THE UNIVERSITY OF ASTON IN BIRMINGHAM

A STUDY IN THE APPLICATION OF DOMESTIC SOLAR ASSISTED HEAT PUMPS FOR HEATING AND COOLING

Thesis submitted for the Degree of

Doctor of Philosophy

by

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SUMMARY

In the present work, the more important parameters of the heat pump system and of solar assisted heat pump systems were analysed in a quantitative way. Ideal and real Rankine cycles applied to the heat pump, with and without subcooling and superheating were studied using practical recommended values for their thermodynamics parameters. Comparative characteristics of refrigerants were analysed looking for their applicability in heat pumps for domestic heating and their effect in the performance of the system. Curves for the variation of the coefficient of performance as a function of condensing and evaporating temperatures were prepared for R12.

Air, water and earth as low-grade heat sources and basic heat pump design factors for integrated heat pumps and thermal stores and for solar assisted heat pump-series, parallel and dual-systems were studied. The analysis of the relative performance of these systems demonstrated that the dual system presents advantages in domestic applications.

An account of energy requirements for space and water heating in the domestic sector in the U.K. is presented. The expected primary energy savings by using heat pumps to provide for the heating demand of the domestic sector was found to be of the order of 7%. The availability of solar energy in the U.K. climatic conditions and the characteristics of the solar radiation were studied. Tables and graphical representations in order to calculate the incident solar radiation over a tilted roof were prepared and are given in this study in section IV.

In order to analyse and calculate the heating load for the system, new mathematical and graphical relations were developed in section V.

A domestic space and water heating system is described and studied. It comprises three main components: a solar radiation absorber, the normal roof of a house, a split heat pump and a thermal store. A mathematical study of the heat exchange characteristics in the roof structure was done. This permits to evaluate the energy collected by the roof acting as a radiation absorber and its efficiency. An indication of the relative contributions from the three low-grade sources: ambient air, solar boost and heat loss from the house to the roof space during operation is given in section VI, together with the average seasonal performance and the energy saving for a prototype system tested at the University of Aston. The seasonal performance was found to be 2.6 and the energy savings by using the system studied 61%. A new store configuration to reduce wasted heat losses, is also discussed in section VI.

Key words: Heat Pump - Solar Energy - Heating Load - Thermal Store - Performance - Energy Savings.

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I-INTRODUCTION

The rate at which the industrial nations in particular have been depleting the world's natural energy resources in the form of fossil fuels has caused some scientists and technologists to be concerned for many years. Demands for energy to cope with improved living standards and creature comforts have been doubling every 10 years whilst little account has been taken of the resultant depletion of resources.

The oil crisis of 1973 has, perhaps more than any other factor, highlighted the problem as far as the general populace is concerned. The escallating costs of energy has drawn attention to the consideration of alternative sources of energy. Whilst nuclear power derived from fission will no doubt make a significant contribution to future energy requirements, fears of the 'Plutonium Economy' will inevitably encourage research into the effective utilization of pollution free sources to provide for a proportion of our energy needs, whether it be from the sun, wind, wave or tide. Apart from conservation aspects there are economic advantages to be ultimately gained in harnessing solar energy for domestic use even in areas of the world with limited direct sunshine.

(1) The source of energy is always available and predictable.

(2) For the user it provides an element of inflation free energy.

(3) For the nation it should mean less capital expenditure on power plant and economy in the use of natural resources although the energy consumed by ancilliary equipment must not be neglected in considering total energy savings.

On the other hand, Heat Pumps have an important contibution to make in helping one to face the coming transitional period in energy use patterns. Since most of the alternative energy sources such as wind power, nuclear power etc. all favour the production of electricity, the heat pump becomes steadily more and more attractive as the transition progresses.

The heat pump principle is not new, but the availability of cheap fossil fuels combined with the low cost of conventional boilers has so far mitigated again its development, except in the field of refrigeration and cooling. However, the increasing cost of conventional fuels now provides an incentive for heat pump development for a variety of purposes; not the least of which is space heating. With the cooler ambient surroundings providing the low grade heat, and electricity, fossil fuels or simply waste heat providing the high grade energy source, the heat pump also offers the potential of substantial energy savings.

A survey of practical experience with heat pumps in buildings in the U.K. over the post-war years has shown that current machines will operate with a coefficient of performance (COP(H)) between 2 and 3 averaged over the year (1). If one could achieve a seasonal COP(H) of 3, and heat pumps could supply the National domestic space heaing load, the resulting annual national consumption of primary energy for this purpose could be reduced from the present 1.49×10^9 GJ to about 0.8×10^9 GJ. This saving would represent about 7 per cent of the total national primary energy consumption. A further 2 per cent would be saved if water heating were accomplished in a similar way, if only a COP(H) of about 2 could be achieved.

Thus, solar energy and heat pump systems are two promising means of reducing the consumption of non-renewable energy resources and hopefully, the cost of delivered energy for domestic space heating and domestic water heating.

The coefficient of performance of a heat pump depends inversely on the source-sink temperature difference, so that solar energy could be used to increase the source temperature thereby increasing the COP. At the same time the lower solar collector operating temperature would result in higher collector efficiencies and increased utilization of solar energy. It seems that a logical extension of each system would be to combine the two in order to have a "Solar Assisted Heat Pump System" which could further reduce the cost of delivered energy.

In spite of the wet weather, the U.K. climatic conditions are good enough for solar assisted heat pump applications. The U.K. lies between latitudes 50°N and 58°N with a high proportion of the population in the south. It has a relatively mild climate and the occasions when temperatures are less than 0°C for several consecutive days are rare. The extremes between Summer and Winter solar radiation levels are $\sim$ 800 and 150 W/m^2 respectively, with a high proportion of diffuse radiation for most of the year. There are few completely cloudless days, even in summer, whereas, there are frequently extended cloudy periods and for this reason a thermal storage element such as a liquid filled tank must be added to the solar assisted heat pump system. This permits the modulation of the heat pump capacity in order to provide for heating at night and during those cloudy periods. Thus, one will have an "Integrated Solar Assisted Heat Pump and Thermal Store System", which is the subject matter of this study.

This theoretical study is a contribution to the research programme over the subject of domestic heating by a heat pump and solar energy system, which is being investigated in the Department of Physics, at the University of Aston.

<u>OBJECTIVES</u> - The main objectives of this study were to:

1) Analyse in a quantitative engineering way, the more important parameters of the heat pump systems and of solar assisted heat pump systems.

2) Consider the capability and the application possibilities of these systems in order to cover the seasonal domestic heating requirements through the U.K. climatic conditions.

3) Consider an integrated solar assisted heat pump system and a new thermal store configuration, which utilises a normal roof as a solar absorber taking advantage of the diffuse radiation, and which fulfilled the seasonal domestic heating requirements for either existing or new dwellings in most areas of the U.K. One of the features of this proposed system is that it manages to reduce both the energy requirement to supply the heat demand, and the environmental load by recycling some of the lost heat back into the building again.

In looking at these objectives one has to consider:

1) The overall design factors including components and system performance.
2) The availability of energy, energy sources and the climatic conditions.
3) The seasonal heating demand.

The development of this study towards these objectives pemits one to identify areas in this field, where research, development, and demonstrations, are needed.

Data on saturation properties for some non-commercial refrigerants is available only in Imperial Units. For this reason the comparisons of refrigerants shown in section II-5 have not been quoted in International Metric Units.

CHAPTER II

HEAT PUMPS. BASIC CONSIDERATIONS

II-1. Principles of Operation

The heat pump (1) is a machine which extracts thermal energy from a low temperature source, upgrades it to a higher temperature level, thus, enabling it to be used for space heating or for water heating in domestic or industrial applications.

There is no fundamental difference (2) between a conventional refrigeration system and the so-called heat pump system. Thermodynamically both systems are heat pumps, employing a compressor, a condenser, an evaporator and an expansion valve.

The main difference between the two systems is the primary objective of the application. A refrigeration installation is concerned with the low temperature effect produced at the evaporator while the Heat Pump is concerned with both the cooling effect produced at the evaporator and the heating effect produced at the condenser. So, a system making use of the rejected heat cannot be properly named a refrigeration system since cooling is not the primary object during the cycle. Such a system has sometimes inaccurately been called the "Reverse Refrigeration Cycle".

The operating principle of the Heat Pump is identical with that of the heat power thermodynamic cycle, governing the conversion between heat energy and mechanical work; but in this case the system reverses the natural direction of the heat flow, so that work must be expended during the cycle. This is illustrated in Fig. 1.

The main feature of the Heat Pump is that it will always provide more energy for heating than is used directly in driving it. In the generation of electrical energy from heat the efficiency is in the region of 33%. The heat pump inverts this process so, for an electrically driven compressor system one can expect to achieve at least a performance of COP(H) = 1/0.33 = 3; i.e. for each kWh of electrical energy expended 3 kWh of heat can be supplied to the space or water to be heated. Thus the heat pump appears to be the best electrical method of supplying a building's heating requirements while consuming the least amount of energy.

The idea of using the heat rejected from a refrigeration system, to warm a space at a higher temperature, is not new, having been first suggested in the 1850s by William Thompson who later became Lord Kelvin. (3) He proposed a scheme for heating houses by extracting energy from the earth.

Various schemes have been proposed over the years and many individuals have in fact been successful in heating their homes with varying degrees of success. Some 20 years ago a few firms attempted to introduce heat pumping schemes by extracting heat from external ambient air and upgrading the temperature for a ducted warm air heating system. The schemes suffered from many problems: Frosting at low ambient temperature necessitated the use of electrical heaters to pre-heat the incoming air, which considerably increased running cost. Noisy fans and a reluctance on behalf of householders to accept warm air heating systems were perhaps other factors.

the use of Heat Pumps for reducing primary energy requirements, particularly to their use in water and space heating applications. It is perhaps useful to relook at the Heat Pump principle in order to identify areas where its characteristics could be exploited. The important aspects of energy consumption need to be matched against requirements whether for industrial or domestic applications.

II-2. Thermodynamic Cycles

In the refrigeration process (4) the interest is to abstract heat from a body or the environment with the minimum expenditure of energy in the form of work. The heat abstracted would be available for other uses but in normal refrigeration practice the heat is rejected to waste. It follows that work must be expended to transfer heat energy from a lower to a higher temperature. The processes of cooling and heating (5) are thus seen to be closely related, and maximum overall efficiencies result, if the two can be combined by using one machine for both purposes. To achieve both cooling and heating, a working substance is taken through a cyclic process. Thompson proposed air, but modern machines use one of a family of organic refrigerant material: two of the most common being R12, dichlorodiflurομethane (CF_2 Cl_2) and R22 monochlorodiflurομethane (CH ClF_2).

II-2-1. The Carnot Cycle

This cycle (6), represents the highest possible performance between two temperatures and is considered the ideal heat pump cycle. In this cycle, as shown in Figure 2, the refrigerant fluid is taken through four operations:

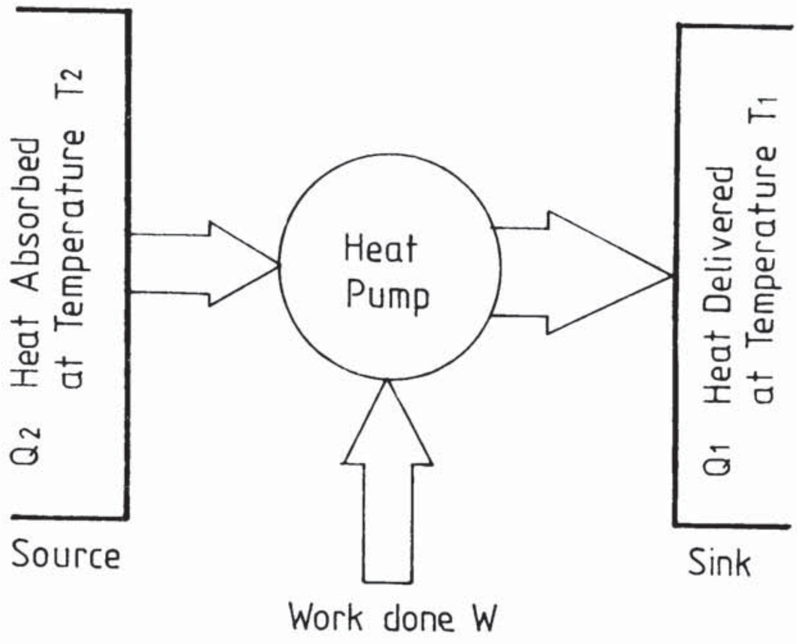


Fig 1 The Heat Pump

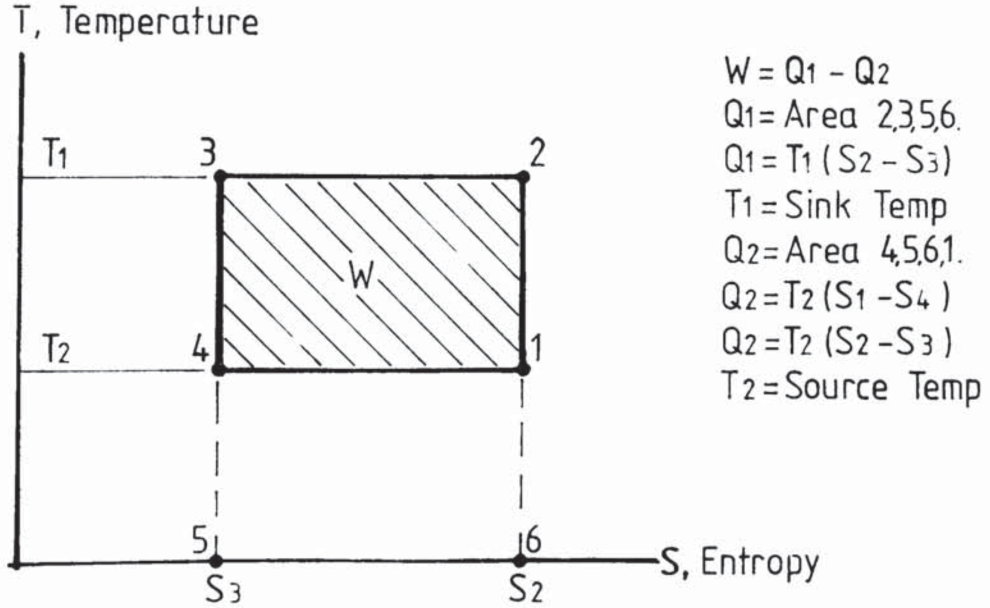


Fig 2 The ideal heat pump Carnot cycle.

Operation 4-1 involves isothermal expansion or evaporation (constant temperature)

Operation 3-4 involves isentropic expansion (constant entropy)

Operation 1-2 involves isentropic compression (constant entropy)

Operation 2-3 involves isothermal compression or condensation (constant temperature).

The heat absorbed by the system (the refrigerating effect) is represented by Q_2 (area 4,5,6,1). The heat rejected by the system (The heating effect) is represented by Q_1 (area 2-3-5-6). The energy added ideally by the compressor at work to accomplish these effects is represented by $Q_1 - Q_2$, (area 1-2-3-4).

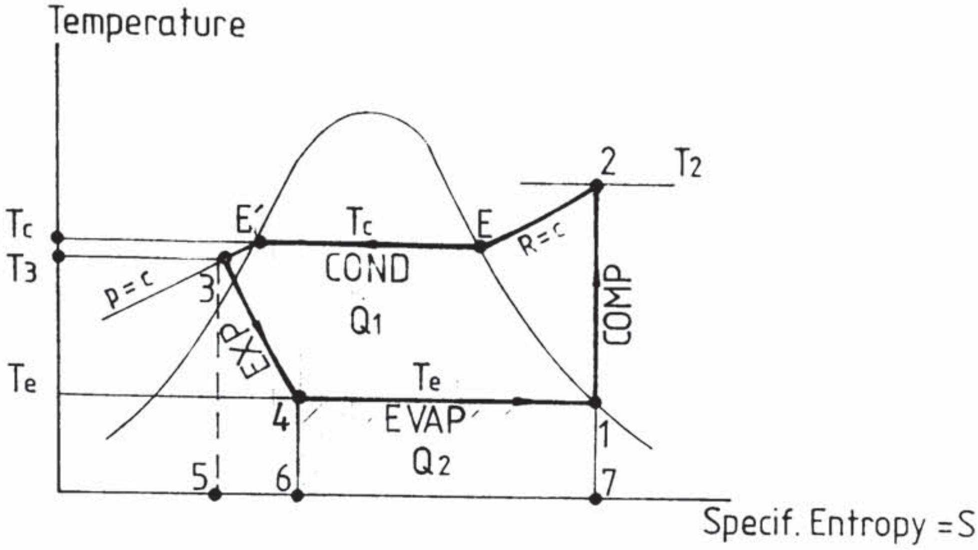
II-2-2. Ideal Heat Pump Rankine Cycle

This cycle (2), which assumes irreversible expansion at constant enthalpy through a throttle valve, is generally chosen as more indicative of a vapour-compression refrigeration cycle for the modern heat pumps. Thermodynamically, the Rankine Cycle can be illustrated by the temperature - entropy diagram of Fig. 3(a), or by the Fig. 3(b), which is an idealised pressure - specific enthalpy curve for a liquid vapour Rankine cycle.

The sequence of the cycle consists of the following thermodynamic operations:

1. The refrigerant is here compressed isentropically from 1 to 2 and work is done on the system.

2. Superheated Vapour is cooled from the temperature T_2, to Tc, condensed isothermally to liquid, from E to E' and then cooled from Tc to T_3 at constant pressure. Heat is rejected from the system.



$Q_1 = T_c(S_2 - S_3)$; T_c = Condensing Temp
$Q_2 = T_e(S_1 - S_4)$; T_e = Evaporating Temp
$W = Q_1 - Q_2$

Fig 3a Ideal heat pump - Rankine cycle

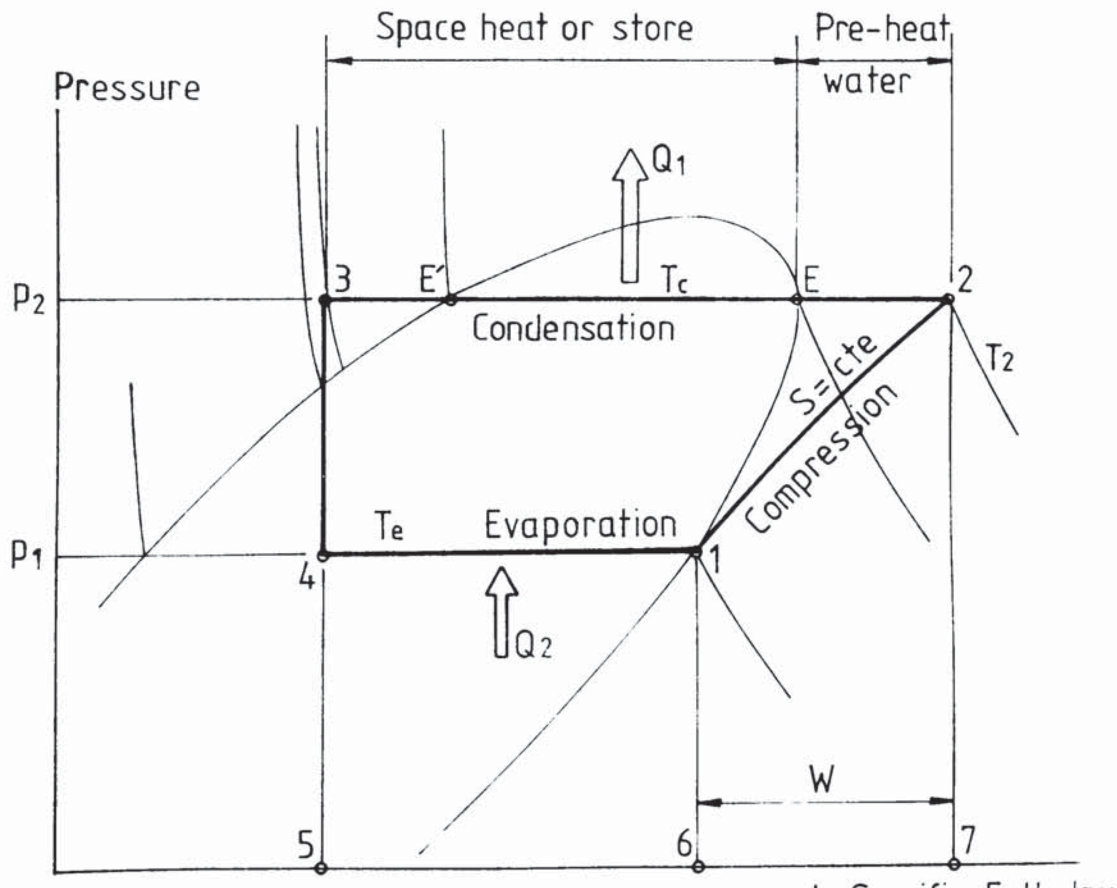


Fig 3b Pressure - Enthalpy diagram for the basic Rankine cycle

3. Liquid is expanded irreversibly at constant enthalpy through a throttling valve, between 3 and 4.

4. Liquid refrigerant is evaporated isothermally to vapour at constant pressure and at the lower temperature Te, absorbing heat in the section 4 to 1.

Figure 4 is a schematic representation of the four basic heat pump components. These are:

1. An Evaporator where heat is collected at the lower temperature.

2. A Compressor which performs the work required.

3. A Condenser where useful heat is released.

4. An Expansion Valve which regulates the flow of refrigerant from the high to the low pressure side of the machine.

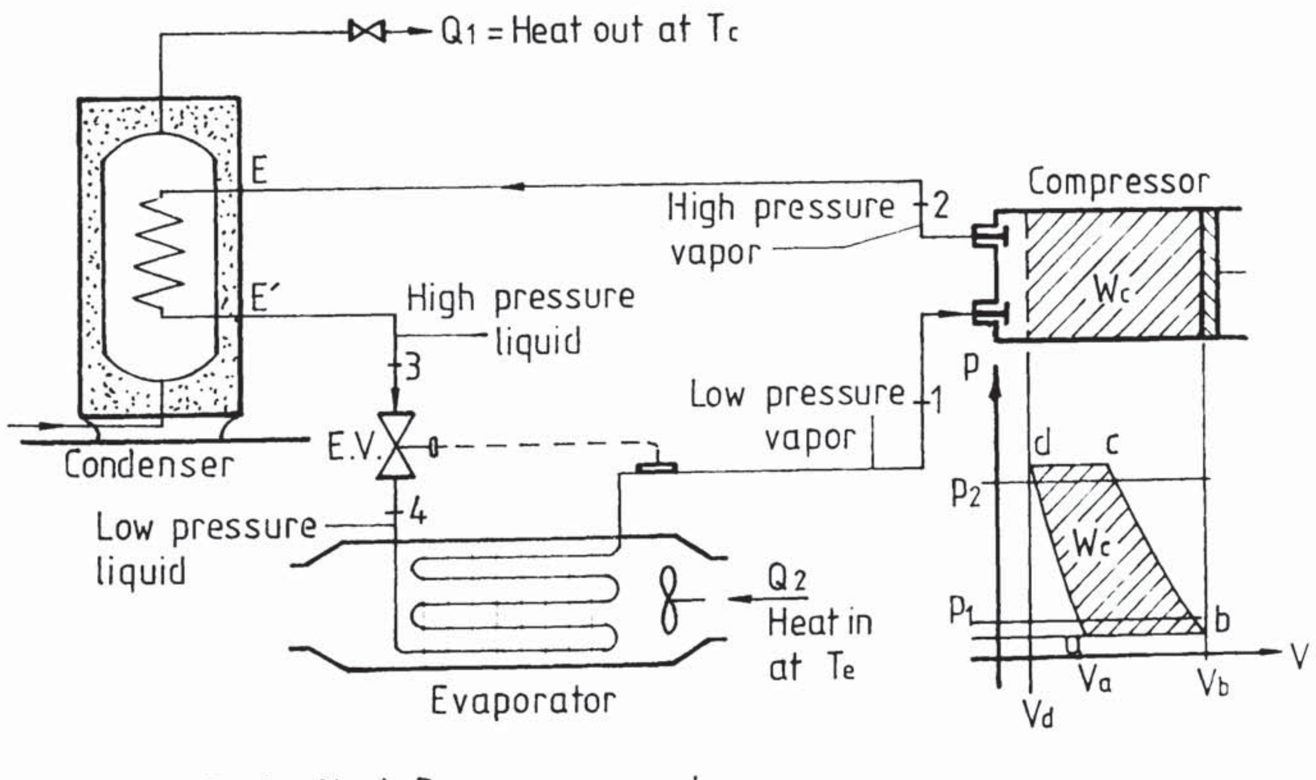


Fig 4 Basic Heat Pump components

In an ideal situation the energy balance of the system is give by:

$$Q_1 = Q_2 + W_c \qquad \text{(II-1)}$$

Where:

Q_1 = The useful energy released

Q_2 = The energy collected from the environment

W_c = The work done in compression

If "h", represents the enthalphy of the refrigerant in each point and if $\dot{m}$ is the mass flow rate of refrigerant in the cycle, the heat absorbed per second in the evaporator at the lower temperature Te is given by:

$$\dot{Q}_2 = \dot{m}\,(h_1 - h_4) = \text{Refrigeranting effect} \qquad \text{(II-2)}$$

The stage from 1 to 2 in the ideal case is an adiabatic-isoentropic compression and the work done on the compressor per second will be:

$$\dot{W}_c = \dot{m}\,(h_2 - h_1) \qquad \text{(II-3)}$$

The heat rejected per second at the condenser at the temperature Tc will be:

$$\dot{Q}_1 = \dot{m}\,(h_2 - h_3) = \text{Heating effect} \qquad \text{(II-4)}$$

The change from 3 to 4 is through the ideal expansion valve at constant enthalpy so that $h_3 = h_4$.

II-2-3 Real Rankine Cycle Operation

In practice there are several reasons why the ideal situation as indicated in Figure 3 cannot be realised. The compression will not be isentropic. The compressor suction pressure has to be less than the evaporator exit pressure and the compressor expulsion pressure has to be greater than the condenser inlet pressure because of the valves. Some heating may take place in the line from the evaporator to the compressor by extracting heat from the surroundings.

Frequently the refrigerant entering the expansion valve may be subcooled so that its temperature is less than the saturation temperature corresponding to its pressure. This cooling may occur in the condenser or in the liquid line by heat exchange with the ambient surroundings. The vapour may be superheated a few degrees in the evaporator and may be further superheated in the compressor suction line. On occasions superheating is arranged following evaporation so that no liquid gets back to the compressor. In addition we should not ignore the pressure drops which occur in both the condenser and the evaporator. Figure 5 shows the realistic cycle taking into account all the factors mentioned, and Figure 6 shows some of the recommended values for this type of cycle (2, 6, 7). In these figures:

A A' represents subcooling in condenser and pipe: Δ T subc. ≈ 8.5°C

C M represents superheating in evaporator and pipe: Δ T suph. ≈ 5.5°C

M C' represents deliverate superheating in suction line

Pressure drop D' to A' represents loss in condenser: Δ p ≈ 0.4 bar

Pressure drop B' to C represents loss in evaporator: Δ p≈ 0.4 bar

C' F represents suction line pressure drop: Δ p = equivalent to 1.1°C

D D' represents discharge line pressure drop: Δ p = equivalent to 1.1°C

The subcooling of the liquid before expansion through the throttle is rejected heat. By heat exchange it is possible to transfer some

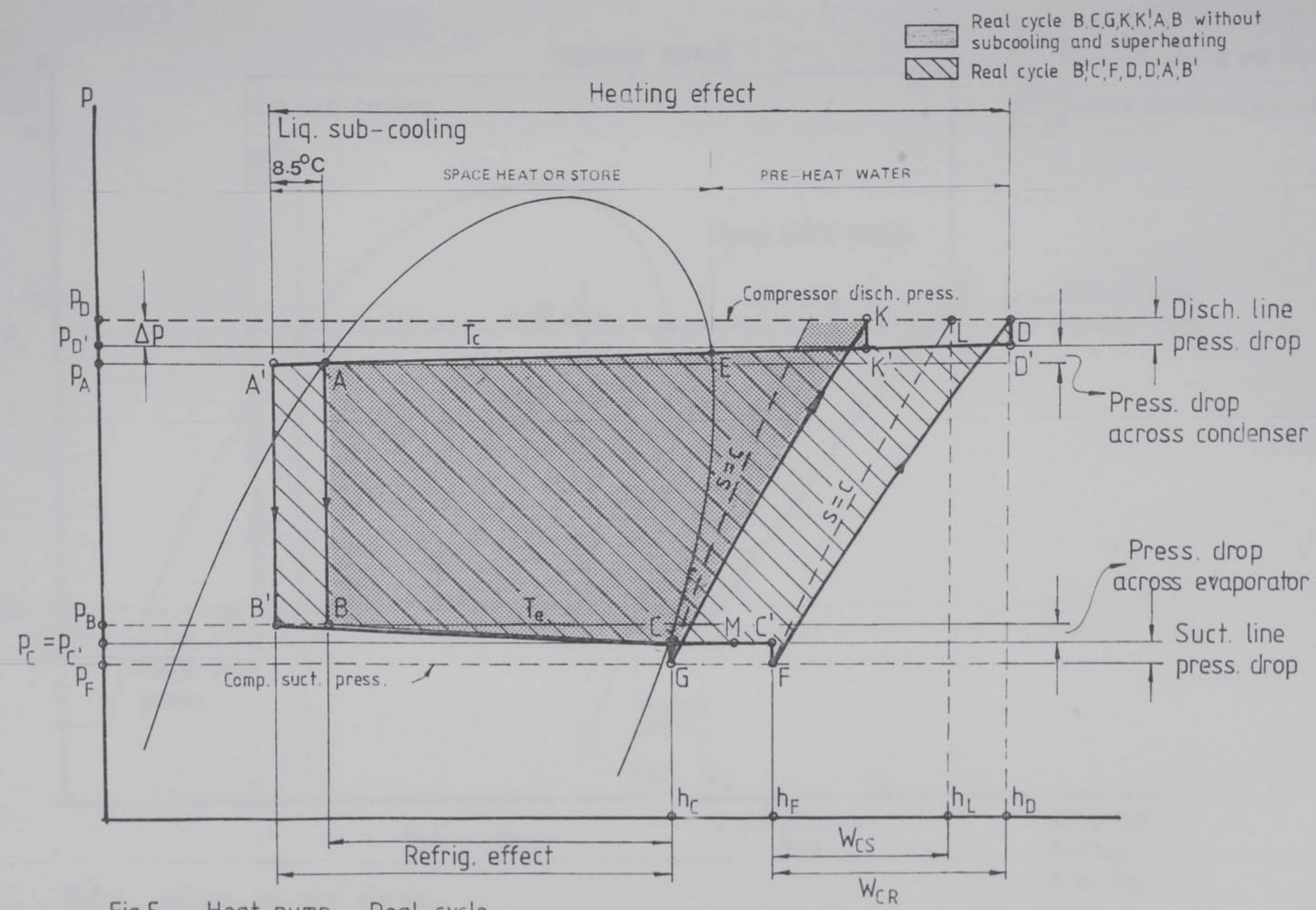


Fig 5 Heat pump - Real cycle

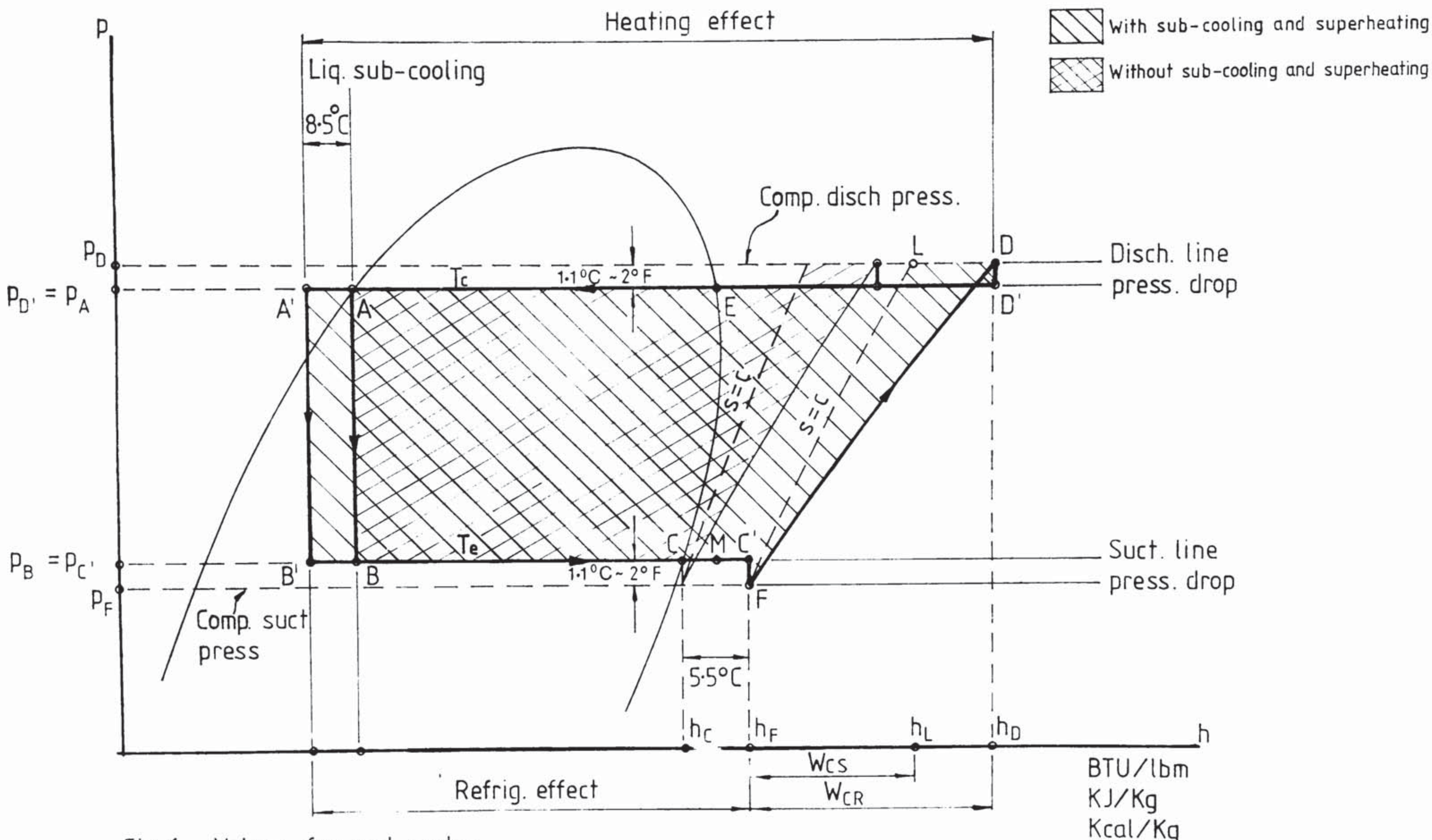


Fig 6 Values for real cycles

of this heat to the vapour MC'. The subcooling can be in the region of 8.5°C (15°F) and the superheating of vapour from the evaporator in the region of 5°C (10°F).

In actual operating conditions (8) the condensing temperature should be about 5 to 12°C above the surroundings and the evaporating temperature 5 to 12°C below the source of heat to allow for heat transfer. The heat transfer characteristics are of course better for higher temperature differences.

II-3 Coefficient of Performance and Efficiency (2,8)

The most useful parameter for comparing the behaviour of Heat Pumps is the coefficient of performance C.O.P. One can define two coefficients of performance:

(a) In the refrigeration or cooling mode, C.O.P. (C) = Heat absorbed/work done

(b) In the heating mode, C.O.P. (H) = Heat rejected/work done

II-3-1 C.O.P. of the Carnot Cycle

Assuming reversibility it is possible to define these two coefficients for such a machine in terms of the absolute temperatures Tc and Te, of the condenser and evaporator respectively between which the machine operates, since the areas representing the energy quantities, Q_1, Q_2 and $Q_1 - Q_2$, are rectangular for the Carnot Cycle (Figure 2).

$$[\text{C.O.P. (H)}]_R = \frac{Q_1}{W} = \frac{Tc}{Tc-Te} \qquad \text{(II-5)}$$

$$[C.O.P.\ (C)]_R = \frac{Q_2}{W} = \frac{Te}{Tc-Te} \qquad (II-6)$$

Where R signifies reversibility. It is seen therefore that:

$$C.O.P\ (H) = 1 + C.O.P.\ (C)$$

It is the fact that this ratio can be several times greater than unity, which makes the Heat Pump attractive when energy economy is sought.

II-3-2. C.O.P. of the Rankine Cycle

The coefficient of performance of the Rankine Cycle is more representative of an actual heat pump operating cycle and can be defined in the following terms:

(a) In the heating mode

C.O.P. (H) = Useful heat rejected at stated conditions/Net energy supplied from external sources to operate the system.

(b) In the cooling mode

C.O.P. (C) = Useful refrigerant effect/Net energy supplied from external sources to operate the system.

The C.O.P. (H) and the C.O.P. (C) of Rankine Cycle can be expressed in terms of the enthalpies of the cycle for both the theoretical or ideal and the real Rankine process.

So, one can calculate, in terms of the enthalpies, an ideal C.O.P.

for the theoretical Rankine Cycle 1-2-3-4, shown in Figure 3b, in which $\dot{m}$ is the refrigerant mass flow rate, $\dot{Q}_1 = \dot{m}\,(h_2 - h_3)$ the total heat rejected, and $W_c = \dot{m}\,(h_2 - h_1)$ the total energy required to drive the compressor:

$$[C.O.P.\ (H)]_I = \dot{Q}_1/\dot{W}_c = (h_2 - h_3)/(h_2 - h_1) \qquad \text{(II-7)}$$

$$[C.O.P.\ (C)]_I = \dot{Q}_2/\dot{W}_c = (h_1 - h_4)/(h_2 - h_1) \qquad \text{(II-8)}$$

And a coefficient of performance for the real Rankin Cycle, in terms of the enthalpies of each point, as shown in Figure 5.

$$C.O.P.\ (H) = \dot{Q}_1/\dot{W}_c = (h_D' - h_A')/(h_D - h_F) \qquad \text{(II-9)}$$

$$C.O.P.\ (C) = \dot{Q}_2/\dot{W}_c = (h_C' - h_B')/(h_D - h_F) \qquad \text{(II-10)}$$

II-3-3 <u>C.O.P. of an actual heat pump cycle</u>

The Rankine Cycle C.O.P. as given by equations II-7 to II-10 will be lower than the Carnot Cycle values given by equations II-5 and II-6. In an actual heat pump system the C.O.P. varies directly as the evaporation temperature which is in turn, determined by the temperature of the heat source, and inversely as the condensation temperature which is defined by the heat sink temperature.

The evaporation temperature and the condensation temperature determine respectively, the suction pressure and the head pressure of the compressor, so that the lower the heat source temperature, the lower the compressor suction pressure and the lower the C.O.P. of the system.

In most cases the actual performance (3) may be 60% to 70% of the ideal value which in itself will be less than the value for a reversible machine working as the Carnot Cycle. The C.O.P. of an actual heat pump may be defined as:

$$C.O.P.\ (H) = \dot{Q}_1/\dot{W}_{HP} = \dot{Q}1/(\dot{W}_C + \dot{W}_A) \qquad (II-11)$$

$$C.O.P.\ (C) = \dot{Q}_2/\dot{W}_{HP} = \dot{Q}_2/(\dot{W}_C + \dot{W}_A) \qquad (II-12)$$

Where $\dot{W}_A$, is the energy required to drive ancillary equipment. (In addition to compressor work, energy is required to move fluids across the evaporator and/or condenser).

The main causes for the lower C.O.P. of an actual heat pump system are:

(a) The temperature gradients which are necessary for the heat transfer from the refrigerant to the heat source and heat sink.

(b) The fan power which is necessary to move the air over the heat transfer surfaces and represents about 10 to 15 percent of the total energy input to the compressor.

(c) The theoretical compression cycles do not take into consideration compressor volumetric efficiency.

(d) The compressor-motor combinations is only 85 to 90 percent efficient.

(e) The pressure drops at the suction and exhaust valves of the compressor.

II-3-4. Efficiency of the heat pump cycle

The efficiency of an actual heat pump is defined in terms of a reversible engine, so the heating efficiency will be:

$$\eta_H = C.O.P.\ (H)/[C.O.P.\ (H)]_R \qquad \text{(II-13)}$$

In practice one can only expect to achieve about 40% to 50% of the ideal Carnot values.

II-3-5. Coefficient of Performance Application

Figure 8, drawn from the elaborated table II-1 and II-2, has been made in order to illustrate the realistic predicted performance which could be expected of an actual heat pump designed for operation in the heating mode, using a refrigerant identified commercially as R12. This figure shows the variation in C.O.P.(H) and the variation in $[C.O.P.(H)]_R$ with different evaporation temperatures and two condensation temperatures of 38°C (100°F) and 49°C (120°F), compared with values for a reversible engine working between the same temperature limits.

Enthalpy values for each point of the Rankine Cycle, indicated in Figure 7, were taken from a Mollier Pressure - Enthalpy Chart for Freon R-12 Refrigerant. Two cases have been considered:

(a) A cycle with subcooling (ΔT = 8.5°C) in the condenser and superheating (ΔT = 5.5°C) in the evaporator.

(b) A cycle without subcooling and without superheating.

In this exercise the energy expended on ancillary equipment has been assumed to be 15% of the energy expended on compression. The compressor was assumed to have an overall efficiency of 70%.

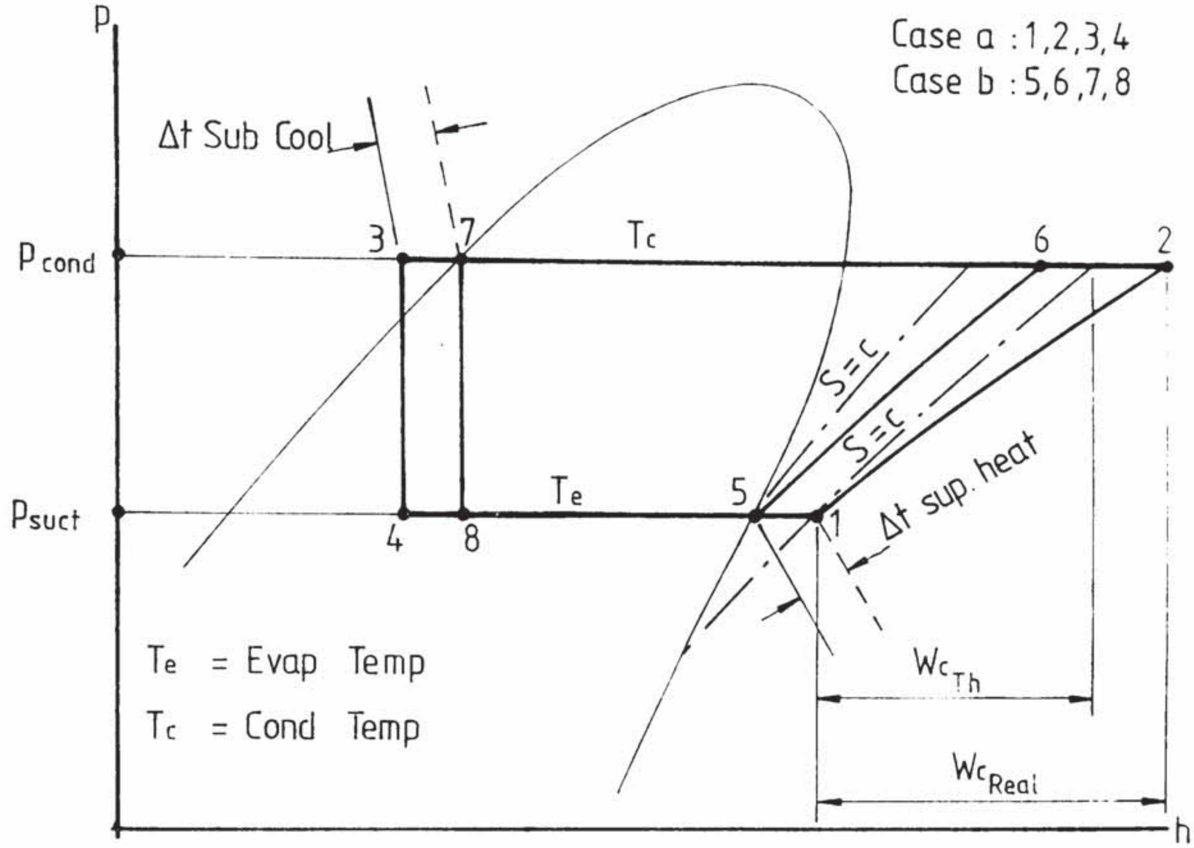


Fig 7 Rankine cycle for COP applications

Reversible machine: Carnot

$$[C.O.P.\ (H)]_R = Tc/(Tc - Te) \qquad (II\text{-}14)$$

Actual Heat Pump: Rankine

$$C.O.P.\ (H) = q_1/(w_C + w_A) = q_1/w_{HP} \qquad (II\text{-}15)$$

Absorbed heat:

$$q_2(a) = h_1 - h_4\ ;\ q_2(b) = h_5 - h_8$$

Rejected heat:

$$q_{1\,(a)} = h_2 - h_3 \; ; \qquad q_{1\,(b)} = h_6 - h_7$$

Work done to drive the compressor:

$$W_c \text{ real} = \frac{W_c \text{ theor}}{\eta \text{ comp}} \quad \begin{cases} a) & = h_2 - h_1 \\ b) & = h_6 - h_5 \end{cases}$$

η comp = 0.70

Energy expended on ancillary equipment:

$$W_A = 0.15\ W_c \text{ real}$$

Total Energy expended

$$W_{HP} = W_c \text{ real} + W_A = W_c \text{ real} + 0.15\ W_c \text{ real} =$$

$$W_{HP} = 1.15\ W \text{ comp real}$$

Table II-1 and II-2 show the values obtained for the cases (a) and (b) which have been represented in Figure 8 for the heating mode only.

Efficiency of the cycles, for given evaporating - condensing temperature combinations, remains about 50% of the Carnot efficiency in most cases. It can be seen that useful energy released is between 2 and 4 times the energy expended.

Table II-1. Carnot and Real C.O.P. (H) values for Freon 12, with a condensation temperature of 38°C

Te °C	q_2 kJ/kg	q_1 kJ/kg	W_C theor kJ/kg	W_C Real kJ/kg	W_A kJ/kg	COP (H) Carnot	COP (C)	COP (H)	Effic. η(H)
Case (a)	Tc = 38°C (100°F); W_A = 15% W_C real, η comp = 0.70		Δt superheating = 5.5°C (10°F); Δt subcooling = 8.5°C (15°F)						
-29	115.1	174.9	41.9	59.8	8.96	4.67	1.67	2.54	0.54
-23	117.7	171.4	37.5	53.5	8.02	5.09	1.91	2.78	0.55
-18	120.3	167.7	33.3	47.5	7.12	5.60	2.20	3.07	0.55
-12	124.0	166.1	29.5	42.1	6.33	6.22	2.56	3.42	0.55
-7	126.1	163.5	26.1	37.2	5.58	7.00	2.95	3.82	0.55
-1	128.9	160.7	22.3	31.9	4.79	8.00	3.52	4.38	0.55
4.5	131.4	157.9	18.6	26.5	3.98	9.33	4.31	5.17	0.55
10	134.0	155.8	15.4	21.9	3.30	11.20	5.33	6.18	0.55
15.5	136.1	153.3	12.1	17.2	2.58	14.00	6.85	7.71	0.55
Case (b)	Tc = 38°C (100°F); W_A = 15% W_C real, η comp = 0.70		Δt subcooling = 0°C; Δt superheating = 0°C						
-29	104.0	160.0	39.1	55.8	8.37	4.67	1.62	2.49	0.53
-23	107.0	158.2	36.1	51.4	7.72	5.09	1.81	2.68	0.53
-18	109.6	154.9	31.9	45.6	6.82	5.60	2.09	2.96	0.53
-12	111.7	151.7	27.9	39.8	5.98	6.22	2.44	3.31	0.53
-7	114.4	148.6	24.0	34.2	5.12	7.00	2.91	3.78	0.54
-1	116.8	147.7	20.9	30.0	4.42	8.00	3.38	4.28	0.54
4.5	119.8	144.7	17.5	24.9	3.74	9.33	4.19	5.05	0.54
10	122.1	142.1	14.0	20.0	3.00	11.20	5.31	6.19	0.55
15.5	124.7	139.1	10.9	15.6	2.35	14.00	6.96	7.74	0.55

Table II-2. Carnot and Real C.O.P. (H) values for Freon 12, with a condensation temperature of 49°C

Te °C	q_2 kJ/kg	q_1 kJ/kg	W_C theor kJ/kg	W_C Real kJ/kg	W_A kJ/kg	COP (H) Carnot	COP (C)	COP (H)	Effic. $\eta_{(H)}$
Case (a)	Tc = 49°C (120°F) W_A = 15% W_C real, η comp = 0.70 Δt subcooling = 8.5°C (15°F) Δt superheating = 5.5°C (10°F)								
-29	103.5	170.0	46.5	66.4	9.98	4.14	1.35	2.22	0.54
-23	106.3	167.1	42.6	60.8	9.12	4.46	1.52	2.39	0.54
-18	109.1	164.2	38.6	55.1	8.28	4.83	1.72	2.59	0.54
-12	112.4	161.5	34.4	49.2	7.37	5.27	1.99	2.85	0.54
-7	144.4	159.3	31.4	44.9	6.72	5.80	2.22	3.09	0.53
-1	117.2	156.3	27.7	39.5	5.93	6.44	2.58	3.43	0.53
4.5	119.8	156.0	24.9	36.2	5.42	7.25	2.88	3.82	0.53
10	122.1	153.7	21.9	31.2	4.68	8.28	3.40	4.28	0.52
15.5	124.4	150.3	18.1	25.9	3.88	9.67	4.18	5.05	0.52

Case (b) $T_c = 49°C\ (120°F)$

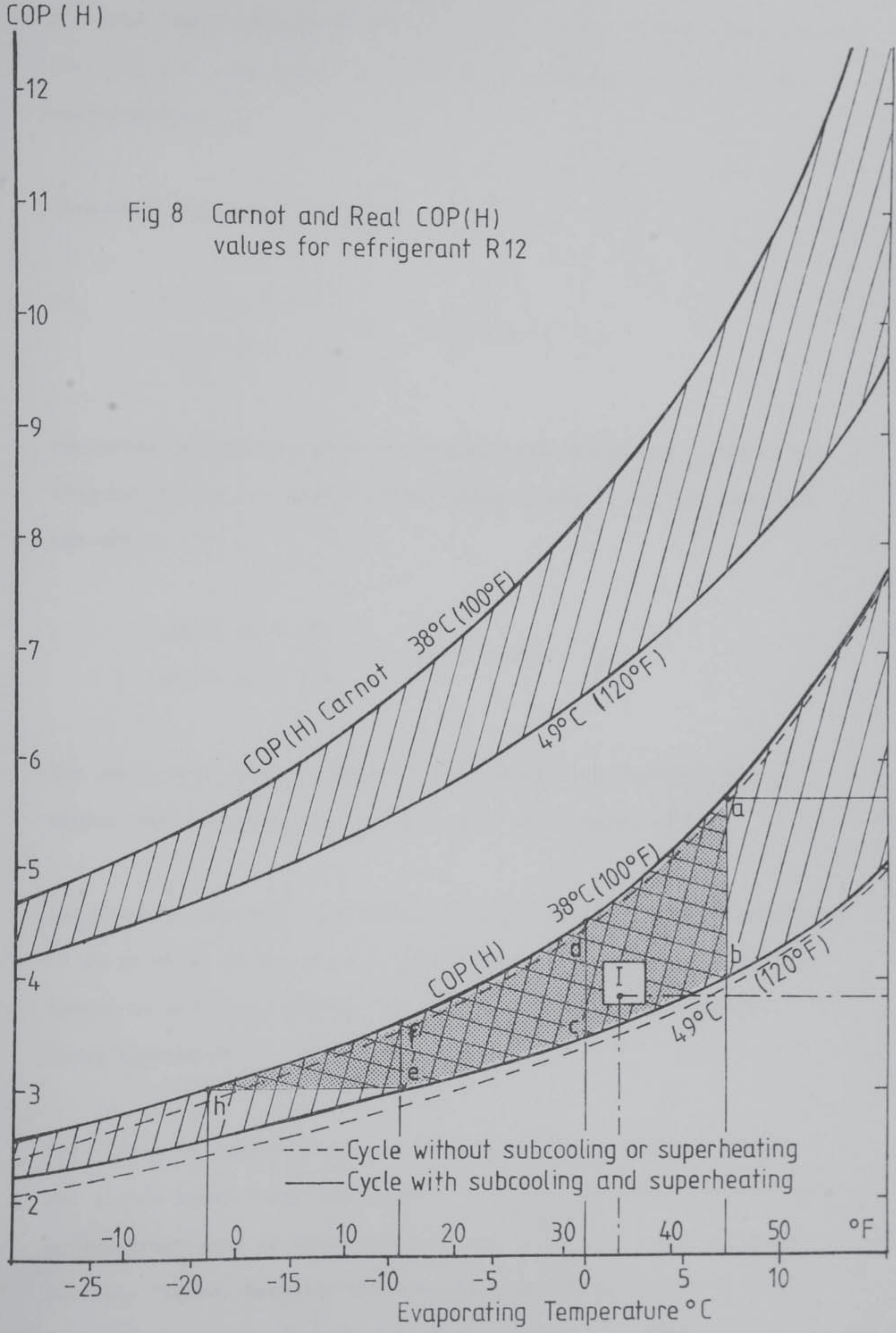
$W_A = 15\%\ W_C$ real; $\eta_{comp} = 0.70$

Δt subcooling = 0°C

Δt superheating = 0°C

-29	92.6	157.0	45.1	64.4	9.68	4.14	1.25	2.12	0.51
-23	95.4	153.8	40.9	58.5	8.77	4.46	1.42	2.29	0.51
-18	97.7	150.8	37.2	53.1	7.98	4.83	1.60	2.47	0.51
-12	100.0	148.2	33.7	48.2	7.23	5.27	1.81	2.67	0.51
-7	103.0	144.6	29.1	41.5	6.23	5.80	2.16	3.03	0.52
-1	105.8	142.0	25.6	36.5	5.49	6.44	2.52	3.38	0.52
4.5	107.7	140.6	22.8	32.6	4.88	7.25	2.88	3.75	0.52
10	110.5	138.7	19.8	28.2	4.23	8.28	3.40	4.27	0.52
15.5	113.0	137.2	16.8	23.9	3.58	9.67	4.11	4.99	0.52

Since the raising of the head pressure and the lowering of the suction pressure reduce the coefficient of performance, it is important to operate the system at minimum head pressure and maximum suction pressure, (i.e. at minimum condensation temperature and at maximum evaporation temperature). However, there are temperature limits to take into account when operating domestic heating systems. Hot water radiators requires a minimum water temperature (3, 8, 9) of about 72°C; underfloor heating requires water temperature about 52°C as a minimum, and warm air heating requires minimum air temperatures of about 27°C. These values limit the minimum condensing temperature. On the other hand for an air source heat pump, at air temperature below 4°C and at high relative humidities, icing can occur on the evaporator coils, resulting in a greatly reduced performance, because removal of ice requires the use of wasteful


COP (H)
12
11
10
9
8
7
6
5
4
3
2
Fig 8 Carnot and Real COP(H) values for refrigerant R12
COP(H) Carnot
38°C (100°F)
49°C (120°F)
COP(H)
38°C (100°F)
49°C
(120°F)
a
b
c
d
e
f
h
I
----Cycle without subcooling or superheating
——Cycle with subcooling and superheating
-10
0
10
20
30
40
50
°F
-25
-20
-15
-10
-5
0
5
10
Evaporating Temperature °C

additional energy.

The area "abcd" (Figure 8) indicates the region of heat pump operation (for the U.K. conditions) which would be expected to give the best performance.

From this figure one can write:

$$\text{If,}\quad \left.\begin{array}{l} 0°C < Te < 7°C \\ 38°C < Tc < 49°C \end{array}\right\} \quad 3.5 < COP\ (H) < 5.5$$

Operating in the area a-b-e-f (without considering the icing on the evaporator) one can expect COP(H) values higher than 3.0. Also one can write:

$$\left.\begin{array}{l} -9°C < Te < 7°C \\ 38°C < Tc < 49°C \end{array}\right\} \quad 3.0 < COP(H) < 5.5$$

The area e-f-h, with Te values $< -9°C$, also permits COP(H) values higher than 3.0, but only for Tc values approaching 38°C.

And finally, for Te $< -20°C$ and Tc $> 38°C$, the COP(H) will be always < 3 as is shown in the figure. Coefficients of Performance therefore range, in practice, between 2.2 and 2.7 for source temperatures lying between 0°C ($\sim$ 30°F) and 10°C ($\sim$ 45°F).

The effect of superheating the refrigerant vapour and subcooling the liquid leaving the condenser to increase the COP(H) is seen to be important only at lower values of Te (Te $<$ 0°C) as is shown in the same figure. Normally the COP(H) decreases by superheating the vapour and increases by subcooling the liquid.

II-3-6. Superheating and Subcooling Effect

(a) Superheating Effect

Even though the superheating of the gas entering the compressor may decrease the COP(H), because of the increase in compression work, there are some practical reasons to do it:

Superheating of the vapour in the evaporator is necessary to avoid liquid refrigerant reaching the compressor and to get a proper operation of the expansion valve, which requires 8 to 10 degrees C. of superheat in the evaporator for satisfactory performance.

An additional gas-liquid heat exchanger may be employed for superheating the vapour by subcooling the liquid leaving the condenser, making it unnecessary to use the evaporator for the superheating purpose. This increases the refrigerating effect of the heat pump. Moreover, in heat pumps with long uninsulated suction lines outside the conditioned space, in which the heat gain can not be avoided, it is economically justifiable to use this additional heat exchanger, since the necessity for insulating the suction line is then eliminated.

(b) Subcooling Effect

Subcooling of the liquid increases the coefficient of performance of the heat pump. If the liquid leaving the condenser is subcooled below the saturation temperature, see point 3 Figure 7, the refrigerating effect and the useful heat output per kg of liquid circulated are increased without any change in the work of compression.

In contrast to the disadvantages resulting from superheating the suction vapour, the subcooling of the liquid refrigerant will always improve the heat output and the COP(H) of the system.

The subcooling of the liquid can be accomplished in the condenser itself or by means of the additional gas-liquid heat exchanger placed in the suction line.

The subcooling of the liquid refrigerant is also necessary to prevent flashing of gas in the liquid line, when excessive pressure drops are present in it.

II-4. Refrigerants Used in Heat Pumps (6, 10, 11)

II-4-1. Organic Halocarbon Refrigerants

The refrigerant is the working fluid of the heat pump. Many refrigerants have been variously proposed and used in heat pump systems, but in modern machines, following refrigeration practice, the working fluid employed is usually one of a family of Halocarbon Compounds, known comercially under the trade name of Freons. This group of halogen compounds is divided into two, the first based on methane, the second on ethane. Refrigerants 10, 11, 12, 13, 14, 20, 21, 22 are some examples of the first group and refrigerants 110, 111, 112, 113, 114 etc., examples of the second group. Table II-3 gives information on some of these organic refrigerants. One can see from this table that refrigerants R113 and R114 have high normal

Table II-3. Organic Refrigerants (Freons)

R	Chemical name	Formula	Mol. Weight $\frac{kg}{k\ mol}$	Normal boiling point		Critical Temperature		Freezing Point
				°C	°F	°C	°F	°C
10	Carbon Tetrachloride	$C\ Cl_4$	153.8	76.7	170.2	283.0	541.4	-22.3
11	Trichloromonofluoromethane	$C\ Cl_3\ F$	137.4	23.8	74.8	198.0	388.4	-111.0
12	Dichlorodifluoromethane	$C\ Cl_2\ F_2$	120.9	-29.7	-21.6	111.5	232.7	-157.3
13	Monochlorotrifluoromethane	$C\ Cl\ F_3$	104.5	-81.4	-114.6	28.8	83.9	-181.1
14	Carbontetrafluoride	$C\ F_4$	88.0	-128.0	-198.4	-45.5	-49.9	-183.9
20	Chloroform	$CH\ Cl_3$	119.4	61.1	142.0	260.0	500.0	-63.5
21	Dichloromonofluoromethane	$CH\ Cl_2\ F$	102.9	8.9	48.1	178.5	353.3	-135.0
22	Monochlorodifluoromethane	$CH\ Cl\ F_2$	86.5	-40.7	-41.4	96.0	204.8	-160.0
23	Trifluoromethane	$CH\ F_3$	70.0	-84.4	120.0	26.1	79.0	-155.0
30	Methylene chloride	$CH_2\ Cl_2$	84.9	40.6	105.2	235.0	455.0	-97.0
40	Methyl chloride	$CH_3\ Cl$	50.5	-23.8	-10.8	143.1	289.6	-97.5
50	Methane*	CH_4	16.0	-161.6	-259.0	-82.2	-116.0	-182.0
110	Hexachloroethane	$C_2\ Cl_6$	236.8	185.0	365.0			
111	Pentachloromonofluorethane	$C_2\ Cl_5\ F$	220.3	137.2	279.0			
112	Tetrachlorodifluoroethane	$C_2\ Cl_4\ F_2$	203.8	92.8	199.0			
113	Trichlorotrifluoroethane	$C_2\ Cl_3\ F_3$	187.4	47.6	117.6	214.1	417.4	-35.0
114	Dichlorotetrafluoroethane	$C_2\ Cl_2\ F_4$	170.9	3.6	38.4	145.7	294.3	-93.9
115	Monochloropentafluoroethane	$C_2\ Cl\ F_5$	154.5	-38.7	-37.7	80.0	176.0	-106.0
116	Hexafluoroethane	$C_2\ F_6$	138.0	-78.2	-108.8			
120	Pentachloroethane	$C_2\ H\ Cl_5$	202.3	162.2	324.0			
142b	Chlorodifluoroethane	$C_2\ H_3\ ClF_2$	100.5	-9.8	14.4	137.1	278.8	-131.1
152a	Difluoromethane	$C_2\ H_4\ F_2$	66.1	-25.0	-13.0	133.5	236.3	-117.0
160	Ethyl chloride	$C_2\ H_5\ Cl$	64.5	12.2	54.0	185.0	365.0	-138.2
170	Ethane*	$C_2\ H_6$	30.0	-88.6	-127.5	33.0	90.0	-133.0

* Not Halocarbons; References: (9), (10), (11) and (12)

boiling points and relatively high critical temperatures, R12 and R22 have intermediate normal boiling points and moderately high critical temperatures, R13 and R14 have low normal boiling points and low critical temperatures.

The "Normal Boiling Point" is referred to as: Liquid and Vapour in equilibrium at the pressure of 1013 millibars.

Due to the possibility of using evaporating temperatures at about 2°C (35°F), a refrigerant having a normal boiling point near to this value will be good for heat pump heating applications.

II-4.2. Azeotropic Refrigerants (10), (11)

An azeotropic solution is one containing at least two components which boil at some given pressure without change in composition. There are two types of azeotropes that must be considered. One has a pressure lower and a boiling point higher than either of the pure constituents; the other has a pressure higher and a boiling point lower than either of the constituents. One is called a minimum pressure or maximum temperature azeotrope; the other is the maximum pressure or minimum temperature type. The series number 500 has been allocated to azeotropes; for example R500 is an azeotropic mixture at 0°C containing 26.2% of R152 and 73.8% of R12. It is the only commercially available azeotrope. Table II-4 shows some azeotropic mixtures.

Table II-4. Azeotropic Refrigerants

Name	Refrigerants 1 - 2	Normal Boiling points (°C) 1	2	Azeotrop.	% by Weight of Comp. 1
R 500	R 152a - R 12	-24.1	-29.8	-33.0	26.2
R 501	R 12 - R 22	-29.8	-40.7	-40.8	25.0
R 502	R 22 - R 115	-40.7	-38.7	-45.6	48.8
R 503	R 23 - R 13	-84.4	-81.4	-89.0	40.1
R 504	R 32 - R 115	-51.9	-38.7	-57.6	48.2
	R 40 - R 12	-23.7	-29.8	-32.0	22.0
	R 21 - R 114	8.9	3.6	1.4	25.0
	R 227 - R 12	-15.9	-29.8	-30.0	13.5
	R 152 - R 115	-24.1	-38.7	-41.1	16.0

Already mixtures of R12 and R22 far from the azeotropic composition are being used as refrigerants primarily because the addition of R12 to R22 increases the oil solubility and makes it possible to get much better oil return to the compressor. One can use up to about 25%, by weight of R12 in R22 without getting a great deal for change in capacity. (10)

II-4-3. Refrigerant Selection (6, 10, 11, 12)

Even though the most important factors influencing the choice of refrigerants, for use in comfort air conditioning installations, are the thermodynamic properties, it is also imperative for the refrigerant to have certain desirable physical and chemical properties. An ideal refrigerant should have some of the following characteristics:

1. Positive evaporating pressure; low-side pressures above 1 atmosphere prevent possible leakage into the system during operation.

2. Moderately low condensing pressures, this means that light weight equipment and piping can be used on the high pressure side.

3. Relatively high critical temperature; the critical temperature should be much higher than normal operating temperatures to reduce power requirements, otherwise the efficiency of operation will be extremely low.

4. Low freezing temperature; to prevent solidification of refrigerant during normal operation.

5. Low Cost.

6. High latent heat of vaporisation; this means a high refrigerating effect per unit mass of refrigerant circulated. (In small capacity systems too low rates of flow may result in control problems).

7. Other features include: stability, high heat transfer characteristics, low water solubility, non toxicity, non-irritability, non-inflamability, easy leakage detection, odour, acidity, reactivity with air, oil, and other substances, corrosiveness and lubricating characteristics.

No one fluid satisfies the requirements for all systems. R12 and R22 are most commonly used in domestic type systems. Comparisons will be made later in terms of heat pump applications. R11 has been used in large capacity equipment for water-chilling machines used in air conditioning units for large buildings.

The choice of the type and design of compressor will be influenced by the compression ratio and the refrigerant volume factors.

The compression ratio in centrifugal compressors is affected by the vapour density. The density is related to the molecular weight of the gas as well as to the operating temperature and pressure. The majority of small and medium size refrigeration systems use positive displacement reciprocating compressors, designed to operate with refrigerants having low specific volume and relatively high pressure characteristics. For a good many years nearly all refrigerants used were toxic, flammable, or explosive when mixed with the appropriate amount of air. In modern practice the halocarbon refrigerants have largely replaced other refrigerants as: carbon dioxide, methyl chloride, sulfur dioxide, the hydrocarbons, methylene chloride, and even ammonia, because many of them are non-toxic, nonflammable, and will not explode when mixed with air. Refrigerants R11, R12, R22, R113, R114, and R500 are the most important commercial refrigerants to-day, having these desirable properties.

The chlorine derivatives of methane without fluorine have a toxic and narcotic effect, but this toxicity decreases with increasing fluorine content. In addition, the chemical stability is improved as the number of fluorine atoms increases. A higher fluorine content results in a lower temperature of adiabatic compression and this reduces the swelling action on rubber seals and decreases the solubility for mineral oils.

One reason why the halocarbons are so widely used today is their comparative safety. Of the most popular inorganics, ammonia forms an explosive mixture with air at concentrations of about 20% by volume and is, in addition, toxic. A concentration of 1% can be fatal within one hour. Sulphur dioxide is non-flammable, but is even more toxic. The hydrocarbons are not toxic, but are all flammable and prone to explosions. By contrast the halocarbons are relatively safe. Refrigerants 11 and 22 are classified as being as safe as carbon dioxide. Safest of all are refrigerant 12 and R13b which are regarded as non-toxic in concentrations of up to 20%. (10)

Another factor favouring the halocarbons is their chemical stability. Ammonia is corrosive and cannot be used with copper alloys. Sulphur dioxide will dissolve in water to form sulphurous acid, which can attack iron and steel. By contrast, the halocarbons can generally be used with most common metals.

Refrigerant characteristics change with operating temperatures. Some of them are advantageous for low refrigeration systems between -40 and -5°C, others are more favourable in the air conditioning range of -5 to 5°C. It may also be found that a particular refrigerant is better for a cooling machine than for a heating machine. Maximum allowable compressor discharge temperature varies depending on design, but is usually fixed at 135°C to 175°C. It has been estimated that the rate of chemical reaction of a system doubles with each 10°C increase, reducing the probable life of the machine.

II-5. Comparison of Refrigerants

In order to compare refrigerants as to performance for heat pump applications, either the operating conditions should be the same as nearly as possible or the same machine should be used and let the conditions change as they will. This will now be demonstrated for the particular operating conditions selected for domestic heating.

II-5-1. Operating Conditions

The operating conditions, for this present study of application on refrigerant comparison, have been selected as 2°C (35°F) on the evaporator and 46°C (115°F) on the condenser, which corresponds to point I for R12 in Figure 8. This is in order to have COP(H) values of about 3.0 and because these temperatures are representative of heat pump operating conditions for domestic use.

Two tables of comparative features, II-5 for ideal cycles and II-6 for real cycles, have been prepared and the data given there are presented to depict the relations existing between various characteristics of refrigerants and to show some of the conflicting merits.

The saturated dry vapour compression ideal cycles were considered for R11, R12, R21, R22, R113, R114, R500, R502, R503 and R504 refrigerants. Real cycles were considered with the commercial available refrigerants R12, R22 and R500. Thermodynamics properties of these refrigerants at the different points of the cycles were taken from references (12), (13) and (14).

Data on saturation properties or pressure-enthalpy diagrams for some non commercial refrigerants are available only in Imperial Units. For this reason these tables and the comparison of refrigerants have not been quoted in International Metric Units.

Figure 9 shows the ideal and real cycles, which have been used for the comparison of refrigerants. Figures 10, 11 and 12 show the data

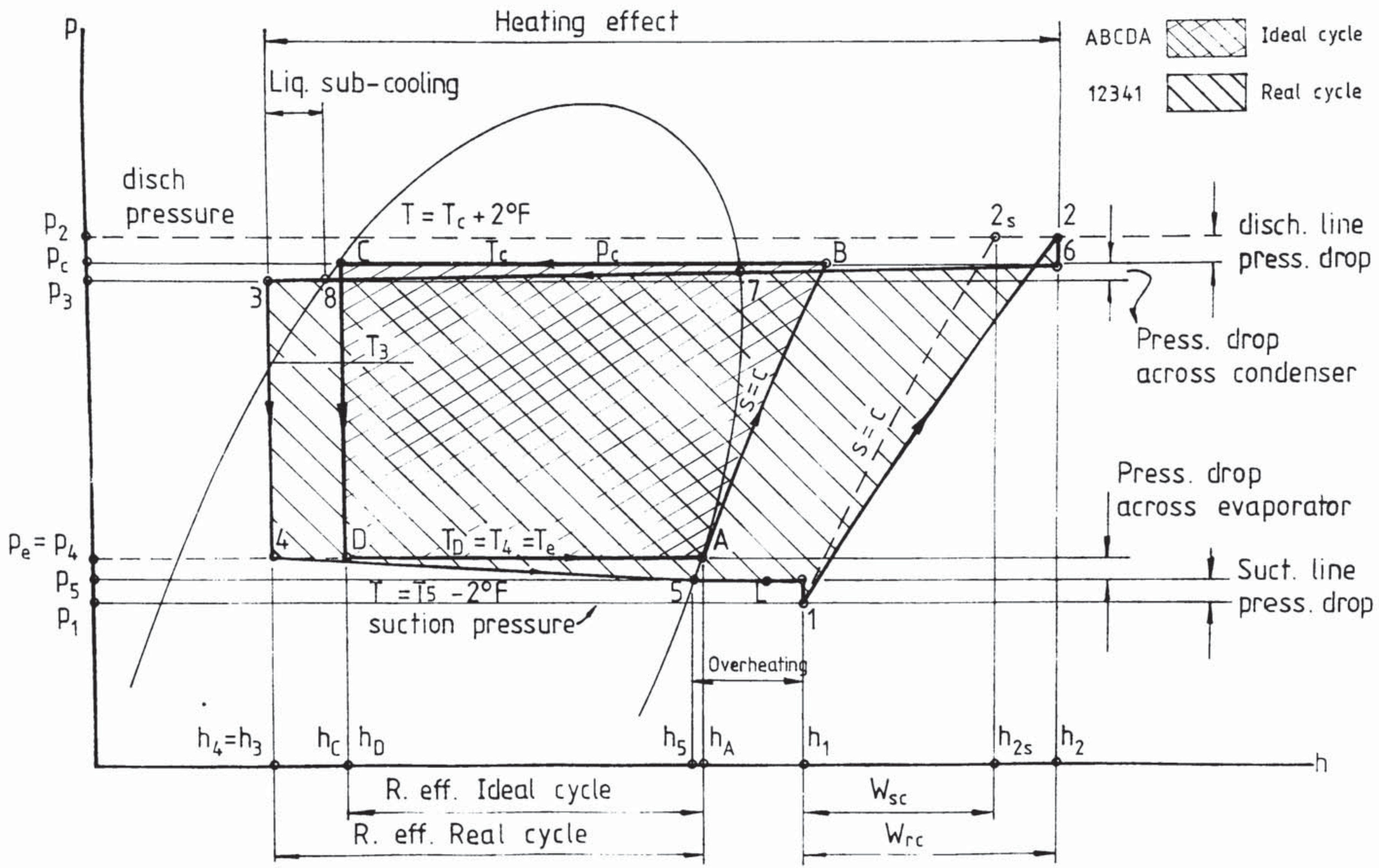


Fig 9 Ideal and Real Rankine cycles for comparison of refrigerants

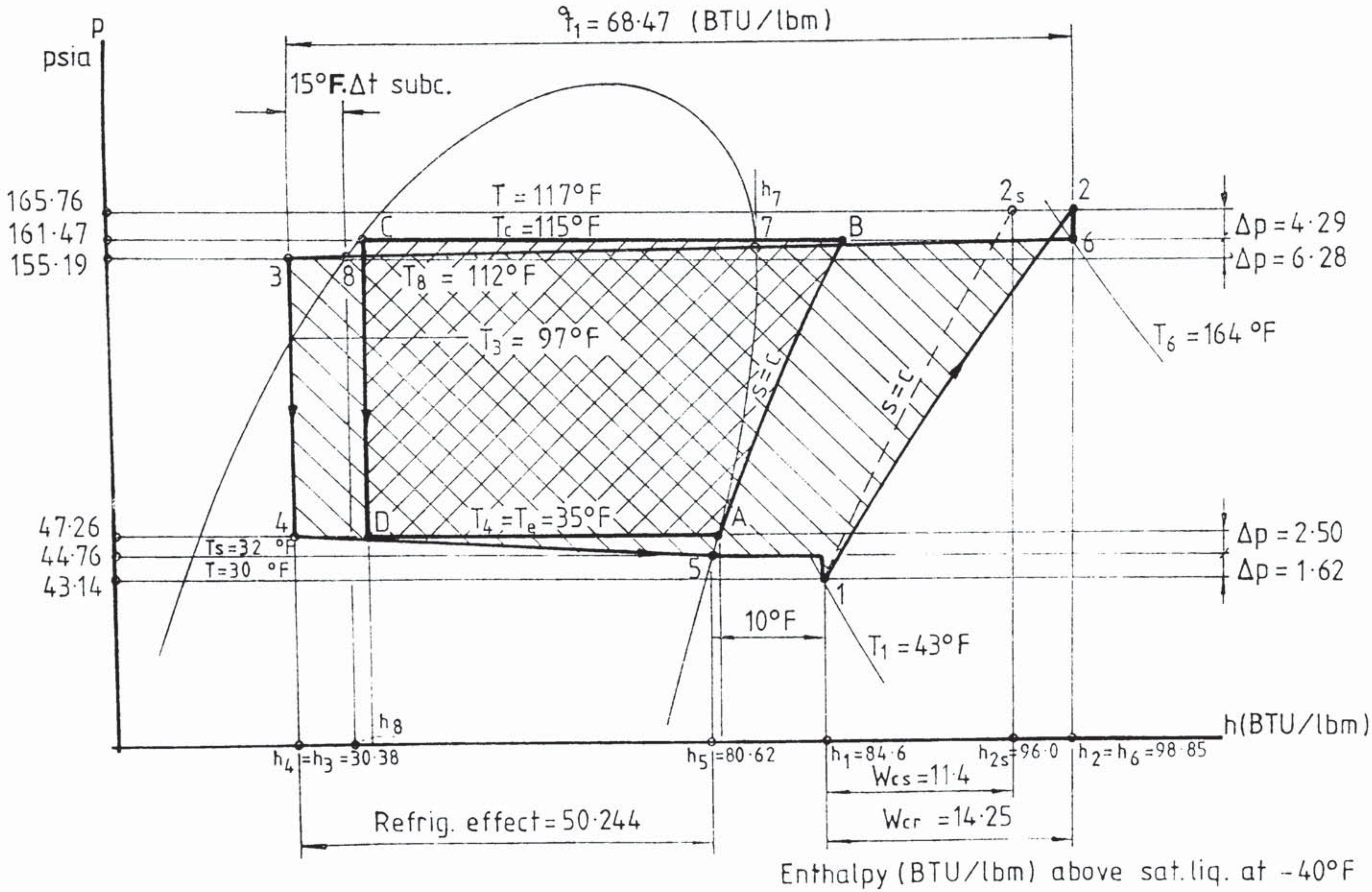


Fig 10 Refrigerant 12 (R 12) real cycle

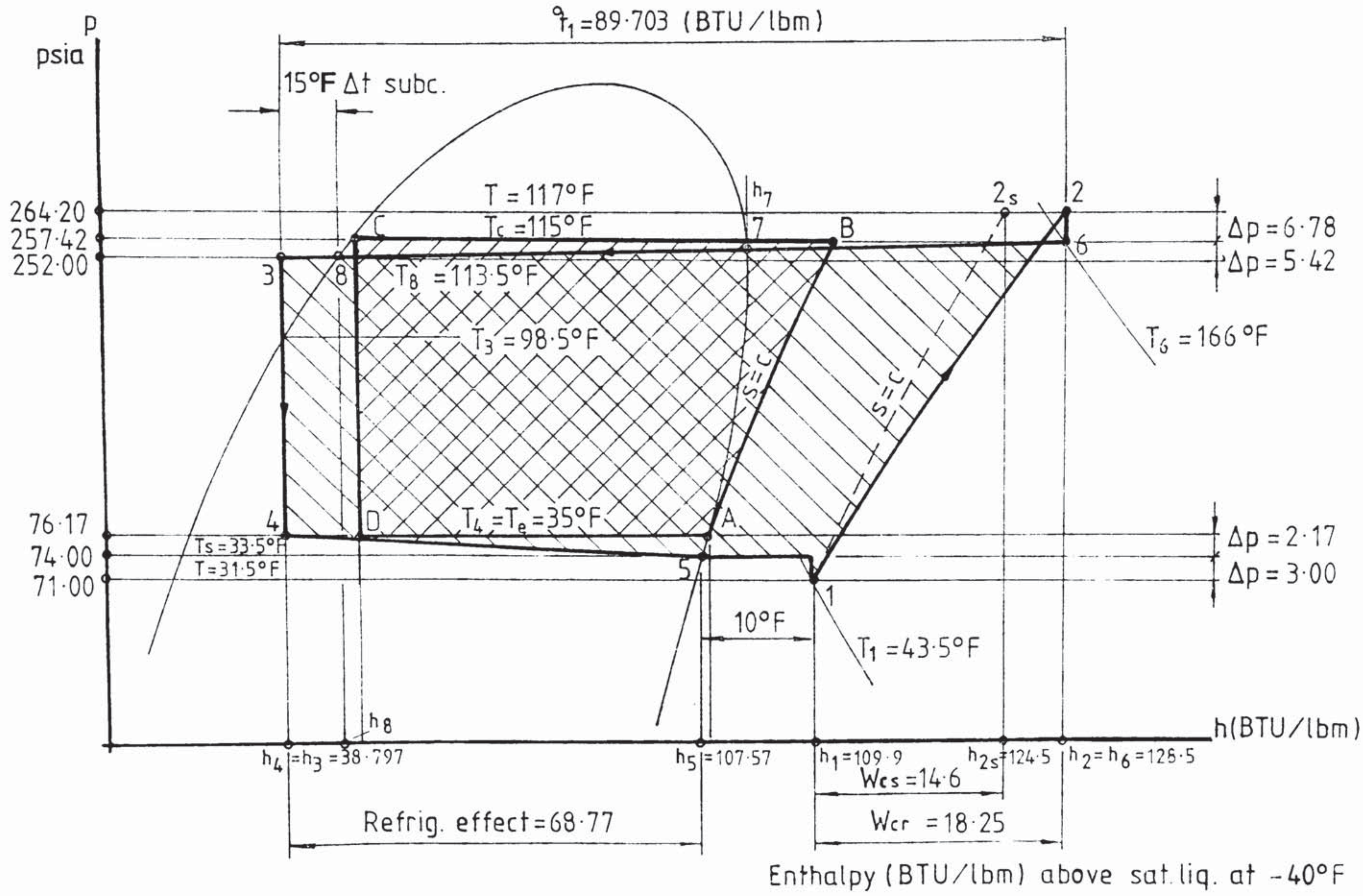


Fig 11 Refrigerant 22 (R 22) real cycle

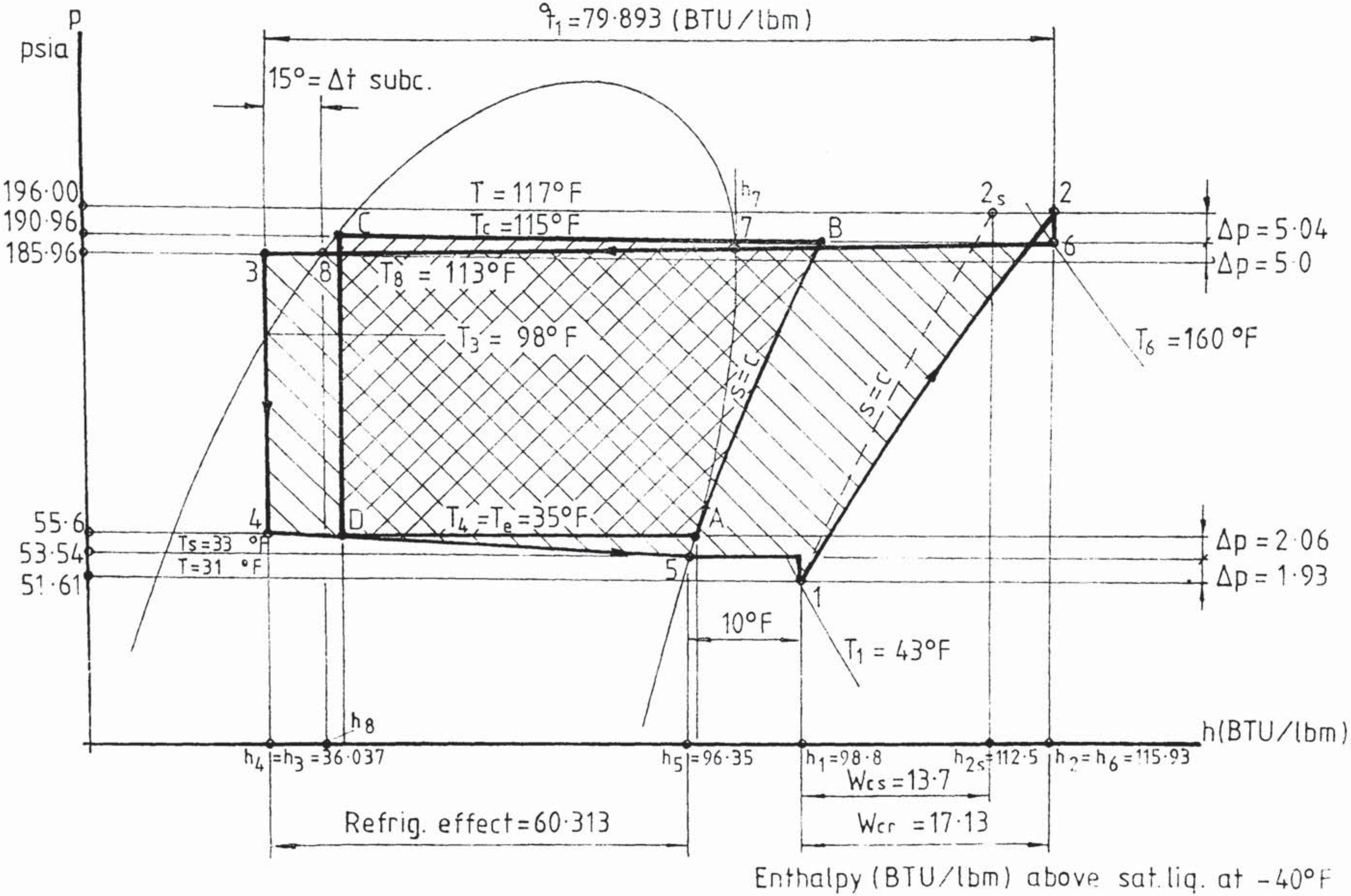


Fig 12 Refrigerant 500 (R500) real cycle

for R12, R22 and R500 respectively, working within a real cycle, which has been used in order to prepare table II-6.

Values of subcooling, superheating and pressure drops were assumed according to the practical heat pump operating systems.

The following assumptions have been made for developing tables II-5 and II-6:

1. The compressor efficiency was taken as:

$$\eta_{comp} = \eta_{iad} \times \eta_{m} \times \eta_{motor} \times \eta_{D}$$

where:

η_{iad} = Adiabatic efficiency = 0.85

η_{m} = Mechanical efficiency = 0.94

η_{D} = Efficiency of the transmission = 0.99

η_{motor} = Efficiency of the electrical motor = 0.87

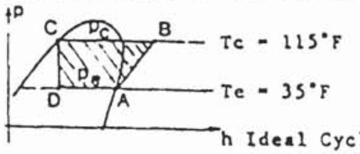
2. The term "Ton of refrigeration" is defined (8, 10, 15) as: The removal in the evaporator of 3517 Watt (200 BTU/min), the concept being derived from the fact that this is the rate at which heat must be removed from liquid water at 0°C to produce one ton of ice at 0°C in 24 hours. Since an American ton is 2000 lb and the Latent heat of fusion of water amounts to 144 BTU/1b one can write:

$$1 \text{ Ton R} = \frac{2000 \times 144}{24} = 12000 \frac{BTU}{h} = 200 \frac{BTU}{min}$$

$$1 \text{ Ton R} = \frac{12000 \ BTU}{h} \times \frac{1.05 \ kJ}{BTU} = 12600 \frac{kJ}{h}$$

$$1 \text{ Ton R} = 12600 \frac{kJ}{h} = 210 \frac{kJ}{min} = 3.52 \ kW$$

Table II-5. Comparison of Refrigerants.- Ideal Cycle



		REFRIGERANTS									
		R11	R12	R21	R22	R113	R114	R500	R502	R504	R503
Normal Boiling Point (1 bar)	°F	74.84	-21.62	48.00	-41.36	117.6	38.40	-28.30	-49.70	-71.60	-127.6
Evaporating Temperature = Te	°F	35	35	35	35	35	35	35	35	35	35
Condensing Temperature = Tc	°F	115	115	115	115	115	115	115	115	115	115
Evaporating Pressure = Pe	psia	6.26	47.26	11.20	76.17	2.32	13.55	55.55	87.52	145.40	
Condensing Pressure = Pc	psia	30.50	161.40	50.51	257.40	12.94	58.11	190.50	279.50	459.00	
Compression Ratio = r = Pc/Pe		4.87	3.42	4.51	3.38	5.56	4.29	3.44	3.19	3.15	
Enthalpy h_C = Enthalpy h_D	BTU/lbm	31.46	34.85	38.12	46.09	32.35	36.10	41.54	42.19	54.30	
Enthalpy point B = h_B	BTU/lbm	108.20	91.74	139.17	121.61	94.12	84.95	108.72	89.86	114.04	
Enthalpy point A = h_A	BTU/lbm	96.51	82.16	123.58	108.62	83.92	76.43	96.96	81.36	103.13	
Refrigerating effect = h_A-h_D	q_2=BTU/lbm	65.05	47.31	85.46	62.52	51.57	40.33	55.42	39.17	48.83	
Heating effect = h_B-h_C	q_1=BTU/lbm	76.73	56.89	101.05	75.52	61.77	48.85	67.18	47.67	59.74	
Ideal Compressor = h_B-h_A	Wcs=BTU/lbm	11.68	9.58	15.59	12.99	10.20	8.52	11.76	8.50	10.91	
Refrigerant circuled per Ton ref.	$\dot{m}=\frac{200}{q_2}=\frac{lbm}{min.Ton}$	3.07	4.22	2.34	3.19	3.88	4.95	3.61	5.10	4.09	
HP/Ton = $(200/q_2)$ (Wc/42.42)	HP/Ton	0.85	0.95	0.86	0.98	0.93	0.99	1.00	1.02	1.05	
C.O.P. (C)		5.57	4.92	5.48	4.81	5.05	4.74	4.71	4.60	4.48	
C.O.P. (H)		6.57	5.92	6.48	5.81	6.05	5.74	5.71	5.60	5.48	
Specific volume at point A	Cu.ft/lbm	6.05	0.86	4.59	0.72	12.09	2.28	0.86	0.47	0.38	
Theoretical piston displacement per Ton	c.f.m./Ton	18.58	3.63	10.74	2.28	46.90	11.28	3.12	2.39	1.54	
Refrigerant circulated per kW of heating	$\dot{m}=\frac{3413}{q_1}=\frac{lbm}{kW.h}$	44.48	59.99	33.77	45.19	55.25	69.86	50.80	71.59	57.11	
Theoretical piston displacement per kW of heating	c.f.m./kW	4.48	0.86	2.58	0.54	11.13	2.65	0.72	0.56	0.35	

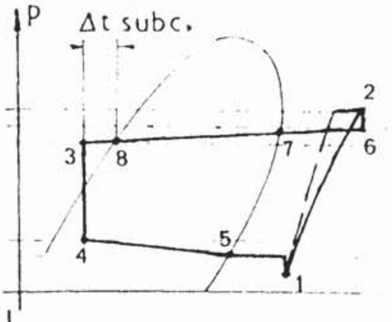
Critical Temperature at 67°F.
For Refrigeration Applications only.

Table II-6. Comparison of Refrigerants

Real Cycle

Te = 35°F ; $\eta_{comp} = \eta_{iad} \times \eta_m \times \eta_D \times \eta_{mot} = 0.85 \times 0.94 \times 0.99 \times 0.87 = 0.69$

Tc = 115°F



Δ P cond ≈ 5 psia; Δ P evap ≈ 2 psia; Δ t subcool = 15°F

Δ P disch ≡ 2°F; Δ P suct ≡ 2°F; Δ t overheat = 10°F

ΔP disch.
ΔP cond.

ΔP evap.
ΔP suct.

		REFRIGERANTS		
		R12	R22	R500
Boiling Point (14.7 psia)	°F	-21.62	-41.36	-28.30
Discharge Pressure = P_2	psia	165.76	264.20	196.00
Condensing Pressure = Pc = P_6	psia	161.40	257.42	190.96
Pressure at Condenser Outlet = P_3	psia	155.20	252.00	185.90
Evaporating Pressure = Pe = P_4	psia	47.26	76.17	55.52
Pressure at evaporator outlet = P_5	psia	44.70	74.00	53.50
Suction pressure = P_1	psia	43.15	71.00	51.61
r compr. = $P_2 \div P_1$		3.84	3.72	3.80
Compressor discharge temp. = T_6	°F	164.00	166.00	160.00
Condensing temperature = Tc	°F	115	115	115
T_7	°F	114.5	114.7	114.7
T_8	°F	112.0	113.5	113.0
Temperature at condenser outlet T_3	°F	97.0	98.5	98.0
Evaporating temp. = Te = T_4	°F	35	35	35
Temp. at evaporator outlet = T_5	°F	33.0	33.5	33.0
Comp. suction temperature = T_1	°F	43.0	43.5	43.0
$h_3 = h_4$	BTU/lb	30.38	38.80	36.03
h_5	BTU/lb	80.62	107.57	96.35
h_1	BTU/lb	84.60	109.90	98.30
h_{2S}	BTU/lb	96.00	124.50	112.50
$h_2 = h_6$	BTU/lb	98.85	128.50	115.93

Table II-6. Continued

Comparative characteristics :		REFRIGERANTS R12	R22	R500
Refrigerating effect = $q_2 = h_5 - h_4$	BTU/1b	50.24	68.77	60.31
Heating effect = $q_1 = h_6 - h_3$	BTU/1b	68.47	99.70	79.83
Ideal Compressor = $W_{cs} = h_{2S} - h_1$	BTU/1b	11.40	14.60	13.70
Real Compressor = $W_{cr} = W_{cs} / \eta_{ad} \cdot \eta_m$	BTU/1b	14.25	18.25	17.13
Shaft Motor Work = $W_{mot} = W_{cr} / \eta_D \times \eta_{mot}$	BTU/1b	16.54	21.19	19.89
Fan Work = W fan = 15% W motor	BTU/1b	2.48	3.18	2.98
Total Work = W motor + W fan	BTU/1b	19.02	24.37	22.87
COP carnot = COP $(H)_R$		7.19	7.19	7.19
COP (C)		2.64	2.72	2.64
COP (H)		3.60	3.62	3.49
Heating efficiency $\eta_{C_H} = COP(H)/[COP(H)]_R$	%	50%	51%	48.6%
Refrigerant circulated per Ton $\dot{m} = \frac{200}{q_2}$	$\frac{1bm}{min.Ton}$	3.96	2.91	3.32
Theoretical piston displacement Ton = $\dot{m}.v_1$	c.f.m./Ton	3.88	2.33	3.25
Total power imput = $(200/q_2) \cdot \frac{(W\ motor)}{42.42}$	B.H.P./Ton	1.54	1.45	1.55

This unit has been readily adopted by air conditioning engineers because of its low value in terms of units, and also that at normal air conditioning levels it requires about one horsepower of compressor work. Modern refrigeration uses the kW as the unit, of the International Units system (S.I.), to express the capacity of cooling of a machine, rather than "Ton of Refrigeration".

3. The values for the efficiency term, "Horsepower per Ton", frequently used to rate cycles, were calculated from the COP as :

$$\frac{HP}{Ton} = \dot{m} \times Wc = \frac{K}{q_2} Wc = K \cdot \frac{W}{q_2} = \frac{K}{COP\ (C)} \qquad (II-17)$$

K = Constant according to the used units.

W = Work on the compressor for the ideal case or work on the motor for the real case

W/q_2 = reciprocal of the COP (C).

4. Other terms used for the comparison of refrigerants were:

1) The "rate $\dot{m}$ of refrigerant circulation necessary per ton of refrigeration (15) and

2) The theoretical piston displacement for reciprocating compressors per ton of refrigeration per minute:

$$\dot{m} = \frac{200}{q_2} = \frac{lbm}{min.\ Ton} \quad \text{with} \quad q_2 = \frac{BTU}{lbm} \qquad (II-18)$$

or

$$\dot{m} = \frac{210}{q_2} = \frac{kg}{min.\ Ton} \quad \text{with} \quad q_2 = \frac{kJ}{kg}$$

The theoretical piston displacement was calculated as:

$$\frac{200}{q_2} \times v_1 = \dot{m} \cdot v_1 = \frac{c.f.m.}{Ton} \qquad (II-19)$$

Were v_1 = is the saturated vapour specific volume at the compressor entrance point.

The ratio of compression, taken in conjunction with the refrigerant volume factors will have considerable bearing on the choice of the compressor type, and the horsepower figures will reflect the operating costs for the driving energy.

II-5-2. Comparative characteristics(10, 12)

Knowledge of the thermodynamic properties of the refrigerant and their effect on the performance are important for the investigation of heat pump behaviour:

1) For the refrigerants and cycles considered in this application, the normal boiling point affects the compression ratio as shown in Figure 13; the relation is almost linear. The compression ratio is seen to increase with the increase of the normal boiling point.

2) The COP (H) is also affected by the normal boiling point, the relation again being linear as shown in Figure 14. The equations of the straight lines appear in the figure and were obtained in each case from two points. In a given class, one can see that increasing the number of carbon atoms lowers the COP and increasing the number of hydrogen atoms increases the COP. In other words, more carbon atoms per molecule increase the horsepower; more hydrogen atoms decrease it, according to the equation II-17.

Fig 14 also permits one to estimate the COP (H) for other refrigerants,

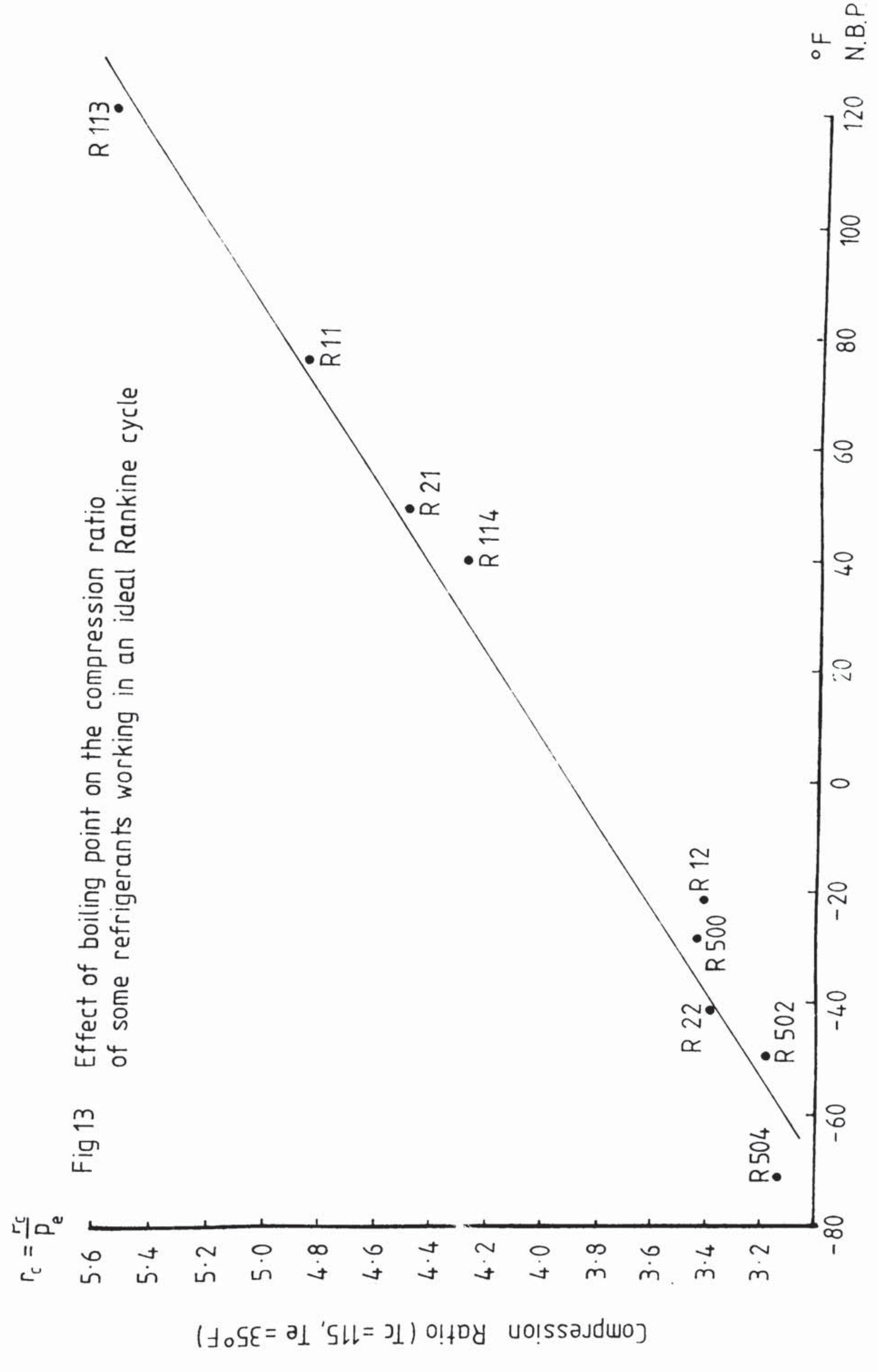


Fig 13 Effect of boiling point on the compression ratio of some refrigerants working in an ideal Rankine cycle

working in a similar cycle, by plotting their N.B.P. over the line of its particular refrigerant class. However, scrutiny of the data in table II-5 and II-6, shows that the coefficient of performance for the Carnot cycle, equal to 7.19 as a heating machine, is approached , within the ideal cycles , with any one of several halocarbons. All of them offer a coefficient of performance in excess of 5. Thus the selection of any common halocarbon refrigerant will require ideally about the same horsepower for operation of the plant, according to the equation II-17. The table demonstrates that power savings, by selection of one refrigerant rather than another, are of only minor significance. Refrigerant choice must rather be based on the consideration of other items such as the operating pressures, fluid volumes to be handled or heating capacity.

3) Suction pressure is a minimum of 2.32 psia (0.16 bar) with R113, and head pressure is a maximum of 459 psia (30.6 bar) with R504. The R11, R21 and R113 systems, as is shown, operate at a negative pressure on the evaporator side with all the problems of maintenance of vacuum. On the condensing or discharge side, R22, R502 and R504 require designing for the higher pressures.

4) The refrigerant mass entering the compressor, per kW of heating produced, is a maximum with R502, 71.59 lbm/h , while R11, R12 R22, R113, R114, R500 and R504, require an amount of 44.5 to 69.9 lbm/h , and the minimum is offered by R21 at only 33.8 lbm/h.

(5) The theoretical piston displacement in the compressor, per kW of heating produced on the condenser, is smallest with R504,

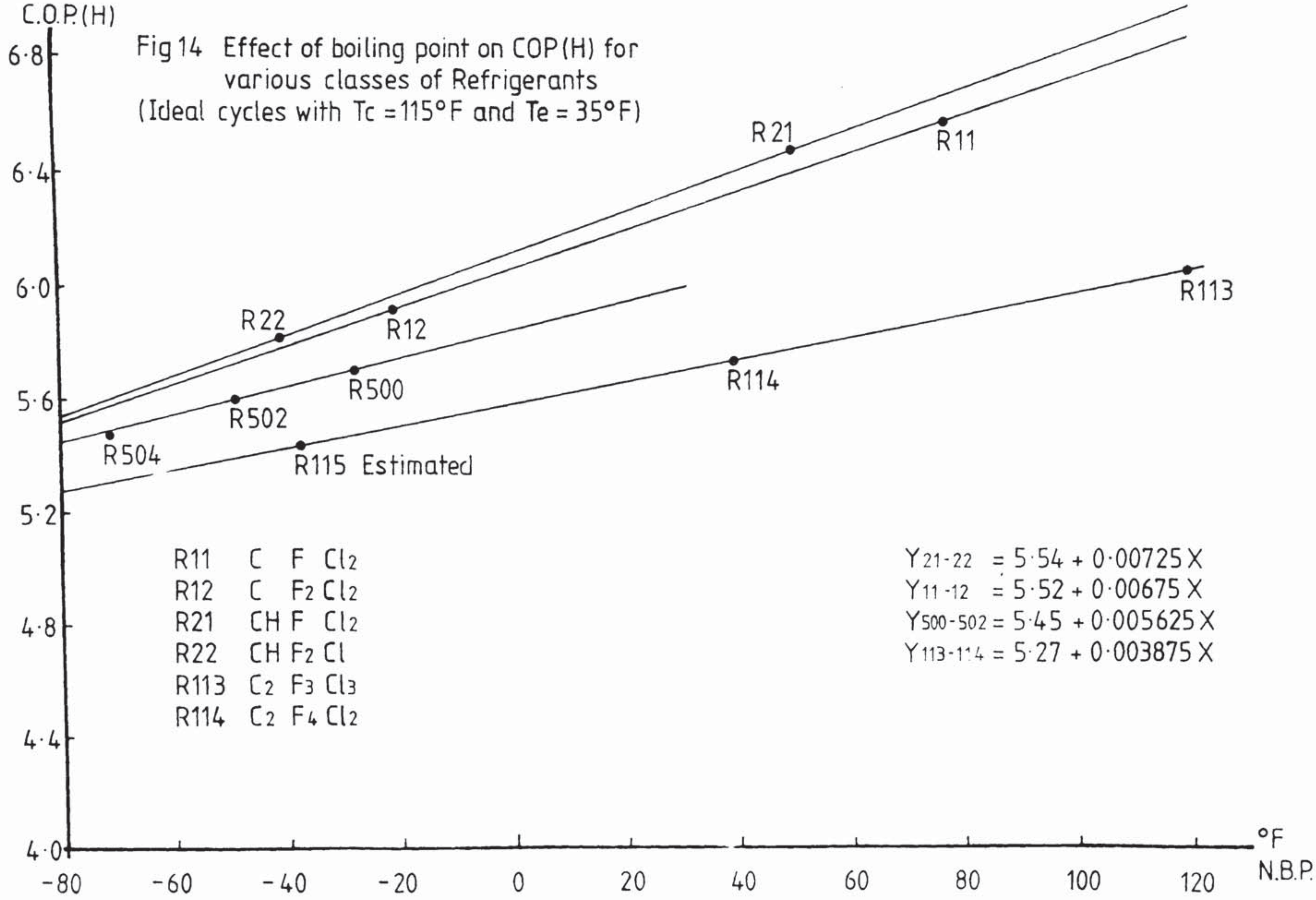


Fig 14 Effect of boiling point on COP(H) for various classes of Refrigerants (Ideal cycles with Tc = 115°F and Te = 35°F)

and favourably competitive with R12, R22, R500 and R502, which means smaller compressors and less limitation on selection design

(6) The compression ratio encountered will also have an influence on the type and on the desing of compressor. This ratio ranges from 3.15 with R504 to 5.56 with R113, and it is a determining factor on the volumetric efficiency of a positive-displacement unit and on the number of stages required in a centrifugal or axial-flow unit.

(7) The heating capacity is another very important point for refrigerant comparison in heating applications. Different refrigerants, where used in identical machines with appropriate motors, will yield a different heating capacity in each case. A 5 kW unit of heating designed for refrigerant 12 will have the following capacities with different refrigerants according to the data on table II-5:

R11:

$$\text{Heating capacity} = \frac{0.86 \text{ c.f.m.}}{\text{kW}} \times 5 \text{ kW} \times \frac{1 \quad \text{kW}}{4.48 \text{ c.f.m.}} = 0.95 \text{ kW}$$

R12:

Heating capacity = 5 kW

R21:

$$\text{Heating capacity} = \frac{0.86 \text{ c.f.m.}}{\text{kW}} \times 5 \text{ kW} \times \frac{1 \quad \text{kW}}{2.58 \text{ c.f.m.}} = 1.67 \text{ kW}$$

With similar calculations one can obtain for the other refrigerants:

R22 : H.C. = 7.96 kW

R113 : H.C. = 0.38 kW

R114 : H.C. = 1.62 kW

R500 : H.C. = 5.97 kW

R502 : H.C. = 7.67 kW

R504 : H.C. = 11.94 kW

Even though the COP (H) of R504 is 8% lower than the COP (H) of R12, this refrigerant could be used in the future for heating purposes, because of its over-whelming relative heating capacity, 2.4 times higher than R12, with a competitive lower compression ratio and a discharge temperature of about, 140°F (60°C) which is sufficient for both water and space heating in domestic applications. Refrigerant 504 is not commercially available at the moment.

Table II-6 shows that R12, R22 and R500, which are commercially available, have very similar characteristics when working in a real cycle, and perhaps R22 is better than the other two because of its higher relative heating capacity.

Research on refrigerants has hitherto been done mainly for refrigeration and cooling purposes, but with the new possibilities of the heat pumps for space heating, fundamental research work on refrigerants, to be used primarily in heating applications is essential as can be seen from this short survey.

C H A P T E R III

H E A T P U M P T Y P E S - H E A T S T O R A G E

III-1. Heat Pumps Classification

Heat Pumps may be classified (16) according to (a) type of heat source and sink, (b) heating and cooling distribution fluid, (c) type of thermodynamic cycle, and (d) size and configuration; but the more common classification is according to the heat source and sink combinations, so that heat pumps are often referred to as air to air, air to water, water to water, etc. Figure 15 illustrates the various possible environments in which the evaporator and condenser can be placed.

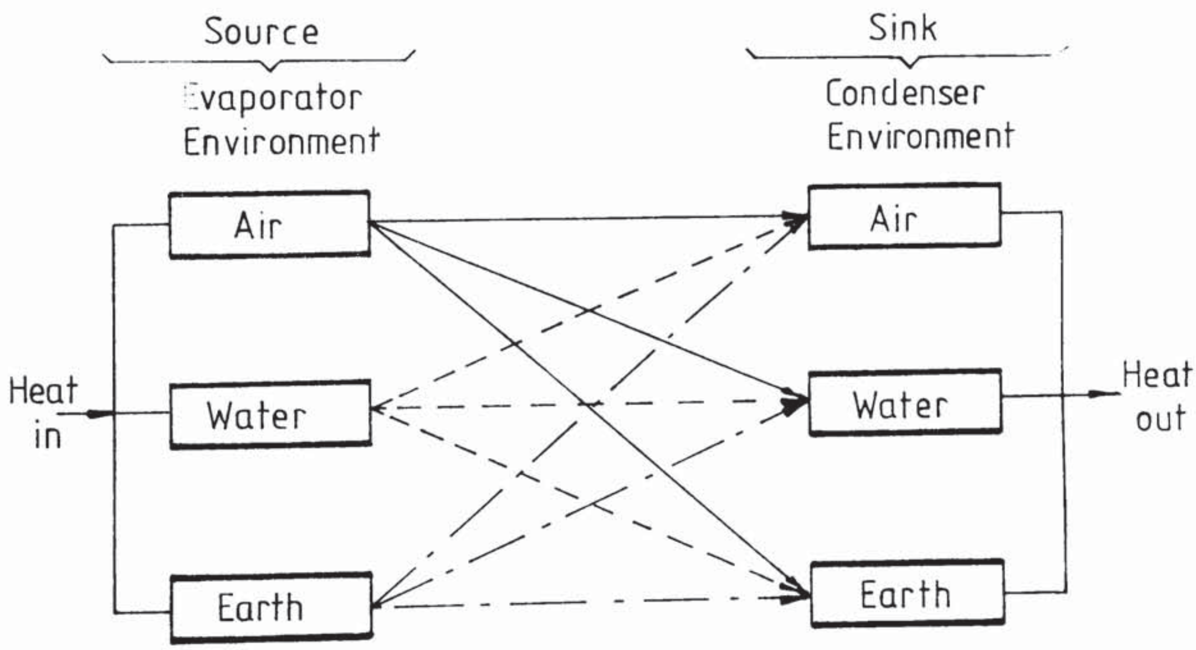


Fig 15 Environmental possibilities for Heat Pump evaporators and condensers

III-2. Low-grade Heat Sources

The low-grade heat for domestic heating can be absorbed from different sources and it is possible to get large variations in source temperature (T_2) depending on climate conditions. An ideal heat source is one which is abundant and inexpensive with an average temperature of 5°C (40°F) to 20°C (68°F) all the year round. Outdoor air, ground and water from wells, lakes and rivers may meet the requirement. Discharged water from manufacturing processes or any other liquid where the temperature is too low for direct utilisation may qualify providing it is chemically suited and does not require special metals for heat exchangers construction or extensive treatment.

III-2-1. Heat from the water

Water has a high specific heat and is a good medium from which to extract heat. Well water is particularly attractive because of its relatively high and nearly constant temperature. Unfortunately there is always the uncertainty of finding water at any given location, and the cost of drilling and the maintenance involved in the use of water, further detract from its use in many places. Water often will either cause corrosion in heat exchangers or it may induce scale formation. Also, the disposal of the water after its use may constitute a serious practical problem. Due to these inconveniences, one of the other heat sources will be more practical.

III-2-2. Heat from the soil

Ground offers possibilities as a heat source for heat pumps systems. Energy from the soil may be recovered by burying a coil evaporator and using the upgraded energy rejected from the condenser for space or water heating. Localised extraction of energy from the soil will cause the temperature of the soil to drop but an equilibrium situation will be reached when heat extraction is balanced by heat conducted to the cooled volume from the surrounding soil. Thus the Earth's surface is acting as a solar collector for the evaporator of the heat pump.

At depths of 1 metre or so there is little daily or seasonal variation in temperature so that T_2 could be constant and with a fixed condenser temperature a constant COP(H) can be achieved.

Published data on soil temperature at different depths are available. (17)

Figure 16 shows the variation at different depths, obtained by Lovering and Goode (18), and clearly illustrates that daily air temperature variations do not penetrate beyond one metre.

The rate at which energy from the sun reaches the Earth is much greater (Average of 1.41 kW/m^2) than the flow of geothermal energy from the Earth (Average heat flow estimated as 0.068 W/m^2(17)), with the greater proportion of the sun's energy re-radiated back to space. The surface characteristics of the ground are an important factor in the proportion of solar energy absorbed. In concequence, the main source of energy in the soil is solar, supplied in daily and annual cycles. The effect of the annual cycle penetrates some 20 to 30 metres, depending upon the thermal characteristics of the soils, so, it is usual to assume that the temperature remains constant at depths greater than 20 to 30 metres. However, Earth as a heat source has not been extensively used. This may be attributed

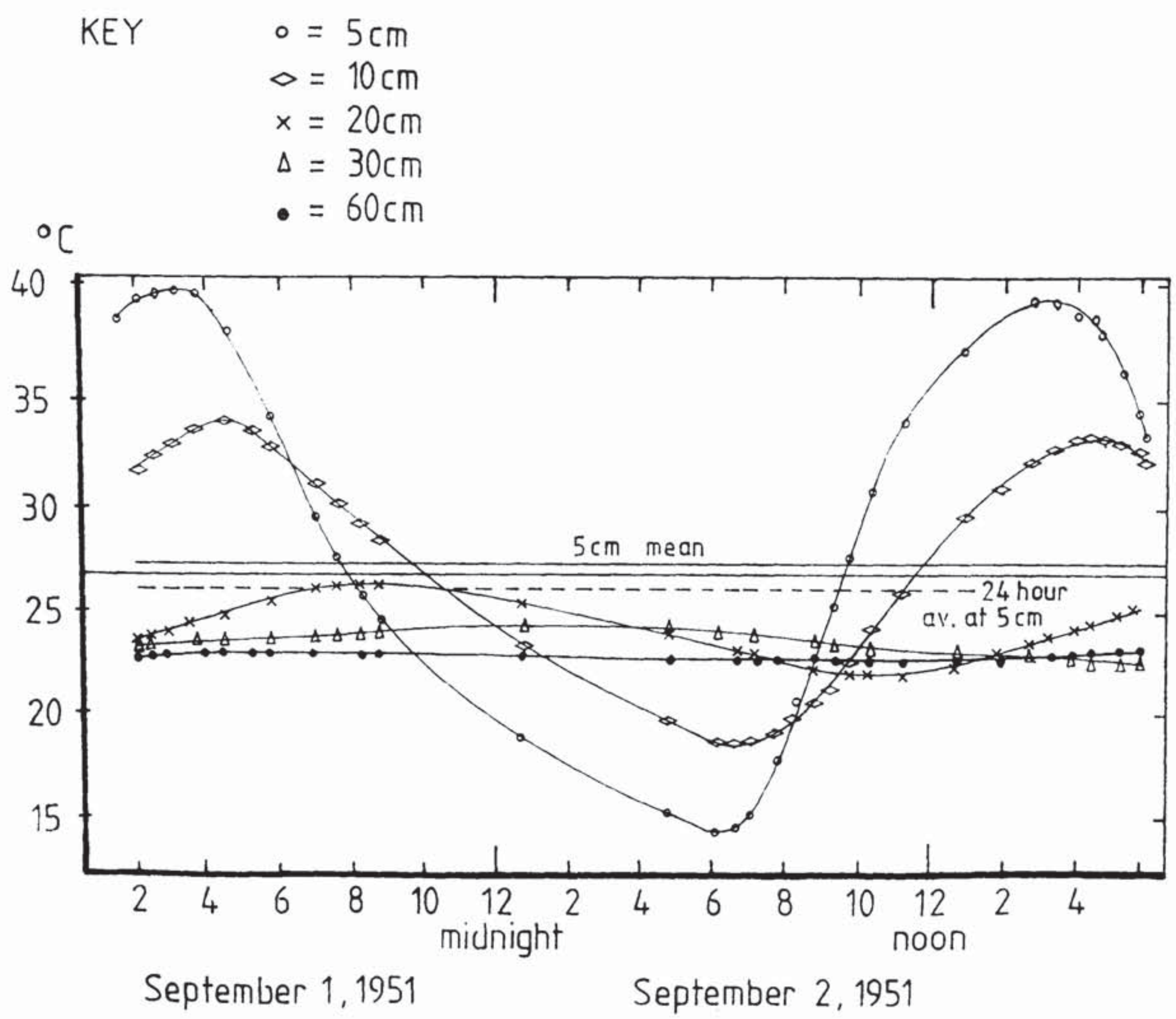


Fig 16 Curves of temperatures measured in a metre hole in dry soil during a 28 hour period (Ref. 18)

to high installation expense, ground area requirements and the difficulty and uncertainty of predicting performance.

Compositions of soil vary widely from wet clay to sandy soil, and have a predominant effect on thermal properties and attendant overall performance. The parametres which most strongly influence the thermal conduction of heat from the soil to the evaporator coil are: Thermal conductivity (k); density (ρ); specific heat (c) thermal diffusivity ($\alpha = k/\rho c$) and moisture content. When moisture content changes, due to either heavy rainfall or moisture migration, so do all the otherparameters.

Many individual studies involving theoretical as well as practical considerations have been conducted [16] to determine earth heat transfer with respect to heat pump possibilities. The heat extraction rate, with horizontal coils, can lie between 25 and 50 W/m of tube, depended on conditions [3,17].

The value of T_2 found in practice, during winter periods, is in the region of 0°C since ice forms on the coil after a short period. With such a system it is customary to circulate antifreeze rather than "Freon" refrigerant through the ground coil. The coil itself is not normally used as the evaporator because of circulatory problems of the oil which is carried in the refrigerant to lubricate the compressor. An additional antifreeze-refrigerant heat exchanger is included in the evaporator unit. This increases pump power and reduces the overall COP(H).

The problem of excessive ground surface area can be largely overcome by using vertical ground coils of U type. [17] Attention is also being given to the use of plastic rather than copper pipes. Several universities and manufacturers are investigating the ground as a heat source and plan to make actual installations (10) to obtain better knowledge on the transfer coefficients, on the cost and performance of the systems.

III-2-3. Heat from the air

Air is a plentiful supply but unfortunately it suffers from the disadvantages that it has a low specific heat and is subject to large variations in temperature and humidity in the United Kingdom.

When selecting or designing an air-source heat pump, two factors in particular must be taken into consideration. 1. The variation in temperature experienced in a given locality, and 2. The formation of frost. It is usually the case that greater heating demands are made on heating systems at the time when the ambient temperatures fall so that the COP(H) of the heat pump will be lower at low air temperature (for the same condensing temperature) and so the heat output of the unit will be less than it would be at other times of the year.

In high humidity conditions it is possible to get ice forming on the evaporator at ambient temperatures in the region of 4°C. This reduces the rate of heat absorption and the performance. A defrosting process has to be utilised and this itself results in a loss of overall performance. The most common methods of defrosting are:

1. Reversing the heat pump to heat the evaporator.
2. Direct electrical heating of the evaporator.
3. Re-cycling some hot gas from the Condenser.
4. Stoping operation to allow defrost in warmer ambient air.

Up to 5% of total energy could possibly be used in defrosting cycles depending on the frequency and duration. The number of defrosting operations will be influenced by the climate, the air-coil design and the hours of operation. Experience has shown that less defrosting is generally required below -5°C if $\varphi < 60\%$ (φ = relative humidity, which can be confirmed by psychrometric analysis; however, it should be noted that, under very humid conditions when small suspended water droplets may be present in the air, the rate of frost deposit may be about three times as great as would be predicted from psychrometric theory. Under such conditions, a heat pump may require defrosting after as little as 20 minutes of operation.

With air as the heat source, evaporators tend to be large and it is necessary to draw large volumes of air past the evaporator. Energy is used to drive fans which results in a reduction of overall COP(H). Using ambient air as a source also leads to difficulties in designing for the size and loading of a compressor which must operate over a wide temperature range in the evaporator. Undersizing causes periodic overloading, and oversizing, whilst not reducing the life, does reduce the overall COP(H). Thus, the unit has to be sized to meet the total demand at some design temperature, and at all temperatures greater than this the normal part load problem of any heating system is compounded by the increased capability of the heat pump to produce heat. The effect is compounded by the way in which the components of any practical system react to changing temperatures. For example the refrigerant vapour density at the inlet of the compressor decreases as the evaporating temperature falls so that the same volumetric displacement represents a lower mass flow and therefore a lower power to the compressor. This combined with the reduced COP results in even less heat being rejected than the COP variation would suggest.

For example (19), a unit which is designed to produce 9 kW of heat at an air source temperature of 0°C. may well produce 12 kW at an air temperature of 10°C. At the same time, the heat demand of the house will have dropped from 9 kW to 4.5 kW (approximately) so that, instead of operating at a load factor of 50% as would be the case for a conventional heating system, this heat pump is operating at a load factor of only 37% (i.e. 4.5/12). This introduces the importance of the integrated heat pump systems and thermal storage to modulate the capacity of the units to better match the conditions under which they are operating.

In an air source heat pump, improvements in performance can be obtained by pre-heating the air before passing it over the evaporator using for example, an air solar panel, or by mixing the external air stream with warm ventilated air leaving the building. These approaches also reduce the probability of frosting.

III-2-4. Heat from solar energy

The use of solar energy (16) as a heat source, either on a primary basis or in combination with other sources, is attracting increasing interest. The principal advantage of employing solar radiation as a heat pump heat source is that, when available, it provides heat at a higher temperature level than other sources, thus resulting in an increase in coefficient of performance.

III-3. Basic Designs and Operating Arrangement

The four basic heat pump design for space heating and cooling are: 1) air to air; 2) water to water; 3) water to air; and 4) air to water. Flow diagrams, together with a brief description, are given for these typical arrangements.

Reversal operation: Where air conditioning is required a heat pump can be operated in a reversal situation where the evaporator and condenser exchange roles. This requires that both heat exchangers (evaporator and condenser) are designed and sized to serve two functions according to the season and usually requires a compromise to be made. Often, equipment which is designed to meet either the cooling or heating load is either oversized or undersized for the other. The trend in the design of reverse operation equipment in the U.S.A. is to match the cooling load. Even in the southern regions of the U.S.A. this usually produces a need for supplementary heating in extreme weather. Such equipment used further north could lead to disadvantageous operation because of unbalanced loads. In some cases the cooling load could be of the order of 25% of the heating load. Equipment used for heating and cooling has a high use factor and needs to be reliable.

III-3-1. Air to Air

This heat pump is the most common type and has been widely used for residential and commercial applications. During the heating cycle air is used as a source of heat, and air is used to remove heat from the condenser. During the cooling cycle air is used to cool the control space and air is used to reject the heat to the outside.

The path of the refrigerant is reversed by means of several 2-way valves as shown in Figure 17. An alternate design can maintain a refrigerant circuit and reverses air flow.

III-3-2. Water to Water

In this type of heat pump water is used as a source of heat, and water is used to transfer heat from the condenser and chiller. Heating-cooling changeover may be accomplished in the refrigerant circuit, but, in manv cases, it is more convenient to perform the switching in the water cicuit as shown in Figure 18.

III-3-3. Water to Air

Water is also used as a heat source in this case and air flow is used to transfer the heat from the condenser to the conditioned space. Figure 19 shows a water to air design.

III-3-4. Air to Water

This type of heat pump is commonly used in large buildings where zone control is necessary, and is also sometimes employed for the production of hot or cold water in industrial applications. An air to water arrangement with air as a source of heat and water to transfer heat from condenser can also be seen in Figure 19. Solid lines show the path of refrigerant and water when performing the function indicated; dotted lines the unused paths when performing as an air to water heat pump.

In addition to the types of heat pump already mentioned,

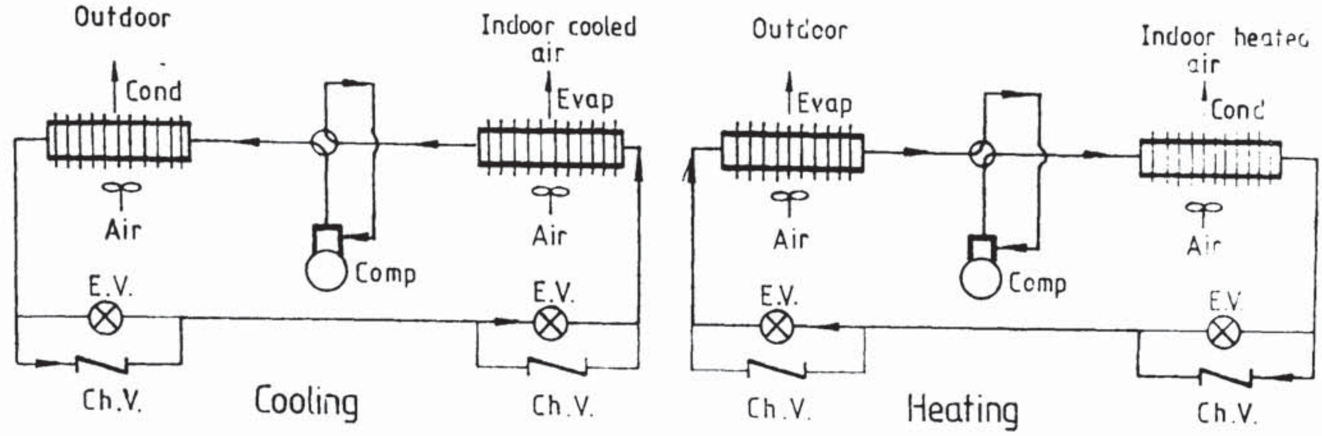


Fig 17 Air to Air Heat Pump

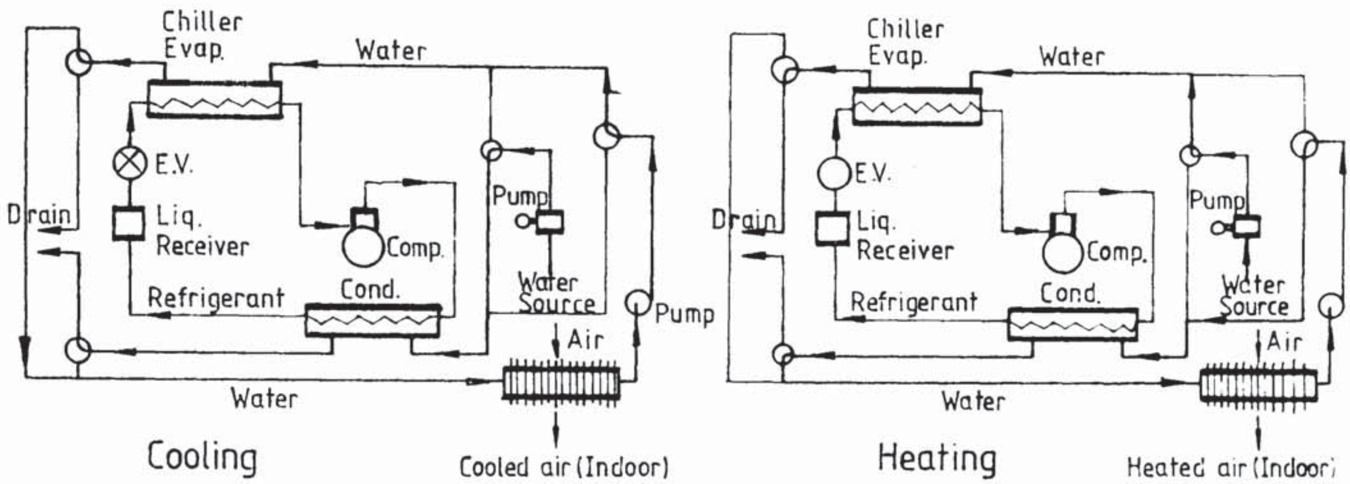


Fig 18 Water to Water Heat Pump

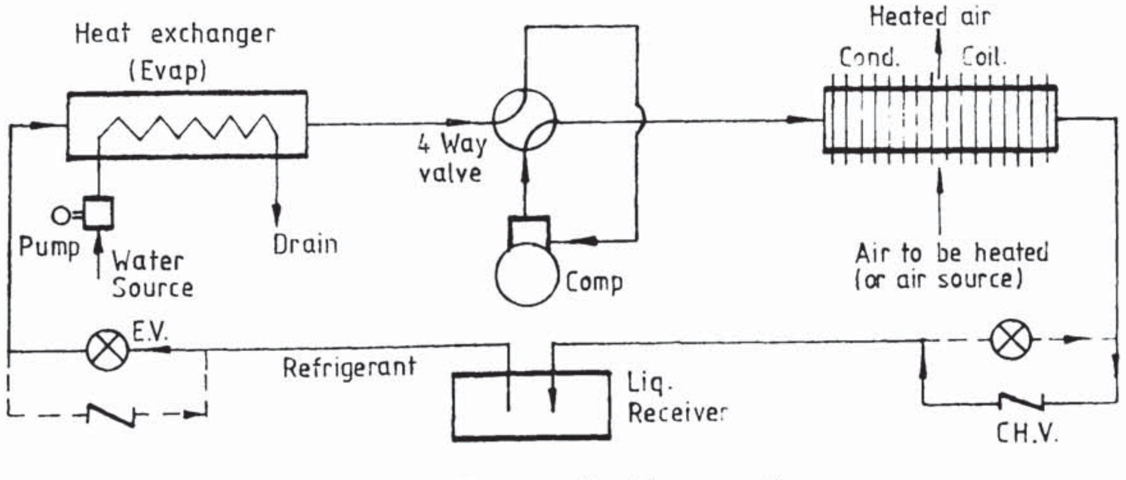


Fig 19 Water to Air Heat Pump - Heating mode
By reversing the refrigerant circuit, with the 4way valve, the system performs as an Air to Water Heat Pump

there are also earth source heat pumps, but these are less commonly used in the U.K., although there is increasing use in Denmark for domestic systems.

Earth to Water and Earth to Air heat pumps are similar. In the former, a refrigerant-water heat exchanger replaces the finned coil on the indoor side of the earth to air system. At the outdoor side both systems may employ direct expansion of the refrigerant in an embedded coil, or both may be of the indirect type, and in this case an antifreeze solution is pumped through an additional circuit consisting of the chiller and pipe coil embedded in the earth.

III-4. Integrated Heat Pump and Thermal Store

The use of thermal storage in a heat pump system can affect its performance characteristics. All materials possess the property of thermal storage to a greater or lesser degree. In the case of a building, the structural materials are almost always in the process of either absorbing heat from or delivering heat to the interior space. Storage tends to reduce the peak equipment requirements. In this sense every heating and cooling system can be said to involve heat storage in some degree.

In the case of the heat pump, a provision for heat storage can serve not only to reduce the size of the heat pump necessary for a given load, but also to provide a more desirable electrical load by shifting part of the load to the time of the day when the cost of power is least. Heat storage in a heat pump system may be utilized on the high side, when heat is available at a temperature suitable for direct heating, or on the low side as an intermittent heat source at temperatures lower than those required for the heated space.

In the case of solar energy applications, the necessary storage required to cope with variations in solar radiation usually adds considerably to the overall system costs. To date there has been little research effort devoted to thermal storage (particularly long term storage) but this is now becoming more important. The obvious application of solar energy and thermal storage, would be for cooling when the maximum radiation levels coincide with maximum cooling loads.

III-4-1. Types of Thermal Store

If one excludes stores involving chemical reactions there are two types of thermal store:

1. Latent heat or phase change stores, in which most of the heat is stored at a constant temperature.

2. Sensible heat stores, in which heat is stored over a range of temperatures.

III-4-2. Comparative volumes required for latent and sensible heat stores in solar energy applications.

Most of the sensible heat stores which have been used in solar energy applications use either rock or water raised in temperature by up to 30°C above the required comfort temperature. There is a range of organic and inorganic substances which have been considered for phase change storage. Table III-1 gives an indication of the relative

Table III-1. * Relative store volumes required for water, rocks and latent heat stores to store 1GJ (278 kWh) of energy (sensible heat $\equiv$ 30°C rise)

Store Material	PROPERTY				
	Density $kg.m^{-3}$	Specific heat $kJ.kg^{-1}\ °C^{-1}$	Latent heat $kJ.kg^{-1}$	Volume m^3	Ratio
Phase change (Average Values)	1600	2.10	2.32×10^2	2.7	0.34
Ice	917	2.10	3.32×10^2	3.3	0.41
Water	1000	4.18	——	8.0	1.0
Rock	2250	0.87	——	17.0	2.1

store volumes required for water, rocks and latent heat stores to store 1GJ (278 kWh) of energy (sensible heat ≡ 30°C rise).

As can be seen from Table III-1 a rock store volume would be more than double the size of a water store for the same energy stored as sensible heat. In fact in practice, the rock store volumes shown would need to be increased by up to 25% to allow for fluid passage [20]. Rock sizes vary in diameter from 5 to 15 cm. A phase change store could be about one-third of the volume of water store and most of the energy would be stored at a constant temperature. Store capacities depend on specific applications and range from energy stores for domestic service hot water for one day use and quick availability, through intermediate stores with sufficient energy for space heating for a period of a few weeks, to long term stores which provide energy for a heating season. Seasonal storage is probably one of the most important problems to be tackled in conjunction with the use of solar energy for the heating and cooling of buildings.

III-4-3. Storage Material Characteristics

There are many factors that must be taken into consideration in the selection of the storage material to be used for a thermal storage unit. Some desirable characteristics would include the following:

1. High specific heat
2. High thermal diffusivity
3. High density
4. Reversible heating and cooling
5. Chemical and geometrical stability
6. Non-combustible, non-corrosive, and non-toxic

7. Low vapour pressure to reduce the cost of containment
8. Low cost material and storage unit fabrication
9. Sufficient mechanical strength to be able to support compression load resulting from the stacking of the storage core.

In addition to the above list, one must give serious consideration to the operating temperature range of the storage material. If the temperature range is very high, many materials may be eliminated from consideration.

III-4-4. Phase Change Storage Units

Storage units that utilise the latent heat of the storage material are receiving considerable attention particularly for solar heating and air conditioning applications. The major advantage of these units is smaller size and lower weight per unit of storage capacity as was indicated in table III-1. The phase change utilised involves the liquid and solid phases. For practical applications there are certain criteria which need to be met by materials for phase change storage.

1. A melting point slightly above the heating temperature or slightly below the cooling temperature. This permits the desired heat transfer to take place.

2. Large heat of fusion. The larger the heat of fusion, the less material is required to store a given amount of energy.

3. Congruent melting point. The material should melt completely so that the liquid and solid phases are identical in composition. Otherwise, the difference in densities between solid and liquid

will cause segregation, which causes changes in the chemical composition of the material.

4. High stability

5. Compatibility with the container. The material should not interact with the container.

6. Non toxicity and preferably non-flammability.

7. Cost appropriate to application.

In practice there are problems due to phase separation resulting from cycling, poor heat transfer characterisation and corrosive properties. There are many potential candidates for use as storage media including inorganic salts and organic materials such as waxes, covering a wide range of melting temperatures, but serious consideration is being given to salt hydrates in the low temperature storage range, 30 to 60°C; sodium hydroxide for storage in the 315°C range; and to eutectic mixtures: Lithium hydroxide, Lithium hydride, and Lithium flouride for high temperature storage systems. (1000 to 1700°C (21)). In general they have low thermal conductivities and high heats of fusion ranging between 600 and 900 kJ/kg at temperature ranging between 450 and 850°C.

Other inorganic eutectic mixtures having melting points in a range most suitable for solar heat pump storage as well as having high heats of fusion have been investigated. Some are presented in table III-2. Table III-2 also gives the melting point ranges of appropriate salt hydrates. Sodium sulphate has received considerable attention because

Table III-2. * Melting point ranges and Latent heat of some phase change heat storage materials.

Phase Change Heat Storage Material	Density $kg.m^{-3}$	Transition Temp. °C	Latent Heat $kJ\ kg^{-1}$
Calcium chloride - Hexahydrate $Ca\ Cl_2 - 6H_2O$	1150	26-35	175
Sodium carbonate - Decahydrate $Na_2\ CO_3 - 10H_2O$		32-36	247
Sodium sulphate - Decahydrate $Na_2\ SO_4 - 10H_2O$	1460	32.4	243
Sodium thiosulphate-Pentahydrate $Na_2\ S_2O_3 - 5\ H_2O$		48-51	209
Eutectic $LiNO_3 - NH_4NO_3 - Na\ NO_3$ (25-65-10)%	1640	80	113
Eutectic $Mg\ (NO_3)_2.6H_2O$ - $Mg\ Cl_2.6H_2O$ - (53-47)%	1610	59	145
Eutectic. $CaCl_2 - Mg\ Cl_2 - H_2O$ (41-10-49)%		25	175
Urea - NH_4NO_3 (45.3 - 54.7)%		46	172
Eutectic Na/K/Mg/Li-F		449	600 To 900
Eutectic Na/Ca/Mg-F		745	
Paraffin Wax	930	49-63	
Ice	917	0	332

of its large latent heat of transition and because its transition temperature can be varied from 32.4°C to as low as 5°C by incorporating additives. There is a tendency for the anhydrous sulphate to settle to the bottom of a storage vessel because it has a higher density than saturated solution. It does not recombine easily when subsequently cooled. Also the solution can be subcooled by a few degrees before crystallisation of the decahydrate takes place. Attapulgite Clay can be added to keep the sodium sulphate in suspension and borax acts as a nucleating agent to reduce subcooling (20). Practical stores using sodium sulphate are often constructed of horizontal layers of shallow containers (like trays) of material with spaces between the layers through which the heat transfer medium (gas or liquid) can pass.

Another approach is to introduce a fluid (into the salt solution and which is immiscible with it) at the bottom of a storage vessel. The fluid is less dense than the solution and bubbles rise through the store exchanging heat with the store medium and agitating it. The fluid passes through solar panels or a heat source during the changing of the store and through a heat exchanger to the load in the discharging mode.

Latent heat storage systems require both a transport liquid and a storage material. Thus a heat exchanger is needed. This presents a distinct disadvantage for these systems when compared with units that use one fluid for both the storage and the transport of energy.

III-4-5. Sensible Heat Storage Units

There are two types of sensible heat storage unit (21, 22).

1. The first type involves systems where the storage material also serves as the energy transporting fluid. This one is often referred to as a Heat Accumulator.

2. The second type involves storage units that require a separate fluid system to transport the energy from one location to another. This type is often referred to as a Heat Regenerator. This type of unit is somewhat more complex, since a heat transfer surface area and a temperature difference between the fluid and the surface of the storage material are required.

1. Heat Accumulator Units - Hot water storage, as used in most homes for domestic water systems or as used in solar energy applications, fall into the first category. Its major advantages are a relatively high storage per unit volume, due mainly to its high specific heat, and the fact that it can be used for both energy storage and as the energy transporting fluid. These types of units store energy in the low temperature range 0 - 110°C. Because of the temperature range used, it is often necessary to use additives to lower the freezing point. The position of the storage tank's inlet and outlet nozzles is very important, since one would ideally like to have thermal stratification present in the tank, with the hot water at the top and the cold water at the bottom.

2. Heat Regenerator Units - The second type of sensible heat storage unit uses a liquid or a solid as store material. There are two major groups of materials one is comprised of firebricks formed from clay, chrome, Feolite (Fe_2O_3), magnesite and various mixtures of these ingredients. The other is composed of castable metals which include gray cast iron and cast iron containing alloying ingredients

such as silicon and aluminum. Rock is another material used for sensible heat storage mainly in domestic applications.

Interseasonal Storage - Seasonal storage is technically possible at U.K. latitudes, but the volumes required are large. Consider, for example a standard house with a typical winter heating consumption 100 kWh/day, which mean about a 15MWh annual heating requirement. If water storage is used then, with 30°C of temperature rise, a volume of $430m^3$ would be required to store all the heat energy needed for one year. A smaller store could be used at a higher temperature but then the store insulation would become more costly and the solar collection efficiency drops with rising temperature. Table III-3 has been compiled from available sources and gives some materials and their characteristics when used for sensible heat storage. This table also shows the relative store volume required for an annual sensible heat store capacity of 15 MWh with a rise of 30°C in temperature, as was calculated in this study.

Prediction of the performance of solar heating systems utilising annual storage are made in reference (26).

III-5. Solar Assisted Heat Pump Types

The combination of a heat pump and a solar energy system would appear to alleviate many of the disadvantages that each has when operated separately.

During winter, the energy that could be collected by the solar system but which would be too low in temperature to be useful for direct heating may be used as a source for the heat pump. It is

Table III-3*. Relative Store Volumes required for sensible heat stores to store a seasonal heat capacity of 15 MWh (≡ 30°C rise)

Sensible Heat Storage Materials	PROPERTY				Volume req. to store 15 MWh(≡ 30°C rise) m^3	Ratio
	Store Operating Temperature	Density kg/m^3	Specific Heat kJ/kg°C	Heat Capacity $kJ/m^3°C$		
Water	20 - 110	1000	4.18	4180	430	1
Rock (Average Value)	20 - 150	2000-3500	0.87	2390	753	1.7
Concrete	60 - 550	2245	0.962	2160	833	1.9
Feolite Bricks	50 - 800	3900	0.92	3588	501	1.1
Magnesite Bricks	50 - 800	3500	0.984	3444	522	1.2
Scrap Cast Iron	50 - 800	7480	0.46	3440	523	1.2
Scrap Aluminum	50 - 600	2690	0.912	2620	687	1.6
Fire-Clay Bricks	60 - 500	2246	0.837	1880	957	2.2

* References: (9-12-20-21-23-24-25)

apparent that a practical and effective heat pump system employing solar energy must have an alternative heat source to be used in severe weather conditions, and some means of heat storage to be used during periods of insufficient solar radiation. For example, in the U.K. during the mid-winter period with radiation level of 2 kWh/m^2 day (see Fig. 27) and typical heating consumption of 100 kWh/day, a Solar Assisted Heat Pump System without storage, in which 25% of the consumption is to be supplied by solar energy with a 30% collection efficiency, would require $(25/2 \times 0.3) = 41m^2$ of collector to provide the 25 kWh of low temperature energy. The cost of such a solar collection system is much greater than that of a slightly larger heat pump, so that the inclusion of a heat store becomes essential.

Since the solar collection-storage system can supply energy at temperatures higher than the ambient outdoor air, the capacity and the COP of the heat pump would increase over that for the heat pump alone, the peak auxiliary load requirement would be reduced, and the combined heating system would be seen to operate more economically. The operation of the solar system at lower collector temperature would reduce the transmission losses and allow a larger percentage of the lower intensity solar energy to be collected thus increasing the efficiency of the solar system. The lower collection temperature may allow the use of collectors without covers or the roof itself as an absorber, which would reduce the first cost from that of a conventional solar system.

All systems combining heat pumps with solar collectors are fundamentally parallel, series or a combination of these: The dual source system. They are named according to the path by which the heat pump and the solar system bring heat into the house. Schematic. configurations of these systems are shown in Figure 20.

The parallel system employs an air-source heat pump to provide an auxilliary heating source for an otherwise undersized, conventional solar system. The series systems uses a water-source heat pump to draw heat from the solar energy storage tank and supplies it to the house. The series sytem, compared to the parallel, has the advantage of operating at lower solar collection temperature and higher heat pump source temperature. However, during periods of low solar collection, the storage temperature of a series system is lowered by the heat pump to a temperature below that of the outdoor air temperature. Under such conditions, the heat pump drawing energy from storage is no longer advantageous. The combinated system is intended to remedy this problem by using a heat pump that takes heat from either outdoor air, or from storage, depending on the highest source temperature.

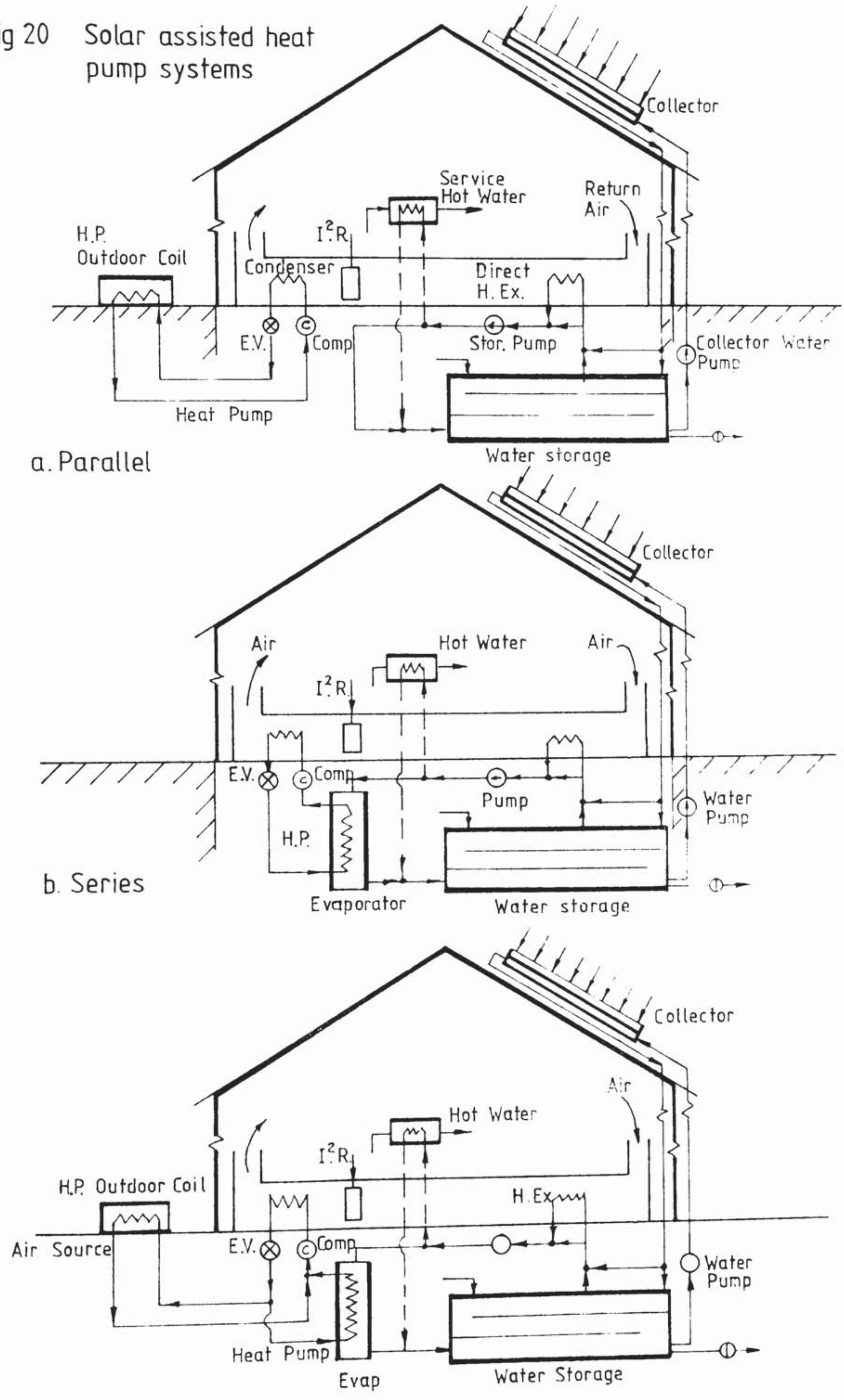
III-5-1. Parallel System

In this system (figure 20-a) solar energy meets most of the heating requirement and the heat pump (air to air) works only when the room temperature falls below a prescribed level. Electrical heaters work only when neither solar nor heat pump can maintain the set temperature. The parallel system does not use solar energy as a source for the heat pump.

III-5-2. Series System

This is also a relatively simple solar-heat pump system which uses a water to air heat pump placed between the solar system and the load

Fig 20 Solar assisted heat pump systems



(Figure 20-b). The evaporator is in the storage water circuit and the condenser is placed in the house air duct. Direct solar heating can be used by bypassing the heat pump when the storage temperature is high enough to deliver heat directly to the house, without expending any heat pump work. Compared to the other systems, this one has the advantage of increasing both the COP(H) and the collector efficiency.

III-5-3. Dual Source System

This system is similar to the series type, but the dual system has an additional evaporator placed outdoors, converting the heat pump into an air to air or a water to air type according to the source from which heat is to be extracted. There are three heating modes for a dual source system. (27). Mode 1: Direct solar heating carried out from the store. In this case the store tank is hotter than the predetermined control temperature value. The Heat Pump is off and any excess solar energy collected is placed into the store - Mode 2: This involves the use of the heat pump to boost the temperature of air that has been preheated by solar energy stored in the tank. It means simultaneous use of the heat pump and stored solar energy. Successful operation of this mode requires the heat pump to have some means of capacity modulation, otherwise pre-heated air feeding its condenser will cause overloading of the heat pump compressor. Capacity modulation is accomplished by the use of a two-speed heat pump in all of the systems. Mode 3: This occurs when the storage is at a low temperature and only the heat pump is useful. When the tank temperature is below the predetermined control value, but higher than the minimum temperature, ambient temperature fluid from the storage tank is pumped through the evaporator and used as a heat source. When the ambient temperature is higher than the tank temperature, ambient air is the heat source for the evaporator and auxiliary heat is supplied when needed. The dual source system appears to take advantage of the best features of the parallel and series systems. Auxiliary heat may be supplied by gas, oil or electrical resistance. The solar system can deliver energy to the service hot water systems when there is no space heating load.

The pre-set temperature limits in the storage water can be considered as: T = 26°C (i.e. the minimum required for preheating the air coming to the building) and T = 45°C (i.e. the minimum required for direct heating of the building from the store). Thus, the 3 operating modes can be written as: $T_S > 45°C$ mode 1

$26°C > T_S < 45°C$ mode 2

$T_S < 26°C$ mode 3

Where T_S = storage water temperature in the tank.

Solar assited heat pump systems can be used to produce hot water only. Figure 21 shows solar water systems including a heat pump to upgrade the heat.

III-6. Solar Assisted Heat Pump System Performance

This heating system is generally evaluated on a seasonal basis, so that the important quantities for its evaluation are the seasonal energy flows, which permit the computation of its Seasonal Overall Performance [$S.O.P.(H)_{SAHPS}$], defined for an electrically driven system as:

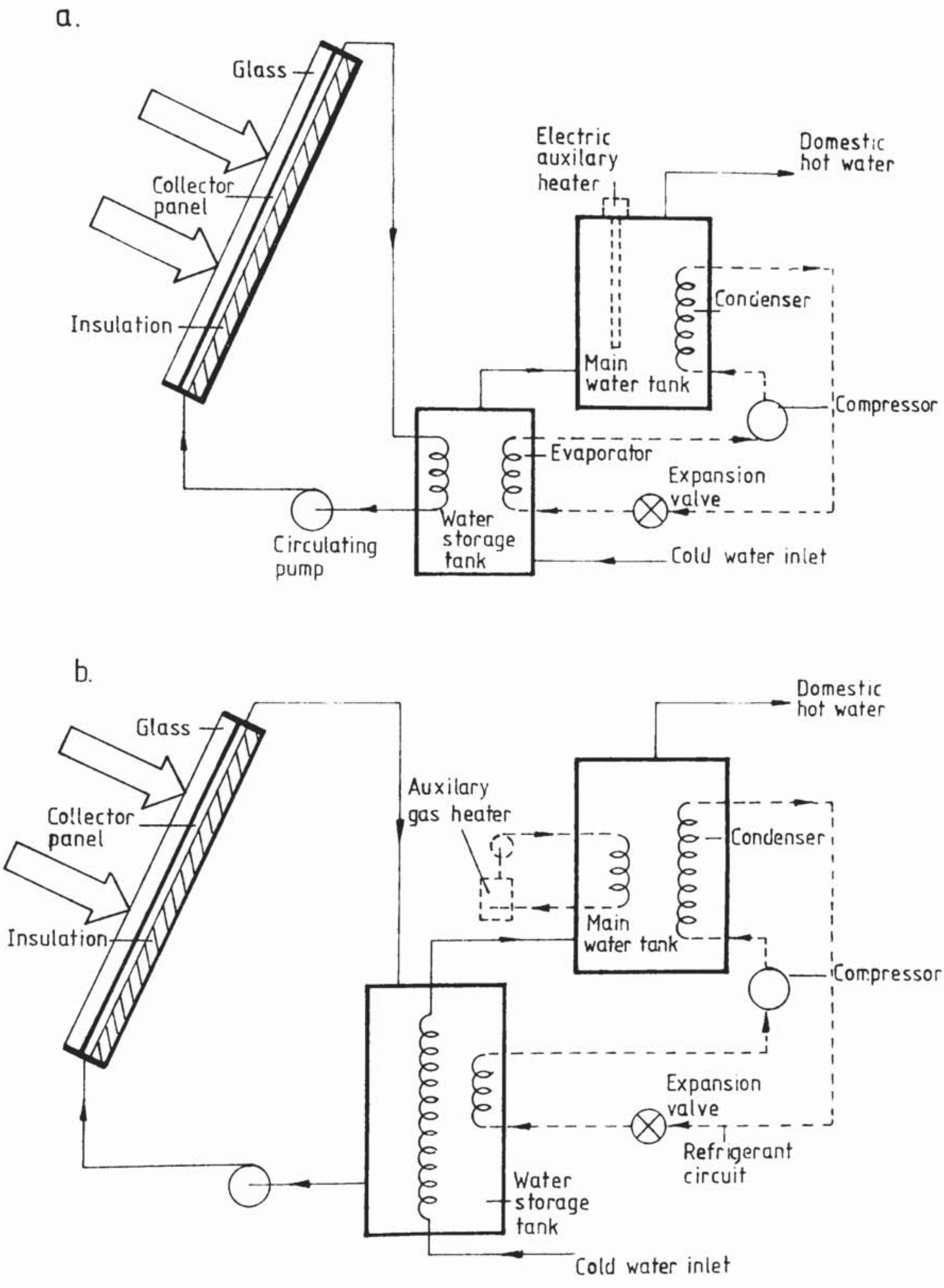


Fig 21. Heat-pump assisted solar water heating systems

$$S.O.P.(H)_{SAHPS} = \frac{\text{Seasonal Heating Load}}{\text{Seasonal Electrical Energy Input}} \qquad \text{(III-1)}$$

The seasonal energy flows to take into consideration are:

Q_{Load} = Total seasonal heating load equal to the total space and water heating requirements.

Q_{Air} = Total heat extracted by the heat pump from the ambient air

Q_{Solar} = Total heat collected by solar energy system

W_{HP} = Total electricity required by the heat pump compressor and fans.

E_{Aux} = Total auxiliary energy (normally electric) required to meet the hot water and space heating loads.

The system energy balance can be written as:

$$Q_{Solar} + Q_{Air} + W_{HP} + E_{Aux} = Q_{Load} \qquad \text{(III-2)}$$

The first two terms (Q_{Solar}, Q_{Air}) are "free energy", while the second two terms (W_{HP}, E_{Aux}) represent "purchased energy". The seasonal overall performance can now be written as:

$$S.O.P.\ (H)_{SAHPS} = Q_{Load}/(W_{HP} + E_{Aux})$$

$$S.O.P.\ (H)_{SAHPS} = 1 + \left[(Q_{Solar} + Q_{Air})/(W_{HP} + E_{Aux})\right] \qquad \text{(III-3)}$$

As can be seen from the equation III-3 the SOP(H) increases directly as term "free energy" ($Q_{Solar} + Q_{Air}$) increases. At the same time, if Q_{solar} collected during the season is higher, then the amount of E_{Aux} will be lower. This readuces the purchased energy term, which also leads to an increase in the overall performance of the system.

Another method to evaluate the thermal performance of an assisted solar heat pump system is by calculating the fraction (F) of the total load that is met by non purchased energy (27, 28) defined as:

$$F = (Q_{Air} + Q_{Solar})/Q_{Load} \qquad \text{(III-4)}$$

$$F \doteq 1 - \left[(W_{HP} + E_{Aux})/Q_{Load}\right] \qquad \text{(III-5)}$$

The thermal performance fraction (F) reflects only energy requirements, and permits the comparison of the performances of the different solar assisted heat pump systems e.g. dual source, series, and parallel systems, as well as conventional solar and conventional heat pump systems:

(a) For a house with neither a solar energy system nor a heat pump, F equals zero.

(b) For a house with only a heat pump, the fraction of the heating requirement supplied by non purchased energy is Q_{Air} divided by the total heating requirement.

In this case the value of F depends on the COP of the heat pump and on the relative sizes of space and water heating loads (Q_{Load}), and is usually between 0.2 and 0.4 (28). The following relations have been developed. Starting with the COP of the heat pump, one can write:

$COP(H)_s$ = Heat Pump Output/Energy Input to drive compressor and fans

$$COP(H)_s = [Q_{Air} + W_{HP}]/W_{HP}$$

And:

$$1/COP(H) = (Q_{Air} + W_{HP} - Q_{Air})/(Q_{Air} + W_{HP}) = 1 - [Q_{Air}/(Q_{Air} + W_{HP})]$$

$$Q_{Air} = (Q_{Air} + W_{HP})\ (1 - (1/COP(H)))$$

Or, dividing by Q_{Load}:

$$Q_{Air}/Q_{Load} = [(Q_{Air} + W_{HP})/Q_{Load}]\ [1 - (1/COP(H)_s)] \qquad \text{(III-6)}$$

$$\text{Air energy contribution} = \frac{\text{Heat Pump Output}}{\text{Heating Load}}\left[1 - \frac{1}{COP(H)_s}\right]$$

And finally:

$$F_{(air)} = \%\ (HP)\ [1 - (1/COP(H)_s)] \qquad \text{(III-7)}$$

Considering the series system (Figure 20-b) for the case in which the heat pump contributes to the domestic hot water system, a mathematical relation for the solar energy contribution in that system can be written:

$$F_{(Solar)} = \%\ (HP)\ [1 - (1/COP(H))] \qquad \text{(III-8)}$$

Where $\%(HP) = (Q_{Solar} + W_{HP})/Q_{Load}$

(c) For a house with only a conventional solar system, F depends on collector area. As collector area increases, F increases from zero asymptotically towards unity.

III-6-1. Comparative performance of the three basic SAHP configurations

An interesting comparative study, made by simulations with TRNSYS (a transient simulation program), of the performance of three basic solar assisted heat pump configurations, as well as conventional solar and conventional heat pump systems, has been reported by T.L. Freeman et al (27). Figures 22 and 23, taken from that report show some of the results that were obtained. The trends are similar to these for other system sizes and other locations.

The fraction F of the annual load met by free energy as a function of collector area is shown in Figure 22 for conventional furnace, solar, conventional heat pump, parallel, series and dual source systems.

The relative contributions from each of the four heat sources, for the parallel, dual source, series solar assisted heat pump systems and for the conventional solar and heat pump systems, is shown in the bar graphs of Figure 23. The combined height of Q_{Solar} and Q_{Air} bars represent the percentage of the total heating requirement supplied by free energy and is therefore equivalent to F. The systems are ranked from left to right in order of decreasing F value.

Some interesting findings are apparent from the results of the simulations as shown in these figures:

(a) The Fraction F asymptotically approaches unity as collector area

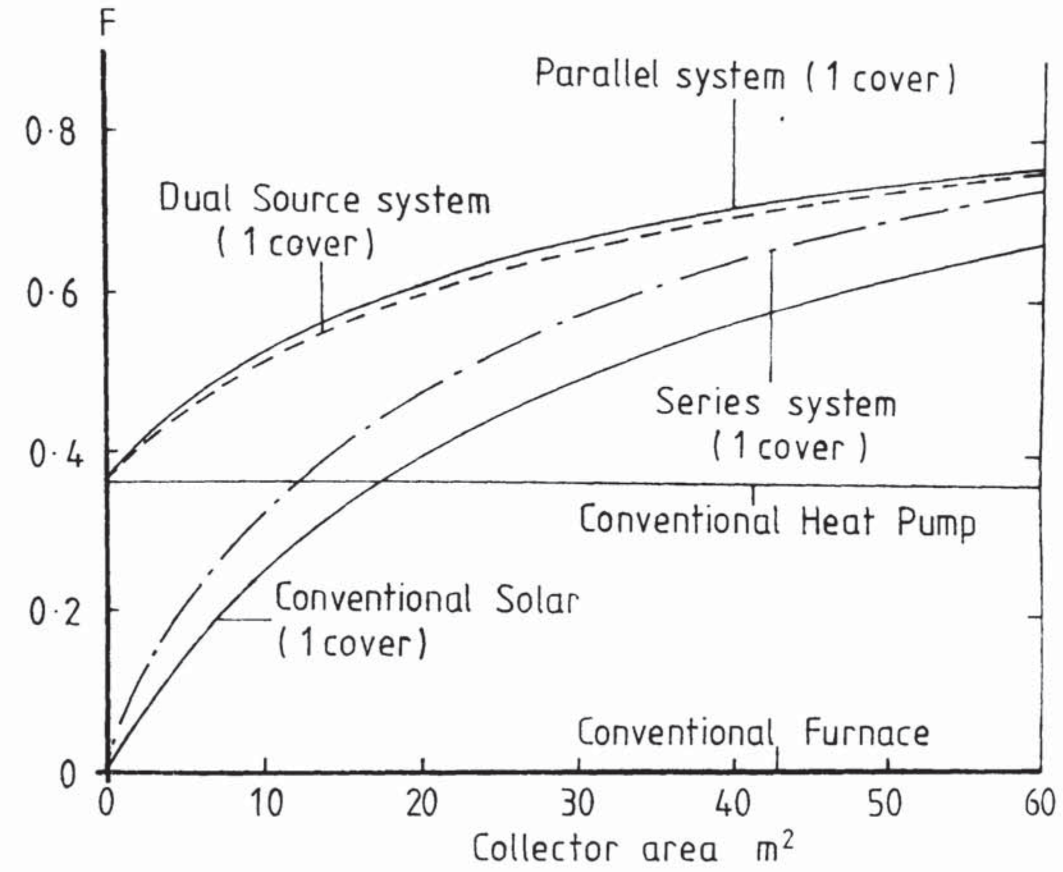


Fig 22 Fraction of free energy as function of the collector area

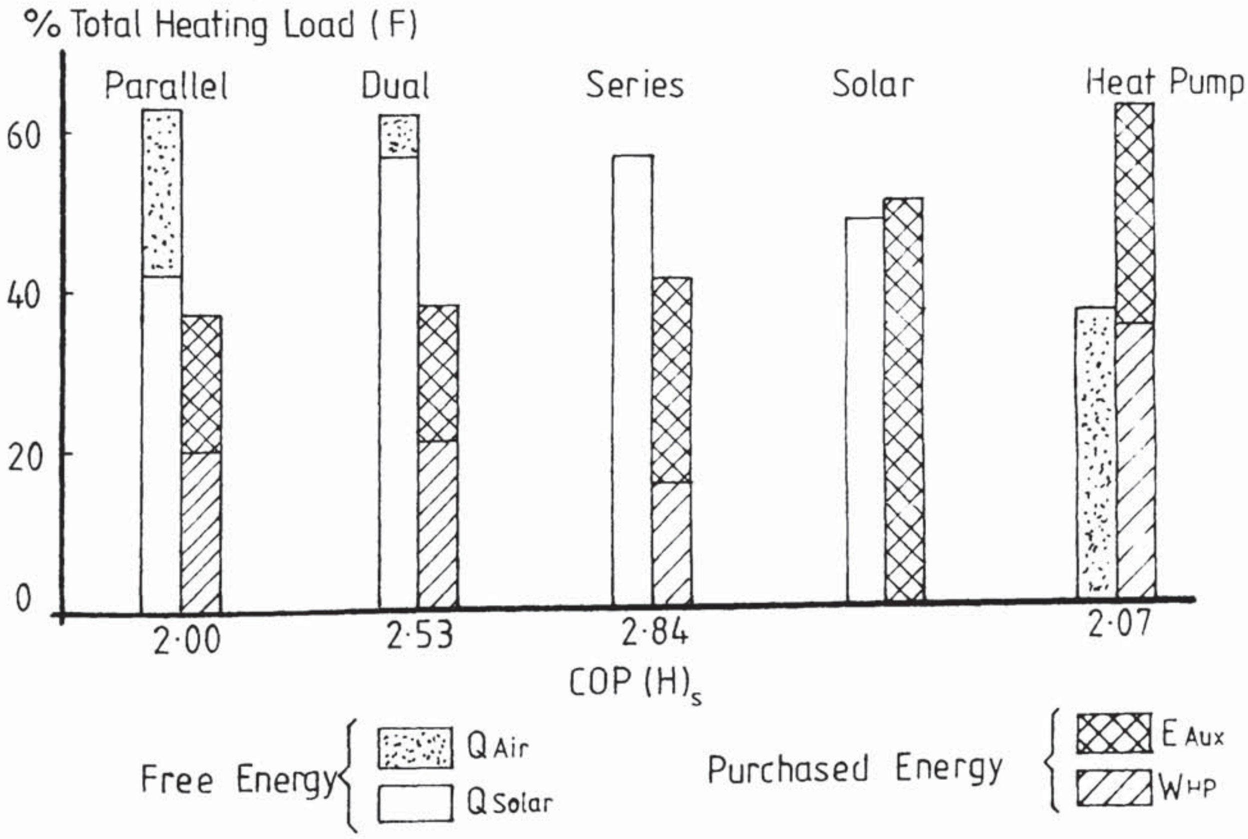


Fig 23 Heat contribution from different sources for Solar Assisted Heat Pumps and Conventional systems

increases, for all three solar assisted heat pump systems. The trend is similar to the conventional solar system but always with greater values for F. As a result the solar assisted heat pump systems perform more and more like the conventional solar system at large collector areas. However, for practical collector areas as used in domestic applications solar assisted heat pump systems always have better performance than conventional ones as was predicted. (higher F values)

(b) The heat pump seasonal COP(H) values given with Figure 23 show how they vary between the different systems and demonstrate that the use of a solar source for the heat pump raises the seasonal COP(H) over that of the conventional heat pump and parallel systems, which achieve heat pump COP(H) values of only 2.1 and 2.0 respectively.

As expected the seasonal COP(H) for the series and dual source heat pump are substantially higher since they utilise the solar heated storage tank as a source.

The seasonal COP(H) of the heat pump for the series system is higher than that of the heat pump for the dual source system because the series heat pump operates only down to a source temperature of 5°C, while the dual source heat pump utlises much colder ambient air as a source when the tank has reached the freezing point. As a result the dual system supplies more heat to the house than the series one but at lower COP(H) in its heat pump. The dual source system has better capability for using the solar source.

(c) According to Figure 23 the parallel system appears to have, although just, better overall performance than the dual system. This is due to the fact that in the dual and series systems heat pump work must be expended to utilise the same energy that is collected and supplied directly to the house without heat pump work in the conventional solar and parallel systems.

During the winter, storage water temperatures, in series and dual systems, are rarely high enough to be used for direct space or water heating, whereas the higher storage temperatures in the conventional solar and parallel systems permit, during this season, direct heating of the house. Thus they use less purchased energy to cover the heating load (Q_{Load}) and therefore they achieve higher overall performance. Nevertheless, the dual source system, because of the lower storage temperature presents other advantages that make it a better system for heating applications e.g. inexpensive solar collectors (possibly systems without covers), less insulation and therefore, less expensive heat storage tanks, better COP(H) of the heat pump and a much better capability of using the solar source as previously mentioned.

It should be remembered that the study referred to, relates to the Wisconsin climate. The parallel system requires a higher store temperature. This in turn means more expensive collectors (to achieve higher efficiencies at the higher temperatures), and higher insulation cost for the store. In addition because of the frosting or icing produced on the evaporator (for an air source Heat Pump) at low temperatures and high relative humidities, the parallel system is less attractive for heating applications in the U.K.

CHAPTER IV

HEAT PUMPS IN THE U.K. SOLAR AND WEATHER CONDITIONS

IV-1. Heat Pump Progress in the U.K.

Air has been the source for nearly all installations (29) in the U.S.A., U.K., Germany and France. Reversible air to air heat pump have been used for many years (30) in the U.S.A., for heating in winter and cooling in summer. They have been used in the U.K. since the mid-60s for space heating, but the major suppliers of heat pumps to the U.K. are based on the U.S.A. models, which are primarily for commercial and industrial applications, and only a few smaller models for domestic heating are available. These machines are electrically driven air to air heat pumps and usually require a three phase power supply.

Typically the performance of an air to air machine is given in table IV-1. The column COP (theory) gives the theoretical performance of a heat pump operated on a Rankine cycle using R.12 with 10°C temperature drop at each heat exchanger and output air at a temperature of 35°C.

Table IV.1 Manufacturer's performance for a typical Heat Pump. Air to Air (1)

Ambient Temperature	Heat Output	Power Consumed	COP(H)	COP(theory)
+ 13°C	7.4 kW	2.9 kW	2.55	6.32
- 1°C	4.6 kW	2.25 kW	2.04	4.59
- 7°C	3.6 kW	2.03 kW	1.77	4.00

The expected performance for a typical machine with a condensing temperature in the region of 55°C to 60°C is given in an empirical relationship by Heap* as:

$$COP\ (H) = 2.186 + 0.058\ T_1$$

Where T_1 is the source temperature in °C

Studies by the Electricity Council indicate the COP(H) in the region of 2.3 to 2.5 for present machines in the U.K. climate averaged over one heating season. In fact an electric driven heat pump with a COP of about 2.0 uses the same amount of primary energy to provide the same amount of heat as a gas fired boiler, working at 60% efficiency, using substitute natural gas. This puts even present machines in a favourable position in terms of utilisation of primary energy resources.

In spite of significant advantages, however, there are still several technical problems (31,32) associated with the heat pump systems in common use. Some of them are:

1. The heat pumps which are commercially available in the U.K. have higher running costs (30) when used for domestic space heating than gas or coal heating appliances.

2. They periodically need defrosting: winter conditions in the U.K. with high humidities, give rise to defrosting requirements when the air temperature drops to around 4°C. In commercial models this is overcome either by direct heating of the evaporator or by the periodic reversal of the operating cycle.

* Heap R.D. Building Services Engineer (1976), 80.

3. They require supplementary heat. It was seen that the performance of a heat pump decreases with the temperature of its heat source, and if ambient air is used the minimum output of the heat pump coincides with the maximum heating load of a building. Thus, the conventional air to air heat pump must be augmented by a fossil fuel appliance if the building demand exceeds the heat pump output. For example an air source heat pump, with an outdoor evaporator, heats very well down to 4°C. Below this temperature, frosting on the coil or merely depressed evaporator temperatures make the use of supplementary heat necessary. Furthermore most existing units in the U.K. were designed primarily for cooling with reversible facilities, and incorporate direct electric supplementary heat for heating in winter. Wide-spread use could pose a problem to the electricity generation network: The peak load problem is exacerbated both when the resistance heaters are in use and through the high start power requirements of the motors to drive the heat pump compressor.

4. Noise levels of today's heat pumps may be too high for the average U.K. housing densities, particularly for operation at night and on many models, this is still not low enough for an acceptable level even for daytime operation (29, 33). So, apart from maximising the heat transfer, the most important design improvement needed for air source heat pumps is for lower noise emission.

These previous four principal problems of existing heat pumps, especially number three, which causes the COP's to fall off, are not favourable for the image of an otherwise efficient system.

Although the design of heat pumps which use the atmosphere as a heat source has been much improved during the last four years (34, 35), contemporary air to air heat pumps, as well as other source/sink combinations, require fundamental engineering desing revision. The objectives being, lower cost through standardisation and large volume production; lower noise levels for application to domestic environments; longer life through more rugged construction; low maintenance costs; simplified "on site installation", and the most important when incorporated in a heating system: a better seasonal coefficient of performance. Examples to improve performances would be the reduction of temperature differences at the evaporator and the condenser, through improved heat transfer surfaces, and the use of better modes of operation. With large temperature differences (36) the compressor operates at a high pressure ratio, producing such a high discharge temperature in the refrigerant fluid that certain lubrication problems can be encountered. Nevertheless for the near future, for certain applications, heat pumps could be more efficient than most conventional systems provided that new technological developments can be incorporated into their design and the seasonal COP's improved.

It was stated that one method of improving the performance on a heat pump for space heating would be by supplying solar energy as a low temperature heat source in conjunction with an associated storage, in order to have a "solar assisted heat pump system". This may also be accomplished by using the roof of the house itself acting as an absorber of solar radiation. The absorbed energy may be used directly or stored for use by the heat pump during times of low ambient temperature.

Several solar assisted heat pump systems, for heating and cooling, using only solar collectors have been developed throughout the world for a range of weather conditions. See for example references (27) to (31) and (37) to (40). Some of them (Ref. 30, 38, 39) are related to the U.K.

These studies consider different aspects of solar assisted heat pump systems, including in some cases an economic analysis. In this last aspect most systems have high capital costs in common with most systems for converting solar energy and in addition storage is needed. Another important conclusion obtained from studies of these publications is that all the modes of operation of the systems proposed, require supplementary heat during the peak load time in winter.

This present theoretical study takes into consideration some of the problems relating to other systems already mentioned, and gives in section VI a solar assisted heat pump system which uses the roof of the house as solar radiation absorber, instead of solar panels.

The proposed system is in principle suitable for adverse climatic conditions due to the possibility of frost reduction over the evaporator. It does not include auxiliary heat supply for space heating, but does include supplementary heating for hot water if required at a temperature greater than 50°C. The system could provide cooling when required and could be installed in existing dwellings without significant changes to their structure.

The principal purpose is to get a system with different modes of operation, and with reasonable temperature differences at the evaporator-condenser, which could be able to achieve maximum energy saving and provide a high seasonal overall performance.

IV-2. Domestic Heating Requirements in the U.K.

In 1972, the U.K. consumed 8.83×10^9 GJ (2455×10^6 MWh) of primary energy (42), which came from four different sources, with percentage contributions from each source as follows: Oil 48%, Coal 37%, Natural gas 12%, Nuclear and Hydropower 3%.

Final users received only 6.16×10^9 GJ (1713×10^6 MWh) equivalent to 70% of the primary energy consumed. The difference of 2.67×10^9 GJ (742×10^6 MWh), was due to losses when converting primary energy into the final utilized form, and also to losses in the distribution of both primary and secondary energy.

The identified final users and the relative amount of primary energy accounted for by each of them were:
Industry 41%, Domestic 29%, Transport 16% and Other Users 14%.

IV-2-1. Performance Energy Ratio

This is the ratio between the amount of the energy received in final form by the users, Eu, and the primary energy, Ep, which went into its production:

$$\text{Performance Energy Ratio} = \frac{Eu}{Ep}$$

The reciprocal value of this ratio, gives therefore, the number of units of primary energy input per unit of final energy output.

In 1972 the performance energy ratio, for each of the final forms of fuel as received by the different users, was (42):

Electricity	=	0.27	(1/Ratio = 3.73)
Manufactured fuels	=	0.72	(1/Ratio = 1.40)
Natural gas	=	0.94	(1/Ratio = 1.06)
Oil	=	0.93	(1/Ratio = 1.08)
Coal	=	0.98	(1/Ratio = 1.02)

Manufactured fuels are: coke, coal gas etc. It can be seen that the efficiency in the generation of electricity from primary fuel is only 27% and that electricity as a source of usable energy is most inefficient in primary energy usage. However, the total demand for electricity in 1972 (38) was 214×10^6 MWh and then, the primary energy required for its production was: $3.73 \times 214 \times 10^6 = 798 \times 10^6$ MWh. Therefore, reducing the demand for electricity, in the U.K. would produce a considerable reduction on the consumption of primary energy.

IV-2-2. Primary Energy Consumption by the Domestic Sector

As it has been indicated domestic energy usage in the U.K. accounts for approximately 29% of the total National primary energy consumption, this can be seen from the following energy statistics, for the domestic sector, compiled from data on reference (42).

a) Net energy received by the domestic sector (1972):

Electricity	$= 0.31 \times 10^9$ GJ	$= 86.16 \times 10^6$ MWh
Manufactured fuels	$= 0.37 \times 10^9$ GJ	$= 102.86 \times 10^6$ MWh
Coal	$= 0.46 \times 10^9$ GJ	$= 127.88 \times 10^6$ MWh
Natural gas	$= 0.24 \times 10^9$ GJ	$= 66.72 \times 10^6$ MWh
Oil	$= 0.16 \times 10^9$ GJ	$= 44.48 \times 10^6$ MWh
Total	$= 1.54 \times 10^9$ GJ	$= 428.12 \times 10^6$ MWh

b) Primary energy consumption by the domestic sector, calculated on the basis of previous values:

Electricity	$= 3.73 \times 0.31 \times 10^9$	$= 1.14 \times 10^9$ GJ	(45%)
Manufactured fuels	$= 1.40 \times 0.37 \times 10^9$	$= 0.51 \times 10^9$ GJ	(20%)
Coal	$= 1.02 \times 0.46 \times 10^9$	$= 0.47 \times 10^9$ GJ	(19%)
Natural gas	$= 1.06 \times 0.24 \times 10^9$	$= 0.26 \times 10^9$ GJ	(9%)
Oil	$= 1.08 \times 0.16 \times 10^9$	$= 0.17 \times 10^9$ GJ	(7%)

Total consumption $= 2.55 \times 10^9$ GJ (100%)

Losses $= (2.55-1.54)\ 10^9 = 1.01 \times 10^9$ GJ

% of primary energy $= (2.55/8.83)\ 100 = 28.88\%$

The domestic sector receives 29% of the National primary energy consumption. It can also be seen that electricity accounts for 45% of the sector's consumption of primary energy. Therefore, reducing the demand for electricity in the domestic sector would have a considerable effect on primary energy consumption in the U.K.

IV-2-3 Distribution of the energy consumption in the domestic sector

The final net energy supplied to the sector in 1972 was 1.54×10^9 GJ

(428×10^6 MWh), and assuming that there are approximately 19 million households in the U.K., this means that each household on average received in 1972 a quantity of energy equal to 81 GJ (22518 kWh) consumed in the following form (42):

Space Heating	= 52 GJ	(14456 kWh)	= 64%
Water Heating	= 18 GJ	(5004 kWh)	= 22%
Cooking	= 8 GJ	(2224 kWh)	= 10%
TV, Lighting, etc.	= 3 GJ	(834 kWh)	= 4%
Total	= 81 GJ	(22518 kWh)	= 100%

The values have been averaged for each form of consumption. One can see that something of the order of 84% of the energy in the domestic sector is used for domestic space and water heating, i.e. roughly 22% of the national energy consumption is required for domestic heating only. Considering a heating season of six months, the typical winter heating energy consumption for an average four person family house in the U.K. will be: (Gross consumption)

$$= \frac{52 \times 278}{180} + \frac{18 \times 278}{365} = \frac{\text{GJ} \times \text{kWh}}{\text{days GJ}} = 94 \ \frac{\text{kWh}}{\text{day}}$$

Normally a heating consumption of 100 kwh/day is considered representative for the whole winter period, as it has been indicated in previous chapters.

IV-2-4. Space Heating Requirements.

Clearly the space heating load is the largest consumer of energy in the domestic sector, and an estimate of the national space heating demand is:

$$0.64 \times 1.54 \times 10^9 = 0.99 \times 10^9 \text{ GJ } (275.22 \times 10^6 \text{MWh})/\text{year}$$

According to the Electricity Council (42), 36% of the electricity sold to the domestic sector is used for space heating purposes and is equal to:

$$0.31 \times 10^9 \times 0.36 = 0.11 \times 10^9 \text{ GJ } (30.58 \times 10^6 \text{ MWh})/\text{year}$$

The corresponding gross consumption of primary energy being:

$$3.73 \times 0.11 \times 10^9 = 0.41 \times 10^9 \text{ GJ } (114 \times 10^6 \text{ MWh})/\text{year}$$

The remaining: $(0.99-0.11)\ 10^9 = 0.88 \times 10^9$ GJ/year, of the space heating load is met from other sources with an average ratio, between units of primary energy imput and units of final net energy output (considering a rough estimate) of :

$(1.4 + 1.08 + 1.06 + 1.02)/4 = 1.14$

Thus, the total consumption of primary energy, converted into space heating for 19 million homes in 1972 was:

$$0.88 \times 10^9 \times 1.14 + 0.41 \times 10^9 = 1.4132 \times 10^9 \text{ GJ } (392.86 \times 10^6 \text{ MWh})/\text{year}$$

This represents about 16% [i.e. (1.4132/8.83) 100] of the national primary energy consumption. The requirements for space heating reach a peak during the winter, when ambient temperature levels are low. However, it was stated (section IV 2.3) that the average household requirement for space heating over the year in the U.K. is about 52 GJ (14,456 kWh) expended over a six month period. Normally this average is taken in the region of 43 GJ (12000 kWh) to 58 GJ (16000 kWh) according to the type of heating system employed. More severe weather in some parts of the U.K. and a general lack of insulation would tend to increase the national average above the figure of 52 GJ (14,456 kWh). Absence of central heating would reduce the average and on balance these factors seem to cancel each other to a large extent. Then, 52 GJ does not seem an unreasonable, estimate for the national "Gross Demand Average" per household per year. For a six month period of heating, the "Daily Gross Demand Average" amounts to: 14456/180 = 80 kWh/day.

The "net demand" of energy for domestic space heating can be estimated as follows:

The total national domestic space heating demand was estimated to be, 0.99×10^9 GJ (275×10^6 MWh) per year. Of this 0.11×10^9 GJ (30.6×10^6 MWh) is accounted for by electricity. Assuming that the remaining 0.88×10^9 GJ (264×10^6 MWh) are supplied by coal, oil, natural gas and manufactured fuels, at an average operational efficiency of 60 per cent, the net demand at the point of use for non-electric useful energy in space heating is:

$0.88 \times 10^9 \times 0.60 = 0.53 \times 10^9$ GJ (146.8×10^6 MWh)/year

If electricity for space heating is taken as operating at 100% efficiency, then the total "Net demand" of useful energy for space heating amounts to:

$(0.11 \times 10^9 + 0.53 \times 10^9) = 0.64 \times 10^9$ GJ (178×10^6 MWh)/year

If one assumes a heating season occuring over a six month period, then the "Daily Net Demand Average" for each of 19 million households can be estimated as:

$0.64 \times 10^9/19 \times 10^6 \times 180 = 0.187$ GJ (52 kWh)/day.

This figure represents losses through fabric and also losses due to infiltration and ventilation.

IV-2-5. Water Heating Requirements

The requirements for water heating are almost constant throughout the year. An average household, in the U.K., requires 0.15 m^3 (150 litres) of hot water daily, at a temperature range between 55 and 65°C. If a mean annual mains water temperature of $\sim$10°C is assumed, then the "net demand" of useful energy for water heating can be estimated as between 7 and 10 kWh per day, i.e. between 9.2 GJ (2555 kWh) and 13.2 GJ (3650 kWh) per year.

Normally a "Net Demand" of 12 GJ (3336 kWh) is taken as an annual average for domestic hot water consumption in different types of dwellings. The energy consumed to provide this hot water will exceed the previous demand since water heating appliances are not 100 per cent efficient. Water heaters in general work with an average efficiency of 67%, then the average energy consumption for water heating can be estimated as:

$$12/0.67 = 18 \text{ GJ/dwelling} \times \text{year} = 5000 \text{ kWh/dwelling} \times \text{year}$$

and the annual consumption of energy for 19 million households will be:

$$18 \times 19 \times 10^6 = 0.34 \times 10^9 \text{ GJ } (94.52 \times 10^6 \text{ MWh})/\text{year}.$$

The Electricity Council estimates (42) that about 25% of the electricity sold to the domestic sector is used in water heating, so that, the demand in electricity for this purpose amounts to:

$$0.31 \times 10^9 \times 0.25 = 0.08 \times 10^9 \text{ GJ } (22.24 \times 10^6 \text{ MWh})/\text{year}$$

and the corresponding gross consumption of primary energy amounts to:

$$0.08 \times 10^9 \times 3.73 = 0.298 \times 10^9 \text{ GJ } (82.95 \times 10^6 \text{ MWh})/\text{year}$$

Summary account of energy requirements for space and water heating in the domestic sector. (Average values)

I) Space Heating

1) By Household:

Net demand (Heat losses) { 0.187 GJ (52 kWh)/day

Gross demand { 0.289 GJ (80 kWh)/day; 52 GJ (14456 kWh)/year

2) Total for 19 million dwellings:

Net demand { 0.64×10^9 GJ (178×10^6 MWh)/year

a) 0.11×10^9 GJ $\equiv$ Electricity

b) 0.53×10^9 GJ $\equiv$ Other fuels

Gross demand { 0.99×10^9 GJ (275×10^6 MWh)/year

a) 0.11×10^9 GJ $\equiv$ Electricity

b) 0.88×10^9 GJ $\equiv$ Other fuels

3) Primary energy consumption 1.4132×10^9 GJ (393×10^6 MWh)/year or 16% of the total national consumption

II) Water Heating:

1) By Household:

Net demand { 0.033 GJ ($\sim$9 kWh)/day; 12 GJ(3336 kWh)/year

Gross demand { 18 GJ (5000 kWh)/year

2) Total for 19 million dwelling:

Net demand { 0.23×10^9 GJ (63.4×10^6 MWh)/year

Gross demand { 0.34×10^9 GJ (94.52×10^6 MWh)/year

a) 0.08×10^9 GJ $\equiv$ Electricity

b) 0.26×10^9 GJ $\equiv$ Other fuels

3) Primary energy consumption { 0.5944×10^9 GJ (165.2×10^6 MWh)/year or 6.7% of the total national consumption

The non-electric energy demand will be:

$0.34 \times 10^9 - 0.08 \times 10^9 = 0.26 \times 10^9$ GJ (72.28×10^6 MWh)/year

and the gross primary energy consumed for non-electric sources, at an average ratio between units of primary energy input and units of final net energy output of 1.14 (See IV-2.4), will be:

$0.26 \times 10^9 \times 1.14 = 0.2964 \times 10^9$ GJ (82.4×10^6 MWh)/year.

So, the total consumption of primary energy amounts to:

$(0.298 + 0.2964)\ 10^9 = 0.5944 \times 10^9$ GJ (165.2×10^6 MWh)/year

which represents 6.7% [i.e. (0.5944/8.83) 100] of the national primary energy consumption.

If an electric immersion water heater is considered 100% efficient, then the net demand of 12 GJ of useful energy for water heating in each dwelling, will consume only an amount of primary energy equal to:

$3.73 \times 12 = 45$ GJ (12 454 kWh)/year.

IV-3. Electrically Driven Heat Pumps for Domestic Heating and Primary Energy Savings in the U.K.

It becomes obvious that any energy conservation measure taken for space and water heating will have great potential. Two approaches can be made:

1st: To decrease energy demand by reducing heat losses through fabric with better insulation and also by heat recovery.

2nd: To provide a more efficient way of providing the energy especially when electricity is used. A heat pump is considered to be one of the alternative technologies available in this second approach.

IV-3-1. Implication of COP(H), of Electrically Driven Heat Pumps, on Primary Energy Savings

For an electrically driven heat pump, the energy lost when generating and distributing the electricity for the compressor must, however, be considered. The overall efficiency of electricity generation is 27% (Performance energy ratio = 0.27), but if distribution losses, which amount to about 5%, are neglected the efficiency in the generation of electricity from primary fuel becomes 32%. Currently, actual systems of electricity generation can reach efficiency values varying from 25 to 30% in the overall case and 30 to 35% when neglecting distribution losses. In general one can assume 27% and 33% respectively as average efficiency values for electricity generation.

The COP(H) of a heat pump was established as:

$$\mathrm{COP(H)} = Q_1/W_{HP},$$

and the performance energy ratio has to be:

$$\mathrm{P.E.R.} = W_{HP}/Ep$$

So,

$$W_{HP} = \mathrm{P.E.R.} \times Ep$$

$$\mathrm{COP(H)} = Q_1/\mathrm{P.E.R.} \times Ep$$

$$Ep = Q_1/\mathrm{P.E.R.} \times \mathrm{COP(H)}$$

Where

P.E.R. x COP (H) = Overall efficiency of the heat pump in terms of primary fuel

W_{HP} = Energy expended on the compressor and on ancillary equipment (kWh)

E_p = Primary energy consumed (kWh)

Q_1 = Energy available for heating (kWh)

For an electrically driven heat pump, assuming that the efficiency in the generation of electricity is 33% (Neglecting distribution losses), one can write:

$W_{HP} = 0.33\ E_p$

$COP\ (H) = Q_1/0.33\ E_p$

$E_p = Q_1/0.33\ COP\ (H)$

If $COP\ (H) = 3$ then $Q_1 = E_p$

If $COP\ (H) < 3$ then $Q_1 < E_p$

If $COP\ (H) > 3$ then $Q_1 > E_p$

In the operation of an electrically driven heat pump, therefore, in order to obtain 100% utilization of primary fuel one needs a COP(H) of at least 3.0. In practice because of distribution losses, the efficiency is only 27% and $W_{HP} = 0.27\ E_p$, so that, a minimum COP(H) of 3.7 is required to have an overall efficiency of 100%. This means that for 100 units of primary fuel consumed, 100 units of heat are available for heating. It can also be seen, in this case, that for a COP(H) of only 3.0, the overall efficiency of the heat pump would be: $0.27 \times 3.0 = 0.81$, and this 81% well exceeds the 60% seasonal efficiency usually achieved by direct fired central heating using natural gas or manufactured fuels.

Natural gas and manufactured fuels have an average performance energy ratio of 0.83 (see IV-2.1), thus the overall efficiency for a direct fired system will be: 0.83 x 0.60 = 0.497 or ∿ 50% . An electrically driven heat pump, with a COP(H) of 1.85, has an overall efficiency of: 0.27 x 1.85 = 0.499 or ∿ 50%, which makes its performance, equivalent in terms of primary energy consumption, to that of the direct fired central heating system.

IV-3-2. Expected Energy Savings in Domestic Space Heating

Basically there are three forms of space heating, each requiring different distribution temperature ranges:

1. Hot water radiators : 72 to 82°C
2. Underfloor heating : 52 to 62°C
3. Warm air heating : 27 to 47°C

The higher the temperature of the heat distribution system, the worse the average performance of the system. The distribution temperature of the water, in the 1st system, depends on radiator size and for existing houses, radiator areas would have to be increased to provide the same emission at a lower temperature than 72°C; i.e. by doubling the radiator size, either by using finned or large panel radiators, the temperature of water can be reduced to 50°C.

Improved house insulation would reduce the demand for energy because of a reduced rate of heat loss. It is possible that a wet system designed to run at higher temperatures might compensate for heat losses, if run at lower surface temperatures, without the need to increase radiator surface area.

Air distribution temperatures can be reduced to 27°C by increasing the flow rate, but this requires larger ducting and greater fan power. Larger air ducts are liable to transmit engine noise and vibration, which may prove a serious disadvantage. On the other hand, a small thermal mass of air could lead to a high cycle time for the heating unit.

If one considered the use of a heat pump to supply these temperatures, with a source at a minimum temperature T_2 of -1°C, then the COP(H) ranges would be: (1) 2.3 - 2.1, (2) 3.0 - 2.6, and (3) 5.3 - 3.3, as is shown in the table IV-2, which gives the ideal and the expected COP (H) figures for varying values of the output temperature T_1.
It can be seen from this table that domestic heat pumps with a COP(H), limit set at 3.0, are more suited to warm air or underfloor heating.

Table IV-2 Expected Values of COP(H) for a heat pump operating with varying output temperature T_1 and fixed source temperature T_2 of 272°K (-1°C)

Type of Heating	T_1 (°K)	ΔT (°K)	COP (H)	
			Ideal	Expected
Hot-water radiators	355	83	4.28	2.1
Hot-water radiators	345	73	4.72	2.3
Underfloor heating	335	63	5.31	2.6
Underfloor heating	325	53	6.13	3.0
Warm-air heating	320	48	6.67	3.3
Warm-air heating	300	28	10.71	5.3

An electrically driven heat pump with a COP(H) of 3.0 has a primary fuel overall efficiency of 0.81. If the present annual domestic space heating load of the country could be met by electrically driven heat pumps working with this overall efficiency of 81%. then the resulting consumption of primary energy would be:

$0.64 \times 10^9/0.81 = 0.79 \times 10^9$ GJ (219.6×10^6 MWh)/year.

This amount of energy, compared with the actual gross national consumption of primary energy for domestic space heating of 1.41×10^9 GJ (392×10^6 MWh) per year for the existing mix of fuels, would represent a saving of: (See IV-2-3) 0.62×10^9 GJ (172.3×10^6 MWh)/year which is equal to 7%, of the total national consumption of primary energy.

IV-3-3. Expected Energy Savings in Domestic Water Heating by using Heat Pumps.

As it was already mentioned, hot water for domestic use is required at a temperature range between 55°C and 65°C. The range of performances which might be expected with a heat pump when extracting heat from different sources and delivering hot water at these temperatures, assuming that its COP(H) is about 0.5 $[COP(H)]_R$, are:

Source temperature T_2	°C	-10	-5	0	5	10	15
	°K	263	268	273	278	283	288
Expected COP (H)	T_1 = 328°K (55°C)	2.52	2.73	2.98	3.28	3.64	4.10
	T_1 = 338°K (65°C)	2.25	2.41	2.60	2.82	3.07	3.38

Source temperatures lower than 5°C are more related to ambient air, whereas, higher source temperature values are related to water storage tanks. In both cases it could be possible to obtain the hot water requirements with a COP(H) of about 2.5. The average net demand of 9 kWh daily (3336 kWh/year) required for water heating in each dwelling could be obtained with an electrically driven heat pump operating with this COP(H) of 2.5 and a net electrical expenditure of about 3.6 kWh daily as compared with the net consumption of about 9 kWh daily using an electric immersion heater.

If the annual net demand for useful energy of 12 GJ (3336 kWh)/dwelling for water heating could be met by electrically driven heat pumps with a COP(H) of 2.0, and therefore with an overall efficiency of $0.27 \times 2.0 = 0.54$, then the national consumption of primary energy would be:

$12 \times 19 \times 10^6/0.27 \times 2.0 = 0.42 \times 10^9$ GJ (116.8 kWh)/year.

This amount of energy, compared with the estimated annual gross consumption (0.59×10^9 GJ) of primary energy for water heating (see IV-2.5), represents a saving of 0.172×10^9 GJ (48×10^6 MWh)/year which is equal to $\sim 2\%$ of the total national consumption of primary energy.

Because of this advantage, the heat pump system presented later on in section VI, incorporates in its operation modes the provision for domestic water heating simultaneously with, or as an alternative to space conditioning.

Major savings of electricity and, therefore, of primary energy can be made by using alternative energy sources: i.e. solar energy, with the solar assisted heat pump systems, or with solar collector systems to cover the hot water requirements.

IV-4. Availability of Solar Energy in the U.K.

The amount of solar energy reaching the United Kingdom per year far exceeds the quantity needed to satisfy demands. The utilization of the energy to suit the requirements suffers from many disincentives and this is why (in a society where energy derived from fossil fuels has been relatively cheap) solar energy has not been exploited. Direct radiation from the sun is restricted in many parts of the world, but particularly in regions such as the United Kingdom because of climatic conditions. Nevertheless, the total energy reaching a horizontal surface in the U.K. (30) is of the order 1 MWh/m^2 per year, and a conventional house roof may be 50 to 100 m^2 in area. Hence the annual solar energy available on a standard house would be 50 to 100 MWh/year. Household heating requirements would be 10 to 30 MWh/year and hence if it could be suitably collected and stored, sufficient solar energy would land on the roof of a dwelling to provide all its heating. However, it can be seen from Fig. 27 that 60 per cent of the solar energy arrives during the four summer months when heating is not required, and on the other hand 50% of the heating is required during the three winter months when little solar energy is available. For this reason, unless one considers interseasonal storage from summer to winter there must either be a shortfall of energy in winter or a surplus in summer. Furthermore the energy supply is at a low level when it is most needed for space heating, i.e. at night during Winter period. Solar heating schemes for existing houses are more suited to water heating rather than space heating and even so the capital costs of equipment and installation costs make most schemes unattractive to the individual. B. McNelis (43) has made an interesting assessment based on capital costs, electricity costs, panel area and efficiency.

The provision of solar panels to supply domestic hot water at a suitable temperature (in the region of 55 to 65°C) even during the summer would have a pay back period in excess of ten years. (The consumer should have a reasonable expectation of recovering capital cost in 5 to 6 years). It has already been mentioned that the efficiency of panels decreases as the temperature difference between outlet and inlet increases and it is estimated that the mean efficiency of collection varies between 30 and 40 per cent. The energy required to raise the temperature of $0.15 m^3$ of water to 65°C from a mean annual mains water temperature of 10°C was 9.57 kWh. The energy reaching a surface in the U.K. in mid summer at a latitude of 51° 30′N, facing south and inclined at 40° to the horizontal is in the region of 7.75 kWh/m^2 day ($\sim$ 28 MJ/m^2 day), see Figure 30; therefore, for panels with efficiencies in the region of 30% an area of 4.12 square metres would be necessary to heat the $0.15 m^3$ of water required per day and per home. In practice a panel area of 4 to $5 m^2$ would be necessary. During the mid-winter period with the radiation level in the region of 2.5 kWh/m^2 day ($\sim$ 9 MJ/m^2 day), the rise in temperature would be less than 15°C above water main temperature.

Thus, if serious consideration is to be given to the utilization of solar energy for both domestic space and water heating the inclusion of a heat store becomes essential. The size of store could be reduced if more energy could be made available and/or if the heat could be upgraded, particularly during periods of low ambient temperatures.

IV-4-1. Primary Energy Savings by using Solar Energy in Domestic Water Heating

If one considers the mid-winter radiation level of 9 MJ/day x m^2, see Fig. 30, the energy that could be collected over the year by a solar panel of $6 m^2$ in area and 30% efficient amounts to:

0.009 (GJ/m^2 day) x 0.30 x $6m^2$ x 365 days/year = 6 GJ/year

National primary energy consumption used in domestic water heating, assuming a net demand of 12 GJ/year x dwelling, has been estimated at 0.5944×10^9 GJ/year, which is equivalent to 6.7% of the total. Thus, the overall efficiency in terms of primary energy for this case must be: $12 \times 19 \times 10^6 / 0.5944 \times 10^9 = 0.384$ (or 38.4%).

If 6GJ/year of solar energy in the form of hot water could be provided for half of the 19 million households in the U.K., the saving in primary energy would be about 2% of the total national consumption, i.e.

Required demand - Solar collected = Net demand

$12 \times 19 \times 10^6 - \frac{1}{2} (19 \times 10^6) \times 6 = 0.171 \times 10^9$ GJ/year

Primary energy required = $0.171 \times 10^9 / 0.384 = 0.445 \times 10^9$ GJ

Primary energy saving = $0.5944 \times 10^9 - 0.445 \times 10^9 = 0.1494$ GJ/year

which is equivalent to: $(0.1494/8.83)\ 100 = 1.7\%$ of the total national consumption.

IV-5. Climatological Data

Correct criteria in the selection of the meteorological parameters is a fundamental pre-requisite for designing Solar Assisted Heat Pumps Systems.

Absolute maximum or minimum conditions are unsuitable for the basis of equipment design, for cooling or heating, since they would lead to oversizing and uneconomic plant capacities. Therefore, it is necessary to take into consideration not only the extreme values of temperature, humidity and wind speed etc., but also their frequency, duration, coincidence and the order of diurnal variations.

The principal characteristic of the climate in the British Isles is its remarkable inconsistency; the monthly average figures conceal wide variations in the recorded values. January is not always the coldest month, it is often February, sometimes December and occasionally March.

In 1916 The Midlands (44) had mean January temperatures of about 7°C, 3°C higher than normal, while in January 1963 mean temperatures averaged about -3°C, 7°C below normal. The highest temperatures likely to occur in England are near to 38°C and about 32°C in Scotland.

The lowest recorded temperature is -27°C at Braemar in Scotland. Its continual variation and its high levels in relative humidity during the winter, makes the weather one of the major inconveniences, to the use of air to air heat pumps for heating purposes in the U.K. This highlights the advantage of an integrated heat pumping and thermal store system.

In spite of its inconsistency, the average values of the climate in the British Isles, remains high enough to make possible the advantageous application of solar assisted heat pumps. The very cold weather conditions (Temperatures far lower than -1.5°C), do not last for periods any longer than ten consecutive days in any region of the whole country. This means that a large sized store is not required in the integrated system.

Table IV-3 extracted from reference (44) gives values of average temperatures for the British Isles based on a 30 year period, from 1921 to 1950. Climatological data are published by the Meteorological Office (45) in London, for most sections of the country; however, the IHVE Guide (46) and the CIBS Guide (47) give in the A2 sections, information about the weather its records, frequency data,

Table IV-3 Average Temperatures (°C) for the British Isles on a 30 year period from 1921 to 1950. Extracted from reference (44)

	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEP	OCT	NOV	DEC
Aberdeen	4	4	5	7	9	12	14	14	12	9	6	4
Bath	5	5	7	9	12	16	17	17	14	11	7	5
Birmingham	4	4	6	8	11	14	16	16	14	10	6	4
Blackpool	4	4	6	8	11	14	16	16	13	11	7	5
Bournemouth	6	5	7	9	12	15	17	17	15	11	8	6
Bradford	3	3	5	7	11	13	16	15	13	9	6	4
Brighton	6	5	7	9	12	15	17	17	16	12	8	6
Cambridge	4	4	6	9	12	15	17	17	14	10	7	4
Cardiff	5	5	7	9	12	15	16	16	14	11	7	6
Dover	5	5	6	9	12	14	17	17	15	12	8	6
Dublin	5	5	6	8	11	14	15	15	13	10	7	5
Dundee	3	4	5	7	9	13	15	14	12	9	6	4
Durham	3	3	5	7	10	13	16	15	13	9	6	4
Edinburgh	4	4	5	7	9	13	15	14	12	9	6	4
Falmouth	7	7	8	9	12	15	16	17	15	12	9	7
Felixstowe	4	4	6	8	11	15	17	17	15	12	7	5
Fort William	4	4	6	8	11	13	14	14	12	9	6	4
Ilfracombe	7	6	8	9	12	14	16	17	15	12	9	7
Inverness	3	4	5	7	9	12	14	14	12	8	6	4
Littlehampton	5	5	6	9	12	15	17	17	15	12	8	6
Liverpool	4	4	6	8	11	14	16	16	13	10	7	5
London (Greenwich)	4	4	7	9	12	16	18	17	15	11	7	4
Luton	3	4	6	8	11	14	17	16	14	10	6	4
Malin Head	6	6	7	8	9	12	14	14	13	11	7	6
Manchester	4	4	7	9	12	15	17	16	14	11	7	5
Margate	5	5	7	9	12	15	17	17	16	12	8	6
Norwich	4	4	6	8	12	15	17	17	14	11	7	4
Nottingham	4	4	6	8	12	14	17	16	14	10	7	4
Oxford	4	4	7	9	12	15	17	17	14	11	5	4
Plymouth	7	6	8	9	12	15	17	17	15	12	9	7
Portsmouth	6	6	7	9	13	16	18	18	16	12	8	6
Renfrew (Glasgow)	3	4	6	7	10	13	15	14	12	9	6	4
Shannon Airport	5	6	7	9	12	14	15	16	14	11	8	6
Sheffield	4	4	6	8	11	14	17	16	14	10	7	4
Shrewsbury	3	4	6	9	11	15	17	16	14	10	7	4
Southend	4	4	7	9	12	16	18	18	16	12	8	4
Stornoway	5	5	6	7	9	11	13	13	12	9	7	6
Weymouth	6	6	7	9	12	15	17	17	16	12	8	7
York	4	4	6	8	11	14	17	18	14	10	6	4

recommended design conditions, etc., for heating in the U.K.

Table IV-4, extracted from reference (45) gives information about relative humidity and maximum and minimum temperatures, in the period from 1931 to 1960, for some selected latitudes of the U.K.; i.e. 50° 21'N (South Western), 51° 28'N (Thames Valley), 52°C 29'N (Midlands), and 57° 29'N (North East Scotland). These are considered to be representative of the climatological data for the whole of the British Isles. The last two columns of temperature in the table are the absolute extremes which have been recorded during the entire period of observational records,used in the table for each location.

Table IV-5, is derived from the data given in reference (46) and gives the maximum number of consecutive days, in the whole twenty-five year period from 1925 to 1950, with daily mean temperatures below the value shown, for several latitudes of the U.K.

Table IV-6, taken from reference (46) section A2, gives the recommended winter external design temperatures for the U.K. provided that three conditions are satisfied:

a) "The building thermal inertia is taken into account in accordance with the classification quoted".

b) "The heating system overload capacity is taken into account as indicated"

c) "The heating system can, if necessary, be operated continuously in severe weather".

By observing these conditions it will be possible to maintain the

Table IV-4. Relative Humidity – Maximum and Minimum Temperature data for several latitudes in the U.K.

Period 1931-60	Temperature °C						Relative humidity % Average of observations at	
	Average daily		Average monthly		Absolute			
	Max.	Min.	Max.	Min.	Max.	Min.	0900	1500
PLYMOUTH/MOUNT BATTEN 50° 21'N 4° 07'W 27m								
January	8.2	4.1	12.0	-2.8	13.9	-8.9	89	81
February	8.1	3.5	11.9	-2.8	15.0	-8.3	88	78
March	10.1	4.6	14.6	-1.3	19.4	-5.0	86	74
April	12.3	6.1	17.7	1.0	22.2	-1.7	78	69
May	15.1	8.4	20.8	2.9	26.1	-0.6	77	71
June	17.7	11.3	24.0	6.4	27.8	1.7	80	73
July	19.0	13.0	24.0	8.4	28.9	7.2	81	74
August	19.3	13.0	24.1	8.1	31.1	3.9	83	75
September	17.7	11.7	22.1	5.8	27.2	2.8	86	75
October	14.6	9.2	18.5	2.2	23.3	-1.7	88	77
November	11.3	6.6	15.0	-0.2	17.2	-3.9	88	79
December	9.2	5.1	12.7	-1.7	14.4	-5.0	89	82
Year	13.6	8.0	26.3	-4.6	31.1	-8.9	84	76
KEW 51° 28'N 0° 19'W 5m								
January	6.3	2.2	11.7	-4.3	14.3	-9.5	86	77
February	6.9	2.2	12.1	-3.6	16.1	-9.4	85	72
March	10.1	3.3	15.5	-2.3	21.4	-7.7	81	64
April	13.3	5.5	18.7	0.1	25.5	-2.1	71	56
May	16.7	8.2	23.3	2.7	30.2	-1.0	70	57
June	20.3	11.6	25.9	6.9	32.7	4.8	70	58
July	21.8	13.5	26.9	9.3	33.8	7.0	71	59
August	21.4	13.2	26.2	8.5	33.1	6.2	76	62
September	18.5	11.3	23.4	5.4	29.9	3.0	80	65
October	14.2	7.9	18.7	0.4	25.6	-3.6	85	70
November	10.1	5.3	14.4	-1.4	19.0	-5.0	88	78
December	7.3	3.5	12.2	-3.2	15.1	-7.0	87	81
Year	13.9	7.3	28.6	-5.7	33.8	-9.5	79	67

Table IV-4. Continued.

BIRMINGHAM/EDGBASTON 52° 29'N 1°56'W 163m								
January	5.3	1.6	11.4	-4.6	13.3	-11.7	89	82
February	6.0	1.5	12.0	-4.0	15.6	-8.9	89	76
March	9.1	2.7	15.6	-2.4	20.6	-7.2	85	68
April	12.2	4.7	18.8	0.0	23.9	-1.7	75	58
May	15.6	7.3	23.1	2.2	29.4	-1.1	74	58
June	18.8	10.4	26.1	6.1	30.6	2.8	74	59
July	20.2	12.3	26.5	8.4	32.2	6.1	75	62
August	20.0	12.1	26.0	8.1	32.8	6.1	80	64
September	17.2	10.3	23.4	5.3	27.2	2.8	84	67
October	13.0	7.3	18.6	1.4	25.0	-2.2	88	73
November	8.8	4.7	13.9	-0.7	19.4	-4.4	90	80
December	6.4	2.9	11.9	-3.1	14.4	-6.1	90	84
Year	12.7	6.5	28.7	-5.8	32.8	-11.7	83	69
ACHNASHELLACH 57° 29'N 5°16'W 67m								
January	6.0	-0.4	11.1	-7.5	14.4	-12.8	-	-
February	6.4	-0.1	11.3	-7.3	15.6	-16.1	-	-
March	8.8	1.5	15.2	-4.8	21.1	-12.2	-	-
April	10.9	2.9	16.7	-2.6	22.8	-6.7	-	-
May	14.7	5.4	21.7	-0.9	27.2	-3.9	-	-
June	16.8	8.0	24.5	2.6	30.0	-0.6	-	-
July	17.5	10.0	23.3	4.8	31.1	-0.6	-	-
August	17.5	9.6	23.4	4.1	27.8	0.6	-	-
September	15.4	7.8	20.9	1.6	26.1	-2.2	-	-
October	12.1	5.3	17.3	-1.0	23.9	-4.4	-	-
November	8.9	2.4	13.9	-3.4	19.4	-10.6	-	-
December	7.0	1.1	12.6	-5.6	18.3	-13.9	-	-
Year	11.8	4.4	26.5	-9.6	31.1	-16.1	-	-

Table IV-5.

Maximum Number of Consecutive days of cold weather with daily mean temperatures below of the values shown

25 Years Period from 1925 to 1950	Temperature °C																
Regions	0	-0.5	-1.0	-1.5	-2.0	-2.5	-3.0	-3.5	-4.0	-4.5	-5.0	-5.5	-6.0	-6.5	-7.0	-7.5	-8.0
Severn Valley (Bristol 51° 31'N)	16	16	14	8	7	-	7	4	4	4	3	3	3	3	-	1	-
Thames Valley (London 51° 28'N)	14	11	10	8	6	-	4	4	4	4	3	3	1	1	-	-	-
Midlands (Birmingham 52° 26'N)	22	-	15	10	10	-	6	5	5	5	5	3	3	1	1	-	-
West Pennines (Manchester 53° 22'N)	14	9	9	7	6	-	5	5	4	4	3	2	1	1	-	-	-
Northern Ireland (Aldergrove 54° 40'N)	9	9	-	8	8	7	7	7	6	5	1	1	1	-	-	-	-
West and East Scotland (Edinburgh 55° 55'N)	28	11	6	5	5	-	4	3	3	1	1	1	-	-	-	-	-
North East Scotland (Aberdeen 57° 10'N)	10	10	9	8	5	-	5	3	2	1	1	1	-	-	-	-	-

Fig 24. Average Winter Temperature for the U.K. (20 years period)

Taken from Ref 48 (Department of Energy) internal design temperature, during those cold spells which occur during the average winter when heat loss calculations are based on the external design temperatures quoted. The operation modes of the system given in the present study will permit the possibility of designing the heat pump with a source temperature T_1 higher than 0°C. e.g., an ambient temmperature of $\sim$ 2°C gives an air temperature of 4°C at the evaporator. The average winter temperatures for the U.K. are given in Figure 24 (Ref. 48).

Table IV-6. Winter External Temperatures (U.K.)

Characteristics		External Design Temp. °C
Type of Building	System overload capacity %	
Multi-storey building with solid intermediate floors and partitions	20	-1
	Nil	-4
Single-storey buildings	20	-3
	Nil	-5

IV-6 Solar Radiation Characteristics and U.K. Solar Data

In order to design the solar assisted heat pump system it is necessary to have adequate knowledge of solar insolation characteristics in the region in which the system is to be situated, so that the amount of energy actually available at the various orientations of the solar absorber (or roof) can be determined.

IV-6.1. Solar Radiation at Outer Limit of Atmosphere

IV-6.1.1 Intensity of Solar Radiation

The intensity of solar radiation, incident upon a surface placed at the outer limit of the atmosphere, expressed in J/hm^2, may be written (49) as:

$$Io = I_N \cdot \text{Cos}\ \theta$$

$$Io = rI_{SC} \cdot \text{Cos}\ \theta \qquad \text{(IV-1)}$$

Where:

I_{SC} = Solar constant, i.e. the intensity of radiation reaching the upper limits of the atmosphere (1353 W/m^2) over a surface normal to the sun's rays. (49, 50, 51).

r = The ratio of extra terrestrial radiation to the nominal Solar Constant. This ratio is function of the mean distance to the actual distance between earth and sun (r max = 1.034 in Jan. and r min = 0.967 in July) (50).

θ = Angle of incidence (Angle between the sun's rays and the normal to the surface under consideration).

If a surface (or roof) is tilted at an angle β from the horizontal plane toward the equator, then the incidence angle θt can be calculated from the following equation (49, 50, 51).

$$\cos\ \theta t = \cos\ (1-\beta)\ \cos\delta \cos\ h + \sin\ (1-\beta)\ \sin\delta \qquad \text{(IV-2)}$$

Where:

l = Latitude (North positive)

δ = Solar declination (i.e. the angular position of the sun at solar noon with respect to the plane of the equator.

h = Hour angle: Angle between meridian plane passing through the sun and the meridian plane passing through the location.

If the surface is horizontal (flat roof), $\beta = 0$, so the equation (IV - 2) is reduced to:

$$\cos\theta_h = \cos l \cos\delta \cos h + \sin l \sin\delta \qquad \text{(IV-3)}$$

Figure 25 shows definitions of the angles: l, β, δ, h and θ

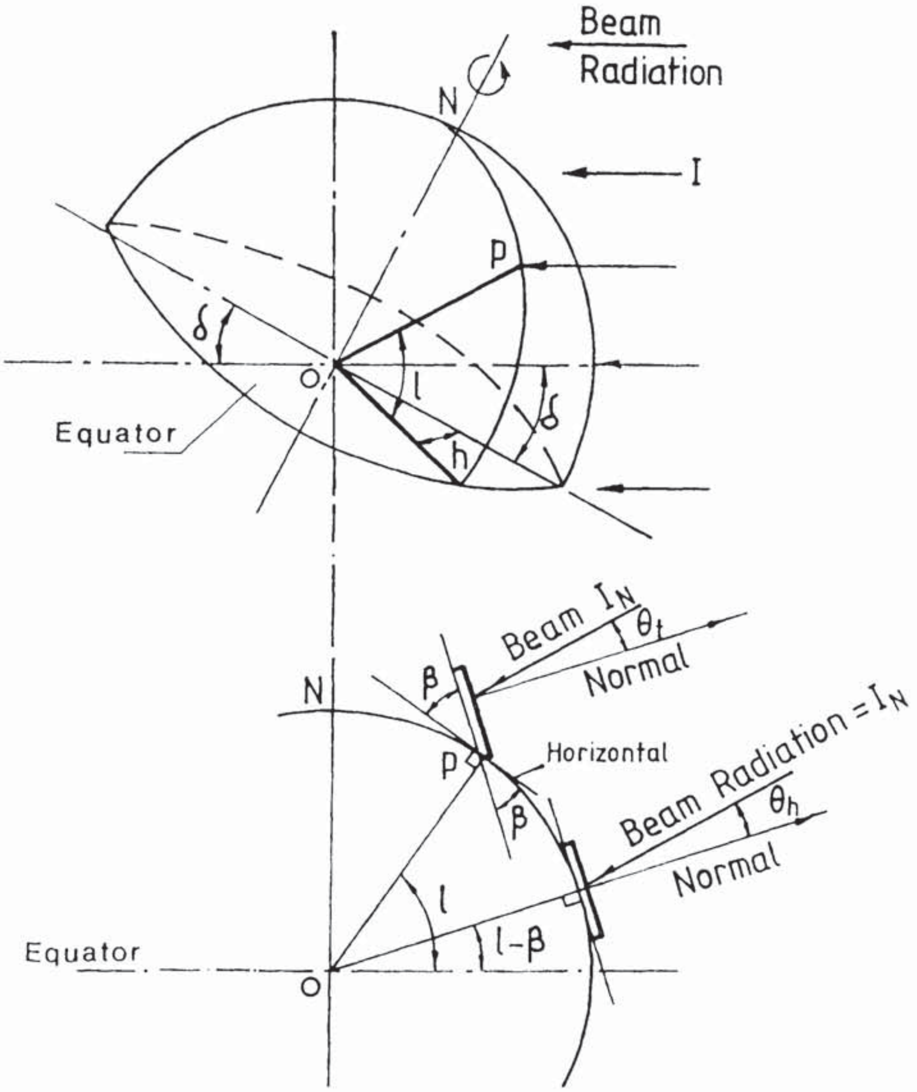


Fig. 25:- Section of earth showing definitions of β, θ, l, δ and h.

IV-6.1.2 <u>Daily and Hourly Radiation upon Horizontal and Tilted Surfaces</u>

The radiation intensity I_0 may be integrated with respect to time to obtain the total energy incident upon a surface during a specified time interval.

1) <u>Daily Radiation:</u>

a) The daily total radiation Ho_h incident upon a horizontal surface of unit area, expressed in J/m^2 day, will be:

$$d\,Ho_h = Io\,d\tau = r\,I_{SC}\cos\theta_h \cdot d\tau$$

Where: Io = Expressed in $J/h.m^2$

$d\tau$ = $(1/\omega)\,dh$

ω = Earth's angular velocity = $(2\pi/24)$ radians/h

dh = The change in hour angle (in radians) during time $d\tau$ (in hours)

$$dHo_h = r\,I_{SC}\cdot\cos\theta_h \cdot d\,(h/\omega)$$

$$dHo_h = r\,I_{SC}\left[\cos 1\cos\delta\cos h + \sin 1\sin\delta\right] d\,(h/\omega)$$

$$\int_o^{Ho} dHo_h = \int r\,I_{SC}\,(\cos 1\cos\delta\cos h + \sin 1\sin\delta)\,d\,({}^{24h}/2\pi)$$

$$Ho_h = r\,I_{SC}\,\frac{24}{2\pi}\int_{-h_s}^{+hs}(\cos 1\cos\delta\cos h + \sin 1\sin\delta)\,dh$$

$$Ho_h = r\,I_{SC}\,\frac{24}{2\pi}\left[\cos 1\cos\delta\sin h + h\sin 1\sin\delta\right]_{-h_s}^{+h_s}$$

$$Ho_h = r\,I_{SC}\,\frac{24}{2\pi}\Big\{(\cos 1\cos\delta\sin hs + hs\sin 1\sin\delta) - (\cos 1\cos\delta\sin(-hs) - hs\sin 1\sin\delta)\Big\}$$

$$Ho_h = r\,I_{SC}\,\frac{24}{2\pi}\Big\{(\cos 1\cos\delta\sin hs + hs\sin 1\sin\delta) + (\cos 1\cos\delta\sin hs + hs\sin 1\sin\delta)\Big\}$$

$$Ho_h = r\, I_{SC}\, \frac{24}{\pi} \left[\cos l \cos\delta \sin h_s + h_s \sin l \sin\delta\right] \qquad \text{(IV-4)}$$

The limits of integration are $\pm h_s$; where h_s is the sunset angle or from equation IV-3, with $\cos\theta_h = 0$:

$$0 = \cos l \cos\delta \cos h_s + \sin l \sin\delta$$

$$\cos h_s = -\frac{\sin l \sin\delta}{\cos l \cos\delta} = -\tan l \tan\delta$$

$$\cos h_s = -\tan l \tan\delta \qquad \text{(IV-5)}$$

b) Similarly, the daily radiation Ho_t incident upon a surface tilted at an angle β from the horizontal plane toward the equator will be given by:

If, $h_s \leqslant h_s'$,

$$Ho_t = \frac{24}{\pi}\, r\, I_{SC} \left[\cos(l-\beta) \cos\delta \sin h_s + h_s \sin(l-\beta) \sin\delta\right] \qquad \text{(IV-6)}$$

Or if $h_s' \leqslant h_s$, the equation of Ho_t will be:

$$Ho_t = \frac{24}{\pi}\, r\, I_{SC} \left[\cos(l-\beta) \cos\delta \sin h_s' + h_s' \sin(l-\beta) \sin\delta\right] \qquad \text{(IV-7)}$$

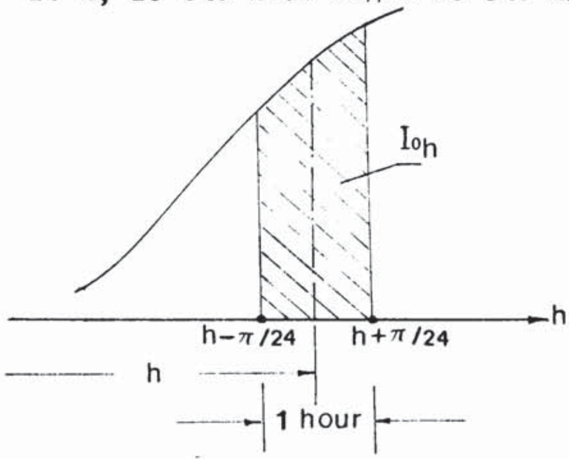
Equations (IV-5) and (IV-6) expressed in J/m^2.day and in which, by setting equation (VI-2) to zero,

$$\cos h_s' = -\tan(l-\beta) \tan\delta \qquad \text{(IV-8)}$$

2) Hourly Radiation

a) By similar integration to the previous cases the hourly radiation Io_h, on a horizontal surface, expressed in $J/m^2.h$, will be:

If h, is the hour angle at the mid-point of the hour.



1 hour = 15° $= \frac{\pi}{12}$ (rad.)

½ hour = 7.5° $= \frac{\pi}{24}$ (rad.)

h = radians.

$$\int_o^{Io_h} d\ Io_h = \int Io\ .\ d\tau = \int Io \cdot d\left(\frac{1}{\omega}\ h\right)$$

$$Io_h = \int_{h-(\pi/24)}^{h+(\pi/24)} r.I_{SC}\ .\ \cos\theta_h\ .\ d\left(\frac{h}{2\pi/24}\right)$$

$$Io_h = r.I_{SC}\ .\ \frac{24}{2\pi}\int_{h-(\pi/24)}^{h+(\pi/24)} (\cos l\ \cos\delta\cos h + \sin l\ \sin\delta)\ dh$$

$$Io_h = \left[r.I_{SC}.\frac{24}{2\pi}\right]\left[\cos l\ \cos\delta\ \sin h + h\ \sin l.\ \sin\delta\right]_{h-(\pi/24)}^{h+(\pi/24)}$$

$$Io_h = \left[r.I_{SC}.\frac{24}{2\pi}\right]\left\{\left[\cos l \cos\delta \sin (h + (\pi/24)) + (h + (\pi/24)) \sin l \sin\delta\right] - \left[\cos l \cos\delta \sin (h - (\pi/24)) + (h - (\pi/24)) \sin l \sin\delta\right]\right\}$$

$$Io_h = \left[r.I_{SC}.\frac{24}{2\pi}\right]\left\{\cos l \cos\delta\left[\sin (h + (\pi/24)) - \sin (h - (\pi/24))\right] + \sin l \sin\delta\left[h + (\pi/24) - h + (\pi/24)\right]\right\}$$

Since:

$$\sin (h + (\pi/24)) = \sin h \cos (\pi/24) + \cos h \sin (\pi/24), \text{ and}$$

$$\sin (h - (\pi/24)) = \sin h \cos (\pi/24) - \cos h \sin (\pi/24),$$

Then, the difference between these two will be:

$$\sin (h + (\pi/24)) - \sin(h-(\pi/24)) = 2 \cos h \sin (\pi/24)$$

By replacing this result into the last expression of Io_h one will have:

$$Io_h = rI_{SC}\left[(24 / \pi) \sin (\pi/24) \cos l \cos\delta \cos h + \sin l \sin\delta\right]$$

but,

$(24/\pi)(\sin(\pi/24)) = 0.9972 \simeq 1$, Then:

$$Io_h = rI_{SC}\left[\cos l \cos\delta \cos h + \sin l \sin\delta\right] \qquad \text{(IV-9)}$$

b) Similarly, the hourly radiation Io_t, on a surface (or a roof) tilted at an angle β from the horizontal plane toward the equator, expressed in $J/m^2.h$, can be written as:

$$Io_t = rI_{SC}\left[\cos(1-\beta)\cos\delta\cos h + \sin(1-\beta)\sin\delta\right] \quad \text{(IV-10)}$$

IV-6.2 Ratios between Daily and Hourly Radiation

Useful ratios can be obtained by means of the previous equations.

IV-6.2.1 Io_h/Ho_h

$$\frac{Io_h}{Ho_h} = \frac{rI_{SC}\left[\cos 1\cos\delta\cos h + \sin 1\sin\delta\right]}{rI_{SC}\,(24/\pi)\left[\cos 1\cos\delta\sin h_s + h_s\sin 1\sin\delta\right]}$$

$$\frac{Io_h}{Ho_h} = \frac{\pi}{24}\left[\frac{\cos h + \tan 1\tan\delta}{\sin h_s + h_s\tan 1\tan\delta}\right]$$

From equation IV-5: $\cos h_s = -\tan 1\tan\delta$, then the ratio ($Io_h/Ho$) becomes:

$$\frac{Io_h}{Ho_h} = \frac{\pi}{24}\left[\frac{\cos h - \cos h_s}{\sin h_s - h_s\cos h_s}\right] \quad \text{(IV-11)}$$

IV-6.2.2 Ho_t/Ho_h = Solar absorber tilt factor

a) If $h_s \leqslant h_s'$

$$\frac{Ho_t}{Ho_h} = \frac{(24/\pi)(rI_{SC})\left[\cos(1-\beta)\cos\delta\sin h_s + h_s\sin(1-\beta)\sin\delta\right]}{(24/\pi)(rI_{SC})\left[\cos 1\cos\delta\cos h_s + h_s\sin 1\sin\delta\right]}$$

$$\frac{Ho_t}{Ho_h} = \frac{\cos(1-\beta)\cos\delta\left[\sin h_s + h_s \tan(1-\beta)\tan\delta\right]}{\cos\delta \,.\, \cos 1\left[\cos 1 \cos h + h_s \tan 1 \tan\delta\right]}$$

From equation IV-5, $\cos hs = -\tan 1 \times \tan\delta$ and from equation IV-8, $\cos h'_s = -\tan\ (1-\beta)\tan\delta$, so the ratio (Ho_t/Ho_h) for $h_s \leqslant h_s'$ will be:

$$\frac{Ho_t}{Ho_h} = \frac{\cos(1-\beta)}{\cos 1}\left[\frac{\sin h_s - h_s \cos h_s'}{\sin h_s - h_s \cos h_s}\right] \qquad \text{(IV-12)}$$

b) Similarly if $h_s' \leqslant h_s$:

$$\frac{Ho_t}{Ho_h} = \frac{\cos(1-\beta)\cos\delta \sin h_s' + h_s' \sin(1-\beta)\sin\delta}{\cos 1 \cos\delta \sin h_s + h_s \sin 1 \sin\delta}$$

$$\frac{Ho_t}{Ho_h} = \frac{\cos(1-\beta)\cos\delta}{\cos\delta\cos 1}\left[\frac{\sin h_s' + h_s' \tan(1-\beta)\tan\delta}{\sin h_s + h_s \tan 1 \tan\delta}\right]$$

And finally with $\tan(1-\beta)\tan\delta = -\cos h_s'$ and $\tan\alpha \,.\, \tan\beta = -\cos h_s$, this ratio, for $h_s' \leqslant h_s$, becomes:

$$\frac{Ho_t}{Ho_h} = \frac{\cos(1-\beta)}{\cos 1}\left[\frac{\sin h_s' - h_s' \cos h_s'}{\sin h_s - h_s \cos h_s}\right] \qquad \text{(IV - 13)}$$

Table IV-7, which has been generated in this study gives the extra-terrestrial solar radiation (Ho_h) on a horizontal surface of unit area for the U.K. latitudes from 50°N to 60°N. The Ho_h values were calculated by means of the equation (IV-4), using solar declination values, shown in the table, for the 15th days of each month. The values

Table IV-7. Extraterrestrial Solar Radiation (Ho_h) at the U.K. Latitudes (1°N)

δ = Solar declination (51). τ_N = Total number of hours between sunrise and sunset $= \frac{2h_s°}{15}$

r = Ratio of extraterrestrial Radiation to the nominal solar constant (50)

Ho_h = Daily Radiation on a Horizontal surface outside atmosphere = Wh/m^2 day. (Equation IV-4)

Month	δ*	r*	1 = 50°N		1 = 52°N		1 = 54°N		1 = 56°N		1 = 58°N		1 = 60°N	
			τ_N	Ho_h	τ_N	Ho_h	τ_N	Ho_h	τ_N	Ho_h	τ_N	Ho_h	τ_N	
Jan	-21.3	1.030	8.31	2514	8.01	2181	7.67	1875	7.29	1556	6.85	1222	6.33	958
Feb	-13.3	1.020	9.82	4111	9.65	3722	9.46	3389	9.26	3056	9.04	2722	8.78	2361
Mar	-2.8	1.006	11.55	6439	11.52	5833	11.48	5536	11.44	5223	11.40	4972	11.35	4653
Apr	9.4	0.988	13.52	8750	13.63	8528	13.76	8334	13.89	8110	14.05	7889	14.22	7639
May	18.8	0.975	15.19	10569	11.44	10417	15.72	10278	16.04	10167	16.40	10083	16.81	10000
June	23.3	0.968	16.12	11528	16.46	11472	16.84	11444	17.29	11389	17.80	11333	18.43	11291
July	21.5	0.967	15.73	11056	16.03	10944	16.37	10861	16.76	10833	17.21	10764	17.74	10694
Aug	13.8	0.976	14.26	9444	14.44	9167	14.63	9028	14.84	8889	15.09	8722	15.35	8528
Sept	2.2	0.991	12.34	6125	12.37	6675	12.40	6389	12.43	6110	12.47	5834	12.50	5556
Oct	-9.6	1.010	10.44	4639	10.33	4278	10.20	3945	10.06	3611	9.91	3278	9.72	3000
Nov	-19.2	1.025	8.73	2778	8.47	2472	8.18	2112	7.85	1778	7.48	1472	7.05	1195
Dec	-23.3	1.032	7.88	2139	7.45	1792	7.15	1500	6.71	1195	6.19	917	5.57	625

* For the midpoints of each month.

of r, for the midpoints of each month, were extracted from the figure No.2.9 given in reference (50). Table IV-7, also gives the possible total number of hours between sunrise and sunset at the U.K. latitudes. The hour-angles h_s were calculated by using the equation IV-5 and the total number of hours from the expression $\tau_N = 2h_s^{\circ}/15$.

Table IV-8, also developed in this study, shows the solar absorber tilt factor, or ratio Ho_t/Ho_h, at the U.K. latitudes (50°N to 60°N), for various β absorber tilt angles. Angles h_s and h_s' were calculated by equationsIV-5 and IV-8, and the Ho_t/Ho_h values from the equations IV-12 and IV-13.

IV-6.3 Solar Radiation at the Surface of the Earth

The previous ratios are for extraterrestrial radiation, but in qualitative form they remain the same when the effect of the earth's atmosphere is taken into consideration.

When solar radiation passes through the atmosphere, part of it may be scattered by constituents such as dry air molecules. Water molecules and dust particles. Another part of the radiation may be absorbed particularly by ozone and by the water vapour present in the atmosphere. The remaining part of the original direct radiation quantity reach the earth's surface unchanged in wavelength.

Some of the radiation intercepted by the atmosphere also reaches the earth's surface but in a different wavelength. This radiation of diffuse nature comes from the entire sky region.

Table IV-8. Solar Absorber Tilt Factor at the U.K. Latitudes

Ho_t/Ho_h = Tilt factor = Ratio of dailv radiation, Ho_t, on a surface tilted at β degrees from horizontal towards south to daily radiation Ho_h on a horizontal surface .(From equations IV-12 and IV-13)
δ = Solar declination for the 15th day of each month.

Month	δ	1° North Latitude							
		50°N		52°N	54°N	56°N	58°N	60°N	
		$\beta = 50°$	$\beta = 70°$	$\beta = 52°$	$\beta = 54°$	$\beta = 56°$	$\beta = 58°$	$\beta = 60°$	$\beta = 70°$
		Hot/Ho_h (Tilt Factor)							
January	-21.3	3.56	3.94	4.10	4.70	5.52	6.65	8.31	8.82
February	-13.3	2.49	2.62	2.71	2.96	3.27	3.65	4.12	4.28
March	-2.8	1.65	1.62	1.79	1.89	2.01	2.13	2.28	2.29
April	9.4	1.17	1.00	1.20	1.23	1.26	1.30	1.34	1.26
May	18.8	0.90	0.64	0.91	0.92	0.93	0.94	0.95	0.84
June	23.3	0.80	0.56	0.80	0.80	0.80	0.80	0.81	0.70
July	21.5	0.84	0.62	0.84	0.84	0.85	0.85	0.86	0.76
August	13.8	1.02	0.83	1.05	1.07	1.09	1.11	1.13	1.04
September	2.2	1.44	1.32	1.51	1.57	1.64	1.72	1.80	1.76
October	-9.6	2.10	2.14	2.32	2.50	2.71	2.97	3.28	3.35
November	-19.2	3.16	3.32	3.62	4.10	4.70	5.51	6.62	6.99
December	-23.3	4.04	4.52	4.67	5.47	6.58	8.23	10.88	11.61

Thus, the surface of the earth receives solar energy in two forms: Direct beam radiation and Diffuse radiation. The reduction of direct solar radiation by the atmosphere is large even during clear days, while with heavy cloud cover almost complete extinction of the direct beam may occur. Thus, the total radiation as measured on a horizontal plane, must be divided into direct and diffuse components.

Meteorological records usually provide the hourly average or hourly total (I_{Th}) intensities, measured on a horizontal plane ($Wh/m^2.h$). Some solar radiation recording stations, measure the hourly total radiation, and the hourly diffuse component (Id_h) separately, but most of them measure only the hourly total horizontal flux. Data on diffuse horizontal radiation is very rare and it is, therefore, necessary to resort to developed correlations (49), between the widely available total and the rarely available diffuse radiation data.

Stations actually measuring and publishing hourly data are rather few; often one can only find daily totals over horizontal surfaces H (Wh/m^2 day), or possible one average day's total for each month, or the total cummulative amount received per month. For this reason it is very common to design solar systems based on long-term monthly averages of radiation. Monthly average daily radiation is quite often written as average daily radiation: $\bar{H}$ = Direct beam $\bar{B}$ + Diffuse $\bar{D}$, which is measured over a horizontal plane.

IV-6.3.1 <u>Other two useful ratios (49) may be established:</u>

a) $K_T = \bar{H}/Ho_h$, which is an index representing the fraction of extraterrestrial radiation arriving at the surface of the earth. Its variation is from 0.2 to a maximum of approximately 0.75, with most data within the range of 0.3 to 0.6 in the U.K.

b) $K_D = \overline{D}/Ho_h$, which is the fraction of extraterrestrial radiation reaching the surface of the earth in diffuse form. In these relations:

$\overline{H}$ = Monthly average daily total radiation on a horizontal surface at the earth level (J/day.m^2).

$\overline{D}$ = Monthly average daily diffuse radiation on a horizontal surface (J/day.m^2)

The results in table IV-9, have been calculated from values given by Liu and Jordan (49), and shows the relationship between K_T and K_D and the ratio $\overline{H}/\overline{D}$, which can be used to estimate the monthly average daily diffuse radiation $\overline{D}$ with a reasonable degree of accuracy. Values of this table were plotted in the Figure 26.

Table IV-9. Relation between K_T, K_D and $\overline{D}/\overline{H}$

$K_T = \overline{H}/Ho_h$	0.2	0.3	0.4	0.5	0.6	0.7	0.75
$K_D = \overline{D}/Ho_h$	0.172	0.178	0.183	0.188	0.174	0.149	0.125
$\overline{D}/\overline{H}$	0.860	0.594	0.457	0.376	0.290	0.213	0.167

Figure 27 shows the daily average solar radiation $\overline{H}$ incident on a horizontal surface in the U.K. for a latitude of 51° 28'N. It can be seen from this figure that about 60% of the total solar radiation arrives in diffuse form and that about 60% of the total solar radiation is incident during the summer season.

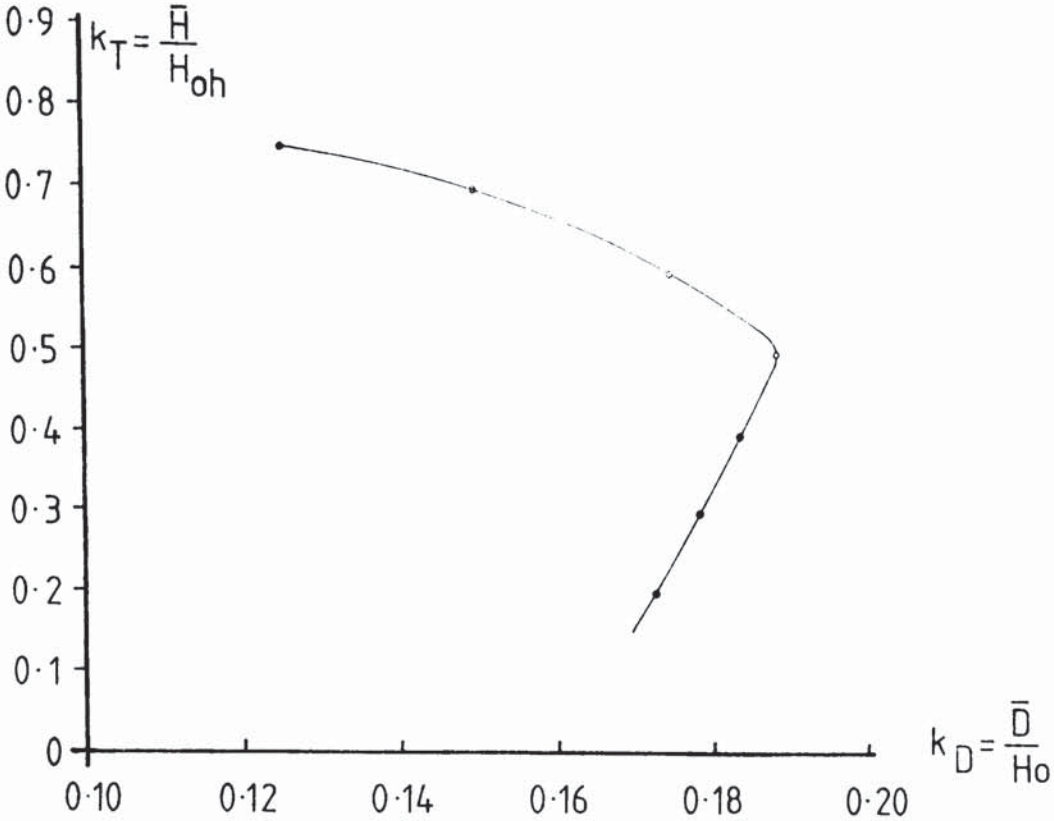


Fig 26 Relation between Monthly-Average Daily Diffuse Radiation $\bar{D}$, and the Monthly-Average Daily Total Radiation $\bar{H}$, on a Horizontal Surface

Monthly average daily totals ($\bar{H}$ = Direct beam $\bar{B}$ + Diffuse $\bar{D}$) of solar radiation over the British Isles are published by the Meteorological Office (52) and these values, in form of maps, are annexed to this study. The average radiation's values given in the maps cover the period from 1941 to 1970.

Table IV-10, prepared in this study, gives the monthly average daily radiation $\bar{H}$, on a horizontal surface at the U.K. latitudes, extracted from reference (52) for each month of the year, and also gives the estimated monthly average values of daily diffuse solar radiation $\bar{D}$ on a horizontal surface for the U.K. latitudes. Values of diffuse radiation $\bar{D}$ were calculated as a function of the ratio $K_D = \bar{D}/Ho_h$, extracted from Figure 26 for the different calculated values of $K_T = \bar{H}/Ho$. The extraterrestrial daily radiation (Ho_h)

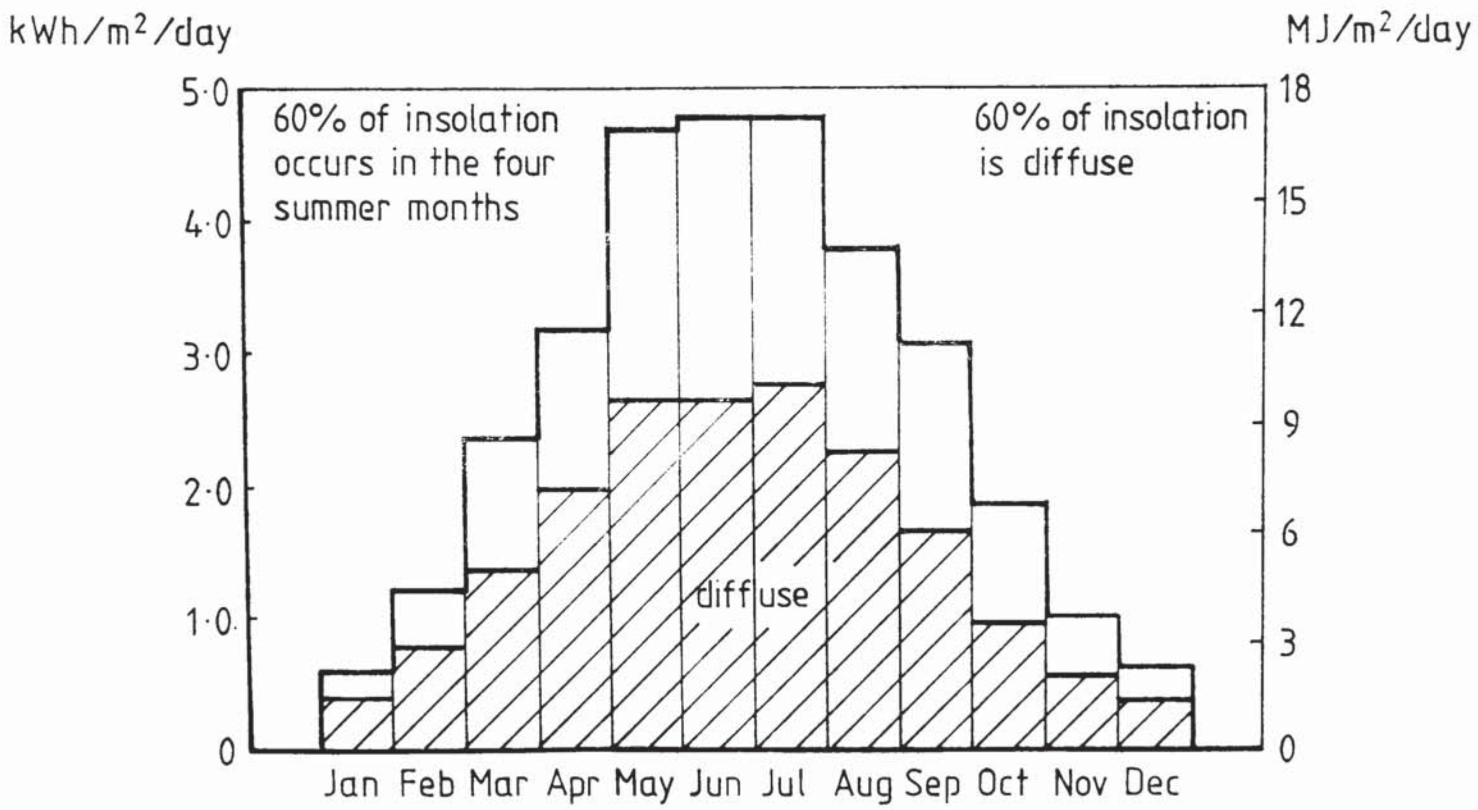


Fig 27 Daily average insolation on horizontal (UK latitude 51° 28′ N) plate (Ref 30)

Table IV-10. Monthly Average Daily Diffuse Radiation $\bar{D}$, and Monthly Average Daily Total Radiation $\bar{H}$, on a horizontal surface, at the U.K. Latitudes.

$\bar{H}$ = Wh/m^2 day. Extracted from Reference (52), $\bar{D} = K_D \times Ho_h = Wh/m^2$ day; $Ho_h = Wh/m^2$ day from Table IV-7.

$K_D = \bar{D}/Ho_h$ calculated from Figure 26 as a function of K_T; 1 = Latitude North.

Month	1 = 50°N				1 = 52°N				1 = 54°N			
	$\bar{H}$	$K_T = \bar{H}/Ho_h$	$K_D = \bar{D}/Ho_h$	$\bar{D}$	$\bar{H}$	$K_T = \bar{H}/Ho_h$	$K_D = \bar{D}/Ho_h$	$\bar{D}$	$\bar{H}$	$K_T = \bar{H}/Ho_h$	$K_D = \bar{D}/Ho_h$	$\bar{D}$
Jan.	700	0.28	0.177	445	600	0.28	0.177	386	500	0.27	0.176	330
Feb.	1500	0.36	0.181	744	1200	0.32	0.179	666	1100	0.32	0.179	607
March	2600	0.40	0.183	1178	2200	0.38	0.182	1062	2000	0.36	0.181	1002
April	3800	0.43	0.185	1619	3200	0.38	0.182	1552	3200	0.38	0.182	1517
May	4800	0.45	0.186	1966	4400	0.42	0.184	1917	4200	0.41	0.183	1881
June	5600	0.49	0.187	2156	5000	0.44	0.185	2122	4800	0.42	0.184	2106
July	5000	0.45	0.186	2056	4600	0.42	0.184	2014	4400	0.41	0.183	1987
Aug.	4000	0.42	0.184	1738	3800	0.41	0.183	1678	3600	0.40	0.183	1652
Sept.	3100	0.49	0.187	1145	2800	0.42	0.184	1228	2600	0.41	0.183	1169
Oct.	1900	0.41	0.183	849	1700	0.40	0.183	783	1500	0.38	0.182	718
Nov.	1000	0.36	0.181	503	800	0.32	0.179	442	700	0.33	0.179	378
Dec.	600	0.28	0.177	379	500	0.28	0.177	317	400	0.27	0.176	264

Table IV-10. Continued

Monthly Average Daily Diffuse Radiation D, and Monthly Average Daily Total Radiation H, on a horizontal surface, at the U.K. Latitudes.

$\bar{H}$ = Wh/m^2 day. Extracted from Reference (52), $\bar{D} = K_D \times Ho_h$ = Wh/m^2 day; Ho_h = Wh/m^2 day from Table IV-7.

$K_D = \bar{D}/Ho_h$ calculated from Figure 26 as a function of K_T; 1 = Latitude North.

	1 = 56°N				1 = 58°N				1 = 60°N			
Month	$\bar{H}$	$K_T = \bar{H}/Ho_h$	$K_D = \bar{D}/Ho_h$	$\bar{D}$	$\bar{H}$	$K_T = \bar{H}/Ho_h$	$K_D = \bar{D}/Ho_h$	$\bar{D}$	$\bar{H}$	$K_T = \bar{H}/Ho_h$	$K_D = \bar{D}/Ho_h$	$\bar{D}$
Jan.	400	0.26	0.176	274	300	0.24	0.174	213	300	0.31	0.178	170
Feb.	1100	0.36	0.181	553	1000	0.37	0.181	493	900	0.38	0.182	430
March	2000	0.38	0.182	950	1800	0.36	0.181	900	1800	0.39	0.182	847
April	3200	0.39	0.182	1476	3200	0.41	0.183	1444	3200	0.42	0.184	1405
May	4000	0.39	0.182	1850	4000	0.40	0.183	1845	4000	0.40	0.183	1830
June	4600	0.40	0.183	2084	4400	0.39	0.182	2063	4400	0.39	0.182	2055
July	4200	0.39	0.182	1972	4000	0.37	0.181	1948	4000	0.37	0.181	1936
Aug.	3400	0.38	0.182	1618	3200	0.37	0.181	1579	3200	0.37	0.181	1543
Sept.	2400	0.39	0.182	1112	2300	0.39	0.182	1062	2100	0.38	0.182	1011
Oct.	1400	0.39	0.182	657	1200	0.37	0.181	593	1000	0.33	0.179	537
Nov.	600	0.34	0.180	320	500	0.34	0.180	265	400	0.33	0.179	214
Dec.	300	0.25	0.175	209	200	0.22	0.173	159	200	0.32	0.179	112

values were taken from table IV-7.

$$\bar{D} = k_D \, Ho_h = (K_D/K_T)\, \bar{H} = (\bar{D}/Ho_h)\, (Ho_h/\bar{H})\, \bar{H} = (\bar{D}/\bar{H})\, \bar{H} \qquad \text{(IV-14)}$$

$\bar{D}$ = Watt.hour/m^2 . day

IV-6.3.2 Estimation of Average Solar Radiation

Whenever information of $\bar{H}$ is not available, it can be obtained through its correlations with hours of bright sunshine, e.g. daily total radiation at the earth level can be estimated by using the expression given by Glover and McCulloch (53) for the U.K. latitudes:

$$\bar{H} = H_{SC}\,(0.29.\cos 1 + 0.52\ {}^{\tau}n/{}^{\tau}N) \qquad \text{(IV-15)}$$

Where:

$\bar{H}$ = Average Daily total (Direct + Diffuse) solar radiation on a horizontal plane (Wh/m^2.day)

H_{SC} = U.K. Solar constant per day ≃9830 Wh/m^2.day

1 = Degress of Geographical latitude (°N)

$^{\tau}N$ = Maximum possible sunshine hours per day (hours)

$^{\tau}n$ = Average actual sunshine hours per day (hours)

For 52° North Latitude (London) one have:

$$\bar{H} = 9830\,(0.29 \cos 52 + 0.52\ {}^{\tau}n/\ {}^{\tau}N)$$

$$\bar{H} = 1755 + 5111\ {}^{\tau}n/\ {}^{\tau}N = \text{Wh/m}^2\text{.day} \qquad \text{(IV-16)}$$

Table IV-11, extracted from reference (44) gives values of average number of hours of bright sunshine a day for the British Isles, based on a 30 year period from 1921 to 1950. Table IV-11a, derived from data on reference (45) gives values of bright sunshine for three locations in the U.K. and also gives the average per cent of possible bright sunshine hours ($\tau n/\tau N$) which can be used in equation IV-15, to calculate the $\bar{H}$ values in those latitudes.

IV-6.3.3 <u>Hourly Radiation to Daily Radiation Ratios</u>

Due to diurnal motion of the sun in the sky it is often desirable that the performance of a solar absorber be evaluated on an hour to hour basis, with the absorber following the path of the sun. Thus, hourly radiation values are more useful than daily radiation values to make performance calculation of solar processes. However, flat solar collectors and roofs acting as solar absorbers are always placed in a fixed orientation and furthermore, only daily radiation data is generally available. For this reason, it is often necessary to start with daily data and estimate hourly values from the daily numbers. It must be recognised that this is not an exact process.

Statistical studies of the time distribution of total radiation on horizontal surfaces through the day, using average data from a number of stations, have led to generalised charts of the ratio of hourly total to daily total radiation as a function of day length and the hour in question. Figure 28 shows such a chart from Liu and Jordan (1967) extracted, for this study, from references (49) and (51). Day length (between sunrise and sunset) as a function of latitude l and declination δ can be calculated by $\tau_N = 2h_s°/15$.

Table IV-11. Average number of hours of bright sunshine a day for the British Isles on a 30 year period from 1921 to 1950. Extracted from reference (44)

	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEP	OCT	NOV	DEC	Total for year	Daily Average
Aberdeen	1.5	2.4	3.4	4.5	5.5	6.1	4.9	4.6	4.1	3.1	1.9	1.2	1317	3.6
Bath	1.6	2.5	4.0	5.2	6.2	7.0	6.1	5.8	4.7	3.3	1.9	1.5	1514	4.1
Birmingham	1.3	2.0	3.2	4.4	5.4	6.1	5.4	5.2	4.0	2.9	1.6	1.2	1302	3.6
Blackpool	1.4	2.3	3.8	5.5	6.7	7.1	5.9	5.5	4.3	3.1	1.8	1.2	1480	4.1
Bournemouth	2.0	2.8	4.4	5.9	6.9	7.7	6.8	6.6	5.1	3.7	2.4	1.9	1711	4.7
Bradford	0.9	1.7	2.8	4.3	5.3	6.3	5.4	4.9	3.7	2.5	1.3	0.8	1217	3.3
Brighton	2.0	3.0	4.8	5.6	7.2	7.7	7.2	6.8	5.7	3.9	2.4	1.8	1766	4.8
Cambridge	1.6	2.4	3.8	4.9	6.1	6.8	6.1	5.8	4.6	3.4	1.9	1.3	1488	4.1
Cardiff	1.7	2.6	4.0	5.5	6.4	7.2	6.2	6.0	4.8	3.4	2.1	1.6	1566	4.3
Dover	1.8	2.8	4.7	5.2	7.0	7.9	7.2	6.8	5.4	3.7	2.1	1.7	1721	4.7
Dublin	1.8	2.5	3.5	5.0	6.0	6.0	4.9	4.9	4.1	3.1	2.3	1.5	1390	3.8
Dundee	1.6	2.7	3.4	4.7	5.4	6.1	5.0	4.6	4.1	3.1	2.1	1.3	1335	3.7
Durham	1.5	2.3	3.4	4.5	5.3	6.0	5.1	4.8	4.0	3.1	2.0	1.4	1325	3.6
Edinburgh	1.7	2.7	3.7	4.6	5.6	6.5	5.2	4.6	4.2	3.2	2.1	1.4	1385	3.8
Falmouth	1.9	2.6	4.3	5.9	6.8	7.7	6.4	6.3	5.1	3.6	2.5	1.8	1672	4.6
Felixstowe	1.9	2.7	4.5	5.7	7.2	7.6	7.1	6.6	5.4	3.8	2.2	1.7	1720	4.7
Fort William	0.4	1.6	2.7	3.6	5.6	5.1	3.7	3.8	2.7	1.8	0.9	0.2	981	2.7
Ilffracombe	1.5	2.4	4.3	5.6	6.7	7.5	6.2	6.0	4.7	3.2	1.9	1.3	1566	4.3
Inverness	1.5	2.4	3.5	4.4	5.3	5.6	4.5	4.4	3.6	2.8	1.7	1.0	1239	3.4
Littlehampton	2.1	2.9	4.6	5.9	7.3	8.0	7.2	6.9	5.5	3.9	2.5	2.0	1792	4.9
Liverpool	1.6	2.3	3.7	5.3	6.4	7.0	5.9	5.5	4.4	3.1	1.9	1.3	1476	4.1
London (Greenwich)	1.1	1.8	3.4	4.4	5.9	6.6	6.2	5.8	4.5	3.1	1.4	0.9	1378	3.8
Luton	1.5	2.3	3.9	4.8	6.1	6.8	6.2	5.9	4.7	3.4	1.9	1.4	1486	4.1
Malin Head	1.2	2.1	3.6	5.2	6.5	5.7	4.6	4.6	3.8	2.5	1.8	1.0	1297	3.5
Manchester	0.7	1.3	2.6	3.9	5.1	5.5	4.5	4.5	3.4	2.1	1.0	0.5	1071	2.9
Margate	1.8	2.8	4.6	6.0	7.3	7.9	7.7	7.0	5.7	3.8	2.1	1.5	1771	4.9
Norwich	1.7	2.4	4.2	5.2	6.7	6.8	6.6	6.1	4.9	3.6	1.9	1.5	1572	4.3
Nottingham	1.3	1.9	3.1	4.3	5.4	6.0	5.5	5.1	4.1	2.9	1.7	1.1	1289	3.5
Oxford	1.7	2.5	4.0	5.0	5.8	6.7	5.9	5.7	4.5	3.4	2.1	1.6	1482	4.1
Plymouth	1.9	2.7	4.4	5.9	6.3	7.5	6.4	6.3	5.1	3.7	2.4	1.8	1674	4.6
Portsmouth	2.0	2.9	4.6	6.0	7.3	8.0	7.2	6.8	5.4	3.9	2.4	1.9	1779	4.9
Renfrew (Glasgow)	1.0	1.9	3.0	4.5	5.5	6.1	4.9	4.3	3.7	2.4	1.5	0.9	1205	3.3
Shannon Airport	1.8	1.9	3.5	4.8	6.8	6.1	4.7	5.4	3.5	2.7	2.3	1.4	1374	3.7
Sheffield	1.2	1.8	2.9	4.2	5.1	6.2	5.5	5.0	3.9	2.7	1.5	1.0	1252	3.4
Shrewsbury	1.6	2.3	3.5	5.0	5.7	6.5	5.4	5.4	4.3	2.9	1.8	1.4	1339	3.7
Southend	1.7	2.6	4.2	5.5	7.0	7.5	7.1	6.6	5.2	3.7	2.0	1.5	1664	4.6
Stonoway	1.0	2.0	3.4	4.9	6.1	5.5	4.4	4.3	3.6	2.5	1.6	0.8	1216	3.3
Weymouth	2.0	2.8	4.6	6.0	7.1	7.8	6.9	6.8	5.2	3.7	2.4	1.8	1742	4.8
York	1.1	2.0	3.2	4.6	5.6	6.3	5.6	5.1	4.0	2.9	1.7	1.2	1310	3.6

Table IV-11a. Bright Sunshine Data for several latitudes in the U.K.

A = Average monthly duration (hrs)

B = $\tau n / \tau N$ = Average per cent of possible (%)

C = Minimum duration in one day (hrs)

D = Average No of days with no sunshine

Period 1931-1960	Plymouth/Mount Batten 50° 21'N, 4° 07'W, 27m				Kew 51° 28'N, 0° 19'W, 5m				Birmingham/Edgbaston 52° 29'N, 1° 56'W, 163m			
Month	A hrs	B %	C hrs	D days	A hrs	B %	C hrs	D days	A hrs	B %	C hrs	D days
January	60	22	8.0	11	46	18	7.1	13	43	16	7.3	14
February	80	28	9.6	7	64	23	9.6	9	58	21	9.0	9
March	133	36	11.4	5	113	31	10.7	6	98	27	11.2	8
April	182	44	13.6	3	160	38	13.3	3	139	33	13.1	3
May	219	46	14.9	2	199	41	15.0	2	167	34	15.3	3
June	222	45	15.4	2	213	43	15.3	1	180	36	15.3	2
July	198	41	15.3	3	198	39	15.5	1	166	33	15.4	2
August	198	44	14.2	2	188	41	13.9	1	159	35	14.0	2
September	152	40	12.1	3	142	37	12.0	2	117	31	11.9	3
October	114	34	10.3	5	98	30	10.2	5	86	26	9.9	6
November	67	25	8.5	8	53	20	8.1	11	48	18	7.7	12
December	52	21	7.3	11	40	16	7.0	14	38	16	7.0	4
Year	1667	37	15.4	62	1514	34	15.5	68	1299	29	15.4	78

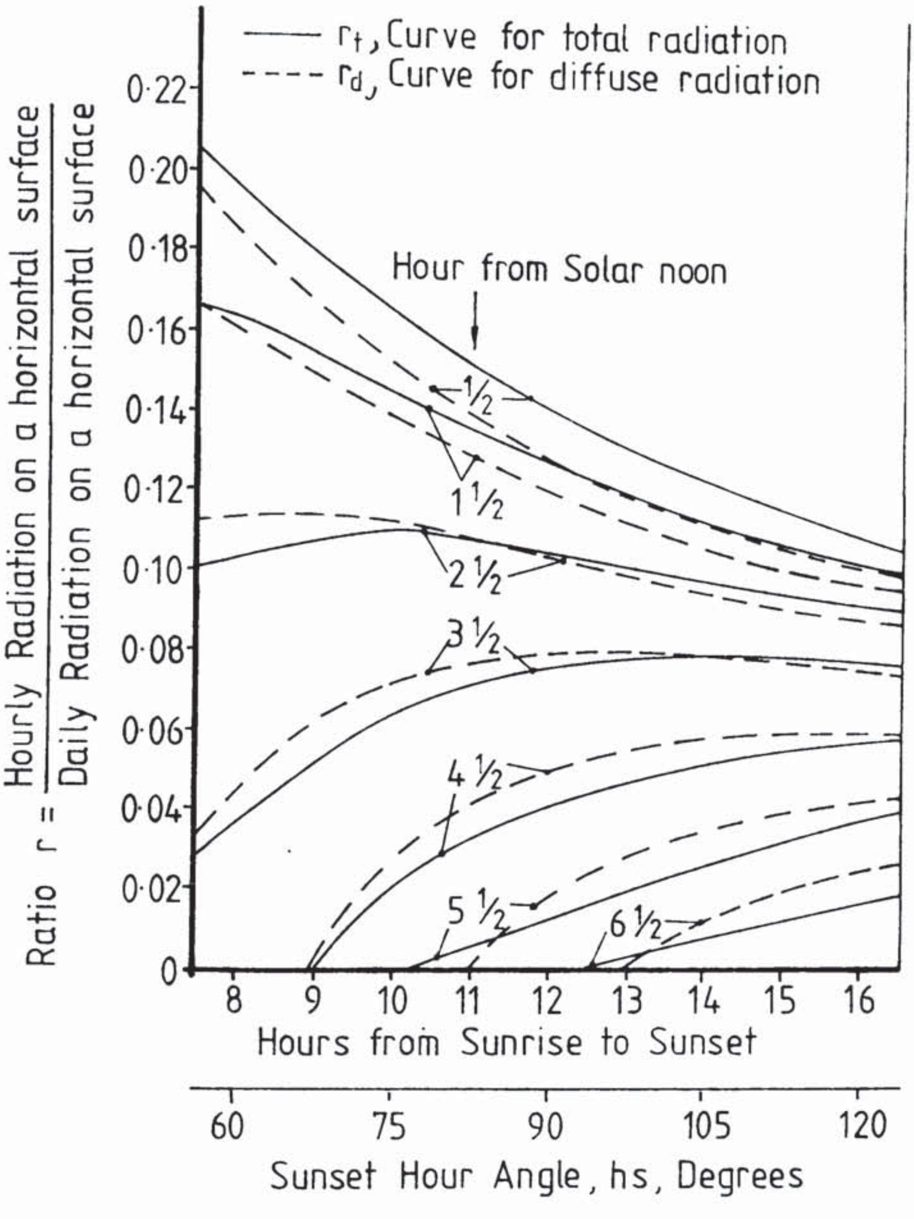


Fig 28. Relationships between hourly radiation and daily radiation on a horizontal surface -(49)

Thus, from a knowledge of day length and daily total radiation, the hourly radiation can be estimated. Two different ratios are represented in Figure 28.

a)

$$r_t = \frac{\text{Average hourly total radiation on a horizontal plane}}{\text{Average daily total radiation on a horizontal plane}}$$

$$r_t = \frac{\bar{I}_{Th}}{\bar{H}}$$

b)

$$rd = \frac{\text{Average hourly diffuse radiation}}{\text{Average daily diffuse radiation}}$$

$$rd = \frac{\bar{I}_d}{\bar{D}}$$

Figure 28 is based on long-term averages and is intended for use in estimating averages of hourly radiation.

IV-6.3.4 Radiation on Tilted Surfaces at the Earth's Surface

A solar absorber tilted at an angle β from the horizontal, absorbs both the direct beam and the diffuse component of solar radiation. In order to use horizontal total radiation data it is necessary to consider the angular correction to convert that data on a horizontal plane to radiation on a tilted surface. This can be done exactly for the direct beam component, and for the diffuse radiation coming from the part of the sky around the sun. The direct radiation beam can be divided in two components, one perpendicular and one parallel to the tilted surface. Only the perpendicular component makes impact on the surface; thus the angular correction for the direct beam can be done by using the "Tilt Factor" according to the equations IV-12 and IV-13, which is equal to the ratio:

$$R_b = \frac{I_{bt}}{I_{bh}} = \frac{I_b \cos\theta_t}{I_b \cos\theta_h} = \frac{\cos\theta_t}{\cos\theta_h} = \frac{Ho_t}{Ho_h}$$

and named "the beam correction factor". See Fig. 29.

The correction factor for the diffuse component (51) depends on the distribution of diffuse radiation over the sky, which generally is not known; this distribution depends particularly on clouds and also on the spatial distribution and amounts of other atmospheric components that determine scattering.

If one assumes that the diffuse component is isotropic, then the diffuse radiation on a surface of other than horizontal orientation is dependent only on how much of the sky the surface sees.

If it can also be assumed that the ground and other surfaces, seen by the tilted surface, reflect solar radiation in such a way as to be a source of diffuse solar radiation equivalent to the sky, then the surface will receive the same diffuse radiation no matter what its orientation. So the correction factor for diffuse radiation is always unity.

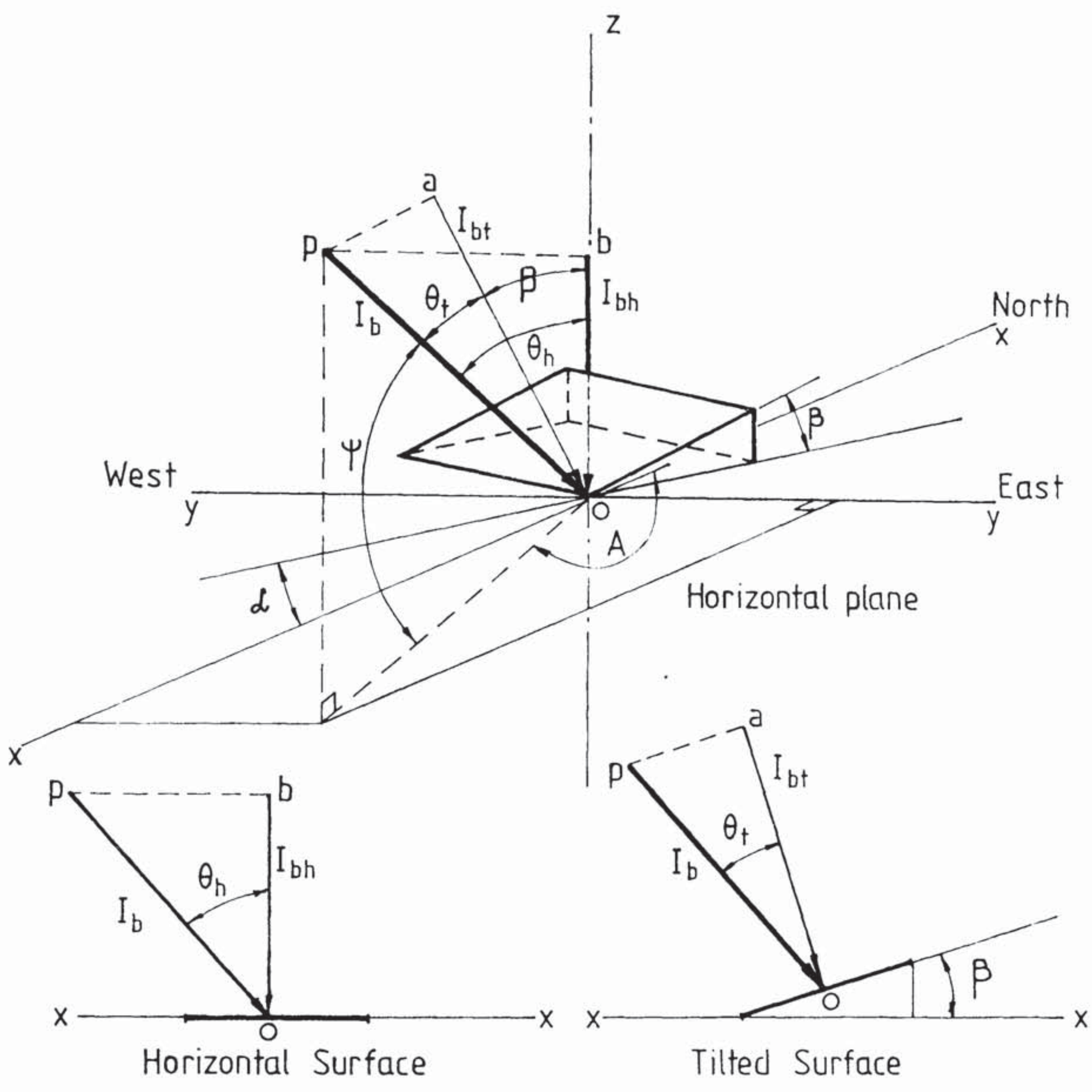
Under these conditions, the radiation on a tilted surface at the earth's level, will be:

a) Total hourly radiation.- ($Wh/m^2.h$)

I_{Tt} = Direct beam + Diffuse radiation

$I_{Tt} = I_{bt} + I_d$

Fig 29 Radiation on horizontal and tilted surfaces and involved angles



α = Surface orientation angle on the horizontal plane = angular distance from North to the projection of the profile of the slope on the horizontal plane (Varies from $-\pi/2$ to 0 to $+\pi/2$)

A = The Solar Azimuth angle

Ψ = The Solar Altitude angle

β = Surface tilt angle from the horizontal plane

θ_t and θ_h = Tilted and horizontal incidence angle

I_b = Direct beam radiation from Sun, at the Earth's level

I_{bt} = Direct beam radiation on tilted surface

I_{bh} = Direct beam radiation on horizontal surface

$$I_{Tt} = I_{bh}\ (\text{Tilt factor}) + I_d$$

$$I_{Tt} = I_{bh} \cdot R_b + I_d \qquad \text{(IV-17)}$$

Where:

I_{Tt} = Total hourly radiation on the tilted surface

I_{bh} and I_d = Beam and diffuse components of hourly solar radiation on the horizontal surface.

I_{bt} = Beam component of hourly solar radiation over the tilted surface. ($Wh/m^2.h$)

b) Total Average daily radiation ($Wh/m^2.day$)

If one takes into account the monthly average daily radiation values, the total radiation on a tilted surface will be:

$$\bar{H}_{Tt} = \text{Direct component} + \text{Diffuse component}$$

$$\bar{H}_{Tt} = (\bar{H} - \bar{D})\ \text{Tilt factor} + (\bar{D}/\bar{H})\ \bar{H}$$

$$\bar{H}_{Tt} = \bar{B} \times R_b + (\bar{D}/\bar{H})\ \bar{H} = Wh/m^2.day \qquad \text{(IV-18)}$$

Where:

$\bar{H}_{Tt}$ = Average daily total radiation on the tilted surface;

$\bar{H}$ = Average daily total radiation on a horizontal surface;

$\bar{B}$ and $\bar{D}$ = Average daily of direct beam and diffuse radiation on a horizontal surface.

Figure 30 shows the average daily total radiation incident on south and north facing roofs inclined at 40° to the horizontal, for the U.K.

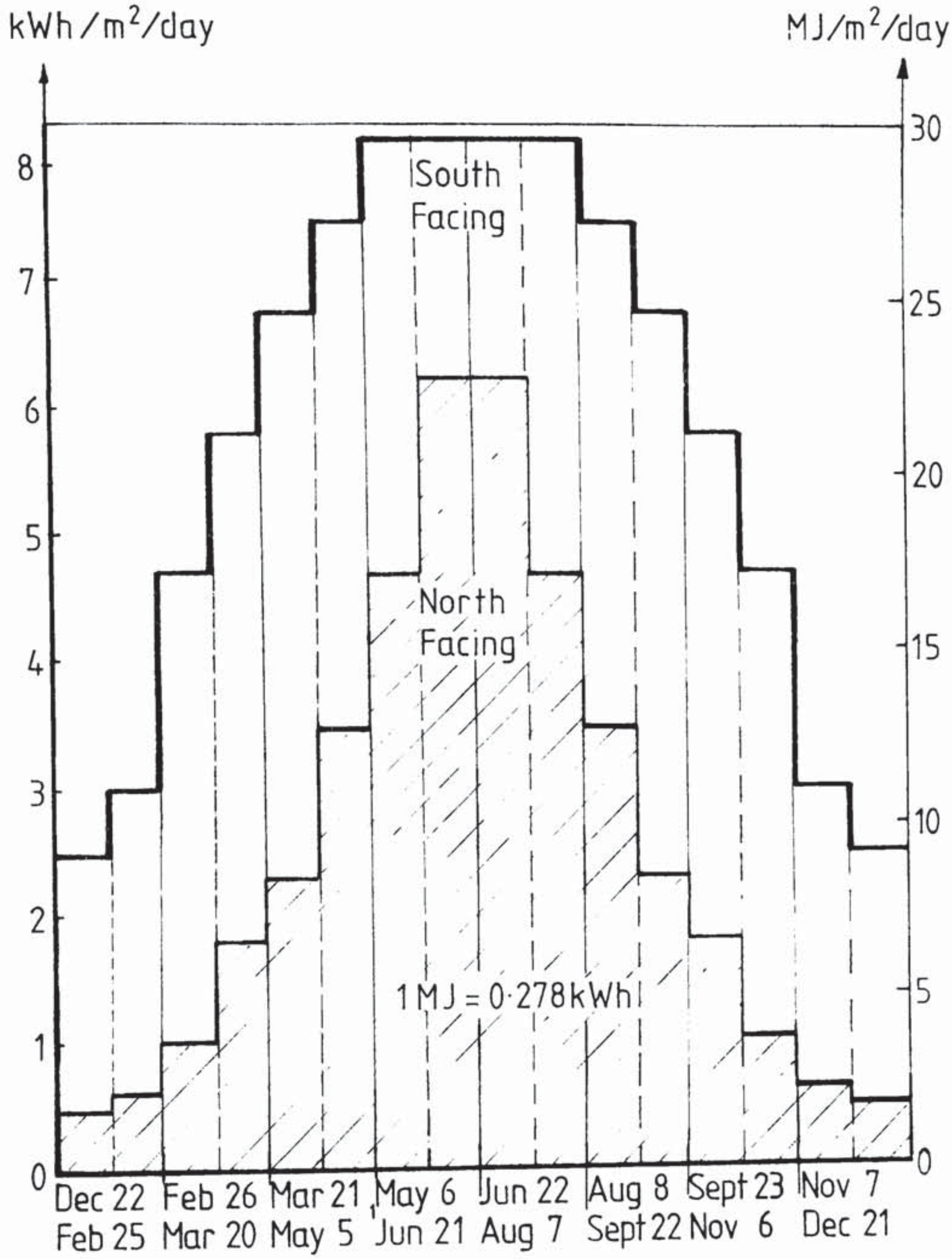


Fig. 30

Bright sunshine available energy daily from South and North facing roofs inclined at 40° to the horizontal at a latitude 51° 28′ N (Ref 4) at a latitude of 51° 28'N. It can be seen from this figure that radiation on a North facing roof, which is only diffuse, can add appreciably to the total available energy.

IV-6.3.5 Actual Radiation Incident on a Titled Roof Corrected for Ground Reflected Radiation

A tilted roof, acting as an absorber of solar radiation, receives not only direct radiation and diffuse radiation from the portion of the sky-dome it actually sees, but also solar radiation reflected from the ground and from the environment which the tilted surface sees. A roof tilted at slope β from the horizontal sees a portion of the sky dome given (51) by $(1 + \cos\beta)/2$, which is the correction factor for diffuse radiation if it is isotropic.

If the surroundings have a diffuse reflectance of ρ for solar radiation, then the reflected radiation from the environment on the surface of the roof from total solar radiation is $(I_{b_h} + I_d)(1-\cos\beta)$ $(\rho/2)$, (51); thus, the total solar radiation on the surface of the tilted roof at any time will be:

$$I_{Tt} = [I_{direct}] + [I_{diffuse}] + [I_{reflected}]$$

$$I_{Tt} = [I_{bh} R_b] + [(1 + \cos\beta) I_d/2] + [(I_{bh} + I_d)(1 - \cos\beta)\rho/2] \qquad \text{(IV-19)}$$

On a long-term average basis the hourly total radiation on a tilted surface or roof can be calculated as a function of the average daily radiation on a horizontal surface $\bar{H}$:

$$\bar{I}_{Tt} = \bar{I}_{direct} + \bar{I}_{diffuse} + \bar{I}_{reflected}$$

$$\bar{I}_{Tt} = (\bar{I}_{Th} - \bar{I}_d) R_b + \tfrac{1}{2} (1 + \cos\beta) \bar{I}_d + \tfrac{1}{2} (1-\cos\beta) \rho \bar{I}_{Th}$$

$$\bar{I}_{Tt} = (\bar{I}_{Th} - \bar{I}_d) (\cos\theta t/\cos\theta h) + (\tfrac{1}{2}) (1 + \cos\beta) \bar{I}_d + (\tfrac{1}{2}) (1 - \cos\beta) \rho \bar{I}_{Th}$$

By using the relationships between hourly and daily radiation given previously one can write:

$$\bar{I}_{Tt} = [r_t\bar{H} - r_d\bar{D}] (\cos\theta t/\cos\theta h) + (\tfrac{1}{2}) (1 + \cos\beta) (r_d\bar{D}) + (\tfrac{1}{2}) (1 - \cos\beta) \rho r_t\bar{H}$$

or:

$$\bar{I}_{Tt} = \left\{(1 - (r_d\bar{D}/r_t\bar{H})) (\cos\theta t/\cos\theta h) + (\tfrac{1}{2}) (1 + \cos\beta) (r_dD/r_t\bar{H}) + (\tfrac{1}{2}) (1 - \cos\beta) \rho\right\} r_t\bar{H}$$

by definition:

$$R = \bar{I}_{Tt}/\bar{I}_{Th} = \left\{(1 - (r_d\bar{D}/r_t\bar{H}))(\cos\theta t/\cos\theta h) + (\tfrac{1}{2}) (1 + \cos\beta)(r_d\bar{D}/r_t\bar{H}) + (\tfrac{1}{2}) (1 - \cos\beta) \rho\right\} \qquad \text{(IV-20)}$$

and then, the hourly total radiation on a tilted roof at an angle β from the horizontal will be:

$$\bar{I}_{Tt} = (R)\ r_t \cdot \bar{H} \qquad \text{(IV-21)}$$

Values of r_t can be obtained from Fig. 28. These were derivated from Liu and Jordan (49), who suggested values of ground reflectance varying between $\rho = 0.2$ when there is no snow and $\rho = 0.7$ when there is snow cover.

CHAPTER V

SPACE HEATING LOAD CALCULATIONS

V-1 Heat Loss Rate and "N kW House"

From the thermal point of view a building is characterised by the specific heat-loss rate concept, which is an index of the thermal properties of a building, given as the total heat loss rate per unit temperature difference between the air inside and outside of a building. (W/°C).

The total heat loss rate from the building can be divided into conductance heat loss rate through the enclosing structure, and heat loss rate due to ventilation.

V-1.1 Thermal Conductance and Thermal Resistance

The heat loss from the building by conduction through the enclosing elements may be expressed as:

$$Qc = (\Sigma\, An\ Un)\ (T_i - T_o)$$

$$Qc = (\Sigma\, An\ Un)\ \Delta T \qquad \text{(V-1)}$$

Where:

Qc = Conductance heat loss in (W)

An = Area of exposed surface of each enclosing element (m^2)

T_i = Inside air temperature (°C or K)

T_o = Outside air temperature (°C or K)

Un = Thermal transmittance coefficient of each element, i.e., the rate of heat loss through $1m^2$ of the element n, when the air temperature at each side differ by 1°C, expressed in (W/m^2°C).

Enclosing elements of the structure are (54):

(a) Any external wall including any doors or windows.

(b) Any wall adjacent to a ventilated (or partially ventilated) space.

(c) A floor having the underside exposed to air or ventilated space.

(d) A roof including any ceiling, roofspace and any ceiling below.

(e) Any glazed area in a wall or roof.

Walls, floors and roofs of buildings are usually composed of several layers, each one of different material. They provide a resistance (R) to the heat loss in order to maintain the temperature difference between the air on each side of the enclosing structure. The actual rate of heat flow through each element depends on the materials used in construction, which have varying rates of thermal conductivity ($k = kJ.m/m^2\ h\ °C$).

The surface resistance is the reciprocal of the surface conductance, which is the rate of heat transfer by radiation and convection, from air to surface or vice-versa. The surface conductance quantified as the surface heat transfer coefficient, depends on the nature of the wall surface itself, and is equal to the amount of heat transferred per square metre and per second, between the surface and the surrounding air for each degree C of temperature difference between them.

The surface resistance, assuming that the surrounding surfaces on each side of the enclosing element are the respective air temperatures, is given (46) by:

$$Rs = 1/(\varepsilon\, hr + hc) \qquad \text{(V-2)}$$

Where:

Rs = Surface resistance ($m^2 °C/W$)

hr = Radiation heat transfer coefficient ($W/m^2 °C$); ($hr = 5.7$ for a mean surface temperature of 20°C and $hr = 4.6$ for a mean surface temperature of 0°C). (46)

ε = Emissivity factor; (for normal temperature radiation $\varepsilon = 0.9$ and an extreme value would be $\varepsilon = 0.05$). (46)

hc = Convection heat transfer coefficient ($W/m^2 °C$); ($hc = 3.0$ for walls; $hc = 4.3$ for upward flow to ceiling and $hc = 1.5$ for downward flow to floors). (46)

Because of the wind speed, which is an important factor in determining the values of the surface conductance, the rate of heat transfer can vary considerably according to the rate of air movement across surfaces. In this case the convection coefficient may be derived (46) from:

$$hc = 5.8 + 4.1\,\nu \qquad (V\text{-}3)$$

Where ν = wind speed. Table V-1 obtained from equation (V-3) by reference (46), gives the values of internal surface resistances.

Table V-1. Internal Surface Resistance (Rsi)

Building Element	Heat Flow	Rsi ($m^2 °C/W$)	
		$\varepsilon = 0.9$	$\varepsilon = 0.05$
Walls	Horizontal	0.123	0.304
Ceilings - Roofs: flat or pitched - Floors.	Upward	0.106	0.218
Ceilings and Floors	Downward	0.150	0.562

Values for conductivities, resistivities of materials, surface conductances, surface resistances of internal and external walls, roofs etc, are given in figures published by the Institution of Heating and Ventilating Engineers, in the IHVE Guide (46) section A, and also by reference (55) in section 27. The values given, for external surface resistance of walls, roofs, etc, are based on plane non-metallic surfaces such as are usually found with thermal building materials used in the U.K; and according to aspect, to situation and degree of exposure. These values conform to the concept of normal wind exposure, i.e. that the wind speed at roof surfaçes is 3.0 m/s:

For "sheltered" and "severe" exposures, the wind speed at roof surfaces have been taken as 1.0 and 9.0 m/s respectively. The external surface resistance values (m^2 °C/W) of walls and roofs, calculated through equations V-2 and V-3 by reference (46) section A3, for sheltered, normal and severe exposures are given in table V-2.

Table V-2. External Surface Resistances Rso

Element	Emissivity of surface ε	Surface Resistance for stated exposure (m^2 °C/W)		
		Sheltered	Normal	Severe
Wall	High = 0.90	0.08	0.055	0.03
Wall	Low = 0.05	0.11	0.067	0.03
Roof	High = 0.90	0.07	0.045	0.02
Roof	Low = 0.05	0.09	0.053	0.02

Areas up to the third floor of buildings in city centres are considered as "sheltered exposure". The term "normal" exposure is applied to the fourth to eighth floors of buildings in city centres and most suburban or country buildings. However the building regulations, 1976, see reference (54), prescribe for dwellings, factories and other buildings, that the sum of the internal and external surface resistances must be taken as: 0.18 (m^2 °C/W) for walls, 0.20 (m^2°C/W) for floors, and 0.15 (m^2°C/W) for roofs.

V-1.2 Computation of Transmittance

The thermal transmittance coefficient (U value) is defined as the number of watts transmitted through one m^2 of structure when there is 1°C temperature difference between the air on each side. ($U = W/m^2$ °C)

Thermal transmittance of walls, roofs and floors composed of parallel layers are obtained by adding thermal resistances of layers and taking the reciprocal. For example the thermal transmittance of a cavity wall construction as shown in Figure 31 may be expressed by:

$$U = 1/R = 1/(R_{si} + R_1 + R_2 + R_a + R_3 + R_{so}) \qquad \text{(V-4)}$$

$$U = 1/\left[(1/hi) + (X_1/k_1) + (X_2/k_2) + (X_a/k_a) + (X_3/k_3) + (1/ho)\right] \qquad \text{(V-5)}$$

Where:

U = Thermal Transmittance (W/m^2 °C)

$Rsi = 1/hi$ = Internal Surface Resistance (m^2 °C/W)

$Rso = 1/ho$ = External Surface Resistance (m^2 °C/W)

Ra = Resistance of Air Gap or cavity, including the surface resistance of both surfaces enclosing the cavity (m^2 °C/W)

$R_1 = X_1/k_1$

$R_2 \;\; = X_2/k_2$

$R_3 \;\; = X_3/k_3$ = Thermal Resistance of structural components ($m^2\,°C/W$)

$h_i \;\; =$ Surface conductance for inside wall ($W/m^2\;°C$):

hri = radiation; hci = convection

$h_o \;\; =$ Surface conductances for outside wall ($W/m^2\;°C$);

h_{co} = convection; hro = radiation

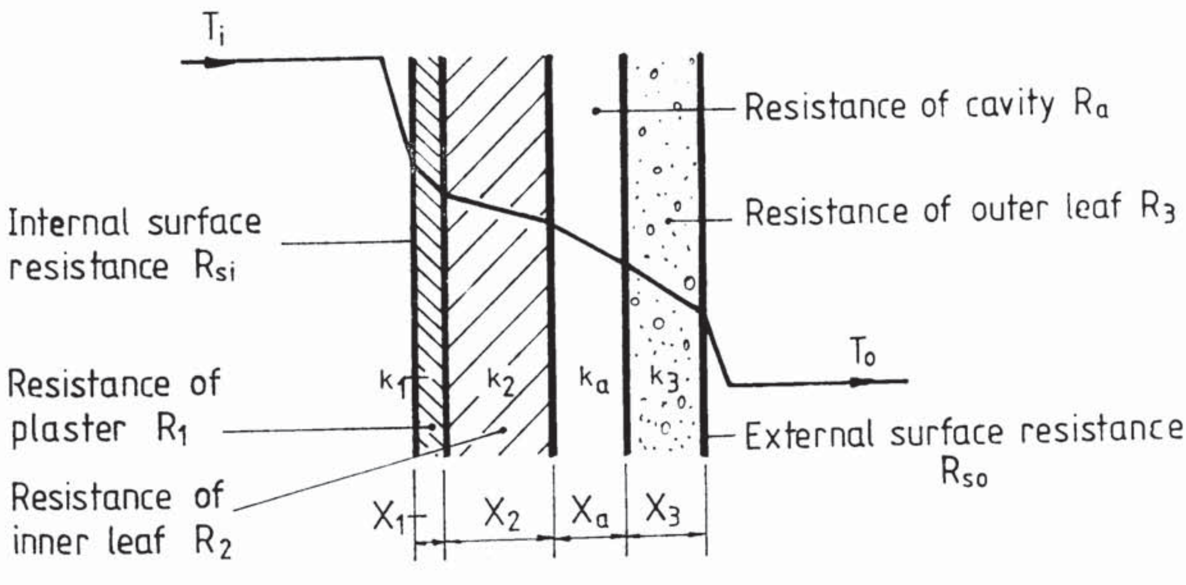
X = Thickness of material (m)

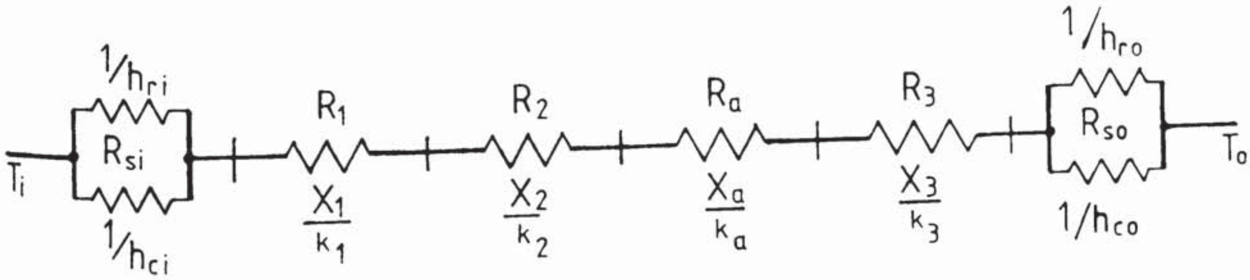
k = Thermal conductivity of material

($kJ.m/m^2\;h\;°C$) or ($W/m.°C$)

1/k = Resistivity of material ($m°C/W$)

Fig 31. Thermal Transmittance Coefficient (U value)





The thermal resistance Ra of an air space in a cavity wall construction may be taken (9) as:

Ventilated Cavity $\quad Ra = 0.12$ (m^2 °C/W)

Unventilated Cavity $\quad Ra = 0.18$ (m^2 °C/W)

There is no point in installing an expensive solar assisted heat pump system for heating the building, if the structure is thermally inefficient. If the building is thermally well designed and has an efficient insulation, the contribution from solar energy will be proportionately greater because of reduced energy requirements. Thermal transmittance values (U) for certain clearly defined building elements, for different categories of exposure to wind, are given in the IHVE Guide (46) section A3; also in references (9) and (55). Nevertheless the building regulations, 1976, prescribed the minima of insulation, i.e., the maximum acceptable U values of walls, floors, roofs and perimeter (Average values) walling. Maximum and average permitted U values (54) only for dwellings (i.e. houses flats and maisonettes); are presented in the table V-3.

V-1.3 Specific heat loss rate by conductance of an "N kW house"

To determine the specific heat loss rate by conductance of a particular house, it is necessary to know the overall conductance heat loss through the enclosing structure, for an established temperature difference between inside and outside of the house.

The established ΔT may be taken as 20°C, which means that the inside temperature is 20°C when the outside is 0°C or that the inside is 18°C when the outside is -2°C. The last ones are an average value of the recommended internal and external design temperatures for the U.K.

Table V-3. Maximum and Average U values permitted by building regulations (1976) for elements of dwelling

Type of Element	U (W/m^2 °C)	
	Maximum	Average
Walls separating the dwelling from the external air, ventilated space or roof space.	1.0	0.6
Walls separating the dwelling from partially ventilated space, or the interior of any building except a dwelling.	1.7	1.7
A partition separating a habitable room in the roof, from any other roof space.	1.0	-
A floor between a dwelling and the exterior or ventilated space.	1.0	-
Roofs including ceilings and any space between.	0.6	-
Fixed value for windows, single glazed.	5.7	5.7
Fixed value for windows, double glazed.	2.8	2.8
Fixed value for walls dividing one dwelling from another.	0.5	0.5

For thermal design purposes and because of the U values from the building regulations, the type of construction, and the size of a house can be established if this is defined as a function of the conductance heat loss rate, for the ΔT selected.

V-1.3.1 Definition of an "N kW House"

An "N kW house" is one which loses by conduction in still air through its enclosing structure $N \times 10^3$ J/s with an outside temperature of 0°C and an inside temperature of 20°C. e.g. a 5 kW house loses 5000 J/s under the mentioned conditions.

Thus:

$$5000 = (\Sigma \text{ Un An}) \ 20$$

$$\Sigma \text{Un An} = 250 \ (\text{W}/°\text{C}) = (\text{Specific heat loss rate by conduction})$$

Taking into consideration the values of the thermal transmittance coefficient from the building regulations, i.e., average U values of perimeter walling, maximum U values of roof, etc., it would be possible to estimate the "Average U value" for the house boundary Area: $Ue = \Sigma \text{ An Un} / \Sigma \text{ An}$. So, if the average U value is for example $Ue = 0.6 \text{ W/m}^2 \ °\text{C}$, the total exposed area of the 5kW house could be:

$$\Sigma \text{Un An} = 250$$

$$\text{Ue} \times \text{Ae} = 250$$

$$\text{Ae} = 250/0.6 = 416.67 \text{ m}^2$$

And for $Ue = 0.8 \text{ W/m}^2 \ °\text{C}$, the exposed area could be $\sim 312 \text{ m}^2$.

The table V-4, gives the specific heat loss rate by conduction for several "N kW houses", and in each case the total exposed area for a range of expected "Average U value" (The average of thermal transmittance for the whole boundary area).

Table V-4. Specific Heat Loss Rate by Conduction and Total Exposed Area for Several "N kW house"

N kW House	Spc. Heat Loss Rate Σ Un An	Average Ue	Total Exposed Area	N kW House	Spc. Heat Loss Rate ΣUn An	Average Ue	Total Exposed Area
kW	W/°C	W/m^2°C	m^2	kW	W/°C	W/m^2°C	m^2
2	100	0.6	166	10	500	0.6	833
		0.8	125			0.8	625
		1.0	100			1.0	500
		1.2	83			1.2	416
		1.4	71			1.4	357
		1.6	62			1.6	312
4	200	0.6	333	12	600	0.6	1000
		0.8	250			0.8	750
		1.0	200			1.0	600
		1.2	166			1.2	500
		1.4	143			1.4	428
		1.6	125			1.6	375
6	300	0.6	500	14	700	0.6	1166
		0.8	425			0.8	875
		1.0	300			1.0	700
		1.2	250			1.2	583
		1.4	214			1.4	500
		1.6	187			1.6	437
8	400	0.6	666	16	800	0.6	1333
		0.8	500			0.8	1000
		1.0	400			1.0	800
		1.2	333			1.2	666
		1.4	286			1.4	571
		1.6	250			1.6	500

The effect of the insulation on the total heat loss by conduction through the enclosing can be seen from this table:
Similar houses with about 400 m^2 of exposed area may have heat losses from 5000 J/s (5 kW house) to 14000 J/s (14 kW house) according to the "Average U value" rising from 0.6 to 1.6.

V-1.4 Ventilation Heat Loss Rate

Natural air infiltration through cracks in walls and around windows and doors, is a direct function of the wind velocity and can be calculated by either the crack method or the air-change method (6, 46, 56).

For hygiene reasons: smoke, odours etc, and because of water vapour, it is necessary in some installations to supply more ventilation air than will be obtained by natural infiltration. In this case the air exchange requirement can be specified in terms of m^3/h. person, depending on the available volume per person. The air infiltration and the air requirement, if the number of occupants is unknown, can be expressed in the number of air changes per hour, which varies between 1 and 3 for normal rooms. In practice a minimum air-change rate of 0.5 to 1.0 per hour is needed in the heating season to remove odours and water vapour and to ensure the satisfactory removal of waste products and an adequate supply of combustion air for heating appliances.

The sensible heat required to raise the temperature of the air supplied by both the natural infiltration and the forced ventilation, if necessary, as well as the Latent Heat of the water vapour must be considered as part of the heat loss. Normally they are included separately as ventilation heat losses.

The following expression is used to assess the heating requirement due to ventilation in winter and is named the ventilation heat loss rate:

$$Qv = 0.36 \quad V \; n \; (T_i - T_o) \qquad \text{(V-6)}$$

Where:

0.36 represents the volumetric specific heat of air at temperatures between 0 and 20°C (Wh/m^3 °C) or (1300 J/m^3 °C)

V Is the volume of the building (m^3)

n Is the hourly air change rate (hour $^{-1}$)

Δ T = $(T_i - T_o)$ = the temperature difference (°C) between inside and outside space.

Qv Is the ventilation heat flow rate (W)

The crack method is usually considered more accurate than the air change method to determine the air infiltration rate and the ventilation heat loss rate providing that the variables entering into the method can be correctly evaluated. The CIBS Guide (57) section A4 gives information to calculate the ventilation heat loss rate by the crack method.

V-1.5 Total Heat Loss Rate

Conduction heat losses from the building are normally one to four times the ventilation heat losses. The total heat loss rate or "Space Heating Load" of the house (Q_L) will be the sum of the conduction rate and ventilation rate:

$$Q_L = Qc + Qv$$

$$Q_L = (\Sigma An\ Un)\Delta T + 0.36\ V\ n \Delta T = (W) \qquad (V-7)$$

And the specific heat loss rate will be:

$$Q_L / \Delta T = [\Sigma An\ Un + 0.36\ V\ n] = W/°C \qquad (V-8)$$

On dividing each of these terms by the internal volume V of the house, one gets the volumetric specific heat loss rate for the house:

$$G_L = Q_L / V \Delta T = (\Sigma An\ Un/V) + 0.36\ n = W/m^3 °C \qquad (V-9)$$

These equations assume an equilibrium situation. In fact, the external temperature varies throughout the day and there are energy inputs to the east, south and west walls and diffuse radiation on the north wall which are not accounted for in the calculations. A more exact approach to the problem would assume the treatment of a none-equilibrium system which takes account of the temperature variations and the energy inputs as a function of time. The expression for Q_L can be written as:

$$Q_L = G_L\ V\ (T_i - T_o(\tau)) - \Sigma Q_{S.G.}(\tau)$$

Where $Q_{S.G.}$ represents the solar gains as a function of time τ. These solar gains occur primarily because of the windows. For a realistic installation which is economic when compared to conventional solutions it is necessary to follow the daily gains and losses in order to predict the supplementary heating and the storage required for the least favourable conditions.

The proportion of heat lost as Qc or Qv by the average house is difficult to specify because of the variation in ventilation requirements and standards of insulation. In practice, Qv, is likely to account for about 20 per cent of the total heat loss rate Q_L.

V-2. Degree Days and Space Heating Requirements of a Building

As it has already been seen, the energy required kWh to maintain the interior of a house at a given temperature of comfort T_i depends on the temperature difference between the interior and the exterior and on the heat transfer properties of the fabric of the building. The comfort temperature is considered as 18.3°C.

The outside temperature varies over the season and it is quite clear that the energy required to maintain the comfort temperature of the house also varies. A useful term in this connection is the "Degree Day" (Do), which permits the prediction of the fuel consumption in a heating system over long periods.

V-2.1 Definition of Degree Days (Do)

Degree Days – The daily differences in degree celsius between a base temperature of 15.5°C and the 24 hour mean outside temperature (when the outside temperature falls below the base temperature). There is a recommendation by the Department of Energy for calculating degree days depending on the variation of temperature.

$$\text{Degree Days} = 15.5-[(T_{max} + T_{min})/2] \qquad \text{(V-10)}$$

In the U.K. by definition, above a mean external temperature of 15.5°C no heating is required, below 15.5°C, heating is required. Ventilation and infiltration losses have to be considered additionally. The explanation of this difference (i.e., 18.3°C − 15.5°C = 2.8°C) lies in the fact that not all the heat requirements of a building are supplied from its heating plant. Other sources such as occupants, motors, cooking and domestic appliances, lights, etc., supply sufficient heat to raise the internal temperature of the house on average by some 2.8°C.

"Degree Days" can be calculated for different ambient temperatures and for different periods. If it is assumed that an outside temperature of 14.5°C prevails for one day (24 hours), then heating requirements are proportional to 1°C for that day (This is equivalent to one degree day). If this condition remains for each day of the week, a total of seven degree days would be accumulated. A plot of mean external temperature against time, calculated with data extracted from table V-5, is given in Figure 32. The shaded area of the diagram gives the degree days for a heating season based on the 15.5 °C temperature.

The Department of Energy publishes monthly and seasonal degree days for different regions of the country; the values vary from 1800 in the South West to 2600 in North East Scotland for an average heating season. The seasonal figure for the Midlands area is near to 2300 degree days (when considering September to May).

The use of degree days for comparison of fuel consumed should not therefore be regarded as leading to highly accurate results because the total heat loss rate (Q_L) is also a function of wind velocity and other factors and not only a function of the external temperature.

Seasonal degree days can also be used to compare the severity and duration of the winter from year to year and from place to place. Another use is to take them as a basis of comparison of the power used for district heating schemes in various parts of the country.

Table V-5 gives monthly average and seasonal average degree day values for several areas of the country, based on the 15.5°C datum temperature and for a 20 year period (58).

Table V-5. Twenty-year Mean Degree Days 1957/77 (58)

	Sept	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Total
1 Thames Valley	56	132	256	336	349	304	285	199	113	2,030
2 South Eastern	84	162	281	359	371	326	307	225	146	2,261
3 Southern	77	145	258	331	342	307	297	216	141	2,114
4 South Western	56	115	215	279	296	272	270	201	131	1,835
5 Severn Valley	69	144	259	332	348	310	294	211	130	2,097
6 Midland	92	172	290	362	375	335	322	235	153	2,336
7 West Pennines	79	155	280	350	363	323	308	225	138	2,221
8 North Western	95	168	295	361	371	333	321	241	163	2,348
9 Borders	108	183	300	364	380	343	334	261	193	2,466
10 North Eastern	88	170	296	364	379	335	320	235	155	2,342
11 East Pennines	77	157	283	354	367	324	307	218	139	2,226
12 East Anglia	74	153	283	363	383	336	319	234	143	2,289
13 West Scotland	105	179	303	359	373	336	319	238	163	2,375
14 East Scotland	106	185	309	371	383	344	328	254	190	2,470
15 North E. Scotland	124	199	322	385	399	360	347	272	205	2,613
16 Wales	73	138	239	305	327	301	296	231	157	2,067
17 Northern Ireland	100	172	288	347	363	326	314	239	168	2,317
Total	1,463	2,729	4,757	5,922	6,169	5,515	5,288	3,935	2,628	38,407
	86	161	280	348	363	324	311	231	155	2,259

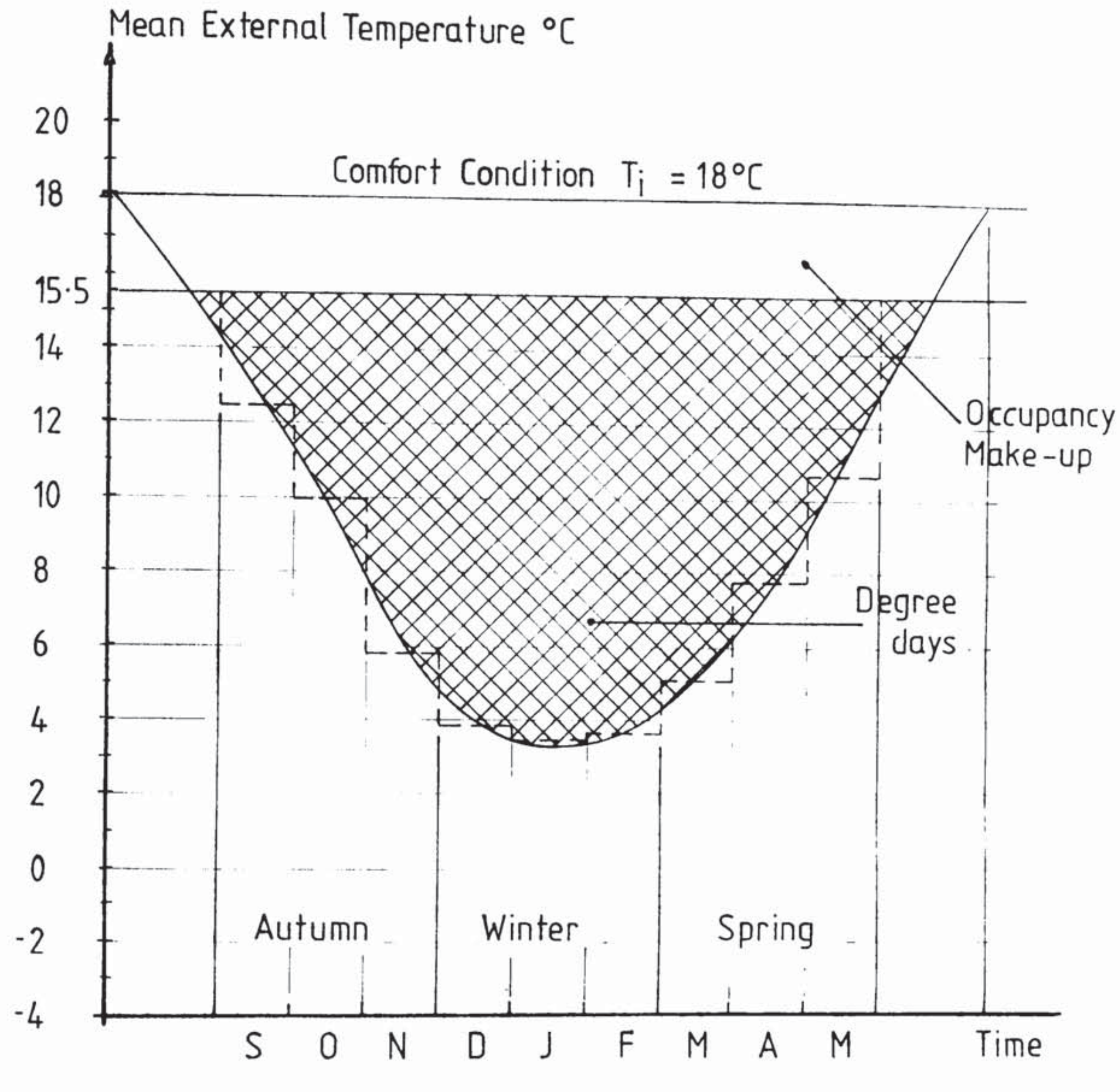


Fig 32 Average Degree Days over a heating season for the Midlands (52^{0} 26′ N) in a 20 years period (Calculated with data extracted from table V - 5)

V-2.2 Space Heating Load as a function of Degree Days

1) The space heating load, in kWh, to compensate for fabric heat loss during a heating period is given by Qc:

$$Qc = \Sigma\, Un\, An\, (Do)\, (24/1000) = (W/°C)\ (°C\ day)\ (kWh/W\ day)$$

$$Qc = 0.024\ Do\ Ue\ Ae = kWh/period \qquad (V-11)$$

Where:

Do = Degree days in the required period (i.e. monhly, yearly, seasonal, etc.)

Ue = Average thermal transmittance (W/m^2 °C)

Ae = Total exposed surface area of the house (m^2)

Table V-6, prepared in this study, gives the monthly and yearly space heating load Qc for both a 5 kW house and a 10 kW house at four diferent U.K. areas. Figure 33, plotted with data from table V-6 shows the variation of Qc through the year at these four different areas of the U.K. The principal factor which determine the pattern of these curves is the general increase in space heating load with increasing latitude.

2) If one considers the volumetric heat loss rate G_L, which takes into account ventilation heat losses, the daily space heating load can be expressed in the following form:

$$Q_L = G_L\ V\ (T_i - T_o)\ 24\ /(1000) \qquad (V-12)$$

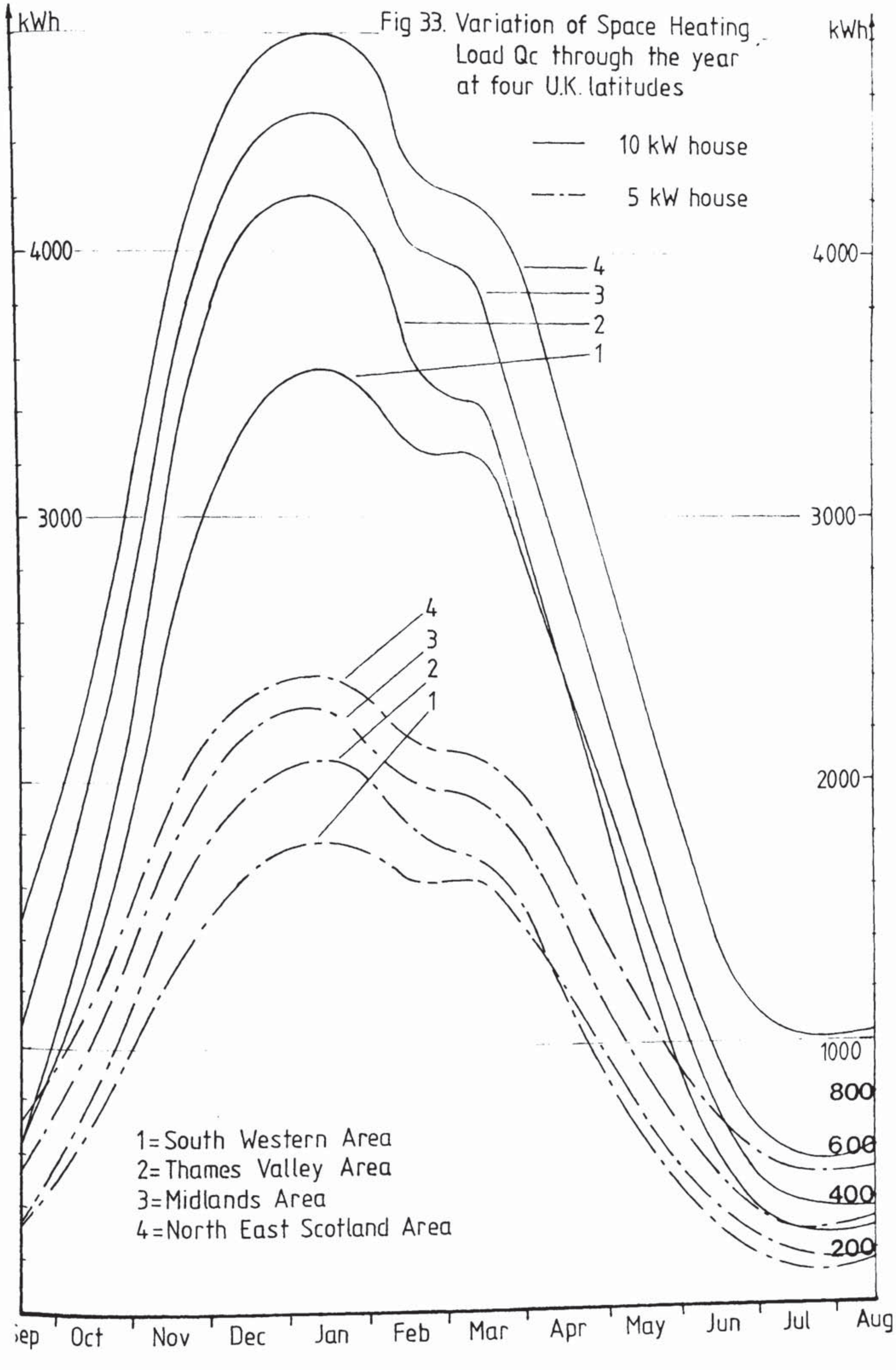
$$Q_L = (W/m^3\ °C)\ m^3\ °C\ h/day\ (W/kW) = kWh/day$$

Introducing degree days Do, in order to determine the space heating load during a given period within the winter season, one can write:

Table V-6. Average Degree Days and Average Monthly Space Heating Load for a 5 kW, and 10 kW House at several U.K. Latitudes. (Qc)

Do = Average Degree Days for a 20 year period (i.e. 1957-1977); A = 5 kW House; B = 10 kW House

Area	South Western Mountbatten - Plymouth 50°10′N			Thames Valley London 51° 30′N			Midlands Elmdon - Birmingham 52°26′N			North East Scotland Dyce - Aberdeen 57°10′N		
Month	Do D.Days	A kWh	B kWh	Do D.Days	A kWh	B kWh	Do D.Days	A kWh	B kWh	Do D.Days	A kWh	B kWh
September	56	336	672	56	336	672	92	552	1104	124	744	1488
October	115	690	1380	132	792	1584	172	1032	2064	199	1194	2388
November	215	1290	2580	256	1536	3072	290	1740	3480	322	1932	3864
December	279	1674	3348	336	2016	4032	362	2172	4344	385	2310	4620
January	296	1776	3552	349	2094	4188	375	2250	4500	399	2394	4788
February	272	1632	3264	304	1824	3648	335	2010	4020	360	2160	4320
March	270	1620	3240	285	1710	3420	322	1932	3864	347	2082	4164
April	201	1206	2412	199	1194	2388	235	1410	2820	272	1632	3264
May	131	786	1572	113	678	1356	153	918	1836	205	1230	2460
June	60	360	720	49	294	588	78	468	936	112	672	1344
July	32	192	384	24	144	288	47	282	564	85	510	1020
August	31	186	372	26	156	312	51	306	612	86	516	1032
Total Year	1958	11748	23496	2129	12774	25548	2512	15072	30144	2896	17376	34752
Total Winter Oct.-Apr.	1533	9198	18396	1861	11166	22332	2091	12546	25092	2284	13704	27408


kWh
Fig 33. Variation of Space Heating Load Qc through the year at four U.K. latitudes
10 kW house
5 kW house
4
3
2
1
4000
3000
2000
1000
800
600
400
200
1=South Western Area
2=Thames Valley Area
3=Midlands Area
4=North East Scotland Area
Sep
Oct
Nov
Dec
Jan
Feb
Mar
Apr
May
Jun
Jul
Aug

$$Q_L = G_L\ V\ Do\ 24/1000 = kWh/period$$

$$Q_L = 0.024\ Do\ G_L\ V = kWh/period \qquad (V-13)$$

The value of G_L varies according to the building type, but regulations could be instituted on the maximum value of G_L considered acceptable at different areas of the country.

V-3 Volumetric Specific Heat Loss Rate G_L related to the "N kW house" concept and degree days Do

It should be remembered that the coefficient G_L is referred to the total space heating load (Q_L = Qc + Qv), whereas the "N kW house concept (using values of Σ Un An) is referred only to the conductance heat loss rate Qc and therefore to the overall U value or insulation characteristics.

One can now define a "specific volumetric heat loss rate by conductance Gc" as the ratio between the specific conductance heat loss rate of an "N kW house" (Σ Un An) and the internal volume V of the building, so:

$$Gc = \Sigma\, Un\ An/V = W/m^3\ °C \qquad (V-14)$$

The coefficient is directly related to the insulation of the building through the overall thermal conductance U.

If one considers the volumetric heat loss rate by ventilation equal to Gv = 0.36 n = W/m^3 °C, then the expression for G_L becomes:

$$G_L = (\Sigma\, Un\ An/V) + 0.36\ n$$

$$G_L = Gc + Gv = W/m^3\ °C \qquad (V-15)$$

Thus the space heating load Q_L can be written as:

$$Q_L = 0.024\ Do\ [(\Sigma Un\ An/V) + 0.36\ n]\ V$$

$$Q_L = 0.024\ Do\ (Gc + Gv)\ V = kWh/period \qquad (V-16)$$

And per unit volume:

$$Q_L/V = 0.024\ Do\ G_L = 0.024\ Do\ (Gc + Gv) = kWh/m^3 \qquad (V-17)$$

V-3.1 Space Heating Load Ratio r_L

This coefficient can be defined as the ratio between the total volumetric heat loss rate G_L and the volumetric heat loss rate by conduction Gc

$$r_L = G_L/G_C \qquad (V-18)$$

This ratio reflects the proportion of ventilation and conduction heat losses, Qv and Qc respectively, into the total space heating load Q_L. From equation V-15 one can obtain:

$$V = \Sigma Un\ An/Gc$$

So, the total heating load Q_L can be written as:

$$Q_L = G_L\ (\Sigma Un\ An/Gc)\ 0.024\ Do$$

$$Q_L = r_L\ (0.024\ Do\,\Sigma\ Un\ An)$$

$$Q_L = r_L\ (Qc) \qquad (V-19)$$

Thus: $r_L = G_L/Gc = Q_L/Q_c = (Qc + Qv)/Qc$

$$r_L = 1 + (Qv/Qc) \qquad (V-20)$$

It can be seen that:

If $Qv = 0$ then $r_L = 1$

If $Qc = 0$ then $r_L = \infty$

Therefore, r_L must vary as: $1 < r_L < \infty$, but in practice, according to Figure 34, r_L varies between 1.1 and 2.5 (i.e., $1.1 < r_L < 2.5$).

According to the definition of r_L or to Figure 34 three cases can be considered:

If $Gc = 1$ then $G_L = r_L$

If $Gc < 1$ then $G_L < r_L$ and if $G_L \downarrow$ then $r_L \uparrow$

If $Gc > 1$ then $G_L > r_L$ and if $G_L \uparrow$ then $r_L \downarrow$

It can be seen in Figure 34, that if $Gc > 1$ and G_L increases then r_L decreases and asymptotically approaches unity. From equation V-20 one can write:

$$r_L - 1 = Qv/Qc$$

and,

$$Qv = (r_L - 1)\, Qc \qquad \text{(V-21)}$$

Dividing (V-19) by (V-21) one can have:

$$\frac{Q_L}{Qv} = \frac{r_L\, Qc}{(r_L - 1)\, Qc}$$

$$Qv = [(r_L - 1)/r_L]\, Q_L \qquad \text{(V-22)}$$

From equations (V-21) and (V-22) one can conclude:

a) If $r_L \uparrow$ then $Qv/Q_L \uparrow$ and $Qv/Qc \uparrow$

$r_L \downarrow$ then $Qv/Q_L \downarrow$ and $Qv/Qc \downarrow$

b) If $r_L < 2$ then $Qv < Qc$

$r_L = 2$ then $Qv = Qc$

$r_L > 2$ then $Qv > Qc$

Table V-7. Values of G_L and r_L as a function of Gc values for different air changes per hour "n" ; $r_L = G_L/Gc$; $Gc = \Sigma Un\ An/V = W/m^3.°C$; $G_L = Gc + 0.36\ n = W/m^3.°C$

Gc	n = 0.5		n = 1.0		n = 1.5		n = 2.0		n = 3.0	
	G_L	r_L	G_L	r_L	G_L	r_L	G_L	r_L	G_L	r_L
0.00	0.18	∞	0.36	∞	0.54	∞	0.72	∞	1.08	∞
0.05	0.23	4.60	0.41	8.20	0.59	11.8	0.79	15.8	1.13	22.6
0.10	0.28	2.80	0.46	4.60	0.64	6.40	0.82	8.20	1.18	11.8
0.20	0.38	1.90	0.56	2.80	0.74	3.70	0.92	4.60	1.28	6.40
0.30	0.48	1.60	0.66	2.20	0.84	2.80	1.02	3.40	1.38	4.60
0.40	0.58	1.45	0.76	1.90	0.94	2.35	1.12	2.80	1.48	3.70
0.50	0.68	1.36	0.86	1.72	1.04	2.08	1.22	2.44	1.58	3.16
0.60	0.78	1.30	0.96	1.60	1.14	1.90	1.32	2.20	1.68	2.80
0.70	0.88	1.26	1.06	1.51	1.24	1.77	1.42	2.03	1.78	2.54
0.80	0.98	1.22	1.16	1.45	1.34	1.67	1.52	1.90	1.88	2.35
0.90	1.08	1.20	1.26	1.40	1.44	1.60	1.62	1.80	1.98	2.20
1.00	1.18	1.18	1.36	1.36	1.54	1.54	1.72	1.72	2.08	2.08
1.20	1.38	1.15	1.56	1.30	1.74	1.45	1.92	1.60	2.28	1.90
1.40	1.58	1.13	1.76	1.26	1.94	1.38	2.12	1.51	2.48	1.77
1.50	1.68	1.12	1.86	1.24	2.04	1.36	2.22	1.48	2.58	1.72
1.70	1.88	1.10	2.06	1.21	2.24	1.32	2.42	1.42	2.78	1.63
2.00	2.18	1.09	2.36	1.18	2.54	1.27	2.72	1.36	3.08	1.54
2.40	2.58	1.07	2.76	1.15	2.94	1.22	3.12	1.30	3.48	1.45
3.00	3.18	1.06	3.36	1.12	3.54	1.18	3.72	1.24	4.08	1.36
4.00	4.18	1.04	4.36	1.09	4.54	1.13	4.72	1.18	5.08	1.27
6.00	6.18	1.03	6.36	1.06	6.54	1.09	6.72	1.12	7.08	1.18
9.00	9.18	1.02	9.36	1.04	9.54	1.06	9.72	1.08	10.1	1.12
∞	∞	1	∞	1	∞	1	∞	1	∞	1

Table V-7, gives values of G_L and r_L as a function of Gc for different air changes per hour n, calculated according to the equations (V-15), (V-18) and (V-14) respectively.

Table V-8, gives the estimated space heating load per unit volume Q_L/V (kWh/m^3) calculated as a function of Gc, G_L and Do for four areas of the U.K., when considering different "N kW house" in each of these areas and a period of heating from November to April inclusive (i.e. Z = 181 days).

Figure 34, plotted with data from tables V-7 and V-8 permits one to obtain values of G_L and r_L for different air changes per hour n and values of Q_L, Qv and Qc for a different "N kW house" with internal volume $V(m^3)$ at several seasonal degree days Do over the heating season in the U.K. Values of Do from 1400 to 2300 were considered to cover the extreme cases.

Values of Q_L/V, as a function of G_L, and values of Qv/Q_L, Qv/V and Qc/V, as a function of r_L were plotted according to the following four relations:

1) $Q_L/V = G_L\ 0.024\ Do = kWh/m^3$

2) $Q_L/V\ Gc = r_L\ 0.024\ Do = kWh/m^3\ (W/m^3 {}^\circ C)$ (V-23)

3) $Qv/V = [(r_L - 1)/r_L]\ Q_L/V$

$Qv/V = [(r_L - 1)/r_L]\ [r_L\ 0.024\ Do \Sigma\ Un\ An/V]$

$[Qv/V]\ [r_L/(r_L - 1)\ Gc] = r_L\ 0.024\ Do$

$(Qv/V)\ \phi v = r_L\ 0.024\ Do = kWh/m^3\ (W/m^3 {}^\circ C)$ (V-24)

With $\phi v = r_L/(r_L - 1)\ Gc = (m^3 {}^\circ C/W)$

4) $Qc/V = (1/r_L)\ (Q_L/V) = (1/r_L)\ [r_L \Sigma\ Un\ An\ 0.024\ Do/V]$

Table V-8. Estimated space Heating Load per unit volume. Q_L/V (kWh/m^3), over the heating season* for different "N kW house" at four different areas of the U.K.

"N kW house"	Σ Un An	House Internal Volume	$Gc = \Sigma$ Un An/V	G_L	Air Changes per hour n = 0.5				
					Q_L/V = kWh/m^3 per season				$r_L = G_L/Gc$
					South Western	Thames Valley	Midlands	N.East Scotland	
					50°10'N	51°30'N	52°26'N	57°10'N	
kW	W/°C	m^3	W/m^3°C	W/m^3°C	Do=1533	Do=1729	Do=1919	Do=2085	
2.0	100	80	1.25	1.43	52.6	59.3	65.9	71.6	1.14
		100	1.00	1.18	43.4	49.0	54.3	59.0	1.18
		120	0.83	1.01	37.2	41.9	46.5	50.5	1.22
4.0	200	120	1.66	1.84	67.7	76.3	84.7	92.1	1.11
		160	1.25	1.43	52.6	59.3	65.9	71.6	1.14
		200	1.00	1.18	43.4	49.0	54.3	59.0	1.18
		240	0.83	1.01	37.2	41.9	46.5	50.5	1.22
5.0	250	150	1.66	1.84	67.7	76.3	84.7	92.1	1.11
		200	1.25	1.43	52.6	59.3	65.9	71.6	1.14
		250	1.00	1.18	43.4	49.0	54.3	59.0	1.18
		300	0.83	1.01	37.2	41.9	46.5	50.5	1.22
		400	0.62	0.80	29.4	33.2	36.8	40.0	1.29
		500	0.50	0.68	25.0	28.2	31.3	34.0	1.36
**		150	2.00	2.18	80.2	90.5	100.4	109.1	1.09
		200	1.50	1.68	61.8	69.7	77.4	84.1	1.12
		250	1.20	1.38	50.8	57.3	63.6	69.0	1.15
6.0	300	300	1.00	1.18	43.4	49.0	54.3	59.0	1.18
		350	0.86	1.04	38.3	43.1	47.9	52.0	1.21
		500	0.60	0.78	28.7	32.4	35.9	39.0	1.30
		600	0.50	0.68	25.0	28.2	31.3	34.0	1.36

** If Gc = 2.0 (i.e. G_L = 2.18) with Do = 1400 and Do = 2300, then Q_L/V = 73.2 and Q_L/V = 120 respectively.

* Heating Season from November to April inclusive.

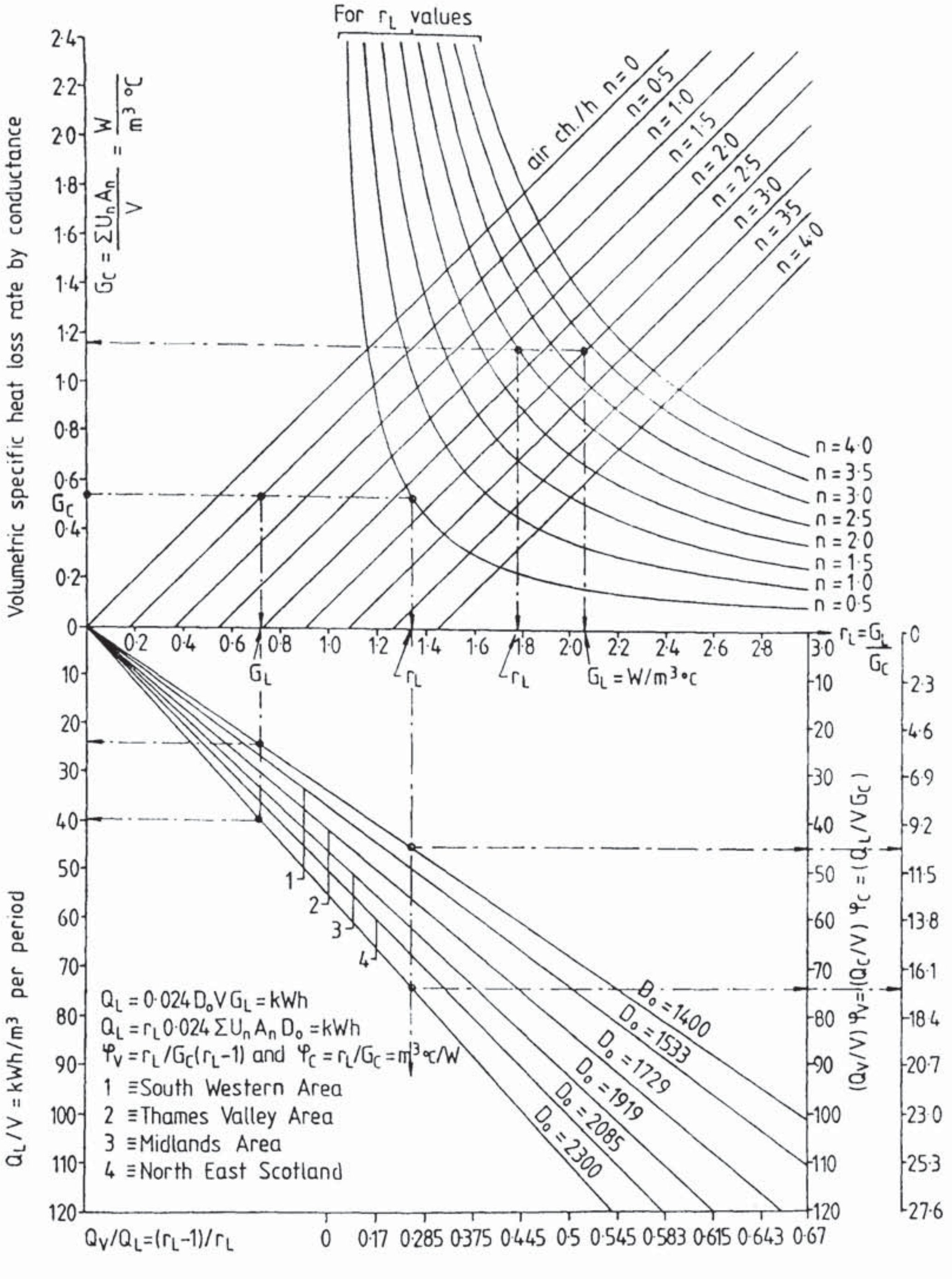


Fig 34 Values of r_L and G_L for different air changes per hour "n" and values of Q_L/V, Q_V/Q_L, Q_V/V and $Q_C/V = kWh/m^3$ for different "N kW house" at several seasonal degree days D_o over the heating season in the UK
D_o for the areas shown correspond to degree days from November to April inclusive (z = 181 days)

$[Qc/V]\ (r_L/Gc) = r_L\ 0.024\ Do$

$$(Qc/V)\ \phi c = r_L\ 0.024\ Do = kWh/m^3\ (W/m^3{}^\circ C) \qquad (V-25)$$

with $\phi c = r_L/Gc = (m^3{}^\circ C/W)$

This figure also permits one to estimate the required insulation for a given house, in order to match it with an established maximum space heating load Q_L during the winter at a particular area of the country. And finally it can be used as a basis to compare insulation requirements for similar houses at different places of the country when using the same amount of energy during a heating period: With Q_L per season, the internal volume V of the house, the number of occupants (or n) and Do in the area given, one could obtain from Figure 34, first G_L and then Gc from which the required thermal transmittance U could be calculated.

Figure 35, which has been calculated for the Midlands area, shows values of G_L, r_L, Q_L/V, Qv/V, and Qc/V for different "N kW house" values, at each month of the year when more or less heating is required.

V-4 Heat Pumps and Space Heating Load

V-4.1 Heat Pumps Energy Consumption and Degree Days

For conduction only, the basic space heating energy consumption equation for a heat pump in a given period can be written as:

$$W_{HP} = \frac{\Sigma\ Un\ An\ 24\ Do}{1000\ COP(H)} = kWh/period \qquad (V-26)$$

Where, Do = Degree Days in the considered period.

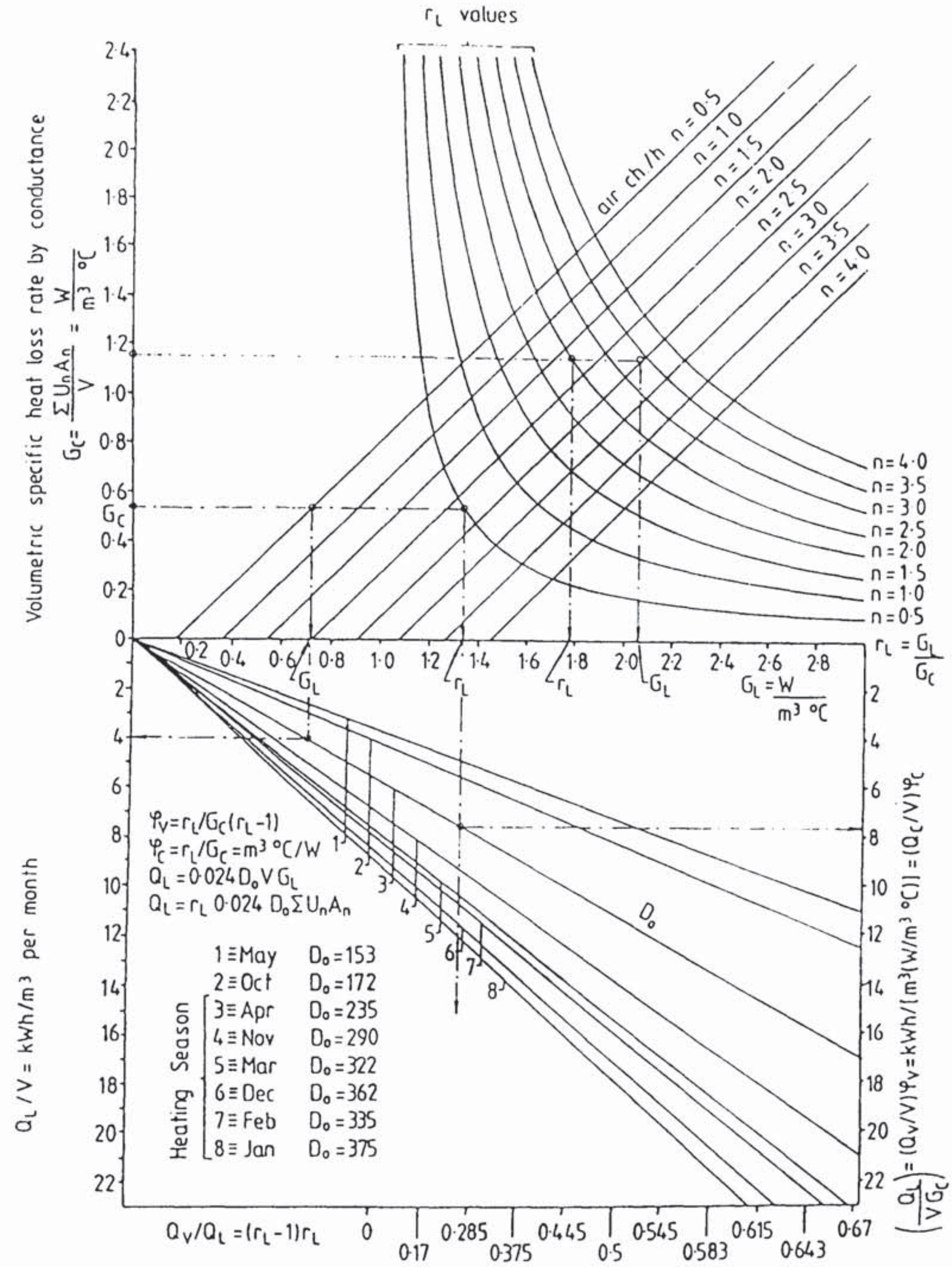


Fig 35 Values of Q_L/V, Q_V/V, Q_C/V (kWh/m³) for different "N kW house" in the Midlands area during some months of the year

Taking into account the ventilation losses the daily energy consumption by the heat pump becomes:

$$W_{HP} = \frac{Q_L}{COP(H)} = \frac{Qc + Qv}{COP(H)}$$

or

$$W_{HP} = \frac{G_L\ V\ (Ti - To)\ 24}{1000 \times COP(H)} = kWh/day \qquad (V-27)$$

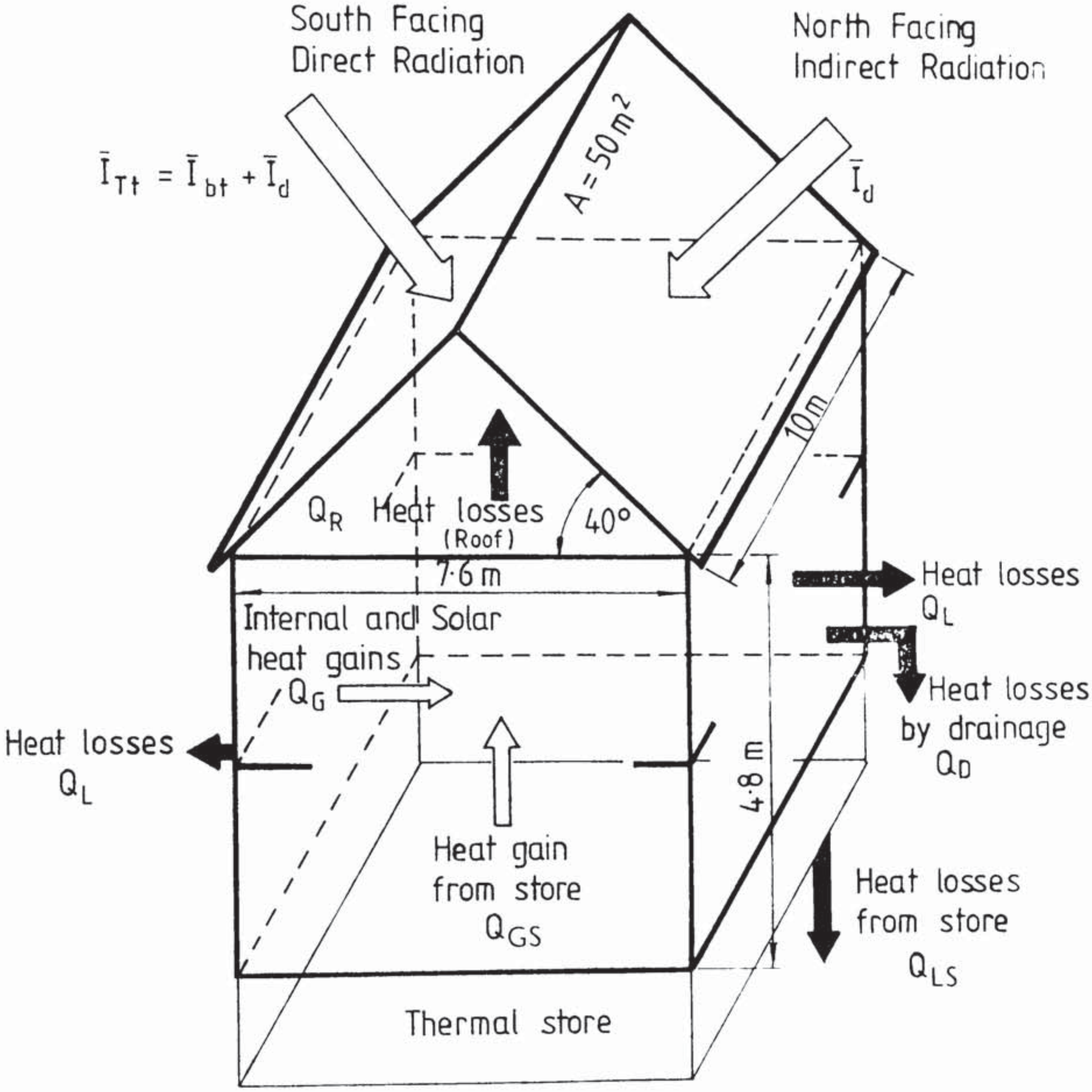
And for a given period during the winter:

$$W_{HP} = \frac{G_L\ V\ 24\ Do}{1000\ COP(H)} = kWh/period \qquad (V-28)$$

Thus, the annual energy consumption for an air to air heat pump or a water heat source-sink heat pump can be found with equations (V-26) or (V-28) by inserting the appropriate coefficient of performance and the proper Q_L, required to cover the space heating requirement, expressed in terms of the annual degree days for the region, and the thermal characteristics of the house.

The performance coefficient for the water heat source-sink heat pump, during the heating cycle particularly, is reasonably constant and independent of the outdoor temperatures. This is not the case, as it was already seen, with the air heat source unit, in which the coefficient of performance is directly related to the outdoor temperature and as a result must be keyed to the expected hours of operation at the various outdoor temperatures to obtain a reliable seasonal coefficient of performance.

The hours of operation and the corresponding outdoor temperatures data, are required for estimating the monthly kWh demand and the maximum possible annual kWh consumption for the heat pump equipment in order to determine the monthly and yearly supplementary heating requirements once the amount of useful internal heat gain has been taken into account.



Wall and Window area $\sum A_n = 321\,m^2$
Internal volume $= 365\,m^3$
Insulation properties of materials $= U_n$
Temperature difference between inside and outside $= T_i - T_o$
If, 5kW House : $\sum U_n A_n = 250$; $U_{AV} = 0{\cdot}78\ W/m^2{}^\circ C$; $G_C = 0{\cdot}685\ W/m^3{}^\circ C$
If, 4kW House : $\sum U_n A_n = 200$; $U_{AV} = 0{\cdot}62\ W/m^2{}^\circ C$; $G_C = 0{\cdot}55\ W/m^3{}^\circ C$

$$\text{For: } \left.\begin{array}{l} G_C = 0{\cdot}685\ W/m^3{}^\circ C \\ n = 0{\cdot}5\ \ ch/h \end{array}\right[\quad G_L = 0{\cdot}86\,\frac{W}{m^3{}^\circ C}\ ;\ r_L = 1{\cdot}26$$

Fig 36 Heat losses and heat gains on a typical "N kW house"

V-4.2 Heat Pumps Design Capacity and Supplementary Heating

It is well known that it is not economically justified to meet peak heat demand by means of a heat pump, because if the heat pump is designed to meet the maximum heating load, it will be over-sized for most of the season. (59)

The Electricity Research Council has estimated that in the U.K. any domestic space heating unit must meet at least 65% of the heat loss of a building when the outside ambient temperature is -1°C. A peaking system relying on electric power or fossil fuel must be provided to meet the remainder of the space heating load. This is in order that supplementary heat usage will depend more on living patterns than on the weather. However, heat pumps designed to meet lower percentages of the conventional maximum required capacity can adequately cover the average seasonal heating requirements in most areas of the U.K.: Normally, a conventional minimum ambient temperature of -1°C is adopted as the "design temperature" for heat need calculations, while the average temperature for the heating season, from October to April inclusive, remains higher than 3°C in most places of the country. This, indirectly implies that a moderate fraction of the peak power, continuously supplied, is sufficient to deliver a large fraction of the total seasonal space heating load. This can be demonstrated by the following example, in which the seasonal space heating requirements have been calculated for a typical 5 kW house, as it is shown in Figure 36, located at three different areas of the U.K.

Table V-9 (calculated with values from Figure 35), and Figure 37, plotted with values from that table, show seasonal space heating requirements for

Table No.V-9(a). Average domestic space heating requirements for a 5 kW house, with a volume of 365 m^3 and n = 0.5 air changes per hour, during the heating season in North East Scotland area; Z = 212 days; Gc = 0.685 $W/m^3{}^\circ C$; G_L = 0.86 $W/m^3{}^\circ C$; r_L = 1.26; $(r_L-1)/r_L$ = 0.21; Q_{LSe} = Total heat requirement per season

Month	Degree Days	$\bar{T}o$=15.5-(Do/Z)	Fabric Losses		Ventilation Losses		Total Heat Losses			Q_{LSe} =	
			Qc/V	Qc	Qv/V	Qv	Q_L/V	Q_L	$Q_L/24.V.Z$	$\Sigma Q_L/24VZ$	$Q_L/24.Z$
	Do	°C*	kWh/m^3	kWh	kWh/m^3	kWh	kWh/m^3	kWh	W/m^3	W/m^3	kW
October	199	9.1	3.26	1190	0.85	309	4.11	1499	5.70		2.1
November	322	4.8	5.27	1925	1.37	500	6.65	2425	8.93	5.70	3.3
December	385	3.1	6.31	2302	1.64	598	7.95	2900	10.68	14.63	3.9
January	399	2.6	6.54	2386	1.70	620	8.23	3006	11.07	25.31	4.0
February	360	2.6	5.90	2152	1.53	560	7.43	2712	11.05	36.38	4.0
March	347	4.3	5.68	2075	1.48	539	7.16	2614	9.63	47.43	3.5
April	272	6.4	4.45	1626	1.16	423	5.61	2049	7.79	57.06	2.8
Total	2284	4.7	37.41	13656	9.72	3551	47.14	17207	9.26	64.85	3.4

* $\bar{T}o$ = Average outside ambient temperature in the area.

Table No.V-9(b). Average domestic space heating requirements for a 5 kW house, with a volume of 365 m^3 and n = 0.5 air changes per hour, during the heating season in the Midlands area;
Z = 212 days; Gc = 0.685 $W/m^3 °C$; G_L = 0.86 $W/m^3 °C$; r_L = 1.26; $(r_L-1)/r_L$ = 0.21;
Q_{LSe}= Total heat requirement per season

Month	Degree Days	$\bar{T}o$=15.5-(Do/Z)	Fabric Losses		Ventilation Losses		Total Heat Losses			Q_{LSe} =	
			Qc/V	Qc	Qv/V	Qv	Q_L/V	Q_L	$Q_L/24.V.Z$	$\Sigma Q_L/V.24.Z$	$Q_L/24.Z$
	Do	°C*	kWh/m^3	kWh	kWh/m^3	kWh	kWh/m^3	kWh	W/m^3	W/m^3	kW
October	172	9.9	2.82	1029	0.73	267	3.55	1296	4.77		1.7
November	290	5.8	4.75	1734	1.23	449	5.98	2183	8.31	4.77	3.0
December	362	3.8	5.93	2164	1.54	562	7.47	2726	10.04	13.08	3.6
January	375	3.4	6.14	2241	1.60	584	7.74	2825	10.50	23.12	3.8
February	335	3.5	5.49	2004	1.42	518	6.91	2522	10.29	33.62	3.7
March	322	5.1	5.28	1927	1.37	500	6.65	2427	8.93	43.91	3.3
April	235	7.7	3.85	1405	1.00	365	4.85	1770	6.74	52.84	2.5
Total	2091	5.6	34.25	12501	8.91	3252	43.16	15753	8.48	59.58	3.1

* $\bar{T}o$ = Average outside ambient temperature in the area.

Table No.V-9(c). Average domestic space heating requirements for a 5 kW house, with a volume of 365 m^3 and n = 0.5 air changes per hour, during the heating season in the Thames Valley area; Z = 212 days; Gc = 0.685 $W/m^3{}^\circ C$; G_L = 0.86 $W/m^3{}^\circ C$; r_L = 1.26; $(r_L-1)/r_L$ = 0.21; Q_{LSe} = Total heat requirement per season

Month	Degree Days	$\bar{T}o$=15.5-(Do/Z)	Fabric Losses		Ventilation Losses		Total Heat Losses			Q_{LSe} =	
			Qc/V	Qc	Qv/V	Qv	Q_L/V	Q_L	$Q_L/24.V.Z$	$\Sigma Q_L/24.V.Z$	$Q_L/24.Z$
	Do	°C*	kWh/m^3	kWh	kWh/m^3	kWh	kWh/m^3	kWh	W/m^3	W/m^3	kW
October	132	11.2	2.16	789	0.56	205	2.72	994	3.66		1.3
November	256	7.0	4.19	1531	1.09	398	5.28	1929	7.34	3.66	2.7
December	336	4.7	5.50	2009	1.43	522	6.93	2531	9.32	11.00	3.4
January	349	4.2	5.72	2087	1.49	542	7.20	2629	9.68	20.32	3.5
February	304	4.6	4.98	1818	1.29	473	6.27	2291	9.34	30.00	3.4
March	285	6.3	4.67	1704	1.21	443	5.88	2147	7.91	39.34	2.9
April	199	8.9	3.26	1190	0.85	309	4.11	1499	5.70	47.25	2.1
Total	1861	6.7	30.48	11127	7.93	2893	38.41	14020	7.54	52.95	2.7

* $\bar{T}o$ = Average outside ambient temperature in the area.

North East Scotland, the Midlands, and the Thames Valley areas. It can be seen in Figure 38, which has been plotted with data from the calculated table V-10, that a heat pump with a design capacity rated at 52% of the conventional peak heating load, supplies about 79%, 86% and 97% of the seasonal heat need in the North East Scotland area, Midlands area and Thames Valley area respectively. Such a heat pump, as it can be seen in Figure 37, could cover the base load in the period Mid-November to end of March in all three cases (Shaded area), and could be employed at part load outside that period. A design capacity rated at only 60% of the conventional peak load could be sufficient to supply 92% of the overall space heating requirements of the season, in the case of North East Scotland area, and 100% in the cases of Midlands and Thames Valley areas.

The selection of moderate heating capacities permits one to use smaller and therefore less costly units. The equation (V-28), which gives the estimated energy consumption for the heat pump as a function of the total degree days Do in a given period, must now be modified by a factor ξ, that takes into account the percentage of the overall seasonal heating requirements rated as heating capacity for the system, in order to calculate the seasonal energy consumption by the heat pump.

$$\dot{W}_{HP} = \xi \ (0.024 \ G.V.Do)/COP(H) = kWh/period \qquad (V-29)$$

Also, the selection of a moderate capacity for the heat pump system makes it easier to increase the radiator area in order to have lower fluid distribution temperatures, which is beneficial for the heat pump performance when under-floor heating systems or hot-water radiator systems are used.

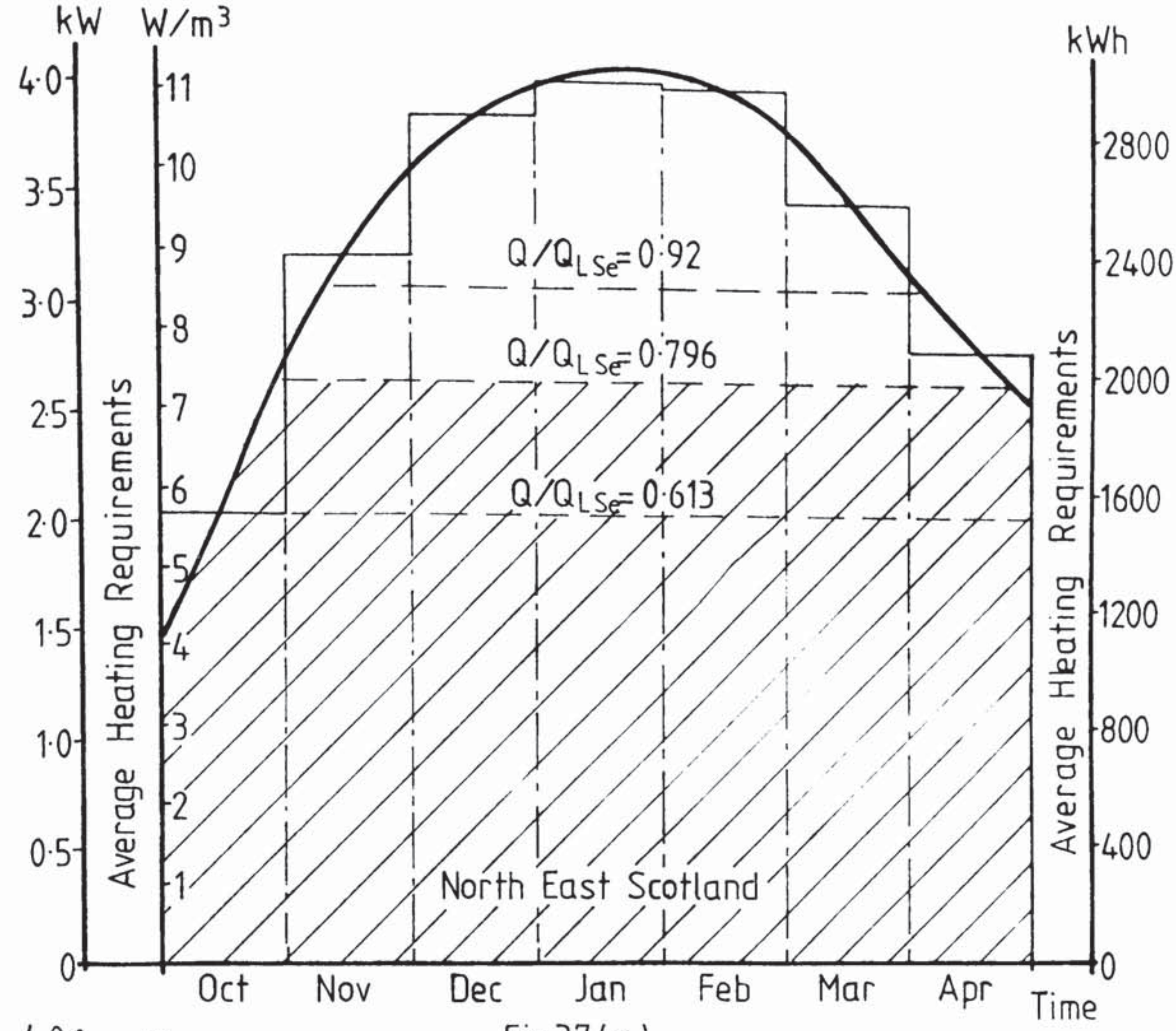


Fig 37(a)

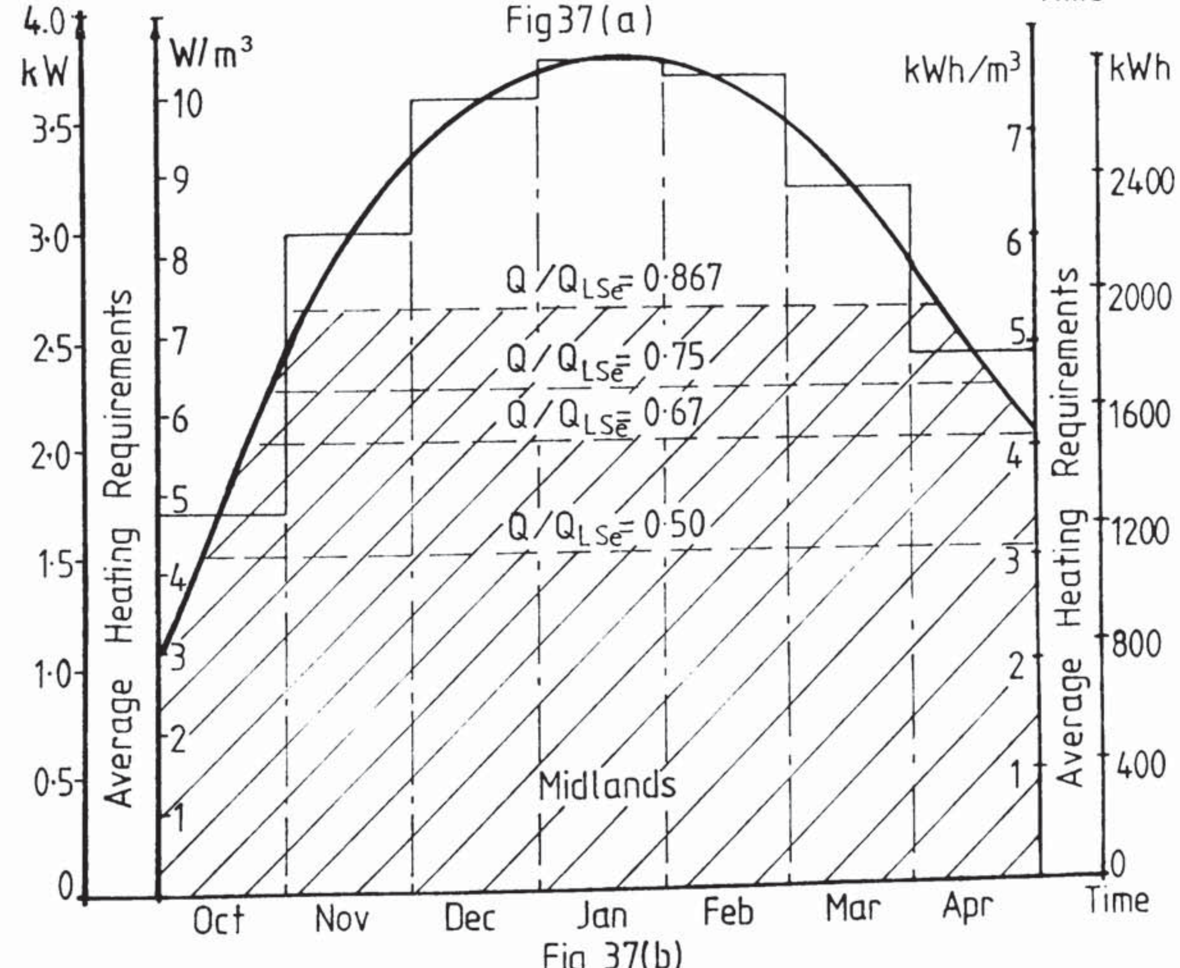


Fig 37(b)

Fig 37 Domestic Space Heating Requirements for a "5kW house" with a volume of 365m³ and 0·5 air changes per hour, during the year in:(a) North East Scotland,(b) Midlands,(c) Thames Valley

Table No. V-10. Design capacity and percentage of the heating load delivered on a 5 kW house at three different areas of the U.K. $G_L = 0.86$; $V = 365\ m^3$, $r_L = 1.26$; $Gc = 0.685$. Maximum capacity rated at T.amb. = $-1°C$ (i.e. $Q_{Lmax} = G_L\ (15.5-(-1)) = 14.2\ W/m^3$)

$\frac{Q\ Design}{Q_L\ (Max)}$	Design Capacity $= Q_{DESIGN}$	Heat Delivered per Season = Q	Heat Delivered/Total heat requirement = Q/Q_{LSe}		
			Thames Valley 51° 30′N	Midlands 52°26′N	N. East Scotland 57°10′N
	W/m^3	W/m^3	$Q_{LSe} = 52.95 W/m^3$	$Q_{LSe} = 59.58 W/m^3$	$Q_{LSe} = 64.85 W/m^3$
0.10	1.42	9.87	0.186	0.166	0.152
0.20	2.84	19.88	0.375	0.334	0.306
0.30	4.26	29.82	0.563	0.500	0.460
0.40	5.68	39.76	0.751	0.667	0.613
0.45	6.38	44.69	0.844	0.750	0.689
0.50	7.10	49.70	0.939	0.834	0.766
0.52	7.37	51.65	0.975	0.867	0.796
0.54	7.66	53.63	1.012	0.900	0.827
0.60	8.52	59.64	1.126	1.010	0.919
0.63	9.00	63.00	-	-	0.971
0.70	9.94	69.58	-	-	1.073
0.80	11.36	79.52	-	-	-
0.90	12.78	89.46	-	-	-

Where a heat pump system directly replaces the original boiler in old buildings, heated by standard hot-water radiators, the heat capacity level to be assigned to the heat pump system must be optimized. The water temperature requested to transfer that capacity to the conditioned space through the existing radiators must be established taking into account that the hot water output of a heat pump is typically at 45-55°C, whereas for a domestic boiler it is at 65-75°C. The lower temperature will mean a reduced heat output, at full load, from the existing radiators, perhaps by as much as one-half. However, this may not be a serious problem: existing heating systems almost always have their radiators oversized relative to the design heat loss of the rooms and it is recommended that steps to reduce heat losses (below the original design values) should have had priority over the change to a heat pump system.

In new buildings, low temperature emitters can be used and heat pumps are particularly suited to under-floor heating systems because of the large effective surface areas. If a heat pump is to be used in conjunction with a fossil-fuel boiler, that covers the peak demand the problems of reduced heat emission can be avoided. The two appliances become alternatives with valving arrangements to isolate the heat pump from the hot water circuit when the boiler is operated.

V-4.3 Space Heat Load v Mean External Temperature

If the heat source is the ambient air then, as the outdoor temperature falls, the heat loss from a building (kept at a constant indoor temperature) rises, and the maximum available output from the heat pump falls. The "balance point B.P." is the outside temperature at which the heat losses from the building are equal to the heat pump

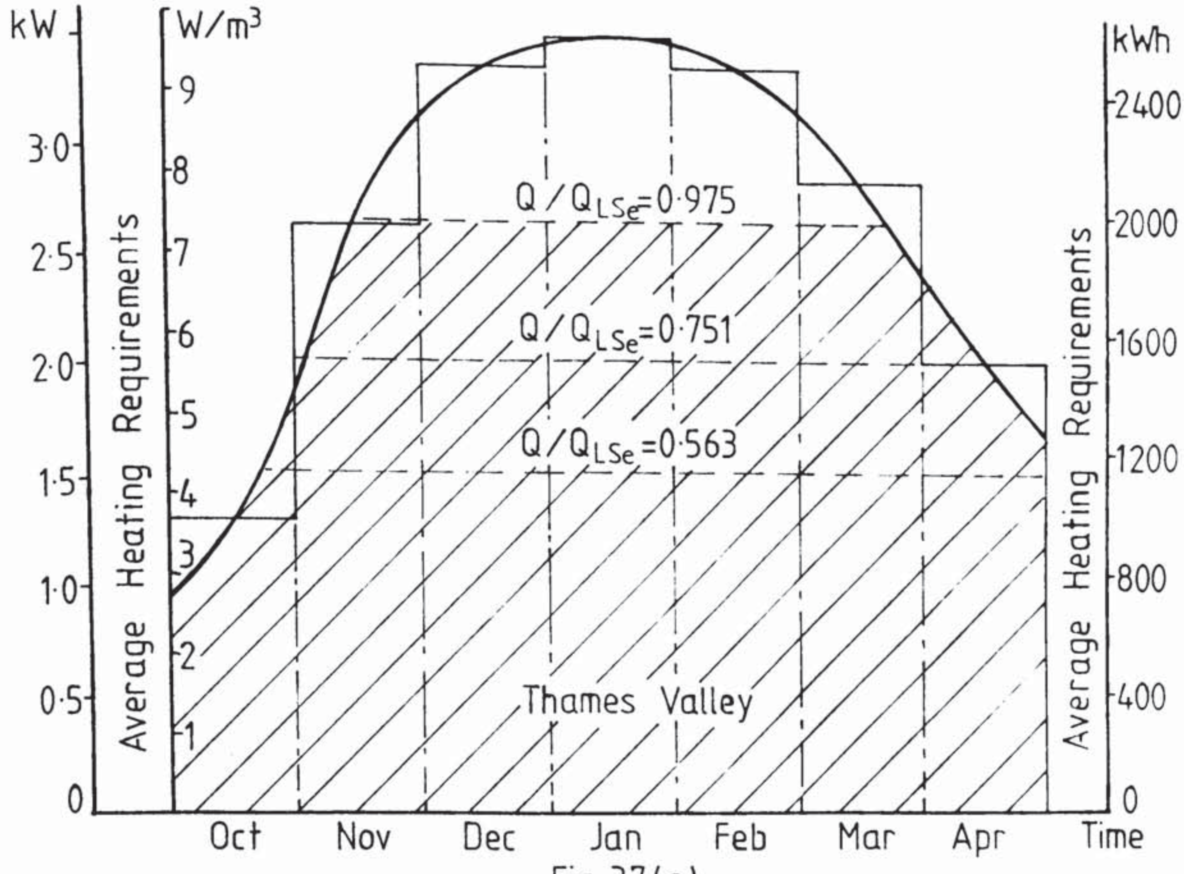
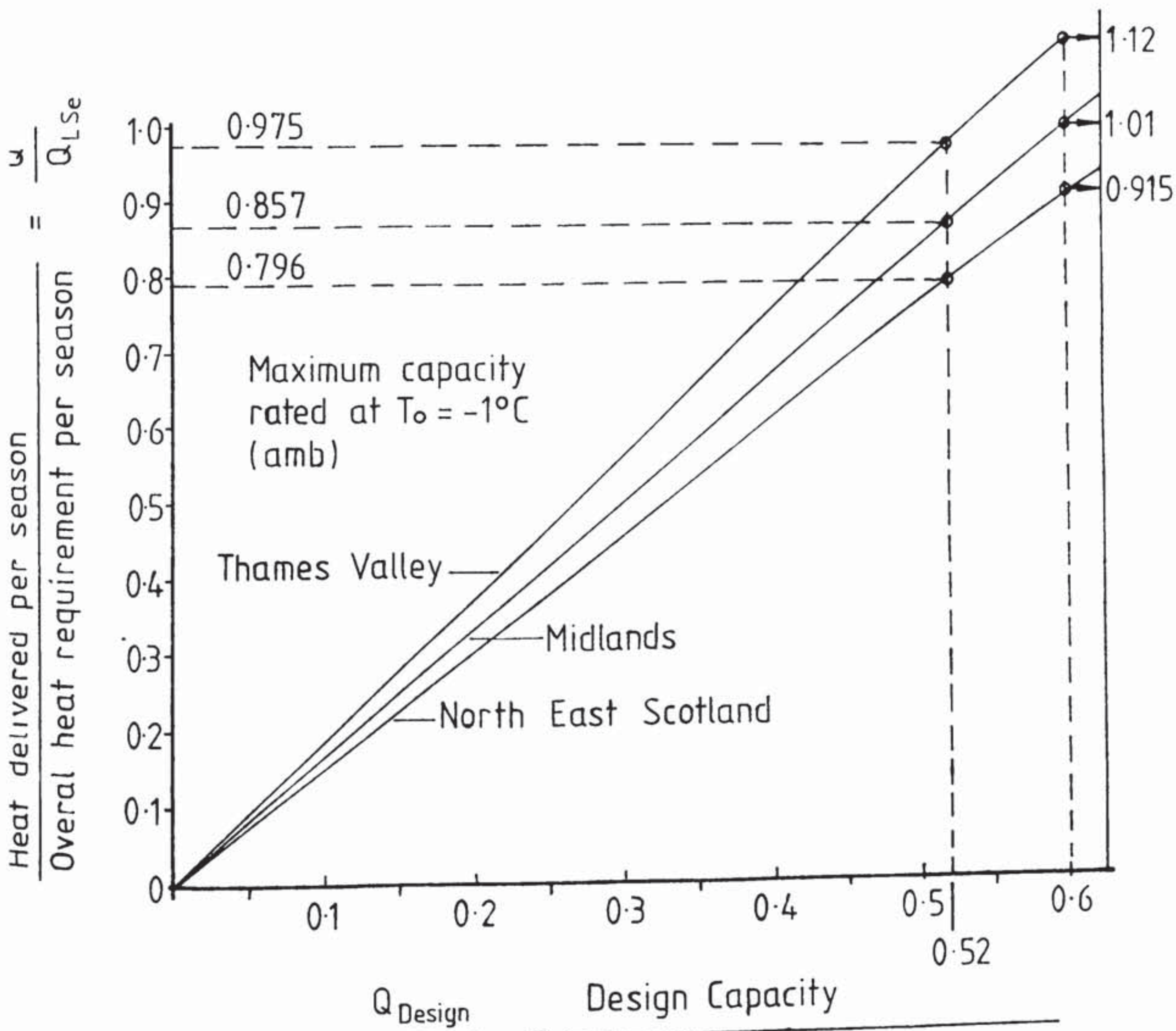


Fig 37(c)



$$\frac{Q_{Design}}{Q_{L\,max}} = \frac{\text{Design Capacity}}{\text{Maximum Requested Capacity}}$$

Fig 38 Seasonal heat delivery as a function of design capacity output during unoccupied periods and equal the heat pump output plus the internal gains during the occupied periods.

Logic tells one that the heat pump should only be sized such as to meet all of the heat demand down to some selected balance point, which is usually at air temperatures between 0 and 5°C. At higher balance points there will be a substantial auxiliary or boost heat requirement on cold days. In the U.K. a suitable choice would be between 0 and 3°C as, over most of the country the monthly average mean external temperature is higher than 3°C (see Table V-9), and the mean temperature falls to less than zero on only few days in the year (see Table IV-5). On average, most places have only about one day per year with mean external temperature lower than -3°C (57). Where the only boost envisaged is from electric resistance heating, a lower balance-point is needed, so that the effective coefficient of performance of the whole installation is not uneconomically low.

Thus, the size of the heat pump has to be optimised to minimise capital cost while maximising the utilization of the installation.

The gross and the net heating requirements of the structure can be superimposed on the heat pump capacity curves, as shown in Figure 39 to obtain the respective balance point. This figure shows the structure gross heat loss, calculated by means of the equation (V-7) at various outdoor temperatures for a "5 kW house", with the characteristics shown on Figure 36 and with the inside comfort temperature assumed constant at 18.3°C.

$$Q_L = 250 + 0.36 \times 0.5 \times 365 \ (18.3 - T_o)/1000 = kW$$

$$Q_L = [(W/°C) + (Wh/m^3°C)\ (ch/h)\ (m^3/ch)]°C/1000 = kW$$

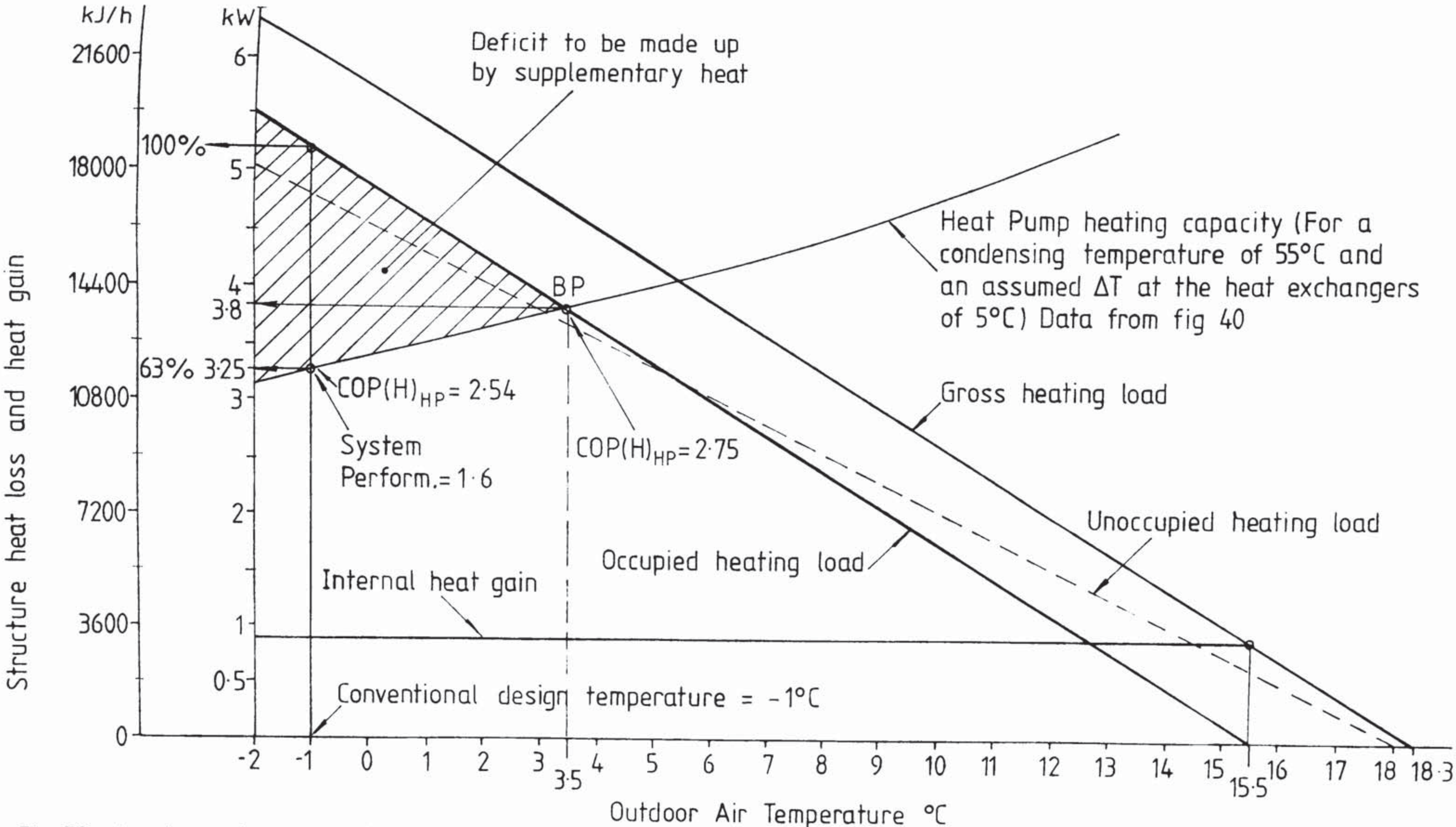


Fig 39 Load requirements at various outdoor temperatures for a 5kW house with a volume of 365m^3 and 0·5 air changes per hour compared with heating capacity of a heat pump.
B.P. = Balance Point

If one assumes that no external heating is required above an outdoor temperature of 15.5°C, then the internal heat gain: from people, appliances and solar, can be estimated as:

$$Q_{gain} = [250 + 0.36 \times 0.5 \times 365]\ (18.3 - 15.5)/1000 = kW$$

$$Q_{gain} = 0.9\ kW$$

The calculated gross heat loss, less the useful heat gain, equals the heat loss during the occupied period. In addition the calculated gross heat loss with the reduced ventilation requirements (neglecting infiltration) equals the heat loss during the unoccupied period. The balance point, as well as the supplementary heating requirements can also be obtained from Figure 39.

It is interesting to note, in connection with Figure 39 and 40 that only small heat pump systems are required to meet the average needs for domestic space heating in most areas of the U.K.

As it can be seen on these figures, the motor power input of the heat pump system employed, varies between 1.2 and 1.5 kW. At the conventional design condition of -1°C outdoor temperature, the heat pump has a heating capacity of 3.25 kW, which represents 63% of the design heating load for this type of house. At this design condition, the motor input is only 1.28 kW, giving a COP(H) of 2.54 for the heat pump and 1.6 for the whole system if one takes into account that 1.95 kW are required as supplementary heating.

The heat pump system presents a balance point, at an outdoor temperature of 3.5°C, with a heating capacity of 3.8 kW for a motor input of only 1.38 kW. This represents a COP(H) of 2.75 for the heat pump and for the system at this point.

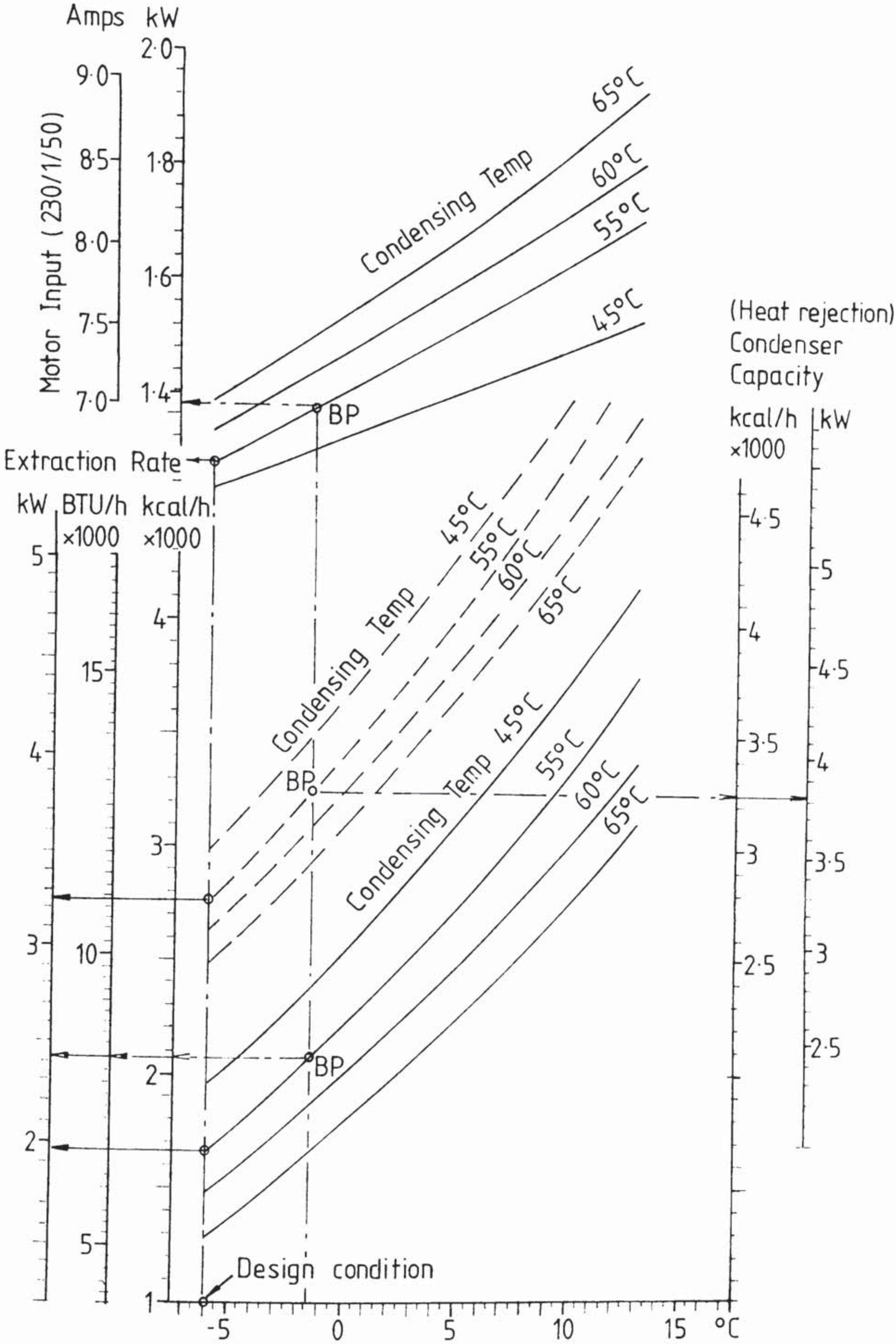


Fig 40 AK26E Heat Pump Capacities. Prestcold Manufacturers data. (For Refrigerant R 22)

According to the averages in Figure 37 the balance point temperature and the balance point capacity obtained with this small heat pump could easily cover the space heating requirement, with the average peak load included, of a "5 kW house", when it is located at the Thames Valley area or at the Midlands area. It is for these reasons that heat pumps, using outdoor air as the heat source, should be receiving more favourable consideration in most areas of the U.K.

Alternatively, for areas of the U.K. with higher space heating requirements, a heat pump can itself be regarded as an auxiliary form of heating, capable of providing the full requirement during Spring and Autumn only. This is particularly so, for water-source machines, which may be unable to operate in very frostly conditions.

V-4.4 The Effect of Storage

The choice of tariff for an electrically-driven heat pump also needs consideration. If a dwelling is likely to be heated intermittently it may consume such a large a proportion of its electricity at full rate, that an "off-peak" tariff heat pump may not be justified. On the other hand where substantial thermal storage can be provided the off-peak tariffs should be chosen. However, it must be remembered that off-peak times also tend to be those of lowest air temperature, so the heat pump's performance will be somewhat reduced. Furthermore, most off-peak tariffs offer an eight-hour period of availability at most, and if this is to provide the same amount of heat as an unrestricted (24 hour) tariff, one can expect to need a heat pump of about three times the heat output. However building occupancy and any differences between night time and day time temperatures which are required internally will affect individual situations. So, thermal storage, with off-peak topping up, could provide only the boost heating in cold spells. Extensive thermal storage becomes necessary in the case of solar assisted heat pump systems, even if charged at full rate, in order to modulate the heating capacity and to reduce reduce external supplementary heating. But, the economic implications of these systems are not yet clear.

Thus, the application design philosophy lies in deciding the balance temperature, and the balance capacity of the heat pump system, the amount of boost heating to install, and whether or not a measure of thermal storage should be included. It is a question of sizing: in general large units are more efficient than small ones, but it would be preferable to run a small unit continuously rather than a higher rated unit intermittently, which could remain oversized most of the season as it has been seen in this chapter.

THus heat pump systems, and solar assisted heat pump systems integrated with thermal store, with all factors considered, usually prove to be very practical and satisfactory future heating systems.

CHAPTER VI

INTEGRATED SOLAR ASSISTED HEAT PUMP AND THERMAL STORE SYSTEM

VI-1. Increasing the Air Evaporator Temperature

VI-1.1 The roof as a solar radiation absorber

Despite the inherent problems connected with air evaporators mentioned in previous sections (see Section III-2.3), these are still the most convenient and in some cases the only suitable means for low-grade heat absorption for domestic use. Normally the heat pump is a split system with the evaporator situated outside the house (outdoor coil) with the condenser (indoor coil) inside the house with insulated vapour and liquid refrigerant lines. In new houses where a south-facing solar roof can be incorporated it is possible to arrange for warmed air flowing under the solar roof to pass to the inlet of the heat pump, as illustrated in the article by S.J. Leach (60). The major modifications (and cost) necessary to convert an existing roof of a house to a solar roof of sufficient area to provide for space and water heating would be a disincentive to most owner occupiers unless a thermal store is included in the heating system.

The normal roof (without specially designed panels) of existing houses with few structural alterations can be used as a solar radiation absorber to raise the air temperature at the evaporator of the heat pump. This method, which is being investigated at the University of Aston, takes advantage of the diffuse radiation, and as it has already been mentioned, diffuse radiation in the U.K. constitutes a high proportion of the total. The system collects diffuse radiation on the north facing roof as well as diffuse and direct radiation on the south facing one.

VI-1.2 Description of the System

The basic system is illustrated in Figure 41 for a house with half its roof facing south and half facing north and comprises three main components: 1) An energy absorber, 2) A heat pump and 3) A thermal store. The energy absorber "A" is the existing roof, whether of slate or tile. Air at an external ambient temperature T_{f1} is drawn from the eaves through channels "B" formed between roof tiles and roof felt, fixed to the tile support structure (61). The temperature of the air is raised during its passage to the apex by heat transferred from the roof which absorbs solar radiation. The warmed air entering the roof space at the apex is passed over the evaporator of a split heat pump at a temperature T_{RS} and then rejected to the outside at a reduced temperature T_{f3}. In general $T_{RS} > T_{f1} > T_{f3}$. Thus, the heat extracted at the evaporator Q_2 is obtained from three low grade sources: (a) Q_{Air}, ambient air (exhausted air is below ambient), (b) Q_{Solar}, heat derived from solar radiation and (c) Q_R, recovered heat losses from the house to the roof space.

The heat pump "C" is split, with the evaporator section "D" situated on the roof space and the main condenser "E" at or below ground level. A subsidiary heat exchanger "F" is located adjacent to a small water storage tank which feeds the main domestic hot-water supply tank. An indirect coil is used to heat the water to 55°C. Figure 41 shows the heat pump located in the roof space but it could

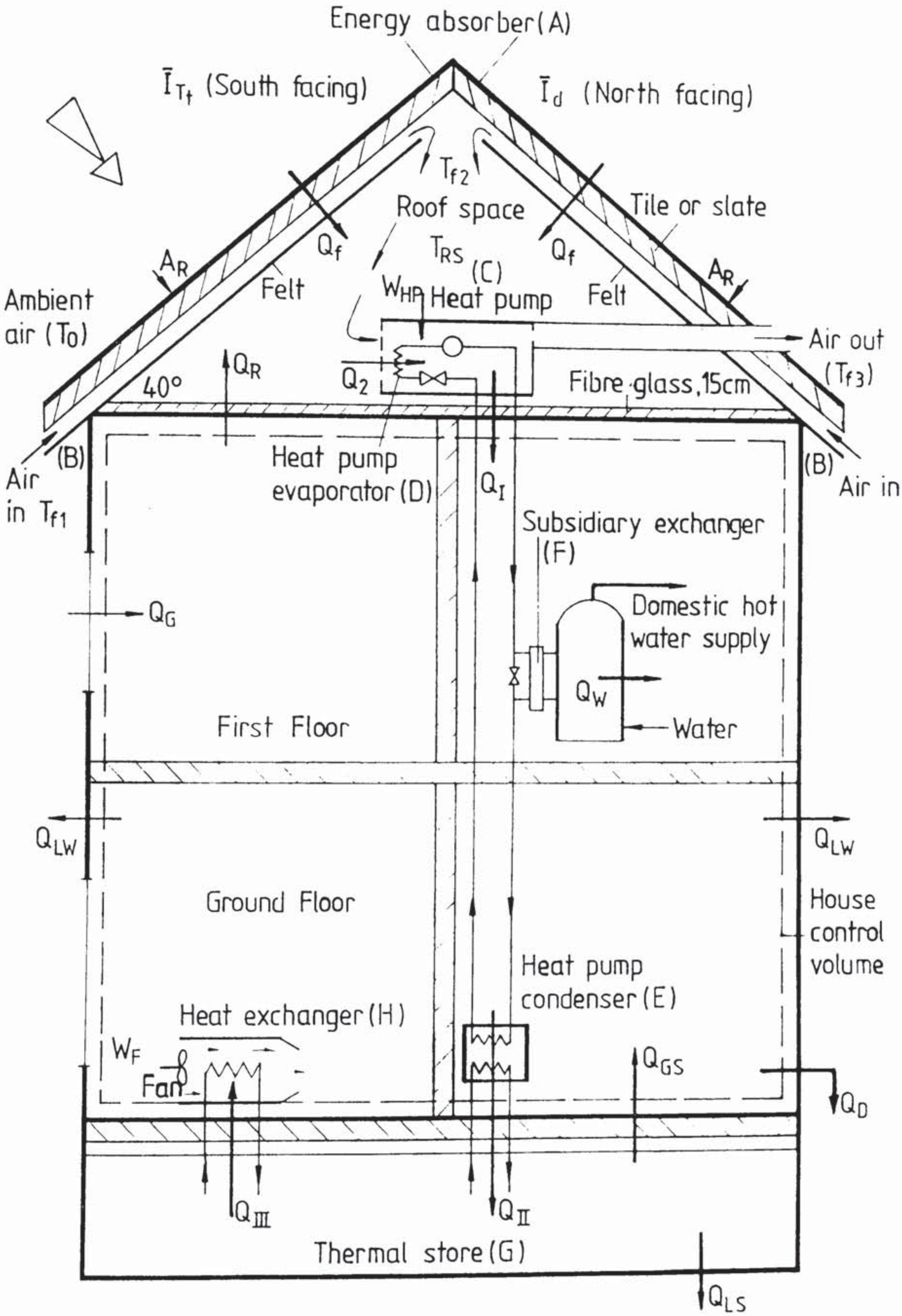


Fig 41 Domestic heating system utilising solar energy, a heat pump and thermal store - Heat balance.

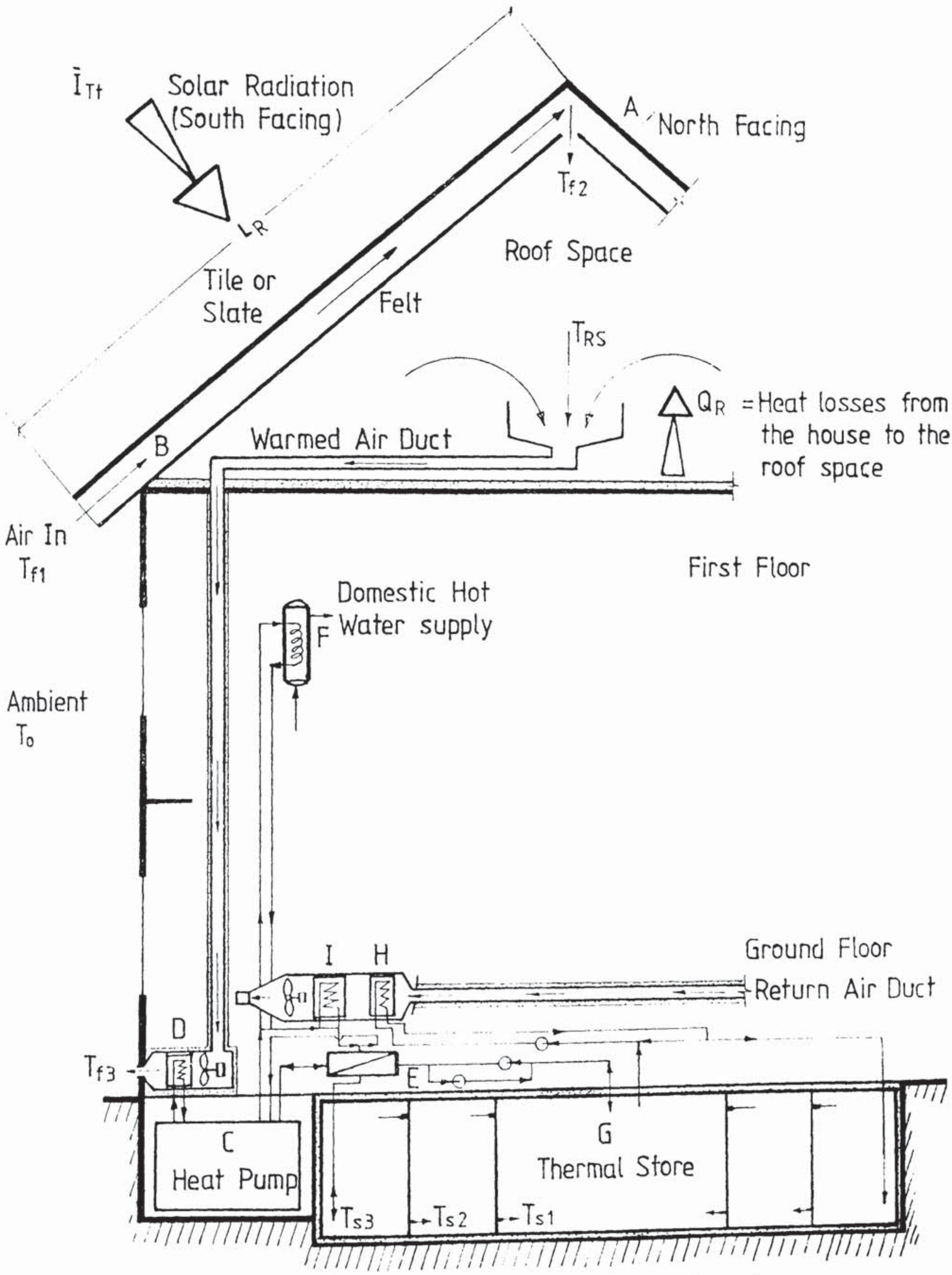
be located elsewhere provided an insultated duct connects the evaporator of the heat pump to the roof space. The intermediate heat store "G" of the system is located below ground level, and a Water to Air heat exchanger "H" permits heat to be extracted from store for space heating.

In the United Kingdom the heat pump would be required to operate principally in the heating mode, but in circumstances where air conditioning is required a reversible pump could operate in the cooling mode. Figure 42 shows the same system with the heat pump located on the ground floor, and with an additional heat exchanger "I" which permits the operation of the system as an"Water to Air" heat pump in order to produce both heating or cooling. In this figure, the system also includes a different distribution on the storage tank, which could permit the reduction of thermal store heat losses. Later on, the different modes of operation of the system, represented in Figure 42 will be described.

VI-1.3 Operational Considerations

The efficiency of roof materials is obviously not as good as that of a specially designed solar panel but the exposed area is much greater than that normally available for installation of solar panels in existing houses. Consider a period during the winter. The maximum available radiation for a south facing roof (at a latitude of 51.5°, inclined at 40° to the horizontal for the month of January is in the region of 317 MJ/m^2 (88 kWh/m^2). If it is assumed that only about 12.5% of this is absorbed by the roof and transferred to the air flowing under the slates or tiles and into the heat pump then the pump receives some 39.6 MJ/m^2 (11 kWh/m^2). Indirect

Fig 42 Integrated Solar Assisted Heat Pump and Thermal Store System for heating or cooling



A = Energy Absorber
B = Roof Gap
C = Heat Pump
D = Outdoor Coil (Evaporator)
E = Main Condenser
F = Subsidiary heat exchanger
G = Thermal Store
H = Air to Water heat exchanger
I = Indoor Coil (Air to Air heat pump)

radiation from each square metre of roof facing north add another 5.8 MJ/m^2 (1.6 kWh/m^2). Exact figures are not available for the effectiveness of roofing materials as absorbers but this is the subject of a separate experimental programme at the University of Aston. The mass of a conventional roof is approximately 50 kg/m^2 (3) so that 42 kJ (0.012 kWh) would be required to raise this mass by 1 degree C. Assuming the average daily radiation figures apply for a period of about 6 hours during January, sufficient direct energy is absorbed by a south-facing tile roof in one hour to raise the temperature by more than 3°C and to maintain it on average for a further 5 hours with an ambient outside air temperature of 0°C. This factor is significant when considering the operation of the heat pump, as has previously been mentioned.

By taking advantage of the energy absorbed (both direct and diffuse) by the roof during hours of daylight it is possible to operate the heat pump with a higher COP(H) than would be otherwise possible by taking air direct from the outside to the evaporator. This in turn reduces the electrical energy to drive the compressor. If the heat pump is programmed to be operating under the most advantageous conditions the average COP(H) can be optimised for the whole heating season provided some provision is also made for heat storage.

VI.2 Heat Exchange Characteristics in the Roof Structure

The final mathematical expressions of the following theoretical analysis will permit the determination of the efficiency η of the roof structure in transferring the incident solar radiation to the air stream, in terms of measurable parameters.

A section of the air channel is illustrated in Figure 43 and the control volume for the analysis, in each of the two considered cases, is shown in Figure 44 (case I) and in Figure 45 (case II).

In a simplified approach the following conditions were assumed:

1. The roof is of infinite width and length so that heat transfer occurs only in the X direction.

2. The tiles conforming the roof are homogeneous with constant material properties.

3. The surface heat transfer coefficients (h) are constant.

4. The solar absorptivity of the outside surface of the roof is independent of angle of incidence and is constant.

5. The pattern of the variation of T_O and I are identical with time on consecutive days.

6. The internal thermal environment of the roof space is constant. And finally,

7. Case I: Tiles as radiation absorbers are considered thin enough so that their entire mass is at the same temperature T_R during the heat transfer process. Also, in this case, the felt temperature T_b is the same in its inside and outside surfaces.

   Case II: Tiles and felt are considered thick enough so that a temperature gradient exists between the outside and inside surfaces.

In this second case, T_R and T_b are the surface temperature, of tiles and felt respectively, adjacent to the air stream.

In the development of this mathematical analysis, the following symbols were used:

T_o Temperature of air outside the house

T_{os} Effective black-body temperature of the sky

T_{f1} Air temperature at the entrance of the air channel

T_{f2} Air temperature at the exit of the air channel

T_f Mean fluid (air stream) temperature

T_R Mean roof absorber (tile) temperature in case I or temperature of the tile-surface adjacent to the air stream in case II.

T_{Ro} Temperature of tile-surface adjacent to outside air in case II.

T_b Mean felt temperature in case I, or temperature of felt-surface adjacent to air stream in case II

T_{bo} Temperature of felt-surface adjacent to roof space in case II

T_{RS} Temperature of the ambient in the roof space.

T_{WR} Average temperature of interior surfaces of the roof space reflecting radiation to the outside surface of the felt.

L_R Channel (roof) length (m)

Lw Channel (roof) width (m)

A_R Half roof area = Lw L_R = (m^2)

h Surface heat transfer coefficients: hc = Convection, hr = Radiation, ho = Combined convection and radiation for wall-outside surface.

U Overall heat transfer coefficient:

Uup = Upward heat loss coefficient between the roof absorber (tile) and the outside air

U_b = Rear heat loss coefficient between the felt-layer and roof-space

U_L = Heat loss coefficient between the roof structure and outside ambient, including back losses ($U_L = Uup + U_b$)

Uo	Overall heat transfer coefficient between the air stream and the outside space
F_R'	Roof absorber efficiency factor = Uo/U_L
F_R''	Roof flow factor
k_R, k_b	Heat conduction coefficients of tiles and felt respectively
$\bar{I}_{Tt}$	Incident solar radiation (W/m^2) (Equation IV-21)
α	Net radiation absorption coefficient
$\dot{Q}a$	Heat absorbed by roof-tile (W)
$\dot{Q}u$, $\dot{Q}u'$	Heat transferred from the absorber tile to the control volume of the fluid and from the controlvolume of the fluid to the felt surface respectively (W)
$\dot{Q}_b$	Rear heat losses to the roof space (W)
$\dot{Q}_f$	Useful heat taken out by the air stream (W)
$\dot{Q}_{IR}$	Heat lost from the roof absorber (tiles) to the outside space.
ε	Emissivity of surfaces: $\varepsilon_R \equiv$ tiles, $\varepsilon_b \equiv$ felt.
C	Specific heat: $Cp \equiv$ Humid air, $C_R \equiv$ roof (tiles); $C_b \equiv$ felt (kJ/kg °C)
ρ	Density: $\rho a \equiv$ air, $\rho_R \equiv$ roof, $\rho_b \equiv$ felt
ν	Air speed (m/s)
$\dot{M}_f$	Air mass flow rate kg/h
G_f	Flow rate of air per unit area of roof (kg/m^2h)
σ	Stefan - Boltzmann constant

VI-2.1 Energy balance on the roof

The analysis of the heat tranfer model would require simultaneous solution of all the energy balance equations from top roof surface to rear felt surface in the roof space. The whole structure could then be treated as an air panel.

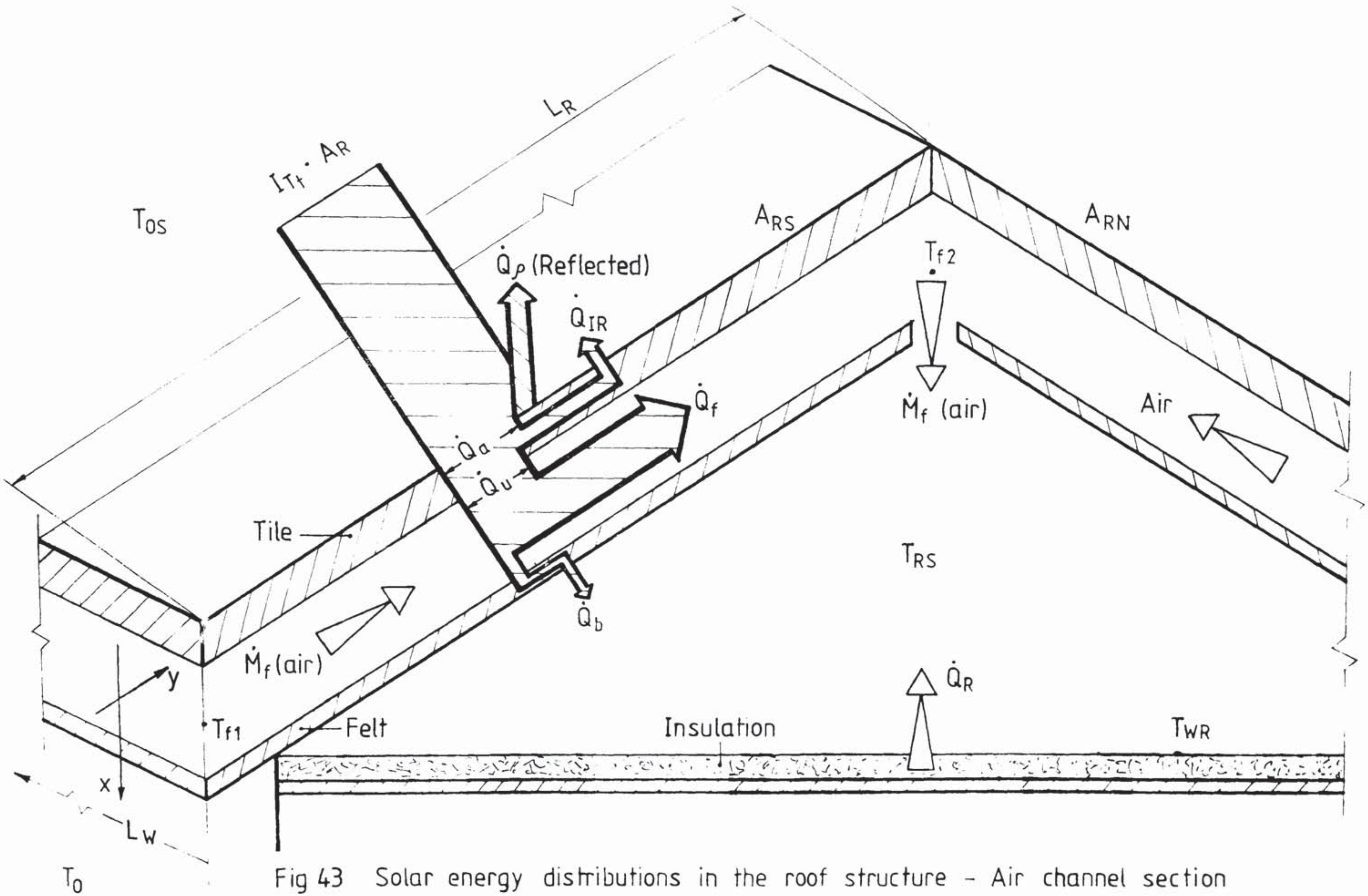


Fig 43 Solar energy distributions in the roof structure - Air channel section

VI-2.1.1 Case I.

The main three energy balances on the roof are: (see Fig. 44)

1) On the absorber roof (tiles):

(a) The solar radiation incident upon the absorber roof $\bar{I}_{Tt}$ can be calculated from equation (IV-21), and the energy absorbed by the tiles as:

$$\delta \dot{Q}a = (\alpha \bar{I}_{Tt}) \, Lw \, \delta y \qquad \text{(VI-1)}$$

(b) The heat lost from the roof absorber (tile) to the outside space can be given by:

$$\delta \dot{Q}_{IR} = Uup \, (T_R - T_o) \, Lw \delta y \qquad \text{(VI-2)}$$

(c) The amount of heat transferred form the roof absorber to the control volume of the air stream will be:

$$\delta \dot{Q}u = \delta \dot{Q}a - \delta \dot{Q}_{IR} = [\alpha \bar{I}_{Tt} - Uup \, (T_R - T_o)] \, Lw \delta y$$

and as a function of heat transfer coefficient at the inside surface of the roof absorber

$$\delta \dot{Q}u = [hc \, (T_R - T_f) + hr \, E \, (T_R - T_b)] \, Lw \delta y \qquad \text{(VI-3)}$$

Thus, the net balance on the roof absorber system will be:

$$\delta \dot{Q}a = \delta \dot{Q}_{IR} + \delta \dot{Q}u$$

$$\alpha \bar{I}_{Tt} \, Lw \delta y = [Uup \, (T_R - T_o) + hc \, (T_R - T_f) + hr \, E \, (T_R - T_b)] \, Lw \delta y$$

$$I_I = Uup_I \, (T_R - T_o) + hc \, (T_R - T_f) + hr \, E \, (T_R - T_b) \qquad \text{(VI-4)}$$

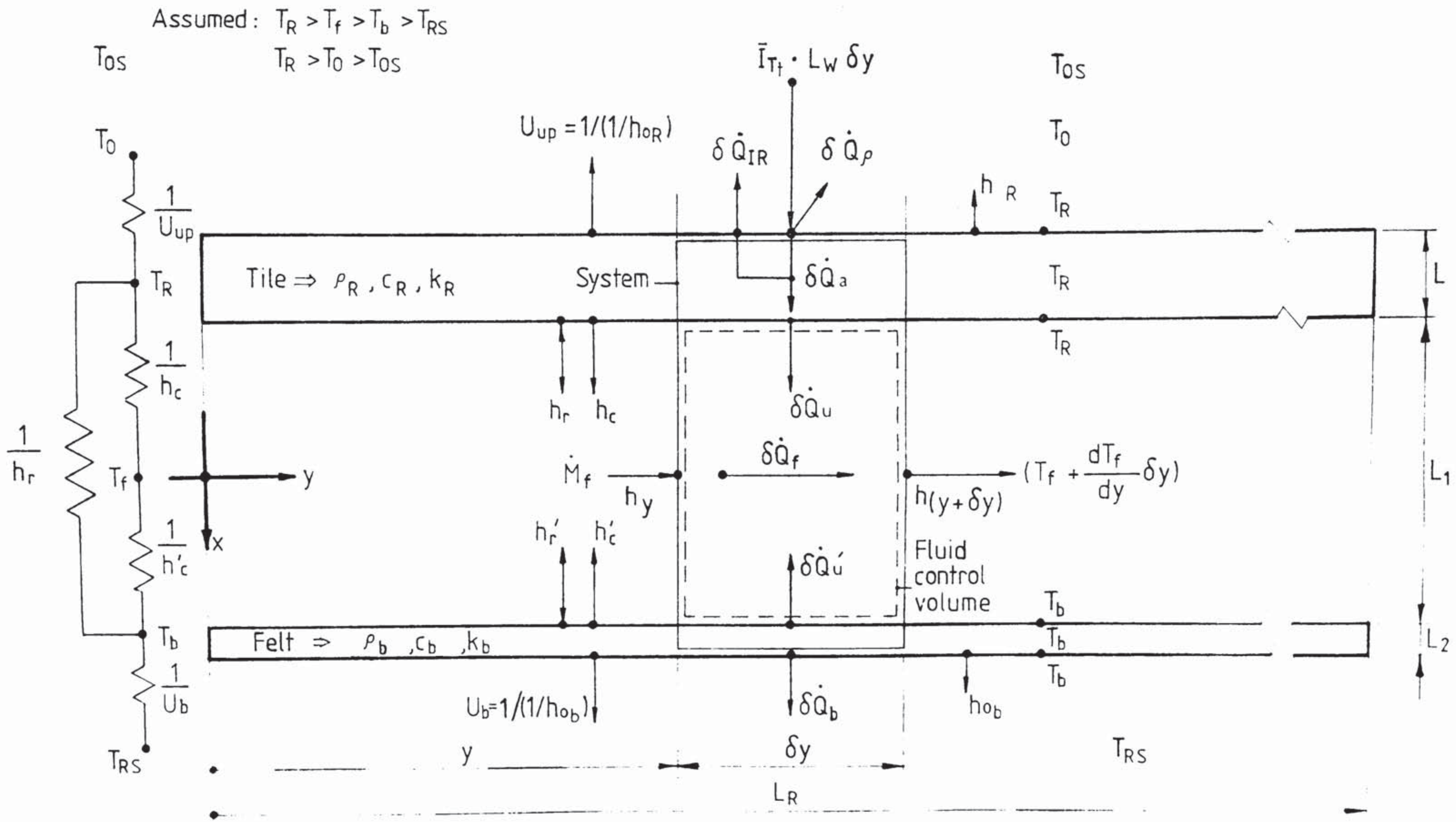


Fig 44 Heat exchanges in the roof - Fluid control volume - Case I

In this equation:

a) $I_I = \alpha I_{Tt}$; $Uup_I = Uup\ (\text{case I}) = 1/(1/ho_R)$

$ho_R = hc_o + \varepsilon_R\ hr_o\ [(T_R - T_{os})/(T_R - T_o)]$

$hr_o = \sigma(T_R^4 - T_{os}^4)/(T_R - T_{os})$ = Radiation heat transfer coeff.

$hc_o = hw$ = convection heat transfer coefficient related to wind speed. ($W/m^2 °C$). From reference (62) for plate surfaces

$hw = 5.7 + 3.8\ \nu$ when $\nu < 5$ m/s.

b) The emissivity factor E, between the inside roof absorber surface and the inside felt surface can be calculated by:

$E = 1/[(1/\varepsilon_R) + (1/\varepsilon_b) - 1]$ and the radiation heat transfer coefficient in this case as:

$hr = \sigma\ (T_R^4 - T_b^4)/(T_R - T_b) = \sigma\ (T_R^2 + T_b^2)\ (T_R + T_b)$

2) On the control volume of the air stream

$$\delta\dot{Q}_f = \delta\ \dot{Q}u + \delta\dot{Q}u'$$

a) Energy transported by the fluid (air): The flow of fluid is steady and may also be regarded as one-dimensional.

$$\delta\dot{Q}_f = \dot{M}_f\ [h_f\ (y + \delta y) - h_f\ (y)]$$

$$\delta\dot{Q}_f = \dot{M}_f\ [C_p\ T_f\ (y + \delta y) - C_p\ T_f\ (y)]$$

$$\delta\dot{Q}_f = \dot{M}_f\ C_p\ [(T_{f(y)} + (dT_f/dy)\ \delta y) - T_{f(y)}]$$

$$\delta\dot{Q}_f = C_p\ \dot{M}_f\ (dT_f/dy)\ \delta y \qquad \text{(VI-5)}$$

b) Energy received by the control volume of the air stream

$$\delta\dot{Q}u + \delta\dot{Q}u' = [hc(T_R - T_f) + hr\ E\ (T_R - T_b)]\ Lw\ \delta y$$
$$+ [hc'(T_b - T_f) + E\ hr'\ (T_b - T_R)]\ Lw\ \delta y$$

but, $hr = \sigma(T_R^4 - T_b^4)/(T_R - T_b) = \sigma(T_b^4 - T_R^4)/(T_b - T_R) = hr'$

So, that:

$$\delta\dot{Q}u + \delta\dot{Q}u' = [hc\ (T_R - T_f) + hc'\ (T_b - T_f)]\ Lw\ \delta y \qquad \text{(VI-6)}$$

The net energy balance on the control volume will be:

$$Cp\ \dot{M}_f\ (dT_f/dy)\ \delta y = [hc\ (T_R - T_f) + hc'\ (T_b - T_f)]\ Lw\delta y \qquad \text{(VI-7)}$$

Values of hc and hc' can be calculated from the Nusselt number $Nu = hc\ L_1/k_R$ and $Nu = hc'\ L_1/k_b$ respectively, with the Nusselt number as a function of Reynolds number (Re) and Prandtl number (Pr).

In this case empirical relations for hc and hc' in forced convection must be developed. One must consider the class of air flow, between the parallel surfaces with different shape and for different type of tiles and felts forming the channels.

As a first approach one could consider the correlation for fully developed turbulent flow between flat plates with one of them heated, derived from Kays' data, given in reference (51).

$$Nu = 0.0158\ Re^{0.8} \qquad \text{(VI-8)}$$

Which can be applied for the ratios of flow length to hydraulic diameter (L_R/D) higher than 10.

For the laminar fully developed case with once surface insulated, Kays gives Nussel numbersof 5.4 and 4.9 for constant heat flux and constant temperature conditions at the surfaces respectively (51).

3) <u>On the felt.</u> (see Fig. 44)

In this case one has:

$$-\delta\dot{Q}u' = \delta\dot{Q}\ \text{back}$$

a) The rear heat losses from the felt surface are:

$$\delta\dot{Q}_b = U_{bI}\ (T_b - T_{RS})\ Lw\,\delta y \qquad \text{(VI-9)}$$

Where:

$$U_{bI} = 1/ho_b = U_{back}\,\text{case I} \qquad \text{(VI-10)}$$

$$ho_b = hc_b + [\varepsilon_b\ hr_b\ (T_b - T_{WR})/(T_b - T_{RS})]$$

$$hr_b = \sigma(T_b^4 - T_{WR}^4)/(T_b - T_{WR})$$

Thus, the net energy balance on the felt will be:

$$-[hc'\ (T_b - T_f) + Ehr'\ (T_b - T_R)]\ Lw\,\delta y = U_{bI}\ (T_b - T_{RS})\ Lw\,\delta y$$

$$\boxed{Ehr\ (T_R - T_b) = hc'\ (T_b - T_f) + U_{bI}\ (T_b - T_{RS})} \qquad \text{(VI-11)}$$

b) It has been assumed in Figure 44 that,

$$T_R > T_f > T_b > T_{RS}$$

and, $T_R > T_o > T_{os}$

However, heat lost $\dot{Q}_R$ from the house and heat gained $\dot{Q}_f$ from warmed air pass through the roof space, so, it is likely that T_{RS} becomes equal to T_b, in which case heat losses from the back of the roof (felt) would be reduced to zero. If one assumes that $T_b \sim T_{RS}$, then $U_b << U_{up}$ and U_L will be equal to U_{up}. This ignoring the small amount of radiation between the felt and the base of the roof space. Furthermore, if the object is to calculate the amount of heat absorbed by the roof, in order to evaluate its efficiency in collecting solar radiation, the small amount of heat losses occurring through the felt to the roof space do not matter. In this case most

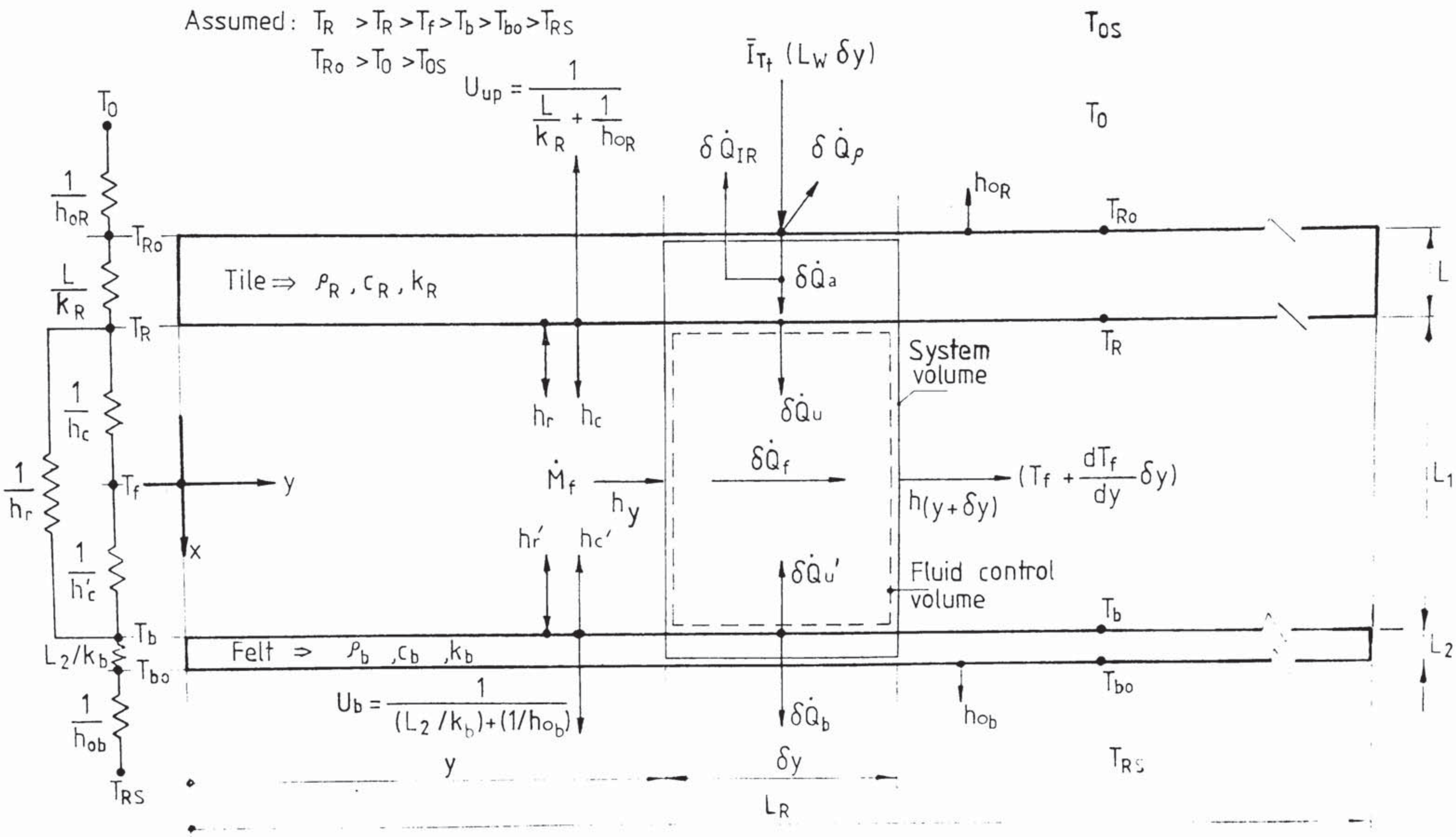


Fig 45.- Heat exchanges in the roof - Fluid control volume - Case II

of these losses will be recovered and at the end the general balance will be the same. If the back heat losses are taken into account, the algebra becomes extremely complicated and the resulting expressions for F_R', U_L, T_f, etc. unnecessarily difficult to handle.

If $T_b \simeq T_{RS}$, equation (VI-11), reduces to:

$$\underline{Ehr\ (T_R - T_b) = hc'\ (T_b - T_f)} \tag{VI-12}$$

Solution of the simultaneous equations (VI-4), (VI-7) and (VI-11), or (VI-12) if one considers $Uup_I \gg U_{bI}$ and $U_{LI} = Uup_I$, will produce the differential equation which defines the temperature variation of the air stream in this case I.

VI-2.1.2 Case II

The main energy balances on the roof are (see Figure 45):

1) On the Absorber Roof (tiles)

a) Solar radiation absorbed:

$$\delta \dot{Q}a = \alpha \bar{I}_{Tt}\ Lw\delta y$$

b) The up heat losses $\dot{Q}_{IR}$

$$\delta \dot{Q}_{IR} = ho_R\ (T_{Ro} - T_o)\ Lw\delta y \tag{VI-13}$$

c) The heat transferred from the roof to the control volume of the air stream

$$\delta\ \dot{Q}u = \delta\ \dot{Q}a - \delta\dot{Q}_{IR} = \delta\ \dot{Q}\ \text{conduction through tile}$$

$$[\alpha \bar{I}_{Tt} - ho_R\ (T_{Ro} - T_o)]\ Lw\ \delta y = (k_R/L)(T_{Ro} - T_R)\ Lw\delta y \tag{VI-14}$$

and as a function of heat transfer coefficients at the adjacent surface to the air stream:

$$\delta \dot{Q}u = [hc\ (T_R - T_f) + hr\ E\ (T_R - T_b)]\ Lw\ \delta y \qquad (VI\text{-}15)$$

Thus, the net balance on the roof absorber will be:

$$[\ \alpha \bar{I}_{Tt} - ho_R\ (T_{Ro} - T_o)]\ Lw\ \delta y = [hc\ (T_R - T_f) + hr\ E\ (T_R - T_b)]\ Lw\ \delta y \qquad (VI\text{-}16)$$

From equation (VI-14) one can have:

$$(k_R/L)\ (T_{Ro} - T_R) = \alpha\ \bar{I}_{Tt} - ho_R\ (T_{Ro} - T_o)$$

$$ho_R\ T_{Ro} = [ho_R\ L/(k_R + ho_R L)]\ \alpha\ \bar{I}_Tt + [k_R\ ho_R/(k_R + ho_R L)]\ T_R + ho_R\ [ho_R L/(k_R + ho_R L)]\ T_o$$

If one replaces $ho_R\ T_{Ro}$ into the left hand side of the equation (VI-16), this side A becomes:

$$A = \alpha \bar{I}_{Tt} - \alpha \bar{I}_{Tt}\ [ho_R L/k_R + ho_R L)] - [k_R\ ho_R/(k_R + ho_R L)]\ T_R + T_o\ ho_R\ [ho_R L/(k_R + ho_R L)] + ho_R\ T_o$$

$$A = \alpha \bar{I}_{Tt}\ [k_R/(k_R + ho_R L)] - [k_R\ ho_R/(k_R + ho_R L)]\ T_R + [ho_R\ k_R/(k_R + ho_R L)]\ T_o$$

But, $ho_R\ k_R/(k_R + ho_R L) = 1/[(L/k_R) + (1/ho_R)] = Uup$

$$A = [Uup/ho_R]\ \alpha\ \bar{I}_{Tt} - Uup\ [T_R - T_o]$$

Equation (VI-16) becomes then:

$$[Uup/ho_R]\,\alpha\,\bar{I}_{Tt} = Uup\,[T_R - T_o] + hc\,(T_R - T_f) + hrE(T_R - T_b)$$

and finally:

$$\boxed{I_{II} = Uup_{II}\,[T_R - T_o] + hc\,[T_R - T_f] + hrE\,(T_R - T_b]} \qquad \text{(VI-17)}$$

In this equation:

a) $I_{II} = (Uup_{II}/ho_R)\,\alpha\,\bar{I}_{Tt}$

$Uup_{II} = Uup\ (\text{case II}) = 1/[(L/k_R) + (1/ho_R)]$

$Uup_{II} = ho_R\,k_R/(k_R + ho_R L)$

$ho_R = hc_o + [\,\varepsilon_R\,hr_o\,(T_{Ro} - T_{os})/(T_{Ro} - T_o)]$

$hr_o = \sigma(T_{Ro}^{\,4} - T_{os}^{\,4})/(T_{Ro} - T_{os})$

hc_o = hw = convection heat transfer coefficient related to wind speed ($W/m^2\,°C$) = $5.7 + 3.8\,\nu$ when $\nu < 5$ m/s, (62).

b) The emissivity factor E and the radiation heat transfer coefficient hr, are the same values as in case I.

2) <u>On the control volume of the air stream</u>

The control volume of the air stream in case II is exactly the same as that in case I, for this reason the net energy balance on the control volume of this case II is similar to equation VI-7.

$$\boxed{Cp\,\dot{M}_f\,(dT_f/dy)\,\delta y = [hc\,(T_R - T_f) + hc'\,(T_b - T_f)]\,Lw\,\delta y} \qquad \text{(VI-18)}$$

3) <u>On the felt layer</u> (see Figure 45)

The energy balance in this case will be:

$$-\,\delta\dot{Q}u' = \delta\dot{Q}_{\,back}$$

a) The rear heat losses from the felt surface are in this case:

$$\delta \dot{Q}_b = U_{bII} (T_b - T_{RS}) L w \delta y$$

Where

$$U_{bII} = 1/[(L_2/k_b) + (1/ho_b)] = U_b \text{ case II}$$

$$ho_b = hc_b + [\varepsilon_b hr_b (T_{bo} - T_{WR})/(T_{bo} - T_{RS})]$$

$$hr_b = \sigma (T_{bo}^4 - T_{WR}^4)/(T_{bo} - T_{WR})$$

$$hc_b = 5.7 + 3.8\,\nu \text{ for } \nu < 5 \text{ m/s, (62)}$$

And the net energy balance on the felt will be:

$$Ehr (T_R - T_b) = hc' (T_b - T_f) + U_{bII} (T_b - T_{RS}) \qquad \text{(VI-19)}$$

b) It has been assumed on Figure 45 that:

$$T_{Ro} > T_R > T_f > T_b > T_{bo} > T_{RS}$$

and, $T_{Ro} > T_o > T_{os}$

The same reasons for nil heat losses from the back of the roof given in case I, apply here in case II, so that, the equation (VI-19) for the net balance on the felt, will be reduced to:

$$Ehr (T_R - T_b) = hc' (T_b - T_f) \qquad \text{(VI-20)}$$

with $U_{upII} >> U_{bII}$ and $U_{upII} = U_{LII}$.

Solution of the simultaneous equations (VI-17), (VI-18), and (VI-20) will produce the differential equation which defines the temperature variation of the air stream in this case II.

VI-2.2 <u>Temperature T_f of the air stream at a distance y from the inlet end of the roof channel</u>

Simultaneous equations (VI-4), (VI-7) and (VI-12) of case I are similar to the simultaneous equations (VI-17), (VI-18) and (VI-20) of case II, respectively. For this reason a general case of simultaneous equations, in which $I_1 = I = I_2$ and $U_{LI} = U_L = U_{LII}$, can be established. This permits one to obtain a general solution of T_f, that can be applied in each particular case by introducing the correct parameters of the roof considered.

Thus, the simultaneous equations to be solved are:

$$(1) \quad I = U_L\,[T_R - T_o] + hc\,[T_R - T_f] + hr\,E\,[T_R - T_b] \qquad \text{(VI-21)}$$

$$(2) \quad (Cp\,\dot{M}_f/Lw)\,(dT_f/dy) = hc\,[T_R - T_f] + hc'\,[T_b - T_f] \qquad \text{(VI-22)}$$

$$(3) \quad 0 = hc'\,(T_b - T_f) - Ehr\,[T_R - T_b] \qquad \text{(VI-23)}$$

Solving the system (1), (2), (3) for T_b and T_R one obtains:

$$T_b = T_f + \left\{\left[(Cp\,\dot{M}_f/Lw)(dT_f/dy)[Ehr/(hc' + Ehr)] \div \left\{hc + [hc'\,Ehr/(hc' + Ehr)]\right\}\right]\right\} \qquad \text{(VI-24)}$$

$$T_R = T_f + \left\{\left[(Cp\,\dot{M}_f/Lw)\,(dT_f/dy)/\left\{hc + [hc'\,Ehr/(hc' + Ehr)]\right\}\right]\right\} \qquad \text{(VI-25)}$$

Let $Ux = hc + [hc'\,Ehr/(hc' + Ehr)]$

$$Ux = hc + \left\{1/[(1/hc') + (1/Ehr)]\right\}$$

Then, T_b and T_R will be:

$$T_b = T_f + (Cp\,\dot{M}_f/Lw)(dT_f/dy)\,[Ehr/(hc' + Ehr)]/Ux \qquad \text{(VI-26)}$$

$$T_r = T_f + (Cp\,\dot{M}_f/Lw)(dT_f/dy)/Ux \qquad \text{(VI-27)}$$

Substituting these into the equation (VI-21) and rearranging one obtains:

$$(Cp\,\dot{M}_f/Lw)[dT_f/dy] = [Ux/(Ux + U_L)][I - U_L\,(T_f - T_o)]$$

Let $F_R' = Ux/(Ux + U_L) = 1/[1 + (U_L/U_x)]$, so that the differential equation becomes:

$$dT_f/dy = (Lw\ F_R'/Cp\ \dot{M}_f)[I - U_L(T_f - T_o)] \tag{VI-28}$$

$$(1/U_L)dT_f/dy = -\ (Lw\ F_R'/Cp\ \dot{M}_f)[T_f - T_o - (I/U_L)]$$

$$dT_f/(T_f - T_o - (I/U_L)) = -\ [U_L\ Lw\ F_R'/Cp\ \dot{M}_f]\ dy$$

The general solution of this differential equation will be:

$$T_f - T_o - (I/U_L) = K.\exp.(-U_L\ Lw\ F_R'\ y/Cp\ \dot{M}_f)$$

If $y = 0$ then $T_f = T_{f1}$, so that K will be:

$$K = T_{f1} - T_o - (I/U_L)$$

Thus, the air temperature T_f at a distance X from the inlet end of the channel is found to be:

$$T_f = [T_{f1} - T_o - (I/U_L)]\ \exp.\ (-U_L\ Lw\ F_R'\ y/Cp\ \dot{M}_f) + T_o + (I/U_L)$$

$$T_f = (I/U_L) - [(I/U_L) - (T_{f1} - T_o)]\ \exp.\ (-U_L\ Lw\ F_R'\ y/Cp\ \dot{M}_f) + T_o \tag{VI-29}$$

The air temperature at the apex of the roof at a distance $y = L_R$ will be:

$$T_{f2} = [T_{f1} - T_o - (I/U_L)]\ \exp.\ (-U_L\ Lw\ F_R'\ L_R/Cp\ \dot{M}_f) + T_o + (I/U_L)$$

$$T_{f2} = [(T_{f1} - T_o) - (I/U_L)]\ \exp.\ (-U_L\ F_R'/G_f\ Cp) + T_o + (I/U_L) \tag{VI-30}$$

Where $G_f = \dot{M}_f/Lw\ L_R = kg/m^2h$

Equations VI-29 and VI-30 can be applied to the South or to the North facing roof, according to the value of I introduced.

VI-2.3 Useful heat transported by the air stream

The heat picked up by the air stream in half area of the roof (North or South, according to the value of I considered) is found to be

$$\dot{Q}_f = A_R \cdot G_f \, Cp \, [T_{f2} - T_{f1}] \qquad \text{(VI-31)}$$

$$\dot{Q}_f / A_R = G_f \, Cp \, [(T_{f1} - T_o - (I/U_L)) \exp. (-U_L \, F_R'/G_f \, Cp) + T_o + (I/U_L) - T_{f1}]$$

$$\dot{Q}_f / A_R = G_f \, Cp \, [-T_{f1} + T_o + (I/U_L)][1 - \exp (-U_L \, F_R'/G_f \, Cp)]$$

$$\boxed{\dot{Q}f / A_R = (G_f \, Cp/U_L)[I - U_L(T_{f1} - T_o)][1 - \exp (-U_L \, F_R'/G_f \, Cp)]} \qquad \text{(VI-32)}$$

The roof as a heat exchanger has an efficiency factor which is the ratio between the heat transfer resistance from the absorber tiles to the ambient air ($1/U_L$) and the heat transfer resistance from the fluid to the outside air ($1/Uo$). Thus F_R' is the ratio of these two heat transfer coefficients.

$$F_R' = (1/U_L)/(1/Uo) = Uo/U_L \qquad \text{(VI-33)}$$

$$Uo = F_R' \, U_L = U_L/(1 + (U_L/Ux))$$

$$Uo = 1/[(1/U_L) + (1/Ux)] \qquad \text{(VI-34)}$$

Substituting F_R' by Uo/U_L into equation (VI-32) one obtains:

$$\dot{Q}_f / A_R = F_R' \, (G_f \, Cp/Uo)[I - U_L(T_{f1} - T_o)] \cdot [1 - \exp (-Uo/G_f \, Cp)]$$

$$\frac{\dot{Q}_f}{A_R} = \frac{F_R' \, [1 - \exp(-Uo/G_f \, Cp)]}{(Uo/G_f \, Cp)} \, [I - U_L \, (T_{f1} - T_o)]$$

$$\dot{Q}f/A_R = F_R' \, F_R'' \, [I - U_L \, (T_{f1} - T_o)] = (W/m^2) \qquad \text{(VI-35)}$$

Where, F_R'' = flow factor, is given by:

$$F_R'' = \frac{1 - \exp(-Uo/G_f \, Cp)}{(Uo/G_f \, Cp)} \qquad \text{(VI-36)}$$

In case I:

$I = \alpha \, \bar{I}_{Tt} = [I_b + I_d + I\rho]_t \cdot \alpha$, South facing

$I = \alpha \, (I_d + I\rho]$, North facing

$$(\dot{Q}_f/A_R)_{south} = F_R' \, F_R'' \, [\alpha \, \bar{I}_{Tt} - U_{LI} \, (T_{f1} - T_o)] \qquad \text{(VI-37)}$$

In case II:

$I = (U_{LII}/h_{oR}) \, \alpha \, \bar{I}_{Tt}$, south facing

$$(\dot{Q}_f/A_R)_{south} = F_R' \, F_R'' \, [(U_{LII}/ho_R) \, \alpha \, \bar{I}_{Tt} - U_{LII} \, (T_{f1} - T_o)] \qquad \text{(VI-38)}$$

VI-2.3 Efficiency of the roof acting as a solar radiation absorber

Equation VI-35 is the same form used for solar panels and shows the possible means of increasing the amount of useful heat collected for a given solar radiation input. The amount of collected heat is increased by increasing F_R', F_R'', α and reducing U_L. α and U_L are related with the characteristics of the roof material and its upper surface. F_R' and F_R'' can be increased by increasing the mass flow rate per unit area and hc the convective heat transfer coefficient between the air stream and the inside surface of the tiles. The heat output can be determined once the radiation intensity, flow rate, ambient and inlet temperatures, and material properties of the roof are given. The solar radiation intensity is the main parameter identifying the performance of the roof absorber. The roof efficiency could be calculated from:

$$\eta = \dot{Q}_f / A_R \, \bar{I}_{Tt} = G_f \, Cp \, \Delta T / \bar{I}_{Tt}$$

Case I:

$$\eta = F_R' \, F_R'' \, [\alpha \, \bar{I}_{Tt} - U_{LI} \, (T_{f1} - T_o)] / \bar{I}_{Tt}$$

$$\eta = F_R' \, F_R'' \alpha - F_R' \, F_R'' \, U_{LI} \, (T_{f1} - T_o) / \bar{I}_{Tt}$$

$$\eta = C - K \, [(T_{f1} - T_o) / \bar{I}_{Tt}] \qquad \text{(VI-39)}$$

Where

$C = \alpha F_R' \, F_R''$ and $K = U_{LI} \, F_R' \, F_R''$

Case II:

$$\eta = F_R' \, F_R'' \, [(U_{LII}/ho_R) \, \alpha \, \bar{I}_{Tt} - U_{LII} \, (T_{f1} - T_o)] / \bar{I}_{Tt}$$

$$\eta = C' - K' \, [(T_{f1} - T_o) / \bar{I}_{Tt}] \qquad \text{(VI-40)}$$

Where

$C' = F_R' \, F_R'' \, \alpha (U_{LII}/ho_R) \, \bar{I}_{Tt}$; $K' = U_{LII} \, F_R' \, F_R''$

If the efficiency η is plotted against $(T_{f1} - T_o)$ then a single generalized curve would represent the roof performance. Air rising over the walls to the roof channel can be pre-heated by heat lost from the house and by reflected radiation over the wall from the surroundings, so that, T_{f1} will be different from T_o.

As the mass flow rate through the channel of the roof increases, the temperature rise through the absorber roof decreases. This causes lower losses and a corresponding increase in the useful energy gain, since the average temperature of the roof is lower. This increase in the useful energy gain must be reflected by an increase in the roof heat removal factor $F_R' \, F_R''$ as the mass flow rate increases. However, as the flow rate becomes very large, the temperature rise from inlet to outlet decreases towards zero and T_{f2}

becomes equal to T_{f1} or T_o, which is a disadvantage for the heat pump because that reduces its COP(H) and if T_o is lower than 4°C, frost on the evaporator could be present.

VI-3. Relative energy contributions from the various sources of low-grade heat.

The amount of power $\dot{Q}_2$ absorbed per cycle by the refrigerant on the evaporator of the heat pump will be equal to:

$$\dot{Q}_2 = \dot{Q}_{Air} + \dot{Q}_{Solar} + \dot{Q}_{Heat\ loss} \qquad (VI-41)$$

These three contributions are depicted in Figure 46 which gives the specific enthalpies of air at the different stages. Ambient air is represented by the point A and in passing through the roof channel one assumes no change in moisture content. The air arriving at the evaporator is represented by B and the move BC takes place at the evaporator. It can be seen, from Figure 46, that Q_{Air} is made up from sensible and latent heat contributions and can be calculated as:

$$\dot{Q}_{Air} = 2\ \dot{M}_f\ \Delta h_{Ambient}$$

where, $\dot{M}_f$ is the mass flow rate in each side of the roof. The other two contributions can be expressed by:

$$\dot{Q}_{Solar} = \dot{Q}_f\ (South) + \dot{Q}_f\ (North)$$

$$\dot{Q}_{Solar} = 2\ \dot{M}_f\ \Delta h_{Solar} = \dot{M}_f\ \Delta h_f\ (South) + \dot{M}_f\ \Delta h_f\ (North)$$

and the heat losses from the house to the roof space:

$$\dot{Q}_R = 2\ \dot{M}_f\ \Delta h_{Heat\ Loss}$$

According to Figure 46 these contributions are:

$$\dot{Q}_{Air} = 2 \dot{M}_f (h_A - h_C) \qquad (VI-42)$$

$$\dot{Q}_{Solar} + Q_R = 2 \dot{M}_f [h_B - h_A] \qquad (VI-43)$$

$$\dot{Q}_2 = 2 \dot{M}_f [h_B - h_C] \qquad (VI-44)$$

The University of Aston has monitored (61) a prototype system which has been installed in an occupied dwelling, in the Thames Valley Area. This is a three bedroomed detached house having a conventional delta-tiled roof, of 55 m^2 each side, facing South East and North West, with a pitch of 25°. The split heat pump is situated in the roof space and the thermal store comprising 22 m^3 of water is located below the ground floor.

The diagram in Figure 47, extracted from reference (61), shows typical temperature rises for air passing under the South East and North West facing roofs. Rises of up to 12°C have been measured on the South East side and up to 7°C on the North West side in April and October, with air flow rates of 740 litres/s. In January and February the corresponding rises can be between 2 and 3 degrees.

The curves in Figure 48 shows the variations of contributions from ambient, solar and heat loss to the roof space, in the prototype system, over the heating season 1977-1978 as it has been reported in reference (61). In those curves it was assumed a 100% heat recovery of the losses from the house to the loft during the operation of the heat pump. The effect of less than 100% recovery would be to increase the solar contribution as can be seen from Figure 46, since the sum of solar and heat loss contributions is

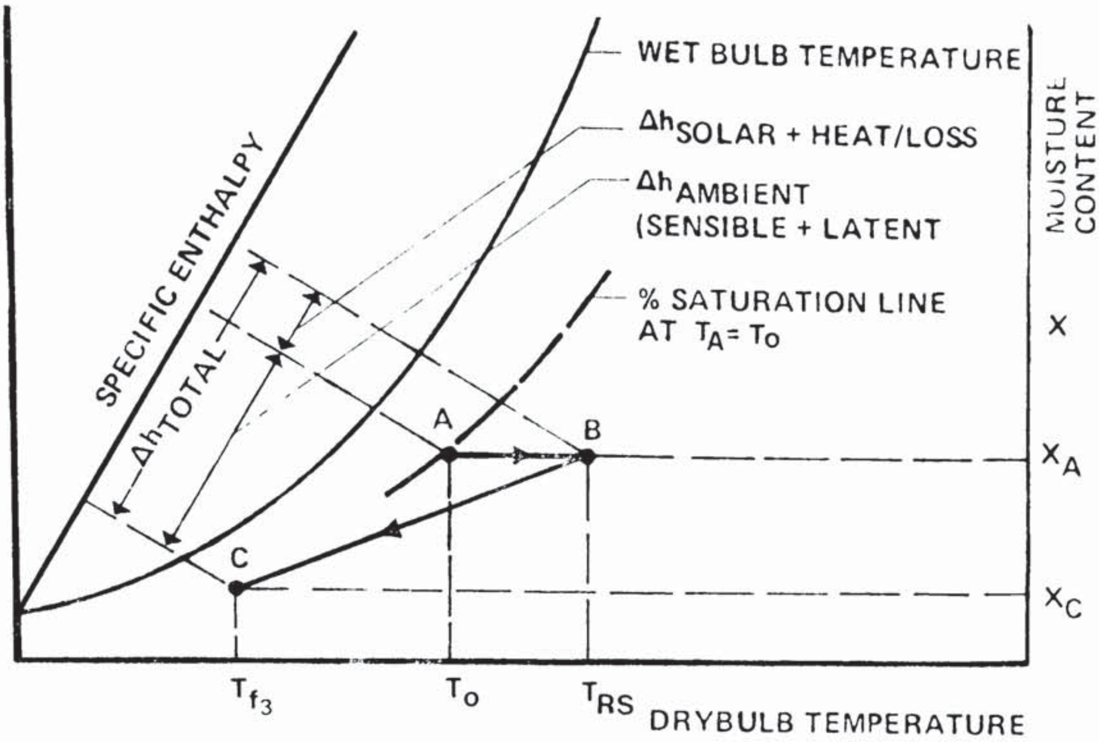


FIG 46. PSYCHROMETRIC CHANGES AS AIR PASSES THROUGH THE SYSTEM

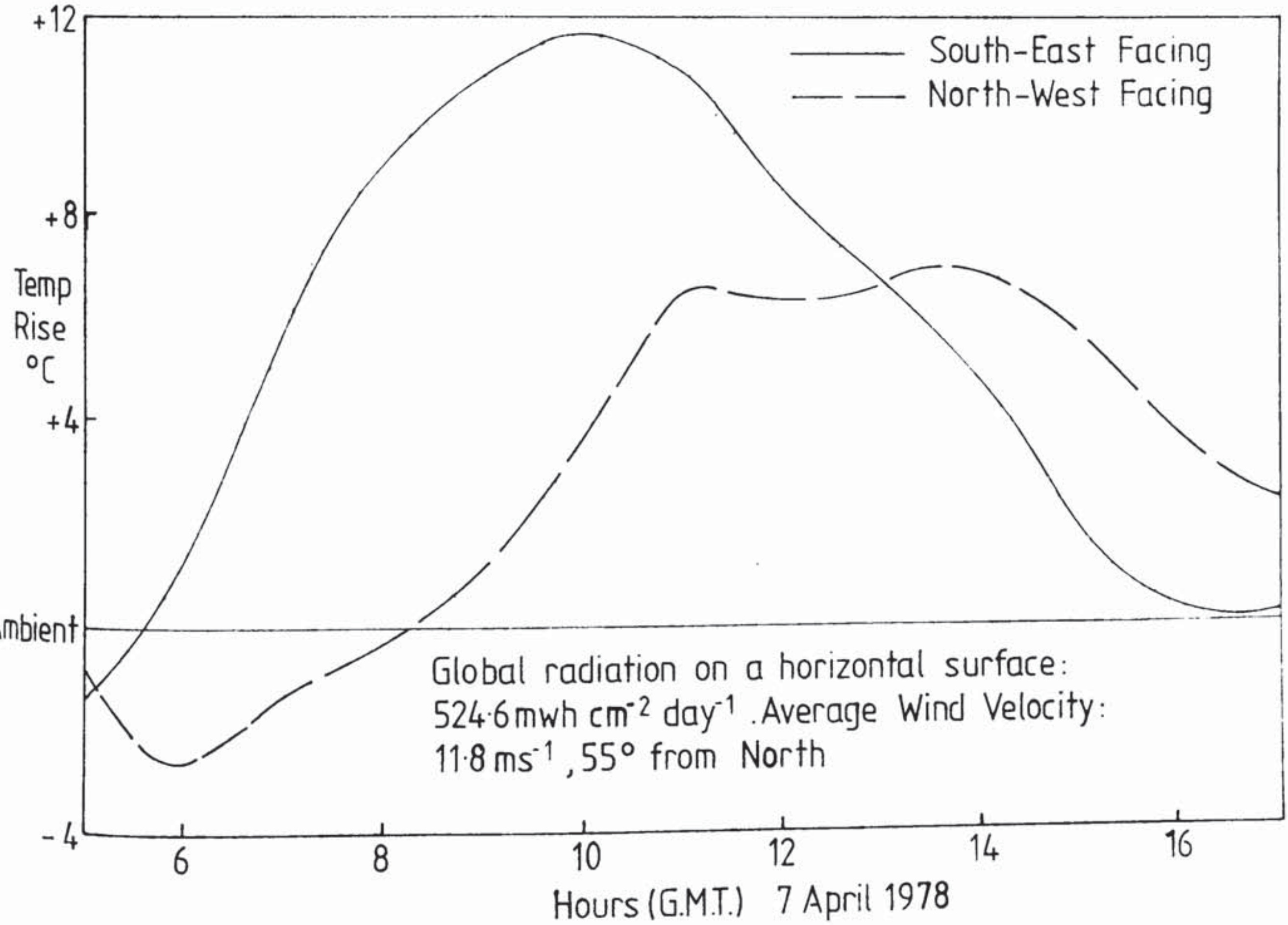


Fig 47 Air temperature rises, South-East and North-West facing sides of the roof

fixed for a given situation and is represented by the change in the air conditions from A to B. The reported curves are for assumed still wind conditions.

The average contributions over the season from either ambient, solar or heat loss is defined as:

$$\frac{\text{Total heat extracted from a given source}}{\text{Total heat extracted from all sources}} \times 100 = \%$$

In the prototype system, for the 1977-1978 heating season the contribution values were found to be 77%, 17.5% and 5.5% for ambient air, solar and heat loss respectively.

There is no doubt that the utilization of a normal roof as a radiation absorber can provide a useful addition to a heat pump system in mild climates.

VI-4 The heat store in the system

The thermal capacity of the heat store for a dwelling depends on a variety of factors including climatic conditions, thermal transmission of structural materials, the number of occupants and house usage.

VI-4.1 The heat pump power rating, and the required running hours

If one considers the heat losses and the heat gains indicated in Figure 41, one can establish the general balance of the system. It has already been assumed that since the heat pump draws air from the roofspace, the small heat losses Q_R from the dwelling into this space are recycled when the pump is operating. Additionally, if it is possible to site the store below ground level underneath the house, a proportion of any heat losses from the store will go into the house itself, although this factor has been neglected in all thermal balances. The amount of heat lost by drainage Q_D is supposed to be equal to the heat Q_W used to produce domestic hot water.

Thus, in equilibrium the daily heat balance over the control volume of the house indicated in Figure 41, will be given by:

$$Q_I + Q_{III} + Q_G = Q_W + (Q_{cw} + Q_R) + Q_v + Q_{II} \qquad \text{(VI-45)}$$

Where

Q_I = Energy transferred from the roof space to the control volume of the house (kWh/day)

Q_{II} = Energy transferred from the store to the control volume of the house (kWh/day)

Q_{III} = Energy transferred from the control volume to the store (kWh/day)

$Q_W = Q_D$ = Heat derived from heat pump to raise the temperature of domestic water (kWh/day)

Q_G = Heat gains from appliances, lights, occupants and solar radiation through windows (kWh/day)

Q_{cw}, Q_R and Q_v = Heat losses by conduction through walls, and roof ceiling and ventilation heat losses respectively (kWh/day)

($Q_L = Q_{cw} + Q_R + Q_v$; and $Q_{Total\ Load} = Q_W + Q_L$)

The equation VI-45 can be written as:

$$Q_I = (Q_W + Q_L) + (Q_{II} - Q_{III})$$
$$Q_I = Q_{T.Load} + Q_{Store} \qquad \text{(VI-46)}$$

The heat pump power rating for this system (kW) can be determined by:

$$Q_2 + \tau \dot{W}_{HP} = Q_{T.Load} + Q_{Store}$$

$$\dot{W}_{HP} = (1/\tau)\ [Q_{T.Load} + Q_{Store} - [Q_{Air} + Q_R + Q_{Solar}]] = kW \quad \text{(VI-47)}$$

Where:

τ = Number of operating hours of the heat pump per day.

Q_{Store} = Heat put into the store (kWh/day)

Under the ideal conditions it would be desirable to operate a heat pump with a variable power rating. Solar assisted heat pumps integrated with a thermal store operate with a constant power rating which is determined by the rate at which the store is required to be charged but, because of the variations in the solar heat available, this in turn depends on the day in the Autumn from which one operates the pump to commence charging the store.

VI-4.2 Available heat for storage and store capacity

If, $Q_{GS} = Q_{LS} = 0$, and if Q_S is the heat available in the store, the daily balance on the heat store can be written as:

$$Q_{II} - Q_{III} \begin{cases} > 0 ; & Q_S \uparrow \text{ (Store is being charged)} \\ = 0 ; & Q_S \rightleftarrows \text{ (Balance point for the heat pump)} \\ < 0 ; & Q_S \downarrow \text{ (Store is being discharged)} \end{cases}$$

From equation (VI-46) one can write:

$$Q_{II} - Q_{III} = Q_I - (Q_W + Q_L)$$

$$Q_{II} - Q_{III} = [Q_{Air} + Q_R + Q_{Solar} + \tau \dot{W}_{HP} - Q_W - Q_L] \quad \text{(VI-48)}$$

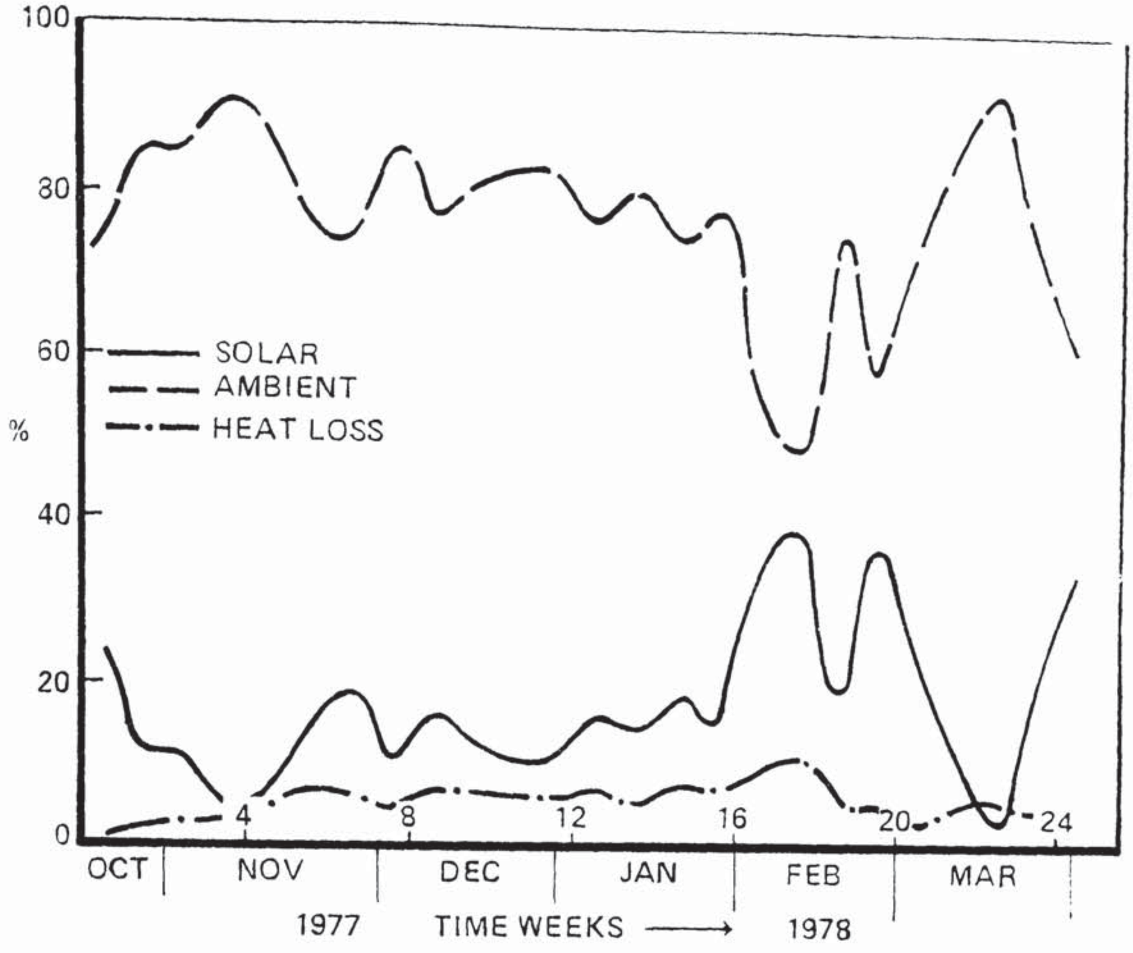


FIG 48. CONTRIBUTIONS OF AMBIENT AIR RADIATION AND HOUSE HEAT LOSS TO COLLECTED ENERGY

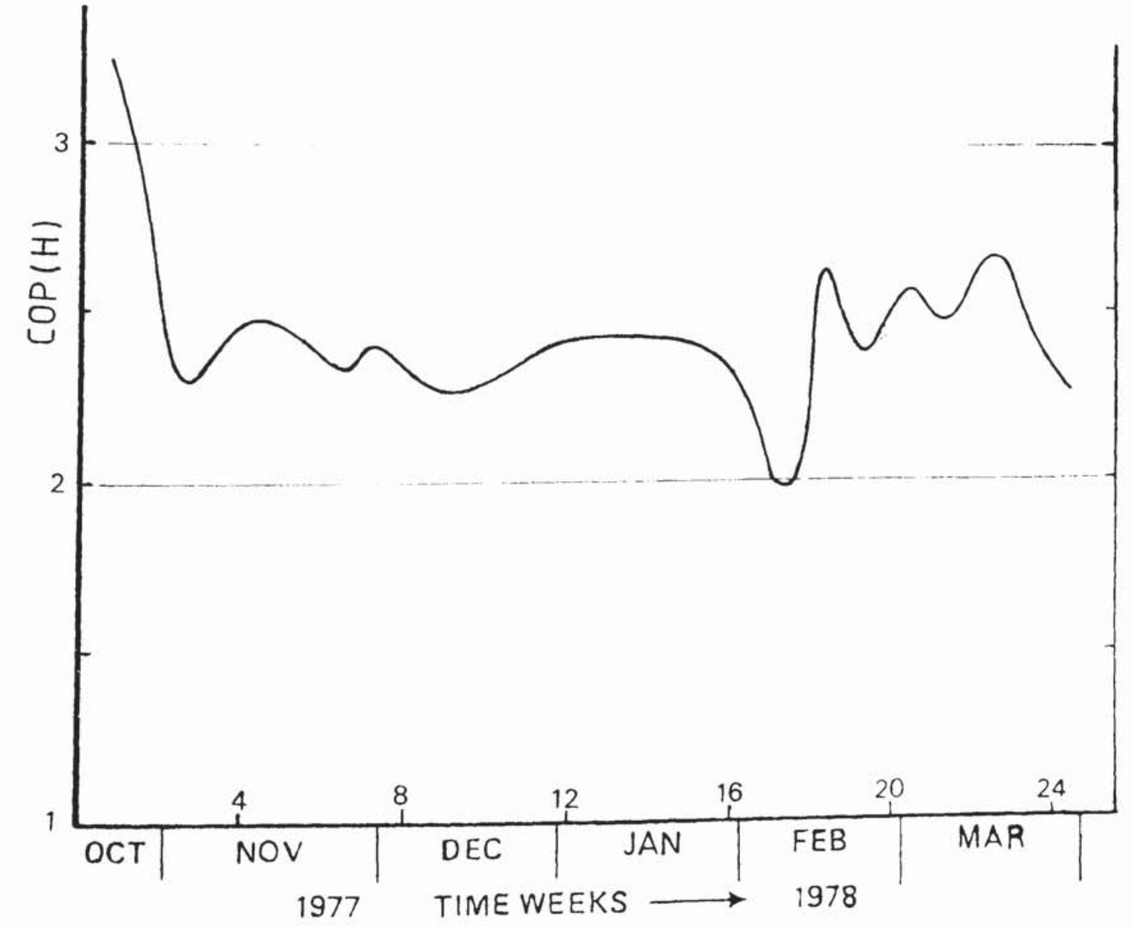


Fig 49 . Coefficient of performance over a heating season

1. One day Store Possibility.

In this case, the heat pump of the system is operated only during the day, taking into account the solar radiation boost in the roof absorber. The system performance would depend on the number of hours τ of operation per day required to supply the 24 hour load. A part of the energy collected during the day would be stored for overnight use. The required running hours per day can be calculated from the equation:

$$Q_{T.Load} = Q_W + Q_L = \dot{W}_{HP} \ COP(H)\ \tau$$

or,

$$\tau = Q_{T.L}/\dot{W}_{HP}\ COP(H) \qquad \text{(VI-49)}$$

A properly matched heat pump would operate for about 12 hours during December, January and February and for 5 to 8 hours per day in October and April.

The power rating of the heat pump and the one day store coupled to the system must have sufficient capacity to cope with the maximum load in January. This, perhaps makes the heat pump to be oversized during the first and last months of the heating season. The one day store capacity is in the region of ∿ 60 kWh per day (i.e. ∿ half of the typical maximum daily load) for most areas of the country (see section IV-2.3). If water is used as a store material and raised to a maximum store design temperature of 55°C, then each cubic metre has a storage capacity of 34.8 kWh above 25°C. Therefore, a store with a volume of about $2m^3$ of water would be sufficient to cover the heating load in this possible case of one day store. Because of the heat losses from the store, a volume higher than $2m^3$ would be required and its real value would depend on the amount of insulation used.

2. Intermediate store possibility

The system could be coupled to an intermediate store of volume Vm^3 of sufficient capacity to provide, for example, the average heating load of the house for ten consecutive days in January or to provide only a fraction "r" of the total heating load of the house in that month, which is in average considered the coldest of the heating season. Thus the heat pump will be required to provide only the fraction (1-r) of the average total heating load of the house in January and this reduces the required power rating of the system. If water is used as the store material and raised, as in the previous case to a maximum store design temperature of 55°C, then the storage capacity will be 35 kWh/m^3 above 25°C, which is the minimum required temperature for warm air distribution heating systems. The thermal store volume to provide for rQ_{TL} (January) and to compensate for wasted heat loss from the store during that month, when the store is being discharged can be calculated then, by:

$$V = [rQ_{TL}\ (\text{January}) + Q_{LS}\ (\text{January})]/35 \qquad \text{(VI-50)}$$

$$V = \text{kWh}/(\text{kWh}/\text{m}^3) = \text{m}^3$$

The store could be charged in October or November, so that the heat pump, in the period before January would need to cope with the total heating load Q_{TL} in each month and with the heat lost Q_{LS} (monthly) from the store to the ground. Q_{LS} in January would be relatively small so that it could be considered nil.

Although the apparent system performance and therefore, the energy saved, by charging in October, when the solar radiation boost is higher, would be greater than for November charging, more heat could be lost to the ground since the storage time is greater, so that the real difference between October and November charging could be in the end minimal.

The provision of an intermediate store for the system described, enables the heat pump to be operated at times when higher COP(H) can be achieved and the slightly higher air temperature over external ambient provided at the evaporator reduces the frosting problem in addition to improving the performance of the system.

VI-5 Installed Heat Pump Performance

The coefficient of performance of an installed heat pump can be expressed to a good approximation (63) as:

$$COP(H) = 0.5\ Tc/(Tx - T_{RS} + 15) \qquad \text{(VI-51)}$$

Where:

Tc is the temperature of condensation (deg. K)

Tx is the exit coolant temperature at the condenser (deg. K)

T_{RS} is the inlet air temperature to the heat pump (deg. K), which can be determined from a psychrometric diagram, as in Figure 46, and the equations (VI-41) to (VI-44). T_{RS} is equal T_o when solar boost is not in consideration.

Figure 49 shows the performance variation of the heat pump, in the prototype system tested at the University of Aston, over the heating season 1977 - 1978 as it has been reported in reference (61).

Table VI-1 has been prepared with data from reported values, for the test house over a heating season, in reference 64, and it

Table VI-1. Energy contributions and monthly average COP(H) values for the Test House over a heating season (64)

τ = Daily heat pump running hours (average) from equation (VI-49).
W_{HP} = Electrical energy supplied = $(3.4 \cdot \tau \cdot Z)$*; Q_2 = Energy extracted from all sources.
F = % free energy contribution (see section III-6) to the system in providing Q_{TL}.
$COP(H)_a$ for daily operation with solar boost and $COP(H)_b$ for day and night operation without solar boost, from equation (VI-51).**

Month	Q_{TL}	$\bar{\tau}$	W_{HP}	Q_2	F	COP (H)		% Saving in Energy
						τ h/day operation	24 h/day operation	
	kWh	h/day	kWh	kWh	%	(a)	(b)	f = (a-b)/b
Oct.	1432	4.50	474	958	67	3.01	2.67	13.10
Nov.	2522	9.33	949	1573	62	2.65	2.50	6.00
Dec.	3159	12.03	1265	1894	60	2.49	2.41	3.30
Jan.	3385	13.16	1391	1994	59	2.41	2.37	2.95
Feb.	2952	12.35	1181	1771	60	2.51	2.41	4.15
Mar.	2795	9.97	1054	1741	62	2.66	2.48	7.26
Apr.	2055	6.92	704	1351	66	2.91	2.57	13.22
Total	18300	-	7018	11282	62	-	-	-

* For heat pump power rating $\dot{W}_{HP}$ = 3.4 kW;

** Calculated in reference (64) for Tc = 338K and Tx = 333K enables one to assess the % improvement of the COP(H) and therefore the saving in energy due to the solar radiation. What is compared in that table, is the situation where the heat pump is operated only during the day coupled to the roof-absorber and a storage system is used for night heating (mode a), to the situation where the heat pump is operated day and night (24 hours) with the air being drawn directly from the outside (mode b). In a real situation the performance during the day, mode (a) would depend on the running hours of the heat pump per day as it has been previously mentioned.

The f value given in the last column on the table VI-1 is the basis for the calculations of energy savings for the particular monthly loads of the system given in the table VI-2, which has also been prepared with reported figures, for the monitored system, in reference 64. It is seen from table VI-1 that an "Integrated solar assisted heat pump and thermal store system", employing the tiled roof as a solar absorber, can improve the seasonal performance. The improved performance figures indicated in table VI-1 for mode (a) operation do not take into account the fact that the energy expended on defrosting, in the case when air is drawn direct from ambient, is reduced. Additional savings from such a consideration could range from 1 to 5%.

Table VI-2 gives the saving per month which could be achieved by operating a day store or an intermediate store compared with a lower rating pump operating 24 hours to supply the required instantaneous load.

The maximum design temperature for the $22m^3$ water store, has been considered at 55°C giving a storage capacity of 765 kWh above 25°C.

Table VI-2. Average Values of heating load and energy saving for the Test House, over a heating season (64).

Q_L for average degree days Do, Thames Valley area, and n = 0.5 air ch/h
Q Supplied load = Heat pump load - Q_{LS}; Q_{LS} = 0.010 Q_S (kWh/day)
Energy Saving = f × Supplied load ; f = from table (VI-1); Q_S = Store capacity = 765 kWh above 25°C.
Volume V = $22m^3$ of water.

Month	Heating Loads			One day Store		Intermediate Store ($22m^3$)			
						October Charge		November Charge	
	Space Heating Load Q_L	Water Heating Load Q_W	Required Heating Load Q_{TL}	Load to Heat Pump	Energy saving	Load to Heat Pump	Energy Saving	Load to Heat Pump	Energy Saving
	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh	kWh
Oct.	1184	248	1432	1432	187	2197	287	1432	187
Nov.	2282	240	2522	2522	151	2752	151	3287	197
Dec.	2911	248	3159	3159	104	3398	104	3398	104
Jan.	3137	248	3385	3385	100	2620	77	2620	77
Feb.	2728	224	2952	2952	123	2952	122	2952	122
Mar.	2547	248	2795	2795	202	2795	202	2795	202
April	1815	240	2055	2055	271	2055	271	2055	271
Total	16604	1696	18300	18300	1138	18769	1214	18539	1160
% Annual Saving					6.21	-	6.46	-	6.25

The power rating of the heat pump was 3.4 kW including 0.4 kW fan power. The domestic hot water load has been assumed constant at 8 kWh per day supplied by the heat pump. The space heating load has been calculated by considering the test house as a "5kW house", and estimating 0.5 air changes per hour in order to calculate the ventilation losses.

It can be seen, in table VI-2, that annual energy savings when mode (a) is compared with mode (b) (for an October to April heating season) amount to 6.2% of the load for an average year. As already mentioned, no account has been taken of savings resulting from fewer defrost cycles as it was seen during the monitoring programme.

The seasonal overall performance for the solar assisted heat pump system in consideration can be calculated by the equation III-1 given in section III-6.

$$\text{S.O.P. (H)}_{\text{SAHPS}} = \frac{\text{Seasonal Heating Load}}{\text{Seasonal Electrical Energy Input}}$$

$$\text{S.O.P. (H)}_{\text{SAHPS}} = 18300/7018 = 2.60$$

This value, without taking into consideration any supplementary heating to increase the hot water temperature above 55°C.

The overall seasonal saving, if compared to the case in which the seasonal heating load is met by electrical heaters with 100% efficiency, amounts to:

$$F = \%\ E_{\text{(saving)}} = [1 - (1/\text{S.O.P(H)})] \times 100$$

$$F = \%\ E_{\text{(saving)}} = [1 - (1/2.6)]\ 100 = 61.5\%$$

or 61.5% of the seasonal electricity bill for heating.

It is apparent from the results shown in table VI-2, that for an existing house with a conventional roof, the direct annual savings obtained by using the roof as a radiation absorber (compared with the extraction of heat from external ambient air) is only of the order of 6% whether a one day store or a medium store is used. The advantage of one day store over a medium term store for an existing house is that because of the smaller volume required it is more likely to be possible to locate it within the building structure so that all heat losses are into the house. A medium term store (of the order of weeks) is likely to give rise to larger difficulties in terms of construction and cost for domestic applications than the short term (day) store. On the other hand a medium term store would permit the reduction of the heat pump power rating which means lower capital cost for the domestic system. Improved performances over the short term store, for winter operation would be possible provided waste heat losses from the store to the ground and side surfaces can be kept low.

VI-5.1 Off peak use of the heat pump

The consumer would not benefit greatly financially from a mode of operation saving only $\sim$ 6% in energy. The more attractive use of a "one day store" from the consumers point of view would be to charge the store by operating the heat pump on "off peak" electricity. However, night operation of the heat pump would reduce its performance by about 3.5% if compared to the performance for similar 12 hours of operation at day light (64). Thus, because of the reduced performance the heat pump rating would need to be increased by 3.5% for 12 hours running at night. Normally "off peak" periods are for less than 12 hours so that the rating would have to be increased in the ratio $12/\tau_o$ where τ_o represents the "off peak" period in hours.

The use of heat pumps in this way might be preferred by the electrical supply industry. The "off peak" rating for the heat pump would be (64):

$$\dot{W}_{HP(OP)} = 1.035\ (12\ \dot{W}_{HP}/\ \tau_o) \qquad \text{(VI-52)}$$

The one day store capacity Q_S would need to be increased to:

$$Q_{S(OP)} = Q_{TL\ (max)}\ (24 - \ \tau_o)/24 = \text{kWh} \qquad \text{(VI-53)}$$

For the test house this would mean an increase from Q_S = 54 kWh to $Q_{S(OP)}$ = 77 kWh for a τ_o value of 7 hours. The heat pump rating would need to be increased from 3.4 kW to 6kW.

VI-5.2 Store heat losses Q_{LS} of the test house

The store heat losses could be divided into three main components and calculated as:

$$Q_{LS} = Q_{LS1} + Q_{LS2} + Q_{LS3}$$

$$Q_{LS1} = U_1\ A_1\ (T_S - T_i) \times 24/1000 = \text{kWh/day}$$

$$Q_{LS2} = U_2\ A_2\ (T_S - T_{earth}) \times 24/1000 = \text{kWh/day}$$

$$Q_{LS3} = U_3\ A_3\ (T_S - T_{earth}) \times 24/1000 = \text{kWh/day}$$

$$Q_{LS} = \Sigma\ U_n\ A_n \cdot \Delta\ T_n \cdot 24/1000 = \text{kWh/day} \qquad \text{(VI-54)}$$

Where:

Q_{LS1} = Upward heat losses into the house

Q_{LS2} = Sideway heat losses through vertical walls

Q_{LS3} = Downward heat losses to the ground

U_1, U_2, U_3 = Overall heat transfer coefficient upwards, sideways and downwards respectively ($W/m^2\,°C$)

T_S, T_i, T_{earth} = Temperature of storage water, inside house and earth surrounding store respectively (°C)

Temperatures T_S, T_i and T_{earth} could be assumed as 55°C, 18°C and 5 to 10°C respectively, for the store in the test house. Coefficients of heat transfer U_1, U_2 and U_3 could be supposed between 0.4 and 0.6.

The amount of heat losses from the store for the test house, over the heating season were evaluated as:

Q_{LS} = 0.010 Q_S = kWh/day

and for Q_S = 765 kWh, Q_{LS} amounts to 230 kWh/month in November and to 239 kWh per month in December.

The power rating of the heat pump in the prototype system is greater than required to provide the daily heating load of the test house. For mode (b) of operation, with a COP(H) of 2.37 (see table VI-1), it can provide 1.7 times the maximum average daily heating load required by the house in January. A heat pump rated at only 1.92 kW could be sufficient to meet the required load. However, small heat pumps of a high efficiency are not yet available in the U.K. and this type of unit still require engineering developments. The heat store capacity should be increased to maintain the heating capability of the system (i.e. $Q_S > 765$ kWh). An appropriate design of the store would optimize the heat pump power rating.

Increasing the heat store capacity also increases the heat loss from the store, so that, if small heat pumps are going to be developed attention needs to be drawn to the design of heat stores with a heat cpacity in the region of 800 to 1000 kWh and with improved insulating properties that can keep Q_{LS} considerably reduced. Using water as the store material, a volume of $\sim$ 22 m^3 to $\sim$ 28 m^3 would be required, but the volume could be reduced if a satisfactory store could be developed.

On the other hand, larger heat pumps offer some advantages that must be taken into consideration.

- A larger unit is inherently more reliable.
- The use of an oversized heat pump reduces maintenance problems and also the number of running hours per year.
- Noise from the unit should be present only for a limited number of hours during the day.
- The unit could be operated only on "off peak" electricity to charge the store, and finally
- Larger units are already available with reasonable efficiency in the heating mode.

It could also be attractive to consider this type of system, with greater power rating units applied for example in schools, in a block of flats or in a group of houses that permits the use of some form of communal heating. In this case, attention should need to be drawn into the engineering design of an integrated solar assisted heat pump and thermal store system relying upon the technological advances of control systems.

VI-6 New Thermal Store Configuration Integrated to the Solar Assisted Heat Pump System using the Roof as a Solar Radiation Absorber.

The experience gained with the analysis of the relatively simple prototype has provided the basis to propose an engineering design of an integrated solar assisted heat pump and thermal store system which includes heat pumping from the store and a new store configuration. Figure 50 shows the general aspect of the system. The objective has been to match the heat pump to the supply and the load in an attempt to maximise the seasonal performance of the heat pump and system and to provide a heat pump system suitable for any house in any locality without the need for auxiliary space heating.

The heat pump of the system would be operated as an air to water unit, modes 1-3-5-7, to charge the store or as a water to air unit, modes 4-6-8-9, to extract heat from the store. Solar panels could also be incorporated to charge the store.

The system would provide the hot water requirements and space heating or cooling when required in the building. The sections of the basic thermodynamic Rankine Cycle where useful heat is released are illustrated in figure 5, in chapter II. To produce domestic hot water, refrigerant vapour is cooled from T_D1 (compressor discharge temperature) to T_E (condensation temperature for compressor discharge pressure) by the vapour to water heat exchanger in the hot water tank. For space heating or storage most of the heat is released through the heat pump condenser at the temperature T_C of condensation from point E to point A' in the diagram of figure 5. The main parts of this system and their position related to the house were shown in Figure 42 and described in section VI-1-2.

The system should be operated at day time, and in order to improve its performance the following temperatures should be considered as a limit in any application:

1) T_o = 2°C. Minimum outdoor air temperature for air to water operation. This will give about 4°C to the air arriving at the evaporator or outdoor coil, if one takes into account the solar boost through the roof. In this form frost formation on the evaporator would be prevented, provided that the relative humidity of the air arriving at the evaporator remains lower than 65%.

2) T_S = 55°C Maximum water temperature in the hot water tank and in the store.

3) T = 25°C Minimum water temperature, required for heating the air, in the indoor coil (water-air heat exchanger)

4) The fact that the heat pump extracts heat from the store (water to air operation), increases the heating capacity of the storage, because its temperature can go down to 4°C or lower values if a brine is used instead of water as storage material. The central part of the store could be cooled down to 0°C in order to increase even more the store heating capacity or to produce very cold water for air conditioning (cooling) application.

VI-6.1 Components of the System

In this section the major components used in the system will be outlined and some characteristics or special considerations which apply to the different operating modes will be pointed out in brief.

The indicative number of each component is referred to the general system shown in Figure 50.

1. Compressor. The compressor can be one of four basic types: reciprocating, rotary sliding vane, centrifugal, or rotating screw. The last two types are used only for large capacity machines while

the majority of small and medium size heat pumps use electric driven positive displacement reciprocating compressors, designed for use with refrigerants having low specific volume and relatively high pressure characteristics (R12, R22, R500). Reciprocating compressors are available in three types: hermetic, semihermetic and open. One can use in the system the hermetic type which has the advantage of suction cooling. This is the method in which cooling of the electric motor is achieved by drawing the cold refrigerant from the evaporator over the motor in its way to the compressor. Hermetic compressors are almost noiseless and very reliable. They are available in capacities up to about 5kW. Semihermetic compressors do exist in higher capacities, they have suction cooling but they are noisier and more expensive than the hermetic model. For the purpose of a monitoring program, the open compressor has the flexibility to allow any suitable type of prime mover to be adopted. The main disadvantage is that the pressurised refrigerant system is not completely enclosed and is potentially subject to leakage through the shaft seals.

Compressors designed to operate at speeds below 1150 rpm are normally connected to motors with V-belt drives (65, 66). Direct driving is used in compressors with higher rotative speeds eliminating transmission inefficiency and giving higher reliability. Direct driven motor speeds normally are: 60 cycle power supply: 1150 and 1750 rpm. 50 cycle power supply: 900 and 1450 rpm. Small compressors are available for 2950 and 3500 rpm at 50 and 60 cycles respectively.

Capacity control. When the evaporator is unusually warm the pressure can become high enough to cause the compressor to overload, which is dangerous to both compressor and motor. This is likely to be a particular problem in solar assisted heat pumps because of the possibility of occasional very sunny days during the heating season. To avoid this problem it is necessary to modulate the capacity of the heat pump. This means to match the condenser output to the building heating requirement over a wide range of outside air temperatures.

The store coupled to the system with the new configuration could permit to achieve the modulation of the heat pump capacity. However it is preferable to consider the possibility of modulating the capacity by varying the speed of the compressor provided that its efficiency is not adversely affected. In mild conditions the condenser operates with a lower temperature difference delivering less heat, for this reason a reduced-speed operation would be necessary in order to reduce the amount of refrigerant arriving to the condenser. The speed reduction would reduce the power input to the compressor and the overall coefficient of performance should be improved. A normal practice is to use a series of fixed speeds rather than a continuously variable speed. The best method of speed control, which has the advantage of being infinitely variable rather than varying in a series of steps, is to vary the frequency of the supply by electronic means. Converters to produce the variable frequency exist only for large motors, from about 10 kW up, and three-phase power supply.

The internal efficiency of the compressor can be considered as high as 86% and the overall efficiency of the set motor-compressor at about 70%.

The power rating of the system and the selection of the compressor should be established on the basis of the operation of the heat pump as a water to air unit in which the output at the refrigerant-air heat exchanger (condenser) must provide the space heating load of the house (See operation modes 4 and 6). In order to prevent the compressor to overload a two different speed unit should be selected.

2. Outdoor Coil : Evaporator - Condenser. Air-source heat pumps have to pass substantial volumes (500 m^3/kWh or more (19)) of external air through their evaporators and so the evaporator coil is usually situated outdoors.

This air-refrigerant heat exchanger could be a standard finned coil with one or two fans. It works as an evaporator during heating operation and as a condenser during cooling (air conditioning) operation. As an evaporator it picks up the heat from the air coming from the roof space. As a condenser, it releases heat to the air from the roof space in a water to air heat pump operation. During the heating or air to water operation this coil could be considered as a nonfrosting evaporator, because of the T_0 limit value assumed equal or greater than 2°C. Occasionally, the evaporator may build up a light coat of frost just before the compressor shuts off. This frost immediately would melt on the off cycle. The evaporator coil would operate at an air temperature of about 4°C, in the lowest periods, while the refrigerant inside the coil would be down to -4 or -6°C. The evaporating temperature could be considered then as -4°C. It could be possible to regulate the effective capacity of the evaporator by sensing the condensing temperature (19) and with this information, controlling the speed of the fans which blow the source air over the evaporator coil. The outdoor coil must be selected on the basis of the required heat extraction and rejection, according to the heating and cooling load of the house.

3. Indoor Coil : Refrigerant to Air Heat Exchanger (Condenser). It could also be a standard finned coil, working at a condensating temperature of 60°C in order to have hot water in the domestic hot water tank and in the store at about 55°C. The output of this condenser must match the space heating load of the house in the water to air operation of the heat pump.

4. Indoor Coil : Is a water to air heat exchanger used to heat or cool the air from the store simultaneously or not with the operation of the heat pump.

5. Heat Pump Water – Refrigerant heat exchanger, which works as a condenser or as an evaporator according to the heat pump operating mode, and which could be of the common shell and tube types.

6. Immersed Coil : Is a refrigerant evaporator, required to cool the water in the central tank of the store (see Figure 50) down to 0°C. This is in order to extract the maximum available heat from the store for heating purposes when outdoor conditions are very low and the store is going down to its minimum capacity. Inside the coil the refrigerant could be at about -6 or -8°C. This coil permits to increase the heat capacity of the store if a part of the water in the central tank is converted into ice at 0°C releasing the latent heat of fusion in its change of phase (i.e. 335 kJ/kg (93 kWh/m^3) of ice formed).

The temperature of maximum density of water is 4°C. This means that the temperature of the water at the bottom of the tank will be 4°C or above. If the water cools below 4°C, it will expand and rise to the surface. A temperature of 0°C will cause it to freeze and ice forms on the surface of the water in the tank. The upper water of the tank has to freeze solid before the temperature of the water at the bottom can be lower than 4°C. So this coil situated at the bottom of the central tank may provide an efficient heat pickup unit for heating and a good heat dissipator during the cooling cycle, see operation mode 6. During this operation heat from the other tanks of the store, in which the water temperature is at 4°C, could be transferred to the central tank, by conduction through the walls. The COP(H) of the heat pump would be reduced because of the lower value of the evaporating temperature. Ice accumulation over the pipes of the immersed coil could be controlled by location of the thermostatic expansion valve thermal bulb, inside the water and fixed to the pipe (see Figure 50).

7. <u>Oil Separator</u>
8. Compressor low and high pressure side service valves.
9. <u>Vibration Absorbers</u>. To reduce any vibration that might travel into the lines.Vibration absorbers may be installed in the suction and discharge lines.
10. <u>Suction Line-heat Exchanger</u>. To subcool the condensed refrigerant before it enters the expansion valve.
11. <u>Accumulator - Receiver Heat Exchanger</u>. The heat exchanger-accumulator installed in the suction line allows more subcooling of the refrigerant. It separates liquid-vapour and allows dry vapour and oil to enter the compressor because the liquid is boiled off by the heat exchanger.
12. Liquid sight glass.
13. Liquid filter drier.
14. Solenoid manifold check valve, to close of open refrigerant circuits according to the desired effect in each operation mode.

15. Three-way solenoid valves. To control two separate refrigerant circuits as required in the operation modes.

16. Pressure drop distributor installed on the outdoor coil, 2, and on the indoor coil, 3, (condenser).

17. Metering type or thermostatic type two-temperature valves. Used as evaporator pressure regulating on the outdoor and immersed coil evaporators to ensure a constant low-side pressure.

18. Thermostatic Valve. Controlled by the temperature of the superheated vapour coming from the compressor to the outdoor coil condenser during the air conditioning (cooling) operation modes. Its function is to divert the condensed liquid stream as it is shown in modes 8 and 9 of Figure 50.

19. Thermostatic water valve (or pressure operated water valve). To modulate the water flow rate entering the refrigerant-water heat exchanger working as a condenser. It is controlled by the temperature of the water leaving the condenser or the temperature of the condensed liquid leaving the condenser. The valve must be set to open at a definite heat pressure on the condenser.

20. Thermostatic Water Valves. These are three-way water regulating valves, used to modulate the water flow in the different operation modes in order to get the appropriate temperature required to heat the air in the indoor coil, 4.

21. Air purge valves.

22. Electric water shut-off valves, to close or open water circuits according to the operation mode.

23. Hand water valves, to fill and empty the store tanks.

24. Air filter.

25. Low and high side pressure service valves.

26. Electric shut-off valves to close or open refrigerant circuits.

27. Hot water tank and coil.

28. Check valves.

29. Receiver pressure relief valve.

30. Return air ducts and conditioned air ducts.

31. Automatic dampers.

32. Air duct from the roof space.

33. Pressure motor control (thermostatic, not in the diagram). To turn on and off the motor driving the compressor.

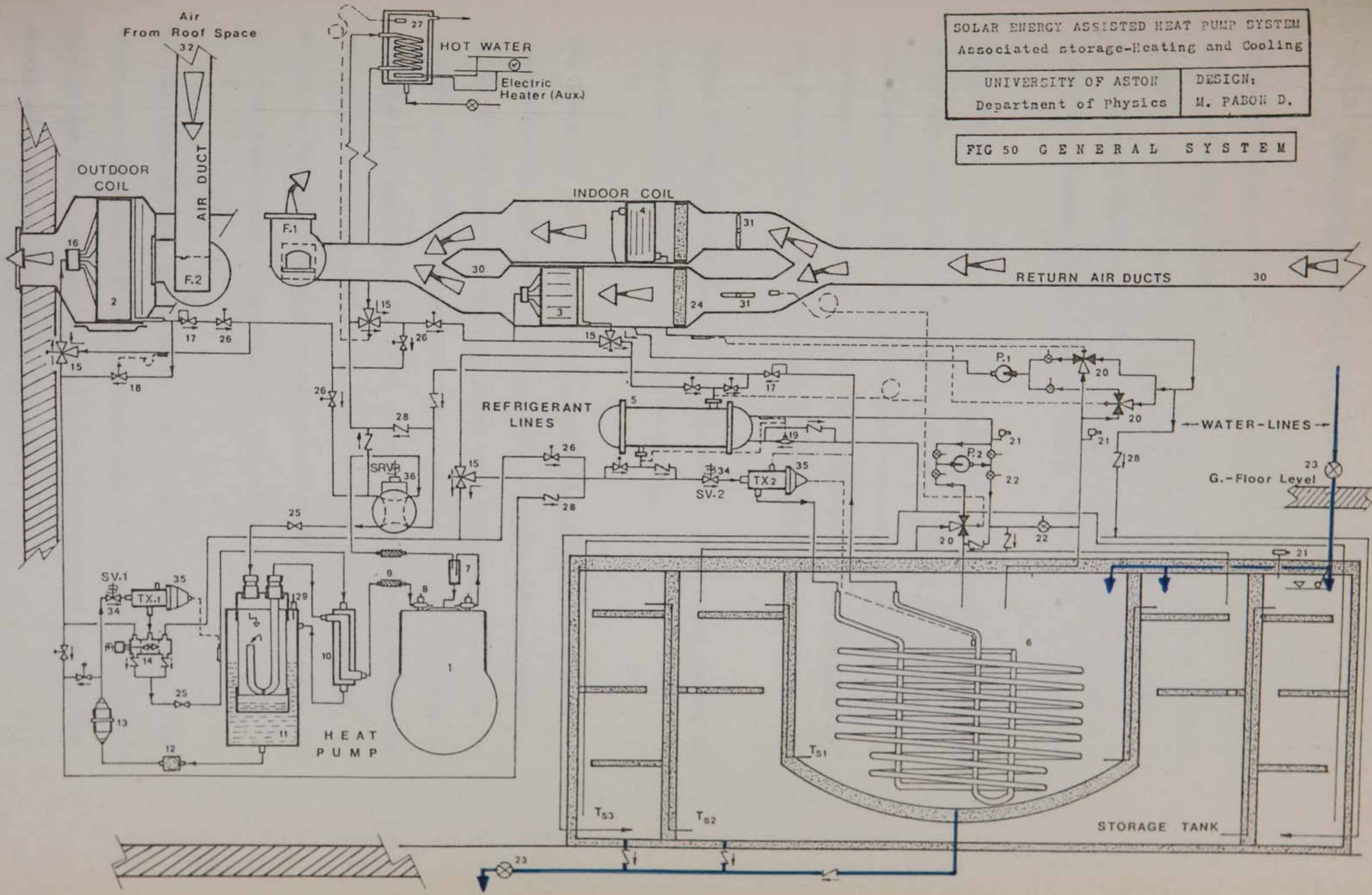
34. Shut-off Solenoid Valves. (SV1 and SV2). Located in the liquid line. They must be electrically connected in parallel with the pressure motor control. They are used to stop flooding the low pressure side during the off cycle.

35. Thermostatic Expansion Valves (TX1 and TX2). For TX1, the location of the refrigerant control is not on the evaporator but on the condensing unit with the thermal bulb mounted on the liquid line (see Figure 50). In this way the liquid is controlled as it leaves the condenser rather than when it enters the evaporator. This permits to have the evaporator (outdoor coil) completely flooded and utilized. Because of the inclusion of the accumulator-receiver it is not required to superheat suction gas on the evaporator as it is in most systems. The Expansion Valve TX2 controls the refrigerant flow to the immersed coil, 6.

36. Solenoid Four-Way Reversing Valve. To reverse the flow of refrigerant according to the operation mode.

P1, P2. Water Pumps. To move the water from the store to the condenser No.5, and to the indoor coil No.4.

F1, F2. Air Fans. To move the air through the distribution system and from the roof space respectively.



SOLAR ENERGY ASSISTED HEAT PUMP SYSTEM
Associated storage-Heating and Cooling

UNIVERSITY OF ASTON Department of Physics	DESIGN: M. PABON D.

FIG 50 GENERAL SYSTEM

VI-6.2 The store configuration and the heat loss rate Q_{LS}

In solar energy applications the problem of heat storage becomes the first priority because of the variability of the sunshine periods. Numerous systems have been proposed to solve this problem but most of them with relatively low efficiency because of their high heat loss rate through the external surface.

Considering a solar assisted heat pump and thermal store system it is estimated that a further improvement in the performance of the system could be achieved if a satisfactory store could be developed. It is on this basis that a different configuration in the store coupled to the system has been proposed.

Figure 50 shows this special configuration, which could permit the reduction of thermal store heat losses Q_{LS}.

The basis for the analysis is to consider the store as a heat regenerator in which the solid storage material and the energy transporting fluid (or heat carrier) have been replaced by the storage water (see section III-4.5). One can consider the store as the type of regenerator in which a front of temperature moves from the inlet "A" to the outlet "B" during charge and in opposite direction during discharge between a maximum inlet temperature T_A and a minimum outlet temperature T_B. See Figure 51.

In this type of heat regenerator, "At Itinerant Temperature", the fluid heated during discharge periods can, theoretically, leave the unit at any point between A and B once it has reached the maximum design temperature required by the application.

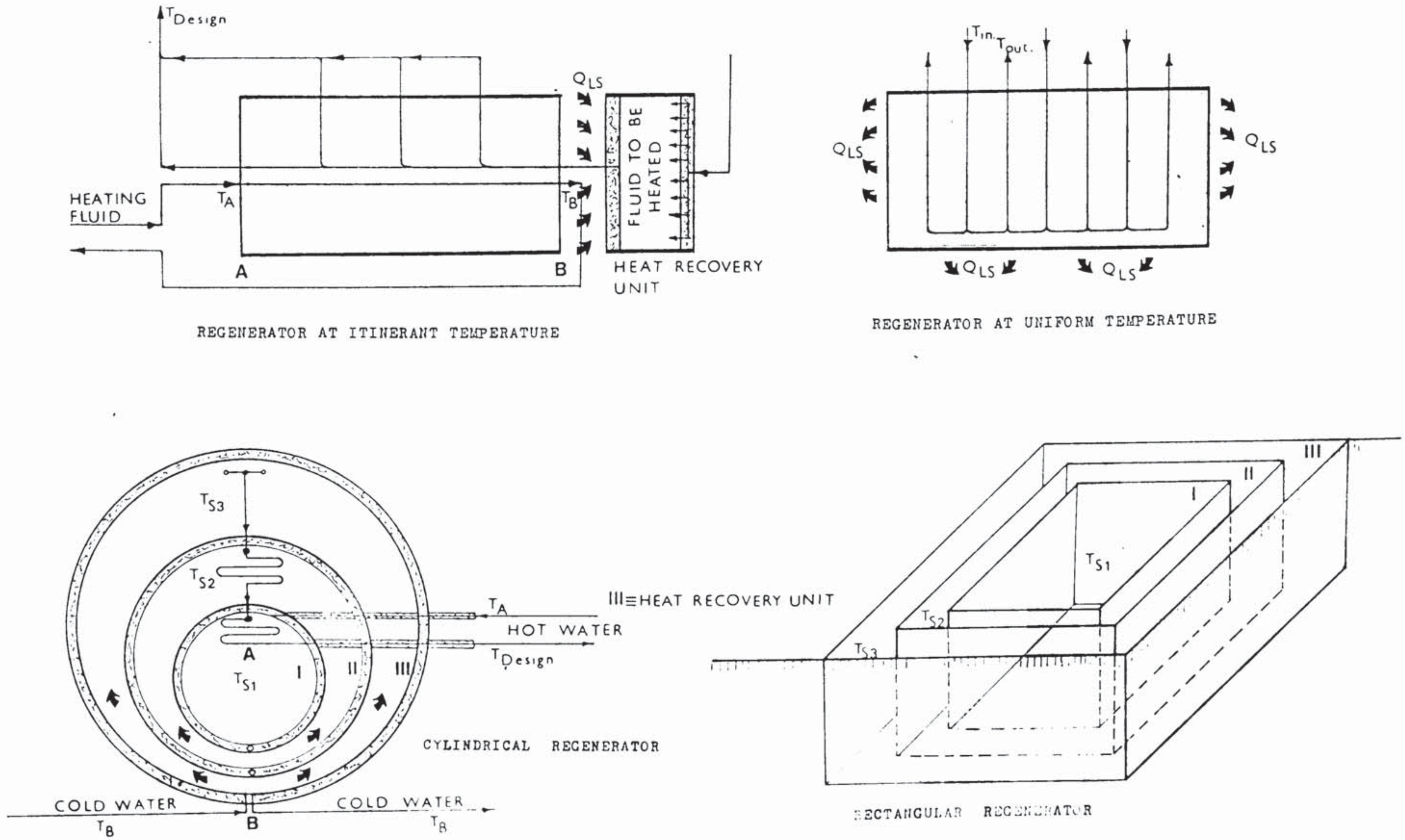


FIGURE 51.- HEAT REGENERATORS - WATER HEAT STORE

Lateral heat losses are considered nil if the regenerator is of infinite width, or if it has a spherical shape with the inlet, A, located at the centre and the outlet, B, at the periphery of the sphere. Thus the only heat losses to be considered occur at the minimum temperature through the outlet surface B, where a heat recoveryunit is coupled to the regenerator.

The fluid to be heated goes first through the heat recoveryunit, ta king with it any heat lost, theoretically, from the outlet surface. The mass of material of the heat recover unit, working as a pre-heater, could be the fluid to be heated itself as is shown in Figure 51.

In a regenerator "At Uniform Temperature", the other type, all the points of its entire mass are supposed to be heated or cooled simultaneously to the same temperature by the heat carrier. The mass temperature increases during charge periods, and decreases during discharge periods. The external surface is always at the temperature of the regenerator, increasing the heat losses. Hot water tanks are considered as regenerators of this type, in which the solid material and the heat carrier are the same.

The ideal new configuration for the store, working according to the principle of regenerators at itinerant temperature should be of spherical shape. Water at the maximum store design temperature (i.e. 55°C) coming from the condenser would be drawn to the centre of the sphere and dispersed towards the periphery during charge periods. During discharge periods cold water returning to the store moves from the periphery to the centre of the sphere increasing its temperature. The water would leave the sphere once it reaches the design temperature required to heat the air in the air-water heat exchanger taking with it the exact amount of heat required.

One would have then, within the liquid, a front of temperature moving from the centre to the periphery of the sphere during charge and in opposite direction during discharge times. The absence of solid material in the regenerator, makes the water play the role of heating and heated fluid and of storage material. In order to obtain the convenient disposition of temperature it would be necessary for the store to be composed of insulated concentric spheres connected between one another, with the external one acting as a heat recovery unit.

VI-6.2.1 Theoretical curves of water temperature variation inside an ideal store

Consider the charge period of an ideal store (concentric spheres) with outer radius R and with the water moving from the centre to the periphery. One assumes that the variation of the temperature and the movement of water occur only in the r direction.

For the mathematical analysis one can consider:

ρ = Density of water (kg/m^3)

c = Specific heat of water (kJ/kg °C)

M = Mass of water (kg)

$\dot{M}$ = Flow rate of water (kg/h)

H = Total enthalpy of water (kJ)

u = Water velocity in the store (m/h)

r = Radius

T = Temperautre of water in the store (°C)

τ = Time (hours)

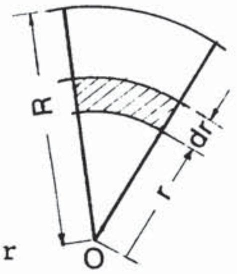
k = Heat conduction coefficient of water

Let one follow a layer of water dr thick in its movement to the periphery.

The amount of heat H (total enthalpy) contained by the mass of water in the layer dr thick can be written as:

$$H = M.c.T = 4\pi r^2.dr.\rho\, c.T$$

and its variation during a time $d\tau$ as:



$$\frac{dH}{d\tau} = \frac{d}{d\tau}\,(4\pi\, r^2.dr.\,\rho\, .c.T)$$

or

$$dH = 4\pi\, c.\,\frac{d}{d\tau}\,(r^2.dr.\rho\,.T)\, d\tau \qquad \text{(VI-55)}$$

This variation dH is equal to the net amount of heat Q transferred to the layer from the neighbouring layers at sections r and r + dr as a result of the thermal conduction which, in the case of the sphere, is expressed as (67):

$$Q_{conduct.} = k\left[\frac{\partial^2 T}{\partial r^2} + \frac{2}{r}\,\frac{\partial T}{\partial r}\right] 4\pi\, r^2.dr.d\tau \qquad \text{(VI-56)}$$

If,

$$dH = Q_{conduction}$$

$$\frac{d}{dt}\,(\rho T r^2.dr)\; d\tau = \frac{k}{c}\left[\frac{\partial^2 T}{\partial r^2} + \frac{2}{r}\,\frac{\partial T}{\partial r}\right] r^2.dr.d\tau \qquad \text{(VI-57)}$$

In the left hand side A of the equation one has:

$$A = \frac{d}{d\tau}\,(\rho T r^2.dr) = \frac{d(\rho\, T)}{d\tau}\, r^2.dr + \rho T\, \frac{d(r^2.dr)}{d\tau}$$

But:

1) $$\frac{d\,(\rho\, T)}{d\tau} = \frac{\partial(\rho\, T)}{\partial\tau} + \frac{\partial(\rho\, T)}{\partial r}\cdot\frac{dr}{d\tau}$$

$dr/d\tau$ = u = Velocity of the layer (variable as a function of r)

2) $$d(r^2.dr) = \frac{\partial (r^2.dr)}{\partial r} dr$$

$$\frac{d(r^2.dr)}{d\tau} = \frac{\partial (r^2.u)}{\partial r} dr$$

Substituting in the expression of A and re-arranging one has:

$$\frac{d}{d\tau}(\rho.T.r^2.dr) = r^2 \frac{\partial (\rho T)}{\partial \tau} dr + \frac{\partial (\rho.T.r^2.u)}{\partial r} dr$$

The product $\rho r^2 u = \dot{M}/4\pi$ =constant. Substituting these results in equation (VI-57) one obtains:

$$\frac{\partial (\rho T)}{\partial \tau} r^2 + \rho.r^2.u \frac{\partial T}{\partial r} = \frac{kr^2}{c}\left[\frac{\partial^2 T}{\partial r^2} + \frac{2}{r}.\frac{\partial T}{\partial r}\right]$$

Let $a = k/\rho c$ = thermal characteristicsof the storage material (i.e. for water, a = 0.0005 (m^2h)), So:

$$\frac{1}{\rho u} . \frac{\partial (\rho T)}{\partial \tau} + \frac{\partial T}{\partial r} = \frac{a}{u}\left[\frac{\partial^2 T}{\partial r^2} + \frac{2}{r} . \frac{\partial T}{\partial r}\right]$$

Finally, for ρ of water considered constant, re-arranging one obtains:

$$\frac{\partial T}{\partial \tau} = a . \frac{\partial^2 T}{\partial r^2} + \frac{(2a - u)}{r} . \frac{\partial T}{\partial r} \qquad \text{(VI-58)}$$

The solution of this differential equation will give the variation of water temperature in the ideal store from the centre to the periphery. To solve it one can separate the equation into two functions of the variables τ and r and write: $T = f(r) . g(\tau)$

Let $F = (2a - ur)/r$. Substituting the new three functions into equation (VI-58) one obtains:

$$fg' = af''g + F.g.f'$$

dividing by g.f one has:

$$\frac{g}{g'} = \frac{af'' + F\,f'}{f} = -\omega^2$$

Where ω is a constant. Finally one can re-arrange to obtain:

$$f'' + (F/a)\,f' + (\omega^2/a)\,f = 0 \qquad \text{(VI-59)}$$

This equation is known as "Sturm Liouville Equation" and its solution is given as (22):

$$T = \Sigma\, f\,(r, \omega)\,\exp.\,(-\omega^2.\,\tau)$$

To calculate T it is necessary to have f as a function of r and ω which is difficult to establish. One can solve the general equation VI-58 by successive approximations using the method of finite differences. Thus one can write:

1) $$\frac{\partial T}{\partial \tau} \sim \frac{\delta T}{\delta \tau} = \frac{1}{\delta \tau}\,(T_{r,\,\tau+\delta\tau} - T_{r,\,\tau})$$

2) $$\frac{\partial T}{\partial r} = \frac{1}{\delta r}\,(T_{r+\delta r,\,\tau} - T_{r,\,\tau})$$

3) $$\frac{\partial^2 T}{\partial r^2} = \frac{1}{\delta r^2}\,(T_{r+\delta r,\,\tau} + T_{r-\delta r,\,\tau} - 2T_{r,\,\tau})$$

These terms can be substituted into equation VI-58 to obtain:

$$\frac{1}{\delta\tau}(T_{r,\tau+\delta\tau} - T_{r,\tau}) = \frac{2a}{\delta r^2}\left(\frac{T_{r+\delta r,\tau} + T_{r-\delta r,\tau}}{2} - T_{r,\tau}\right)$$

$$+\frac{(4\pi\cdot 2a\rho r - \dot{M})}{4\pi\cdot\rho r^2}\cdot\frac{1}{\delta r}(T_{r+\delta r,\tau} - T_{r,\tau})$$

Dividing the radius R in N layers $\delta r = R/N$, one has for the i layer:

$$T_{i,\tau+\delta\tau} = T_{i,\tau} + \frac{2a\cdot\delta t}{\delta r^2}\left(\frac{T_{i+1,\tau} + T_{i-1,\tau}}{2} - T_{i,\tau}\right)$$

$$+\frac{(4\pi\cdot 2a\cdot\rho\cdot r_i - \dot{M})}{4\pi\cdot\rho r_i^2}\cdot\frac{\delta\tau}{\delta r}\left(T_{i+1,\tau} - T_{i,\tau}\right)$$

Let $B = 2a\cdot\delta\tau / \delta r^2$ which gives $\delta\tau = (B)\cdot\delta r^2/2a$. If one takes $B = 1$ and considers the water characteristics: $\rho = 1000$ and $a = 0.0005$ the equation becomes:

$$T_{i,\tau+\delta\tau} = T_{i,\tau} + \left(\frac{T_{i+1,\tau} + T_{i-1,\tau}}{2} - T_{i,\tau}\right)$$

$$+\frac{\delta\tau}{\delta r}\cdot\frac{10^{-3}(r_i - (\dot{M}/4\pi))}{r_i^2}\left(T_{i+1,\tau} - T_{i,\tau}\right) \quad \text{(VI-60)}$$

5

In order to determine the form of the curves of the temperature variations in an ideal store, a computer programme, in FORTRAN formulation, has been written to solve equation VI-60, taking into consideration the following values:

$\delta r = 2.8 \times 10^{-5}$ m; $\delta\tau = 7.84 \times 10^{-7}$ hours

$B = 1$; $\dot{M} = 45$ kg/h

$\rho = 1000$ kg/m^3 ; $C = 4.18$ kJ/kg°C

r_i varying from 0.05 m to 0.075 m; the maximum temperature T_A at the centre of the sphere was assumed equal to 55°C and the minimum T_B was assumed as 5°C.

The following notations were used in the FORTRAN programme:

$$LI = 2a\ (\ \delta\tau\ /\ \delta r^2)$$

$$DER = 10^{-3}\left[(r_i - (\dot{M}/4\pi))/r_i^{\,2}\right]\ (\ \delta\tau\ /\ \delta r\ )$$

Figure 52 shows the curves obtained. This figure permits one to appreciate the front of temperature created inside of an ideal store.

VI-6.2.2 Functioning of the store coupled to the system

The practical shape of the store working on this principle, would be far from the spherical one because of the operational difficulties. In a real situation the store would take a cylindrical or rectangular shape in which the height, width and length could be limited, therefore lateral losses, although reduced, are inevitable.

In order to approach the ideal model, a hot water storage system, functioning as a regenerator at itinerant temperature, should have an inner hot, an intermediate warm, and an outer cold zone. It should be composed of several containers of either cylindrical or rectangular shape with different diameters and lengths assembled in increasing order of dimension one within the other in an eccentric position (see Figures 50 and 51). In the cross section of the store represented in Figure 50 the containers are not totallyconcentric due to the fact that heat losses occurring through the upper surface of the store are assumed to be entering the living space if the store is located underneath the house.

One supposes that the store is divided into two parts: the first is the heat accumulator and the second is the heat losses recoveryunit.

FORTRAN Programme

```
      REAL LI
      DIMENSION TA (1000),TB (1000)
      I1 = 2
      J1 = 2
      LI = 1.0
      DR = 2.8 E - 05
      DT = 7.84 E - 07
      TA (1) = 55.0
      TB (1) = 55.0
      N = 0.
      R1 = 0.05
      DO 30 K = 2.5
   30 TA (K)  = 5.0
      R = R1
   51 DO 10 I = I1.1000
      R = R + DR
      DERA = DT*0.001*(45-R)
      DERB = DR*R**2
      DER = DERA/DERB
      TB (I) = TA(I)+LI*((TA(I+1)+TA(I-1))/2-TA(I))-DER*(TA(I+1)-TA(I))
      IF (TB(I)-54.990) 333,444,444
  444 J1 = I
      TB(J1-1) = 55.000
      R1 = R
  333 continue
      IF (TB(I) -  5.0000) 11,11,10
   10 CONTINUE
   11 CONTINUE
      N = N+1
      TB(I+1) =  5.0
      TB(I+2) =  5.0
      R = R1
      DO 20 J = J1,1000
      R = R+DR
      DERB = DR*R**2
      DERA = DT*0.001*(45-R)
      DER = DERA/DERB
      TA(J) = TB(J)+LI*((TB(J+1)+TB(J-1))/2-TB(J))-DER*(TB(J+1)-TB(J))
      IF(TA(J)-54.990) 111,222,222
  222 I1 = J
      TA (I1-1) = 55.000
      R1 = R
  111 CONTINUE
      IF (TA(J)- 5.0000) 21,21,20
   20 CONTINUE
   21 CONTINUE
      N = N+1
      TA (J+1) =  5.0
      TA (J+2) =  5.0
      R = R1
      CALL DATSW (0.IA)
   40 CONTINUE
      IF (N-500) 51,50,50
   60 CONTINUE
      END
```

The heat losses recover unit would be essentially constituted by the external container, whereas the remainder internal containers would constitute the heat accumulator or the heat store itself.

One can consider two periods: Charge and discharge. During the charge period the hot water would arrive to the innermost container in its upper part displacing the cold water. This would cause a water movement from the centre to the periphery through all the other containers (see Figure 50).

During the charge period of the first container, heat losses would occur through its walls to the second container and also heat would be carried with the water leaving the container. Thus the temperature of the water on the second container (i.e. T_{S2}) would increase slightly while the temperature of water in the first (i.e. T_{S1}) is being increased. It can be seen in Figure 50 that the water leaving a container would be sent immediately to the upper part of the following container.

Because of the eccentricity of the containers within the store, the heat losses occurring from the first to the second, through the walls, heat up, in the lower part of the second tank, a thin layer of fluid. The thin layer at the bottom becomes hotter than the thicker upper layer, creating a difference in densities. This also creates a flux of heat by convection from the bottom to the top of the tank, opposing the propagation of heat diffusion from upper layers within the liquid. So that the hot water should always be found in the upper part of the container, far from the external surface, where any coil should be located in order to take heat from the store during discharge periods.

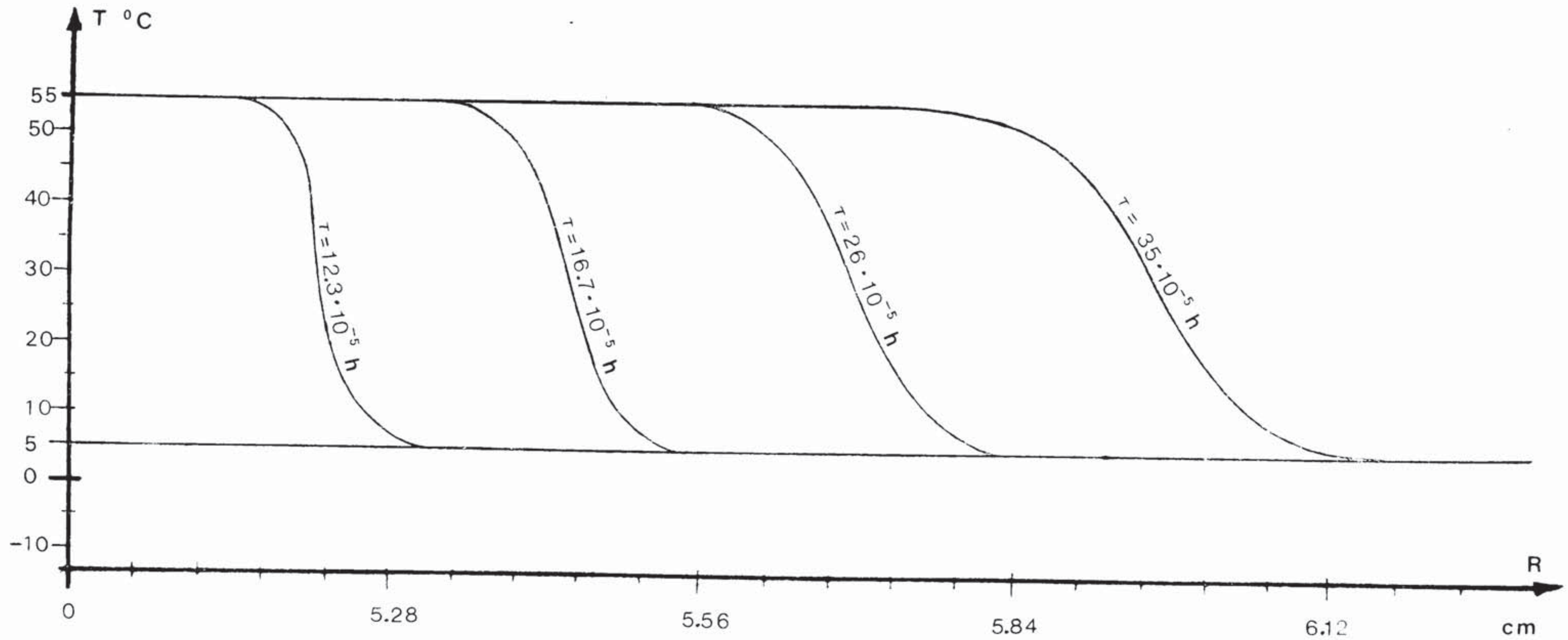


FIG. 52.- THEORETICAL CURVES OF THE TEMPERATURE VARIATION IN AN IDEAL SPHERICAL STONE DURING THE CHARGE PERIOD

In the case of the store coupled to the heat pump system, shown in Figure 50, this natural convection process would make the hot water to be found always near to the floor of the house, reducing the back losses from the store to the earth.

Thus one would have a front of temperature moving from the inlet of the inner tank to the outlet of the heat recover unit, as in the regenerators at itinerant temperature. The temperature of the water at the inlet of the first tank would be constant whereas the temperature of the water leaving the recoveryunit would be variable between the initial temperature of the water in this unit and the final temperature reached, at this point, at the end of the charging period. The heat recoveryunit, would be used then to retain during the charging period the heat losses that occur from the heat accumulator tanks.

The use of the heat pump during daylight, in order to have the solar boost, implies that the charging of the store cannot be realized continuously. Therefore, the water enthalpy in each tank would increase during the day and decrease slightly during the night or during any interruption of the charging period. The process would be repeated everyday, until each container reaches the maximum enthalpy of equilibrium. In his case the enthalpy of equilibrium of container (n) will be lower than that of container (n-1), numbered from the innermost tank, their difference depending in part on the insulation used. The greater the difference the higher the efficiency of the storage because the rate of heat losses from the store Q_{LS} would be lower.

In order to increase that difference it is suggested to use for the entire walls of the containers insulating materials such as fibre-glass. Alternatively, one could use some type of thin cavity walls (e.g. fibre-glass cavity walls) for the containers, filled with a salt or latent heat storage material having an adequate transition temperature see table III-2. The salt would work as an insulating material and would increase the heat store capacity.

The second period, constitutes the discharge. If the conformation of the store is by eccentric cylinders, coils must be located in the free space obtained by their eccentricity as is shown in Figure 51. In that case water to be heated would be taken from the periphery and circulated inside the coils through the other internal cylinders, increasing its temperature all along its way to the centre of the store. Water from the heat recover unit would be the first to leave, heated at a temperature near to the maximum temperature at the centre of the store, recovering then most of the heat losses from the heat accumulator part of the system. Cold water arriving to the store would pass first through the periphery and then through the coils to continue the discharging process. When the temperature of the water in a inner tank becomes to low for an efficient heat transfer through the coil it would be necessary to use this water directly rather than the water from the periphery by starting the discharge from this tank.

In the configuration of the store, shown in Figure 50, the discharge process is similar to the charge process. Hot water would be extracted from the upper part of the inner tank at T_{S1} causing a water movement from the periphery to the centre through all the other containers. In this case, the warmer water at the top of each container would be sent to the lower part of the following container which must be at about the same temperature. Thus a front of temperature would be moving from the periphery to the upper part of the innermost tank. Heat losses stored in the heat recover tank would pass to the intermediate tank. Cold water returning from the heat exchanger would arrive at the bottom of the recover unit to start again the process.

Returning water from the heat exchanger could be introduced to the intermediate tank if its temperature is higher than that of the water in the outer tank.

Water could also be extracted from the intermediate tank to be mixed with hot water from the inner tank in order to produce the right temperature of the water required in the evaporator during a water to air heat pump operation (see operation mode No.4). This is in order to prevent overloading the compressor and to regulate the extraction of the stored heat, which is not possible in conventional heat store tanks assisting a heat pump.

Heat losses Q_{LS} from the store system or heat losses from one tank to the neighbouring tank, could be calculated with equation (VI-54) once the equilibrium temperature in each tank has been established.

A computer modelling exercise on the new store configuration would be the next first stage if any future development is envisaged.

VI-6.3 Operation modes of the general system as indicated in Figure 50

This sytem would operate as an air-source heat pump down to the ambient outdoor air temperature of $T_o = 2°C$. Below that, the system would become a water-source heat pump which could use, if necessary, the latent heat of fusion given up by water in its change of phase to ice. The system could operate with a minimum temperature of evaporation of -4°C in the outdoor coil and with -6 to -8°C in the immersed coil. The condensing temperature could be fixed at 60°C in order to have domestic hot water at about 55°C.

Relying on the recently technological advances a microprocessor could be coupled to the system. This microprocessor would periodically accumulate data from appropriate sensing locations and execute COP calculations, determining if some other modes of operation at that moment could be employed which would result in the use of less energy. This would optimize the performance of the system.

In order to facilitate the comprehension of the operation modes and to avoid long explanations the diagrams, with the refrigerant and water circuits of each suggested mode of operation, are presented in colour. This also will permit one to determine more easily which components are operating in each of the different cycles.

The system would have six basic operation modes: Four of these basic modes would be used to produce heat for one use or another (i.e. to produce domestic hot water, to charge the store or to heat the house); and the remaining two would be used to produce cooling.

<u>Basic Mode I</u>

Diagram mode 1: $T_o > 2°C$; $T_{S1} < 25°C$

This first operation mode would be employed to produce domestic hot water and to charge the store when the outdoor temperature remains not lower than 2°C. Heat would be removed from the outdoor air during daylight operation in order to take advantage of the solar boost. To accomplish this, the system would perform as a typical air to water heat pump: Air coming from the roof space would pass over the outdoor coil -2- (acting as an evaporator) transferring heat to the refrigerant. The refrigerant within the coil would vapourize and this suction vapour (yellow lines) would be then compressed to a superheated vapour (deep red lines) in the compressor -1-. The superheated vapour would be transferred to the refrigerant-water heat exchanger -5- acting as a condenser. There the hot vapour would condense giving its heat to the cold water (light blue lines) coming from the storage tanks. The resulting hot water (scarlet lines) would transport the transferred heat to the central tank of the store. The condensed refrigerant in the high pressure line (purple lines), would be dropped to the receiver -11- on its way to the expansion valve TX1 -35-. Once the refrigerant is expanded the resulting liquid (orange lines) would flow back to the outdoor coil to complete the cycle.

In the store, during this operation, hot water (red shade) would be moving from the central tank to the outer containers filled with cold water (light blue shade).

During this operation mode, as in the others, the temperature of the domestic hot water in the hot water tank would control the opening and closing of the three-way solenoid valve -15-, located in the high pressure superheated vapour line, to regulate the amount of superheated vapour required to pass through the coil in the hot water tank. Dotted lines in the diagram indicated connections between valves and the required sensor positions.

Basic Mode II

Diagram Mode 2: $T_o \gtrless 2°C; T_{S1} > 25°C$

This operation mode would be employed at night in order to heat the house from the store. In this mode the heat pump would stop working.

A three way thermostatic water valve -20- would regulate the hot water flow required to heat the air in the water-air heat exchanger -4-. This valve would mix the right amount of hot water coming from the tank with the cold water returning from the indoor coil -4-. It can be seen from the diagram that the store would have cold water (light blue shade) in the outer tank, warm water (yellow shade) in the intermediate tank, and hot water (red shade) in the central container. The flow of water in the store should be moving from the periphery to the centre tank.

Basic Mode III

Diagram mode 3: $T_o \geqslant 2°C$; $25°C < T_{S1} < 55°C$

Diagram mode 5: $T_o > 2°C$; $4°C < T_{S1} < 25°C$

Diagram mode 7: $T_o > 2°C$; $4°C > T_{S1} \geqslant 0°C$

It can be seen from the diagrams of mode 3, mode 5 and mode 7, which integrate the basic mode III, that they are essentially the combination of the two previous basic operation modes I and II. Their refrigerant and water circuits are the same as in basic modes I and II.

The diagrams mode 3 and mode 5 do not include the part of the system with the immersed coil -6- and the expansion valve TX2 -35- in order to indicate the simplified system if it is not required for air conditioning in the cooling mode.

Mode 3, mode 5 and mode 7, would supply domestic hot water, heating for the house and heat to charge the store, to accomplish this, the system would perform as a typical air to water heat pump.

The water movement in the store for all three operation modes would be from the centre to the periphery. The difference between mode 3 and mode 5 is in the temperature of the water in the central tank of the store. The diagrams show the water circuit required in each mode of operation to heat the house.

In operations 3, 5 and 7 it is assumed that the house would not need all the heat output to satisfy its demands. The excess heating capacity would be put then into the store as is shown in the diagrams.

In operation mode 7 the excess of hot water would be discharged into the intermediate tank because of the low water temperature in the central tank i.e. $4°C > T_{S1} \geqslant 0°C$. It could be possible that ice has been accumulating on the top part of the central tank in previous operation periods, so that the inlet pipes to that tank could be frozen. It is supposed that operation mode 7 would be employed immediately after a period of very heavy winter with very low outdoor temperatures.

Basic Mode IV

Diagram mode 4: $T_o < 2°C$; $4°C < T_{S1} < 25°C$

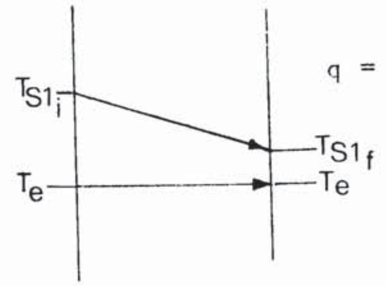
Diagram mode 6: $T_o \ll 2°C$; $4°C > T_{S1} \geqslant 0$

These two operation modes, integrating the basic operation mode IV, are similar because they permit the heating of the house from the store through the heat pump. It can be seen in the diagrams that operation mode 4 is using the expansion valve TX1 and the refrigerant-water heat exchanger -5- as an evaporator, whereas operation mode 6 is using the expansion valve TX2 and the immersed coil -6- as the evaporator.

These two modes would be used during periods of severly cold weather. Mode 4 would increase the COP(H) because of the higher temperature of the store in relation to the outdoor conditions. Mode 6 would increase the heating capacity of the system although reducing its COP(H). It can also be seen in the diagrams of these operations that superheated vapour from the compressor is transferred to the indoor coil or condenser -3- where the vapour condenses transferring heat to the air to provide the heating load. The condensed refrigerant drops to the receiver on its way to the expansion valve TX1 for mode 4 and to expansion valve TX2 for mode 6; once the expansion is done, the liquid refrigerant is carried to the refrigerant-water heat exchanger -5- acting as an evaporator in operation mode 4, or to the immersed coil -6- in operation mode 6. Heat stored in the water is transferred to the liquid refrigerant, in the respective evaporators according to the operation mode, vapourizing it. This vapour at low pressure goes back to the compressor to complete the cycle. Hot water from the central tank and warm water from the intermediate tank are mixed at the three way water temperature valve -20- to produce the right amount of heat required on the evaporator during operation mode 4. The expansion vlave TX1, is controlled by the temperature in the liquid vapour line. In this way the amount of refrigerant circulating is controlled by the valve as it leaves the condenser. The sensor to control the three way water temperature valve is located in the return air ducts. This would permit to modulate the heat output from the condenser to provide the space heating load. This is possible because of the configuration of the store, which allows different temperatures in the stored water. This also would prevent the overloading of the compessor. The diagram of operation mode 4 also shows that water in the store should be moving from the periphery to the centre of the store during that operation.

It can be seen in the diagram of operation mode 6 that the water in the store should not require movement from tank to tank. The amount of refrigerant through the immersed coil is controlled by the expansion valve TX2 with the sensor bulb located over the line on the vapour side within the liquid. This would limit the amount of ice that could be formed during the operation. The average temperature of the store during that period could be between 0°C and 4°C.

The heat balance across the immersed coil could be established as:



$$q = Ue\ Ae\ \Delta\ T_{LM} = \dot{m}\ \Delta\ h_{fg} = M_{S1}\cdot C\cdot\frac{dT_{S1}}{d\tau} \qquad \text{(VI-61)}$$

Since dT_{S1} is very small, $\Delta\ T_{Lm}$ could be approximated by $\Delta\ T = (T_{S1} - Te)$, so that:

$$d\tau = \frac{M_{S1} C}{Ue\cdot Ae} \cdot \frac{dT_{S1}}{(T_{S1} - Te)} \qquad \text{(VI-62)}$$

Where:

U_e = Overall heat transfer coefficient between refrigerant and water in the central tank.

T_e = Evaporation temperature of the refrigerant in the coil.

A_e = Coil heat transfer surface

$\dot{m}$ = flow rate of refrigerant.

$\Delta\ h_{fg}$ = Refrigerant enthalpy

M_{S1} = Mass of water in the central tank.

T_{S1} = Water temperature in the central tank.

The integrations limits would be:

$$\tau = 0 \rightarrow T_{S1} = T_{S1i} \text{ (initial temp.)}$$

$$\tau = \tau \rightarrow T_{S1} = T_{S1f} \text{ (final temp.)}$$

By integrating equation (VI-62) one has:

$$\tau = \frac{M_{S1}\ C}{U_e\ A_e} \cdot \ln \frac{T_{S1f} - T_e}{T_{S1i} - T_e} \qquad \text{(VI-63)}$$

If $K = U_e \cdot A_e / M_{S1} \cdot C$, the water temperature in the central tank would be:

$$T_{S1f} = (T_{S1i} - T_e) \exp (K \cdot \tau) + T_e \qquad \text{(VI-64)}$$

Equation VI-64 permits the calculation of the temperature variation of the water in the tank when the immersed coil is used. If T_{S1f} is lower than 4°C, ice could be formed at the upper part of the central tank.

The amount of heat extracted could be calculated (without taking into consideration the possible formation of ice) as:

$$\frac{d\ T_{S1}}{d\tau} = 0 + K\ (T_{S1i} - T_e) \exp (K\tau)$$

$$\frac{d\ T_{S1}}{d\tau} = K\ (T_{S1i} - T_e) \exp (K\tau)$$

$$q = M_{S1} \cdot C \cdot K\ (T_{S1i} - T_e) \exp (K\tau) \qquad \text{(VI-65)}$$

<u>Basic Modes V and VI</u>

Diagram mode 8: $T_o > 20°C; T_{S1} \sim 5°C$

Diagram mode 9: $T_o > 20°C; 4°C > T_{S1} \geq 0°C$

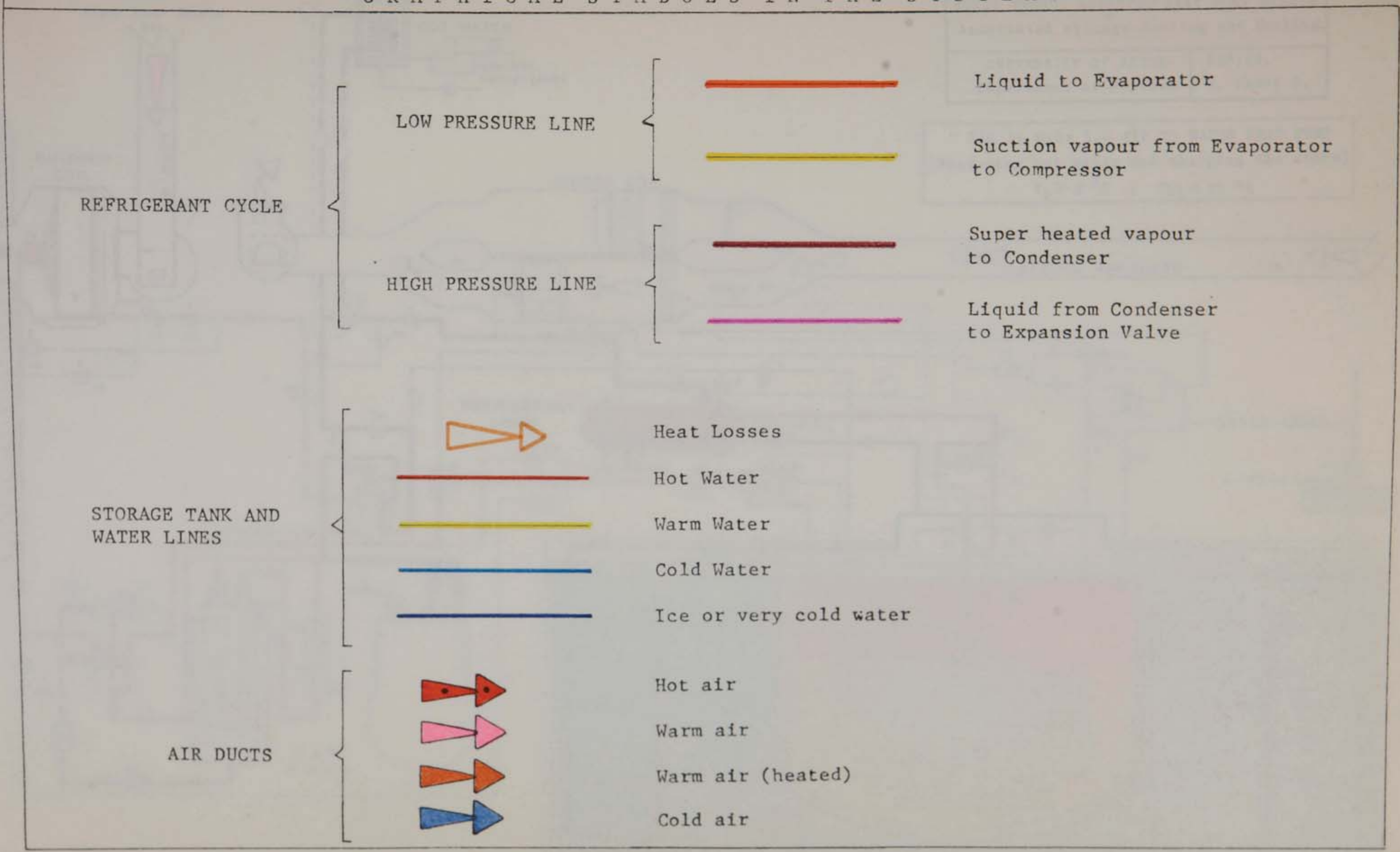
The remaining two of the six basic modes of operation are cooling modes. Operation mode 8 is similar to operation mode 4 and operation mode 9 is similar to operation mode 6. The differences are, that in both cases the condenser unit is now the outdoor coil -2- and that the cold water that would be produced when employing these operation modes is taken to the indoor coil -4- to cool the air coming from the house through the return ducts. Water circuits required to produce the air conditioning can be seen in the diagrams.

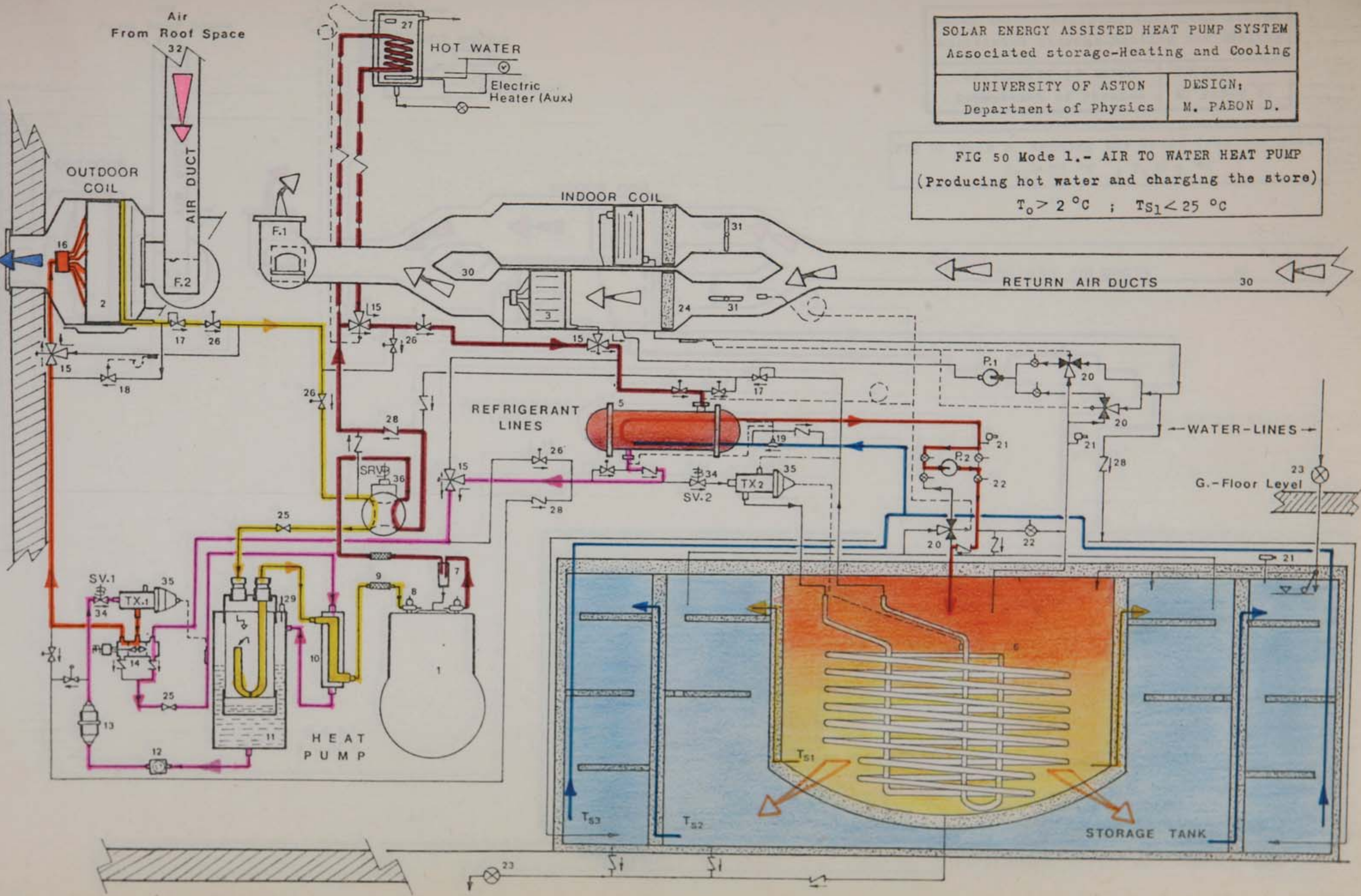
Warmer water, during these operation modes, should be located at the outer tank of the store and its movement should be from the periphery to the centre.

Operation mode 9 should be employed principally at night times, to obtain a better heat transfer efficiency in the outdoor coil, and to use the "off peak" power. Cold water produced at night would be used during the following day.

With the previous operation modes described for the system, it should be possible to improve the performance of the system, but a computer modelling exercise of the system must be realized to evaluate the real possibilities.

GRAPHICAL SYMBOLS IN THE SYSTEMS

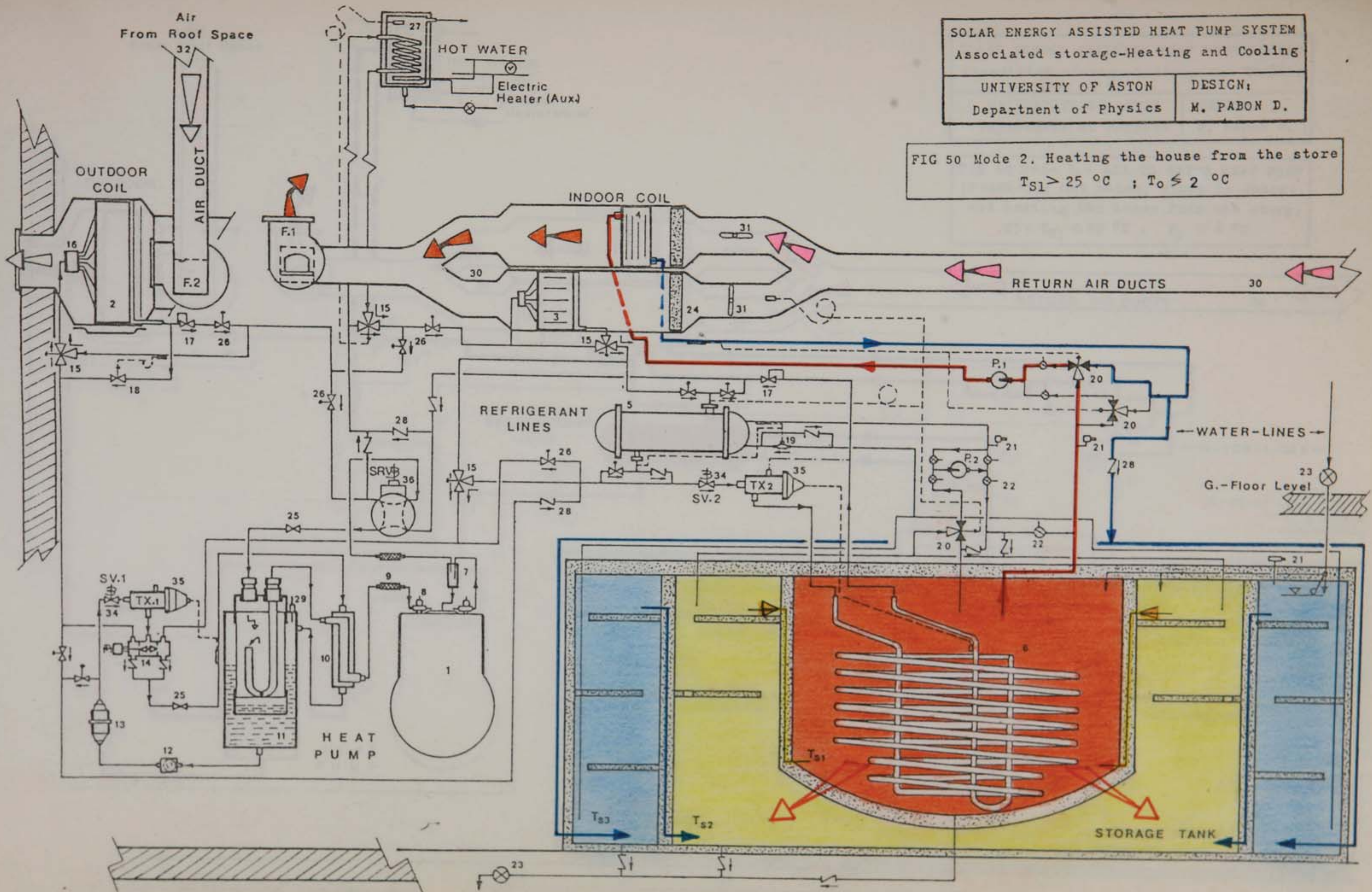




SOLAR ENERGY ASSISTED HEAT PUMP SYSTEM
Associated storage-Heating and Cooling

UNIVERSITY OF ASTON Department of Physics	DESIGN: M. PABON D.

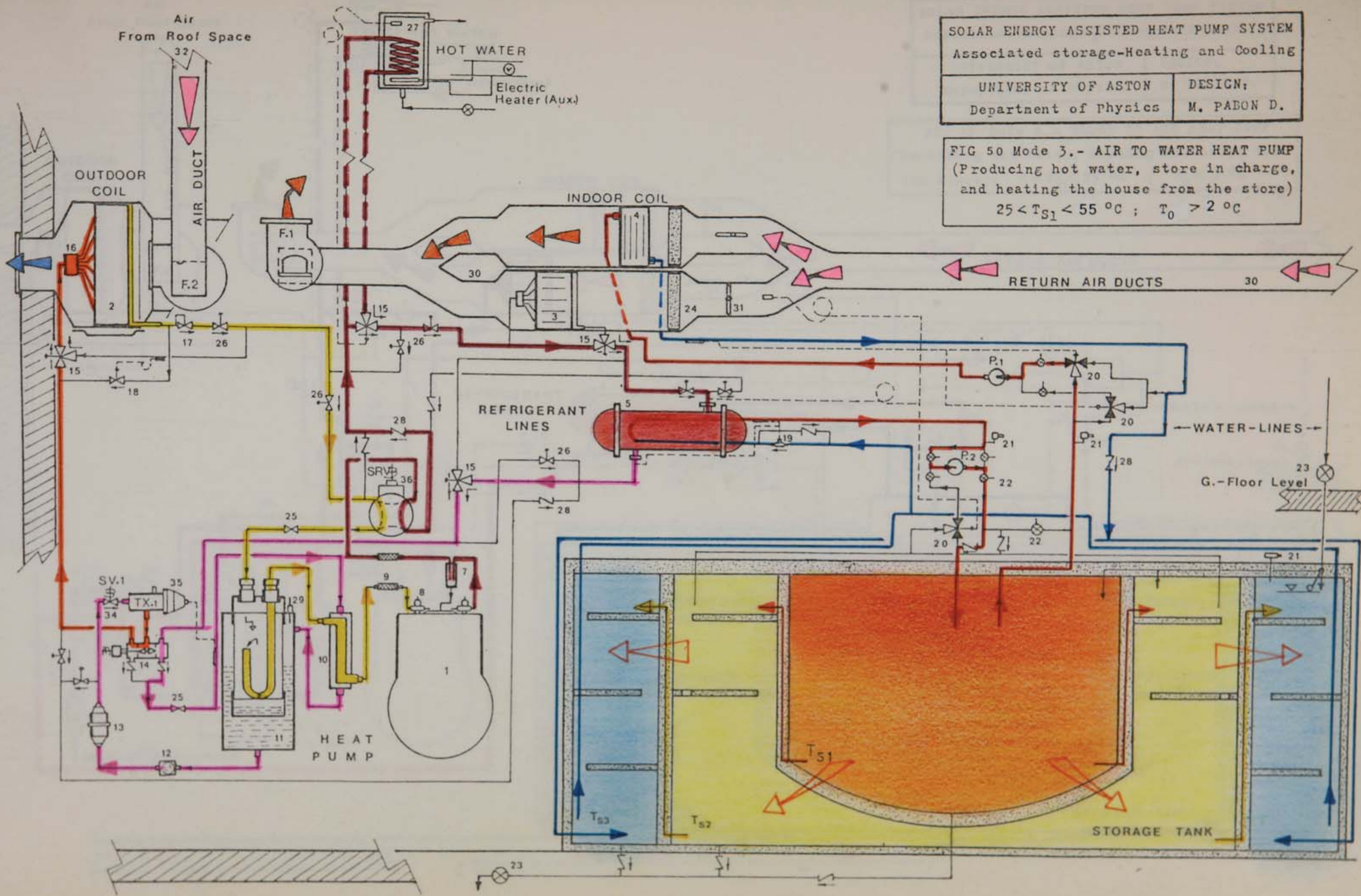
FIG 50 Mode 1.- AIR TO WATER HEAT PUMP
(Producing hot water and charging the store)
$T_o > 2\ ^oC$; $T_{S1} < 25\ ^oC$



SOLAR ENERGY ASSISTED HEAT PUMP SYSTEM
Associated storage-Heating and Cooling

UNIVERSITY OF ASTON Department of Physics	DESIGN: M. PABON D.

FIG 50 Mode 2. Heating the house from the store
$T_{S1} > 25\ °C$; $T_o \leq 2\ °C$



SOLAR ENERGY ASSISTED HEAT PUMP SYSTEM Associated storage-Heating and Cooling	
UNIVERSITY OF ASTON Department of Physics	DESIGN: M. PABON D.

FIG 50 Mode 3.- AIR TO WATER HEAT PUMP (Producing hot water, store in charge, and heating the house from the store) $25 < T_{S1} < 55\ ^{o}C$; $T_0 > 2\ ^{o}C$

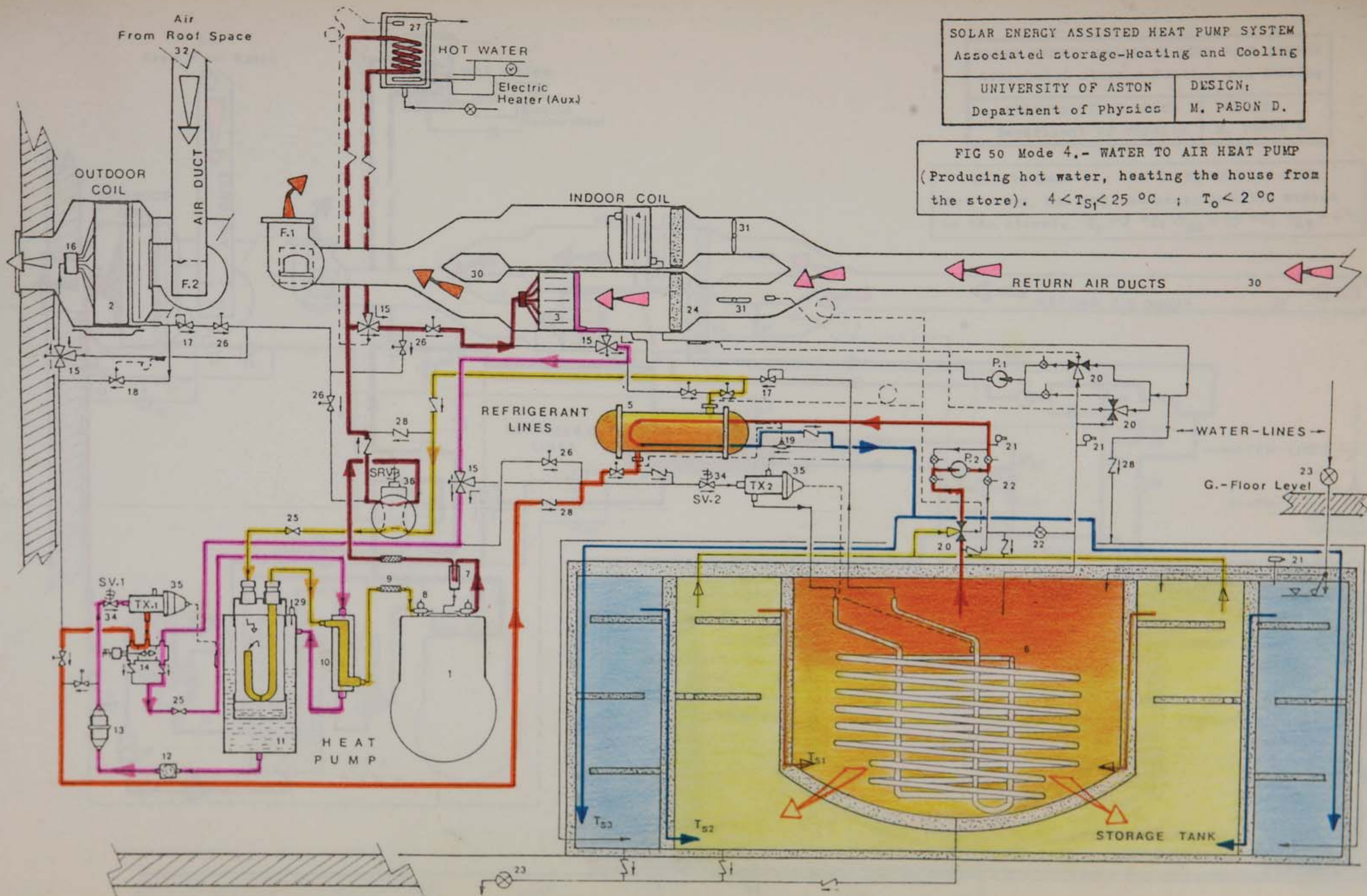
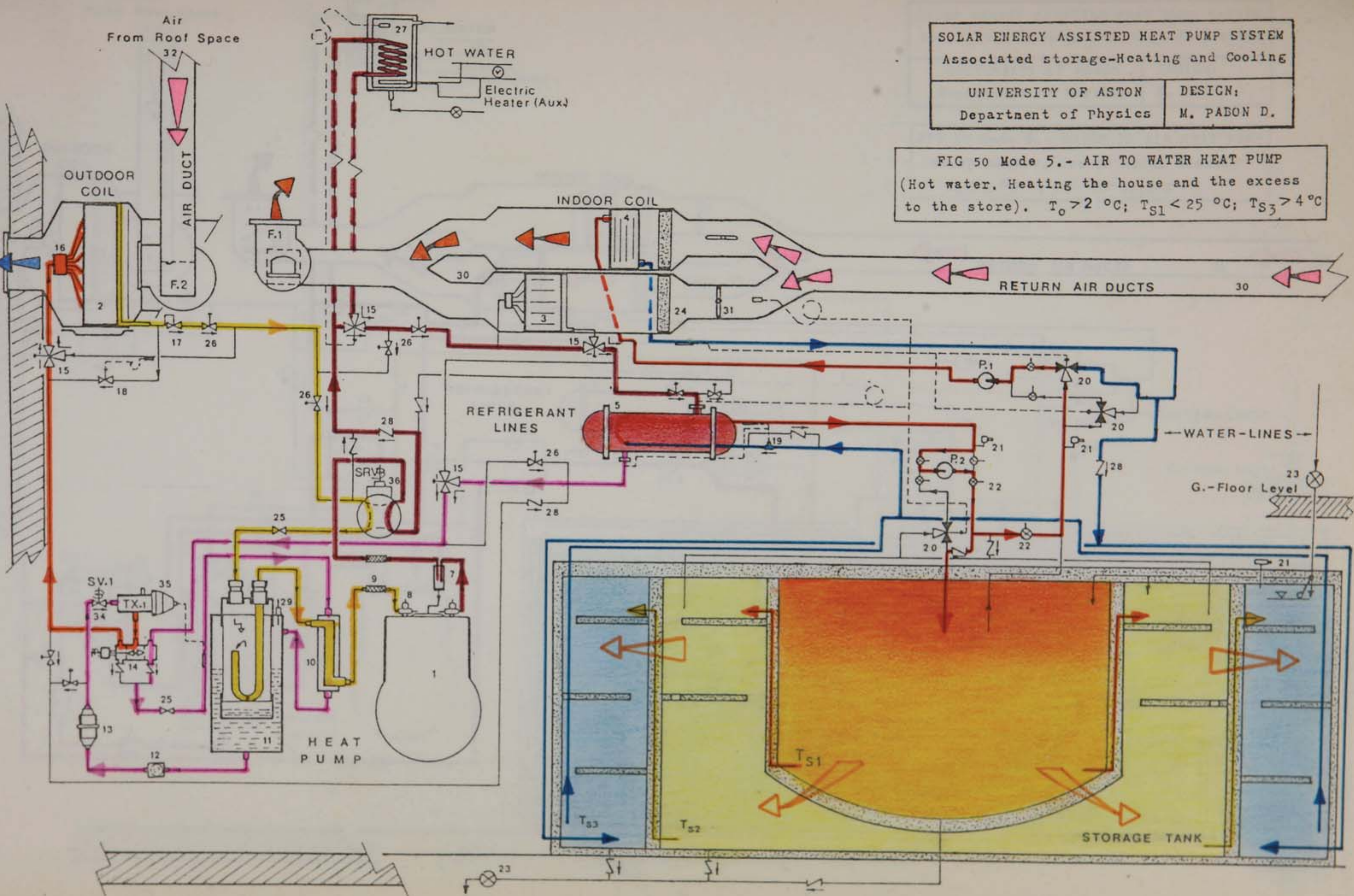


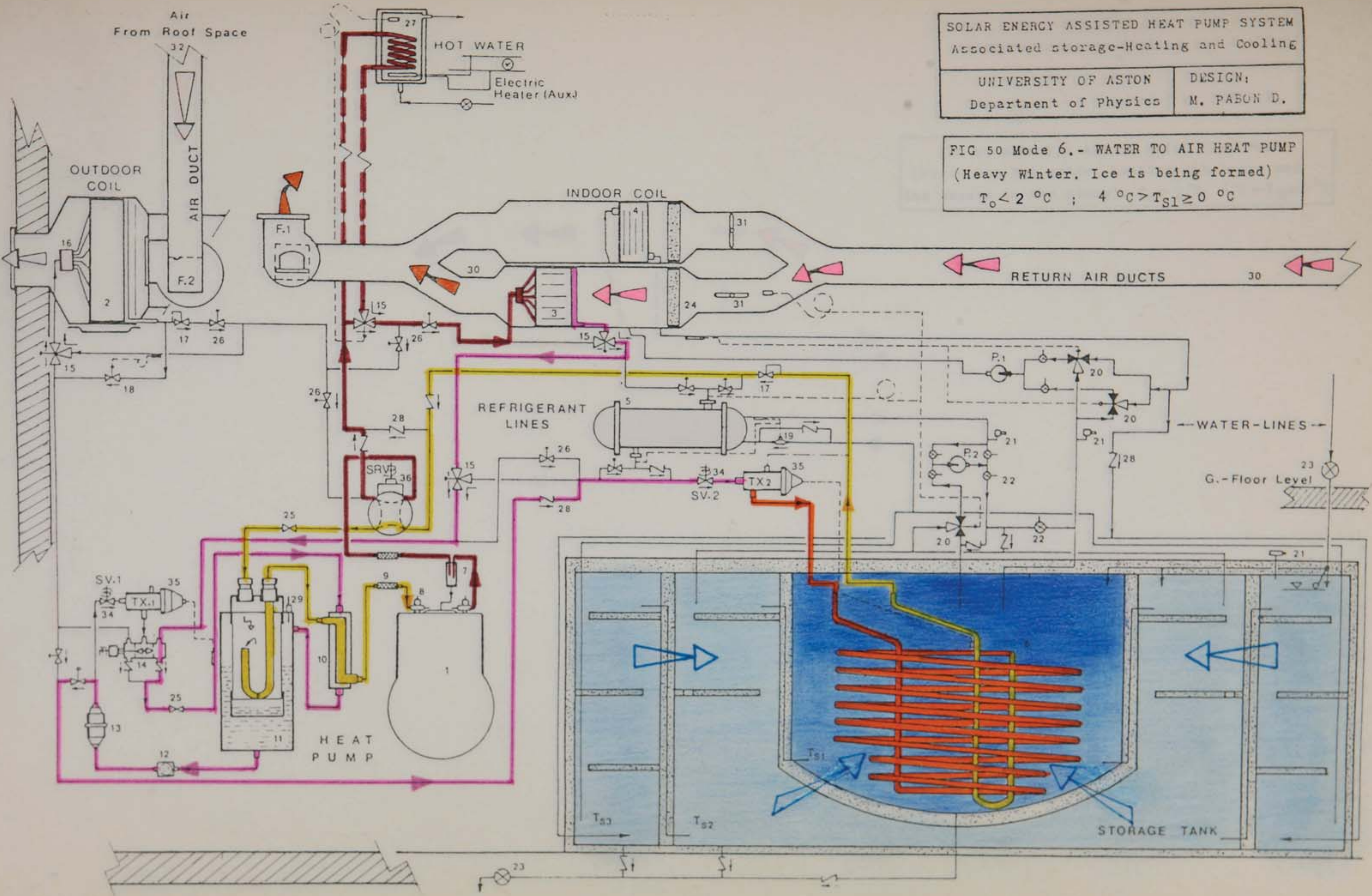
FIG 50 Mode 4.- WATER TO AIR HEAT PUMP (Producing hot water, heating the house from the store). $4 < T_{S1} < 25$ °C ; $T_o < 2$ °C



SOLAR ENERGY ASSISTED HEAT PUMP SYSTEM
Associated storage-Heating and Cooling

UNIVERSITY OF ASTON Department of Physics	DESIGN; M. PADON D.

FIG 50 Mode 5.- AIR TO WATER HEAT PUMP (Hot water, Heating the house and the excess to the store). $T_o > 2$ °C; $T_{S1} < 25$ °C; $T_{S3} > 4$ °C



SOLAR ENERGY ASSISTED HEAT PUMP SYSTEM
Associated storage-Heating and Cooling

UNIVERSITY OF ASTON Department of Physics	DESIGN: M. PABON D.

FIG 50 Mode 6.- WATER TO AIR HEAT PUMP
(Heavy Winter. Ice is being formed)
$T_o < 2\ ^{\circ}C$; $4\ ^{\circ}C > T_{S1} \geq 0\ ^{\circ}C$

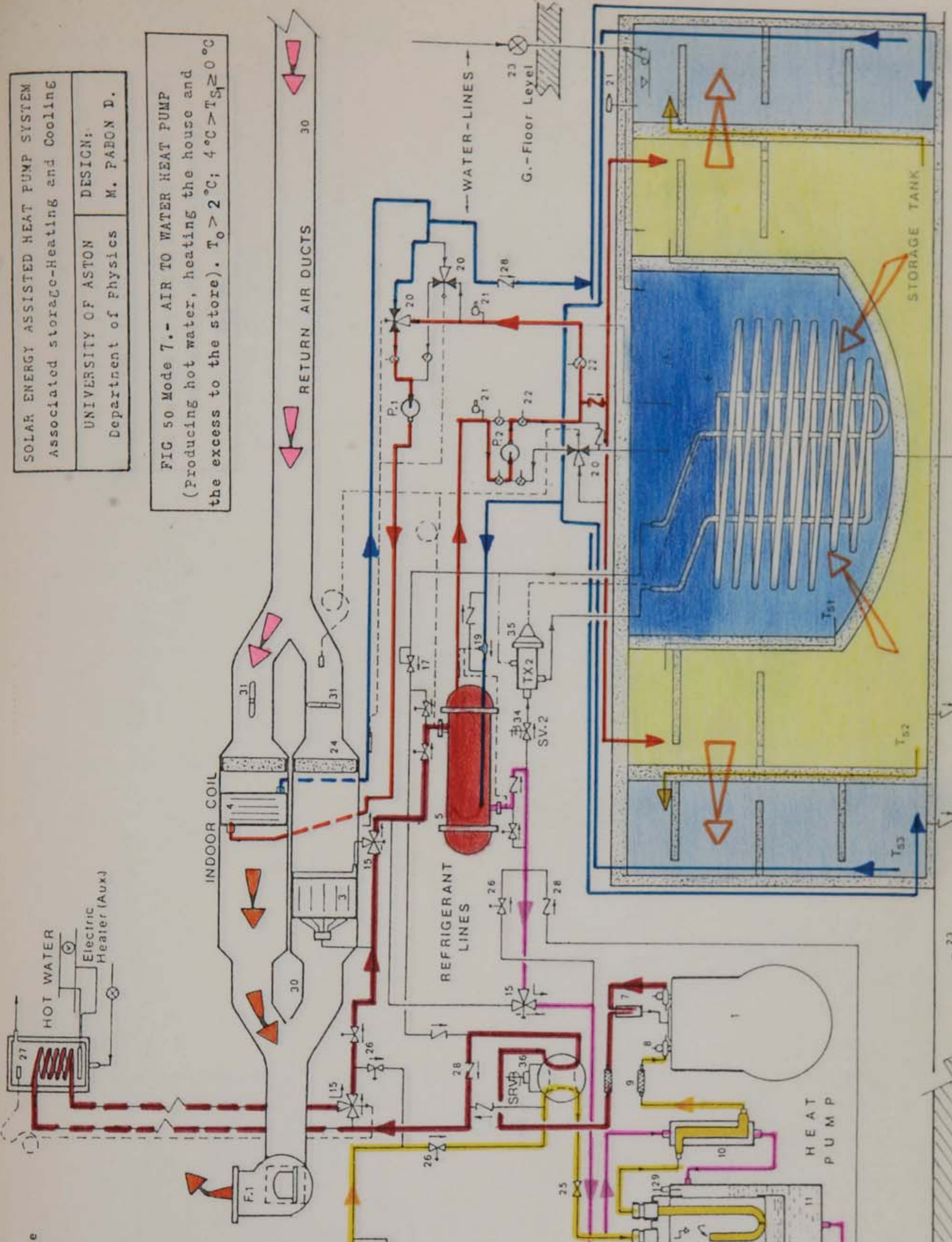


FIG 50 Mode 7.- AIR TO WATER HEAT PUMP (Producing hot water, heating the house and the excess to the store). $T_o > 2\,°C$; $4\,°C > T_S \geq 0\,°C$

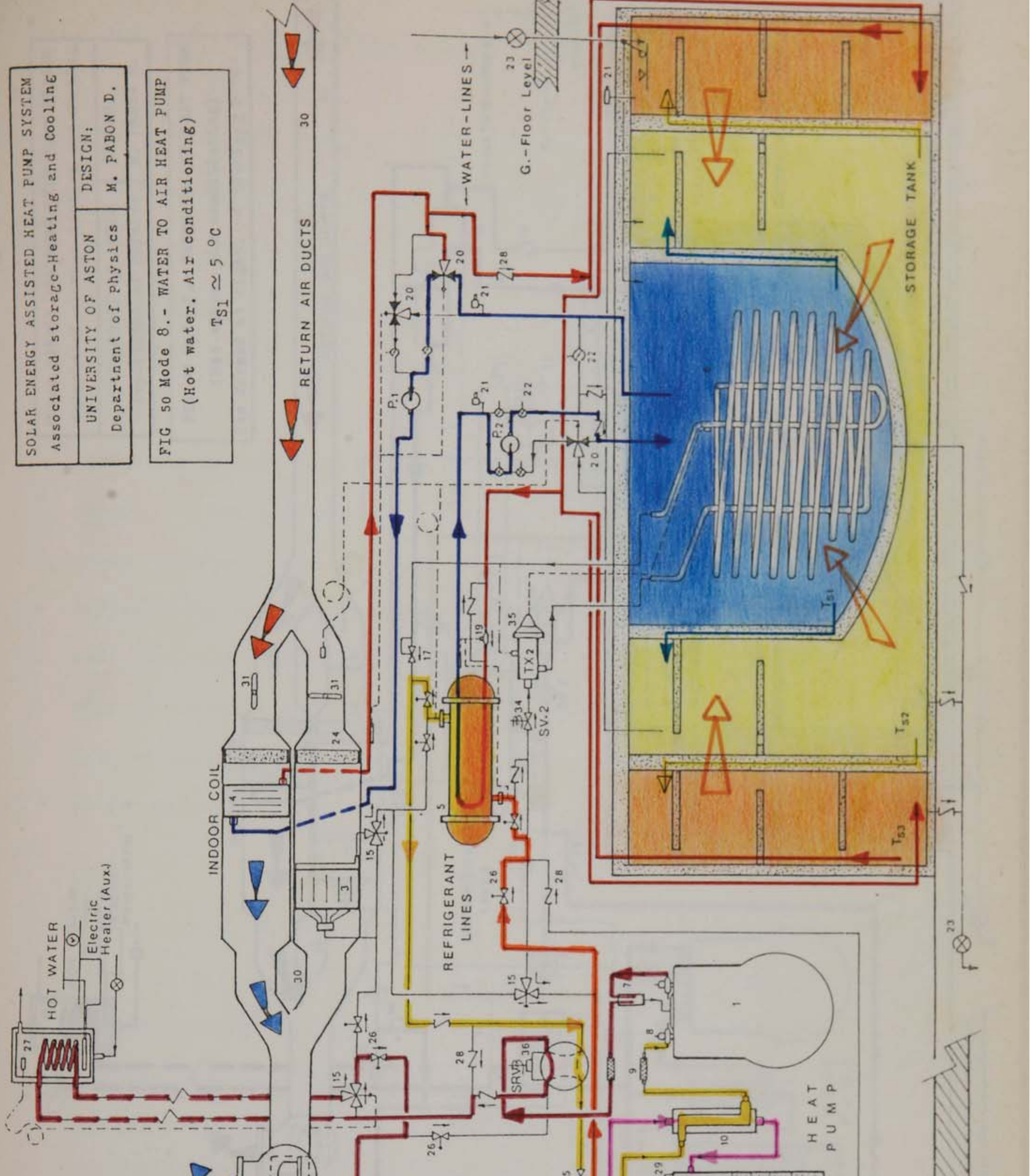
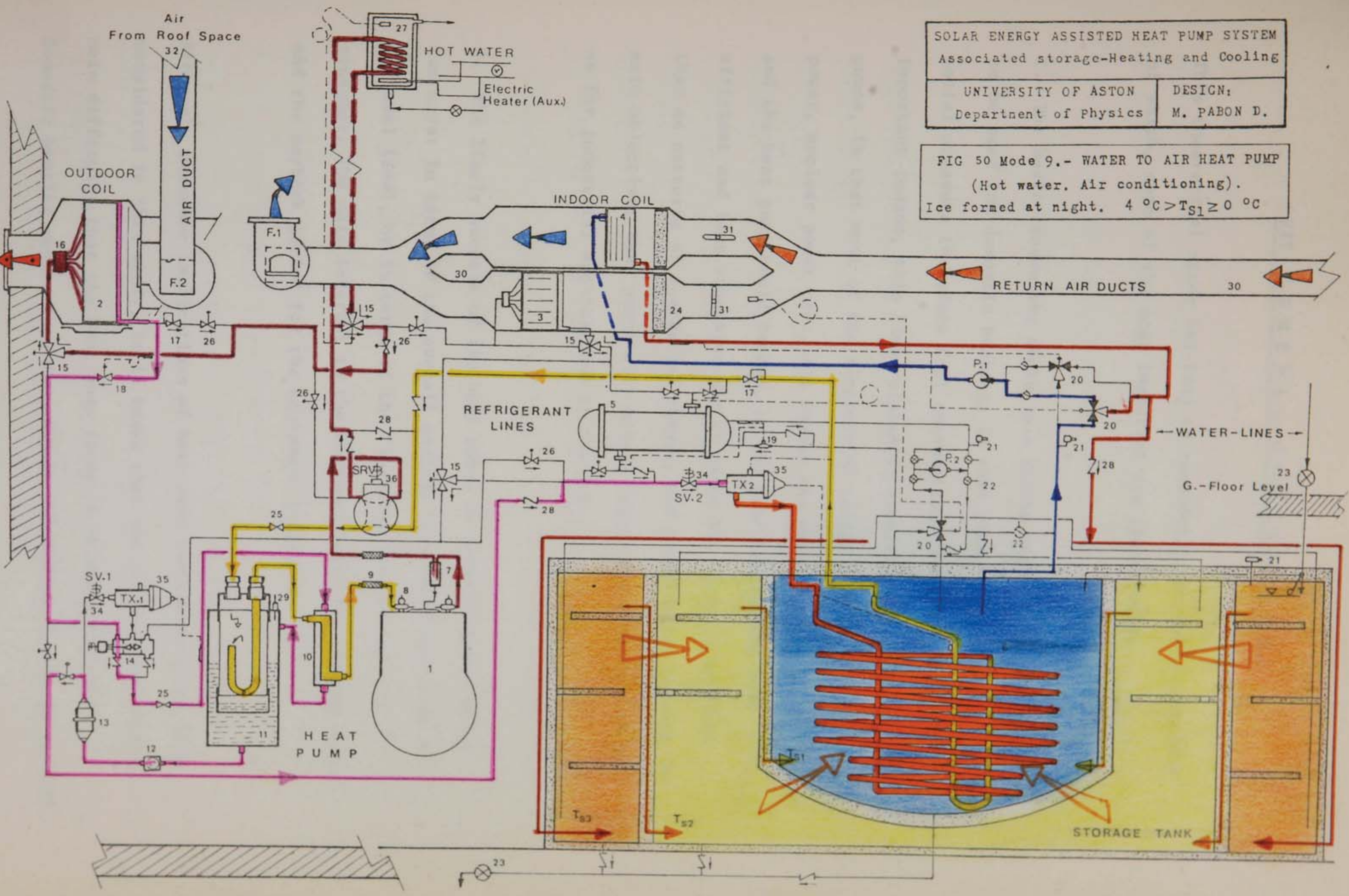


FIG 50 Mode 8.- WATER TO AIR HEAT PUMP (Hot water. Air conditioning)
$T_{S1} \simeq 5\ ^{o}C$



SOLAR ENERGY ASSISTED HEAT PUMP SYSTEM
Associated storage-Heating and Cooling

UNIVERSITY OF ASTON Department of Physics	DESIGN: M. PABON D.

FIG 50 Mode 9.- WATER TO AIR HEAT PUMP
(Hot water. Air conditioning).
Ice formed at night. 4 °C > $T_{S1} \geq 0$ °C

VII GENERAL CONCLUSIONS

This theoretical study has lead to considerations and conclusions of which, some of the most important are listed below.

1. The basic reason why there is a rising interest in heat pumps, to be used for domestic heating, is that there is an economic and social pressure to reduce fuel consumption. However, another important reason, this time related to electrically driven heat pumps, is that most of the alternative energy sources such as wind power, nuclear power etc., all favour the production of electricity, and the heat pump could become, in the near future, the most efficient and less expensive way of using it. There is no doubt that as patterns of energy use change, and conservation becomes the main objective, heat pumps will have a future for domestic, as well as for industrial applications in the U.K.

2. The likely success of the heat pumps is normally measured in two ways: In terms of the overall savings in primary energy at a national level, or in terms of the financial benefits to the consumer. The main interest to the consumer being the running costs and the payback period for the equipment installed.

Although, economic implications of heat pump systems have not been considered in this study, it is known that the initial cost is the main difficulty that the heat pump faces in breaking into the domestic heating market (see for example references: 1, 19, 30 and 38).

Oil or gas-fired boiler systems are half the initial price of heat pump with similar heating power. At the present time the running costs of a heat pump heating system are considered smaller than those of an oil-fired boiler system, but comparable to those of a gas-fired system. Thus, the only incentive to change existing gas boiler system by heat pumps would be a net energy saving associated with the heat pump.

Electrically driven heat pumps have been found to offer a potential energy saving when used for space heating which amounts to 7% of the national primary energy consumption if a COP(H) of 3 could be reached. A further 2% would be saved if water heating was accomplished in the same way with an estimated COP(H) of 2. Nevertheless, these figures, which have been derived from the analysis of using electrically driven heat pumps instead of alternative systems, do not take into account the relative value of the quality of the fuel that would be consumed in each case. Thus, in the test house considered in the last section of this study, the heat pump used 7018 kWh of electrical energy for the heating season, while the heat produced was 18300 kWh. At 60% efficiency, this would have required a quantity of oil equivalent to 30.500 kWh. If one assumes that the electricity was generated and delivered at an efficiency of 30% (see section IV-3.1), then the 7018 kWh delivered would have required the equivalent of 25393 kWh of fuel at the power station. The heat pump system has therefore, with a COP(H) of 2.6, saved the equivalent of 5107 kWh in fuel. Unfortunately, this is a too simple analysis as it takes no account of the fact that electricity is not generated on a large scale from the light oil that is used to fire domestic boilers. The fuel for power stations is either very heavy residual oil or low-grade coal or uranium, none of which can really be used in a large scale for any other purpose. At the moment a factor does not exist to quantify the quality of the fuel that is being consumed, and the consideration of the relative fuel value, in the energy analysis seems set to remain an intractable problem for some time.

3. In considering the capability and the application possibilities of heat pumps for the U.K. climatic conditions (see sections V-4.2 and V-4.3) it was concluded that only small heat pumps are required to provide the average domestic heating needs in most areas of the U.K. (i.e. a power rating of about 2 kW with $U_{AV} \sim 0.8$ W/m^2°C). If one increases the insulation, then smaller heat pumps would be required, with consequent smaller total primary energy savings, smaller fuel cost savings, and a new set of estimates to be made of capital cost, before the economics can be reassessed.

Although an extensive analysis must await further research experience in the use of heat pumps in the U.K. conditions, it is worth while to evaluate the cost effectiveness of domestic heat pump installations trying to strike a balance between the economic assessment and the real terms of an energy analysis that takes into account those problems concerned with the optimum use of the energy.

4. Although the electricity research council has estimated that any domestic space heating unit must meet at least 65% of the design

heat loss of a building when the outside ambient is -1°C, a design capacity rated at only 60% of the conventional peak load could be sufficient to supply 92% of the overall space heating requirements of the season, in the case of North East Scotland area and 100% in the cases of the Midlands and the Thames Valley areas (see section V-4.2). Higher values than 60% could lead to oversized units.

5. Considering the adoption of heat pumps in new buildings, or assuming that the replacement of existing heating units is economically justified, any effort should be made to employ low temperature space heaters. Recent regulations reducing the admissible heating loads contribute directly to a better acceptability of the loss of capacity caused by a lower temperature heat carrier. Under-floor water heating systems are perfectly adequate to heat pump intrinsic requirements of base load - low temperature operation.

6. One of the problems with heat pump development is that there is a tendency for engineers to believe that, since the heat pump is nothing but a refrigerator there is no research needed or development work to be done. There are still technical developments required to increase seasonal performances of heat pumps, particularly for domestic machines with power ratings from 2 to 6 kW. At least part of the heat pump research effor must be directed towards reducing the initial cost, while the rest must aim at improving the heat transferred per unit of power supplied. e.g. new heat exchangers of improved efficiency, variable speed compressors, and modern controls coupled to the systems. More field trials are needed to monitor

performance particularly for air source heat pumps in which performance is dependent on the outside ambient temperature. Investigations need to be made into the optimum use of heat pumps in hybrid systems and in conjunction with short term and medium term stores.

7. The integrated solar assisted heat pump and thermal store system using the roof as a solar radiation absorber has been demonstrated to be technically feasible. It would permit energy savings in the region of 50 to 62% on domestic space and water heating (see section VI-5). The roof acting as an absorber of solar radiation could be modified from the existing tiles to metal tiles in order to improve the solar boost and improve the % of energy savings. Heat transfer coefficient between air and tiles, for different materials must be evaluated.

8. Winter conditions in the U.K. with high humidities give rise to high defrosting requirements. Above 5°C frost does not form on the outdoor coil. Below -5°C the moisture content of air is low enough to give slow frost formation. The solar assisted heat pump system used in the test house permits the reduction of this problem, because the air reaching the roof space decreases in relative humidity.

9. When the tile roof of the test house is used as a large area radiation air panel in series with an air source heat pump and operated in conjunction with a store during the daytime only to provide the heating load, then the seasonal performance (October to

April) is only $\sim$ 6% better on average than a heat pump operated as required over a 24 hour period to provide the load. Values range between 6.2% for a one day store to 6.4% for an intermediate store changed in October to provide for some of the January load. The effective improved performance would be higher because there would be fewer defrost cycles for this mode of operation. This latter contribution has not been quantified although it could increase the above figure to between 7% and 10%.

10. Large electically driven heat pumps are more efficient and reliable and could be employed in domestic heating applications if some form of communal heating can be developed. Large units could be coupled to larger thermal store systems reducing the problem for auxiliary heating because then little or no boosting would be required. Research in this field should be done.

APPENDIX I

Maps of average daily totals of global solar radiation ($\bar{H}$) over the British Isles.

These maps prepared in 1976, by the National Radiation Centre, Meteorological Office, Department of Energy, show only the broadscale features of the distribution of global solar radiation on a horizontal surface, over the British Isles for the period 1941 to 1970.

The principal factors which determine the pattern or isopleths are:

1) The general decrease in radiation levels with increasing latitudes.

2) The general decrease in radiation levels in inland areas associated with the increase in cloud amount, particularly over high ground.

3) The accentuation of radiation loss over a broad central urban area.

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