The actual benefits of thermally stratified storage in a small and a medium size solar system

THE ACTUAL BENEFITS OF THERMALLY STRATIFIED STORAGE IN A SMALL AND A MEDIUM SIZE SOLAR SYSTEM.

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INTRODUCTION

Although the advantages of the thermally stratified storage have been noted early in the development of solar heating, until recently only limited progress had been made in the actual realisation of this type of storage, particularly for liquid systems. A thermally stratified pebble bed storage can relatively easily be realised for air systems. However, it should be noted that the benefits claimed for this type of stratified storage in |1| are not only due to the stratification, but also to the low temperature of the return air in air heating systems. This distinction should also be kept in mind when radiator or floor heating systems are being considered. The slow introduction of the thermally stratified liquid storage has been partly due to a common misinterpretation of the Hottel-Whillier equation |2| for the efficiency of solar collectors, leading to the application of unduly large mass flow rates, and partly to some unsolved design problems related to the required suppression of mixing phenomena in the storage tank. The H.W. equation reads (for $F' = 1$)

$$\eta_c = F_R\{\alpha\tau - U(T_b - T_o)/F_R E\} \qquad (1)$$

(F_R = heat removal factor, $\alpha\tau$ = combined transmission absorption factor, U = total heat loss coefficient of collector, E = insolation intensity perpendicular to collector, F' = plate efficiency factor).

The adjoint equation for F_R is:

$$F_R = |1 - \exp(-U/M_c c)|M_c c/U \qquad (2)$$

(M_c = mass flow rate per unit collector area, c = specific heat fluid).

Equation (2) indicates that a large M_c leads to a high F_R (i.e. close to 1) and (1) indicates that a high F_R leads to a high collector efficiency, thereby also, but wrongly suggesting that a large M_c leads to a high energy gain. The main causes for this fallacy are that the system behaviour is overlooked (particularly the collector-storage interaction and the benefits of stratification) and that sight is lost of the temperature level required for an efficient utilisation of the heat. It is evident that heat at a low temperature is of limited usefulness for space heating and hot water supply. Such heat requires too large volumes for storage, too large heat exchange areas and/or mass flows for its distribution, too large pumping powers and, in many cases, extra auxiliary heating. As shown below the consequence of the "large mass flow approach" is the replacement of heat at a high temperature by heat at a lower temperature. In its turn this loss of quality (from the viewpoint of thermodynamics aequivalent to a loss of exergy) leads to a decrease in the performance of the system. This will be shown for three cases: a simplified simulation system, in order to clarify the physical interactions involved; the system of the Solar House of the Eindhoven University of Technology, chosen to demonstrate the practicability of stratification and the system of the Laboratory of the Food Inspection Department at Enschede, The Netherlands, to show o.a. the effect of the pumping power on the optimal flow rate.

An essential component of all three systems is the so-called floating inlet for the storage. The floating inlet is a wide, thin-walled, flexible plastic hose connected to the inlet stud |3|. Laboratory experiments at the E.U.T. and several years of practical experience in the two last mentioned systems have proved that the floating inlet is a cheap, reliable and very effective device for delivering the hot water from the collector at exactly the level of equal temperature in the storage tank. Thus preventing any mixing of hot and cold water and promoting stratification. By limiting the flow velocity in the inlet to 0.1 ms^{-1} all tendencies to swinging of the hose can be eliminated.

The working fluid in all three systems is water; for the simplified simulation system water was chosen because it is the most widely used working fluid nowadays; moreover the working fluid as such does not essentially change the relevant physical interactions.

In the first two systems also a so-called selective outlet system is used, i.e. a system in which the level at which water is extracted from the storage is continuously adapted to the momentaneous heat demand of the distribution system. A selective outlet system offers slightly better system performance and an easy control.

THE SIMPLIFIED SYSTEM

The simplified system on which the low optimal flow rate will be demonstrated first is depicted in fig. 1.

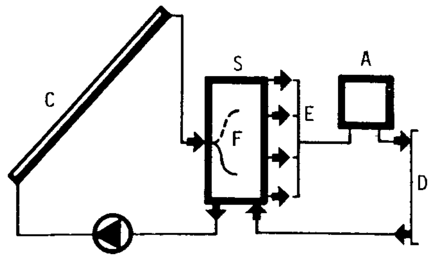

Fig. 1. Simplified system for calculation of optimal flow. C = collectors, S = storage (water), A = auxiliary heater, D = distribution system, F = floating inlet, E = selective extraction.

The flows and the temperature distribution in the storage tank are shown in figure 2.

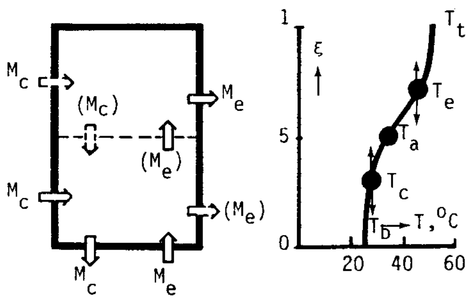

Fig. 2. Flows and temperature distribution in the thermally stratified storage; T = temperature; b = bottom storage; c = collector (top), a = average storage, e = extraction, t = top storage; M = mass flow; ξ = dimensionless heigth in storage.

The temperature is assumed to increase harmonically from the bottom of the tank to the top. In reality the temperature gradients in a stratified storage are often considerably larger than those corresponding to an harmonic temperature increase |3|. In order not to overestimate the benefits of stratification the simple harmonic approximation was preferred for the simulation.
Regarding the flows in the storage tank it should be noted that the mass flow through the collector is associated with a slow downward displacement of the water in the region between the (floating) inlet point and the bottom of the tank. The feeding of hot water from the storage to the distribution system similarly evokes a slow upward displacement in the region between the bottom of the tank and the (selected) extraction point. Depending on the position of the inlet (extraction) point these displacements will or will not lead to convective transport of heat from the upper (lower) half to the lower (upper) half of the tank. The associated heat flows enter into the heat balances of the two parts of the tank.
In order to simplify the calculations all secondary effects will further be neglected. Taken as secondary effects are the heat losses of the pipes and the storage, the non-linearities in the performance of the collector, the conduction of heat and any mixing in the storage (see |3|), the thermal capacity of the collector, the absorber plate efficiency factor, the time-lags in the control of the system etc. In the same framework of simplification and concentration on the essentials the intensity of the light radiation perpendicular to the collector is assumed to vary in harmonic way, between 6 a.m. and 6 p.m. (solar time), with the amplitude E_o.
The remaining simulation model permits a rapid step to step calculation of the behaviour and the main characteristics of a solar heating system with a thermally stratified storage.

GOVERNING EQUATIONS

Heat delivered to storage by collector

During a time step Δt (t in s., starting at solar midnight) the heat delivered by the collector per collector area ΔQ_c is given by:

$$Q_c = \{\alpha\tau\ E_o \sin(\frac{2\pi(t-6.3600)}{24.3600}) + U(\frac{T_c+T_b}{2} - T_o)\}\Delta t \qquad (3)$$

under the condition that the form between braces is positive (U = total heat loss coefficient of collector; T_o = ambient temperature; further see fig. 2).
The heat delivered to the upper half of the storage, ΔQ_u, follows from the equations:

$$\Delta Q_u = Q(T_c - T_a)\ ,\ T_c > T_a \qquad (4a)$$

$$\Delta Q_u = 0 \qquad T_c < T_a \qquad (4b)$$

and the amount of heat delivered to the lower part, ΔQ_l, follows from

$$\Delta Q_l = \Delta Q - \Delta Q_u \qquad (5)$$

The temperature of the water at the top of the collector, T_c, is directly related to the heat delivered, for:

$$\Delta Q = (T_c - T_b) c\, M_c\, \Delta t \qquad (6)$$

Heat extracted from storage for distribution

If the heating power equals δ the heat to be delivered by the storage tank and, as far as necessary, by the auxiliary heater during the time step Δt is given by:

$$\Delta D = \delta\, \Delta t \qquad (7)$$

When $T_d \leq T_t$ (see fig. 2) all the heat can be extracted from the storage. However a distinction has to be made between the cases $T_d > T_a$ and $T_d < T_a$. For the heat extracted from the upper half of the tank, ΔD_u, the heat balance of the upper half together with the assumption that the return temperature of the water from the distribution system equals 20°C leads to the equations:

$$\Delta D_u = \Delta D\left(\frac{T_d - T_a}{T_d - 20}\right), \quad T_a < T_d < T_t \qquad (8a)$$

$$\Delta D_u = 0 \qquad , \quad T_d < T_a < T_t \qquad (8b)$$

The heat extracted from the lower half, ΔD_1, is given in both cases by:

$$\Delta D_1 = \Delta D - \Delta D_u, \quad T_d < T_t \qquad (9)$$

When $T_d > T_t$ only part of the required heat can be extracted from the storage and the equations for determining ΔD_u and ΔD_1 are:

$$\Delta D_u = \Delta D\, \frac{T_t - T_a}{T_d - 20} \quad , \quad T_d > T_t \qquad (10)$$

and

$$\Delta D_1 = \Delta D\, \frac{T_a - 20}{T_d - 20} \quad , \quad T_d > T_t \qquad (11)$$

In the derivation of (8) through (11) it is assumed that the inlet temperature of the distribution system, T_d, is equal to the required heating temperature under all circumstances. This implies auxiliary heating when $T_d > T_t$ and backmixing of the return water into the water extracted from the storage when $T_d < T_1$ (the mass flow from the storage being reduced in the last case). Starting from the assumed harmonic temperature increase in the storage tank it can easily be shown that the heat contents of the lower and upper half of the tank, W_1 and W_u respectively, are given by:

$$W_1 = \tfrac{1}{2} c\, U\left\{\frac{T_t + T_b}{2} - \frac{(T_t - T_b)}{\pi}\right\} \qquad (12)$$

and

$$W_u = \tfrac{1}{2}\, cV \left\{\frac{T_t + T_b}{2} + \frac{T_t - T_b}{\pi}\right\} \qquad (13)$$

(V denoting the mass content of the storage per unit area of collector).

During a time step Δt (12) and (13) hold for the increments of the involved quantities. The increments of the heat contents of the lower and upper half of the tank, ΔW_1 and ΔW_u respectively, follow from:

$$\Delta W_1 = \Delta Q_1 - \Delta D_1 \qquad (14)$$

and

$$\Delta W_u = \Delta Q_u - \Delta D_u \qquad (15)$$

The equations (1) through (15) permit a straightforward step by step calculation of the behaviour of the simulation system for various values of the main parameters T_d and M_c. It makes little sense to vary $\tau\alpha$ as this parameter is almost constant in practice. Varying U, V and E_o might additionally be considered but in practice U and V are rather firmly fixed on economic grounds, and for E_o a value corresponding to clear sky conditions is to be preferred, because most of the solar heat is gained in that situation.

RESULTS OF CALCULATIONS

For the calculations E_o was fixed at 800 Wm^{-2}, T_o at °C, $\alpha\tau$ at 0.80, U at 5 $Wm^{-2}\ {}^oC^{-1}$ and V at 80 kg per m^2 collector area. For the parameter T_d two values where chosen, 40°C and 60°C. The first value may represent the required distribution temperature for a well designed air or floor heating system, the second value is appropriate for hot water supply. The heat demand was taken constant and related to the average (24 hrs) utilizable solar radiation, $\bar{E}$. The latter was calculated with a critical (threshold) radiation intensity on the absorber plate of 100 Wm^{-2} (= 5. (distribution return temperature - ambient temperature)). With $(\alpha\tau)\ E_o$ = 640 Wm^{-2} the value of $\bar{E}$ was found to amount to 156,2 Wm^{-2}. For the ratio, φ, between $\bar{E}$ and the heat demand per m^2 collector area the values 0,6 and 1,0 were used in the calculations. To study the effect of stratification as such some additional calculations were made in which the stratification was suppressed. All calculations were made for steady conditions, i.e. for strictly periodical phenomena from day to day.
The calculated heat gains as a function of the mass flow rate are shown in fig. 3 and 4. From fig. 3 it appears that the optimal flow rate is as low as 11 kg m^{-2} hr for a high required T_d and a high radiation/demand ratio, φ, and gradually shifts to a value of 25 $kgm^{-2}hr$ for low T_d and low φ. Fig. 4 illustrates that the unusually low optimal flow rates are associated with and entirely

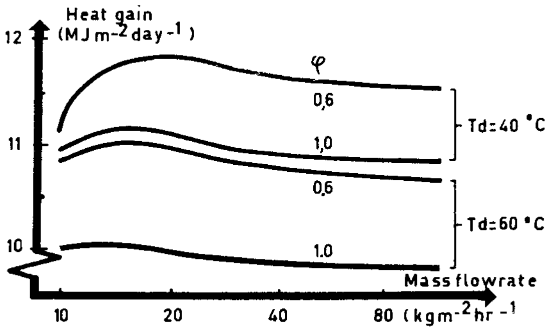

Fig. 3. Energy gain (excl. pumping power) as a function of mass flow rate for two demand/ solar radiation ratios (φ) and two distribution temperatures (T_d).

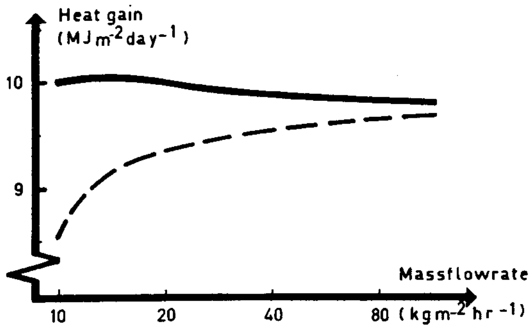

Fig. 4. Comparison of heat gain for stratified (—) and uniform (---) storage, as a function of mass flow rate. Demand/solar radiation ratio = 1, distribution temperature = 600°C.

due to the thermal stratification. It also appears from fig. 4 that the benefits of stratification are small when the mass flow rate is large. The reason simply being that the high fluid turnover suppresses virtually all stratification in the tank. Consequently it makes little,if any sense to consider the application of thermal stratification in the framework of a large mass flow approach.

REAL SYSTEMS AND OTHER CLIMATES

It may be argued that the conclusions reached with the aid of the simulation model are of limited significance because of toc many simplifications or because of the rather specific, clear sky type of solar radiation used in the model. However, the results of the simulation are substantiated by calculations and observations made (earlier) on two real solar heating systems under the rather unfavorable Dutch climatic conditions. The latter are characterised by a yearly global radiation of about 1000 kWh/m^2, of which some 65% is diffuse light, and a percentage of clear sunshine varying between 15% in winter and 45% in summer. The real systems that will be considered here are the space heating and hot water systems of the Solar House of the Eindhoven University of Technology and of the Laboratory of the Food Inspection Department at Enschede. The first system has a net collector area of 51 m^2 and a thermally stratified storage with a volume of 4.1 m^3. The collector area of the second system is 400 m^2 and the volume of the storage amounts to 31 m^3. In the last system not only the hot water from the collectors but also the water returning from the distribution system is fed into the storage tank via a floating inlet. A special feature in the Eindhoven installation is the pre-heating of the fresh ventilating air by passing this air along the bottom of the storage tank; the heat exchange reduces the temperature of the water in this part of the tank to values below 10°C in winter. The net effect on the output of the installation has been calculated at about +5%. Another rather exceptional feature is the preservation of a high temperature (60°C) in the top part of the tank by means of the auxiliary heater. No significant penetration of this heat to lower parts of the tank has been observed in practice. More details on the installations can be found in |4| and |5|.

Starting from hourly weather data over the period 1961–1970 calculations have been made concerning the benefits of the nearly perfect stratification in both systems and on the optimal flow rate. Compared to a perfectly mixed storage tank the heat gain per m^2 collector area at constant mass flow rate was found to increase from 204 kWh to 238 kWh per year in the Enschede installation (+17%) and from 192 to 233 kWh in the Eindhoven system (+16%). For the Enschede installation the influence of the flow rate on the heat gain is given in fig. 5. The optimal flow rate

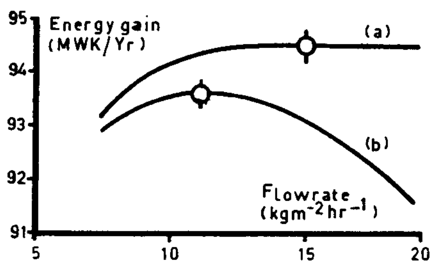

Fig. 5. Energy gain of the Enschede solar installation versus mass flow rate. a = thermal energy only, b = corrected for pumping power (reduced to primary energy).

appears to be 15 $kgm^{-2}hr^{-1}$ when only the thermal heat gain is considered, and 12 $kgm^{-1}hr^{-1}$ after correcting for the pumping power (reduced to primary energy). The large influence of the pumping power is due to the

flatness of the first optimum (the reduced pumping power itself is small, only about 0.75 Wm^{-2} collector area, near the optimum). A similar flat optimum was found for the Eindhoven installation at a collector flow rate of 12 $kgm^{-2}hr^{-1}$.
Regarding the percentages justmentioned it should be noted that at very high flow rates the performance of a system with a uniform storage temperature is only a few present lower than the performance of a system with a stratified storage at low flow rate (see fig. 4). Taking into account the pumping power the last system again shows considerably advantages however.

DISCUSSION, CONCLUSIONS

The good agreement between the results obtained on the basis of detailed calculations and measurements on two operating installations and the results of the simulation model lend strong support to the general statement that a more widespread utilisation of thermally stratified storage in combination with much lower collector flow rates than usual will lead to considerable improvements in the performance of solar space heating and hot water supply systems. As little is known about the extent to which stratification arises "naturally" in existing systems, no definite figure on the possible improvements can be given. However, the experiments in the Laboratory of the E.U.T. have shown that the floating inlet |6| improves the stratification by a factor of 2, compared to "natural" stratification. Therefore, it seems justified to expect an increase in the heat gain of 5 to 10% (half the percentages found above) and/or a very attractive reduction of the pumping power. Further study is required and worth while.

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