"SIMULATION OF THE BEHAVIOUR OF SPACE AND WATER HEATING SOLAR SYSTEM"

BY

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SYNOPSIS

Computer flow diagram is constructed to get the net heat gain of the collector used for a space and water heating system. In addition, the flow and return temperatures, the space and water heating demands are computed. The storage tank is subdivided into four regions. Dummy variables are assumed to simulate the regions situated in between these 4 regions. The flow chart gives the computer simulation of these regions with respect to volumes, heat flow quantities and the corresponding temperatures.

I. INTRODUCTION

Some 25 % of all energy consumed is used for the heating of buildings and domestic hot water. Space and water heating requires the lower grade of energy. The highest collection efficiencies are obtainable with low temperature collection(7). Thus, with the consideration of these facts, the use of solar energy for space and water heating seems to be an obvious proposition.

Focusing devices require a tracking mechanism and respond to direct radiation only. Flat plate collectors can utilize both diffuse and direct radiation and may be fixed in one particular position. They may become part of the building envelope, replacing an enclosing element, such as a wall or a roof. There are lar areas involved. A domestic building may have a flat plate colle as much as 80 m^2 in area. It will not only dominate the appe of the house, but will also impose quite strict design cons

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Any solution applying a solar heating system as an afterthought to a building already designed (let alone built) cannot hope to have the same degree of success as one which is an organic part of the design.

Solar radiation affects buildingsin two ways:

1) Entering through windows, absorbed by surfaces inside the building, thus causing a heating effect.
2) Absorbed by outside surfaces of the building, creating a heat input into the fabric, which will be partly emitted to the outside, mostly by convection, but will partly be conducted through the fabric and subsequently emitted to the inside.

Both effects can be influenced (if not determined) by the designer.

The transmission through windows is assigned by:

a) orientation of the window thus the intensity of radiation incident on its surface.
b) size of the window.
c) type of glazing i.e. clear, heat absorbing, heat rejecting or photochromatic glasses.
d) shading devices either:
   d.1) external i.e. grilles, louvres, canopies, awnings and shutters .
   or d.2) internal i.e. blinds, curtains.

If the intensity of radiation (I) incident on the window is known, the rate of heat gain will be

$$Q_s = I\ A\ \tau$$

Where
Q_s = solar heat gain (watt)
A = Window area (m^2)
τ = Solar gain factor.

The solar gain factor includes the transmission coefficient (θ) plus an experimentally determined proportion of the absorption coefficient, corresponding to the proportion of the heat absorbed by the glass that will subsequently be emitted inwards.

In addition to the solar gain, there will be a conduction heat flow through the window:

$$Q_c = A\ U\ \Delta t$$

where U is the transmittance (watt/m^2 deg c)

Δt is the temperature difference ($t_o - t_i$ for heat gain) (deg c)

Q_s and Q_c may have opposite signs, e.g. on a sunny day in winter, when there may be a simultaneous solar gain and conduction heat loss.

The same equation is used to determine heat flow through solid walls or roofs. If there is solar radiation incident on opaque surfaces, its heating effect (over and above the effect of air temperature) can be expressed as the "sol-air excess" temperature:

$$t_e = \frac{I \times a}{f}$$

Where a = absorption coefficient

f = surface (film) conductance (W/m^2 deg c)

Which is the same expression for the equilibrium temperature of irradiated surfaces.

The sol-air temperature is the sum of this and the outdoor temperature (t_o)

$$t_s = t_e + t_o = \frac{I \times a}{f} + t_o$$

The heat flow due to the combined effect of warm air and solar radiation can be found if for Δt we use $t_s - t_i$ in lieu of the $t_o - t_i$.

It will be seen from this expression that the rate of heat flow through solid walls and roofs is determined by:

a) orientation of the surface i.e. the intensity of radiation incident on it.

b) the area of the exposed surface.

c) the absorption coefficient of the surface.
d) the surface (film) conductance, governing the heat emission, depending partly on surface qualities (texture, colour) but more on the velocity of air movement passing the surface, i.e. on the degree of exposure. The surface of film conductance is a composite quantity, allowing for radiant as well as convective heat transfer. Its value for building surfaces is between 8 and 80 Watt/m^2 degc, normally taken between 12 and 20 Watt/m^2 degc. It varies with air velocity.

Definitions[7]

$\frac{a}{e}$ ratio: is the ratio of absorption coefficient for solar radiation to the amission coefficient at operating temperature, a measure of the selectivity of absorber surfaces.

selective absorber: is a dark surface having a high absorption coefficient for short wave solar radiation but a low emission coefficient for long wave low temperature radiation.

specific heat loss rate: is an index of the thermal properties of a building, given as the total (conduction + convection) heat loss rate per unit temperature difference (W/degc).

Equations

When radiant energy falls on a matt black surface, much of it is absorbed. This may be a complex process, which varies with the type of absorber material. It involves scattering, photon absorption, acceleration of electrons, multiple collisions, but the end effect is that the radiant energy of all grades (all wave lengths) is degraded to heat. Molecules of the surface will be excited, a temperature increase is caused. The absorption coefficient of various types of black absorber varies from 0.8 to 0.98 (the remaining 0.2 or 0.02 is reflected). Some of this molecular movement (i.e. heat) is transmitted to other parts of the body by conduction and some of it is re-emitted to the environment by convective and radiant processes. This emission of heat (heat loss) depends on the difference in temperature between the surface and the environment. Thus, as the surface is heated, the heat loss is increasing[1,2]

1) When the rate of radiant heat input is equalled by the heat loss, an equilibrium temperature is reached i.e.

when $Q_i = Q_L$

i.e. $I \times a = f \times \Delta t$

$$\Delta t = \frac{I \times a}{f}$$

which yields the equilibrium temperature

Where Q_i = heat input rate (watt/m^2)
Q_L = heat loss rate (watt/m^2)
I = incident intensity (watt/m^2)
a = absorption coefficient
f = film, or surface conductance for emission (watt/m^2 degc)

It depends on several factors (as explained before), one of them is the temperature of surfaces opposite the absorber (at any distance); thus it allows for both convective and radiant heat transfer process. Beyond the normal range zero to 40°C, f is no longer constant. In actual fact, its value varies with air velocity and it is sufficient to assume that

$$f = 11 + 0.85\, V$$

where V is the air velocity (m/S)

2) In a given time (e.g. one hour) the temperature increase can be found from

$$H_i = H_l + H_g$$

where H_i is the heat input (Wh)

$$= Q_i \times h \times A$$
$$= I \times a \times h \times A$$

H_l is the heat loss (Wh)

$$= Q_l \times h \times A$$
$$= f \times \Delta t \times h \times A$$

H_g is the heat gain (Wh)
$= C \times \Delta t$

where C = heat capacity of body (Wh/degc)
= mass x specific heat or
= volume x volumetric specific heat.

Taking 1 hour and 1 m^2, the h and A terms may be omitted, then we have:

$$I \times a = f \times \Delta t + \Delta t \times c$$

$$\Delta t = \frac{I \times a}{f \times c}$$

Note that the specific heat of steel radiator
= 0.13 Wh/Kg degc.
and the volumetric specific heat = 1.16 Wh/litre degc.

3) Having a glass cover, the heat loss from the plate will be almost exclusively convective - conductive, thus the "air-to-air transmittance" concept (the U - value) used in building heat loss calculations will be applicable. This can be taken as: $U = 5 W/m^2$ degc for single glass
$= 2.7\ W/m^2$ degc for double glass

For low temperature applications, the heat loss rate can be taken as:

$$Q_L = U \times \Delta t$$

and the total heat loss as

$$H_L = U \times \Delta t \times h \times A$$

Some authors suggest that when the value of Δt exceeds about 20 degc, it will be more accurate to use the non-linear relationship:

$$Q_L = c \times \Delta t^{1.25}$$

where
$c = 2.38\ W/m^2$ degc for single glass
and $= 1.70\ W/m^2$ degc for double glass

4) If some thermal fluid, e.g. water or air is circulated as a carrying medium in thermal contact with the absorber plate, it will be heated and thus some of the heat absorbed by the

plate will be removed. The temperature of the plate is thereby reduced to below the above calculated equilibrium temperature, and this will reduce the heat loss.

In normal operation, we must distinguish 2 temperature differences:

a) for effective gain,

$$\Delta t = t_f - t_r$$

where t_f is the flow temperature from plate.

t_r is the return temperature to plate

b) for heat loss,

$$\Delta t = \frac{t_f + t_r}{2} - t_o$$

where $\frac{t_f + t_r}{2}$ is denoted as the mean plate temperature

and t_o is the outdoor air temperature.

Since $H_g = H_i - H_l$

Heat gain $H_g = C \times \Delta t = C\ (t_f - t_r)$

and $C = F \times s$

where F is the flow rate (litre/h)

s is the volumetric specific heat of water

= 1.16 Wh/litre degc.

Heat input $H_i = I \times a \times A \times \theta$

where I is the incident intensity (W/m^2 or $Wh/m^2\ h$)

a is the absorption coefficient of plate

A is the area of plate (m^2)

θ is the transmission coefficient of glass.

Heat loss $H_l = U \times \Delta t \times A$

$$= U \times A \times \left(\frac{t_f + t_r}{2} - t_o\right)$$

$$C\ (t_f - t_r) = I \times a \times A \times \theta - U \times A \left(\frac{t_f + t_r}{2} - t_o\right)$$

$$t_f = \frac{I\ a\ A\ \Theta + t_r\ (C - \frac{AU}{2}) + AU\ t_o}{C + \frac{AU}{2}}$$

II. WATER SYSTEMS

Any solar heating system will consist of five major components: collector, storage, auxiliary heater, distribution system (including emitters), and controls (including pumps and fans). The simplest form can be represented as in figure 1. Heat transfer from the collection circuit into the storage may take place by direct flow or through a coil.

Asystem designed to be fully operative on all days of the year, would be uneconomical. In one instance, it has been found that if a given area of collector provides all the heat required for 320 days of ayear, a doubling of the area will be necessary to cope with a further 30 days, and asecond doubling is needed to provide enough heat for the remaining 15 days(7). The low of diminishing returns prevails. It is more economical to choose the lesser size and rely on an auxiliary heater of some kind for the mid-winter days. This auxiliary heater may be a calorifier (fed by hot water froma boiler), a boiler connected in series or even an electric immersion heater. Three alternative positions may be considered as shown in figure 2.
Position 1 needs the smallest heat output rate, as more time is available for its functioning but it will have to heat a large volume of water unnecessarily and it may prevent collection the following day. It is essentially a slow response system.

Position 2 would make it possible to isolate the heating circuit from the storage and run it as a conventional central heating system, when the storage temperature is below the level required for heating.

Position 3 is the most favoured one, having "the best of both worlds", having the advantages of (2) but making it possible to

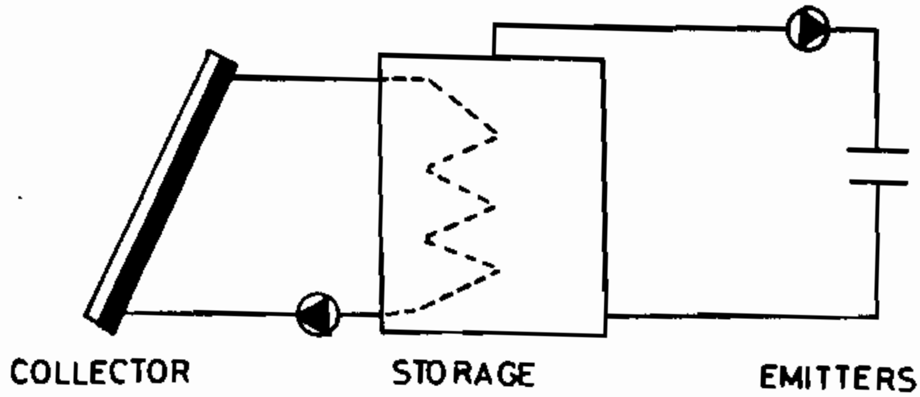


FIG.(1). A SOLAR HEATING SYSTEM.

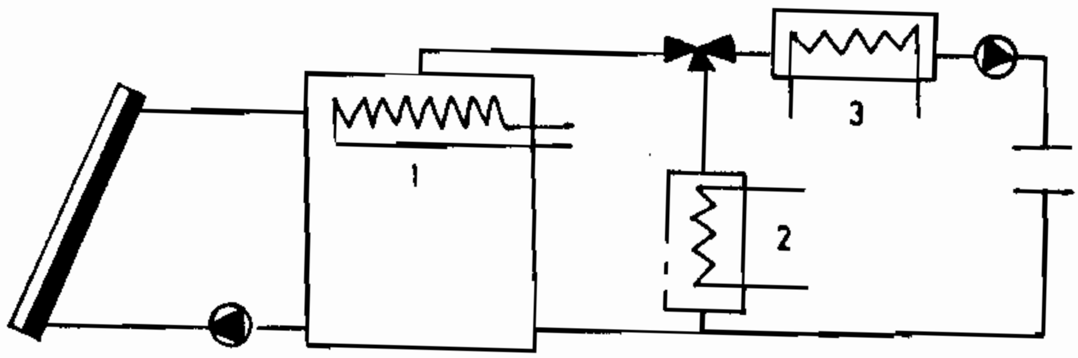


FIG.(2) POSITIONS OF THE AUXILIARY HEATER.

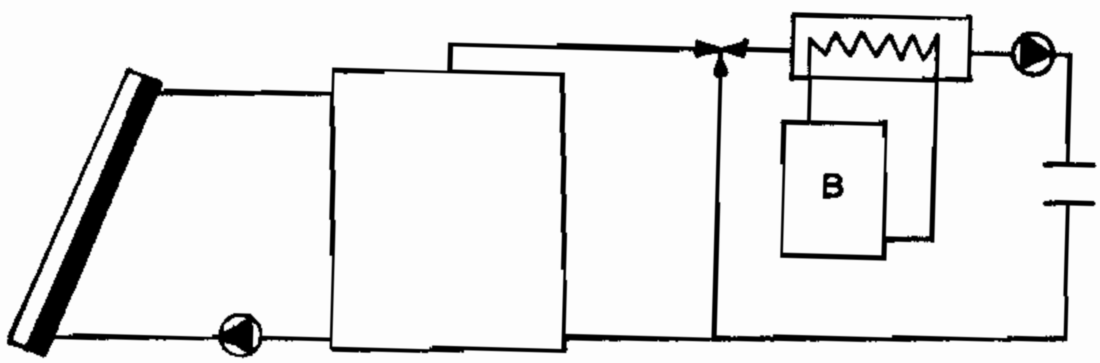


FIG.(3). A 3- PHASE SYSTEM.

utilize the solar collector system as a pre-heater, thus reducing the heat input requirement. Eg if the emitters are designed to use 40°C water (resulting in a return water temperature of, say 25°C) the operation will have three phases (figure 3);

a) when collection - (or upper storage) temperature is above 40°C the auxiliary heater does not operate.
b) As the collection temperature is between 40°C and 28°C, the circulation still flows through the storage and the auxiliary heater works as a topping-up device, bringing the water temperature up to 40°C .
c) When the collection (or storage) temperature drops below the set limit of 28°C, the water flow is short-circuited and the auxiliary heater will work as a conventional central heating system, independently of the solar collector and storage.

To allow an optimum collection efficiency under widely varying conditions, a 2- stage collection system can be adopted, relying at times on a heat pump system for the transfer and upgrading of the heat (figure 4). With high intensity radiation, when a flow temperature greater than (say) 55°C is achieved, the flow is directed into the top tank. Here it will discharge some of its heat to the stored water then the flow enters the lower tank. The coolest part of this is fed back to the collector. With a lesser radiation intensity, thus lower collection temperature, the flow is directed into the lower tank. A compressor type heat pump will use this lower tank as the "source" and the upper tank as the "sink".

III. SPACE HEATING

The utilization of solar radiation in space heating is generally attained by the three following methods:

First method in which the building is used as a collector. This is the common - sense approach in building design. Its basis is a thermally very efficient building, with good insulation

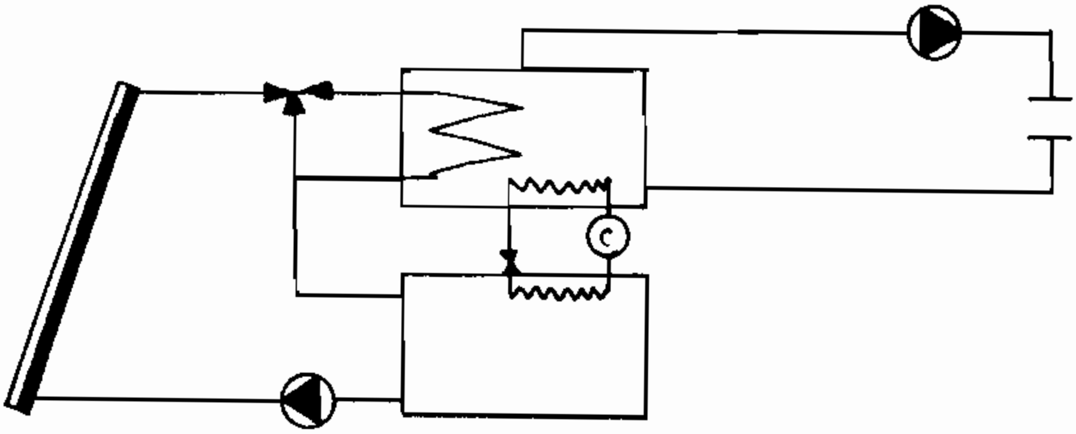


FIG.(4). 2_ PHASE COLLECTOR WITH HEAT PUMP.

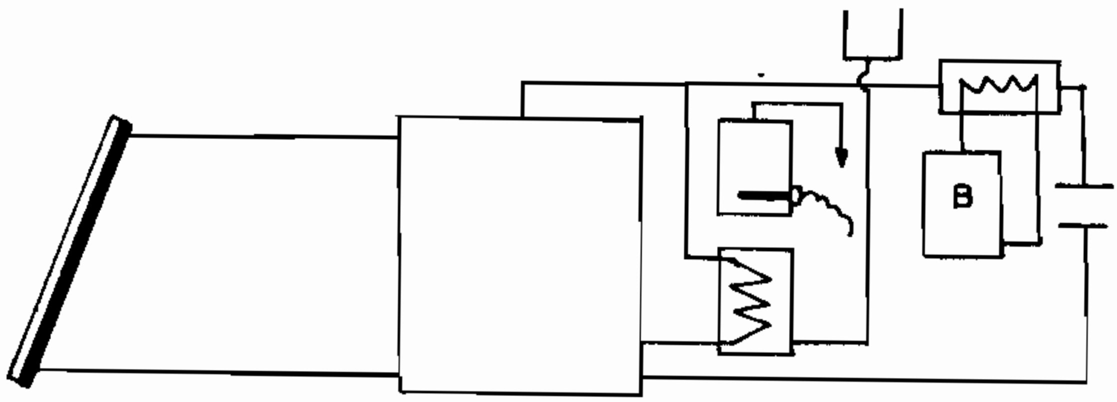


FIG.(5.a).

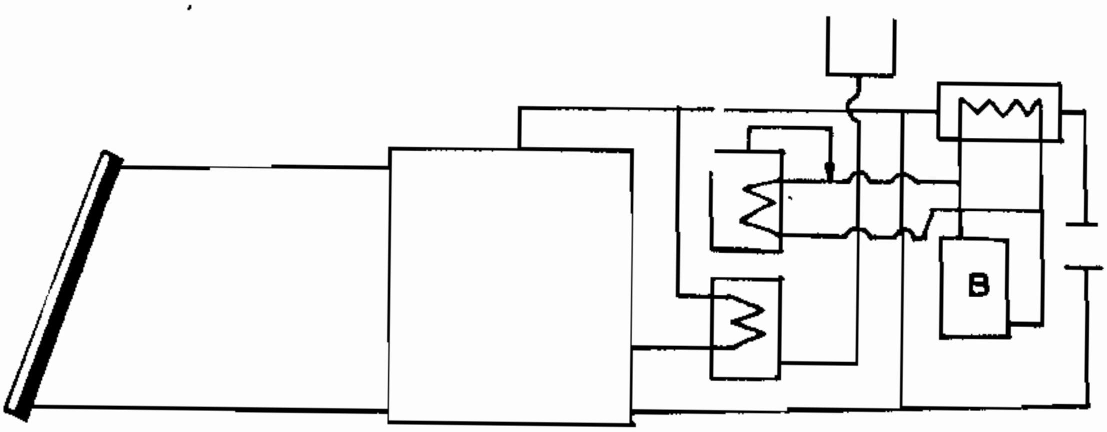


FIG.(5.b.).

placed outside the main mass of wall and roof elements. It has large windows facing the equator, which can be closed by shutters or heavy curtains when there is no solar gain, in order to reduce heat loss. Without such shutters or curtains the glass surface may give an annual cumulative heat loss greater than the annual cumulative solar gain. The second method in which an external enclosing element of the building (a wall or a roof) may be designed in such a way as to act as a collection device. Eg, a massive wall, with the outside painted black, may be covered by one or two sheets of glass. It will act as an absorber, storing some of the heat in its mass and providing an output mechanism through the convection currents induced.

Both methods 1 and 2 can significantly improve the thermal conditions and can substantially reduce the energy requirement of the building, but neither provides a flexibility of controls and neither is precisely predictable in its performance. Given the tolerances and accuracy of building work and the unpredictability of user behaviour, any performance prediction can only provide a rough guidance.

The third method uses flat plate collectors in which 2 basic types must be distinguished water and air systems. The operating principles of water systems are essentially the same as of water heating. The difference is only in magnitude: a single family dwelling may have as much as 80 m^2 collector area. It is usual to provide for the storage of a few days of heat collection, which would bridge over a short period of insufficient collection when the sky is heavily clouded.

IV. COMBINED WATER AND SPACE HEATING

If the water heating system is to be linked with this heating system, further difficulties arise. As the domestic hot water supply should be 60 or 65°C (for kitchen and laundry purposes; for bathroom use 45°C would suffice) there will have to

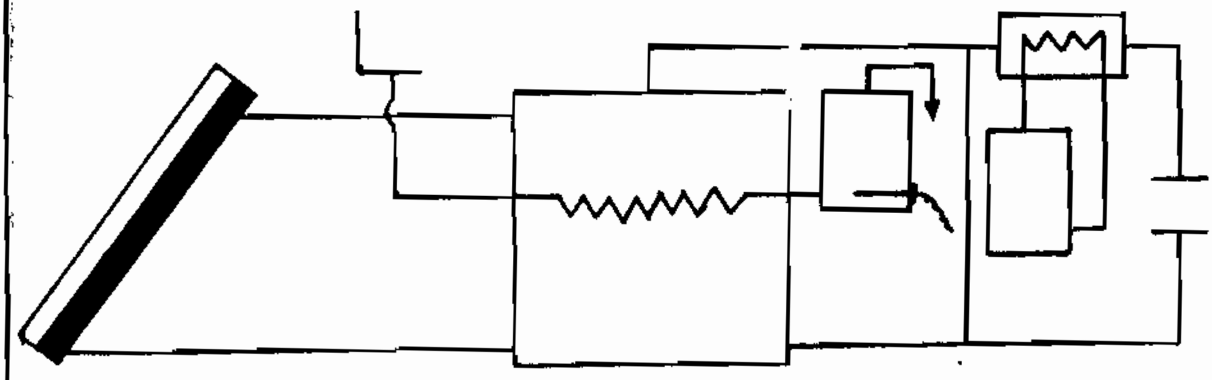


FIG.(5C)

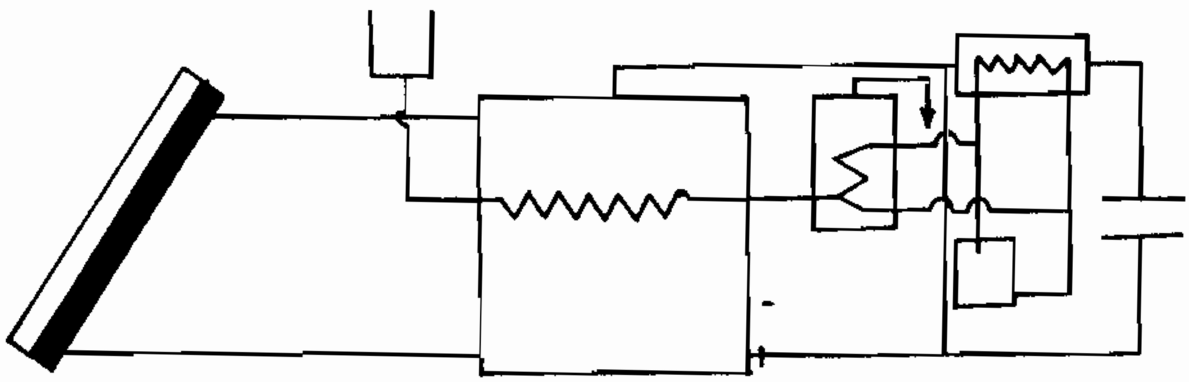
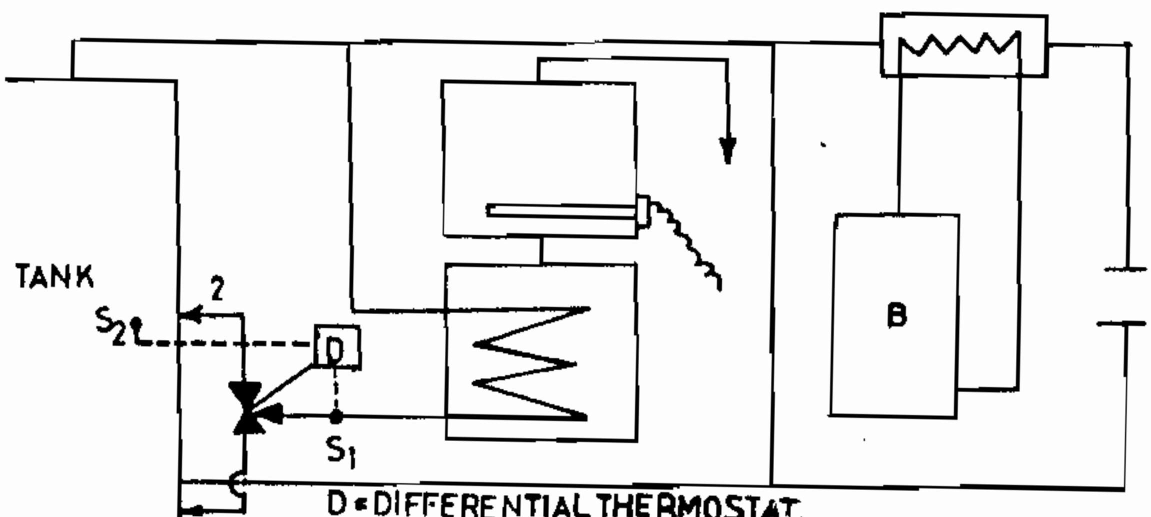


FIG.(5.d)

FIG.(5).COMBINED WATER AND SPACE HEATING SYSTEM.



D=DIFFERENTIAL THERMOSTAT.
S=TEMPERATURE SENSORS.

FIG.(6). TWO WAY INLET OF THE RETURN CIRCIUT TO ASSIST STRATIFICATION.

be a topping-up device. If the heating operates at 40°C, the same medium could bring up the cold water to almost the same temperature, but the remaining 20°C must be added by this topping-up device. This could be an independent electric immersion heater or a calorifier fed by the same boiler as the heating. The pre-heating itself can be a flow-through or a storage type system.

Thus we have four possible variants, shown in Fig. 5

a) storage type pre-heater electric topping-up
b) storage type pre-heater boiler topping-up
c) flow-through pre-heater electric topping-up
d) " " " " boiler topping-up

In the summer where there is no space-heating draw-off, the whole of the storage temperature may increase to above 80°C thus the pre-heater will do all the heating required.
The storage type pre-heaters (a and b above) have the advantage that more time is available for heating the given volume of water, thus a lesser heat transfer surface will suffice. There is however the need to operate a separate circulating pump. The temperature of water returning into the storage tank from the pre-heating circuit will vary between wide limits. At the beginning of the pre-heating cycle it will approach the cold water temperature (just above 10°C) whilst towards the end of the cycle it will be very near to the flow temperature. This may disturb the stratification pattern in the main storage tank. A system has been devised, using a thermostatically controlled 3-way valve, to discharge into the storage tank near its bottom in the first half of the pre-heating cycle and much further up towards the end of the cycle (Fig. 6). There was also a suggestion to use a motorized swingarm tube (Fig. 7) which, controlled by a differential thermostat, would always discharge at a level in the storage where the temperature of the particular layer matches the in - flow temperature. The same system could also be used at the entry of collector flow into the storage tank. The flow - through type pre - heaters (c and d) are much simpler, but must be capable of heating

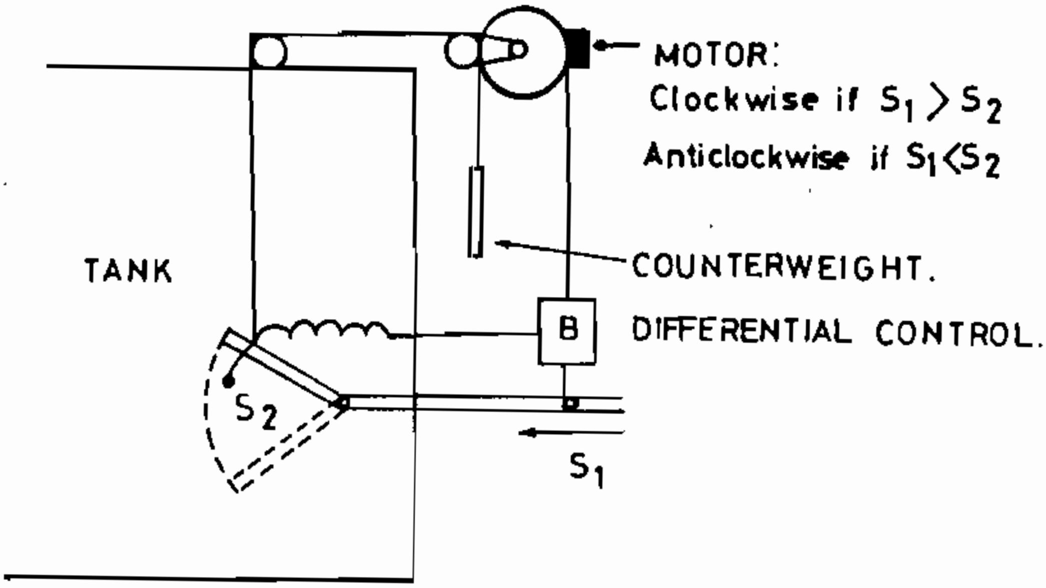


FIG.(7). SWING_ARM TUBE INLET OF THE RETURN CIRCUIT TO ASSIST STRATIFICATION.

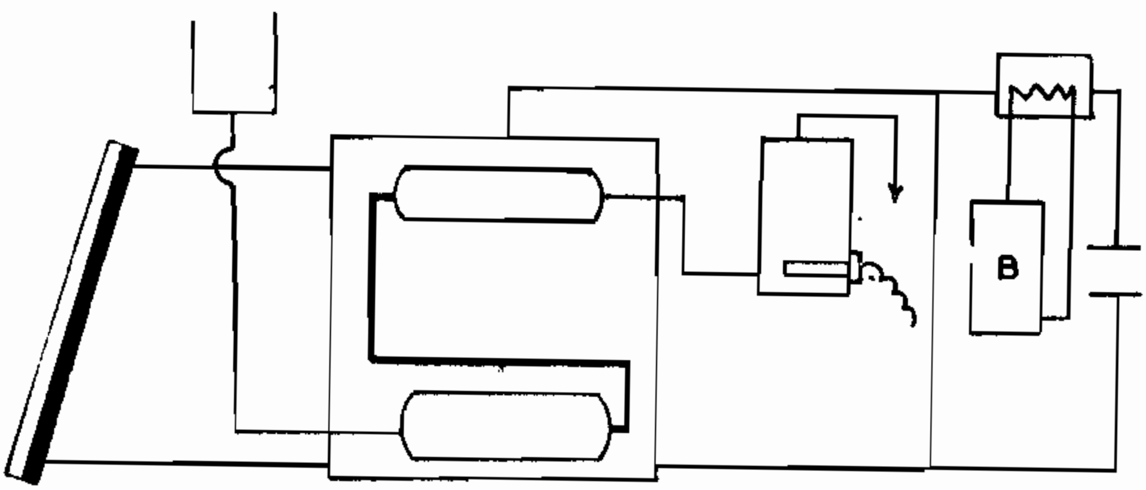


FIG.(8). SEMI_STORAGE TYPE PRE_HEATER T THE HOT WATER SYSTEM.

a certain amount of water whilst flowing through. With a large rate of draw - off they may not give as great a heating effect as they should. A semi - storage type pre-heater is also possible. This consists of a container placed inside the storage tank. The volume of this should correspond to the volume of the maximum demand likely to occur within the period of recovery (Fig. 8).

V. SOLAR CONTRIBUTION

The supply of solar heat is out of phase with the heating demand. Much of the energy available in the summer will be wasted, as there is no simultaneous demand (in fact the collection system will be self-regulating to some extent: when the water temperature reaches the balance point, there will be no further collection, as the heat loss from the collector will equal the solar gain).

The amount of energy usefully collected will depend on the following parameters:

1) Incident energy.
2) Optical loss through transparent cover.
3) Absorption properties of the receiving surface.
4) Heat transfer properties of the absorber (from surface to fluid, i.e. the plate efficiency).
5) Thermal transmittance of transparent cover, which is a factor of heat loss .
6) collection temperature, which in turn depends on:
   6.1) fluid flow rate
   6.2) fluid temperature at entry to collector.
7) External air temperature[7].

The collection efficiency, i.e. the ratio of utilized energy to incident energy

$$\eta = \frac{H_g}{I \times A \times time}$$

will be the result of all these parameters. Under favourable conditions, it may be as highas 70% but it can be as low as 30%.

Accurate prediction of the efficiency can only be achieved with a full simulation of the system's behaviour, taking into account all seven of the above parameters. This is clearly at task for the computer.

The intensity of radiation constantly changes, but the cumulative total for a day or even for a month will provide a sufficient basis for an estimate. The average monthly totale measured on a horizontal plane, totals for a vertical south facing wall, and for an optimally tilted plane at certain country are calculated and tabulated for each month. It may be seen that the optimally tilted plane receives (on average) about 1.5 times as much energy as the horizontal surface. If the values for a horizontal plane are available, these may be multiplied by the product of this ratio and the 40% assumed efficiency: 1.5 x 0.4 = 0.6. This would give the amount of collected energy for the period taken.

Even with manual calculations, it will be necessary to work out at least manthly balances (both demand and collection) in order to establish the amount of effective solar contribution which is calculated for each month having obtained:

1) the monthly amounts of radiation (collected energy) in KWh/m^2 at certain location.
2) the specific heat loss rate of a certain house in KW/degc. (say 0.25 KW/degc)
3) the annual number of (deg. - day) or (degc. h) of our location.
4) the solar collector area in m^2. Then the calculation of the effective solar contribution for each month is executed as follows:
   a) Space heat requirement is computed as 0.25 x (degc - h) for each month.
   b) The unit collection in KWh/m^2 is calculated as the horizontal total radiation in KWh/m^2 multiplied by 0.6 (1.5x0.4)
   c) Total collection is equal to the unit collection in KWh/m^2 multiplied by the collector area in m^2.
   d) The solar contribution is given by item (c) but not more than (a).

Temperatures (°C)

TST starting temperature
TEO outdoor air temperature
TMP mean plate temperature
TPP same in "previous" hour
TEF flow temperature (from plate)
TER return temperature (to plate)
TTU top, upper
TTL top, lower
TLU lower, upper
TLL lower, lower
T (1) → T (5)

layers in storage

dummy storage temperatures

T D 1 → T D 5

temperature difference settings

THF flow temperature in heating circuit
TSN space heating "normal" return temperature
TSR space heating return temperature
TPH temperature in pre-heating cylinder
TWR water heating return temperature
TRM room temperature

Heat flow quantities (watt or W)

QIN radiation absorbed
QLO heat loss from collector
QSM space heating max. output
QSH space heating output
QGN heat gain or loss by storage
Q (1) → Q (10)

dummy variables for stratification

QSO heat content of storage (Wh)
QSN "new" value for QSO (Wh)
HCP heat capacity of collector (Wh/degc)
HLR heat loss rate of house (W/degc)
QAS auxiliary to space heating
QAW auxiliary to water heating
QWH total water heating

VI. SIMULATION OF WATER AND SPACE HEATING SYSTEM

(A) System under research

The combined water and space heating system shown in Fig.5.a is simulated to derive the different requirements by the aid of computer. It is essentially consisted of a collector, storage, heating circuit (water and space), distribution circuit, auxiliary heater and electric immersion pre-heater.

(B) Glossary of symbols.

Constants:

KSH coefficient of heat transfer inspace heating fan-coil.
KWH coefficient of heat transfer in water pre-heater coil.

Flow quantities (litre/h)
FLO in collection circuit.
FTO total flow into heating circuit.
FWH in water heating branch
FSH in space heating branch
FSR in space heating return
FCW "consumed" water (in H/W system)
FPH set value for FWH
FL 1
FL 2 } Low and high settings for FLO

Volumes (litre)
VTU top, upper
VTL top, lower
VLU lower, upper
VLL lower, lower } Storage layers

V(1) → V(5)
VS (1) → VS (9)
UTU
UTL
ULU
ULL } dummy variables for stratification in storage

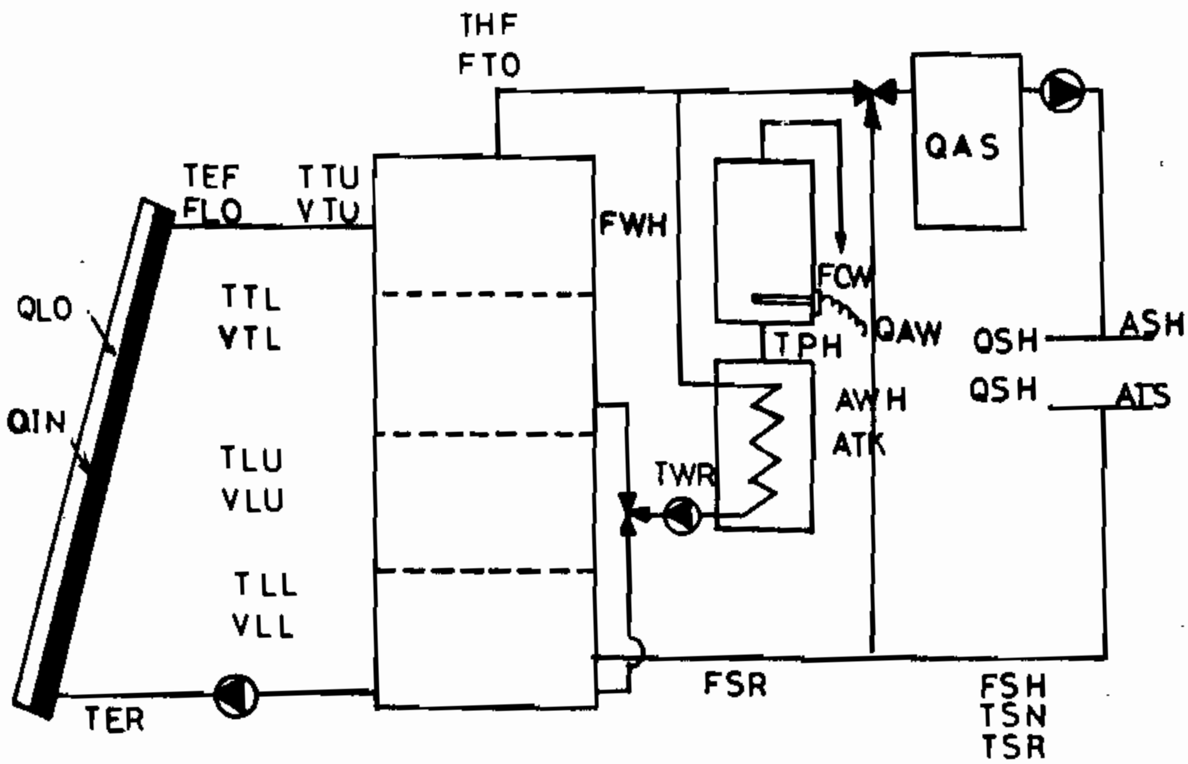
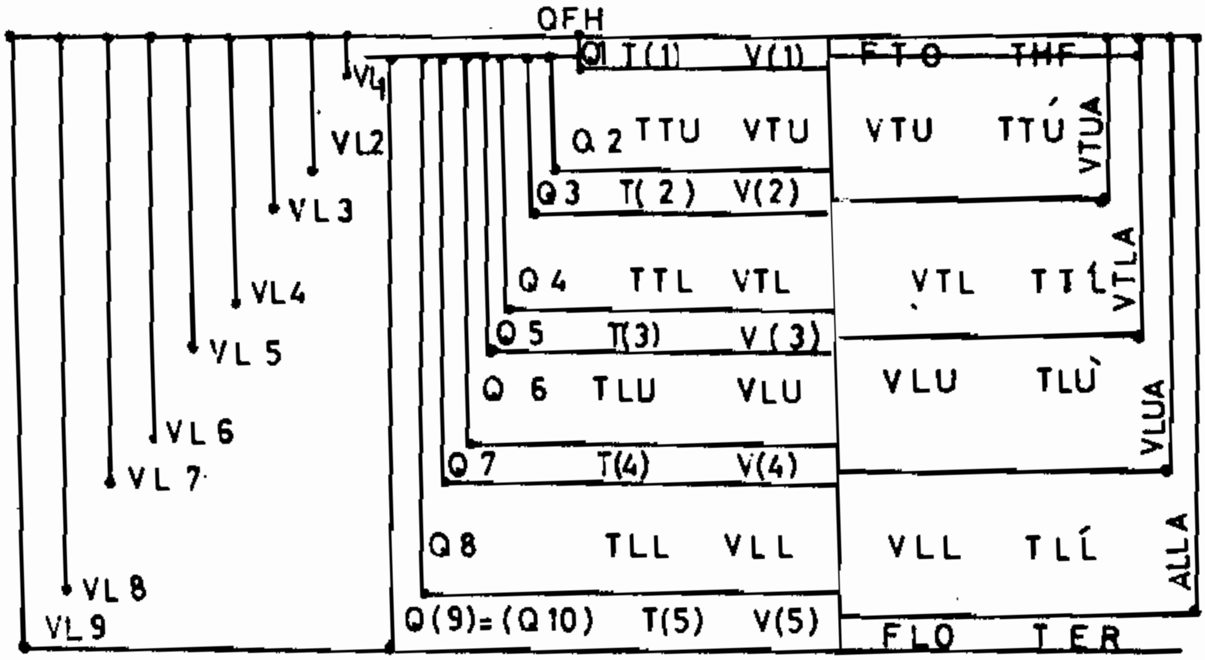


FIG.(9). SYSTEM UNDER RESEARCH.



DUMMY VARIABLES FOR TEMPERATURE STRATIFICATION IN STORAGE TANK.

Areas (m^2)

ARP collector plate

AWH water pre-heating coil

ASH space heating coil

Time

HON hour "on"

HOF hour "off" } time settings for boiler

TIM fraction of hour worked for space heating pump.

Others

ATS attenuation coeff. in space heating coil

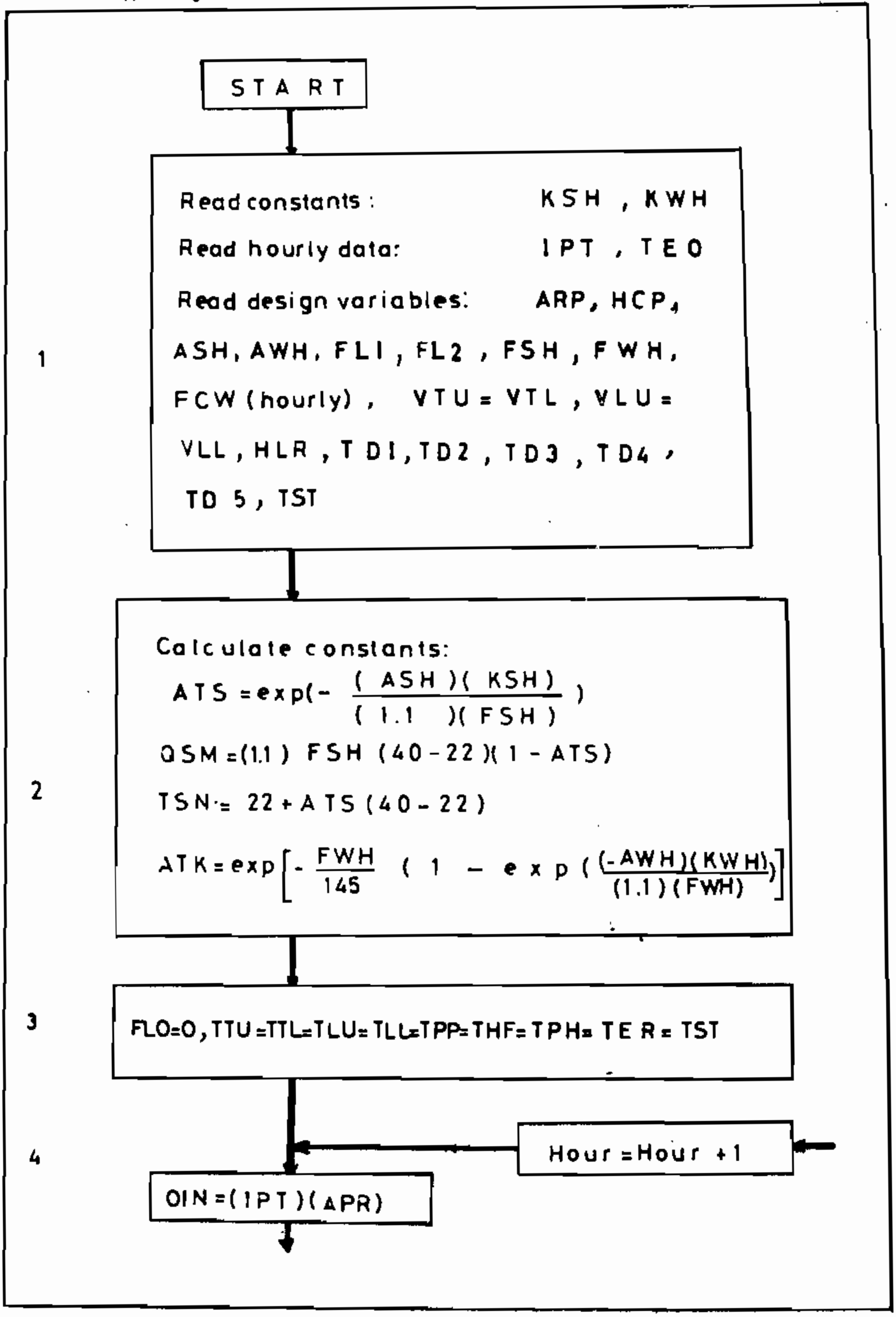
ATK attenuation coeff. in water heating coil

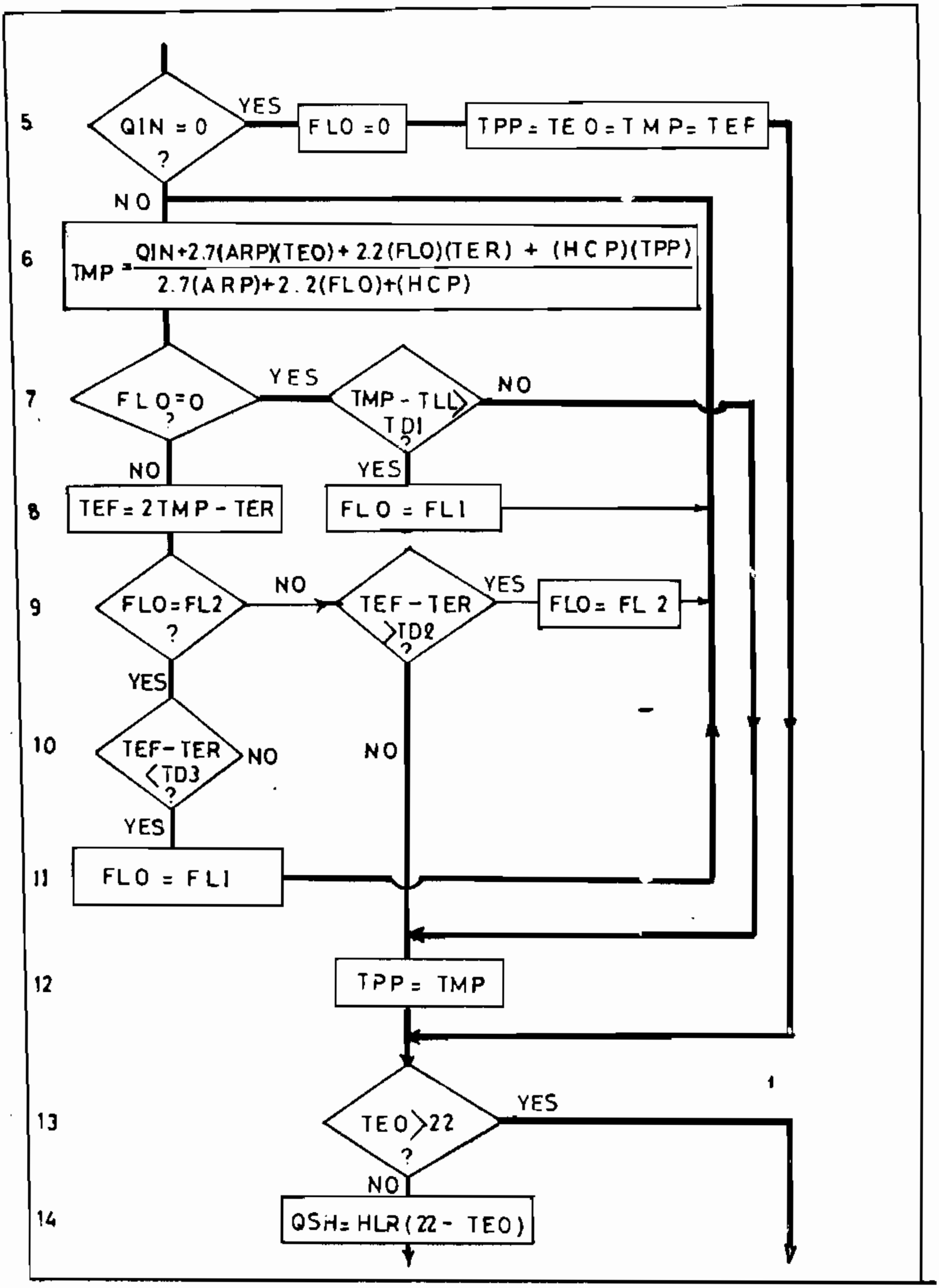
SOC solar contribution factor.

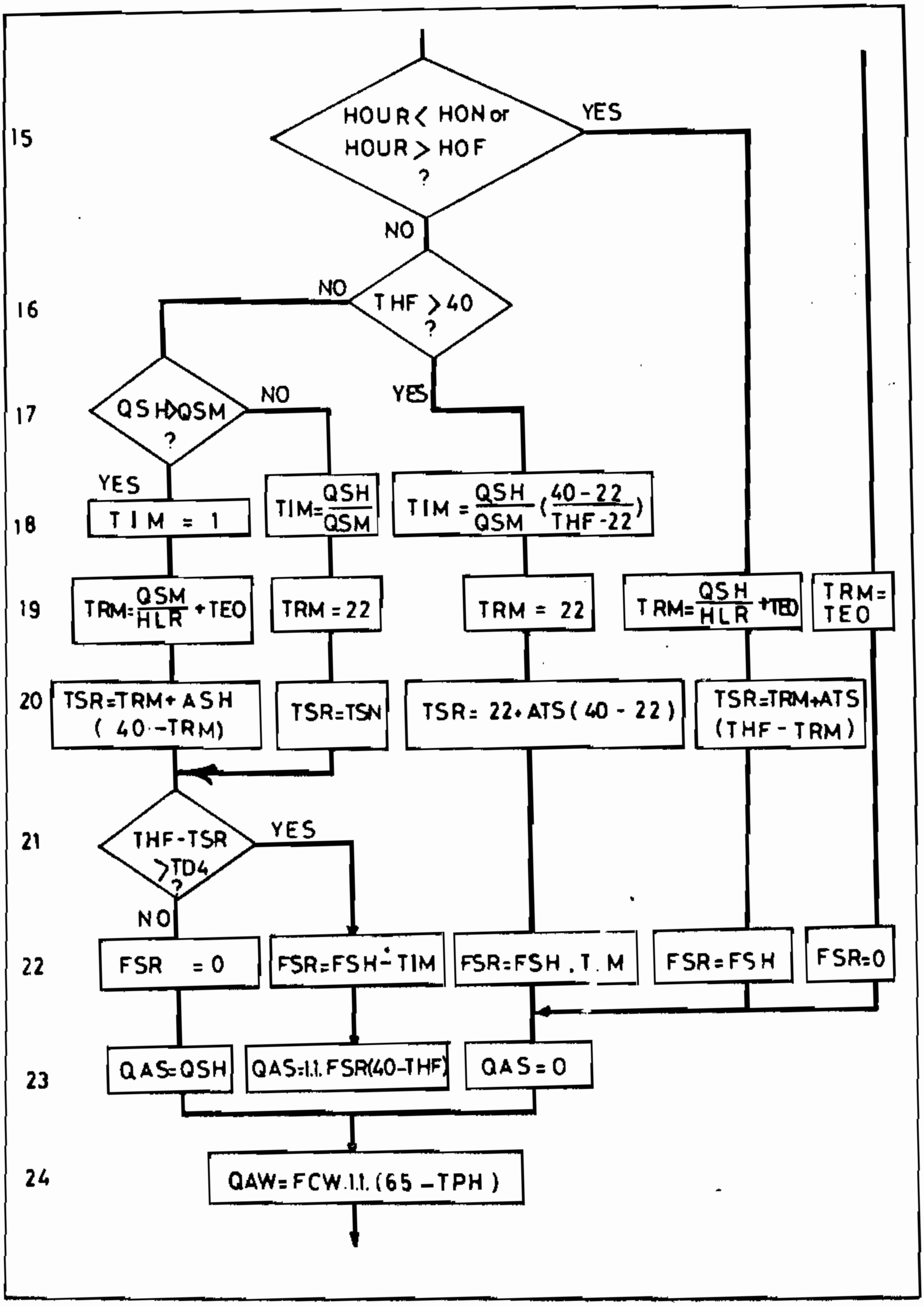
(C) computer flow chart

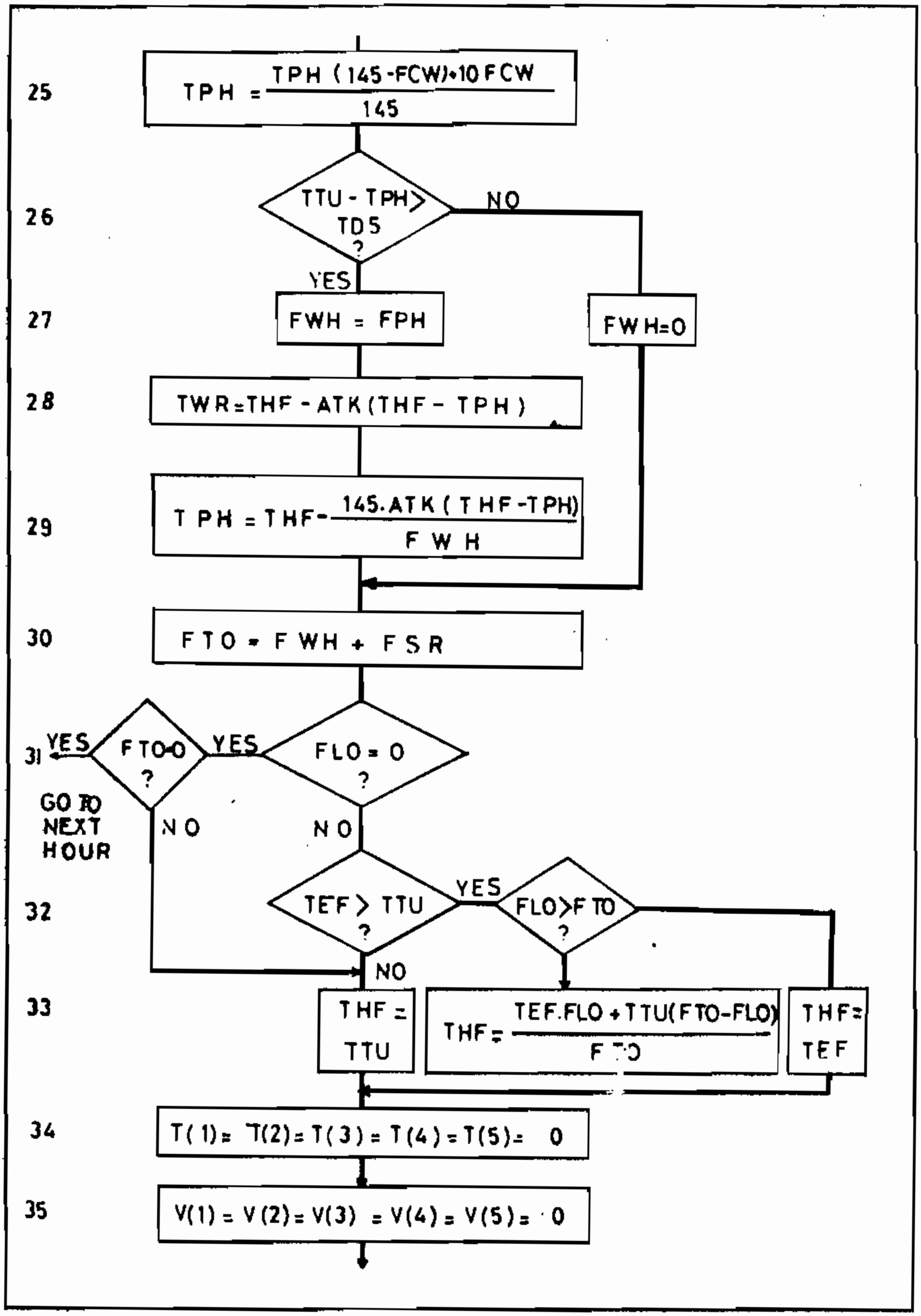
The program simulates the behaviour of our system. Its base data are the hourly IPT values, simulataneous radiation and temperature values for every day of an actual year. Settings of thermostats and controls are put in as "design variables", together with the collector area, the specific heat loss rate of the building, the volume of storage and an assumed hot water consumption pattern. The net heat gain of the collector is calculated, allowing for heat losses from the collector. This amount enters the storage. The space and water heating demand is established as well as the flow and return temperatures. The simulation of stratification in the storage tank is presented in the central part of the program.

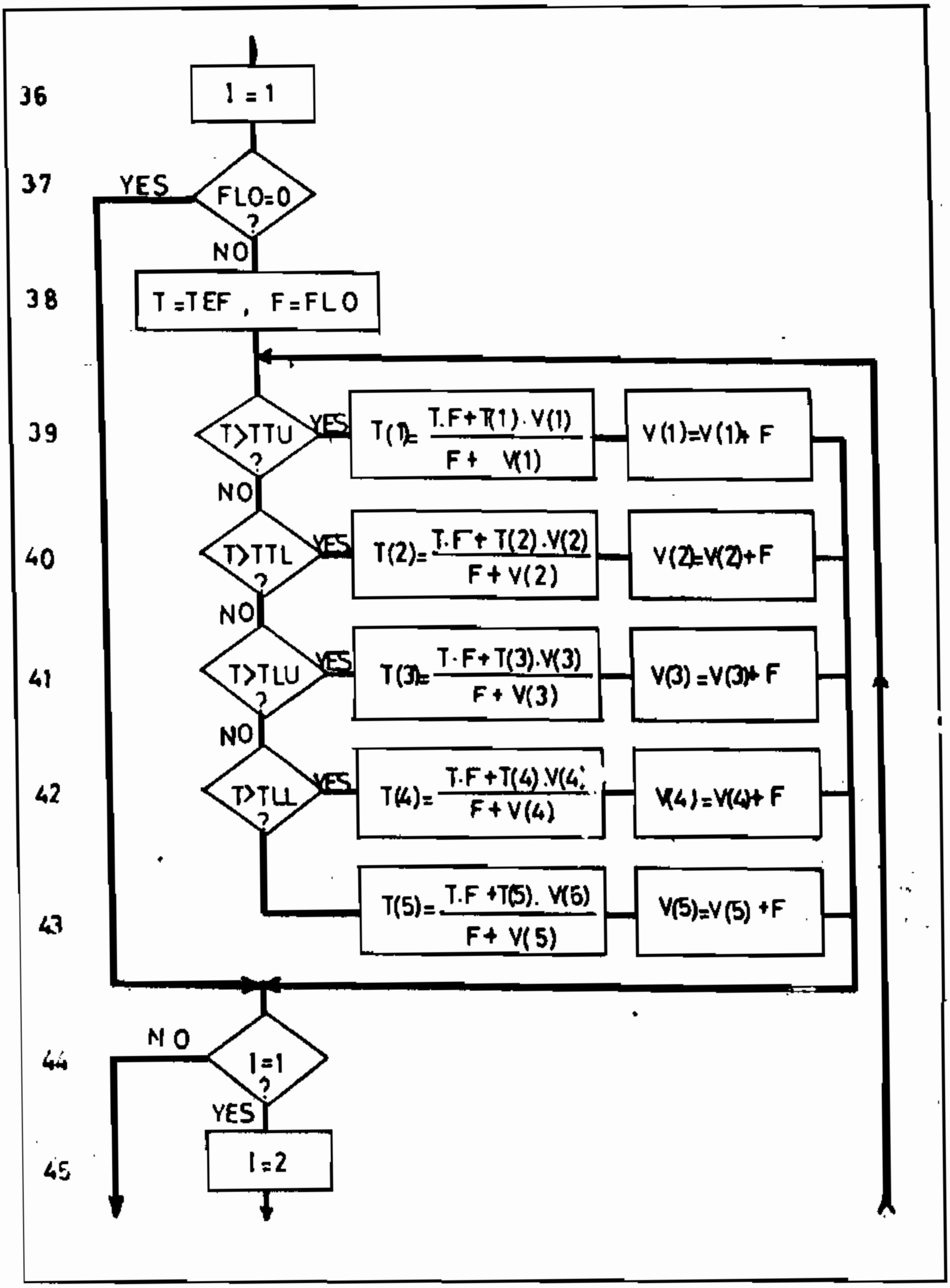
In the following section, the computer flow chart is introduced:

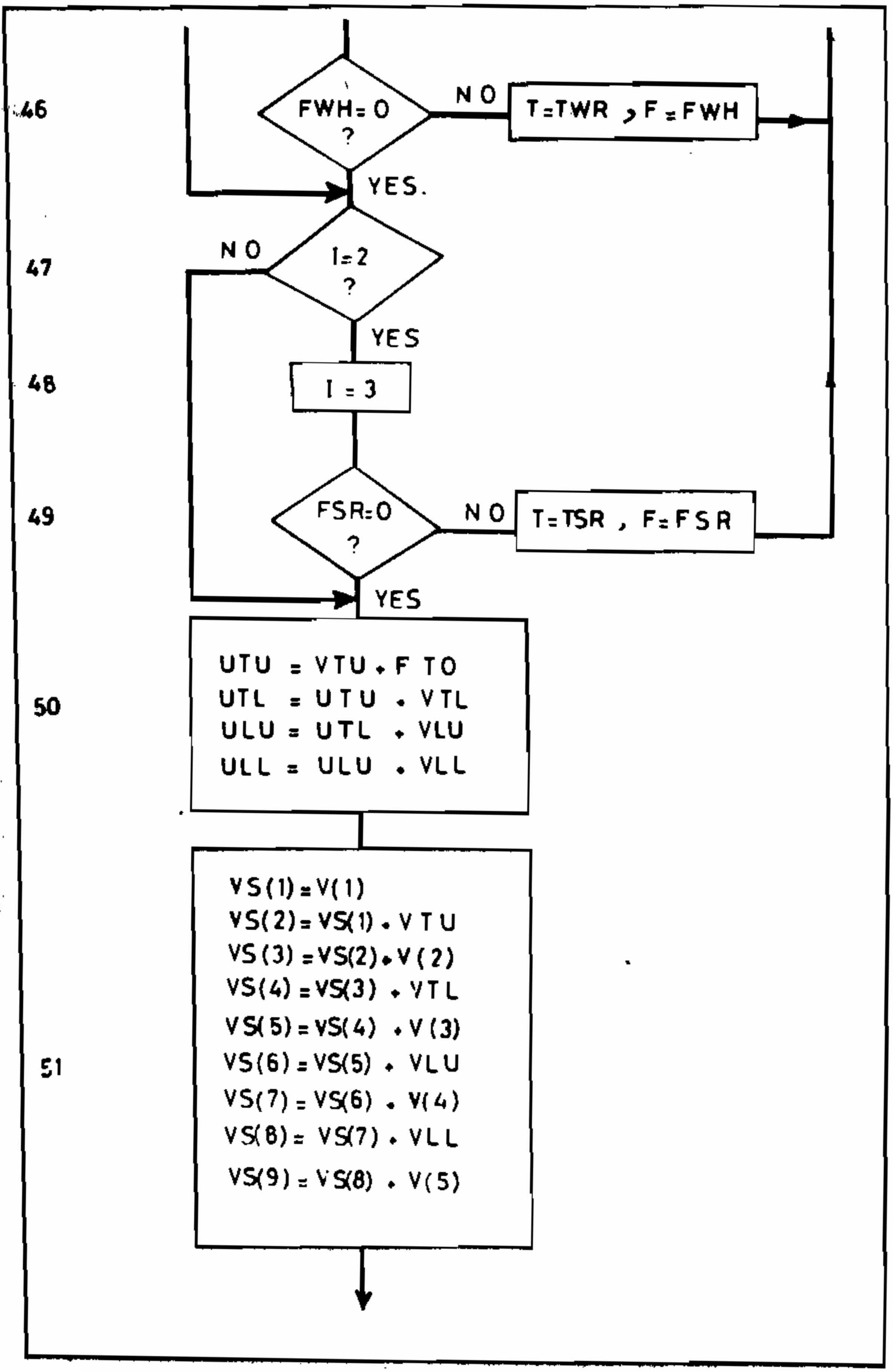

START
Read constants : KSH , KWH
Read hourly data: IPT , TEO
Read design variables: ARP, HCP,
1
ASH, AWH, FL1 , FL2 , FSH , FWH,
FCW (hourly) , VTU = VTL , VLU =
VLL , HLR , TD1, TD2 , TD3 , TD4 ,
TD 5 , TST
Calculate constants:
ATS = exp(- (ASH)(KSH) / (1.1)(FSH))
QSM = (1.1) FSH (40 - 22)(1 - ATS)
2
TSN = 22 + ATS (40 - 22)
ATK = exp [- FWH / 145 (1 - exp ((-AWH)(KWH) / (1.1)(FWH))]
3
FLO=O, TTU=TTL=TLU=TLL=TPP=THF=TPH=TER=TST
4
Hour = Hour + 1
OIN = (IPT)(APR)

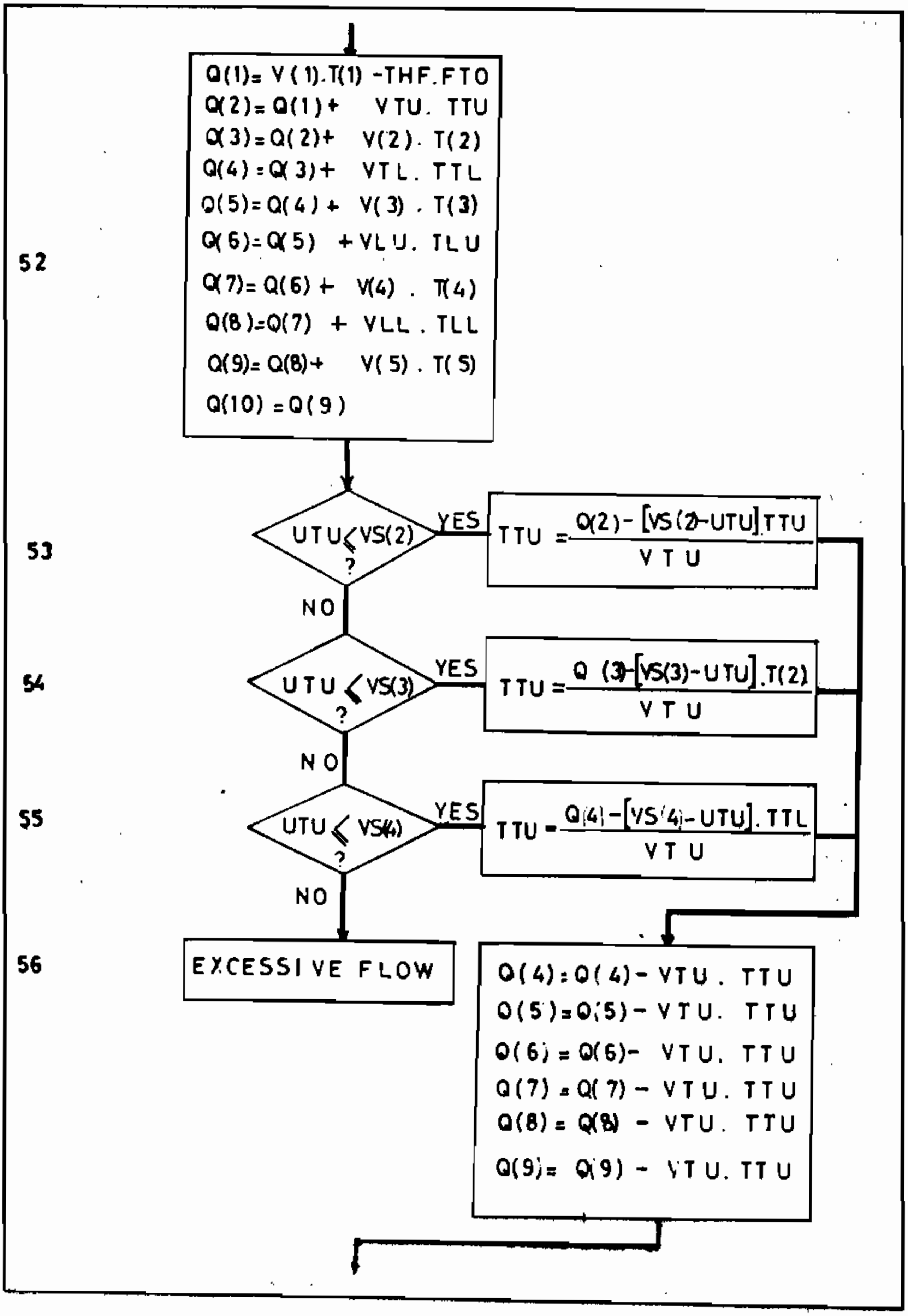

5
QIN = 0 ?
YES
FLO = 0
TPP = TEO = TMP = TEF
NO
6
TMP = QIN+2.7(ARP)(TEO) + 2.2(FLO)(TER) + (HCP)(TPP) / 2.7(ARP)+2.2(FLO)+(HCP)
7
FLO=0 ?
YES
TMP - TLL > TDI ?
NO
NO
YES
8
TEF = 2TMP - TER
FLO = FL1
9
FLO=FL2 ?
NO
TEF - TER > TD2 ?
YES
FLO = FL 2
YES
10
TEF - TER < TD3 ?
NO
NO
YES
11
FLO = FL1
12
TPP = TMP
13
TEO > 22 ?
YES
NO
14
QSH = HLR (22 - TEO)


15
HOUR < HON or HOUR > HOF ?
YES
NO
16
THF > 40 ?
NO
17
QSH > QSM ?
NO
YES
YES
18
TIM = 1
TIM = QSH/QSM
TIM = QSH/QSM (40-22/THF-22)
19
TRM = QSM/HLR + TEO
TRM = 22
TRM = 22
TRM = QSH/HLR + TEO
TRM = TEO
20
TSR = TRM + ASH (40 - TRM)
TSR = TSN
TSR = 22 + ATS (40 - 22)
TSR = TRM + ATS (THF - TRM)
21
THF - TSR > TD4 ?
YES
NO
22
FSR = 0
FSR = FSH - TIM
FSR = FSH . T . M
FSR = FSH
FSR = 0
23
QAS = QSH
QAS = 1.1. FSR (40 - THF)
QAS = 0
24
QAW = FCW.1.1. (65 - TPH)

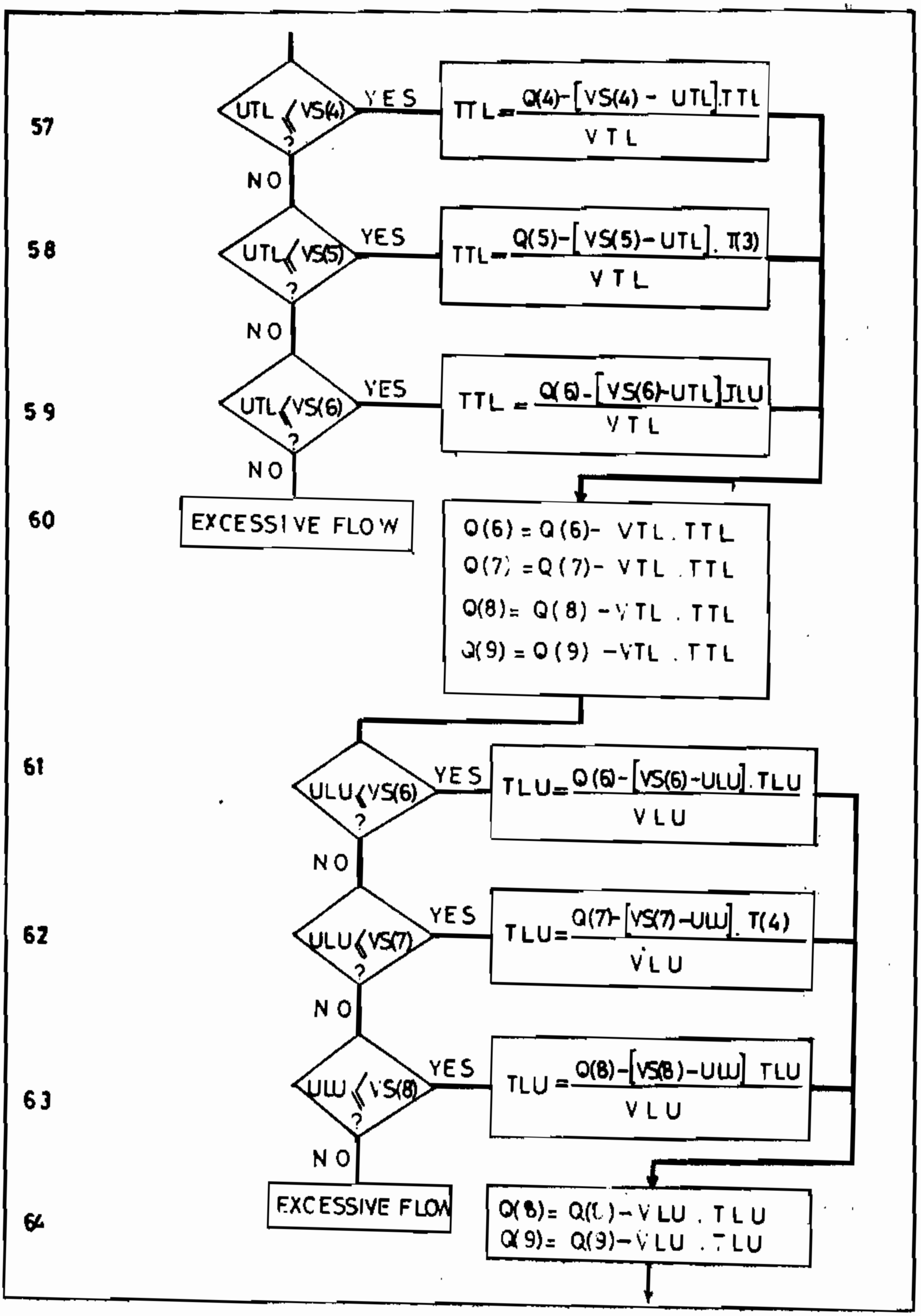

25
TPH = TPH (145-FCW)+10 FCW / 145
26
TTU - TPH > TD5 ?
NO
YES
27
FWH = FPH
FWH=0
28
TWR=THF - ATK (THF - TPH)
29
TPH = THF - 145.ATK (THF-TPH) / FWH
30
FTO = FWH + FSR
31
YES
FTO=O ?
YES
FLO = O ?
GO TO NEXT HOUR
NO
NO
32
TEF > TTU ?
YES
FLO>FTO ?
NO
33
THF = TTU
THF = TEF.FLO + TTU(FTO-FLO) / FTO
THF= TEF
34
T(1)= T(2)= T(3) = T(4) = T(5)= 0
35
V(1) = V(2)= V(3) = V(4) = V(5)= 0

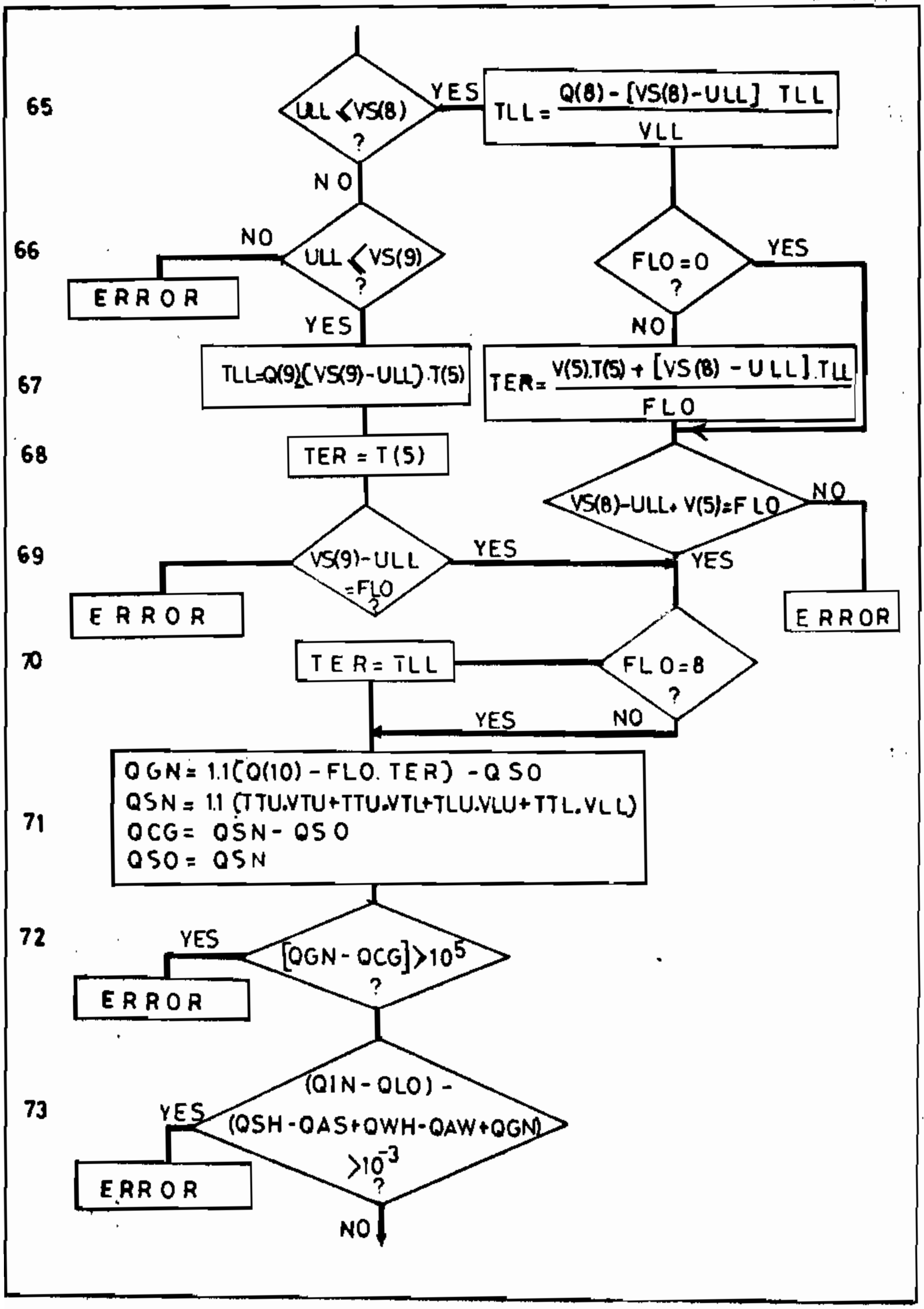

36
I = 1
37
YES
FLO=0 ?
NO
38
T=TEF , F=FLO
39
T>TTU ?
YES
T(1)= T.F+T(1).V(1) / F+V(1)
V(1)=V(1)+F
NO
40
T>TTL ?
YES
T(2)= T.F+T(2).V(2) / F+V(2)
V(2)=V(2)+F
NO
41
T>TLU ?
YES
T(3)= T.F+T(3).V(3) / F+V(3)
V(3)=V(3)+F
NO
42
T>TLL ?
YES
T(4)= T.F+T(4).V(4) / F+V(4)
V(4)=V(4)+F
43
T(5)= T.F+T(5).V(5) / F+V(5)
V(5)=V(5)+F
44
NO
I=1 ?
YES
45
I=2


46
FWH=0 ?
NO
T=TWR , F=FWH
YES.
47
NO
I=2 ?
YES
48
I = 3
49
FSR=0 ?
NO
T=TSR , F=FSR
YES
50
UTU = VTU + F TO
UTL = UTU . VTL
ULU = UTL . VLU
ULL = ULU . VLL
51
VS(1) = V(1)
VS(2) = VS(1) . V T U
VS(3) = VS(2) . V (2)
VS(4) = VS(3) . V T L
VS(5) = VS(4) . V (3)
VS(6) = VS(5) . V L U
VS(7) = VS(6) . V(4)
VS(8) = VS(7) . V L L
VS(9) = VS(8) . V(5)



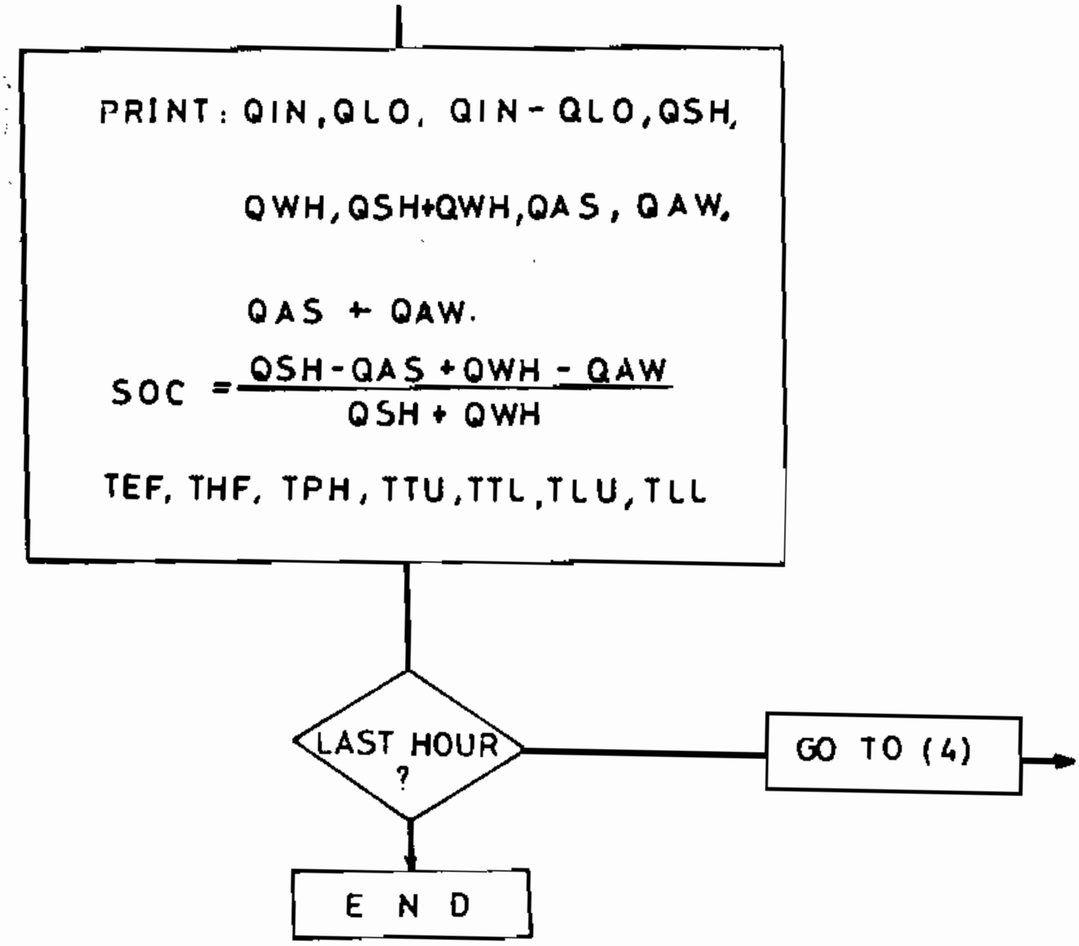
52
Q(1)= V(1).T(1) -THF.FTO
Q(2)= Q(1)+ VTU. TTU
Q(3)= Q(2)+ V(2). T(2)
Q(4)= Q(3)+ VTL. TTL
Q(5)= Q(4)+ V(3). T(3)
Q(6)= Q(5) +VLU. TLU
Q(7)= Q(6)+ V(4). T(4)
Q(8)= Q(7) + VLL. TLL
Q(9)= Q(8)+ V(5). T(5)
Q(10) = Q(9)
53
UTU ≤ VS(2) ?
YES
TTU = (Q(2) - [VS(2)-UTU].TTU) / VTU
NO
54
UTU ≤ VS(3) ?
YES
TTU = (Q (3)-[VS(3)-UTU].T(2)) / VTU
NO
55
UTU ≤ VS(4) ?
YES
TTU = (Q(4)-[VS(4)-UTU].TTL) / VTU
NO
56
EXCESSIVE FLOW
Q(4)= Q(4)- VTU . TTU
Q(5)= Q(5)- VTU. TTU
Q(6)= Q(6)- VTU. TTU
Q(7)= Q(7) - VTU. TTU
Q(8)= Q(8) - VTU. TTU
Q(9)= Q(9) - VTU. TTU


57
UTL ≤ VS(4) ?
YES
TTL = [Q(4) - [VS(4) - UTL].TTL] / VTL
NO
58
UTL ≤ VS(5) ?
YES
TTL = [Q(5) - [VS(5) - UTL]. T(3)] / VTL
NO
59
UTL ≤ VS(6) ?
YES
TTL = [Q(6) - [VS(6) - UTL].TLU] / VTL
NO
60
EXCESSIVE FLOW
Q(6) = Q(6) - VTL . TTL
Q(7) = Q(7) - VTL . TTL
Q(8) = Q(8) - VTL . TTL
Q(9) = Q(9) - VTL . TTL
61
ULU ≤ VS(6) ?
YES
TLU = [Q(6) - [VS(6) - ULU]. TLU] / VLU
NO
62
ULU ≤ VS(7) ?
YES
TLU = [Q(7) - [VS(7) - ULU]. T(4)] / VLU
NO
63
ULU ≤ VS(8) ?
YES
TLU = [Q(8) - [VS(8) - ULU] TLU] / VLU
NO
64
EXCESSIVE FLOW
Q(8) = Q(8) - VLU . TLU
Q(9) = Q(9) - VLU . TLU


65
ULL < VS(8) ?
YES
TLL = Q(8) - [VS(8)-ULL] TLL / VLL
NO
66
NO
ULL < VS(9) ?
ERROR
YES
FLO = 0 ?
YES
NO
67
TLL=Q(9)(VS(9)-ULL).T(5)
TER= V(5).T(5) + [VS(8) - ULL].TLL / FLO
68
TER = T(5)
VS(8)-ULL+ V(5)=FLO
NO
69
VS(9)-ULL =FLO ?
YES
YES
ERROR
ERROR
70
TER= TLL
FLO=8 ?
YES
NO
71
QGN= 1.1(Q(10) - FLO. TER) -QSO
QSN = 1.1 (TTU.VTU+TTU.VTL+TLU.VLU+TTL.VLL)
QCG= QSN- QSO
QSO= QSN
72
YES
[QGN - QCG]>10^5 ?
ERROR
73
YES
(QIN-QLO) - (QSH-QAS+QWH-QAW+QGN) >10^-3 ?
ERROR
NO

74

75


PRINT: QIN, QLO, QIN-QLO, QSH,
QWH, QSH+QWH, QAS, QAW,
QAS + QAW.
SOC = (QSH-QAS+QWH-QAW) / (QSH+QWH)
TEF, THF, TPH, TTU, TTL, TLU, TLL
LAST HOUR ?
GO TO (4)
E N D

VII. CONCLUSIONS

Precise and logic simulation of water and space heating system under consideration is presented in this article. This system is a complex type comprising ..e possible means of additional and auxiliary heating. Complex and hour-by-hour data of intensity of radiation perunit area, and the outdoor air temperature are recorded. Also settings of thermostats and controls are put in as "design variables", together with the collector area, the specific heat loss rate of the building, the volume of storage and an assumed hot water consumption pattern. Allowing for heat losses from the collector, the net heat gain of it is computed. The flow and return temperatures in addition to the space and water heating demand are established. The simulation of stratification in the storage tank is presented.

VIII. REFERENCES

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