler type

The Hamworthy R320 is a conventional, relatively high thermal-capacity boiler. It was decided to assess the Stelrad Super Concord, which is a semi-condensing low thermal-capacity boiler with a stacking design, which aims to capture losses from unfired boilers. It also employs a forced-draught burner (as opposed to Hamworthy's natural-draught design) which should cut dramatically the ventilation heat losses [Ref.6.7]. All these factors might be expected to reduce the advantage of storage. The result (see Figure 6.16) actually predicts that storage can improve the performance of this type of boiler by 5%. However, it must be stressed that this figure is only based on estimates for the boiler-off heat loss rate of the Stelrad modular arrangement.

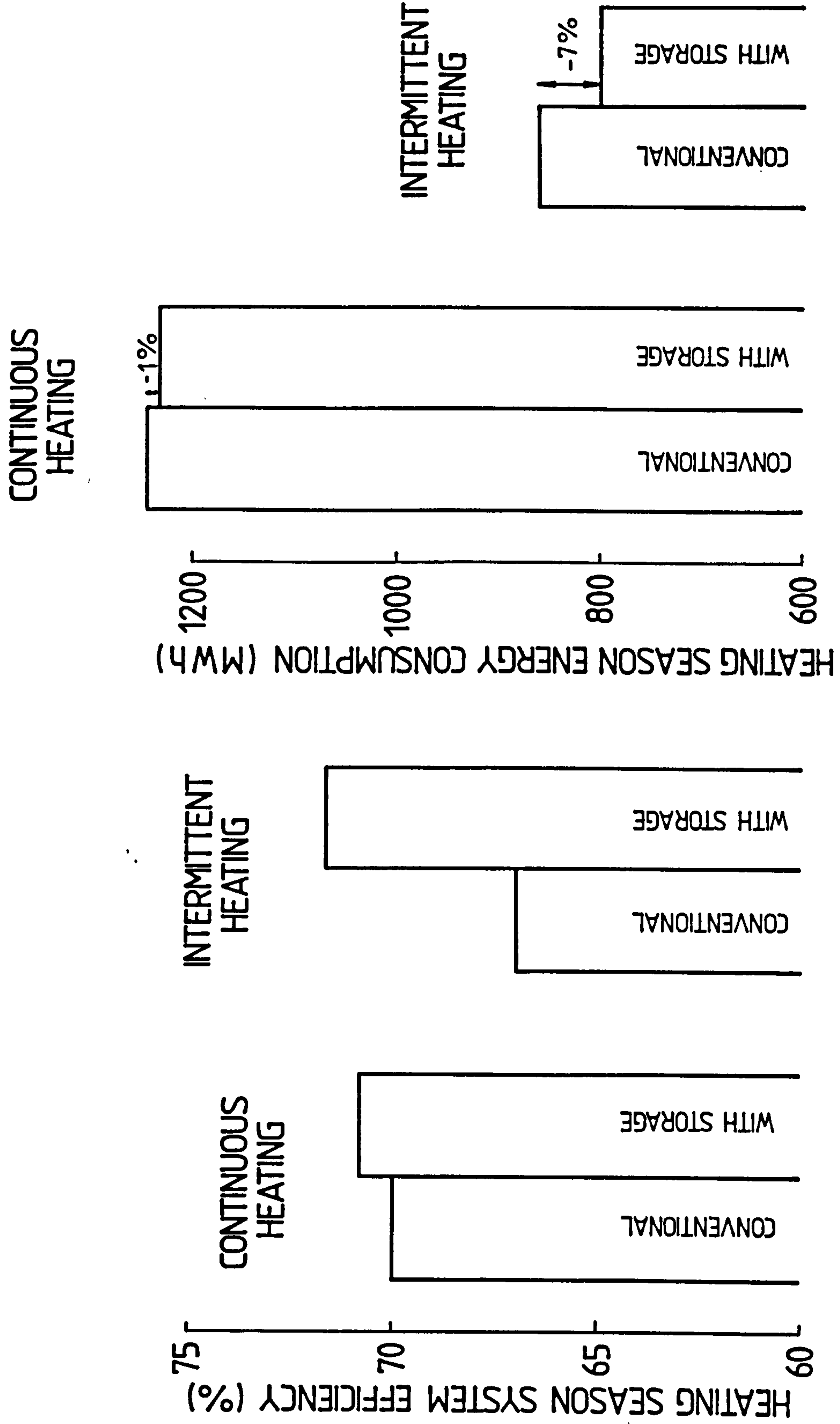
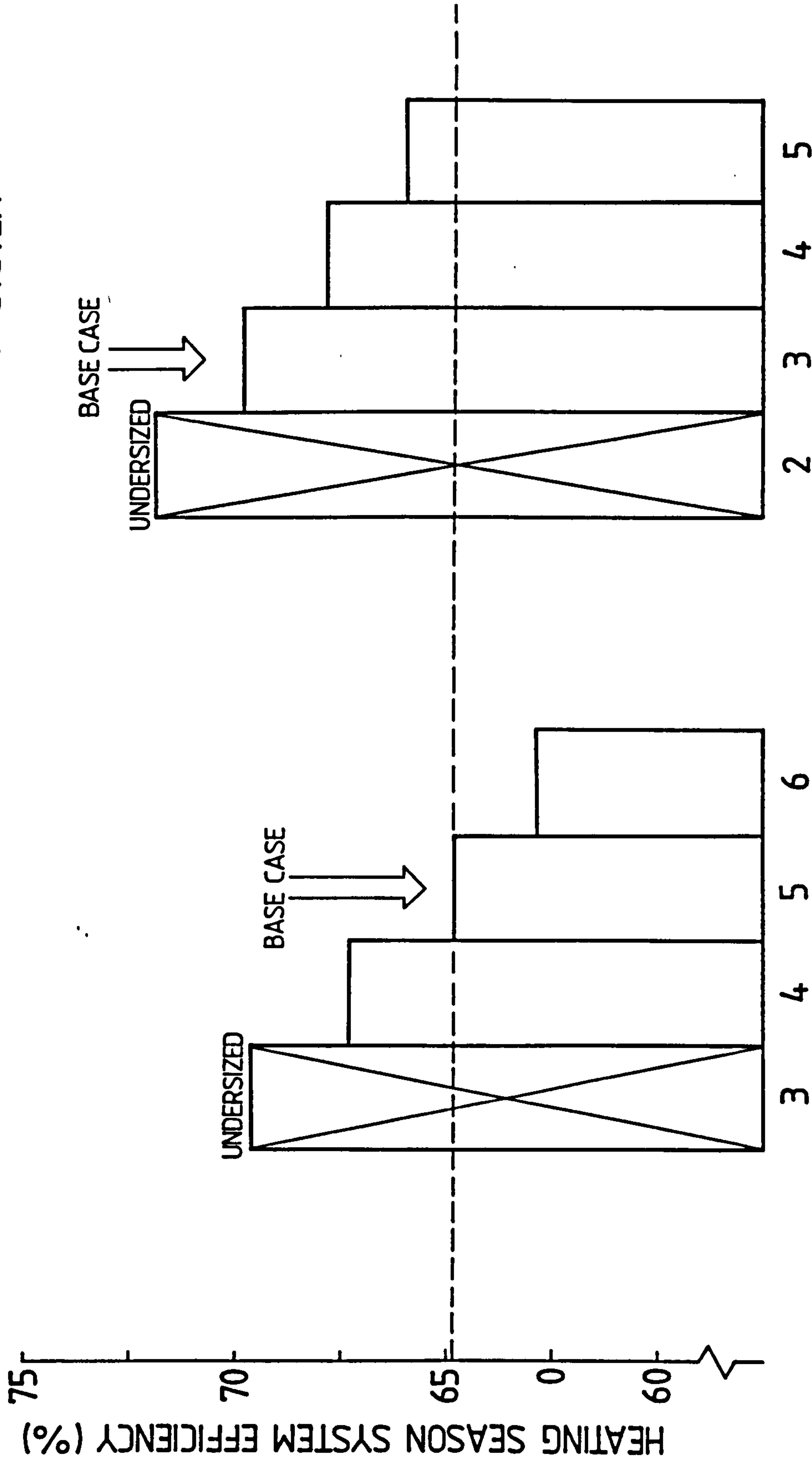


FIGURE 6.14 UNDER A CONTINUOUS HEATING REGIME, CONVENTIONAL AND STORAGE SYSTEMS HAVE A SIMILAR PREDICTED EFFICIENCY (ASSUMING THEY HAVE THE SAME SIZED BOILER PLANT).

CONVENTIONAL SYSTEM

STORAGE SYSTEM



NUMBER OF BOILERS IN HEATING SYSTEM

FIGURE 6.15 BOILER OVERSIZING IS PREDICTED TO HAVE A SIMILARLY DELETERIOUS EFFECT ON THE EFFICIENCY OF EACH SYSTEM.

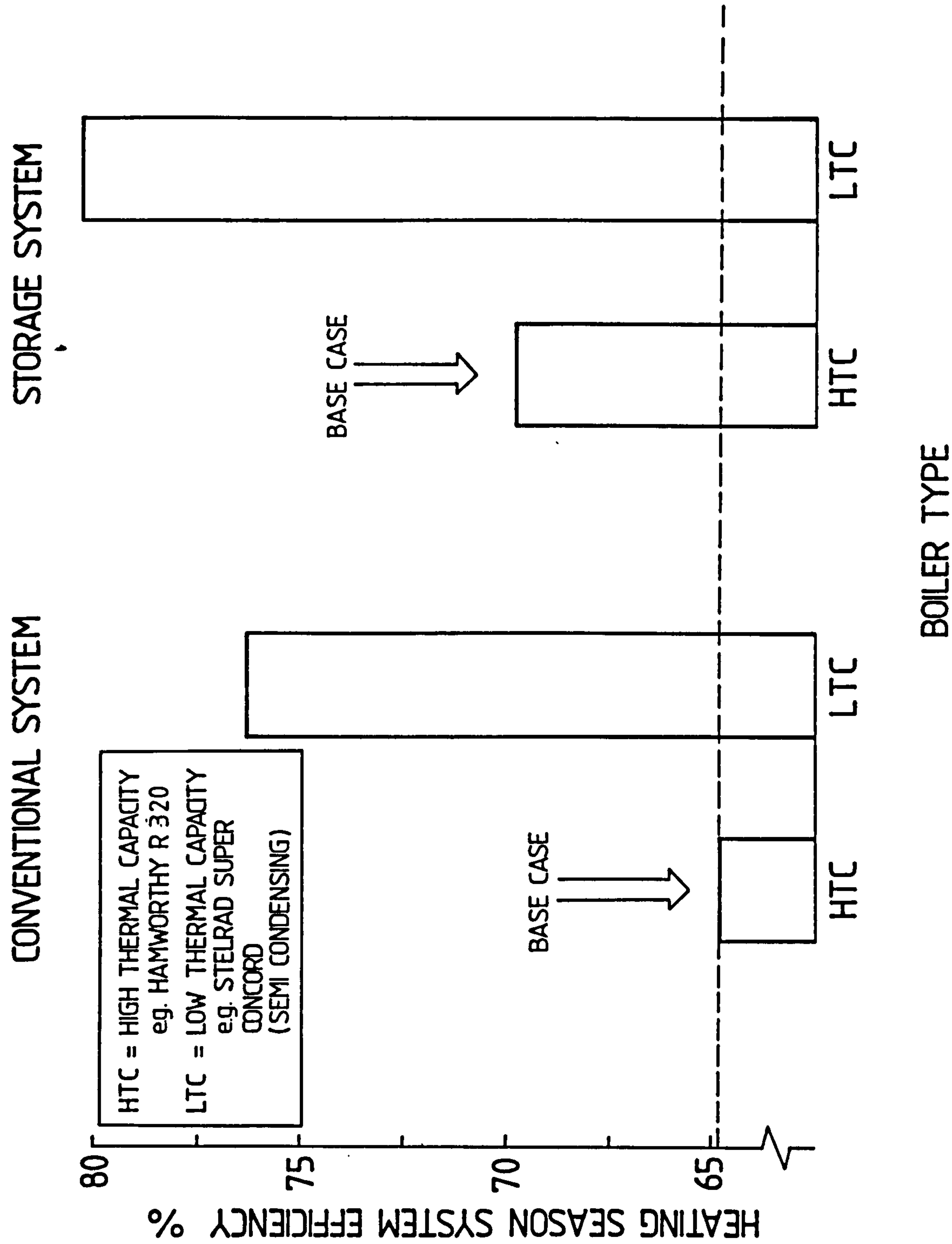


FIGURE 6.16 STORAGE IS PREDICTED TO IMPROVE SIGNIFICANTLY THE EFFICIENCY OF EVEN LOW THERMAL CAPACITY BOILERS.

(e) Store volume

The program predicts only a small improvement (1.6%) in performance as the store volume is varied from 1m^3 to 10m^3 (see Figure 6.17). This confirms that reduced boiler cycling is being off-set by a growing heat loss. The figure also shows the situation of the store loss being useful or, equivalently, a store with perfect insulation. Under this circumstance, an improvement of 3.9% is predicted to occur over the range 1m^3 to 10m^3 .

Even this is a relatively modest gain, and the reason is that as the store volume increases, the more oversized the boiler-store combination becomes. The 10m^3 store could be operated with 2 boilers instead of 3, in which case the seasonal efficiency is predicted to be 3.1% higher than for the 3 boilers with 1m^3 store combination, assuming the store losses are not useful and 5.6% higher if the store losses are useful.

(f) Store temperature swing

The thermostat at the bottom of the store and the minimum allowable temperature at the top of the store determine the swing in temperature in the store during a charge/discharge cycle. The results (Figure 6.18) predict that these settings have little influence on system performance. If the temperature swing is increased by allowing the store to charge to a higher temperature, there is a modest reduction in boiler cycling but also a greater heat loss. The highest seasonal efficiency (70.4%) is achieved with both limits set as low as possible, i.e. the externally compensated temperature of the flow to the radiators.

(g) Store insulant thickness

This study (Figure 6.19) predicts that the store heat losses increase significantly if the insulant thickness drops below about 40mm (of mineral wool), but that there is little benefit in insulating the store with more than about 80mm of mineral wool.

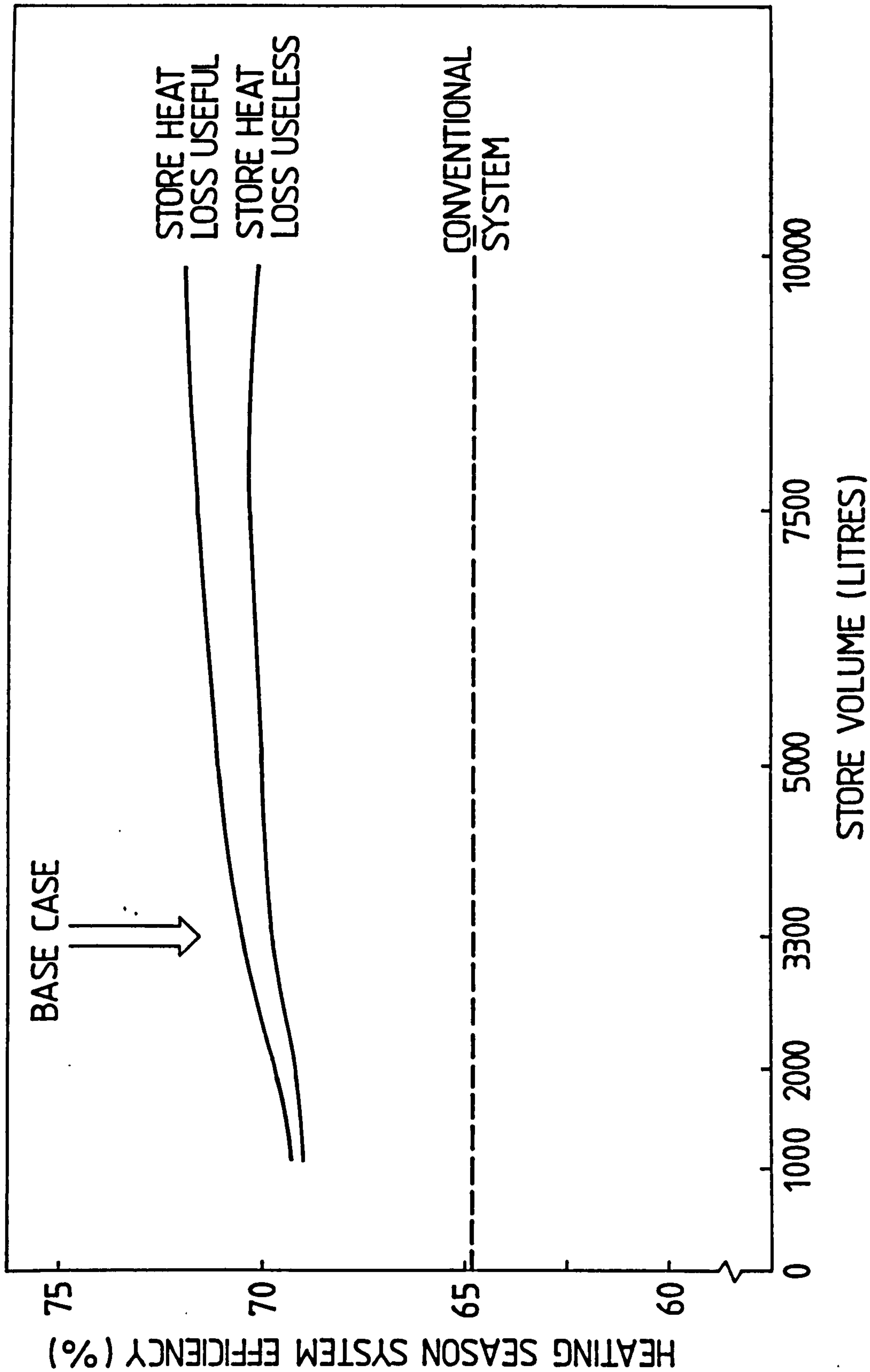
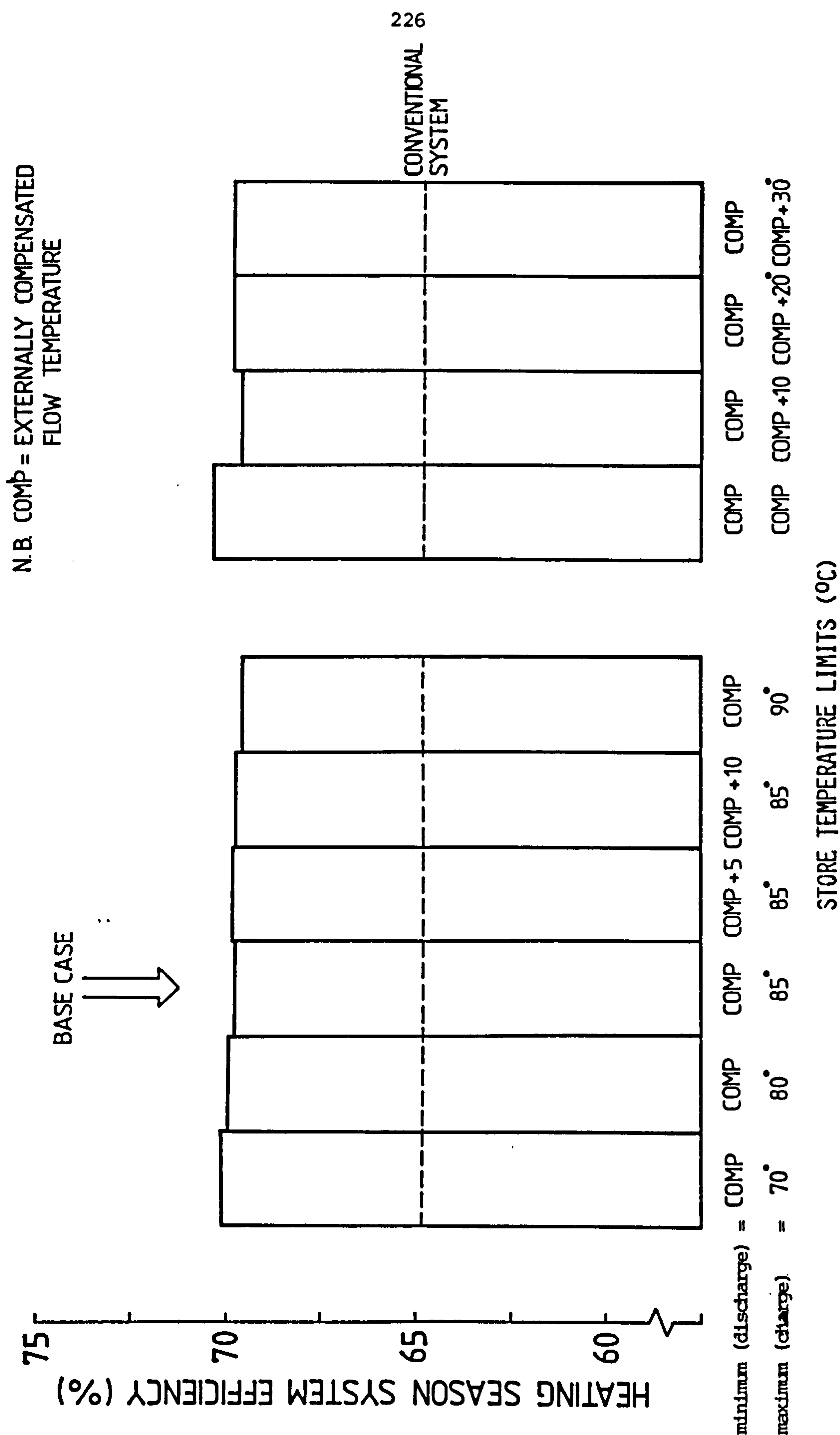


FIGURE 6.17 THE EFFICIENCY OF A STORAGE SYSTEM IS PREDICTED TO IMPROVE ONLY MODESTLY AS THE STORE VOLUME IS INCREASED.



minimum (discharge) = COMP COMP COMP COMP COMP COMP COMP COMP

maximum (charge) = 70° 80° 85° 85° 85° 90°

STORE TEMPERATURE LIMITS (°C)

FIGURE 6.18 THE MAXIMUM AND MINIMUM SETTINGS OF STORE TEMPERATURE ARE PREDICTED TO HAVE LITTLE EFFECT ON SYSTEM EFFICIENCY.

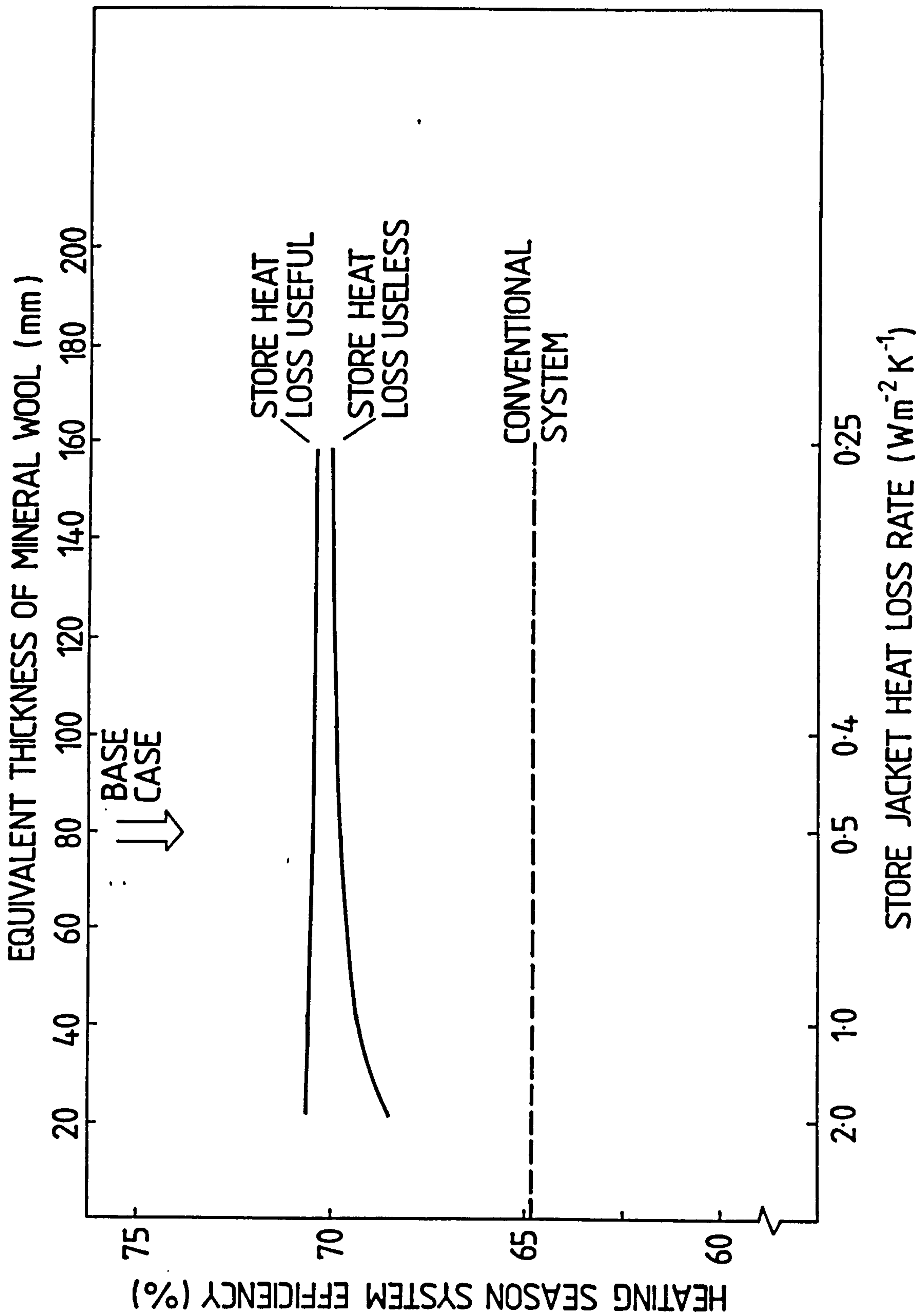


FIGURE 6.19 THE OPTIMUM THICKNESS OF STORE INSULATION IS PREDICTED TO LIE IN THE RANGE OF 40 TO 80 MM OF MINERAL WOOL.

(h) Temperature Stratification in the store

The degree of mixing in the store is assumed to be determined by the depth of penetration of the inlet flow. This study predicts that the system performance is almost independent of the degree of mixing over the complete range from a fully-mixed store to a perfectly stratified store (see Figure 6.20). However, the program did not allow short circuiting, even when the predicted depth of penetration exceeded the store height. Furthermore, it was not possible to take account of the faster pre-heat produced when stratification ensures that all of the cold water from the radiators is 'trapped' in the store during the initial discharge.

6.5.3 Conclusions on the Simulation Results

The computer simulations predict that incorporating hot water storage into a well-controlled, modern central heating system will enhance the seasonal efficiency by about 8% except where the building is heated continuously. This improvement is not particularly sensitive to changes in most of the system parameters, although the performance of either system suffers if the boilers are oversized. The result is however sensitive to the assumed rate of heat loss from a boiler during the off portion of a cycle. Further work is necessary to confirm the accuracy of the estimated value for this parameter.

The results verified that major reductions in installed boiler capacity are made possible by the use of storage in intermittently-heated buildings.

The study indicated that the performance of the store in terms of stratification might only be significant during the first discharge of the day, when the building is being pre-heated.

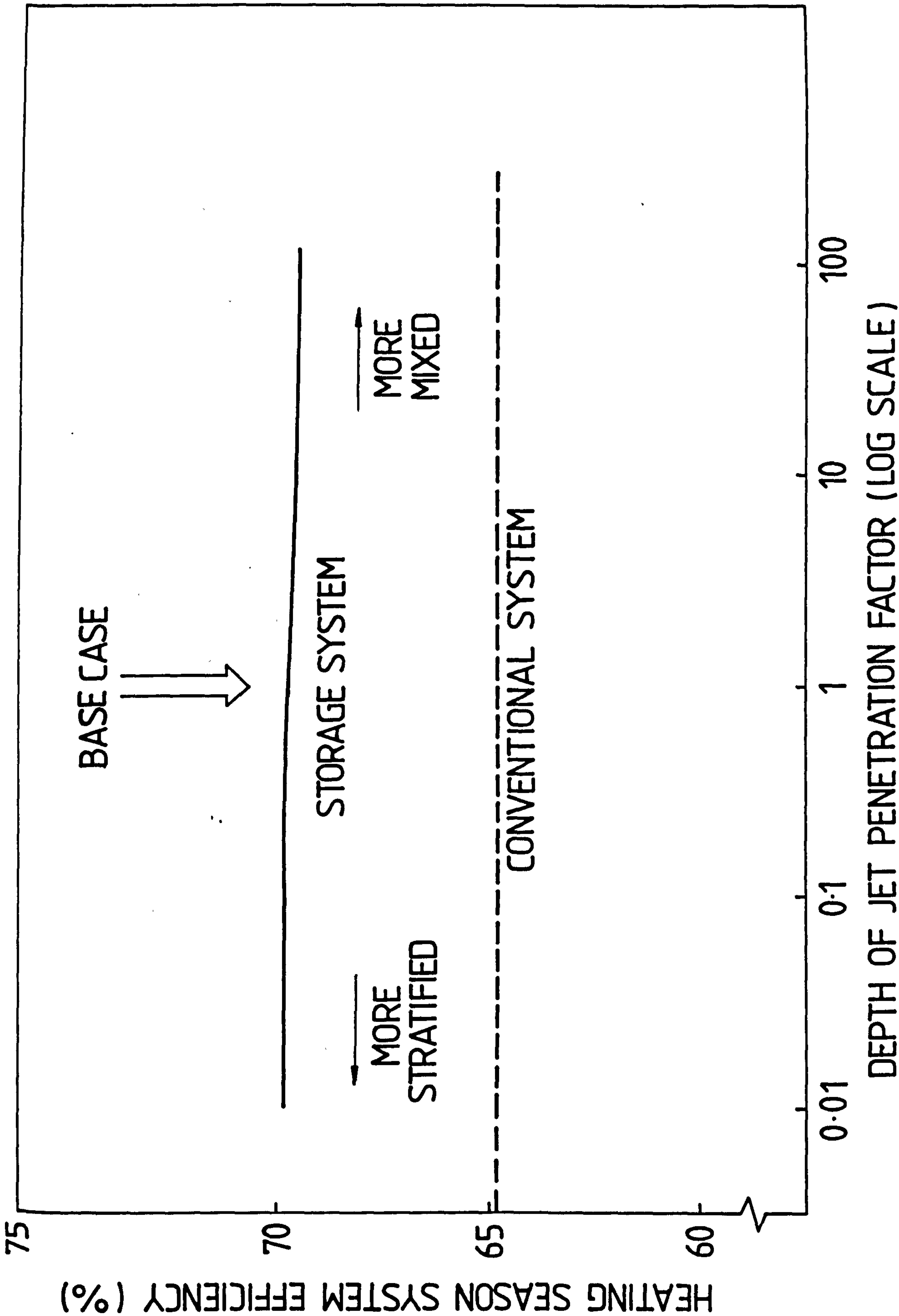


FIGURE 6.20 STORE STRATIFICATION IS PREDICTED TO IMPROVE SYSTEM EFFICIENCY ONLY MARGINALLY DURING NORMAL OPERATION.

6.6 ECONOMIC APPRAISAL

6.6.1 Costing Method

Estimation of materials costs and installation costs of boilers or complete heating systems is a notorious minefield. It is common knowledge, for example, that reputable quotations for exactly the same job can vary by a factor of two or more. This position is particularly endemic to the domestic sector. However matters are no easier in the commercial and institutional sector, where buildings and their heating systems are so different from one another that, for example, the same boiler room plant might cost twice as much to install in one building as another due to the space available, accessibility, etc.

To try to make some sense of such a hazy area, it was decided to employ marginal cost analyses wherever possible. This involved calculating only the difference in costs between those for storage and conventional systems, not the absolute costs of each. Thus the materials and installation costs of all the common elements of the two systems could be ignored.

In view of the current position, the economic analysis has used the present price of fuel: no attempt has been made to allow for possible fuel price increases (in real terms) which would clearly make the higher efficiency of heat storage systems that much more valuable.

6.6.2 The Installation Circumstances

The incorporation of energy storage within a heating system is considered under four circumstances :-

- (a) A Completely new heating system for a new building or a building previously without central heating. This will undoubtedly be the most attractive case as there are no restraints on installing a purpose-built system with the most suitable boiler, store, plumbing and controls.

- (b) A Replacement system for inefficient or worn-out boilers in an existing heating-system. In this case, the cost of the storage system can be offset against the cost of a conventional replacement system. However extra plumbing may be needed and many of the controls would have to be changed.
- (c) A Store as an add-on component to enable existing boiler(s) to accept increased heat loads resulting from building or heating-system extensions. In this application, the cost of the store can be off-set against the cost of a replacement, larger boiler. However there is likely to be comparatively more plumbing involved and some new controls will be required.
- (d) A Retrofit system to improve the performance of existing boiler(s). Under this circumstance, the cost of the store, associated plumbing and any new controls required would all have to be justified by reductions in running costs.

6.6.3 Definition of Equivalent System Designs

This economic assessment has taken as its subject the same building as the simulation studies. However it could be considered more generally as an intermittently heated building with a seasonal heat demand of 225,000 kWh. Based on the computer simulations described in Section 6.5, it was possible to nominate one conventional system and four systems with storage, capable of meeting this demand (see Figure 6.21). Although the analysis assumes that modular boilers are employed, the trend of the results should be applicable to other types of boiler.

The capital and running costs of each system are summarised in Table 6.1.

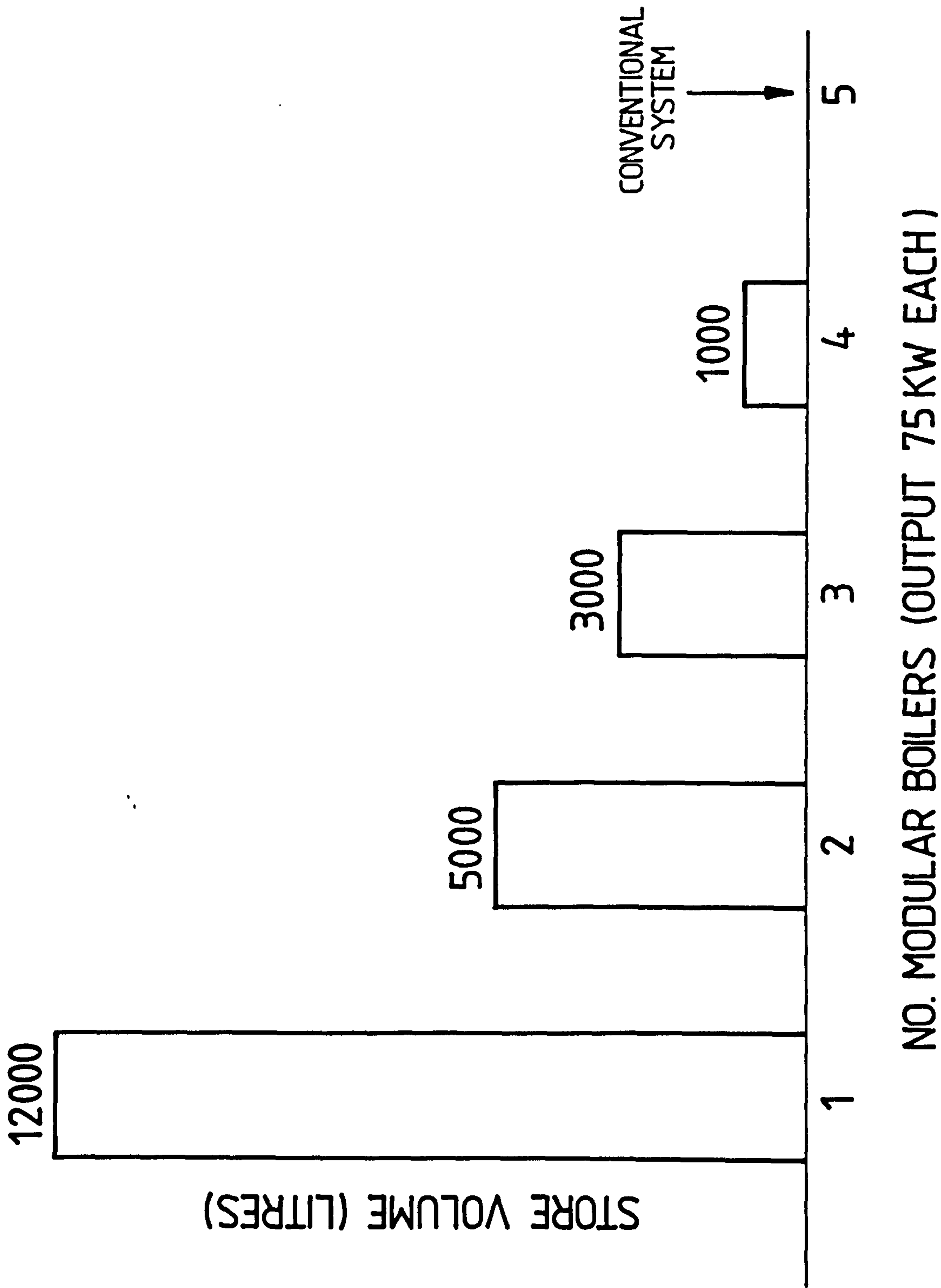


FIGURE 6.21 THE FOUR BOILER-STORE COMBINATIONS AND THE CONVENTIONAL SYSTEM ASSUMED SUITABLY SIZED FOR THE BUILDING IN THE ECONOMIC APPRAISAL.

System	No. of boiler Modules	Cost of boilers* (£)	Store Volume (ltrs.)	Cost of Store** (£)	Total Materials Cost (£)	Seasonal Efficy. (%)	Seasonal Fuel Cost*** (£)
Conven- tional	5	3114	0	0	3114	64.8	4171
TES 1	4	2492	1000	475	2967	68.3	3958
TES 2	3	1868	3000	1125	2993	69.8	3873
TES 3	2	1246	5000	1720	2966	71.8	3765
TES 4	1	624	12000	3400	4024	72.1	3749

* Cost of boilers based on Hamworthy price list April, 1983.

** Cost of stores based on quotations from Doulton Industrial Products Ltd., August, 1984.

*** Based on a gas price of 1.201p/kWh (i.e. 35.2p/therm)

Table 6.1 Costs for equivalent gas-fired systems

6.6.4 The Economics of Heat Storage Under The Four Situations

(a) New installations

New purpose-designed installations incorporating heat storage should not incur any extra plumbing costs compared with a conventional system. The cost of any extra pipework for the store is offset by there being less pipework for the smaller boiler plant capacity. It is therefore assumed that installation costs for the 5 alternative systems are equal. Table 6.2 gives the extra capital cost, annual running cost savings and the pay-back periods for the 4 storage options.

System	No. of boiler modules in storage system	Store Volume (ltrs.)	Extra capital cost (£)	Annual running cost saving (£/yr)	Pay-back period (yrs.)
TES 1	4	1000	-147	213	*
TES 2	3	3000	-121	298	*
TES 3	2	5000	-148	406	*
TES 4	1	12000	910	422	2.2

* Means that the storage system is both cheaper with respect to capital cost and achieves annual fuel savings.

Table 6.2 Economics of new installations

(b) Replacement of old boilers

In this situation, there are different installation costs for each of the 5 alternative systems, because they have to be integrated into existing pipework. It is assumed that the cost of installing a store is given by $\pounds(3000 + V/10)$, where V is the store volume in litres; and that the cost of plumbing-in new boilers, whether for a conventional or storage system, is $\pounds200 N$, where N is the number of boiler modules. These costs include an allowance for modifying the control system to operate with a store. Table 6.3 gives the extra initial cost, annual running cost saving and pay-back period for the 4 storage options.

System	No. of boiler modules in storage system	Store volume (ltrs.)	Extra installation cost (£)	Extra initial cost (£)	Annual running cost saving (£/yr)	Pay-back period (yrs.)
TES 1	4	1000	2900	2753	213	12.9
TES 2	3	3000	2900	2779	298	9.3
TES 3	2	5000	2900	2752	406	6.8
TES 4	1	12000	3400	4310	422	10.2

Table 6.3 Economics of boiler replacement

(c) Adding storage to meet an increased heat load (e.g. system extension)

To assess this circumstance it is conjectured that there are 4 different buildings which have all been extended so that their new heat loads are each 225,000 kWh. To meet this extra demand, the option is either to add extra boiler modules or to install a store. The installation cost of a store is again given by $\pounds(3000 + V/10)$. The installation cost of extra boiler modules is given by $\pounds500 N$, more than for case (b) since there is no existing plumbing for extra boilers. Table 6.4 gives the extra initial cost, annual running cost saving and pay-back period for the 4 different installations.

System	Existing No. of boilers	Extra No. of new boilers for conventional system	Store volume (ltrs.)	Extra installation cost of store option (£)	Extra initial cost (£)	Annual running cost saving (£/yr)	Pay-back period (yrs.)
Bldg.1	4	1	1000	2600	2451	213	11.5
Bldg.2	3	2	3000	2300	2179	298	7.3
Bldg.3	2	3	5000	2000	1852	406	4.6
Bldg.4	1	4	12000	2200	3108	422	7.4

Table 6.4 Economics of meeting an increased heat demand

(d) Retrofitting a store to improve the boiler performance

In this case, the performance of the existing, conventional boilers is to be improved by adding one of the 4 stores, and switching-off surplus boilers. The installation cost of a store is again given by $\pounds(3000 + V/10)$. The retrofit cost, annual running cost saving and pay-back period for the 4 alternative storage sizes are given in Table 6.5.

System	Store volume (ltrs.)	No. of boilers switched off	Installation cost of store (£)	Retrofit cost (£)	Annual running cost saving (£/yr)	Pay-back period (yrs.)
TES 1	1000	1	3100	3575	213	16.8
TES 2	3000	2	3300	4425	298	14.8
TES 3	5000	3	3500	5220	406	12.9
TES 4	12000	4	4200	7600	422	18.0

Table 6.5 Economics of adding a store to a set of existing boilers

6.6.5 Economic Conclusions

1. Installation costs dominate

Modifications or additions to existing pipework are expensive in Institutional buildings, both due to the commercial nature of the market and to the fact that pipe sizes are often large. It is only in new installations where storage incurs no extra installation costs, and it is then an extremely attractive option.

2. Reducing the required boiler plant capacity is the key economic benefit of storage

In the above analysis of applications (b), (c) and (d), it is significant that in each case the shortest pay-back period occurs for a fairly large store, i.e. 5000 litres, which is capable of replacing some 60% of the original boiler-plant capacity. The main reasons for this situation are :-

- O installation costs are fairly insensitive to store volume
- O large stores benefit from very significant economies of scale

- O the cost of modular boiler plant does not exhibit economies of scale
- O running-cost savings flatten out as store volumes become very large; i.e. in this case, for store volumes greater than 5000 litres.

3. There is likely to be an economic optimal size of store

Continuing the above argument, larger stores can replace more boiler plant, but there is a steep rise in required store volume as boiler plant becomes very small. This factor combined with the flattening out of running cost savings, means that there is an optimal size of store, which may well be fairly large.

6.7 GENERAL CONCLUSIONS ON THIS APPLICATION OF HEAT STORAGE

1. Using a heat store in tandem with a conventional boiler will inevitably raise the overall annual boiler efficiency. Simultaneously the store enables a lower output boiler to be used for providing the same heat demand.
2. The magnitudes of the annual energy savings and boiler size reductions attainable clearly depend on the system design and the individual circumstances. For example, in any particular building there is a wide choice of boiler size and store volume that could satisfy the heat demand, each combination having its own capital and running costs. However this study indicates that, in comparison with a well-controlled, efficient, conventional system, energy savings of about 8% are feasible, due to an enhanced system efficiency, whilst savings of a further 5% are likely due to a faster building pre-heat. At the same time, the study predicts that boiler plant capacity can be reduced by around 40%.
3. A costing exercise, whilst unavoidably approximate, highlights new installations as the most financially attractive: the heat storage option is both cheaper to install and more efficient to run.

4. Adding a store to an existing system, whether or not the boiler is changed, involves extra plumbing costs, which, in economic terms, make pay-back periods unacceptably long, e.g. 7 - 18 years.

CHAPTER SEVENCONCLUSIONS7.1 EXPERIMENTAL STUDIES

Evaluations have been made of the behaviour of a 3m^3 hot water store under varied temperature step input conditions using a micro-processor-controlled test facility, with the following results:

- (i) The store was observed to contain two distinct regions: a fully-mixed region adjacent to the inlet port, and a region of smooth 'plug-flow' in the remainder of the store. This distinctive response occurred whether the inlet/outlet ports were horizontal, vertical or distributors.
- (ii) There was a close correlation with each port configuration between the initial depth of the mixed region and the parameter

$$\frac{\dot{m}}{\rho} r^{-3/2} (g \beta \Delta T)^{-1/2}$$

for all conditions excluding very low inlet flowrates.

- (iii) The performance of the water store under step input conditions was defined by the initial depth of the mixed region, and therefore was also closely correlated with this parameter.
- (iv) The horizontal ports were found to produce less mixing than the vertical ports under the same conditions, but dead space beyond the horizontal outlet port radically impaired the overall performance. The depths of the mixed region produced by the distributors were smaller than those with

vertical ports, but greater than those with horizontal ports. However, the problem of deadspace, incurred by the horizontal ports, determined that the performance of the distributors was superior to that of both plain ports under the same conditions.

7.2 COMPUTER SIMULATIONS OF THE STORE

A one-dimensional numerical computer model of the store was developed, with reference to the observed behaviour and derived correlations. The conclusions drawn from the exercise are:

- (i) The relatively uncomplicated model accurately simulated the thermal behaviour of the store under step input conditions, except at very low inlet flowrates. The error in predicting the store performance coefficient and the outlet temperature was of the order of 3%, whilst the internal temperature distribution during a test could be estimated to within 10%.
- (ii) The influence of vertical heat conduction on the store performance was shown to be negligible when a charge/discharge cycle takes less than one day.

7.3 COMPUTER SIMULATION OF A SYSTEM INCORPORATING STORAGE

Computer simulation of a storage system (using the store model) and of the alternative conventional system was used to evaluate one promising application of hot water storage, viz: gas-fired central-heating systems. The conclusions of this study were that:

- (i) The storage system was about 8% more efficient owing to a dramatic cut in the number of boiler on/off cycles.
- (ii) Additional energy savings of around 5% seemed likely due to a faster building pre-heat.

- (iii) Boiler plant capacity could be reduced by around 40%.
- (iv) Following economic analysis, storage would appear to represent a very attractive financial option in new installations, but would be more difficult to justify as a retrofit measure, owing to the extra installation costs involved.

7.4

FURTHER WORK

- (i) A logical and valuable progression from the present study would be to initiate the experimental validation of the computer programs simulating storage and conventional central heating systems.
- (ii) Further detailed study with low inlet flowrates could more precisely define the range of conditions for maximum store performance, whilst investigating the predicament whereby peak results are presently obtained when the store is charged or discharged at a relatively slow rate.
- (iii) In this work, the depth of penetration of an inlet jet was shown by a theoretical argument to correlate with a parameter defined by the inlet flow conditions. By experiment, the initial depth of a fully-mixed region in the tank was shown to correlate with the same parameter. It can be induced that these two depths were essentially identical, in the circumstances considered.

However, if the ratio of the tank diameter to the nozzle diameter were much larger, it is likely that the initial depth of the fully-mixed region would no longer correspond to the jet's depth of penetration (which should still be defined by the theoretical correlation). In this situation, the one-dimensional numerical computer model would be less realistic as it uses the theoretical correlation for the depth of penetration to calculate the initial depth of the fully-mixed region. Similarly, the accuracy of the performance coefficient correlations would suffer.

Further experimental work could usefully be carried out therefore on a tank with a transparent wall (care being taken to ensure a low store heat loss rate). A flow-visualisation technique could then be employed to observe both the depth of penetration of a jet and the development of the fully-mixed zone, under a full range of inlet flow conditions. The range of applicability of the computer model and the performance coefficient correlations could be determined by steadily reducing the nozzle diameter.

REFERENCES

- 1.1 MEADOWS, D.H., MEADOWS, D.L., RANDERS, J. and BEHRENS, W.W., "The Limits to Growth". Published by Potomac Associates, Washington D.C., U.S.A., 1972.
- 1.2 DEPARTMENT OF ENERGY, "Energy Technologies for the United Kingdom". Energy Paper Number 39, H.M.S.O., 1979.
- 1.3 CHAPMAN, P., "Fuel's Paradise". Published by Penguin Books Ltd., England, 1975.
- 1.4 LEACH, G., "A low Energy Strategy for the United Kingdom". Published by Science Reviews Ltd., London, 1979.
- 1.5 THE INSTITUTE OF FUEL, "Energy for the Future". The Institute of Fuel, 1972.
- 1.6 DEPARTMENT OF ENERGY, "Digest of United Kingdom Energy Statistics". H.M.S.O., 1985.
- 1.7 DEPARTMENT OF ENERGY, "Low Energy Futures Assessed and Compared". RE NEWS Issue 7, Dept. of Energy, October 1982.
- 1.8 LEACH, G., "The Energy Costs of Food Production". In 'The Man-Food Equation', ed. A.Bourne, Academic Press, England, 1974.
- 1.9 BEIJENDORFF, A.J., "Energy Efficiency". Shell International Petroleum Co., London, 1979.
- 1.10 "B.P. STATISTICAL REVIEW OF WORLD ENERGY, 1982". The British Petroleum Company, London, May 1983.
- 1.11 LOVINS, A.B., "World Energy Strategies". Friends of the Earth Ltd., London, 1975.

- 1.12 **CHESTER, P.F., "Coal and our Atmosphere". 1985 Robens Coal Science Lecture, The Royal Institution, London, October 1985.**

- 1.13 **MINISTRY OF INDUSTRY, SWEDEN, "Guidelines for the Swedish Energy Policy". Ministry of Industry, Stockholm, Sweden, February 1985.**

- 1.14 **FELLS, I., "Energy Provision : Policies and Technologies". Energy World Number 118, Institute of Energy, London, October 1984.**

- 1.15 **RUSSELL, F.M., "The Techniques, Magnitudes and Opportunities for Development of Energy Storage (U.K.)". Science and Engineering Research Council, England, 1981.**

- 1.16 **JONES, B.W., "A Solar Power System (Prometheus) to Provide 100% of Low-grade Heat Needs". Applied Energy 6, pp 329-346, 1980.**

- 1.17 **HAWKER SIDDELEY POWER ENGINEERING LTD., "Method and Means for Heat Storage". Patent Specification, June 1983.**

- 1.18 **GOLDSTERN, W., "Thermal Energy Storage in Industry and Power Stations". International Conference on Energy Storage, organised by BHRA at Brighton, U.K., April 1981.**

- 1.19 **TELKES, M., "Thermal Storage for Solar Heating and Cooling". Proceedings of Workshop on Solar Energy Storage for Heating and Cooling of Buildings, Charlottesville, U.S.A., April 1975.**

- 1.20 **PAGE, J.K.R. and SWAYNE, R.E.H., "Phase Change Thermal Storage for Solar Applications". International Conference on Energy Storage, organised by BHRA at Brighton, U.K., April 1981.**

- 1.21 MARQUETTE, C.J., "Ice Storage Systems that Cost Less Than Conventional Systems - A Breakthrough for Building Owners". 3rd International Conference on Energy Storage for Building Heating and Cooling, Toronto, Canada, September 1985.
- 1.22 FRANCIS, C.E., "An Annual Storage Ice System for Summer Cooling". Idem, Toronto, Canada, September 1985.
- 1.23 BENSON, D.K., CHRISTENSEN, C.B., BURROWS, R.W., and SHINTON, Y.D., "New Phase Change Thermal Energy Storage Materials for Buildings". Idem, Toronto, Canada, September 1985.
- 1.24 BAKKEN, K., "System Tepidus, High Capacity Thermochemical Storage/Heat Pump". International Conference on Energy Storage, organised by BHRA at Brighton, U.K., April 1981.
- 1.25 MAIER-LAXHUBER, P., ROTHMEYER, M. and ALEFELD, G., "Zeolite Heat Pump and Heat Storage". International Conference on Energy Storage, organised by BHRA at Stratford, England, May 1983.
- 1.26 BINDON, F.J.L., "Central Electricity Generating Board. Summary of Report and Accounts 1984/85". Energy World Number 129, Institute of Energy, London, October 1985.
- 1.27 ENERGY EFFICIENCY OFFICE, "Pattern of Energy Use in the U.K., 1980", H.M.S.O., 1984.
- 1.28 COHEN, R.R., TANTON, D.M., HALE, R.A., and PROBERT, S.D., "An Assessment of the Benefits of Heat Storage in Central Heating Systems". Final Report to ETSU, Cranfield, January 1985.
- 1.29 SODERSTROM, M.F. and JOHANSSON, T.R., "Use of Energy Storage in Industry - potential in Sweden". International Conference on Energy Storage, organised by BHRA at Stratford, England, May 1983.

- 1.30 **ENERSTOCK 85**, 3rd International Conference on Energy Storage for Building Heating and Cooling, Toronto, Canada, September 1985.
- 1.31 **HYDEN, H. and KELNER, J.**, "The Suncourt Project". Idem., Toronto, Canada, September 1985.
- 1.32 **HULTMARK, G.**, "Sunclay - Operation and Optimization". Idem. Toronto, Canada, September 1985.
- 2.1 **GUPTA, S.K. and JALURIA, Y.**, "An Experimental and Analytical Study of Thermal Stratification in an Enclosed Water Region due to Thermal Energy Discharge". Energy Conversion and Management, Vol. 22, No. 1, pp. 63-70, 1982.
- 2.2 **TURNER, J.S.**, "Jets and Plumes with Negative or Reversing Buoyancy". J. Fluid Mech., Vol. 26, part 4, pp. 779-792, Great Britain, 1966.
- 2.3 **SCHWIND, R.G. and VLIET, G.C.**, "Observations and Interpretations of Natural Convection and Stratification in Vessels". Proc. of 1964 Heat Transfer and Fluid Mechanics Institute, Berkeley, California, U.S.A., June 1964.
- 2.4 **MILLER, C.W.**, "The Effect of a Conducting Wall on a Stratified Fluid in a Cylinder". 12th AIAA Thermophysics Conference, Albuquerque, New Mexico, U.S.A., June 1977.
- 2.5 **SHERMAN, C., WOOD, B.D. and MASON, J.**, "Effect of Vertical Wall Conductance on Temperature Relaxation in Thermally Stratified Liquid Thermal Storage Tanks". Proc. of ISES Conference, Atlanta, Georgia, U.S.A., May 1979.
- 2.6 **VISKANTA, R. and HALE, N.W.**, "Free Convection Circulation in a Thermal Energy Storage System Driven by Heat Loss from the Walls". Proc. of 14th Southeastern Seminar on Thermal Sciences, North Carolina State University, U.S.A., April 1978.

- 2.7 LEYERS, H.J., SCHOLZ, F. and THOLEN, A., "Analytical and Experimental Determination of the Temperature Distribution in Stratified Hot Water Stores". International Seminar on Heat Transfer in Buildings, Dubrovnik, Yugoslavia, 1977.
- 2.8 JALURIA, Y. and GUPTA, S.K., "Decay of Thermal Stratification in a Water Body for Solar Energy Storage". Solar Energy, Vol. 28, No. 2, pp. 137-143, Great Britain, 1982.
- 2.9 VAN GALEN, E. and VAN DEN BRINK, G.J., "Recommendations for European Solar Storage Test Methods". Final Report, No. 203. 212, TNO, Delft, The Netherlands, November 1984.
- 2.10 NAKAJIMA, Y., "Studies on the Weighting Functions of Thermal Storage Water Tanks, Part I". Transactions of the Japanese Architectural Society, Vol. 199, pp. 37-47 (Japanese), September, 1972.
- 2.11 MATSUDAIRA, H. and SAKAKURA, Y., "Analysis on the Thermal Capacity of Heat Storage Water Tanks, Part I". Journal of the Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, Vol. 46, No. 4, pp. 11-17 (Japanese), April 1972.
- 2.12 HILL, J.E., STREED, E.R., KELLY, G.E., GEIST, J.C. and KUSUDA, T., "Development of Proposed Standards for Testing Solar Collectors and Thermal Storage Devices". NBS Technical Note 899, Washington, DC, U.S.A., February 1976.
- 2.13 HILL, J.E., KELLY, G.E. and PEAVY, B.A., "A Method of Testing for Rating Thermal Storage Devices Based on Thermal Performance". Solar Energy, Vol. 19, pp. 721-732, Great Britain, 1977.
- 2.14 ASHRAE STANDARD 94-77, "Methods of Testing Thermal Storage Devices Based on Thermal Performance". ASHRAE, New York, February 1977.

- 2.15 HUNT, B.J., RICHMYER, T.E., HILL, J.E. and FRANKLIN, E.A.,
"Testing of Water Tanks for Thermal Storage According to
ASHRAE Standard 94-77". NBS Report No. NBSIR-78-1548,
Washington, DC, U.S.A., October 1978.

- 2.16 MARSHALL, R.H., "Experimental Experience with ASHRAE/NBS
Procedures for Testing a Phase Change Thermal Storage
Device". 1st BHRA International Conference on Energy
Storage, Brighton, U.K., April 1981.

- 2.17 LAVAN, Z. and THOMPSON, J., "Experimental Study of Thermally
Stratified Hot Water Storage Tanks". Solar Energy, Vol. 19,
pp. 519-524, Great Britain, 1977.

- 2.18 SLIWINSKI, B.J., MECH, A.R. and SHIH, T.S., "Stratification
in Thermal Storage During Charging". 6th International Heat
Transfer Conference, Toronto, 1978.

- 2.19 SLIWINSKI, B.J., "Investigation of Methods to Predict
Thermal Stratification and its Effect on Solar Energy System
Performance". Construction Engineering Laboratory, Special
Report E-160, Illinois, U.S.A., May 1980.

- 2.20 WOOD, R.J., AL-MUSLAHI, S.M., O'CALLAGHAN, P.W. and PROBERT,
S.D., "Thermally Stratified Hot Water Storage Systems".
Applied Energy 9 pp. 231-242, 1981.

- 2.21 DAVIS, E.S. and BARTERA, R., "Stratification in Solar Water
Heater Storage Tanks". Proc of The Workshop on Solar Energy
Storage Subsystems for the Heating and Cooling of Buildings,
Charlottesville, VA, U.S.A., 1975.

- 2.22 VAN KOPPEN, C.W.J., FISCHER, L.S. and DIJKMANS, A.,
"Stratification Effects in the Short and Long Term Storage
of Solar Heat". International Solar Energy Conference, New
Delhi, India, 1978.

- 2.23 LOEHRKE, R.I., GARI, H.N., SHARP, M.K. and HABERSTROH, R.D., "A Passive Technique for Enhancing Thermal Stratification in Liquid Storage Tanks". ASME paper 78-HT-50, 2nd AIAA/ASME Thermophysics and Heat Transfer Conference, May 1978.
- 2.24 LOEHRKE, R.I., HOLZER, J.C., GARI, H.N. and SHARP, M.K., "Stratification Enhancement in Liquid Thermal Storage Tanks". Journal of Energy, Vol. 3, No. 3, pp, 129-130, May-June 1979.
- 2.25 GARI, H.N., LOEHRKE, R.I. and HOLZER, J.C., "Performance of an Inlet Manifold for a Stratified Storage Tank". ASME paper 79-HT-67, 18th ASME/AICHE National Heat Transfer Conference, San Diego, California, U.S.A., August 1979.
- 2.26 GARI, H.N. and LOEHRKE, R.I., "A Controlled Buoyant Jet for Enhancing Stratification in a Liquid Storage Tank". Journal of Fluids Engineering, Vol. 104, pp. 475-481, December 1982.
- 2.27 WU, S.T., "The Experimental and Numerical Studies of Liquid Storage Tank Thermal Stratification for a Solar Energy System". Semi-Annual Progress Reports, EG-77-S-02-4479, University of Alabama, U.S.A., September 1977 - February 1978 and March 1978 - August 1978.
- 2.28 HAN, S.M., WU, S.T. and CHRISTENSEN, D.L. "The Effects of Thermal Stratification in a Water Storage Tank for the Performance of a Solar Hot Water System". Proc. of 14th Southeastern Seminar on Thermal Sciences, North Carolina State University, U.S.A., April 1978.
- 2.29 PHILLIPS, W.F. and DAVE, R.N., "Effects of Stratification on the Performance of Liquid-Based Solar Heating Systems". Solar Energy, Vol. 29, No. 2, pp. 111-120, Great Britain, 1982.
- 2.30 BOWMAN, N.T., REDFERNE, W.B. and ELDESSOUKY, E., "Stratified Solar Storage for Use in Domestic Scale Systems". Sun at Work in Britain, No. 12/13, pp 39-42, 1981.

- 2.31 MINE, T., "Determination of Heat Transfer Characteristics of Immersed Coils". MSc Thesis, School of Mechanical Engineering, Cranfield, 1984.
- 2.32 HUGHES, D.G. and THRING, J.B., "Water Based Heat Stores for Active Solar Domestic Space and Water Heating Systems". 2nd BHRA International Conference on Energy Storage, Stratford, England, May 1983.
- 2.33 SHA, W.T., LIN, E.I.H., SCHMITT, R.C., LIU, K.V., HULL, J.R., ORAS, J.J. and DOMANUS, H.M., "COMMIX-SA-1: A Three-Dimensional Thermohydrodynamic Computer Program for Solar Applications". Argonne National Laboratory Report No. ANL-80-8, U.S.A., November 1980.
- 2.34 LIN, E.I.H. and SHA, W.T., "Effects of Baffles on Thermal Stratification in Thermocline Storage Tanks". Proc of ISES Conference, Atlanta, Georgia, U.S.A., May 1979.
- 2.35 CABELLI, A., "Storage Tanks - A Numerical Experiment". Solar Energy, Vol. 19, pp. 45-54, Great Britain, 1977.
- 2.36 MATSUDAIRA, H., TANAKA, Y. and TAKAHASHI, H., "Dynamic Characteristics of a Heat Storage Water Tank". Numazu Technical College, Shizuska-Ken, Japan, 1977.
- 2.37 DUFFIE, J.A. and BECKMAN, W.A., "Solar Energy Thermal Processes". Published by John Wiley & Sons, New York, 1974.
- 2.38 MARSHALL, R.H., "Modelling of Thermal Storage for Solar Heating Systems". Proc. of EC Contractors' Meeting, Athens, November 1981, Published in Solar Energy Applications to Dwellings Series A Vol. 1, D. Reidel, Holland, 1982.
- 2.39 SHARP, M.K., "Thermal Stratification in Liquid Sensible Heat Storage". M.S. Thesis, Mechanical Engineering Dept., Colorado State University, U.S.A., 1978.

- 2.40 KUHN, J.K., VON FUCHS, G.F. and ZOB, A.P., "Developing and Upgrading of Solar System Thermal Energy Storage Simulation Models". Final Report No. DOE/CS/34482-5, Boeing Computer Services Company, Seattle, Washington, U.S.A., August 1980.
- 2.41 WUESTSLING, M.D., "Investigation of Promising Control Alternatives for Solar Water Heating Systems". M.S. Thesis, Mechanical Engineering Dept., University of Wisconsin - Madison, U.S.A., 1983.
- 2.42 SHARP, M.K. and LOEHRKE, R.I., "Stratified Thermal Storage in Residential Solar Energy Applications". Journal of Energy, Vol. 3, No. 2, pp. 106-113, March-April 1979.
- 2.43 VELTKAMP, W.B., "Thermal Stratification in Heat Storages". Proc of International TNO-Symposium on Thermal Storage of Solar Energy, Amsterdam, November 1980, Published by Martinus Nijhoff, The Hague, The Netherlands, 1981.
- 3.1 BAYLEY, F.J., OWEN, J.M. and TURNER, A.B., "Heat Transfer", P.163. Published by Nelson Ltd., London, 1972.
- 4.1 RAJARATNAM, N., "Turbulent Jets", P.36. Published by Elsevier Scientific Publishing Company, Amsterdam, The Netherlands, 1976.
- 4.2 BRADSHAW, P., "Turbulence", P.5. Published as "Topics in Applied Physics". Volume 12, by Springer-Verlag, Berlin, Germany, 1976.
- 4.3 CROOME-GALE, D.J. and ROBERTS, B.M., "Air Conditioning and Ventilation of Buildings". Pergamon Press, Oxford, 1975.
- 5.1 CHOW, C.Y., "An Introduction to Computational Fluid Mechanics". pp. 331-336. Seminole Publishing Company, Colorado, U.S.A., 1983.
- 6.1 Energy Efficiency Series No. 2, "Pattern of Energy Use in the U.K., 1980". Energy Efficiency Office, HMSO, 1984.

- 6.2 COHEN, R.R., WOOD, R.J. and PROBERT, S.D., "Thermal-Energy Storage in Fossil-fuel Fired Central-Heating Systems". 2nd BHRA International Conference on Energy Storage, Stratford, England, May 1983.

- 6.3 GIBSON, N.M., "Performance of the Watson House Designed Start-up Interval Controller". Report No. WH/T/R&D/78/83, British Gas Corporation, London, May 1978.

- 6.4 GIBSON, N.M. and BARNES, J.V., "Field Study of the Heating System Installed at Spa School, Bermondsey, SE1". Watson House Report No. WH/R/R&D/82/116, July 1982.

- 6.5 COHEN, R.R. and O'CALLAGHAN, P.W., "Field Trial of a Thermal-Energy Storage Regulated Central-Heating System in a Secondary School". Proposal reference 950/RRC/152, Cranfield Institute of Technology, January, 1984.

- 6.6 GIBSON, N.M., Private communication.

- 6.7 DAN, R.G., LOVELACE, D.E. and PAGE, M.W., "The Effect of Flue Dampers on Natural Ventilation Heat Losses from Boilers". H.A.C. May 1984.

- 6.8 BARNES, J.V., "Further Efficiency Measurements on a Hamworthy Modular-Boiler System". Watson House Report No. WH/T/R&D/76/123, August 1976.

- C.1 STREETER, V.L., "Fluid Mechanics". 4th Edition, P. 256. McGraw-Hill Book Company, New York, 1966.

APPENDIX C1

INSTRUMENTATION CALIBRATION DATA FOR P.R.T.'s

ACTUAL TEMP (°C)	TEMPERATURE READ BY THE COMPUTER (°C) P.R.T. NUMBER					
	1	2	3	6	7	8
0	1.2	0.2	1.4	1.4	1.2	1.1
10	10.9	10.1	11.4	11.2	11.1	11.0
20	20.7	20.1	21.1	21.1	21.0	20.8
30	30.6	30.1	30.9	30.9	30.7	30.7
40	40.3	39.8	40.5	40.6	40.3	40.3
50	50.2	49.9	50.4	50.5	50.3	50.3
60	59.9	59.8	60.0	60.3	59.9	60.0
70	69.8	69.8	70.1	70.3	69.8	70.0
80	79.7	79.8	79.8	80.1	79.6	79.8
90	89.4	89.7	89.6	89.9	89.4	89.6
100	99.2	99.7	99.4	99.8	99.2	99.5

APPENDIX C.2CALIBRATION DATA AND CURVES FOR FLOWMETERS

Flowmeter No. 1	
Pulses/sec P	Litres/sec F
118.99	2.284
79.53	1.542
58.12	1.15
38.12	0.768
18.1	0.378
7.76	0.17
Best fit straight line: $F = 0.019P + 0.0356$	

Flowmeter No. 2	
Pulses/sec P	Litres/sec F
69.9	0.408
56.6	0.332
41.5	0.246
28.2	0.1695
14.1	0.0897
6.3	0.0407
Best fit straight line: $F = 0.00575P + 0.0068$	

Flowmeter No. 3	
Pulses/sec P	Litres/sec F
120.89	2.278
79.84	1.519
59.05	1.141
38.72	0.767
17.8	0.372
7.68	0.181
Best fit straight line: $F = 0.0185P + 0.0441$	

Flowmeter No. 4	
Pulses/sec P	Litres/sec F
69.68	0.4035
54.9	0.3203
42.13	0.2488
27.27	0.1643
14.01	0.0887
6.81	0.0415
Best fit straight line: $F = 0.00572P + 0.0063$	

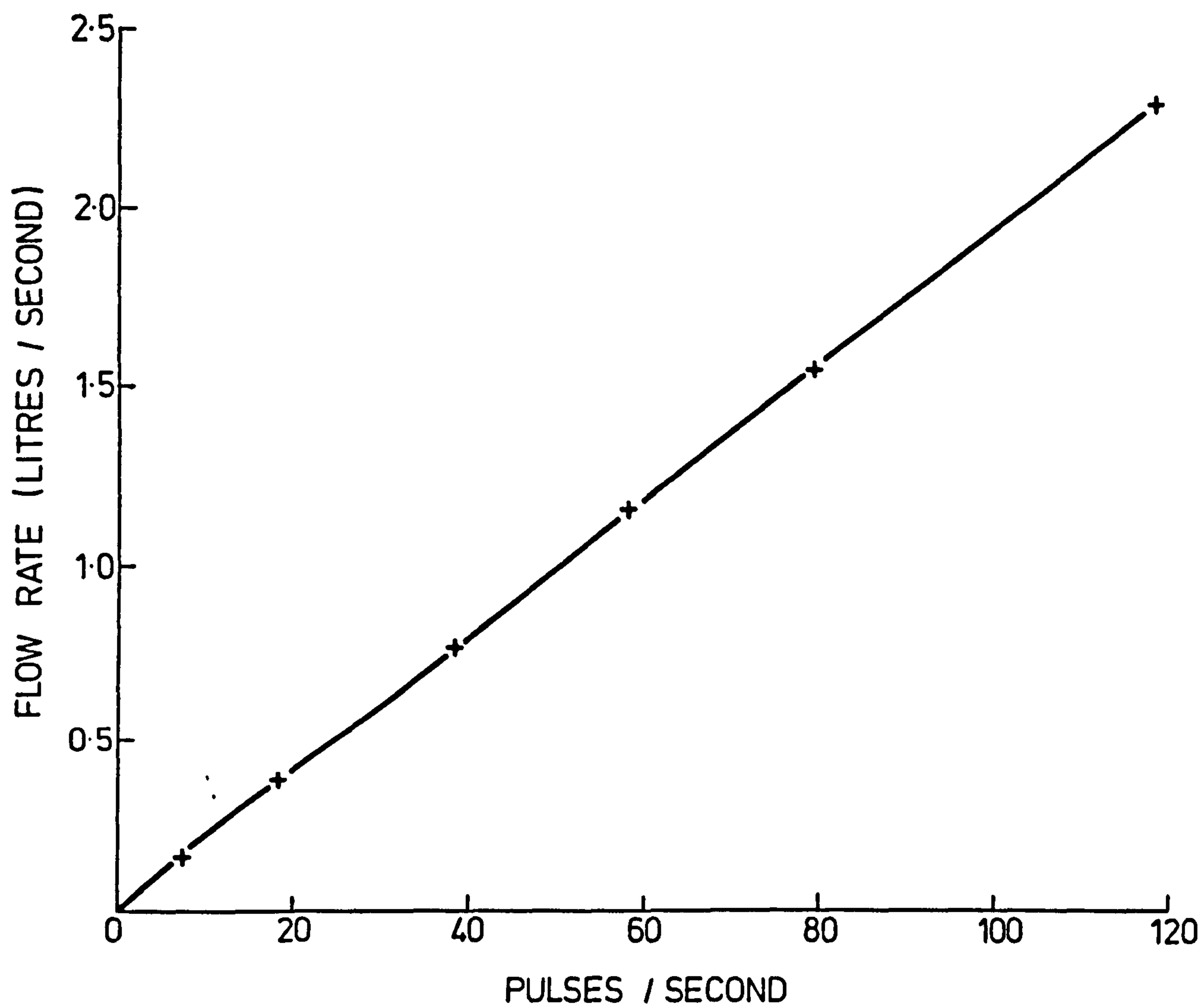


FIGURE C.1 CALIBRATION CURVE FOR FLOWMETER NO. 1

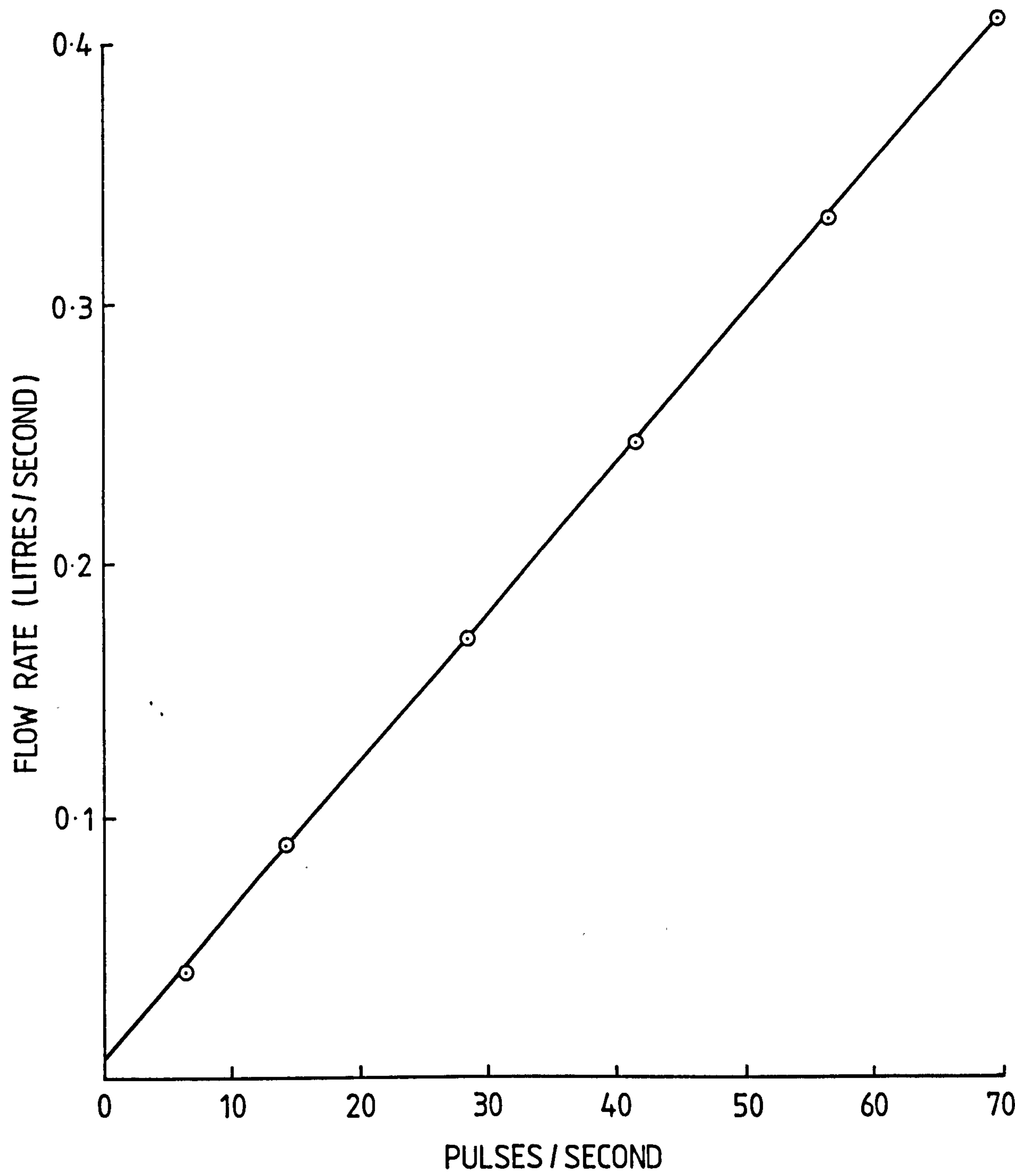


FIGURE C.2. CALIBRATION CURVE FOR FLOWMETER NO. 2.

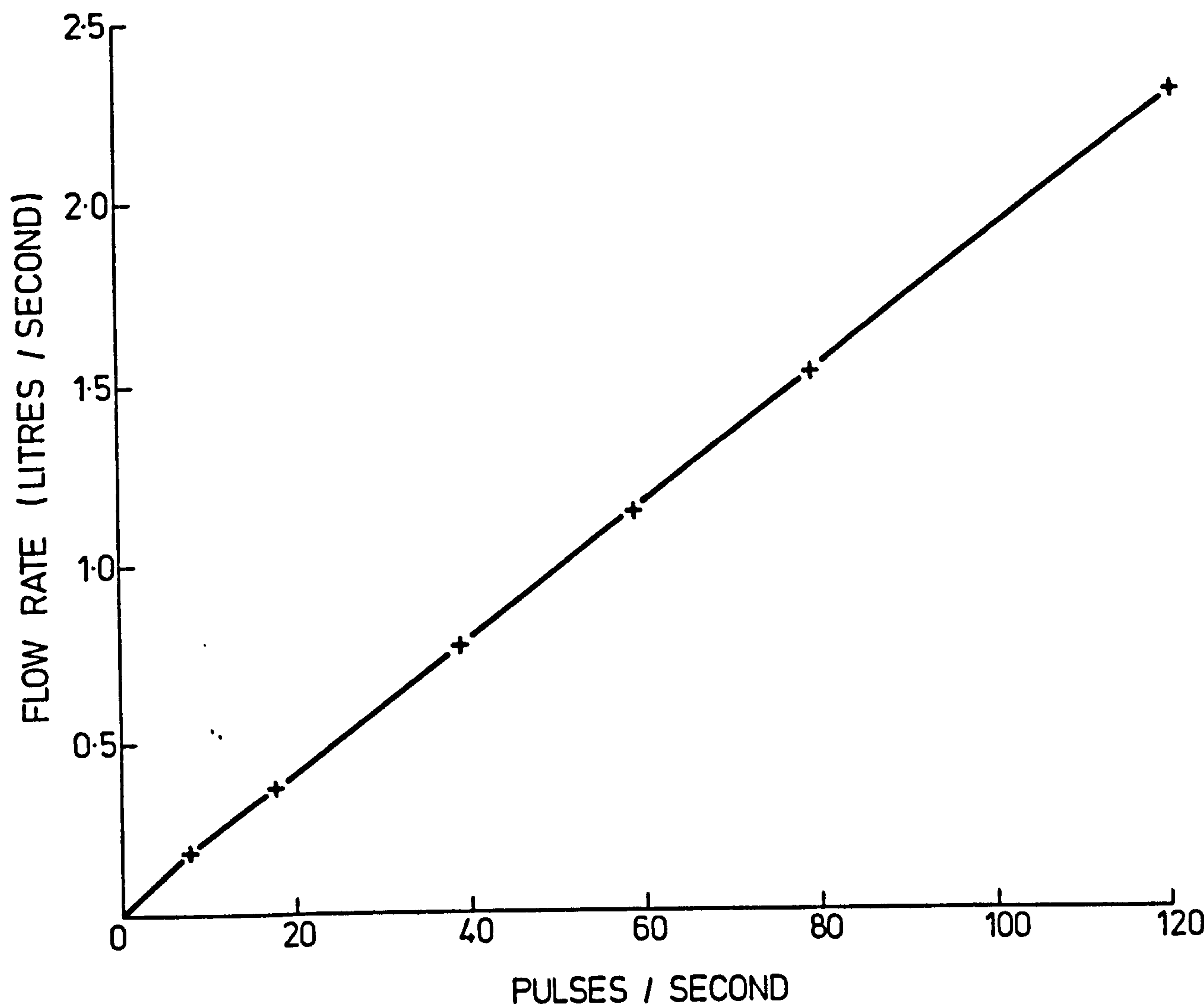


FIGURE C.3 CALIBRATION CURVE FOR FLOWMETER NO. 3

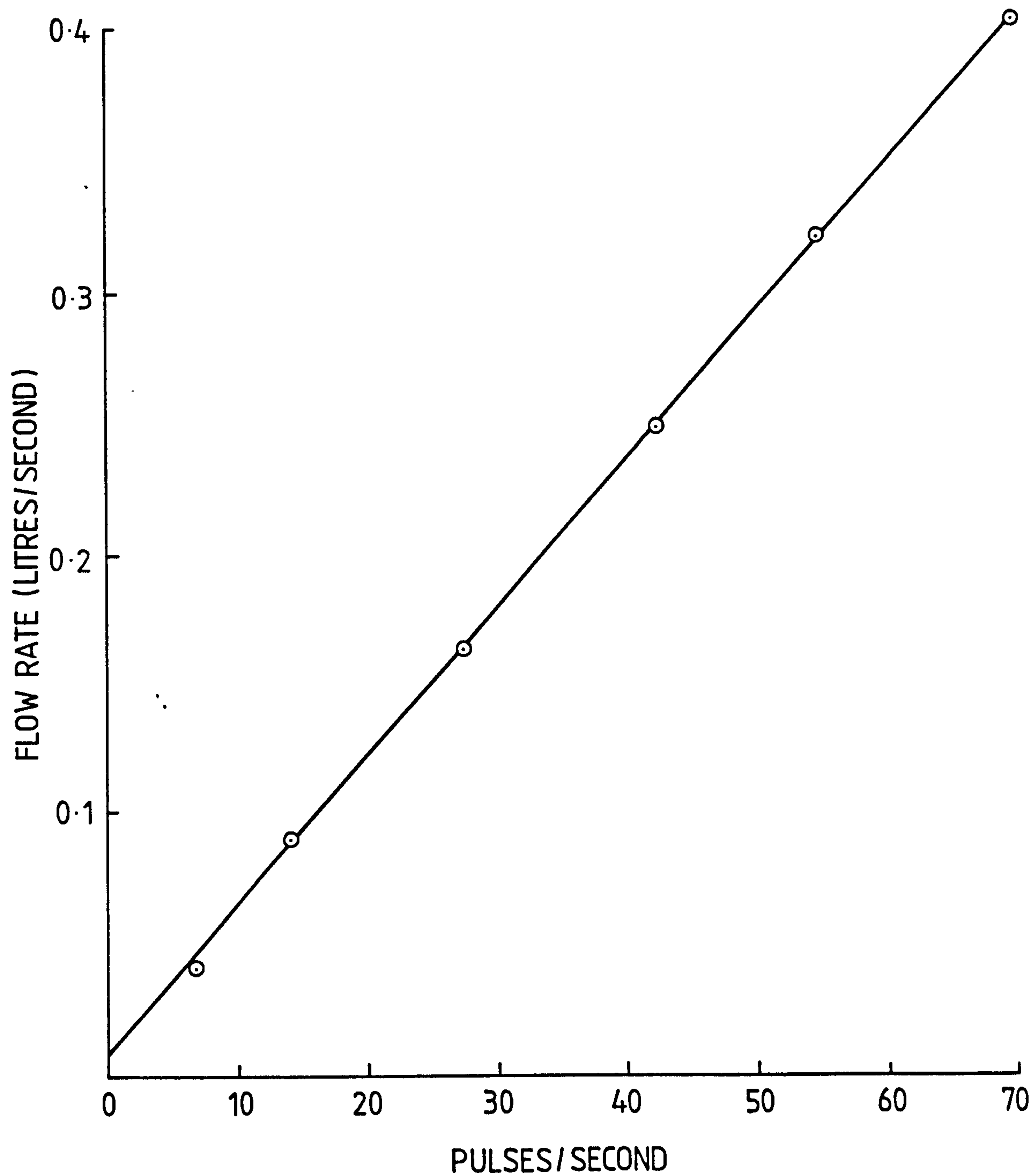
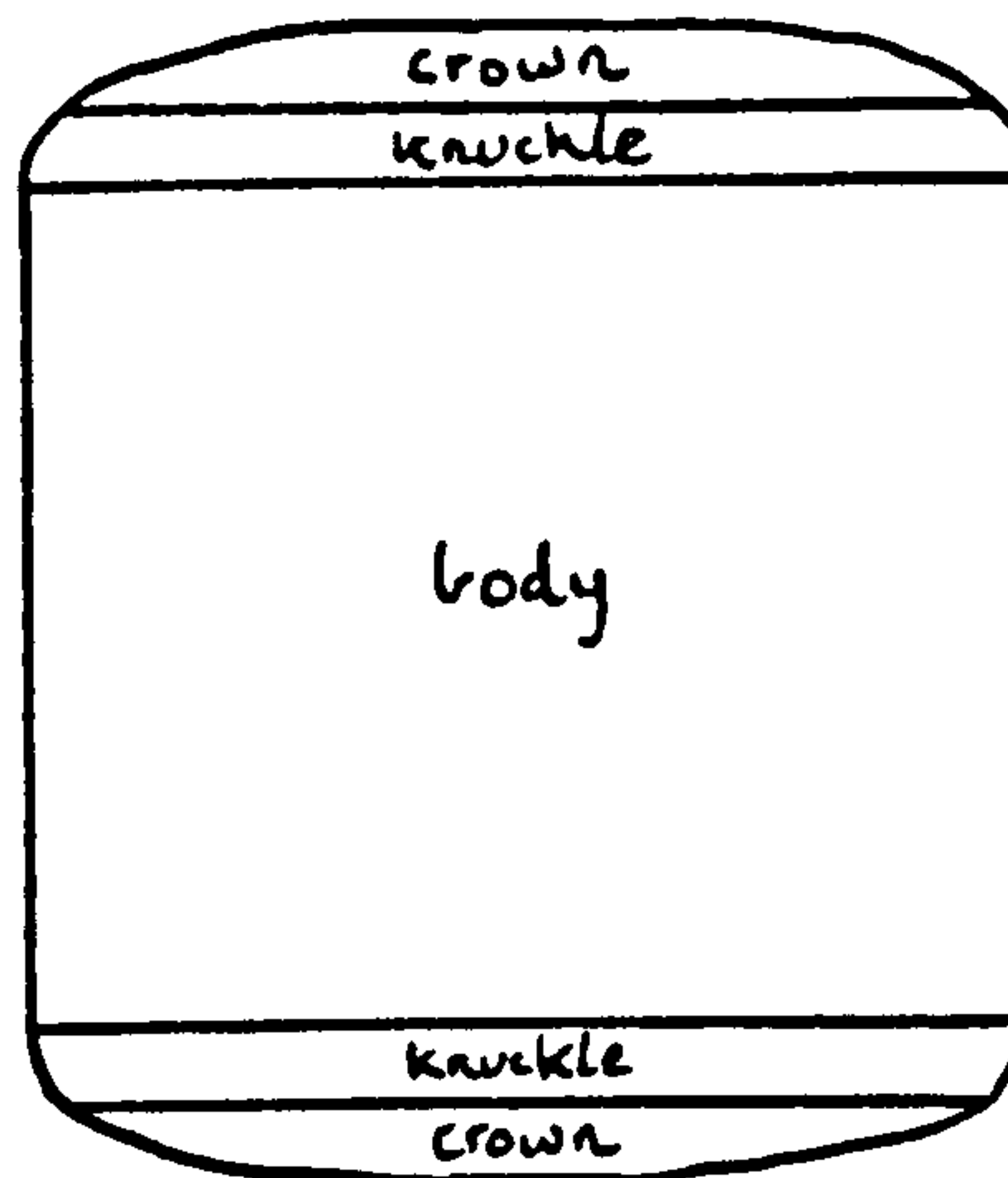


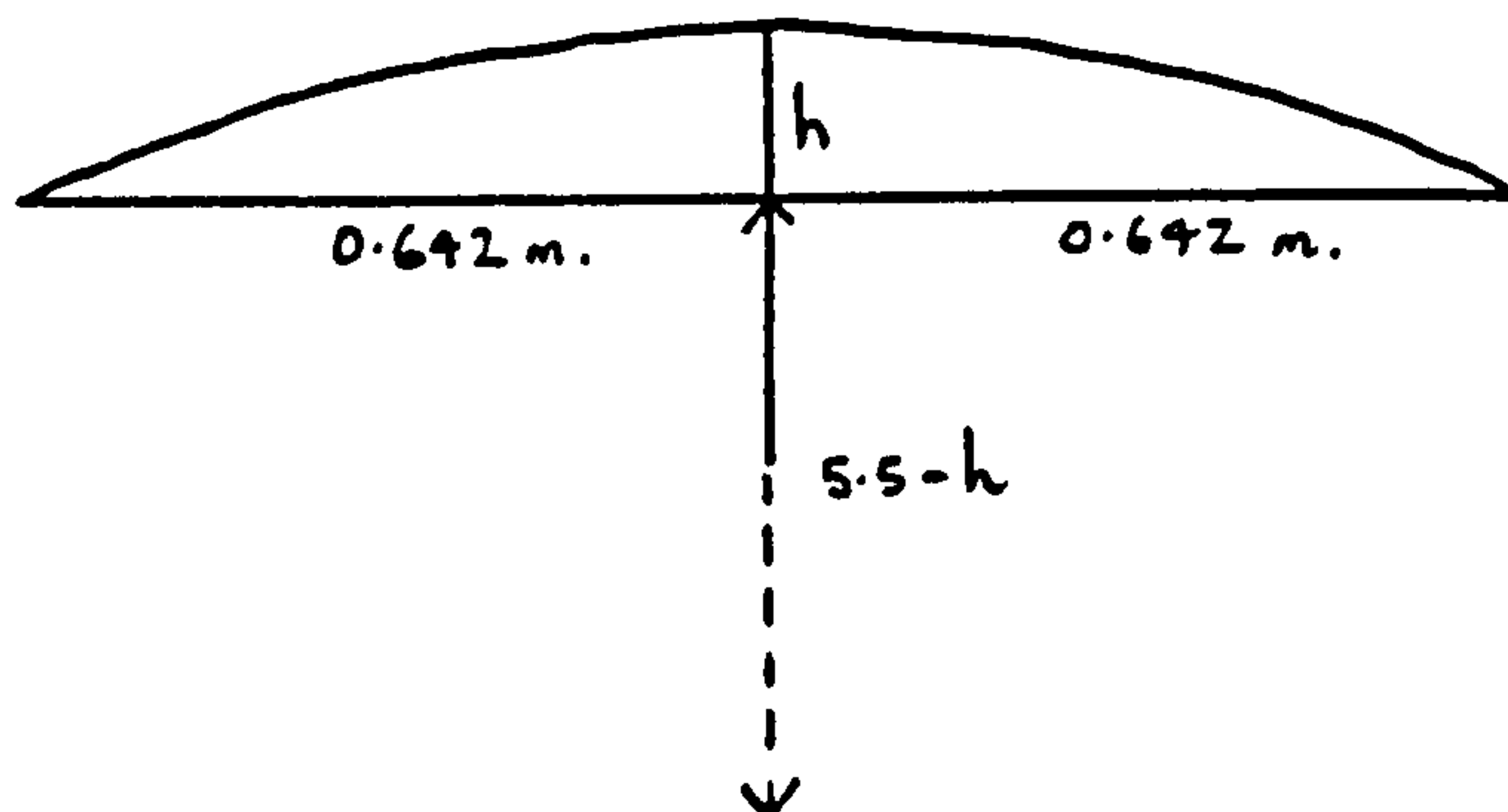
FIGURE C. 4 CALIBRATION CURVE FOR FLOWMETER NO. 4.

APPENDIX C.3CALCULATION OF VOLUME OF WATER STORAGE VESSEL

The vessel can be considered to be made up from 5 sections : 2 crowns, 2 knuckles and a body :

Volume of crown

The crown is a segment of a sphere of radius 2.75m, and its width is 1.284m :

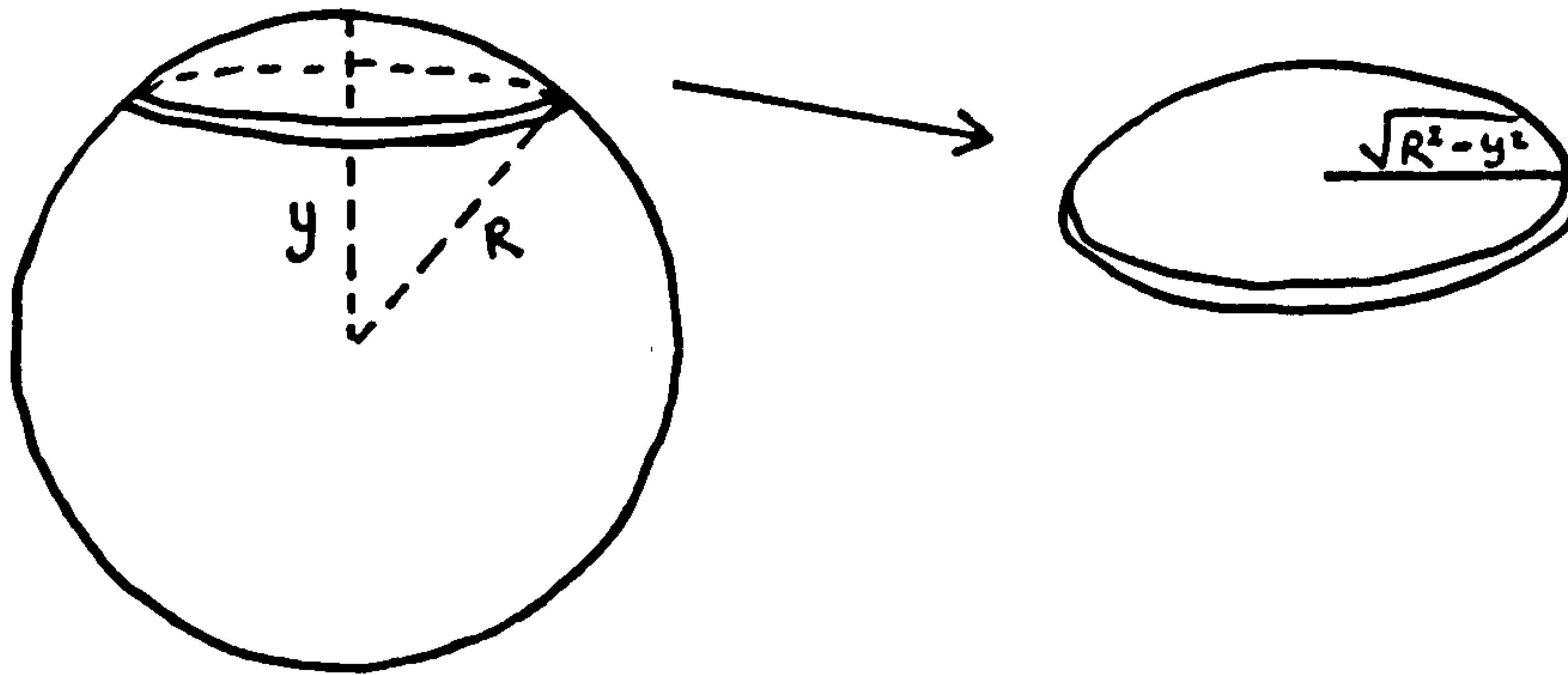


Let the height of the segment be h metres

$$\begin{aligned} \text{Then } h(5.5 - h) &= .642^2 \\ h^2 - 5.5h + 0.412 &= 0 \end{aligned}$$

$$\therefore h = \frac{5.5 \pm \sqrt{5.5^2 - 4 \cdot 0.412}}{2} = 0.076$$

The volume of the crown may now be calculated by integration :



disc at height y from sphere centre has radius $\sqrt{R^2 - y^2}$

where R = sphere radius

$$\therefore \text{volume of disc} = \pi (R^2 - y^2) dy$$

where dy = disc thickness

$$\therefore \text{volume of crown} = \int_{2.674}^{2.750} \pi (R^2 - y^2) dy$$

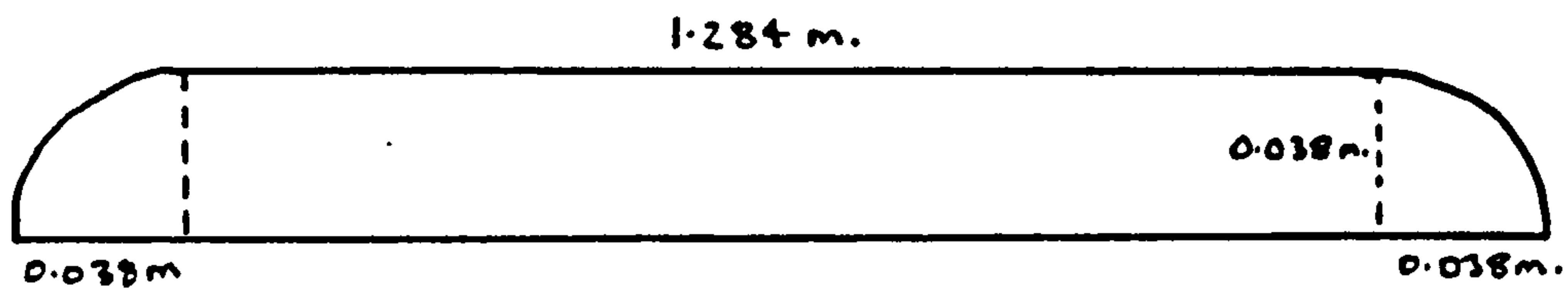
$$= \left[\pi R^2 y - \frac{\pi y^3}{3} \right]_{2.674}^{2.750}$$

$$= 43.55697 - 43.50743$$

$$= \underline{0.0494 \text{ m}^3}$$

Volume of knuckle

The knuckle consists of two components : a cylinder and a quadrant of a torus.



$$\begin{aligned} \text{The volume of the cylinder} &= \frac{\pi D^2 H}{4} & \text{where } D &= \text{diameter} = 1.284\text{m} \\ & & \text{and } H &= \text{height} = 0.038\text{m} \\ &= 0.049 \text{ m}^3 \end{aligned}$$

The volume of the torus may be taken as the volume of a cylinder of radius 0.038m and height πD where D = the diameter of the torus's central axis = 1.284.

$$\text{i.e. volume} = \pi \cdot 0.038^2 \cdot \pi \cdot 1.284 = 0.0183$$

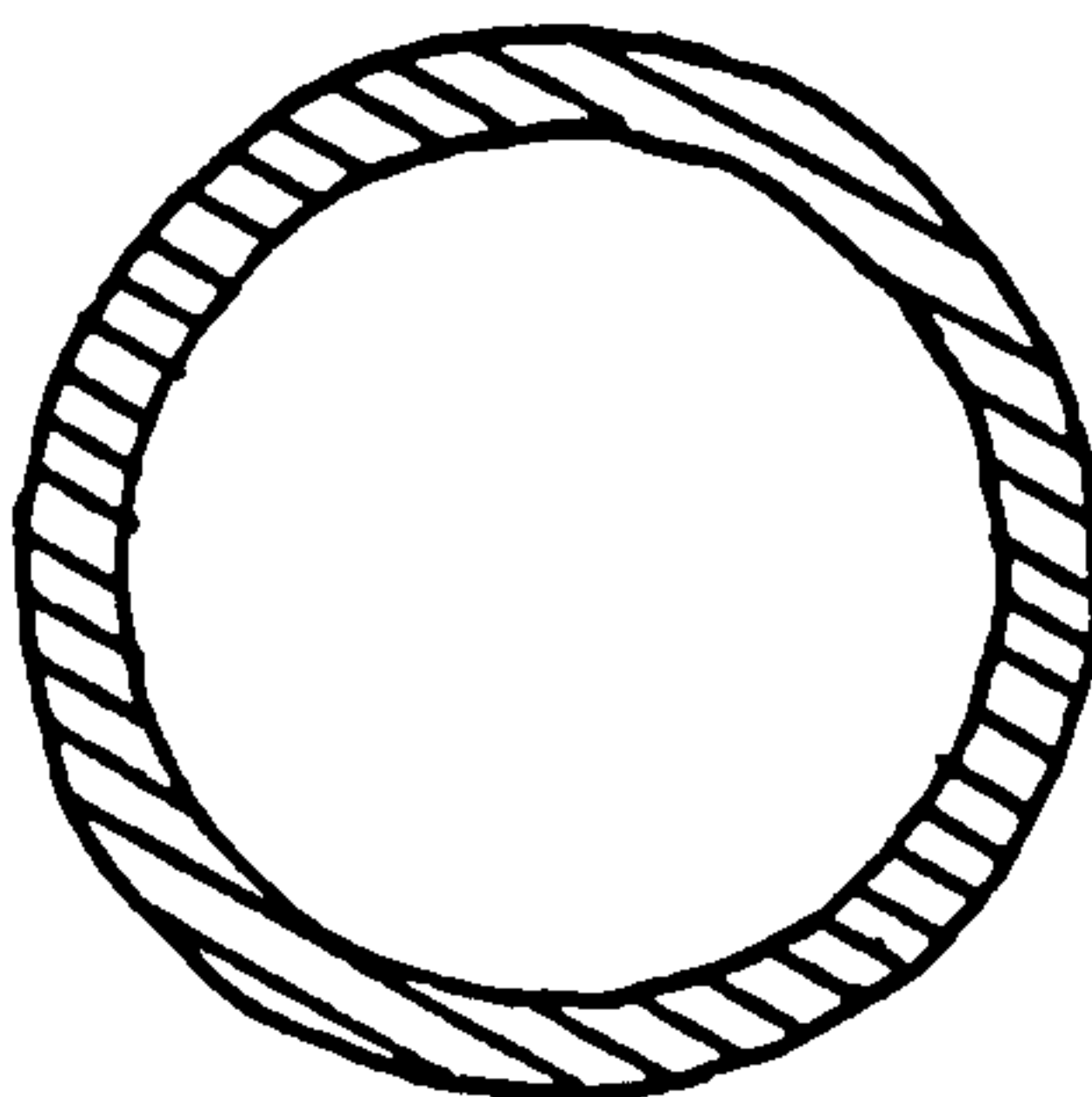
$$\therefore \text{volume of torus quadrant} = 0.0183/4 = 0.0046\text{m}^3$$

$$\therefore \text{The volume of the knuckle} = \underline{0.054 \text{ m}^3}$$

Volume of body

The body comprises a cylinder of diameter 1.36m and height 1.826m. Its volume is therefore 2.6526m^3 .

However from this must be deducted the volume of the internal flanges. There are 4 of these, 2 with inside diameter 1.3m and 2 with inside diameter 1.342m. All of them have outside diameter 1.36m and thickness 0.02m. Their volume can be calculated by subtraction of disc volumes :



$$\text{volume of disc, diameter 1.3m} = \frac{\pi \cdot 1.3^2}{4} 0.02 = 0.0265 \text{ m}^3$$

$$\text{" " " 1.342m} = \frac{\pi \cdot 1.342^2}{4} 0.02 = 0.0283 \text{ m}^3$$

$$\text{" " " 1.36m} = \frac{\pi \cdot 1.36^2}{4} 0.02 = 0.0291 \text{ m}^3$$

$$\therefore \text{The larger flange has volume} = 0.0291 - 0.0265 = 0.0026 \text{ m}^3$$

$$\text{and the smaller flange has volume} = 0.0291 - 0.0283 = 0.0008 \text{ m}^3$$

$$\text{So total volume of flanges} = 0.0068 \text{ m}^3$$

$$\therefore \text{The effective volume of the body} = \underline{2.646 \text{ m}^3}$$

Volume of vessel

The vessel comprises a body, 2 crowns and 2 knuckles.

$$\therefore \text{The vessel volume} = 2.646 + 2 (0.0494 + 0.054)$$

$$= \underline{2.853 \text{ m}^3}$$

APPENDIX C.4CALCULATION OF VOLUME OF STEEL AND INSULANTUSED FOR WATER STORAGE VESSEL(a) Steel

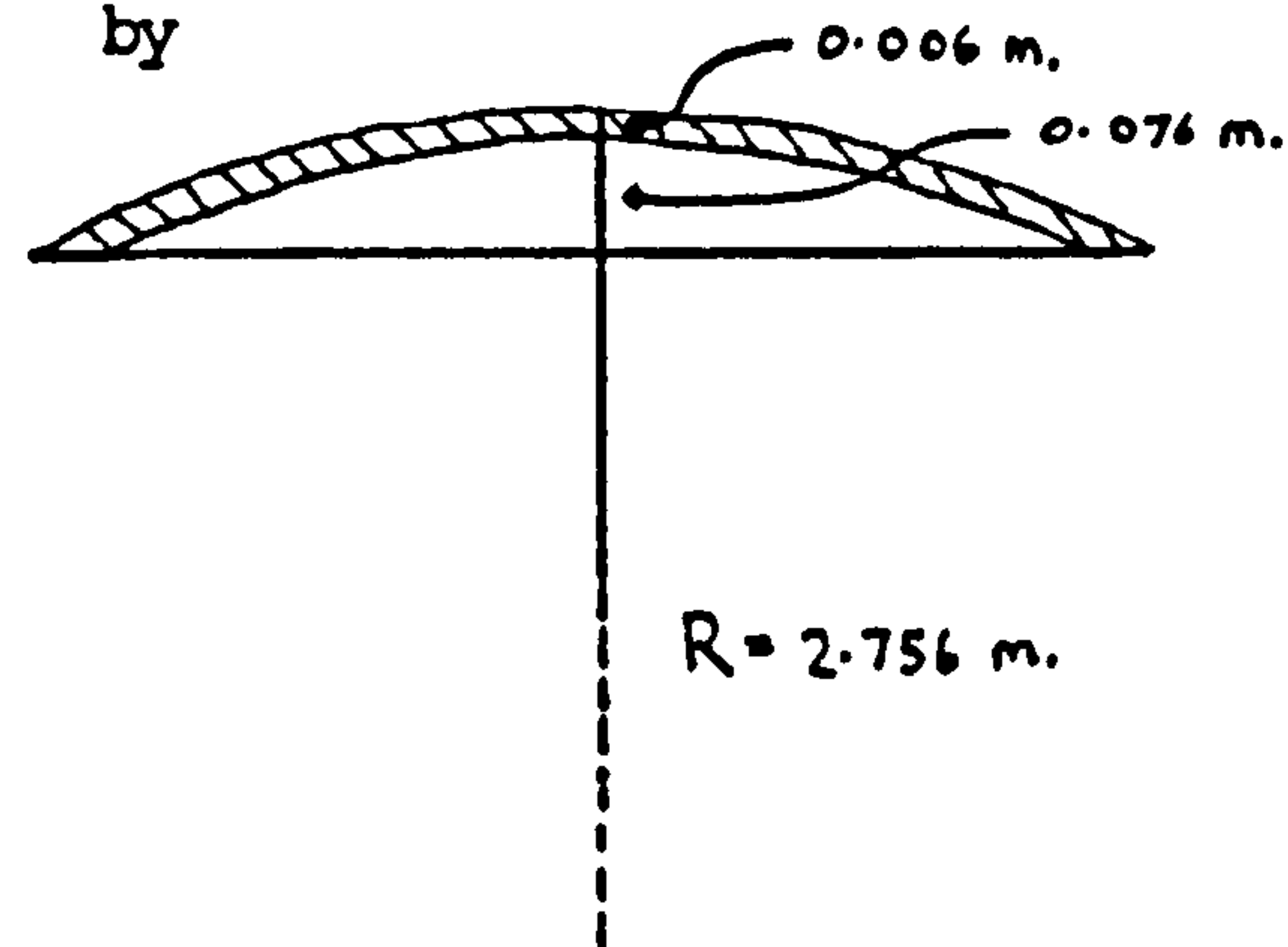
The calculation uses the principles of the previous appendix, determining the volume contained within the external dimensions of the vessel and deducting the internal volume (i.e. the water volume).

Volume of Crown

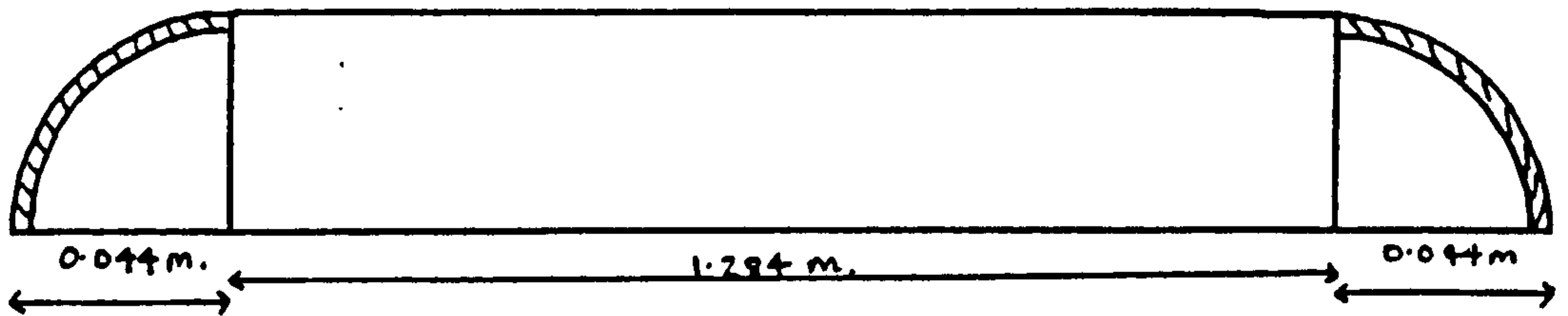
External crown is segment of a sphere of radius 2.756m and width $0.076 + 0.006$, i.e. 0.082m.

As before, the volume is calculated by integration:

$$\begin{aligned}
 \text{Volume} &= \int_{2.674}^{2.756} \pi (R^2 - y^2) dy \\
 &= \left[\pi R^2 y - \frac{\pi y^3}{3} \right]_{2.674}^{2.756} \\
 &= 43.8426 - 43.7850 \\
 &= 0.0576 \text{ m}^3
 \end{aligned}$$



$$\begin{aligned}
 \therefore \text{Volume of steel in crown} &= 0.0576 - 0.0444 \\
 &= \underline{0.0082 \text{ m}^3}
 \end{aligned}$$

Volume of knuckle

Assuming (a slight over estimate) that the steel is the difference between the quadrants of two toruses, the volume of the external quadrant is :

$$\frac{1}{4} \cdot \pi \cdot 0.044^2 \cdot \pi \cdot 1.284 = 0.0061 \text{ m}^3$$

$$\therefore \text{volume of steel in knuckle} = 0.0061 - 0.0046 = \underline{0.0015 \text{ m}^3}$$

Volume of body

For the purposes of calculating the volume of steel in the body, the internal cylinder has diameter 1.36m and height 1.813m and the external cylinder has diameter 1.372m and height 1.813m.

(N.B. The height is 13mm shorter than in the calculation of water volume since there is a gasket 6.5mm thick on each flange which obviously must be excluded).

The internal volume is	2.6337 m ³
and the external volume is	2.6804 m ³
giving a steel volume of	<u>0.0467 m³</u>

Volume of flanges

As calculated previously the total volume of the internal flanges is 0.0068 m³.

There are 4 external flanges each with inside diameter 1.372m, outside diameter 1.472m and thickness 0.02m.

$$\text{Volume of disc diameter 1.372m} = \frac{\pi \cdot 1.372^2}{4} \cdot 0.02 = 0.02957 \text{ m}^3$$

$$\text{Volume of disc diameter 1.472m} = \frac{\pi \cdot 1.472^2}{4} \cdot 0.02 = 0.03404 \text{ m}^3$$

$$\begin{aligned} \therefore \text{The total volume of the external flanges is } & 4 \cdot 0.00447 \text{ m}^3 \\ & = \underline{0.0179 \text{ m}^3} \end{aligned}$$

Total volume of steel

$$\begin{aligned} \text{Adding 2 crowns, 2 knuckles, the body and flanges give the total} \\ \text{volume of steel} &= 2(0.0082 + 0.0015) + 0.0467 + 0.0068 + 0.0179 \\ &= \underline{0.0908 \text{ m}^3} \end{aligned}$$

(b) Insulant

The outside of the vessel was sprayed with foam to an approximate depth of 100mm. Due to its very low density, an approximate estimate of its volume will suffice. This can be done by assuming the outside of the vessel to be a perfect cylinder of diameter 1.372m and height 2.066m and the insulated vessel to be a perfect cylinder of diameter 1.572m and height 2.266m.

$$\text{This makes the uninsulated vessel volume} = 3.054 \text{ m}^3$$

$$\text{and the insulated vessel volume} = 4.398 \text{ m}^3$$

$$\text{So that the volume of insulant} = \underline{1.344 \text{ m}^3}$$

APPENDIX C.5STORE HEAT LOSS RATE AND TIME CONSTANT

For a store at a uniform temperature, T :

rate of heat lost from the store = rate of heat transferred through the insulation

$$\text{So, } -M C_p dT = UA (T - T_{\text{amb}}) dt$$

Integrating between $t = 0$ (corresponding to $T = T_0$) and $t = t$,

$$\int_{T_0}^T \frac{dT}{T - T_{\text{amb}}} = -\frac{UA}{M C_p} dt$$

$$\therefore UA = \frac{M C_p}{t} \ln \left| \frac{T_0 - T_{\text{amb}}}{T - T_{\text{amb}}} \right|$$

The time constant, τ_c , of the store is defined as the time taken for the temperature excess of the store over its surroundings (and so also the energy stored) to have decayed to $1/e$ of their initial value.

$$\text{i.e. } t = \tau_c \text{ when } \frac{T - T_{\text{amb}}}{T_0 - T_{\text{amb}}} = \frac{1}{e}$$

$$\therefore \tau_c = \frac{M C_p}{UA}$$

APPENDIX C.6CALCULATION OF POSITION AND SIZE OF DISTRIBUTOR HOLES(a) Deciding the number of holes

The aim was to have the same flow through each hole, creating as little turbulence as possible. It was decided to try to keep the velocity, u , of the flow from each hole the same as the velocity of the total flow entering the distributor. A higher velocity would have created more turbulence and a lower velocity might have caused the flow from the holes towards the ends of the distributor arms to be less than that from the holes at the start of the arms.

To keep the velocity the same, the number of holes, N , had to be inversely proportional to the square of the hole radius;

$$\text{i.e. } u = \frac{\dot{V}}{NA_{in}} = \text{const.} \rightarrow NA_{in} = \text{const.} \rightarrow N \propto \frac{1}{r_o^2} \quad (A_{in} = \pi r^2)$$

The results of the plain vertical port experiments had shown that vertical jet penetration was given by

$$P = C \dot{V} r^{-3/2} (g \beta \Delta T)^{-1/2}$$

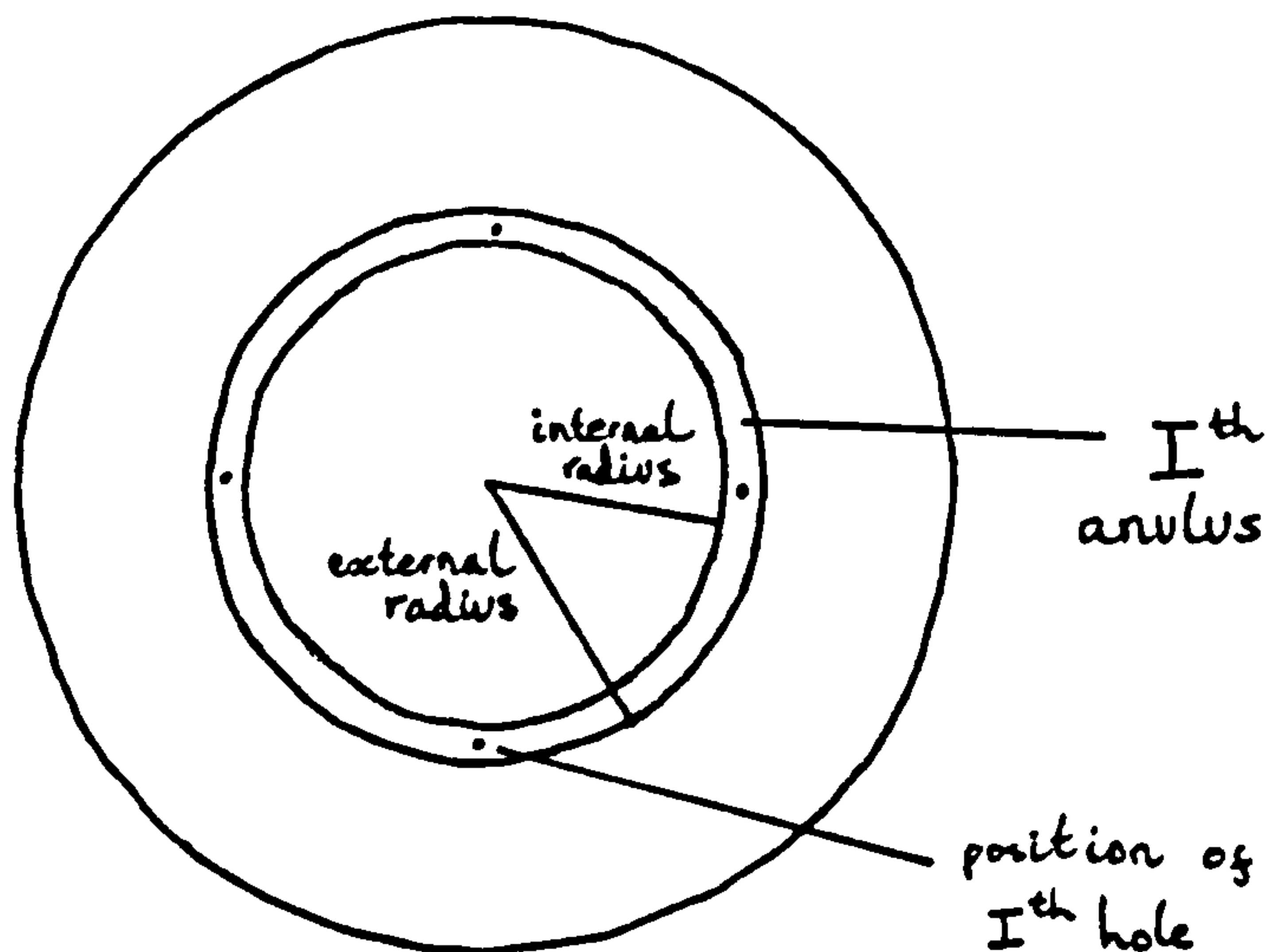
$$\text{So that } P \propto \dot{V} r^{-3/2} = u A_{in} r^{-3/2}$$

$$\text{so } P \propto r^{1/2} \quad (u = \text{const.} ; A_{in} = \pi r^2)$$

Thus the penetration reduces as the hole radius decreases, as long as the number of holes is increased to maintain a constant velocity. Three factors counterbalance the argument for reducing the hole radius : (i) as the hole radius decreases the pressure drop increases; (ii) similarly, the risk of blockage increases and (iii) as the number of holes goes up, the cost of manufacturing the distributor increases. For this work the compromise design of 256 1/16" nominal diameter holes was selected.

(b) Calculating the position of each hole

The position of the holes is determined so that each hole serves an equal cross-sectional area of the vessel. With 4 arms, and holes on each side of each arm, the tank cross-section must be divided into 32 $(256/(4 \times 2))$ equal annuli, with a hole at the centre of each annulus.



Let total cross-sectional area of vessel = A

then, area of each annulus = $A/32$

area up to start of I^{th} annulus = $\frac{(I-1)A}{32}$

area up to end of I^{th} annulus = $\frac{I A}{32}$

$\therefore$ internal and external radii of I^{th} annulus are $\sqrt{\frac{(I-1)A}{32\pi}}$ and $\sqrt{\frac{I A}{32\pi}}$

$\therefore$ radius of I^{th} hole, r_I = $\frac{1}{2} \left\{ \sqrt{\frac{(I-1)A}{32\pi}} + \sqrt{\frac{I A}{32\pi}} \right\}$

A computer program incorporating this formula was created and it gave the following results :

DISTRIBUTOR DESIGN FOR STORE DIAMETER 1.36M
POSITION OF HOLES IN DISTRIBUTOR

HOLE NO. RADIUS IN MM

1	60
2	145
3	189
4	224
5	254
6	281
7	306
8	329
9	350
10	370
11	389
12	407
13	424
14	441
15	457
16	473
17	488
18	502
19	516
20	530
21	544
22	557
23	570
24	582
25	594
26	606
27	618
28	630
29	641
30	652
31	663
32	674

(c) Calculating the diameter of each hole

Although it was initially assumed that each hole had a diameter of 1/16", it was felt that holes towards the end of the distributor might need to be slightly larger, to compensate for the extra head loss suffered by the water in travelling further down the arm. It was therefore decided to calculate the size of each hole to ensure that HL, the total head loss (or resistance to flow) from the start of the distributor to the exit from each hole, was constant. A condition which would necessarily produce a constant flow from each hole.

Taking the position of the first hole as a common starting point, we can say [ref.C.1], that the head loss through the first hole is given by :

$$HL = \frac{u_1^2}{2g} \quad \text{where} \quad u_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V} \cdot 4}{256 \cdot \pi D_1^2}$$

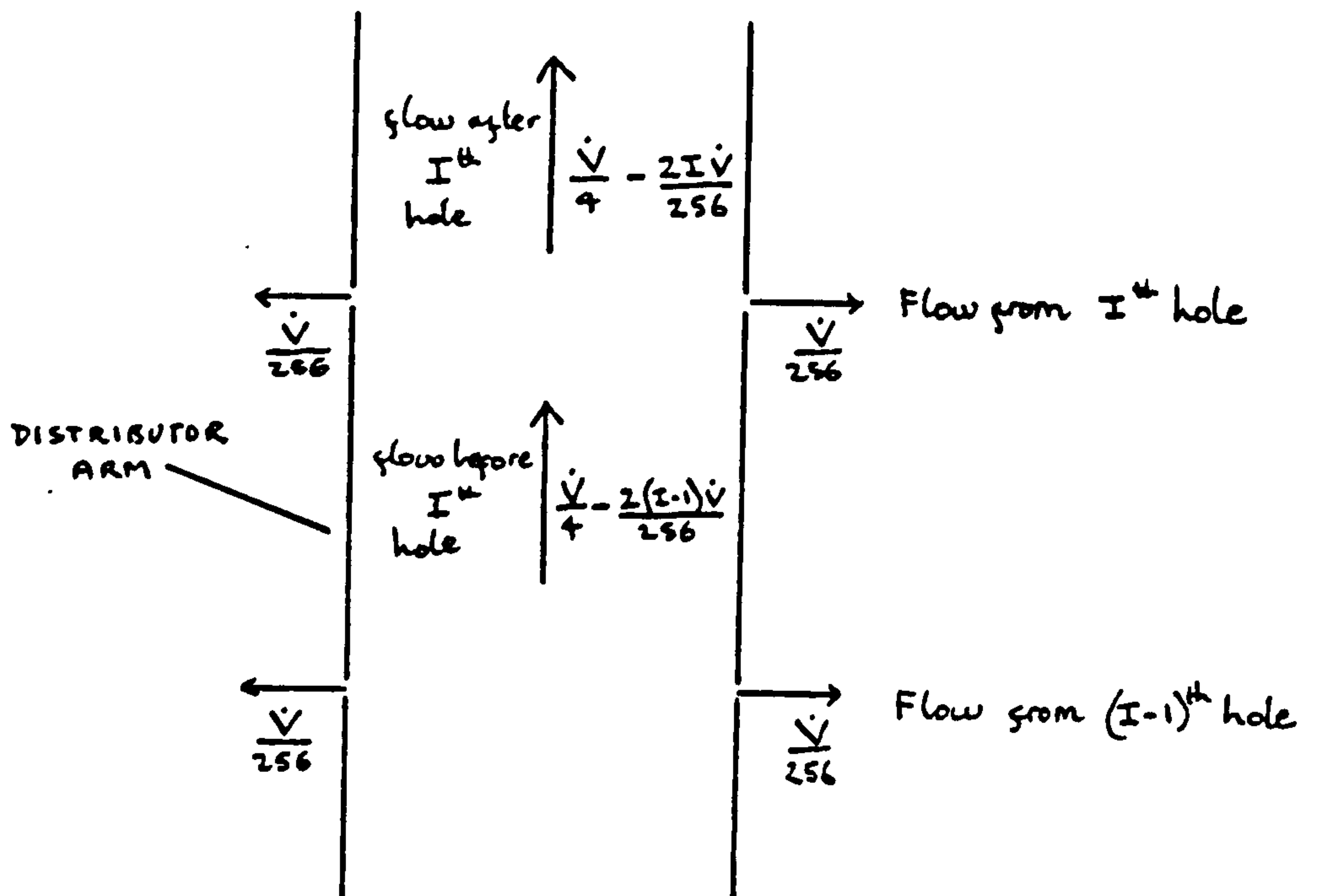
Now we assume that $D_1 = 1/16'$ (1.59mm) and take a typical value for $\dot{V}$, the total flow into the vessel, as $0.4 \cdot 10^{-3} \text{ m}^3/\text{s}$.

$$\text{Hence } u_1 = 0.79 \text{ m/s}$$

$$\text{and } HL = 3.16 \cdot 10^{-2} \text{ m.}$$

Thus the total head loss for each hole is a fixed, known value. To calculate the diameter of each hole after the first, we simply move down the distributor taking each hole in turn, calculating the head loss in the arm of the distributor up to the hole, deducting this from the fixed total head loss ($3.16 \cdot 10^{-2} \text{ m}$) and equating the result to the head loss through the hole, thereby fixing that hole's diameter.

Consider the I^{th} hole :



The flow between the $(I-1)^{\text{th}}$ hole and the I^{th} hole is given by

$$\dot{V}_I = \frac{\dot{V}}{4} - \frac{(I-1)\dot{V}}{128}$$

and the velocity in this section $u_I = \frac{\dot{V}_I}{A_A}$

where A_A = cross sectional area of distributor arm.

The length of this section $L_I = R_I - R_{I-1}$ and thus the head loss in this section can be calculated from

$$h_I = f_I \frac{L_I}{D} \frac{u_I^2}{2g} \quad \left(A_A = \frac{\pi D^2}{4} \right) [\text{ref.C.1}] \quad (1)$$

where f is the friction factor for the pipe which is found from a Moody diagram [ref.C.1], by knowing the Reynolds number of the flow and the relative roughness of the pipe.

The roughness height of drawn copper tubing, $\epsilon = 5.10^{-6}$ ft.

The diameter of the distributor arm, $D = \frac{1}{12}$ ft

$\therefore$ the relative roughness of the pipe $\frac{\epsilon}{D} = 6 \cdot 10^{-5}$

The flow in the arm decreases from 0.1 litres/s before the first hole to $3.1 \cdot 10^{-3}$ litres/s before the last hole.

Thus the Reynolds number decreases from about 9000 to 300.

For $Re = 9000$, i.e. turbulent flow, the Moody diagram gives
 $f = 0.032$ under these conditions.

For $Re = 300$, i.e. laminar flow, the friction factor is independent of ϵ and is given by :

$$f = \frac{64}{Re} ; \quad \text{so in this case } f = 0.21$$

For the purposes of this calculation, it was decided to assume a linear increase in f along the arm corresponding to the linear decrease in velocity and hence Reynolds number.

i.e. Defining f_I as the friction factor in the section between the $(I-1)^{th}$ and I^{th} hole,

$$f_I = \frac{0.21 - 0.032}{31} (I-1) + 0.032 = 0.0057 (I-1) + 0.032$$

Thus all the parameters in equation (1) can be calculated to give h_I . The total head loss up to the I^{th} hole is then found by summing the head losses of all the sections up to the I^{th} hole :

$$\text{i.e.} \quad \text{HL (I)} = \sum_{I=1}^I h_I$$

and, since the starting point on the arm for the calculations is the first hole,

$$\text{HL (1)} = h_1 = 0$$

Now, to calculate the diameter of the I^{th} hole, the head loss through the I^{th} hole, $H(I)$, is fixed so that :

$$H(I) + \text{HL}(I) = \text{HL} \quad (2)$$

$$\text{Now } H(I) = \frac{u_I^2}{2g} \quad \text{and} \quad u_I = \frac{\dot{V}}{256 A_I} = \frac{\dot{V} \cdot 4}{256 \cdot \pi D_I^2}$$

Thus, by calculating $H(I)$ using equation (2), we can determine u_I , A_I and therefore D_I .

A computer program incorporating these formulae was written and it gave the results below. It can be seen that the head loss in the distributor is much less than the head loss through a hole. Indeed, the head loss along the whole distributor arm is only 3% of the head loss of one hole. This means that the diameter of the last hole need only be 0.7% greater than that of the first hole to compensate for the head loss in the arm. (Remember : $H(I) \propto D_I^{-4}$). This is, of course, negligible.

CALCULATION OF HEAD LOSS IN DISTRIBUTOR UP TO EACH HOLE

PARAMETERS OF DISTRIBUTOR

DIAMETER OF DISTRIBUTOR ARM= .0254 M

TOTAL FLOW INTO DISTRIBUTOR= .4 LITRES/SEC

ASSUMED FRICTION FACTOR ALONG ALL OF DISTRIBUTOR, F= .0395

HOLE NO.	RADIUS IN MM	FLOW RATE AFTER HOLE IN M ³ /SEC	HEAD LOSS BETWEEN HOLES IN M	HEAD LOSS UPTO NXT HOLE IN M
1	60	9.6875E-05	2.46259E-04	2.46259E-04
2	145	9.375E-05	1.19384E-04	3.65643E-04
3	189	9.0625E-05	8.8739E-05	4.54382E-04
4	224	.0000875	7.09068E-05	5.25289E-04
5	254	8.4375E-05	5.93392E-05	5.84628E-04
6	281	8.125E-05	5.09492E-05	6.35577E-04
7	306	7.8125E-05	4.33369E-05	6.78914E-04
8	329	.000075	3.64663E-05	7.15381E-04
9	350	7.1875E-05	3.1896E-05	7.47277E-04
10	370	6.875E-05	2.77236E-05	.000775
11	389	6.5625E-05	2.3931E-05	7.98931E-04
12	407	.0000625	2.05002E-05	8.19431E-04
13	424	5.9375E-05	1.85015E-05	8.37933E-04
14	441	5.625E-05	1.56284E-05	8.53561E-04
15	457	5.3125E-05	1.39402E-05	8.67501E-04
16	473	.00005	1.15766E-05	8.79078E-04

17	488	4.6875E-05	9.49644E-06	8.88575E-04
18	502	4.375E-05	8.27245E-06	8.96847E-04
19	516	4.0625E-05	7.13288E-06	9.0398E-04
20	530	.0000375	6.07772E-06	9.10057E-04
21	544	3.4375E-05	4.74219E-06	.0009148
22	557	3.125E-05	3.91917E-06	9.18719E-04
23	570	2.8125E-05	2.93033E-06	9.21649E-04
24	582	.000025	2.31532E-06	9.23965E-04
25	594	2.1875E-05	1.77267E-06	9.25737E-04
26	606	1.875E-05	1.30237E-06	9.2704E-04
27	618	1.5625E-05	9.04422E-07	9.27944E-04
28	630	.0000125	5.30595E-07	9.28475E-04
29	641	9.375E-06	2.98459E-07	9.28773E-04
30	652	6.25E-06	1.32649E-07	9.28906E-04
31	663	3.125E-06	3.31621E-08	9.28939E-04
32	674			

CALCULATION OF DIAMETERS OF EACH HOLE TO ACHIEVE EQUAL FLOW

ASSUMED DIAMETER OF FIRST HOLE= 1.5875 MM

HOLE NO.	HOLE DIAMETER IN MM	HOLE AREA IN M ²	VELOCITY THRO HOLE HOLE IN M/SEC	HEAD LOSS IN M
1	1.5875	1.97933E-06	.789411	.0317619
2	1.59059	1.98704E-06	.786344	.0315157
3	1.5921	1.99082E-06	.784854	.0313963
4	1.59323	1.99364E-06	.783744	.0313076
5	1.59413	1.9959E-06	.782856	.0312366
6	1.59489	1.9978E-06	.782112	.0311773
7	1.59554	1.99943E-06	.781472	.0311264
8	1.5961	2.00082E-06	.780928	.031083
9	1.59657	2.002E-06	.78047	.0310466
10	1.59698	2.00303E-06	.780069	.0310147
11	1.59733	2.00392E-06	.77972	.0309869
12	1.59764	2.0047E-06	.779419	.030963
13	1.59791	2.00536E-06	.779161	.0309425
14	1.59815	2.00596E-06	.778928	.030924
15	1.59835	2.00647E-06	.778731	.0309084
16	1.59853	2.00692E-06	.778556	.0308944
17	1.59868	2.0073E-06	.77841	.0308829
18	1.5988	2.00761E-06	.77829	.0308734
19	1.59891	2.00788E-06	.778186	.0308651
20	1.599	2.00811E-06	.778096	.030858
21	1.59908	2.00831E-06	.778019	.0308519
22	1.59914	2.00846E-06	.777959	.0308471
23	1.59919	2.00859E-06	.77791	.0308432
24	1.59923	2.00868E-06	.777873	.0308403
25	1.59926	2.00876E-06	.777844	.030838
26	1.59928	2.00882E-06	.777821	.0308362
27	1.5993	2.00886E-06	.777805	.0308349
28	1.59931	2.00889E-06	.777794	.030834
29	1.59932	2.00891E-06	.777787	.0308335
30	1.59932	2.00891E-06	.777783	.0308332
31	1.59932	2.00892E-06	.777782	.030833
32	1.59932	2.00892E-06	.777781	.030833

APPENDIX C.7LISTING OF THE MONITORING AND CONTROL COMPUTER PROGRAM

```

10 ON ERROR GOTO 3670
15 START=9.2E+07
20 OUT 0,1
30 OUT 6,48
40 OUT 6,32
50 DIM THERM(59),MAT(59),STOR(929)
60 I=67
70 READ CONST
80 OUT 1,85:OUT 1,CONST
90 I=I-1:IF I>63 GOTO 70
100 DATA 255,60,60,100
110 FOR I=1 TO 8
120 FOR J=0 TO 10
130 READ TCALIB(I,J)
140 NEXT J
150 NEXT I
160 DATA 1.2,10.9,20.7,30.6,40.3,50.2,59.9,69.8,79.7,89.4,99.2
170 DATA 0.2,10.1,20.1,30.1,39.8,49.9,59.8,69.8,79.8,89.7,99.7
180 DATA 1.4,11.4,21.1,30.9,40.5,50.4,60.0,70.1,79.8,89.6,99.4
190 DATA 0.1,10.5,20.9,31.3,41.3,51.6,61.4,71.5,81.3,90.9,100
200 DATA 0.0,10.3,20.9,31.5,41.6,52.0,62.1,72.4,82.3,92.2,100
210 DATA 1.4,11.2,21.1,30.9,40.6,50.5,60.3,70.3,80.1,89.9,99.8
220 DATA 1.2,11.1,21.0,30.7,40.3,50.3,59.9,69.8,79.6,89.4,99.2
230 DATA 1.1,11.0,20.8,30.7,40.3,50.3,60.0,70.0,79.8,89.6,99.5
240 FOR I=1 TO 4
250 READ FMCAL(I),FMCON(I)
260 NEXT I
270 DATA .0190,.0356,.00575,.0068,.0185,.0441,.00572,.0063
280 FOR I=1 TO 24
290 PRINT
300 NEXT I
310 PRINT"****STEP INPUT TESTING OF WATER ENERGY STORE****"
320 INPUT"HAVE YOU PLACED A DATA DISC IN DRIVE B";A$
330 OPEN "D",*1,"B:CSTORE"
340 INPUT"EXPERIMENT REFERENCE NO.(FORMAT:CN.DDMMYY)";E#
350 C=INT(E#/10)
360 IF C=0 OR C>2 THEN PRINT "INVALID REF. NO.;TRY AGAIN":GOTO 340
370 IF C=1 THEN PRINT "YOU HAVE SELECTED A STEP CHARGE EXFT.":GOTO 390
380 PRINT"YOU HAVE SELECTED A STEP DISCHARGE EXFT."
390 INPUT"IS THIS CORRECT(PRESS N IF NOT)";N$
400 IF N$="N" GOTO 340
410 INPUT"HAVE YOU SWITCHED OVER TO AUTO CONTROL";A$
420 PRINT"VALVES WILL NOW BE SET IN CORRECT POSITION FOR START OF EXPERIMENT"
430 MPSET(1)=10
440 MPSET(2)=10
450 MPSET(3)=10
460 MPSET(4)=10
470 MPSET(5)=.15
480 MPSET(6)=.1
490 PRINT STRING$(70,45)
500 PRINT"VALVE NO.", "REQ.D FOS", "CURRENT FOS"
510 FOR I=1 TO 6
520 GOSUB 1860
530 PRINT I,MPSET(I),MP(I)
540 NEXT I
550 PRINT STRING$(70,45)
560 FLAG=0
570 FOR I=1 TO 6
580 GOSUB 1860
590 MPDIF=MPSET(I)-MP(I)
600 IF ABS(MPDIF))>.2 THEN GOSUB 1970 ELSE GOTO 610
610 NEXT I
620 IF FLAG=1 GOTO 560

```



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630 FOR I=1 TO 6
640 OUT(79+I),0
650 NEXT I
660 PRINT"ALL VALVES ARE NOW IN CORRECT INITIAL POSITION
670 I=4:GOSUB 2120
680 PRINT"HOT HEADER TEMP IS NOW";T(4);"DEG C"
690 INPUT"REQUIRED HOT HEADER TEMP TO START EXPERIMENT";TSET(4)
700 IF T(4)>TSET(4) GOTO 770
710 PRINT"HOT HEADER TEMP WILL NOW BE RAISED TO REQUIRED VALUE"
720 IF T(4)<TSET(4) GOTO 770
730 FOR I=1 TO 10000
740 NEXT I
750 I=4:GOSUB 2120
760 PRINT"HOT HEADER TEMP IS NOW";T(4);"DEG C  -----";TSET(4)-T(4);"DEG C TO GO":GOTO 720
770 PRINT STRING$(70,45)
780 IF C=1 THEN TCONT=6:VT=82:VF=83:FCONT=2
790 IF C=2 THEN TCONT=1:VT=80:VF=81:FCONT=3
800 PRINT"CHECK DATALOGGER IS SCANNING CORRECTLY"
810 GOSUB 2950
820 INPUT"IS SCAN SATISFACTORY";SCAN$
830 IF SCAN$="N" GOTO 800
840 PRINT"NOW INPUT VALUES OF SET PARAMETERS FOR THIS EXPERIMENT"
850 INPUT"INITIAL TEMP. OF STORE(FORMAT:XX.X DEG.C)";TINIT
860 IF C=1 THEN INPUT"STEP RISE IN TEMP.(FORMAT:XX.X DEG.C)";TRISE:TSTEP=TINIT+TRISE:GOTO 880
870 INPUT"STEP DROP IN TEMP.(FORMAT:XX.X DEG.C)";TDROP:TSTEP=TINIT-TDROP
880 PRINT STRING$(70,45)
890 INPUT"FLOWRATE(FORMAT:X.XX KG/SEC)";FSET
900 FTOL=.004
910 IF FSET<.2 THEN FTOL=.02*FSET
920 PRINT STRING$(70,45)
930 INPUT"TEST TIME(FORMAT:XXX.XXX HOURS)";H
940 H=H*3600
950 Y=2962/FSET
960 PRINT "H/Y =";H/Y
970 IF H>Y AND H<2*Y GOTO 1010
980 PRINT"TEST TIME IS <1 OR >2 TIMES EXPECTED TIME BASED ON SET FLOWRATE"
990 INPUT "ARE THEY CORRECT(PRESS Y FOR YES)";A$
1000 IF A$(">Y" GOTO 840
1010 P=H/99
1020 PHR=(CINT(P/3.6))/1000
1030 PRINT"PERIOD BETWEEN SCANS=";PHR;"HOURS"
1040 PRINT"EXPERIMENT WILL NOW PROCEED"
1050 PRINT"PUMP WILL NOW BE SWITCHED ON":OUT (86+2*(C-1)),1:OUT (87+2*(C-1)),1
1060 PRINT"CONDITIONS FOR EXPERIMENT WILL NOW BE PRINTED"
1070 LPRINT"EXPERIMENT REFERENCE NO. =";E*
1080 LPRINT"INITIAL TEMP. OF STORE =";TINIT
1090 IF C=1 THEN LPRINT"STEP RISE IN TEMP. =";TRISE
1100 IF C=2 THEN LPRINT"STEP DROP IN TEMP. =";TDROP
1110 LPRINT"FLOWRATE =";FSET
1120 LPRINT"TEST TIME =";H/3600;"HOURS"
1130 LPRINT"PERIOD BETWEEN SCANS =";PHR;"HOURS"
1140 PRINT#1,USING"**,*****";E*
1150 PRINT#1,USING"**,.*";TINIT
1160 PRINT#1,USING"**,.*";TSTEP
1170 PRINT#1,USING"**,.*";FSET
1180 PRINT#1,USING"**,.*";H/3600
1190 PRINT#1,USING"**,.*";P/3600
1200 IF C=2 AND FSET<.25 THEN FCONT=FCONT+1
1210 PRINT"STORE WILL NOW BE TAKEN TO ITS CORRECT INITIAL TEMP."
1220 TSET=TINIT+1
1230 GOSUB 2220
1240 LAP=TIME
1250 GOSUB 2220

```



```

1260 IF (LAP-TIME)/100<300 GOTO 1350
1270 LAP=TIME
1280 GOSUB 2950
1290 FLAG=0
1300 FOR I=0 TO 49
1302 IF I=6 OR I=10 GOTO 1330
1304 IF I=12 OR I=13 GOTO 1330
1310 IF I=15 OR I=43 GOTO 1330
1320 IF ABS(THERM(I)-TINIT))1 THEN FLAG=1
1330 NEXT I
1340 IF FLAG=0 GOTO 1370
1350 GOSUB 3500
1360 GOTO 1250
1370 PRINT"TEMP. THROUGHOUT STORE IS NOW WITHIN 1 DEG. C OF REQUIRED INITIAL"
1380 PRINT"VALUE. THE FLOWRATE WILL NOW BE TAKEN TO ITS SET VALUE."
1390 GOSUB 3590
1400 IF ABS(FDIF)<.01 GOTO 1420
1410 GOTO 1390
1420 PRINT"FLOWRATE IS NOW CORRECT. THE STEP CHANGE EXPERIMENT WILL NOW PROCEED"
1430 PRINT"FIRST A STABILISATION PERIOD !"
1440 GOSUB 2220
1450 BEGIN=TIME
1460 FLAG=0
1470 GOSUB 3500
1480 IF ABS(TDIF))1 THEN FLAG=1
1490 GOSUB 3590
1500 IF ABS(FDIF))1 THEN FLAG=1
1510 IF FLAG=1 GOTO 1460
1520 GOSUB 2220
1530 IF (BEGIN-TIME)/100<300 GOTO 1460
1540 PRINT"STABILISATION PERIOD COMPLETE"
1550 TSET=TSTEP
1560 GOSUB 2220
1570 START=TIME:LAP=TIME:TINSUM=0:TOUTSUM=0:FSUM=0:TDEV=0:FDEV=0
1580 LPRINT"START TIME FOR EXPERIMENT =";START
1590 LPRINT "ELAPSED TIME","TIN AVE. ","TOUT INSTANT","FLOW AVE. ","T HOT INSTANT","T CLD INST"
1600 LOOPS=0
1610 FLAGA=1
1620 GOSUB 2220
1630 LOOPS=LOOPS+1
1640 IF (LAP-TIME)/100>F THEN GOSUB 2560
1650 IF (START-TIME)/100>H GOTO 1770
1652 OUT 96,12
1654 LB=INP(96)
1656 IF LB<128 GOTO 1654
1658 MB=INP(97)
1660 OUT 96,13
1662 LSB=INP(96)
1663 IF LSB<128 GOTO 1662
1664 MSB=INP(97)
1665 GOSUB 1940
1666 I=8:GOSUB 2140
1668 LSB=LB:MSB=MB:GOSUB 1940
1670 T(7)=CINT(CALC*25)/10
1672 IF C=1 THEN TIN=T(8):TOUT=T(8)-T(7)
1674 IF C=2 THEN TIN=T(8)-T(7):TOUT=T(8)
1680 TINSUM=TINSUM+TIN
1685 TOUTSUM=TOUTSUM+TOUT
1690 TDEV=TDEV+(TSET-TIN)^2
1700 GOSUB 3500
1710 GOSUB 3590
1720 FSUM=FSUM+FLOW(FCONT)
1730 FDEV=FDEV+(FSET-FLOW(FCONT))^2

```



```

1740 IF FLAGA=1 THEN GOSUB 2360
1745 IF FLAGA=1 THEN LOOPS=LOOPS-1
1750 FLAGA=0
1760 GOTO 1620
1770 PRINT"END OF EXPERIMENT"
1780 PRINT"PUMP WILL NOW BE SWITCHED OFF"
1790 GOSUB 2360
1800 LPRINT" END TIME FOR EXPERIMENT =";TIME
1810 OUT 86,0:OUT 87,0:OUT 88,0:OUT 89,0
1850 END
1860 ADC=I-1
1870 GOSUB 1900
1880 MP(I)=INT(CALC*100+.5)/100
1890 RETURN
1900 OUT 96,ADC
1910 LSB=INP(96)
1920 IF LSB<128 GOTO 1910
1930 MSB=INP(97)
1940 CALC=LSB-16*INT(LSB/16)+MSB*16
1950 CALC=CALC/409.5
1960 RETURN
1970 IF MPDIF>0 THEN DRIVE=253 ELSE DRIVE=255
1980 OUT(79+I),DRIVE
1990 FLAG=1
2000 RETURN
2010 I=TCONT:GOSUB 2120
2020 TCHNG=ABS(T(I)-A):IF TCHNG>=.2 GOTO 2050
2030 IF TDIF>0 THEN DRIVE=27:GOTO 2090
2040 IF TDIF<0 THEN DRIVE=25:GOTO 2090
2050 IF TDIF>0 AND T(I)>A THEN DRIVE=27:GOTO 2090
2060 IF TDIF<0 AND T(I)>A THEN DRIVE=253:GOTO 2090
2070 IF TDIF<0 AND T(I)<A THEN DRIVE=25:GOTO 2090
2080 IF TDIF>0 AND T(I)<A THEN DRIVE=255
2090 OUT VT,DRIVE
2100 REM LPRINT A,T(1),TDIF,DRIVE,MP(1)
2110 RETURN
2120 ADC=I+5
2130 GOSUB 1900
2140 T(I)=CALC*10
2145 IF I=7 THEN T(I)=T(I)/4:GOTO 2200
2150 FOR J=0 TO 10
2160 IF T(I)<TCALIB(I,J) GOTO 2190
2170 NEXT J
2180 GOTO 2200
2190 T(I)=10*(J-1)+10*((T(I)-TCALIB(I,J-1))/(TCALIB(I,J)-TCALIB(I,J-1)))
2200 T(I)=INT(T(I)*10+.5)/10
2210 RETURN
2220 HOURS=INP(67):MINS=INP(66):SECS=INP(65):XTMS=INP(64)
2230 IF XTMS=CONST GOTO 2220
2240 TIME=360000**HOURS+6000**MINS+100*SECS+XTMS
2242 XTIM=(START-TIME)/360000!
2244 IF XTIM>64 THEN XTIM=64+CINT((XTIM-64)*1000)/1000:GOTO 2250
2246 IF XTIM>32 THEN XTIM=32+CINT((XTIM-32)*1000)/1000:GOTO 2250
2248 XTIM=CINT(XTIM*1000)/1000
2250 RETURN
2260 I=VF-79:GOSUB 1860
2270 COUNT=3*INT(MP(I)/3)
2280 IF COUNT=0 THEN COUNT=1
2290 OUT(14+F*2),COUNT
2300 GOSUB 2220
2310 JAM=TIME
2320 MSB1=INP(14+F*2)
2330 GOSUB 2220

```



```

2340 IF JAM-TIME>200 THEN FLOW(F)=0:RETURN
2350 IF MSB1<128 GOTO 2320
2360 LSB1=INP(15+F*2)
2370 MSB2=INP(14+F*2)
2380 IF MSB1=MSB2 GOTO 2400
2390 LSB2=INP(15+F*2):COUNTIME=LSB2+(MSB2-128)*256:GOTO 2410
2400 COUNTIME=LSB1+(MSB1-128)*256
2410 FLOW(F)=10000*FMCAL(F)*(COUNT+1)/COUNTIME
2420 FLOW(F)=FLOW(F)+FMCON(F)
2430 RETURN
2440 F=FCONT:GOSUB 2260
2450 SUM=SUM+FLOW(F)
2460 K=K+1
2470 IF K<10 THEN RETURN
2480 DAC1=CINT(25.6*SUM)
2490 OUT 98,DAC1
2500 SUM=0:K=0
2510 RETURN
2520 IF FDIF>0 THEN DRIVE=25:GOTO 2540
2530 IF FDIF<0 THEN DRIVE=27:GOTO 2540
2540 OUT VF,DRIVE
2550 RETURN
2560 REM DATAC SUB
2570 GOSUB 2950
2580 PRINT#1,USING"***";DQ
2590 PRINT#1,USING"***.***";XTIM
2600 TINAVE=TINSUM/LOOPS:TDEVAVE=(TDEV/LOOPS)^.5
2605 TOUTAVE=TOUTSUM/LOOPS
2610 TINAVE=(CINT(TINAVE*10))/10
2615 TOUTAVE=CINT(TOUTAVE*10)/10
2620 PRINT#1,USING"***.***";TINAVE
2630 PRINT#1,USING"***.***";TDEVAVE
2640 PRINT#1,USING"***.***";TOUTAVE
2660 FLAVE=FSUM/LOOPS:FDEVAVE=(FDEV/LOOPS)^.5
2670 FLAVE=(CINT(FLAVE*1000))/1000
2680 PRINT#1,USING"***.***";FLAVE
2690 PRINT#1,USING"***.***";FDEVAVE
2700 FOR I=0 TO 49
2710 PRINT#1,USING"***.***";THERM(I)
2720 NEXT I
2730 PRINT#1,USING"***.***";TCIN
2740 PRINT#1,USING"***.***";TCOUT
2750 AMB=0
2760 FOR I=50 TO 57
2762 IF I=52 GOTO 2780
2770 AMB=AMB+THERM(I)
2780 NEXT I
2790 AMBAVE=AMB/7
2800 AMBDEV=0
2810 FOR I=50 TO 57
2812 IF I=52 GOTO 2830
2820 AMBDEV=AMBDEV+(AMBAVE-THERM(I))^2
2830 NEXT I
2840 AMBDEVAVE=(AMBDEV/7)^.5
2850 FOR I=4 TO 5
2860 GOSUB 2120
2870 NEXT I
2880 PRINT#1,USING"***.***";AMBAVE
2890 PRINT#1,USING"***.***";AMBDEVAVE
2900 PRINT#1,USING"***.***";T(4)
2910 PRINT#1,USING"***.***";T(5)
2920 LPRINT XTIM,TINAVE,TOUTAVE,FLAVE,T(4),T(5);DP$
2930 TOUTSUM=0:LAP=TIME:LOOPS=0:TINSUM=0:FSUM=0:TDEV=0:FDEV=0

```



```

2940 RETURN
2950 REM TAKE A SCAN
2960 DP$=" ":DQ=0
2970 OUT 0,0
2980 OUT 6,48
2990 OUT 6,32
3000 L=1
3010 IF (INP(6) AND 2) = 2 THEN 3030
3020 IF (INP(6) AND 1) = 1 THEN 3100 ELSE GOTO 3010
3030 IF L=930 THEN PRINT "DATALOGGER PROBLEM":DP$="*":DQ=1:GOTO 3100
3040 STOR(L)=INP(0)
3050 IF STOR(1)()64 GOTO 3070
3060 L=L+1
3070 OUT 6,48
3080 OUT 0,1
3090 GOTO 3010
3100 PRINT "END OF SCAN "
3110 OUT 6,32
3120 FOR I=0 TO 59
3130 IF STOR(37+15*I)=32 THEN STOR(37+15*I)=48
3140 THERM(I)=10*(STOR(37+15*I)-48)+STOR(38+15*I)-48+(STOR(40+15*I)-48)/10
3150 NEXT I
3160 THERM(6)=99.9:THERM(10)=99.9:THERM(12)=99.9:THERM(13)=99.9:THERM(15)=99.9
3162 THERM(43)=99.9:THERM(59)=99.9
3170 FOR R=9 TO 0 STEP -1
3180 FOR COL=0 TO 4
3190 PRINT 10*COL+R;THERM(10*COL+R),
3200 NEXT COL
3210 NEXT R
3215 PRINT
3220 PRINT "THERMOCOUPLE TEMPS.",
3230 IF C=1 THEN TCIN=THERM(58):TCOUT=THERM(52)
3240 IF C=2 THEN TCIN=THERM(52):TCOUT=THERM(58)
3250 PRINT "IN";TCIN,"OUT";TCOUT
3400 PRINT
3405 PRINT "AMB. TEMPS.";
3410 FOR I=50 TO 57
3412 IF I=52 GOTO 3430
3420 PRINT I;THERM(I);
3430 NEXT I
3435 PRINT
3440 PRINT "ELAPSED TIME =";XTIM,
3480 PRINT STRING$(70,45)
3490 RETURN
3500 I=TCONT:GOSUB 2120
3510 A=T(I)
3520 TDIF=TSET-T(TCONT)
3530 IF ABS(TDIF)<10 GOTO 3570
3540 IF TDIF>0 THEN DRIVE=255 ELSE DRIVE=253
3550 OUT VT,DRIVE
3560 RETURN
3570 IF ABS(TDIF)>.3 THEN GOSUB 2010 ELSE OUT VT,0
3580 RETURN
3590 F=FCONT:GOSUB 2440
3600 FDIF=FSET-FLOW(FCONT)
3610 IF ABS(FDIF)<.01 GOTO 3650
3620 IF FDIF>0 THEN DRIVE=253 ELSE DRIVE=255
3630 OUT VF,DRIVE
3640 RETURN
3650 IF ABS(FDIF)>FTOL THEN GOSUB 2520 ELSE OUT VF,0
3660 RETURN
3670 PRINT "ERROR NO. ";ERR;"AT LINE ";ERL
3680 END

```


APPENDIX D.1THEORETICAL ANALYSIS OF A FULLY-MIXED STORE

Consider a store which is being charged with an inlet temperature T_{in} , and which remains fully-mixed at all times. Under these conditions let the store temperature increase from T to $T + dT$ during the time from t to $t + dt$.

By conservation of energy,

the increase in stored energy = net energy lost by the flow into and out of the store

$$\text{So, } \Sigma M C_p dT = \dot{m} C_p (T_{in} - T_{ex}) dt$$

and $T_{ex} = T$, since at time t the whole store is at temperature, T ,

So from time 0 to τ ,

$$\Sigma M C_p \int_0^{\tau} \frac{dT}{T_{in} - T} = \dot{m} C_p \int_0^{\tau} dt$$

$$\text{Hence } \Sigma M C_p \ln \left| \frac{T_{in} - T_{initial}}{T_{in} - T_{final}} \right| = \dot{m} C_p \tau$$

But by definition (Chapter 4, equation (24)), $\Sigma M C_p = \dot{m} C_p \tau$

$$\text{So, } \ln \left| \frac{T_{in} - T_{initial}}{T_{in} - T_{final}} \right| = 1$$

$$\text{and, } \frac{T_{in} - T_{initial}}{T_{in} - T_{final}} = e. \quad \dots(D.1)$$

$$\text{Now } E_c(\tau) = \Sigma M C_p (T_{\text{final}} - T_{\text{initial}})$$

$$\text{and } E_{\text{max}}(\tau) = \text{TSC} = \Sigma M C_p (T_{\text{in}} - T_{\text{initial}})$$

$$\text{So } \eta(\tau) = \frac{E_c(\tau)}{E_{\text{max}}(\tau)} = \frac{T_{\text{final}} - T_{\text{initial}}}{T_{\text{in}} - T_{\text{initial}}}$$

$$\text{Now from (D.1) } \frac{1}{e} = \frac{T_{\text{in}} - T_{\text{final}}}{T_{\text{in}} - T_{\text{initial}}}$$

$$\text{So that, } 1 - \frac{1}{e} = \frac{T_{\text{final}} - T_{\text{initial}}}{T_{\text{in}} - T_{\text{initial}}}$$

$$\text{Hence } \eta(\tau) = 1 - \frac{1}{e} = 0.632.$$

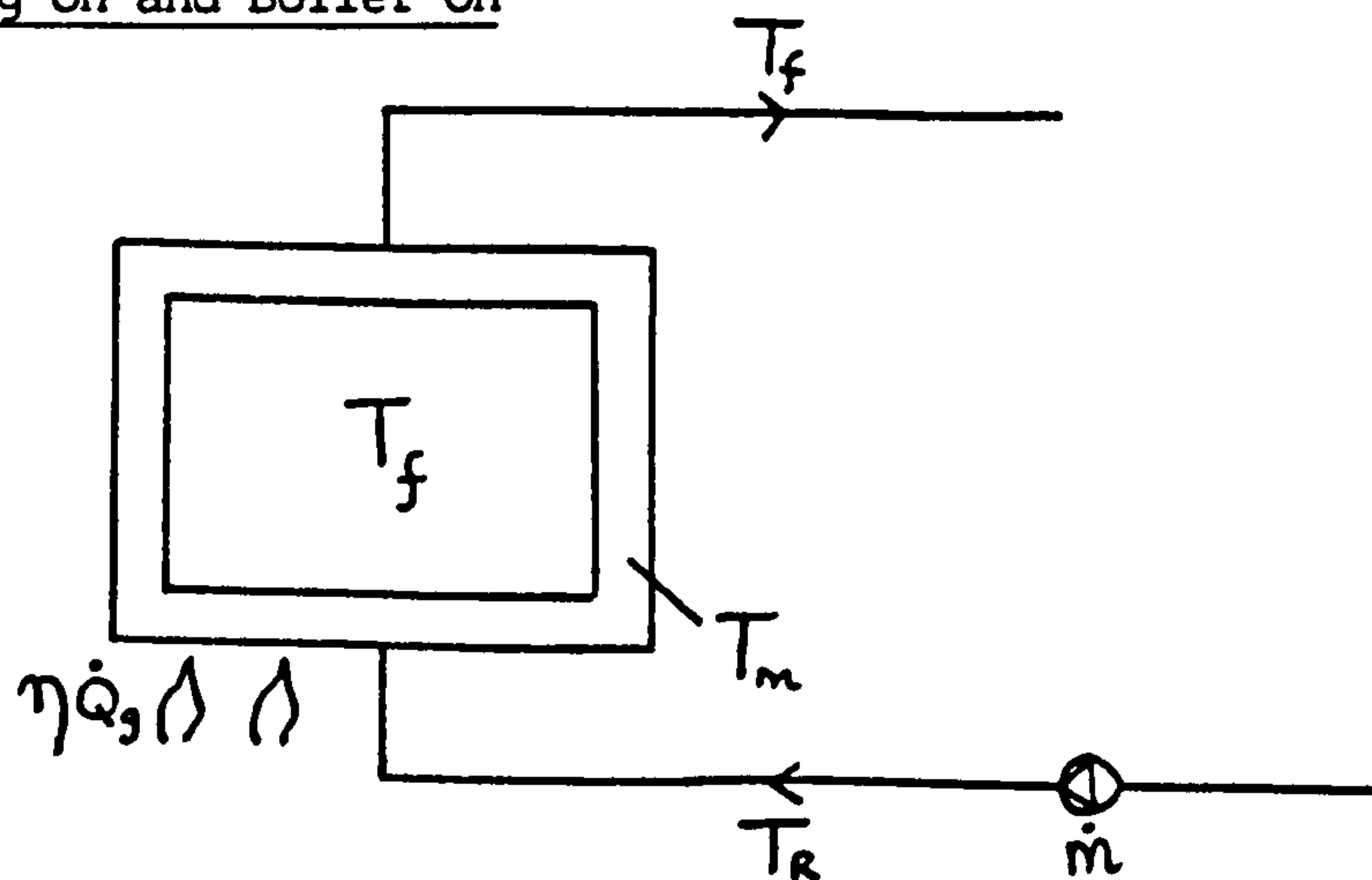
APPENDIX F.1

HEAT TRANSFER EQUATIONS FOR THE SYSTEM SIMULATION MODELS

F.1.1 Conventional System Model

The system comprised 5 boilers (in the base case) in parallel. When there was flow in the system ('heating on') the model assumed it to be split equally between the boilers. The heat transfer in each boiler was simulated independently to produce the outlet temperature for each boiler, and the flow was then recombined, to give a flow temperature to the radiators equal to the mean of the 5 boiler outlet temperatures. The control logic in the model simulated a sequence controller, switching on or off the marginal boiler in order to minimise the difference between the calculated flow temperature and the target flow temperature, obtained from the input data. The equations for one boiler under the various possible operating conditions were formulated as follows :

(a) Heating On and Boiler On



Let the boiler comprise a casing or metal component of thermal capacity $(MC_p)_m$, at a temperature, T_m , and a mass of water of thermal capacity $(m C_p)_w$, at a temperature T_f , which is also the temperature of the flow out of the boiler. Then,

for the metal :

heat gained by metal = heat gained from gas - heat lost to water

$$(M C_p)_m \frac{\partial T_m}{\partial t} = \eta \dot{Q}_g - (UA)_{mw} (T_m - T_f) \quad \dots(F.1)$$

where $\dot{Q}_g$ is the rate of heat output from the gas burner, $(UA)_{mw}$ is the heat transfer rate per degree temperature difference from the metal to the water and η is the efficiency of heat-transfer from the gas's combustion products to the metal of the boiler.

Now η is taken to be related to the boiler's peak efficiency, $\eta_{\max}$, by the following equation :

$$\eta = \frac{1800 - T_R}{1790} \cdot \eta_{\max} \quad \dots(F.2)$$

where T_R is the temperature of the water returning to (i.e. entering) the boiler, and $\eta_{\max}$ is the laboratory-measured full-load efficiency when T_R is a constant 10°C (taken as 77.3% for these boilers). 1800°C in equation (F.2) is supposed to represent the temperature of the gas's combustion products around the metal surface, and (F.2) is construed to yield $\eta = \eta_{\max}$ when $T_R = 10^\circ\text{C}$. The effect of (F.2) is to cause about a 3.4% drop in the boiler full-load efficiency when the return temperature increases from 10°C to 70°C .

In finite-difference form (F.1) becomes :

$$\frac{(M C_p)_m (T_m^{t+1} - T_m^t)}{\Delta t} = \frac{1800 - T_R^t}{1790} \cdot \eta_{\max} \dot{Q}_g - (UA)_{mw} (T_m^t - T_f^t)$$

Hence

$$T_m^{t+1} = \left[1 - \frac{(UA)_{mw} \Delta t}{(M C_p)_m} \right] T_m^t + \left[\frac{(UA)_{mw} \Delta t}{(M C_p)_m} \right] T_f^t + \frac{1800 - T_R^t}{1790} \eta_{\max} \dot{Q}_g$$

$$\text{and for stability } \Delta t \leq \frac{(M C_p)_m}{(UA)_{mw}} \quad \dots(F.3)$$

for the water :

heat gained by water = heat gained from metal - flow heat loss

$$(M C_p)_w \frac{\partial T_f}{\partial t} = (UA)_{mw} (T_m - T_f) - \dot{m} C_p (T_f - T_R) \quad \dots(F.4)$$

where $\dot{m}$ is the mass flow rate of the water through that boiler.
In finite-difference form (F.4) becomes :

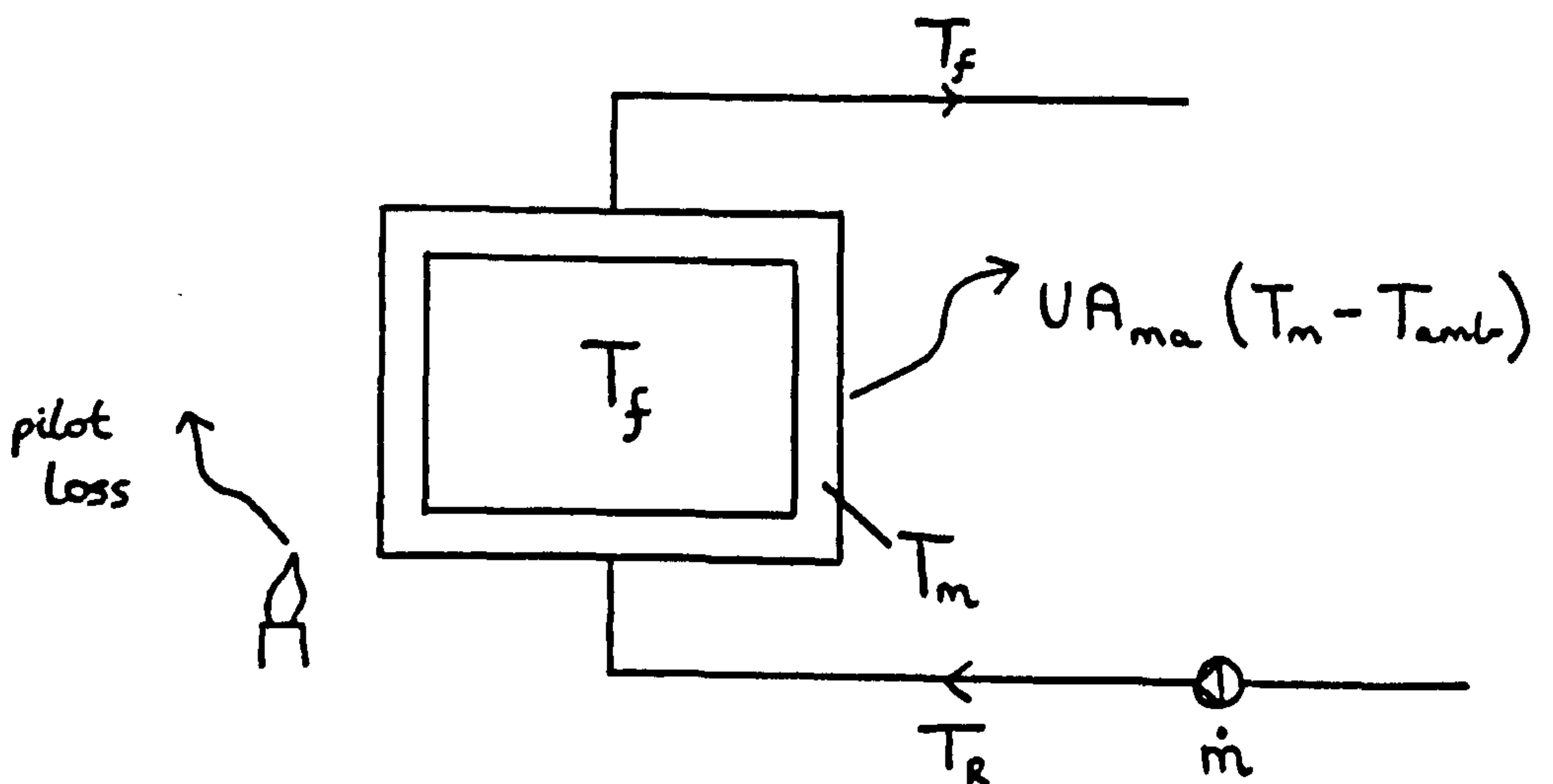
$$\frac{(M C_p)_w (T_f^{t+1} - T_f^t)}{\Delta t} = (UA)_{mw} (T_m^t - T_f^t) - \dot{m} C_p (T_f^t - T_R^t)$$

$$\begin{aligned} \text{Hence } T_f^{t+1} = & \left[1 - \frac{\Delta t}{(M C_p)_w} \left\{ (UA)_{mw} + \dot{m} C_p \right\} \right] T_f^t + \left[\frac{(UA)_{mw} \Delta t}{(M C_p)_w} \right] T_m^t \\ & + \left[\frac{\dot{m} C_p \Delta t}{(M C_p)_w} \right] T_R^t \end{aligned}$$

$$\text{and for stability } \Delta t \leq \frac{(M C_p)_w}{(UA)_{mw} + \dot{m} C_p} \quad \dots(F.5)$$

(b) Heating On and Boiler Off

This simulates the water flowing through a boiler which is not firing.



for the metal :

heat lost by metal = - heat lost to ambient - heat lost to water

$$(M C_p)_m \frac{\partial T_m}{\partial t} = - (UA)_{ma} (T_m - T_{amb}) - (UA)_{mw} (T_m - T_f) \quad \text{..(F.6)}$$

where $(UA)_{ma}$ is the heat transfer rate per degree temperature difference from the boiler to the ambient air, at temperature T_{amb} . $(UA)_{ma}$ is called the boiler flue-loss factor in Chapter 6 and is supposed to take account of heat losses up the flue to the outside air as well as heat losses from the boiler case to the boiler room air.

In finite-difference form (F.6) becomes :

$$\frac{(M C_p)_m (T_m^{t+1} - T_m^t)}{\Delta t} = - (UA)_{ma} (T_m^t - T_{amb}) - (UA)_{mw} (T_m^t - T_f^t)$$

Hence

$$T_m^{t+1} = \left[1 - \frac{\Delta t}{(M C_p)_m} \left\{ (UA)_{mw} + (UA)_{ma} \right\} \right] T_m^t + \left[\frac{(UA)_{mw} \Delta t}{(M C_p)_m} \right] T_f^t + \frac{(UA)_{ma} \Delta t T_{amb}}{(M C_p)_m}$$

$$\text{and for stability } \Delta t \leq \frac{(M C_p)_m}{(UA)_{mw} + (UA)_{ma}} \quad \text{..(F.7)}$$

for the water :

The equations are identical to those for the water with the boiler on, e.g. (F.4).

for the pilot light :

The pilot light is assumed not to effect the temperatures in the boiler and is simply treated as lost heat.

(c) Heating Off and Boiler Off

This simulates night-time, weekends, etc. Because these periods are so long, the boilers are assumed to lose all their heat to ambient. This assumption was supported by observation of the monitored data. The total heat lost, Q_{loss} , by a boiler is therefore given by

$$Q_{loss} = (M C_p)_m (T_m^t - T_{amb}) + (M C_p)_w (T_f^t - T_{amb})$$

where t is the time the heating is switched off.

During this period the pilot light is again taken as lost heat.

F.1.2 Stability of the Conventional System Model

As shown above, the maximum time step for stability depends on the thermal properties of the boiler in question, i.e. a Hamworthy R320. These properties had been measured by British Gas in their laboratory [Ref.6.6] with the following results :

$$\begin{aligned} (M C_p)_w &= 115 \text{ kJ/}^\circ\text{C} \\ (M C_p)_m &= 105 \text{ kJ/}^\circ\text{C} \\ (UA)_{mw} &= 1140 \text{ W/}^\circ\text{C.} \end{aligned}$$

As discussed in Section 6.4.5, it was decided to take

$$(UA)_{ma} = 116 \text{ W/}^\circ\text{C.}$$

Inserting these values in equations (F.3), (F.5) and (F.7) produces the following stability criteria :

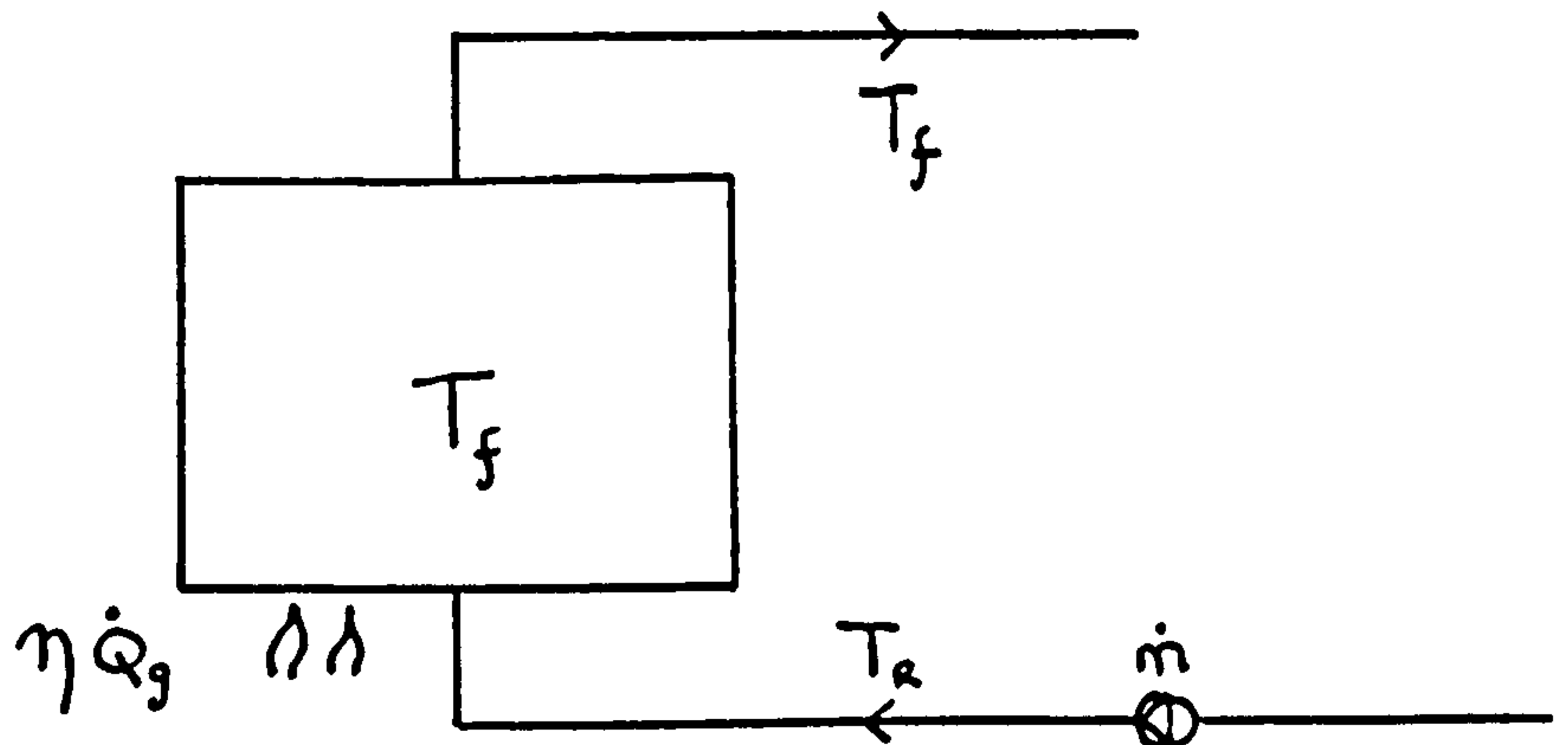
$$\begin{aligned} \Delta t &\leq 92 \text{ secs} && \text{for the boiler metal (boiler on)} \\ \Delta t &\leq 14 \text{ secs} && \text{for the boiler water} \\ \text{and } \Delta t &\leq 84 \text{ secs} && \text{for the boiler metal (boiler off).} \end{aligned}$$

The simulation therefore adopted a time step of 14 seconds for the periods when the heating was on. This incidentally meant that there were four time steps in the period during which the simulated sequence controller was prevented from changing the number of boilers that were firing (see Section 6.4.4).

F.1.3 Storage System Model

The algorithm for the boilers in this case could be made much simpler than in the conventional system model. This is because : (a) all of the boilers operated together (i.e. they were either all on or all off) and performed identically, so that it was only necessary to model one boiler; and (b) the long periods of operation in the 'on' mode or 'off' mode meant that differences in temperature between the metal and water were irrelevant and these two components could be 'lumped' together. The differential equations for a boiler under the two possible operating conditions were formulated as follows :

(a) Boiler On



heat gained by boiler = heat gained from gas - flow heat loss

$$(M C_p) \frac{\partial T}{\partial t} = \eta \dot{Q}_g - \dot{m} C_p (T_f - T_R) \quad \dots (F.8)$$

where $(M C_p) = (M C_p)_m + (M C_p)_w$.

In finite-difference form (F.8) becomes :

$$(M C_p) \cdot \frac{(T_f^{t+1} - T_f^t)}{\Delta t} = \frac{1800 - T_R^t}{1790} \cdot \eta_{\max} \dot{Q}_g - \dot{m} C_p (T_f^{t+1} - T_R^t) \quad \dots(F.9)$$

Hence

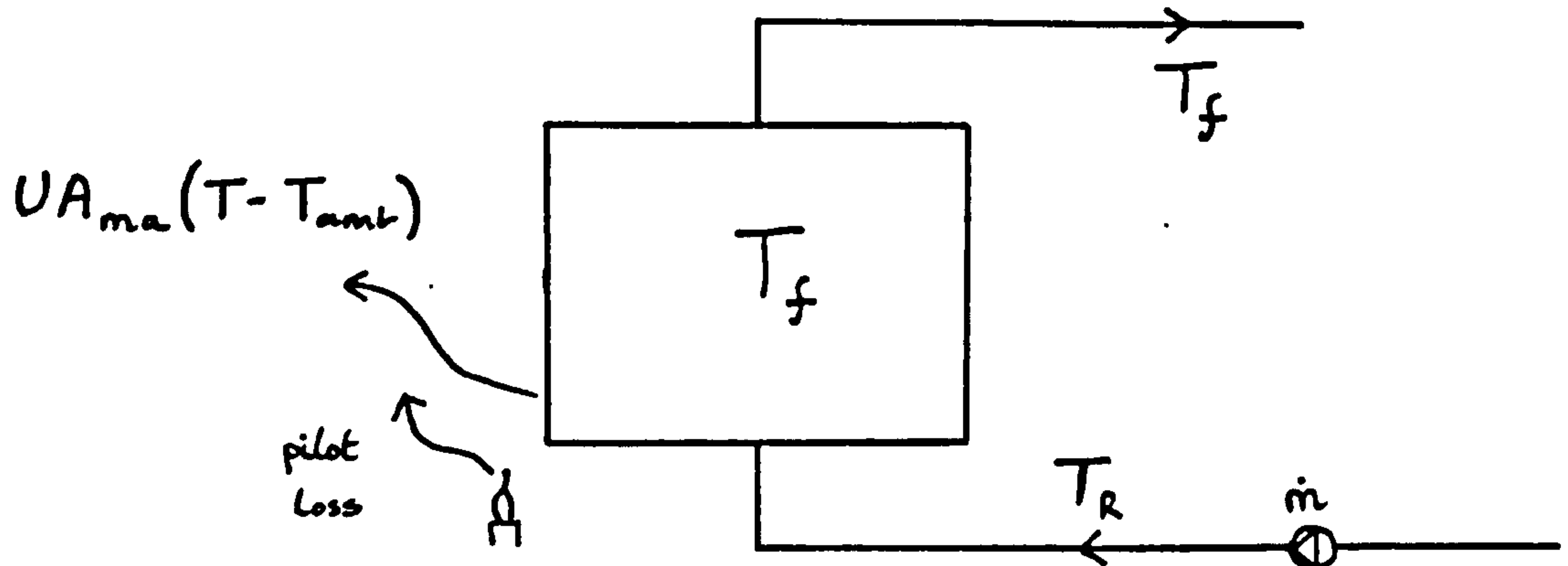
$$T_f^{t+1} = \left[\frac{1}{1 + \frac{\dot{m} C_p \Delta t}{(M C_p)}} \right] T_f^t + \frac{\Delta t / (M C_p)}{1 + \frac{\dot{m} C_p \Delta t}{(M C_p)}} \cdot \frac{1800 - T_R^t}{1790} \cdot \eta_{\max} \cdot \dot{Q}_g$$

$$+ \left[\frac{\frac{\dot{m} C_p \Delta t / (M C_p)}{1 + \frac{\dot{m} C_p \Delta t}{M C_p}}}{1 + \frac{\dot{m} C_p \Delta t}{M C_p}} \right] T_R^t$$

so that

$$T_f^{t+1} = \frac{(M C_p) T_f^t + \Delta t (1800 - T_R^t) \eta_{\max} \dot{Q}_g / 1790 + \dot{m} C_p \Delta t T_R^t}{(M C_p) + \dot{m} C_p \Delta t}$$

This formulation has the characteristic of being stable for all values of the time step, Δt . This is very important for the model, since the store algorithm generates a variable time step determined by the flow rate through the store (see Section 5.1.3 and Equation (5.14)). The essential difference between this formulation and the one employed in the conventional system model is that, during the first time step, part of the heat input from the gas goes into the flow water and part into the boiler, instead of it all going into the boiler.

(b) Boiler Off

heat lost by boiler = - heat lost to ambient

$$(M C_p) \frac{\partial T}{\partial t} = - (UA)_{ma} (T - T_{amb})$$

which in finite difference form becomes :

$$(M C_p) \frac{(T_f^{t+1} - T_f^t)}{\Delta t} = - (UA)_{ma} (T_f^t - T_{amb})$$

Hence

$$T_{f..}^{t+1} = T_f^t + \frac{(UA)_{ma} \Delta t}{(M C_p)} (T_f^t - T_{amb})$$

$$\text{and for stability } \Delta t \leq \frac{(M C_p)}{(UA)_{ma}^p} \quad \dots (F.10)$$

Inserting the measured values into (F.10) gives $\Delta t \leq 32$ minutes. In fact the store algorithm was construed to ensure a maximum Δt of 5 minutes (i.e. a minimum store flow rate of 0.33 litres/sec, since each store zone had a volume of 100 litres).