

CRANFIELD INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL ENGINEERING

Ph.D. THESIS

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SOME ASPECTS OF THE USE OF  
WATER - FILLED HEAT STORES IN  
GAS-FIRED CENTRAL-HEATING SYSTEMS

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April 1986

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THE SPINE**

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## ABSTRACT

Water-filled heat stores present a convenient, relatively inexpensive means of optimising the use of diminishing gas stocks for the central-heating of buildings. The British Gas Corporation recently launched a series of central-heating units with storage, for use in the domestic sector, whose benefits include:-

- reduced boiler size,
- more efficient boiler operation,
- load-levelling at the hours of peak gas demand.

This thesis is divided into three parts. Part I examines the inherent advantage of a with-storage, domestic, central-heating system over a conventional system, by means of two simple computer-simulation programs. A minimum efficiency advantage of about 5% is anticipated; the variation of this advantage with the values of certain key parameters has been assessed. Part II is an interim report of a full-scale field trial in the commercial sector; a large ( $3.3\text{m}^3$ ) store was fitted in the heating system of a London school, and its performance during the first weeks of its operation is presented here. Returning to the domestic sector, Part III presents a study of the use of two integral heat exchangers in the storage vessels of the above domestic units, whereby hot water can be drawn instantaneously. An attempt to optimise this domestic hot-water facility has been made.



## ACKNOWLEDGEMENTS

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At British Gas (Watson House), to Dr. D. Nevrala for guidance of the work throughout; to S. Gataora, G. Menzies, G. Wright and especially to N. Gibson for tireless support during the Spa School field trial, and of course to the British Gas Corporation itself for financial support.

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Lastly, to a series of friends at Russell Park Church in Bedford, notably Theo Willcocks, for support in the final stages. And to Barbie Gooch for understanding and encouragement.

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## GENERAL INTRODUCTION

### 1. Preamble

There is heated debate at present as to the course that future energy production (or conversion) should follow. The "hard path" (ref.I.1) favours a high nuclear contribution, whereas its "soft" alternative advocates the harnessing of increasing amounts of renewable energy. Compared with the current domination of fossil-fuel combustion, both courses have considerable advantages and disadvantages.

The nuclear option avoids the emission problems of the fossil-fuels, and can offer electricity and heat economically. It is evident, though, that the problem of nuclear waste disposal has yet to be resolved satisfactorily. In the longer term, uranium reserves are finite, and the advent of fusion technology is not likely to occur until well into the next century.

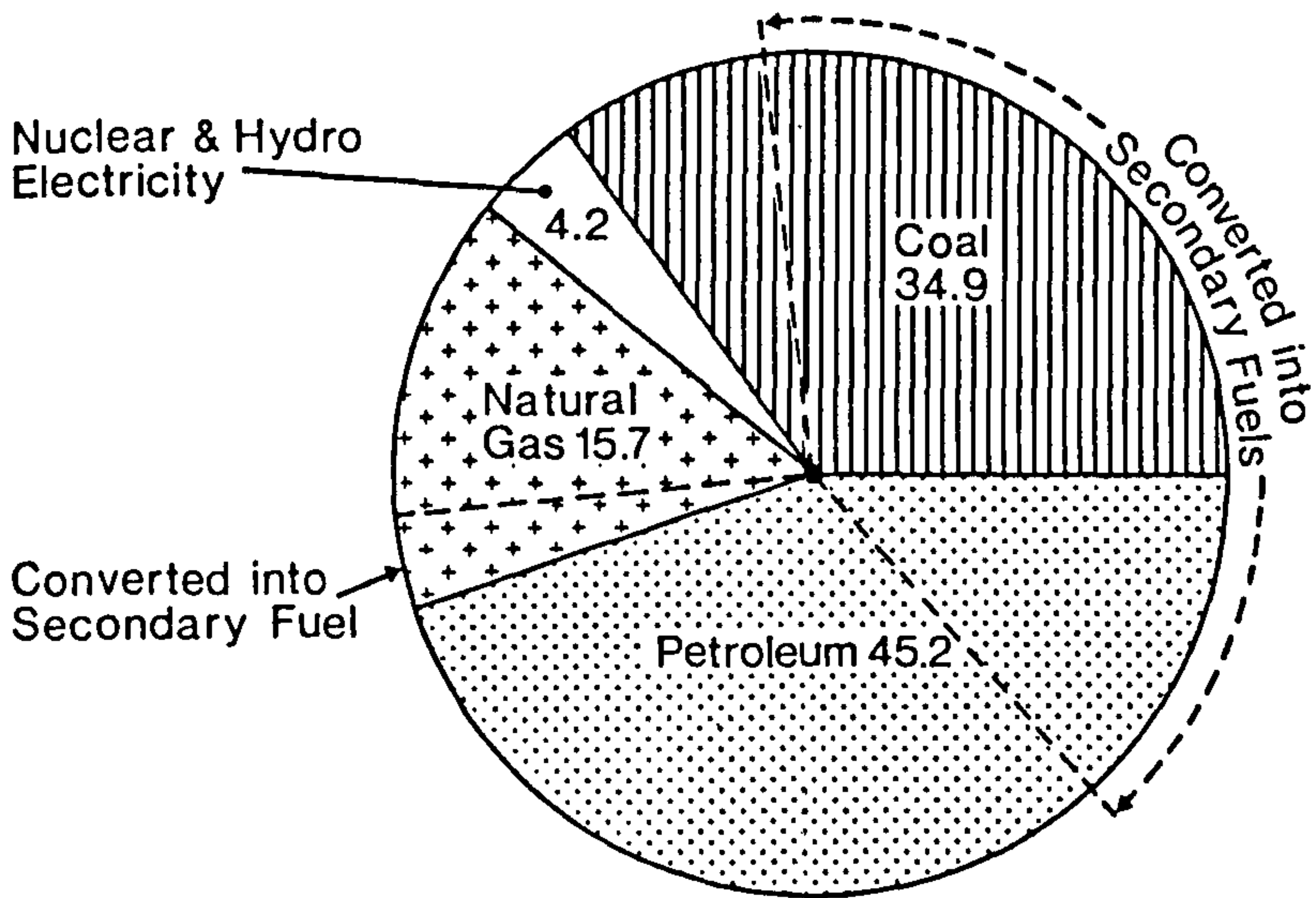
Renewable energies are either direct solar radiation or solar energy in an indirect form (wind, hydroelectric, etc.). Dr. O'Callaghan (ref.I.2) suggests that 0.009% of the solar radiation incident on the Earth, if suitably harnessed, could replace the current fossil-fuel consumption. Solar-derived energies are therefore plentiful, clean and relatively inexhaustible, with minimal adverse effects on the environment. The technologies required for their widespread utilisation, however, are in their infancy.

The advocacy of these contrasting future "scenarios" will continue to be discussed; the entry of the debate into world politics has long since been accomplished, and there is thus little expectation that the respective arguments will be presented with any reduction of acuity (or any increase in disinterested reason). It is, however, universally acknowledged that the profligate use of fossil-fuels which existed until 1973 (and which has since been checked somewhat) can never again be contemplated. At the current rate of consumption, the world's oil and gas reserves will be severely depleted in two or three decades, though coal stocks will last considerably longer. The dependence of



**1974 - 337.5 MILLION  
TONNES COAL  
EQUIVALENT**

PERCENTAGE SHARES



**1984 - 312.2 MILLION  
TONNES COAL  
EQUIVALENT**

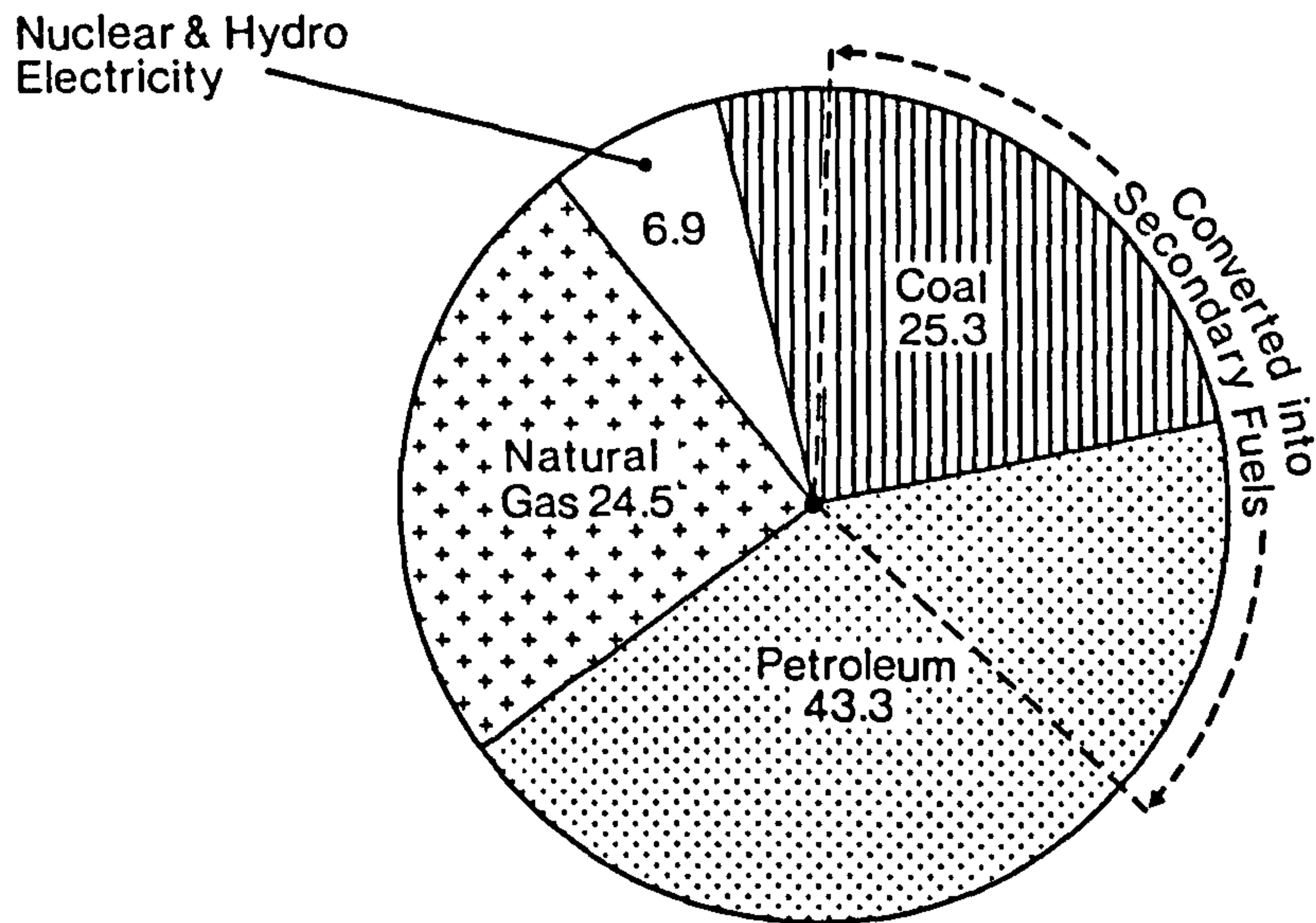


FIG I.1: RECENT FOSSIL-FUEL DEPENDENCE IN U.K.

(FROM REF I.4)

the U.K. on fossil-fuels is shown on Fig.I.1. As the alternative energy sources (whichever are adopted eventually) are progressively called upon to relieve the current dependence on fossil-fuels, the energy industries must direct their attention to the rational use of the latter. Though political decisions will doubtless cause market fluctuations (the current low oil price is due to O.P.E.C.'s reduction from 31.7 million barrels per day in 1979 to 18.6 in 1984, ref.I.3), conservation must remain the watchword, and the implemented policy.

## 2. Gas and the Heating of Buildings

The context of this study is the heating of buildings, uniquely from the point of view of the Gas Industry in the U.K.

Since the advent of North Sea "natural gas", the portion of the total U.K. inland energy consumption met by natural gas rose from less than 2% in 1968 to nearly 25% in 1984, see Fig.I.2. Due to its very competitive unit price, an increasing share of the market is anticipated. Ref.I.4, however, suggests that total recoverable reserves of natural gas originally in place on the U.K. continental shelf might be as little as  $1380 \cdot 10^9$  cubic metres. The 1984 consumption of natural gas was some  $78 \cdot 10^6$  t.c.e.; if the same rate were maintained, the above reserve would be exhausted by about the year 2010 A.D. It is evident that gas stocks must be used very cautiously from the present moment; the British Gas Corporation is therefore making efforts to economise on its use.

From a recent publication at Cranfield, Fig.I.3 shows the breakdown of all energy consumed in the U.K. in 1984. 52% is supplied for use in buildings, well over half of which is required for space and water heating. Of the natural gas marketed in 1984, almost half was supplied direct to domestic consumers, predominantly for heating purposes. (All the above figures have been drawn from ref.I.4.) With the heating of buildings being the major end-use of gas in the U.K., it is therefore important that (i) buildings (and their hot-water systems) are designed rigorously for energy economy, and (ii) gas burning devices are run at their optimal efficiencies.

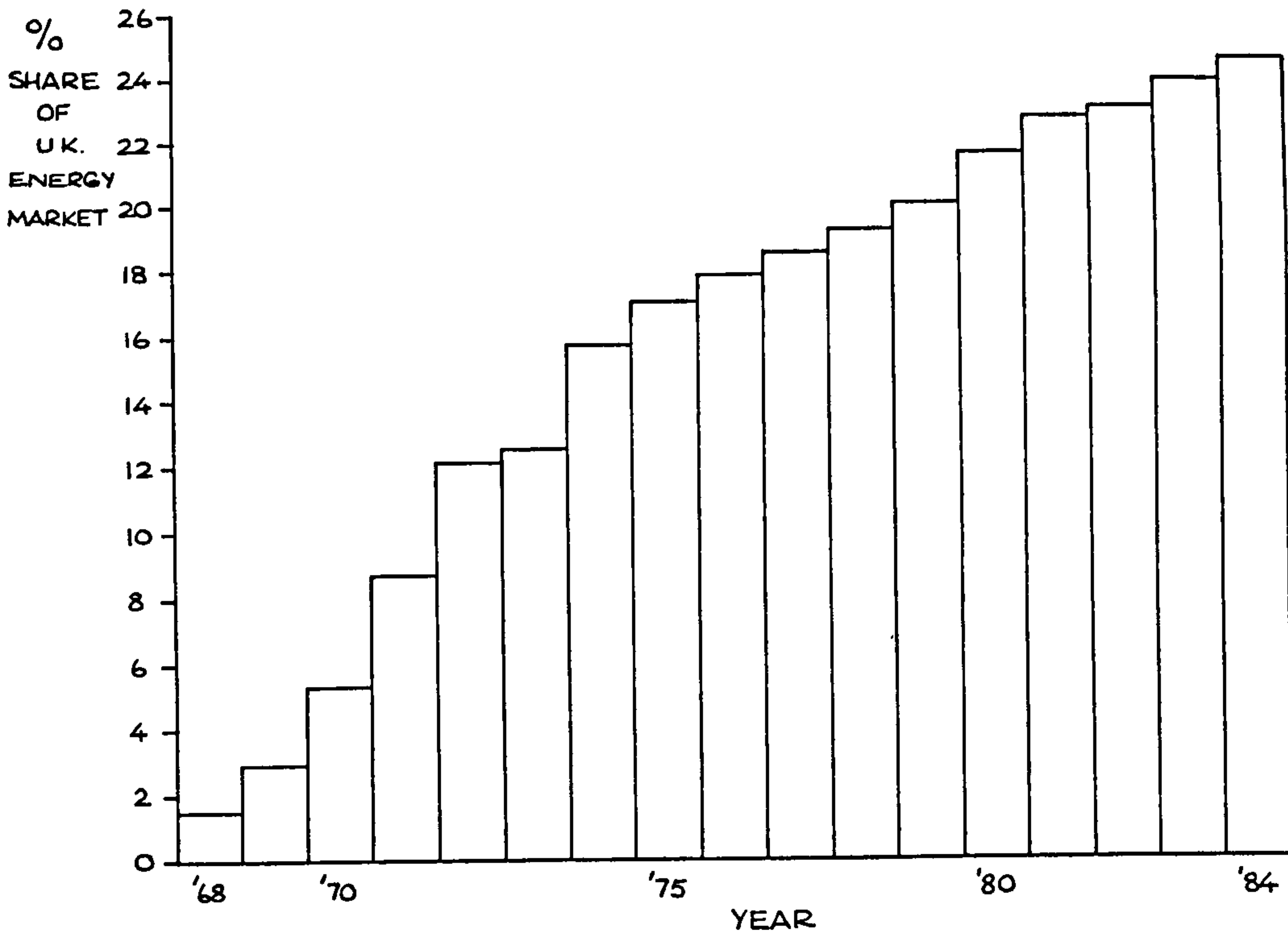


FIG I.2 : NATURAL GAS MARKET PENETRATION (U.K.)

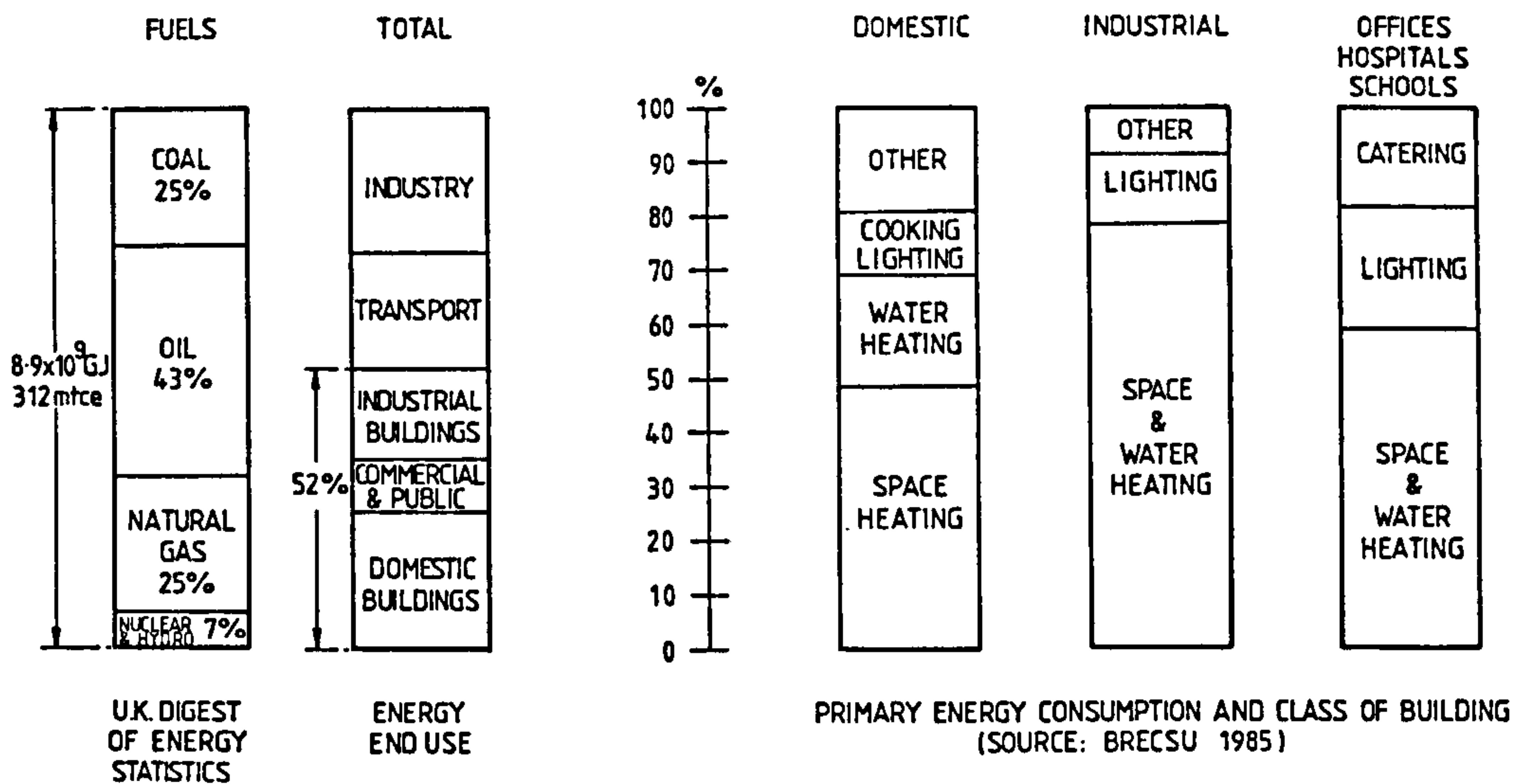


FIG I.3 : BREAKDOWN OF U.K. ENERGY CONSUMPTION



(i) There are now many means available to the designer of a new building whereby he is able to reduce radically the rate of energy consumption (by insulation, passive solar design, etc.) compared with traditional structures. Two-storey houses with a 3kW design heat-load (D.H.L.) are now achievable. Indeed, the "superinsulation" now used in Canada and Sweden can virtually eliminate the need for a space heating plant, even with ambient temperatures substantially below zero Celcius. "Retrofit" economy measures for existing buildings offer less substantial rewards, but are nevertheless necessary such that government aid is available for such improvements. In spite of the U.K. government's national target of a 20% energy saving by the end of the century, Fig.I.4 shows that the U.K. remains well down the league of European domestic energy thrift, (ref. I.5).

(ii) This study deals with the other half of the measures pertinent to building energy economy, namely the efficient use of natural gas supplied to the boiler plant, and is principally concerned with domestic heating and hot-water production, although a study was undertaken in the commercial sector, as will be seen.

### 3. Efficiency of Gas-Fired Boilers

In late 1982, at the start of the study covered by this thesis, G.Pickup of the British Gas Corporation (Watson House) issued a paper delineating the "state of the art" in the development of gas-fired heating appliances, ref. I.6 . From this paper, the following salient features have been drawn.

(i) Conventional Boilers: In the twenty years or so from 1960, the full-load efficiency of domestic central-heating boilers increased steadily from well below 70% to about 80%. At the same time the number of centrally-heated dwellings in the U.K. increased from a negligible number to about nine million. Pickup emphasized that an efficiency of 80% for a modern, lightweight boiler was likely to be near the achievable limit for a highly-developed, conventionally-designed unit for the following reasons:-

- the majority of appliances being of a natural-draught

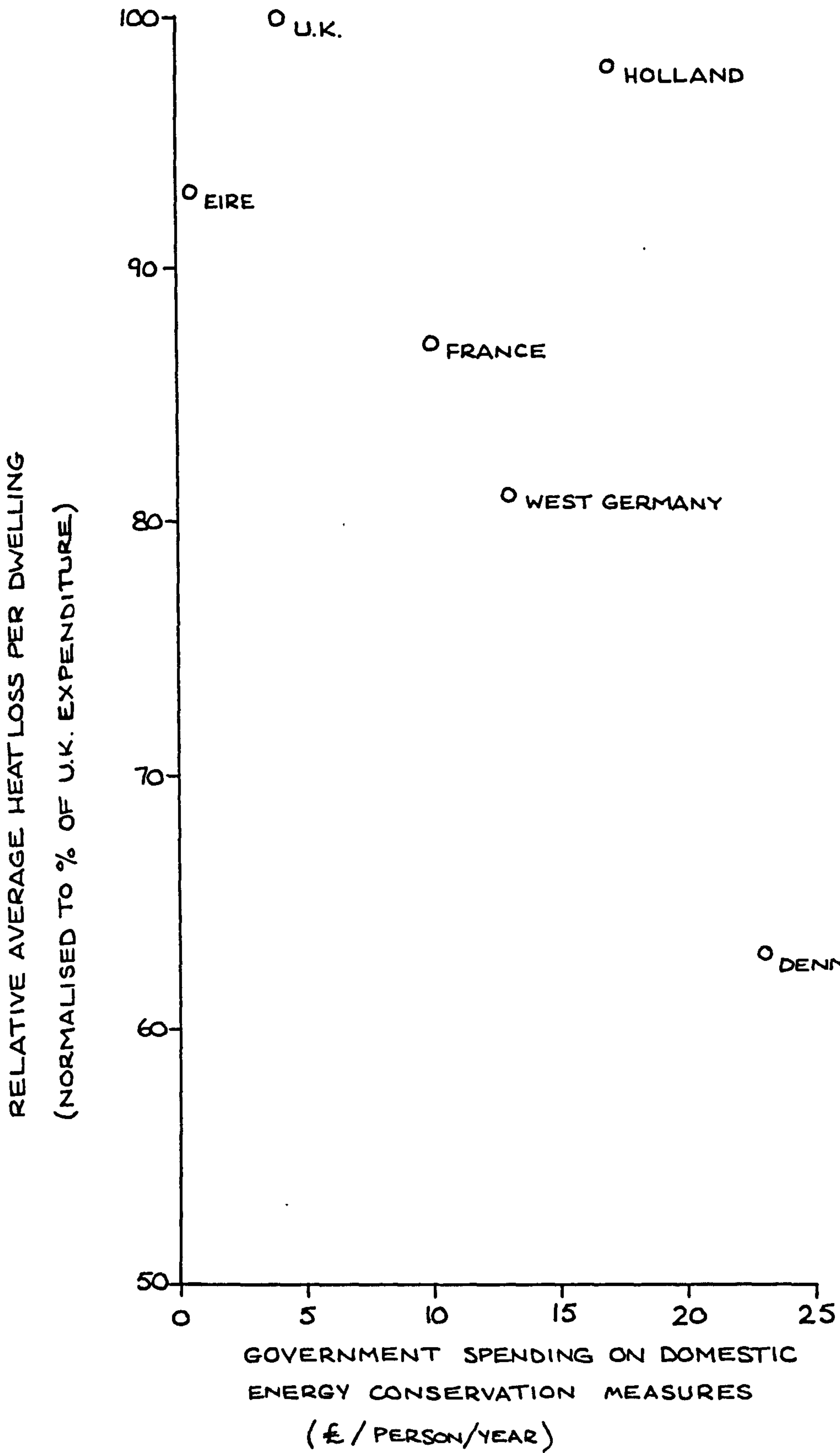


FIG 14 : EUROPEAN DOMESTIC ENERGY CONSERVATION



principle, a quantity of heat is required to pass into the flue to achieve a buoyancy-driven evacuation of the combustion products,

- any further increase in efficiency is likely to provoke an acidic condensation in the flue, and heat exchanger materials are usually not sufficiently corrosion-resistant, and
- the general, vertical disposition of these boilers would allow condensate to fall onto the heat exchanger and burner, and eventually escape from the bottom of the appliance.

In any application, a boiler of even 80% full-load efficiency will have an operating efficiency below this figure, due to its unavoidable reduction at part-load, though this reduction is commendably small for the most efficient devices (e.g. lightweight, balanced-flue, damped-flue, etc.) To begin to use this combustion loss (i.e. over 20% of gas energy input), therefore, a new departure in boiler plant design was necessary. Two such departures are currently undergoing development - the condensing boiler, and the use of heat storage.

(ii) Condensing Boilers: By passing the flue gases downwards over a suitable heat exchanger through which the returning water flows, water vapour in the flue can be made to condense and deliver its latent heat to the incoming water. The cooler is the return flow, the greater is the fraction of the more-than-20% of gas energy (hitherto wasted) which can be recovered. Thus the efficiency of a condensing boiler increases as the return water temperature falls. What is more, the downward flow of the flue gas, together with a damper (usually), greatly impedes the passage of heat out of the flue after boiler shut-down (a.s.d.). The condensing boiler therefore offers an efficiency increase under part-load operation, an altogether radical advantage over the condensing boiler.

Even when operating in the non-condensing mode, i.e. at high return temperatures, the extra heat exchanger can boost the boiler's efficiency to 85 - 90%. Under fully condensing conditions, the efficiency can rise to as high as 98%. It would appear at face value, therefore, that the efficiency problem is thereby overcome, or

potentially so. There remain, however, a number of obstacles before the condensing boiler is widely adopted.

The acidic condensate was considered likely to corrode both the boiler's working parts, and the domestic effluent pipe. Again, however, the corrosion resistance of the boiler can be raised satisfactorily (at a cost), and Watson House has found that the alkalinity of sewage is more than enough to neutralise the condensate.

The cost (and size) of condensing boilers remains comparatively high, though continuing development should overcome these problems.

Clearly the condensing boiler is best suited to low temperature operation, which implies a large radiator area. Again, Pickup states that conventional radiator systems are already oversized for the vast majority of a typical heating season, and quotes a French study which concluded that large radiator systems gave only a marginal increase in efficiency.

With potential efficiency gains of over 15%, a successful future for the condensing boiler appears assured, though the U.K. has been slower to accept it than has occurred in Europe.

Although the condensing boiler is mentioned briefly later in this study (see "Parameter Sensitivity Study" of Part I below), the thesis will henceforth deal solely with the second above-mentioned means of boiler plant optimisation, namely heat storage.

(iii) A Heat Store with a Gas-Fired Boiler: A store, in general, is a method of decoupling supply and demand. Examples are rife, ranging from a domestic hot-water tank to a bank account. Energy can be stored in any of its forms:-

- Kinetic energy by means of e.g. a flywheel
- Potential " mountain reservoir
- Chemical " combustible fuel
- Electrical " accumulator, etc.,etc...



In the context of the heating of buildings, the "Trombe wall" is an ingenious means of storing solar heat (i.e. the supply) until it is required at night (i.e. the demand). For water-filled central heating systems, however, thermal energy can be stored as "low temperature" heat, using a store for either sensible heat or phase-change heat.

The use of encapsulated phase-change materials (P.C.M.) can reduce the required store volume to below a half of that required for a water store. Recent studies, undertaken at Cranfield, on P.C.M. storage are described in references I.7, I.8, and I.9.

This study deals entirely with the use of water stores, likely to be the cheapest and most convenient means of storing heat in a gas-fired central-heating system. The basic tenets of the storage principle are as follows. If a suitable water store is interposed between a building's gas boiler and its radiator system, the three fundamental potential advantages of such a system are:-

#### (a) Load-Levelling

The boiler now heats the store, not the building. The peaks and troughs of any building's heat demand can be met from a suitably-sized store, which can be replenished independently by the boiler. If storage were widely adopted among natural gas users, then the network's peak demands (at about 08.00 and 18.00 hours) could be reduced, and more customers supplied before the network became saturated.

#### (b) Reduced Boiler Size

As the boiler is no longer required to meet such high demands kilowatt-for-kilowatt, its rated output can be smaller than for a non-storage system. The amount of reduction depends on the energy capacity of the store.

#### (c) More Efficient Boiler Use

The boiler of a conventional central-heating system will be sized

to meet the building's maximum demand (at least), and will thus be grossly oversized for most of a heating season. To satisfy ordinary heat demands, a domestic boiler therefore cycles some 100 - 300 times per day, losing a percentage of its heat content after every switch-off. Hence, the boiler of a conventional system operates on a part-load characteristic similar to Fig.I.5. Characteristics of the form of Fig.I.5 are usually published for any available boiler, supposing multi-switching operation at part-load. A simple boiler computer model was set up to examine boiler efficiency over a range of load-factors and switch frequencies (independently). It was readily shown that Fig.I.5 could be expanded to Fig.I.6 if the frequency domain was investigated. As frequency falls, the part-load curve is seen to separate from the multi-switching curve (or asymptote) at a frequency determined by the boiler parameters.

The boiler of the equivalent storage system, as will be seen later, might have some 0.6 of the rated power of the conventional system's boiler, and cycle less than five times per day. Thus the boiler of the conventional central-heating system might return a heating season overall efficiency at "A" of Fig.I.6. Its replacement is likely to operate

- at a proportionally higher load-factor (50% in the place of 30%), and

- at a radically lower switch frequency.

The combination of these two effects will give a heating season at "B" on Fig.I.6, reflecting an inherent advantage of the storage concept. This reasoning is central to the use of storage to optimise boiler operation, and will be called upon repeatedly throughout the thesis.

#### 4. A Domestic Boiler/Store Unit with an Integral Hot-Water Facility

In 1984, 30% of new dwellings had a D.H.L. of 3kW or less. Domestic hot water (d.h.w.) requirements are thus becoming a very significant part of a low-energy dwelling's heat load, ref.I.10.

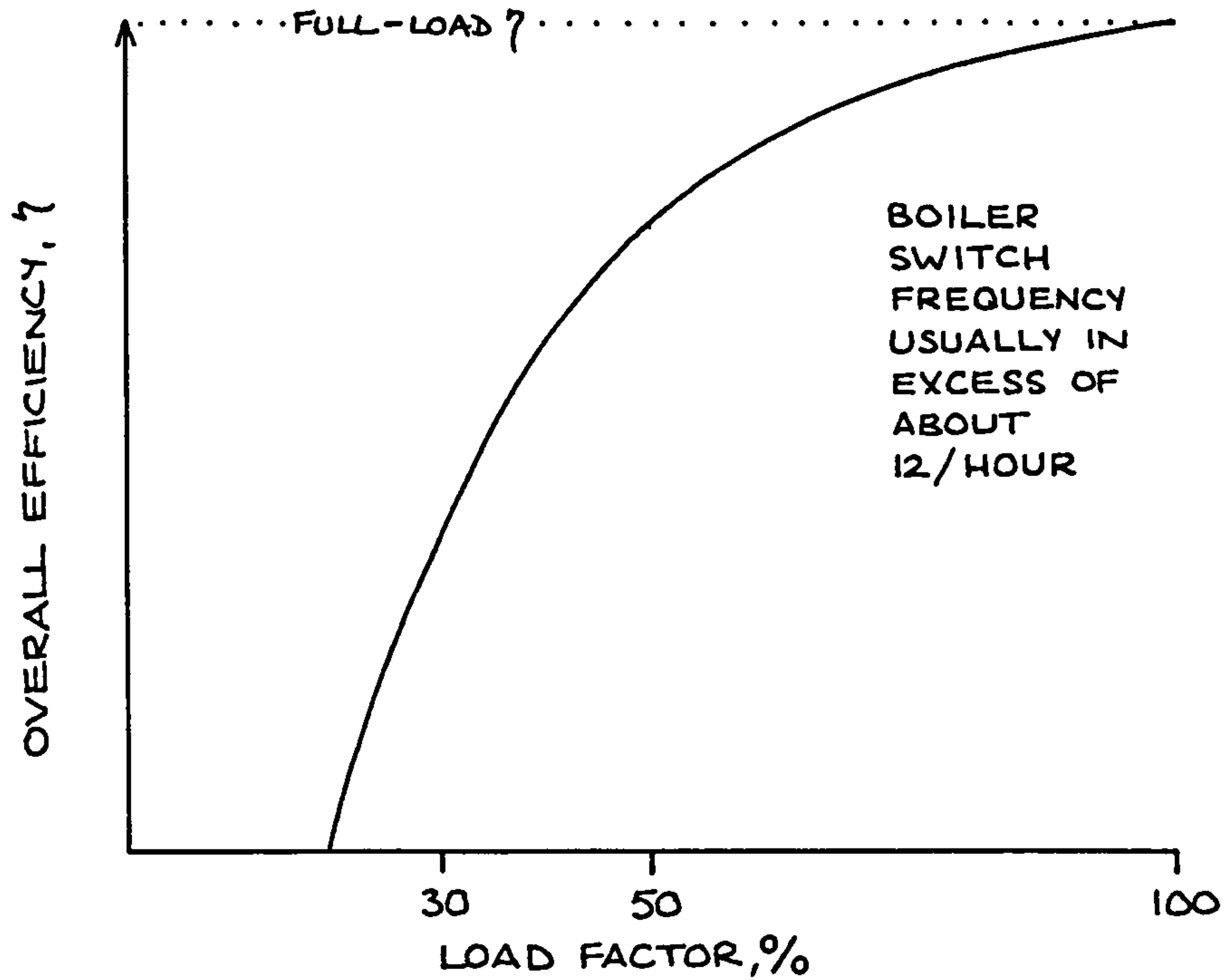


FIG. I.5: CUSTOMARY PART-LOAD BOILER EFFICIENCY CURVE, ASSUMING HIGH SWITCH FREQUENCY.

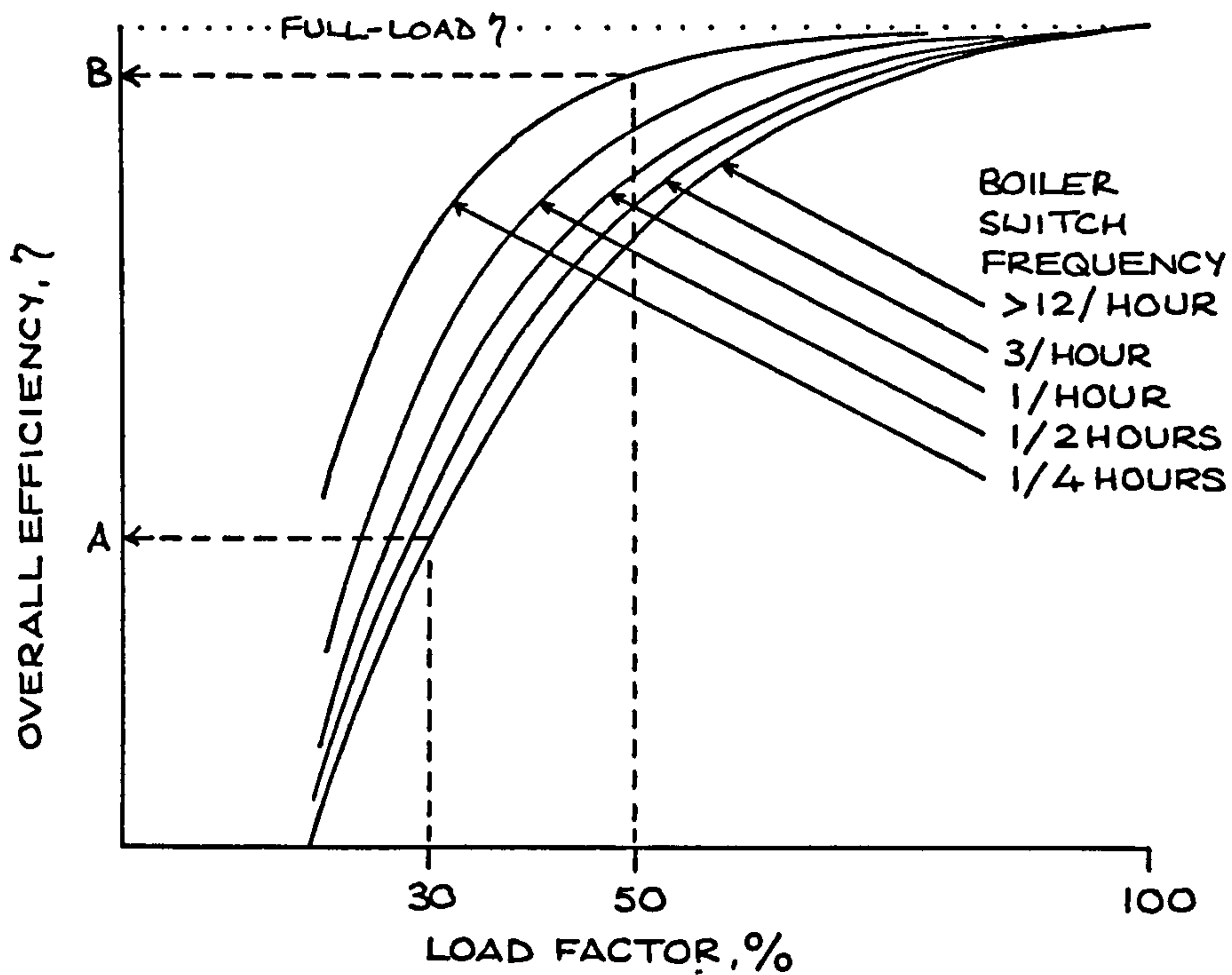


FIG. I.6: EFFECT OF LOW SWITCH FREQUENCY ON EFFICIENCY FOR BOILER OF FIG. I.5.



In 1982, at the start of the period covered by this thesis, the British Gas Corporation, in conjunction with Gledhill Water Storage Ltd., had fabricated prototype domestic boiler/water store combinations which contained (in the store) a heat exchanger for hot water (tap) use. These prototypes were eventually launched under the trade name "Cormorant". The boiler and radiator circuits are connected independently to the store, see Fig.I.7. Hot water for the taps is generated by passing the cold mains supply directly to the heat exchanger(s) arranged inside the store. Initially, large heat exchangers made from one-inch copper piping were used (see Part I below). Eventually, these were replaced by exchangers made from "Integron" material, a finned copper piping with greatly enhanced heat exchange properties, (see Part III).

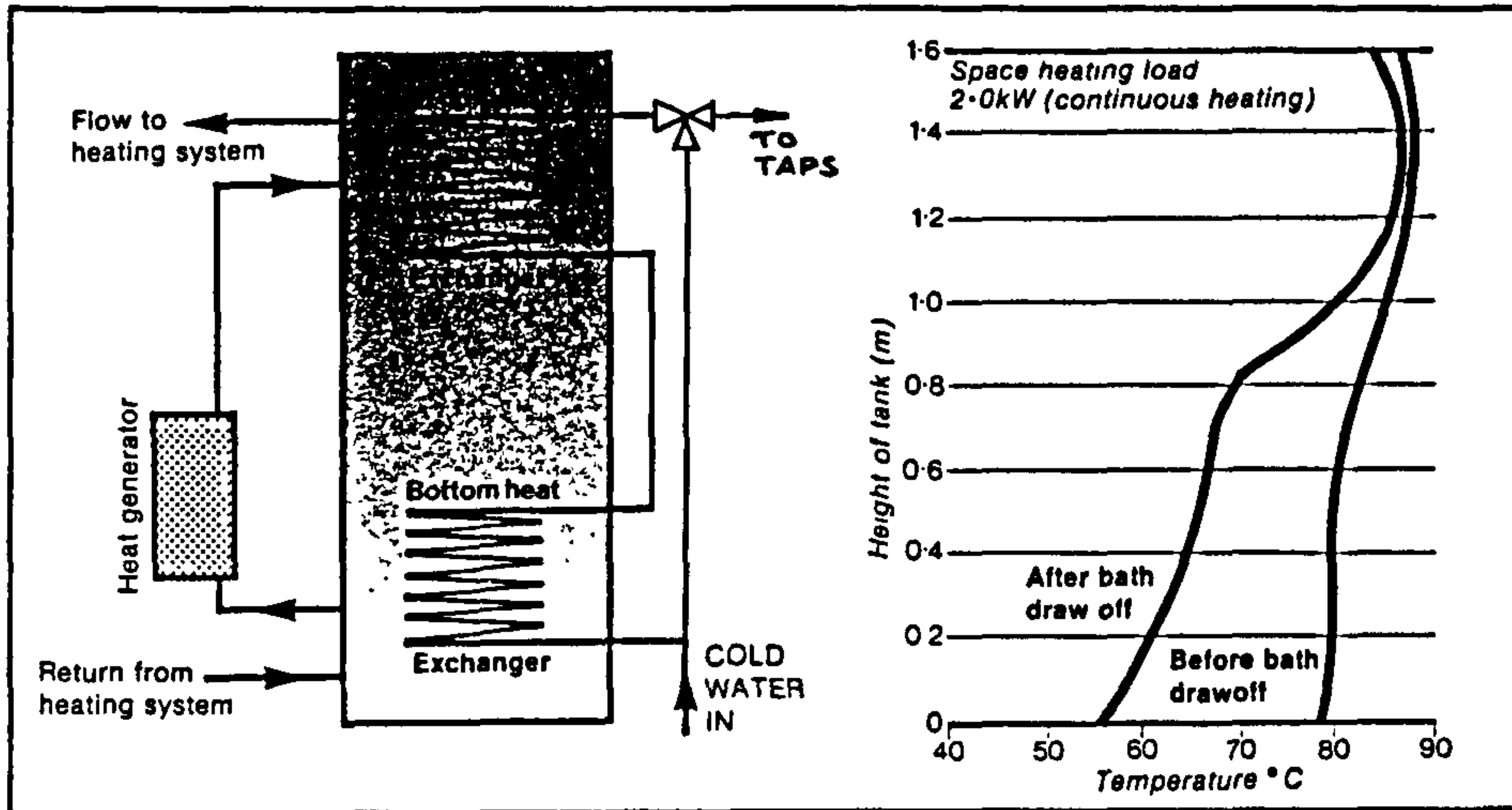
Such an arrangement has the following advantages over a conventional domestic central-heating system:-

- the boiler/store combination offers the gas-usage benefits outlined in section 3.(iii) above i.e. load-levelling, associated with a small boiler unit (many of only 4kW rated input have been installed), together with the high efficiency of such a combination,

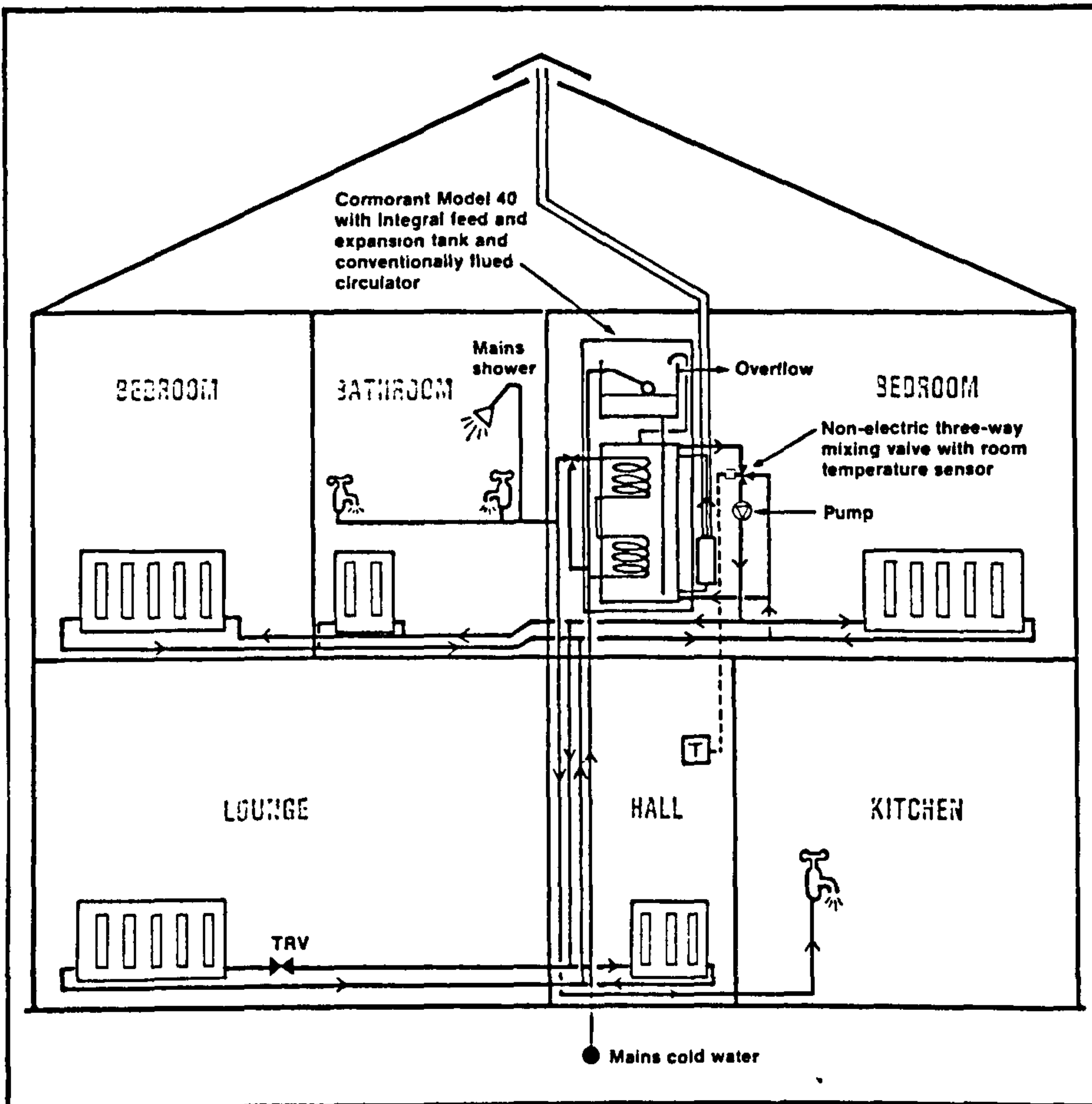
- the cold mains feed into the d.h.w. circuit obviates the need for a separate hot-water tank,

- the above two advantages imply that there is no longer the need for any installation in the loft space of the dwelling, thus reducing the cost of piping and labour, etc.,etc., see Fig.I.8.

Preliminary tests conducted by British Gas, Watson House, indicated that the Cormorant system, as well as being simple to install, operated at least as effectively as its conventional counterpart, and that the boiler rated output could successfully be reduced to a value considerably below the D.H.L. of the dwelling in question. For example, a storage unit powered by a 4.5kW boiler satisfied the requirements (space heating and d.h.w.) of a house of 8.4kW D.H.L. (The equivalent conventional system would doubtless incorporate a boiler rated higher than 8.4kW.) The Cormorant concept



FIGI.7: THE "CORMORANT" INTEGRATED D.H.W. STORAGE UNIT



FIGI.8: DOMESTIC C.H. ARRANGEMENT WITH "CORMORANT"

offered, therefore, considerable potential. It was in such a context that the work reported in this thesis was set underway.

## 5. Study Covered by the Thesis

The principal domain of this investigation has been the use of a water heat-store in a domestic, gas-fired, central-heating system, expressly in the context of the above Cormorant design. It transpired that an opportunity arose to participate in a similar investigation in the commercial sector, see below. Three major studies have been undertaken; each will be presented as a "mini-thesis", i.e. with its own introduction, conclusions etc. within the body of the thesis, as Parts I, II, and III.

### (i) Part I

This is the most important component of the thesis. With the considerable potential of the Cormorant design described above, British Gas were very keen to evaluate the inherent, likely advantage of such a domestic, storage system over its conventional counterpart. This has been accomplished by means of two comparable computer simulation programs. A quantitative assessment of this advantage has been evaluated, together with its dependence on several of the design parameters of each system.

### (ii) Part II

In 1982/3 British Gas, in conjunction with the Energy Technology Support Unit (Dept. of Environment) were planning a full-scale field trial of a central-heating system incorporating a large water store in a London school. The author was called upon to assist at the design stage (principally by developing and using programs for computer predictions), to help with monitoring and to conduct preliminary data analysis of the project. It had been hoped that the readings obtained from the 1984/5 heating season would suffice to complete this investigation; in fact, the monitoring has continued through the current (1985/6) season, so that the results of Part II are "interim" results only.



## (iii) Part III

Reverting to the Cormorant boiler/store unit, it is important that stored heat be used wisely, especially with regard to the d.h.w. heat exchanger such that high-temperature heat is conserved for the radiator system. British Gas, having adopted the "Integron" heat exchanger material, suggested that two exchangers, each independently served by a three-way (mixing) valve outside the storage cylinder, might afford the best use of stored energy. Accordingly, a rig was built at Cranfield to simulate an entire domestic system with storage; a series of experiments has been carried out to try to optimise the d.h.w. heat exchanger system, by varying the heat exchanger's size and position.

The study reported here therefore encompasses computer modelling and data analyses, full-scale field trial work with its attendant practical problems, and a laboratory experimental investigation; the use of heat storage in both the domestic and commercial sectors has been explored. At no stage has the author had the intention of "selling the storage case", but it is hoped that all the work reported here has been dispassionately reasoned. It is further hoped that, in dividing the thesis into three Parts:-

- the reader will not form the impression that the three tasks undertaken were independent, but very much inter-related, and

- the format of presentation as "mini-theses" will prove to be convenient; graphs etc. which were considered of immediate relevance to the flow of the text have been incorporated in the text, while others have been included in appendices.

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**PART I**

**THE COMPUTER SIMULATION OF  
DOMESTIC, GAS-FIRED CENTRAL-HEATING  
SYSTEMS WITH AND WITHOUT A WATER-  
FILLED HEAT STORE**

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#### ABBREVIATIONS/SYMBOLS

The following abbreviations and symbols have been used intermittently in the text and on the figures:-

D.H.L. Design Heat Load of a building

B.O.D. Boiler "over-design" ratio

c.h. Central heating

d.h.w. Domestic hot water

3WV Three-way, or mixing, valve

## Abbreviations/Symbols (cont.)

h.e. Heat exchanger

a.s.d. After shut-down

t.r.v. Thermostatic radiator valve

$\eta_{GU}$  Overall boiler efficiency (gas usage)

$\eta_{FL}$  Full load boiler efficiency

kW Kilowatt

hr Hour



## INTRODUCTION

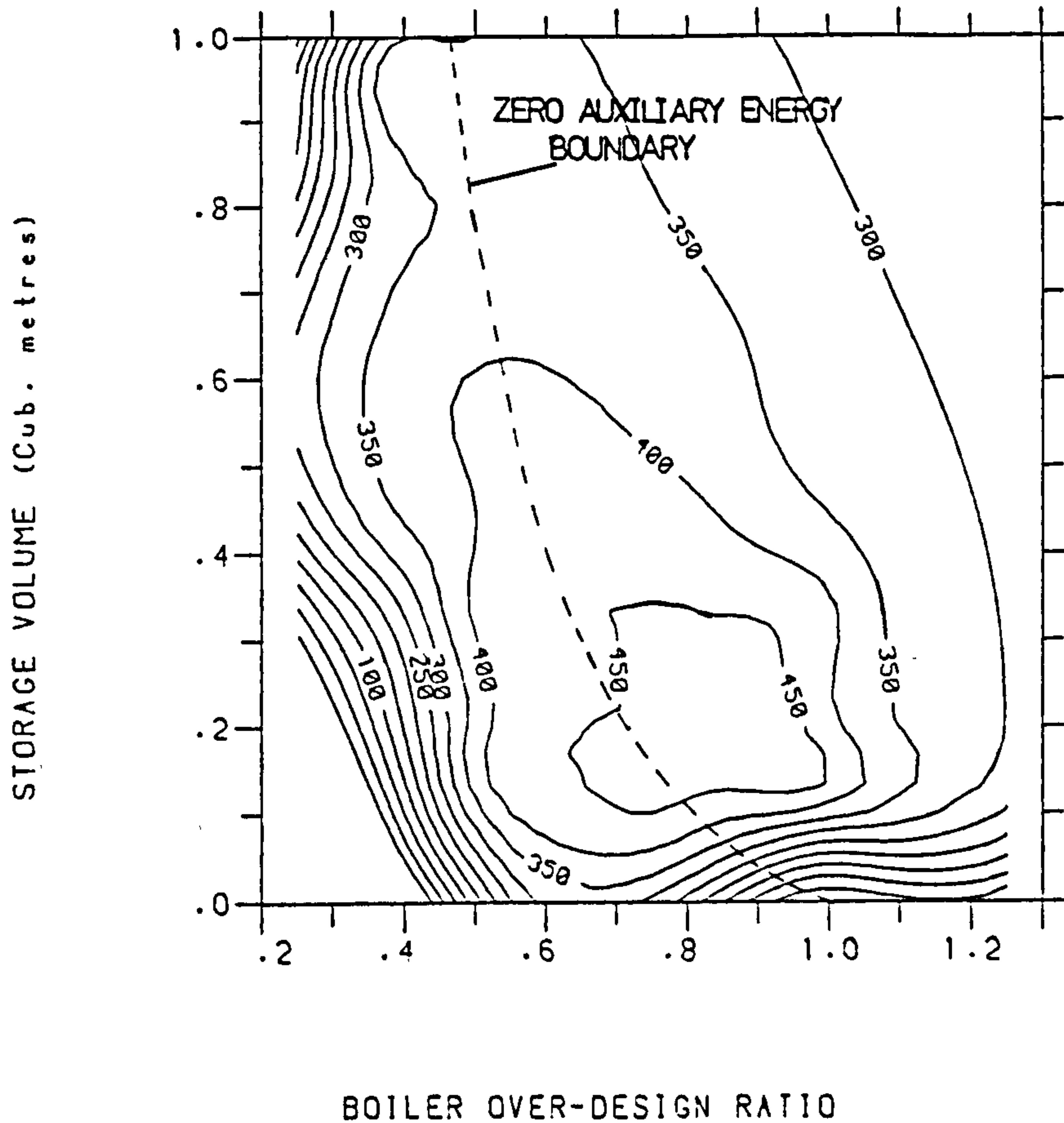
In May 1983, R.R. Cohen and R.J. Wood (of the Cranfield Applied Energy Dept.) presented a paper outlining the principles of thermal energy storage in the context of gas-fired central heating systems, ref. 1.1. For this paper, Cohen and Wood developed a simple computer simulation model to examine the adequacy, or otherwise, of a range of boiler and store sizes to meet the space heating demands (only) of a house whose peak heat demand was 6 kW (its design heat load, or D.H.L., will have been well below this). Comparing the estimated life-cycle savings of any storage system with the conventional system to be replaced, Fig. 1.1 was produced (reproduced from ref. 1.1). "Boiler Over-Design Ratio" was defined by Cohen and Wood to be:-

$$\text{B.O.D.} = \frac{\text{boiler full-load power output}}{6 \text{ kW}}$$

Fig. 1.1 suggests that, for the dwelling in question:-

- considerable savings could be effected by the use of a store of very modest volume, even as little as 100 litres,
- whatever the boiler power chosen, there was little point in increasing store volume beyond about 200 litres,
- maximum savings would be achieved with a range of boiler and store sizes, a good optimum being a store of 150 litres at a B.O.D. of 0.8,
- a B.O.D. of 0.8 was just over half the size of the boiler power fitted to the equivalent conventional system,
- for any chosen store volume, there would be a minimum B.O.D. below which an auxiliary heat supply would be called upon.

These findings were very encouraging as a basis for examining the "Cormorant" design of the integral boiler/store unit described in the General Introduction, above. The drawbacks of this computer study were:-



## SIMULATION PARAMETERS

- TOTAL ANNUAL SPACE HEATING DEMAND = 7335 kWh
- STORE COST =  $240 V^{-0.38} \text{ £/m}^3$
- BOILER COST = £50/kW
- BOILER FUEL COST = £0.023/kWh
- AUXILIARY FUEL COST = £0.05/kWh
- DISCOUNT RATE = 10%
- FUEL INFLATION = 3% REAL TERMS

Life cycle savings = difference in total capital and running costs over a 20 year lifetime between storage boiler system and conventional system with a boiler over design ratio of 1.5.

FIG 1.1: CONTOURS OF LIFE - CYCLE SAVINGS

(FROM REF 1.1)



- the program dealt only with "energy transfers", i.e. it did not model the physical characteristics of system components, such as heat transfer etc.,
- the boiler model was simply an arithmetic form of a published part-load efficiency curve, and thus took no account of the effect of the switch frequency domain (see section 3 (iii) of General Introduction above),
- the costing formulae were necessarily of an approximate nature.

At the outset of this study, therefore, it was required to set up computer models of a similar nature as the above program (i.e. similar input was used), but with a far more detailed modelling of system components. Two detailed models were sought with the aim of accurately predicting the performance of gas-fired central systems, with and without the use of thermal energy storage. When two such models had been developed, parallel runs with each model were used to evaluate the inherent advantage of the storage system, using typical values for each component size, flow rate etc. of the two systems, see "Base Case Study".

With such models developed, an assessment of the dependence of either system on any parameter value could quickly be effected. For example, a designer must know if radiator size, or boiler output is going to greatly influence system performance, particularly when he is obliged to choose from a range of discrete component sizes. The storage system would therefore be advantageous if its performance should prove insensitive to e.g. radiator size. Having established the "Base Case" evaluation of the systems, therefore, a parameter sensitivity study was conducted.

DATA AVAILABLE(i) House Space Heating Requirement

The setting up of a full computer model to simulate the performance of a heating system (whether storage or conventional) in a specified building for a given year's weather conditions would have been a very lengthy exercise. Indeed, it would also have been largely pointless because several programs were already operating to simulate any given building and predict its likely heat load requirement, e.g. the "Tas" program of Jones-Cassidy-Mellor (ref. 1.2), and in particular the "Therm" program of British Gas, Watson House (ref. 1.3). The "Therm" output was made available to the author on request. This program predicted the mean temperature in each of up to ninety rooms of the building under investigation, in any weather conditions. It was described by Watson House in 1982 as being a "cumbersome" program to use; a simplified version entitled "Microtherm" was subsequently developed by Watson House to be compatible with a small desk computer. The author therefore made no attempt to become familiar with the use of "Therm". It was possible to ask Watson House to effect any required building simulation on "Therm", and make the relevant output available in a form compatible with central heating (c.h.) system models that would be developed at Cranfield.

Again, the purpose of the computer modelling to be undertaken for this study was to compare domestic central-heating system operation with and without water heat storage, i.e. simulation of a building's performance was very much of secondary interest. Thus, availability of "Therm" output would avoid unnecessary work. Moreover, it was intended to construct two heating system simulation programs, one for a conventional system, one for a system with storage, such that the same input data could be imposed on each program, thus effecting a direct comparison of the performance of the two c.h. system types. The comprehensive and complex output obtained from "Therm" could therefore be reduced to a very simple form, and still represent a given house (in a chosen year's weather) adequately for the purpose of this study.



British Gas furnished a total of seven building simulation outputs from Therm, reduced to an hour-by-hour summary of the house space heating requirements, in each case for a 212 day heating season (Oct. 1st to Apr. 30th). Each hour was simply represented by two figures:-

- a total space heating requirement for that hour,
- a mean room temperature for that hour.

Thus it was supposed that if the c.h. system provided the quoted heat requirement, the house would be maintained at the quoted temperature. The building performance characteristics were therefore encapsulated in the above two parameters: the modelling at Cranfield would be required to simulate c.h. system operation from fuel input to heat-to-room at the required room temperature.

The tapes available represented the following houses and heating schedules (Table 1.1):-

Tape No.	No. of Bed-rooms	Type of House	Wall Construction	D.H.L. kW	Heating Period,hr	No. of Persons	Seasonal Heat Load,kWh
1	4	Detached	Cavity	7.994	17	4	19300
2	4	Detached	Cavity	11.780	17	4	29111
3	3	End Terr.	Solid	7.423	24	4	22286
4	3	End Terr.	Solid	7.423	8	2	13174
5	3	End Terr.	Solid	5.668	17	4	14355
6	3	End Terr.	Solid	7.423	17	4	17794
"3a"	3	End Terr.	Solid	6.1	17	4	13138

Notes: Tapes 3, 4 and 6 represented the same house.

Tape 5 represented an insulated version of the same house as tapes 3, 4 and 6.

Tapes 1 and 2 are for quite different houses.

Tape 3a is for a different house from tapes 3-6.

(D.H.L., the "design heat load" for the house, supposes a constant environmental ambient temperature of  $-1^{\circ}\text{C}$ ).

Table 1.1 : Houses modelled by "Therm" to provide Space Heating Requirements

Most of the above tapes represented houses taken to be heated from 06.00 hrs to 23.00 hrs, i.e. with a heating period of 17 hours each day. Tape 3 represented a continuously heated house, and tape 4 a house whose occupants worked during the day, reducing the heating period to eight hours a day. Tape "3a" was the first supplied by British Gas, and was used for preliminary runs to develop the programs, and as the space heating requirement for the "base case" runs prior to the parameter sensitivity investigation, see below. To produce each tape, the same year's weather conditions (for 1964-65) had been fed into "Therm", the year being considered a good "average" year, such that the tapes directly represented the performances described.

(ii) Domestic Hot-Water Requirement

It was decided to impose a domestic hot-water (d.h.w.) draw-off schedule on any central-heating system modelled, such that d.h.w. should be quite independent of the load schedules of the above data tapes. British Gas supplied standard d.h.w. schedules as in Appendix 1.1, which describes two days' requirements - a low and a high days' schedule. These standard days' d.h.w. use were altered slightly in terms of timing, so that, for ease of programming, no more than one d.h.w. draw-off took place in any clock hour. The revised schedules were as in Table 1.2 below:-

<u>Day A</u>		<u>Day B</u>	
Time	Draw (litres)	Time	Draw (litres)
08.45	4	08.45	4
09.30	5	09.45	4
19.45	4	10.45	10
20.45	10	11.30	25
21.30	65	12.30	25
Total	<u>88</u>	13.01	25
		14.01	4
One week's requirement for:-		15.01	5
Working couple, AAAAAAA = 616 litres		16.01	2
Family, ABABABA = 1156 litres		18.50	4
Average, BAAABAA = 976 litres		19.50	5
		20.30	25
		21.45	65
		22.15	65
		Total	<u>268</u>

Table 1.2 : Modified D.H.W. Schedule for Specimen Days

By building up sequences of the specimen days' schedules of Table 1.2, d.h.w. data tapes of hour-by-hour requirements were created to cover entire heating seasons of 212 days for each of the "working couple", "family" and "average" profiles.

Thus any of the seven house space-heating schedules could be combined with any of the above three d.h.w. schedules, and independently fed as input into the heating system simulation models, whose construction will now be described briefly.



## CENTRAL-HEATING SYSTEM SIMULATION MODELS

Two computer programs were developed to model the thermal behaviour of domestic central heating systems (i) with a water heat store, and (ii) with a conventionally arranged boiler/radiator/d.h.w. configuration. Each program was designed to accept any of the above space heat load and d.h.w. load input tapes, reading in requirements for each simulated hour. The programs were kept as simple and yet as reliable as possible, with the aim of assessing the inherent benefit of the storage concept.

### (i) The Storage System Program "HS212"

In November 1982, Robert Cohen of the Applied Energy Group at Cranfield was testing a prototype boiler/store combination named a "Mini", designed by British Gas and fabricated by Gledhill. The store consisted of a copper tank of capacity 125 litres, containing a d.h.w. heat exchange coil made from one-inch copper pipe, see photo 1.1. This heat exchanger was designed to offer minimal resistance to flow, and was of massive construction, incorporating two double-helix coils; its internal capacity was some 11.3 litres. The store was heated by a 4 kW gas input, buoyancy-driven, "circulator" boiler. At this time, British Gas were hoping to develop the "Mini" and the smaller "Micro" (100 litre store) together with a 270 litre variant, the "Tall-Boy". All these would be powered by the same circulator, and all would contain the same d.h.w. heat exchanger. (In fact, none of these was retained for long, the "Cormorant" model, with an "Integron" heat exchanger, being eventually preferred for commercial exploitation.)

In November 1982, however, the start of the period covered by this thesis, it was thought that the "Mini" etc. range would be in widespread use for field trials, and the above tests on the "Mini" (ref. 1.4) were used as a basis for the storage model to be developed. Thus computer models were first sought for the various elements of the "Mini", to agree with the experimental curves of ref. 1.4, particularly Figs. 6 and 7, which are reproduced here by kind permission of R. Cohen as Figs. 1.2a and 1.2b. Being the very start of the research scholarship, this exercise served to introduce the





PHOTO 1.1: ORIGINAL DOUBLE-HELIX D.H.W.

HEAT EXCHANGER IN "MINI" TESTED IN REF 1.4



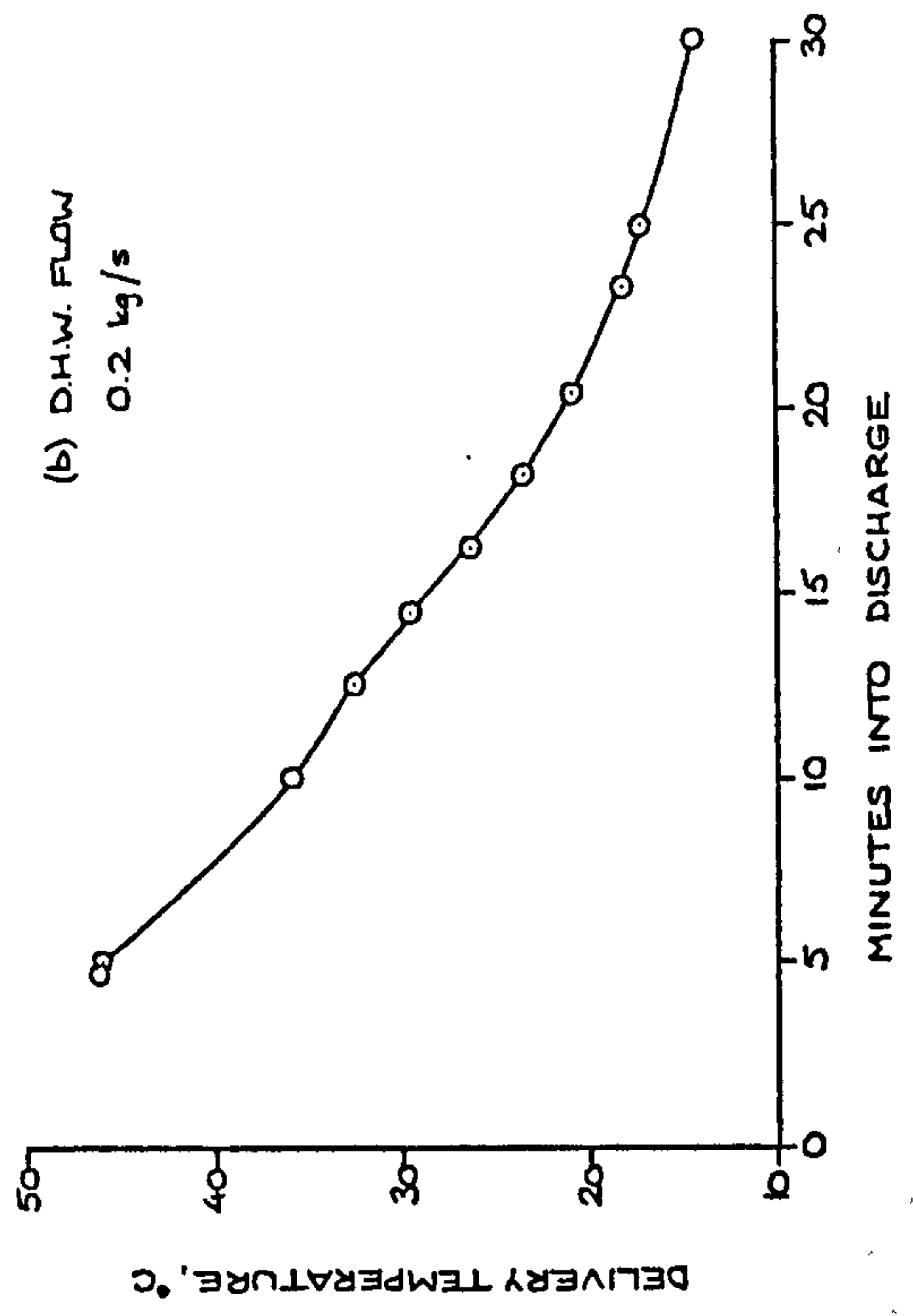
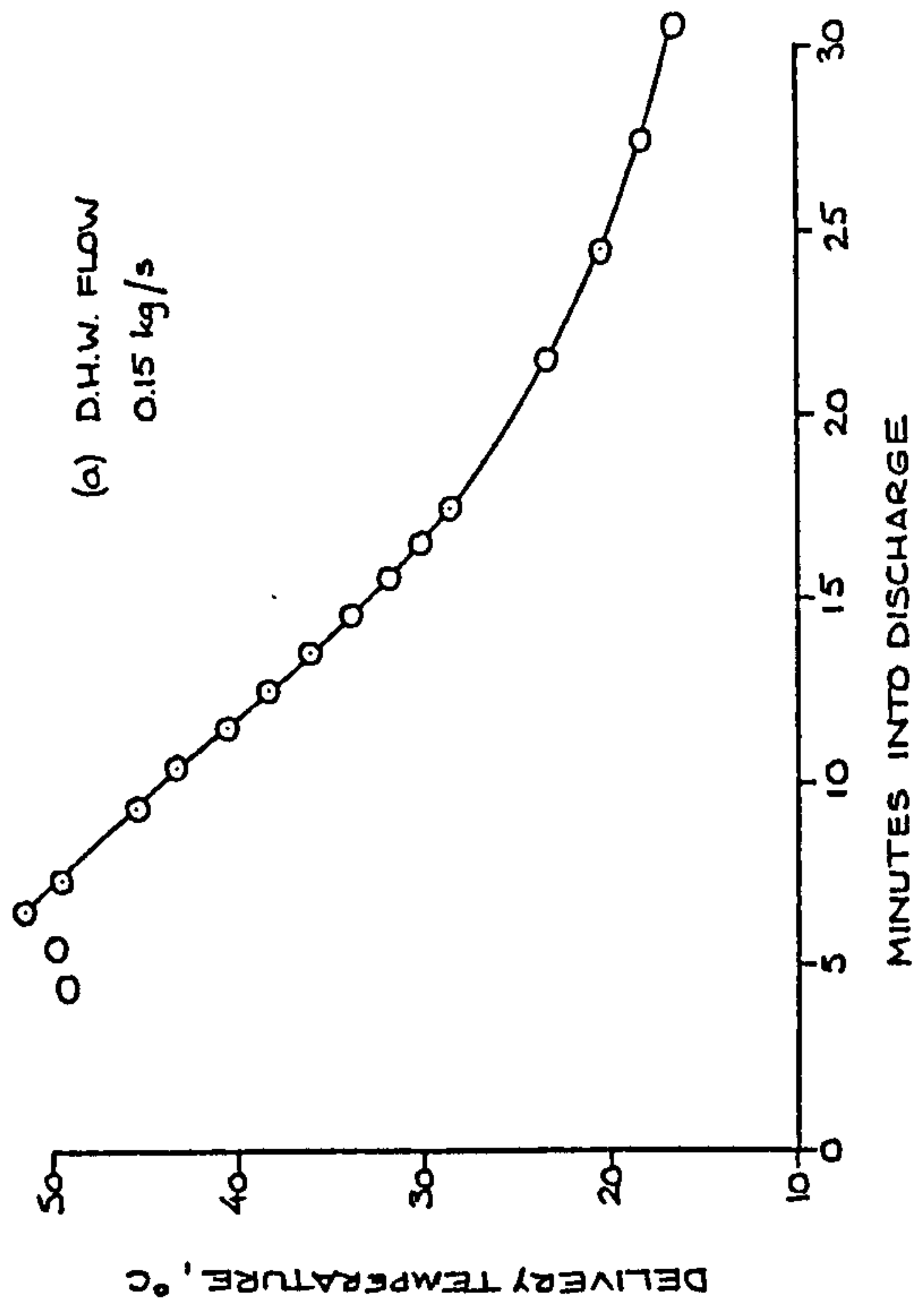
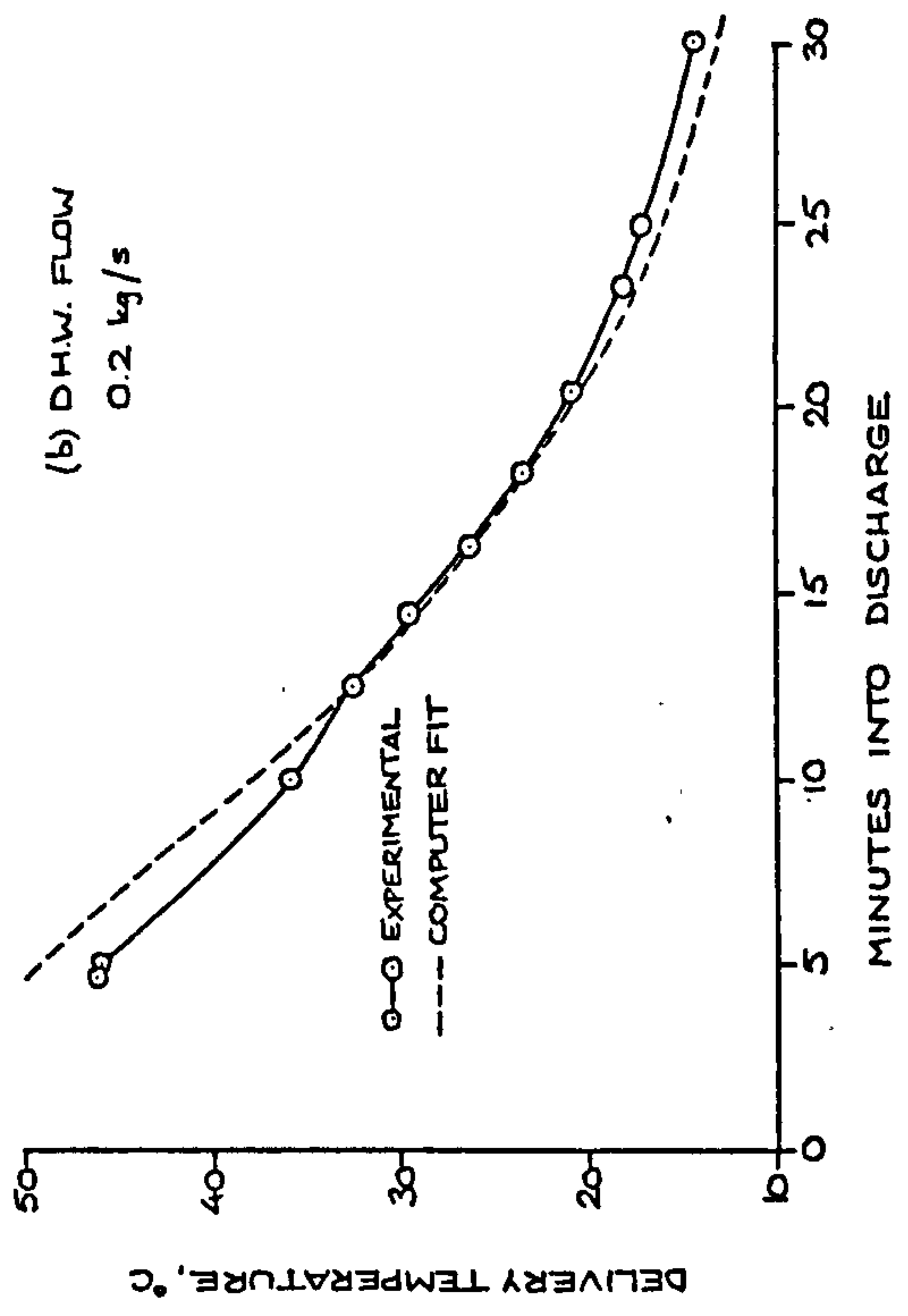
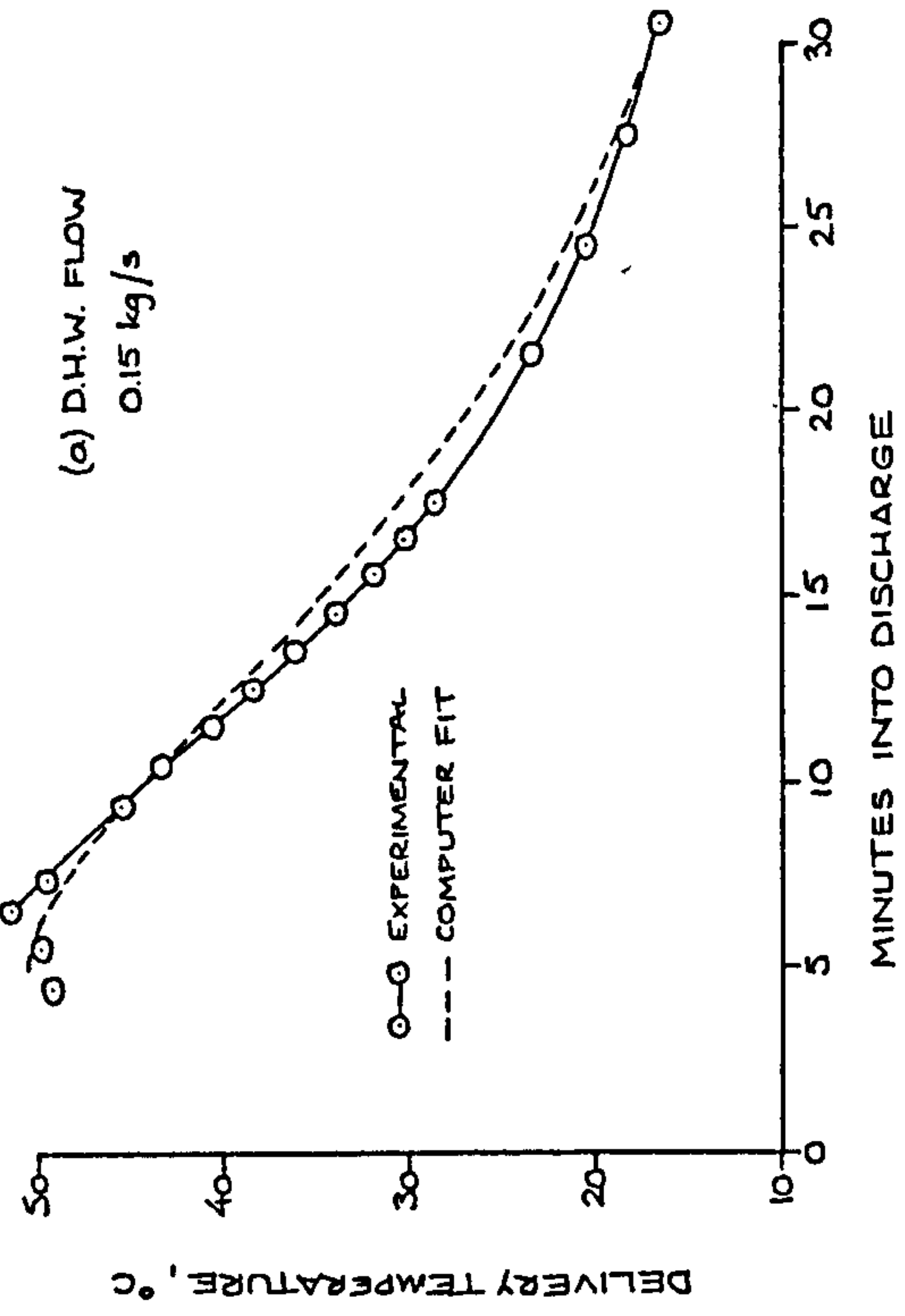


FIG1.5: COMPUTER FITS

FIG1.2: EXPERIMENTAL DISCHARGES - REF.1.4



author to the Cranfield VAX 785 computer.

(a) Model of Store : the first task was to set up a computer model of the heat transfer from the store water to the cold d.h.w. supply as it passed through the coil. Simple discharge tests of the uniformly heated store had produced Fig. 1.2a and 1.2b at 9 and 12 litres/minute flow rate in the d.h.w. circuit. The three-way-valve (3WV) action at the outlet of the coil modulated the part of this flow which passed through the coil, such that delivery temperature was controlled to approximately  $47^{\circ}\text{C}$ . Thus the model set up for the store

was as in Fig. 1.3. One glance at photo 1.1 reveals that accurate modelling of the heat exchange of such a coil would be extremely difficult; the exchanger is seen, however, to be almost uniformly disposed from top to bottom. Its heat exchange with the surrounding water in the store was therefore assumed to be characterised approximately by a uniform heat exchanger (h.e.) extending from the bottom of the store to a height corresponding with the measured uppermost coil of the real h.e. It was further supposed that the volume

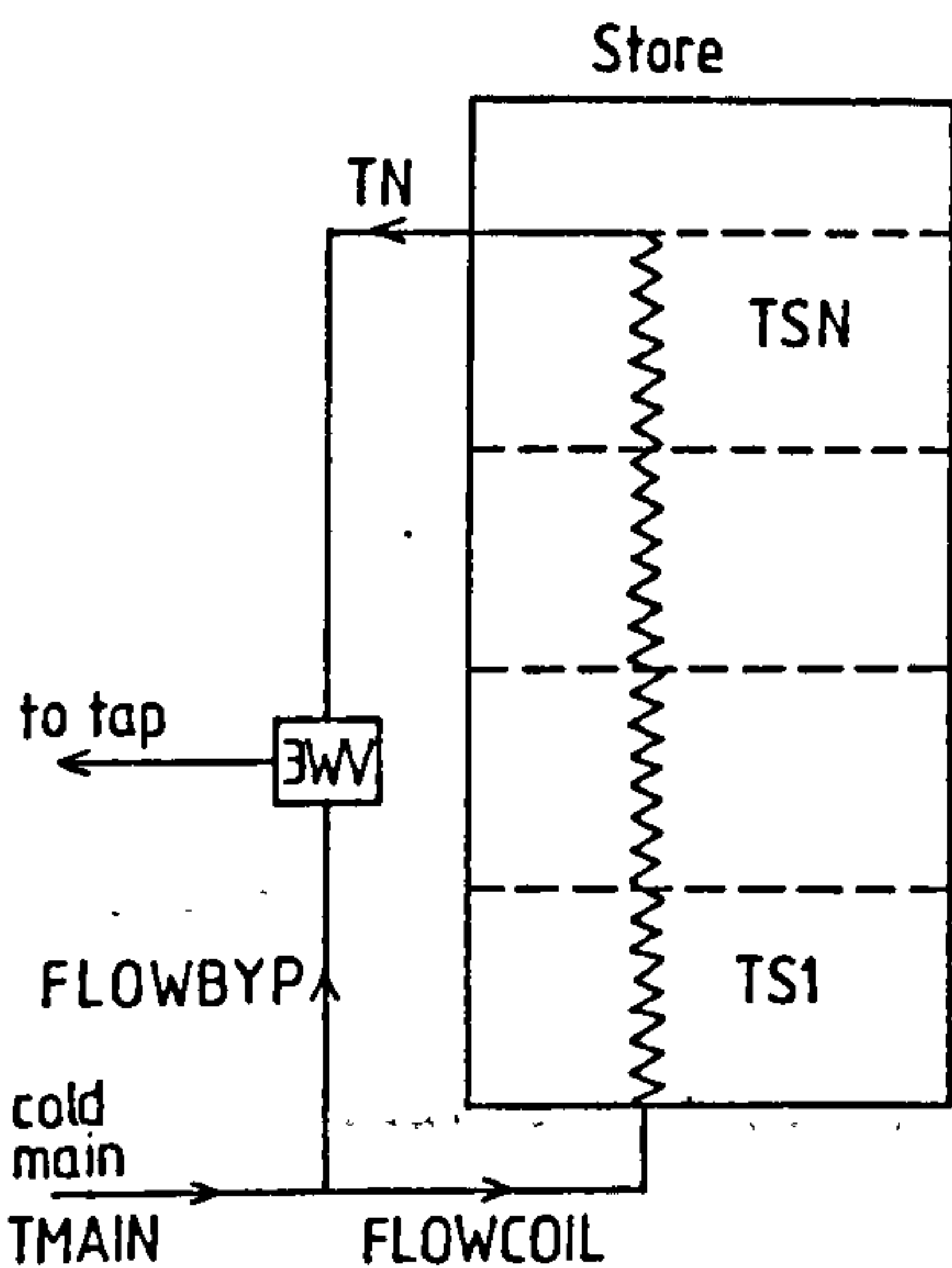


Fig.1.3: Store model

of store water above the h.e. did not exchange heat with the coil, and that the h.e. and the rest of the store could both be divided into an equal number of identical, fully mixed zones ( $N$ ). Each zone of the store would exchange heat with the water inside that portion of the coil contained in the zone, throughout one timestep. The time-step itself ( $\text{DELTA}$ ) was determined as the time required by the water flow through the coil ( $\text{FLOWCOIL}$ ) to completely fill one zone's part of the coil;  $\text{FLOWCOIL}$  itself was determined by the three-way-valve, into which a leak factor was included. No attempt was made to deduce a more refined model for the three-way-valve, i.e. for example with a proportional and integral type of transfer function.

Taking a single zone of the store, which contained its own portion of the "uniform" h.e. through which the d.h.w. circuit water was flowing, an algorithm for heat transfer from the store to the d.h.w. had to be chosen and developed such that computer predictions of the outlet temperature aligned well with the experimental curves of Figs. 1.2. A total of five algorithms were tried, the fifth of which was adopted, and was as follows, see Fig. 1.4.

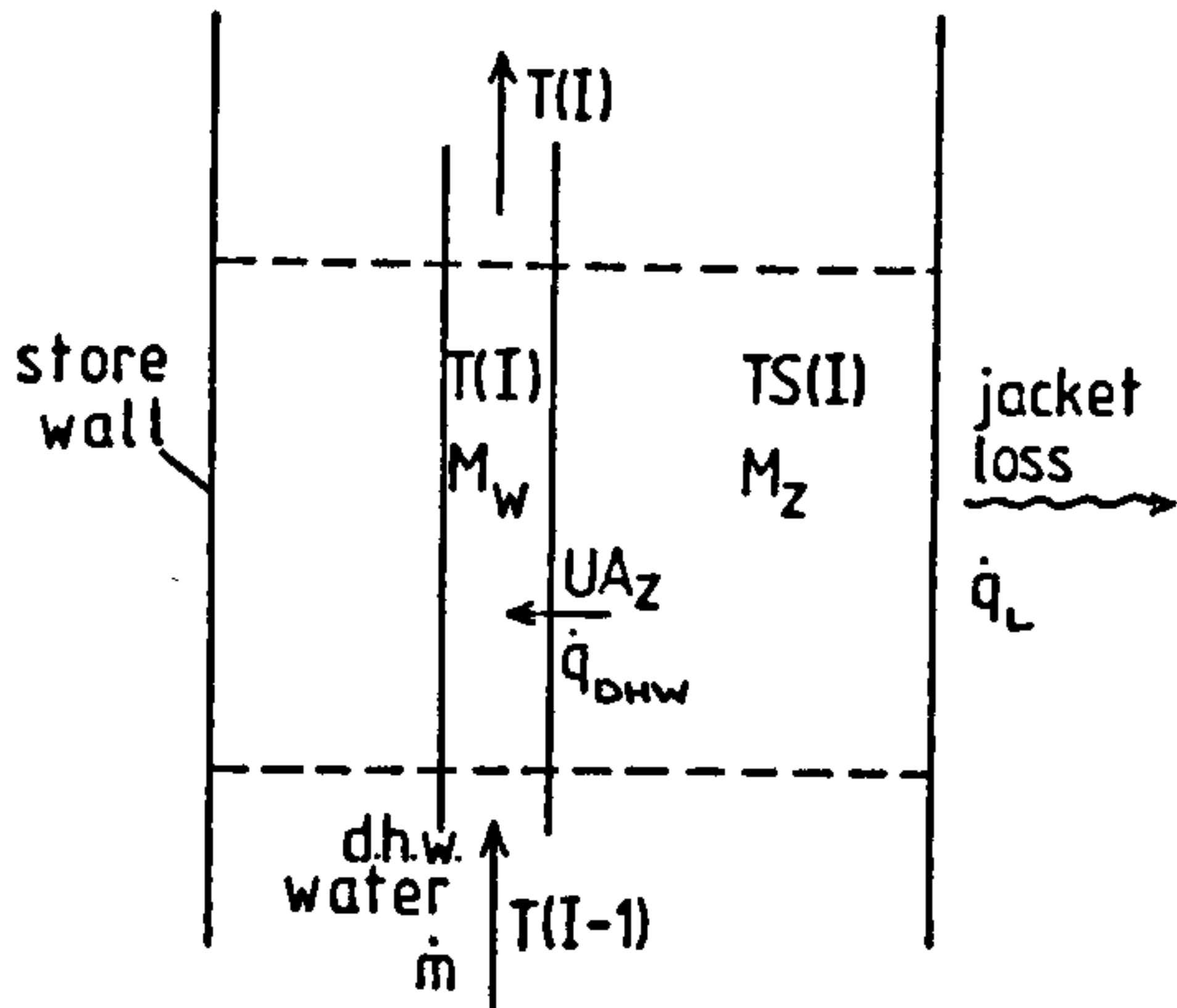


Fig.1.4: Single zone model

A uniform zone, zone no. I, is at temperature  $TS(I)$  and contains a mass  $M_z$  of store water. It is losing heat through its fraction of the jacket insulation at a rate  $\dot{q}_L$ . The portion of the d.h.w. heat exchanger contained inside the zone is filled with water at temperature  $T(I)$  (instantaneously), receiving a flow of cooler water at temperature  $T(I-1)$  and rate  $\dot{m}$ . Water is flowing out of the d.h.w. coil, upwards into the next zone at temperature  $T(I)$ .

The coil element is receiving heat from the store at a rate  $\dot{q}_{DHW}$ . The effective heat transfer coefficient between the store zone and the coil element is  $UA_z$ . The heat balances on the zone give:-

$$\text{for the zone:} \quad -M_z Cp \frac{\partial TS(I)}{\partial T} = \dot{q}_L + \dot{q}_{DHW} \quad (1)$$

$$\text{for heat exchange at the coil:} \quad \dot{q}_{DHW} = UA_z \cdot (TS(I) - T(I)) \quad (2)$$

for the control volume of the water instantaneously contained by the coil element:

$$M_w Cp \frac{\partial T(I)}{\partial t} + \dot{m} Cp (T(I) - T(I-1)) = \dot{q}_{DHW} \quad (3)$$

Of the terms of these equations,  $\dot{m}$  and  $Cp$  are known.  $M_w$ ,  $M_z$  and  $UA_z$  are calculated from the choice of the number of zones (assuming a trial value of  $\Sigma UA_z$ , the total effective heat transfer coefficient of the coil, which is the principal parameter being evaluated).  $\dot{q}_L$  is



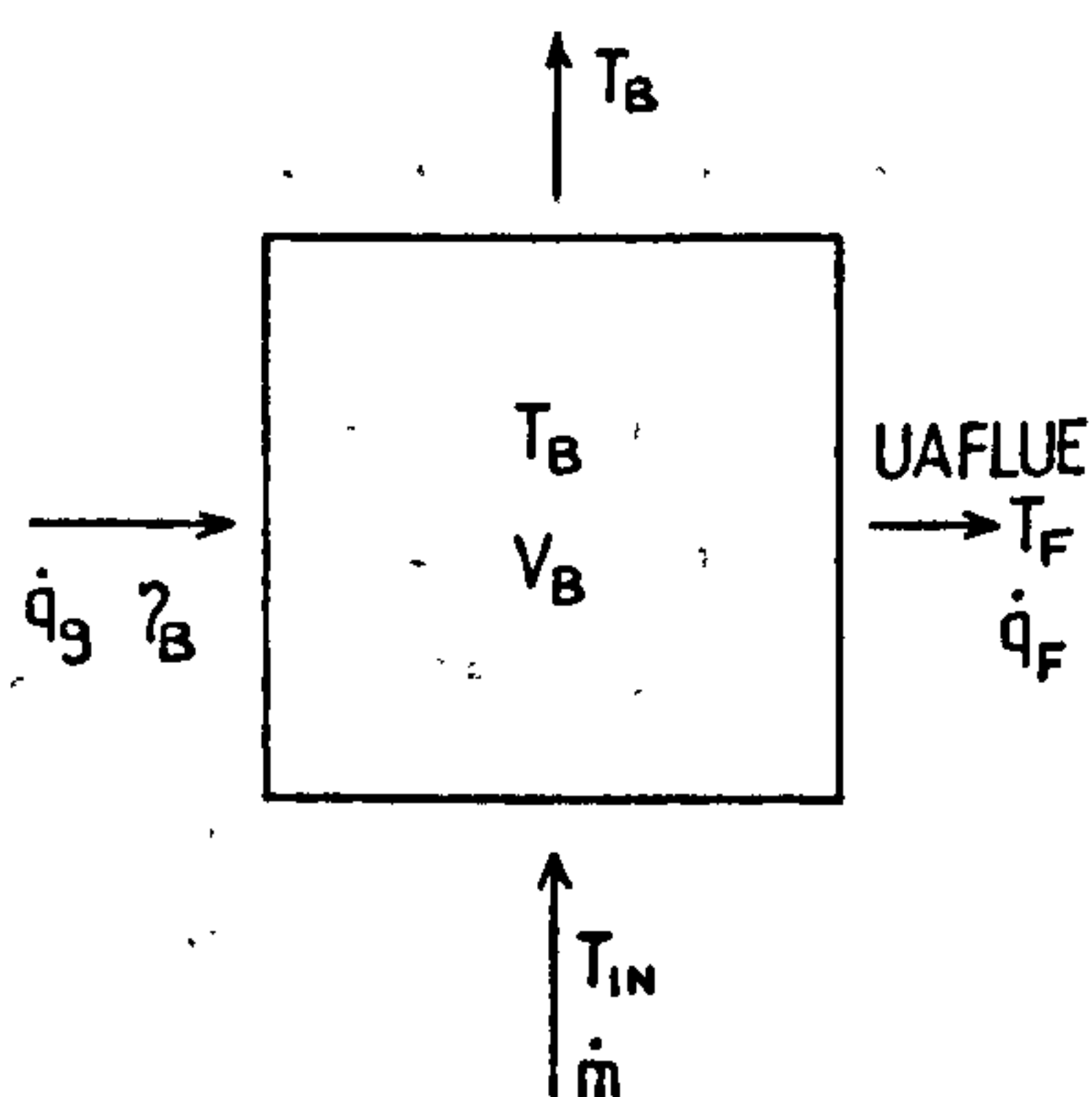
eliminated from equations (1) and (2). Forward finite-difference expressions were substituted for  $\frac{\partial TS(I)}{\partial t}$  and  $\frac{\partial T(I)}{\partial t}$ , and the remaining variables,  $TS(I)$ ,  $T(I)$ ,  $T(I-1)$  were evaluated from the two remaining equations by setting up a simple progressive time-step calculation, to establish all the temperatures of the store and d.h.w. circuit after each time-step.

On the advice of Robert Cohen, who had previous experience of the effect of zone number on such computer models, the number of zones was always fixed at four (for all store sizes, see later), plus one zone above the d.h.w. heat exchanger.

Thus a discharge model was developed to simulate the curves of Figs. 1.2; a series of values of the total heat-exchanger heat-transfer coefficient ( $\sum UA_z$ ) were tried to obtain an optimal match at the two flow rates of Figs. 1.2. The theoretical "fit" was as depicted on Fig. 1.5. For this, a time-step factor of two was chosen (again on the advice of Robert Cohen), such that the system temperatures were calculated twice per time-step. A value of 1800  $W/^\circ C$  was found to be adequate for the effective coefficient of the coil, and this model of the store was retained for subsequent, expanded programs.

(b) Model of boiler: the bouyancy-driven circulator boiler (a "Maxol" unit of some 4 kW input and heat output about 2.8 kW) would have been difficult to model accurately. The buoyancy principle is notoriously difficult to model, requiring accurate geometric representation to even approach a reliable model. For this reason, for the sake of simulation purposes, boiler models were always assumed to be pumped, i.e. run at a constant water-flow rate.

An accurate, previously-devised, model of a pumped boiler was sought but not found. This was a particular drawback, as, in the forthcoming simulations, a good model of the boiler performance would have been a great help. The writer, in the absence of any such model, resorted to the simplest of models, as follows:-



The boiler was considered to approximate to a single, fully-mixed water container of water equivalent  $V_B$  and instantaneous temperature  $T_B$ . When the boiler was firing, it received heat at the gas rate  $\dot{q}_g$ , multiplied by the full-load bench efficiency  $\eta_B$ . It received water at temperature  $T_{IN}$  and rate  $\dot{m}$ , and its output flow temperature was  $T_B$ , see Fig. 1.6.

Fig.1.6: Boiler model

At shut-down, it would lose its stored heat from  $T_B$  through an overall, effective heat-transfer coefficient,  $UAFLUE$ , to a representative flue or room temperature  $T_F$ . (This heat loss,  $\dot{q}_F$ , was to include both heat loss to flue and boiler case loss.) Thus, the boiler operation was governed by the following equations:-

$$\text{Boiler on: } \dot{q}_g \times \eta_B = V_B C_p \frac{\partial T_B}{\partial t} + \dot{m} C_p (T_B - T_{IN})$$

$$\text{Boiler off: } \dot{q}_F = UAFLUE \cdot (T_B - T_F) = -V_B C_p \frac{\partial T_B}{\partial t}$$

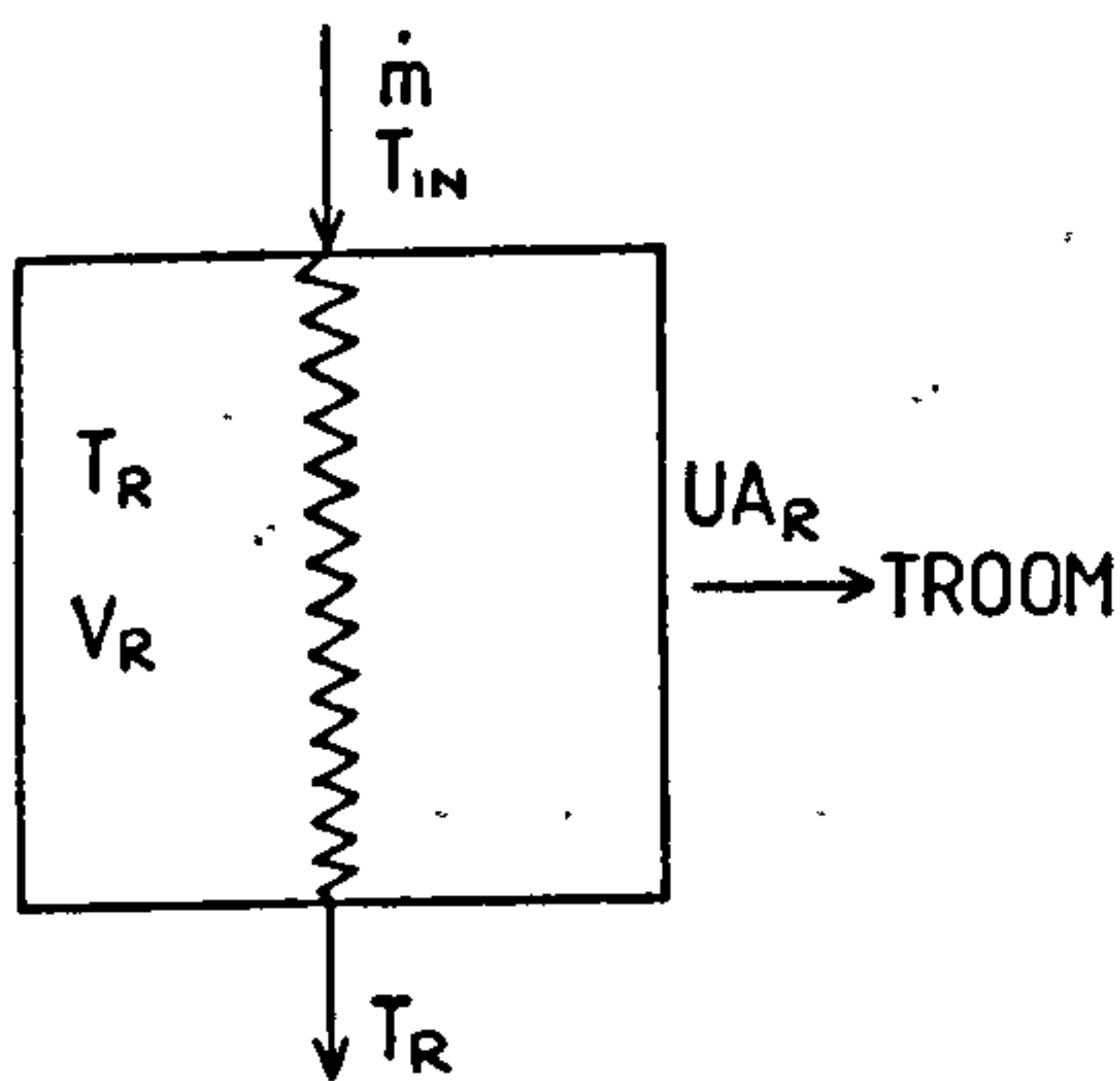
As for the store model, the above parameters were either known or assumed (e.g.  $T_F$ ) and the rate of change of  $T_B$  was substituted by a forward finite-difference. The boiler model could thus be added to the store model, using the same time-step. The coefficient  $UAFLUE$  was scarcely quantifiable, and its value was important in the calculation of boiler loss after shut-down (a.s.d.). It was therefore the subject of much discussion, as will be seen below in the "Reliability" section, as was the adequacy of the boiler model itself. No more sophisticated model was found; however, the author is persuaded that for this (rather elementary) study, the above, simple model was adequate, and that the subsequent results are valid. The model is clearly amenable to certain refinements, which will be among the first



tasks of any researcher who wishes to continue this work; the following are plainly required:-

- a more thorough model of the heat loss after shut-down (a.s.d.),
- a model of heat transfer from the flame to the boiler volume, such that efficiency variation with inlet temperature is allowed for. (A modulated efficiency equation was used for the circulator of the "Mini", constructed from early measurements by Robert Cohen in the test period reported in ref. 1.4.)

(c) Model of Radiator: Again, an elementary model was used, based on the tests of ref. 1.4. A house's radiator system was supposed to behave in a manner similar to a single heat exchanger, of water-equivalent  $V_R$ , and overall heat transfer coefficient  $UA_R$  through which heat was passed from the radiator to the room (temperature  $T_{ROOM}$ ) in a linear relationship. Initially the radiator was supposed to be at a temperature  $T_R$ , equal to the return temperature, see Fig. 1.7.



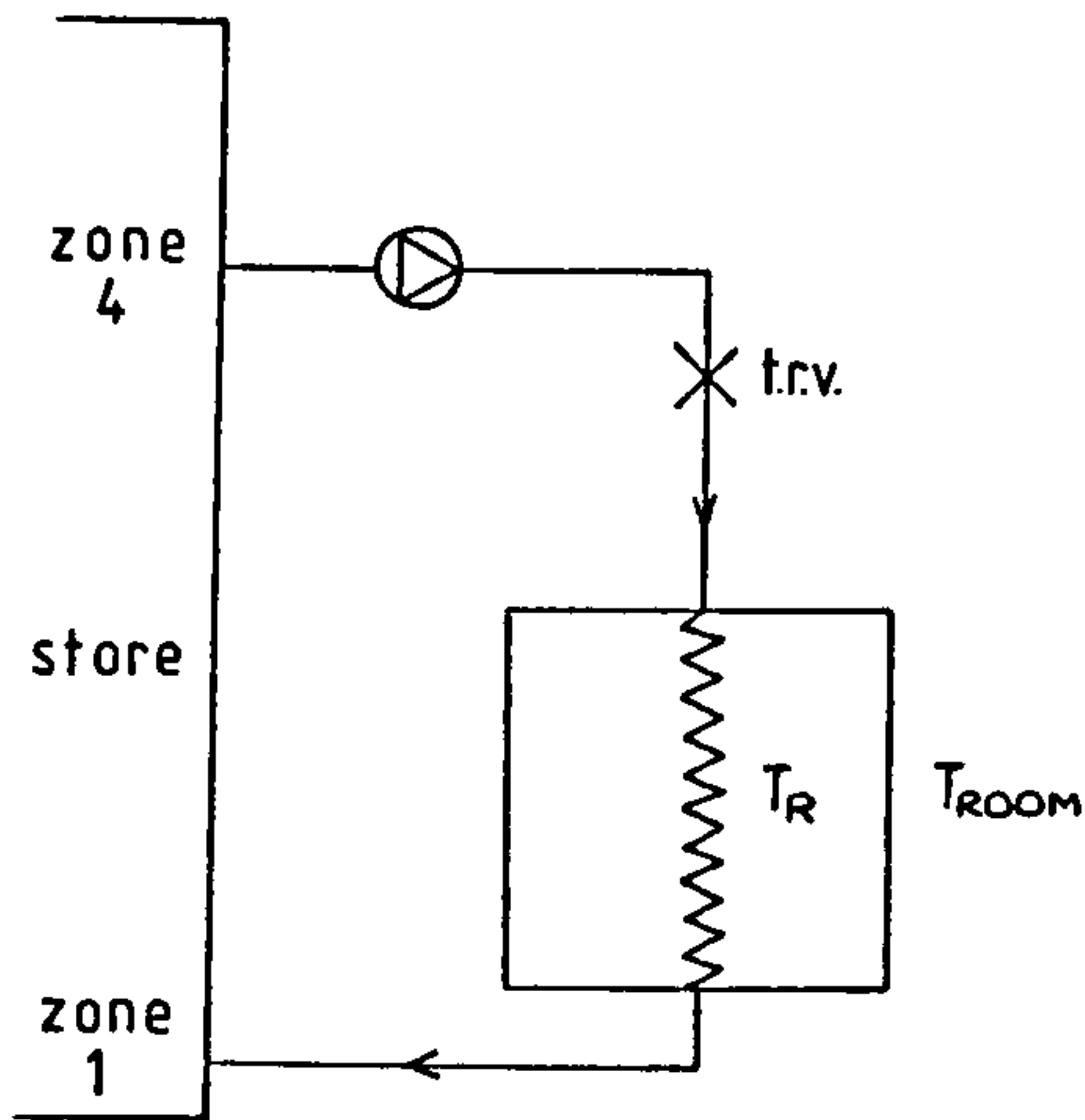
Thus the heat balance equation was

$$\dot{m}Cp(T_{IN}-T_R) = V_R Cp \cdot \frac{\partial T_R}{\partial t} + UA_R \cdot (T_R - T_{ROOM}).$$

The water flow into the radiator was taken to be water from the zone adjacent to the top of the d.h.w. heat exchanger (merely from the geometry of the "Mini"), thus  $T_{IN} = TS(I)$  of section (a) above.

Fig.1.7: Radiator model

In addition, for the storage system,  $\dot{m}$  was taken to be a pumped flow rate, but to make optimal use of the store, a perfect thermostatic radiator valve (t.r.v.) was assumed downstream of the radiator pump and at inlet to the radiator system, as in Fig. 1.8. It will be remembered that a



heat demand to maintain room temperature was to be read in for each simulated hour from the data tape (see above). Thus at the beginning of each simulated hour, the appropriate value of  $T_R$  would be calculated, and the T.R.V. setting would be adjusted each time-step such that the flow  $\dot{m}$  was throttled to that value which would just maintain  $T_R$ .

Fig.1.8: T.R.V. control

Thus the three basic elements of the storage program were developed. All three were put into a preliminary program to model the boiler/store/radiator system, to check against Fig. 1.9. The dotted line on Fig. 1.9 shows the theoretical fit for this dynamic, two-bath test.

These elements had now to be gradually built up from the above combination model, so that firstly a model of a day's central-heating operation was achieved. Two days of Data tape 3a were isolated (one mild, one severe), and the model developed to give suitable summations of e.g. overall boiler efficiency, cycles, heat to radiators, heat to d.h.w. etc.

Finally, the model was expanded to simulate a full heating season's operation of 212 days, and to accommodate any of the three d.h.w. demand tapes above. The circuit supposed was thus as in Fig. 1.10, with the nomenclature used in the seasonal simulation program, "HS212", which is included in Appx. 1.2, in a form which catered for the parameter sensitivity study undertaken later (see below). The salient features of the heating season program will now be presented briefly.

(d) Storage Heating Season Program: As above, the temperatures of the whole system were calculated at the end of each time-step. The time-step duration was determined by the d.h.w. circuit, as follows. If the d.h.w. flow was zero, a time-step of 20 seconds was chosen; if



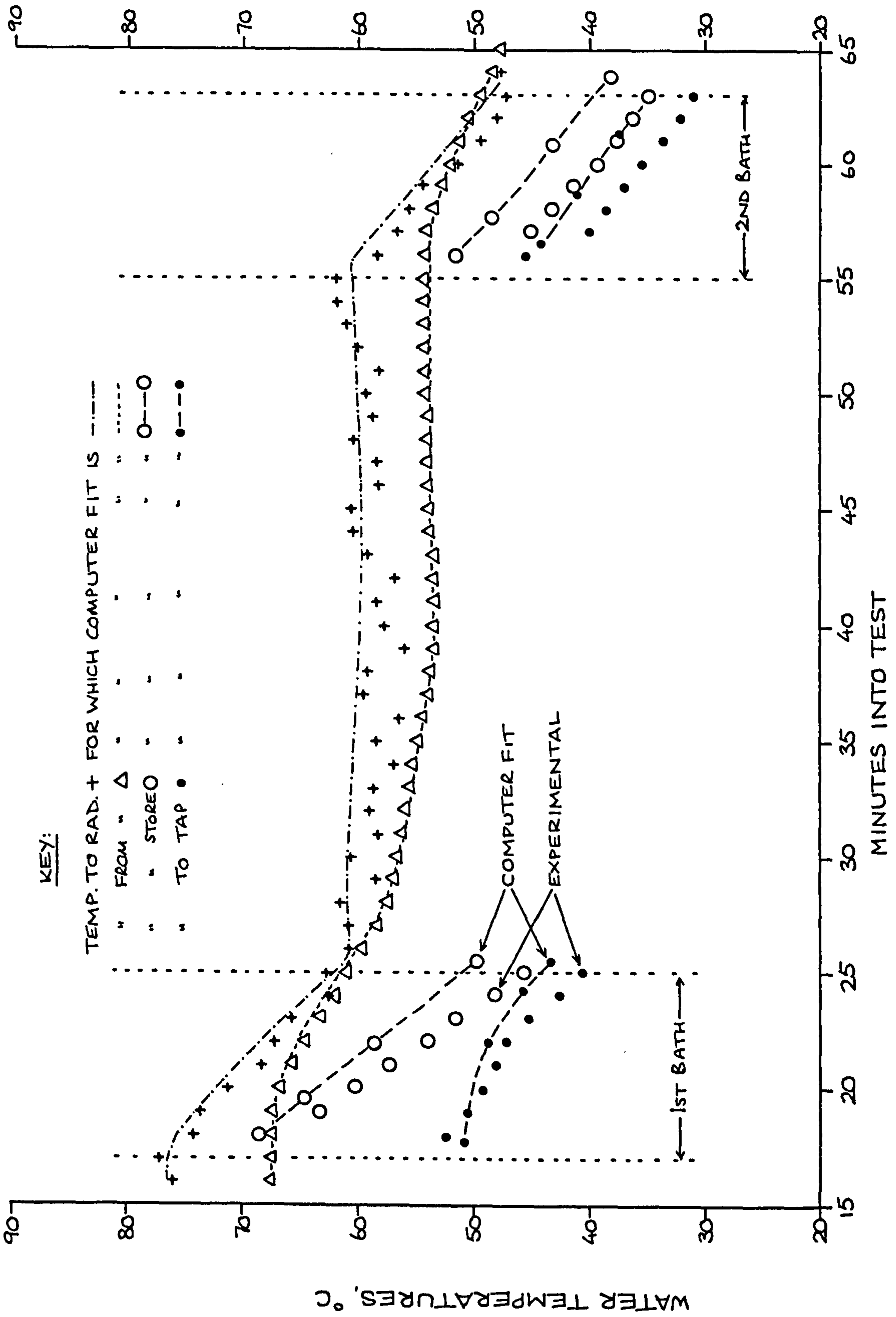


FIG1.9: TWO-BATH DYNAMIC TEST - REF1.4

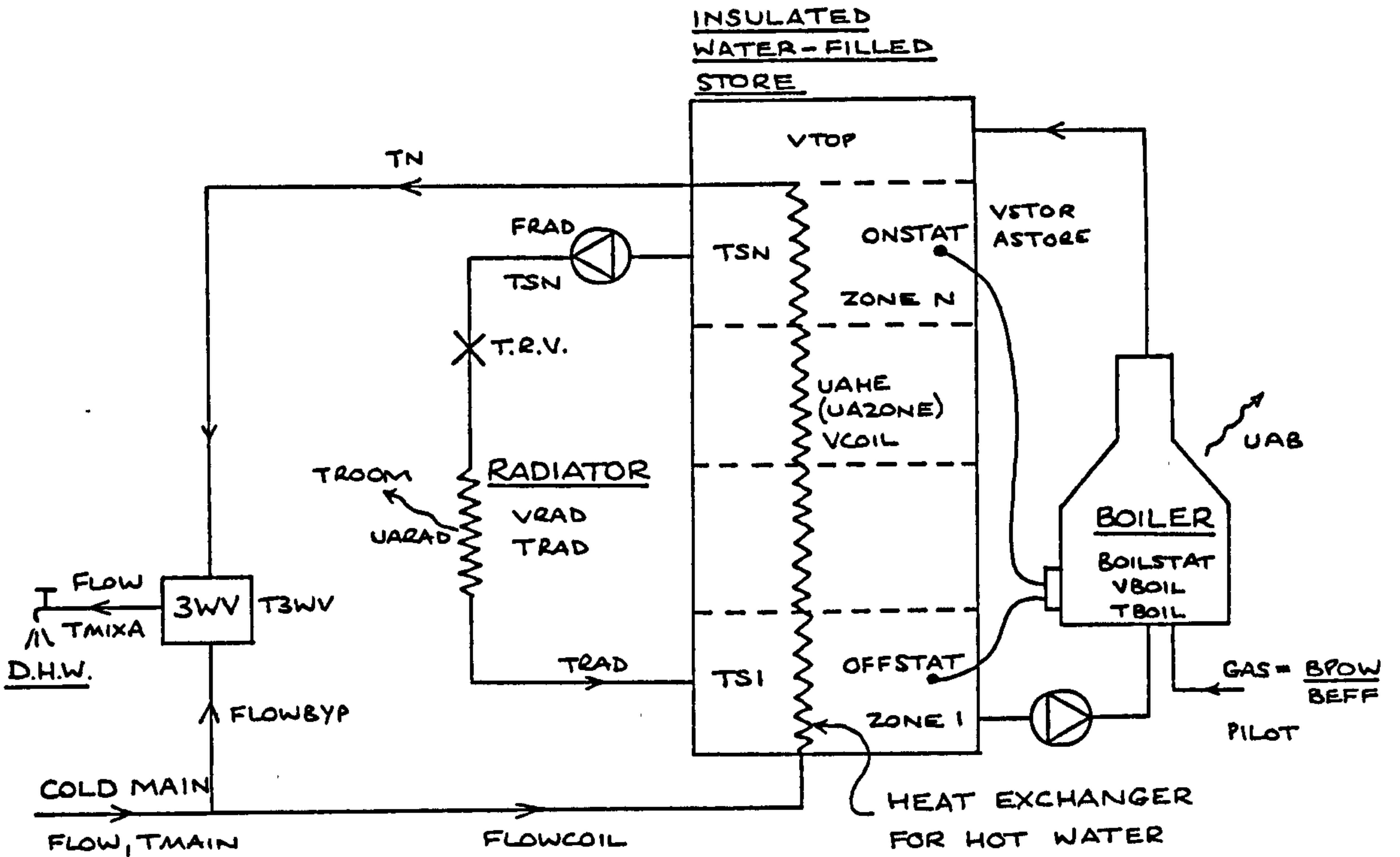


FIG 1.10 : SCHEMATIC DOMESTIC C.H. SYSTEM WITH STORAGE

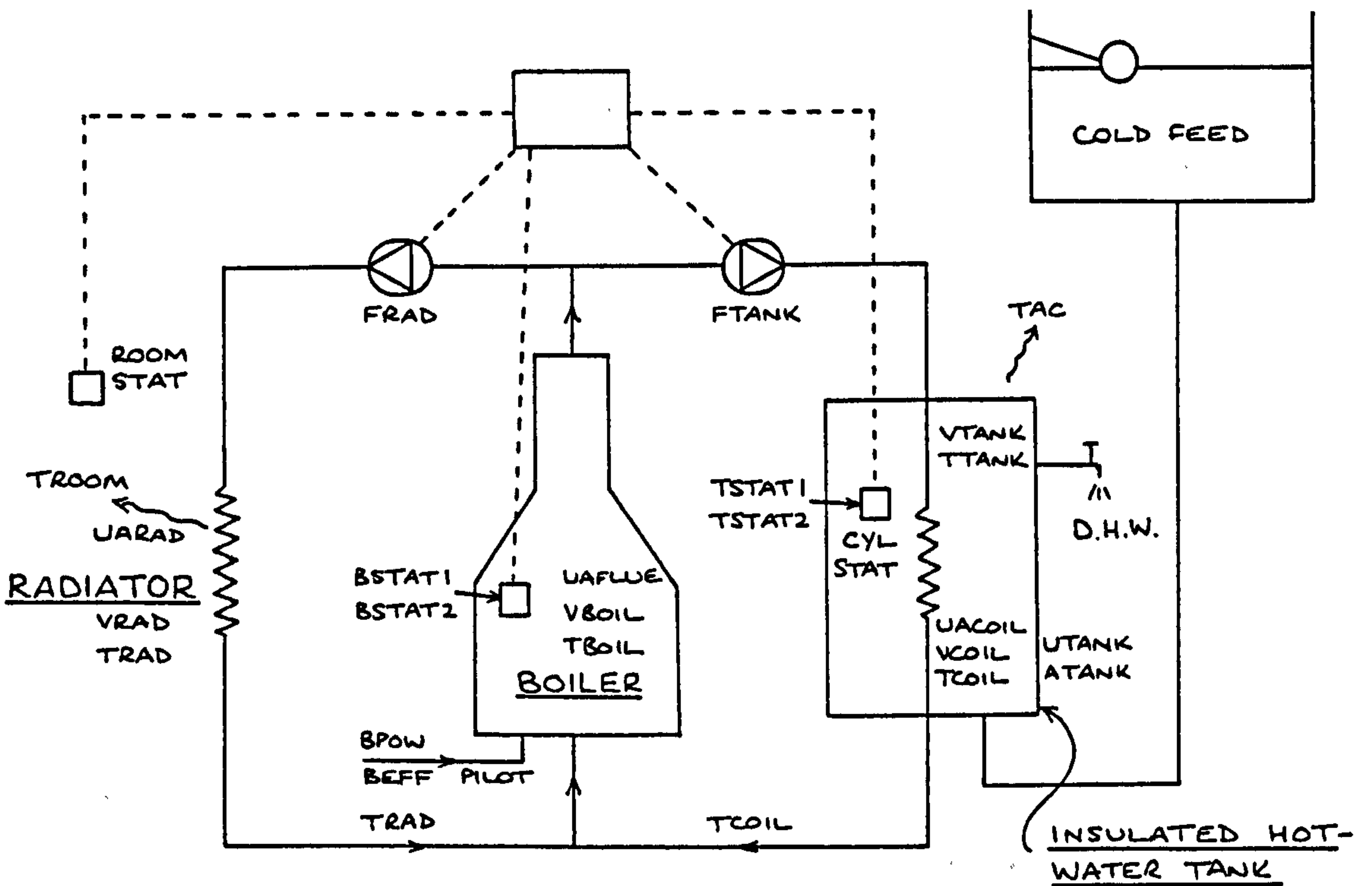


FIG 1.12 : CONVENTIONAL DOMESTIC C.H. SYSTEM

the d.h.w. was being drawn, the time-step was calculated each time as that period required just to fill one zone's portion of the heat exchanger, i.e. referring to Fig. 1.10,

$$\text{DELTA} = \text{VCOIL}/(\text{N} \times \text{FLOWCOIL})$$

FLOWCOIL was itself calculated by the mix at the d.h.w. three-way-valve, so the FLOWCOIL and DELTA values varied for each step.

All heating season simulation programs used comprised a nest of three basic loops; these were:-

- the DELTA loop, a single time-step loop, where the radiator, boiler and store temperatures were recalculated, and sums of various heat fluxes made, etc. This was, of course, the first loop to be established. In the store calculation was the d.h.w. circuit calculation (if necessary), and a temperature inversion clause which allowed cooler water higher up in the store (should it arise) to mix with the zone below. Temperatures were reset at the end of the DELTA loop, which proceeded until the accumulated time allowed for the HOUR loop to be reintroduced.
- the HOUR loop, where required values were read in from the heat load and d.h.w. data tapes. Sums of heat passed, e.g. from the radiators to the rooms were initialised, and printed out if required. After 24 HOUR loops, the program proceeded to
- the DAY loop, where daily totals and efficiencies were printed out if required, and adjustments made to the controls, etc.

After the 212-day season, efficiencies and totals for the entire season were calculated and printed. Thus a typical heating season could be simulated in about 13 minutes on the computer.

The simple controls applied to the storage model were as follows:- The boiler was allowed to heat the store until zone 1 of the store attained the temperature set on "OFFSTAT". The boiler was switched on when zone 4 (strictly zone N) fell to "ONSTAT". It was supposed that

the boiler's own thermostat "BOILSTAT" could be set to a very high value, over 90°C, such that no boiler "short-cycling" was produced before the store was completely charged. ONSTAT and OFFSTAT could be adjusted (assuming normal controls) to boosted values for very cold weather, or to lower values (especially a low "ONSTAT") for mild weather. The t.r.v. action for the radiator circuit has already been described. It was decided to set a mixed-tap delivery temperature to 47°C on the three-way valve (T3WV), and introduce a cold mains feed at 10°C (TMAIN) to the d.h.w. circuit. TMAIN might have been determined more accurately, and a monthly (say) mean value fed to the program. A given d.h.w. draw was terminated after the required volume had been drawn, except where the delivery temperature could not be maintained by the coil and fell to below 40°C, when a d.h.w. failure was recorded.

If there was any shortfall on heat delivered to the radiators in a given hour, the deficit was added to the subsequent hour's requirement. Thus the early morning peak (warm-up period) for the usual 17-hour heating period commencing at 06.00 hrs, would usually be smoothed acceptably. If, however, any deficit remained at 10.00 hrs, the radiator system was deemed inadequate for that day, and a radiator failure was recorded.

Other adjustable features incorporated into the program were:-

- a store preheat before the radiator pump started at 06.00 hrs.
- a pre-bath override to charge fully the store before a bath was drawn (a manual "over-rider" button was envisaged).

Other than day-by-day printouts, most notably of failures, the end of season summary contained the following values:-

- Total Gas used (boiler and pilot), HSGAS
- Heat supplied to the system water by the boiler (after flue and case loss), HSQWAT



- Overall boiler efficiency  $\eta_{GU} = HSQWAT/HSGAS$
- Load factor, defined as total gas used divided by gas that would have been consumed had the boiler fired during all the hours of the heating period (only, normally 17 hours), LOFACS.
- Heat emitted by radiator, HSRAD.
- Heat passed to taps, HSTAP
- Total Store jacket loss, HSQLC.
- Heat lost by boiler a.s.d., HSQBCOOL.
- Total gas used by the pilot light, HSPILOT.
- Number of boiler cycles in season, NBOFFS.

An energy balance, consisting of

- Heat made available to system,  $SUPPLY = HSQWAT + \text{loss of system enthalpy over season.}$
- Heat distributed by system,  $DIST = HSRAD + HSTAP + HSQLC.$
- Energy not accounted for,  $ERROR = (SUPPLY - DIST)/SUPPLY \%$

Thus any design of system could be assessed in any of the seven houses whose heat load tapes were available, with any of the three d.h.w. schedule tapes, and the seasonal performance could be assessed at a glance. As will be explained, the most significant of the above values was overall boiler efficiency,  $\eta_{GU}$ .

(e) Summer Simulation with Storage : by running the above program through the remaining 135 days of the summer, with the heat to radiator requirement maintained at zero throughout, it was a simple matter to assess the storage system when it supplied d.h.w. only. For this purpose, the program controls were trimmed to allow for likely

adjustments made by the householder, e.g. OFFSTAT and ONSTAT, would be reduced so as to reduce jacket loss (less useful in the summer months). By combining heating season and summer results, annual figures could be obtained very quickly.

The development of the storage program has been described rather briefly; many details of its evolution especially during the preliminary modelling to approximate to the test results, have been omitted for brevity. The author took some eight months to achieve a reliable heating season program, and to run off the first full simulation. This period included, of course, preliminary reading and familiarisation with the Vax 785 computer. It was then necessary to evolve a parallel model of a conventional central-heating system; this was achieved far more rapidly as follows.

(ii) The Conventional System program, "CHS"

A glance at the British Gas Central Heating manual (ref. 1.5) reveals that the number of possible layouts of a conventional (i.e. non-storage) gas-fired central heating (c.h.) system is very large indeed. Further, once the layout has been chosen, the householder can choose from one of a number of boiler types (modulated, balanced-flue etc), and in recent years he has had the option of fitting more effective controls such as "energy management" systems. The efficiency of a modern gas-fired conventional system can now be very high indeed; a sensibly-sized light-weight, balanced-flue boiler, or (especially) a condensing boiler, together with thermostatic radiator valves offers him a system of considerable running economy, though good economy often incurs a high installation cost.

In assessing the likely benefit of a storage system, therefore, it was not easy to choose a "typical" conventional system as a basis for comparison. It was hoped, though, to isolate the inherent advantages of the storage system i.e. whatever conventional system was chosen, the conversion to storage, leaving as many as possible of the chosen elements untouched, should reveal "in-built" advantages in favour of the storage principle. Thus a conventional system model was sought with as much in common as possible with the above storage model. The

following characteristics were retained from the storage model:-

- the same data (heat load and d.h.w.) were used,
- the same boiler and radiator models were used,
- the model was constructed to be as simple as possible, the plethora of conventional options making it pointless to choose a complex one.

Thus a simple model was constructed, its only requirement being that, as far as possible, it should accurately represent a real system. Doubtless, a more efficient conventional system could be envisaged easily, but this was not the object of the exercise. The only element of the model different to the storage model (other than controls) was the hot water tank, which was simulated as follows, see Fig. 1.11.

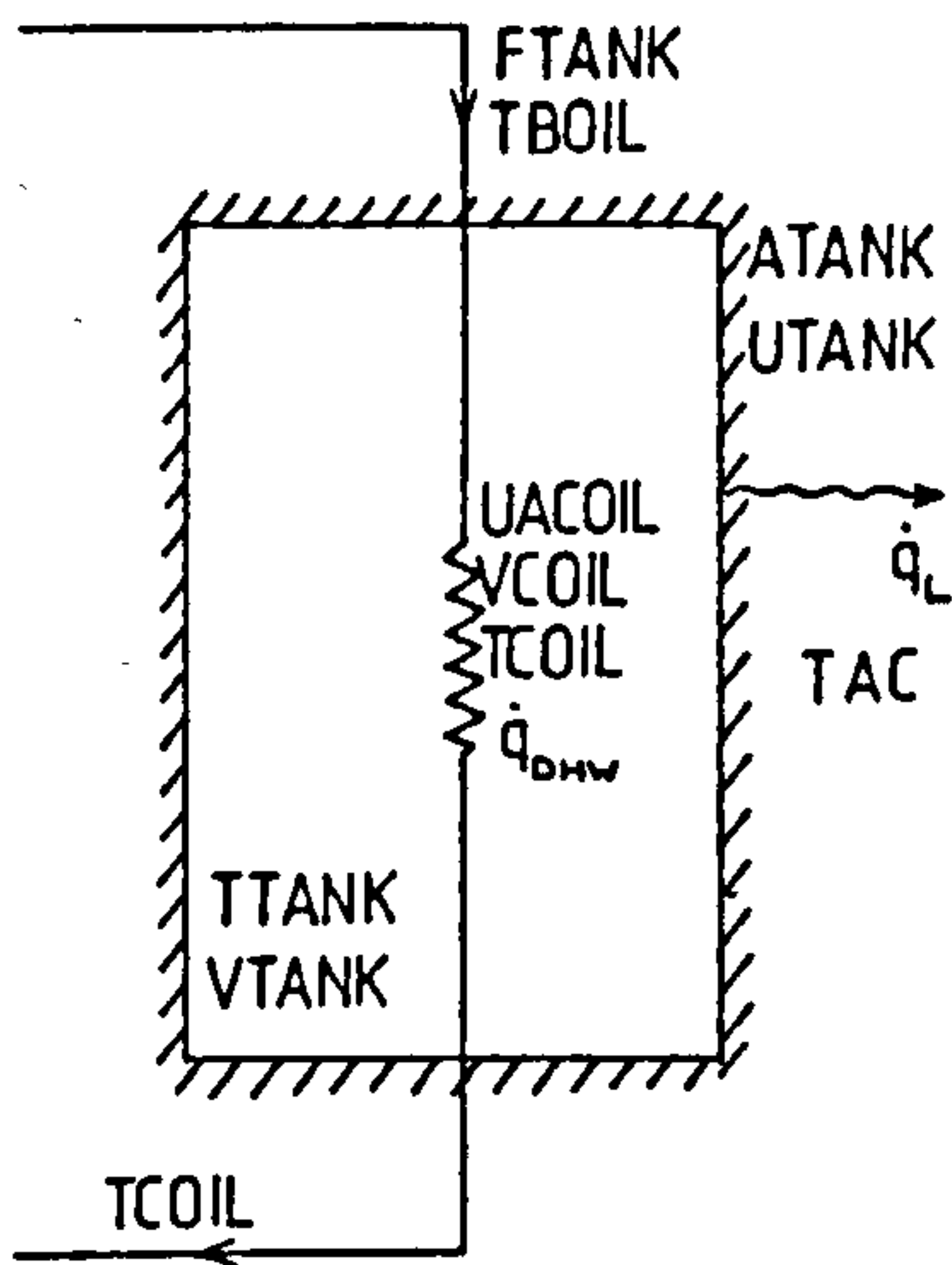


Fig.1.11: D.h.w. tank model

Using the nomenclature of the final heating season simulation program (CHSPS.FOR in Appx. 1.3), hot water from the boiler was pumped at a rate  $FTANK$  and at temperature  $TBOIL$ , into a heat-exchanger coil of heat transfer coefficient  $UACOIL$ , internal volume  $VCOIL$  and temperature  $TCOIL$ , inside the hot-water tank, itself of volume  $VTANK$  at temperature  $TTANK$ . Heat exchange from coil to tank was at a rate of  $\dot{q}_{DHW}$ . The surface area of the cylindrical tank,  $ATANK$ , was insulated to a heat-transfer coefficient  $UTANK$ , losing jacket heat to an airing cupboard at temperature  $TAC$  at a rate  $\dot{q}_L$ . Then:-

$$\text{For the tank surface: } \dot{q}_L = UTANK \cdot ATANK \cdot (TTANK - TAC) \quad (1)$$

For the heat exchange between the coil and the tank:

$$\dot{q}_{DHW} = UACOIL.(TCOIL-TTANK) = VTANK.Cp. \frac{\partial(TTANK)}{\partial t} + \dot{q}_L \quad (2)$$

For the coil contents:

$$FTANK.Cp.(TBOIL-TCOIL) = VCOIL.Cp. \frac{\partial(TCOIL)}{\partial t} + \dot{q}_{DHW} \quad (3)$$

As with the storage program, rates of change of temperature were replaced by forward finite-differences, and the equations (1), (2) and (3) allowed values of TTANK and TCOIL to be computed at the end of each time-step.

So the schematic diagram of the conventional system is as on Fig. 1.12. Note that the hot-water tank was always approximated to being fully-mixed; no attempt at zone division was attempted in this case. For further simplicity, no attempt was made to model the d.h.w. draw-offs; these were approximated to an instantaneous energy transfer to the tap, and the tank was supposed refilled (and cooled) with a corresponding quantity of cold mains water.

The controls of this system were rather critical in that it was necessary to induce a realistic amount of boiler cycling. They were as follows:-

- the radiator received water at boiler outlet temperature, and transferred heat to the rooms until the heat required for that hour was satisfied. At this point the radiator pump was switched off, and the radiators cooled naturally during the remainder of the hour in question. Thus, the first major assumption in this system was effectively that the room thermostat was satisfied once per hour, which is rather fewer than a real system. Further, a "running surplus or deficit" feature was included such that heat shortfalls or surpluses obtained at the end of each hour were added to the requirement of the following hour, as for the storage program. No thermostatic radiator valve was incorporated.



- it was supposed that, during the entire heating period of any given day, the hot water tank was to be heated as required. Heat from the boiler would be received to maintain the tank temperature within the range of the tank thermostat, TSTAT1 to TSTAT2. This thermostat operated the d.h.w. supply pump (not the boiler).

- thus the two pumps, delivering FRAD and FTANK on Fig. 1.12, operated when there was a need for heat in either the radiator or the hot water tank respectively. The boiler, therefore, was subjected to the following possible water flow rates through its heat exchanger:-

Zero, if neither circuit required any heat

FRAD, if radiator alone required heat

FTANK, if hot water tank alone required heat

FRAD, + FTANK, if both radiator and hot water tank required heat.

The boiler fired only if this flow rate was non-zero. When the delivery temperature reached the upper boiler thermostat limit (BSTAT1) the boiler cut out (but the flow was maintained); the boiler re-fired when its thermostat's lower limit was reached (BSTAT2). This was "short-cycling" of the boiler.

Thus a full heating season simulation program was constructed using the three nested loops of the storage program, DELT, HOUR and DAY, and similar printouts. Radiator failures were again recorded, this time on individual hours where heat emitted was substantially lower than that required. The configuration chosen did not produce any failures to meet d.h.w. demand. A full heating season simulation yielded a summarised output similar to that obtained from the storage program, listed in section (i) (d), above. A simple modification again allowed summer runs to be undertaken.

As has been mentioned above, modifications to the layout or to the controls adopted for this model could readily be suggested. The question arising at this point of the study was whether both the storage and conventional c.h. simulation models behaved in a sufficiently realistic manner to afford a reliable comparison of the two suggested system principles. Before making this assessment, it was necessary to choose a set of parameter values for both systems, to establish a "base case" comparison as follows.

## THE "BASE CASE" SIMULATIONS

Having developed the storage and conventional c.h. system programs, it was intended to proceed through a large number of heating season (and annual) simulations to explore the influence on either system of the magnitude of several parameters. Throughout, the critical calculated figure would be the seasonal boiler efficiency, as it has already been suggested that the storage concept should improve this by several percentage points. A figure of secondary interest would, of course, be gas used, to confirm that the storage system actually effected energy savings.

The performance of each system was clearly dependent on many design parameters, e.g. boiler power, store size, thermostat settings, etc. To fully quantify the performance of either system in terms of a reasonable selection of values of all its parameters would have been a massive undertaking. It was therefore proposed to do a "parameter swing" on several key parameters, by running both programs at (normally) three values of the parameter in question, all other parameters remaining constant. The central value of the three had to be the most likely choice; values of all the parameters of both systems had first likewise to be established to set up what came to be known as the "base case" of both systems. One parameter of the base case would then be chosen, and varied to higher and lower values to explore its effect in a "parameter swing".

Arbitrary values were therefore assigned to both systems as in Fig. 1.13. At this time, only tape "3a" of Table 1.1 above was available, and this was chosen as the "base case house", albeit rather smaller than would have been preferred. The "average" d.h.w. schedule tape was used. For the storage "base case", it was decided to select the "Tall-boy" store of 270 litres capacity in preference to the "Mini", which had been tested at Cranfield, ref. 1.4. At the time, a store of this volume appeared to be appropriate, although much smaller stores have more recently be exploited commercially. The "Tall-boy" contained the same d.h.w. heat exchange as the "Mini", so the heat-exchange algorithm was retained. It was decided to "fit" a commercially documented boiler, with its variable power output set to

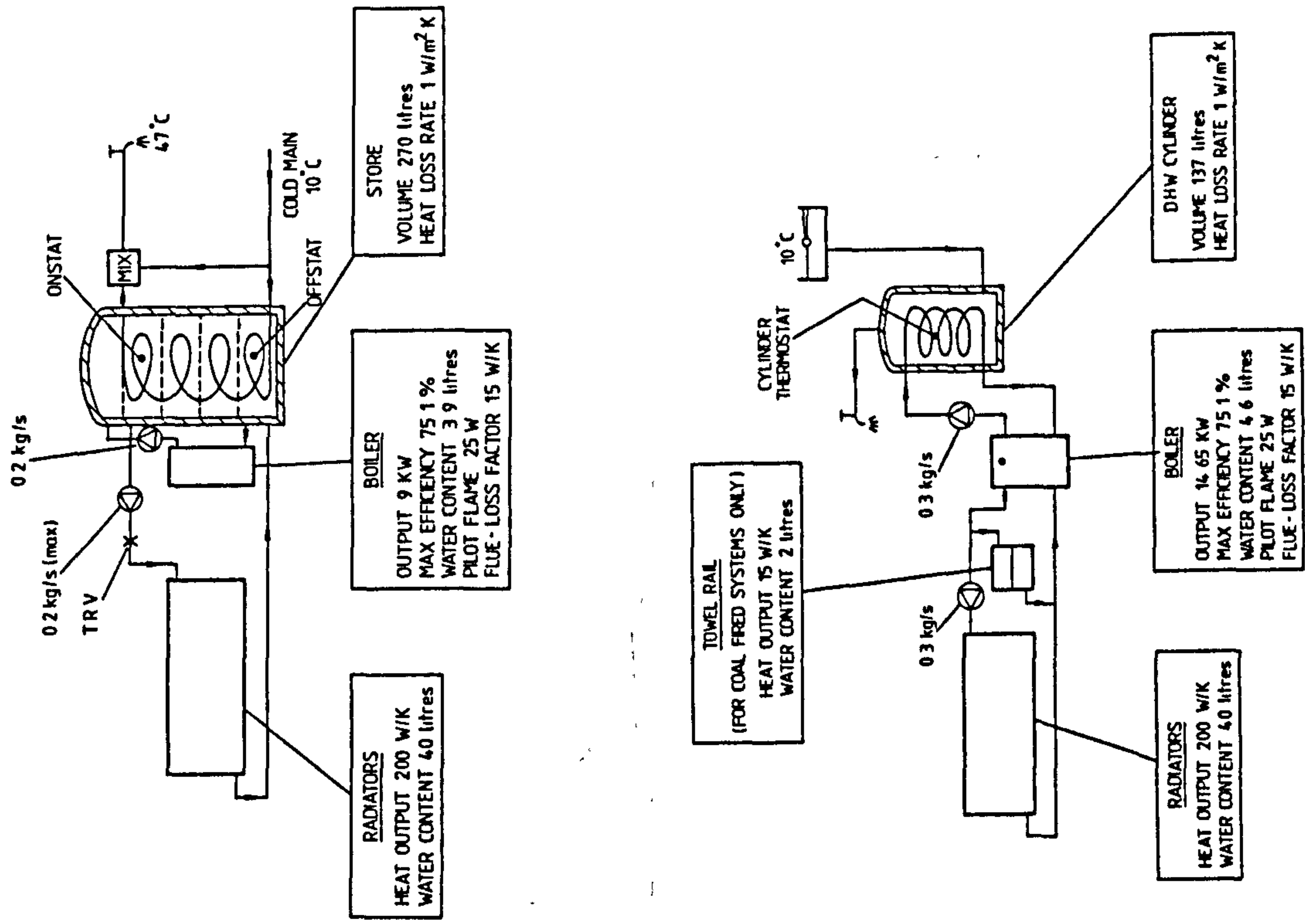


FIG 1.13: PARAMETER VALUES - 'BASE CASES'

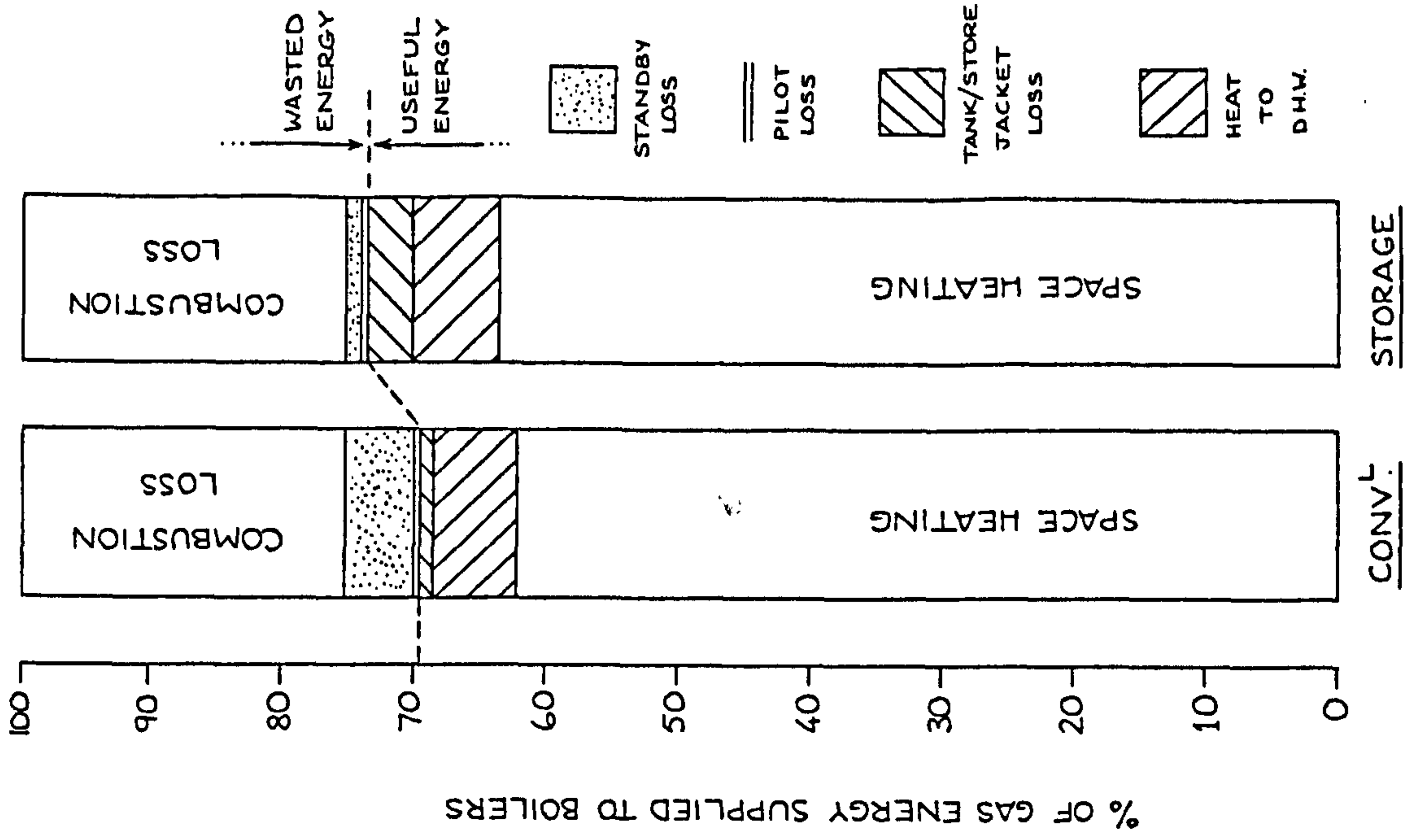


FIG 1.14: 'BASE CASE' RESULTS



9 kW. A pumped flow of 0.2 kg/s was therefore assumed; this is compatible with the British Gas preference for a 10°C differential across the boiler. The heat-loss coefficient to the flue was assumed for the time being to be 15 W/°C - see discussion under "Reliability" section below. After many preliminary runs, the total radiator heat-transfer coefficient was chosen as 200 W/°C, with a water content of 40 litres. A flow of 0.2 kg/s (maximum) was available for the radiator, throttled where appropriate by the thermostatic radiator valve. Both the store of the storage system and the d.h.w. tank of the conventional system were supposed lagged to reduce the jacket-loss coefficient to 1 W/m<sup>2</sup>K.

The conventional system "base-case" differed from the storage system principally in the choice of boiler. This was again a commercially described unit, of output 14.65 kW, i.e. a boiler power ~0.6 times the conventional system boiler power was chosen for the storage system, a figure which was later found to give a good equivalence. Both boilers were of low thermal capacity, and both assumed a full load bench efficiency of 75.1%. The re-adoption of 15 W/°C for heat transfer to flue for the conventional system was slightly charitable, as will be seen. British Gas gave typical values of 137 litres, and 425 W/°C for the d.h.w. tank and heat exchange coil.

Thermostat settings were critical, especially for the conventional system. Ten-degree differentials were adopted for the conventional system's boiler and d.h.w. tank. The thermostats on the store were found to be well placed at 80°C/55°C (OFFSTAT /ONSTAT), but boosted to 82.5°C/65°C for the coldest hundred days of the heating system.

The parameter values chosen for the "base-cases" of the two systems are therefore as in Table 1.3:-

	Storage	Conventional
Boiler output, W	9000	14650
Boiler full load efficiency, %	75.1	75.1
Boiler volume, litres	3.86	4.55
Boiler thermostat, °C	(94)	75/65
Pilot light, W	25	25
Heat transfer to flue, UA, W/°C	15	15
Boiler water flow rate, kg/s	0.2	-
Radiator, UA, W/°C	200	200
Radiator volume, litres	40	40
Radiator water flow rate, kg/s	0.2 (max)	0.3
Water tank flow, kg/s		0.3
Water tank volume, litres		137
Water tank h.e. UA, W/°C		425
Water tank thermostat, °C		60/50
Store and tank skin U-value, W/m <sup>2</sup> °K	1.0	1.0
Store volume, litres	270	
Store d.h.w. h.e. UA, W/°C	1800	
Store OFFSTAT/ONSTAT, °C	80/55	
Store OFFSTAT/ONSTAT, boosted, °C	82.5/65	

Table 1.3 : Base Case Values

When the "base case" systems were run through an entire heating season, using programs HS212 and CHS above, the following analysis was obtained, see Table 1.4:-

	Storage	Conventional
Gas used, kWh	21036.06	21279.18
Heat into system water, kWh	15505.75	14839.57
Efficiency of the boiler, %	73.71	69.74
$\eta_{GU}$ as % of $\eta_{FL}$	98.15	92.86
Overall load factor, %	45.37	30.12
Heat emitted by radiator, kWh	13390.89	13255.84
Heat to d.h.w., kWh	1386.83	1342.32
Store/tank jacket loss, kWh	735.25	241.35
Flue loss, boiler off, kWh	229.32	1065.92
Pilot loss, kWh	83.49	100.06
Boiler cycles	744	42219
Mathematical computing error, %	0.08	0.00039
Failures: (i) Radiator	5	4
(ii) D.h.w.	2	0

Table 1.4 : Results of heating season simulations with "base-cases"

Let us first examine these results before passing on to an assessment of the reliability of the programs. The storage system passed about 700 kWh more heat from the boiler to the heating system (i.e. 4.5% more heat). This is partly due to the nature of the approximations of the programs, whereby radiator heat and d.h.w. heat were always slightly overestimated for the storage system (180 kWh combined in this case). The major cause of this excess heat requirement for the storage system, however, was the store jacket loss incurred here, nearly 500 kWh more than for the conventional system. The significance of this discrepancy will be discussed later (see Parameter Swing on Insulation UA-Value).

Otherwise, the boiler cycles have been radically reduced from some 200 to 3 or 4 per day. This is reflected in over 700 kWh of heat being saved which would otherwise pass out via the flue. The seasonal



boiler efficiency approaches more nearly to maximum efficiency with storage, as the ratio  $\eta_{GU}/\eta_{FL}$  is improved from about 93% to about 98%. Hence a 5% rule-of-thumb improvement is evident at this stage - a most important figure. It is reasonable to deduce that, if radiator and d.h.w. heats were kept strictly equal for the two systems, and the larger store were insulated to equalise the jacket losses, the 5% efficiency advantage would be directly translated to a 5% gas saving.

It is further evident that both systems, as designed, are adequate for the duty imposed here; failures are very infrequent. Lastly, the programs give a good energy balance over the season; the simpler conventional program is virtually perfect in this respect, whereas a 0.5% or so error inherent in the d.h.w. heat-exchanger algorithm of the storage program is "diluted" to 0.08% energy-balance error by the far greater radiator load.

The results may be represented by a simple bar chart showing the distribution of gas energy supplied; this is shown as Fig. 1.14. The storage advantage is shown, but the magnitude of this advantage depends on the usefulness or otherwise of the store jacket loss. This will all be discussed at greater length during the parameter sensitivity study; let us first examine the reliability of the models.

## RELIABILITY OF THE SEASONAL SIMULATION PROGRAMS

It must be borne in mind that the nature of this study was comparative. A comparison of the boiler efficiencies obtained from the heating season simulations was expected to yield a more reliable estimate of gas savings than direct comparison of gas consumption. By way of validation of the models, few data were available even for conventional c.h. systems, and no field trial had been carried out on any storage system in the domestic sector. The following is a brief discussion of the probable adequacy of the programs already presented, from various points of view.

### (i) Results of "Base Case" Simulations

The results appeared reasonable at this stage. The conventional system was cycling at an average of 200 times per day; British Gas had suggested that real boiler cycling was probably between 100 and 300 per day. With the storage system, cycling was reduced to between 1 and 6 per day, i.e. less than one cycle per three hours of heating period. Also the load factor was increased to nearer 50% than 30%. The programs were therefore exhibiting the now familiar two-fold advantage of storage, reflected in the close approach of boiler efficiency to its full-load value. The remaining quantities of Table 1.4 were of the anticipated order of magnitude.

### (ii) Mathematical Validity

Base case results suggested that the magnitude of the inherent storage advantage was of the order of 5%. Thus the models would have to be accurate mathematically to within 1%, so as not to invalidate the results by bad programming. The energy balances of the base case study were well within 1% accuracy (and remained comfortably so for all succeeding simulations).

Very simple forward finite-differences were used for all the first-order differential expressions encountered. No more accurate forms of finite-differences, e.g. central differences, iterations, etc., were used as it was known that some of the parameter values to

be used would be speculative.

The time-step, an important consideration in finite-difference programming, was checked by Robert Cohen as having little effect if its value was maintained within the range 5-20 seconds (ref. 1.6, p.24) as used here.

### (iii) Elements of the Models Chosen

The models of system components were deliberately kept very simple. The following points might be presented as criticisms or possible areas for future improvements to the programs.

The radiator model assumed a linear heat-transfer coefficient, UARAD, an overall, effective, constant coefficient, when the radiating component of the heat emitted from a radiator is known to be very non-linear. In fact a relationship of the sort

$$\text{Heat emitted} = \text{constant } (\Delta T)^{1.3}$$

might have been more appropriate if reliable radiator data had been available at the time. The simple "bath" model, which supposes that all the radiators are at a unique temperature is also plainly a considerable simplification. Thus in the adoption of UARAD = 200 W/°C, it is suggested that this value is compatible with the "bath" model only, and that it could not readily be translated into a real surface coefficient (whose value is likely to be rather lower for the base case house). The volume of 40 litres is felt to be too low, though it was chosen after reading a manufacturer's literature. British Gas suggested that boiler cycling on the room thermostat was typically >2 cycles/hour. The 1 cycle/hour adopted here is thought to be the first of several approximations that slightly favour the predicted performance of the conventional system. As regards the radiator distribution system, British Gas now favour the use of a three-way-valve (3WV) at the outlet of the store, as in Fig. 1.15. A particularly effective further control option is an external compensator (E.C.), which governs the 3WV action to give a delivery



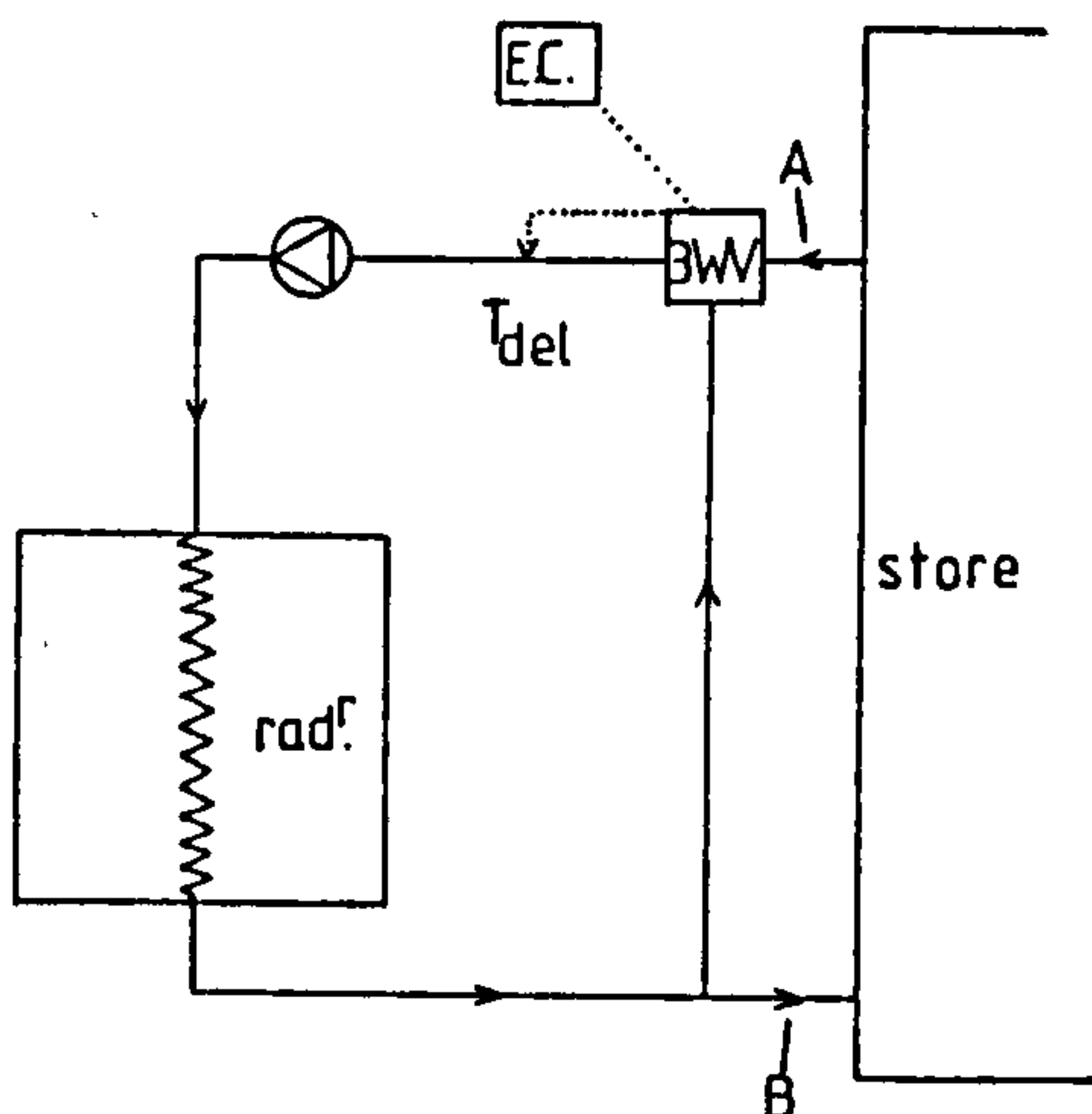


Fig.1.15: 3WV control

temperature ( $T_{del}$ ) appropriate to the environmental ambient temperature outside the house. This system was used successfully on the field trial for the school as reported in part II of this thesis. Thus the flow to and from the store at points A and B is restricted to a trickle at low loads. Though this system would have been simple to program into the storage model, it was argued that the use of a perfect t.r.v. gave an equivalent radiator circuit. Radiator temperature and flow to and from the store

should have been the same for both configurations.

The store and d.h.w. heat exchanger were considered to be adequately modelled for this study, though further checking of the heat exchange algorithm might have led to a better fit to Fig. 1.9 above, and eliminated the persistent  $\sim 1/2\%$  heat-balance error involved. It was suspected, though, that the cumbersome heat exchanger would eventually be redesigned and so further investigation of the algorithm would have been unwarranted. The author regrets not having the opportunity, at a later stage, of remodelling the heat exchanger for the design incorporated in the commercially-exploited "Cormorant" (ref. 1.7). A further possible improvement would be to investigate the results of Robert Cohen's previous storage experience (ref. 1.8), and assess whether a more sophisticated zone division of the many store sizes tried should be incorporated in the program. With such a knowledge, a stratified model of the d.h.w. tank could be developed, though this would probably affect only slightly the results here.

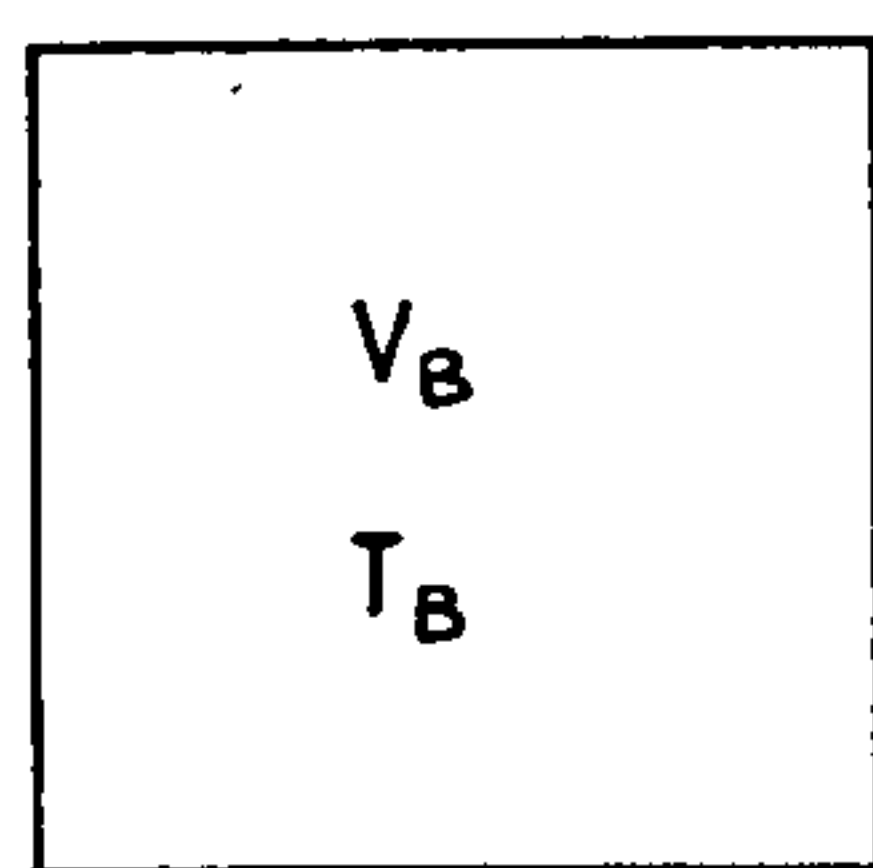
On the conventional system model, performance was likely to be highly dependent on thermostat settings. The values of  $75^{\circ}/65^{\circ}\text{C}$  (boiler) and  $60^{\circ}/50^{\circ}\text{C}$  were selected on the advice of colleagues at

British Gas; certainly it was necessary to adopt one set and persist with it. A ten-degree differential on both thermostats is probably favourable to this system, so that it will probably produce an optimistically low number of cycles, and hence if anything enhance the boiler's seasonal efficiency.

The adequacy of the boiler model has been briefly discussed in paragraph (b) of section (i) above. The author is of the opinion that the weaknesses of the programs presented so far do not invalidate the results, but are mere "fleabites", as it were, compared with the uncertainty as to the value of the boiler heat-transfer coefficient to the flue (after shut-down) UAFLUE, or UAB in the storage program. A justification of the choice of a value of 15 W/K in both programs will now follow.

(iv) Heat Loss to Flue (Boiler Standing Loss)

The reader is reminded that it was hoped to model the boiler with the simplest of models, see Fig. 1.16; after shut-down (a.s.d.), the



$T_F$  boiler, of water equivalent  $V_B$  and at temperature  $T_B$  would cool down at a rate determined by an overall heat transfer coefficient UAFLUE (called UAB in the storage program).  $T_F$  would be a representative flue temperature, to which heat was lost. The choice of a reliable

value for the constant UAFLUE provided the most animated discussion of this study. In the base case study above, a value of 15 W/°C was chosen for both of the boilers incorporated in the storage and conventional programs; the reasoning behind this was as follows:-

Armed with the newly-constructed annual simulation programs, it was necessary to find out whether the value of UAFLUE was of any significance. Thus the base-case systems, described above, were run through the 212-day heating season, varying values of UAFLUE (alone) from 7.5 W/°C to 60 W/°C, a range which, it was hoped, would include the most appropriate value(s) of UAFLUE (should, indeed, a unique value exist). This produced the most important plot, Fig. 1.17. As

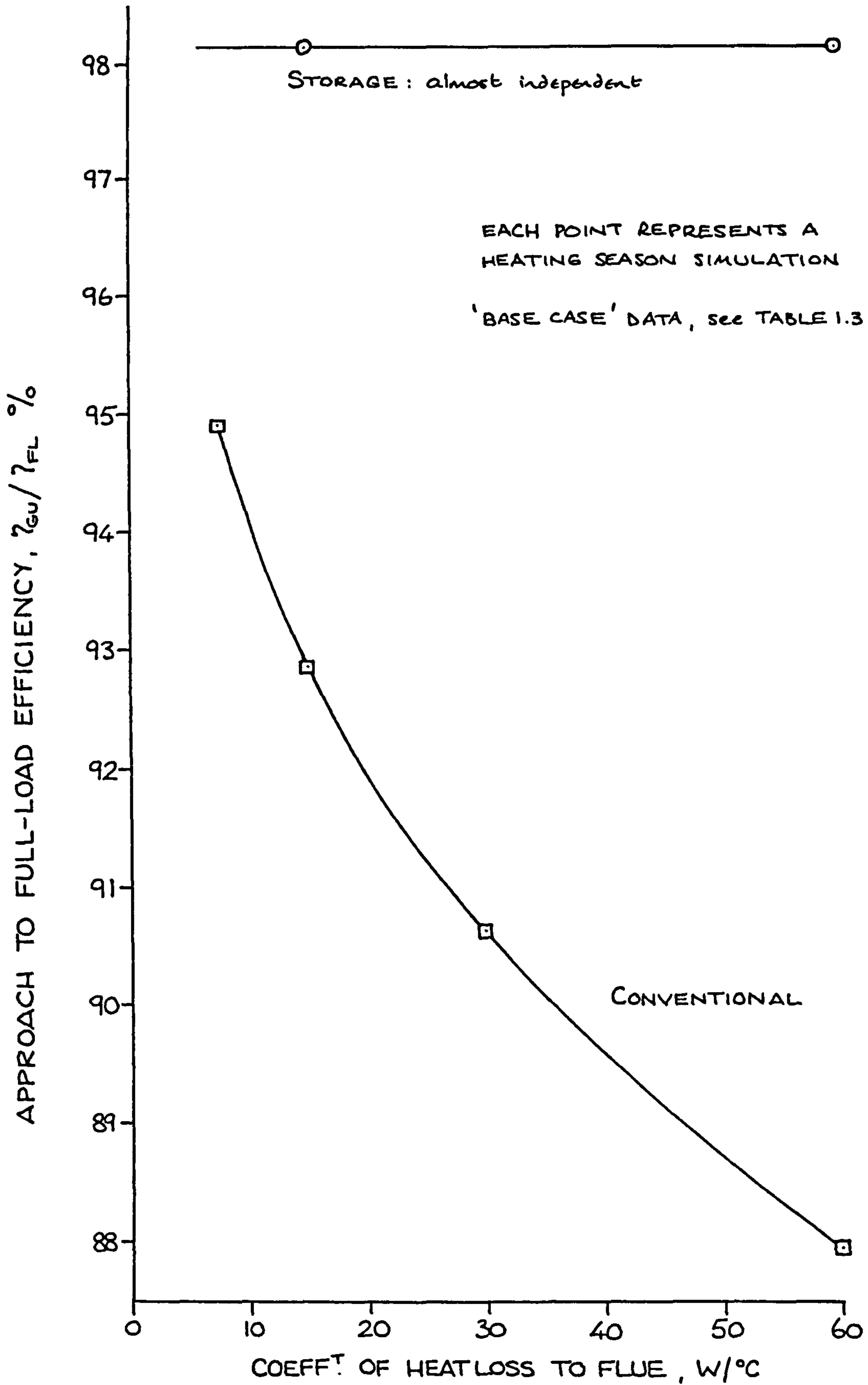


FIG 1.17 : DEPENDENCE OF CONVENTIONAL SYSTEM  
ON RATE OF HEAT LOSS TO FLUE, 'A.S.D.'



above, we are more concerned with boiler efficiency than with gas totals, so the ratio  $\eta_{GU}/\eta_{FL}$  was plotted against the swing of UAFLUE values.

The first very important conclusion to be drawn from Fig. 1.17 is that a storage system will maintain its very close approach to maximum efficiency, almost independent of UAFLUE-value. The reason for this is that cycling is so slow for the storage system that the boiler will lose almost all its heat a.s.d. whatever the UAFLUE-value, because the time between switch-off and switch-on is likely to be long (perhaps several hours). So the storage system is insensitive to rate of heat loss a.s.d., but more sensitive to the number of cycles.

The second conclusion from Fig. 1.17 is that the performance of the conventional system is highly dependent on UAFLUE-value, and thereby on the rate of cooling. For all subsequent modelling therefore, it was necessary to find a good, representative value of UAFLUE for conventional system modelling.

With almost no information to hand at this stage, the following considerations were evaluated:-

- In ref. 1.9, Bennett and Barooah list the details of several boiler types, including a "Heat Transfer Coefficient - Metal/Air". The values quoted suggest that UAFLUE might be very small, perhaps even below  $5 \text{ W/}^\circ\text{C}$ . These values were discussed at length, and it was thought that the quoted figures referred to a steady-state heat transfer (i.e. stagnant air). Intuitively, it was thought that the representative value of UAFLUE sought in this context might be several times a "stagnant air" figure, due to the likely buoyancy effect and turbulence in the flue at and shortly after shut-down.

- In ref. 1.10, R. Dann of British Gas, Watson House, presents a cooling curve (a.s.d.) for a suitable boiler, based on tests at Watson House. Examination of this graph showed firstly that it was very nearly exponential, which would justify the use of a single, overall UAFLUE-value. Secondly, the cooling rate as estimated from this curve suggested a value of UAFLUE of about  $29 \text{ W/}^\circ\text{C}$  for the boiler chosen for

the conventional system base case. Thus the likely value of UAFLUE had been narrowed to within the range 5-29 W/°C, still far too large a span for reliable simulations.

- The Open University were at this time conducting field trials of lightweight conventional boilers over full heating seasons in Milton Keynes. Upon contacting them, they were kind enough to give certain details of preliminary results of these tests, from which it was apparent that a typical heating season value of  $\eta_{GU}/\eta_{FL}$  was emerging as being some 91-92%, ref. 1.11. Referring to Fig. 1.17, this value would suggest a UAFLUE value of between 20 and 26 W/°C.

- At B.S.R.I.A., seasonal boiler efficiencies were evaluated for a low thermal capacity boiler, using laboratory test cycles to simulate the duties of specimen days, ref. 1.12. For combined space and d.h.w. heating, the seasonal values for  $\eta_{GU}/\eta_{FL}$  quoted was 92.9%, which suggests a UAFLUE value of some 15 W/°C, from Fig. 1.17. This value, though, referred to a boiler of only 0.18 litres, compared to the capacity used for the simulations here, 4.55 litres, and so might be considered optimistic.

It now remained to choose the single value of UAFLUE which would subsequently be used in the conventional system simulations. A realistic working figure of 20-25 W/°C seemed justifiable. The author, however, was anxious to assess the minimum inherent advantage, achievable by applying the storage principle, such that a conservative estimate could be published, in the anticipation that a real system would prove slightly more advantageous. Remembering the very low likely value of the steady-state coefficient, a figure of 15 W/°C was adopted for UAFLUE as being the minimum operating overall heat-transfer coefficient. Thus predictions using 15 W/°C are hoped to be, if anything, favourable to the conventional system, as has been the "policy" adopted in other considerations already described. Percentage efficiency benefits of storage will henceforth be taken as a minimum, with an expectation that up to 1 1/2 % further advantage (from Fig. 1.17) may be realised in practice.

With the programs and base case designs thus set up to be as

accurate and reliable as was possible at the time, it was intended to examine the sensitivity of each system to a variation in the value of a series of key parameters. A basic expectation of a minimum storage advantage of 5% had been proposed; would this advantage be maintained if the arbitrarily chosen parameter values of the base cases were altered?



## PARAMETER SENSITIVITY STUDY

A total of eight principal parameters were varied one-by-one. These were as follows, with the appropriate figure and table numbers:-

- (i) D.h.w. demand (Fig. 1.18),
- (ii) Store and tank lagging U-value (Fig. 1.19)
- (iii) Radiator UA-value (Fig. 1.20)
- (iv) Boiler power and thermal capacity (high or low) (Figs. 1.21, 1.21a)
- (v) Store volume (Fig. 1.22)
- (vi) Store thermostat settings (Fig. 1.23)
- (vii) Building size (space heating demand), (Fig. 1.24)
- (viii) Occupancy pattern (space heating period), (Fig. 1.25)

In addition, the storage program was expanded to demonstrate its use as an aid for designing a storage system appropriate for a given house and d.h.w. demand, (ix), Fig.1.26.

The figures have been included in the text of this section of the thesis.

### (i) D.H.W. Demand

The "base-case" systems, again supposed fitted in the house described by tape "3a" above, were run through three annual heating season simulations (i.e. including the summer months) to obtain Fig. 1.18. The boiler efficiency is seen to remain almost invariant as the d.h.w. demand increases from the "couple" demand (616 litres/week) to the "family" demand (1336 litres/week). Note that the storage system's approach to maximum boiler efficiency is about 97.6% for an annual simulation, whereas the above base case analysis gave a figure of 98.15%. Thus the less efficient summer operation has caused a half percent or so drop in this approach to maximum efficiency. The storage system's 5% advantage is seen to be, if anything, increased.

Boiler efficiency is very close to the annual distribution efficiency, the difference being due to slight programming inaccuracies. This distribution efficiency would assume, however,

N.B. THESE ARE ANNUAL EFFICIENCIES

KEY: ——— STORAGE

----- CONVENTIONAL

(i) ○ BOILER EFFICIENCY

(ii) □ A DISTRIBUTION EFF?

$$\eta_1 = \frac{R + D + J_w}{G}$$

WHERE R IS RADIATED HEAT

D IS HEAT TO D.H.W.

$J_w$  IS STORE/TANK JACKET

LOSS, IN WINTER ONLY

G IS TOTAL GAS INPUT

(iii) △ A SECOND DISTRIBUTION EFFICIENCY,  $\eta_2$  WHERE

$$\eta_2 = \frac{R + D}{G}$$

WHICH ASSUMES ALL JACKET LOSS WASTED.

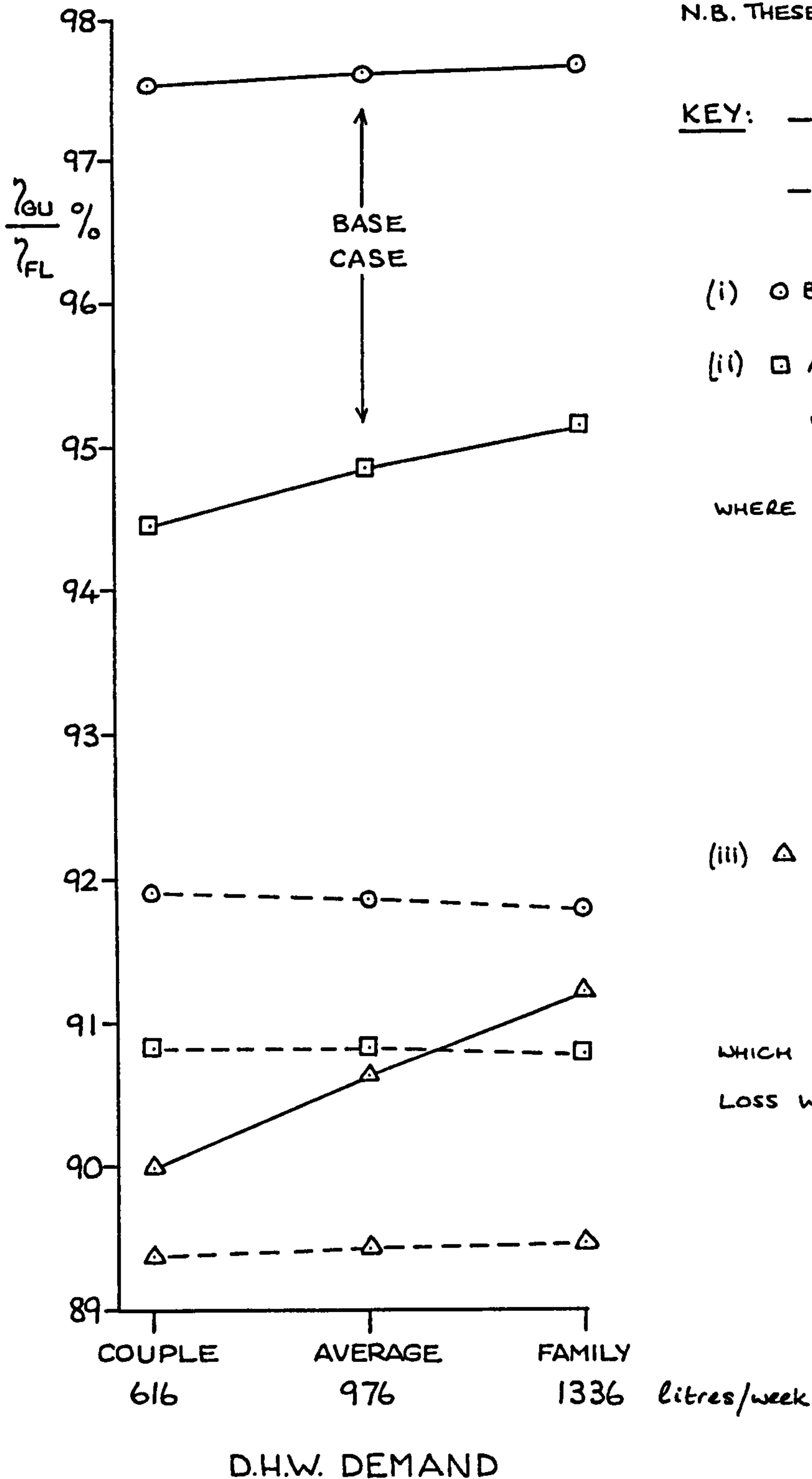


FIG 1.18 : EFFECT OF D.H.W. DEMAND ON ANNUAL OPERATING EFFICIENCIES

that store jacket loss is useful all the year round, which may well be the case. Supposing that the store or tank jacket loss is wasted in summer, the storage system's advantage (because the store chosen was far larger than the hot water tank of the conventional system) is reduced to some 3.5-4%. If the jacket losses are wholly wasted (which is unlikely), the storage advantage is reduced to 0.6-1.8%. The definitions are clearly arbitrary. A criticism sometimes levelled at the storage concept is that the energy saved at the boiler is lost through the store jacket. The study shows how this criticism arises, but careful placing of the store, and, if necessary, further insulation should avoid such an erosion of the storage advantage.

In general, the efficiency is seen to be little dependent on d.h.w. demand. Failures were insignificant throughout this study.

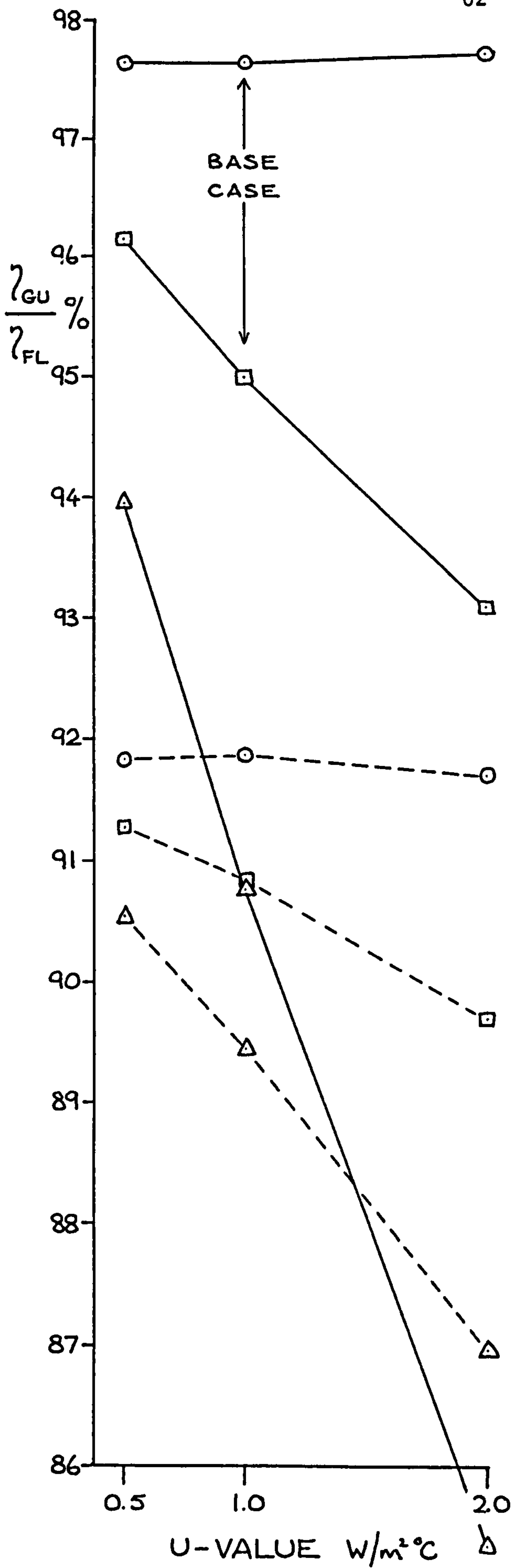
#### (ii) Store/Tank Insulation

The above d.h.w. study prompted an investigation of the effect of store or tank insulation thickness (or U-value), this time using the "Average" d.h.w. demand throughout. Simulated insulation thicknesses of 10, 5, and 2.5 cm of standard insulation, equivalent to U-values of 0.5, 1.0 and 2.0  $\text{W/m}^2\text{K}$  were explored, see Fig. 1.19. Again, annual boiler and distribution efficiencies are presented.

Taking the full boiler efficiencies first (circles), the values are virtually unaffected by the insulation U-value, and the storage advantage of well over 5% is maintained. Remembering that the store has an area of some 1.86 times that of the tank, then as the insulation thickness is reduced, the storage system's distribution efficiency falls off much faster than that of the conventional system if all or part of the jacket loss is supposed wasted. The situation is exacerbated by the fact that the store is charged to beyond about 85°C, whereas the tank's temperature does not exceed 60°C.

This study, and the preceding one, indicate that if excessive jacket loss is a problem (a choice which is under the control of the designer) and if the store is significantly larger than the equivalent tank, then an additional thickness of lagging will be required by the





N.B. THESE ARE ANNUAL EFFICIENCIES

KEY: AS FOR FIG 1.18

0.5  $W/m^2°C$   $\doteq$  10cm INSULATION THICKNESS

1.0 "  $\doteq$  5cm "

2.0 "  $\doteq$  2.5cm "

FIG 1.19 : EFFECT OF STORE/  
TANK JACKET  
U-VALUE ON ANNUAL  
OPERATING  
EFFICIENCIES

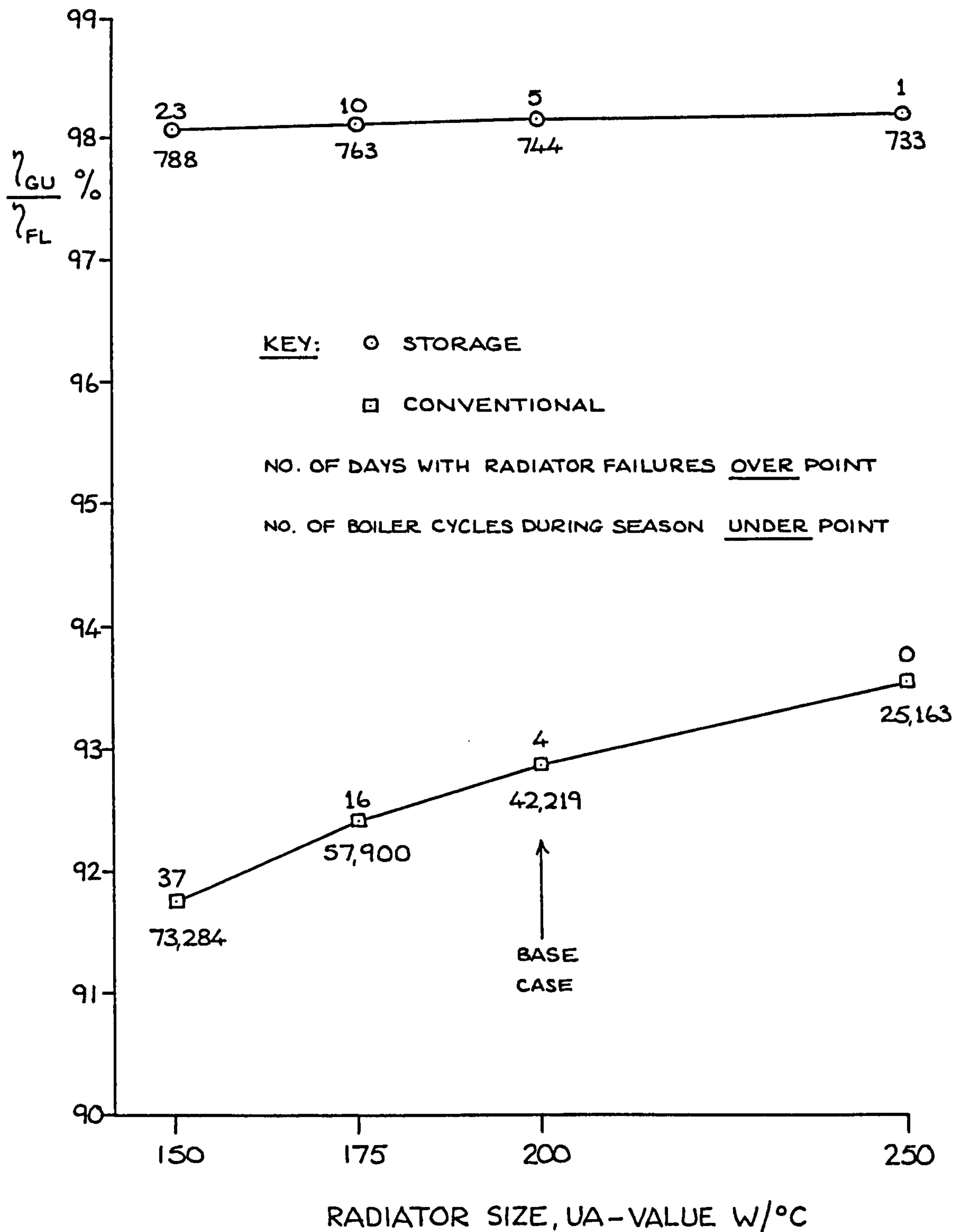


FIG 1.20: EFFECT OF RADIATOR SIZE ON HEATING SEASON BOILER EFFICIENCY

store. Jacket loss is usually useful, in an airing cupboard for example; indeed, it might be subtracted from the house heating load during the heating period and thus be very largely useful. It is certain that with suitable design the store's jacket loss need not be any more of a penalty to a system than that of the replaced hot-water tank.

### (iii) Radiator Size

The radiator's UA-value was varied from 150 to 250 W/°C (the base case was 200 W/°C): the radiator volume was assumed to be 0.2 litres per unit of UA. Simulations for the heating season only (of course) were considered for this study, see Fig. 1.20.

The storage system is seen to be insensitive to radiator size, there being only a marginal increase in cycling and decrease in efficiency as the radiator size is diminished. The conventional system is more sensitive to a reduction of radiator capacity, though not disastrously so. With an undersized radiator, the conventional system cycles excessively. Both systems record an unacceptable number of radiator failures as the radiator size is reduced; there is a suggestion that the least acceptable radiator size is smaller for the storage system than for the conventional system.

### (iv) Boiler Size and Type

(a) Traditional Boilers: It was desired to compare lightweight boiler performance with heavyweight boiler performance, in each case assuming that the boilers could be used throughout a wide power range. The lightweight boilers were taken to be those of the base-case designs above, i.e. 9 kW, 3.86 litres and 14.65 kW, 4.55 litres respectively for the storage and conventional systems. Swung through the power settings on either side of their original powers, the lightweight lines of performance are the solid lines on Fig. 1.21. Heavyweight boilers were each taken to be of 12 litres water equivalent, and UAFLUE-values were sought such that the cooling time constant of each boiler (a.s.d.) i.e.  $V_B C_V / UAFLUE$  remained as for the lightweight boilers. The heavyweight boilers gave the dotted lines of



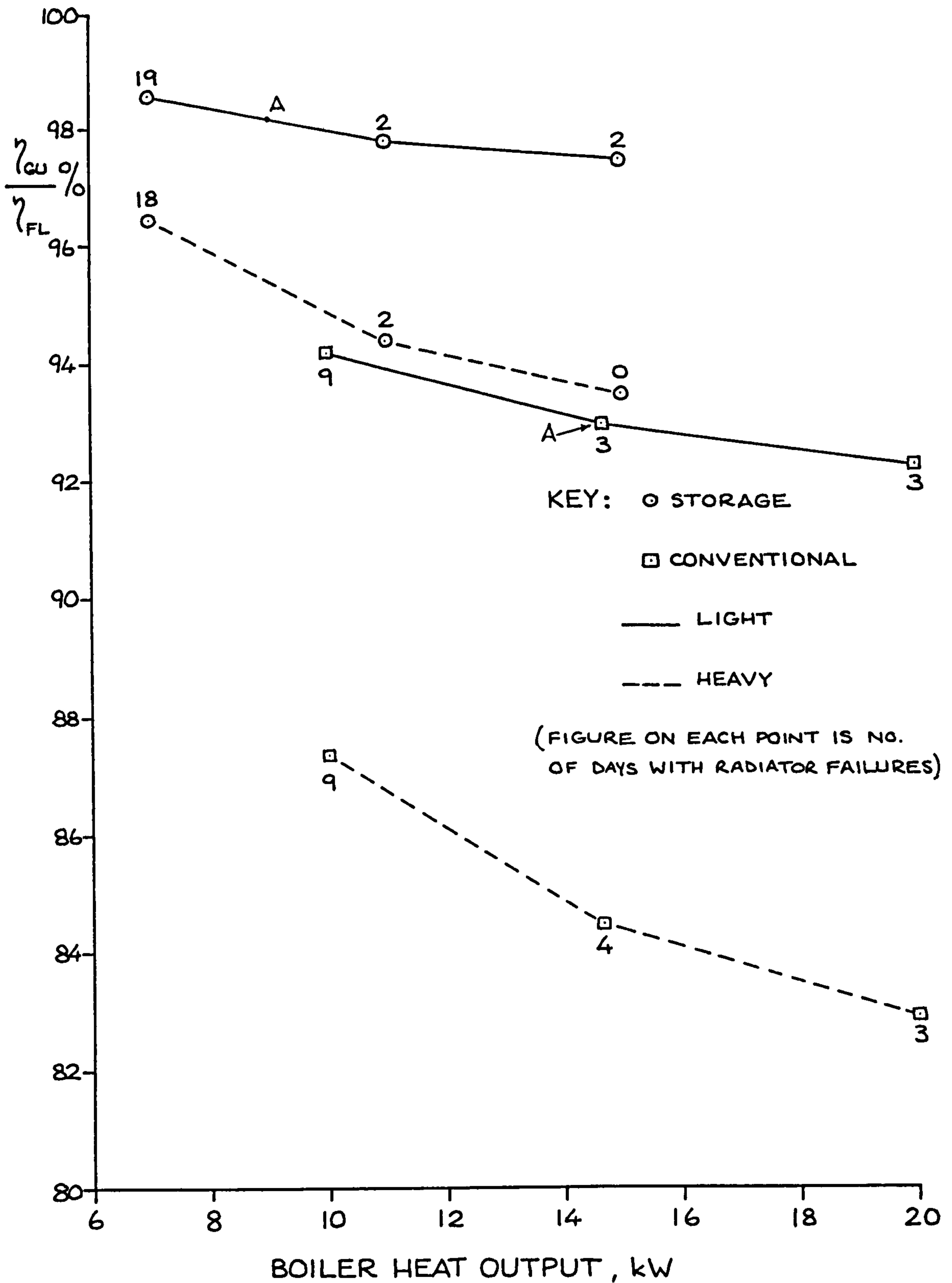


FIG 1.21 : EFFECT OF BOILER POWER AND THERMAL CAPACITY ON HEATING SEASON EFFICIENCY - TRADITIONAL BOILER.

Fig. 1.21. The graph is a little confusing at first glance; note that points A correspond to the base-case boilers. Note also that there is no vertical equivalence when comparing storage and conventional systems; a storage system boiler might be two-thirds the size of its conventional system equivalent. The following observations are drawn from Fig. 1.21.

- All four lines fall as the boiler rating increases, i.e. the seasonal efficiency is invariably impaired by increasing boiler output. Conversely, if boiler power is reduced, a threshold power is reached beyond which radiator failures became unacceptable.
- For conventional systems, over-rating the boiler induces a larger reduction in the boiler's efficiency for a heavyweight than for a lightweight boiler; a heavyweight boiler is likely to be some 9% worse with respect to seasonal efficiency.
- For storage systems, over-rating the boiler has a smaller effect than for conventional systems; a heavyweight boiler is here likely to be about 3.5% worse than a lightweight boiler.
- It would appear that, if a designer had to use a heavyweight boiler, the storage option would be very attractive indeed, perhaps with a 10% advantage over the equivalent conventional system. (All the other assessments of this study refer to lightweight boilers, whose 5% storage advantage is again borne out by Fig. 1.21.)

(b) **Condensing Boilers:** One of the most exciting developments in the field of gas central heating in recent years has been the advent of the condensing boiler, which has an efficiency characteristic rising sharply at low return temperatures to a value well in excess of 90%. The device is thus well suited to large low-temperature radiator systems, as are favoured on the Continent. At the moment, the condensing boiler has been widely accepted in Europe, but not so readily in England, despite the development of a British version (called "Trisave" ref. 1.13). "Trisave" is, by all accounts, an excellent boiler, and is available at a price not too much in excess of its conventional counterparts. It is, perhaps, a matter of time

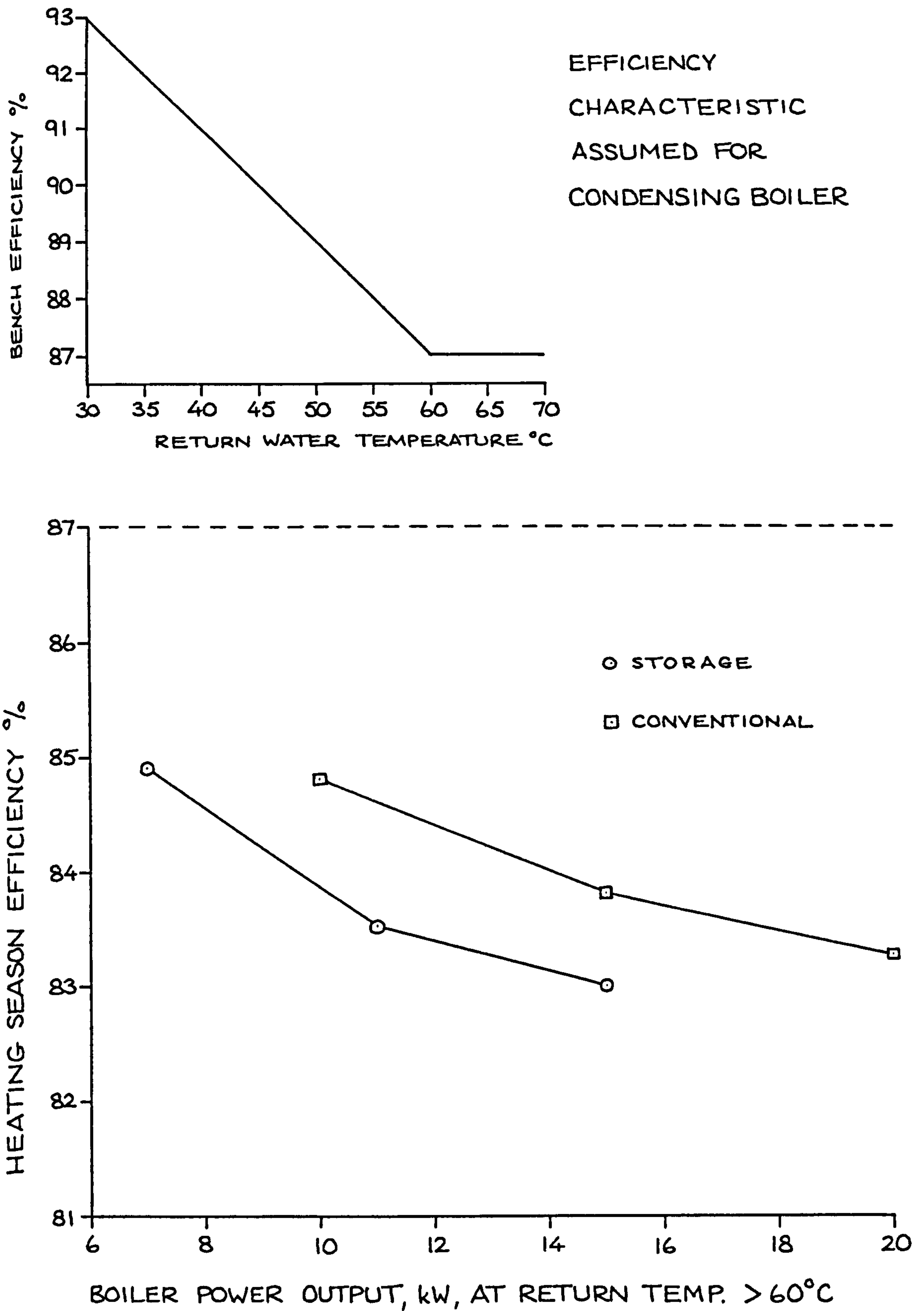


FIG 1.21a : EFFECT OF BOILER POWER ON HEATING

SEASON EFFICIENCY - CONDENSING BOILER

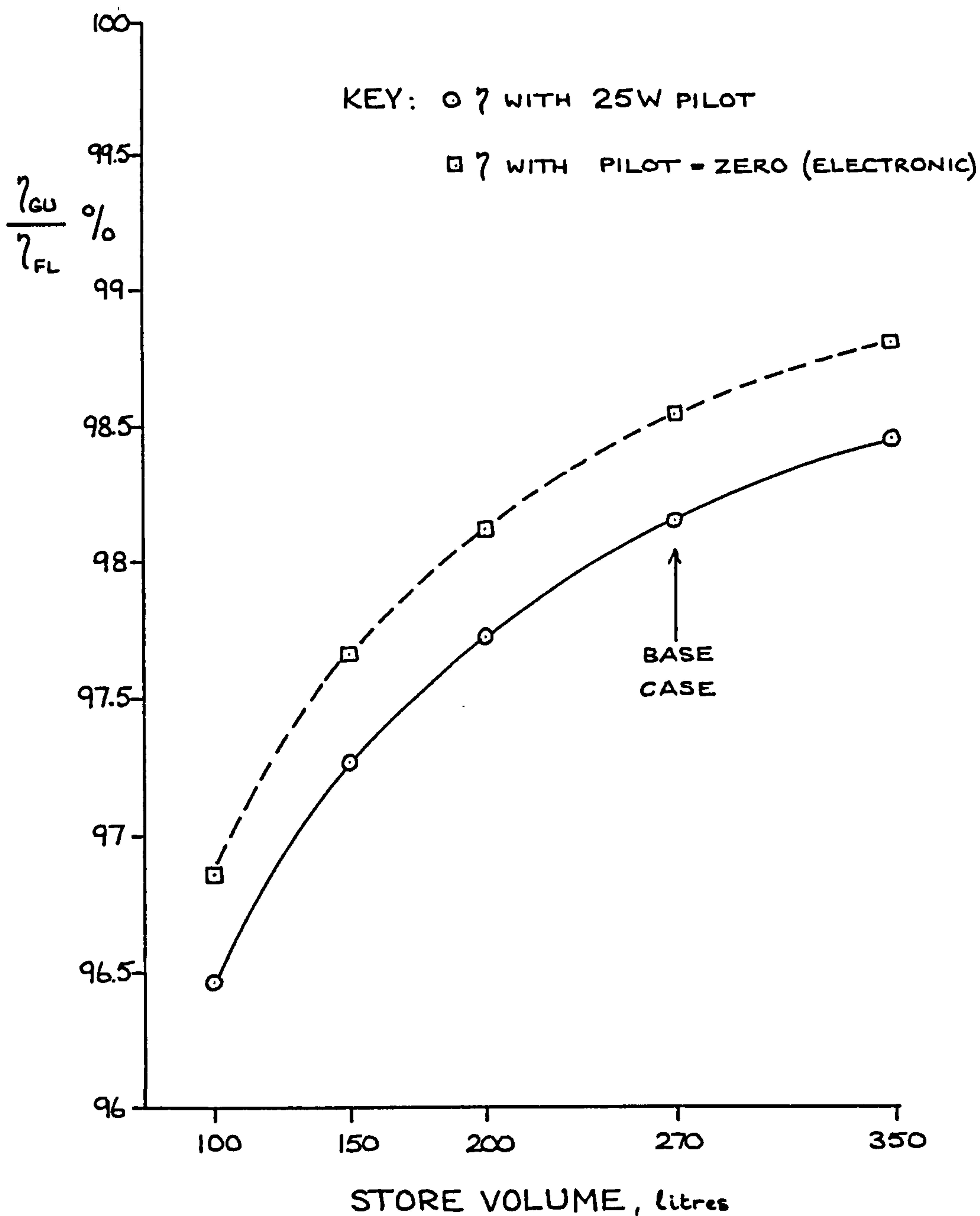


before the condensing boiler makes considerable inroads into British central-heating practices.

In that the condensing boiler operates best at low temperature, it is perhaps less well suited in a storage context as, to reduce cycling, it is favourable to heat the store to very high temperatures. An attempt was made, however, to set up a condensing boiler model based on the rather scant data available at the time. British Gas suggested that a condensing boiler known to them (Watson House) had the characteristic shown on the upper half of Fig. 1.21a. Its thermal mass was approximately 16 kg water equivalent. It had an upward damped flue (whereas the "Trisave" does not), and the author chose a figure of  $5 \text{ W/}^\circ\text{C}$  for UAFLUE with no more justification than that ref. 1.10 suggests a 1% of rated power heat loss to flue for a damped-flue boiler. This altogether speculative assessment of the condensing boiler yielded the heating season simulations whose boiler efficiencies are plotted on the lower graph of Fig. 1.21a. With these approximations, there appears to be little if any advantage of incorporating storage with a condensing boiler, unless the capital cost of a smaller boiler warrants the inclusion of a store. It must be stressed that this is an altogether preliminary study.

#### (v) Store Capacity

Believing that the store volume of 270 litres was rather high, the base-case design was simulated with store volumes of 100, 150, 200 and 350 litres. The same d.h.w. heat exchanger was assumed. The results are presented in Fig. 1.22. As anticipated, a bigger store enhances the efficiency, but diminishing returns mean that there is little to be gained in increasing store size from 250 litres to 350 litres say (but the larger store might well require a smaller boiler - see (ix) below). A 2% improvement is observed overall. Somewhat remarkably, in spite of a rapidly increasing rate of boiler cycling as the store becomes small, the 100 litre store appears to provide radiator heat as well as the 350 litre store. The smaller stores, however, cannot satisfy d.h.w. demands, and failures here become unacceptable. The upper curve of Fig. 1.22 assumes that pilot gas loss is zero, i.e. spark ignition.



D.H.W. 34	13	5	2	1
FAILURES / RAD. <sup>R</sup> 7	3	3	5	6
BOIL. <sup>R</sup> CYCLES 1591	955	954	744	629

FIG 1.22 : EFFECT OF STORE VOLUME ON HEATING SEASON EFFICIENCY

(vi) Store Thermostat Settings

The store was always taken to be fitted with two thermostats, OFFSTAT and ONSTAT as above, whose settings could be "tuned" for optimal operation, see Fig. 1.23. Too low a setting risks failures; too high a setting induces high jacket losses. If the thermostats are set at temperatures close together, the rate of cycling is increased. It has been suggested that to reduce cycling to a minimum, OFFSTAT might be dispensed with altogether, and the boiler thermostat set up to switch control to ONSTAT. This would allow very high charge temperatures to be attained, with long, infrequent cycles, but would incur a high jacket loss unless lagging were sufficient. Fig. 1.23 shows a progressive "tuning" to the base case design; further tuning might be possible. The annual boiler efficiency is improved by 1% overall. It is interesting to note that the heating-season efficiency of the storage system is reduced by the summer efficiency to give an annual efficiency some 1/2% lower. From e.g. Figs 1.19 and 1.20, the corresponding drop for the conventional system was about 1%.

For the remaining parameter sensitivity studies, the base-case design was discarded, as suitable variants on the data tape "3a" were not available, as will become apparent. Heating seasons only were simulated.

(vii) Building Size

Data tapes 1, 2, 5 and 6 were by now available, and used to investigate the effect of building size on the storage/conventional system comparison. All these tapes represented a 17-hour heating period, and supposed a four-person occupancy which prompted the use of the "family" d.h.w. schedule in all simulations. The results appear as Fig. 1.24.

No attempt was made to rationalise the store-to-radiator ratio, for example, the system designs being selected rather randomly using convenient component sizes. The with-storage boiler efficiency line is therefore seen to decrease by about 1% as the building seasonal space heating requirement is doubled over the span of the simulations.



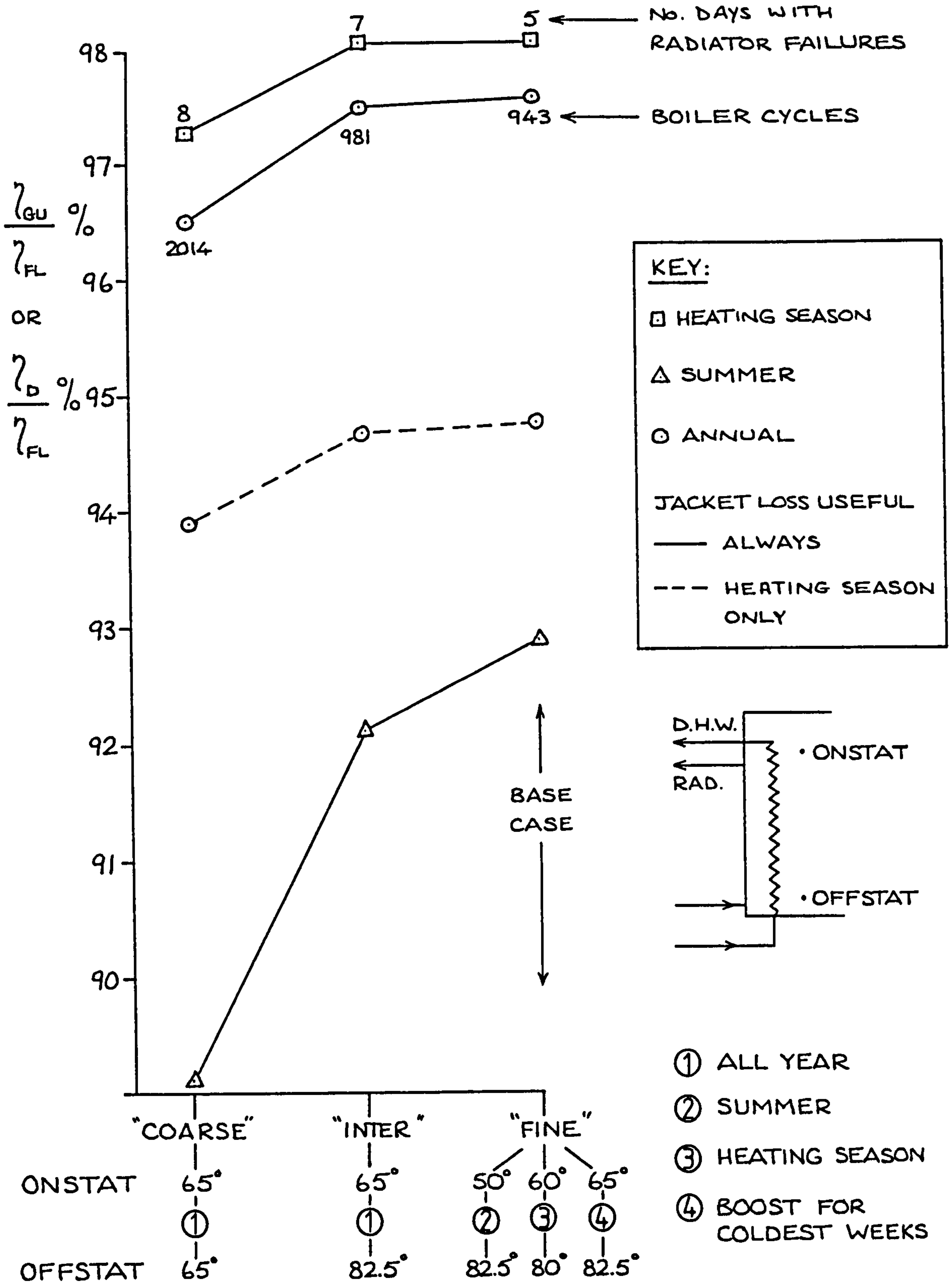


FIG 1.23 : EFFECT OF STORE THERMOSTAT SETTING OPTIMISATION.

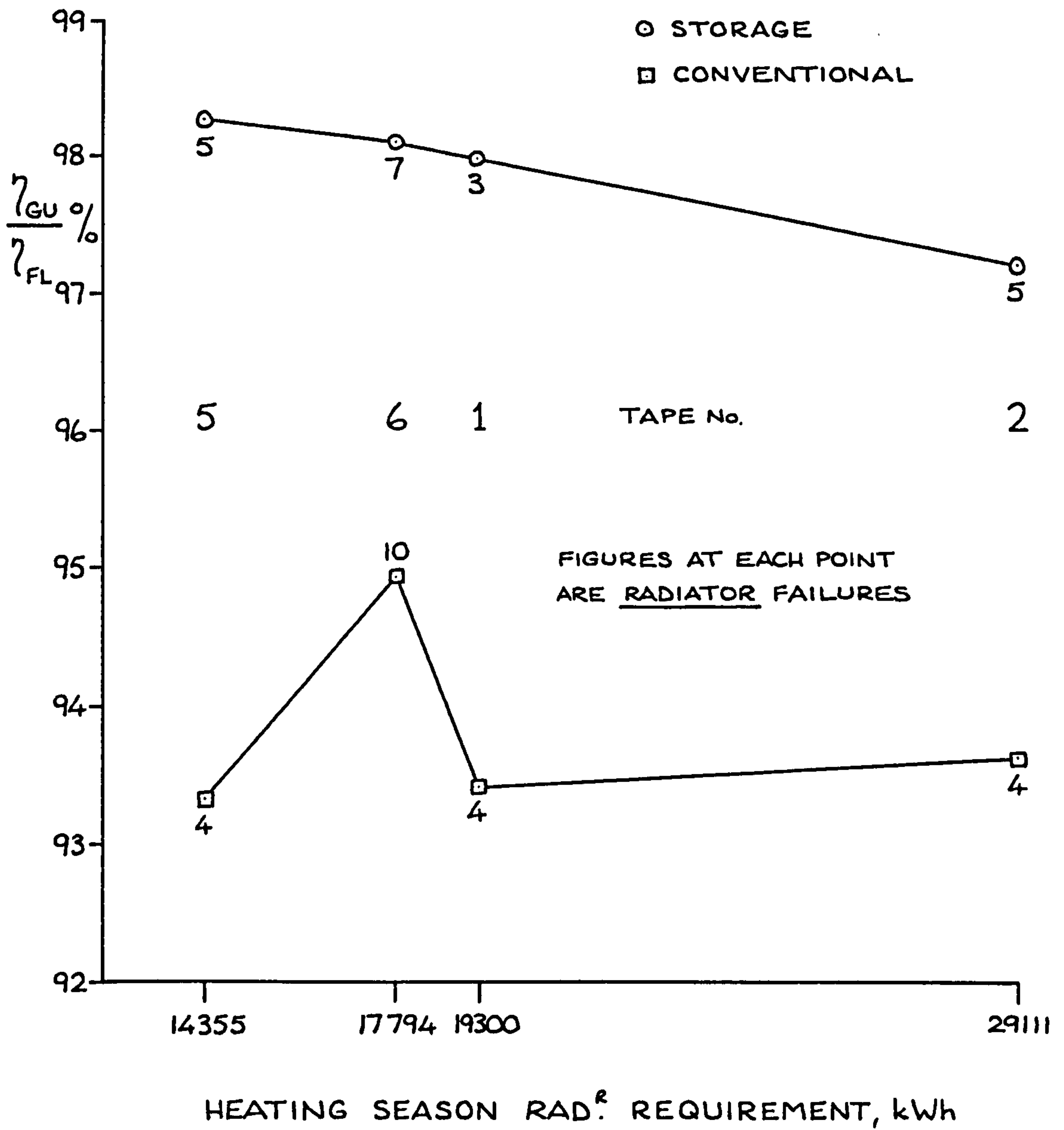


FIG 1.24: EFFECT OF BUILDING SIZE

The largest house modelled (tape 2) would have required a very large store indeed to maintain the same store volume/radiator demand ratio as that supposed for the house modelled by tape 5. Had this huge store been assumed, it is probable that the boiler efficiency of the storage system would have remained independent of house size. It may well be that an upper acceptable limit on store size (due solely to physical considerations) of say 400 litres would slightly penalise a storage system to be fitted into a large house. Though the store size chosen was proportionally smaller for the large house modelled, there is no indication that the adequacy of the radiator system is in any way impaired. This emphasises the conclusion of study (v) above, where it was found that radiator failures were independent of store size. Thus, to fulfil space heating demands, boiler rating is the more important design parameter.

The conventional system shows a non-linearity of its boiler efficiency line, at a point where the boiler rating chosen was rather undersized, hence giving a good efficiency at the expense of too many occurrences of radiator failure. Had the boiler rating been rather larger, the four failures of the other three simulations would have been achieved, but boiler efficiency would have been reduced.

This study does not suggest that the performance of a well-designed system of either sort need be affected by house size. The 5% storage advantage should be available throughout.

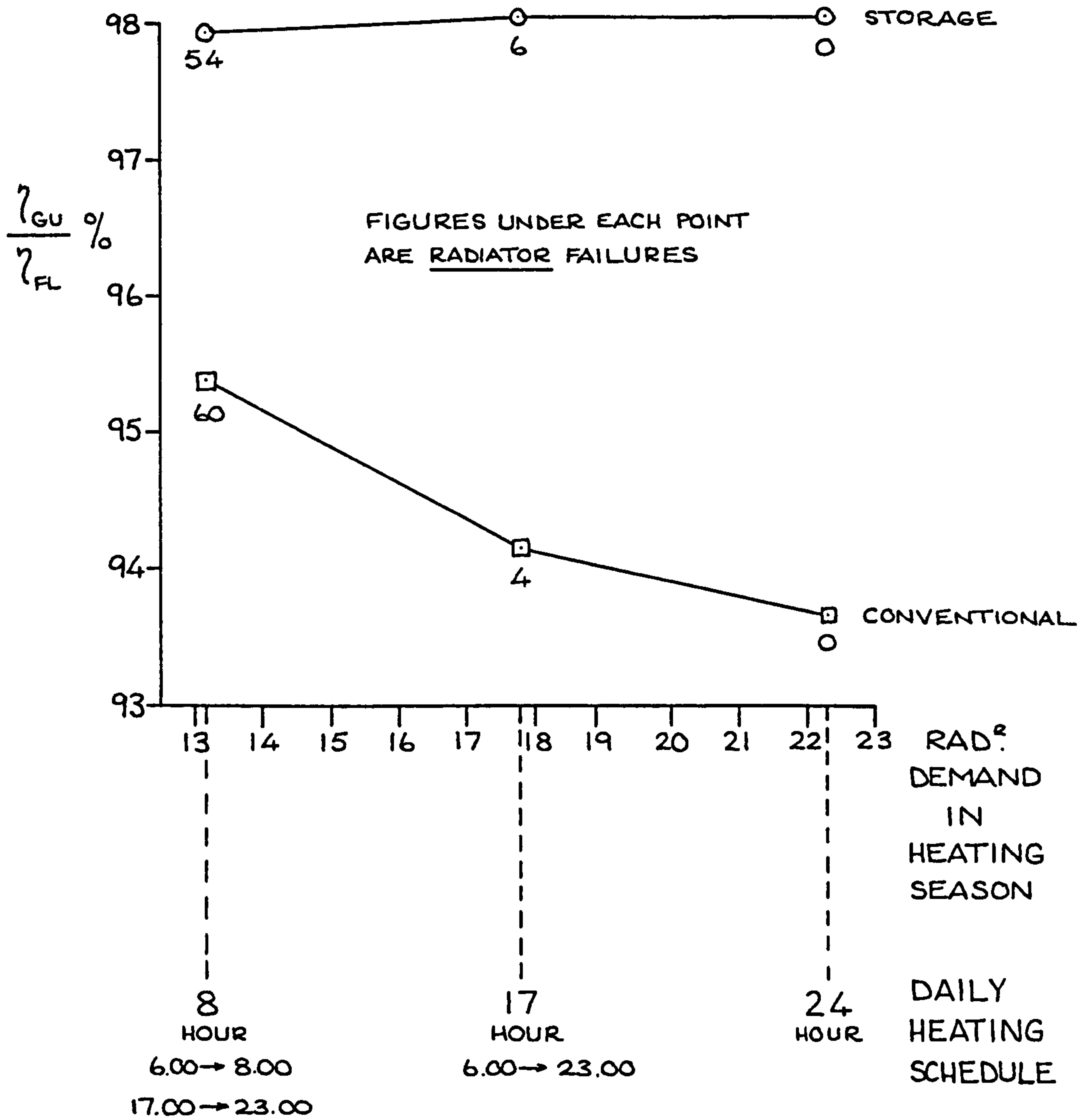
#### (viii) Occupancy Pattern

The heating period quoted for a given data tape was taken to reflect the occupancy pattern of that house. Data corresponding to three heating periods for the same house were available. For the end-of-terrace solid wall house, D.H.L. 7.423 kW:-

Tape 6 was for 17-hour heating, 17794 kWh seasonal load  
 Tape 3 was for 24-hour heating, 22286 kWh seasonal load  
 Tape 4 was for 8-hour heating, 13174 kWh seasonal load

The two systems were designed to be adequate for the 17-hour





<u>DATA USED:</u>	BOILER OUTPUT	RADIATOR "UA"	STORE/TANK
STORAGE	11.5 kW	270 W/°C	270 ℓ
CONVENTIONAL	16 kW	270 W/°C	137 ℓ

FIG 1.25: EFFECT OF OCCUPANCY PATTERN ON BOILER EFFICIENCY FOR END-OF-TERRACE SOLID WALL HOUSE.

heating period, i.e.  $270 \text{ W/}^\circ\text{C}$  radiator coefficient and boilers of 11.5 and 16 kW for the storage and conventional systems respectively. The store and d.h.w. tank sizes for the preceding "base-case" designs were retained. The maximum hourly load on the 17-hour tape was some 11.7 kWh. The 8-hour tape was far more severe, entailing a very high two-hour demand from 06.00 - 08.00 hrs (maximum hourly load 17.9 kW), and a six-hour evening demand. The average hourly demand of the 8-hour tape was 50% higher than that of the 17-hour tape. The 24-hour (continuous) tape was less severe (maximum hourly load 8.8 kW), but accumulated an excessive seasonal demand. The results of the simulations are presented on Fig. 1.25.

The storage boiler efficiency advantage is, not surprisingly, enhanced for continuous running, but reduced for the 8-hour schedule, where the high load factor suits the conventional system. Storage system performance is almost independent of duty; conventional system efficiency falls off as load factor decreases. Both systems fail badly, and are underdesigned, for the 8-hour schedule. If the systems had been designed to satisfy the 8-hour demand, a 98% or so approach to maximum boiler efficiency might be anticipated for the storage system, whereas conventional system efficiency would necessarily be impaired, and severely so as the load factor decreased.

#### (ix) The 'Failure Map' Design Aid

It is evident that for a given house (and d.h.w. demand), a variety of boiler ratings and store sizes could be combined as components of a proposed central-heating system with storage. The program HS212 was thus expanded to undertake heating season simulations for each boiler rating and store size as these were varied from 6 kW to 14 kW (in 2 kW steps) and 100 to 400 litres (in 50 litre steps) respectively. Representing each combination by a point on the "failure map", see Fig. 1.26, the totals of the radiator and d.h.w. failures can be written adjacent to each point,. If it is supposed that, for a system to be adequate, it should produce no more than 5 d.h.w. failures and 5 radiator failures, then a failure boundary can loosely be defined, as in Fig. 1.26. Any combination to the upper right part of the boundary would then, theoretically, satisfy the

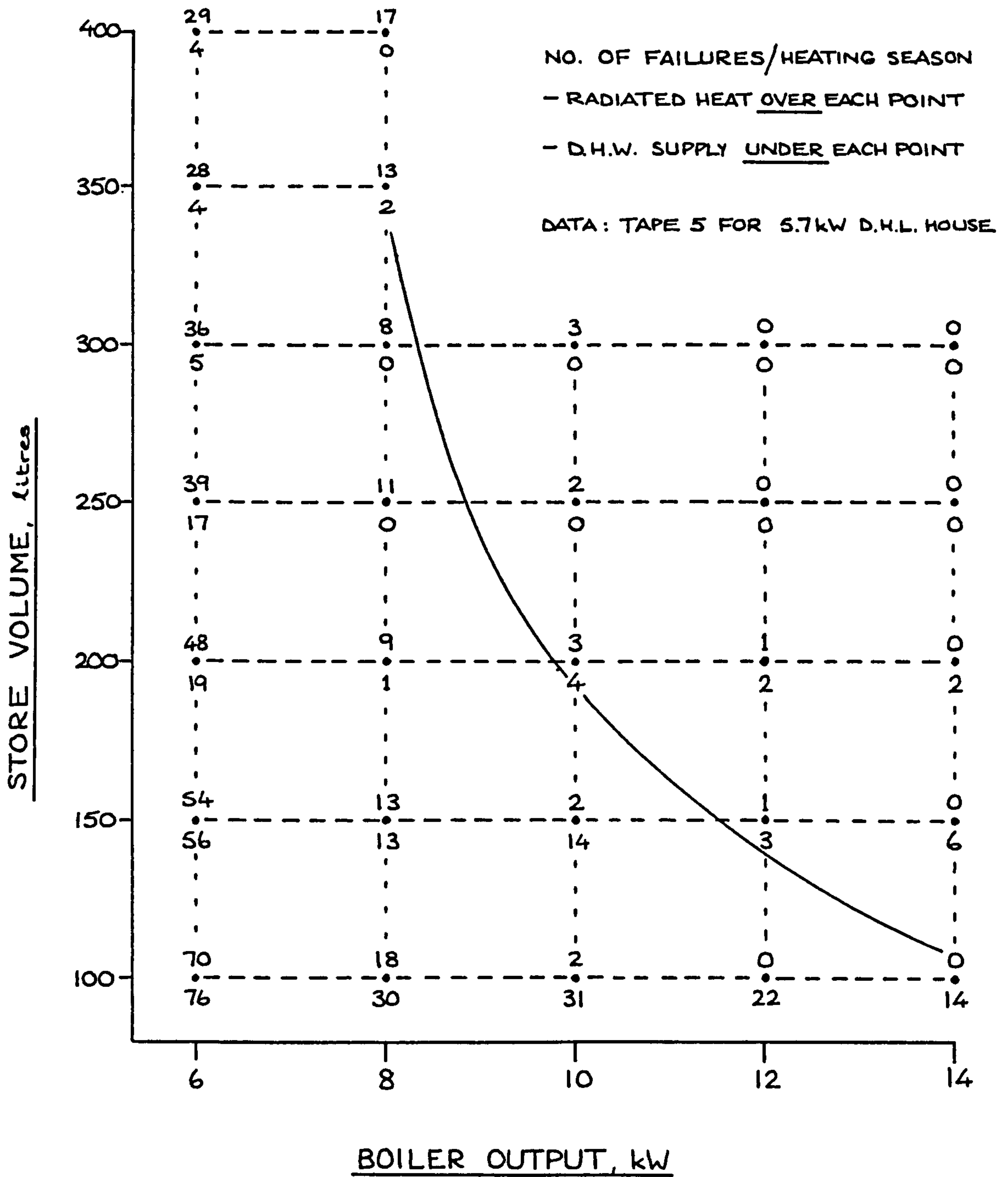
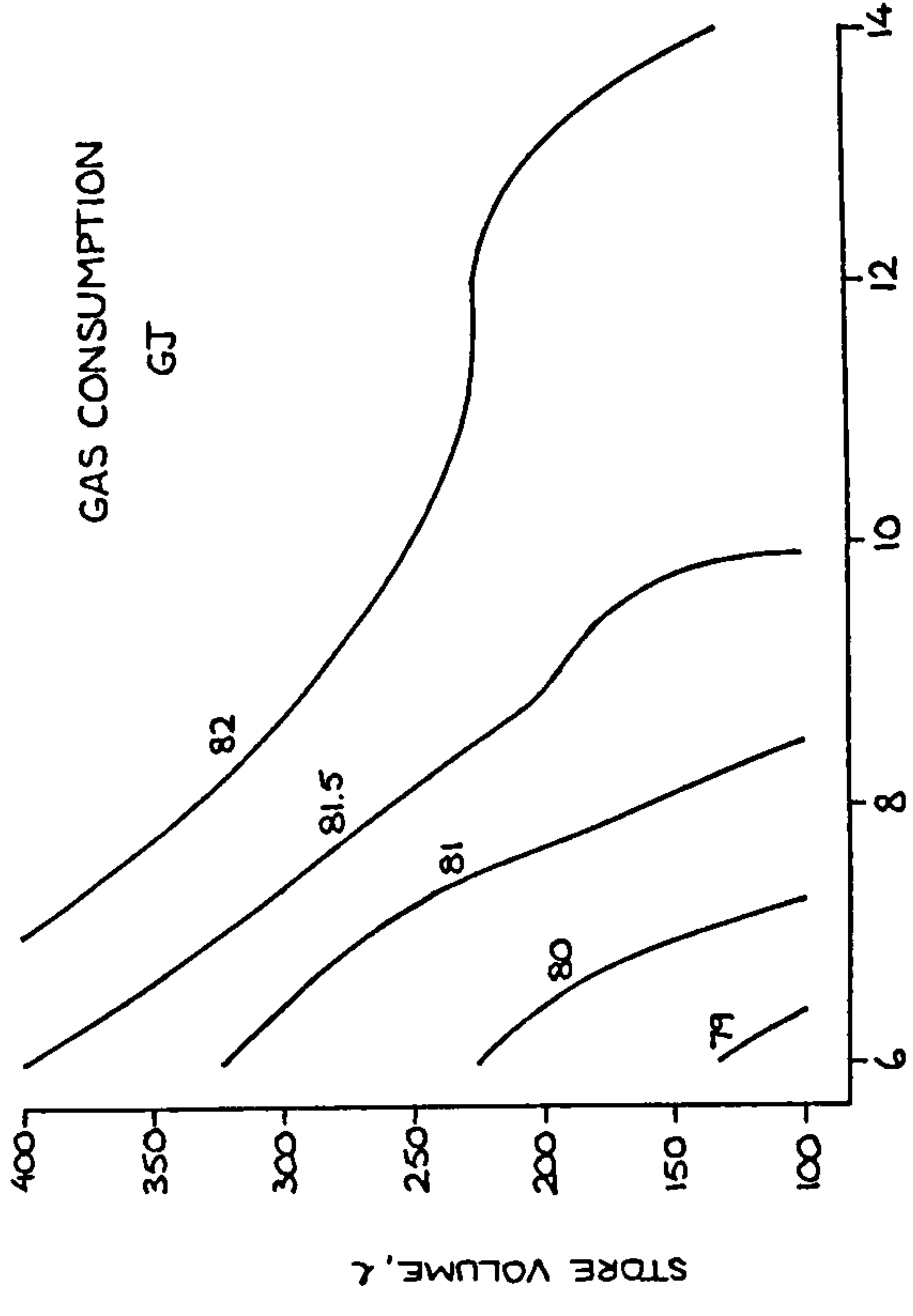
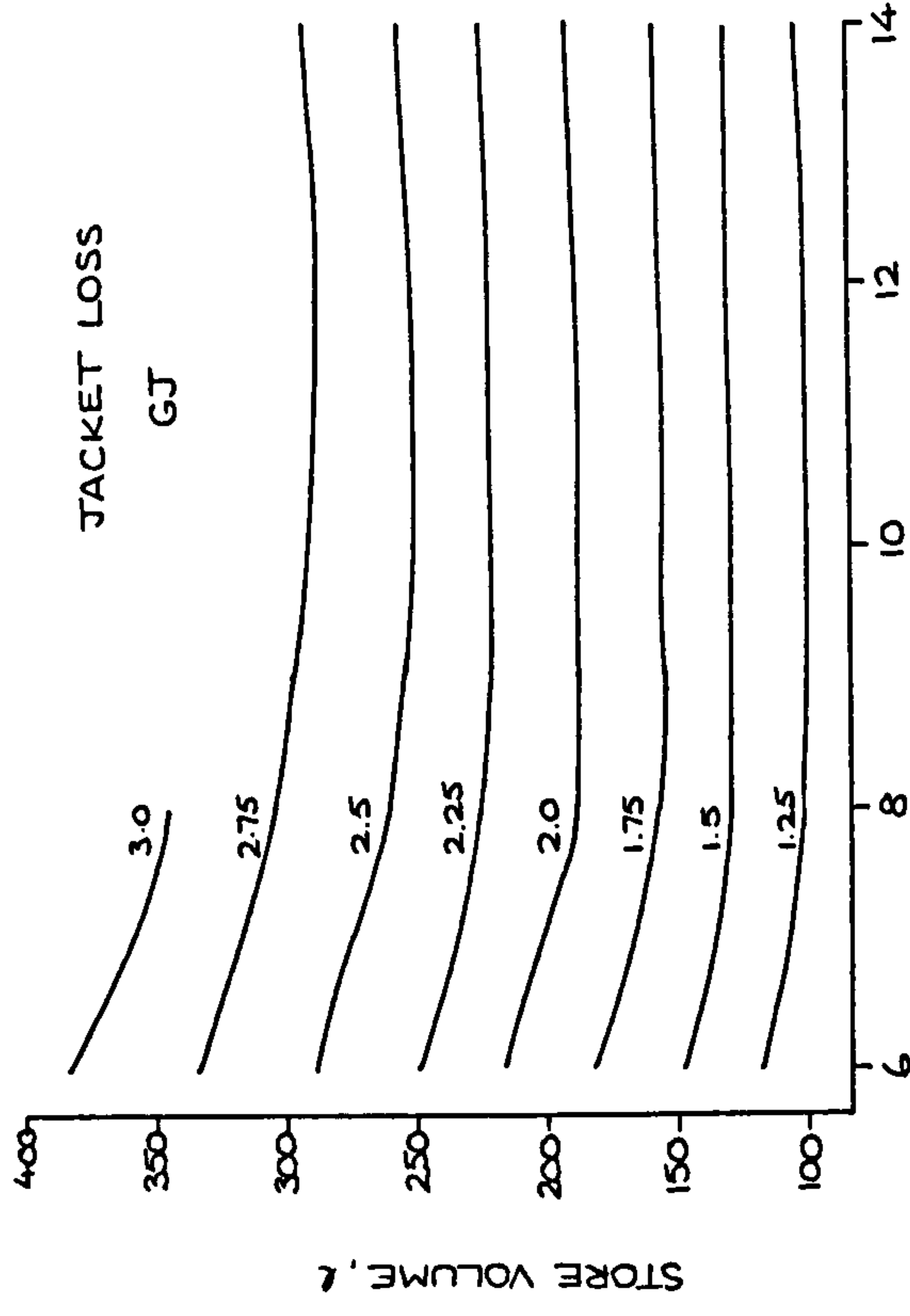
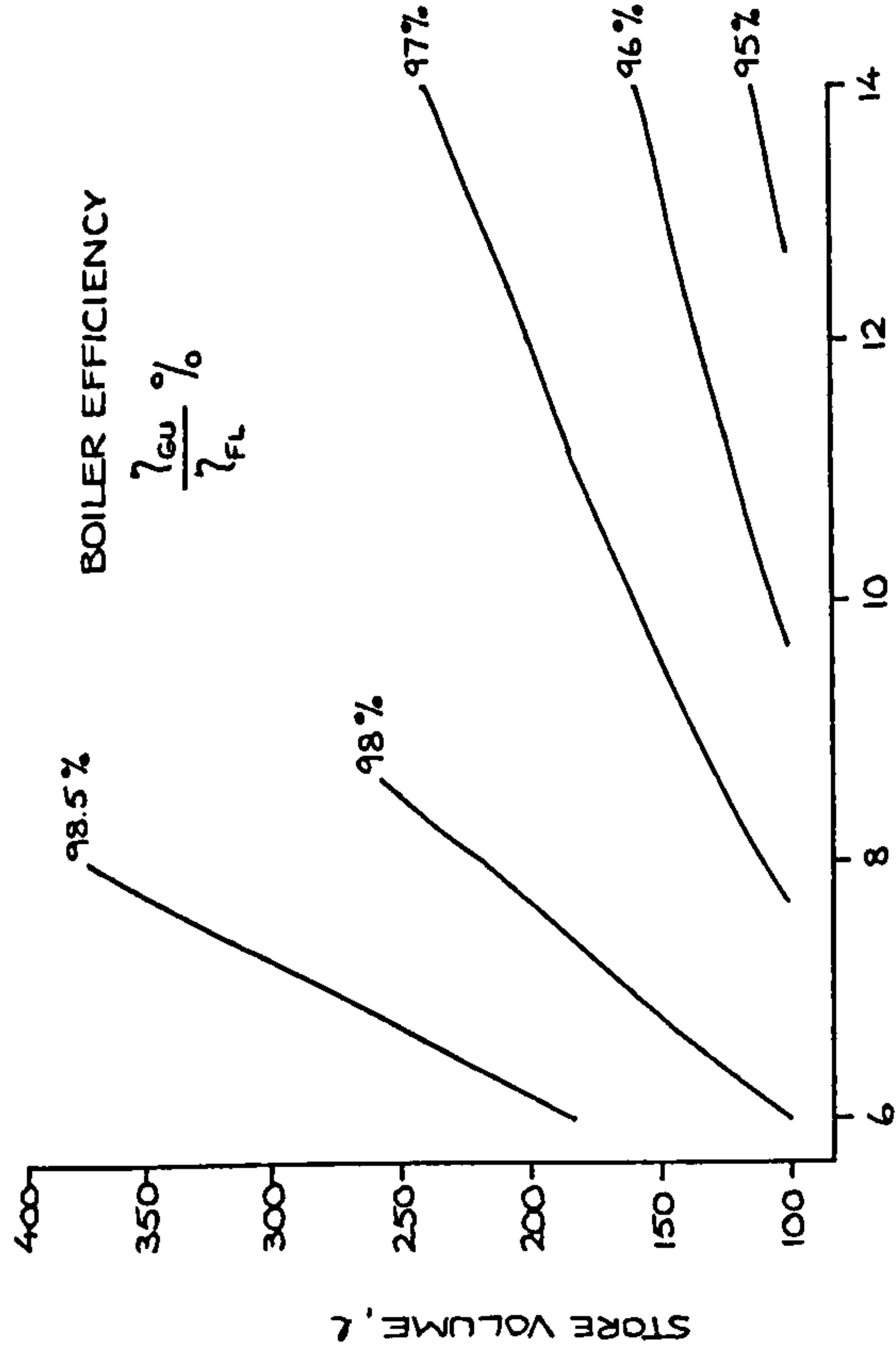
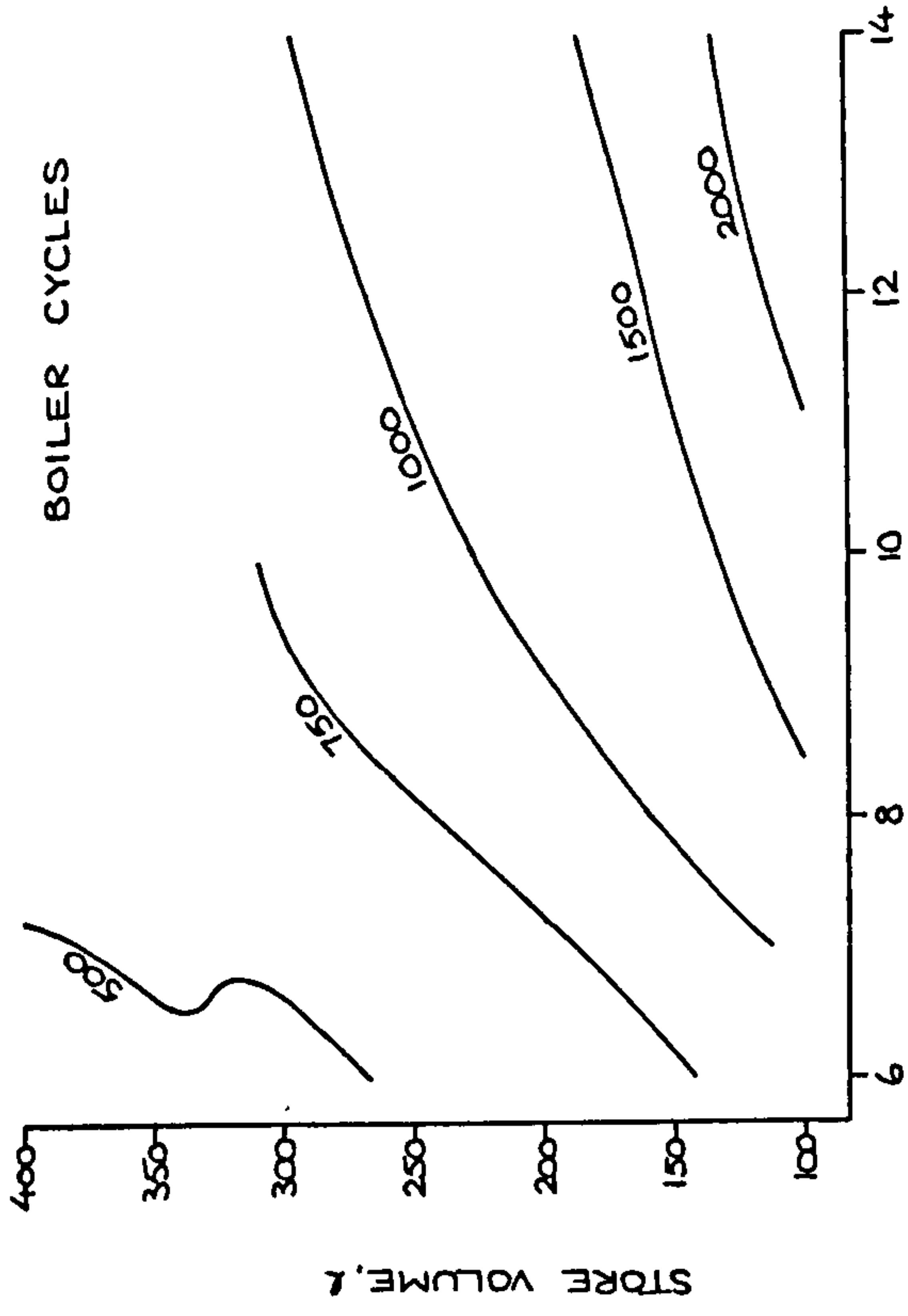


FIG 1.26 : EXAMPLE OF "FAILURE MAP" FOR A STORAGE SYSTEM





**FIG1.27: HEATING SEASON SUMMATIONS FOR STORAGE SIZING STUDY OF FIG1.26**

house's requirements.

It is evident again, that too small a store will enhance the risk of d.h.w. failures, whereas too low a boiler rating will produce an inadequate space-heating capacity.

A number of such maps are of course easily constructed from one such (rather long) simulation; maps of efficiency, gas consumption, boiler switchings and jacket loss are presented as Fig. 1.27. It would be a simple matter, though not undertaken here, to cost each combination of the above map, and combine installation cost with a 20-year running cost to give a map of the same form as Fig. 1.1, described in the introduction to this part of the thesis. The work has now come full circle from Fig. 1.1, expanding the preliminary study of ref. 1.1 to allow for a physical representation of system component operation in the computer models. It is interesting to note that the "failure boundary" of Fig. 1.26 is of the same form as the "zero auxiliary heat" line of Fig. 1.1.

The storage program would thus lend itself well to design work.

## CONCLUSIONS

Computer programs have been developed to model the seasonal performance of gas-fired domestic central-heating systems, (i) for a conventional system incorporating a domestic hot-water tank, and (ii) for a system including a water heat store with integral d.h.w. heat exchanger (to the commercially available "Cormorant" design). Typical values of the parameters for both systems were chosen for the "base-case" study, a straightforward comparison of the two principles. Salient conclusions drawn from this study were as follows:-

1. Lightweight boilers were chosen for both storage and conventional system computer simulations; the with-storage simulation indicated a heating-season boiler efficiency increase of 5% over the conventional system simulation. The ratio  $\eta_{GU}/\eta_{FL}$ , the approach of seasonal to full-load boiler efficiency, was about 98% for the storage system simulation and about 93% for the conventional system simulation. The

nature of the approximations incorporated in this study being conservative, 5% is thought to be a minimum figure likely to be achievable in practice (Table 1.4).

2. The above 5% advantage of the storage system can be eroded if both:-

- store and domestic hot water tank jacket losses are considered largely wasted, and

- lagging of the store (likely to be larger and hotter than the tank) is insufficient to restrict store jacket loss to the order of magnitude of the tank jacket loss. It is reasonable, however, to suppose that store jacket loss (if not excessive) would be largely, if not wholly, useful (Fig. 1.14).

3. The storage system can operate at a load factor at least 1.5 times that of the conventional system, if boiler ratings are suitably chosen. Conversely, the boiler power required for the storage system is likely to be 0.5 to 0.8 that required for the conventional system. There is thus a considerable potential benefit for the gas network due to load-levelling.

4. The storage system allows boiler cycling to be reduced from 100 - 300 cycles per day to less than 5 cycles per day.

From the subsequent parameter sensitivity study, the following further salient conclusions may be drawn:-

5. The conventional system's simulated performance is highly dependent on the value chosen for the "overall, effective heat-transfer coefficient to flue", after boiler shut-down. The storage system's simulated performance is almost independent of this parameter. As this parameter is (as yet) difficult to quantify, the modelling of a storage system is likely to be more reliable than that of a conventional system (Fig. 1.17).

6. Simulations of both systems revealed that the level of domestic

hot water demand had very little effect on the seasonal boiler efficiency of either system (Fig. 1.18).

7. As radiator size was diminished from 250 W/°C to 150 W/°C (overall UA-value), the conventional system's simulated seasonal boiler efficiency fell by over 1.5% due to boiler cycling being nearly trebled, whereas the storage system's simulated seasonal efficiency and cycling rates were only very slightly affected, (Fig. 1.20).

8. For traditional boilers:-

- oversizing will incur a greater seasonal efficiency penalty for the conventional system than for the storage system, the more so if the boilers are of heavyweight construction.

- in considering a conversion of a conventional system to a storage system, the seasonal efficiency gains will be most marked if heavyweight boilers are used (Fig. 1.21). (Note: all other studies conducted here involved lightweight boilers.)

It is possible that the use of a water store with a condensing boiler will not enhance seasonal efficiency.

9. With a suitable storage volume, building size should have no effect on seasonal efficiency, and the 5% storage advantage should be maintained. If structural considerations limit store volume, the storage advantage may be slightly reduced in the case of large houses, (Fig. 1.24).

10. An occupancy pattern incurring high hourly radiator demands (e.g. an 8-hour heating period) will be to the advantage of the seasonal efficiency of the conventional system. The seasonal efficiency of the storage system is largely unaffected by occupancy pattern (Fig. 1.25).

11. Summer boiler efficiencies are lower than heating season efficiencies for both systems. If summer and winter figures are combined for annual boiler efficiencies, the heating season efficiency of the storage system drops by about 0.5%, while that of the



conventional system drops by about 1%.

The following conclusions are drawn from the parameter sensitivity studies relating to the storage system alone:-

12. An increase in store volume from 100 to 350 litres gave a seasonal boiler efficiency rise of 2%, the rate of improvement being more marked at low volumes. Again, there is a threshold store volume below which the domestic hot water facility becomes inadequate, (Fig. 1.22).

13. By progressive "tuning" of the settings of the store thermostats from an initial "coarse" setting to a final "fine" setting, annual efficiency was enhanced by ~1%, (Fig. 1.23).

14. At the design stage, for the "Cormorant" concept examined here, too small a boiler will induce space heating inadequacy, whereas too small a store will result in an inadequate domestic hot water facility. For any given house, a range of boiler and store sizes will combine to give a suitable central-heating plant. The choice is likely to be reduced to the questions of availability and cost, (Fig. 1.26).

#### RECOMMENDED FUTURE WORK

It is suggested that, if a researcher wishes to develop this study, he or she might like to consider the following most useful improvements to the current study:-

1. The d.h.w. heat exchanger has already been redesigned using the "Integron" copper finned-tube material mentioned above. The storage program requires algorithms to be sought corresponding to the use of these horizontal-axis, Integron h.e.'s.

2. Further investigation, perhaps based on an already published detailed boiler model (should such a one be found), should be carried out to confirm that the use of a single-valued, constant, heat-exchange coefficient from boiler to flue a.s.d. is in fact valid.

An independent verification of the value used here (esp. for the conventional system program) would be very worthwhile.

3. A more sophisticated version of the radiator model, acknowledging the non-linearity of its characteristic, should be sought.

4. The overriding need for a continuation of the work is for detailed seasonal data from systems (both storage and conventional) operating in the field. A correlation of program output with field data is a necessity before the possibility of the use of the programs as a commercial design tool arises.

5. With suitable, updated costing data, the "failure map" of Fig. 1.26 could be established for any given house, and a lifetime cost-optimum sought.

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**PART II**

**COMMERCIAL AND INDUSTRIAL SECTOR:**

**THE DESIGN OF A STORAGE HEATING SYSTEM**

**FITTED IN SPA SCHOOL, BERMONDSEY, AND A**

**PRELIMINARY APPRAISAL OF ITS PERFORMANCE**

**DURING THE 1984-85 HEATING SEASON**



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ABBREVIATIONS

During Part II of the thesis, the following abbreviations have been used intermittently in the text:-

c.h.	Central heating
3WV	Three-way, or mixing, valve
P.R.T.	Platinum resistance thermometer
t/c	Themocouple
UA	Overall heat transfer coefficient; U is a coefficient per unit area, in $W/m^2K$ , and A the area in $m^2$
ℓ	Litres
η	Efficiency
LF	Load Factor
η <sub>FL</sub>	Full load (bench) boiler efficiency
hr	Hour

## INTRODUCTION

During the summer of 1983, after the preparatory work for the modelling of Part I of this thesis had been completed, an opportunity arose to become involved in a field trial in the Commercial and Industrial Sector. It had been decided by Cranfield, the British Gas Corporation and the Energy Technology Support Unit (Dept. of the Environment) to embark on a feasibility study into, and subsequent design and operation of, a full-scale storage system to evaluate the potential benefits of the storage principle when applied to a large central-heating system. Working alongside colleagues at both British Gas (Watson House) and Cranfield, the author was called upon to assist in the design stage (at Cranfield), and in data retrieval (in London) and preliminary analysis (at Cranfield). The author's contribution at these various stages of the field trial will be presented in this part of the thesis. No attempt will be made to write a full account of the field trial to date; this will be covered in other material, of which the most notable references to date are refs. 2.1, 2.2 and 2.3. A full report of the field trial will be published at Cranfield after the data at present being obtained from the 1985-86 season have been analysed.

The site chosen for this field trial was the Spa School, Monnow Road, Bermondsey, East London (by permission of the G.L.C.) see Photo 2.1. The school is a medium-sized, two-storey, heavyweight building, built about AD 1900, heated by a modular gas-fired system. The choice of a school was particularly apposite as it offered a predictable occupation pattern, such that heating seasons with and without the storage system could be compared directly. A further advantage was that the school's hot-water system was separate from the heating system.

### 1. Heating System Conversion

The original, conventional system comprised five Hamworthy R320 modular boilers, of normal heat input 94 kW each, see Fig. 2.1 and photo 2.2. An externally-mounted compensator determined the required radiator flow temperature according to the external ambient





PHOTO 2.1: EAST WALL OF  
THE SPA SCHOOL, BERMONDSEY





PHOTO 2.2: THE  
HAMWORTHY BOILERS



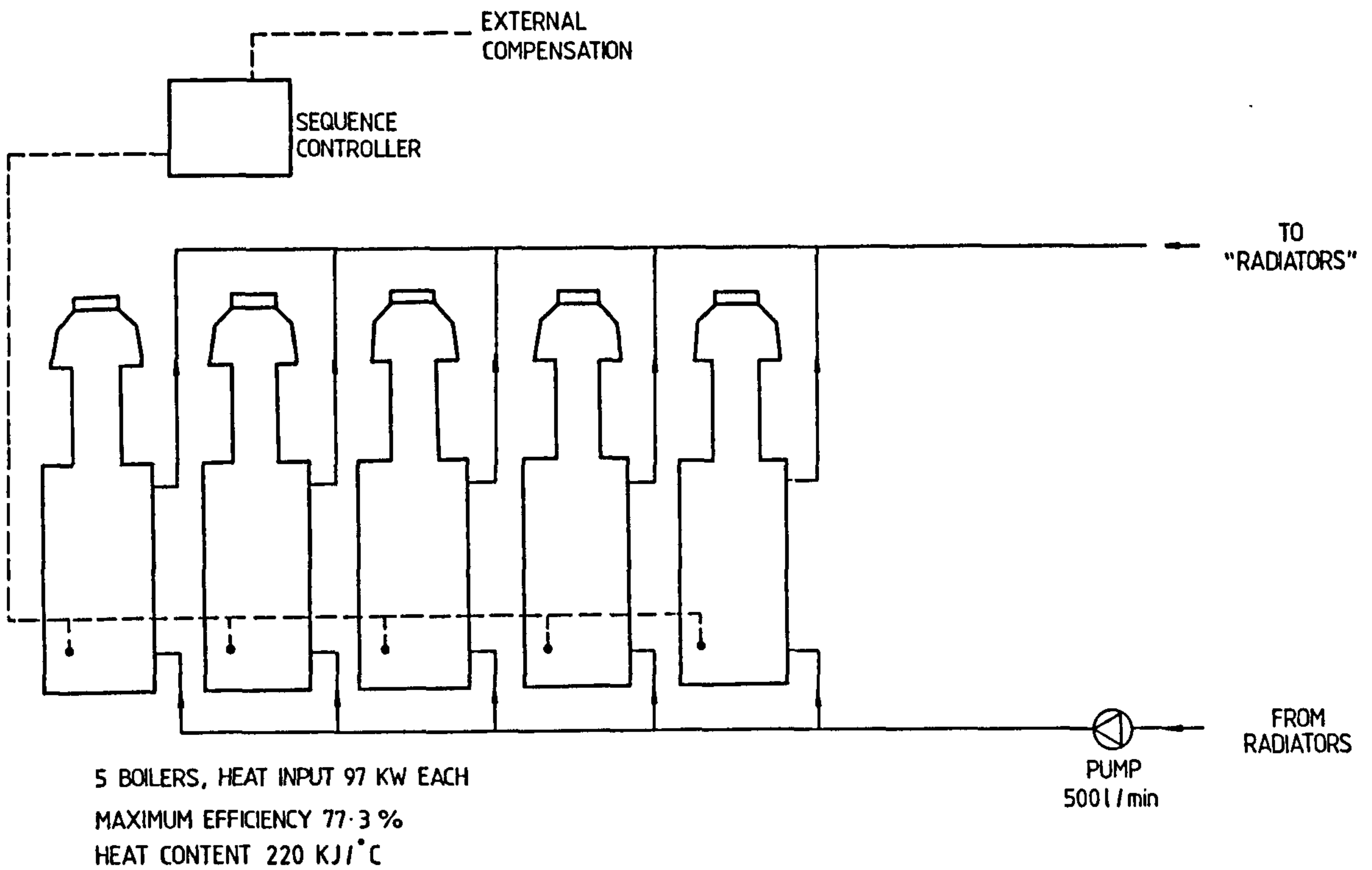


FIG 2.1: ORIGINAL, CONVENTIONAL C.H. SYSTEM

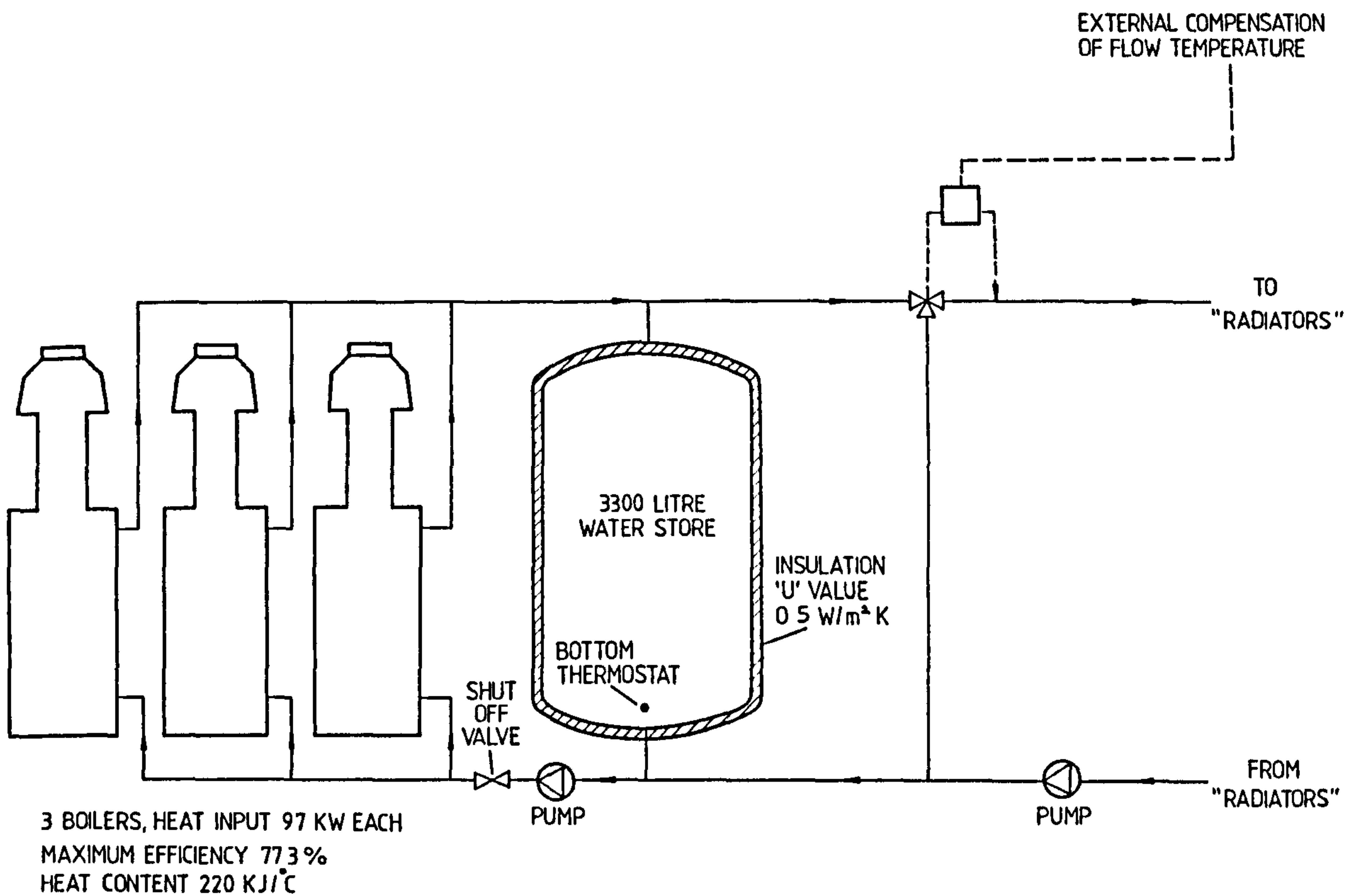


FIG 2.2: PROPOSED, STORAGE C.H. SYSTEM

temperature, and the boilers were sequence-controlled to provide this required ("compensated") temperature. The heating period was set on a timing-clock, normally 03.30 hrs until 15.00 hrs on weekdays other than Mondays, when the heating started at about 01.00 hrs. This system was reckoned to be a good, efficient, conventional system, such that any increase in efficiency when it was converted to a storage system would accurately reflect the inherent advantage of the storage concept. (This had not been the case in a parallel, smaller-scale study undertaken by B.G.C. Watson House where the considerable efficiency gains after conversion to storage were in part due to the poor efficiency of the original system - see ref. 2.4).

The school's original system had been monitored by Watson house in the early 1982 half-season; after finalising agreements on the field trial it was again carefully monitored for the early 1984 half-season, to set up an adequate appraisal of the conventional system for subsequent comparison with the storage system results.

The proposed, storage heating system was to the layout on fig. 2.2. Preliminary checks on the original system showed that there was so much part-load duty that the incorporation of a store would allow the number of boilers to be reduced from five to three. The sequence controller would be disconnected and the three boilers would operate as one large boiler, the surplus two boilers now being blanked off so as not to contribute to standing losses. Thus a nominal gas rating of 3 x 94 kW would be available (in fact the eventual gas rate was measured to be 241 kW, an even further power plant reduction to just over 50% of the original system's maximum rating).

The original distribution system would remain, in particular the external compensator would be retained untouched (if possible), such that exactly the same radiator temperatures would be supplied as before. This would be achieved by incorporating a distributing three-way valve ("3WV"), or mixing valve, on the feed to the radiators to recycle radiator return temperature so as to control the feed temperature to compensated temperature. Thus the boiler/store combination would be required to reproduce the original system's heating, and a reduction in the gas used would indicate the inherent advantage of the storage system. Moreover, in the first stages of

investigation, the above timing schedule of the original system would be retained. As experience accrued, it was hoped to optimise the use of the storage system to explore its full potential, for example, by starting the warm-up of the school as late in the early morning as possible, as the storage system should prove to be capable of heating the school faster (see below).

## 2. The Hamworthy R320 Boilers

Details of the school's boilers were supplied by Watson House, and were as follows, in Table 2.1:-

<u>Hamworthy R320 Boiler</u>	
Thermal Capacities :	Water 115 kJ/°C
	Metal 105 kJ/°C
Output, E	70 kW
Heat Transfer Coefficients:	
Metal - Air, K	29 W/°C
Metal - Water, U	1.14 kW/°C
E/U (= $T_{\text{metal}} - T_{\text{water}}$ )	62°C
E/K (= $T_{\text{gas}} - T_{\text{metal}}$ )	2414°C
K/U	0.025
Heat transfer to flue and room	22.8% of gas input

Table 2.1 : Details of Hamworthy R320 Boiler



A part-load curve for this boiler was not available. However, as described in the General Introduction of this thesis, a short, highly simplified computer model had been set up, which predicted boiler efficiency as a function of both load factor (the conventional representation), and switch frequency (assumed independent of load factor). The above data were applied to this model, and the program yielded the arrays of curves shown in fig. 2.3. It was found in Part I that the heat loss to the flue after each turn-off is highly dependent on the parameter "UAFLUE", the overall, effective, heat-transfer coefficient from the boiler to the flue after shut-down, and that a value of this parameter is singularly difficult to isolate. Table 2.1 gives a steady-state figure of  $29 \text{ W/}^\circ\text{C}$ ; the first set of curves of Fig. 2.3 assumes  $\text{UAFLUE} = 29 \text{ W/}^\circ\text{C}$ . Note that the part-load curve is almost independent of switch frequency if this latter is greater than about 1 switch/hour. Much discussion at this stage led to the intuitive assumption that it was likely that the effective value of UAFLUE (which includes case loss after shut-down), would be much higher than  $29 \text{ W/}^\circ\text{C}$  by virtue of the fact that at and after boiler cut-out, the airflow in the flue would be vigorous, and that the buoyancy draw would greatly increase the heat-transfer coefficient by convection. Robert Cohen of the Applied Energy Unit at Cranfield, used a value of four times the above value,  $116 \text{ W/}^\circ\text{C}$ , for computer simulation subsequent to the work presented here, which he justified in ref. 2.2, p.30. Much later in this study, therefore, the frequency effect on the part-load efficiency curves was re-evaluated with  $\text{UAFLUE} = 116 \text{ W/}^\circ\text{C}$ ; these curves are plotted as the second part of Fig. 2.3.

The two sets of curves show the great effect of UAFLUE-value in these computer simulations. If it is supposed that a value of  $116 \text{ W/}^\circ\text{C}$  is approximately representative of the boilers used at Spa School, then the following improvement might be anticipated when storage is introduced:-

(i) Load factor should be approximately doubled by reducing plant capacity to about 50% of the original, as described above. On a day where load factor had been 0.25, the efficiency (% of  $\eta_{FL}$ ) will increase from about 82% to about 94% due to plant sizing alone.

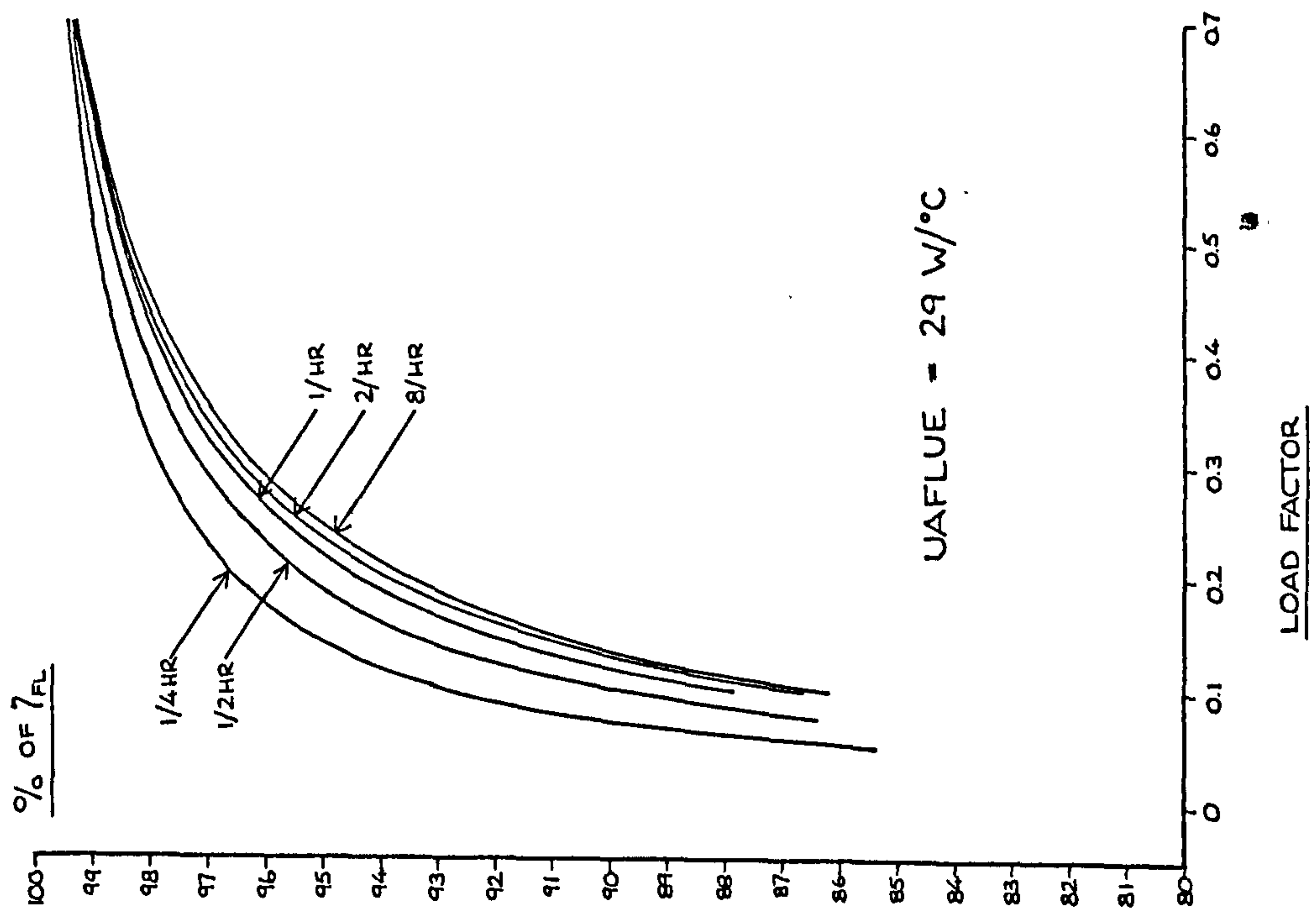
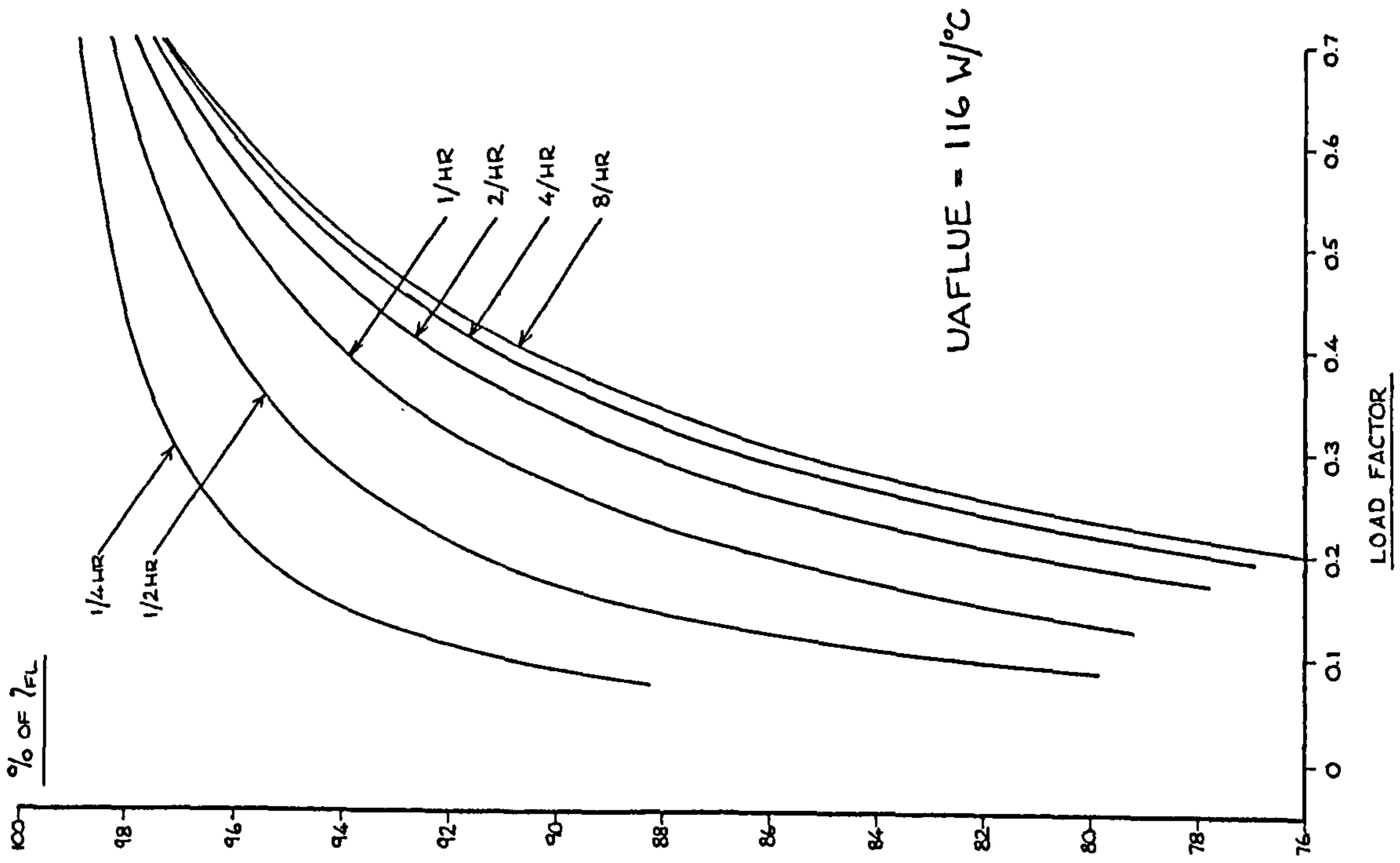


FIG 2.3: THEORETICAL BOILER  $\eta$  AS FUNCTION OF BOTH L.F. AND CYCLE FREQUENCY

(ii) On the basis of the results of Part I, it was anticipated that boiler switchings could be reduced to about five or less per heating period, i.e. less than one every two hours. Thus boiler efficiency might be further increased from 94% of  $\eta_{FL}$  to about 97%, for the day considered. Thus it was hoped that the introduction of the store would have this significant, two-fold effect on boiler efficiency.

Full, accurate analysis of the field trial measurements would yield values of boiler efficiency, and switch rate. It was therefore hoped that experimental points could be placed on Fig. 2.3, and justify, or otherwise, the adoption of  $UAFLUE = 116 \text{ W/}^\circ\text{C}$  in subsequent computer models of the thermal behaviour of Spa School.

Lastly, it is evident from Fig. 2.3 that the efficiency curves of least frequent switch rates (1 switch every 2 or 4 hours) are only slightly lowered as the  $UAFLUE$  is increased from 29 to  $116 \text{ W/}^\circ\text{C}$ . Thus, if switch frequency could be kept to such a very low value, the efficiency values predicted from computer modelling of the storage system would not be invalidated by an unreliable value of  $UAFLUE$ . Modelling of the conventional system, however, remains highly vulnerable to errors in  $UAFLUE$ .

### 3. Storage System Operation and Store-Sizing Considerations

The "modus operandi" of the storage system has been described in section 3.3 of ref. 2.5. Essentially, the system operates as follows:-

- (a) During the night, the boilers charge the store alone.
- (b) When the heating period commences, the store discharges to replace rapidly the cold radiator water with water at the compensated temperature.
- (c) When the store is discharged, the boilers fire and heat both the radiators and the store, supplying heat to the building to counterbalance the fabric loss (quasi steady-state), and recharging the store.
- (d) When the store is fully charged again, the boilers cut out



and the store alone supplies building heat as in (b) above, except that the heat required is this time nearly steady-state fabric loss.

- (e) A cycle of phases (c) and (d) repeats throughout the duration of the heating period, the larger the store, the fewer the cycles.

These features will be discussed later (see "Graphical Presentation", below).

Let us examine phase (b), above, called the "warm-up" period. At this time (early morning, say 05.00 - 07.00 hrs), the building is likely to be cold, not having received any heating for over twelve hours, and the outside ambient environment is often the coldest of the whole day. It was required to heat the school to a minimum of  $18^{\circ}\text{C}$  by 09.00 hrs. Being of a high thermal inertia construction, the warm-up was likely to place the most severe demand on the heating system, and the boiler/store combination had to be sufficient to meet this demand for a design day of constant  $-1^{\circ}\text{C}$ .

The original heating system fired at about 03.30 hrs, and often required over an hour to achieve the compensated temperature in the radiators. The room temperature rose slowly; on a very cold day the system might well require until 07.00 hrs to raise room temperature to  $18^{\circ}\text{C}$ , i.e. after about 3.5 hours of heating and fabric loss, although this latter was from intermediate class temperatures. (In fact, the start-up being so early, overheating was frequent.)

The storage system, however, while the store is being charged, loses a negligible amount of energy (store jacket loss only). When the building heating system starts, the store should provide water at compensated temperature to the whole radiator system in a matter of minutes (flow time through the radiators). Thus a very rapid building warm-up should be possible, a fraction of the original system's 3.5 hours. Thus, further, the start of the heating period can be delayed beyond 03.30 hrs, the building still achieving  $18^{\circ}\text{C}$  by 09.00 hrs. During the delay period, the fabric loss is negligible. In theory,

therefore, the storage system should provide an opportunity to eliminate a certain portion of the heating period, perhaps up to an hour or so, see section 12.6 of ref. 2.2. In this instance the building heat demand would be reduced by an amount equal to the fabric loss otherwise incurred during this delay period. Note also that with zero delay, the building would be heated to  $18^{\circ}\text{C}$  unnecessarily early, and fabric loss and radiator requirement would actually be enhanced.

The problem confronting the Cranfield team at the start of the design stage was therefore to render the boiler/store combination adequate for the school's warm-up requirements; if the combination could meet these requirements it would certainly be sufficient to satisfy the rest of the day's demands.

With the boiler power chosen, it remained to select an appropriate storage volume. The store's major task, again, would be to replace all the radiator's cold water with hot water at the compensated temperature. The principle of the compensator was that it was calibrated to determine its required temperature according to the (almost) steady-state fabric loss of the school; i.e. at an ambient temperature of  $0^{\circ}\text{C}$ , for example, a compensated temperature of say  $65^{\circ}\text{C}$  would maintain class temperature at a comfortable level. Thus, during warm-up, the compensated temperature was of little significance (in terms of meeting the school's requirements), but it would still be imposed on the boiler/store. Ideally, therefore, the store's energy released during first discharge (phase (b) above), should be equal to the energy required to raise the radiator system from cold to the compensated temperature. This was the criterion adopted for the design study; brief details of the method by which the store volume was calculated will follow in the next section.

#### DETERMINATION OF A SUITABLE STORE VOLUME

As stated above, with the boiler power for the proposed storage system at Spa School fixed at up to  $3 \times 94$  kw equivalent via natural gas (corresponding to the maximum rating of the Hamworthy boilers), it remained to choose a suitable store size, adequate for the early

morning warm-up period. It was necessary first, however, to investigate the capacity of the school's existing radiator system, such that a compatible store design could be sought. This preliminary study was commenced in early autumn 1983, the author working alongside Robert Cohen at Cranfield. The author's contribution will be presented here; the final choice of the store size, and subsequent design of the store, were undertaken by Mr. Cohen alone during the Spring of 1984, and will be reported by himself in the full report of the field trial later this year.

The only data available at this stage were those obtained by Mr. N. Gibson of Watson House, during the early 1982 half-season; he provided a graphical presentation of each day monitored, and a computer data bank of the parameters recorded. He selected two days during this period as being representative of the school's operation on "cold" and "warm" days. Fig. 2.4 shows 24th February 1982, a cold day where the environmental ambient temperature hovered near  $0^{\circ}\text{C}$  in the early morning; this day was taken to be close to the notional "design day" of constant  $-1^{\circ}\text{C}$  ambient throughout, and would be adequate for assessing the required boiler/store capacity. Fig. 2.5 shows 11th May 1982, a mild end-of-season day, used as a cross-check for values deduced from Fig. 2.4. From both figures, the large number of boiler switch-offs is evident.

The work outlined here will be in two parts:-

(i) assessment of the parameters required to model adequately the school's radiator system, deduced from the above graphs. The results of this assessment will be seen to be somewhat inconclusive, and the work is presented more for methodology than absolute results, and,

(ii) the use of the above model in a full computer model of the school's proposed storage system to evaluate an adequate storage capacity.



HEATING SYSTEM OPERATION AT SPA SCHOOL

WEDNESDAY 24. 2.82

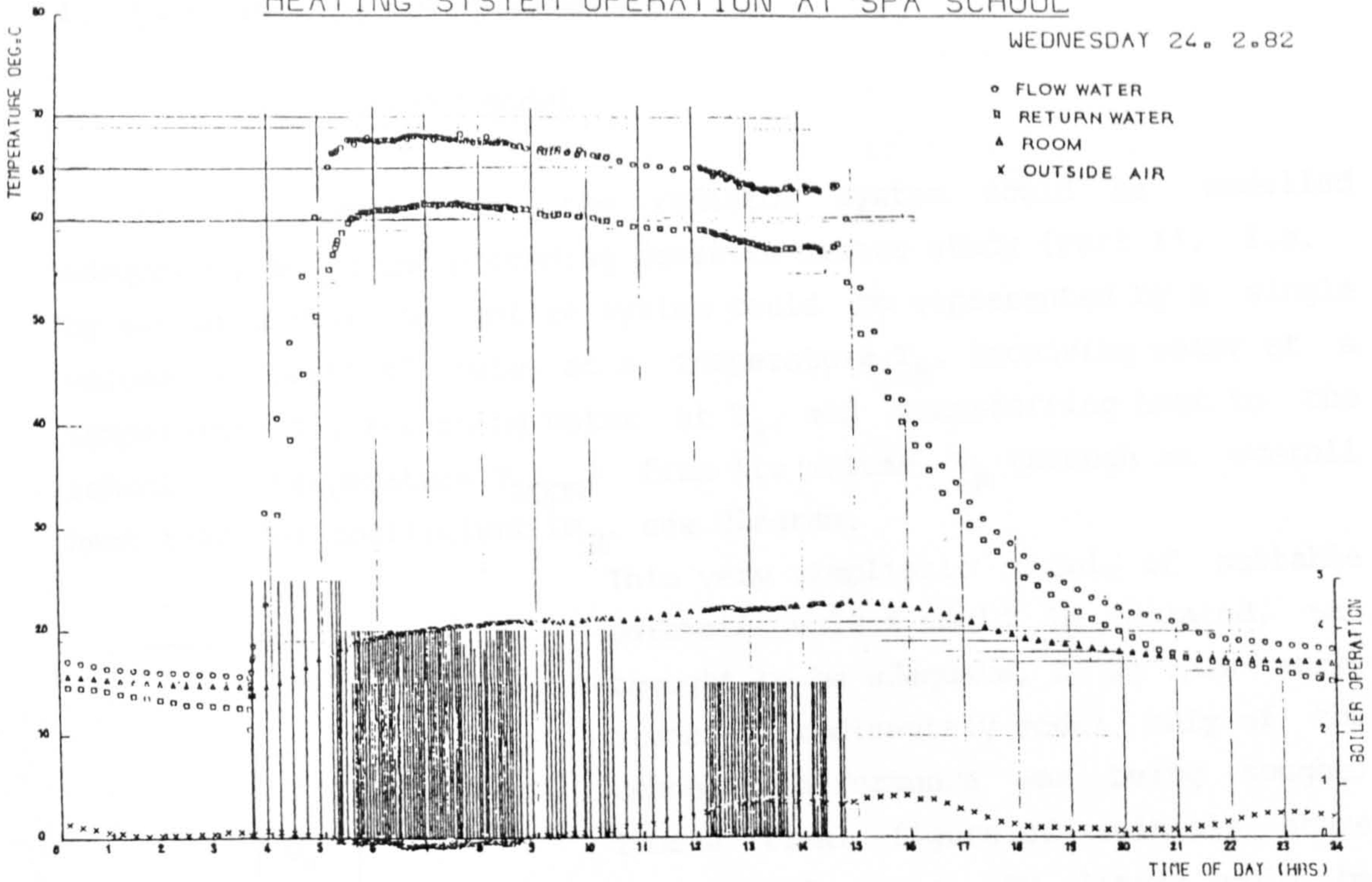


FIG 2.4: CONVENTIONAL SYSTEM - "COLD" DAY

HEATING SYSTEM OPERATION AT SPA SCHOOL

TUESDAY 11. 5.82

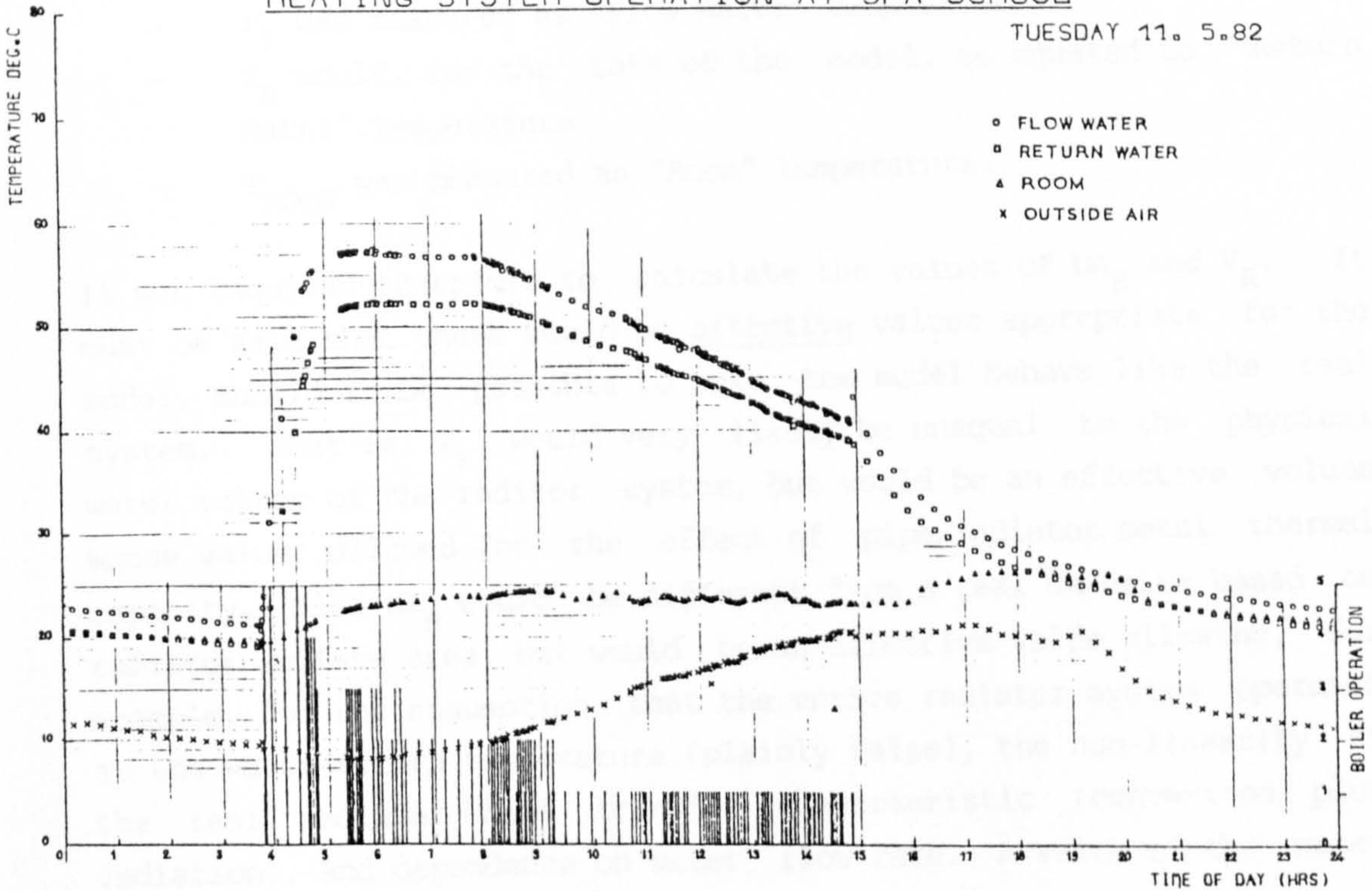


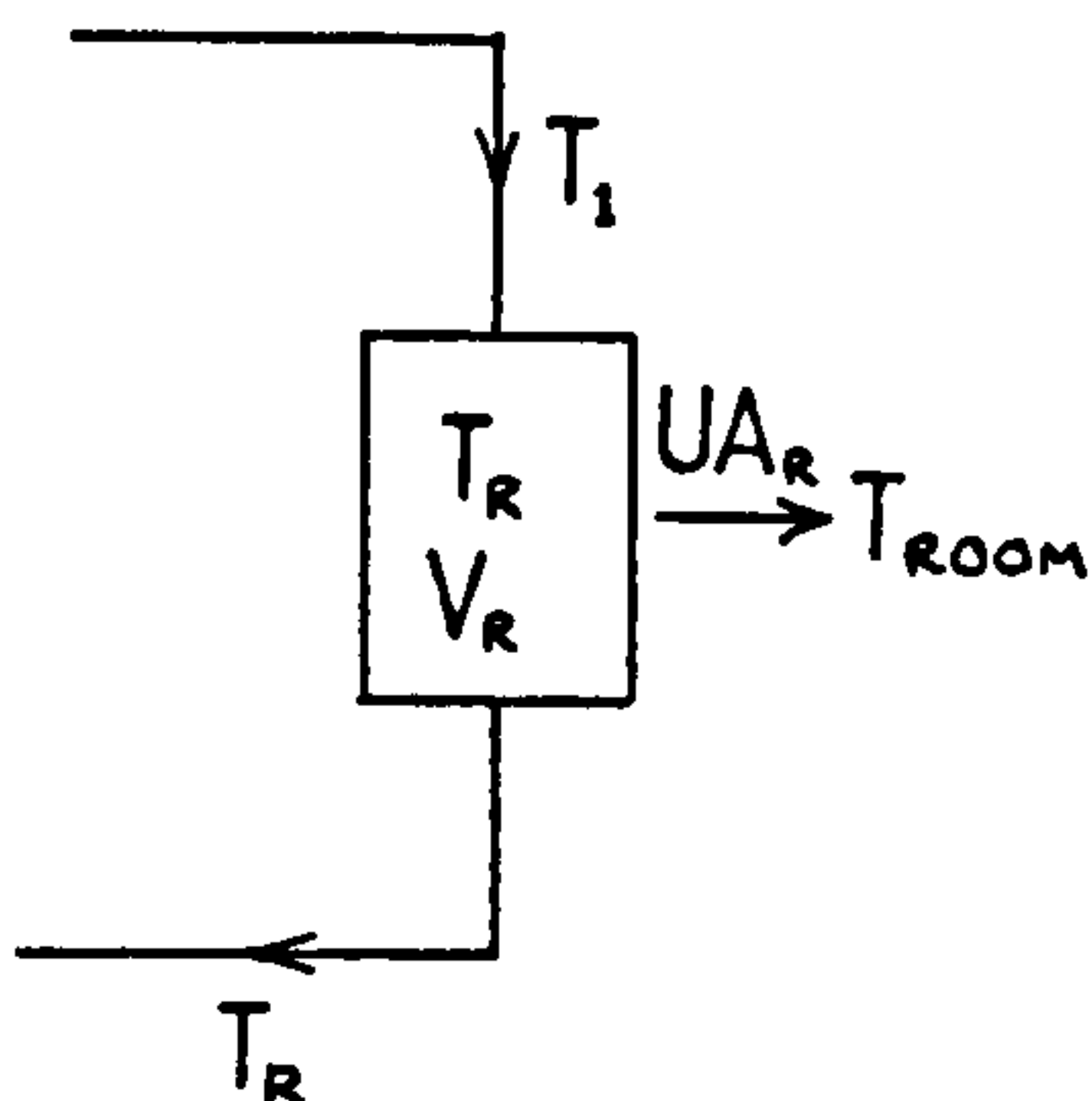
FIG 2.5: CONVENTIONAL SYSTEM - "MILD" DAY



## 1. Assessment of Radiator System

### (a) Simple "Bath" Model

It was hoped that the radiator system could be modelled adequately as in the preceding domestic-sector study (Part I), i.e. by assuming that the entire system could be represented by a single volume or "bath" of water at a temperature  $T_R$ , receiving water at a temperature  $T_1$ , returning water at  $T_R$ , and transferring heat to the school (at temperature  $T_{ROOM}$ ) from its volume  $V_R$  through an overall heat transfer coefficient  $UA_R$ , see diagram.

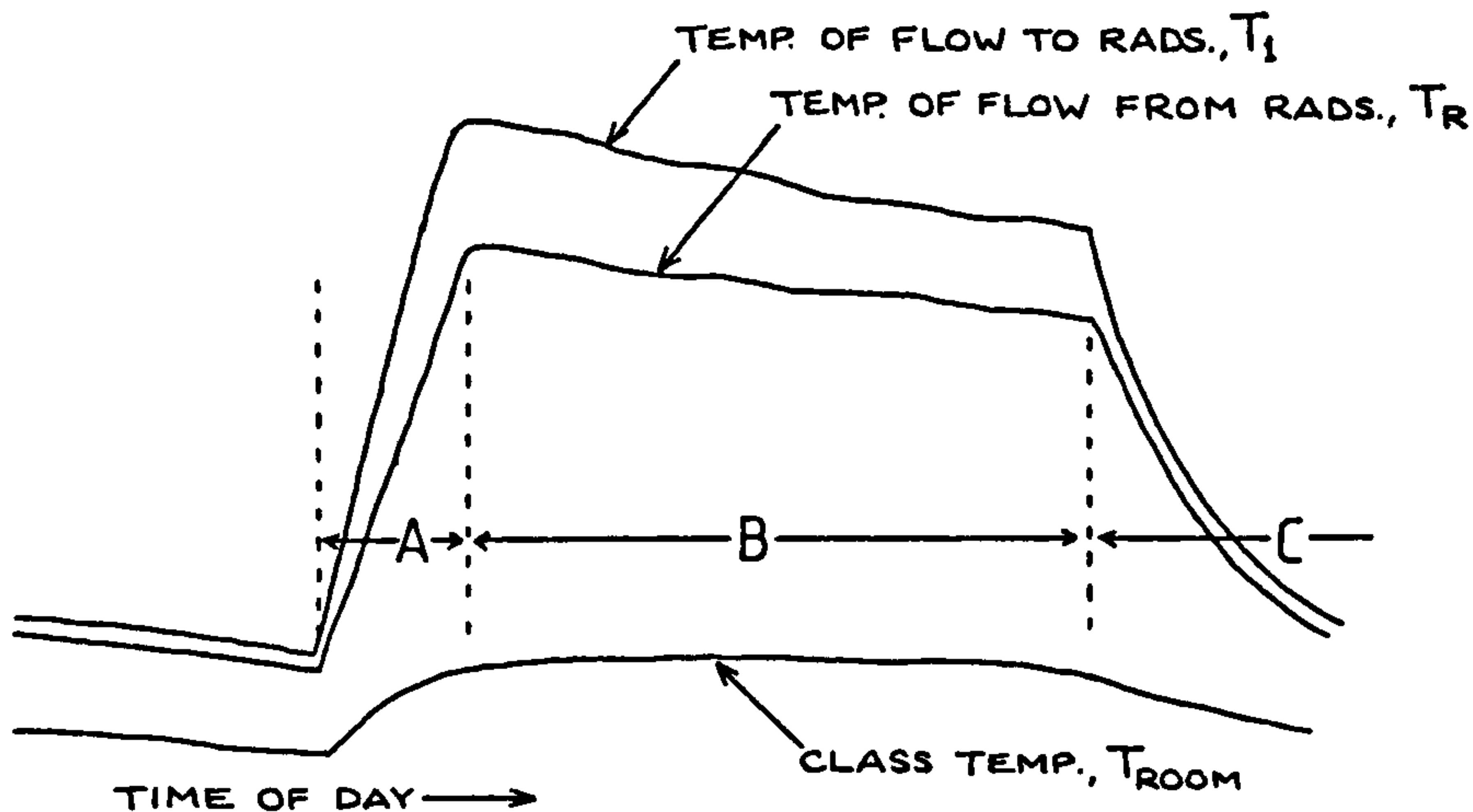


This very simplistic model, if suitable parameter values could be isolated, was thought to be adequate, as at this early stage, a rudimentary model only of the school's performance was being sought. Thus a final figure of suitable store capacity of say  $\pm 200$  litres would be acceptable for the time being. Referring to Figs. 2.4 and 2.5, therefore:-

- $T_1$  was measured as "Flow Water" temperature
- $T_R$  would, for the sake of the model, be equated to "Return Water" temperature
- $T_{ROOM}$  was measured as "Room" temperature.

It was required therefore to calculate the values of  $UA_R$  and  $V_R$ . It must be said that these would be effective values appropriate for the model, should it be possible to make the model behave like the real system. That is,  $V_R$  would very likely be unequal to the physical water volume of the radiator system, but would be an effective volume whose value allowed for the effect of pipe/radiator metal thermal capacity. Also  $UA_R$  would be different from a real UA-value based on radiator surface area, but would be an effective value allowing, for example, for the assumption that the entire radiator system operated at the water return temperature (plainly false), the non-linearity of the real radiator heat transfer characteristic (convection plus radiation), and dependence on water flow rate. A value of the water

flow rate through the boilers and distribution system was eventually measured reliably to be 5.27 litres/second. Values of  $UA_R$  and  $V_R$ , for the "bath" radiator model, were sought by means of the following simple methods. The form of Figs 2.4 and 2.5 is as follows:-



Section A of the diagram represents system warm-up, when the boilers are supplying heat both to heat the system contents and to heat the school. During Section B, the compensated temperature has been achieved, and the boilers' output is all passed into the school.

$UA_R$  : If individual hours of Section B of Figs. 2.4 and 2.5 are chosen such that flow and return temperatures are approximately constant then the change in heat content of the boilers and radiator systems is very small, and heat supplied by the boilers equals heat transferred to the building:-

$$\dot{m}C_p (T_1 - T_R) \approx UA_R (T_R - T_{ROOM}) \quad (1)$$

where  $\dot{m}$  is the water flow rate and  $C_p$  is the specific heat capacity of water. All terms in equation (1) are recorded on Figs. 2.4 and 2.5 except  $UA_R$ . Taking four approximately steady-state hours from Fig. 2.4, and one from Fig. 2.5, an average value of 3550 W/°C was given for  $UA_R$ .



$V_R$  : During Section A of Figs. 2.4 and 2.5, the bulk of the radiator system being at  $T_R$  (consistent with the model), we can say that the rate of system warm-up is  $(V_B + V_R) \cdot Cp \partial T_R / \partial t$ , and equation (1) is amplified to:-

$$\dot{m}Cp (T_1 - T_R) = (V_B + V_R) \cdot Cp \cdot \frac{\partial T_R}{\partial t} + UA_R (T_R - T_{ROOM}) \quad (2)$$

where  $V_B$ , the volume of the boilers is taken to be around 137 litres of water (from the supplied Hamworthy data above). Again, taking four points from figs. 2.4 and 2.5, the average value of  $V_R$  was calculated to be about 3690 litres.

(b) "Plug flow" model

It was believed preferable to consider a second radiator model, whereby the incoming radiator water at  $T_1$  is supposed to cool gradually, as it passes through the radiator, to emerge at  $T_2$ . It is further supposed that there is a minimum of mixing during the passage, hence the term "plug flow".

Consider a particle of this flow, volume  $V_p$ , temperature  $T$ , losing heat through a transfer coefficient  $(UA)_p$  to room temperature  $T_{ROOM}$ . Its time of passage will be  $t_p = V_R / \dot{m}$ . Thus, at any instant:-

$$\begin{aligned} \text{Rate of heat loss from particle} &= (UA)_p \cdot (T - T_{ROOM}) \\ &= -V_p \cdot Cp \cdot \partial T / \partial t \end{aligned}$$

Thus

$$\frac{dT}{T - T_{ROOM}} = - \frac{(UA)_p}{V_p \cdot Cp} \cdot dt$$

Integrating:

$$\left[ \log_e (T - T_{ROOM}) \right]_{T_1}^{T_2} = - \frac{(UA)_p}{V_p \cdot Cp} \cdot t_p$$

Thus

$$\log_e \left( \frac{T_1 - T_{\text{ROOM}}}{T_2 - T_{\text{ROOM}}} \right) = \frac{(UA)_p \cdot V_R}{V_p \cdot \dot{m} \cdot C_p}$$

or

$$\dot{m} C_p \log_e \left( \frac{T_1 - T_{\text{ROOM}}}{T_2 - T_{\text{ROOM}}} \right) = \frac{(UA)_p \cdot V_R}{V_p}$$

We have assumed a uniform radiator, where the particle is able to transfer heat to the school through a constant  $(UA)_p$ . Thus  $(UA)_p/V_p$  is the  $UA/\text{unit volume}$  of any small portion of the radiator.

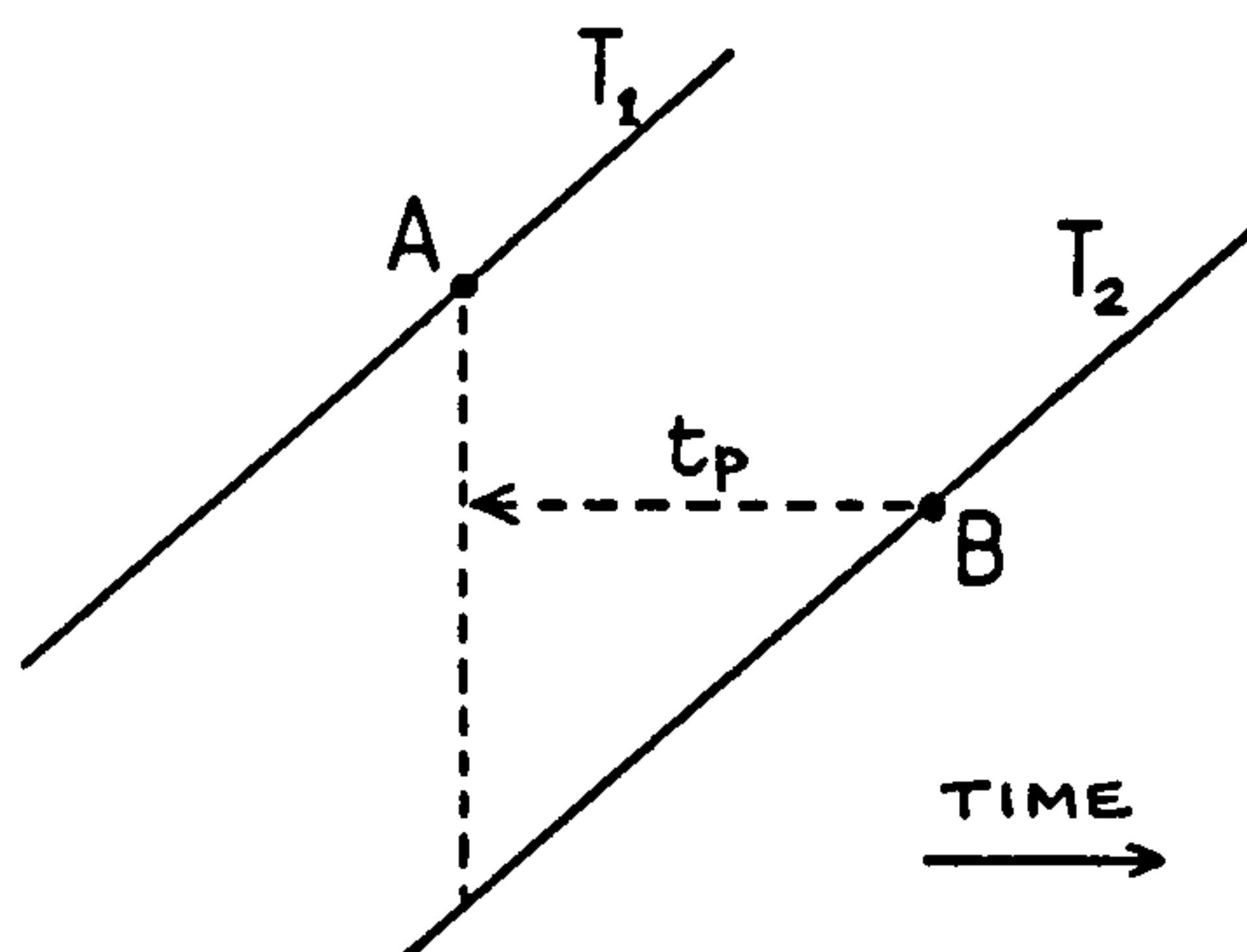
Thus

$$UA_R = \dot{m} C_p \log_e \left( \frac{T_1 - T_{\text{ROOM}}}{T_2 - T_{\text{ROOM}}} \right) \quad (3)$$

$UA_R$ : This time a total of eight points were taken from steady-state periods of Section B of Figs. 2.4 and 2.5, again taking  $\dot{m} = 5.27$  litres/second. The fraction  $(T_1 - T_{\text{ROOM}})/(T_2 - T_{\text{ROOM}})$  averaged 1.1606 with a maximum value of 1.1664, and minimum of 1.1541, i.e. a range of values equal to 10% of the minimum value. This translates via equation (3) to a value of  $3280 \text{ W}^\circ\text{C}$  for  $UA_R$ ,  $\pm 230 \text{ W}^\circ\text{C}$ . The small scatter of the eight carefully prepared values perhaps indicates that the plug flow model is more appropriate.

$V_R$ : To estimate  $V_R$  for plug flow, a different method is required. Equation (3) may be rewritten:-

$$\frac{T_1 - T_{\text{ROOM}}}{T_2 - T_{\text{ROOM}}} = e^{\frac{UA_R}{\dot{m} C_p}} = 1.1606 \quad (4)$$



Section A of Figs 2.4 and 2.5

Taking the warm-up period, Section A of Figs. 2.4 and 2.5, the particle of heated water enters the radiator at point A, see diagram above. It will emerge several minutes later at temperature  $T_2$ , which is calculated from equation (4), and point B is located on the diagram. The time of passage  $t_p$ , can then be read off a carefully plotted, magnified graph of  $T_1$  and  $T_2$ . Then

$$V_R = \dot{m} \times t_p \quad (\text{as above})$$

A careful evaluation from five points taken from Figs. 2.4 and 2.5 gave a mean value of  $V_R$  of 3580 litres.

These very approximate evaluations of the system therefore yield the following figures:-

	$UA_R (W/^\circ C)$	$V_R$ (litres)
Bath model	3550	3690
Plug flow model	3280	3580

Within the limits of accuracy inherent in these calculations there appears to be little to choose between the two models; the plug flow model was preferred, however, for the subsequent modelling. What was encouraging, though, was that the two methods gave almost the same figure for  $V_R$ , taken to represent approximately the water equivalent of the radiator system, at about 3600 litres. Thus a first idea of required store capacity could be ventured. If the store was to fill the radiator system at initial warm-up on the day of figure 2.4, a compensated temperature of some  $68^\circ C$  would be called for. Suppose that the radiators were initially at the minimum acceptable classroom temperature of  $10^\circ C$  (maintained by the frost thermostat), and that the store could be heated to  $90^\circ C$  with careful adjustment to the boiler thermostats. The required store volume would be  $V_S$ , where

$$V_S = \frac{3600 (68-10)}{90-10} = 2600 \text{ litres}$$

This was likely to be a minimum storage requirement, as the calculation implied that the entire store content was perfectly transferred to the radiators, a condition probably very difficult even



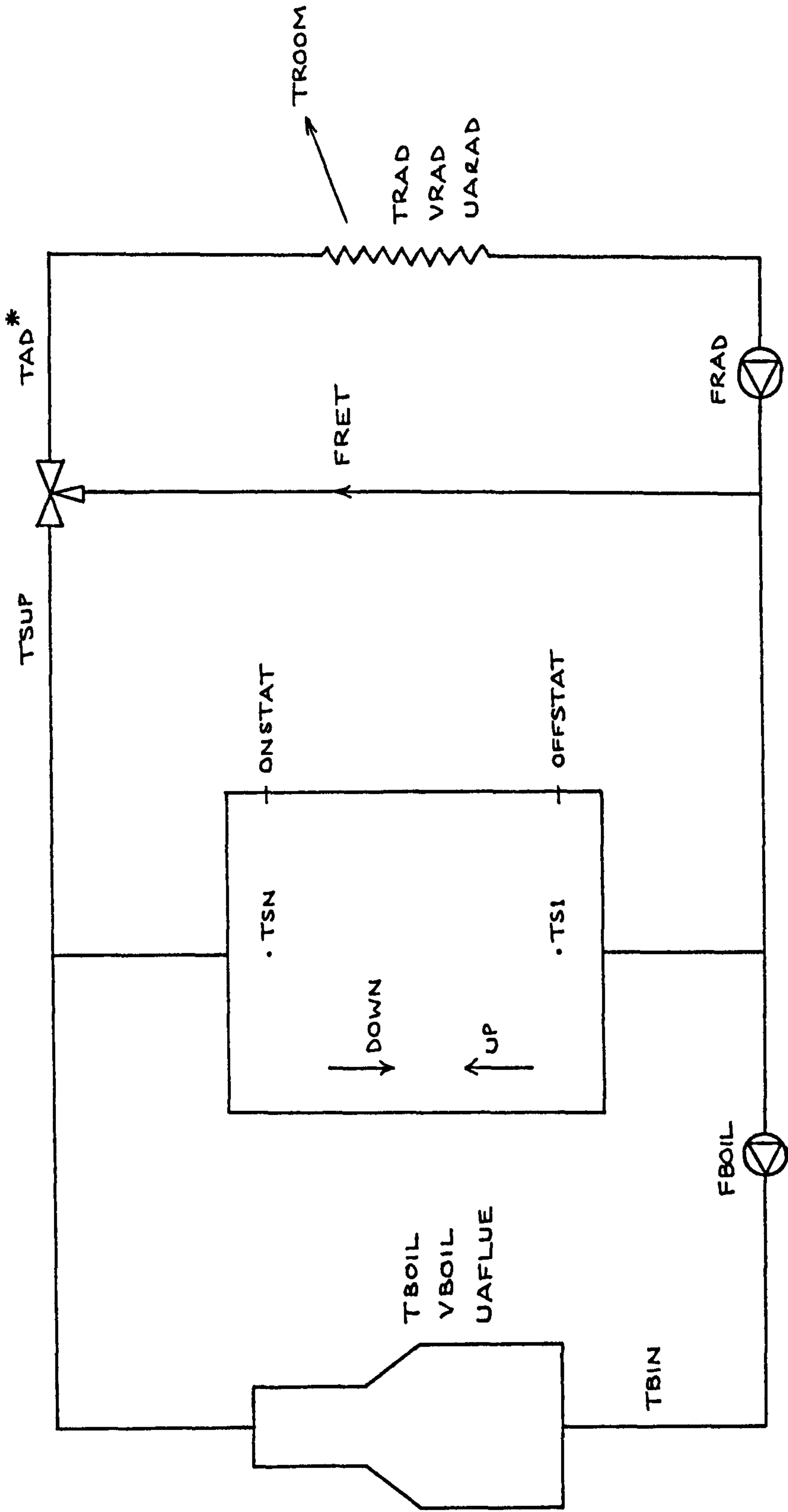
to approach in practice. A volume of about 3000 litres was thought to be near the eventual requirement.

To investigate the school's likely performance with the inclusion of storage, a computer model of its behaviour was evolved.

## 2. Computer Model of the Thermal Behaviour of Spa School with Storage

A simple, time-step program was set up to model the school's operation after the inclusion of a large water store in the heating system, see fig. 2.6. It comprised the following elements:-

- a boiler model similar to those used in Part I of the thesis, supposing the three Hamworthy boilers to be represented by a single, fully-mixed volume of water. A value of  $29 \text{ W/}^\circ\text{C}$  was used for the heat transfer coefficient to flue, UAFLUE, though this was subsequently considered too small. As discussed above, however, the value of UAFLUE is not critical to an accurate assessment of a storage heating system, providing the cycling frequency is kept low.
- a water store model incorporating two zones only. It was thought ill-advised to represent stratification of the store by more than two zones because the flow rate through the store was likely to be relatively high. This decision was taken on the advice of Robert Cohen, whose relevant experience (presented in April 1984, ref. 2.6) was gained on tests on a 2800 litre store, at Cranfield.
- a plug flow radiator, whose volume was assumed to be divided into a large number of small, fully-mixed radiators, of equal volume and UA-value, to approximate to the plug-flow principle.
- two independent, fixed-rate water pumps serving the boiler circuit and the distribution circuit.
- a three-way valve (3WV) to mix the hot water available from the boiler and/or store with the radiator return water to achieve the compensated temperature at radiator inlet. If boiler/store delivery temperature was inadequate, the 3WV would open fully to maximise the



\* TAD - TEMP. ADMITTED TO RAD<sup>o</sup>.

TREQ - COMPENSATED TEMP. FROM FIGS. 2.4 AND 2.5

FIG 2.6: NOMENCLATURE FOR MODEL OF STORAGE SYSTEM, PROG. CALLED "SPAP.FOR", APP<sup>x</sup> 2.1

temperature at the entrance to the radiators.

- the boilers would switch off when the lower zone of the store achieved a fully charged temperature (OFFSTAT). The boilers would switch on when the store's upper zone fell to the prevailing compensated temperature (ONSTAT was automatically controlled to equal compensated temperature). Thus the store's capacity was enhanced for mild days.

It was decided that the proposed storage system should, in the first instance, supply the same heating schedule as the conventional system. Figs. 2.4 and 2.5, therefore, were reduced to hour-by-hour averaged values of:-

- compensated temperature (TREQ in the program)
- room temperature (TROOM)
- total heat required for that hour (QREQ)

TREQ and TROOM could then be imposed as input data on the model (expanded to simulate an entire day's running) and the hourly heat delivered by the boiler/store compared with QREQ. Thus inadequacies of heat supply could be identified quickly.

Values of store size, radiator volume and radiator UA-value could thus be manipulated, and it was hoped to match the design to the school's requirement. The program is included as Appx. 2.1. In fact, it was found difficult to make the hour-by-hour output from this program agree satisfactorily with the data of Figs. 2.4 and 2.5, i.e. though the TREQ profile drawn from Fig. 2.4 (say) was imposed as input on the program, a poor correlation with the return temperature profile of Fig. 2.4 was obtained. Considerable "juggling" with values of e.g.  $V_R$  and  $UA_R$  did not prove satisfactory; normally, if a good match was obtained for the transient operation (Section A) of fig. 2.4, it would be to the detriment of the results for Section B, and vice versa. The model in this form, therefore, was not developed such that reliable predictions of daily performance figures (gas consumption, efficiency etc.) were produced, and the program's output will not be presented here for this reason. At this early stage it was sufficient to



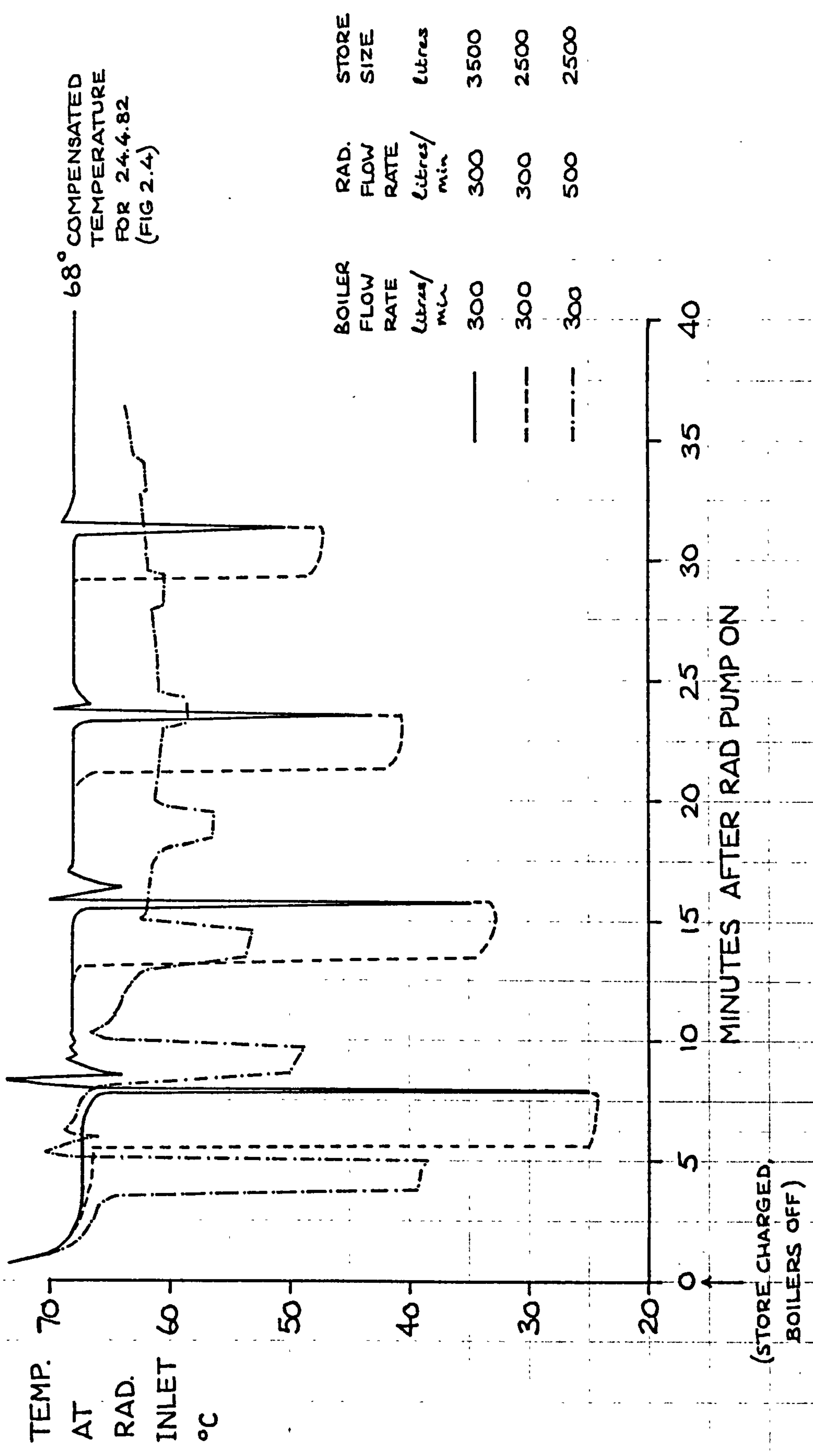


FIG 2.7: COMPUTER PREDICTIONS OF FIRST STORE DISCHARGE FOR COLD DAY OF FIG 2.4.

observe that the program's output suggested:-

- the 'hoped-for' low boiler cycling, perhaps less than six cycles/day
- good values of efficiency, up to about 97% of  $\eta_{FL}$ .

The model was reworked subsequently by Robert Cohen, and results from its revised form are more trustworthy. Again, the reader is referred to the anticipated full report on the field trial, and to ref. 2.2, section 3.2.

For the purpose of sizing the store at this early stage, the program was used solely to examine the heat interchange between the store and the radiators during the first discharge, the process which is critical to store sizing, as described above.

Fig. 2.7 shows the computer prediction of this interchange, supposing that the charged store were required to supply water at the compensated temperature for the day of Fig. 2.4, i.e. at 68°C. It was found that store size would have to be about 3500 litres to maintain this 68°C uninterrupted at the radiator inlet, and this figure was chosen. The author's participation in the project's course ended here for the time being.

Computer predictions apart, however, the physical limitations of the space available in the school's boiler room and the limited access to the boiler room, necessitated the final choice of store size to be 3300 litres. Robert Cohen, working with British Gas engineers, was responsible for drawing up relevant diagrams, and installing the store, during the spring and summer of 1984. Photo 2.3 is a view of the installed storage system, showing the store (note the thermostats and profile P.R.T's) and some of the considerable length of pipework required. The author was again called upon when the system was started up, in November 1984, to collect data and commence preliminary analyses, as follows.



PHOTO 2.3:  
VIEW OF BOILER  
ROOM WITH STORE





MONITORING

During installation, the storage system was fitted to accommodate some 45 sensors, which included the following:-

- gas meter reading
- four water flow rates : into boilers, into radiator circuit, upstream of the distributing three-way valve, and from boilers to store/radiators. These were all electromagnetic flowmeters, providing either integrated or instantaneous output.
- temperatures to and from both the boilers and the distribution circuit, measured by both thermocouple (t/c) and platinum resistance thermometer (P.R.T.).
- a series of ten P.R.T.'s equally spaced to measure the store's vertical temperature profile.
- flow temperatures in the three limbs adjacent to the distributing three-way-valve (t/c)
- temperature at store inlet and outlet ports (t/c)
- temperature in a selected classroom (two sensors), temperature on the school wall (shaded) for outside ambient, and three sensors at various points of the boiler room itself.

Each sensor was fed to a channel of a Solartron "Orion" datalogger. Scans were programmed into the Orion, each containing fifteen of the above readings, as follows, referring to Fig. 2.8:-

FAST SCAN : TA, TB, TC, TBMIX, TSTU, TSTL, TFRAD, TTRAD, TTBOIL,  
TFBOIL, FBOIL, FRAD, GAS, FSTORE, FSIDE.

This scan was labelled either 2 or 4 on the logger, and set to operate every 6 minutes while either water pump operated, and also every minute (for 10 minutes) when either pump was switched on or off. The

• Tc (class)

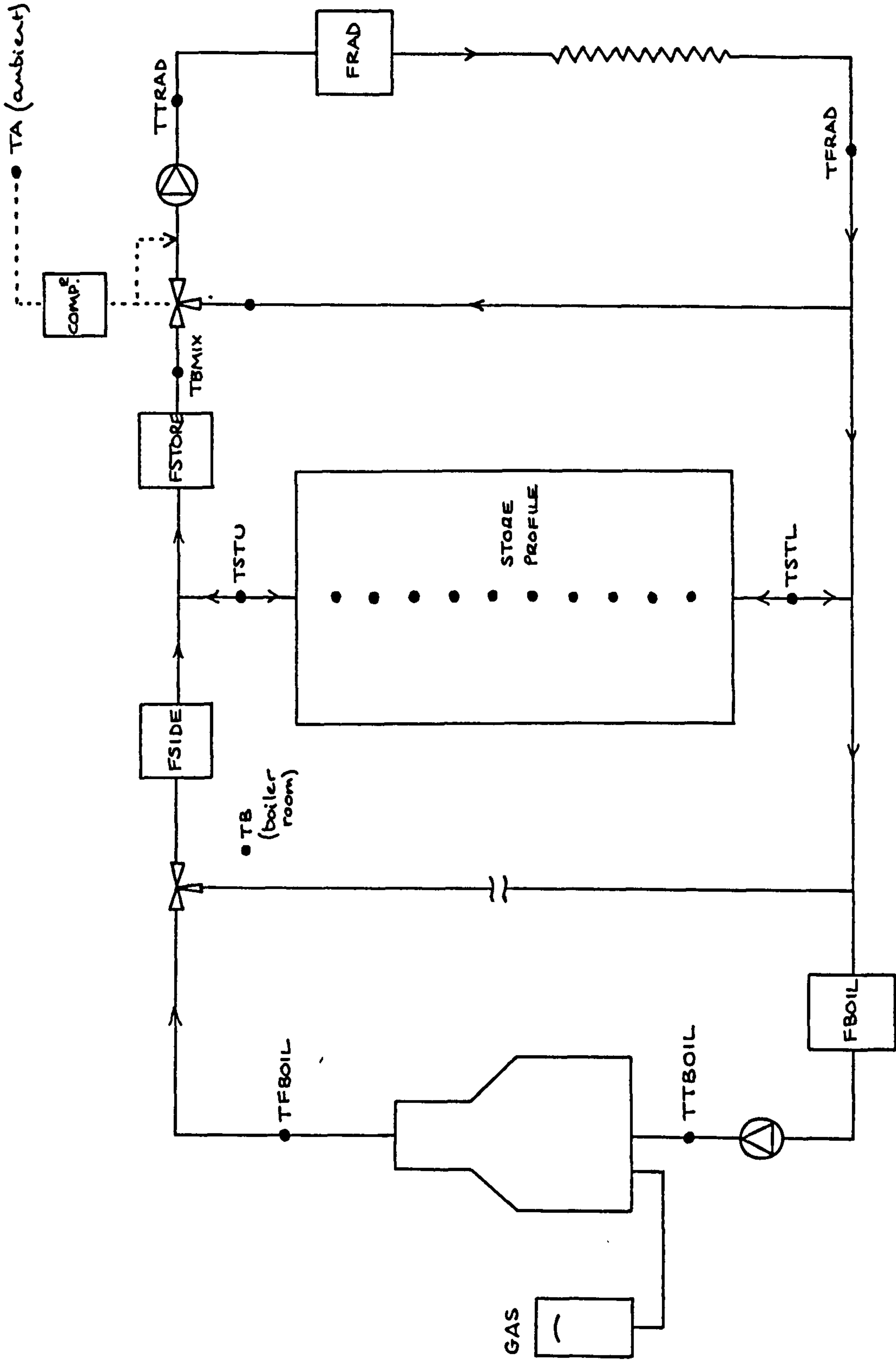


FIG 2.8: PARAMETERS SAMPLED ON FAST AND SLOW SCANS

latter timing gave an opportunity to examine the transient flows more accurately.

SLOW SCAN : this scan read the store profile temperatures (bottom to top) in place of the temperatures of the FAST SCAN. It was labelled scan 6 on the logger, and set to operate every hour on the hour, and was thus the only overnight reading taken.

The Orion recorded each scan onto a "Scotch" mini-cassette, which under normal operating conditions, had sufficient capacity to retain about two weeks' data. Cassettes were therefore changed as appropriate; recorded data were read at British Gas through a Quantex decoder onto the Watson House Harris computer, and then read off onto a seven-inch magnetic tape compatible with the VAX 785 computer at Cranfield.

#### DATA ANALYSIS AND INTERPRETATION

As described above, the data were stored on the VAX computer at Cranfield, largely in blocks of some two weeks' readings, consistent with visits to Watson House. These blocks could easily be subdivided, and chosen days of the heating season were isolated for analyses. All the analysis presented here refers to individual days; the analysis has since been extended to week-by-week analysis (by Robert Cohen at Cranfield). The analysis which follows is therefore a very preliminary look at the figures available from the 1984-5 monitoring, and is included as much to present the technique of analysis used, as to try to formulate numerical conclusions. Indeed, during the 1985-6 season currently being monitored, slight measurement inaccuracies (which will have been operative during the 1984-5 season) have been pinpointed. No attempt has been made to apply correction factors to the 1984-5 data recorded here, as more authoritative, quantitative results will be drawn from analysis of the current season. For example, it is thought that the boiler flow rate was recorded some 3% higher than its actual value, a considerable drawback when trying to quantify the benefit of introducing a storage system, which might be as low as 5-10% (ref. 2.2, section 1.17). In addition, the data from before 5th February 1985 have been discarded (the day of a flowmeter



change); the season investigated has been thus limited to 6th February to 20th May 1985. (See also discussion in section 2 below, "Numerical Analysis".)

A total of 26 individual days were isolated and analysed. These included six Mondays; due to the school's cooling during a weekend, the Monday morning start-up time was set some 2.75 hours earlier than the other weekdays. Mondays were almost invariably the days of heaviest boiler duty, as will be explained. Three analysis programs were written; the school's behaviour during the above part of the 1984-85 heating season will now be examined by means of the results from two of these programs, viz:

- (i) GRAPH21.FOR - a simple graphics program to depict any particular day's c.h. performance, and
- (ii) SPAEFF.FOR - a brief summation program devised to quickly evaluate gas consumption, efficiencies, etc.

### 1. Graphical Presentation

GRAPH21 was developed to operate on the data recorded during any entire day, and plot six of the 14 (or 15 after March 18th) parameters recorded on each scan of the datalogger. These six parameters were:-

- Temperature available at the store/boiler side of the distributing 3WV ("T before 3WV")
- Temperature admitted into radiator circuit, regulated by external compensator ("T rad. flow")
- Temperature returned from radiator circuit ("T rad. return")
- An index showing the boiler operation ("Boiler on").
- Classroom temperature ("T class").
- Outside ambient temperature ("T ambient").

Thus a day's operation could be quickly, visually assessed by this program. To qualitatively examine the variation of the storage system's operation over the cold-to-mild half-season, graphs depicting six individual days have been selected, and are presented on Figs. 2.9 to 2.14 arranged in descending order of severity (i.e. not chronologically):-

Fig. 2.9	:	15th February,	19.43 °days (rel 18°C)
Fig. 2.10	:	8th February,	14.70 °days
Fig. 2.11	:	13th March,	12.06 °days
(Fig. 2.11a	:	13th March,	start-up only)
Fig. 2.12	:	22nd April,	7.81 °days
Fig. 2.13	:	19th April,	4.09 °days
Fig. 2.14	:	20th May,	2.65 °days

(°day figures were produced by the SPAEFF analysis which follows this section).

The system's operational sequence was as follows:-

- Boilers on to heat store only 04.00 hrs. (until late season), but 2.75 hrs. earlier on Mondays, e.g. Fig. 2.12, 22nd April.
- Radiators on, boilers off, as soon as the store is fully charged (to some 90°C).
- Radiators discharge store alone until "T before 3WV" falls to the temperature demanded by the compensator, at which point the boilers fire again.
- Boilers supply radiators and store until the store is recharged, when boilers cut out again and the store is discharged;

charge/discharge cycles supply heat all day.

- Radiators remain on until 15.00 hrs.
- Boilers off 14.00 hours; heat supplied to radiators by store only 14.00 - 15.00, to deplete store.

Central to all the graphs are the curves for the three temperatures at the distributing 3WV; "T before 3WV" is mixed with "T rad. return" to admit "T rad. flow" into the radiator circuit. Thus if the "T before 3WV" and "T rad. flow" curves are coincident, then the temperature of the flow produced by the boiler/store combination is inadequate, and the 3WV is fully open. Again if the "T rad. flow" and "T rad. return" curves are coincident, then radiator temperature is too high, and the 3WV is fully closed with no admission from the boiler or store. Usually the 3WV will be in a position between the two extremes and the upper three curves of Figs. 2.9 to 2.14 are mostly separate. The school's heating system was required to give a minimum of  $18^{\circ}\text{C}$  between 09.00 and 15.00, in the classroom. Empirically,  $20$  or  $21^{\circ}\text{C}$  was found to be preferable.

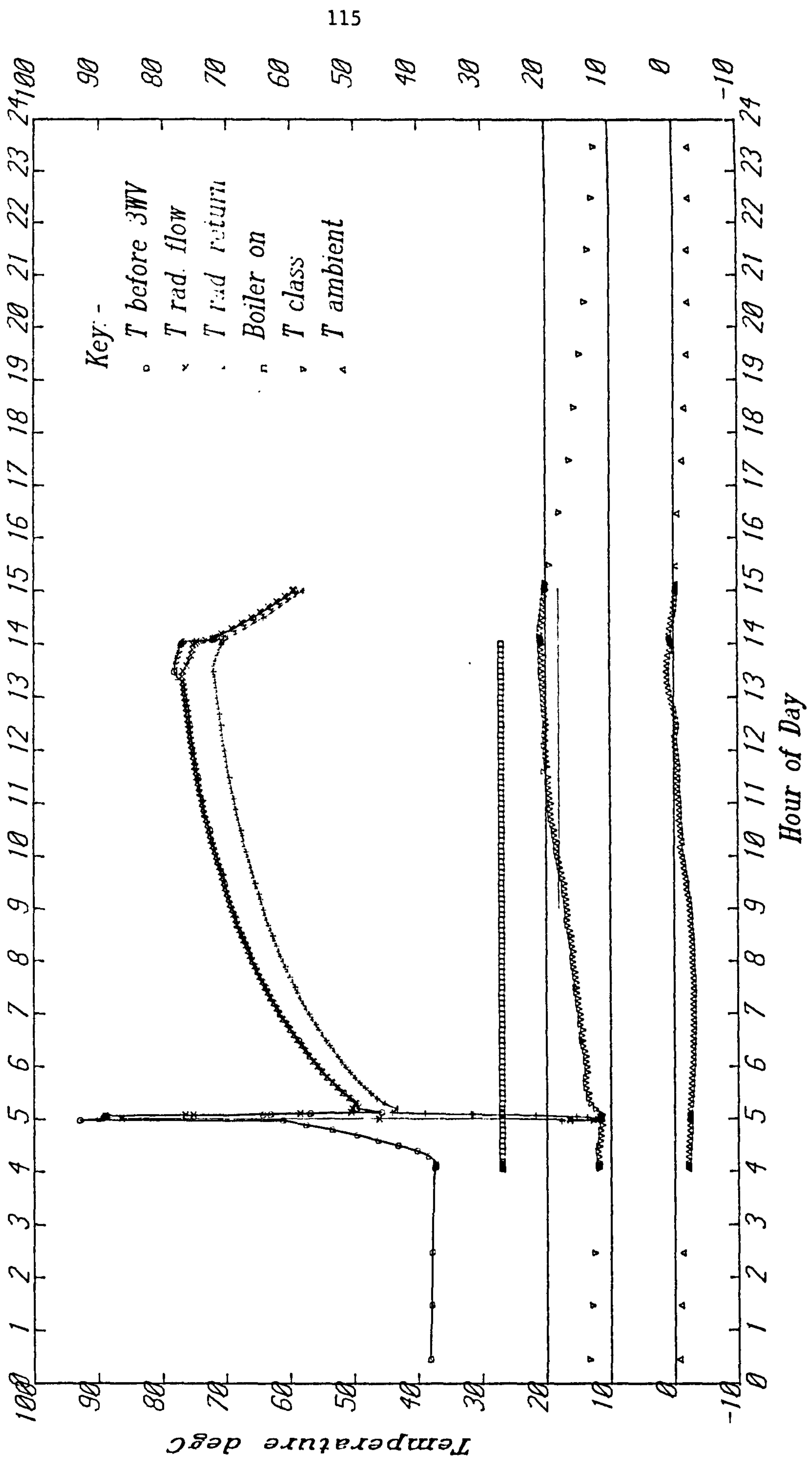
On a cold day, the compensated temperature is so high that the initial discharge of the store is accomplished very rapidly, sometimes in as little as five minutes. Indeed, on such a day, it would be preferable to leave the boilers running during this discharge. Fig. 2.11a shows such an early-morning discharge from mid-season.

Each of the graphs for the six days chosen will now be examined briefly.

(i) Fig. 2.9, 15th February,  $19.43^{\circ}$  days : this day is close to the design day of a constant  $-1^{\circ}\text{C}$  ambient temperature, i.e. a cold day. Fig. 2.9 shows that the class temperature is about  $12^{\circ}\text{C}$  at start-up and outside ambient about  $-3^{\circ}\text{C}$ . After the initial very rapid discharge of the store, the boilers are required not only to provide radiator heat (itself a huge demand under these conditions), but also to recharge the store. The storage principle requires that the store should if possible be of sufficient thermal capacity to replace the cold radiator water with water at the compensated temperature



FIG 2.9: SPA SCHOOL : 15-02



(here  $> 75^{\circ}\text{C}$ ). In this instance the store is inadequate for this purpose. What is more, the inadequacy is compounded by:-

- the increased thermal inertia of the system (storage volume)
- the reduced boiler power (three boilers rather than five).

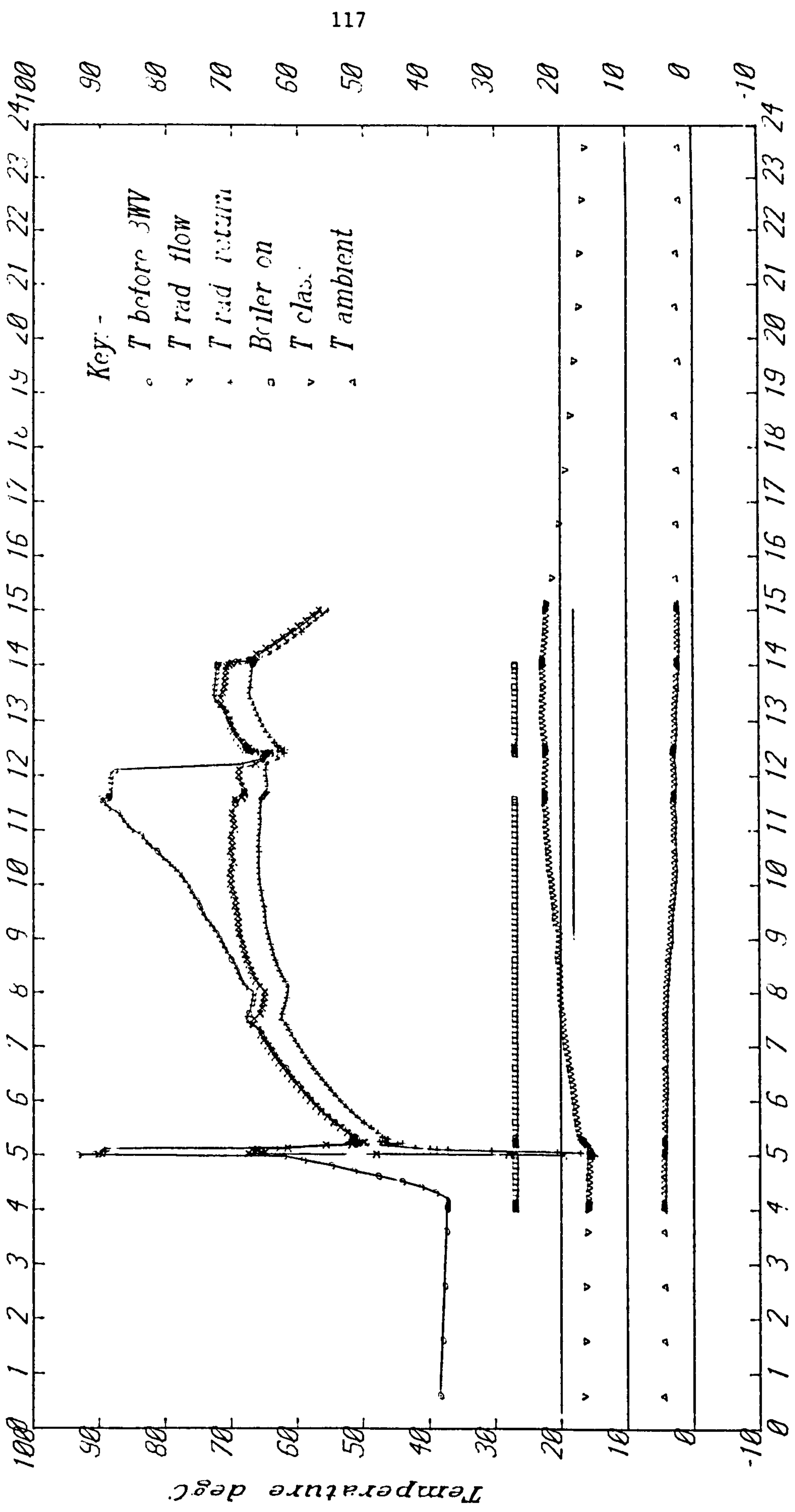
So the "lugging" phase of the recovery, i.e. 05.20 a.m. onwards, proceeds very slowly, and compensated temperature is only finally reached at 13.20. It is clear that, in such an instance, it would be preferable to install a shut-off valve to isolate the discharged store at 05.20, such that the boilers raised the radiator temperature more quickly up to the compensated temperature.

Thus the classroom temperature rises quickly as a result of the initial discharge, but climbs very slowly thereafter, failing to achieve the required  $18^{\circ}\text{C}$  by 09.00 a.m. If the store had been isolated during the "lugging" phase, the school may well have been adequately heated.

This case shows that the installation of a storage system does offer potential benefits (reduced cycling, reduced boiler plant, fast release of hot water initially, etc.), but these benefits can be overturned by inadequate control of the system. In fact, for the current 1985-6 season, a valve has been fitted to isolate the discharged store when necessary. Note also that the store volume would have had to be very large indeed (probably  $> 5000$  litres) to replace fully the radiator cold water initially. With any system, unless a huge store is fitted, there are likely to be days when this full use of the storage principle is approached rather than achieved; it is on these days that skill is required to make the best use of the installed system.

(ii) Fig. 2.10, 8th February,  $14.70^{\circ}$  days : a cold, typical winter day, with outside ambient temperature almost constant at between  $5^{\circ}\text{C}$  and  $2^{\circ}\text{C}$ . This time, however, the class temperature is quite high, some  $16^{\circ}\text{C}$ , at the start-up. After the initial store discharge, which again is very rapid the boilers again "lug" both the radiators and store up to the compensated temperature. As the rate of heat

FIG 2.10:SPA SCHOOL : 08-02





delivered to the class is less than on Fig. 2.9 at this stage, the rate of warm-up of the system is slightly higher, and the less severe ambient environment allows the lower compensated temperature of about  $68^{\circ}\text{C}$  to be attained much sooner (at 07.20 hrs.). After this, radiator demand falls below the boiler output, and the store is charged beyond compensated temperature, the upper two curves of Fig. 2.10 separating. The rate of climb of the top ("T before 3WV") curve at this stage is an indication of the rate of charge of the store. The second store discharge, starting at 11.30 hrs. is far longer this time, and a second recharge starts from about 12.20 hrs. Note that, just after 12.00, the cold front travelling up the store reaches the distribution 3WV, and there is a brusque fall of delivery temperature while the signal is conveyed to the boilers to restart and they attain a sufficient temperature to satisfy the compensator's demand. This period might be reduced with more positive controls, perhaps anticipating the store's depletion.

The compensator regulates flow temperature into the radiators such that this latter is in a one-to-one relationship with the ambient environmental temperature (see Fig. 2.15), i.e. compensated temperature is independent of class temperature. Thus, though the class started from as high as  $16^{\circ}\text{C}$ , it receives heat beyond its requirements, and attains  $23^{\circ}\text{C}$  or so at about 13.00 hrs. Though this is hardly excessive, if a temperature of  $18^{\circ}\text{C}$  -  $19^{\circ}\text{C}$  say in the classroom is comfortable, considerable savings would be achieved by ensuring that the system did not heat the school beyond about  $20^{\circ}\text{C}$ . Thus a slightly more sophisticated controller, perhaps a microprocessor, which assessed radiator flow temperature as a function of both class and ambient temperatures, would be a potential large-scale energy saver.

Further, after start-up, Fig. 2.10 shows that the  $18^{\circ}\text{C}$  required in the class was achieved very soon after the radiator pump was started, i.e. at about 06.00 hrs., or after about one hour's heating of the school. If  $18^{\circ}\text{C}$  were not required, theoretically until 09.00 hrs. then the heating start-up might have been left as late as 07.40 hrs. say (allowing for the further cooling of the school). Thus over two hours of heating would have been unnecessary.

Thus two possible large-scale energy savers have been identified:- (a) class temperature input into the calculation of radiator inlet temperature, and (b) delay of heating start-up according to initial classroom temperature. These would stand or fall on the engineer having access to a representative class temperature. The "black bulb" temperature sampler used here was installed in a west-facing room allegedly one of the warmest, sensibly, in the building. It was suggested that, as the two potential improvements above may prove to be great economisers of energy, the relevant extra costs of a comprehensive classroom temperature measuring system and a microprocessor (together possibly with the use of T.R.V.'s on all radiators) might be recovered very quickly.

(iii) Fig. 2.11, 13th March, 12.06 °days : a slightly less severe day than 8th February above, but displaying the typical early spring feature of a cold night, but a much warmer day due to solar incidence. The main features of Fig. 2.11 are the customary rapid first store discharge, followed by a "lugging" spell similar to that of Fig. 2.10 above, and then as the outside ambient rises quickly, two charge/discharge cycles, the discharge times extended by the correspondingly diminishing compensated temperature. Note that the third store charge of the day (12.00 hrs.) takes place at a more rapid rate than the second (08.00 hrs.), the slope of the uppermost curve of Fig. 2.11 at these stages reflecting the proportion of boiler output being absorbed by the store.

Note also, again, that class temperature at the start is such that 18°C could have been achieved very quickly indeed; with a rigorous control of class temperature to just above 18°C, the start-up time of the radiator circuit could have been considerably delayed, to the exclusion of over two hours of early-morning heating.

One further suggestion arises from the fact that the store's available energy at 04.45 a.m., when the heating came on, was proportional to  $\Delta T$ , where

$$\Delta T = (\text{charge temperature, } \sim 90^{\circ}\text{C}) - (\text{compensated temperature, } \sim 72^{\circ}\text{C})$$

FIG 2.11: SPA SCHOOL : 13-03

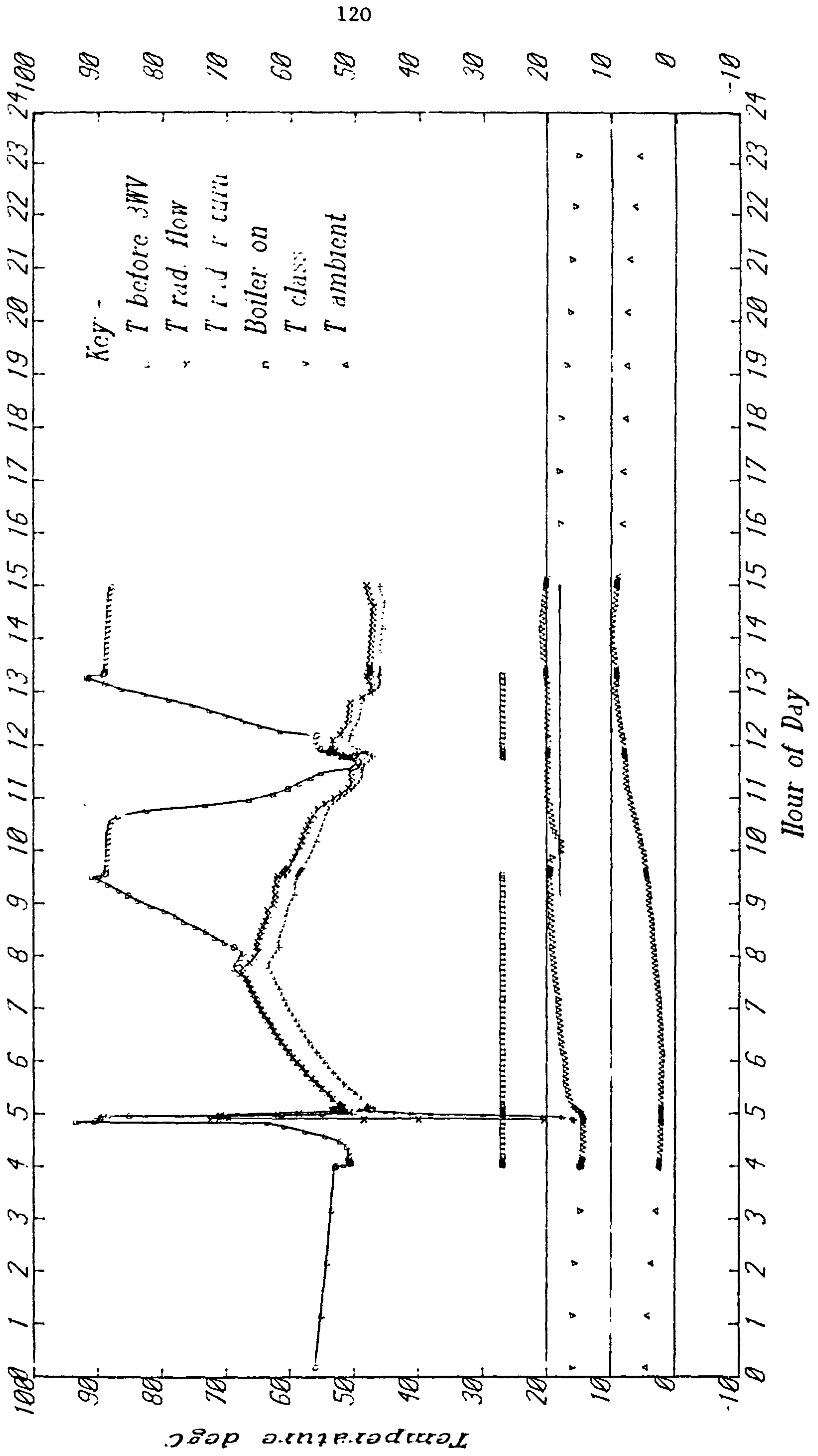
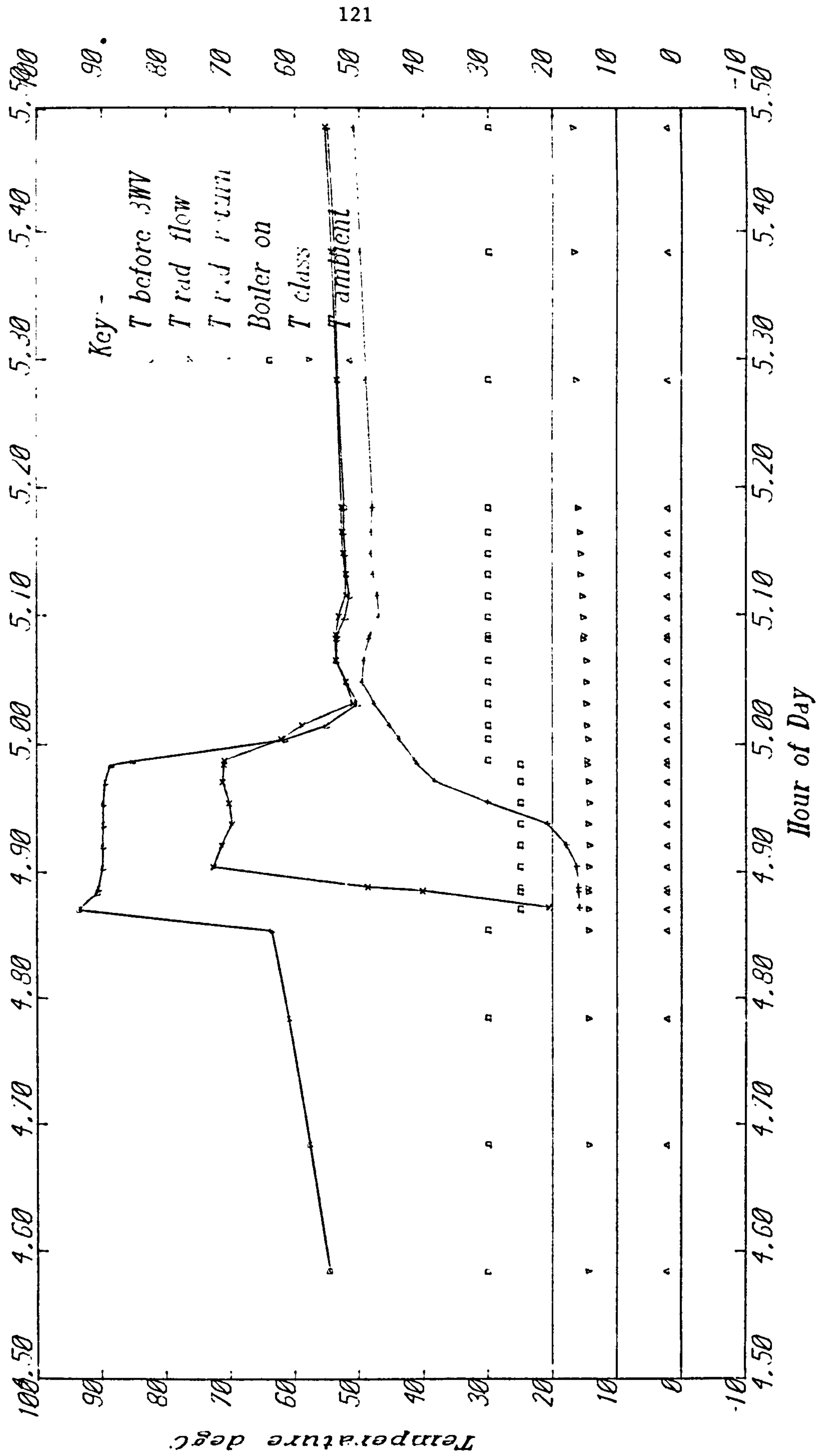




FIG 2.11a:SPA SCHOOL : 13-03



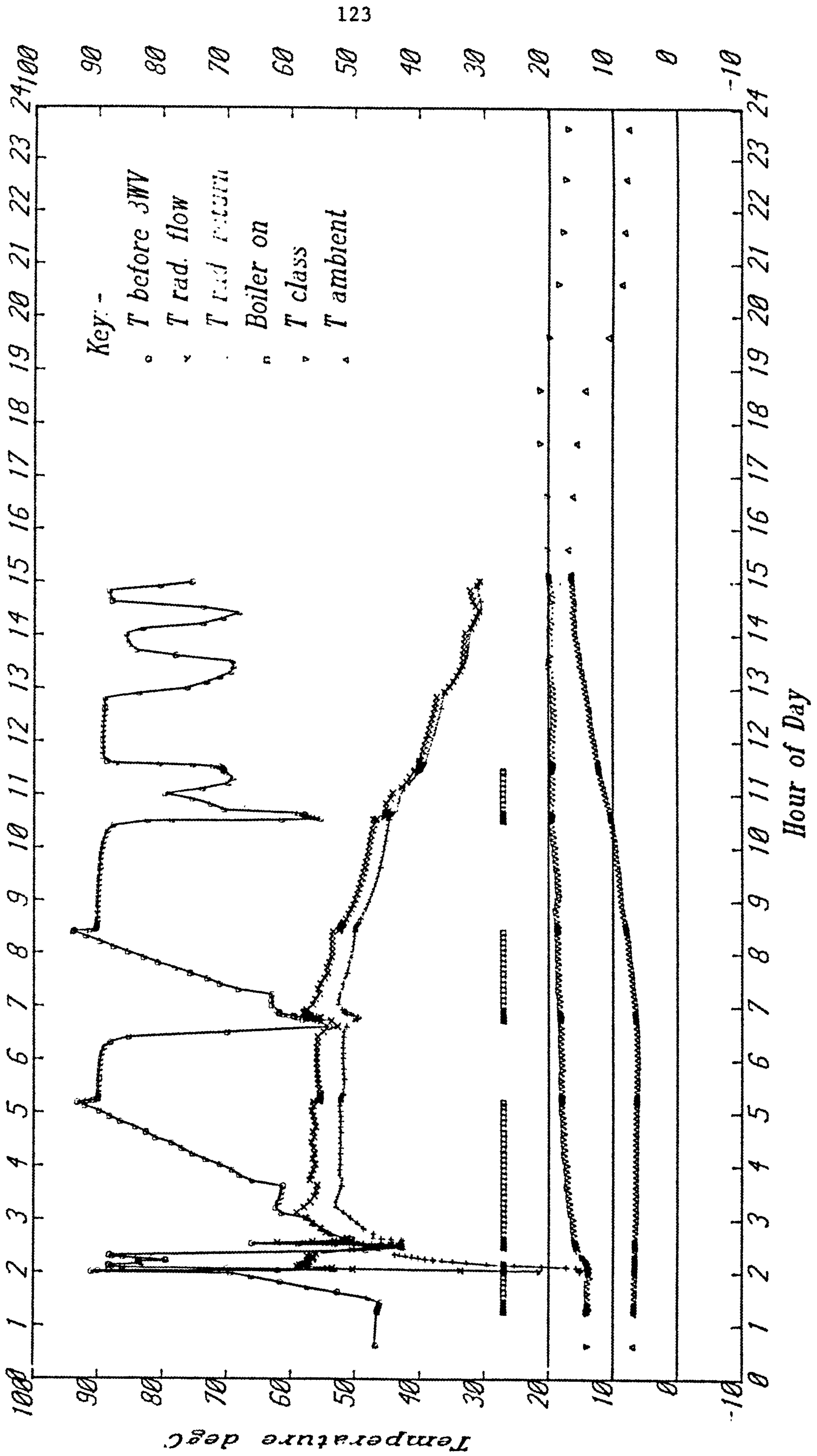
This available energy was just insufficient to heat the school to  $18^{\circ}\text{C}$  (a class temperature of about  $17^{\circ}\text{C}$  was attained at the start of the "lugging" period). Suppose now that a sophisticated controller, aware of the school's total requirement to attain  $18^{\circ}\text{C}$ , calculated a revised  $\Delta T$  such that the store was discharged to a temperature where the total energy released equalled the requirement of the building. The new  $\Delta T$  might be  $30^{\circ}\text{C}$  say, i.e. the store would discharge to  $60^{\circ}\text{C}$  before being depleted. Then, if the initial discharge to the building were released at  $60^{\circ}\text{C}$ , instead of the customary  $72^{\circ}\text{C}$ , the extra energy available from the store might itself raise the building to  $18^{\circ}\text{C}$  before the boilers came on again. If, now, the store were blanked off, the boilers could proceed to maintain radiator temperature (or increase it if necessary). Thus the building could probably be heated all the way to  $18^{\circ}\text{C}$  by the store only, albeit at a lower radiator temperature and therefore at a rather lower rate. The present rate at first discharge is so high, however, that a warm-up at even half this rate would still be very fast. For the day recorded here, a warm-up of the entire school to  $18^{\circ}\text{C}$  might require no more than a half-hour and this from a single store discharge.

It is further evident that, even if the present compensator principle is retained where the radiator temperature is dependent on the outside ambient environment only, there is little need for very hot temperatures to be called for before 09.00 hrs. A relaxation to, say,  $60^{\circ}\text{C}$  maximum might give a better use of the store, though warm-up rate would be slightly impaired.

Fig. 2.11a shows the first discharge of the store for 13th March 1985, to an expanded scale. Note that each time division is for six minutes here. The second curve, "T rad. flow", shows very positive 3WV control to compensated temperature, no overshoot being recorded. The store is discharged in 6 - 7 minutes, its available energy being restricted by the very high value of the temperature demanded by the compensator.

(iv) Fig. 2.12, 22nd April, 7.81  $^{\circ}\text{C}$  days : a cool night ( $7^{\circ}\text{C}$ ) followed by a warm day, such that the heating might well have been switched off after about 13.00 hrs. Being a Monday the system was started at 01.15

FIG 2.12: SPA SCHOOL : 22-04





hrs. to conform with the schedule of the previous conventional system. A glance at Fig. 2.12 shows that very little of the day's heating requirement was delivered after about 10.00 hrs., the bulk being during the early-morning warm-up. Class temperature is seen to rise from the overnight  $14^{\circ}\text{C}$  to  $18^{\circ}\text{C}$  in about three hours of heating, by about 05.00 hrs., and so a full four hours of unnecessary heating has here been used. Class temperature is thereafter well controlled, indicating a good compensator action for this day. Fig. 2.12 does show, though, that poor management with respect to start-up time can lead to a wasteful use of energy.

The unsteadiness of the uppermost curve is due to stagnant flow upstream of the 3WV when this is shut off; here, radiator water is being circulated to cool down to the compensated temperature which is dropping faster than the natural rate of cooling of the radiators as the ambient environmental temperature rises quickly.

(v) Fig. 2.13, 19th April, 4.09  $^{\circ}\text{days}$  : a mild end-of-season day with a  $10^{\circ}\text{C}$  night ambient environmental temperature, but very warm afternoon temperatures. There is an early morning warm-up requirement only. The class is already at  $18^{\circ}\text{C}$  at the time of start-up, and, with good control, it is probable that a short burst after 07.00 hrs. would have been sufficient to warm the school for the morning. A single store discharge might have sufficed with good timing and radiator temperature control. As it is, the inadequacies of the compensator (single temperature input) and of the preheat schedule are displayed. It is possible that, with lax control on these end-of-season days, several times as much heat (albeit a small quantity) as that strictly required is passed to the radiators.

(vi) Fig. 2.14, 20th May, 2.65  $^{\circ}\text{days}$  : a very mild end-of-season Monday, with an ambient curve close to that of 19th April above. This day leads to similar conclusions as for the previous case. In particular that an input of the class temperature to the compensator would probably have reduced the day's energy use to a single store discharge and the building would not have been overheated. With suitable timing, the second boiler firing might not have been necessary. The total gas use might be halved (at least) by suitable

FIG 2.13: SPA SCHOOL : 19-04

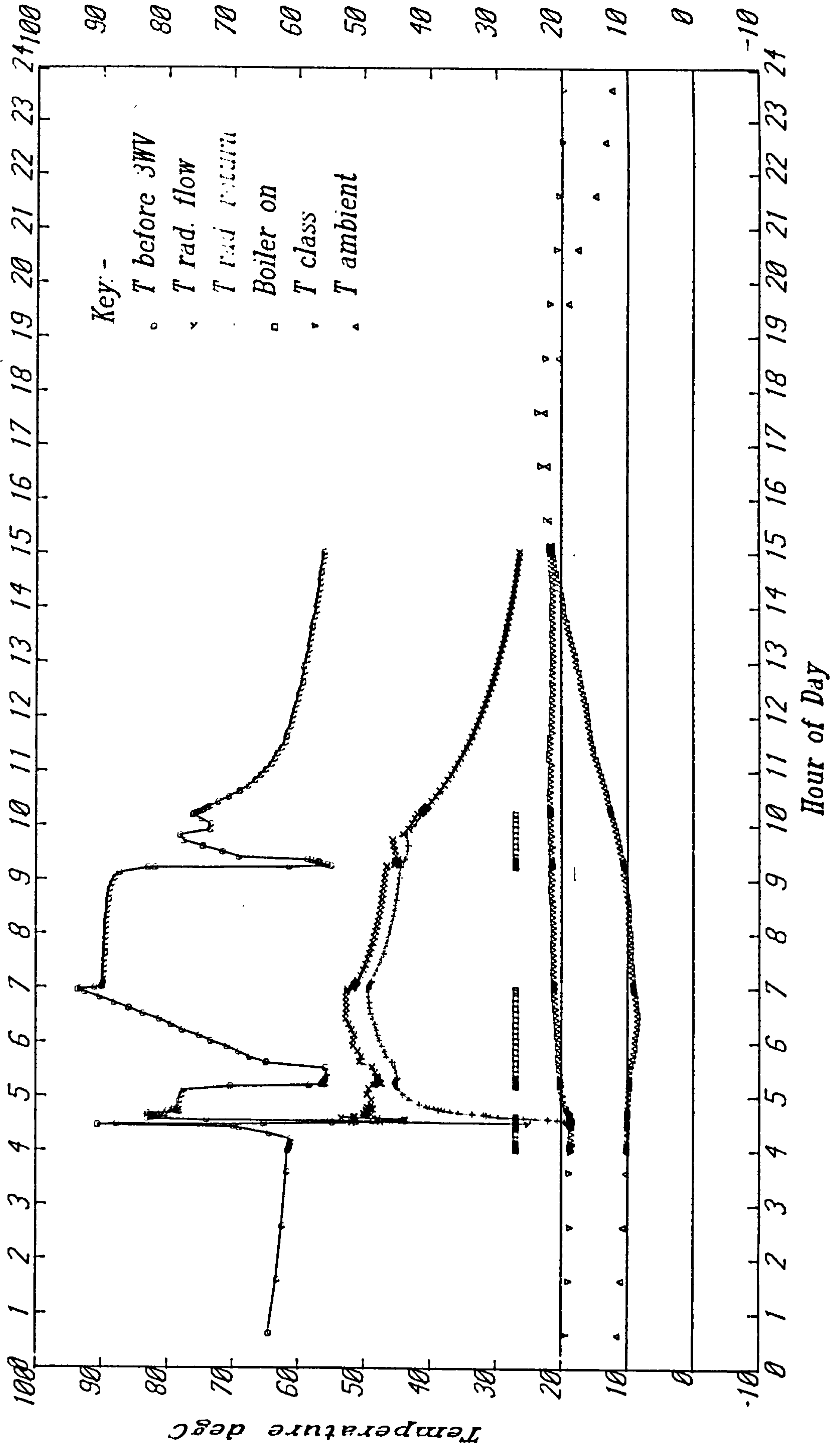
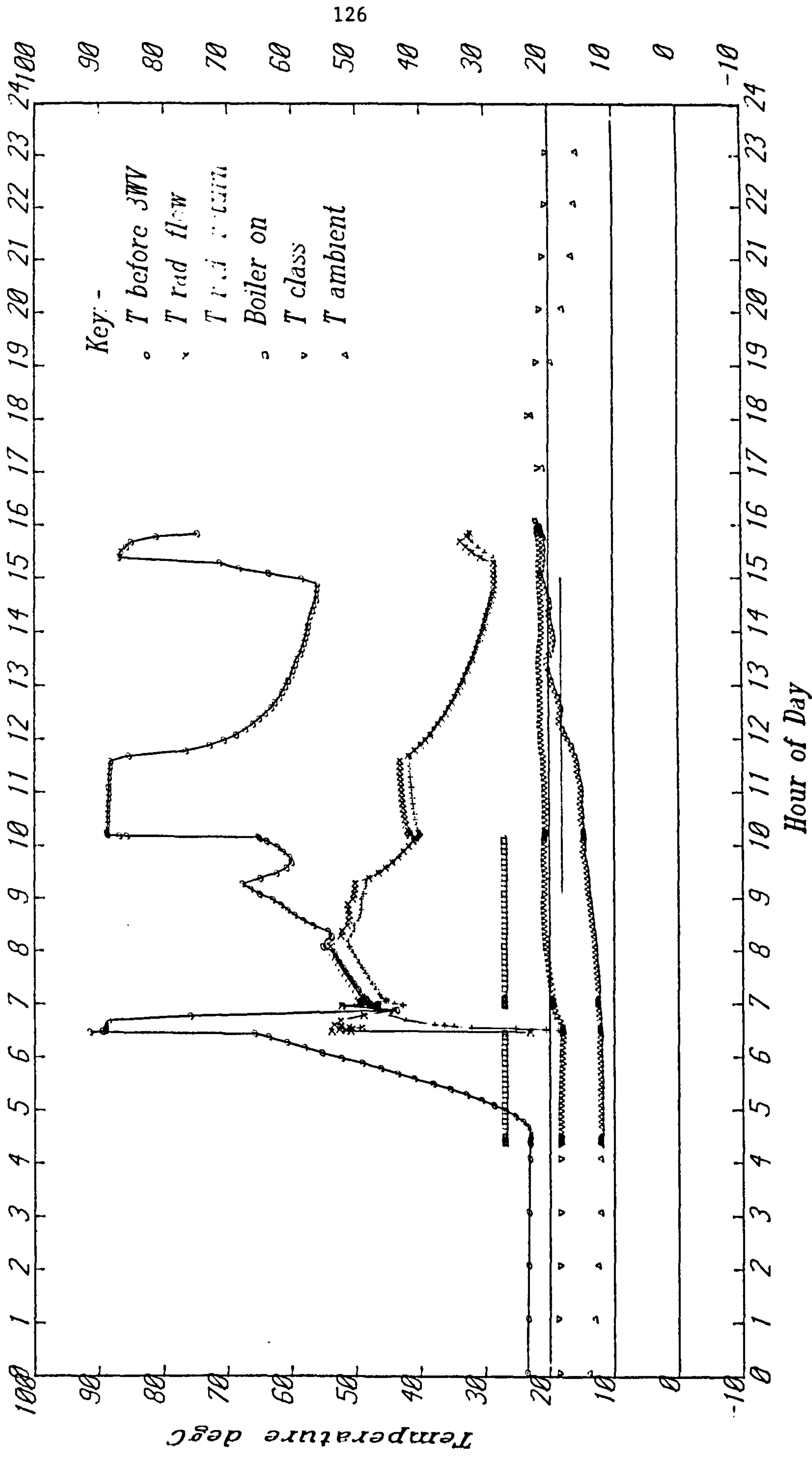
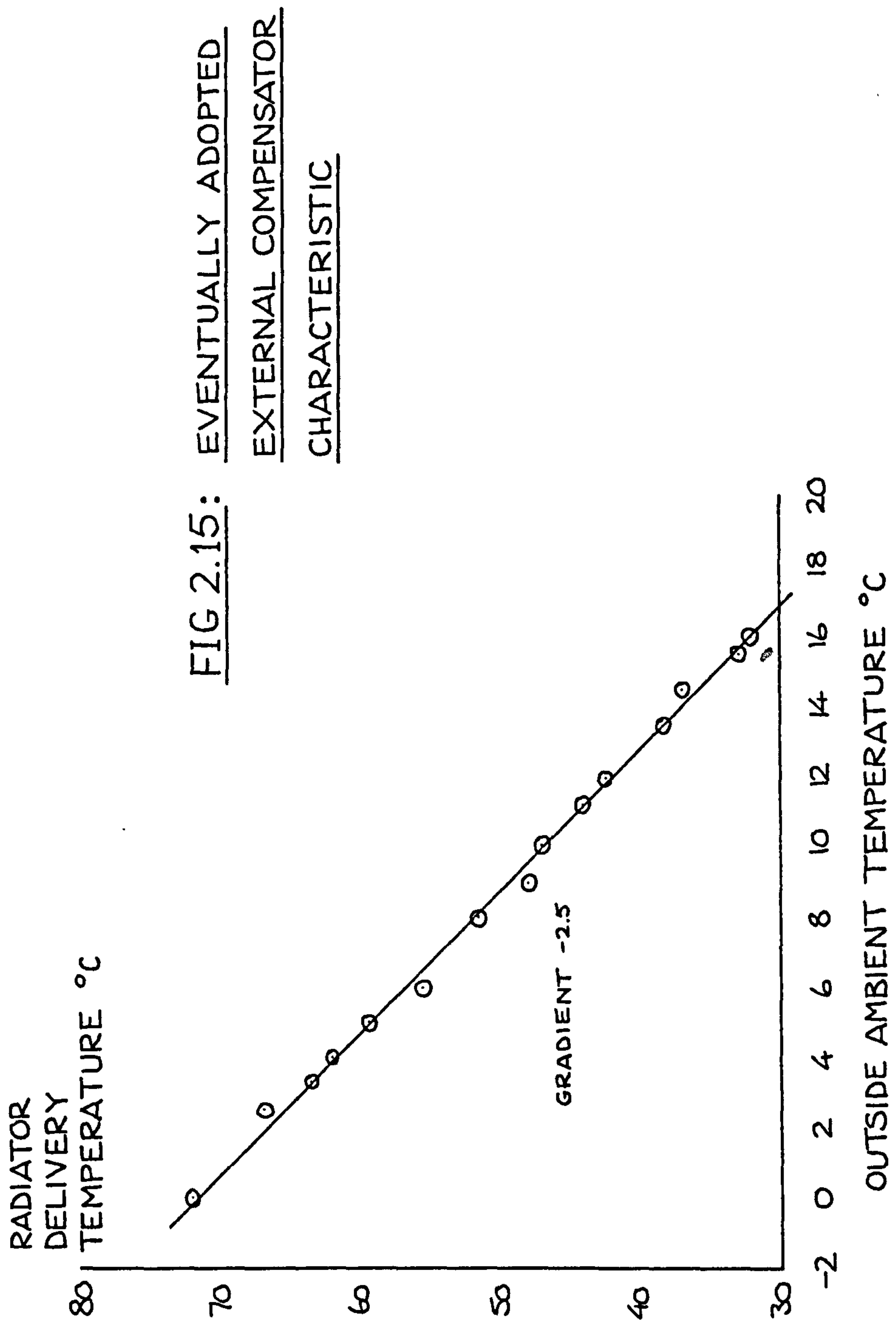


FIG 2.14: SPA SCHOOL : 20-05







controls for these mild days.

As a corollary to this graphical study, fig. 2.15 shows the compensator characteristic employed over most of the days involved. The gradient and vertical position of the line could be altered via a manual control to match the needs of the school. Values of -2.0 and -2.3 for the gradient were used at times, especially early in the season, when the reaction of the school's staff was being monitored.

## 2. Numerical Analysis

A short program, SPAEFF, was composed to analyse numerically a given day's data; this program is presented as Appx. 2.2. It evaluates:-

(a) Boiler operation in terms of total hours on, extent of heating period, and no. of cycles; load factor is defined here as being that fraction of the heating period when the boilers were on. The heating period itself was taken to be the time between the initial boiler firing in the early morning, and the radiator pump shut-down time (normally 15.00 hrs.). Other definitions of load factor are often preferred (e.g. based on gas use, 24 hr. denominator etc.), but it was thought that this definition was most appropriate to define boiler load factor in terms of the frequency map of Fig. 2.3 above.

(b) "Degree-days" were calculated to a base of the 18°C stipulated as being the minimum operable temperature in the class from 09.00 hrs to 15.00 hrs. The deficit of the outside ambient environmental temperature below 18°C was integrated over the 24 hrs of a given day; negative increments were set to zero.

(c) A day's gas usage was calculated from start and finish meter readings and converted to GJ using a factor supplied by Watson House together with a calibration correction factor.

(d) Heats passing from the boilers and to the radiators were both integrated by a simple " $\dot{m} \cdot C_p \cdot \Delta T$ "-type finite-difference technique.

(e) The discrepancy between the above heats from the boilers and to the radiators was partly accounted for by the change in store energy over the 24 hr. period. Values of net increase in store energy content were estimated (only) from temperature profile figures at 00.00 and 23.00 hrs.

(f) The remaining error in the equation

$$\text{heat from boilers} = \text{heat to radiators} + \text{store energy gain}$$

was calculated; this includes store jacket loss, pipe jacket loss etc.

(g) Three efficiencies were calculated:-

- a 24 hr. boiler efficiency,

$$\eta_B = \frac{\text{heat from boilers}}{\text{total gas used}},$$

- an "operational" boiler efficiency,

$$\eta_B' = \frac{\text{heat from boilers} + (\text{hours boiler operation} \times 241 \text{ kW})}{\text{total gas used}}$$

(241 kW was the calculated gas rate to the boilers, see (i) below and Fig. 2.16),

- a 24 hr. distribution efficiency,

$$\eta_D = \frac{\text{heat to radiators}}{\text{total gas used}} \times \text{correction factor to consider store contribution, see below}$$

These checks were applied to the sample of 26 days chosen from the 1984-5 heating season and the results appear in Table 2.2.

Certain features are evident immediately. Boiler cycles are kept to between 2 and 4 per day with this system; it would be possible to reduce the cycles by at least one for many of these days by the use of more sophisticated controls to improve storage management, for which some suggestions have been put forward in the preceding section. The store gain in heat content can be very large (>600 MJ), leaving a hot



DATE (M-MONDAY)	HRS ON	HRS HTG. PERIOD	LOAD FACTOR	B. <sup>r</sup> CYCLES	HEAT FROM BOILER GJ	HEAT TO RADS. GJ	GAS AT METER GJ	GAS (CORRECTED) GJ	STORE $\Delta U$ MJ	ERROR MJ	$\eta_B$ %	$\eta_B'$ %	$\eta_D$ %	$\eta_{DAY}$ (REL. 18°C)
FEB 7	7.59	11.04	0.689	4	5.036	4.782	6.816	6.659	96.0	155.9	75.63	76.43	73.24	12.63
8	8.92	11.00	0.811	3	5.936	5.493	8.024	7.840	38.3	402.9	75.70	76.68	70.53	14.70
13	9.86	11.03	0.894	2	6.599	6.301	8.841	8.637	48.1	249.9	76.40	77.13	73.49	20.01
14	9.13	11.01	0.830	2	6.085	5.949	8.187	7.999	-100.6	236.3	76.07	76.81	73.16	18.81
15	9.79	10.95	0.894	2	6.661	6.332	8.732	8.531	76.7	252.8	78.08	78.45	75.08	19.43
28	5.78	11.01	0.524	3	3.837	3.818	5.204	5.085	-94.8	114.5	75.47	76.58	73.27	11.27
MAR 1	7.31	10.99	0.655	3	4.924	4.172	6.587	6.436	272.8	479.8	76.51	77.66	68.62	11.50
4M	8.25	13.77	0.599	4	5.441	5.029	7.502	7.329	22.6	390.1	74.24	75.99	68.90	10.64
6	6.42	11.01	0.584	3	4.346	3.877	5.792	5.659	62.0	406.9	76.79	77.98	69.50	11.07
8	6.12	11.02	0.556	3	4.111	3.448	5.520	5.393	346.3	316.7	76.23	77.41	69.82	9.68
13	6.88	10.98	0.627	3	4.674	4.256	6.195	6.053	-1.9	419.9	77.22	78.28	70.28	12.06
14	6.29	11.02	0.571	3	4.228	4.117	5.683	5.553	-137.5	247.6	76.14	77.46	71.82	12.39
15	7.56	11.02	0.686	3	5.099	4.489	6.827	6.670	131.9	478.9	76.46	77.76	69.09	13.46
20	8.78	11.01	0.798	3	6.026	5.285	7.926	7.744	92.5	649.3	77.82	79.12	69.30	15.67
22	5.90	10.99	0.537	3	3.893	3.646	5.389	5.265	-63.5	310.1	73.94	76.07	68.14	10.59
25M	8.05	13.76	0.585	3	5.409	4.944	7.273	7.106	7.0	458.3	76.13	77.46	69.67	11.54
28	7.16	10.99	0.652	3	4.813	4.187	6.478	6.329	123.4	502.7	76.05	77.47	67.90	12.35
APR 1M	4.22	13.76	0.307	3	2.775	2.144	3.887	3.797	322.1	308.8	73.07	75.79	63.87	4.71
19	3.35	11.01	0.304	3	2.054	1.826	3.016	2.947	44.9	182.8	69.71	70.67	63.37	4.09
22M	6.22	13.76	0.452	4	3.912	3.635	5.574	5.446	-48.1	325.2	71.82	72.53	65.93	7.81
24	6.09	11.00	0.554	3	3.940	3.286	5.422	5.297	317.9	335.6	74.37	74.55	67.48	8.71
MAY 2	4.86	11.00	0.442	4	3.125	2.924	4.464	4.361	16.0	185.0	71.64	74.10	67.38	7.62
6M	5.27	13.76	0.383	3	3.420	3.177	4.801	4.691	74.2	168.6	72.91	74.76	69.24	5.44
15	2.69	9.09	0.296	2	1.708	2.013	2.471	2.415	-351.6	46.8	70.73	73.26	69.12	4.48
16	2.94	8.58	0.342	2	1.882	1.132	2.689	2.627	683.8	66.8	71.65	73.92	67.66	3.37
20M	5.24	11.49	0.456	2	2.321	1.904	3.266	3.191	-160.5	160.5	72.73	N/A	67.08	2.65

TABLE 2.2: "SPAEEFF" RESULTS FOR THE 26 INDIVIDUAL DAYS ANALYSED

store to stand overnight with the attendant jacket loss. The store ought ideally to be left depleted overnight by suitable controls. The "error" is seen to be variable and sometimes very large. Including jacket and pipe loss, it is bound to vary due to the variability of these constituents; a further explanation is taken to be that there was some doubt during early 1985 as to the accuracy of boiler water flow and radiator temperature drop measurements (prompted by this very problem), see below. The elimination of pilot gas use in the calculation of  $\eta_B'$  renders this boiler efficiency slightly higher than  $\eta_B$  for colder days, but the increase is more marked on mild days. Distribution efficiency,  $\eta_D$ , is alarmingly lower than the boiler efficiencies. Degree day values extend from 20 down to 2.7. Load factor is encouragingly high, mostly above 0.5, falling below 0.4 only on mild days where degree days fall below about 6.

The large "error" values obtained called into question the accuracy of the measurements used to calculate the boiler and radiator heat flows. These were, in both cases, an electromagnetic flowmeter, and a pair of P.R.T.'s for temperature differential. The boiler flowmeter was strongly suspected of being inadequate, and this is borne out by the fact that the high load factor days of Table 2.2 indicate a value of  $\eta_B$  in excess of the manufacturer's quoted bench full-load efficiency (~ 78% against 74.5%). Thus it is altogether probable that the values measured for "heat from boiler" in Table 2.2 are a few percentage points too high. With regard to the heat to radiator values, the P.R.T. readings were subsequently found to have drifted very slightly, though P.R.T.'s were generally found to be very reliable. A slight drift of one P.R.T., however, is very damaging in a temperature differential reading of very few degrees, especially for the milder days where the differential was often of the order of 1°C. Thus there exist inaccuracies in the as-received data used for this study; no attempt has been made to correct these inaccuracies here (by the use of correction factors and careful energy auditing), as the study presented here is a preliminary numerical examination of the system's performance. These inaccuracies have, however, been resolved for the monitoring of the current season, and the results from this far more reliable data will supercede the results obtained in this study. In the graphical presentation of the numerical results of

Table 2.2 which follows, therefore, the graphs should be taken to illustrate trends rather than yield absolute figures for e.g. efficiency.

Clearly, a great many more parameters than those presented in Table 2.2 could have been calculated to characterise a day's performance; likewise the number of graphical configurations that could be drawn even from the contents of Table 2.2, is very great. The following are presented as a sample, exploratory series of graphs only; the analysis here is thus by no means exhaustive. (The author believes, for example, that with the inaccuracies of the data here, it would be ill-advised to try to proceed to the evaluation of the crucial value for seasonal, or half-seasonal, efficiency.)

(i) Gas Consumption : in Figs. 2.16 to 2.18, the days' total gas usages have been plotted against the hours of boiler duty, load factor and number of degree days.

Fig. 2.16 was plotted to verify that boiler power input remained constant. The clear straight line shows that boiler gas rate remained effectively invariant over the several months covered by this study. Gas consumption is composed of boilers-on gas use plus pilot use. From the gradient and position of the straight line of Fig. 2.16, boiler gas input was estimated to be equivalent to 241 kW.

Fig. 2.17 shows gas consumption against a base of load factor. For weekdays other than Mondays where the heating period was left untouched at about 11 hours (clock setting), a good straight line is produced, as would be expected. Fig. 2.17 shows the increased gas usage of Mondays in the period defined by the days lying on the weekday straight line; these Monday points lie some 26% above the previous line. At the end of the season a start-up delay optimiser was tried, reducing the length of the heating period, and the rather arbitrary definition of load factor is inadequate to cater for these days; they fall below the above straight line.

Fig. 2.18 is a rather more important plot, giving gas consumption to a base of degree-days. It must be borne in mind that the



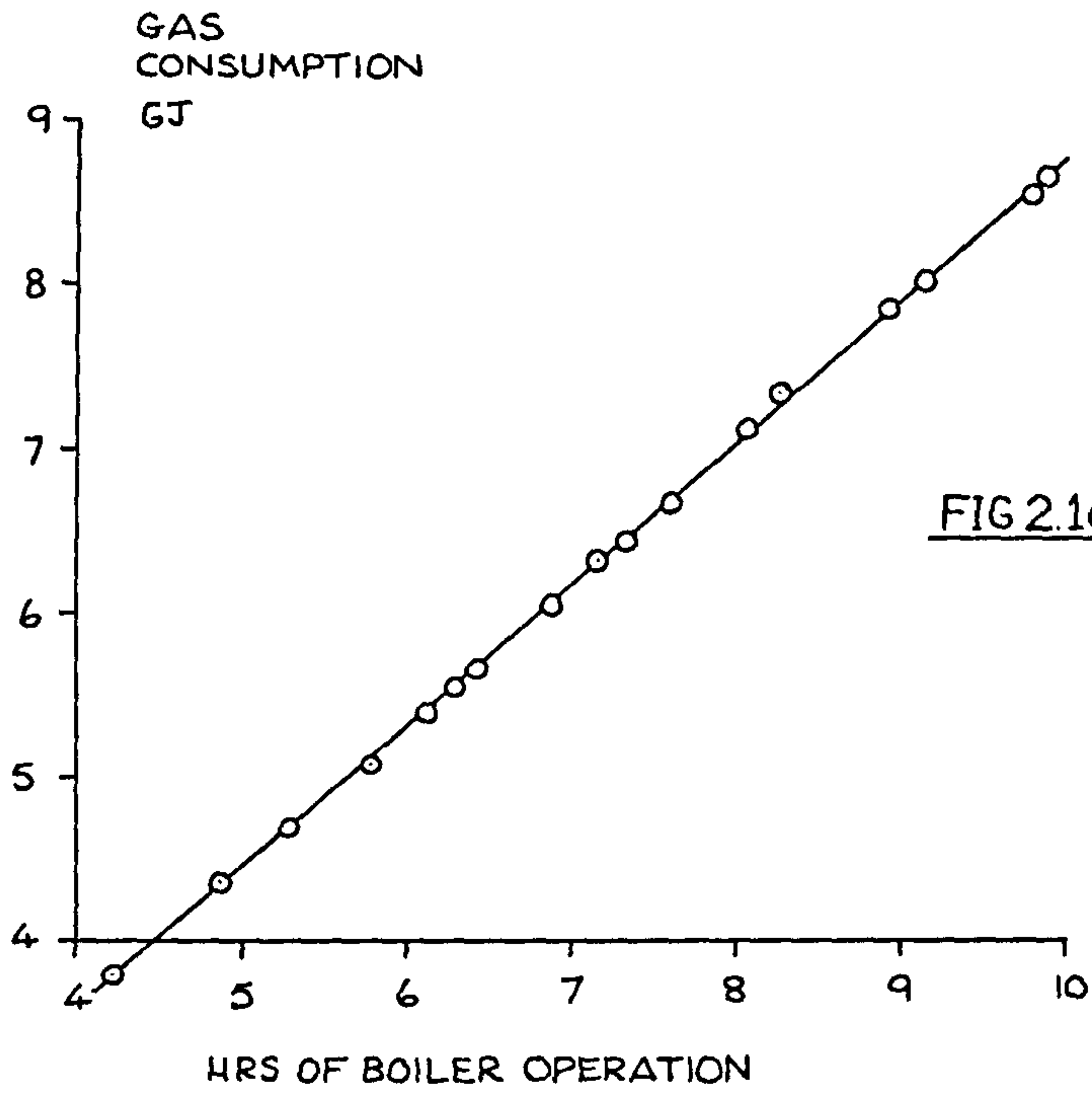


FIG 2.16: CONSTANT GAS RATE TO BOILERS

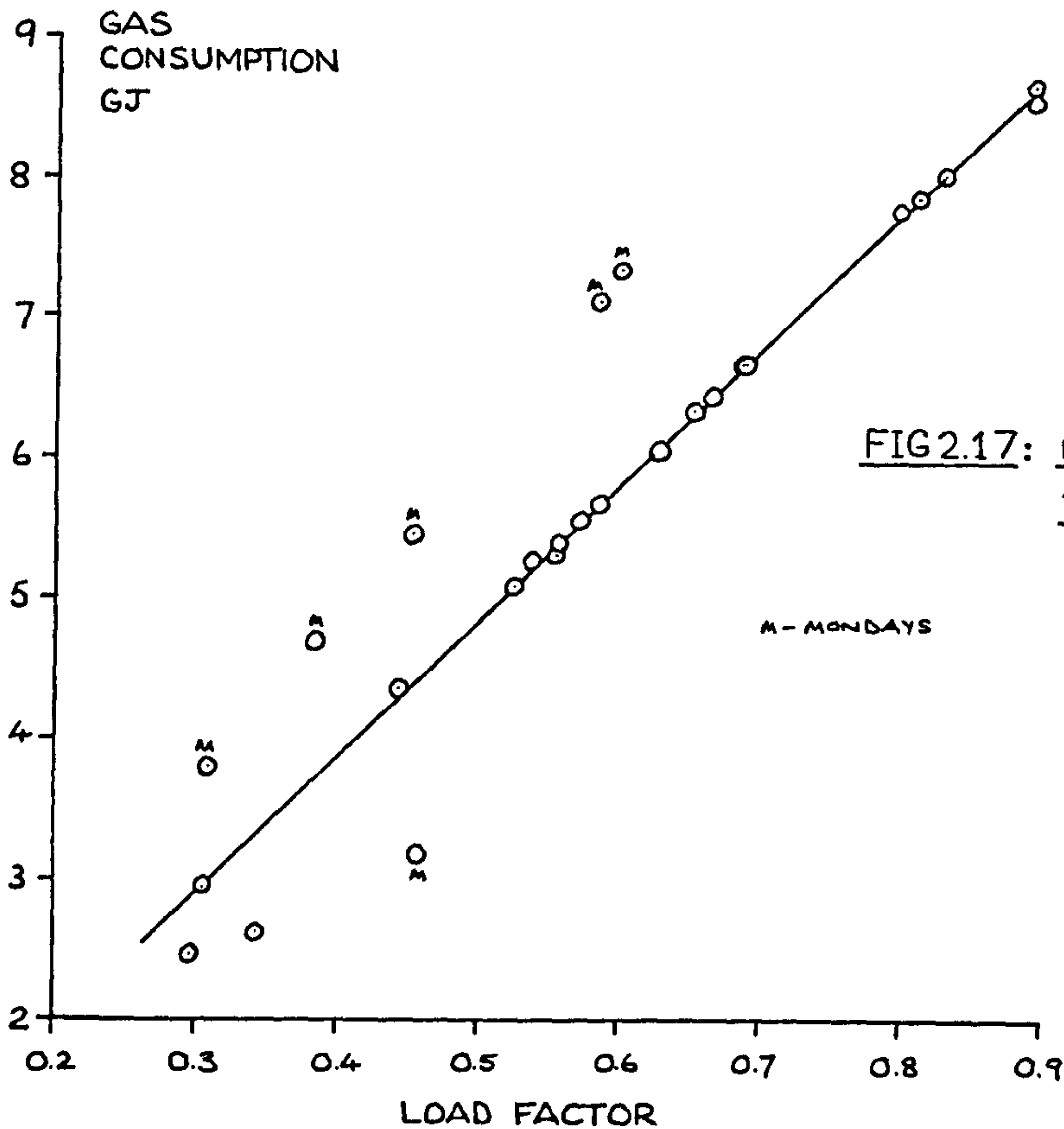
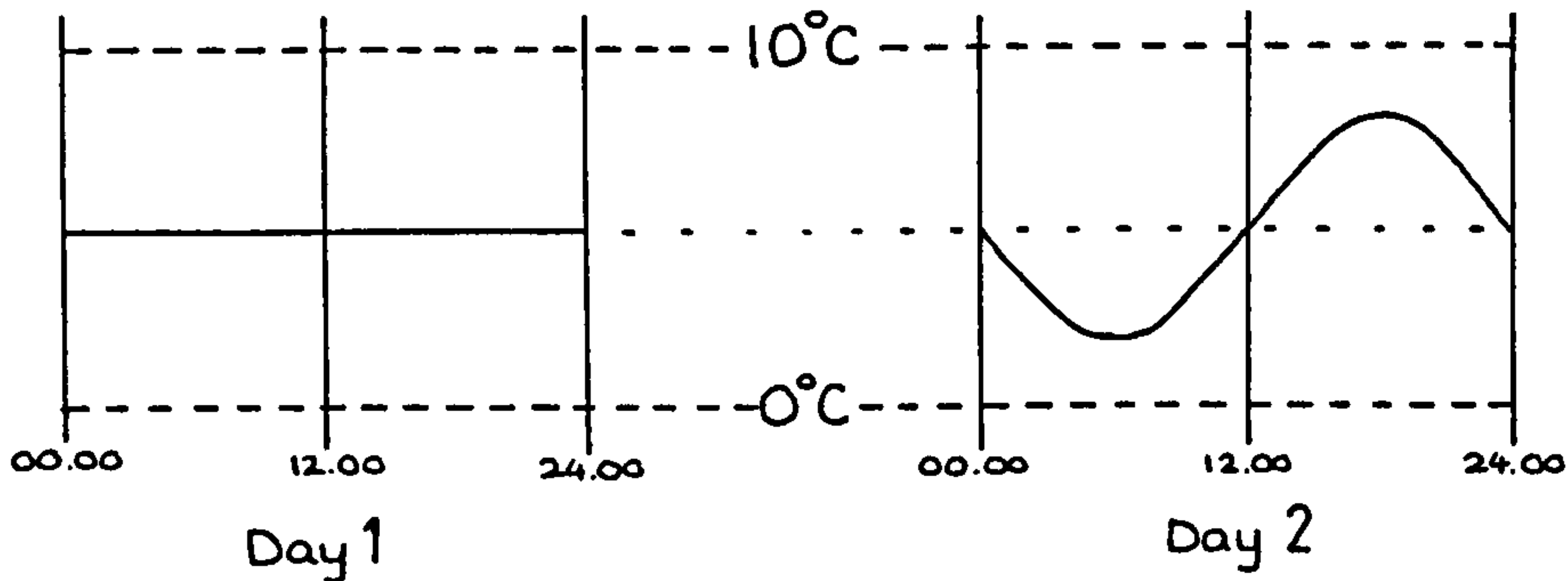


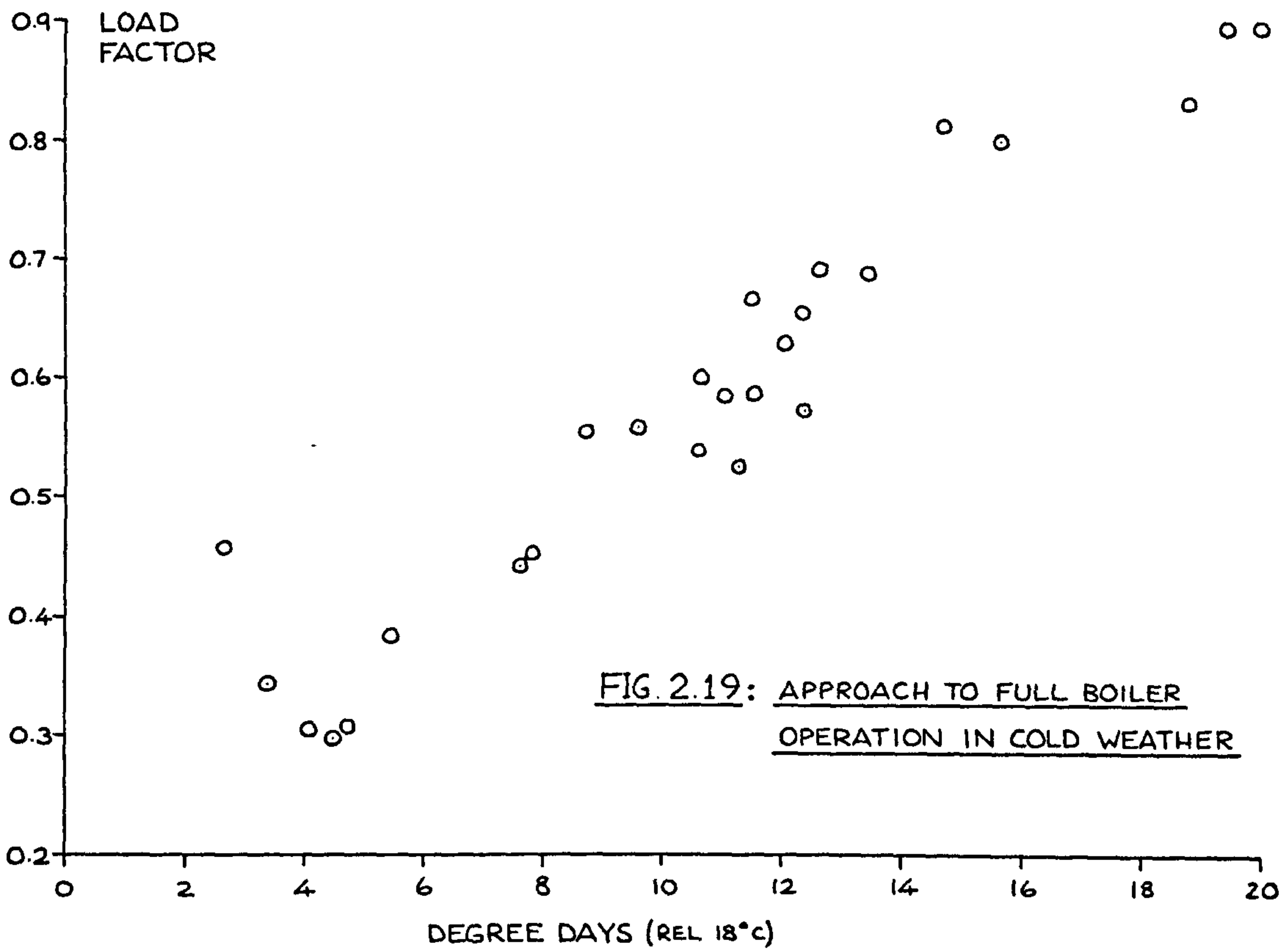
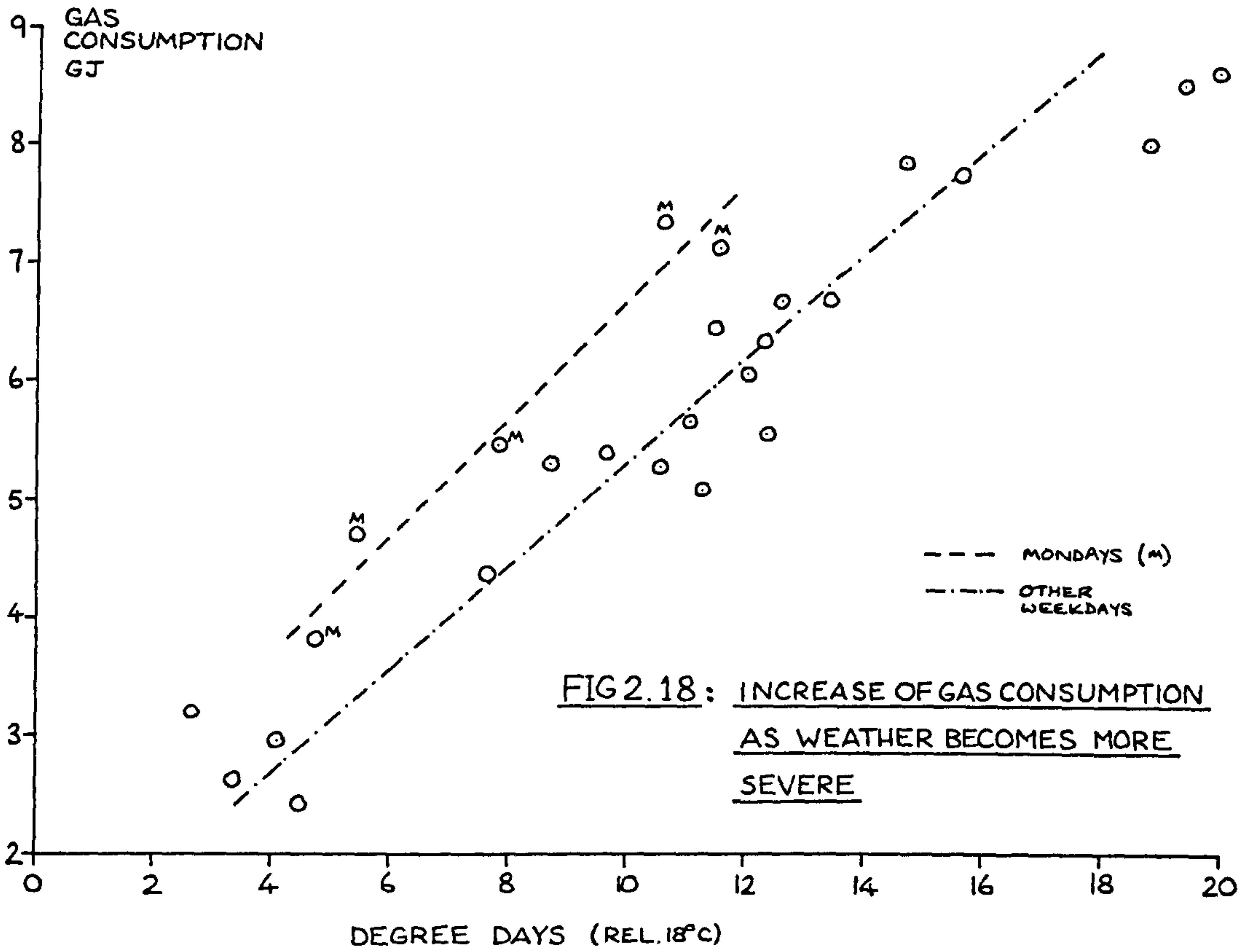
FIG 2.17: EXPANDED FORM OF FIG 2.16 TO BASE OF LOAD FACTOR

degree-day figure for a given day is only an indication of the severity of that day, rather than a strict measure of severity. Thus if two days with the same degree-day figure have the following ambient temperature profiles:-



Day 2 will be more severe than Day 1, as most of its temperature deficit coincides with the most arduous part of the system's heating period. Thus, with regard to Fig. 2.18, two such days would legitimately have differing gas requirements, and a close approximation to a curve or straight line will not result. However, the days' points do lie roughly distributed about two lines, one for Mondays, and the other for other weekdays. A more precise, time-weighted degree-day parameter might well be worth seeking. Fig. 2.18 suggests that, at very high values of degree days, the gas consumption falls away from the approximate straight line fit, probably because the compensated temperature demand was in excess of that which the system could deliver for these very severe days. Though the compensated temperature may be too high for the boilers/store during several hours of a day, the school might still be heated adequately. It is difficult to identify a degree-day limit beyond which the school's 09.00 to 16.00 hrs. minimum requirement would not be met.

(ii) Load Factor : an attempt was made to isolate a parameter on which load factor was dependent, and in choosing degree days on fig. 2.19 a rather artificial plot has been produced, due to the arbitrary nature of the definition of load factor. Fig. 2.19 adds very little





to the information of Fig. 2.18.

(iii) Efficiencies : thus far, the graphs have not examined the quality of operation of the storage system; in Figs. 2.20 to 2.24, the three efficiencies calculated have been plotted against certain parameters to try to investigate the effectiveness of the boilers and of the distribution system.

Figs. 2.20 and 2.21 show the boiler efficiencies against load factor; these are central to an assessment of boiler performance. Fig. 2.20 shows that the 24 hr. efficiency (i.e. the ratio of heat from boiler/total gas used) falls off along a curve similar to the customary  $\eta/LF$  curves published for boilers (though these refer to frequent switching operation, i.e. the asymptotic curves of Fig. 2.3). The points seem to lie scattered about a curve which falls from about 77% at full-load operation down to some 71% at 30% load factor. A proportion of this decline will be due to unavoidable pilot loss and the removal of this (by equating the denominator of the efficiency to "HRS ON" x 241 kW, see section 2(g) above) generated the values of  $\eta_B'$  used for Fig. 2.21. This latter is seen to be more favourable as  $\eta_B'$  falls from about 77.5% at full load down to some 73% at 30% load (very approximately). It is the performance of fig. 2.21 which is to be compared with the theoretical curves of Fig. 2.3. The 77.5%/73% decline of Fig. 2.21, gives a value of "% of  $\eta_{FL}$ " of ~ 94% at 30% load factor; Fig. 2.3 therefore confirms that the UAFLUE value crucial to the boiler model used to date is likely to be nearer 116 W/°C than the 29 W/°C quoted for the steady-state heat loss to the flue. Indeed, if the boilers are supposed to operate between the 1 switch/4 hours and 1 switch/2 hours curves of Fig. 2.3, UAFLUE may be in excess of 116 W/°C. (Table 2.2 indicates that the number of switchings is 2, 3 or 4, and that the typical duty period is some 11 hours). However, the superimposition of an approximate curve from Fig. 2.21 onto the arrays of curves of Fig. 2.3 is a very inaccurate method of estimating a value for UAFLUE because, at this very low frequency, efficiency is little dependent on UAFLUE. Conversely, it is plain that the storage concept is here validated, as switch frequency is rendered so low that the value of UAFLUE of the boiler is of little consequence. More important, at low load factor, is that the switch frequency should be

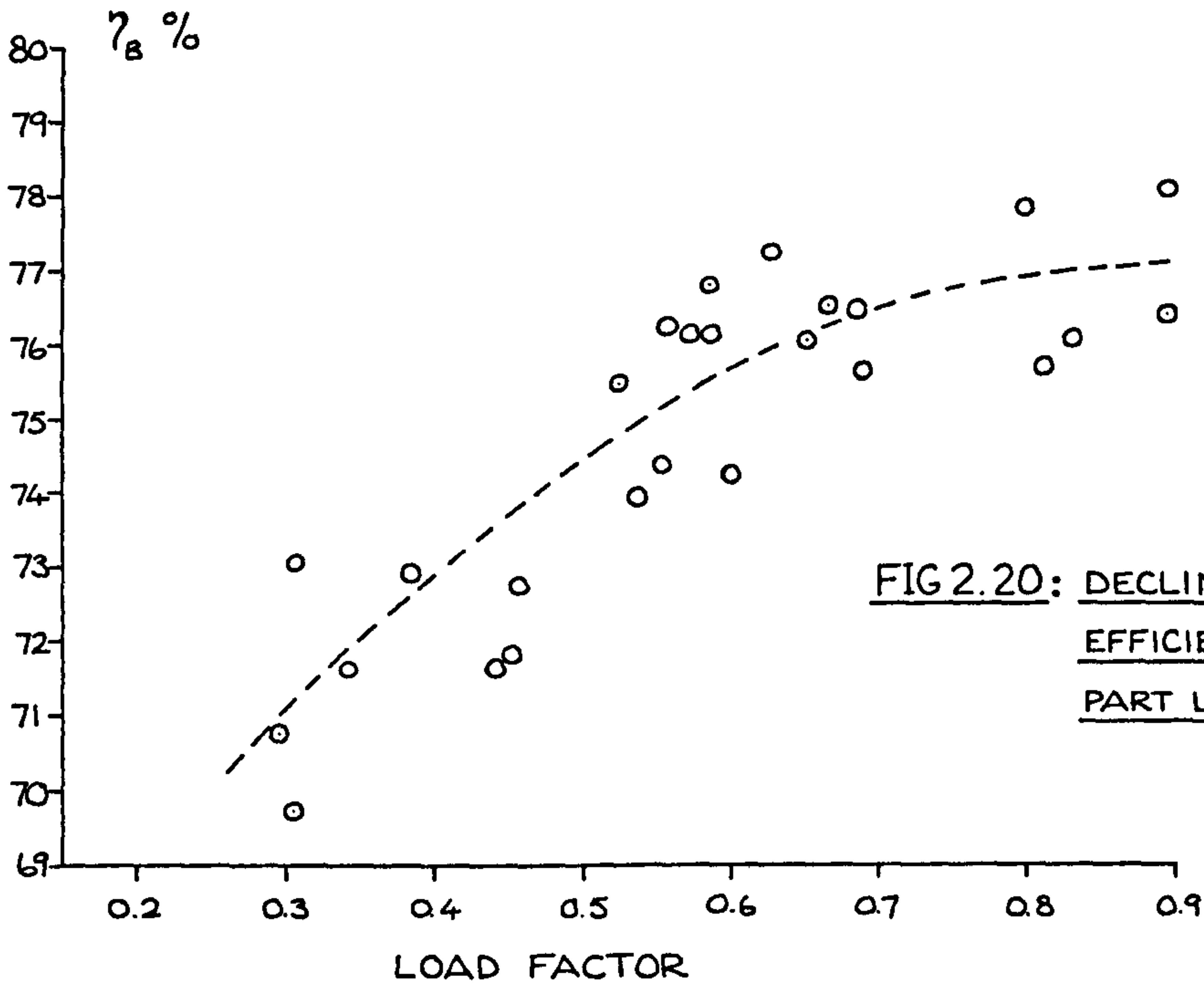


FIG 2.20: DECLINE OF 24HR  
EFFICIENCY AT  
PART LOAD

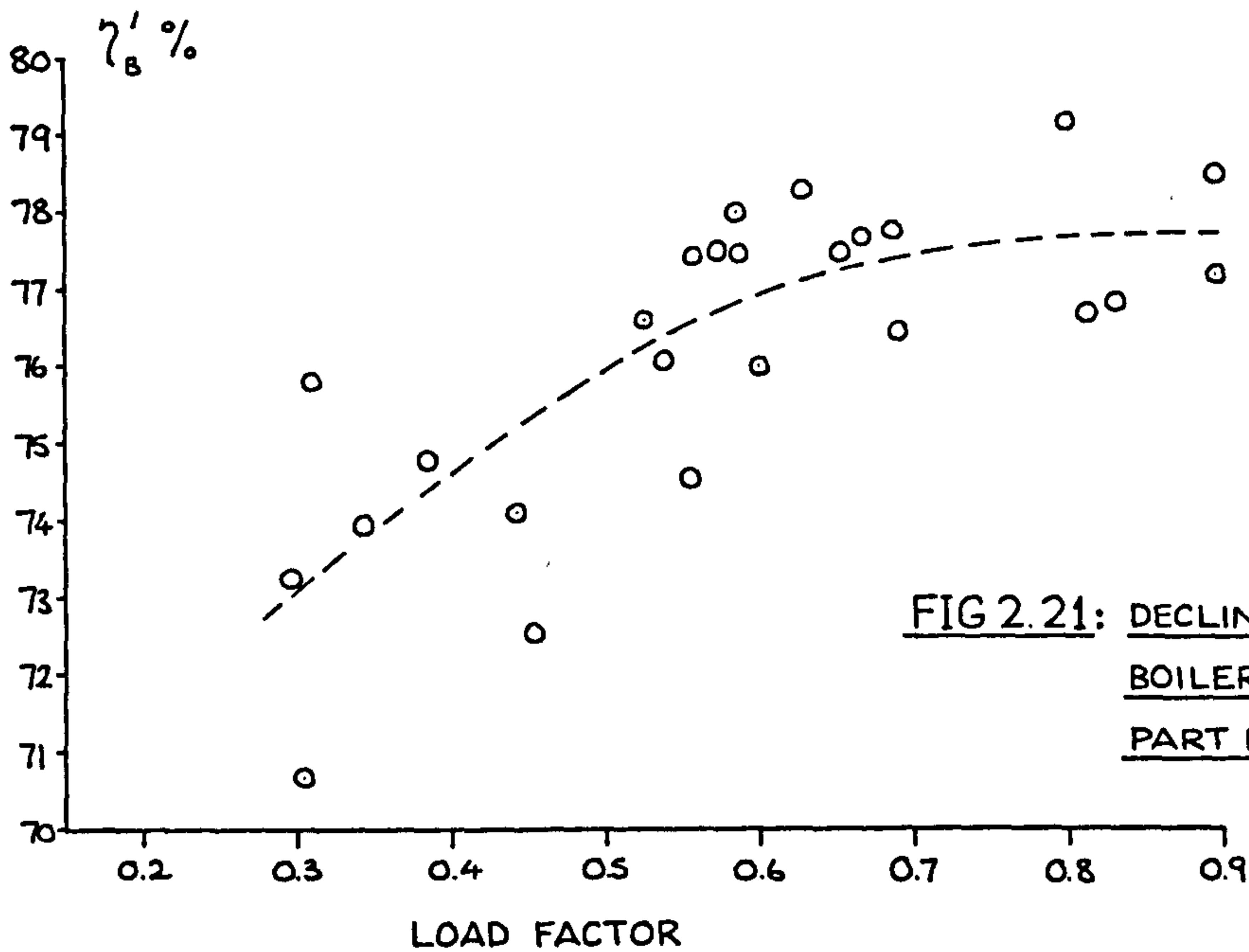


FIG 2.21: DECLINE OF "OPERATIONAL"  
BOILER EFFICIENCY AT  
PART LOAD

reduced to 1 - 3 per day if possible - achievable by good controls, as outlined above.

The adoption of  $116 \text{ W/}^\circ\text{C}$  for UAFLUE, a crucial assumption, used for the continuation of the work at Cranfield, appears to be justified.

It is possible that the increased scatter of Fig. 2.21 compared with Fig. 2.20 is due to the use of a constant 241 kW as boiler gas input in the denominator of  $\eta_B'$ . Though this ought to remain fairly constant as indicated by Fig. 2.16 above, a slight day-to-day drift is not unlikely, and the values of  $\eta_B'$  for the mildest days would be most in error - hence perhaps the increased scatter at the bottom end of Fig. 2.21.

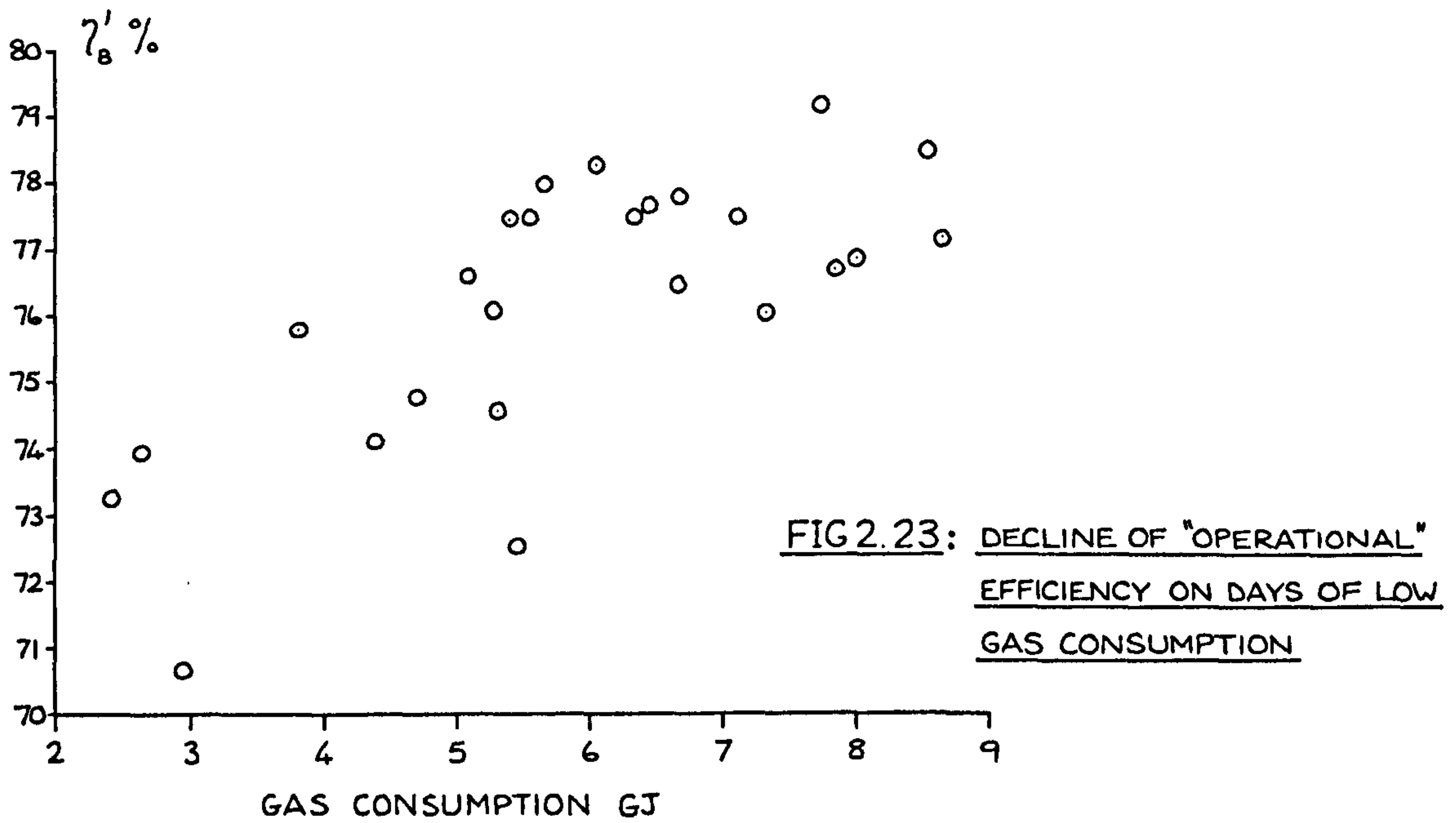
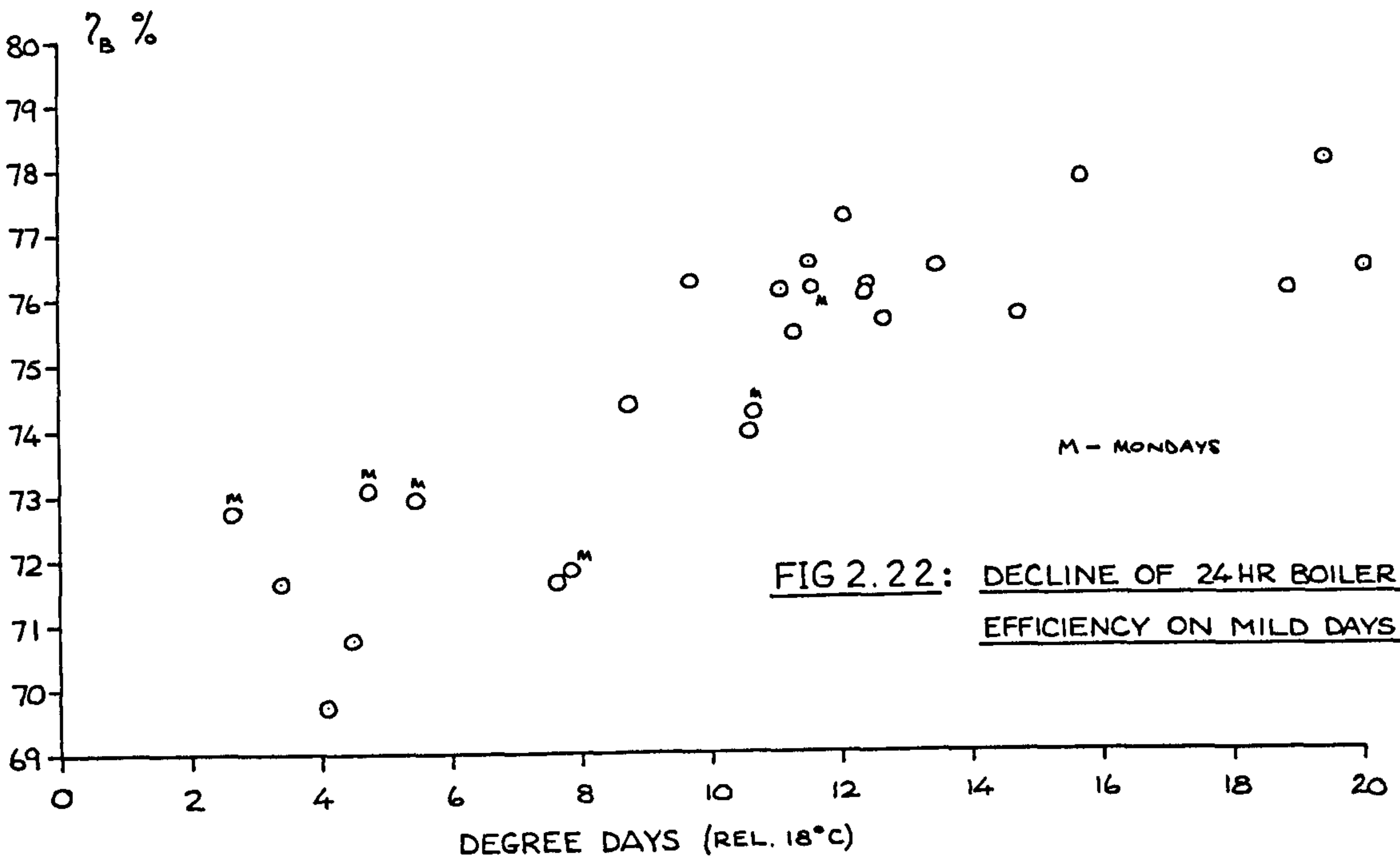
Due to the unreliability of the boiler flow rate data, as discussed above, no value of  $\eta_{FL}$  has been presented here to convert Figs. 2.20 and 2.21 into the form of Fig. 2.3. Assuming, however, that the boiler-flow error was consistent, the trends of these and the following efficiency plots remain valid.

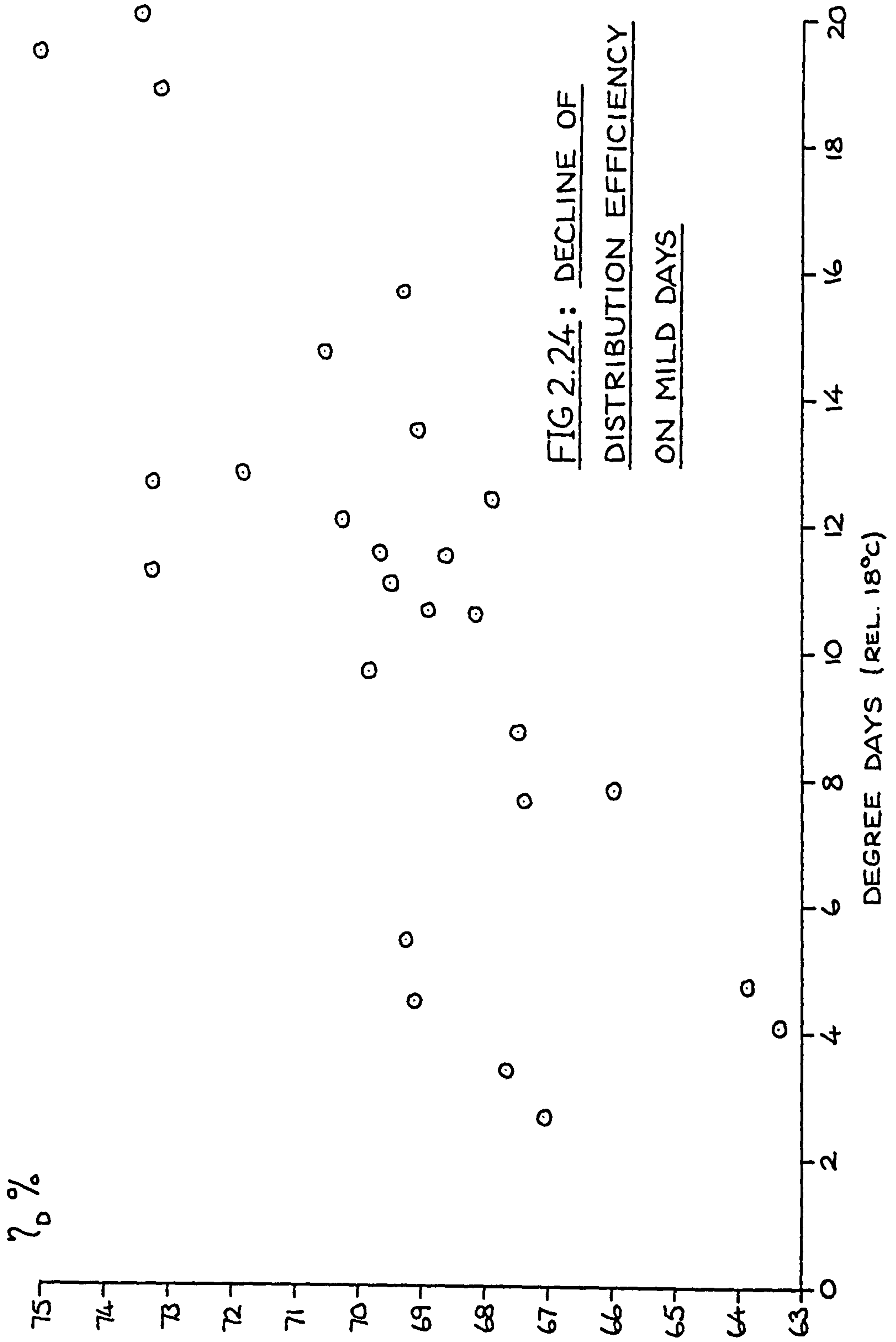
Fig. 2.22 shows an attempt to relate  $\eta_B$  to the severity of the respective days. Because, again, the degree-day value is an approximate indicator of severity, the scatter of the graph is considerable. The points indicate a curved decline rather than a straight-line relationship. The Mondays of the study seem to be scattered with the other points except for mild days, where their increased boiler efficiency is probably due to the generous gas usage of these days, which better controls would reduce.

Fig. 2.23 is a sequel to Fig. 2.21, and an attempt to relate  $\eta_B'$  to gas consumption rather than load factor. There is seen to be a poorer-defined decline.

Fig. 2.24 relates  $\eta_D$ , the distribution efficiency, to the number of degree days. With a storage system,  $\eta_D$  can be defined in a variety of ways. The simple definition, heat to rads./total gas input, is going to be unfairly low if a quantity of the boiler heat passes into







the store, e.g. see the data for 16th May 1985 on Table 2.2. Thus the following definition was adopted:-

$$\eta_D = \frac{\text{heat to rads.}}{\text{total gas input}} \times \frac{\text{heat from boilers}}{(\text{heat from boilers} + \text{net store contribution})}$$

The customary decline as shown on Fig. 2.24; the scatter of the points is considerable, probably for two reasons:-

- the degree-day base will always produce scatter,
- a more representative definition of  $\eta_D$  is probably necessary and,
- the energy audit has not been satisfactorily resolved here, and the above inaccuracies will contribute to give a false value of  $\eta_D$ .

The low values obtained at the end-of-season indicate that further investigation of this parameter is necessary; the storage principle ought to yield good values of  $\eta_D$  at low load factor if the store and pipe jacket loss is not excessive. (Pipe loss is likely to have been disproportionately large in this instance as the "retrofit" design necessitated considerable lengths of large-diameter piping.)

## CONCLUSIONS

1. During the early 1985 half heating-season, the data collected from Spa School were adequate to allow for a graphical, qualitative analysis to be set up together with a preliminary numerical analysis routine. The known inaccuracies of the data, however, prevented the numerical analysis to be extended such that significant, quantitative conclusions could be drawn (seasonal efficiency, boiler efficiency as percentage of full-load efficiency, etc.). The numerical analysis does permit parameter trends to be investigated.



From the graphical analysis it has been suggested that the following control modifications might lead to a more effective use of the storage principle:-

2. A motorised valve, placed so as to isolate the store if necessary once it has been depleted, would greatly reduce the "lugging" time required to re-establish the compensated temperature after the first store discharge on cold days.
3. A series of class temperature sensors, set up in several rooms in the school, from "warmest" to "coolest" would give a more representative measurement of building temperature.
4. The incorporation of this representative classroom temperature, together with outside ambient temperature, as input to a suitable controller to replace the current external compensator, would provide an opportunity for closer control of room temperature. Thus the stipulated classroom temperature could be exceeded by a minimum amount, and overheating avoided.
5. A fixed-time start to the heating period is often wasteful, and can lead to several hours of superfluous heating. Large energy savings would be realised by the incorporation of a suitable optimiser, ideally calculating start-up time as a function of ambient and building temperatures (see 3. above) and store heat content.
6. It may be worthwhile over-riding the compensated temperature during early-morning warm-up in certain instances, to allow for an improved use of the store's heat content during first discharge.

From the numerical analysis for the 26 individual days' data considered, the following conclusions have been drawn:-

7. The boilers cycled between 2 and 4 times per day (three boilers). It is believed that suitable controls could further reduce this by one or two cycles in many instances.
8. Load factor has been defined as the fraction of the daily period

of heating system operation during which the boilers operate. On this basis, no day of the 17 investigated from February and March fell below a load factor of 0.5. Of the remaining days, the minimum load factor recorded was about 0.3.

9. With the aid of a theoretical efficiency/load factor plot, the assumption of a heat-transfer to flue figure ("UAFLUE") of about  $116 \text{ W/}^\circ\text{C}$ , used in subsequent computer simulation at Cranfield, appears to be justified. The storage technique, however, in reducing boiler cycles as in 7. above, and load factor being so often above 0.5, has (i) rendered the boiler efficiency far less sensitive to "UAFLUE" value, and hence made it possible for computer simulations to be more trustworthy; and (ii) lifted the boiler efficiency considerably above the customary boiler part-load curve, especially at low load factors. Reliable efficiency figures will be released after analysis of the current (1985-6) heating season, during which more accurate data are being obtained.

The following overall conclusions have been drawn:-

10. In attempting to estimate the inherent advantage of a storage system over a conventional system, perhaps 5 - 15% energy savings (ref. 2.2) it is vital to have access to reliable measurements of flow rates. The calibrations of the electromagnetic flowmeters used here tended to drift slightly; the analysis stands or falls on the accuracy of these readings. Further, when measuring small temperature differences, such as those across the radiator system at mild ambient temperatures (of the order of  $1^\circ\text{C}$ ), the accuracy even of the P.R.T.'s must be verified. For the radiator and boiler temperature differentials, it is worth doubling the number of P.R.T.'s at each station.

11. The boiler/store combination is thought to be well sized for the Spa School. A store isolating motorised valve ( see conc. 2 above) has been fitted for the current season.

CURRENT/FUTURE WORK

As above, the current season (1985/6) is being carefully monitored, the measurement inaccuracies of 1984-5 having been resolved. Robert Cohen of the Cranfield Applied Energy Unit is in the process of analysing the data, and will publish figures for efficiency etc. later in 1986. He will also compare the storage system results with the results from the monitoring of the conventional system (1982 and 1984) to obtain an estimate of the inherent benefit of the storage system.



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**PART III**

**AN EXPERIMENTAL INVESTIGATION**

**OF A DOMESTIC BOILER/STORE**

**COMBINATION WITH TWO INTEGRAL**

**HOT WATER HEAT EXCHANGERS**

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ABBREVIATIONS

For conciseness in the text, the following abbreviations have often been employed:-

h.e.	heat exchanger
c.h.	central heating
d.h.w.	domestic hot water
3WV	three-way, or mixing, valve
P.R.T.	platinum resistance thermometer
t/c	thermocouple
D.C./A.C.	direct/alternating current

The following symbols appear in the text or on the graphs:-

ℓ	litre
m	metre
cm	centimetre
"	inch
kWh	kilowatt-hour
°C	degrees Celsius
ΔT	temperature differential
min	minute
mV	microvolts
p.s.i.	pounds per square inch

## INTRODUCTION

The early prototype domestic boiler/heat store unit was described in part I of this thesis: it was manufactured by Gledhill and tested briefly at Cranfield during late 1982 (ref. 3.1). It contained the cumbersome d.h.w. heat exchanger of photo 1.1. This comprised two double-helical coils in series, positioned on the same vertical axis central to the tank, and spanning almost the entire water depth. A single 3WV moderated its output to the desired tap temperature. To maintain a d.h.w. temperature rise of say  $40^{\circ}\text{C}$  at some 9 /min, the surface area of the copper tubing of this h.e. had to be so large that its weight and cost (and indeed its internal volume) were unsatisfactorily high. The diameter of the tubing had to be large (2.54 cm) so that the considerable length required for adequate heat exchange still allowed for mains pressure to generate sufficient water flow.

The British Gas Corporation (Watson House) therefore redesigned the h.e. using a finned copper tube marketed under the name "Integron". This material has a heat-transfer coefficient per unit length, and per unit weight, several times higher than that of plain copper tubing, and was the subject of a study conducted in this context by T. Miné at Cranfield in summer 1984 (ref. 3.2). The redesigned heat exchanger was in the form of two interconnected coils, supported horizontally near the top and bottom of the tank. The same water flow passed through both coils. Fitted with the same circulator-boiler and 3WV configuration as its predecessor, this unit is at present being marketed under the trade name "Cormorant", see Fig. I.7. Available in a series of tank capacities, it has been installed successfully in several hundred domestic properties; its performance is under continuous appraisal at Watson House (refs. 3.3 and 3.4).

It has been noted earlier that the current practice in the U.K. tends to reduce c.h. system costs by installing as little radiator area as possible, in conjunction with a high operating temperature. This practice reduces the potential benefit of a storage system as conceived for this study, where the use of a mixing valve as in fig.



FIG. 3.1:

CUSTOMARY RADIATOR  
CIRCUIT USING SINGLE 3WV

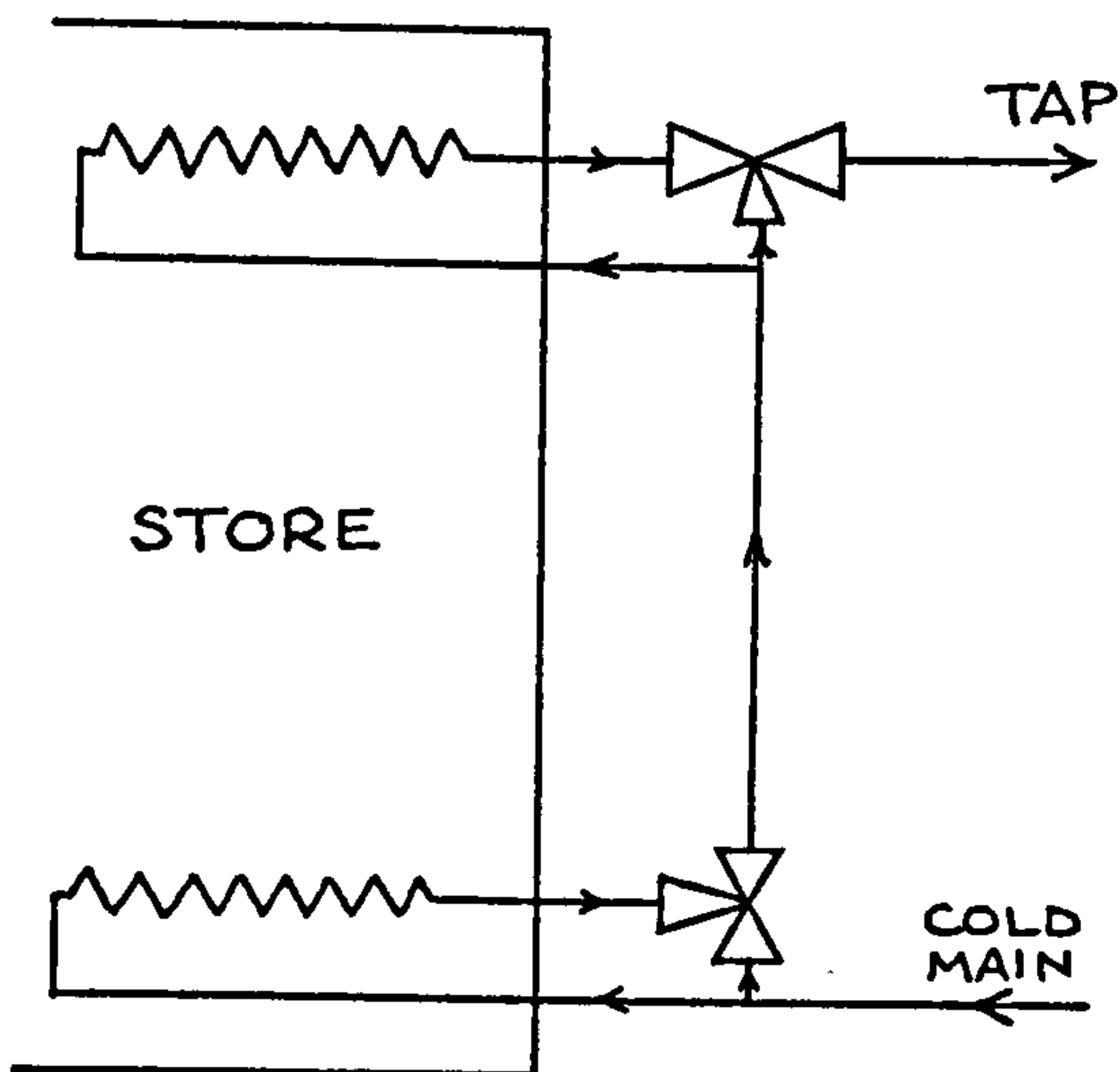
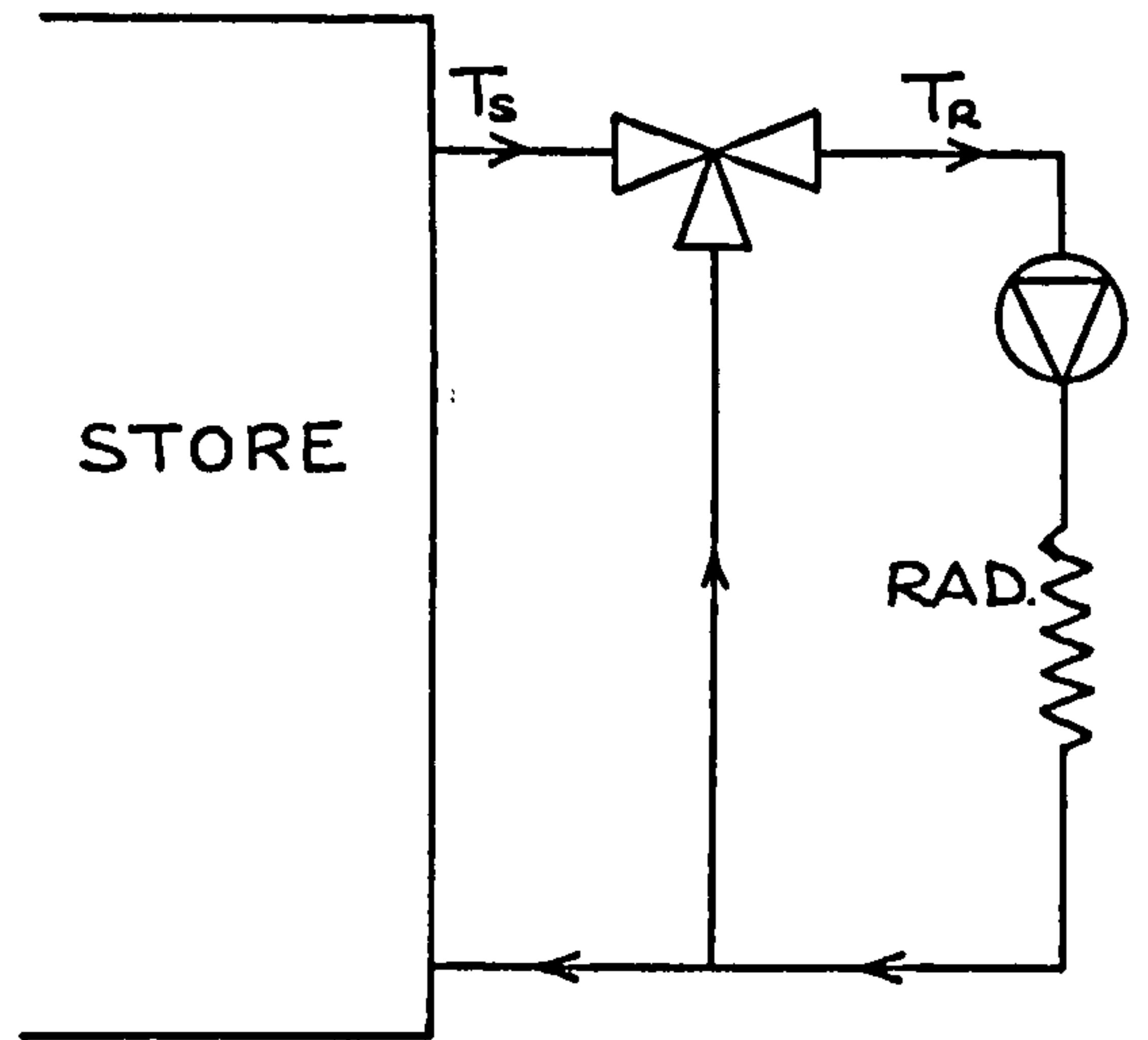
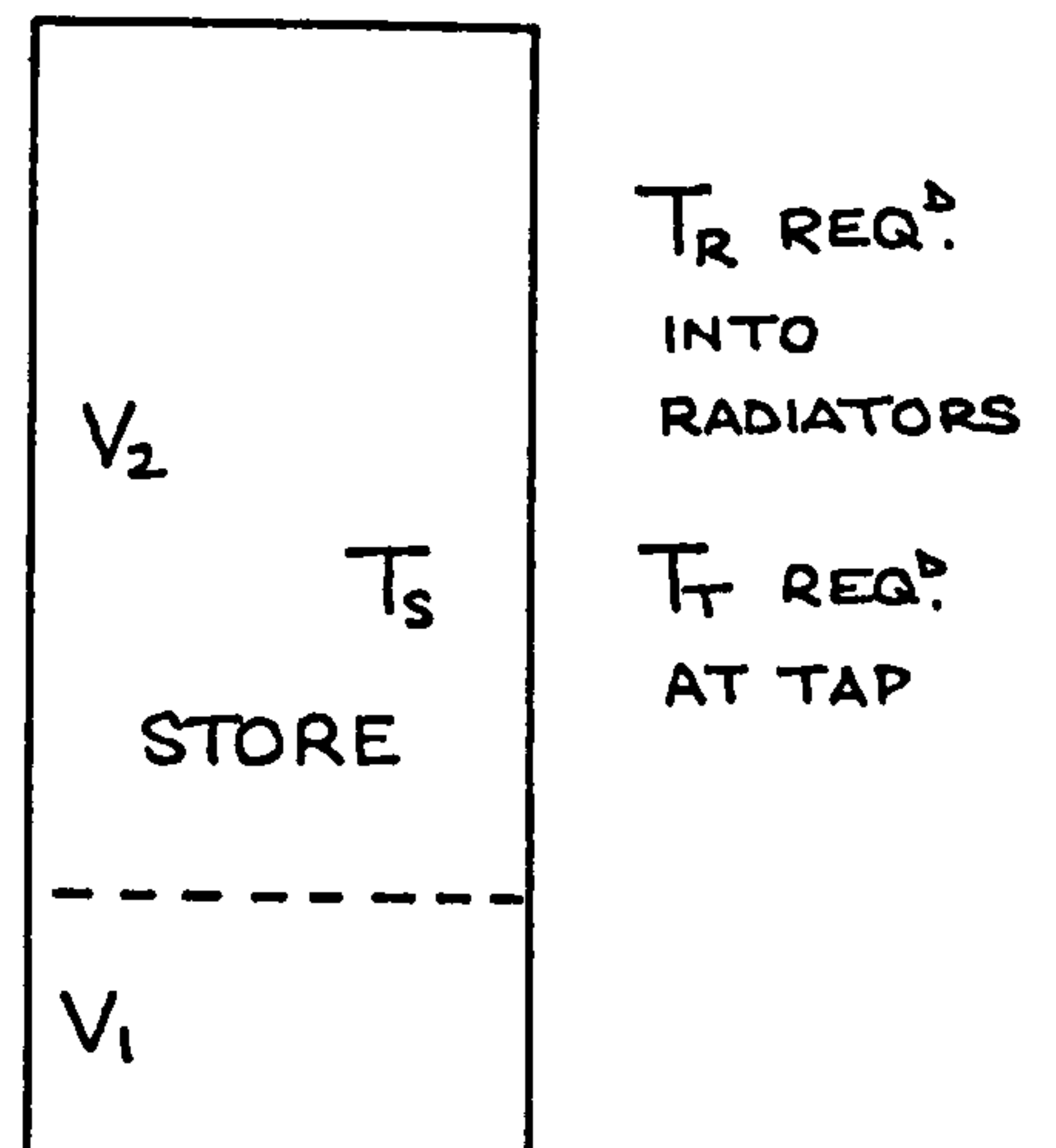


FIG 3.2:

USE OF DOUBLE 3WV'S AND  
H.E.'S ON D.H.W. CIRCUIT

FIG 3.3:

THEORETICALLY,  $V_1$  IS  
DISCHARGED FROM  $T_s$  TO  $T_r$   
TO MEET D.H.W. DEMAND,  
LEAVING  $V_2$  UNTOUCHED



3.1 allows for the store discharge cycle to continue until  $T_S$  is reduced to  $T_R$ , the required radiator flow temperature. If  $T_R$  is high, the store, which operates between a temperature approaching the boiler cut-off temperature and  $T_R$ , is limited in that it must be charged more often. (The common practice on the Continent, however, is to design for a greater radiator area and lower flow temperatures, which results in a system more immediately conducive to the gains which can be obtained from storage.) Hot water in the store is therefore at a premium if the storage capacity is not to be under-exploited.

During a d.h.w. draw-off, the upper coil (of the two interconnected coils) will supply heat to d.h.w. at an increasing rate, due to the fact that the interconnecting link obliges all d.h.w. flow in the store to pass through both coils. The h.e.'s therefore deliver d.h.w. to the mixing valve at temperatures ( $T_S$ ) far higher than  $T_T$ , and the "dilution" of this flow to  $T_T$  indicates a wasteful use of high grade heat at the top of the store. It was therefore suggested at British Gas that this h.e. configuration might be improved if a second 3WV, adjacent to the lower h.e., were incorporated. Thus, while the available energy of the lower zone is sufficient, the heat for small d.h.w. draw-offs could be met solely by the lower h.e.. High temperature heat stored in the upper part of the tank would remain undisturbed for subsequent release to the radiator system, see Fig. 3.2.

The double interconnected h.e. configuration of the current Cormorant might, at worst, be supposed to operate like a single h.e. at the top of the store, drawing heat from the whole store. A rudimentary theoretical evaluation of the advantage of the double-3WV system of Fig. 3.2 is as follows.

Theory : Suppose a store is fully charged to a uniform temperature  $T_S$ . The system of fig. 3.2 functions such that the lower heat exchanger draws its heat from a volume  $V_1$  of the store (whose uppermost layer coincides with the top of this h.e.). The remainder of the store is of volume  $V_2$ , see fig. 3.3. The heated store liquid will remain static for a period, and will supply a series of light d.h.w. draw-offs before the c.h. pump is switched on subsequently.  $V_1$  is

cooled first by d.h.w. draw-off, and there is no mixing with  $V_2$ . Supposing an externally-compensated system, when the c.h. pump starts the required radiator flow temperature will be  $T_R$ . The required temperature of d.h.w. at the tap is  $T_T$ , regulated by the lower 3WV. Suppose now that the lower h.e. has been so positioned that a discharge of  $V_1$  from  $T_S$  to  $T_T$  just supplies the various, light d.h.w. demands during this period. Then,

$$\text{Heat to d.h.w.} = \text{heat lost by } V_1 = V_1 \text{ Cp } (T_S - T_T) \quad (1)$$

Available energy remaining in the store w.r.t.  $T_R$  when the radiator pump starts is:

$$AE = V_2 \text{ Cp } (T_S - T_R) \quad (2)$$

Now let us suppose that there were only one h.e. (effectively) positioned at the top of the tank, drawing heat from the volume  $(V_1 + V_2)$ . The same heat would pass to the d.h.w. as in (1) above. The heat available w.r.t.  $T_R$  is now:

$$AE' = (V_1 + V_2) \text{ Cp } (T_S - T_R) - V_1 \text{ Cp } (T_S - T_T) \quad (3)$$

The advantage of the double-3WV h.e. system is  $AE - AE'$ , and

$$AE - AE' = V_1 \text{ Cp } (T_R - T_T) \text{ from (2) and (3)} \quad (4)$$

Further, dividing (4) by (2) yields

$$\frac{AE'}{AE} = 1 - \frac{V_1}{V_2} \cdot \frac{(T_R - T_T)}{(T_S - T_R)}$$

Therefore, the double-3WV h.e. system makes more heat available to the radiators than the supposed single h.e. system by an amount which increases as

$$\frac{V_1}{V_2} \cdot \frac{(T_R - T_T)}{(T_S - T_R)} \text{ increases, i.e.}$$

(i) if  $(T_R - T_T)$  is large; this is likely to occur in high flow temperature systems, where  $T_R$  might be  $70^\circ\text{C}$  and  $T_T$   $55^\circ\text{C}$ , in particular the high temperature systems now favoured in the U.K.;



(ii) if  $(T_S - T_R)$  is small, i.e. the store temperature  $T_S$  is near to  $T_R$ , and any "pilfering" by the d.h.w. system is clearly going to reduce heat available to the radiators; and

(iii) if  $V_1/V_2$  is large;  $V_1$  was sized to correspond to the d.h.w. demand,  $V_1/V_2$  increases as the lower h.e. is raised, or as the d.h.w. demand to be catered for by  $V_1$  increases. Thus the advantage of the double-3WV h.e. system increases as the d.h.w. requirement increases. It is plain, though, that a high d.h.w. demand during this period will reduce the likelihood of either system meeting the subsequent radiator demand:  $V_1$  ought to be as small as the d.h.w. demand will allow.

This simplistic theoretical analysis suggests that a second 3WV, together with independent h.e.'s, may be an advantageous modification to the "Cormorant" design. It is, however, plain that the performance of the current double interconnected h.e. configuration will lie somewhere between the two examined here, i.e. better than a single h.e., but worse than the proposed double-3WV system of Fig. 3.2. On the basis of this reasoning, it was decided to build a rig at Cranfield to try to evaluate such a system. The British Gas Corporation (Watson House) supplied a storage tank drilled to accommodate any two of a total of six Integron heat exchangers, together with three-way valves (3WV's) and other components, with a view to testing various h.e. sizes and positions, and assessing the performances of the respective combinations. The rig would be, as far as possible, a model of a complete domestic c.h. system, such that, once constructed, tests of increasing complexity would be possible. Thus, with suitable instrumentation, any chosen d.h.w. schedule and radiator requirement might be imposed over, say, a 24-hour period.

## THE EXPERIMENTAL RIG AND TESTING

### 1. Layout

As explained above, the rig was designed to accommodate several tests; only a few of these were conducted in the time available. The schematic layout is presented as Fig. 3.4. Central to the rig was the copper storage tank, see below. This was connected to three flow circuits (any or all of which could be run when desired):- (i) the boiler circuit, (ii) the radiator circuit and (iii) the d.h.w. circuit. Each circuit contained its own pump; the water flow rate could be varied in any circuit by means of gate valves if desired.

The boiler circuit charged the tank to any desired temperature, to a maximum of about 94°C. The tank could be discharged either through the simulated radiator circuit, or through the d.h.w. circuit. An attempt was made to simulate cold mains pressure at the entry to the d.h.w. circuit, see 2(vi) below.

### 2. Principal Components

(i) Tank: British Gas supplied an open-topped copper cylinder, of diameter 39.0 cm and height 160 cm, slightly domed internally at the bottom, see Fig. 3.5. On one vertical plane, the tank was drilled for 4 x ¾" B.S.P. fittings to connect to the boiler and radiator circuits. On a plane at 90° to this, the tank was drilled for 8 x ¾" B.S.P. fittings to accommodate the two d.h.w. heat exchangers, as shown. Thus the connections of the upper heat exchanger could be in one of two positions, and those of the lower heat exchanger in one of four positions. (In addition the asymmetry of each heat exchanger doubled the number of effective vertical positions available - see below). The outer surface of the tank was lagged using glass fibre insulant ("Super Saver"). To maximise the rate of heat loss through the loose-fitting metal lid, a wire cage was supported some 4 cm below the lid, and this was covered with a large number of small, hollow, plastic balls. A water depth of some 156 cm (cold) was maintained throughout the tests, i.e. a volume of some 190 litres.

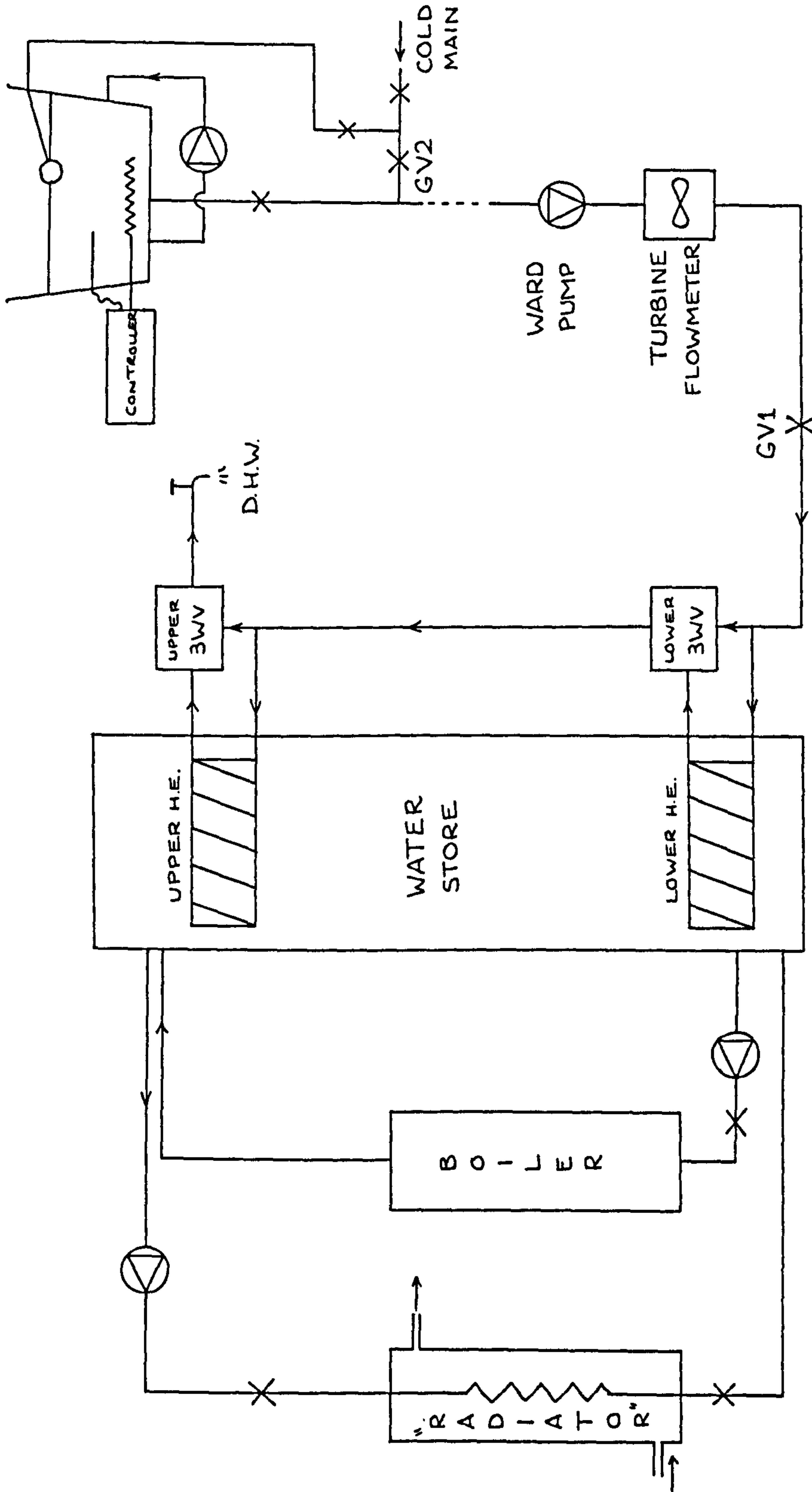


FIG 3.4: SCHEMATIC LAYOUT OF THE RIG



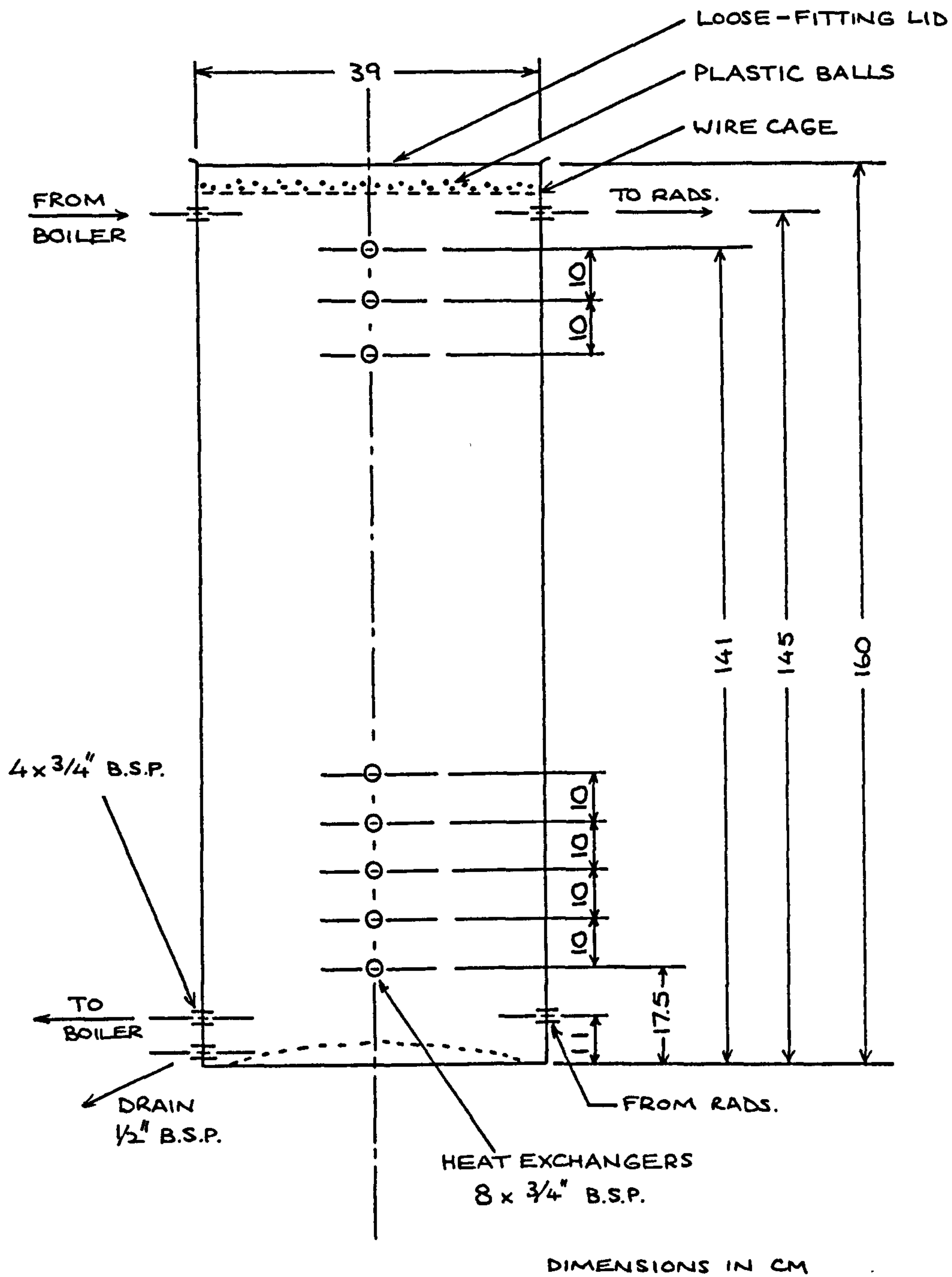


FIG 3.5: CYLINDRICAL COPPER STORAGE TANK

(ii) Boiler : The facilities of the laboratory into which the rig was to be built would scarcely have accommodated a gas-fired boiler (or circulator) flue. Further, the tests envisaged for this study required a "slave" boiler only, whose characteristics were not going to be studied, and which would be used only as a means to pre-heat the tank prior to discharge-testing (i.e. boiler off). To simplify the rig therefore, British Gas supplied an electrically-powered 3 kW boiler simulator, used extensively in testing at Watson House. The behaviour of this device is apparently very similar to the customary gas-powered circulator used for the Cormorant, and it would suffice as an adequate replacement for the circulator in many dynamic test situations. Though the boiler circuit was always pumped for this work, the rig could also be used for buoyancy-driven boiler studies. This heater was fitted with a safety thermostat, which was in fact never used as the limit of warm-up was better controlled by temperature observation than by a thermostat cut-out.

(iii) Radiators : A domestic radiator system was simulated by a simple shell/tube water/water heat exchanger (of the counter-flow type), of cooling capacity up to 5kW. The radiator load could thereby be set up by an appropriate flow rate of the subsidiary water circuit through this heat exchanger. For this study, the radiator system was never used, but is immediately available for any further study; the circuit was left blanked off always.

(iv) D.h.w. Heat Exchangers : Six heat exchangers were supplied, one pair in each of three sizes, see photos 3.1. Each heat exchanger was fabricated by coiling a length of "Integron" heat exchanger material into a helix of diameter approximately 20 cm and length approximately 32 cm. The sizes of heat exchanger supplied were designated;-

- LARGE - from an 8 m length of Integron tubing
- MEDIUM - from a 6 m length of Integron tubing
- SMALL - from a 4 m length of Integron tubing

Any two of these six heat exchangers could be mounted horizontally in any of the various positions available in the tank described above.

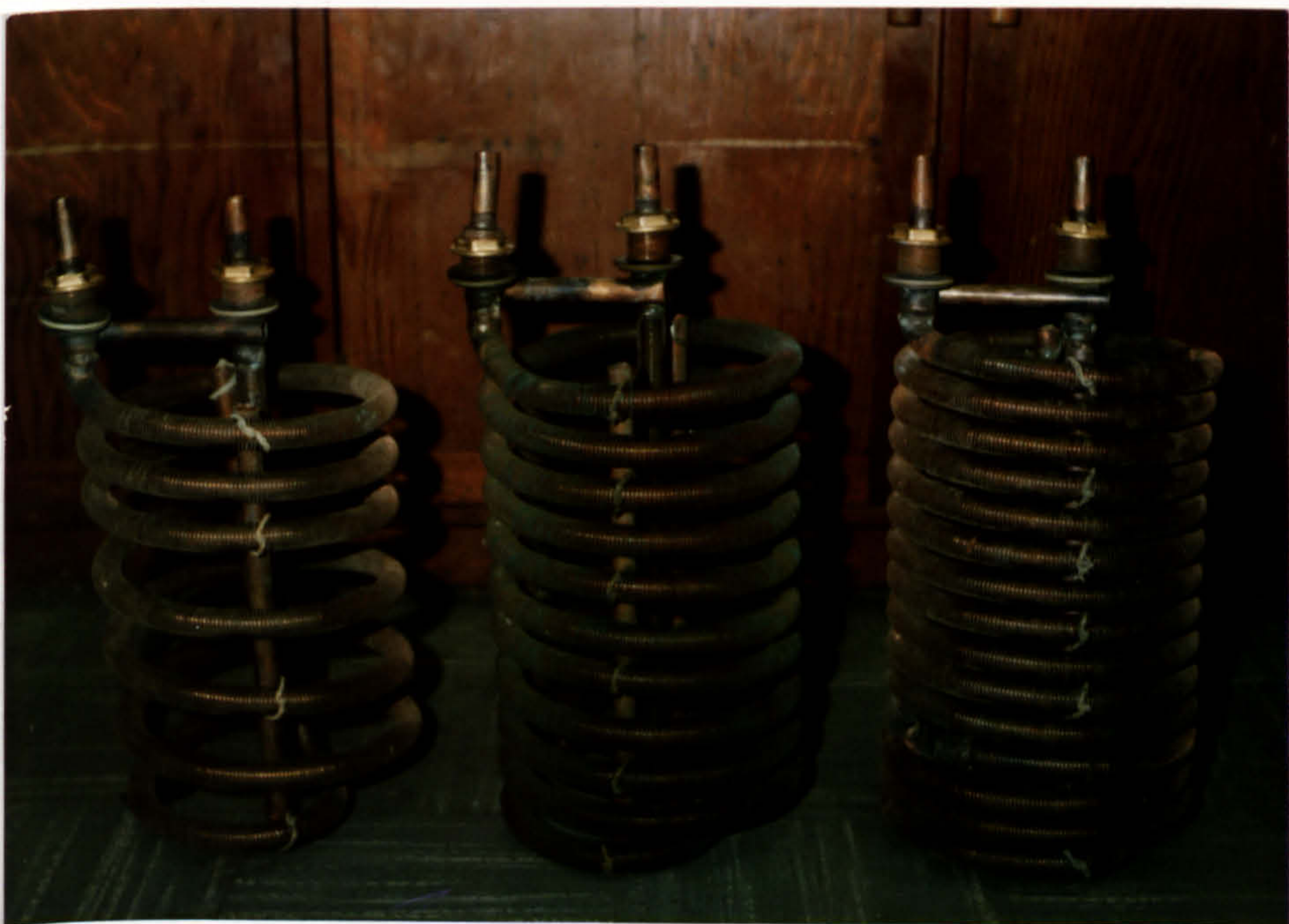




SMALL

MEDIUM

LARGE



PHOTOS 3.1: THE "INTEGRON" HEAT EXCHANGERS



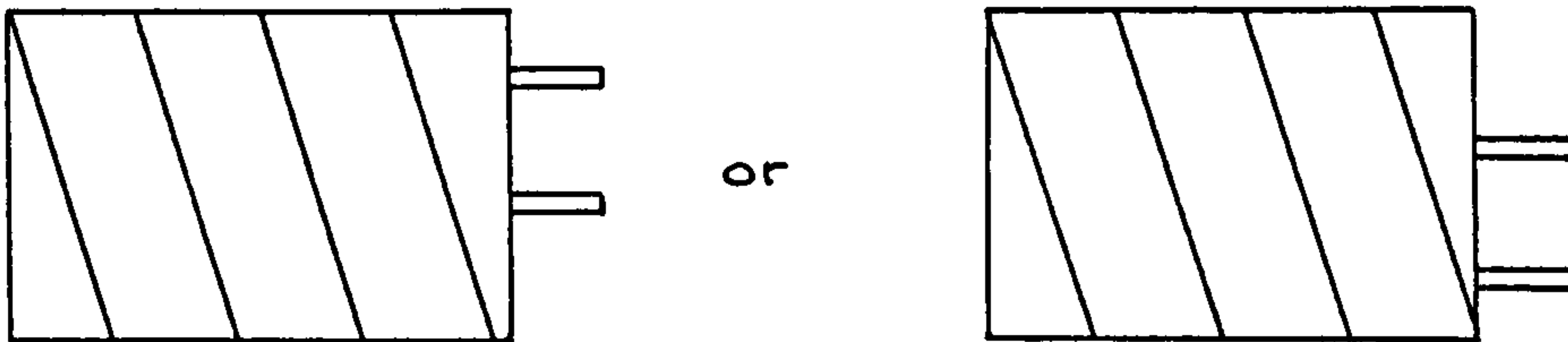
Thus (a) nine permutations of pairs of heat exchangers were available:-

Bottom h.e.	with	Top h.e.
SMALL	"	SMALL, MEDIUM or LARGE
MEDIUM	"	" " "
LARGE	"	" " "

and (b) for any one of these nine h.e. configurations:-

- the top h.e. could be placed in either of two vertical positions, and
- the bottom h.e. could be placed in any of four vertical positions.

The number of vertical positions of each h.e. could be further increased by fitting the asymmetrical h.e. in either of the following orientations:-



(v) Three-way Valves (3WV's): In conjunction with each d.h.w. heat exchanger a three-way mixing valve was fitted to reduce h.e. output temperature to no more than that set on the valve. Prototype Oventrop valves were used; each comprised a shuttle activated by a small wax-filled cylinder. It was hoped that they would (a) have a rapid response, (b) control well to the set temperature and (c) do so without interacting with one another.

(vi) Cold Water Feed: As shown on the layout diagram (see Fig.

3.4), a cold water settling tank was used so as to ensure a constant head for the Ward pump, which was to generate a pressure roughly equivalent to mains pressure to feed the d.h.w. system. This tank was an ordinary domestic 60/40 gallon (plastic) tank, and it was fitted with three modifications so as to allow for control of the cold water temperature at inlet to the d.h.w. system:-

- (a) a mixing loop, consisting of a 1/2" pipe used to circulate the water in the tank, powered by a small Laing pump,
- (b) a standard domestic immersion heater,
- (c) a 4-wire thyristor, coupled to a Eurotherm controller.

The tank was mounted on a 2-metre-high angle-iron frame. Cold water was received via a ball-cock at, say, 15°C. A set, working temperature was dialed into the Eurotherm controller such that this temperature was slightly above the expected daily variation of cold water temperature from the building supply, say 18°C. The Eurotherm would inject pulses of heat via the immersion heater into the tank water until the temperature indicated by the thyristor equalled that set on the Eurotherm. It was then hoped to operate all tests at a fixed 18°C irrespective of supply temperature variations. In fact, the whole tank/control assembly was discarded for the tests, and supply cold water was passed directly into the eye of the Ward pump, see below ("Pretesting").

### 3. Instrumentation

Discharge tests (only) were planned for this study, for which the following parameters would have to be monitored:-

- (i) several temperatures, at various points in the water flows; a fast response would be required,
- (ii) d.h.w. flow rate; this would vary during a discharge

A scanning data-logger would be required for monitoring these

parameters.

(i) Temperatures

Though P.R.T.'s were used with considerable success at Spa School, thermocouples were preferred for this laboratory investigation due to their considerably lower cost (as so many were needed), and due to their ease of insertion into the water flows. The rig circuits were therefore monitored by means of thermocouples (t.c's) as indicated on Fig. 3.6.

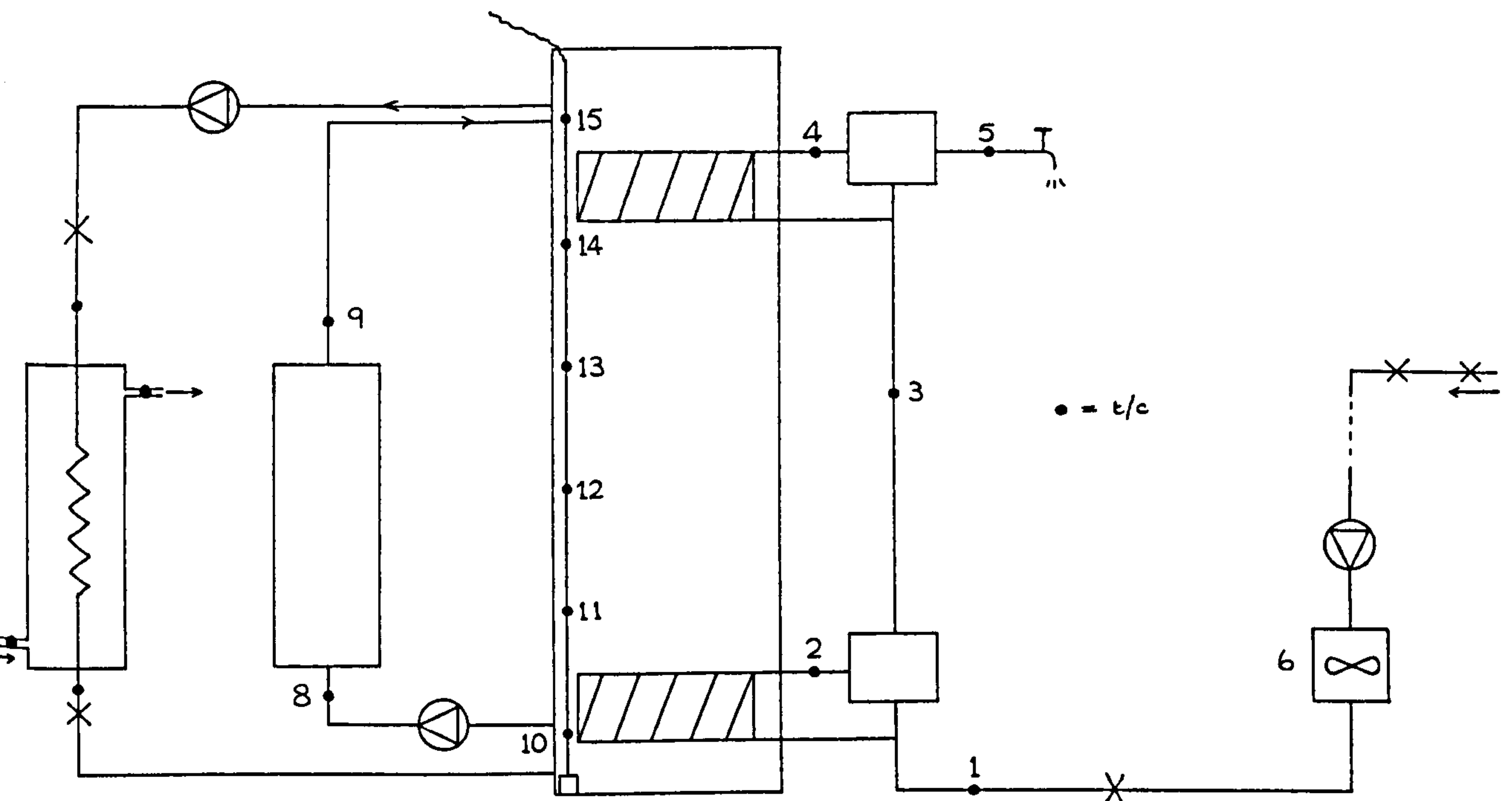


Fig. 3.6 : Thermocouple and turbine positions, with channel numbers.

D.h.w. circuit temperatures were recorded by t/c's 1-5 on Fig.



3.6, i.e. at both inlet ports of both 3WV's (nos. 1-4), and the delivered temperature at the tap (5). Boiler circuit temperatures were recorded by t/c's 8 and 9 at either end of the boiler. Store profile temperatures were indicated by t/c's 10-15, fitted on a Perspex rod, held vertically in the tank adjacent to the h.e.'s. These were equally spaced along the Perspex rod, and were intended to give a rough guide (only) to stratification in the store. The rod was maintained in place by an aluminium weight attached to its bottom end, and it passed through a small aperture cut in the wire mesh to its top. The store lid closed down over the Perspex rod.

(ii) Flow rate

For simple discharge tests, the only flow rate of interest was the d.h.w. flow at the tap. The 3WV action would vary the hydraulic impedance of the d.h.w. circuit, thus for a given (approximately constant) head supplied by the Ward pump, flow rate would fall during a discharge. The rate of decay of this flow and its likely fluctuations, were not known.

Experience at Spa School (see Part II) revealed that the quality of the results would be as good as the accuracy of the flow measurements. Visual methods (rotameter) would be inadequate. An electromagnetic flowmeter might have been available through British Gas, but reconditioning and calibration would have held up the work for a considerable time. A digital device would be accurate only if the flow increments counted were small. An analogue, continuous output would be preferable. British Gas supplied a small electromagnetic turbine-flowmeter compatible with these tests, together with a signal processing unit. The turbine was soon found to be unserviceable, but a replacement was quickly obtained from its manufacturers (AOT Systems of Andover, Hants). A drawing of this device is presented in Appendix 3.1 as Fig. 3.7, together with a copy of its test certificate. The turbine generated a signal whose frequency was proportional to the flow rate. The signal processing unit was a frequency-to-DC converter, and thus a DC analogue signal was available.

It was hoped to measure the d.h.w. flow rate on two devices for enhanced reliability of this critical parameter. To this end a German-built "Pollux" heat meter was fitted across the hot and cold limbs of the d.h.w. flow. The pulse output of the flowmeter of this device would have required the construction of a converter to generate an analogue output, which time did not permit. The heat meter was, however, left in the circuit, though little use was made of it. In particular, its kWh display is rather coarse (necessarily) - a more accurate integral of heat flow during a given discharge would have been a beneficial double-check.

#### (iii) Data-logger

A "Fluke" data-logger was available, capable of receiving electrical inputs on up to 60 channels. Thus all the thermocouples together with the output from the turbine's signal converter were each fed into a channel of the Fluke (the numbers on Fig. 3.6 denote the associated channel nos.). The Fluke was capable of being programmed for scan content, interval between scan and method of scanning (continuous, interval, single etc.). It would have been possible to feed the scan data into a "floppy disc" recorder, but this presented certain problems at the time, which led to the decision to simply print each scan onto paper tape. Though more laborious, this method proved wholly reliable and was adopted for all the tests.

#### 4. Redefined Test Objectives

As mentioned in the Introduction, the rig was designed such that the storage tank could be subjected to a wide variety of charge and/or discharge tests. For example, with the addition of a suitable boiler-actuating control (depending on the state of charge of the tank), a full day's domestic demand could have been simulated and monitored. During Spring 1985 it became evident that such tests as were going to be attempted would have to be closely defined, and that the testing possible in the time available would fall well short of the rig's capacity.

It was finally decided that testing would be restricted to an investigation of the various heat exchangers available, with a view to choosing an optimal combination of h.e.'s, and if possible choosing their optimal vertical position in the tank. This would be influenced by

- (i) blanking off the simulated radiator circuit,
- (ii) using the boiler circuit as a "slave" merely to heat the storage tank - i.e. no properties of the boiler circuit (e.g. buoyancy flow rates) would be investigated,
- (iii) adopting a single test procedure - a simple discharge of the preheated store through the d.h.w. circuit. Comparisons of the discharge curves related to the various h.e. configurations available would allow d.h.w. take-off to be optimised.

In addition, the action of the Oventrop 3WV's would be briefly assessed.

## 5. Pretesting

### (i) Turbine Calibration

The pulse signal from the turbine flowmeter, when processed by the frequency - DC converter, gave a rapidly fluctuating signal to the Fluke, at a constant d.h.w. flow rate. The amplitude of this fluctuation was small - a good, sensibly constant flow rate was being monitored - but the sensitivity of the turbine was such that the corresponding digital display of the Fluke was quite unreadable. If, however, the Fluke was made to sample the signal every few seconds, the flow was found to be nearly steady, and a mean value was easily obtainable.

The turbine was thus carefully calibrated over the flow range available to the rig, using the customary receptacle/stopwatch method. The calibration is plotted on fig. 3.8, Appendix 3.1, and is clearly linear. Its equation is:-

$$\text{Flow rate, l/s} = (\text{mV at Fluke} - 1.27)/613.3$$



Thus, for computational purposes, the processed turbine signal (scanned by the Fluke) was translated directly into litres/second with a high degree of confidence in this value.

The nature of the tests being comparative, it was imperative that the performance of the turbine remained constant throughout the entire testing period (even if its absolute readings were in error). As a check on this, the points corresponding to the three greatest flow rates on fig. 3.8 were taken after the testing had been completed. (A higher flow rate was by this time available, due to the inlet modification described below.) They are seen to fall onto the calibration line very satisfactorily. It is therefore assumed that (a) the turbine's action remained constant for all the h.e. configurations investigated, and (b) the calibration line is linear and accurately expressed by the above equation, over the full flow range of the tests.

The writer is therefore satisfied that the d.h.w. flow rate measurements recorded here are altogether adequate to satisfy the objectives of the tests. This turbine flowmeter is an excellent device, with a very rapid response to transients, and is considered the most appropriate means of monitoring the flow rates in an experiment of this kind. Its performance under prolonged testing, perhaps in a field test in a domestic c.h. system with rather dirtier water than that used here, cannot, of course, be commented upon. Here, and in other similar investigations, an accurate value of flow rate is critical to a valid analysis (see also the energy-balance results later).

(ii) D.h.w. inlet pressure

It must be borne in mind that:-

(a) British Gas take 9 l/min and 12 l/min as representative, selected values of d.h.w. draw rate at the tap, and,

(b) one advantage of this integral d.h.w. facility is that it is powered by cold mains pressure. The cold main can operate under a pressure as high as about 50 m head (depending on location), but this

is very often greatly reduced by internal furring of the pipes.

The cold water supply to the laboratory was fed, not directly by a main, but by a tank on the roof of the building; it could thus offer a pressure of no more than about 20 m head (25 p.s.i.).

When the rig was first started up, the cold water tank (see layout diagram, Fig. 3.4) supplied water at a head of only about 2 m to the Ward pump. It had been hoped, based on experience at British Gas Corporation (Watson House), that the Ward pump would generate enough head to allow the upper d.h.w. rate of 12 l/min to be exceeded, so that the flow could be throttled, by manual adjustment of a gate valve, to a constant chosen flow of say 9 l/min for the entirety of a discharge test, i.e. compensating continuously for the increasing h.e. impedance incurred by 3WV action. The Ward pump could not meet this expectation. Preliminary tests using the MEDIUM/MEDIUM h.e. combination yielded a flow (with all gate valves fully open) of about 9 l/min. at start-up, reducing to some 6 l/min. at full flow through the h.e.'s, clearly an inadequate flow rate. Further, the cold-water tank was slightly heated, to control inlet temperature to some 18°C, a fixed temperature higher than any expected (varying) laboratory supply temperature. This control was found to be rather approximate in that it was liable to vary through 1°C or so while a discharge progressed. In addition, 18°C was already departing from the likely real domestic cold main temperature, which might easily be as low as 10°C. So on the basis of poor pressure, temperature control and high inlet temperature, the original cold water feed configuration was rather inadequate.

It was decided to discard the cold-water tank, and to feed the laboratory cold water supply directly into the eye of the Ward pump. The 20 m head available to the pump, compared with the previous 2 m head, greatly enhanced flow rate such that 9 l/min could often be attained at full h.e. flow, though the luxury of throttling down to a constant 9 l/min throughout a discharge was not available. Discussion with Watson House suggested that d.h.w. take-off performance was rather insensitive to flow rate; thus a slightly reduced flow rate would induce a slightly greater  $\Delta T$ , such that power take-off remained

largely unaffected. The slightly differing flow rates produced with the various combinations of h.e. would therefore hardly affect the h.e. performance, and the comparison would remain valid. Thus, this cold feed arrangement was preferred and adopted - no gate valve throttling was attempted. The inlet to the first h.e. would be at a pressure which would remain largely constant throughout all the tests - it could be said to simulate the mains pressure in a given house.

This arrangement produced two considerable further advantages, and one setback, all unexpected. The temperature of the laboratory supply was found to be remarkably constant: certainly during a discharge, it remained sensibly constant except for the first minute or two during settling down. From day-to-day it scarcely changed more than  $0.5^{\circ}\text{C}$  or so, a bonus for such a set of comparative tests. Secondly, its value was typically  $15^{\circ}\text{C}$ , certainly cooler than that available using the cold water settling tank, and more nearly representative of a domestic situation. The quite unavoidable setback was that the laboratory supply was called upon to meet the needs of other users, whose usages (notably those of the toilets) were unpredictable. The familiar phenomenon named the "bog effect" was to influence the stability of the flow rate curves of many discharge tests.

### (iii) 3WV setting

It had been decided to conduct all the discharge tests with the 3WV combination set to deliver water at  $60^{\circ}\text{C}$ , as nearly as possible, at the tap. This is considered at Watson House a good, high, d.h.w. draw-off temperature, although rather too high in some instances. It was appreciated that, particularly when much higher temperatures were available from the store, good control to limit delivery temperature to  $60^{\circ}\text{C}$  would be required to avoid possible scalding.

With the tank charged, the lower 3WV was wound to a low setting to ensure a full bypass of the lower h.e. Under discharge flow, the top 3WV was then adjusted to deliver tap water at some  $60^{\circ}\text{C}$ . This setting was maintained throughout the tests (apart from the 3WV exchange necessitated by failure - see results section below). It had been hoped to set the lower 3WV also to  $60^{\circ}\text{C}$  to delay the use of the top



h.e. for as long as possible into a discharge. When the lower 3WV was thus set, the initial burst of d.h.w. during a discharge was delivered at up to  $65^{\circ}\text{C}$ , indicating a considerable leak at the top 3WV. The lower 3WV was therefore reset to about  $55^{\circ}\text{C}$ , and the discharge was thereby kept to about  $62^{\circ}\text{C}$  initially. This combination of 3WV settings was, of course, arbitrarily chosen, and maintained for the whole of the discharge tests, so as not to compromise the comparison of h.e. performance.

## 6. The Test Procedure

The writer believes that, for the sake of anyone continuing this work, a detailed account of the experimental procedure ought to be included at this stage. All tests were discharge tests, the store being preheated to a selected uniform temperature, and then discharged using cold water passed through the d.h.w. h.e.'s to emerge at  $60^{\circ}\text{C}$  (if possible). After fitting the selected h.e. pair, (i) the rig was reassembled, (ii) the boiler heated the rig up to  $80^{\circ}\text{C}$ ,  $70^{\circ}\text{C}$  or  $60^{\circ}\text{C}$  as appropriate, and (iii) with the boiler circuit shut-down, the Ward pump produced d.h.w. flow for discharge, as follows:-

(i) Fitting/preparation: Fitting of the top h.e. was a simple matter of draining the tank down below the h.e. ports. The wire mesh retaining the plastic balls was carefully removed so as not to disrupt the thermocouple wiring from the Perspex rod. The top h.e. was carefully removed and the new h.e. fitted, ensuring that its gaskets were well positioned, and the securing nuts tightened hard. To change the lower h.e. the top h.e. had, of course, to be removed, and special care taken not to damage the Perspex rod. The tank needed to be drained down, and the exchange h.e. lowered using a support rod. When a large h.e. was fitted to either position, its remote end was supported by string from a small hole drilled at the top of the tank - the couple required to support these large exchangers at the ports was excessive for the copper of the tank. The assembly was rebuilt and the tank refilled (to the level of the wire mesh).

ii) Warm-up: The boiler and pump were switched on; the tank charged at a rate of some  $13 - 16^{\circ}\text{C/hr}$ . Warm-up time could be reduced

by ensuring that the lagging was well positioned, and the lid covered. Any leaks at the h.e. ports were prevented by further tightening-up when the tank water was warm. By switching on the laboratory supply to the d.h.w. circuit, any leaks in the d.h.w. connections at the h.e.'s were corrected. Each h.e. configuration was tested with the store heated to each of three temperatures -  $80^{\circ}\text{C}$ ,  $70^{\circ}\text{C}$  and  $60^{\circ}\text{C}$ . The store temperature profile was monitored by channels 10 - 15 on the Fluke logger; it was found empirically that, if heating to  $80^{\circ}\text{C}$ , when channel 11 had reached  $79.9^{\circ}\text{C}$ , it was appropriate to switch off the boiler (only). At this point the boiler output temperature was some  $82^{\circ}\text{C}$ , and the boiler pump would then circulate water such that the cooler water at the bottom of the store was delivered to the top of the store, while the slightly overheated upper layers passed to the bottom. Buoyancy mixing ensured a remarkable uniformity of temperature. While this continued a trickle of cold water was admitted to the d.h.w. side such as to cause the 3WVs to adjust to their "mixed" positions, in order to minimise transients at the beginning of the discharge. The d.h.w. side was thus filled with water warmed to the "ready-to-go" temperatures. For a few minutes before the boiler switch off, a sink tap was run in the laboratory, downstream of the rig, to drain off cold water warmed slightly during its rest in the pipework; this ensured a rapid assumption of the steady, cold delivery temperature from the laboratory water supply. With everything switched off (including the sink tap) except the boiler pump, scans of the store profile showed its gradual descent, due to jacket heat loss; its arrival just above  $80^{\circ}\text{C}$  could be anticipated. At this point the discharge was set underway.

(iii) Discharge: It was crucial to establish an accurately repeatable procedure at the start of discharge. When the thermocouples indicating the store temperature profile converged on  $80.1 - 80.2^{\circ}\text{C}$ , the boiler pump was switched off (except in one instance - see subsequent section). The rig was left for a few seconds so that momentum flow would cease. The following manoeuvres were carried out as quickly as possible (5 - 10 seconds) always in the same order:

- the laboratory cold water supply tap was switched fully on

(GV2 of Fig. 3.4 was left fully open)

- the Ward pump was switched on
- GV1 of fig. 3.4 was opened fully to admit full d.h.w. flow
- when GV1 was about 3/4 open, the Fluke logger was set off on "interval scan", set to about 12 second intervals, such that at the moment it scanned the water flow, full flow had just started.

The discharge test was deemed to start at this instant, at the moment GV1 was fully open. Each scan was printed onto paper tape by the Fluke. As the discharge progressed, the rate of change of temperatures and flow became less marked and longer scanning intervals were used. The interval time was continually reset on the Fluke during the discharge. For an 80°C discharge, the 12 sec. initial scanning interval would be progressively increased to 90 seconds or 2 minutes; for a 60°C discharge, the maximum interval reached would be 30 or 40 seconds. In all, some 25 - 30 scans would be recorded per discharge, which was allowed to continue until the d.h.w. delivery temperature fell below 43.3°C, a temperature used at Watson House to denote the end of useful d.h.w. take-off. As 43.3°C was approached, the scan interval was reduced to more accurately pinpoint the end of discharge.

At this point, the Ward pump was switched off and the d.h.w. circuit isolated. If further tests were required that day, the boiler and pump were immediately set in action. It was possible sometimes to complete the three tests on a particular h.e. configuration (80°, 70° and 60°C) in a single day.

As mentioned above, casual uses of the laboratory water supply by other users often affected d.h.w. flow rate, usually slightly and for a short period. A small effect during a discharge could be tolerated; many tests however, were repeated due to what might be termed disproportionate (even irresponsible!) use of the ground floor ablutions, notably at 12.50 hrs. and 17.20 hrs. These times were avoided as far as possible; the best curves were obtained outside working hours.



## 7. Data Processing

(i) Acquisition: each scan, printed onto paper tape by the Fluke, comprised the following elements:-

Date and time (in hrs, mins, seconds)

Channel Nos. 1-5 : d.h.w. circuit temperatures (T1 - T5),

Channel No. 6 : converted voltage from the turbine - d.h.w. flow rate,

Channel Nos. 8-9 : temperatures across boiler (T8, T9),

Channel Nos.10-15: store profile temperatures (bottom to top, T10 - T15).

(Channel No. 7 was left unused in case it should be required to record a second value of d.h.w. flow rate from another device).

Each discharge test contained some 25-30 scans; these were typed from the paper tape into the Institute VAX 785 computer to form a short datafile corresponding to each test. Conserving a fixed, convenient format enabled the data of each test to be quickly sifted on the computer, and graphs and derived data obtained.

(ii) Graphics programs: these were written so that any chosen set of the above test parameters could be plotted by the computer. Three graph formats were created under the following titles:-

RIGPLOT : a graph of d.h.w. inlet temperature, T1,  
d.h.w. outlet temperature to tap,  
T5,  
store temperature adjacent to top  
h.e., T14,  
water flow rate, and  
accumulated heat gained by d.h.w.

D.H.W. TEMPS.: a graph showing the five temperatures in the  
d.h.w. circuit, T1-T5.

TEMP.PROFILE : a graph showing the temperatures of channels 1, 3 and 5 of the d.h.w. circuit (inlet, intermediate and delivery temperatures), together with the store temperature profile on Channels 10-14 (not Ch. 15, see below).

Every discharge test, whether presented here or not was recorded and plotted to the RIGPLOT format. Certain tests, where appropriate, were plotted to D.H.W. TEMPS or TEMP PROFILE formats, or both. Each graph produced was headed using both the program title (one of the three titles above) and also the date the test was carried out. (The date and the initial store temperature of the discharge avoided any confusion of the sort "which graph - which test?" - all the tests were catalogued day-by-day.)

(Channel 15 : this thermocouple was near the top of the store, the top of the six mounted on the Perspex rod, giving the store temperature profile. It consistently indicated a temperature below those indicated by Channels 10-14, which, when the store was charged and fully mixed before discharge, agreed with each other remarkably, all five often being no more than  $0.1^{\circ}\text{C}$  apart. Thus, incidentally, the starting conditions of the store could be repeated well, very nearly at the required charge temperature of  $80^{\circ}\text{C}$  say. The channel 15 thermocouple was verified as being well submerged, and it was anticipated that it would indicate if anything a higher temperature than the others. To check, a second thermocouple was mounted alongside it on the Perspex rod, and this exhibited the same tendency. A different channel on the Fluke was tried with no improvement. Finally, supposing there to be a cold spot somehow induced by the in-store flow, the store temperature was checked alongside the channel 15 thermocouple using a mercury-in-glass thermometer - this agreed substantially with channels 10-14. Defeated by this, the writer ignored the Channel 15 reading, and supposed the channel 15 value to be equal to that of channel 14.)

## 8. The Test Schedule and Results

The rig was received with the two MEDIUM h.e.'s fitted, the lower

h.e. being fitted in the bottom of the four positions available for it. This configuration was arbitrarily chosen as suitable for the first tests; with hindsight, the lower h.e. might well have been fitted somewhat higher up.

The tests conducted fell into four exercises, comprising a total of 44 relevant computer graphs, which for convenience are presented in Appendix 3.2. The four exercises were:-

(i) A study of the tank profile during discharge: graph no. 1 shows the TEMP. PROFILE plot of a 70°C discharge with the store quiescent (as it was in all other discharges except the following). Graph no. 2 shows a corresponding 70°C test, but this time with the boiler pump left running. For the next section (i.e. the Analysis), the reader is asked to bear in mind, from graph no. 1, that temperatures "T11 to T14" are sensibly identical during discharge, whereas T10 in communication with the lowest h.e. falls far below T11 to T14; this is taken to denote that the quiescent store is well represented by two fully-mixed zones at all times.

(ii) A repeatability study : graphs nos. 3-5 are from tests which were direct repeats of those described in graphs nos. 18-20, see below.

(iii) The main study to compare the various h.e. configurations; graphs nos. 6-32 represent 80°, 70° and 60°C tests for all the nine available h.e. configurations.

(iv) A study of the influence of the vertical position of the lower h.e.: all using the MEDIUM/MEDIUM h.e. configuration, graphs nos. 33-41 examine three such lower h.e. positions (RIGPLOT), while graphs nos. 42-44 examine the effect of this position on the d.h.w. circuit temperatures (D.H.W. TEMPS.)

A summary of the tests and corresponding graphs reported here is presented in Table 3.1 below;



Test No	Test Title	Graph Nos	Graph Title	Comments/ Breakdown
1	Storage tank profile	1-2	TEMP. PROFILE	-
2	Repeatability	3-5	RIGPLOT	c.f. Nos. 18-20
3	H.E. combinations	6-32	RIGPLOT	Upper Lower 6-8 SMALL/SMALL 9-11 SMALL/MEDIUM 12-14 SMALL/LARGE 15-17 MEDIUM/SMALL 18-20 MEDIUM/MEDIUM 21-23 MEDIUM/LARGE 24-26 LARGE/SMALL 27-29 LARGE/MEDIUM 30-32 LARGE/LARGE
4	Lower h.e. vertical pos <sup>n</sup> . (All MEDIUM/MEDIUM)	33-41  42-44	RIGPLOT  D.H.W.TEMPS.	33-35 BOTTOM POS <sup>N</sup> . 36-38 MIDDLE POS <sup>N</sup> . 39-41 TOP POS <sup>N</sup> . -

Table 3.1 : Summary of Test Exercises

## 9. Data Analysis

The preceding section presented the test results in a graphical form. The nature of this work being comparative, these graphs do not present a convenient means of performance comparison.

It was therefore found necessary to extend the graphics program such that a quantitative summary of each test was available. A given test was characterised by a number of parameters derived from the raw data of the datafile and the above graphs. These parameters, and a brief explanation of their calculation, were as follows, (Program

designations are given in brackets):-

(i) Minutes to 43.3°C (TIMEND): the test duration. Scans were taken with T5 on either side of 43.3°C, and TIMEND calculated by interpolation.

(ii) Litres d.h.w. delivered (SFLOWEND): the total flow of d.h.w. to TIMEND calculated by a simple finite-difference integration.

(iii) Mean temp. °C (TMEAN): the mean temperature of the d.h.w. delivered, calculated from SFLOWEND (above) and the d.h.w. heat gain SEWEND (below).

(iv) Store temperature start °C (TSTART): the mean store temperature (mixed) at the start of a discharge. N.B. Mean of T10 - T14 only. Usually very close to 80° or 70° or 60°C.

(v) Upper zone end temperature °C (TENDZ2): the section of the store above the top coil of the lower h.e. was designated the upper zone, and was found to be effectively fully mixed, see graph 1 and explanation in paragraph 8 (i) above. TENDZ2 was the mean temperature of this zone at the end of a discharge.

(vi) Lower zone end temperature °C (TENDZ1): the lower zone was that volume of the tank below the top of the lower h.e. TENDZ1 was its temperature at the end of a discharge.

(vii) Store heat loss kwh (QPASSED): an estimate (only) of the heat lost by the store control volume during a discharge, based on TSTART, TENDZ2 and TENDZ1.

(viii) Water heat gain kwh (SEWEND): a finite-difference integral of the heat passed to the d.h.w., based on T1, T5 and the d.h.w. flow.

(ix) Error % (ERROR): theoretically, QPASSED and SEWEND should be identical (except for jacket loss). ERROR is the % difference between them, an indication of the accuracy of the instrumentation.

(x) Efficiency to TMAIN (EFF): an efficiency based on the assumption that the entire store might be discharged down to the cold d.h.w. inlet, temperature (TMAIN or T1). This is clearly pessimistic. The numerator is QPASSED, an estimate only, so EFF is also approximate.

(xi) Efficiency to an ideal discharge state (EFF43): this efficiency assumes that, theoretically the store might be discharged until the lower zone temperature reaches d.h.w. inlet (TENDZ1 → T1), and the upper zone temperature reaches 43.3°C (TENDZ2 → 43.3°C). The numerator is again QPASSED, but the denominator is lower than that of EFF above.

Hence,  $EFF43 > EFF$  and EFF43 is more realistic, though both are approximations only. These efficiencies may be more readily understood in relation to the following parameters:-

(xii) Store energy start kwh (TOTENAV): the initial store charge relative to d.h.w. inlet temperature.

(xiii) Store energy end kwh (TOTENEND): the final store charge relative to d.h.w. inlet temperature.

(xiv) Store energy end, ideal, kwh (TENIDEAL): the store charge relative to d.h.w. inlet temperature when TENDZ1 = T1 and TENDZ2 = 43.3°C.

(xv) Store energy unused, kwh : the store charge at the end of the test still remaining relative to the ideal dead state of (xiv) above.

Thus for the two efficiencies:

$$QPASSED = TOTENAV - TOTENEND$$

and  $EFF = 100 * QPASSED / TOTENAV$

$$EFF43 = 100 * QPASSED / (TOTENAV - TENIDEAL)$$



Thus, any test could be easily represented by these 15 derived parameters. Performance comparisons could now be effected simply in terms of a chosen parameter, e.g. efficiency or total heat passed to d.h.w.

The comparisons of the four test exercises will now be presented by means of tables of these derived parameters, and discussed.

NUMERICAL TEST RESULTS AND DISCUSSION

1. Storage Tank Profiles During Discharge

Graphs 1 and 2 of Appx. 3.2 show the store temperature profile developments of two discharges from 70°C. Their numerical analyses are presented in Table 3.2:-

Test Title	AUG 16B	AUG 6A
Features	70°, Boiler Pump <u>Off</u>	70°, Boiler Pump <u>On</u>
(i) Time to 43.3°C, min (TIMEND)	13.158	20.05
(ii) Litres d.h.w., l (SFLOWEND)	107.03	123.59
(iii) Mean d.h.w. temp. °C (TMEAN)	51.68	53.30
(iv) Store temp. start, °C (TSTART)	69.85	69.83
(v) Upper zone temp. end, °C (TENDZ2)	53.73	46.89
(vi) Lower zone temp. end, °C (TENDZ1)	30.00	45.90
(vii) Store heat loss, kWh (QPASSED)	4.736	5.116
(viii) Heat to d.h.w., kWh (SEWEND)	4.608	4.910
(ix) Error, 1 - (vii)/(viii), % (ERROR)	2.78	4.20
(x) Efficiency to T1, % (EFF)	38.81	45.66
(xi) Efficiency to 43.3/T1, % (EFF43)	65.00	72.48
(xii) Store energy start, kWh (TOTENAV)	12.202	11.204
(xiii) Store energy end, kWh (TOTENEND)	7.466	6.088
(xiv) Store energy ideal, kWh (TENIDEAL)	4.916	4.145
(xv) Unused store energy, kWh	2.550	1.943

Table 3.2 : Numerical results for the tank profile developments studied

In this instance, it is useful to include the discharge curves of the above tests; reduced copies of graphs 1 and 2 are included as Fig. 3.9 here:-

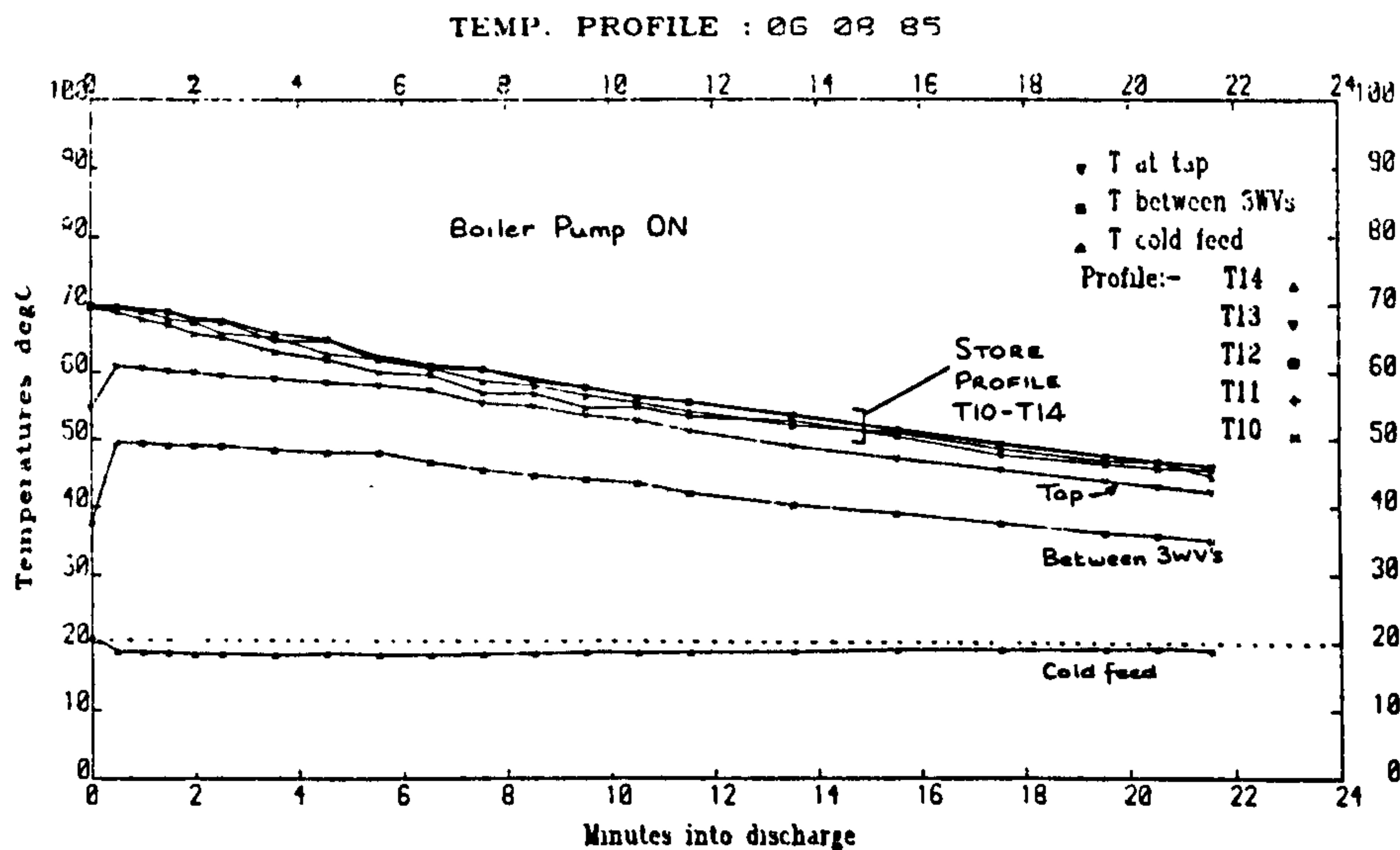
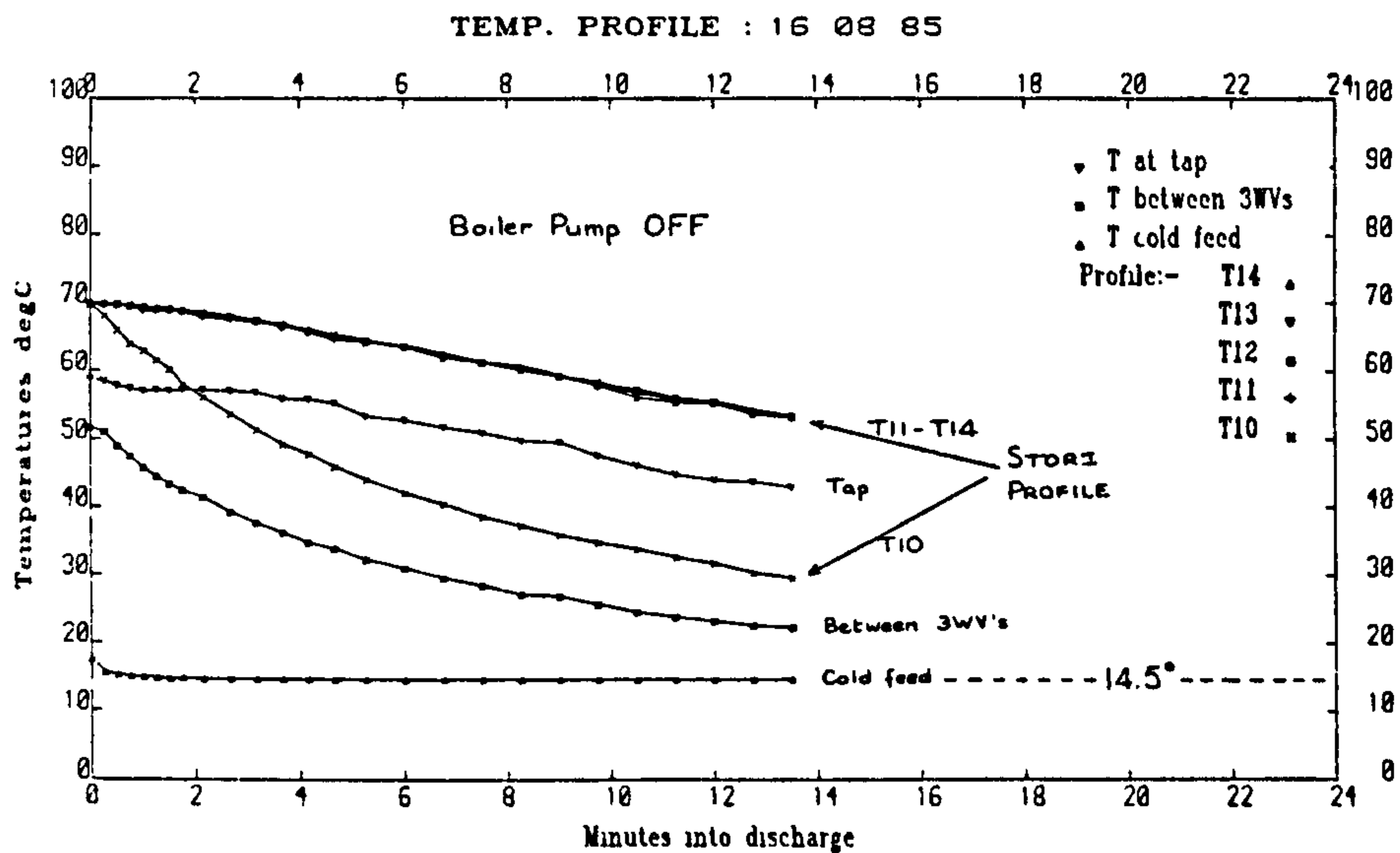


Fig. 3.9 : Temperature Profile Plots of the Tests Studied

The first column of Table 3.2 and the upper graph of Fig. 3.9



represent a typical 70° discharge, i.e. with the boiler pump OFF and the storage tank quiescent, run on 16th August 1985. Of principal interest here is the store temperature profile development, shown on the upper graph, where T11 - T14 are all seen to decay almost along the same line. T10 falls quickly away from T11 - T14. Of the thermocouples supported by the Perspex rod, only T10 was positioned below a horizontal on the upper coil of the lower h.e. This h.e. received colder d.h.w. water than the upper h.e., and the lower zone of the store was cooled much faster than the rest, its capacity being much smaller. It is assumed that the lower zone was all at temperature T10, and that its volume was that cylinder below the upper coil of the lower h.e. Its depth was thus taken (by measurement) to be 35 cm for these early tests. The virtual coincidence of the curves of T11 to T14 suggests that the rest of the tank (i.e. the upper zone) was effectively fully-mixed, and that the quiescent tank may be accurately represented by a two-zone model. (It is probable that some layers of tank water above the upper h.e. remained hotter than T11 - T14, but this h.e. was quite near the water surface.) Thus, as stated previously, the tank was taken to discharge from a uniform TSTART to TENDZ1 ( $\equiv$  T10) for the bottom zone, and TENDZ2 ( $\equiv$  mean of T11 to T15, with T15 = T14) for the upper zone; and hence, evidently, to the calculation of efficiencies, EFF and EFF43.

Included alongside the above test is an early discharge from 70°C, of 6th August 1985, this time with the boiler pump left ON during the test. Thus store water was circulated and the store was no longer composed of two quiescent zones. In fact, as shown on the lower graph of Fig. 3.9, T10 was maintained only slightly below T11 - T14, the spread of the entire profile being less than 3°C for most of the test. The effect of this was that the temperature out of the lower h.e. (between 3WV's) was maintained higher than for the quiescent tank, so enabling the delivery d.h.w. temperature to be closer to the bulk store temperature. A more effective discharge was thereby produced. This is reflected in the numerical analysis of Table 3.2 where the efficiencies of the pumped store are rather higher.

It must be pointed out though, that the two tests are not directly comparable numerically, as the pumped test was conducted using the

early cold d.h.w. water feed, incorporating the controlled-temperature settling tank. The lower feed pressure of this early system gave a d.h.w. flow that stabilised at only 0.10 l/s, whereas the system of 16.8.85 gave 0.13 l/s (having discarded the cold settling tank). The store mixing and low d.h.w. flow rate here will have falsely enhanced heat transfer coefficient. So a performance comparison must be only tentative. It does suggest however, that a more effective transfer of store heat is possible with the boiler pump running, although this is at the expense of high grade heat for the radiator circuit.

The graphs do show the preferable cold d.h.w. feed of the system of 16.8.85; this was both at a steadier temperature and at a lower temperature ( $14.5^{\circ}\text{C}$  against 18 or  $19^{\circ}\text{C}$ ).

[ Quite apart from the subject of this section (store profile evolution), it is worth noting the following facts at this stage:-

From the 16.8.85 graph,

- (i) the temperature curves are smooth and well-defined, the least so being that of the delivery temperature which fluctuates due to 3WV action, and
- (ii) a d.h.w. temperature close to  $60^{\circ}\text{C}$  is maintained only for about 5 minutes of this discharge from  $70^{\circ}\text{C}$ .

From the 16.8.85 analysis:

- (i) the upper zone remains over  $10^{\circ}\text{C}$  higher than  $43.3^{\circ}\text{C}$  at the end of the discharge,
- (ii) the store heat loss, QPASSED, evaluated solely from T10 - T14, ought to be slightly more than SEWEND (the heat passed to the d.h.w.) by an amount equal to the jacket heat loss during the test. The author was pleased with the low values of the errors calculated during these and subsequent analyses, and took the substantial agreement in QPASSED and SEWEND to indicate the integrity of both energy assessment techniques, i.e. turbine flowmeter with (T5 - T1) for SEWEND, and T10 to T14 and the two-zone model for QPASSED.

(iii) for this test some two-thirds of theoretically available energy was conveyed to d.h.w. (EFF43). ]

N.B. For clarity: for this above test of 16.8.85, and for all subsequent tests reported in this part of the thesis, (i) the boiler pump was OFF during discharge (tank quiescent), and (ii) the cold water settling tank was discarded, and the laboratory supply fed directly into the eye of the Ward pump.

## 2. Accuracy and Repeatability

Early in the testing, the MEDIUM/MEDIUM h.e. configuration was tested twice in discharges from 80°C, 70°C and 60°C. The relevant graphs are nos 3-5 and nos 18-20 of Appx. 3.2. Their numerical analyses are presented by start temperatures in Table 3.3:-

Title	AUG15A	AUG14M	AUG15B	AUG16B	AUG15C	AUG15C1
Graph no./Temp.	3/80°C	18/80°C	4/70°C	19/70°C	5/60°C	20/60°C
(i) TIMEND, min	17.43	17.32	13.71	13.16	7.94	8.22
(ii) SFLOWEND, l	145.54	141.23	110.26	107.03	62.24	64.99
(iii) TMEAN, °C	54.22	53.66	51.54	51.68	48.52	48.28
(iv) TSTART, °C	80.20	80.03	69.85	69.85	60.08	60.07
(v) TENDZ2, °C	54.95	55.12	53.48	53.73	53.03	52.41
(vi) TENDZ1, °C	27.93	28.56	29.71	30.00	35.38	35.18
(vii) QPASSED, kwh	6.915	6.819	4.794	4.736	2.433	2.545
(viii) SEWEND, kwh	6.695	6.372	4.716	4.608	2.437	2.534
(ix) ERROR, %	3.28	7.01	1.64	2.78	-0.15	0.43
(x) EFF, %	47.33	47.33	39.36	38.81	24.33	25.40
(xi) EFF43, %	72.25	71.59	65.85	65.00	47.53	49.71
(xii) TOTENAV, kwh	14.487	14.406	12.179	12.202	10.000	10.019
(xiii) TOTENEND, kwh	7.572	7.588	7.386	7.466	7.568	7.474
(xiv) TENIDEAL, kwh	4.916	4.882	4.899	4.916	4.882	4.899
(xv) Unused, kwh	2.656	2.706	2.487	2.550	2.686	2.575

Table 3.3 : Two tests from each of 80°C, 70°C and 60°C were undertaken in order to check the repeatability.



Each pair of tests should theoretically, of course, be identical. The extent of the difference between the various parameters of each test is an indication of the reliability of the test process - these differences, either as simple numerical differences or as percentages, see below, are presented in Table 3.4:-

	80°	70°	60°
(i) TIMEND %	0.6	4.0	3.4
(ii) SFLOWEND %	3.0	2.9	4.2
(iii) TMEAN °C	0.56	0.14	0.24
(iv) TSTART °C	0.17	0.00	0.01
(v) TENDZ2 °C	0.17	0.25	0.62
(vi) TENDZ1 °C	0.63	0.29	0.20
(vii) QPASSED %	1.4	1.2	4.4
(viii) SEWEND %	4.8	2.3	3.8
(ix) ERROR	-	-	-
(x) EFF % points	0.40	0.55	1.07
(xi) EFF43 % points	0.66	0.85	2.18
(xii) TOTENAV kWh	0.089	0.023	0.019
(xiii) TOTENEND kWh	0.016	0.080	0.094
(xiv) TENIDEAL kWh	0.034	0.017	0.017
(xv) Unused Energy kWh	0.050	0.063	0.111

Table 3.4 : Differences between equivalent parameters for the pairs of tests of Table 3.3.

Table 3.4 requires rather careful interpretation; each feature will be examined individually. In general, the smaller the values of Table 3.4 are, the more accurate and repeatable the test process may be considered to be. A value of zero is of course regarded as a perfect repeat.

Temperature variations on Table 3.4 are presented in simple arithmetical differences in °C (percentages being of little meaning). These are rows (iii) - (vi) of Table 3.4. TSTART is very nearly reproduceable test-to-test by careful mixing of the store just prior

to discharge; thus row (iv) of Table 3.4 shows very slight errors, which might possibly in all tests be reduced to less than  $0.1^{\circ}\text{C}$ . The end of test store zone temperatures TENDZ2 and TENDZ1 show fair agreement. These differences are of course a function of all the variables called into play during a discharge test.

Rows (i), (ii), (vii) and (viii) of Table 3.4 are variables calculated by integration over the span of a test, and their differences are presented as percentages; thus the two values of TIMEND for the  $80^{\circ}\text{C}$  tests were 17.43 mins and 17.32 mins, with a difference of 0.6% on Table 3.4. They are seen to repeat one another to within 5% in all cases. Just discernible here is the more accurate repeatability of the longer  $80^{\circ}\text{C}$  tests (c.f.  $60^{\circ}\text{C}$  tests) where the effect of end-errors in the derived parameter integrations is reduced.

Efficiency value variations of rows (x) and (xi) of Table 3.4 are presented as simple differences between the percentages of Table 3.3, expressed in "percentage points". Again, the accuracy of repeatability is reduced for the shorter tests. These differences are to be taken as the order of confidence which can be placed in any quoted efficiency value, i.e. Table 3.4 suggests that "whole number" efficiency estimates might be realistic.

Store energy parameters in rows (xii) to (xv) of Table 3.3 are all calculated to a base of cold inlet temperature ( $T_1$ ), and differences in Table 3.4 are expressed as simple arithmetic energy differences, so that the variations of all four parameters may be compared directly. The differences are usually well below 0.1 kWh, which, in quantities of say 5-10 kWh (Table 3.3), lead to repeatabilities ordinarily below some 2%.

The magnitude of a departure from zero on Table 3.4 is composed of two factors:-

- (i) errors in thermocouple and turbine readings, and the inaccuracies in the subsequent finite-difference techniques for their processing, and

(ii) inherent non-repeatability, which would still exist even if measurement etc. were perfect. This latter would be due to e.g. varying the cold-water inlet conditions (temperature, pressure), slight variations in the characteristic of the Ward pump, slight variations in 3WV action, accumulation of impurities in the store water affecting heat exchange parameters, etc.

Of these, (i) could be improved by more rigorous monitoring, e.g. a second flow-measuring device, doubling-up of thermocouples, increased scanning rate, more refined computational techniques, etc. However, (ii) above would be more difficult to improve, and would require alterations to the rig to effect a tighter control on e.g. supply temperature. The issue here is to determine whether the repeatability of Table 3.3 is adequate for the object of the tests, namely to compare the h.e. configurations available. If it were not adequate, refinements to the measurements and to the rig would be required; in fact the author is persuaded that the accuracy available is sufficient to satisfy the test objective. In other words, the difference between the respective performances of h.e. combinations should be sufficiently marked to allow a comparison of these combinations to be altogether feasible, notwithstanding the test-for-test repeatability variations recorded in Table 3.3. The two tests of Tables 3.3 and 3.4 were early tests, where the test procedure (scanning rates, pre-discharge mixing) had not been perfected; the author believes that the repeatability studied here is likely to be pessimistic, and that the tests proper of the following section were conducted at least as accurately.

### 3. Heat Exchanger Comparison

This being the principal study of Part III of the thesis, a brief recap of the test schedule will be included here. Six d.h.w. heat exchangers (h.e.) were available; three pairs fabricated using 4 m (SMALL), 6 m (MEDIUM) and 8 m (LARGE) lengths of "Integron" material. These gave nine possible h.e. configurations, all of which were tested in turn. Each configuration was used for three discharge tests, to discharge the storage tank from starting temperatures of 80°C, 70°C



and 60°C. Thus 27 individual tests are recorded here; the graphical presentation of these is graphs nos. 6-32 of Appx. 3.2. They are arranged as follows from Table 3.1 above:-

Top h.e.	Bottom h.e.	Graph No. from start temp. of		
		80°	70°	60°
SMALL	SMALL	6	7	8
SMALL	MEDIUM	9	10	11
SMALL	LARGE	12	13	14
MEDIUM	SMALL	15	16	17
MEDIUM	MEDIUM	18	19	20
MEDIUM	LARGE	21	22	23
LARGE	SMALL	24	25	26
LARGE	MEDIUM	27	28	29
LARGE	LARGE	30	31	32

#### (i) Parametric Analysis

As before, the parametric analysis was carried out on each of these tests. The h.e. configurations are best compared by collecting the numerical analyses into three tables; Table 3.5 shows the tests started from a store temperature of 80°C, Table 3.6 those from 70°C and Table 3.7 those from 60°C. The parameters are listed in the same order as before, (i) to (xv).

Inspection of both the above list and Tables 3.5 - 3.7 shows that the order of presentation of the tests differs from chronological test order. During testing, the MEDIUM h.e. was fitted at the bottom, and the top h.e. changed through SMALL, MEDIUM and LARGE, as it was far simpler to change the top h.e. than the lower. Subsequently, the same changes of the top h.e. were carried out with the SMALL h.e. at the bottom, then with the LARGE h.e. at the bottom. In the following graphical analysis presented here, it was found preferable to "invert" the permutation of h.e.'s as the size of the top h.e. was the more significant because it exchanged heat with a much larger zone of the store than the lower h.e. Thus more continuous graphs were produced by arranging the h.e. configurations as follows:-

H.e. combination (top/bottom)		SM/SM	SM/MED	SM/L	MED/SM	MED/MED	MED/L	L/SM	L/MED	L/L
Test Title		SEP05A	AUG22A	SEP20A	SEP09A	AUG14M	SEP17A	SEP10A	SEP02	SEP16A
(i)	Mins to 43.3°C, TIMEND	12.242	14.717	15.930	15.792	17.317	18.892	18.103	18.433	20.755
(ii)	Litres d.h.w., SFLOWEND	118.981	129.974	137.176	145.654	141.228	159.168	160.940	158.555	171.630
(iii)	Mean temp. °C, TMEAN	52.63	53.26	53.49	53.39	53.66	54.25	54.30	55.11	53.67
(iv)	Store temp. start °C, TSTART	80.05	79.90	80.02	79.95	80.03	79.90	80.03	80.02	79.93
(v)	Upper zone end °C, TENDZ2	61.78	59.83	59.23	53.86	55.12	53.10	49.88	50.80	49.73
(vi)	Lower zone end °C, TENDZ1	35.35	30.07	25.44	32.42	28.56	24.41	30.59	29.10	23.87
(vii)	Store heat loss, kWh, QPASSED	5.345	5.906	6.264	6.824	6.819	7.339	7.615	7.527	7.951
(viii)	Heat to d.h.w., kWh, SEWEND	5.295	5.834	6.227	6.577	6.372	7.309	7.438	7.366	7.806
(ix)	Error %, ERROR	0.93	1.24	0.60	3.75	7.01	0.41	2.38	2.19	1.85
(x)	Efficiency to T1, EFF	36.81	40.96	43.23	47.21	47.33	50.97	52.62	52.51	55.02
(xi)	Efficiency to 43.3/T1, EFF43	55.95	62.14	65.66	71.67	71.59	77.26	79.83	79.19	83.54
(xii)	Store energy start, kWh, TOTENAV	14.520	14.421	14.491	14.454	14.406	14.399	14.473	14.336	14.450
(xiii)	Store energy end, kWh, TOTENEND	9.176	8.515	8.227	7.630	7.588	7.060	6.857	6.809	6.500
(xiv)	Store energy ideal, kWh, TENIDEAL	4.968	4.916	4.950	4.933	4.882	4.899	4.933	4.830	4.933
(xv)	Unused store energy, kWh	4.208	3.599	3.277	2.697	2.706	2.161	1.924	1.979	1.567

Table 3.5: Results of discharges from start temperature of 80°C

H.e. combination (top/bottom)	SM/SM	SM/MED	SM/L	MED/SM	MED/MED	MED/L	L/SM	L/MED	L/L
Test title	SEP05B1	AUG23B	SEP20B	SEP06B	AUG16B	SEP 18B	SEP10B	AUG30B	SEP17B
(i) Mins to 43.3°C, TIMEND	8.500	10.000	11.075	12.142	13.158	14.350	13.476	14.769	15.800
(ii) Litres d.h.w., SFLOWEND	79.496	86.947	93.511	106.785	107.028	116.943	116.307	121.856	125.781
(iii) Mean temp. °C, TMEAN	49.56	51.42	51.43	51.90	51.68	52.49	52.42	52.84	52.69
(iv) Store temp. start °C, TSTART	69.97	69.95	69.97	70.02	69.85	69.83	70.12	69.95	70.02
(v) Upper zone end °C, TENDZ2	59.86	58.84	58.61	52.67	53.73	52.51	50.36	49.38	49.79
(vi) Lower zone end °C, TENDZ1	39.00	34.54	29.30	34.30	30.00	26.28	33.42	30.10	26.23
(vii) Store heat loss, kWh, QPASSED	3.266	3.660	3.960	4.741	4.736	5.126	5.203	5.498	5.634
(viii) Heat to d.h.w., kWh, SEWEND	3.246	3.707	4.025	4.613	4.608	5.145	5.107	5.325	5.549
(ix) Error %, ERROR	0.61	-1.27	-1.61	2.79	2.78	-0.36	1.87	3.25	1.53
(x) Efficiency to T1, EFF	26.61	29.99	32.27	38.81	38.81	42.02	42.43	45.47	46.12
(xi) Efficiency to 43.3/T1, EFF43	44.60	50.11	54.09	64.80	65.00	70.40	70.84	75.55	77.00
(xii) Store energy start, kWh, TOTENAV	12.271	12.202	12.271	12.216	12.202	12.198	12.260	12.091	12.216
(xiii) Store energy end, kWh, TOTENEND	9.006	8.542	8.311	7.476	7.466	7.072	7.058	6.593	6.582
(xiv) Store energy ideal, kWh, TENIDEAL	4.950	4.899	4.950	4.899	4.916	4.916	4.916	4.813	4.899
(xv) Unused store energy, kWh	4.056	3.643	3.361	2.577	2.550	2.156	2.142	1.780	1.683

Table 3.6 : Results of discharges from start temperature of 70°C



H.e. combination (top/bottom)	SM/SM	SM/MED	SM/L	MED/SM	MED/MED	MED/L	L/SM	L/MED	L/L
Test Title	SEP05C	AUG22C	SEP 20C	SEP06C	AUG15C1	SEP17C	SEP10C	AUG30C	SEP16C
(i) Mins to 43.3°C, TIMEND	3.350	4.995	5.809	7.158	8.220	9.308	8.950	10.050	10.685
(ii) Litres d.h.w., SFLOWEND	30.325	42.682	48.073	60.917	64.989	72.793	74.412	78.975	80.754
(iii) Mean temp. °C, TMEAN	46.21	47.68	48.49	47.76	48.28	49.43	48.67	49.86	50.03
(iv) Store temp. start °C, TSTART	60.05	60.03	60.07	59.95	60.07	59.87	59.98	59.97	59.93
(v) Upper zone end °C, TENDZ2	57.18	56.45	56.15	52.38	52.41	51.54	49.32	49.23	49.22
(vi) Lower zone end, °C, TENDZ1	45.90	39.89	35.68	38.67	35.18	30.47	36.40	33.13	29.40
(vii) Store heat loss, kWh, QPASSED	1.193	1.613	1.878	2.351	2.545	2.882	2.995	3.169	3.348
(viii) Heat to d.h.w., kWh, SEWEND	1.117	1.634	1.897	2.338	2.534	2.927	2.943	3.179	3.322
(ix) Error %, ERROR	6.83	-1.32	-0.98	0.57	0.43	-1.52	1.75	-0.30	0.77
(x) Efficiency to T1, EFF	11.86	16.11	18.67	23.53	25.40	28.96	29.88	32.06	33.44
(xi) Efficiency to 43.3/T1, EFF43	23.27	31.55	36.62	46.16	49.71	56.84	58.65	62.47	65.70
(xii) Store energy start, kWh, TOTENAV	10.059	10.012	10.063	9.993	10.019	9.953	10.023	9.886	10.012
(xiii) Store energy end, kWh, TOTENEND	8.867	8.399	8.185	7.642	7.474	7.070	7.027	6.717	6.664
(xiv) Store energy ideal, kWh, TENIDEAL	4.933	4.899	4.933	4.899	4.899	4.882	4.916	4.813	4.916
(xv) Unused store energy, kWh	3.934	3.500	3.252	2.743	2.575	2.188	2.111	1.904	1.748

Table 3.7: Results of discharges from start temperature of 60°C

Top h.e.	SMALL	SMALL	SMALL	MEDIUM	
Bottom h.e.	SMALL	MEDIUM	LARGE	SMALL	etc

Hence the preferred order of the above list and Tables 3.5 - 3.7.

The nine h.e. combinations have been arranged from the smallest to the largest, left-to-right, on Tables 3.5 - 3.7. Thus they may be compared, in terms of any of the 15 derived parameters, by considering the relevant row, again left to right. For example, referring to Table 3.5, it will be seen that the duration (TIMEND) of an 80°C discharge varies from 12.242 minutes for the SMALL/SMALL combination, up to 20.755 minutes for LARGE/LARGE. For most parameters of Tables 3.5 - 3.7, glancing along a row from left-to-right will show a progression of values, although (i) certain rows are either variations on an intended constant value (e.g. TSTART, TOTENAV) or random values (ERROR); (ii) there are exceptions; and (iii) the values along a row should in no way be expected to yield a continuous curve, but rather a set of discrete values describing a progression only by dint of the chosen order of presentation.

Before converting these parametric tables into a graphical format, it is worth emphasizing the following features of Tables 3.5 - 3.7 making similar observations to section 2 above: TSTART was very close indeed to the desired charged store temperature, usually being less than 0.1°C adrift. Thus the initial charge conditions could be repeated. ERROR, the discrepancy between QPASSED and SEWEND, is encouragingly small. ERROR represents an assessment of the integrity of the energy-flow measurement technique used in either case; QPASSED was calculated from estimates of store energy content from the temperature profile, thermocouples T10-T14, while SEWEND was an integral of the heat gained by the d.h.w. from the turbine flow rate and (T5-T1). The value of ERROR should be small and positive (the effect of store jacket loss). It is seen to be just this, in most cases, straying over 4% on just two occasions. TOTENAV and TENIDEAL were based on TSTART, but show rather more variation because their calculation included d.h.w. inlet temperature T1, which varied slightly day-to-day.

The parametric comparisons are perhaps easiest to visualise when plotted: nine of the fifteen will be presented and examined.

(ii) Graphical Presentation of Test Parameters

In each of the following graphical presentations of Tables 3.5 - 3.7:

- a single parameter is plotted on a given graph
- values of this parameter for all the initial store charge temperatures (i.e. 80° 70° and 60°C) are plotted on each graph
- the chosen order of h.e. configurations is maintained throughout, with one exception being the tenth of this series concerning unused store heat - see later. Again, the points of each plot are individual values placed alongside each other, and do not form continuous curves (the plots are equivalent to bar-charts, but such a representation would in certain cases be confusing).

(a) Figs 3.10 - 3.13 ; TIMEND, SFLOWEND, QPASSED, SEWEND: these parameters display very similar trends. As would be expected, the effect of the upper h.e. is dominant: with a LARGE upper h.e., the results are better (largely) than with a MEDIUM upper h.e., for which the results are better than those with a SMALL upper h.e. For a given upper h.e. the effect of the use of a SMALL, MEDIUM or LARGE lower h.e. is similar. Thus for the order of the h.e. configurations chosen for the horizontal axis of these graphs, there tends to be an improvement in performance from left to right, in terms of each of these parameters. On closer inspection, it is evident that more can be gained by increasing the upper h.e. from SMALL to MEDIUM than from MEDIUM to LARGE. The diminishing returns here suggest that the MEDIUM h.e. is well-sized for the upper h.e. for this storage tank. For the lower h.e., there is perhaps an indication that the advantage of changing from SMALL to MEDIUM is less than that of changing from MEDIUM to LARGE. These observations will be re-examined later.

Looking at the vertical spacing of the curves, the 80°C/70°C separation is generally nearly equal to the 70°C/60°C separation,



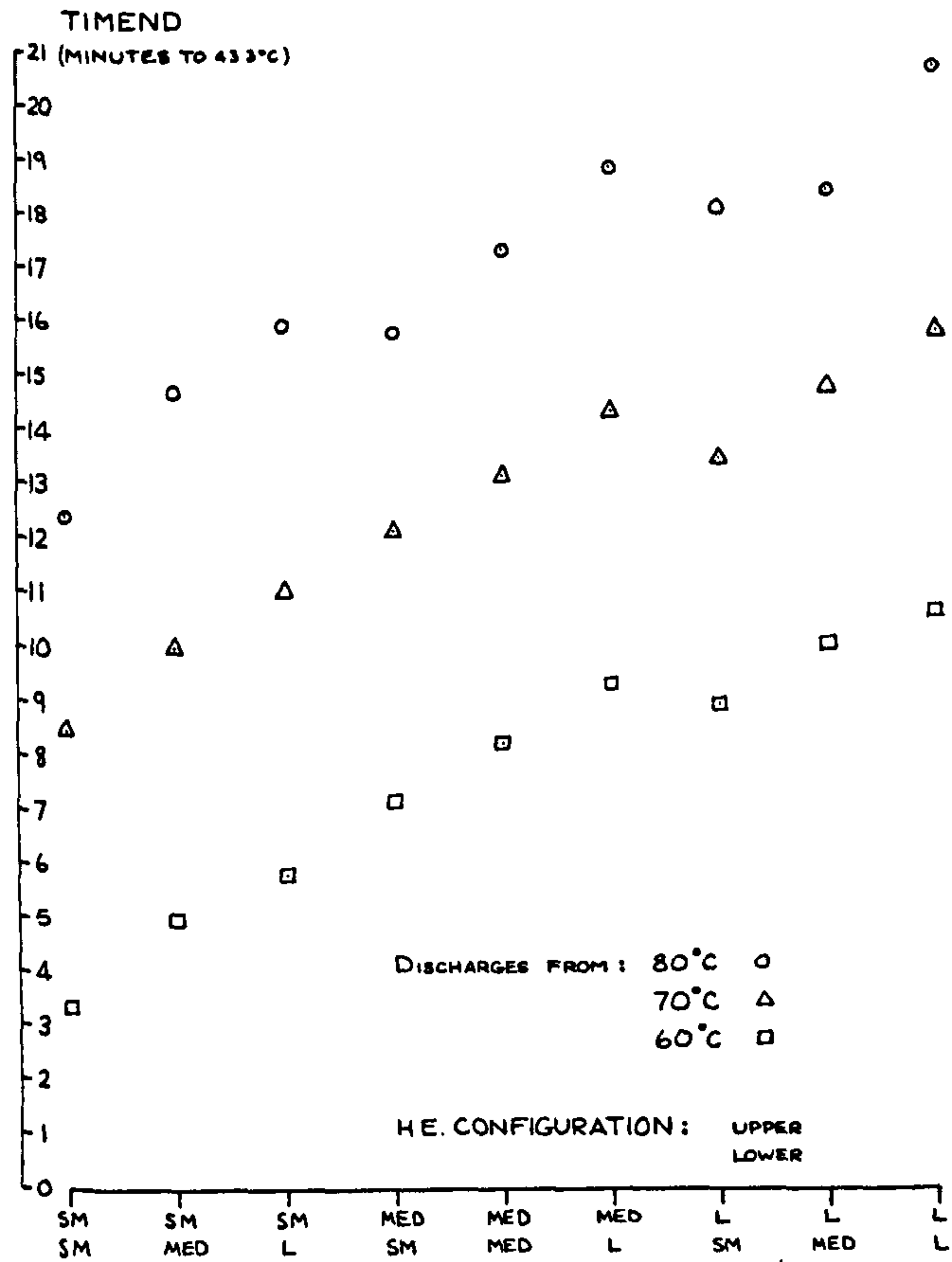


FIG 3 10: TEST DURATIONS

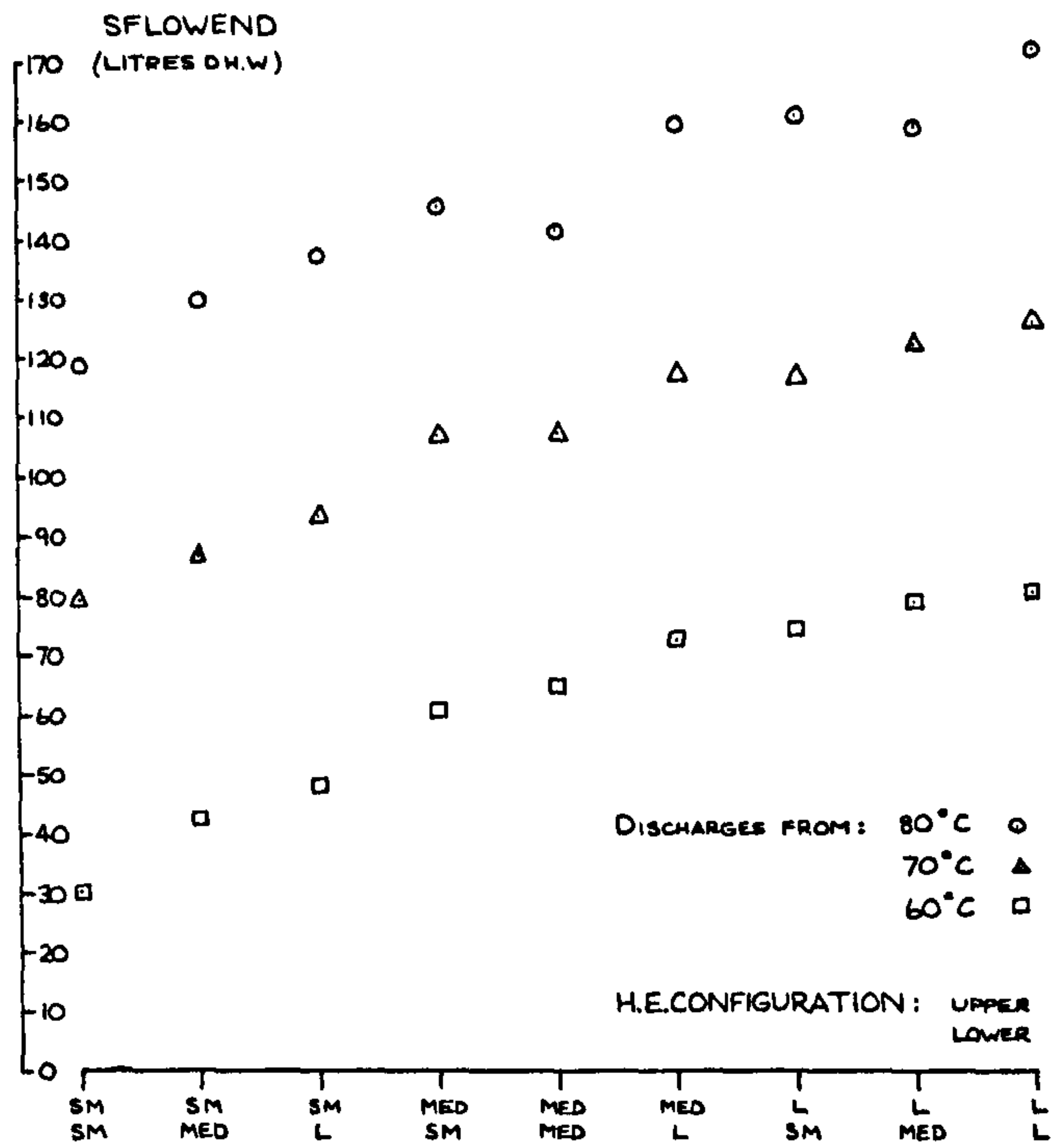


FIG 3 11: VOLUMES OF D.H.W DRAWN

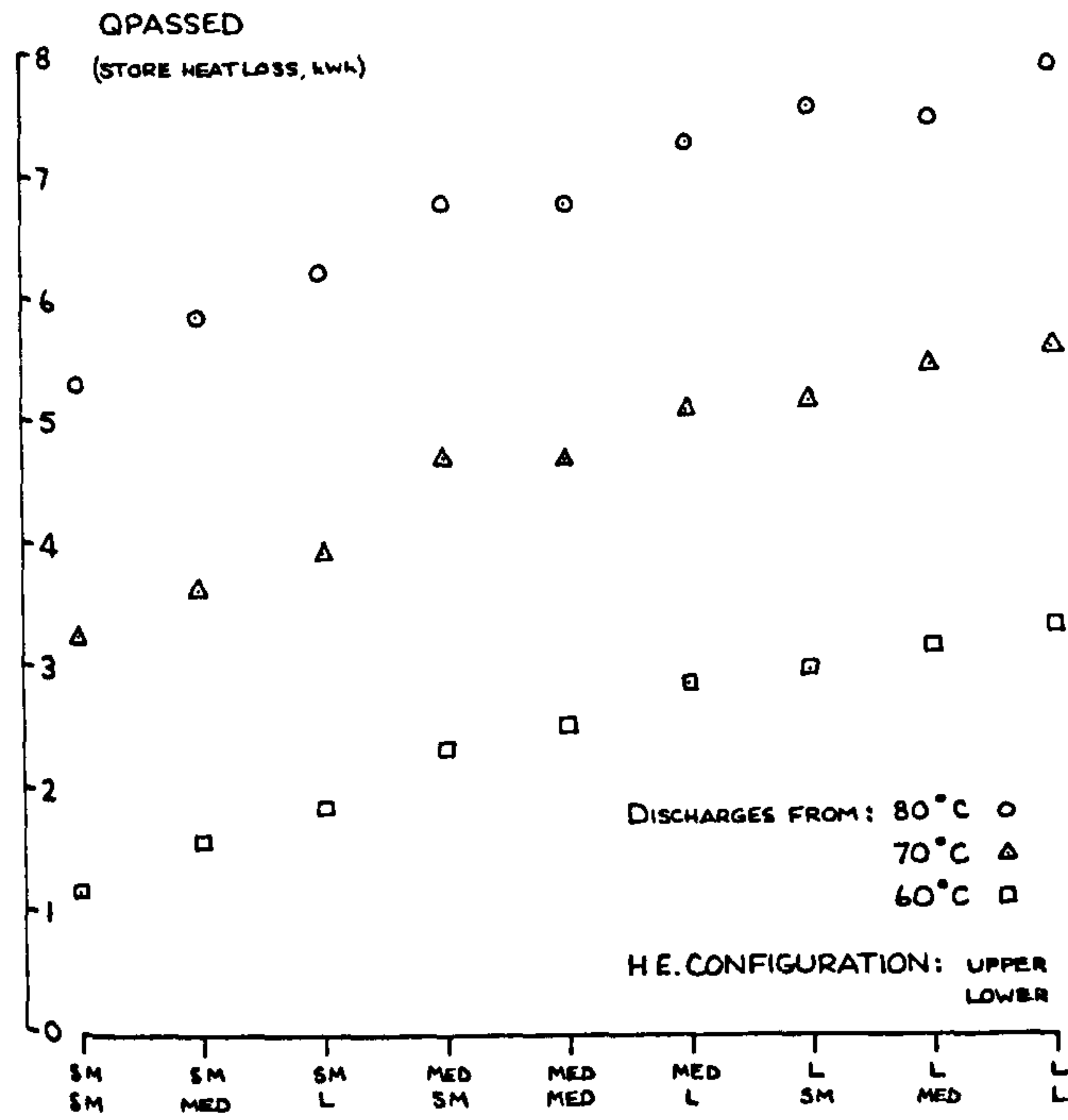


FIG 3 12: HEAT PASSED FROM STORE WATER

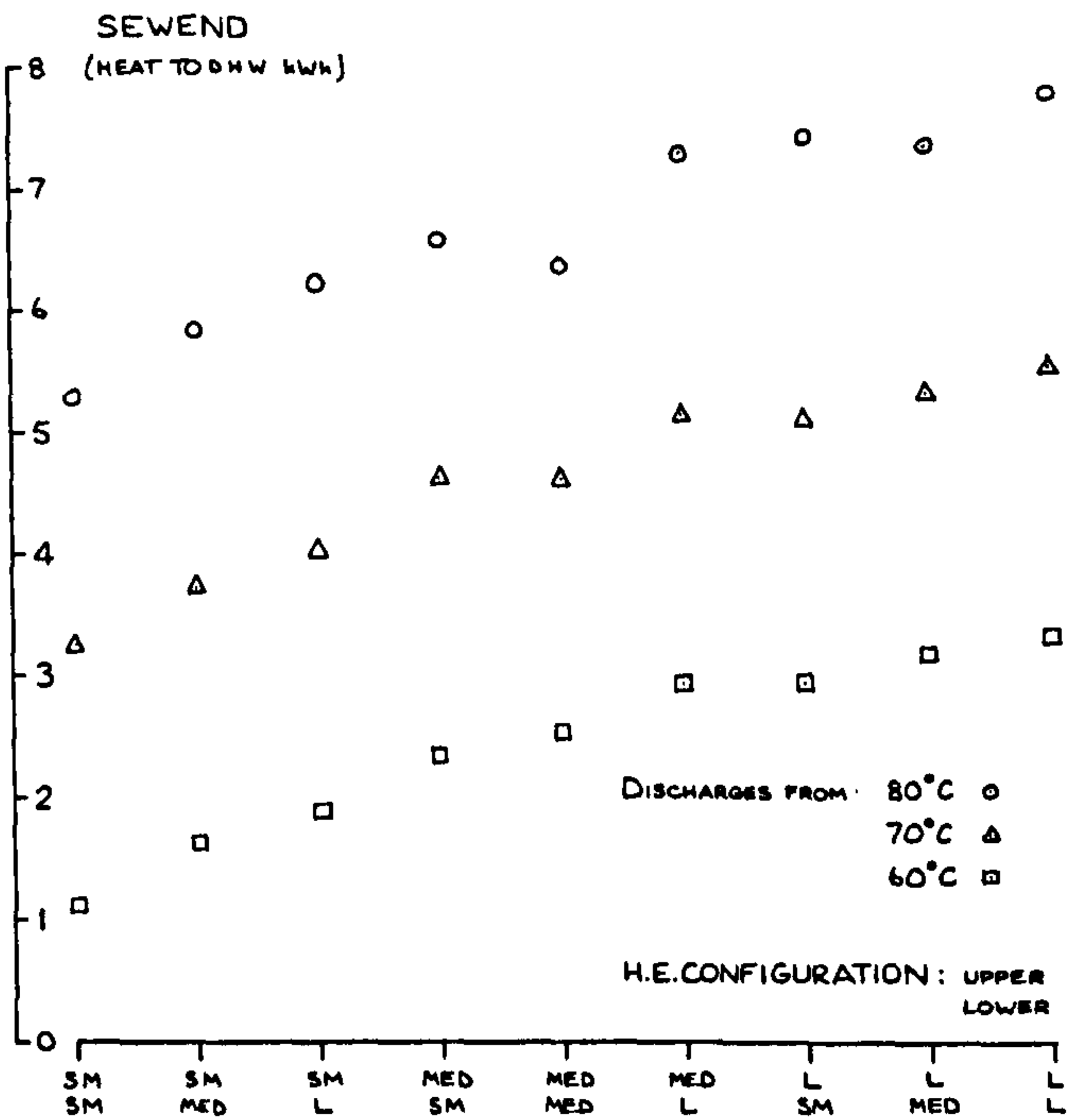


FIG 3 13: HEAT GAINED BY D.H.W

suggesting a linearity of performance with initial charge temperature.

(b) Figs 3.14 - 3.15; EFF and EFF43: these parameters, again, are only approximate, due to the nature of their derivation. They have been said to be trustworthy to perhaps 1%. They show the same trends as Figs 3.10 - 3.13, but being ratio-based, the inefficiency of a 60°C discharge is apparent. Also, the use of a small upper h.e. is shown here to be markedly inefficient, especially at lower initial charge temperatures. EFF43 represents the fraction of "available energy" extracted; it is thought that values over 70% and 80% for the larger h.e.'s from higher charge temperatures reflect on the suitability of this mode of d.h.w. heat extraction.

(c) Figs 3.16 - 3.17; TENDZ1, TENDZ2: these are the final zone temperatures after discharge, and of course reflect the opposite trend to the previous plots - this time, the lower the better. Starting with the simpler plot, that for TENDZ2, for the upper zone, the considerable advantage of the MEDIUM h.e. compared with the SMALL h.e. is evident. Again a smaller further improvement is produced by changing to a LARGE upper h.e. The higher efficiency of the LARGE upper h.e. is shown by the fact that TENDZ2 is here almost independent of initial store charge temperature.

Fig. 3.16 for TENDZ1 (lower zone) is more difficult to interpret. TENDZ1 is obviously more dependent on the lower h.e. size, so that the order for the configurations on the horizontal axis might usefully have been "inverted" (to SM/SM, MED/SM, L/SM, SM/MED. etc.) So triplets of points have been joined (though they do not lie on curves), and the triplets are seen to overlap. Also, 80°C points here lie below those for 70°C and 60°C. The large h.e. is seen to be very efficient in the lower position.

(d) Figs 3.18 - 3.19; unused store energy: these plots, perhaps the most illuminating of this series, show the heat remaining in the store after discharge, assuming that it might ideally have been discharged down to a dead state of 43.3° (upper zone) and T1 (lower zone). On Fig. 3.18 the expected trend is shown: the amount of unused energy falls as the h.e. capacity increases, from some 4.1 kwh

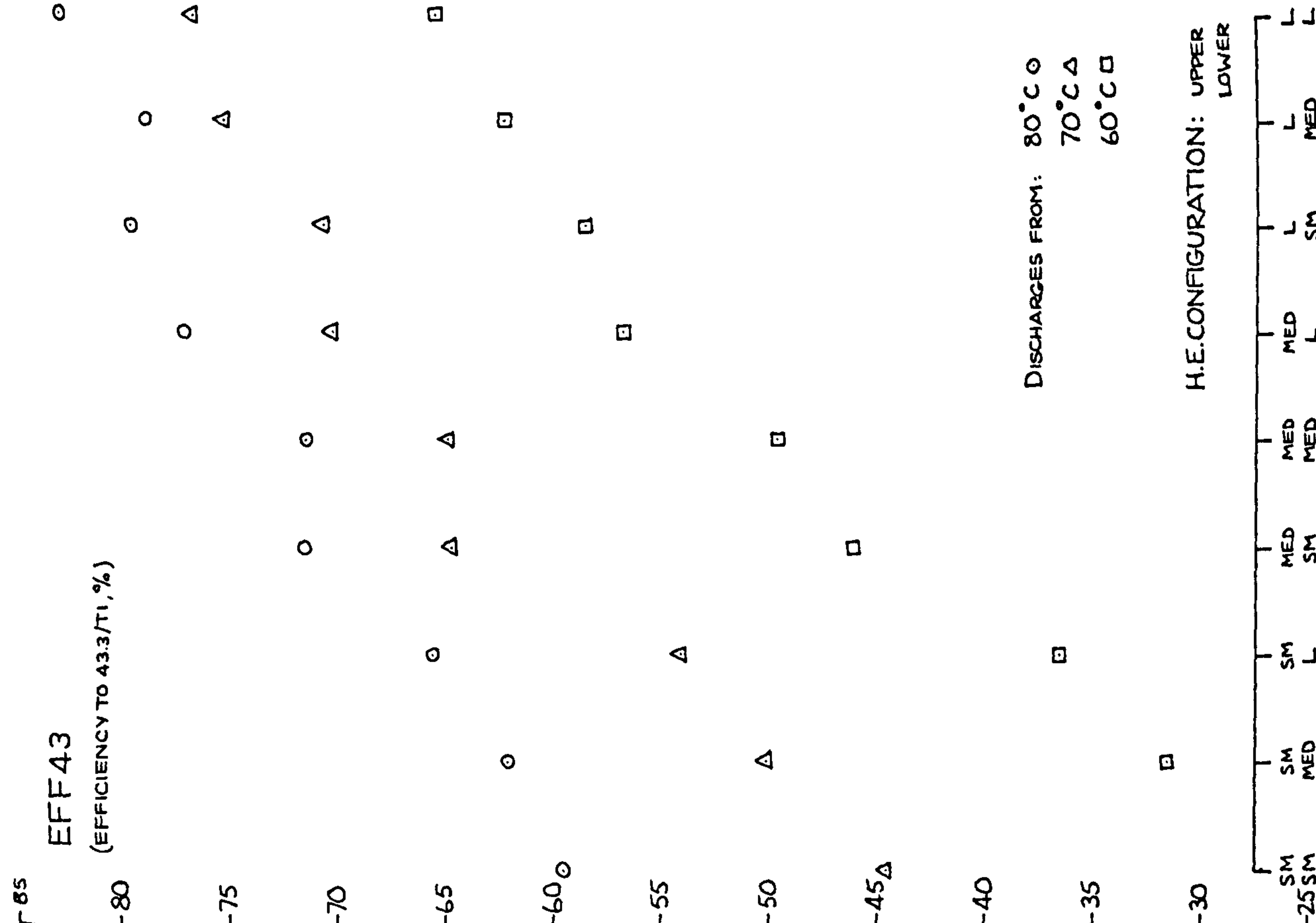


FIG 3.15: EFFICIENCY TO BASE OF IDEAL DEAD STATE

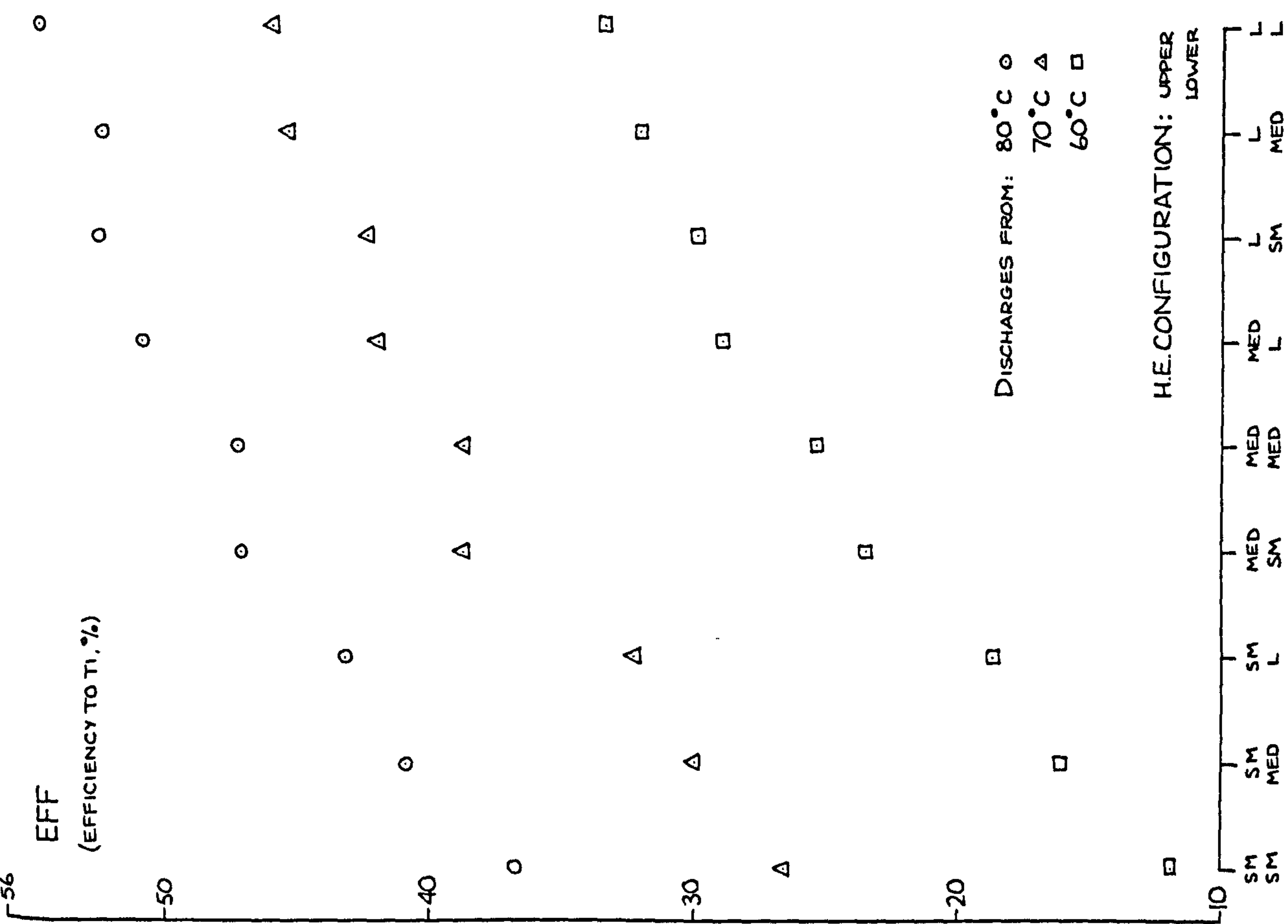


FIG 3.14: EFFICIENCY TO BASE OF COLD INLET TEMPE.



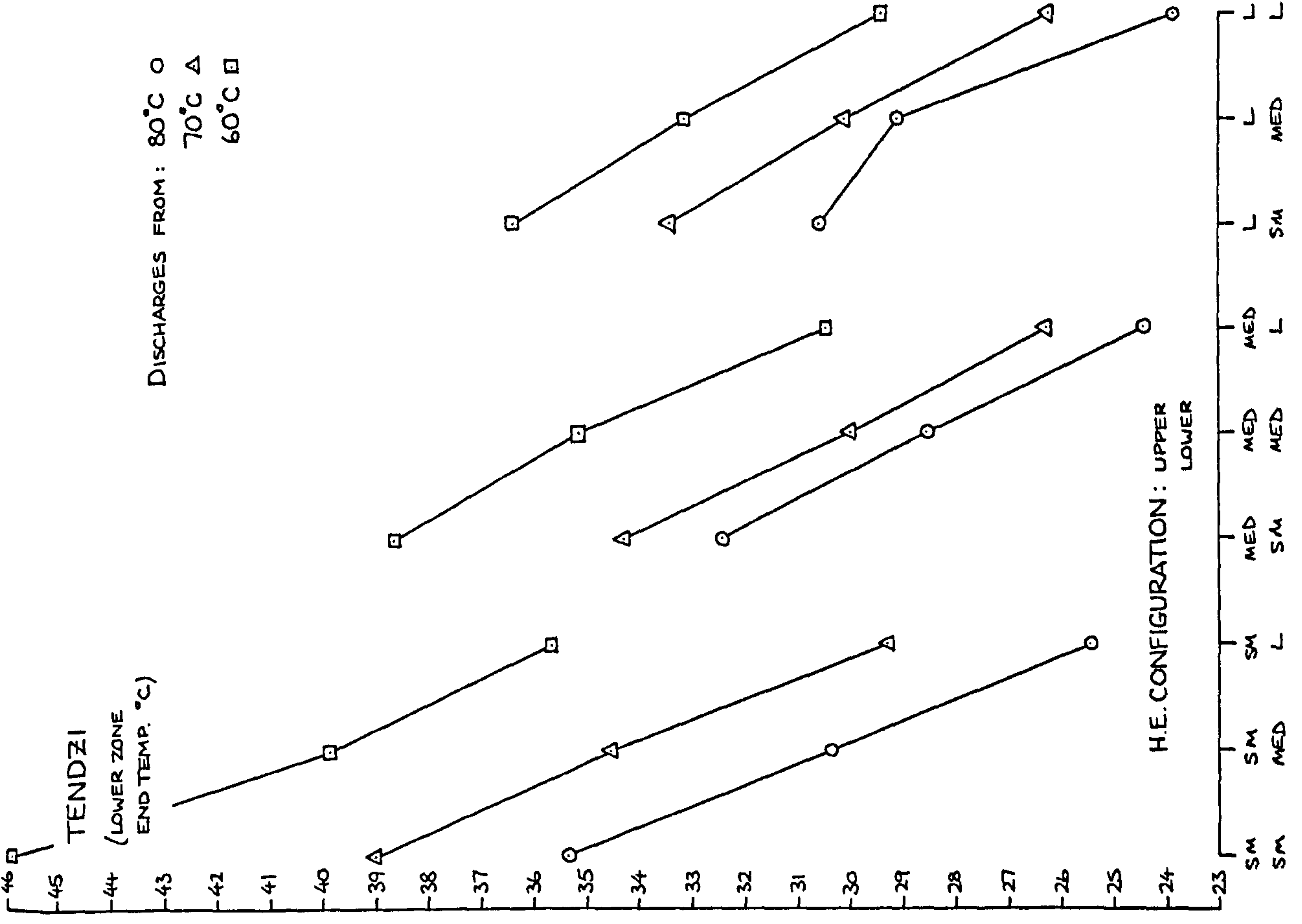


FIG 3.16: TEMP. OF LOWER ZONE AFTER DISCHARGES

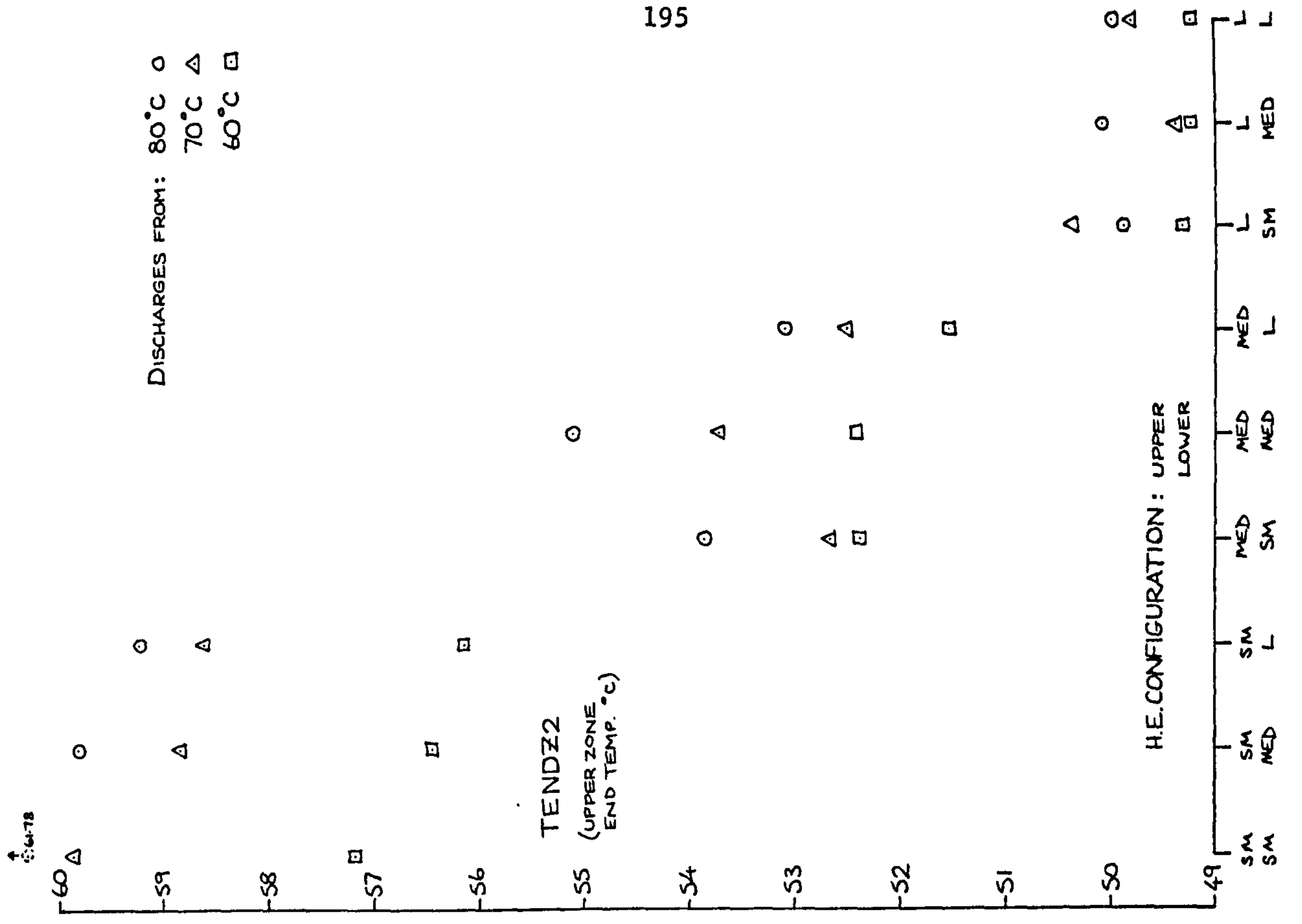


FIG 3.17: TEMP. OF UPPER ZONE AFTER DISCHARGES

for SMALL/SMALL, to through some 2.6 kWh for MEDIUM/MEDIUM, to some 1.7 kWh for the LARGE/LARGE system. The "diminishing returns" of increasing h.e. capacity is clearly shown here. In addition all but two of the h.e. configurations have a spread of unused energy values of less than 0.2 kWh, over the three tests. These show no particular ordering of the discharged states, i.e. for example the 80°C discharges do not appear to proceed to a lower discharge state than the 70° or 60° tests; this may be taken to argue that the discharge condition of the store is independent of the initial charge level over the span of these experiments. (There is, however, a very small dependence for the most inefficient h.e. configurations tested.)

Fig. 3.19 shows an attempt to reduce the points of Fig. 3.18 onto a single continuous curve. The average of the three values of unused energy from Fig. 3.18 for each h.e. configuration was taken. In order to represent each h.e. configuration by a numerical parameter, a "heat exchanger index" was devised as follows. The overall heat-transfer coefficient of a given h.e. was approximately proportional to its length (4m, 6m or 8m) as all the h.e.'s were fabricated from the same "Integron" tubing. Of the two h.e.'s of a given configuration, the upper was dominant, as it drew its heat principally from the upper store zone, which was of depth 121cm. The lower h.e. drew its heat from the lower zone, of depth 35 cm. Thus, as a crude notional approximation, the capacity of the upper h.e. ought to be 121/35 times more critical than that of the lower h.e. in terms of heat extraction. A figure, N, was thereby assigned to each h.e. configuration.

$$N = (\text{length of upper h.e.} \times 121/35) + \text{length of lower h.e.}$$

Each of the nine values of N was then divided by the value for the SMALL/SMALL configuration to give a value for the h.e. index:-

$$\text{H.e. index} = N/(\text{value of } N \text{ for SMALL/SMALL})$$

This gave the following values for the index:-





SMALL/SMALL	1.000
SMALL/MEDIUM	1.112
SMALL/LARGE	1.224
MEDIUM/SMALL	1.388
MEDIUM/MEDIUM	1.500
MEDIUM/LARGE	1.612
LARGE/SMALL	1.776
LARGE/MEDIUM	1.888
LARGE/LARGE	2.000

This slight weighting for each point produced Fig. 3.19, slightly smoother than Fig. 3.13, so that this time it was legitimate to draw a continuous curve. The same improvement in performance is displayed, diminishing as h.e. capacity increases. This curve might usefully be used to assess the likely performance of any other h.e. configurations compatible with this tank. For example, if the "Integron" manufacturers decided to supply their tubing in 5m lengths, instead of the now customary 6m lengths (see later), the possible performance of e.g. 2 x 5m h.e.'s could be predicted from Fig. 3.19.

Clearly, many other graphs could be plotted from the derived parameters of Tables 3.5 - 3.7, and indeed the tests could be represented by further parameters if desired. The author believes though, that these ten graphs characterise the tests adequately for a specified assessment of the h.e. performance to be carried out, according to the experimental objectives.

### (iii) Selection of Optimal H.E. Configuration

From the preceding plots, Figs. 3.10 - 3.19, it is evident that, from a theoretical point of view, the larger the size of the heat exchangers chosen, the better will be the performance of the d.h.w. heat-extraction system. However, before selecting an optimal configuration for the continuation of testing, it was necessary to consider certain practical aspects, in the context of industrial development.

First, the "Integron" tubing is available in lengths of 6m only.

Thus to fabricate a 4m h.e. (SMALL), 2m of tubing is wasted (though it might be possible to rework two such 2m lengths to produce a further 4m h.e.). 8m (LARGE) h.e.'s present a similar problem. It is clear that, in terms of cost/unit h.e. capacity, the 6m (MEDIUM) h.e.'s are likely to be the cheapest to produce (unless, as remarked above, the manufacturers should alter their marketing policy, and change either to say 5m, or make more than one length of tubing available commercially). Accepting this likely, current constraint, however, 6m (MEDIUM) h.e.'s are to be preferred from a production standpoint. Further, the MEDIUM/MEDIUM configuration is plainly greatly preferable from this point of view. So the selection of an optimal configuration was approached from the somewhat pre-judged opinion that MEDIUM/MEDIUM would be adopted, unless a different configuration proved significantly preferable in terms of heat take-off.

Secondly, British Gas Corporation (Watson House) indicated that it would be advantageous if the heat content of the quiescent tank was sufficient to provide two baths of d.h.w. It is reasonable to suppose that a domestic boiler would be able to charge the store to the 80°C of the longest discharges examined here. Current boiler thermostats are often set to the low 80's °C. Referring therefore to Fig. 3.12 and 3.13, and taking the heat requirement of one bath to be some 3 kWh, it seems necessary to rule out the use of the SMALL h.e. in the upper position.

The use of a LARGE h.e. in the upper position would have yielded an enhanced performance as discussed above. However, the MEDIUM h.e. was chosen because of its likely production cost advantage, together with its lower flow impedance. Thus the MEDIUM/MEDIUM combination was confirmed as a suitable operating selection, and used as the basis for the position variation of the succeeding tests, as described below.

#### 4. Lower Heat Exchanger Position

Having selected the MEDIUM/MEDIUM h.e. combination as a suitable, likely choice for this storage tank, a brief study was carried out to evaluate the effect of the vertical position of the lower h.e. The three discharge tests applied to each h.e. configuration previously

discussed (at 80<sup>o</sup>, 70<sup>o</sup> and 60<sup>o</sup>C) were repeated with the lower h.e. in each of three positions, giving the following depth of the lower zone:-

BOTTOM	35 cm (as before)
MIDDLE	48.5 cm
TOP	65 cm

To produce the MIDDLE position, the h.e. was inverted "port-to-port" using its asymmetry to give a lower zone depth as near as possible to the mean of the BOTTOM and TOP lower zone depths (calculated from the uppermost extent of the lower h.e. coils).

This investigation was of interest for the following reason:- Many d.h.w. draw-offs in real life require relatively small quantities of water and heat, e.g. sinkfulls for washing-of-hands, etc. The double 3WV system tested here allows (theoretically) for small draw-offs to be met solely by the lower h.e. in order to leave the upper zone's higher temperature undisturbed for the subsequent filling of the radiator system. Varying the size of the lower zone would allow an estimate to be made of the amount of energy available from the lower h.e. only, and the height of the lower h.e. could be chosen to suit the particular secondary d.h.w. requirement, if necessary.

As before, the graphical representation of these tests is presented in Appendix 3.2 this time as graphs nos. 33-44. Of these, graphs nos. 33-41 are to the familiar "RIGPLOT" format, and are from the nine discharge tests relating to the above three positions of the lower h.e. To examine the extent of the d.h.w. draw-off satisfied by the lower h.e. only, the 80<sup>o</sup>C discharges have been replotted to the "D.H.W. TEMPS." format for graphs nos. 42-44.

The customary parameter analysis was carried out on the results of each of these tests; these parametric representations are included as Table 3.8, with discharges from the same charge temperature placed alongside each other for ease of comparison.



Start Temperature °C	80			70			60		
	BOTTOM	MIDDLE	TOP	BOTTOM	MIDDLE	TOP	BOTTOM	MIDDLE	TOP
Lower h.e. position	SEP24A	SEP15A	SEP26A	SEP23B	SEP25B	SEP30B	SEP23C	SEP25C	SEP30C
Test Title									
(i) Mins to 43.3°C, TIMEND	17.950	18.239	18.311	14.317	13.914	14.156	8.033	8.917	9.050
(ii) Litres d.h.w., SFLOWEND	158.124	160.142	162.599	118.196	117.929	121.625	64.886	73.500	74.796
(iii) Mean temp. °C, TMEAN	53.40	54.56	54.73	51.82	52.23	52.28	48.99	48.87	48.74
(iv) Store temp. start °C, TSTART	79.78	80.02	79.97	69.77	70.02	70.02	60.03	60.02	60.00
(v) Upper zone end °C, TENDZ2	53.11	53.10	52.20	51.92	52.10	51.41	52.66	51.38	50.88
(vi) Lower zone end °C, TENDZ1	27.07	29.15	35.01	28.46	31.46	37.01	34.18	35.23	40.50
(vii) Store heat loss kWh, QPASSED	7.181	7.589	7.713	5.104	5.374	5.434	2.544	3.017	2.970
(viii) Heat to d.h.w. kWh, SEWEND	7.141	7.448	7.557	5.107	5.166	5.307	2.591	2.925	2.939
(ix) Error %, ERROR	0.56	1.89	2.07	-0.07	4.03	2.39	-1.81	3.14	1.05
(x) Efficiency to T1, EFF	49.81	52.45	53.52	41.89	43.83	44.48	25.35	30.08	29.82
(xi) Efficiency to 43.3/T1, EFF43	75.72	75.25	71.89	70.23	68.22	63.69	49.71	53.28	47.13
(xii) Store energy start kWh, TOTENAV	14.417	14.469	14.414	12.183	12.260	12.216	10.034	10.030	9.960
(xiii) Store energy end, kWh, TOTENEND	7.236	6.880	6.700	7.079	6.886	6.783	7.490	7.013	6.990
(xiv) Store energy ideal kWh, TENIDEAL	4.933	4.383	3.684	4.916	4.383	3.684	4.916	4.368	3.659
(xv) Unused store energy, kWh	2.303	2.497	3.016	2.163	2.503	3.099	2.574	2.645	3.331

Table 3.8: Results from discharges, using three positions of the lower h.e.

From Table 3.8, certain parameter trends can be seen and so the data have been plotted, as follows. (Rather less parameters than for the previous section have been plotted as the results are somewhat less conclusive.) The abscissa for all these plots is formed by the lower h.e. positions, these being three discrete positions. No continuity of a given parameter is attempted for this axis, so the points again do not describe continuous curves. The lines joining the points on most of the graphs are included merely to differentiate between like points (e.g. the two sets of points corresponding to QPASSED and SEWEND for  $80^{\circ}\text{C}$  discharges on Fig. 3.20).

Fig. 3.20 shows heat to d.h.w. and unused store heat. There is a slight increase in the rate of heat take-off as the lower h.e. is raised, as might be anticipated with the first d.h.w. heating phase (lower h.e.) being able to sustain a high temperature at delivery to the upper h.e. (T3) for longer into the discharge. Unused store heat is seen to rise significantly as the lower h.e. is raised, which at first sight is paradoxical. This derives from the fact that the definition of the "ideal dead state" after a discharge is the upper zone temperature (TENDZ2) to  $43.3^{\circ}\text{C}$ , lower zone temperature (TENDZ1) to T1 (cold feed), as on Fig. 3.23. Thus as the lower h.e. is raised, the lower zone volume is increased, and the theoretical available energy of the charged store is increased accordingly.

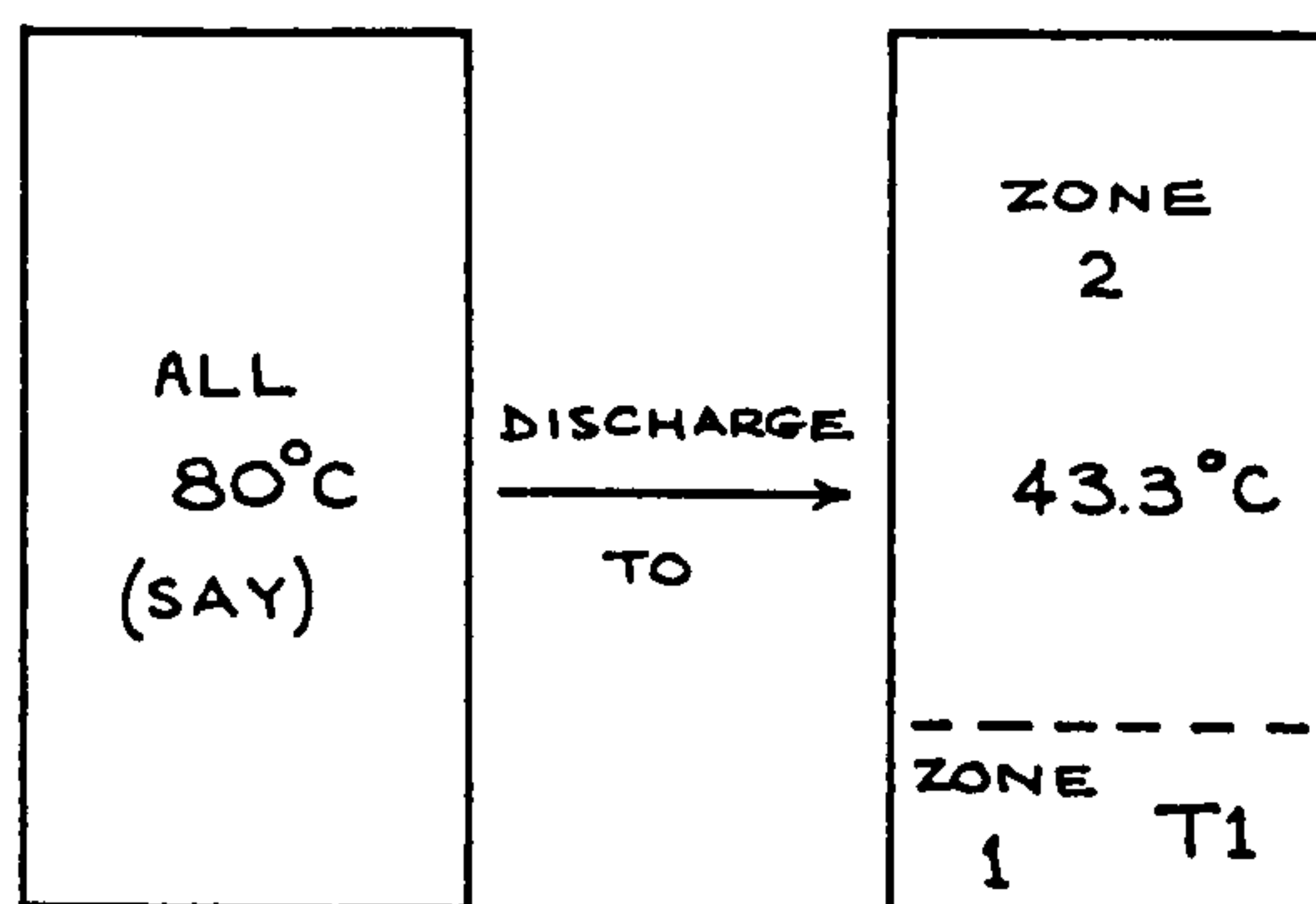


Fig. 3.23 : Ideal Dead State

Fig. 3.20 shows that, although the available energy is increased by moving the lower h.e. from "middle" to "top" say, only a small amount

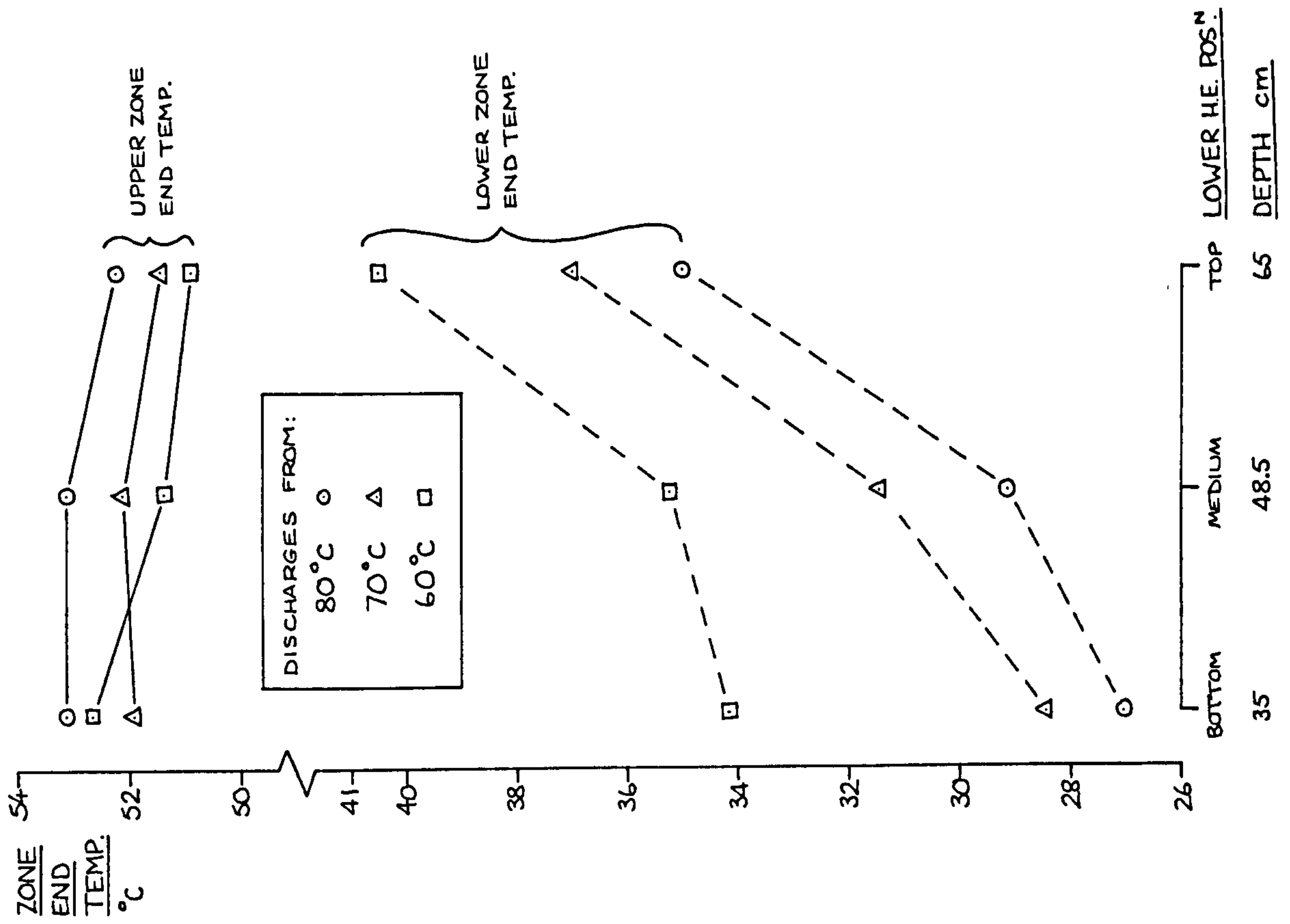


FIG 3.21: ZONE TEMPERATURES AFTER DISCHARGES

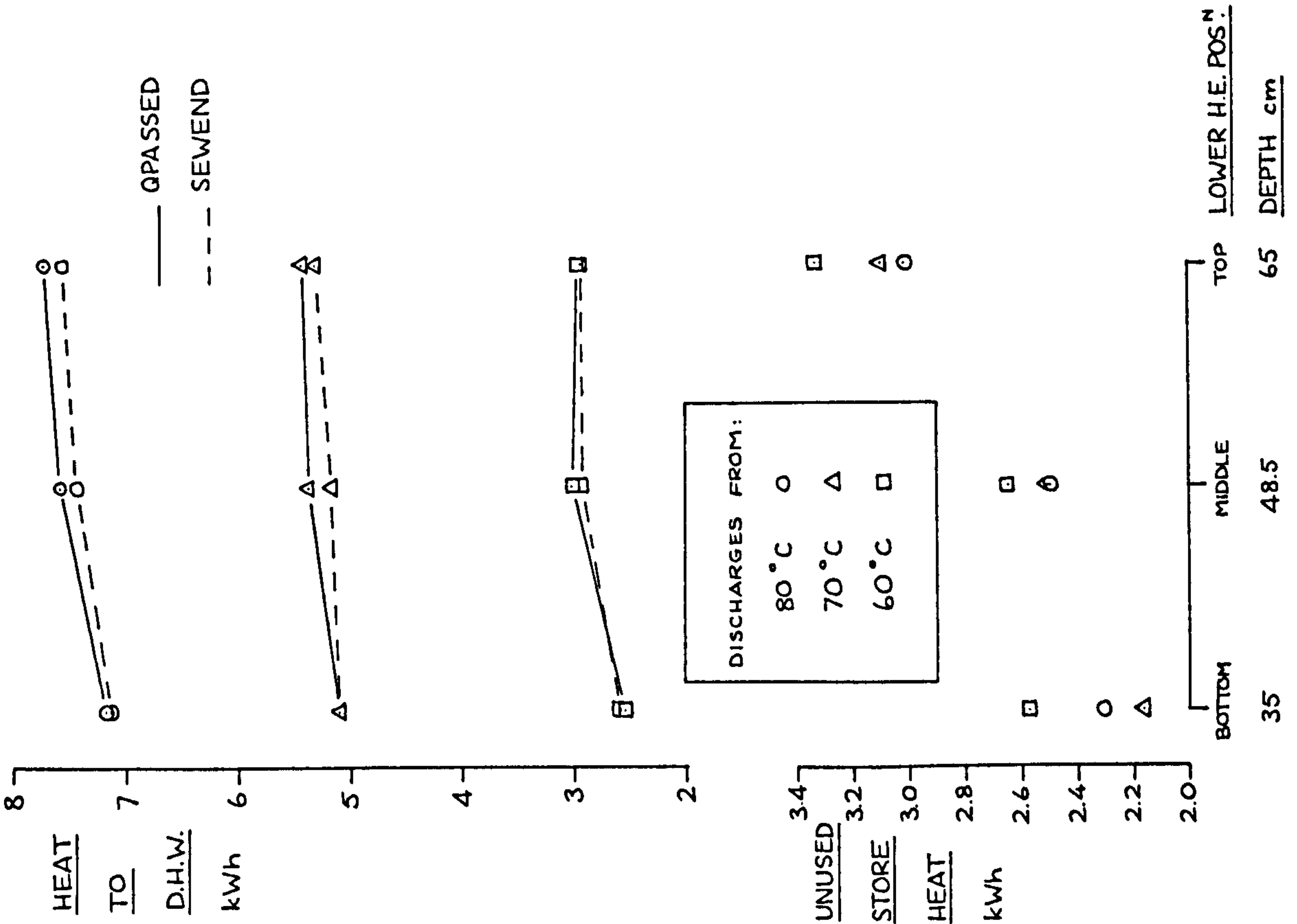


FIG 3.20: HEAT TRANSFERRED AND HEAT REMAINING



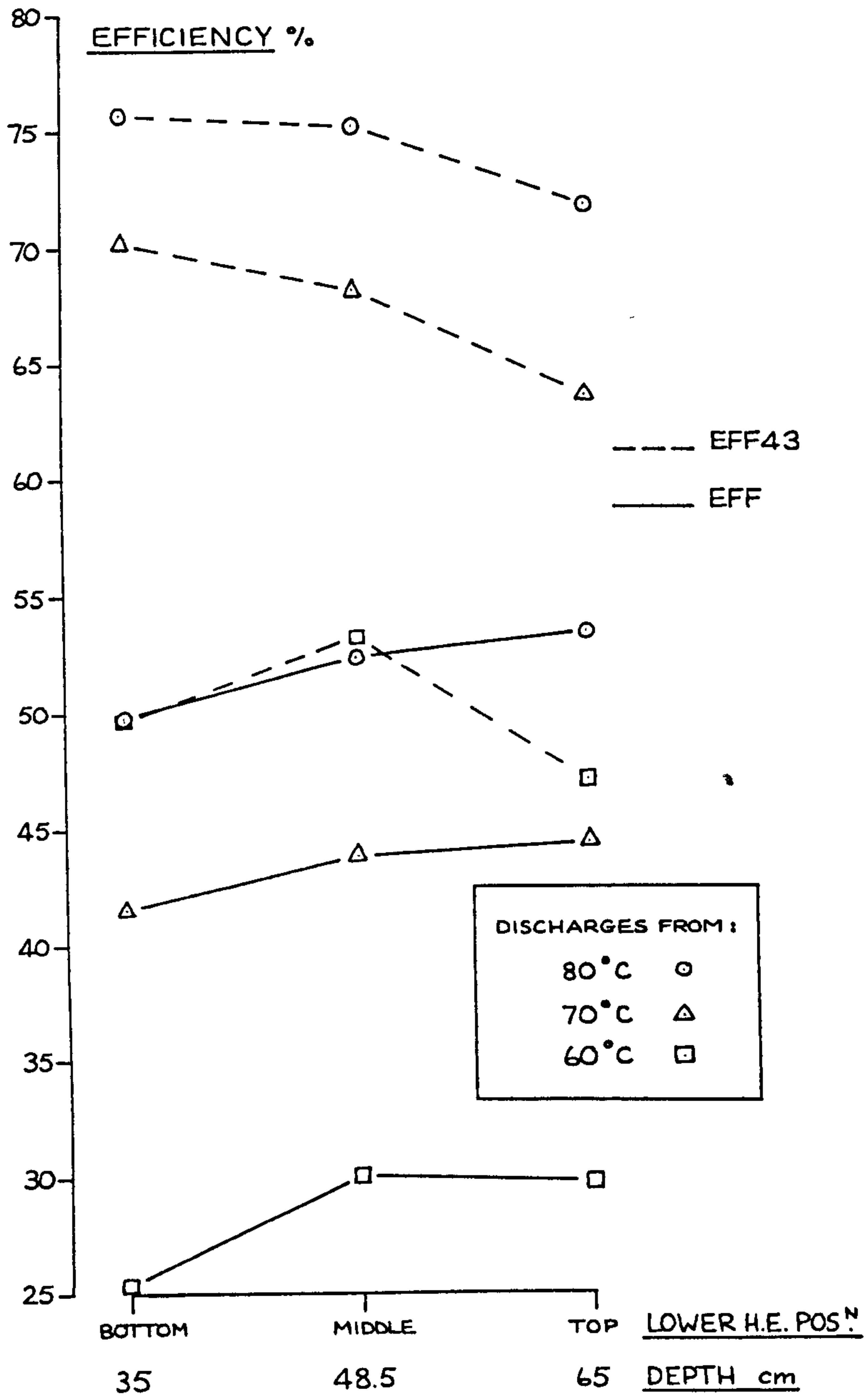


FIG 3.22: EFFICIENCIES

of this increase is in fact transferred to the d.h.w. take-off, and the remainder stays in the store to contribute to the unused store heat value (although the mean end temperature of the store will doubtless be reduced.) This effect is displayed in the "efficiency" plots below. Note also the fact that the unused store heat rises slightly when the lower h.e. is raised from "bottom" to "middle", but more significantly when it is raised from "middle" to "top".

Figure 3.21 shows the zone end temperature, after each discharge. The upper zone temperatures (TENDZ2) fall slightly as the lower h.e. is raised, by  $1^{\circ}\text{C}$  or so over the span of the tests, with little dependence on the initial charge temperature. However, the lower zone temperature (TENDZ1) is highly dependent both on the lower h.e. position (and thus zone volume) and on the initial charge temperature. This perhaps suggests that the MEDIUM h.e. is not of sufficient exchange capacity to realise the potential improvement made available by raising the lower h.e. It would have been interesting to test the MEDIUM/LARGE configuration with the LARGE h.e. in e.g. the middle lower position. This might be the first test to be conducted in a continuation of this investigation.

Figure 3.22 shows the "efficiencies" of these tests, EFF43 and EFF, whose variations with lower h.e. position are largely due to their respective definitions. EFF, based on a "dead state" of  $T_1$  throughout, is seen to rise slightly as the lower h.e. is raised. This corresponds with the rise in the heat to d.h.w. of Fig. 3.20. EFF43 is seen to fall, however, as the energy of the "ideal dead state" ( $43.3^{\circ}\text{C}/T_1$ ) falls off faster than the heat to d.h.w. increases. This is due to the same argument as that for the rise in unused store energy, Fig. 3.20 above. Thus raising the lower h.e. illustrates the "diminishing returns" axiom. in this sense.

Consider now the intended use of the lower h.e. alone to meet the requirements of the d.h.w. supply for small draw-offs. Graphs 42-44 and the relevant data show that the lower 3WV fully opens, and the emerging hot water temperature falls below the  $55^{\circ}\text{C}$  set for this 3WV, as follows;-

BOTTOM position; after ~ 1.2 minutes, or 12 litres or 0.6 kWh  
 MIDDLE position; after ~ 1.5 minutes, or 15 litres or 0.8 kWh  
 TOP position; after ~ 2.0 minutes, or 20 litres or 1.1 kWh

(approximate values at the point where T2 falls to about 1°C above T3, the delivery temperature to the upper h.e. and 3WV). At this point in the discharge, the lower zone has been depleted such that it can no longer maintain the temperature set on the lower 3WV, and the energy deficit must start to be supplied by the upper zone. Evidently, the above values would be influenced by an alteration in either of the 3WV settings, and would certainly be lower if it had been possible to set the lower 3WV setting exactly on the 60°C of the upper 3WV. But the improvement with increasing lower zone size is evident. Thus what we might call the "small draw" capacity of the installation is almost doubled by raising the lower h.e.; this capacity may be simply proportional to the depth of the lower zone.

In approaching the question of an optimal position of the lower h.e. we may summarise qualitatively the results of the above tests as follows:-

- (a) a high lower h.e. position gives - increased "small draw" capacity
  - slightly enhanced heat to d.h.w. before 43.3°C is reached, and therefore slightly higher "EFF";
- (b) a low lower h.e. position gives - larger upper zone untouched during "small draw"
  - slightly higher "EFF43".

Thus, as might have been anticipated, the respective interests of the radiator system and the d.h.w. system are somewhat opposed, and a compromise is necessary to suit the priorities of each particular case at the design stage.

Let us explore one possible design approach. A given house is to



be fitted with an appropriate boiler/store system. Estimates of its radiator water capacity and "small draw" d.h.w. requirement are available. If a variety of store sizes could be fitted, the design might be reasoned as follows. The upper zone would be sized such that its charged heat capacity could just raise the temperature of the entire radiator system from 15°C to 60°C say, using the customary 3WV at store outlet. The lower zone would be sized such that a MEDIUM h.e. supplied 1 kWh of sundry d.h.w. take-offs (small draw), assuming both d.h.w. 3WV's could be perfectly set, before the upper zone supplied any heat to d.h.w. The two zone sizes would give the required store size.

If, on the other hand, a given storage tank is to be used and this is below the size indicated by the above reasoning, then a compromise judgement as to the priority of radiator or d.h.w. system requirement must be made. The performance could be slightly enhanced by fitting a LARGE h.e. in the lower position, somewhat lower in the tank than the MEDIUM h.e. would be fitted.

This conjecture is clearly beyond the scope of this study. The author believes, however, that the results presented might form a useful basis for such a design procedure.

## CONCLUSIONS

The nature of this study has yielded conclusions which may be divided into two categories:- (i) those which are general observations of a largely qualitative nature, and (ii) those specific to the tank and heat exchangers used here and which are largely quantitative.

### (i) General Conclusions

1. It is possible to measure energy flows to an accuracy of better than 5% using a turbine and thermocouples, even though both water flow rate and temperature change continuously. Accuracies of better than 2% should be attainable with careful monitoring.

For the double h.e. concept with two independent 3WV's, tested here:-

2. During discharge via the d.h.w. circuit, the quiescent tank may be characterised accurately by two fully-mixed zones, whose interface coincides with the uppermost coil of the lower h.e.
3. The 3WV's should be chosen so that they do not leak, to avoid delivering hotter water than desired. If a certain leak is unavoidable, the lower 3WV setting is likely to have to be several degrees below that of the upper 3WV to avoid possible scalding. With non-leaking 3WV's, the lower might usefully be set to a slightly higher temperature than the upper, to avoid any initial draw from the top h.e.
4. The Oventrop prototype valves tested here gave good control to the required  $60^{\circ}\text{C}$ , though one suffered mechanical failure during the testing.
5. The lower h.e. may be raised to increase the sundry d.h.w. take-off from the charged tank, while leaving the upper zone of the tank at as high a temperature as possible. The value of this sundry demand is likely to be approximately proportional to the volume of the lower zone thus created. If the lower h.e. is thus raised, the volume of the high-temperature upper zone available for subsequent release to the radiators is reduced accordingly: a design compromise is therefore necessary.

(ii) Conclusions Specific to the Tank and H.E.'s Used Here

1. If the boiler charges the tank to a uniform  $80^{\circ}\text{C}$  temperature, discharge through the d.h.w. circuit by cold water at about  $15^{\circ}\text{C}$  will result in heats gained by d.h.w. of :-
  - 5.3 kWh for the SMALL/SMALL h.e. configuration
  - 6.4 kWh for the MEDIUM/MEDIUM h.e. configuration
  - 7.8 kWh for the LARGE/LARGE h.e. configuration
 before the d.h.w. delivery temperature falls below  $43.3^{\circ}\text{C}$ .

2. If the "Integron" manufacturers continue to supply only the 6m lengths of tubing, it is probable that the MEDIUM/MEDIUM h.e. configuration will remain the most appropriate to this size of tank (190 litres).
  
3. With the MEDIUM/MEDIUM h.e. configuration and the 3WV's set to upper 60°C, lower 55°C, the following d.h.w. energy takes-offs were recorded before the lower 3WV fully opened, as the lower h.e. was raised:-
  - With lower zone 43 , 0.6 kWh d.h.w. take-off
  - With lower zone 59 , 0.8 kWh d.h.w. take-off
  - With lower zone 79 , 1.1 kWh d.h.w. take-off.
  
4. The benefit available through raising the lower h.e. for more "small-draw" d.h.w. would be enhanced if a LARGE lower h.e. were fitted, should this prove economically viable. Further testing would quantify this advantage.

#### REFERENCES

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- 3.2 T. Mine, "Determination of Heat Transfer Characteristics of Immersed Coils", MSc. Thesis, Cranfield, September 1984.
  
- 3.3 R.R. Cohen, D.M. Tanton, R.A. Hale, S.D. Probert, "An Assessment of the Benefits of Heat Storage in Central Heating Systems", Appendix G, Cranfield, January 1985.
  
- 3.4 D.J. Nevrala, S.D. Probert, "What the Cormorant has in Store", from British Gas Corporation, Watson House "Bulletin" Publication, Vol. 48, No. 2, 1984.



COROLLARY

The scope of this work has been (i) a detailed (though not exhaustive) examination of the potential advantages of the British Gas storage concept (i.e. with integral d.h.w. facility) in the domestic sector, and (ii) a rather more cursory and less conclusive investigation of the use of storage in the commercial sector, where d.h.w. is supposed to be supplied independently from the central-heating plant. The following overall remarks might be suggested.

In the domestic sector, the use of storage does seem to fulfil in some measure the aspirations outlined in the General Introduction, namely that depleting gas reserves would be more rationally used if heat storage were widely adopted for home heating (a major gas user). Boiler sizes could be smaller (perhaps greatly so), network peak loads would be reduced, and each appliance should run at least 5% or so more efficiently than its conventional counterpart. In addition, new systems should be simpler to install with no penetration of the loft space. In ref. 1.6, Cohen suggests that economics should be favourable towards storage systems, particularly in the case of new dwellings. The full use of the storage concept, where the design-day's load is just met by the boiler's operation over the full 24-hours, is likely to require a large store; experience will show how large a store can practically be tolerated in the field. A large store, charged to a high temperature for maximum efficiency, will incur enhanced jacket losses; suitable design should ensure that this does not penalise a system with a large store, either by extra insulation (and cost), or by arranging for jacket loss to be wholly useful. Clearly, some advantage can be gained from a "partial" use of the storage concept (i.e. small store and only slightly smaller boiler), though boiler efficiency and d.h.w. capacity will be compromised.

The use of two independent "Integron" heat exchangers and three-way-valves, in stores of such capacity that such a feature is worthwhile, appears a sensible option, provided that the extra cost of the second three-way-valve does not substantially increase unit cost.

A mains-powered d.h.w. supply is a definite advantage.

The proposed boiler/store/heat-exchanger design does appear to be validated here, and should offer a low-cost improvement to the current procedure; it must also be pointed out that little new technology is involved, which might favour its rapid, widespread adoption. Certainly from the point of view of stratification in the store (particularly for "full" storage systems with small boiler and large store), a tall, slender cylinder is preferable; floor-loading apart, there appears to be no reason why the cylinder should not extend to near the ceiling of the dwelling. In terms of space, therefore, it may be that a good system, with a small boiler (perhaps a circulator) and a large tall store, should not occupy more useful house-room than a conventional floor-mounted boiler.

It does seem also, that such a system would reduce (perhaps greatly) the sensitivity of its conventional counterpart to the key parameters of radiator size and boiler rating.

The condensing boiler, rival to the storage system in effecting a more rational use of gas, is likely to enhance efficiency considerably more than the storage system, but (i) may well cost more, (ii) will not give the load-levelling benefits of storage, and (iii) involves a certain amount of new technology such that it is likely that its adoption in the U.K. will not be particularly swift. Due to its low temperature advantage, the condensing boiler may not be used together with a store; if, however, a good, small, condensing boiler is developed, an add-on store would extend its applicability into larger houses.

For the commercial sector field trial at Spa School, it does appear that a comparable storage advantage ought to be potentially available, but that its realisation will depend on the adoption of rigorous energy-management controls. The storage advantage will be more noticeable in a new plant, as here pipework (and its associated losses) can be kept to a minimum. Under typical, intermittent heating and occupation, it is likely that store jacket loss will be useful less often than in the domestic case, and that adequate insulation will be required.

## APPENDIX 1

(RELEVANT TO PART I)

- 1.1 "British Gas Standard Domestic Hot Water Draw-off Schedule", the basis for the d.h.w. tapes used as input to the central-heating simulation programs.
- 1.2 "HS212PS", the heating-season simulation program for the domestic central-heating system using an integral heat-exchanger inside the water store.
- 1.3 "CHSPS", a program parallel to 1.2 for a conventional central-heating system.



BRITISH GAS STANDARD DOMESTIC HOT WATER DRAW-OFF SCHEDULE

Time	'A' Day (litre)	'B' Day (litre)	
07.00	-	-	
08.30	2	2	
09.00	2	2	
09.30	5	2	
10.00	-	2	
10.30	-	5	KEY
11.00	-	5	
11.30	-	25	Hand Washing
12.00	-	25	2 litres
12.30	-	25	
13.00	-	2	Dishwashing
14.30	-	2	5 litres
15.00	-	5	
15.30	-	-	Washing Machine
16.00	-	2	25 litres
16.30	-	-	
19.00	-	2	Bath
19.30	2	2	65 litres
20.00	2	5	
20.30	5	25	
21.00	5	-	
21.30	65	-	
22.00	-	65	
22.30	-	65	
23.00	-	-	
23.30	-	-	
Daily Usage	88 ltrs	268 ltrs	

Weekly Consumption Schedule

Working couple      7 x A Days = 616 litres

Family                4 x A Days + 3 x B Days = 1156 litres

National Average    5 x A Days + 2 x B Days = 976 litres

APPENDIX 1.1

## Appendix 1.2

HS212PS - Heating-season simulation program  
for the domestic central-heating  
system with storage.

```

C   ATTEMPT TO MODEL 212-PERIOD HEATING-SEASON           ALGO 5
C
C   FEATURES:-
C   1. TNEXT USED IN FLEAK EQN. - PERFECT 3-W.V.
C   2. (a) ENBATHR D.H.W. CUT - OFF
C       (b) TMIXA < 40 D.H.W. CUT - OFF
C   3. ENERGY SUMS, BALANCE
C   4. CALLS FOR DATA (CH.15) OF HR-BY-HR ROOM
C       TEMPS, SPACE HEATING, AND D.H.W. USAGE
C   5. PREHEAT BEFORE RADS. SURGE, FROM 4.59 -
C       TO BE SET - SEE SHUT-DOWN
C   6. PREHEAT BEFORE BATH
C   7. STAT. BOOST IN COLD SPELL
C   8. HOT BOILER TEMP MOD - PILOT = 25 W
C   9. RAD & BOILER OFF 23H - 6H
C  10. DEFICIT TO ADJUST QADR EACH HOUR

```

```

C   THIS IS HS212 MODIFIED TO DO A PARAMETER SWING
C

```

```

C   DIMENSION T(5,2),TS(5,2),A(5),QIN(5),QRESID(5)
C   INTEGER BOILER, HOUR
C   REAL LOFAC, LOFACS
C  1. STORE DATA
C     DATA N, KF, VSTOR, VTOP, CP/4, 2, 243.0, 27.0, 4180./
C  2. BOILER CCT. DATA
C     DATA VBOIL, BPOW, BEFF, PILOT, UAB, FBOILS/3.86, 9000., .751, 25., 15.,
C     1 0.2/
C  3. THERMOSTATS
C     DATA ONSTAT, OFFSTAT, BOILSTAT/55., 80., 95./
C  3a. BOOST SPELL
C     DATA ND1, ND2, ONBOOST, OFFBOOST/61, 160, 65., 82.5/
C  3b. SHUT-DOWN
C     DATA NHROFF, NHRON, TIMEON/24, 5, 59./
C  4. RAD. CCT. DATA
C     DATA VRAD, UARAD, FRADMAX/35., 175., .20/
C  5. D.H.W. CCT. DATA
C     DATA UAHE, FLOW, TMAIN, VCOIL, T3WV/1800., .15, 10.0, 11.3, 47./
C  6. HEAT LOSS DATA
C     DATA PIPL, SJL, THOT, TAMB/0., 200., 85., 15./
C  7. DERIVED DATA
C     VZONE=VSTOR/FLOAT(N)
C     UAZONE=UAHE/FLOAT(N)
C  SET AREA MATRIX 'A', AND HEAT LOSS CONSTANTS
C     A(N+1)=0.400
C     AZ=2.345/FLOAT(N)
C     A(1)=0.135+AZ
C     DO 100 I=2,N
100    A(I)=AZ
C     ASTORE=2.875
C     NCOUNT=0
4000    NCOUNT=NCOUNT+1
C     IF(NCOUNT.EQ.1)SJL=200.
C     IF(NCOUNT.EQ.2)SJL=400.
C     OPEN(UNIT=15, FILE='BG1.DAT', STATUS='OLD')
C     OPEN(UNIT=16, FILE='DHW212.AVE', STATUS='OLD')
C     USTORE= SJL/(ASTORE*(THOT-TAMB))
C     UAPIPE=PIPL/(THOT-TAMB)
C  SET I.C.'S AT MIDNIGHT - ARBITRARY
C     TS(1,1)=49.66
C     TS(2,1)=59.36
C     TS(3,1)=66.88
C     TS(4,1)=73.28
C     TS(5,1)=73.28

```



```

101 DO 101 I=1,4
    T(I,1)=TS(I,1)
    TRAD1=16.90
    TBOIL=20.00
    BOILER=0
    TIME=0.
    HOUR=0
850 QST=0.
    DO 102 I=1,N
        ENZ=CP*VZONE*TS(I,1)
        ENC=CP*(VCOIL/FLOAT(N))*T(I,1)
102 QST=QST+ENZ+ENC
        QST=QST+CP*TS(N+1,1)*VTOP
        ENB=VBOIL*CP*TBOIL
        ENR=VRAD*CP*TRAD1
        IF(HOUR.EQ.24)GOTO860
        QSYS0=(QST+ENB+ENR)/1000.
        START=QSYS0
        WRITE(20,5)USTORE
5        FORMAT(///5X,'-----USTORE=',F6.1,' W/m2K-----'/)
        WRITE(20,9)OFFSTAT,ONSTAT
9        FORMAT(/5X,'OFFSTAT = ',F5.1,'      ONSTAT = ',F5.1,/)
        WRITE(20,15)TMAIN,VRAD,UARAD,VBOIL
15        FORMAT(5X,'TMAIN = ',F4.1,3X,'VRAD = ',F4.1,3X,'UARAD = ',F5.1,
1        3X,'VBOIL = ',F5.2,/)
        WRITE(20,16)UAB,PILOT
16        FORMAT(5X,'BOILER UA TO FLUE',F6.1,'      PILOT (W)',F6.1/)
        WRITE(20,6)VSTOR,VTOP,BPOW,BEFF,BOILSTAT
6        FORMAT(/5X,'VSTOR',F6.1,'  VTOP',F5.1,'  BPOW',F8.1,
1        '  BEFF(MAX)',F7.4,'  BOILSTAT',F5.1/)
        WRITE(20,7)FBOILS,FRADMAX,FLOW
7        FORMAT(5X,'FLOWS:-  BOILER',F5.2,'      RAD(MAX)',F5.2,
1        '  D.H.W.',F5.2/)
        WRITE(20,8)(NHROFF-1),(NHRON-1),TIMEON
8        FORMAT(5X,'SHUT-DOWN:-',I6,'  OCLOCK TO',I4,':',F5.2/)
        WRITE(20,10)
10        FORMAT(//5X,'AT START,MIDNIGHT,STORE TEMP. PROFILE IS :'/)
        WRITE(20,11)TS(5,1),TS(4,1),TS(3,1),TS(2,1),TS(1,1)
11        FORMAT(10X,5F8.1/)
        WRITE(20,14)BOILER
14        FORMAT(14X,'BOILER INDEX = ',I1,'  (0=OFF, 1=ON)'/)
        WRITE(20,12)TRAD1,TBOIL
12        FORMAT(14X,'TRAD1= ',F6.2,5X,'TBOIL= ',F6.2,/)
        WRITE(20,13)QSYS0
13        FORMAT(14X,'SYSTEM ENERGY,QSYS0,IS ',F8.1,'  kJ'//)
        TBATH=47.
        ENBATHR=65.*CP*(TBATH-TMAIN)/1000.
        GASMAX=18.25*3600.*BPOW/(1000.*BEFF)
        NF=0
        TRV=0.
        NRUN=0
        NBOFFS=0
        HSQWAT=0.
        HSGAS=0.
        HSRAD=0.
        HSTAP=0.
        HSQLC=0.
        HSQBCCOOL=0.
        HSPILOT=0.
        TBOILM=0.

```

```

C
C DAY LOOP _____
C

```

```

3000  READ(15,*)NDAY
      NRUN=NRUN+1
C     WRITE(20,18)NDAY
18    FORMAT('      DAY',I4,'      ')
      ONSTAT=55.
      OFFSTAT=80.
COLDEST PART OF SEASON - STATS. ON 'MAX' TO BOOST STORE
      IF(NDAY.GE.ND1.AND.NDAY.LE.ND2)ONSTAT=ONBOOST
      IF(NDAY.GE.ND1.AND.NDAY.LE.ND2)OFFSTAT=OFFBOOST
      SQRAD=0.
      SBOOST=0.
      SDRAW=0.
      QWATER=0.
      SQTAP=0.
      GAS=0.
      SQBCOOL=0.
      SQPILOT=0.
      QLC=0.
      NBOFF=0
      DEFICIT=0.

```

```

C
C  HOUR LOOP -----
C

```

```

2000  READ(15,*)HOUR,QRADR,TROOM
      READ(16,*)HOUR,WON,DRAWREQ
      TRADR=TROOM+1000.*QRADR/UARAD
      HEATR=QRADR*3600.
      QRADRA=QRADR+DEFICIT/3600.
      HEATRA=HEATR+DEFICIT
      TRADRA=TROOM+1000.*QRADRA/UARAD
      IF(HOUR.LT.7.OR.HOUR.EQ.24)HEATRA=0.
      IF(HEATRA.LT.1.)TRADRA=TROOM
      IFAIL=0
      TAP=SQTAP
      RAD=SQRAD
      REC=SBOOST
      TMIXA=50.
      TIMEO=0.
      DRAW=0.
      INDEX=1
      IF(HOUR.EQ.NHROFF.OR.HOUR.LE.NHRON)INDEX=0

```

```

C
C  DELT LOOP-----
C

```

```

1000  IF(NF.EQ.0)DELT=20.
      IF(NF.EQ.1)DELT=VCOIL/(FLOAT(N)*FLOWCOIL)
      TIME=TIME+DELT/60.
      IF(NF.EQ.1)TIMEO=TIMEO+DELT

```

```

C  RADIATOR CALCULATION (DELT)-----

```

```

      QRAD=UARAD*(TRAD1-TROOM)*DELT
      SQRAD=SQRAD+QRAD/1000.
      IF(TS(N,1).GE.TRAD1)GOTO190
      TRV=0.
      GOTO203
190   EA=(TS(N,1)-TRAD1)*DELT*FRADMAX*CP
      IF(TRAD1-TRADRA)200,201,202
200   ED=QRAD+VRAD*CP*(TRADRA-TRAD1)
      IF(ED.GE.EA)TRV=1.
      IF(ED.LT.EA)TRV=ED/EA
      GOTO203
201   TRV=QRAD/EA
      GOTO203
202   ES=VRAD*CP*(TRAD1-TRADRA)

```

```

IF(ES.GE.QRAD)TRV=0.
IF(ES.LT.QRAD.AND.EA.LE.(QRAD-ES))TRV=1.
IF(ES.LT.QRAD.AND.EA.GT.(QRAD-ES))TRV=(QRAD-ES)/EA
203 IF(TRV.GT.1.)TRV=1.
IF(HOUR.LT.7.OR.HOUR.EQ.24)TRV=0.
FRAD=TRV*FRADMAX
IF(INDEX.EQ.0)FRAD=0.
BOOST=FRAD*DELT*CP*(TS(N,1)-TRAD1)
SBOOST=SBOOST+BOOST/1000.
TRAD2=TRAD1+(BOOST-QRAD)/(CP*VRAD)
C BOILER CALCULATION (SUBDELT)-----
C STATS.....
IF(BOILER.EQ.1)GOTO300
IF(HOUR.NE.NHRON)GOTO320
IF(TIME.GT.TIMEON.AND.TIME.LT.(TIMEON+0.5))INDEX=1
320 IF(HOUR.NE.7.AND.HOUR.NE.22)GOTO321
IF(TIME.GT.0..AND.TIME.LT.0.5)BOILER=1
321 IF(TS(N,1).LT.ONSTAT.AND.INDEX.EQ.1)BOILER=1
GOTO301
300 IF(TS(1,1).GT.OFFSTAT)BOILER=0
IF(INDEX.EQ.0)BOILER=0
IF(BOILER.EQ.0)NBOFF=NBOFF+1
301 IF(BOILER.EQ.0)GOTO310
C BOILER ON.....
FBOIL=FBOILS
X=FBOIL*DELT
KX=1
305 V=X/FLOAT(KX)
IF(V.LT.VBOIL)GOTO306
KX=KX+1
GOTO305
306 SUBDELT=DELT/FLOAT(KX)
Y=FBOIL*SUBDELT
TS(N+1,2)=TS(N+1,1)
L=0
307 L=L+1
QLTOP=(A(N+1)*USTORE+UAPIPE)*(TS(N+1,2)-TROOM)*SUBDELT
QLC=QLC+QLTOP/1000.
TS(N+1,2)=((VTOP-Y)*TS(N+1,2)+Y*TBOIL)/VTOP-QLTOP/(CP*VTOP)
T1=((VBOIL-Y)*TBOIL+Y*TS(1,1))/VBOIL
TBOIL=T1+BPOW*SUBDELT/(CP*VBOIL)
IF(TBOIL.GE.BOILSTAT)BOILER=0
IF(BOILER.EQ.0)NBOFF=NBOFF+1
IF(TBOIL.GT.TBOILM)TBOILM=TBOIL
308 GAS=GAS+BPOW*SUBDELT/(BEFF*1000.)
QWATER=QWATER+BPOW*SUBDELT/1000.
IF(L.NE.KX)GOTO307
GOTO400
C BOILER OFF.....
310 QBCOOL=UAB*DELT*(TBOIL-20.)
SQBCOOL=SQBCOOL+QBCOOL/1000.
TBOIL=TBOIL-QBCOOL/(CP*VBOIL)
QWATER=QWATER-QBCOOL/1000.
QPILOT=PILOT*DELT
SQPILOT=SQPILOT+QPILOT/1000.
QLTOP=(A(N+1)*USTORE+UAPIPE)*(TS(N+1,1)-TROOM)*DELT
QLC=QLC+QLTOP/1000.
TS(N+1,2)=TS(N+1,1)-QLTOP/(CP*VTOP)
GAS=GAS+QPILOT/1000.
FBOIL=0.
C STORE CALCULATION-----
400 DOWN=FBOIL-FRAD
UP=FRAD-FBOIL

```



```

IF(DOWN.LT.0.)DOWN=0.
IF(UP.LT.0.)UP=0.
IF(NF.EQ.1)STEP=DELT/FLOAT(KF)
IF(NF.EQ.0)STEP=DELT
TINIT=T(N,1)
M=0
410 M=M+1
Z=TMAIN
QIN(1)=STEP*CP*(FRAD*TRAD1+DOWN*TS(2,1))
QIN(N)=STEP*CP*(FBOIL*TS(N+1,1)+UP*TS(N-1,1))
QRESID(1)=(VZONE-STEP*(FRAD+DOWN))*CP*TS(1,1)
QRESID(N)=(VZONE-STEP*(FBOIL+UP))*CP*TS(N,1)
DO 420 I=2,N-1
QIN(I)=STEP*CP*(DOWN*TS(I+1,1)+UP*TS(I-1,1))
420 QRESID(I)=(VZONE-STEP*(DOWN+UP))*TS(I,1)*CP
IF(NF.EQ.1)GOTO440
C D.H.W. OFF.....
DO 430 I=1,N
QL=A(I)*USTORE*(TS(I,1)-TROOM)*DELT
QLC=QLC+QL/1000.
QCOIL=VCOIL*CP*T(I,1)/FLOAT(N)
TS(I,2)=(QRESID(I)+QIN(I)+QCOIL-QL)/(CP*(VZONE+VCOIL/FLOAT(N)))
430 T(I,2)=TS(I,2)
GOTO500
C D.H.W. ON.....
440 DO 450 I=1,N
QL=A(I)*USTORE*(TS(I,1)-TROOM)*STEP
QLC=QLC+QL/1000.
AVDELT=TS(I,1)-T(I,1)
QDHW=UAZONE*STEP*AVDELT
TS(I,2)=(QRESID(I)+QIN(I)-QL-QDHW)/(CP*VZONE)
IF(I.GT.1)Z=T(I-1,1)
R=QDHW-FLOWCOIL*STEP*CP*(T(I,1)-Z)
450 T(I,2)=T(I,1)+R/(FLOWCOIL*DELT*CP)
IF(M.EQ.KF)GOTO500
DO 460 I=1,N
TS(I,1)=TS(I,2)
460 T(I,1)=T(I,2)
GOTO410
C RESET TEMPS-----
500 DO 510 I=1,N
TS(I,1)=TS(I,2)
510 T(I,1)=T(I,2)
TS(N+1,1)=TS(N+1,2)
TRAD1=TRAD2
C TEMPERATURE INVERSIONS-----
IF(TS(N+1,1).GE.TS(N,1))GOTO600
TS(N+1,1)=(TS(N+1,1)*VTOP+TS(N,1)*VZONE)/(VTOP+VZONE)
TS(N,1)=TS(N+1,1)
600 DO 610 J=1,N-1
IF(TS(N+1-J,1).GE.TS(N-J,1))GOTO610
TS(N+1-J,1)=0.5*(TS(N+1-J,1)+TS(N-J,1))
TS(N-J,1)=TS(N+1-J,1)
610 CONTINUE
C CALCULATE TMIX, FLOWCOIL, etc-----
IF(NF.EQ.0)GOTO700
TOUT=(TINIT+T(N,1))/2.
TMIXA=(FLOWBYP*TMAIN+FLOWCOIL*TOUT)/FLOW
QTAP=(TMIXA-TMAIN)*FLOW*DELT*CP/1000.
SQTAP=SQTAP+QTAP
TNEXT=1.5*T(N,1)-0.5*TINIT
FLEAK=(T3WV-TMAIN)/(TNEXT-TMAIN)
IF(FLEAK.GT.1.)FLEAK=1.

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FLOWCOIL=FLOW*FLEAK
FLOWBYP=FLOW-FLOWCOIL
GOTO800
700  TOUT=(TINIT+T(N,1))/2.
      FLEAK=(T3WV-TMAIN)/(T(N,1)-TMAIN)
      IF(FLEAK.GT.1.)FLEAK=1.
      FLOWCOIL=FLOW*FLEAK
      FLOWBYP=FLOW-FLOWCOIL
C  PRINT/TIMING-----
800  IF(NF.EQ.1)DRAW=TIMEO*FLOW
      NF=0
      IF(TIME.GE.WON.AND.DRAW.LT.DRAWREQ)NF=1
      IF(NF.EQ.1.AND.TMIXA.LT.40.)GOTO805
      GOTO810
805  TAPHR=SQTAP-TAP
      TDRAW=1000.*TAPHR/(DRAW*CP)+TMAIN
      NF=0
      IF(IFAIL.EQ.0)WRITE(20,84)NDAY,(HOUR-1),TIME,DRAW,TDRAW
      IFAIL=1
84   FORMAT(5X,'---T<40:DHW CUT OFF:DAY',I3,'HR',I3,F6.2,4X,F6.2,
1    ' lit,',F7.2,' degC')
810  IF((SQTAP-TAP).GE.ENBATHR)NF=0
      IF(TIME.LT.59.99)GOTO1000
C  ....1. HOUR CALCS.....
      TIME=TIME-60.
      SDRAW=SDRAW+DRAW
      IF(QRADR.LT.0.1)GOTO815
      RADHR=SQRAD-RAD
      DEFICIT=HEATRA-RADHR
      IF(HOUR.NE.10)GOTO815
      IF(DEFICIT.GT.100.)WRITE(20,83)NDAY,DEFICIT
83   FORMAT(5X,'4H RAD FAILURE, DAY',I4,' : DEFICIT AT 10H',F9.1,
1    ' kJ')
815  IF(HOUR.LT.24)GOTO2000
C  ....2. DAY CALCS.....
      GOTO850
860  QSYSF=(QST+ENB+ENR)/1000.
      SYSDEP=QSYS0-QSYSF
      GUEFF=100.*QWATER/GAS
      NBOFFS=NBOFFS+NBOFF
      LOFAC=100.*GAS/GASMAX
C  WRITE(20,85)OFFSTAT,ONSTAT,NBOFF
85   FORMAT(5X,'OFF',F6.1,' ON',F6.1,5X,'BOILER SWITCHED OFF',I3,
1    ' TIMES')
CC  WRITE(20,86)QSYS0,QSYSF,SYSDEP
86   FORMAT(5X,'QSYS0 ',F8.1,3X,'QSYSF ',F8.1,4X,'SYS DEPLTN ',F8.1,
1    ' kJ')
      QSYS0=QSYSF
C  WRITE(20,87)GAS,QWATER,GUEFF,LOFAC
87   FORMAT(5X,'GAS',F10.1,' QWATER',F10.1,' G.U.EFF',F6.2,' %',
1    ' L.F.',F5.2,' %')
C  WRITE(20,88)SQRAD,SQTAP,QLC,SDRAW
88   FORMAT(5X,'RADIATOR',F9.1,' DHW',F8.1,' JACKET',F8.1,' DRAW',
1    F6.1,' l')
C  WRITE(20,90)SQBCOOL,SQPILOT
90   FORMAT(5X,'OFFLOSS: BOILER T.C.',F8.2,' PILOT',F8.2)
      DIST=SQRAD+SQTAP+QLC
      SUPPLY=QWATER+SYSDEP
      ERROR=100.*(SUPPLY-DIST)/SUPPLY
C  WRITE(20,89)SUPPLY,DIST,ERROR
89   FORMAT(5X,'SUPPLY',F9.1,' DISTRIB',F9.1,4X,'ERROR',F5.2,' %')
      HSQWAT=HSQWAT+QWATER/1000.
      HSGAS=HSGAS+GAS/1000.

```

```

HSRAD=HSRAD+SQRAD/1000.
HSTAP=HSTAP+SQTAP/1000.
HSQLC=HSQLC+QLC/1000.
HSBCOOL=HSBCOOL+SQBCOOL/1000.
HSPILOT=HSPILOT+SQPILOT/1000.
IF(NRUN.LT.212)GOTO3000
C   ....3. YEAR CALCS.....
WRITE(20,9)OFFSTAT,ONSTAT
WRITE(20,15)TMAIN,VRAD,UARAD,VBOIL
WRITE(20,71)
71  FORMAT(///,15X,'HEATING SEASON SUMMARY - MJ'//)
HSDEP=(START-QSYSF)/1000.
SUPPLY=HSQWAT+HSDEP
DIST=HSRAD+HSTAP+HSQLC
ERROR=100.*(SUPPLY-DIST)/SUPPLY
GUEFF=100.*HSQWAT/HSGAS
GASAN=GASMAX*(FLOAT(NRUN))/1000.
LOFACS=100.*HSGAS/GASAN
WRITE(20,72)HSDEP
72  FORMAT(5X,'(SEASON DEPLETION',F6.2,')')
WRITE(20,73)HSGAS,HSQWAT,GUEFF,LOFACS
73  FORMAT(/24X,'GAS USED',F10.1,/10X,'HEAT INTO SYSTEM WATER',F10.1
1    ,/18X,'(GAS EFF.',F6.2,' % LOAD FACTOR',F6.2,' %)')
WRITE(20,74)HSRAD,HSTAP,HSQLC
74  FORMAT(/24X,'RADIATOR',F10.1,/23X,'TO D.H.W.',F10.1,/21X,
1    'JACKET LOSS',F10.1)
WRITE(20,75)HSBCOOL,NBOFFS,HSPILOT
75  FORMAT(/5X,'BOILER OFF LOSS',F9.3,' (',I4,' TIMES)',4X,
1    'PILOT LOSS',F9.3)
WRITE(20,76)SUPPLY,DIST,ERROR
76  FORMAT(/27X,'SUPPLY',F9.1,/22X,'DISTRIBUTED',F9.1,///30X,'ERROR',
1    F5.2,' %')
WRITE(20,77)TBOILM
77  FORMAT(//5X,'MAX. BOILER TEMP.',F7.2)
IF(NCOUNT.EQ.1)GOTO999
REWIND15
REWIND16
GOTO4000
999  STOP
END

```



### Appendix 1.3

CHSPS - Heating-season simulation program  
for the conventional central-heating  
system.

```

C
C SIMPLE MODEL OF CONVENTIONAL C.H. SYSTEM
C   1. SINGLE "BATH" RADIATOR
C   2. HOT WATER TANK - H.T. TO TANK ASSUMES PERFECT H.E. WITH
C     LOW FLOW RATE
C   3. D.H.W. DRAW ASSUMED INSTANTANEOUS
C   4. ONE ENFORCED ROOM STAT CYCLE PER HOUR
C     -----

C           HEATING SEASON PROGRAM
C THIS VERSION OF CONHS PRINTS ONLY HEATING SEASON SUMMARIES,
C AND HAS BEEN MODIFIED TO RUN PARAMETER SWINGS.

C           INTEGER BOILER
C 1. BOILER DATA
C   DATA BPOW,BEFF,PILOT,VBOIL,UAF LUE,TFLUE,BSTAT1,BSTAT2/14650.,
C   1 .751,25.,4.55,15.,15.,75.,65./
C 2. RAD DATA
C   DATA VRAD,UARAD,FRADS,CP/30.,150.,0.3,4180./
C 3. TANK DATA
C   DATA VTANK,UTANK,ATANK,FTANKS,TSTAT1,TSTAT2/137.,1.,1.545,0.3,
C   1 60.,50./
C 4. D.H.W. DATA
C   DATA TCOLD,TTAP,TAC,VCOIL,UACOIL,PIPE/10.,47.,25.,2.0,425.,1./

C           NCOUNT=0
4000 NCOUNT=NCOUNT+1
C           IF(NCOUNT.EQ.1)UARAD=150.
C           IF(NCOUNT.EQ.1)VRAD=30.
C           IF(NCOUNT.EQ.2)UARAD=200.
C           IF(NCOUNT.EQ.2)VRAD=40.
C           IF(NCOUNT.EQ.3)UARAD=250.
C           IF(NCOUNT.EQ.3)VRAD=50.
C           OPEN(UNIT=15,FILE='TAPE3.DAT',STATUS='OLD')
C           OPEN(UNIT=16,FILE='DHW212.FAM',STATUS='OLD')

C I.C.'S
C   TBOIL1=TFLUE
C   TRAD1=16.9
C   TTANK1=40.
C   TCOIL1=40.
C   ETANK=CP*(VTANK+VCOIL)*TTANK1
C   EBOIL=CP*VBOIL*TBOIL1
C   ERAD=CP*VRAD*TRAD1
C   ESYS0=(ETANK+ERAD+EBOIL)/1000.

C           WRITE(20,14)UARAD
14  FORMAT(/////6X,'----- UARAD = ',F6.1,' W/degC -----'/)
C           WRITE(20,15)
15  FORMAT(/5X,'HEATING SEASON - CONVENTIONAL SYSTEM, 1 SW/HR')
C           WRITE(20,20)BPOW,BEFF,PILOT,UAF LUE,TFLUE,VBOIL,BSTAT1,BSTAT2
20  FORMAT(/5X,'BOILER :',F8.1,F8.4,3F6.1,F6.2,2F6.1)
C           WRITE(20,21)VRAD,UARAD,FRADS
21  FORMAT(/5X,'RAD :',F6.1,F7.1,F6.2/)
C           WRITE(20,22)VTANK,UTANK,ATANK,FTANKS,TSTAT1,TSTAT2
22  FORMAT(5X,'TANK :',2F6.1,F8.3,F7.3,2F6.1/)
C           WRITE(20,23)TCOLD,TTAP,TAC,PIPE
23  FORMAT(5X,'D.H.W. :',4F8.1/)

C           WRITE(20,26)TBOIL1,TRAD1,TTANK1,TCOIL1

```

26       FORMAT(/5X,'INITIAL CONS.: BOIL',F6.2,' RAD',F6.2,' TANK',  
1       F6.2,' COIL',F6.2,' degC'/)

221

HSDRAW=0.  
HSDHOT=0.  
HSENDRAW=0.  
HSQREQ=0.  
HSQR1=0.  
HSQR0=0.  
HSETLOSS=0.  
HSGAS=0.  
HSQW=0.  
HSQB=0.  
HSQP=0.  
NHSBOFF=0

BOILER=0  
DELT=5.  
WRITE(20,24)DELT  
24       FORMAT(5X,'DELT: ',F6.2,' SECONDS'/)  
SURP=0.  
NDAY=0

C   DAY LOOP-----

3000   NDAY=NDAY+1  
TSD=100.  
SQR1=0.  
SQR0=0.  
SSQRAD=0.  
SQREQ=0.  
SDRAW=0.  
SDHOT=0.  
SENDRAW=0.  
SETLOSS=0.  
SGAS=0.  
SQW=0.  
SSQB=0.  
SSQP=0.  
NBOFFS=0  
SURP=0.

IF(NDAY.EQ.1)ESYS1=ESYS0

C   HOUR LOOP-----

2000   READ(15,\*)QREQ,TROOM  
      READ(16,\*)NHR,WON,DRAWREQ  
      NBOFF=0

      SQREQ=SQREQ+QREQ  
C       WRITE(20,30)NHR

30       FORMAT(/5X,'----- HOUR NO.',I2,' -----',//)

C       WRITE(20,31)QREQ,TROOM,WON,DRAWREQ

31       FORMAT(5X,'RAD HT',F7.3,' KWH TO',F6.2,' degC',/5X,  
1       'D.H.W. ON AFTER',F6.2,' MINS, DRAW',F6.2,' 1'/)

IF(DRAWREQ.GT.0.5)GOTO185  
DRAWREQ1=0.  
DRAWHOT=0.  
ENDRAW=0.

185   INDEX=1  
IF(INDEX.EQ.1.OR.BOILER.EQ.0)GOTO190  
BOILER=0

C       WRITE(20,45)(NHR-1),TIME  
      NBOFF=NBOFF+1

190   IF(INDEX.EQ.1)GOTO191



```

FRAD=0.
FTANK=0.
FBOIL=0.

191  IF(NHR.EQ.7)SURP=0.
      QREQA=QREQ*3600.-SURP
      IF(QREQA.LT.0.)QREQA=0.
      SURP=SURP-(QREQ*3600.-QREQA)
      SURP1=SURP
C     WRITE(20,35)(QREQ*3600.),QREQA,SURP
35    FORMAT(5X,'CALLED FOR',F8.1,' ADJUSTED TO',F8.1,
1     ' ; RESIDUAL SURPLUS',F8.1)
      SQRAD=0.
      GAS=0.
      QWATER=0.
      SQBCOOL=0.
      SQPILOT=0.
      TIME=0.
      TIMEO=0.

C     WRITE(6,*)NHR

C     DELT LOOP -----

1000  IF(INDEX.EQ.0)GOTO250

C     D.H.W. DRAW
200   IF(TIME.LT.WON)GOTO210
      DRAWREQ1=DRAWREQ+PIPE
      IF(TTANK1.LT.TTAP)WRITE(20,40)TTANK1,DRAWREQ1
40    FORMAT(5X,'D.H.W. FAILURE; TANK TEMP',F6.2,' degC, DRAW',F6.2,
1     ' 1 TAKEN')
      DRAWHOT=DRAWREQ1*(TTAP-TCOLD)/(TTANK1-TCOLD)
      IF(DRAWHOT.GT.DRAWREQ1)DRAWHOT=DRAWREQ1
      SDRAW=SDRAW+DRAWREQ1
      SDHOT=SDHOT+DRAWHOT
      ENDRAW=DRAWHOT*CP*(TTANK1-TCOLD)/1000.
      SENDRAW=SENDRAW+ENDRAW
      TTANK1=(TTANK1*(VTANK-DRAWHOT)+DRAWHOT*TCOLD)/VTANK
      WON=70.

C     CONTROLS
210   IF(SQRAD.LT.QREQA)FRAD=FRADS
      IF(SQRAD.GE.QREQA)FRAD=0.
      IF(QREQA.LT.300.)FRAD=0.

      IF(FTANK.LT.0.005)GOTO220
      IF(TTANK1.GT.TSTAT1)FTANK=0.
      GOTO221
220   IF(TTANK1.LT.TSTAT2)FTANK=FTANKS

C     24H SHUT-DOWN
221   IF(NHR.NE.23)GOTO230
      IF(TIME.GE.TSD)FRAD=0.
      IF(TIME.GE.TSD)GOTO230
      EARAD=VRAD*CP*(TRAD1-TROOM)/1000.
      IF(EARAD.GT.(QREQA-SQRAD))TSD=TIME

230   IF(BOILER.EQ.1)GOTO240
C     BOILER OFF
      IF(TIME.GT.0.05)GOTO235
      IF((FRAD+FTANK).GT.0.005.AND.TBOIL1.LT.BSTAT1)BOILER=1
235   IF((FRAD+FTANK).GT.0.005.AND.TBOIL1.LT.BSTAT2)BOILER=1

```

```

      GOTO250
C     BOILER ON
240   IF(TBOIL1.GT.BSTAT1)BOILER=0
      IF((FRAD+FTANK).LT.0.005)BOILER=0
      IF(BOILER.EQ.0)NBOFF=NBOFF+1
C     IF(BOILER.EQ.0)WRITE(20,45)(NHR-1),TIME,TBOIL1,FRAD,FTANK,TTANK1
45    FORMAT(5X,'BOILER OFF AT',I4,':',F5.2,'      TBOIL',F6.2,'  FRAD'
1     ,F5.2,'  FTANK',F5.2,'  TT',F6.2)

250   TIME=TIME+DELT/60.

C     RADIATOR .....
      QRAD=UARAD*DELT*(TRAD1-TROOM)
      BOOST=FRAD*DELT*CP*(TBOIL1-TRAD1)
      IF(SQRAD.LT.QREQA)SQRAD1=SQRAD+QRAD/1000.
      IF(SQRAD.GE.QREQA)SURP=SURP+QRAD/1000.
      IF(SQRAD.LT.QREQA)SQRAD=SQRAD1
      TRAD2=TRAD1+(BOOST-QRAD)/(CP*VRAD)

C     D.H.W. TANK.....
      QDHW=UACOIL*DELT*(TCOIL1-TTANK1)
      ETLOSS=UTANK*ATANK*DELT*(TTANK1-TAC)
      SETLOSS=SETLOSS+ETLOSS/1000.
      TTANK2=TTANK1+(QDHW-ETLOSS)/(CP*VTANK)
      TCOIL2=TCOIL1+(FTANK*DELT*CP*(TBOIL1-TCOIL1)-QDHW)/(CP*VCOIL)
      IF(BOILER.EQ.0)GOTO260

C     BOILER ON
      ERESID=(VBOIL-DELT*(FRAD+FTANK))*CP*TBOIL1
      ERET=CP*DELT*((FRAD*TRAD1)+(FTANK*TCOIL1))
      T1=(ERESID+ERET)/(CP*VBOIL)
      TBOIL2=T1+(BPOW*DELT)/(CP*VBOIL)
      GAS=GAS+BPOW*DELT/(BEFF*1000.)
      QWATER=QWATER+BPOW*DELT/1000.
      GOTO270

C     BOILER OFF
260   QBCOOL=UAFLUE*DELT*(TBOIL1-TFLUE)
      SQBCOOL=SQBCOOL+QBCOOL/1000.
      ERESID=(VBOIL-DELT*(FRAD+FTANK))*CP*TBOIL1
      ERET=CP*DELT*((FRAD*TRAD1)+(FTANK*TCOIL1))
      TBOIL2=(ERESID+ERET-QBCOOL)/(CP*VBOIL)
      QWATER=QWATER-QBCOOL/1000.
      QPILOT=PILOT*DELT
      SQPILOT=SQPILOT+QPILOT/1000.
      GAS=GAS+QPILOT/1000.

C     INITIALISE
270   TBOIL1=TBOIL2
      TCOIL1=TCOIL2
      TRAD1=TRAD2
      TTANK1=TTANK2
      TIMEO=TIMEO+DELT
      IF(NHR.NE.12)GOTO280
C     IF(TIME.GT.43..AND.TIME.LT.46.)WRITE(20,47)TIME,BOILER,FRAD,
C     1 FTANK,SQRAD,SURP,GAS,TBOIL2,TRAD2,TTANK2
47    FORMAT(5X,F5.2,I3,2F6.2,3F8.1,F8.2,2F7.2)
      IF(TIMEO.GE.59.)TIMEO=0.

280   IF(TIME.LE.59.99)GOTO1000

C     HOUR CALCS/PRINT-----
C     WRITE(20,50)DRAWREQ1,DRAWHOT,ENDRAW

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50 1  FORMAT(/5X,'HOT WATER: ',F6.2,' 1 AT TAP,',F6.2,' 1 FROM TANK,',
      F9.2,' kJ TO TAP')
      ETAGU=100*QWATER/GAS
      IF(ETAGU.LT.0.)ETAGU=0.
      IF(SQRAD.LT.QREQA)SURP=SURP-(QREQA-SQRAD)
      COOL=SURP-SURP1
      IF(NHR.NE.1.AND.COOL.LT.0.)COOL=0.
      SQRADT=SQRAD+COOL
      IF(QREQA.GT.15000.AND.SURP.LT.-100.)WRITE(20,55)NDAY,NHR,SURP
55 1  FORMAT(5X,'RADFAIL: DAY',I4,' HOUR',I3,' : BALANCE',F9.1,' kJ')
      C  WRITE(20,51)SQRADT,SURP
51 1  FORMAT(5X,'RADIATOR: kJ RADIATED',F10.1,' NEW SURPLUS',F9.1)
      C  WRITE(20,52)GAS,QWATER,SQB COOL,SQPILOT
52 1  FORMAT(5X,'GAS',F10.2,' QWATER',F10.2,' FLUE ASD',F8.2,
      ' PILOT',F6.2,' (ALL kJ)')
      C  WRITE(20,53)NBOFF,ETAGU
53 1  FORMAT(5X,'BOILER OFF',I4,' TIMES, G.U.EFF.',F7.3,' %')

300 1  NBOFFS=NBOFFS+NBOFF
      SSQRAD=SSQRAD+SQRADT
      IF(INDEX.EQ.1)SQR1=SQR1+SQRADT
      IF(INDEX.EQ.0)SQR0=SQR0+SQRADT
      SGAS=SGAS+GAS
      SQW=SQW+QWATER
      SSQB=SSQB+SQB COOL
      SSQP=SSQP+SQPILOT

      IF(NHR.LT.24)GOTO2000

C  DAY CALCS/PRINT-----
C  WRITE(20,60)NDAY
60 1  FORMAT(5X,'-----DAY NO.',I4,'-----')
C  WRITE(20,61)SDRAW,SDHOT,SENDRAW
61 1  FORMAT(5X,'D.H.W.: ',F8.2,' 1 AT TAP,',F8.2,' 1 FROM TANK',
      F10.2,' kJ TO TAP')
C  WRITE(20,65)SETLOSS
65 1  FORMAT(5X,'TANK JACKET LOSS',F8.2,' kJ')
      SQREQ=SQREQ*3600.
C  WRITE(20,62)SQREQ,SQR1,SQR0
62 1  FORMAT(5X,'RADS: REQD',F10.2,' RAD',F10.2,' kJ;',4X,
      '(',F9.2,' HEATING OFF)')
C  WRITE(20,63)SGAS,SQW,SSQB,SSQP
63 1  FORMAT(5X,'GAS',F10.2,' QWATER',F10.2,' FLUE',F9.2,' PILOT',
      F8.2)
      EGUF=100*SQW/SGAS
      FRAC=EGUF/BEFF
C  WRITE(20,64)NBOFFS,EGUF,FRAC
64 1  FORMAT(5X,'TOTAL BOILER S.O.',I5,' TIMES : OVERALL EFF',F7.3,
      '% (',F6.3,' % BEFF)')
      ETANK=TTANK2*CP*VTANK
      ECOIL=TCOIL2*CP*VCOIL
      EBOIL=TBOIL2*CP*VBOIL
      ERAD=TRAD2*CP*VRAD
      ESYS2=(ETANK+ERAD+EBOIL+ECOIL)/1000.
      SYSGAIN=ESYS2-ESYS1
      ESYS1=ESYS2
C  WRITE(20,67)SYSGAIN
67 1  FORMAT(/5X,'SYSTEM EN. GAIN, kJ',F10.2)
      DISTRIB=SSQRAD+SETLOSS+SYSGAIN+SENDRAW
      ERROR=100*(DISTRIB-SQW)/SQW
C  WRITE(20,66)ERROR
66 1  FORMAT(5X'----- ERROR',F10.6,' % -----')

```



```

C      WRITE(20,68)SSQRAD,SQR1,SQR0,(SQR1+SQR0)
68     FORMAT(5X,4F15.4)
      HSDRAW=HSDRAW+SDRAW
      HSDHOT=HSDHOT+SDHOT
      HSENDRAW=HSENDRAW+SENDRAW/3600.
      HSETLOSS=HSETLOSS+SETLOSS/3600.
      HSQREQ=HSQREQ+SQREQ/3600.
      HSQR1=HSQR1+SQR1/3600.
      HSQR0=HSQR0+SQR0/3600.
      HSGAS=HSGAS+SGAS/3600.
      HSQW=HSQW+SQW/3600.
      HSQB=HSQB+SSQB/3600.
      HSQP=HSQP+SSQP/3600.
      NHSBOFF=NHSBOFF+NBOFFS

      IF(NDAY.LT.212)GOTO3000

```

```

C      FINAL PRINT/CALCS-----

```

```

      WRITE(20,70)NDAY
70     FORMAT(////,5X,'-----HEATING SEASON SUMMARY AFTER',I4,' DAYS-----'
1      ,//15X,'CONVENTIONAL C.H. SYSTEM'//)

      WRITE(20,71)HSDRAW,HSDHOT,HSENDRAW
71     FORMAT(5X,'D.H.W.: ',F11.2,' 1 AT TAP',F10.2,' 1 FROM TANK',
1      F9.2,' kWh TO TAP'/)
      WRITE(20,72)HSQREQ,HSQR1,HSQR0
72     FORMAT(5X,'RADS.: REQD',F10.2,' kWh, RADIATED',F10.2,' kWh; ('
1      F7.2,' HEATING OFF)'/)
      WRITE(20,73)HSETLOSS
73     FORMAT(5X,'TANK JACKET LOSS',F8.2,' kWh'/)
      WRITE(20,74)HSGAS,HSQW,HSQB,HSQP
74     FORMAT(5X,'GAS',F10.2,' QWATER',F10.2,' FLUE',F9.2,' PILOT',
1      F7.2,' ALL kWh'/)
      FEGUF=100.*HSQW/HSGAS
      FFRAC=FEGUF/BEFF
      WRITE(20,75)NHSBOFF,FEGUF,FFRAC
75     FORMAT(5X,'H.S. BOILER S.O.',I6,' TIMES: OVERALL EFF.',F7.3,' %
1      ('F7.3,' % BEFF)'/)
      SYSGAIN=(ESYS2-ESYS0)/3600.
      DISTRIB=HSQR1+HSQR0+HSETLOSS+SYSGAIN+HSENDRAW
      ERROR=100.*(DISTRIB-HSQW)/HSQW
      WRITE(20,66)ERROR
      IF(NCOUNT.EQ.1)GOTO999
      REWIND15
      REWIND16
      GOTO4000

999    STOP
      END

```

## APPENDIX 2

(RELEVANT TO PART II)

2.1 "SPAP", the computer program of the heating system of Spa School, Bermondsey, including a large water store.

2.2 "SPAEFF", the program used to analyse a single day's data from Spa School.





```

      DO 80 I=1, NRAD
80      TR(I,1)=TRAD1
C      SYSTEM ENERGY REL. 0 DEGC
      STEN=0.
      DO 81 I=1, N
      ENZ=VZONE*CP*TS(I,1)/1000.
81      STEN1=STEN1+ENZ
      BEN=CP*VBOIL*TBOIL/1000.
      RADEN=0.
      DO 82 I=1, NRAD
      SREN=CP*TR(I,1)*VSRAD/1000.
82      RADEN=RADEN+SREN
      QSYS1=STEN1+BEN+SREN
      WRITE(20,15)
15      FORMAT(/5X,'KEY (ALL IN kWh):-',/10X,'A. ENERGY TO RADIATOR',
1 /10X,'B. ENERGY EMITTED',/10X,
2 'C. CONTRIBUTION OF STORE (REL. TRAD)',/10X,
3 'D. GAS AT METER',/10X,'E. HEAT INTO SYSTEM WATER'/)
C      SUMS TO ZERO
      SQPILOT=0.
      SQBCOOL=0.
      QLC=0.
      GAS=0.
      QWATER=0.
      SENIN=0.
      SQRAD=0.
      SENST=0.
C
      TIME=0.
      TSUP=80.
      TAD=67.
      TBOILM=0.
      L=0
C      HOUR LOOP -----
2000      READ(15,*)HR, TROOM, TREQ
          WRITE(5,*)HR
95      WRITE(20,40)HR
C      ZERO "INDEX" INDICATES THAT WHOLE SYSTEM IS OFF
          IF(HR.LT.3.OR.HR.GT.15)INDEX=0
          GASP=GAS
          QWATP=QWATER
          SEN=SENIN
          SQR=SQRAD
          SENSTP=SENST
          ONSTAT=TREQ
C      DELT LOOP -----
C      CONTROLS.....
C      STORE STATS START CONTROL OF BOILERS
1000     IF(HR.EQ.3.AND.TIME.GE.35.)INDEX=1
          IF(INDEX.EQ.1)GOTO100
          FBOIL=0.
          FRAD=0.
          BOILER=0
          GOTO130
100     IF(BOILER.EQ.0)GOTO110
          IF(TS(1,1).GT.OFFSTAT)BOILER=0
C      TIME OF FINAL BOILER SWITCH-OFF

```

```

IF(HR.EQ.14.AND.TIME.GE.50.)BOILER=0
IF(BOILER.EQ.0)WRITE(20,29)(HR-1),TIME
29  FORMAT(5X,'<-----BOILER OFF AT',I4,' HR',F7.2,' MINS')
IF(BOILER.EQ.0)WRITE(20,31)(TS(I,1),I=1,N)
31  FORMAT(13X,'STORE PROFILE (UPWARDS):',8F8.2)
GOTO120

110  IF(TS(N,1).LT.ONSTAT)BOILER=1
C    TIME OF FINAL BOILER SWITCH-OFF
IF(HR.EQ.14.AND.TIME.GE.50.)BOILER=0
IF(HR.EQ.15)BOILER=0
IF(BOILER.EQ.1)WRITE(20,30)(HR-1),TIME
30  FORMAT(5X,'----->BOILER ON AT',I4,' HR',F7.2,' MINS')
IF(BOILER.EQ.1)WRITE(20,31)(TS(I,1),I=1,N)

120  IF(BOILER.EQ.0)FBOIL=0.
IF(BOILER.EQ.1)FBOIL=FB/60.
FRAD=FRADS/60.
IF(HR.LT.4)FRAD=0.
IF(HR.NE.4)GOTO130
C    RADIATOR PUMP ON
IF(TIME.LT.55.)FRAD=0.
IF(TIME.GT.54.9.AND.TIME.LT.55.1)WRITE(20,32)(HR-1),TIME
1  , (TS(I,1),I=1,N)
32  FORMAT(5X,'AT',I3,':',F5.2,' STORE PROFILE (UPWARDS):',8F8.2,
1  4X,'D.PUMP ON')
130  TIME=TIME+DELT/60.
C    RADIATOR.....
IF(FRAD.LT.0.1)GOTO141
DO 140 I=1,NRAD
QRAD=UASR*(TR(I,1)-TROOM)*DELT
SQRAD=SQRAD+QRAD/1000.
140  TR((I+1),2)=TR(I,1)-QRAD/(VSRAD*CP)
TR(1,2)=TAD
TRAD=TR((NRAD+1),2)
ENIN=FRAD*DELT*CP*(TAD-TRAD1)
SENIN=SENIN+ENIN/1000.
GOTO145
141  DO 142 I=1,NRAD
QRAD=UASR*(TR(I,1)-TROOM)*DELT
SQRAD=SQRAD+QRAD/1000.
142  TR(I,2)=TR(I,1)-QRAD/(CP*VSRAD)
TRAD=TR(NRAD,2)
C    STORE.....
145  UP=FRAD-FRET-FBOIL
DOWN=-UP
IF(DOWN.LT.0.)DOWN=0.
IF(UP.LT.0.)UP=0.
C
ENST=CP*UP*DELT*(TS(N,1)-TRAD1)
SENST=SENST+ENST/1000.
C
TS(N+1,1)=TBOIL
QIN(1)=DELT*CP*(UP*TRAD1+DOWN*TS(2,1))
DO 150 I=2,N
150  QIN(I)=DELT*CP*(UP*TS(I-1,1)+DOWN*TS(I+1,1))
DO 160 I=1,N
QL=A(I)*USTORE*DELT*(TS(I,1)-(TROOM+5.))
QRESID=(VZONE-DELT*(UP+DOWN))*CP*TS(I,1)
TS(I,2)=(QRESID+QIN(I)-QL)/(CP*VZONE)
160  QLC=QLC+QL/1000.
C    BOILER.....
C    ON

```

```

IF(BOILER.EQ.0)GOTO300
TBIN=((FRAD-FRET)*TRAD1+DOWN*TS(1,1))/(FRAD-FRET+DOWN)
TBOIL=TBIN+BPOW*DELT/(CP*VBOIL)
GAS=GAS+BPOW*DELT/(BEFF*1000.)
QWATER=QWATER+BPOW*DELT/1000.
IF(TBOIL.LE.TBOILM)GOTO400
TBOILM=TBOIL
HRM=HR
TIMEM=TIME
GOTO400
C OFF
300 QBCOOL=UAFLUE*DELT*(TBOIL-20.)
SQBCOOL=SQBCOOL+QBCOOL/1000.
TBOIL=TBOIL-QBCOOL/(CP*VBOIL)
QWATER=QWATER-QBCOOL/1000.
QPILOT=PILOT*DELT
SQPILOT=SQPILOT+QPILOT/1000.
GAS=GAS+QPILOT/1000.
C RESET TEMPERATURES.....
400 TRAD1=TRAD
DO 410 I=1,N
410 TS(I,1)=TS(I,2)
DO 411 I=1,NRAD
411 TR(I,1)=TR(I,2)
C 3WV OPERATION FOR NEXT DELT.....
FRET1=FRET
IF((FBOIL+UP).LT.0.001)GOTO420
IF(FRAD.GT..1)TSUP=(FBOIL*TBOIL+UP*TS(N,1))/(FBOIL+UP)
420 FRET=FRAD*(TSUP-TREQ)/(TSUP-TRAD1)
IF(FRET.LT.0.)FRET=0.
IF(FRET.GT.FRAD)FRET=FRAD
IF(TSUP.LT.TREQ)FRET=0.
FRET=(FRET+FRET1)*0.5
IF(FRAD.GT..1)TAD=(FRET*TRAD1+(FRAD-FRET)*TSUP)/FRAD

IF(HR.NE.4.AND.HR.NE.5)GOTO450
IF(HR.EQ.4.AND.TIME.LT.54.)GOTO450
IF(HR.EQ.5.AND.TIME.GT.30.)GOTO450
WRITE(20,35)TIME,UP,DOWN,TBIN,TS(1,1),TS(N,1),TBOIL,TRAD,TSUP,TAD,FRET
35 FORMAT(5X,F6.2,3X,3F6.2,3X,4F6.2,3X,3F6.2)
C WRITE(20,36)FRAD
36 FORMAT(5X,'FRAD =',F6.2)

C HOUR PRINT -----
450 IF(TIME.LT.59.99)GOTO1000
RADEN=(SENIN-SEN)/3600.
REMIT=(SQRAD-SQR)/3600.
STO=(SENST-SENSTP)/3600.
GASH=(GAS-GASP)/3600.
WATERH=(QWATER-QWATP)/3600.
EFFH=WATERH*100./GASH
EFFO=EFFH/BEFF
40 FORMAT(/5X,'HOUR TO ',I2,' OCLOCK:-')
WRITE(20,41)RADEN,REMIT,STO,GASH,WATERH
41 FORMAT(5X,'A',F8.3,' B',F8.3,' C',F8.3,' D',F8.3,' E',
1 F8.3)
WRITE(20,35)TIME,UP,DOWN,TBIN,TS(1,1),TS(N,1),TBOIL,TRAD,TSUP,
TAD,FRET
WRITE(20,31)(TS(I,1),I=1,N)
TIME=TIME-60.

C DAY PRINT -----

```



```

IF(HR.LT.24)GOTO2000
WRITE(20,45)TRAD,TBOIL
45  FORMAT(20X,'TRAD',F7.2,'      TBOIL',F7.2)
C (STORE ENERGY AFTER 24H REL. 20C)
  STEN2=0.
  DO 500 I=1,N
  ENZ=VZONE*CP*TS(I,1)/1000.
500  STEN2=STEN2+ENZ
  BEN=CP*VBOIL*TBOIL/1000.
  RADEN=0.
  DO 501 I=1,NRAD
  SREN=CP*VSRAD*TR(I,2)
501  RADEN=RADEN+SREN/1000.
  QSYS2=STEN2+BEN+RADEN
C
  RADEFF=(SQRAD*100.)/SENIN
  OVEFF=QWATER*100./GAS
  CFFLE=OVEFF/BEFF
  DIST=QSYS2-QSYS1+QLC+SQRAD
  SYSGAIN=(QSYS2-QSYS1)/3600.
  ERROR=(QWATER-DIST)*100./QWATER
C
  WRITE(20,50)
50  FORMAT(///5X,'-----SUMMARY-----',//)
  WRITE(20,59)TBOILM,(HRM-1),TIMEM
59  FORMAT(5X,'MAX. BOILER TEMP',F6.2,' C AT ',I3,':',F5.2/)
  WRITE(20,51)(SENIN/3600.),(SQRAD/3600.),(RADEFF
51  FORMAT(5X,'RADIATOR: RECEIVED',F9.3,' kWh',/,15X,'RADIATED',
1  F9.3,' kWh',/,10X,'EFFICIENCY',F6.2,' %'/)
  WRITE(20,52)(QLC/3600.),(SENST/3600.)
52  FORMAT(5X,'STORE: JACKET LOSS',F9.3,' kWh',/,12X,
1  'CONTRIBUTION(REL. TRAD)',F9.3,' kWh'/)
  WRITE(20,53)(GAS/3600.),(QWATER/3600.)
53  FORMAT(5X,'BOILER: GAS AT METER',F9.3,' kWh',/,13X,
1  'HEAT INTO SYSTEM WATER',F9.3,' kWh'/)
  WRITE(20,54)OVEFF,CFFLE
54  FORMAT(13X,'OVERALL EFFICIENCY',F6.2,' %',/,13X,
1  'C.F. BOILER F.L.E.',F6.2,' %'/)
  WRITE(20,55)(SQBCOOL/3600.),(SQPILOT/3600.)
55  FORMAT(13X,'FLUE HEAT LOSS A.S.D.',F7.3,' kWh',/,13X,
1  'PILOT GAS LOSS',F7.3,' kWh',/)
  WRITE(20,56)(QWATER/3600.)
56  FORMAT(5X,'HEAT BALANCE: HEAT TO WATER',F9.3,' kWh')
  WRITE(20,57)SYSGAIN,(QLC/3600.),(SQRAD/3600.)
57  FORMAT(19X,'DISSIPATED: 1.SYSTEM GAIN',F9.3,/30X,
1  ' 2.JACKET LOSS',F9.3,/30X,' 3.RADIATOR EMISSION',F9.3)
  WRITE(20,58)(DIST/3600.),ERROR
58  FORMAT(32X,'TOTAL',F9.3,' kWh',/19X,'-----ERROR',F9.5,' %-----')
  STOP
  END

```

C PROG. TO SUM RAD AND BOILER HEATS OVER SINGLE DAY  
C CAN THEN EASILY PROCEED TO EFFICIENCY.

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C ALGORITHM - HAVE USED AREA OF BACKWARD TRAPEZIUM.

N=0  
QBOILMJ=0.  
QRADMJ=0.  
DGHR18=0.  
NSWON=0  
NSWOFF=0  
TIMBRUN=0.  
TIMOFF=0.

## APPENDIX 2.2

### ONE-DAY ANALYSIS -

#### "SPAEFF"

```
65      READ(15,65)NDAY,MONTH
      FORMAT(16X,I2,1X,I2)

1000    READ(15,70,END=400)NC,NHR,MIN,SEC,ND,TA,TB,TC,TBMIX0,TSTU,TSTL,
1      S,TFRAD0,TTRAD0,TTBOIL,TFBOIL,FBOIL,FRAD,GAS,FSTORE,FSIDE
70      FORMAT(3X,I1,1X,I2,1X,I2,1X,I2,1X,F4.1,2X,I5,11F7.2,2F6.2,3F7.0)
      IF(NC.EQ.6.AND.NHR.EQ.0)TST1=(TB+TC+TBMIX0+TSTU+TSTL+S+TFRAD0
1      +TTRAD0+TTBOIL+TFBOIL)/10.
      IF(NC.EQ.6.AND.NHR.EQ.23)TST2=(TB+TC+TBMIX0+TSTU+TSTL+S+TFRAD0
1      +TTRAD0+TTBOIL+TFBOIL)/10.
      IF(NC.EQ.6)GOTO1000
      TIME=FLOAT(NHR)+FLOAT(MIN)/60.+SEC/3600.
      N=N+1
      IF(N.EQ.1)GASINIT=GAS

      IF(N.EQ.1)DELDHR=(18.-TA)*TIME
      IF(N.NE.1)DELDHR=(18.-0.5*(TAX+TA))*(TIME-TIMEX)
      TIMEX=TIME
      TAX=TA
      IF(DELDHR.LT.0.)DELDHR=0.
      DGHR18=DGHR18+DELDHR

      IF(IBOIL.GE.1.AND.FBOIL.LT.2.)NSWOFF=NSWOFF+1
      IF(FBOIL.LT.2.)IBOIL=0
      IF(FBOIL.GE.2.)IBOIL=IBOIL+1
      IF(IBOIL.EQ.1.AND.IRAD.EQ.0)TIMON=TIME
      IF(IBOIL.EQ.1)NSWON=NSWON+1
      IF(IBOIL.GT.1)TIMBRUN=TIMBRUN+(TIME-TIM)
      IF(IBOIL.EQ.0)GOTO210
      IF(IBOIL.EQ.1)GOTO200
      IF(FB.LT.6.)FB=FBOIL
      IF(FBOIL.LT.6.)FBOIL=FB
      DELQKJ=3762.*(TIME-TIM)*(FBOIL+FB)*(TFBOIL-TTBOIL+TFB-TTB)
      TBBAR=0.25*(TFBOIL+TTBOIL+TFB+TTB)
      DELQKJ=DELQKJ*(1.01111-0.0005*TBBAR)
      QBOILMJ=QBOILMJ+DELQKJ/1000.

200    TTB=TTBOIL
      TFB=TFBOIL
      FB=FBOIL
      TIM=TIME

210    IF(FRAD.LT.1.3)IRAD=0
      IF(FRAD.GE.1.3)IRAD=IRAD+1
      IF(IRAD.EQ.1)JRAD=0
      IF(FRAD.LT.3..AND.FR.GT.3.)JRAD=JRAD+1
      IF(JRAD.NE.1)GOTO220
      TIMOFF=TIME
```

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```

220  IF(IRAD.EQ.0)GOTO1000
      IF(IRAD.EQ.1)GOTO300
      DELPKJ=3762.*(TIME-TEM)*(FRAD+FR)*(TTRAD0-TFRAD0+TTR-TFR)
      TRBAR=0.25*(TTRAD0+TFRAD0+TTR+TFR)
      DELPKJ=DELPKJ*(1.01111-0.0005*TRBAR)
      QRADMJ=QRADMJ+DELPKJ/1000.

300  TTR=TTRAD0
      TFR=TFRAD0
      FR=FRAD
      TEM=TIME
      TA1=TA

      GOTO1000

400  GASDAY=(GAS-GASINIT)*10.*1032*1.055/1000.
      IF(TIMOFF.LT.1.)TIMOFF=24.
      FLOAD=TIMBRUN/(TIMOFF-TIMON)
      WRITE(20,80)NDAY,MONTH
80  FORMAT(/5X,'DAY',I3,' OF MONTH',I3,/)
      WRITE(20,79)FLOAD,TIMBRUN,(TIMOFF-TIMON),NSWON,NSWOFF
79  FORMAT(8X,'LOAD FACTOR',F6.3,5X,'HRS ON',F7.3,' HRS OVLL',F7.3,
1  /8X,'SWITCHINGS:  ON',I3,'  OFF',I3,/)
      WRITE(20,81)QBOILMJ,QRADMJ
81  FORMAT(5X,F10.2,' MJ DELIVERED BY BOILER'//5X,F10.2,
1  ' MJ DELIVERED TO RADIATORS'/)
      WRITE(20,82)GASDAY
82  FORMAT(5X,F10.2,' MJ OF GAS FED TO BOILER - BY METER')

      GASCORR=GASDAY*0.977
      WRITE(20,83)GASCORR
83  FORMAT(5X,F10.2,' MJ ASSUMING 0.977 METER CORRECTION FACTOR'/)
      STCONMJ=(TST1-TST2)*3.3*4.18
      ERROR=(QBOILMJ+STCONMJ-QRADMJ)
      WRITE(20,86)STCONMJ,ERROR
86  FORMAT(8X,'STORE EN. REDN. OVER DAY',F10.2,' MJ: ERROR',F10.2,
1  ' MJ')
      EFFB=100.*QBOILMJ/GASCORR
      EFFD=EFFB*QRADMJ/(QBOILMJ+STCONMJ)
      WRITE(20,84)EFFB,EFFD
84  FORMAT(/8X,'BOILER EFFICIENCY',F7.2,' %',/8X,
1  ' DISTRIBUTION EFFICIENCY',F7.2,' %'/)
      END=(18.-TA)*(24.-TIME)
      IF(END.LT.0.)END=0.
      DGHR18=DGHR18+END
      DGDAY18=DGHR18/24.
      WRITE(20,85)DGDAY18
85  FORMAT(/5X,F8.3,' DEGREE DAYS OVER WHOLE DAY REL. 18degC')
      STOP
      END

```



APPENDIX 3

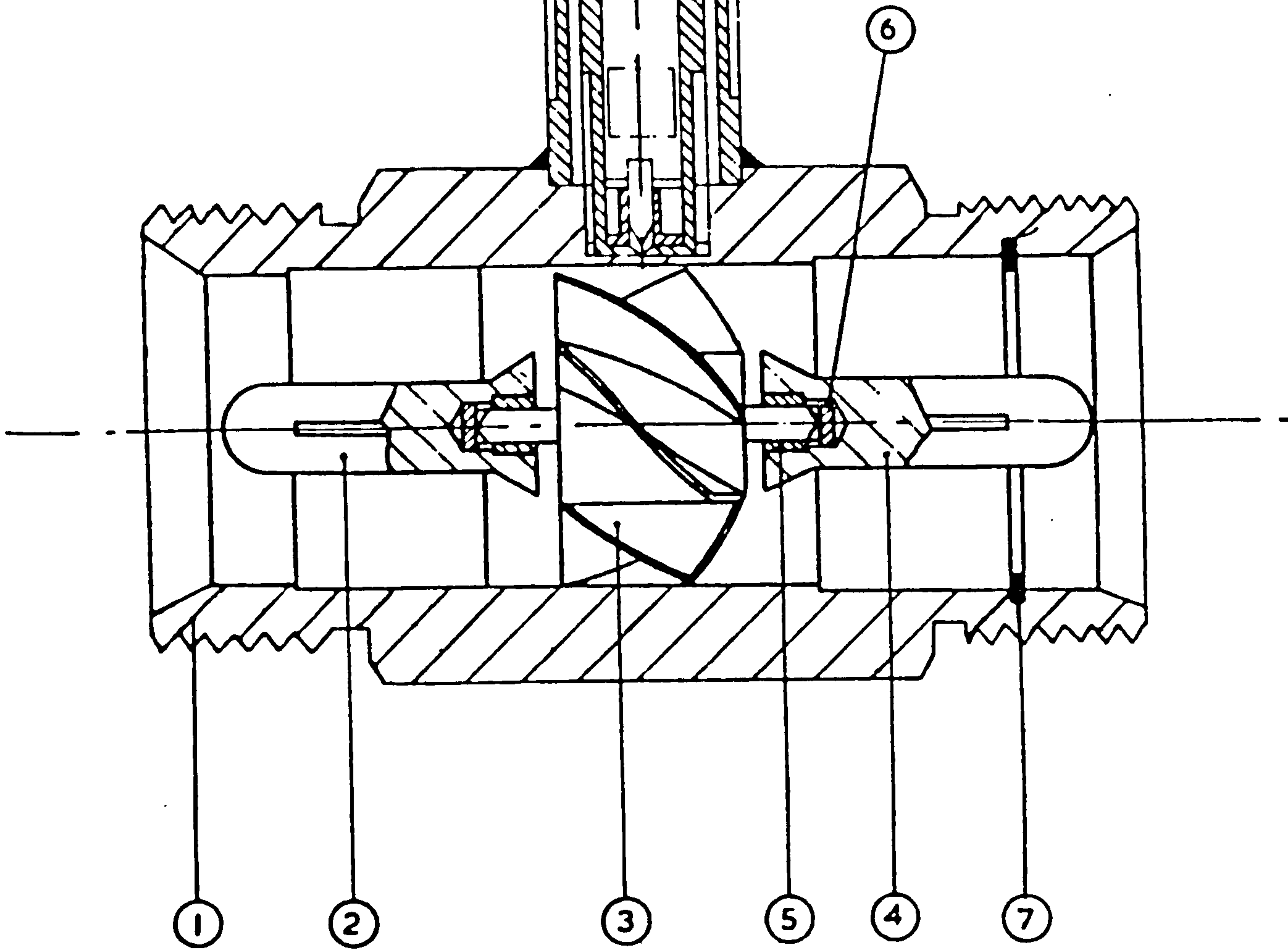
(RELEVANT TO PART III)

- 3.1 (a) Fig.3.7, manufacturer's drawing of electromagnetic flowmeter.
- (b) Manufacturer's test certificate for flowmeter.
- (c) Fig.3.8, rig calibration of flowmeter.
- 3.2 The graphs (nos. 1 - 44) relevant to the analysis of the Part III discharge tests.

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CANNON CONNECTOR  
SHOWN FITTED

APPENDIX 3.1 (a)



- 1. - BODY.
- 2. - REAR BEARING SUPPORT.
- 3. - ROTOR & SHAFT ASSEMBLY
- 4. - FRONT BEARING SUPPORT.
- 5.- SLEEVE BEARING.
- 6. - END STONE.
- 7. - CIRCLIP.

FIG 3.7: GENERAL ASSEMBLY - FLOWMETERS NOMINALLY SIZED  $\frac{1}{2}$ in-2in (12-50mm)

HYDRIL UK LIMITED  
 FOT SYSTEMS DIVISION

TEST CERTIFICATE

CUSTOMER: CRANFIELD INSTITUTE

DATE: 3.6.85

WORKS ORDER NO.: 850335

METER TYPE: B/.5/4

TEST VISCOSITY: 1 CS

BEARINGS: TC/STELL

SPECIFIC GRAVITY: .998

SERIAL NO.: 35954 H2

TOTAL PULSES	PULSES PER IMP GALL	FREQUENCY HZ	RATE OF FLOW IMP GAL /MIN
	11355.908	100.000	.528
	11451.374	250.000	1.309
	11392.276	500.000	2.633
	11365.000	750.000	3.959
	11342.270	1000.000	5.289

AVERAGE PULSES PER IMP GALL 11381.36

CALIBRATED BY

FREQUENCY FOR 4 IMP GALLS /MIN = 758 HZ



REMARKS

\_\_\_\_\_  
 \_\_\_\_\_  
 \_\_\_\_\_  
 \_\_\_\_\_

APPENDIX 3.1(b)



APPENDIX 3.1(c)

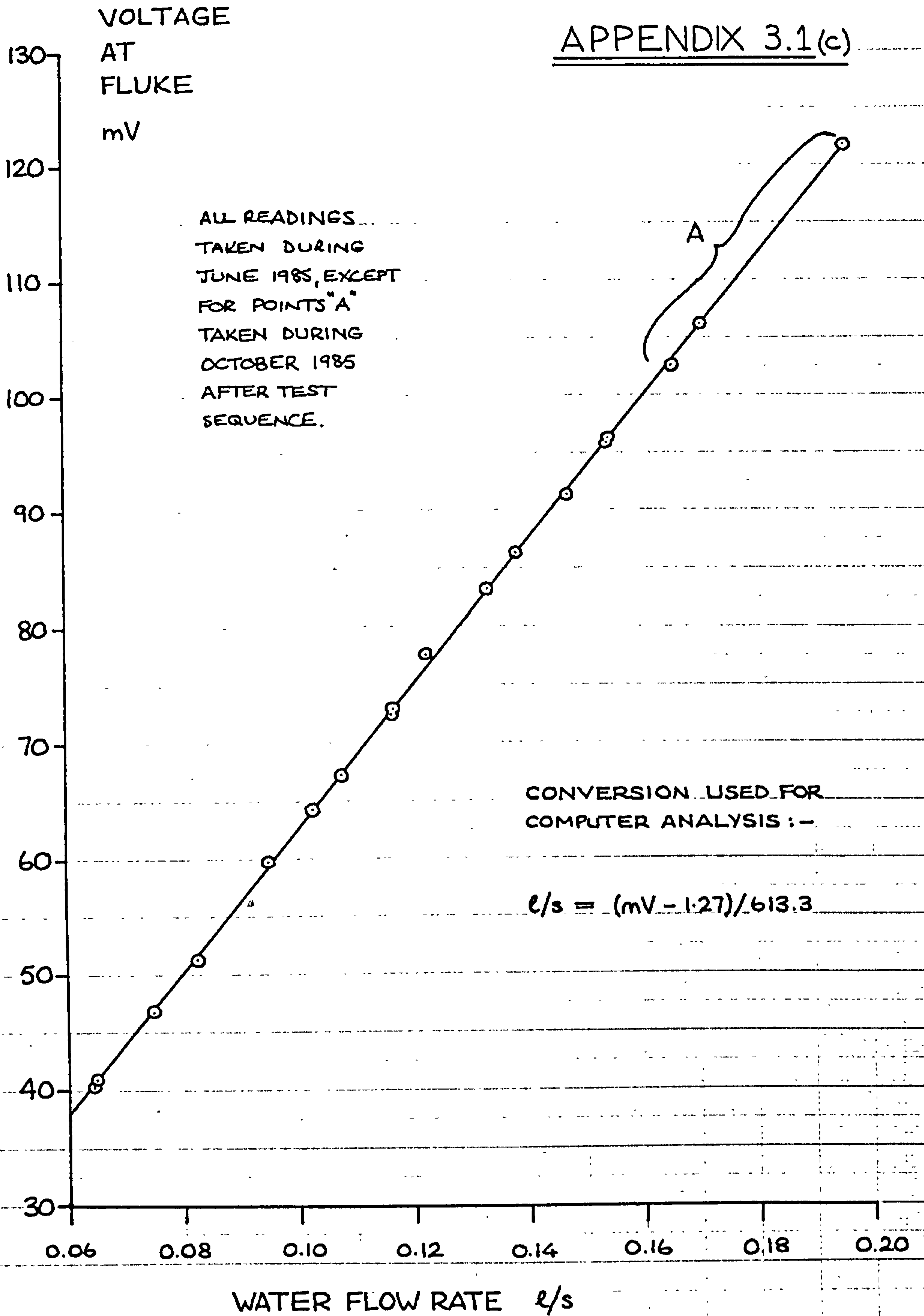


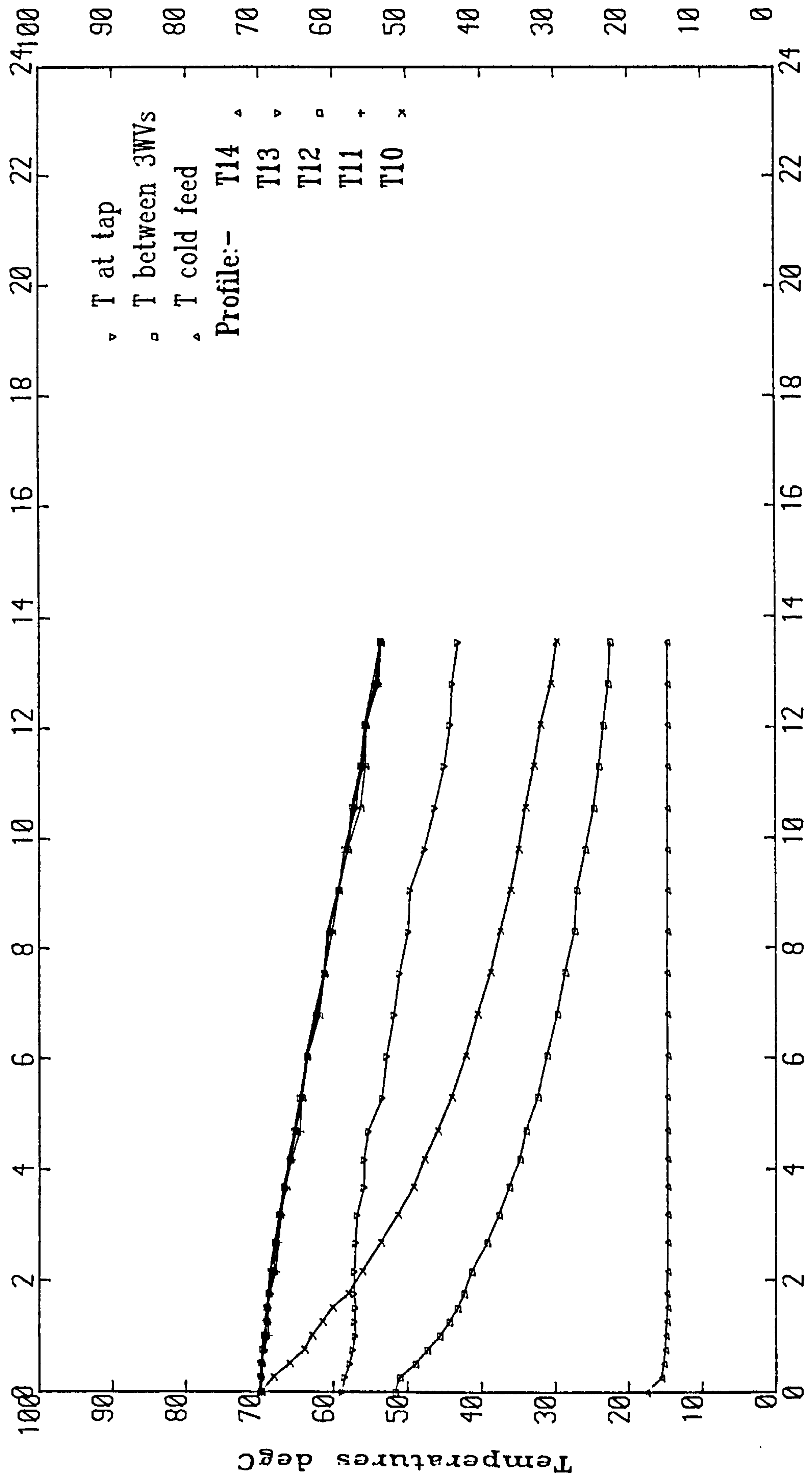
FIG 3.8 : CALIBRATION OF WATER TURBINE

### Appendix 3.2

As on Table 3.1, graphs nos. 1-44 corresponding to the analysis of Part III are here presented in the following order:-

Graph nos.	Test Title	Comments
1-2	Storage tank profile	-
3-5	Repeatability	c.f.nos.18-20
6-32	H.E. combinations	Upper/Lower 6-8 SMALL/SMALL 9-11 SMALL/MEDIUM 12-14 SMALL/LARGE 15-17 MEDIUM/SMALL 18-20 MEDIUM/MEDIUM 21-23 MEDIUM/LARGE 24-26 LARGE/SMALL 27-29 LARGE/MEDIUM 30-32 LARGE/LARGE
33-41	Lower H.E. position	33-35 bottom posn. 36-38 middle posn. 39-41 top posn.
42-44	"	

TEMP. PROFILE : 16 08 85

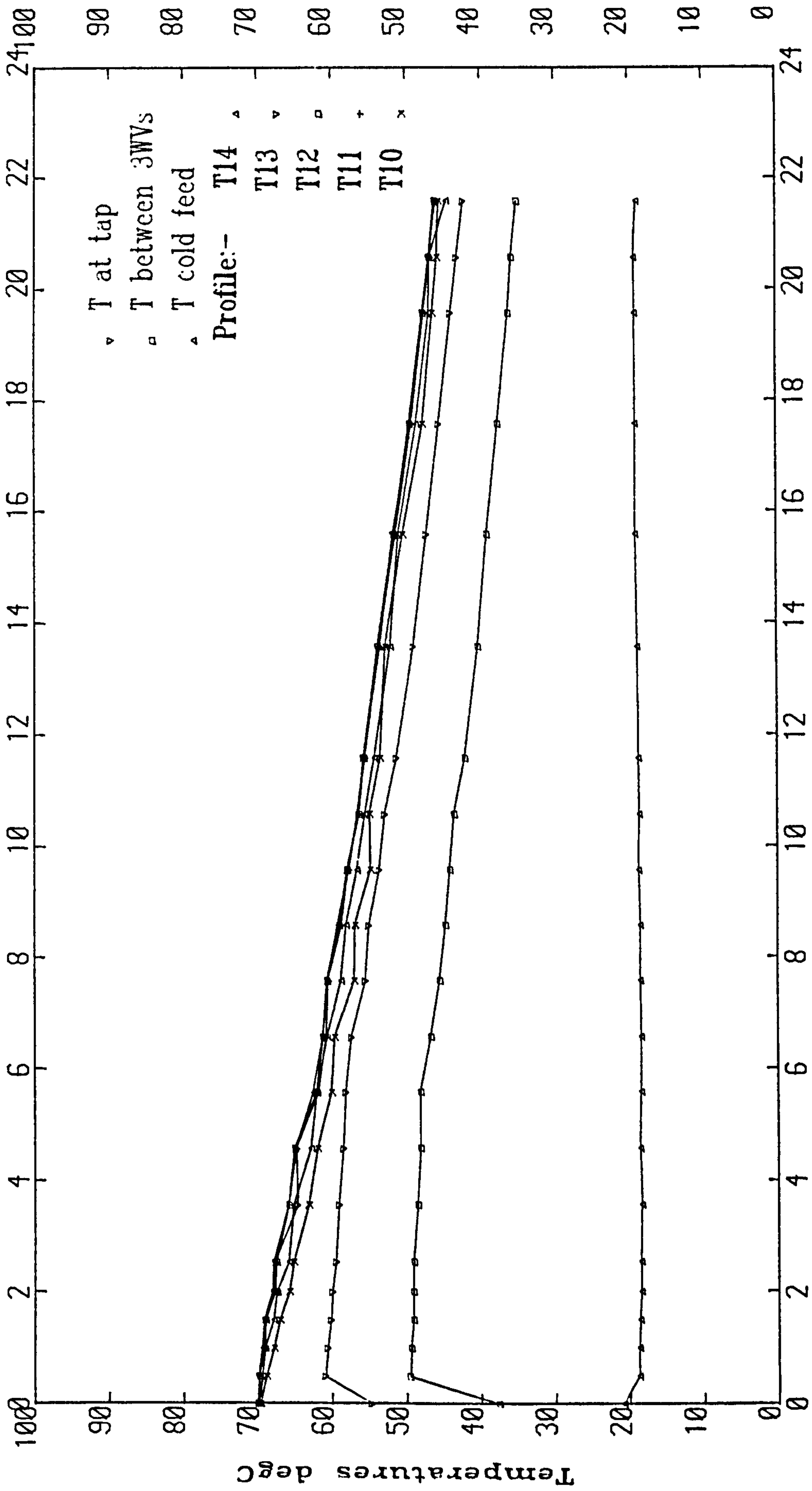


Minutes into discharge

GRAPH No. 1



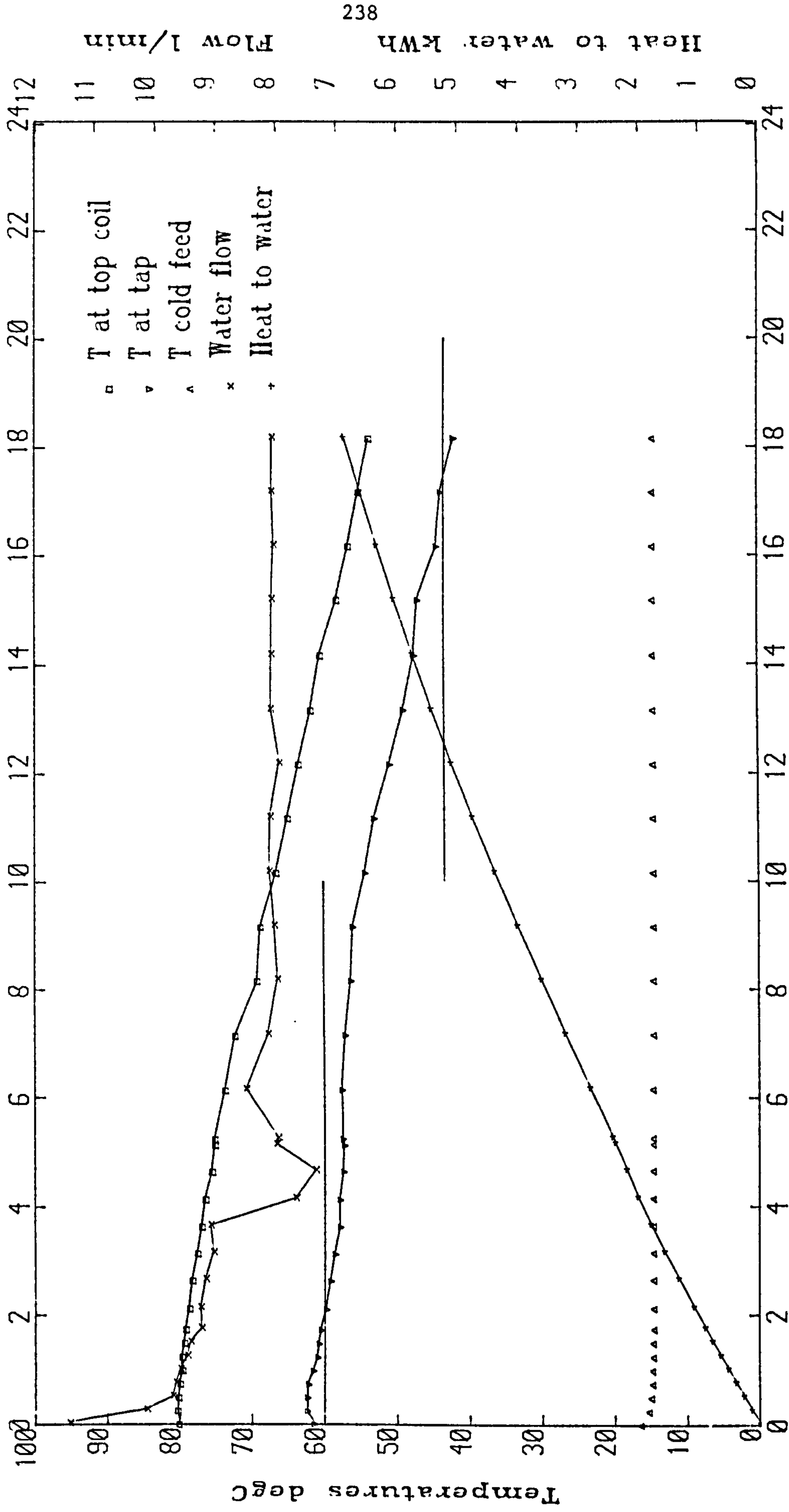
TEMP. PROFILE : 06 08 85



Minutes into discharge

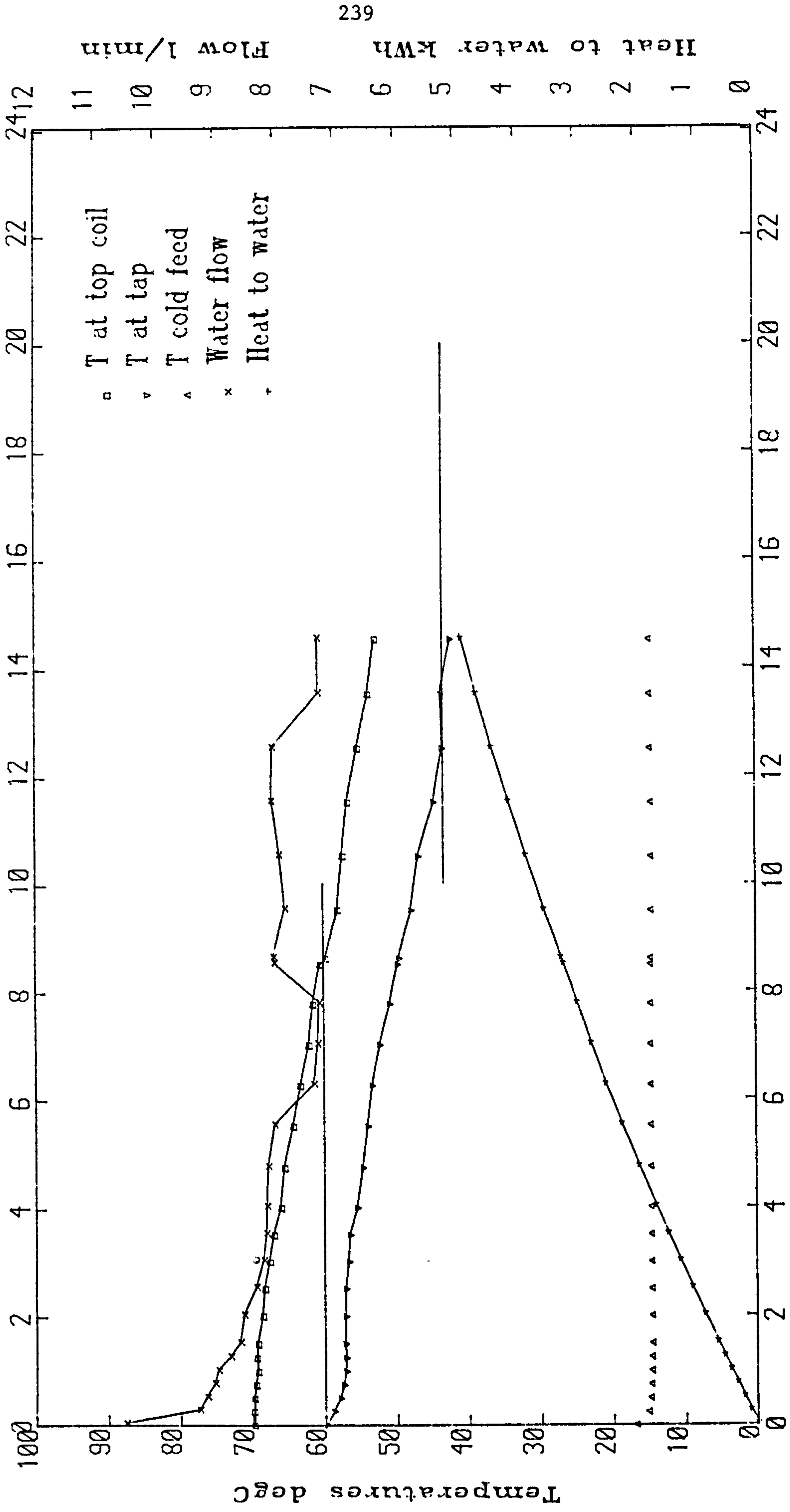
GRAPH No. 2

RIG PLOT : 15 08 85



GRAPH No. 3

RIG PLOT : 15 08 85

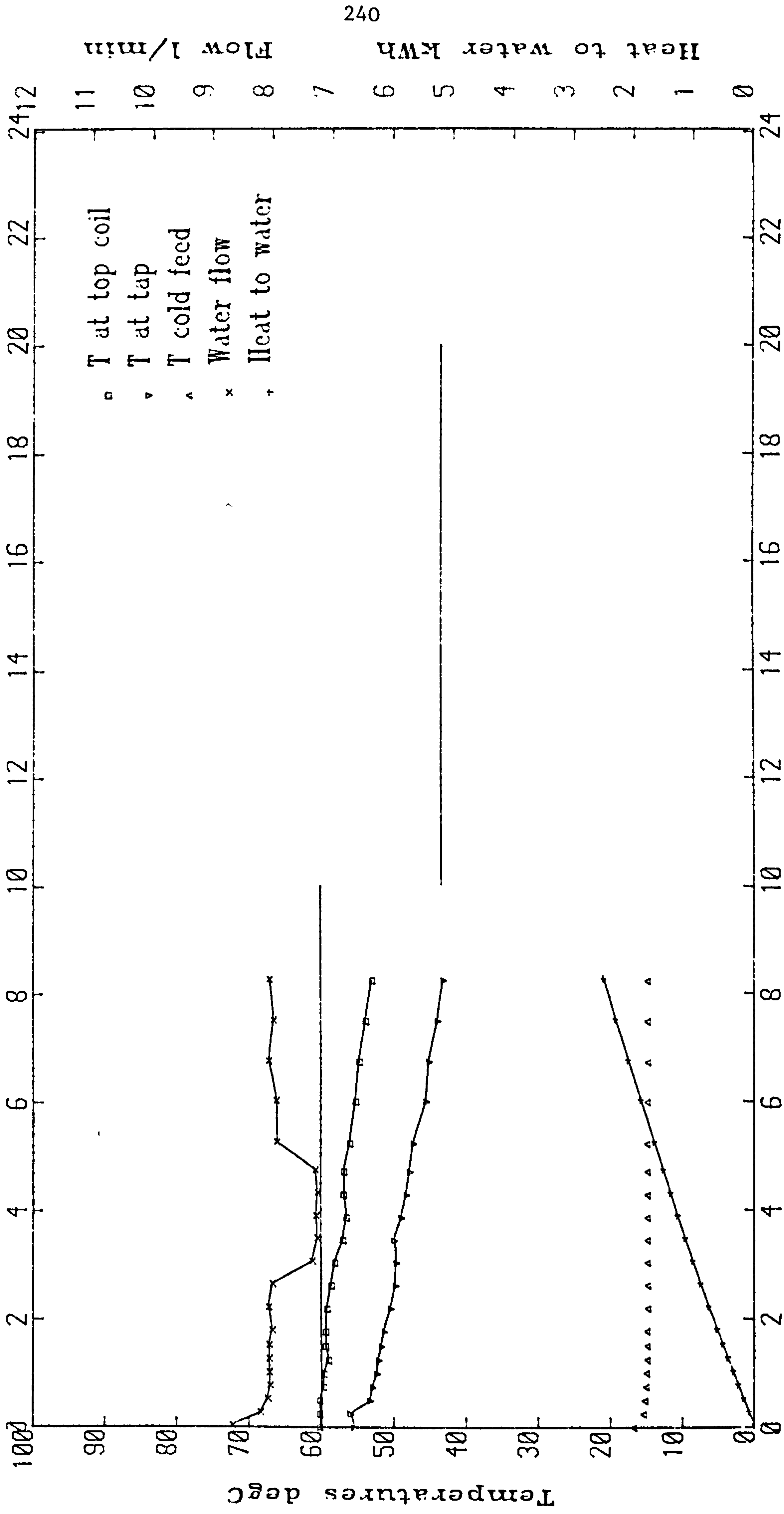


GRAPH No. 4

Minutes into discharge



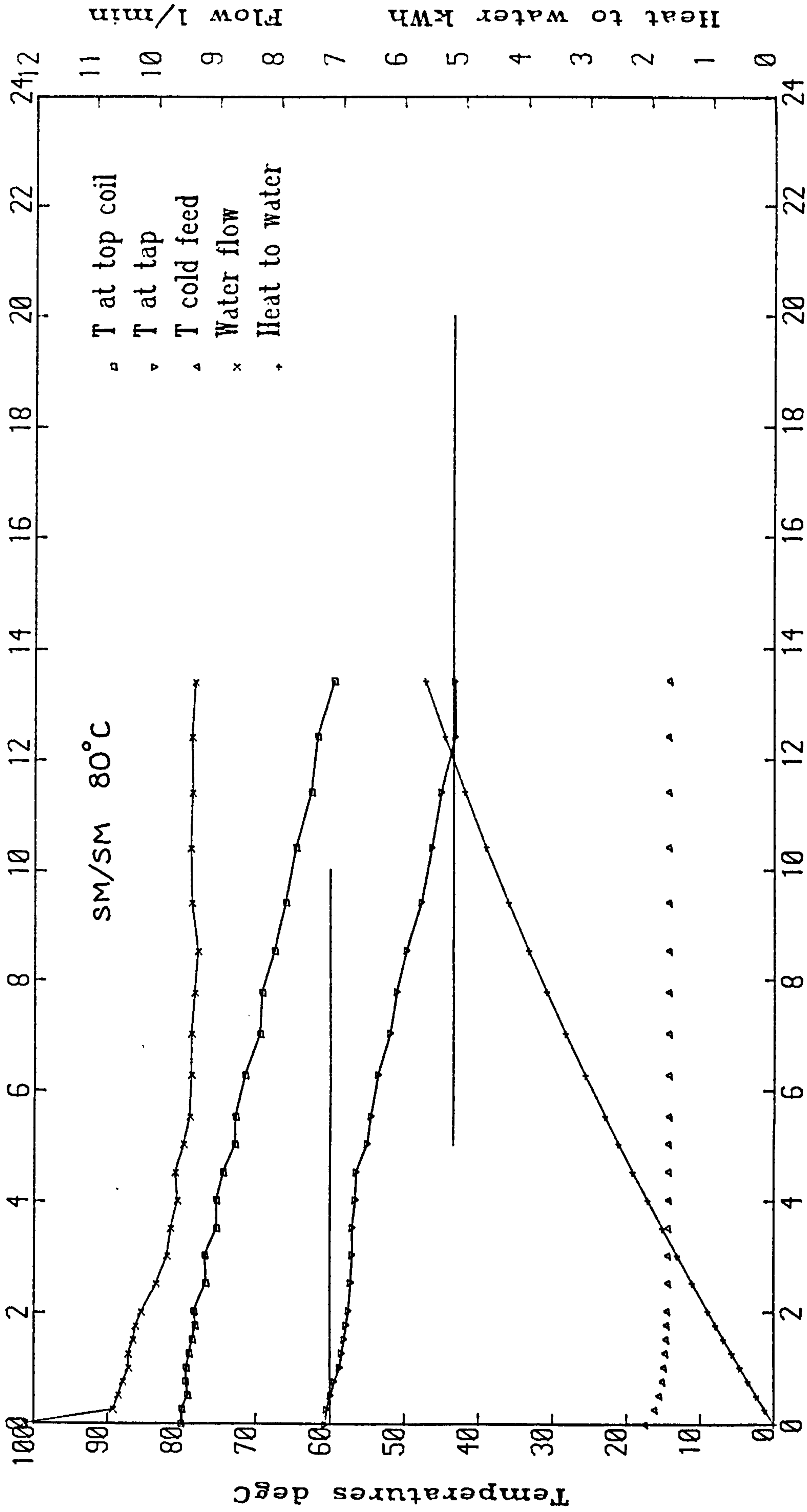
RIG PLOT : 15 08 85



GRAPH No. 5

Minutes into discharge

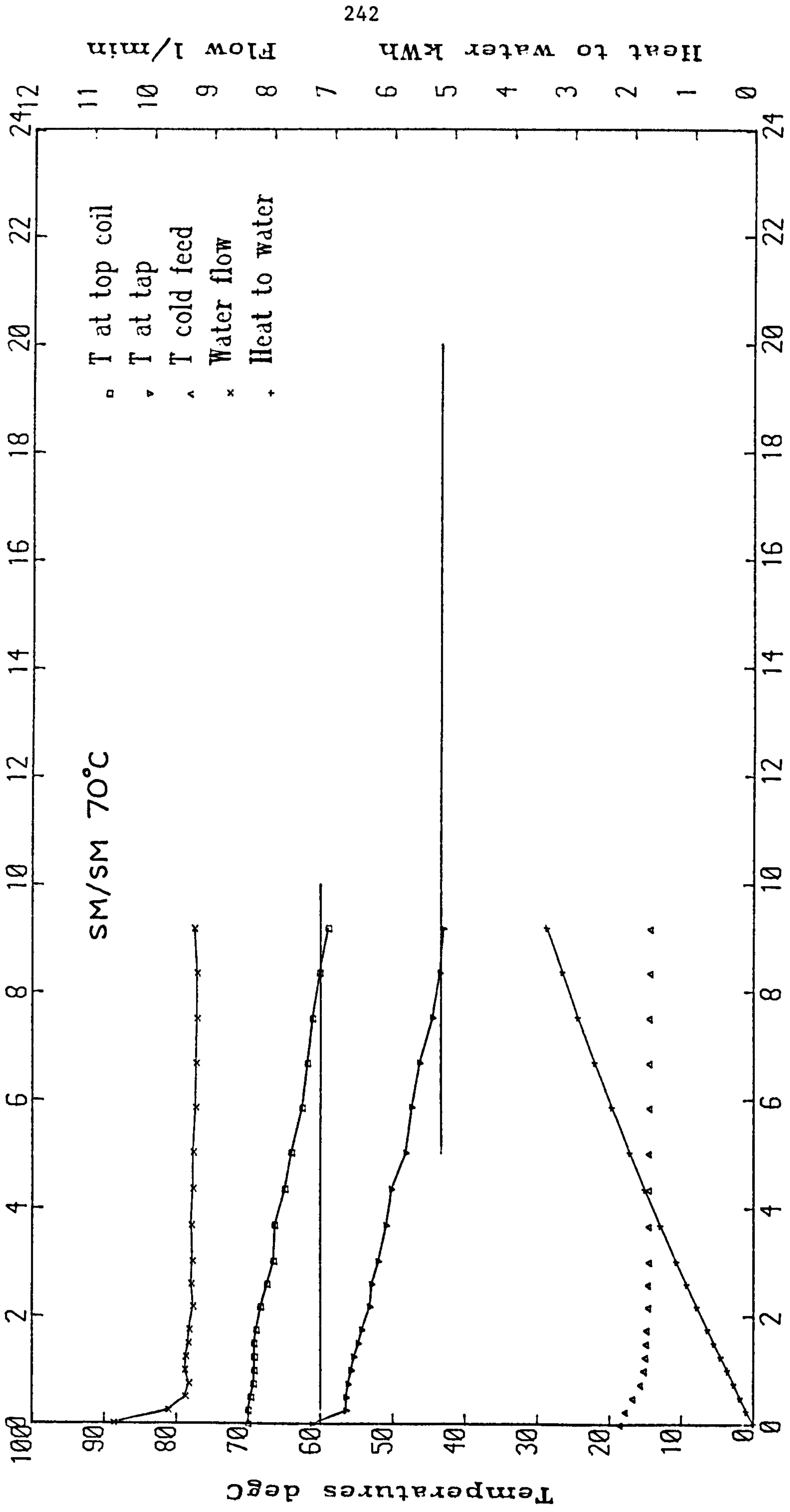
RIG PLOT : 05 09 85



Minutes into discharge

GRAPH No. 6

RIG PLOT : 05 09 85

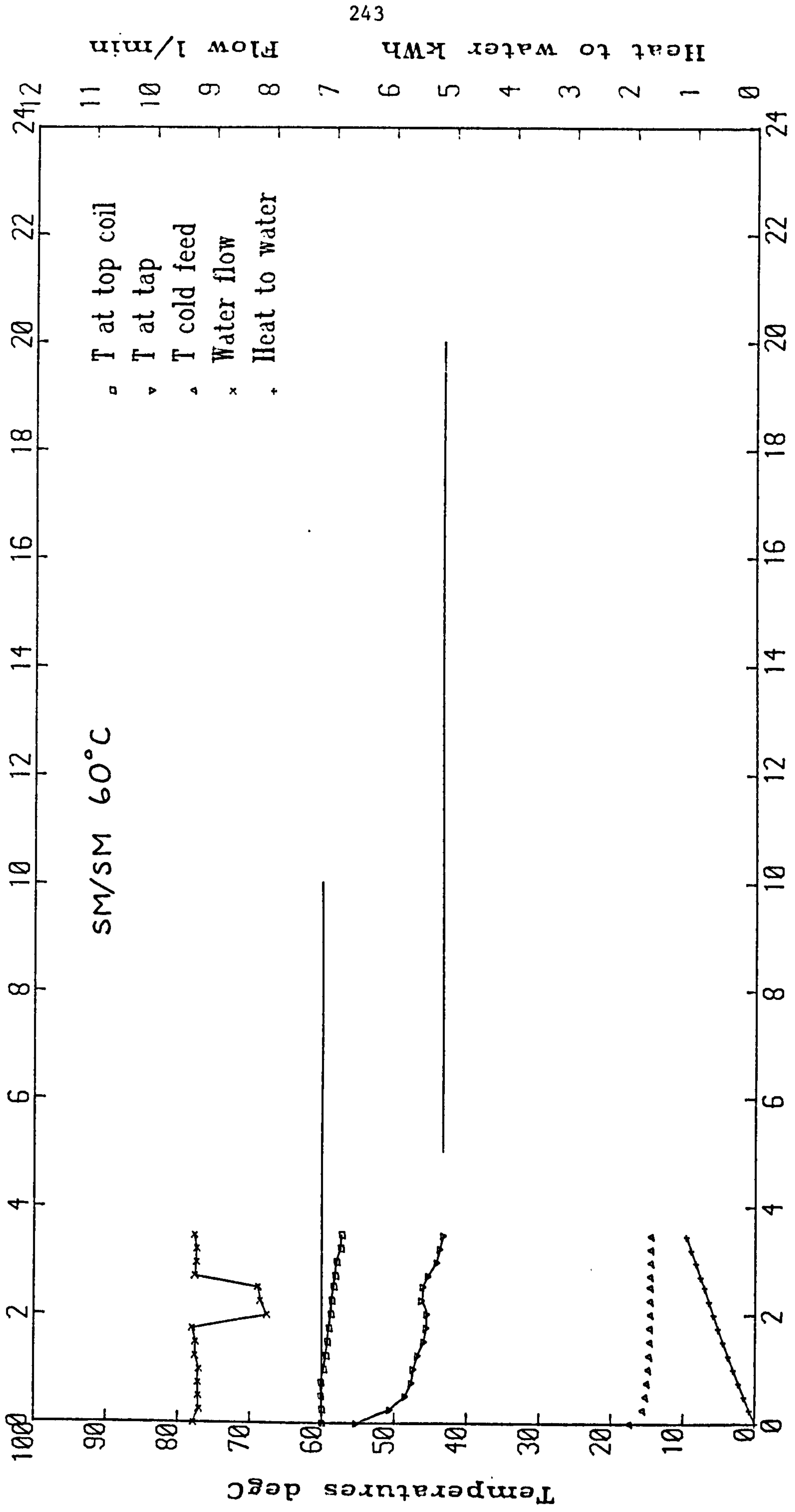


GRAPH No. 7

Minutes into discharge



RIG PLOT : 05 09 85



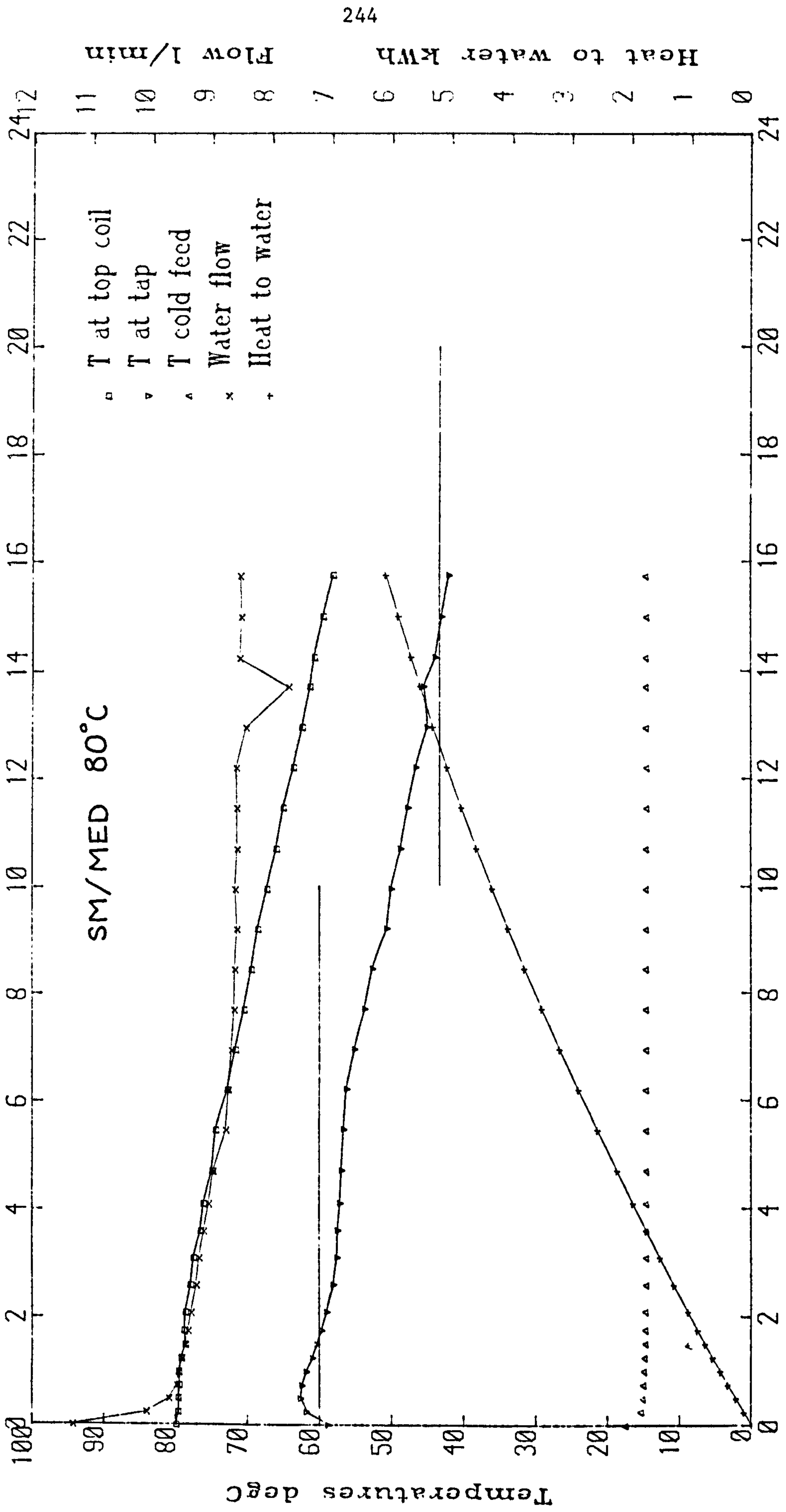
243

Minutes into discharge

GRAPH No. 8

RIG PLOT : 22 08 85

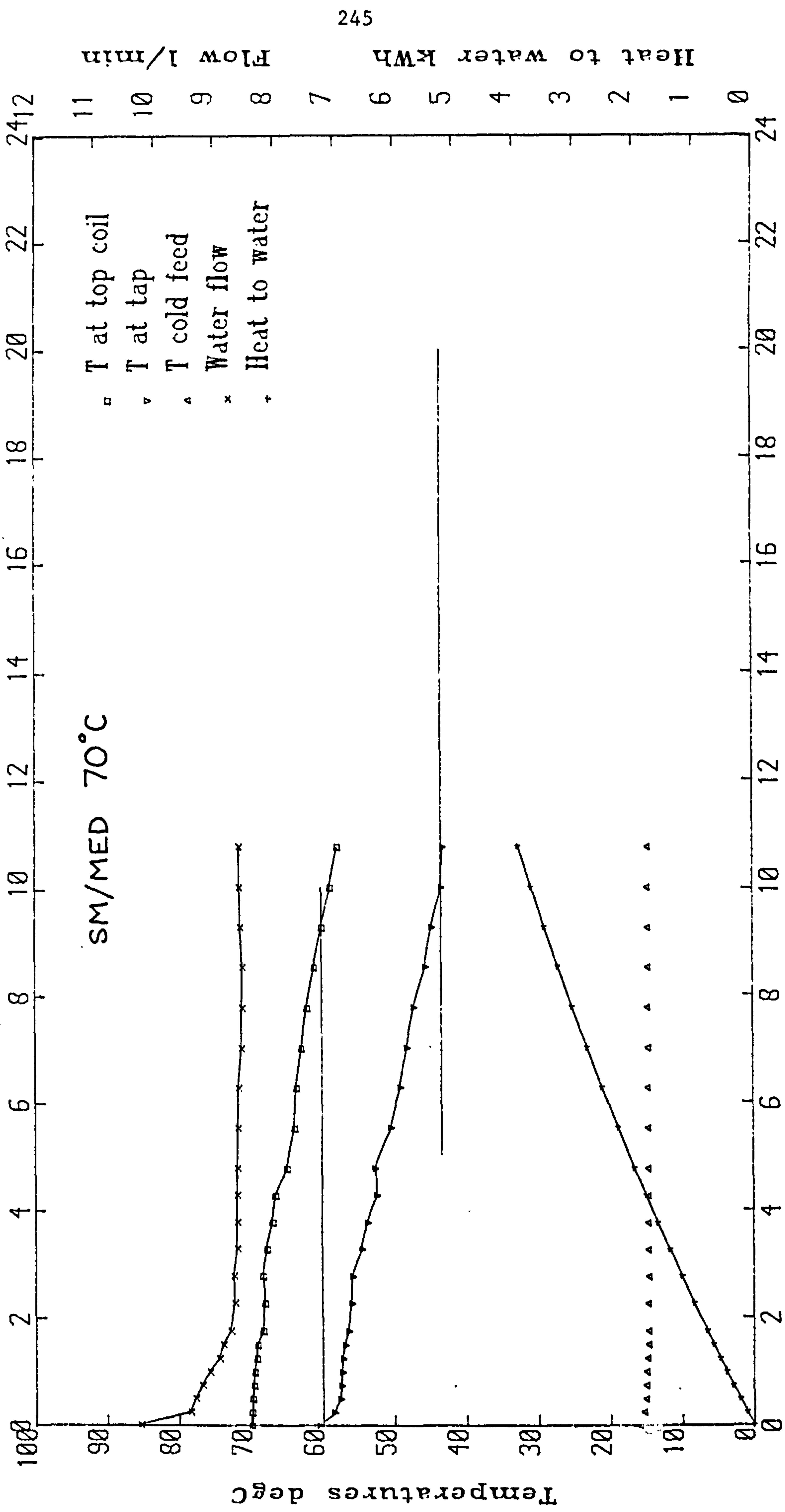
SM/MED 80°C



Minutes into discharge

GRAPH No.9

RIG PLOT : 23 08 85



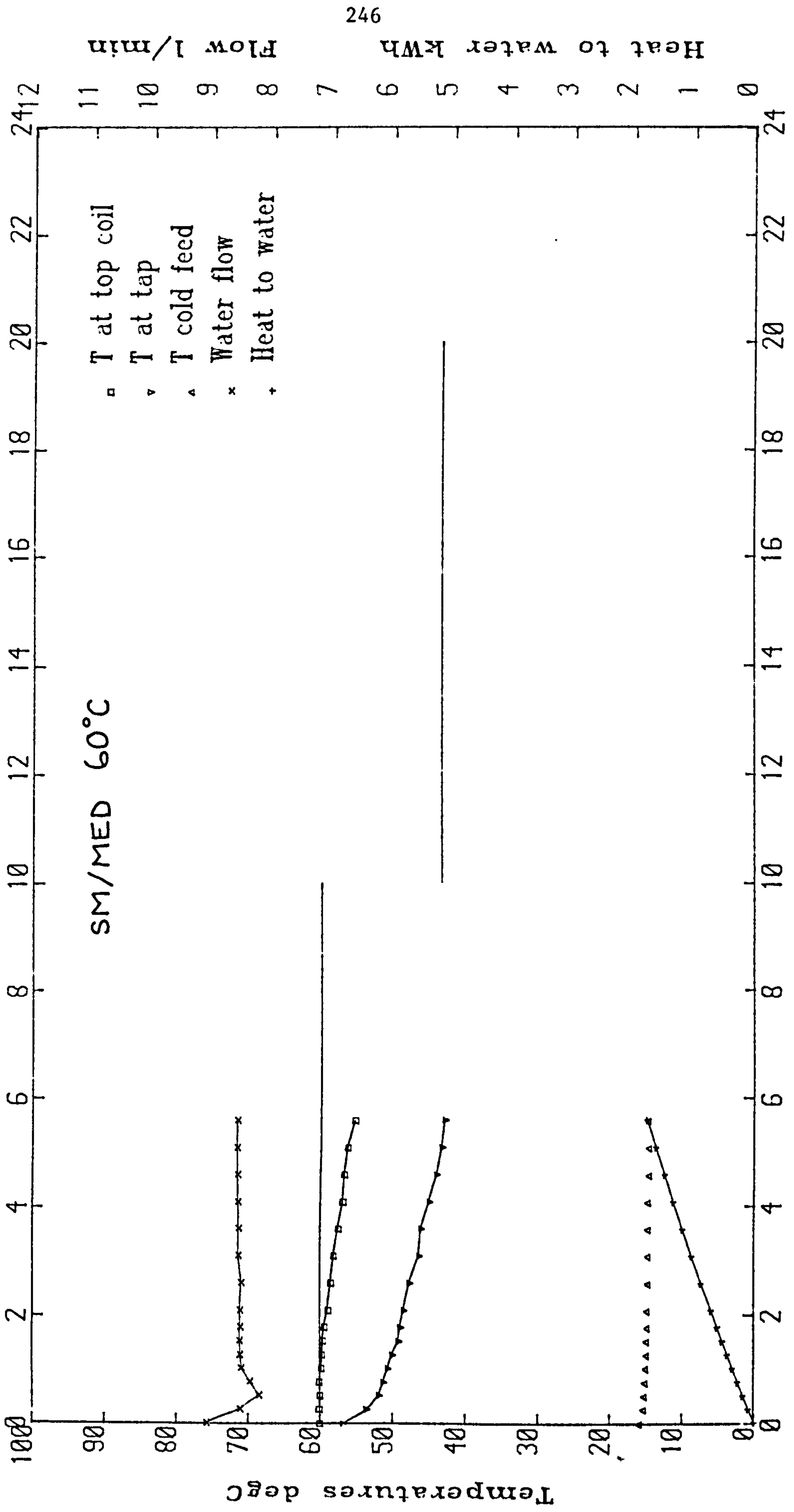
Minutes into discharge

GRAPH No. 10



RIG PLOT : 22 08 85

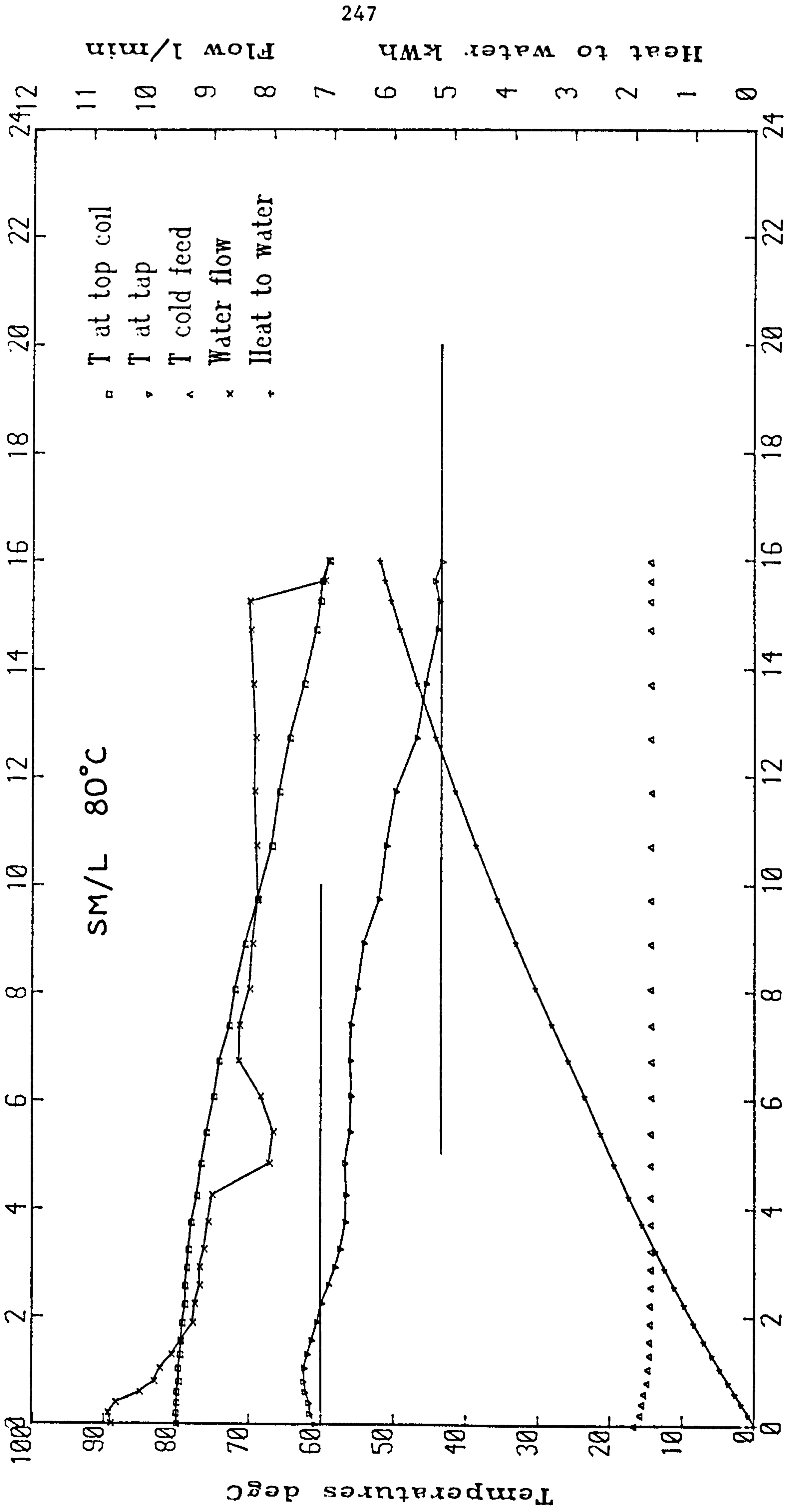
SM/MED 60°C



GRAPH No. 11

Minutes into discharge

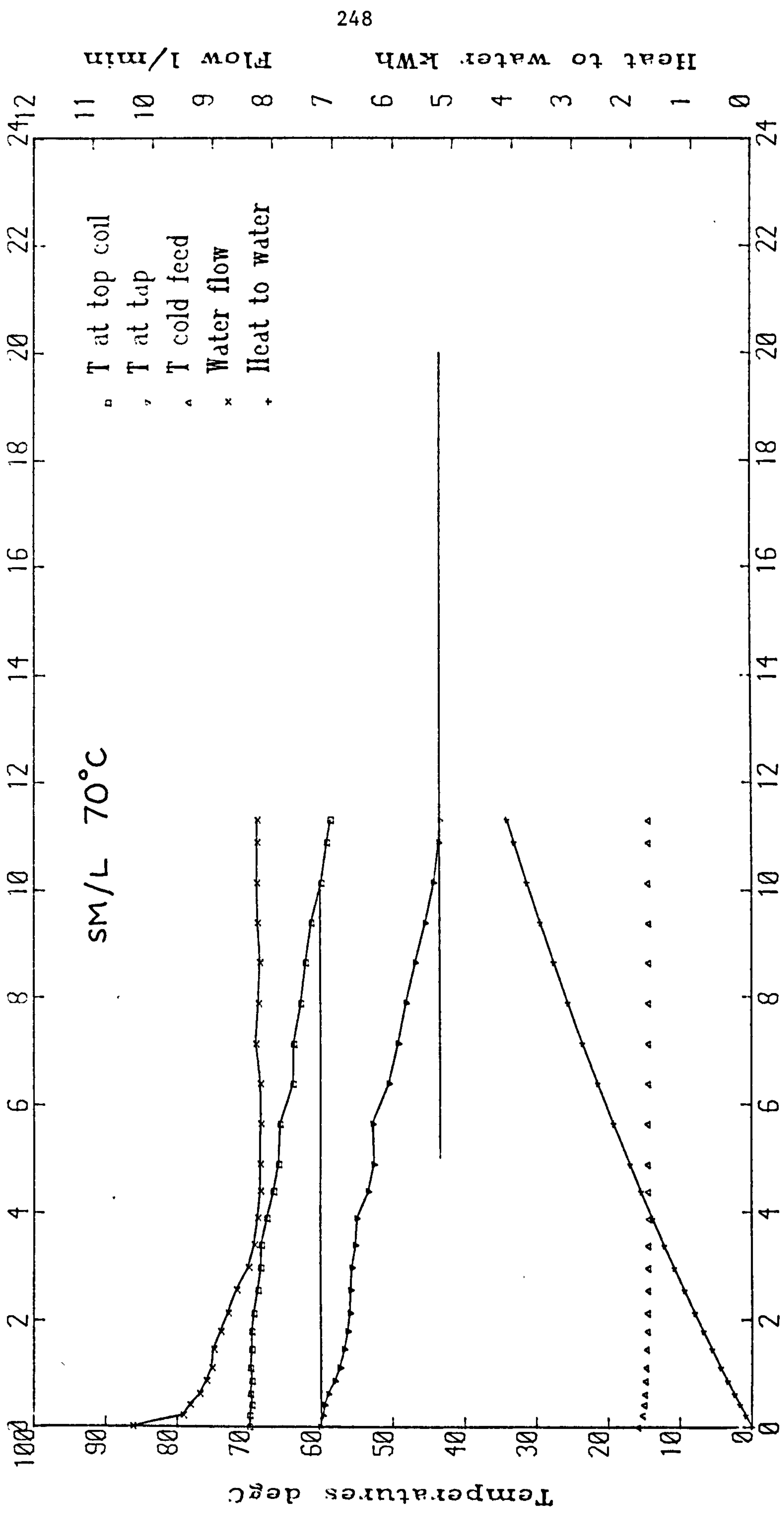
RIG PLOT : 20 09 85



Minutes into discharge

GRAPH No. 12

RIG PLOT : 20 09 85

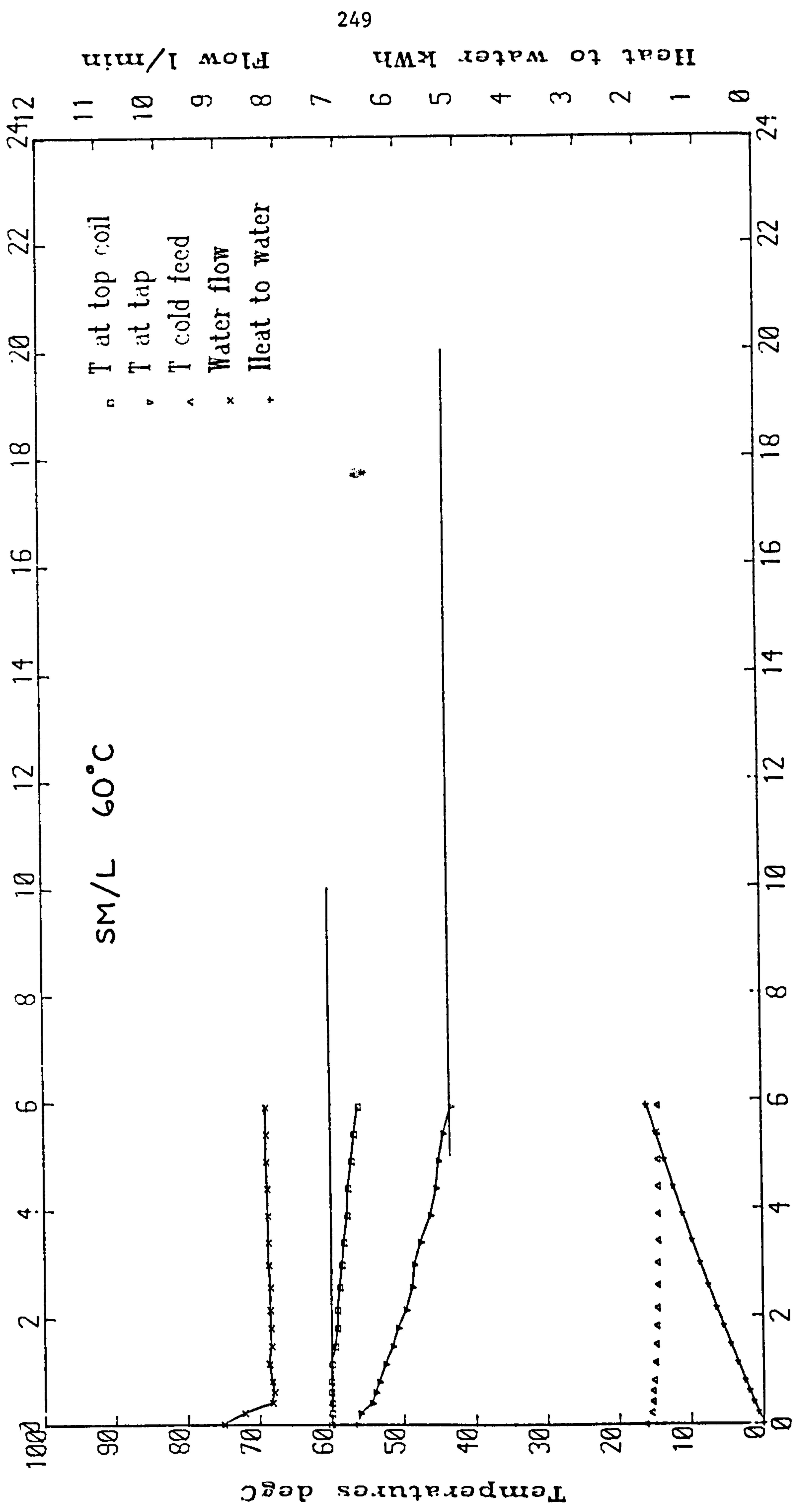


Minutes into discharge

GRAPH No. 13



RIG PLOT : 20 09 85

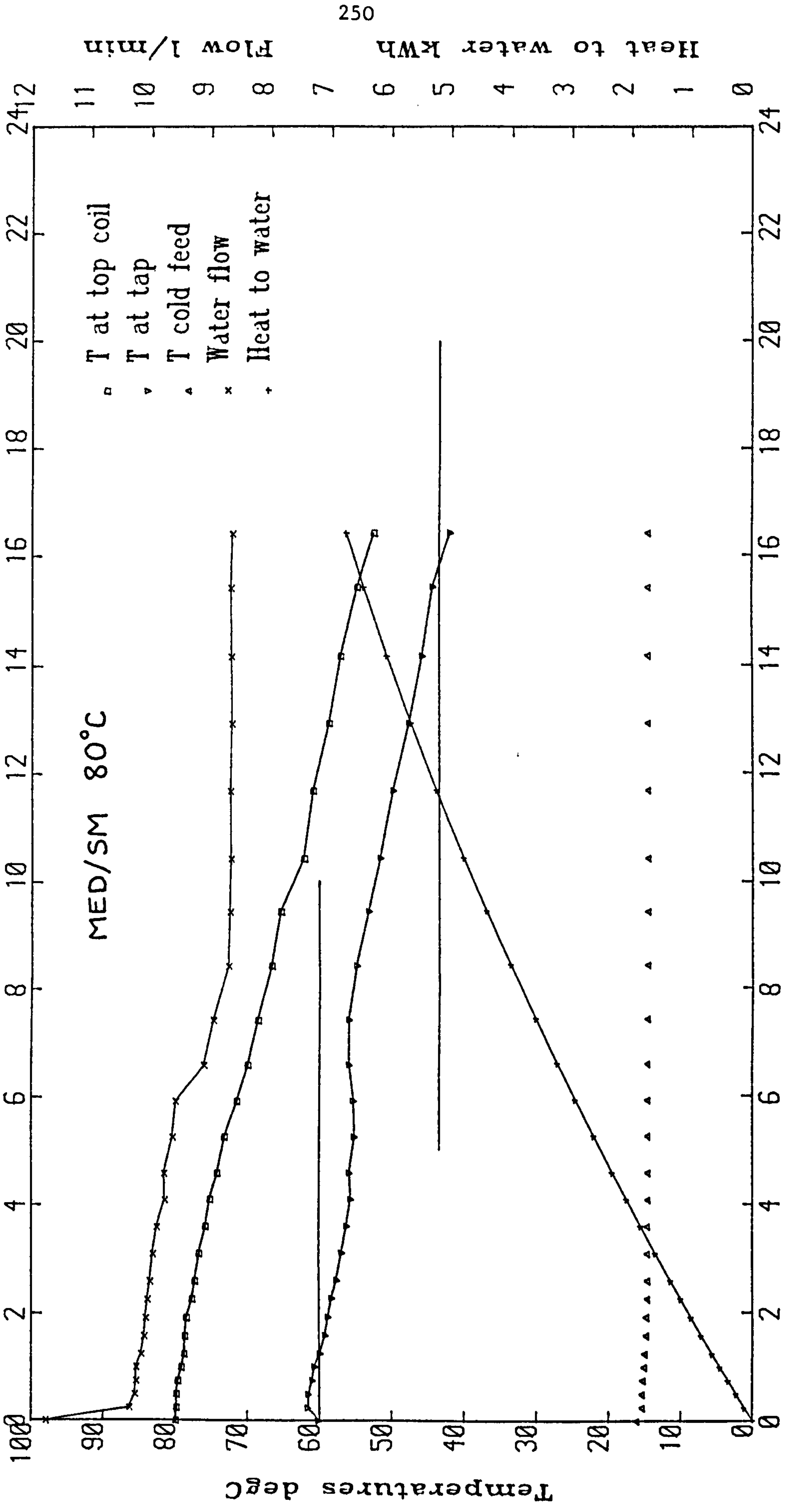


GRAPH No. 14

Minutes into discharge

RIG PLOT : 09 09 85

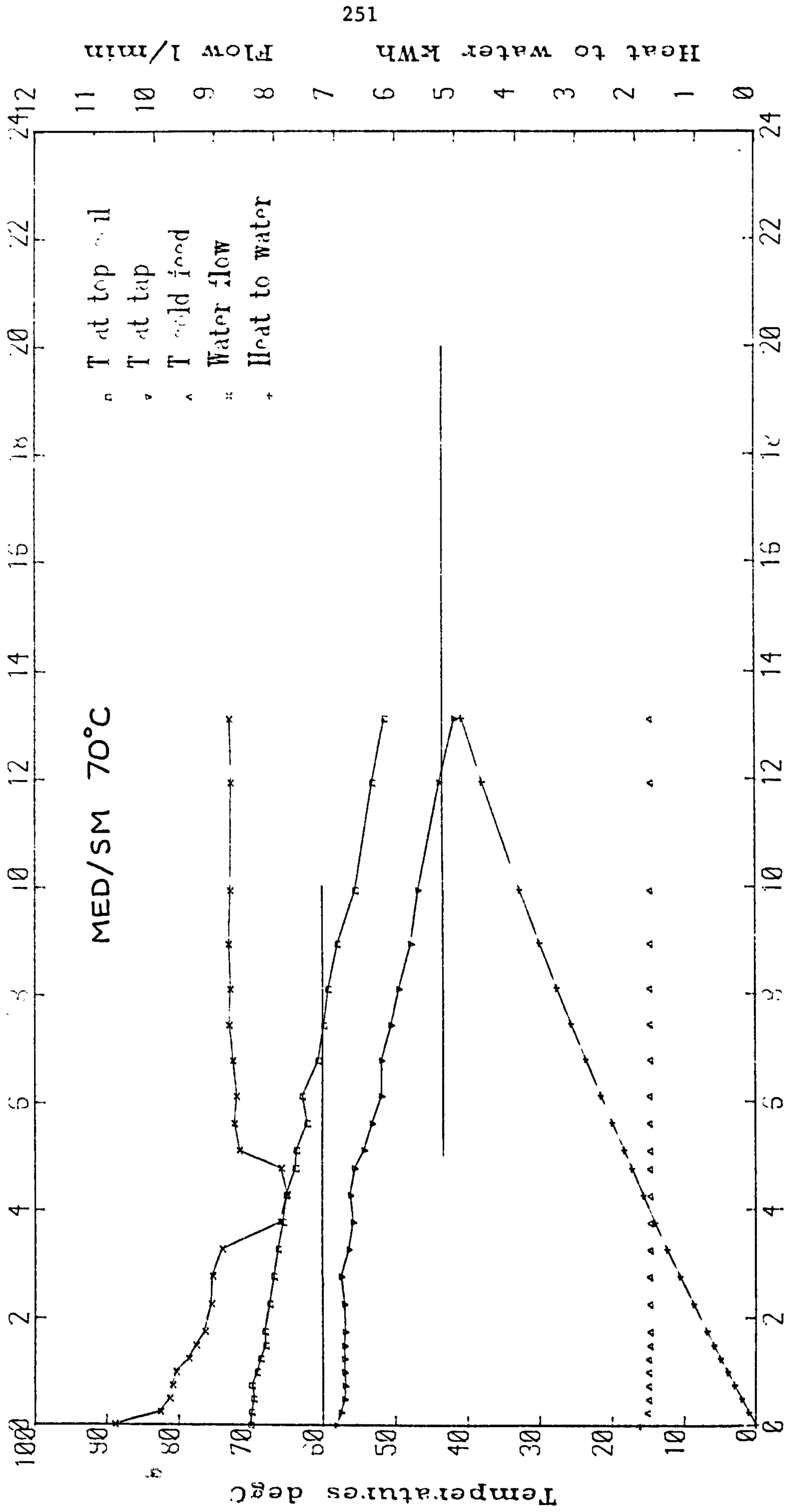
MED/SM 80°C



Minutes into discharge

GRAPH No.15

RIG PLOT : 06 09 85



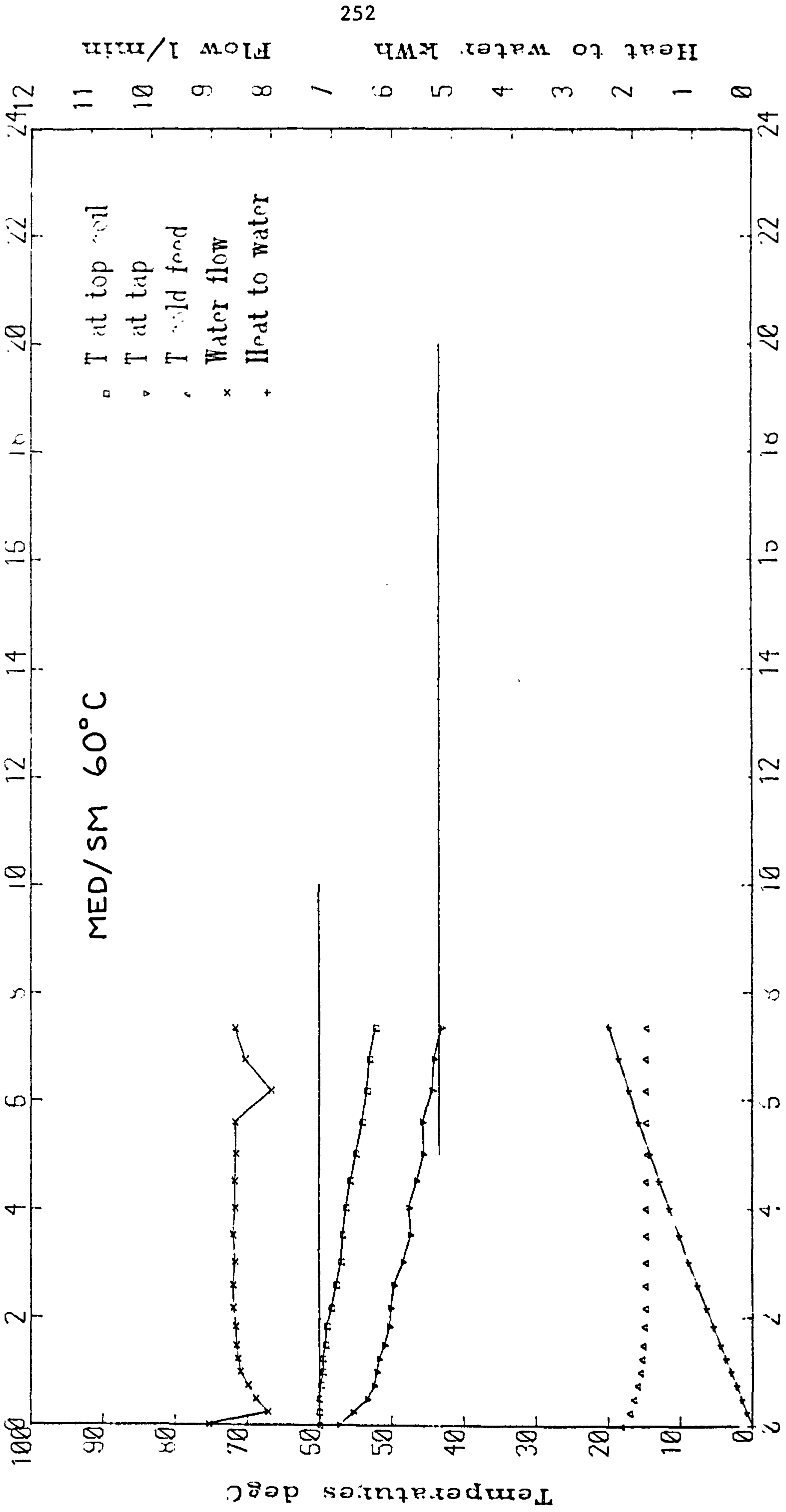
Minutes into discharge

GRAPH No. 16



RIG PLOT : 06 09 85

MED/SM 60°C

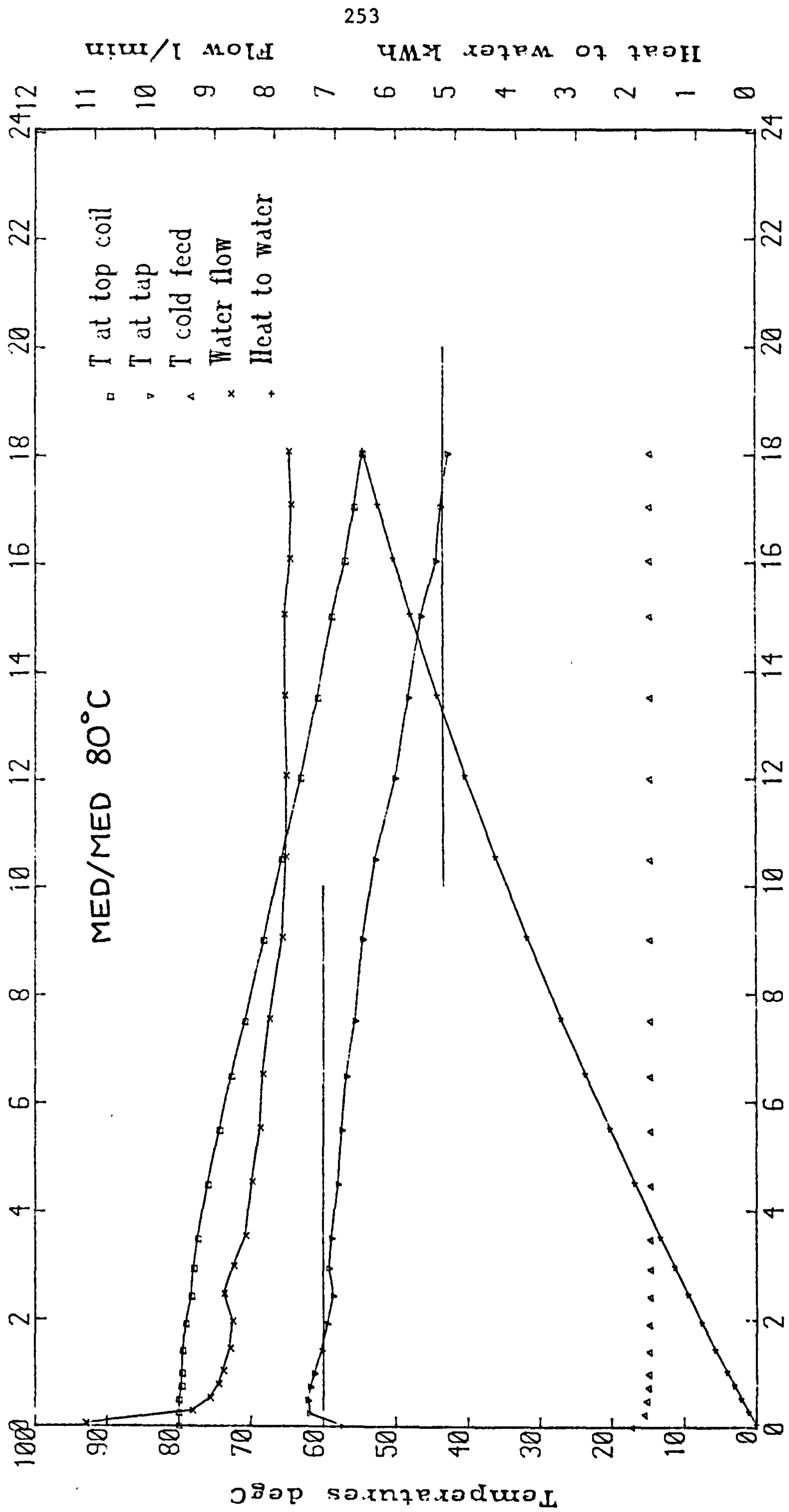


Minutes into discharge

GRAPH No. 17

RIG PLOT : 14 8 85M

MED/MED 80°C

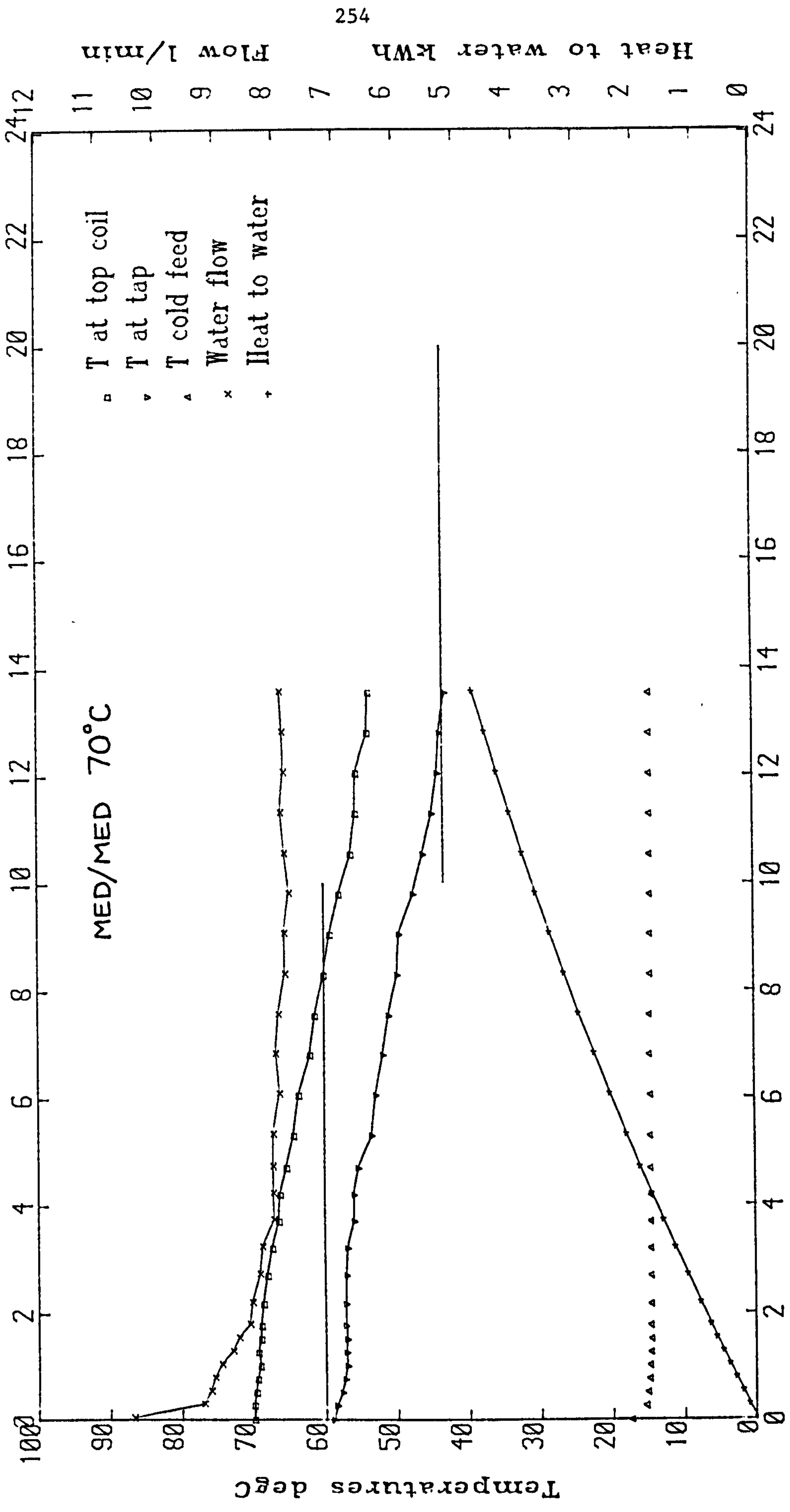


Minutes into discharge

GRAPH No.18

RIG PLOT : 16 08 85

MED/MED 70°C

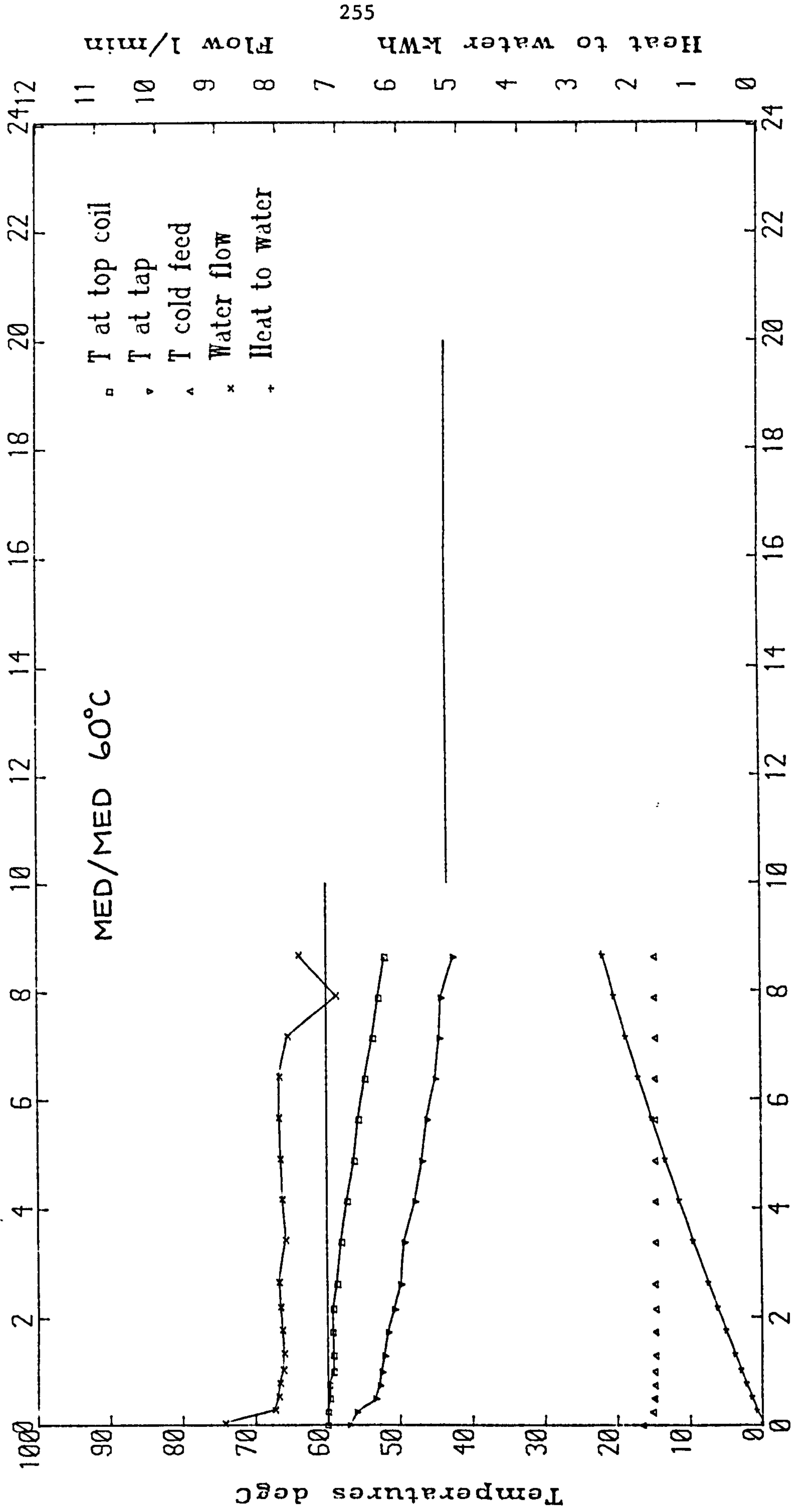


Minutes into discharge

GRAPH No. 19

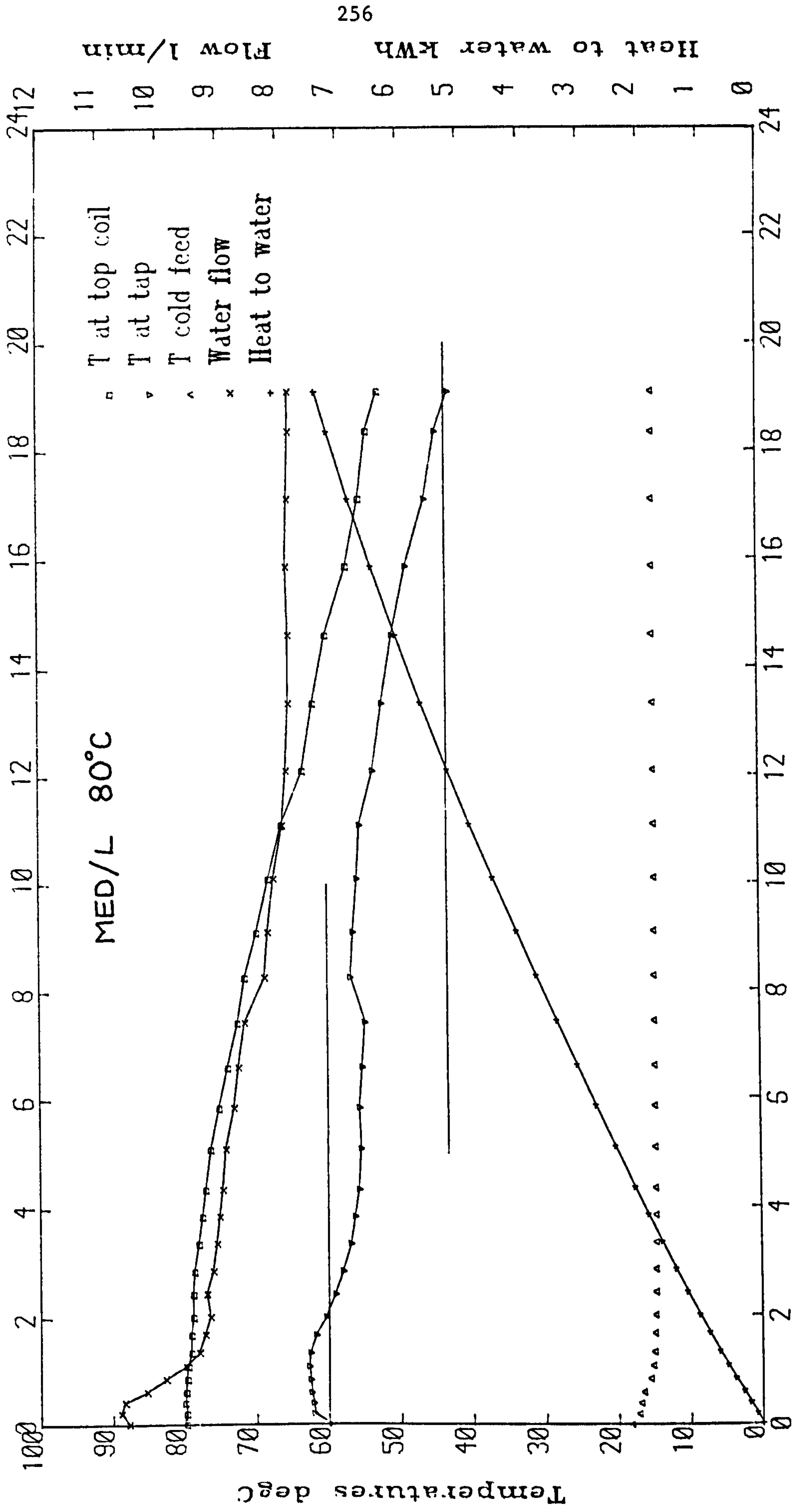


RIG PLOT : 15 08 85



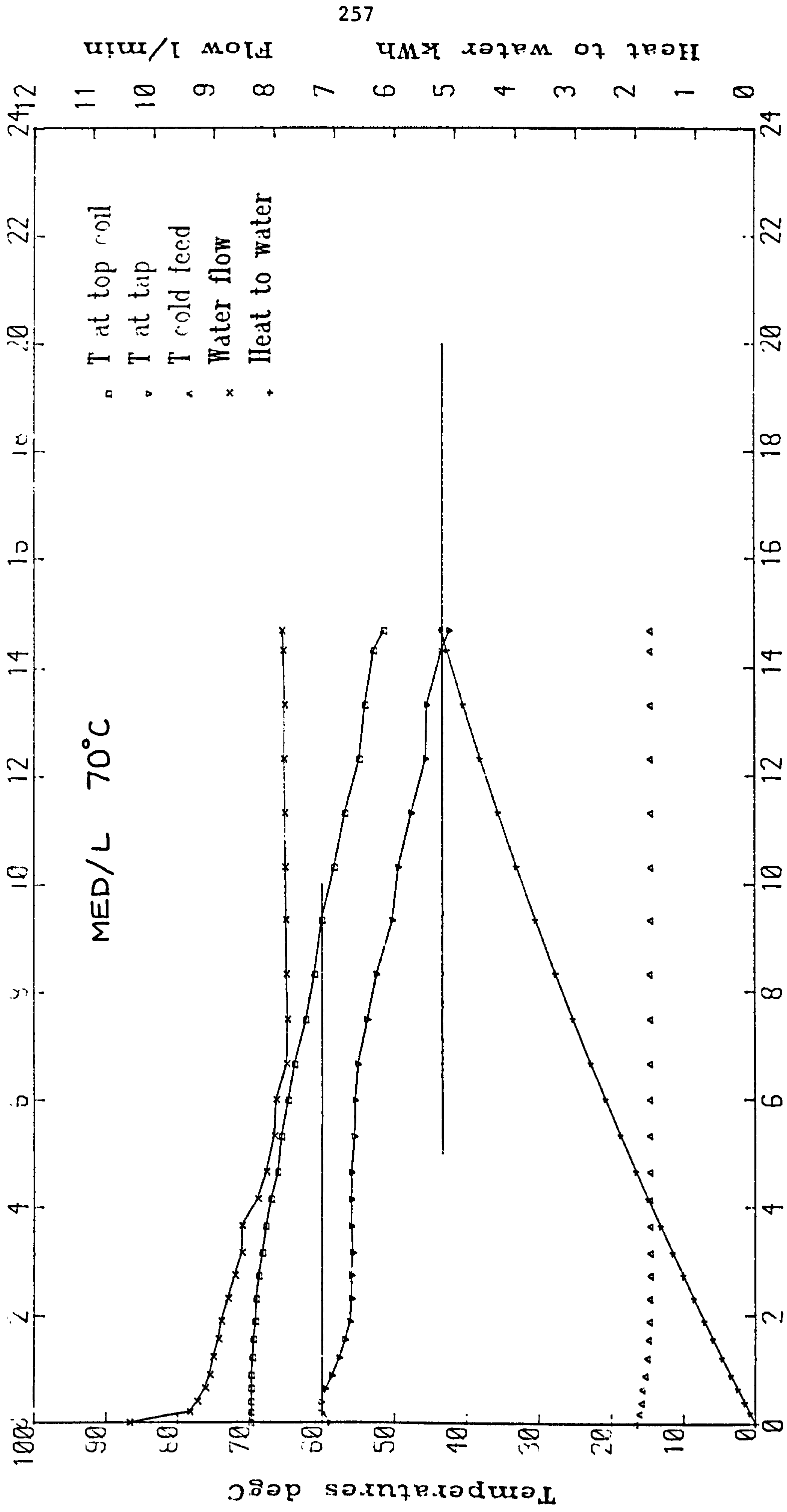
GRAPH No. 20

RIG PLOT : 17 09 85



GRAPH No.21

RIG PLOT : 18 09 85



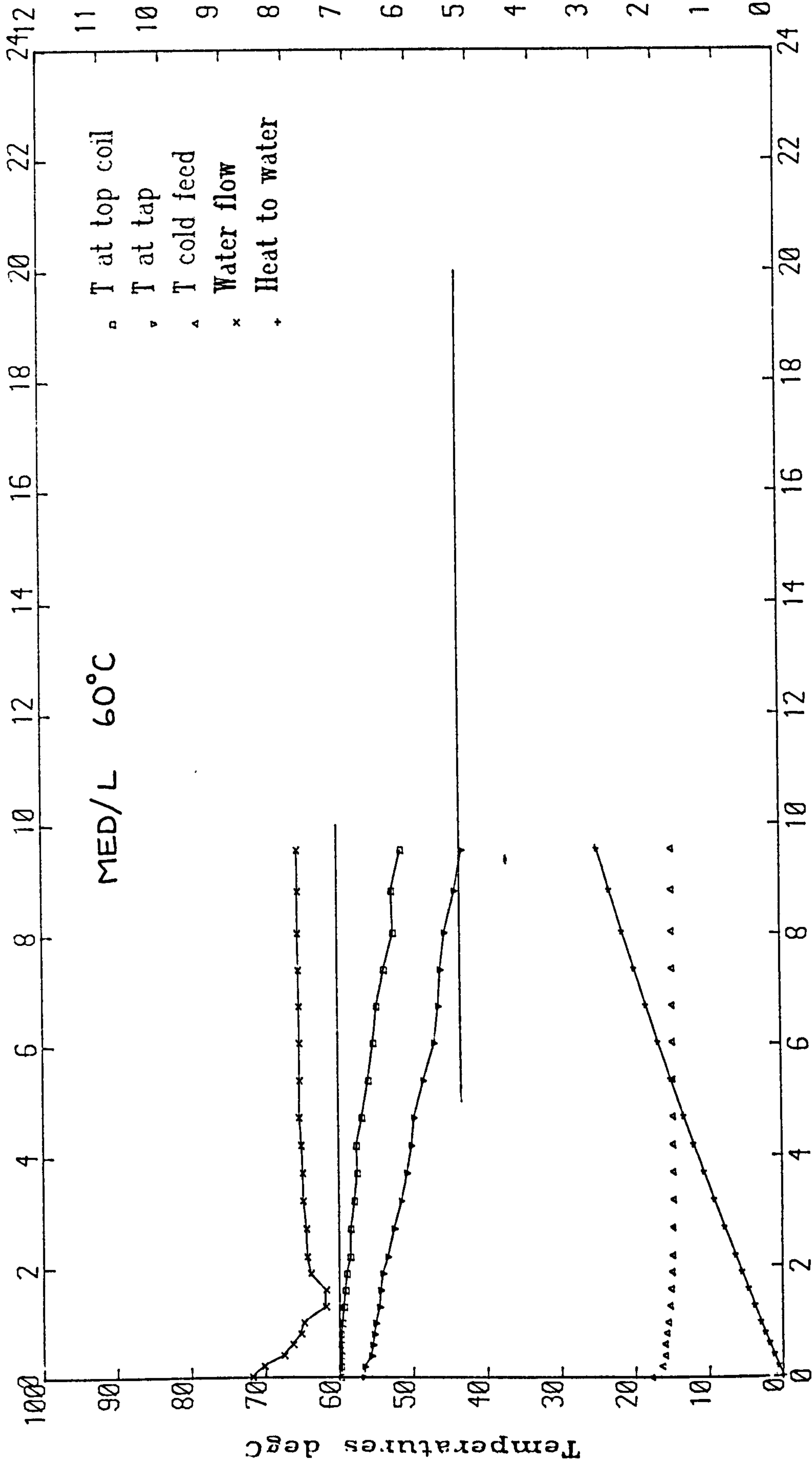
Minutes into discharge

GRAPH No. 22



RIG PLOT : 17 09 85

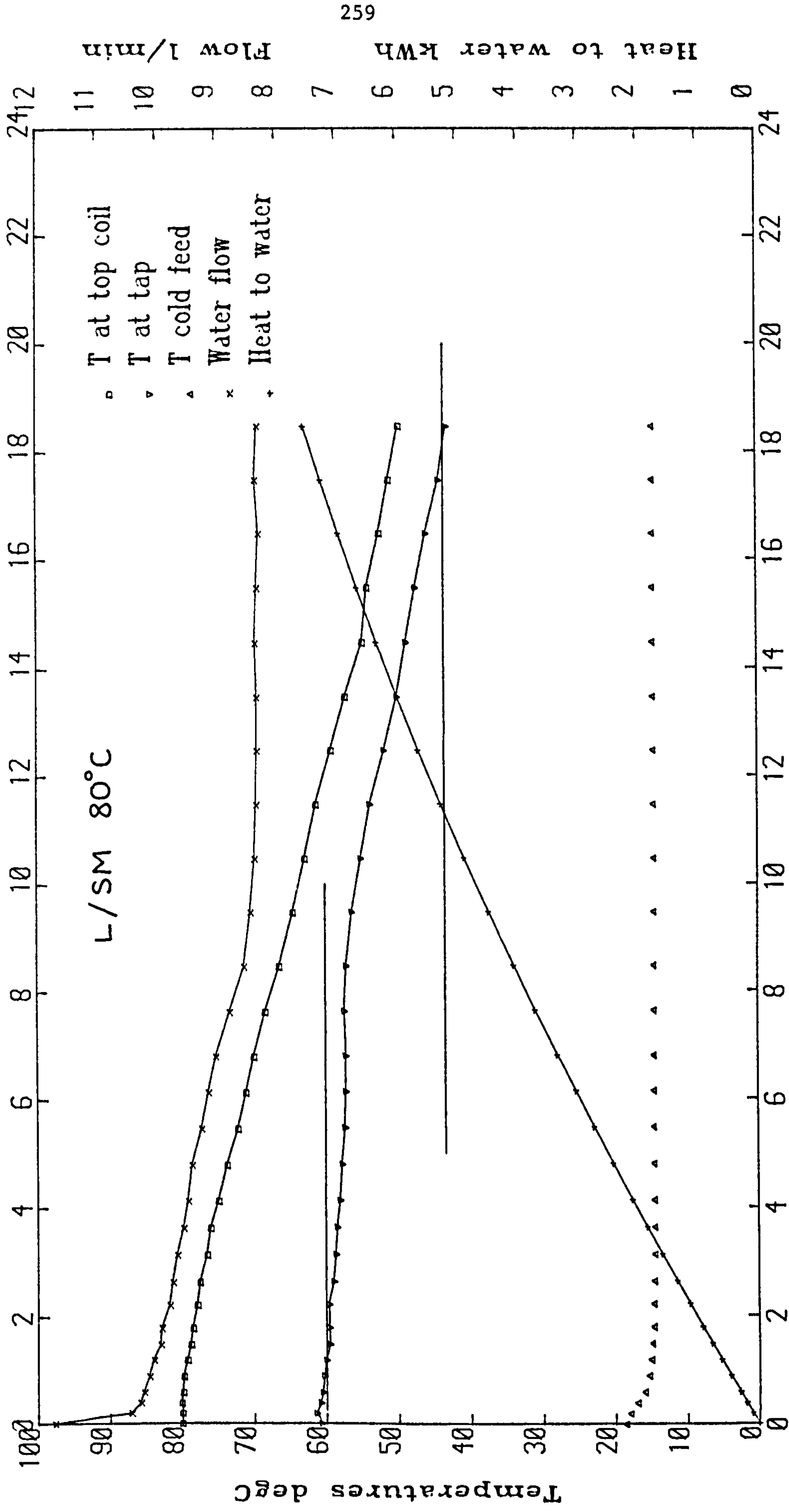
MED/L 60°C



Minutes into discharge

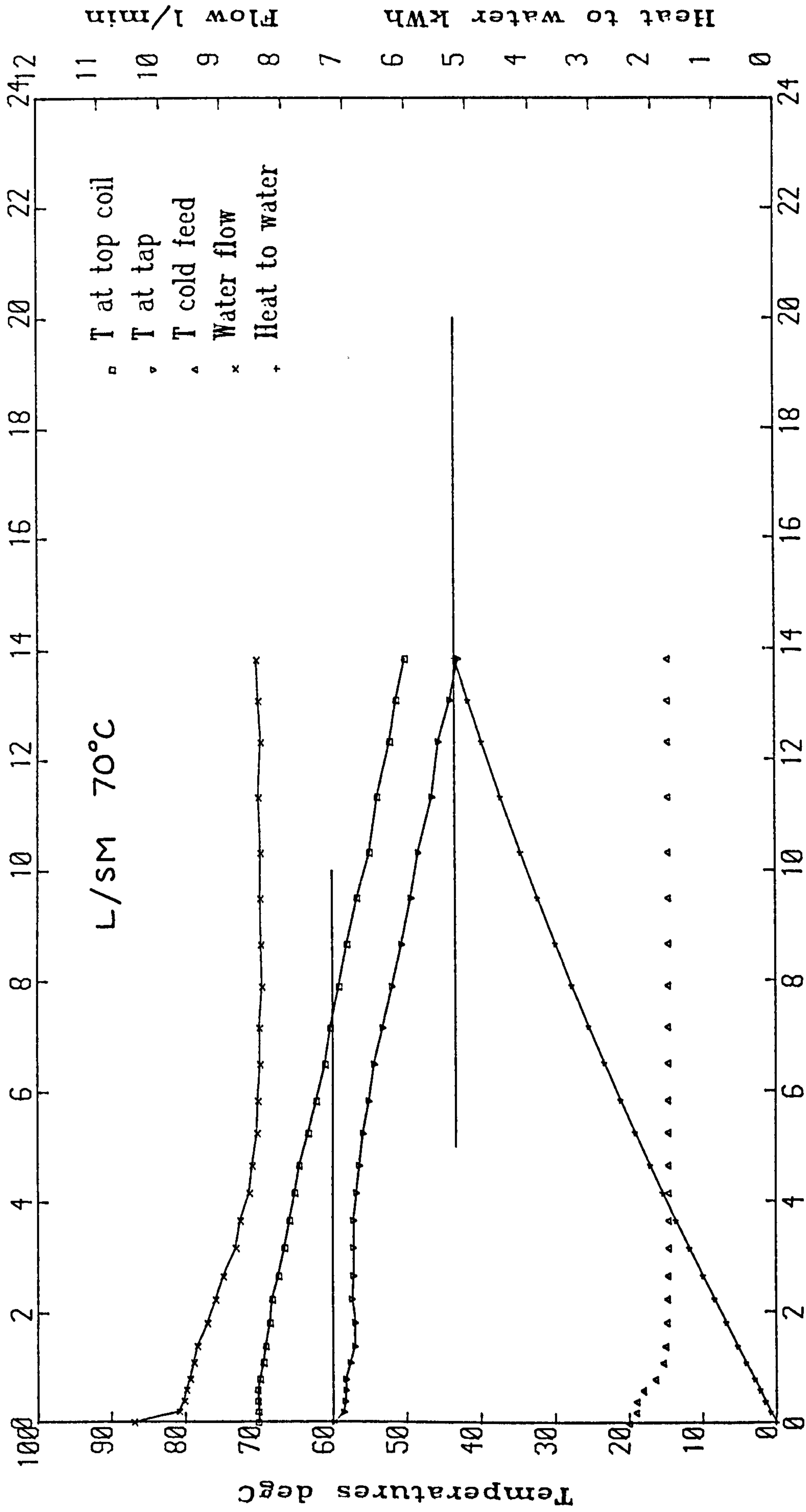
GRAPH No. 23

RIG PLOT : 10 09 85



GRAPH No. 24

RIG PLOT : 10 09 85



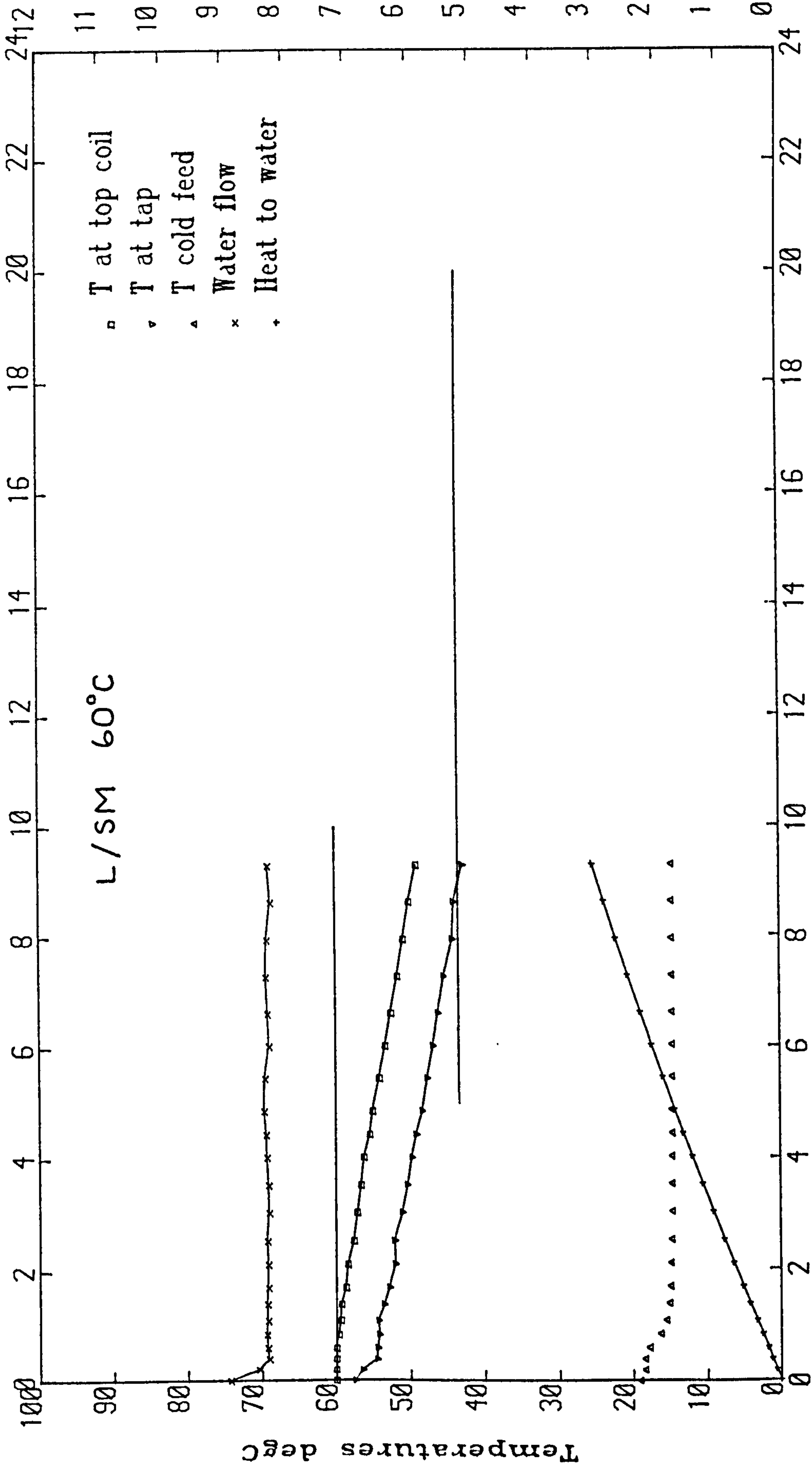
Minutes into discharge

GRAPH No. 25



RIG PLOT : 10 09 85

L/SM 60°C

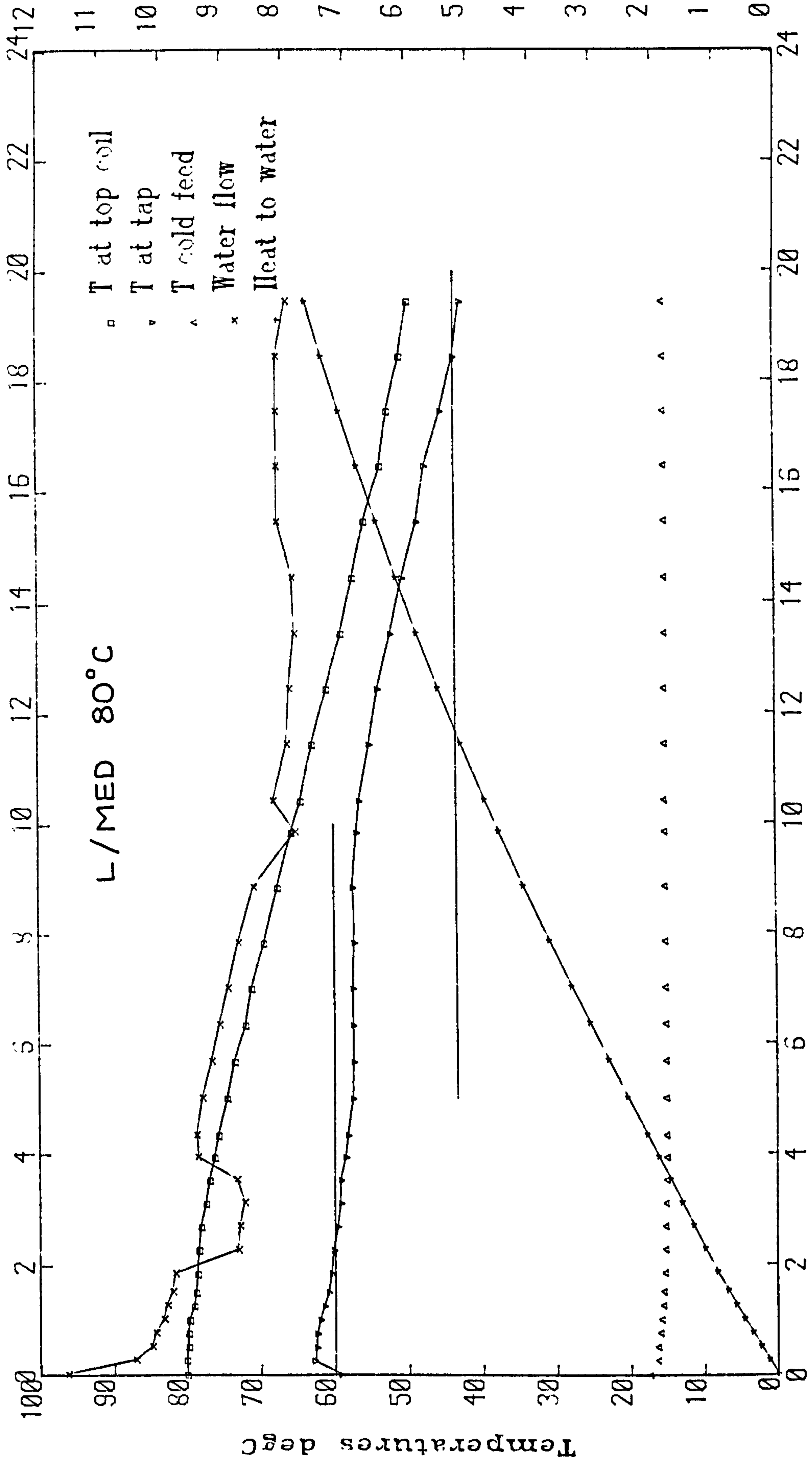


Minutes into discharge

GRAPH No. 26

RIG PLOT : 02 09 85

L/MED 80°C

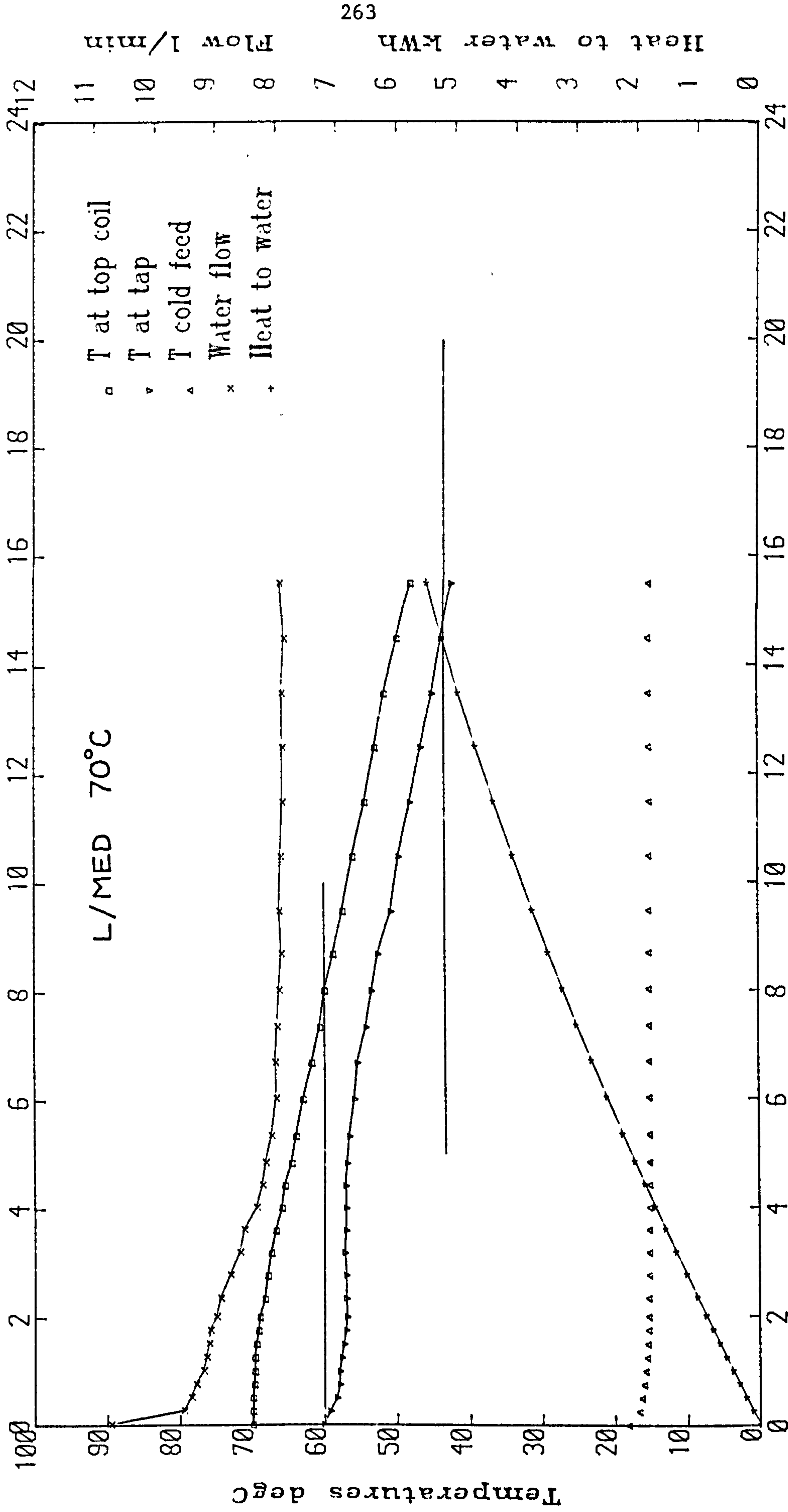


Minutes into discharge

GRAPH No. 27

RIG PLOT : 30 08 85

L/MED 70°C



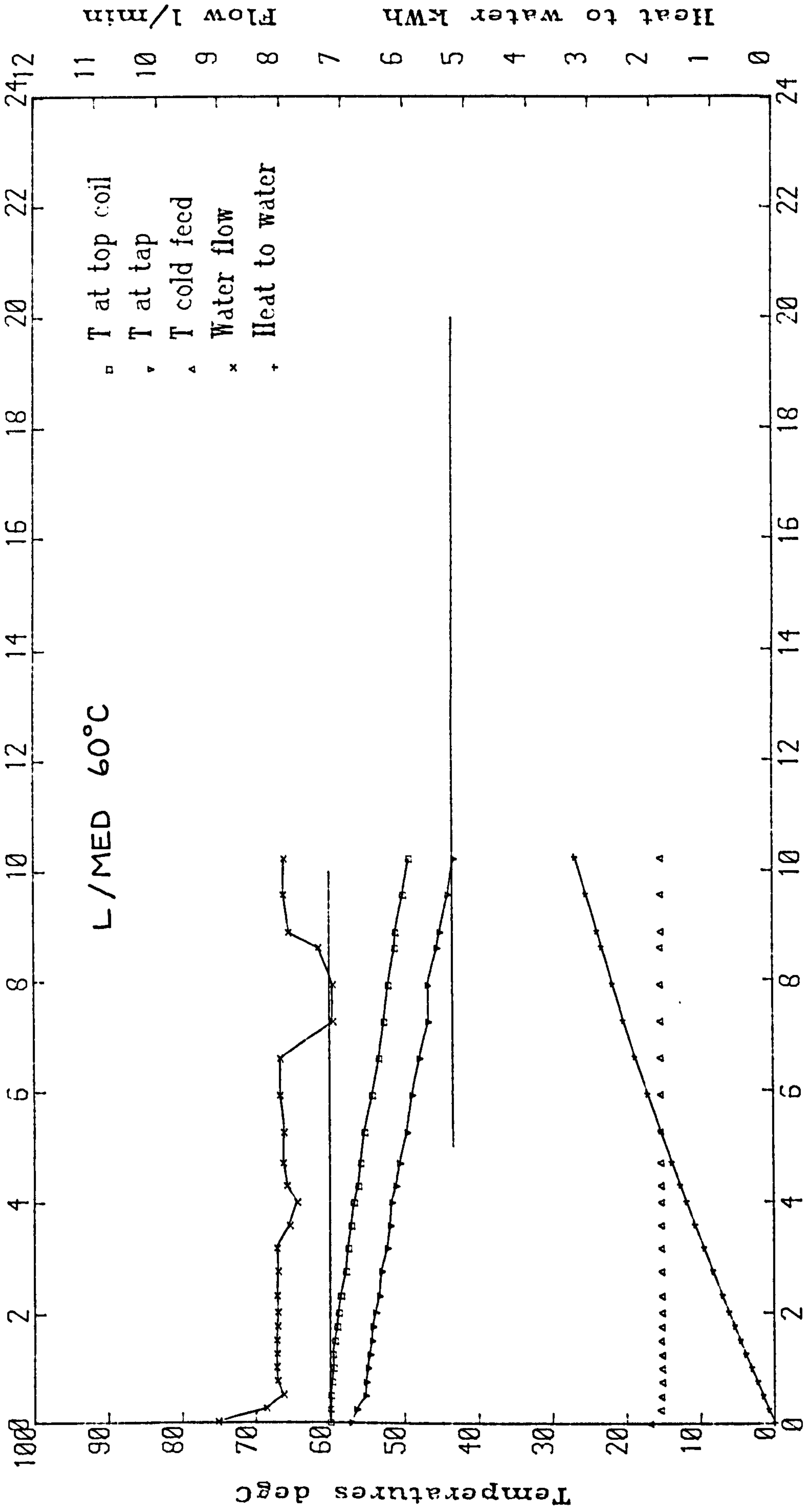
Minutes into discharge

GRAPH No. 28



RIG PLOT : 30 08 85

L/MED 60°C

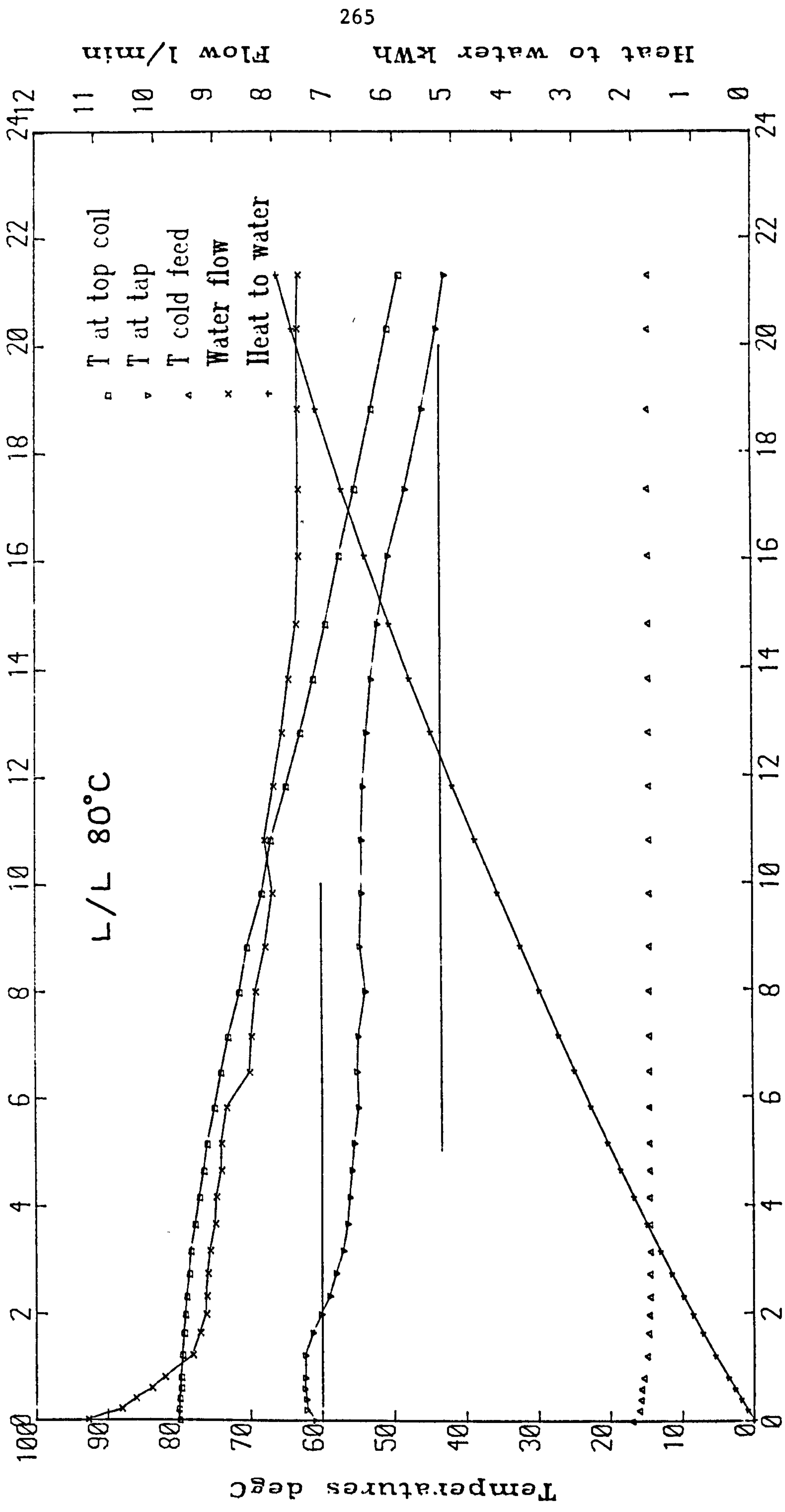


Minutes into discharge

GRAPH No. 29

RIG PLOT : 16 09 85

L/L 80°C

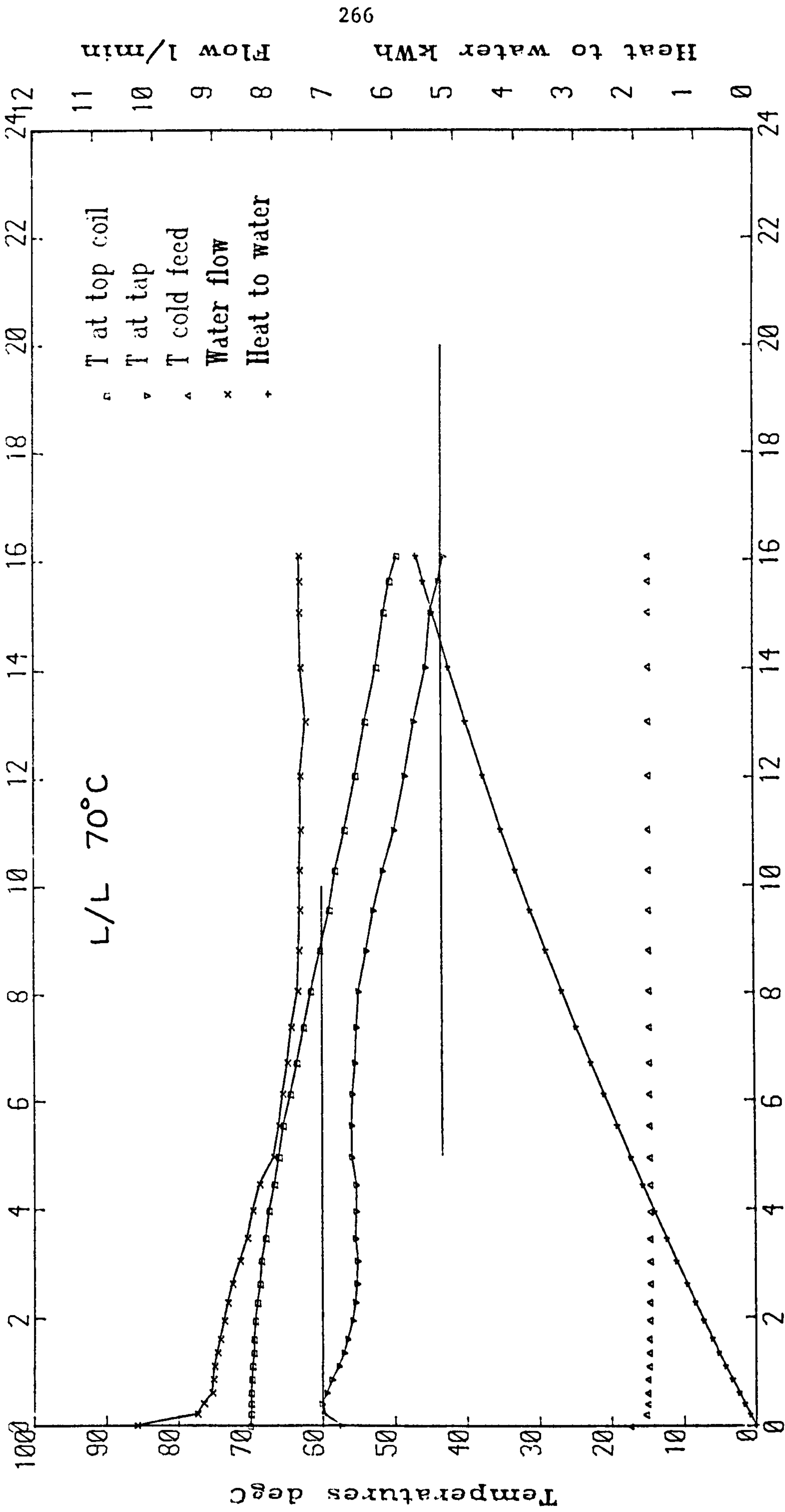


Minutes into discharge

GRAPH No. 30

RIG PLOT : 17 09 85

L/L 70°C

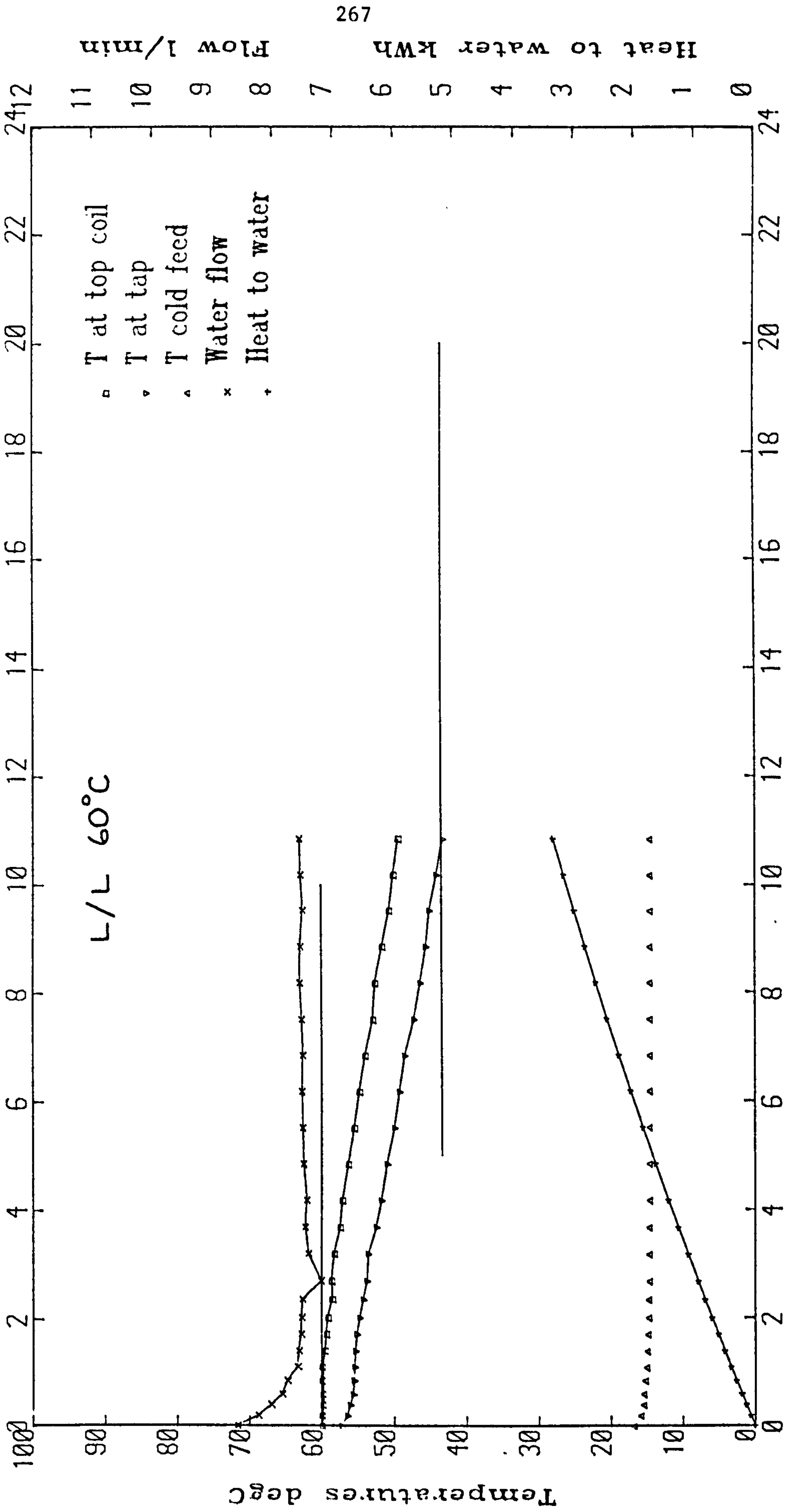


Minutes into discharge

GRAPH No. 31

RIG PLOT : 16 09 85

L/L 60°C



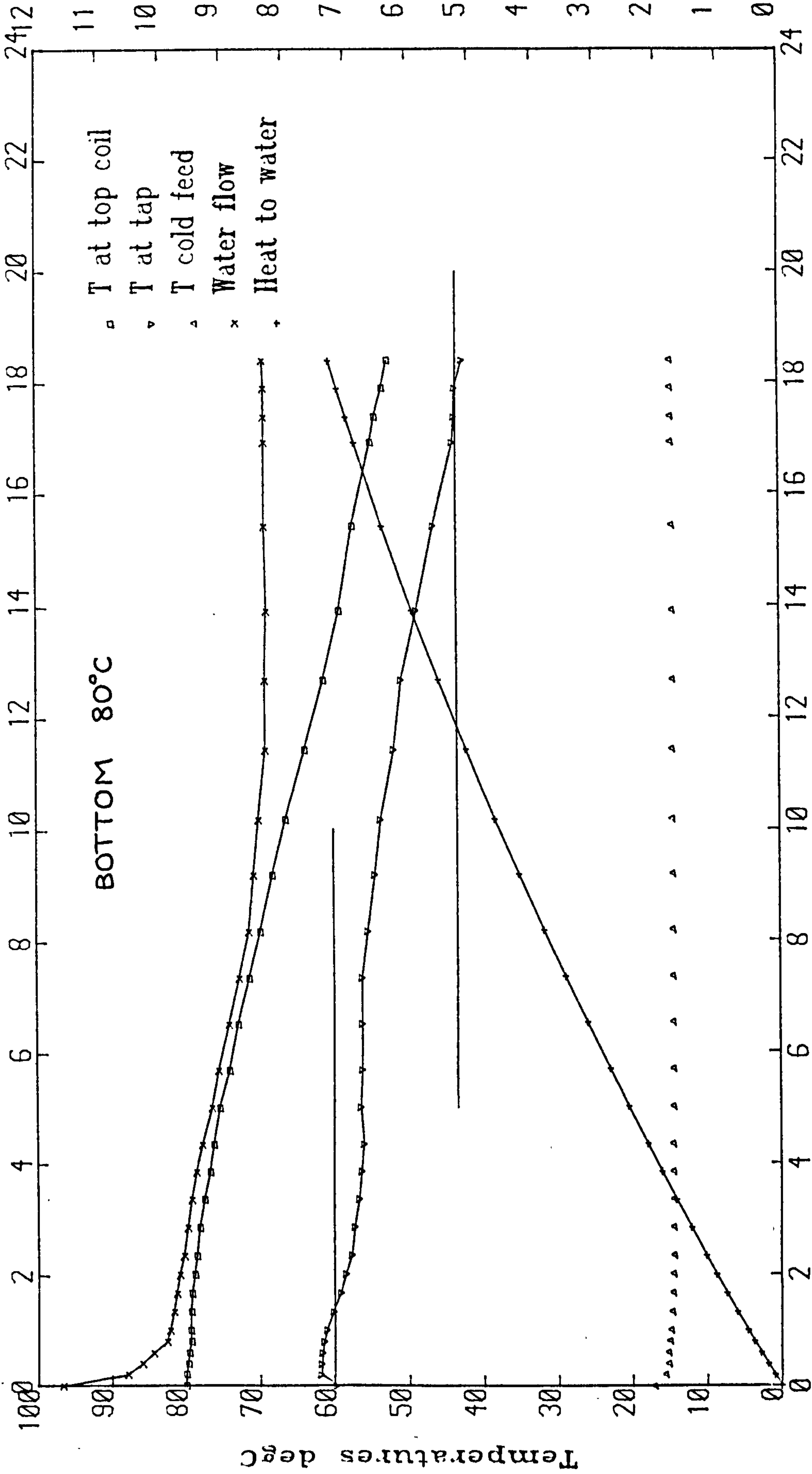
Minutes into discharge

GRAPH No. 32



RIG PLOT : 24 09 85

BOTTOM 80°C

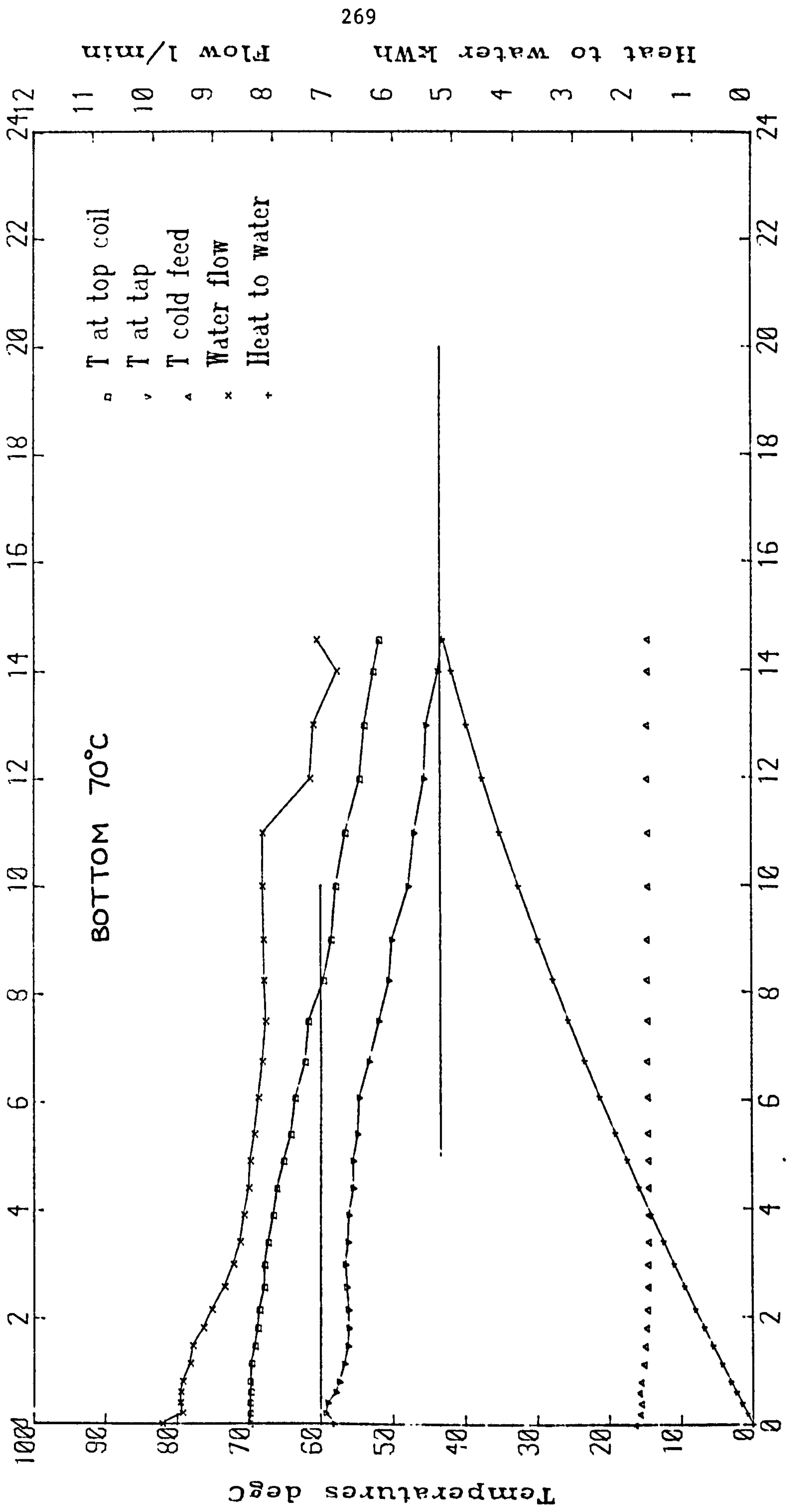


Minutes into discharge

GRAPH No. 33

RIG PLOT : 23 09 85

BOTTOM 70°C

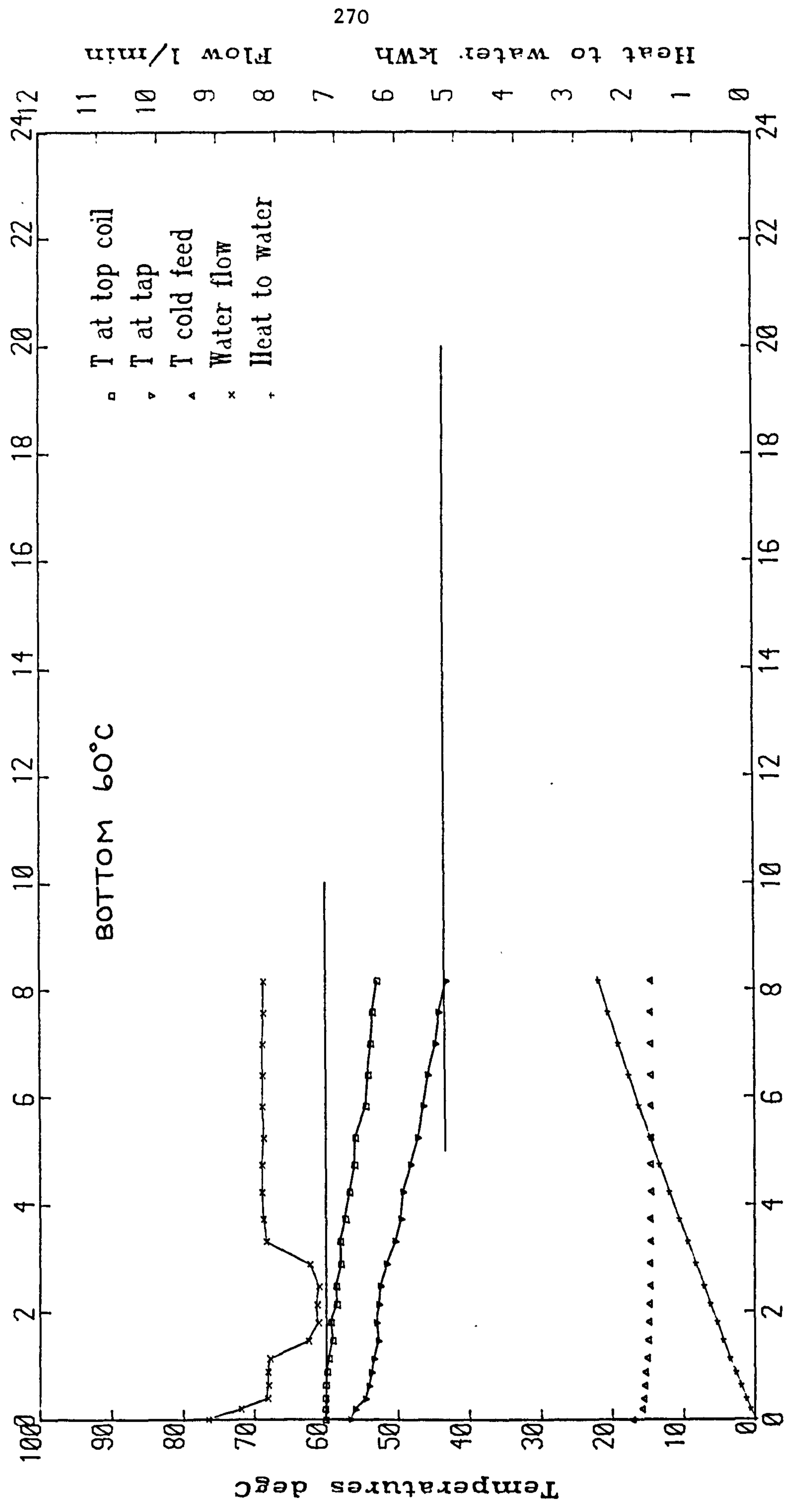


Minutes into discharge

GRAPH No. 34

RIG PLOT : 23 09 85

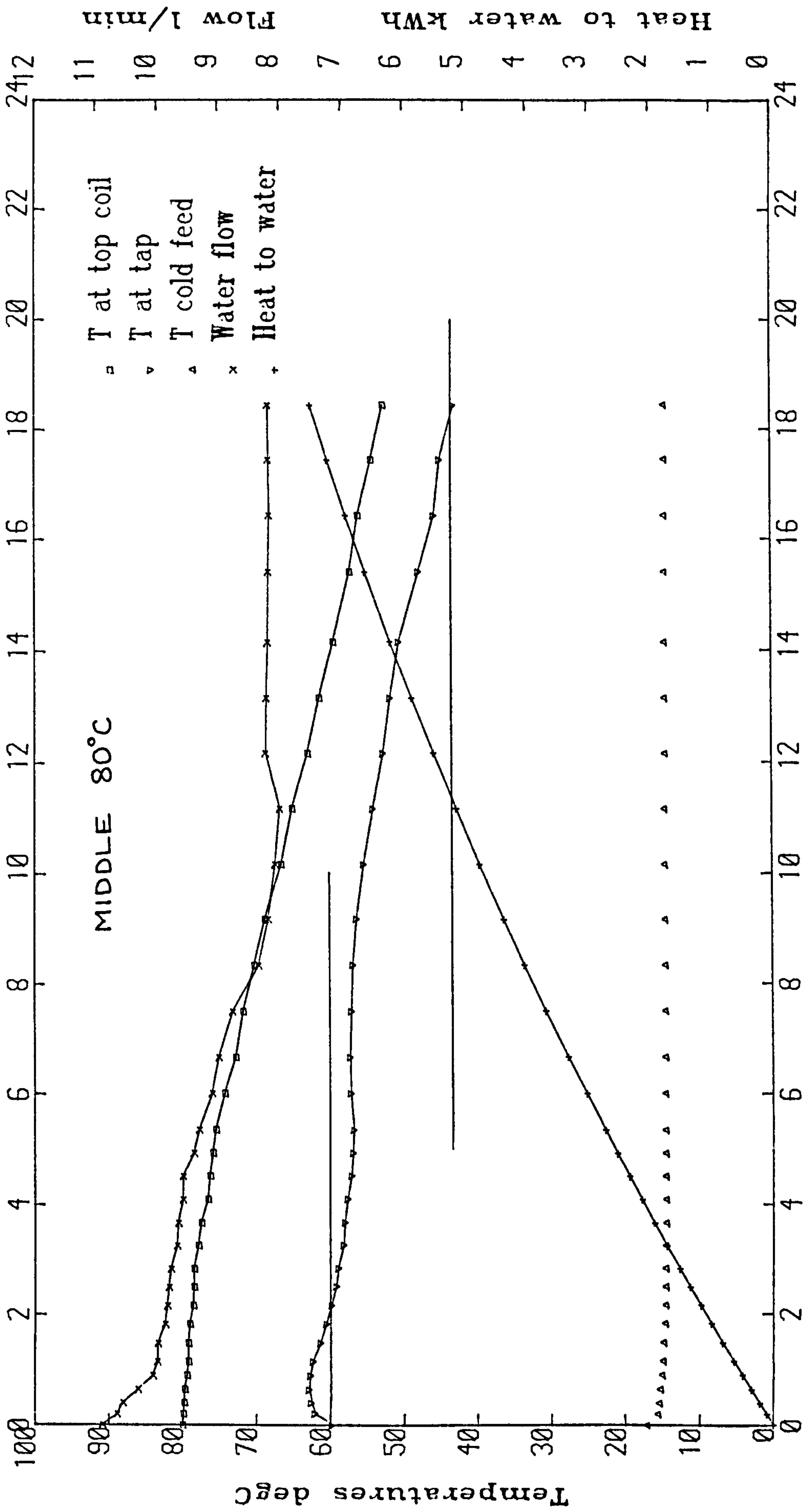
BOTTOM 60°C



Minutes into discharge

GRAPH No. 35

RIG PLOT : 25 09 85



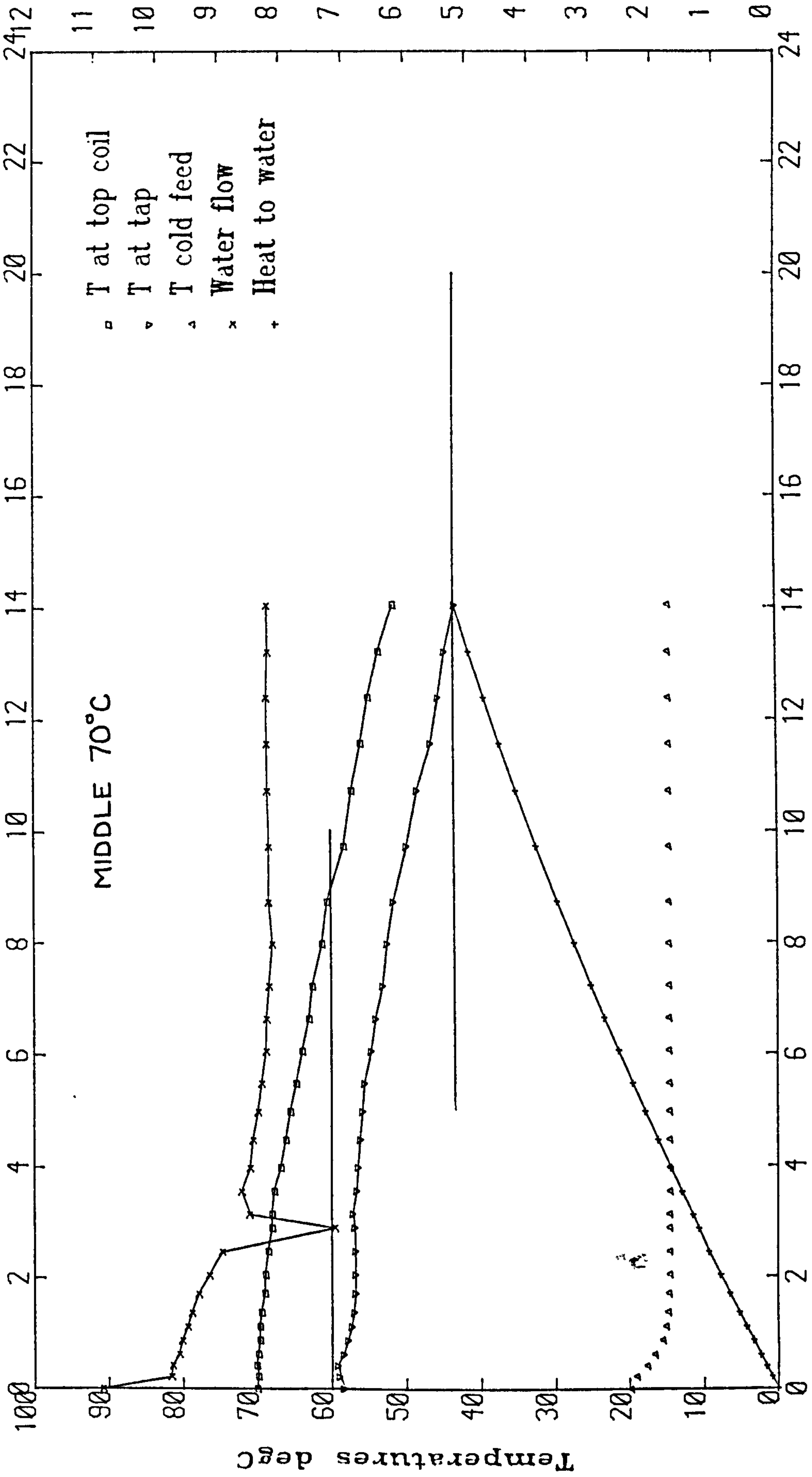
GRAPH No. 36

Minutes into discharge



RIG PLOT : 25 09 85

MIDDLE 70°C

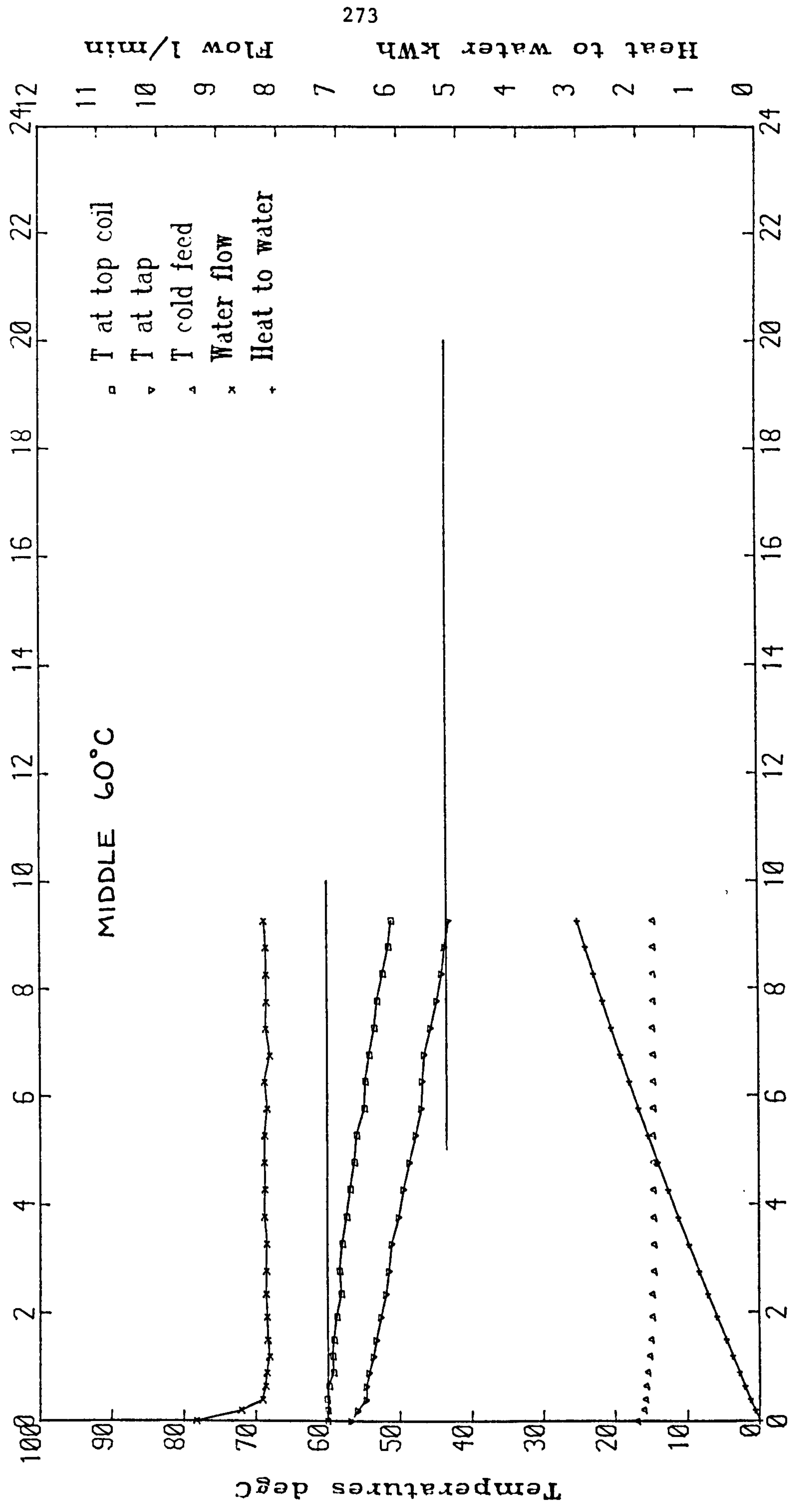


Minutes into discharge

GRAPH No. 37

RIG PLOT : 25 09 85

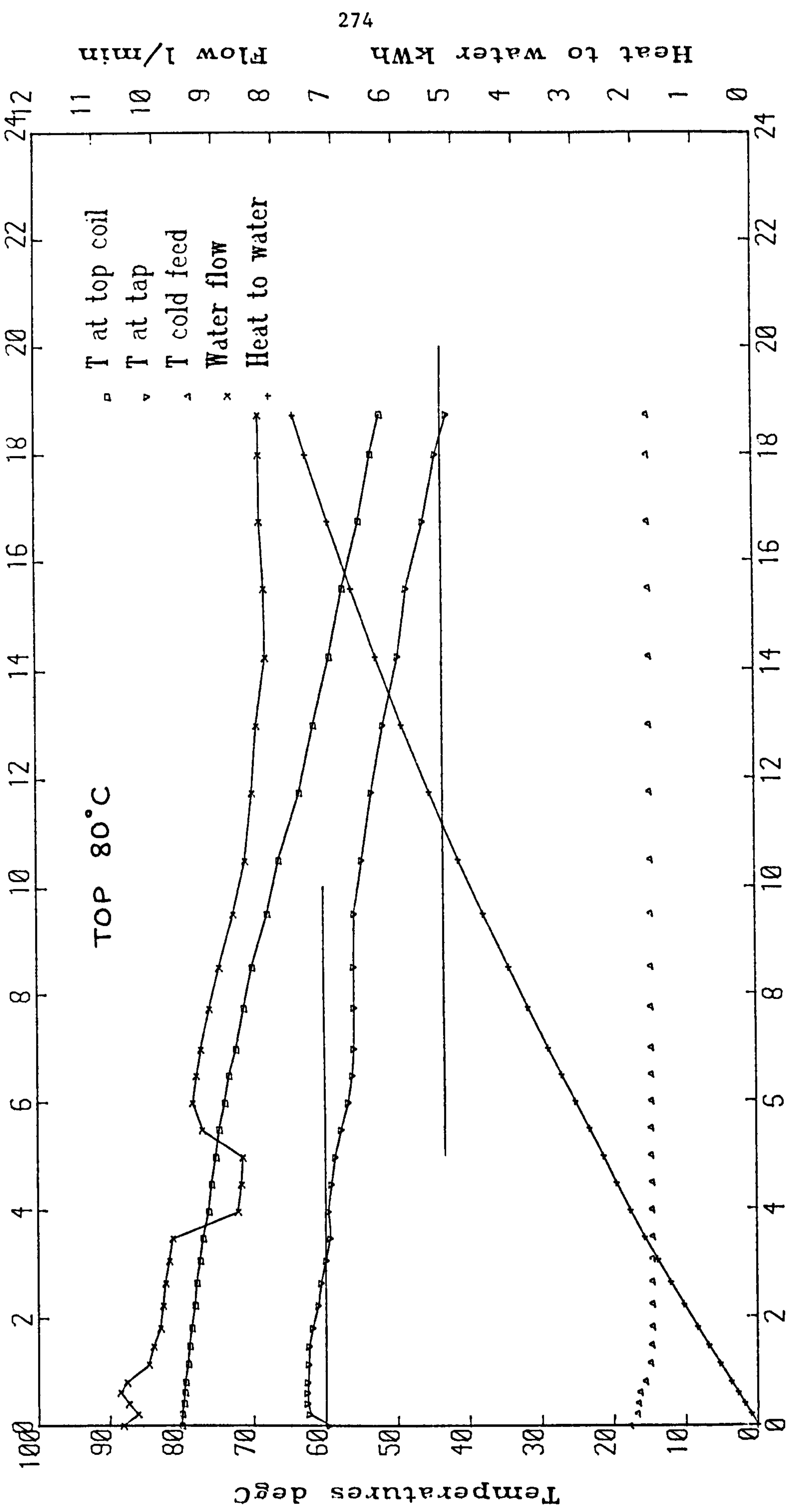
MIDDLE 60°C



Minutes into discharge

GRAPH No. 38

RIG PLOT : 26 09 85

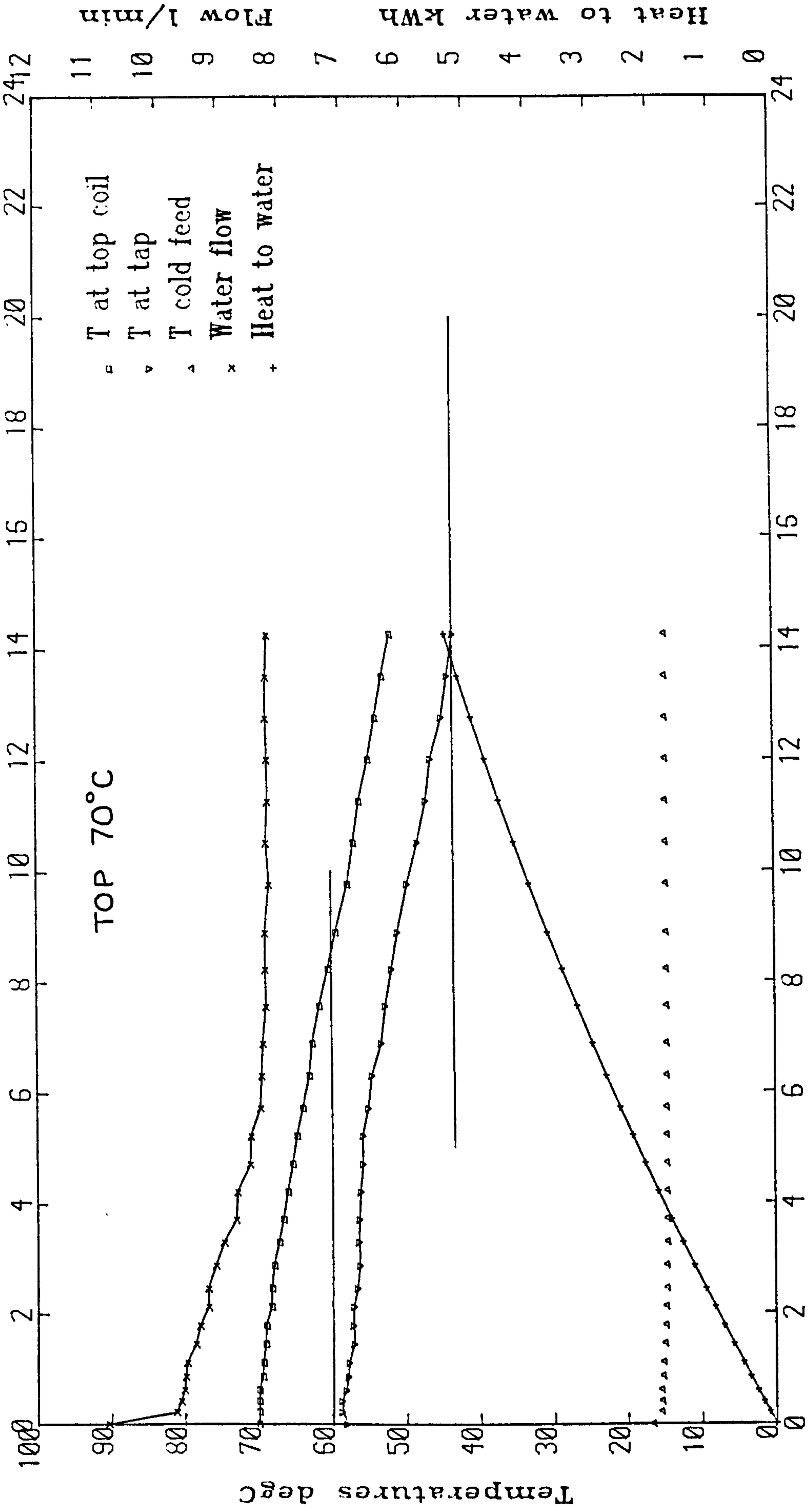


Minutes into discharge

GRAPH No. 39

RIG PLOT : 30 09 85

TOP 70°C



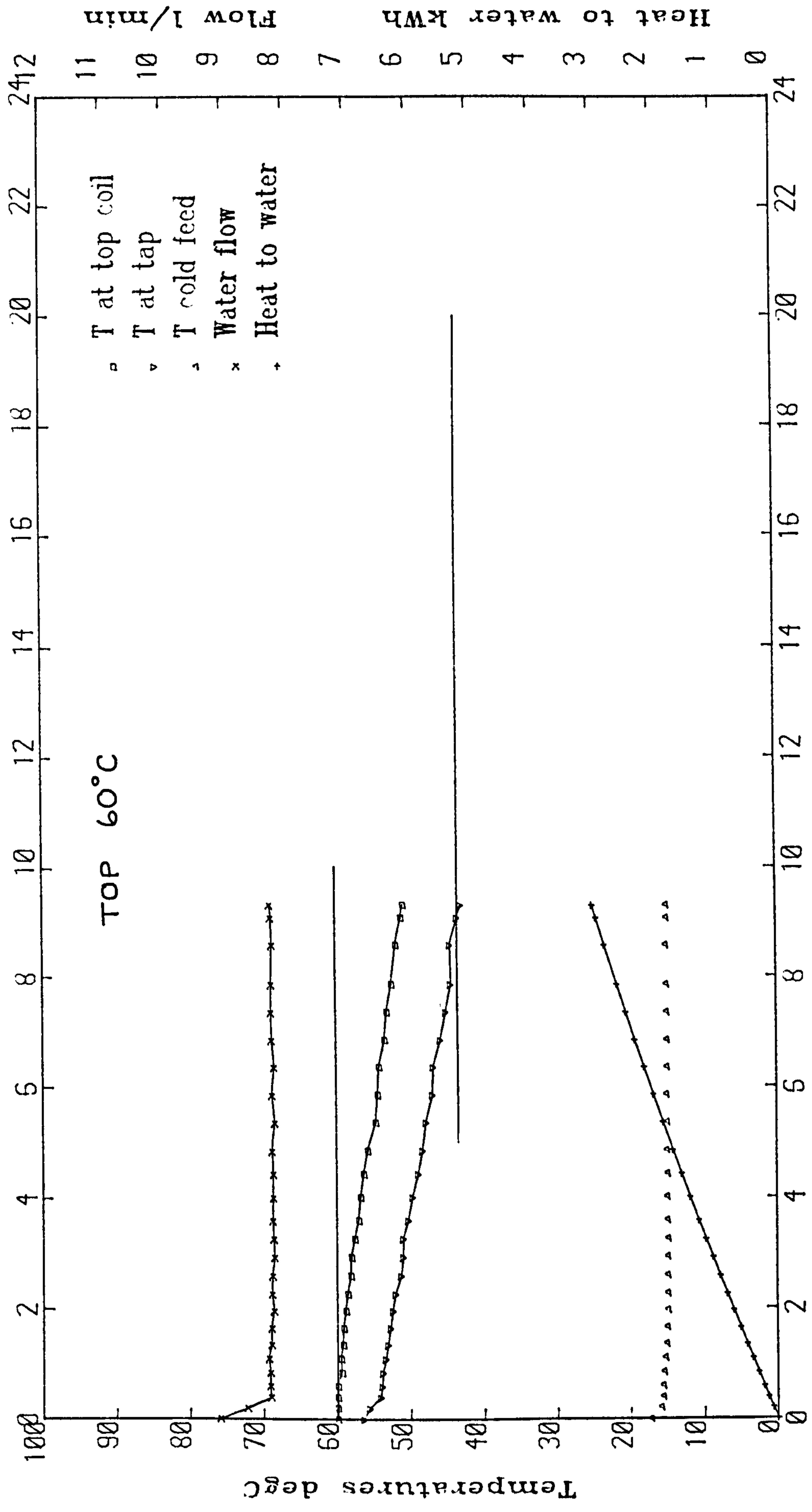
Minutes into discharge

GRAPH No. 40



RIG PLOT : 30 09 85

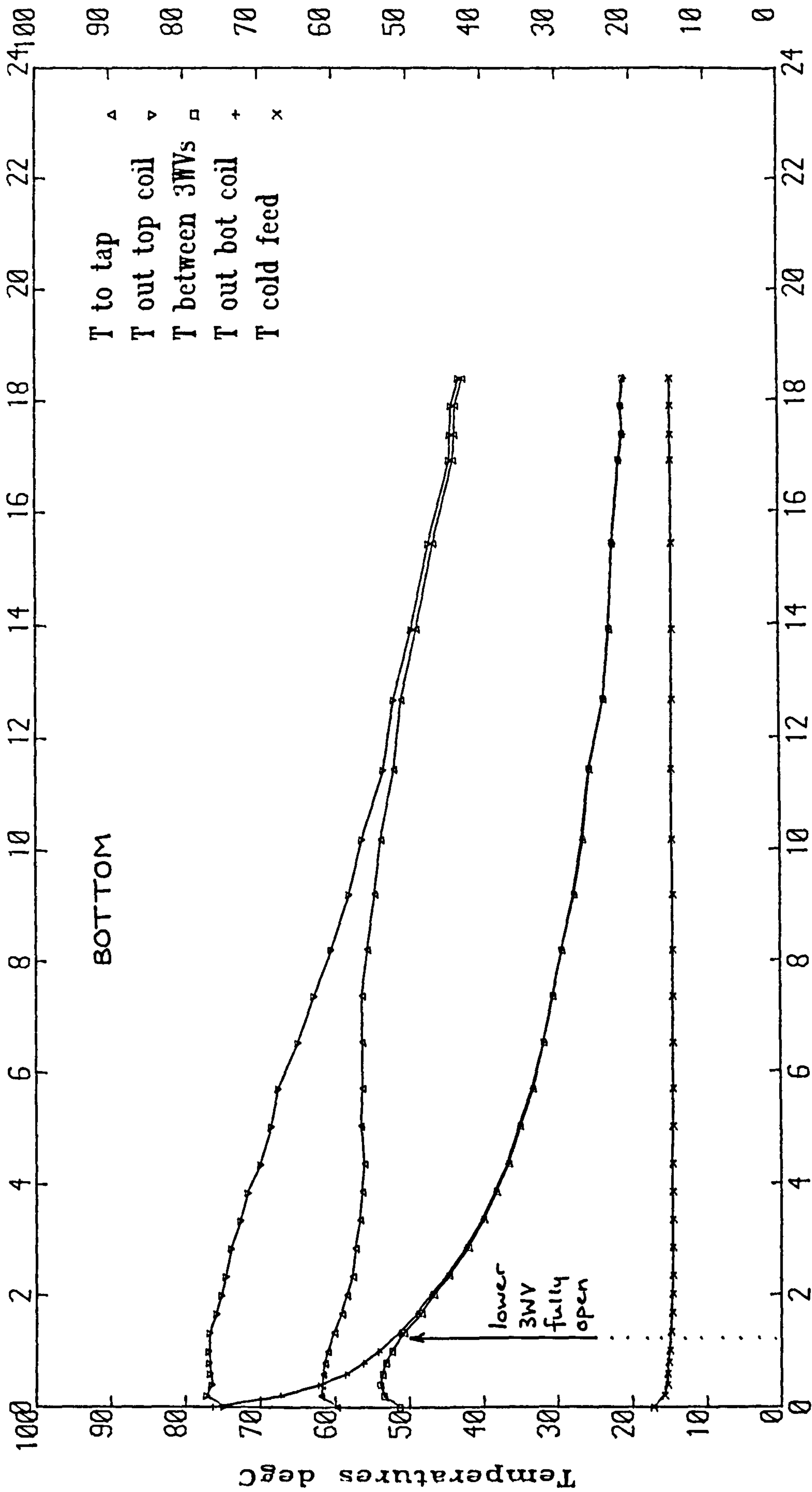
TOP 60°C



Minutes into discharge

GRAPH No. 41

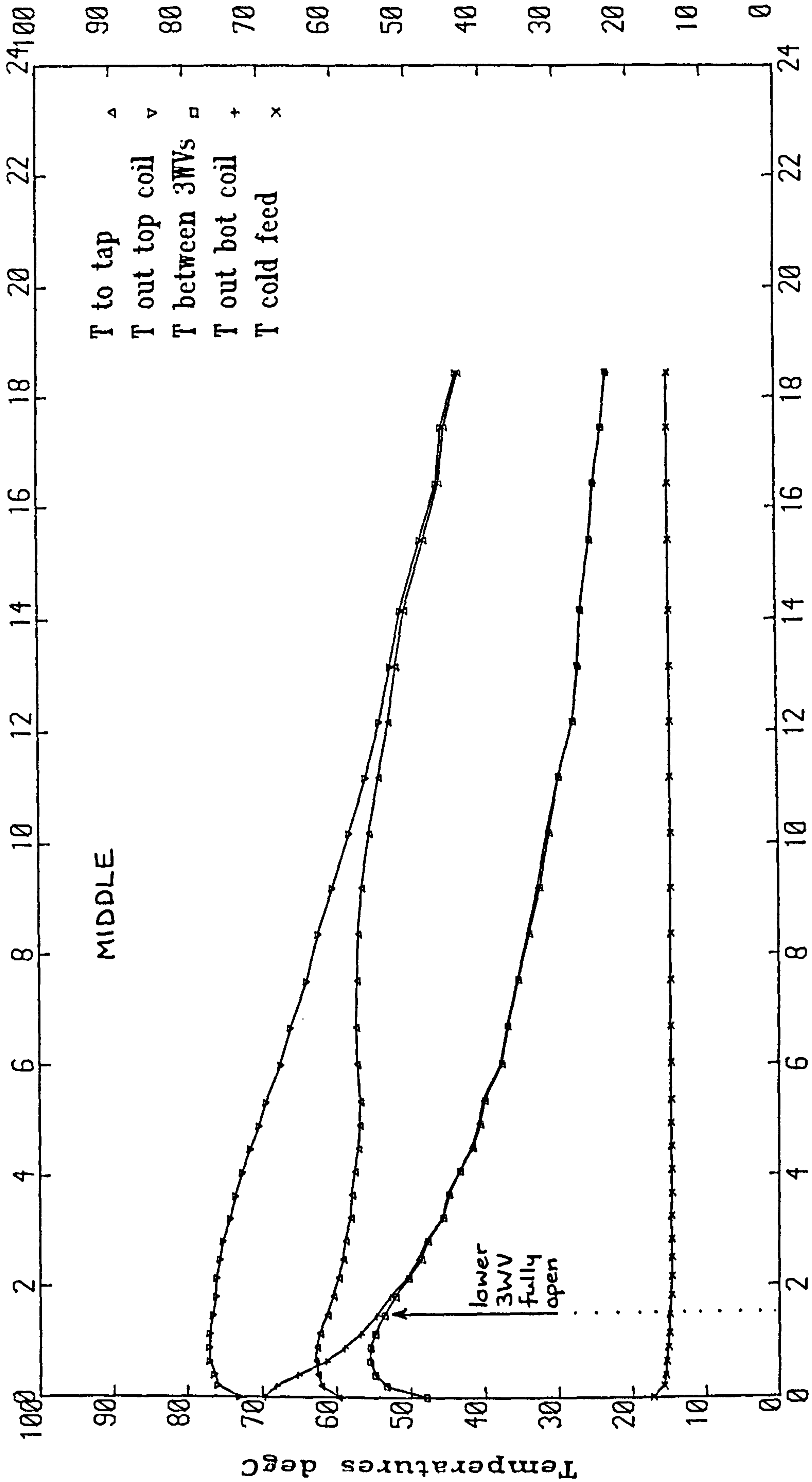
D.H.W. TEMPS : 24 09 85



Minutes into discharge

GRAPH No. 42

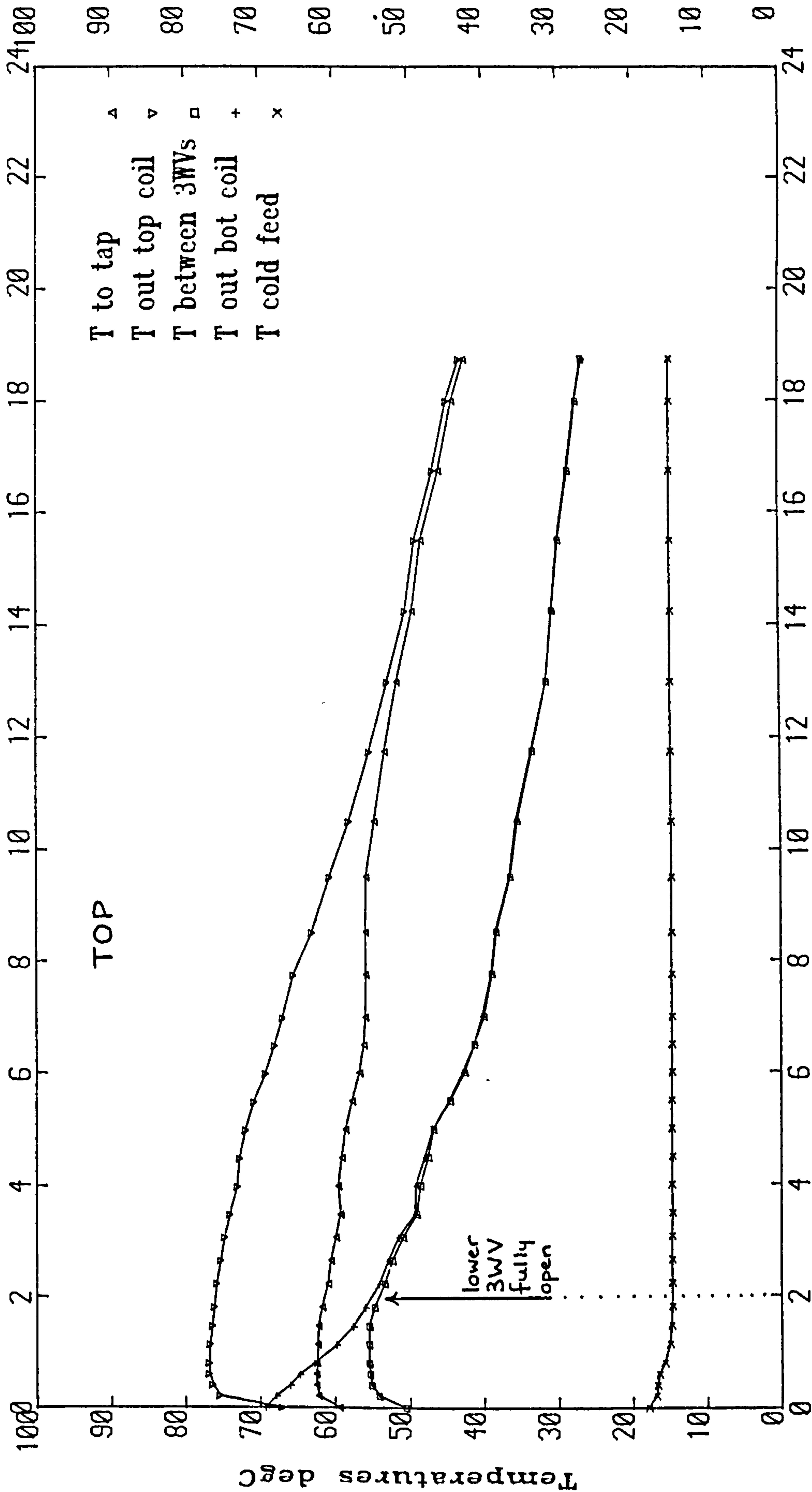
D.H.W. TEMPS : 25 09 85



Minutes into discharge

GRAPH No. 43

D.H.W. TEMPS : 26 09 85



Minutes into discharge

GRAPH No. 44