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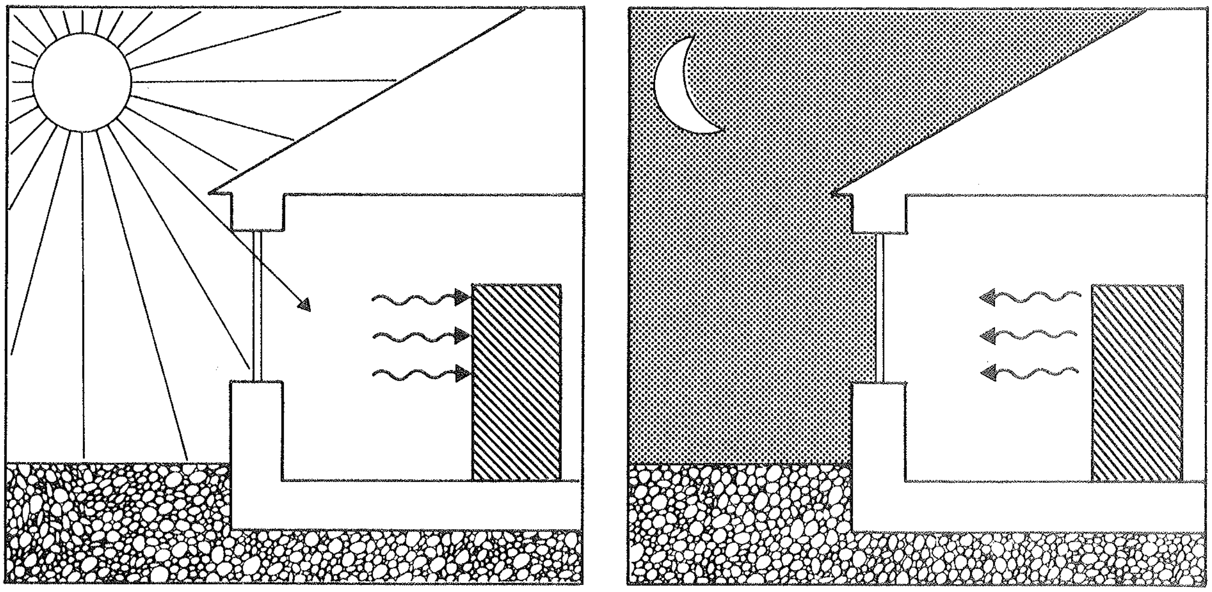
APRIL 1, 1982 - DECEMBER 31, 1983
FINAL REPORT

UNIT FOR LOCAL STORAGE OF SOLAR GAIN



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TECHNICAL UNIVERSITY OF DENMARK
THERMAL INSULATION LABORATORY

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RESUMÉ

Projektets formål har været at udvikle en unit, der er i stand til at lagre en betydelig del af overskudsvarmen fra solindstrålingen i rum med store sydvendte vinduer og ringe varmekapacitet. Den varmeakkumulerende unit placeres "lokalt", d.v.s. inde i det pågældende rum, der i dette projekt tænkes at være opholdsrummet i et eksisterende én-familie-hus. Udover at give en væsentlig energibesparelse vil lagerunit'en sørge for, at de høje rumtemperaturer begrænses i perioderne med stort solindfald, hvormed den termiske komfort forbedres.

For at unitén skal være så kompakt som mulig, er det vigtigt, at lagermediets varmeindhold pr. volumenhed er så stort som muligt indenfor det interval, hvor rumtemperaturen varierer. I projektet er der valgt at benytte en blanding af $CaCl_2$ og vand, der har en høj lagringskapacitet på grund af faseændring (dannelse eller smeltning af saltkrystaller). Det var oprindeligt hensigten at udføre unit'en, så den fungerede helt "passivt", d.v.s. at varmetransporten til eller fra lagermediet udelukkende fandt sted ved naturlig konvektion af rumluft, for dermed at opnå en særlig lav fremstillingspris. Det viste sig imidlertid at være vigtigt med en mere effektiv varmeoverførsel til unitén, og dette opnås ved at forsyne unit'en med en mindre ventilator.

Idéen med den lokale lager unit blev undersøgt ved hjælp af en række forsøg og beregninger. Indledningsvis blev det undersøgt hvilke lagermedier, der kunne komme på tale, og det mest velegnede blev udvalgt på grundlag af småskala forsøg. Derefter blev to forskellige udformninger af unit'en foreslået, og da ingen af dem på forhånd kunne siges at være mest fordelagtig, blev begge lager units opbygget og testet under realistiske driftforhold. En model af varmeovergangen til og fra unit'en blev indpasset i et EDB-program, med hvilket der blev foretaget beregninger af energibesparelserne i afhængighed af forskellige parametre.

På baggrund af resultaterne fra eksperimenterne og beregningerne blev det besluttet at konstruere en tredie unit, der havde en bedre varmeoverføringsevne mellem rumluften og lageret, end det var tilfældet med de to tidligere opbyggede unit's. Lager unit' ens varmeoverføringsevne blev målt i laboratoriet. Endelig blev der foretaget en undersøgelse af, hvorvidt en lokal lager unit vil være økonomisk fordelagtig.

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SUMMARY

The aim of this project has been to develop a unit for storing a considerable amount of the surplus heat due to solar gain in rooms with large windows facing south and with a low heat capacity. The heat accumulating unit is placed "locally", i.e. inside the room in question, which in this case means the living room in a single-family dwelling. Apart from appreciable energy savings, the storage unit will cause a reduction of the high room temperature during periods with large solar gains and thereby improve the thermal comfort.

In order to obtain a compact design of the unit, it is essential to use a storage medium with a large heat content per unit volume within the interval where the room temperature varies. In this project a mixture of $CaCl_2$ and water has been chosen because of the large heat content which is due to that phase change takes place (formation or melting of salt crystals). Originally, it was intended to construct the unit in such a way that it would function completely "passive", i.e. the heat transfer to or from the storage medium would occur by natural convection of the room air. This method would result in a very low manufacturing cost. It became apparent, however, that a more efficient heat transfer to the unit was very important. This was obtained by providing the unit with a small ventilator.

The scheme of the local storage unit was investigated by means of a series of experiments and calculations. Initially it was investigated what storage mediums could be used, and the most appropriate was selected from small scale experiments. Next, two different designs of the unit were suggested and as none of them seemed to be superior to the other, they were both constructed. The two units were installed in a test house and tested under realistic conditions. A model of the heat transfer to and from the unit was prepared for a computer program and

the dependence of the energy savings on different parameters was investigated by calculations. In the light of the results from the experiments and the calculations, it was decided to construct a third storage unit with a better heat transfer coefficient from the room air to the storage than the two units used for the realistic testing. The heat transfer coefficient for this unit was measured at the laboratory. Finally an examination of the economical benefits of a local storage unit was carried out.

1. INTRODUCTION

A considerable part of the houses built in the 1960'ies and early 70'ies are characterized by large window areas that provide large solar gains. As the walls, floor and ceiling usually are constructed from materials without any significant thermal mass, most of the heat due to the transmitted solar radiation will be wasted on days with much sunshine. Furthermore, the increase of the room temperature will give rise to a poor thermal comfort. If there was a possibility for storing some of the heat, the need for auxiliary energy during the following night would be reduced and the thermal comfort would be improved.

In this project it is proposed to provide the necessary amount of storage capacity by means of a "local storage unit". The unit is supposed to be placed in a room which is exposed to overheating. The unit consists of a box containing a heat storage medium and a system that is capable of providing an efficient heat transfer from the room air to the storage medium during periods with excess heat due to the solar radiation, and in the opposite direction during the night. The box is insulated in order to prevent significant heat loss during periods without any heating requirement. The space required for the unit ought to be modest and the aim must therefore be to obtain a unit of a compact design. This implies that the heat content per unit volume of the storage medium in the interval, within which the room temperature is allowed to vary, must be as large as possible. A high heat content can be obtained by using a phase changing material, and for the local storage unit it was decided to use a salt-water mixture consisting of salt hydrate and extra water. The high heat content per unit volume is provided by the heat of fusion of the salt hydrate. The extra water will ensure that a large part of the phase change takes place within the desired interval and at the same time it ensures that the solution remains stable.

During this project the energy savings provided by a local storage unit were investigated by experiments and calculations. The working principle of the unit is shown in fig. 1.1.

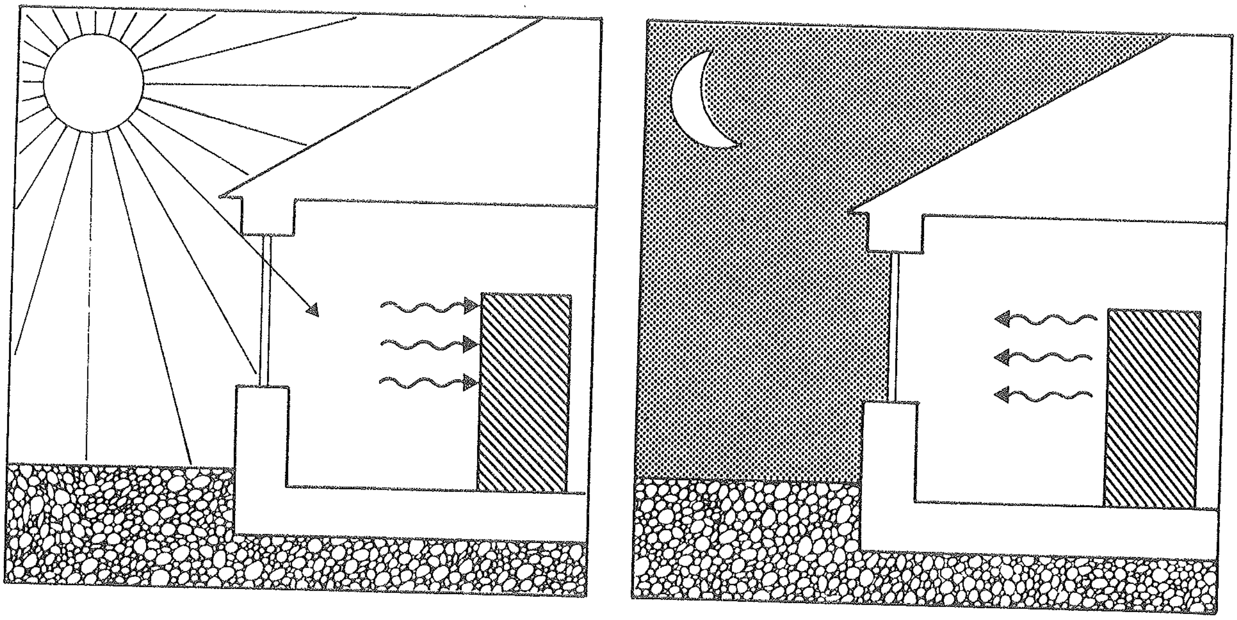


Fig. 1.1 The local storage unit stores the surplus heat collected during the hours with sunshine. The surplus heat can then be used to cover part of the heating requirement during the following period

2. THE HEAT STORAGE MEDIUM

2.1 The use of salt hydrates for storage of heat

A salt-water mixture must fulfil the following requirements in order to be a suitable medium for heat storage in connection to this project.

(a) The heat content in the temperature interval, in which the room temperature varies, must be as large as possible.

(b) Phase separation must be avoided.

(c) Supercooling must be negligible.

(d) The salt hydrate must be inexpensive.

(e) The salt hydrate must be safe to use, i.e. it must not be toxic or inflammable.

In the following an explanation of the requirements (b) and (c) is given.

The phase separation is a problem usually connected to the salt hydrates (i.e. the anhydrous salt and its corresponding crystal water), the reason being the incongruently or partially incongruently melting of the salt hydrate.

Salt hydrates are melting incongruently when the solubility of the salt is too small to dissolve all the anhydrous salt at the melting point. Because of this, three different phases exist at temperatures below the melting point: A solution saturated with anhydrous salt, the salt hydrate crystals and anhydrous salt at the bottom of the container. The anhydrous salt precipitates due to its higher density. At temperatures above the melting point, the mixture consists of a saturated solution and undisolved anhydrous salt at the bottom of the container. When this mixture is cooled down until the temperature reaches the melting point, the salt crystals will begin to form at the dividing line between the solution and the anhydrous salt. The crystals are functioning as a diffusion barrier preventing water from reacting to the anhydrous salt and hereby this salt remains inactive. At a succesive heating of the mixture, the crystals will melt and as the rate of natural mixing is slow, once more all of the anhydrous salt from the crystals will not be dissolved. The amount of sediment increases and consequently the heat storage capacity will decrease. For example, in the case of Glauber salt ($Na_2SO_4 \cdot 10\ H_2O$), the decrease of the storage capacity can be up to 16% for each melting/solidification cycle (see [1], page 94).

Another consequence of the phase separation is that the temperature at which solidification takes place, will be lowered for each cycle. The reason is that solidification can occur only in a saturated solution whereas the solubility decreases with decreasing temperature.

Some types of salt melt partially incongruently. Here the solubility is too small to dissolve all the anhydrous salt at the melting point, but in case the temperature is increased with a few degrees celcius, all the salt will be dissolved. Phase separation will occur due to precipitation of the solid phase formed in the short temperature interval above the melting point.

There are different methods for prevention of the phase separation. A method usually convenient is to add extra water to the system so that all the salt can be dissolved at the melting point. It is also necessary to provide a soft stirring of the mixture preventing sedimentation due to the slow rate of natural mixing. The method is designated "the extra water principle". In [2], [3] and [4] investigations with promising results on this principle are described. However, a disadvantage of adding extra water is that a large part of the anhydrous salt remains dissolved in the water when the storage temperature is at its minimum, with the result that a considerable quantity of the potential heat of fusion is not utilized. In connection to the previously mentioned relation between the temperature and the solubility, the melting (or solidification) will take place continuously at a temperature interval reaching from the minimum storage temperature to a temperature that is dependent on the amount of additional water. The more water added, exceeding what is necessary for dissolving all the anhydrous salt at the melting point, the more this upper limit will be decreased. In fig. 2.2 this is illustrated by plots of the heat content of a system with $CaCl_2$ and water as a function of the temperature for different salt-water mixtures.

Another way of preventing phase separation is microincapsulation of the salt hydrate, i.e. placement in a container less than 1 cm high [2]. With this method it is possible for the water to get in contact with the phase at the bottom by diffusion through the crystals during the solidification process.

As mentioned, it is possible to lower the upper limit of the temperature interval, in which melting takes place, by adding a sufficient amount of extra water to the system. By doing this, the quantity of the heat content in a given temperature interval (for example the interval of the room temperature) below the melting point can be maximized. Another way of increasing the quantity of heat content in the desired temperature interval is to add a suitable second salt to the mixture and here the whole compound is called an "eutectic". The second salt remains inactive, that is it does not melt or crystallize in the considered temperature interval. As far as the effect of adjusting the temperature interval, in which melting takes place, is considered, the second salt has the same effect as though only extra water was added. However, at the same time it is possible to reduce that part of the anhydrous salt which can be dissolved at the minimum temperature of the storage. By this, a comparetively high heat content in the temperature interval of interest is obtained.

Supercooling of the salt solution means that the solidification does not begin at the predicted temperature. In the actual case (the local storage unit) this implies the risk that no heat of crystallization is produced above the minimum acceptable room temperature. The reason for the phenomena of supercooling is that some kind of "granular" surface is necessary if the crystals are to be formed. Normally salt hydrates of a technical quality contain a certain amount of impurities which will provide the requisite "surface" so that the supercooling will not be more than a few degrees. If the pure quality of laboratory salt is used, the supercooling can be of the magnitude 10-20^{o}C, and special measures must be taken. The most common way of preventing supercooling is to add a "nucleating agent", i.e. a chemical with a crystal structure similar to that of the salt crystal. This will induce the solidification to occur at the "correct" temperature.

2.2 Evaluation of the heat storage medium

The requirements mentioned in the beginning of section 2.1 limitates the selection of usable salt hydrates considerably. Especially when considering that a rather large part of the phase-change for the salt hydrate shall occur within the interval of the room temperature. For the mixture of salt hydrate and water, this is not fulfilled if the melting point of the salt hydrate is more than a few degrees celcius above the upper limit of the room temperature.

Three different salt-water mixtures and three different eutectics were chosen for further investigations. The salt hydrates are $Na_2SO_4 \cdot 10\ H_2O$, $Na_2CO_3 \cdot 10\ H_2O$ and $CaCl_2 \cdot 6\ H_2O$. The two first salt hydrates melt incongruently, whereas the third melts partially incongruently, and for all the mixtures phase separation is prevented due to a surplus of water. The composition of the six systems appears from table 2.1 where also some characterizing information are given on the salt-water mixtures. For the eutectics it was not possible to obtain much information in advance. All the salts were of technical quality, and thus supercooling was expected to be insignificant.

Whether the salts would melt and solidificate as predicted had to be tested under conditions that reminded of the supply and withdrawal of heat in the storage unit. At that time it was desired to try to provide the stirring, which is necessary for prevention of phase separation, only by the movements in the melted salt caused by natural convection. The mixtures were placed in the 2½ litre plastic containers planned to be used in one of the storage units. The containers were exposed to several cycles of heating and cooling. For these experiments a special set-up was constructed, consisting of an insulated box in which the containers were placed while they were heated by air. The air was heated by a small electric radiator placed at one end of the box and conducted along the containers by means of a ventilator. After being heated, the mixtures were cooled by placing them outdoors.

From these experiments it appeared that all the systems showed more or less a tendency to supercool, but in the case of the mixture with 46% $CaCl_2$ and 54% H_2O this tendency was less remarkable. Also for this system the dissolution of the salt during the melting process seemed to be very easy as the movements in the liquid caused by natural convection was sufficient to provide the necessary stirring. It was therefore decided to choose the mixture of $CaCl_2$ and H_2O for the further work. The concentration of $CaCl_2$ was then varied and a mixture consisting of 43.6% $CaCl_2$ and 56.4% H_2O turned out to work in the most convincing way, as the time for dissolution was very short and no supercooling was observed at all. Consequently this mixture was selected for the heat storage units constructed for the realistic testing.

Heat storage material	Fraction of salt and water (weight %)	Heat content in the temperature interval 20-26°C (kWh/m^3)	Fraction of solid phase based on weight at 20°C	Melting point of salt hydrate (°C)
Na_2SO_4 H_2O	22 78	27	0.23	33
Na_2CO_3 H_2O	24 76	34	0.32	32
$CaCl_2$ H_2O	46 54	32	0.37	29
Na_2CO_3 K_2CO_3 H_2O	21 16 63	Information not available		32
Na_2SO_4 NaCl H_2O	11 17 72	Information not avaiable		33

Table 2.1 Investigated heat storage materials

2.3 The performance characteristics of $CaCl_2$ with extra water as a heat storage medium

Since the salt hydrate $CaCl_2 \cdot 6H_2O$ was selected as the phase-change material for the storage units, a further discussion on this chemical is given in this section.

The physical data for $CaCl_2 \cdot 6H_2O$ is derived from [5] and they are listed in table 2.2.

In fig. 2.1 is shown part of the phase diagram for a mixture of $CaCl_2$ and H_2O in accordance with [5].

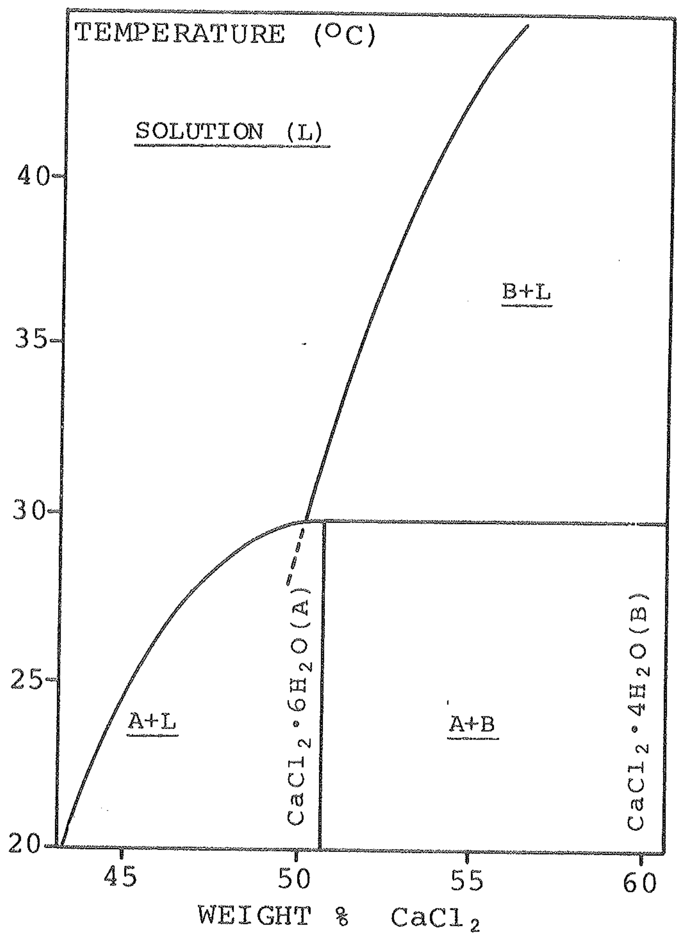


Fig. 2.1 The phase diagram for a system of $CaCl_2$ and water in the range of 43-60 weight % $CaCl_2$ in accordance with [5].

Specific heat of the solid phase	C_{sh} = 1,44 J/(g·K)
Specific heat of the liquid phase	C_{lh} = 2,32 J/(g·K)
Heat of fusion	L = 170 J/g
Density of solid phase	ρ_{sh} = 1712 kg/m^3
Density of liquid phase	$\rho_{\ell h}$ = 1519 kg/m^3
Thermal conductivity of the solid phase	k_{sh} = 1.1 W/(mK)
Thermal conductivity of the liquid phase	$k_{\ell h}$ = 0.54 W/(mK)

Table 2.2 Data concerning $CaCl_2 \cdot 6H_2O$ in the solid and in the melted phase in accordance with [5]

It shows that two different solid phases can appear: The hexahydrate $CaCl_2 \cdot 6H_2O$ and the tetrahydrate $CaCl_2 \cdot 4H_2O$. Phase separation for a system with the composition of the hexahydrate (corresponding to 50.66 weight % $CaCl_2$) occurs due to the formation of tetrahydrate between 29.8°C and 32°C. Thus, when a solution of this partially incongruent system is cooled to 32°C, the tetrahydrate crystals will begin to form and precipitate. When the temperature reaches 29.8°C, this is succeeded by the precipitation of hexahydrate and this phase will prevent the water from getting in contact with the tetrahydrate. That way the tetrahydrates will not be converted into hexahydrates as they are supposed to according to the phase-diagram, and after repeated cycles the avaiable amount of latent heat will decrease significantly.

It should be mentioned that it is possible to obtain a stable function without phase separation if 2 weight % of $SrCl_2 \cdot 6H_2O$ is added to the system when the salt hydrate is of laboratory quality and 1.3% $Ca(OH)_2$ when it is of technical quality [5]. These additives have the effect of suppressing the formation of the tetrahydrates by increasing their solubility. At the same time the heat and the temperature of crystallization for the hexahydrates are almost unchanged. However, due to this, all the phase-changes are still occuring above the upper limit of the room temperature and the method is not suitable in relation to this project.

By addition of extra water, the formation of tetrahydrates is suppressed and at the same time part of the melting and crystallization will occur in the interval of the room temperature. In fig. 2.2 is shown the quantity of heat content for different concentrations of $CaCl_2$. The curves for the mixtures with 46% and 43.6% $CaCl_2$ are calculated from the following formulas given in [2].

$$t \leqq t_s: \; Q_s = L \cdot \left(F_{Sol}(t_o) - F_{Sol}(t) \right) + \int_{to}^{t} C_{ml}(t)\,dt$$

$$t > t_s: \; Q_s = L \cdot F_{Sol}(t_o) + \int_{t_o}^{t_s} C_{ml}(t)\,dt + \int_{t_s}^{t} C_{mn}(t)\,dt$$

where:

t = the temperature of the mixture

t_o = the temperature at which the heat content of the mixture is put to zero

Q_s = the heat content (per unit of mass) at the temperature t

L = the heat of fusion

$F_{Sol}(t)$ = the weight fraction of the salt hydrate that is in the solid phase at the temperature t

t_s = the lower limit of the temperature interval in which all the salt hydrate is in the liquid phase

$C_{ml}(t)$ = the specific heat of the mixture for $t \leqq t_s$

$C_{mn}(t)$ = the specific heat of the mixture for $t > t_s$

The fraction of the salt hydrate in the solid phase is found from:

$$F_{Sol}(t) = \frac{F_{am} - S_a(t)}{F_{as} - S_a(t)}$$

where:

F_{am} = the weight fraction of anhydrous salt in the mixture, i.e. the concentration

F_{as} = the weight fraction of anhydrous salt in the salt hydrate (0.5066 in case of $CaCl_2 \cdot 6H_2O$)

$S_a(t)$ = the solubility of the anhydrous salt in water at the temperature t, i.e. the weight fraction of the anhydrous salt in a saturated solution

From a regression of table values the following expression for the solubility can be obtained:

$$S_a(t) = 0.31104 \cdot \exp(0.015427 \cdot t)$$

The specific heat for $t \leqq t_s$ is calculated from:

$$C_{ml}(t) = F_{Sol} \cdot C_{sh} + (1-F_{Sol}) \cdot \left(\frac{S_a(t)}{F_{as}} \cdot C_{lh} + \left(1 - \frac{S_a(t)}{F_{as}}\right) \cdot C_w \right)$$

where:

C_{sh} = the specific heat of the solid phase salt hydrate

C_{lh} = the specific heat of the liquid phase salt hydrate

C_w = the specific heat of water

For $t > t_s$ we have the specific heat:

$$C_{mh} = \frac{F_{am}}{F_{as}} \cdot C_{lh} + \left(1 - \frac{F_{am}}{F_{as}}\right) \cdot C_w$$

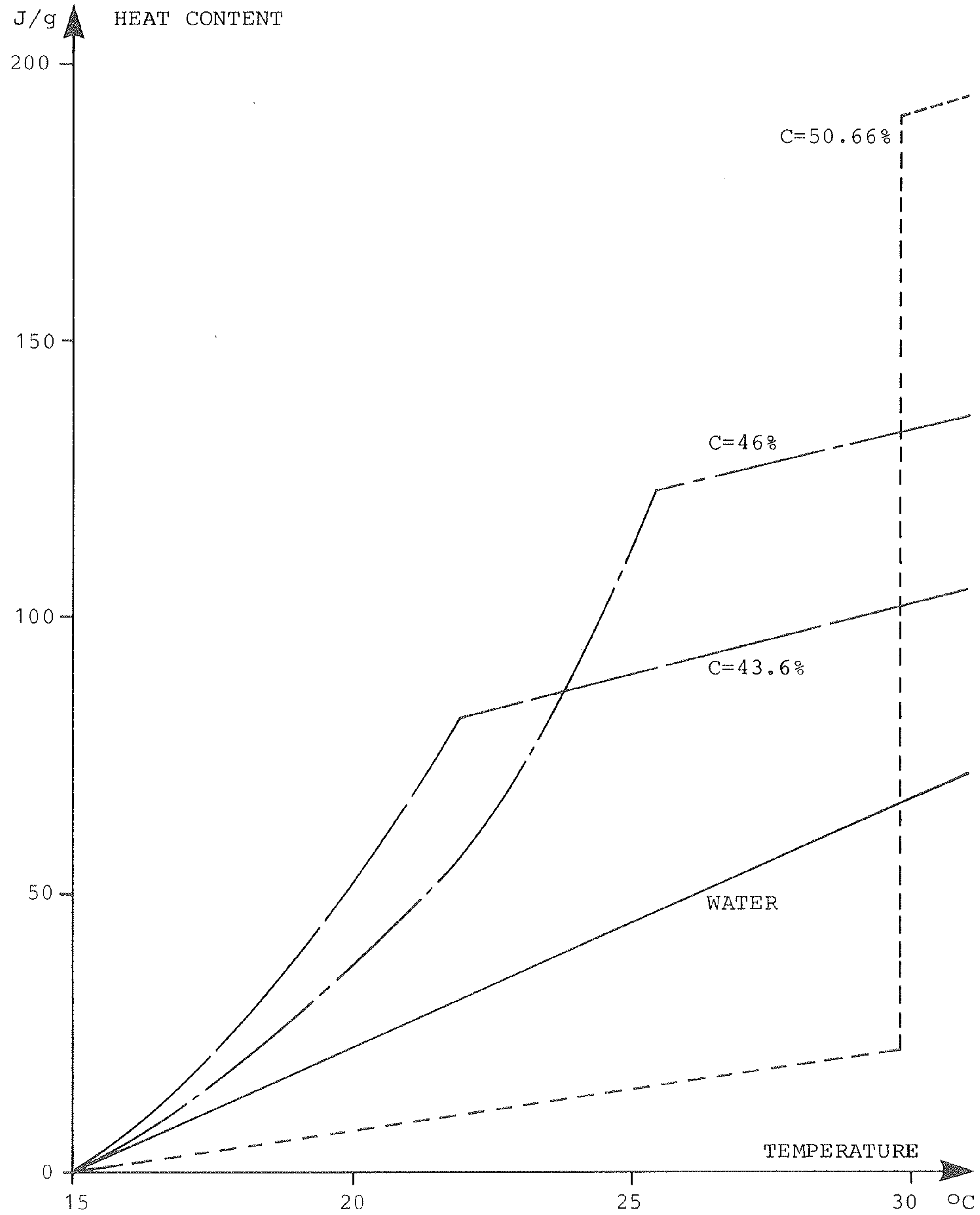


Fig. 2.2 The heat content of a mixture of $CaCl_2$ and water for different concentrations (C) of $CaCl_2$. The heat content is put at zero at 15°C

In the temperature interval $20^{o}C$ - $26^{o}C$ the change in heat content of the mixture with 46% $CaCl_2$ is 88 J/g which corresponds to 52% of the heat of fusion for the salt hydrate without additional water. The 46% $CaCl_2$ appeared to be very close to the optimal concentration as far as the quantity of heat content between $20^{o}C$ and $26^{o}C$ is considered. For the 43.6% $CaCl_2$, the change in heat content is somewhat less, 42 J/g, corresponding to 25% of the heat of fusion for the hexahydrate. The change in heat content per unit volume in the mentioned temperature interval was found to be about 5.2 times as large as water for the 46% $CaCl_2$ mixture and 2.4 times as large for the 43.6% $CaCl_2$ mixture.

3. REALISTIC TESTING OF TWO DIFFERENT STORAGE UNITS

In order to demonstrate that local storage units will function and to gain some experience from them, two storage units were constructed and monitored under realistic running conditions.

3.1 The construction of the storage units

Two local storage units, based on different working principles, were constructed. In the first unit the heat transfer between the room air and the heat storage medium was effected by simply conducting the air along the containers with the salt-water mixture. In order to obtain an improved heat transfer, the second heat storage unit was constructed with a separate heat exchanger between the room air and the water, so that the heat was transported to and from the storage containers by means of a water loop.

The general criteria defined in connection with the construction of the storage units are mentioned below.

(a) To secure storage of a considerable amount of the solar radiation, a satisfactory heat transfer between the storage medium and the room air is necessary.

(b) It must be economically advantageous to produce the storage unit. This implies that the unit should be simple to construct and that the materials and components are inexpensive.

(c) The unit should be well insulated, making the losses from the unit neglegible during periods with no heat demand.

(d) The dimensions and the weight of the storage unit must be suitable. The unit should be as compact as possible and with a height close to the ceiling height of the room limiting the needed floor space. The unit should not be difficult to move.

(e) The inlet of the storage unit's heat exchanger must be situated at the top, as the temperature of the room air is highest near the ceiling.

(f) As the storage unit is supposed to be placed in the living room, attention must be paid to the aesthetics.

The two storage units are described in the following. They were both constructed at the Thermal Insulation Laboratory in cooperation with industrial companies and based on commercial products. It was attempted to fulfil the above mentioned requirements, but it must be pointed out that the units in several ways are not fully optimized prototypes. As described in chapter 5, a storage unit with an improved design was later developed.

From initial investigations it became clear that it would be impossible to obtain a sufficient heat transfer between the room air and the storage medium if the heat had to be transferred only by natural convection of the room air. It was therefore necessary to induce a forced air flow by a ventilator, and for this purpose a cross stream ventilator with a power consumption of 65 W was used for both units. The ventilator is controlled by a differential thermostat, so that it is switched on if either the room temperature is higher than 21°C and higher than the storage temperature (charging of the unit), or if the room temperature is lower than 21°C and lower than the storage temperature (discharging of the unit). In the unit with the separate air-to-water heat exchanger, the pump for the water loop is switched on simultaneously with the ventilator. If the room temperature falls to 20°C, the auxiliary heating by electric radiators will begin to operate.

As mentioned in the previous chapter, a salt-water mixture with 43.6% $CaCl_2$ and 56.4% H_2O was chosen as heat storage medium, and from initial calculations it was found that a storage volume of approx. 100 litre was appropriate for the rooms.

The heat storage unit based on direct heat transfer
<u>between the room air and the storage containers</u>

This storage unit is illustrated by the drawings in fig. 3.1 and the photograph in fig. 3.2. The salt-water mixture was poured into 44 plastic containers of 2.5 litre, leaving a small air space open at the top in order to allow the mixture to expand. The containers were arranged on shelves with small air gaps between. The air is flowing downwards through holes in the shelves and holes at the top and bottom of the box. Behind the opening at the top is mounted a thin plastic film to prevent heat losses due to reverse air flow (caused by natural convection).

The ventilator is placed at the bottom of the unit. The containers are placed in an insulated box which is a commercial deepfreezer with the measures 0.6 x 0.6 x 1.7 m.

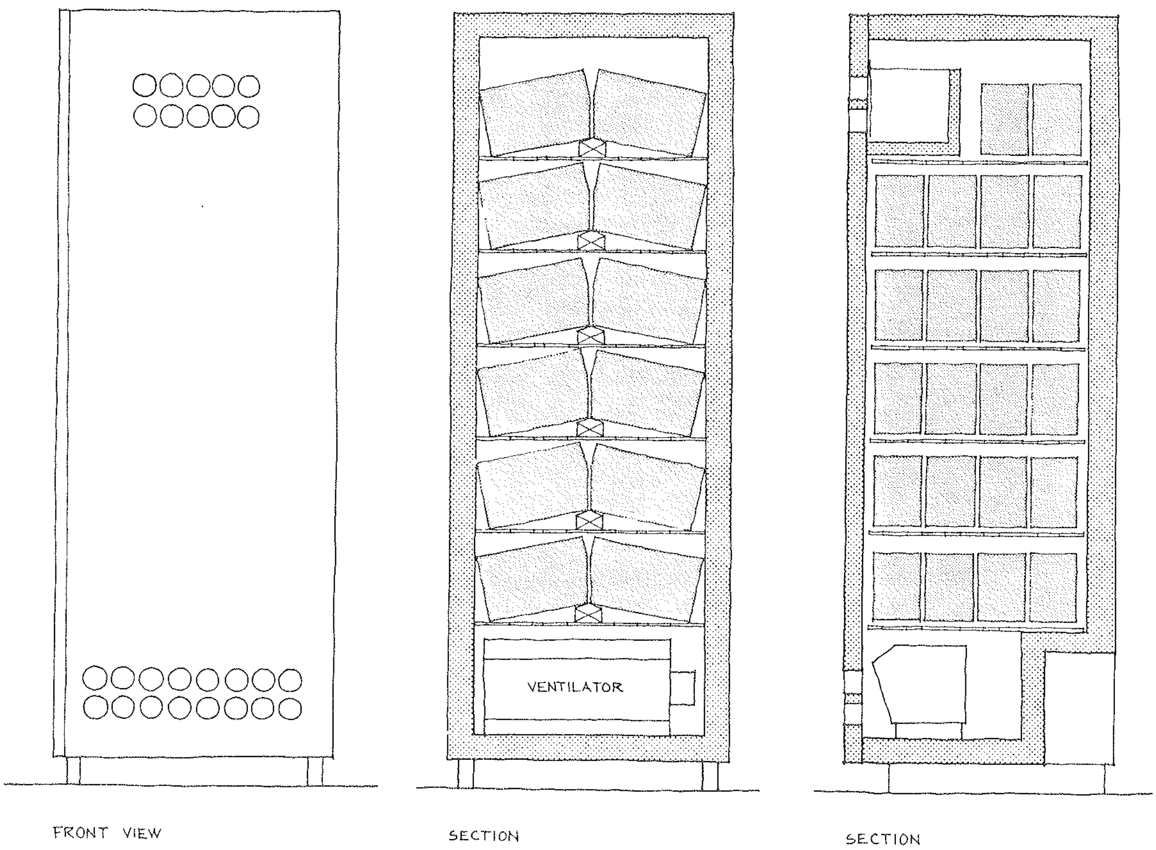


Fig. 3.1 Storage unit consisting of plastic containers placed in an insulated box

Fig. 3.2 The storage unit with the salt-water mixture placed in 2.5 liter containers

The heat storage unit based on separate container and heat exchanger

The principle of the second storage unit is shown in fig. 3.3 and fig. 3.4 is a photo of the unit. In this case 126 kg salt-water mixture was used and it was placed in three vertical tubes with a height of 2 m and a diameter of 0.16 m (see fig. 3.5). The tubes, normally used as energy absorbers for heat pump systems, are made from PVC. In the walls are extruded small channels through which the water is flowing when the pump is switched on. The channels are connected by a manifold in both ends. A small circulating pump (GRUNDFOSS UPS 15-35) with a power consumption of 35 W was used for the water loop. The water flows in upward direction through the tube walls. Thereby the temperature of the salt-water mixture will decrease in upward direction when heat is transferred to the storage, and the natural convection in the containers will cause the soft stirring which is necessary if phase separation is to be avoided. Unwanted flow in the water loop, due to natural convection, is prevented by a springloaded check-valve. Polyurethane, which was foamed in situ, was used as insulation for the tubes.

The heat exchanger is of the finned-coil type made by the Nordic Ventilator Company. It consists of two elements (LFA 1051-4 POS L) connected in series. The dimension of each element is 1.0 x 0.5 x 0.1 m, and the distance between the fins (aluminum) is 3 mm and between the pipes (copper) 50 mm. The heat exchanger is placed in a 2.25 m high channel made from plywood boards. The ventilator is situated at the top, and the inlet and outlet is at the top and at the bottom, respectively.

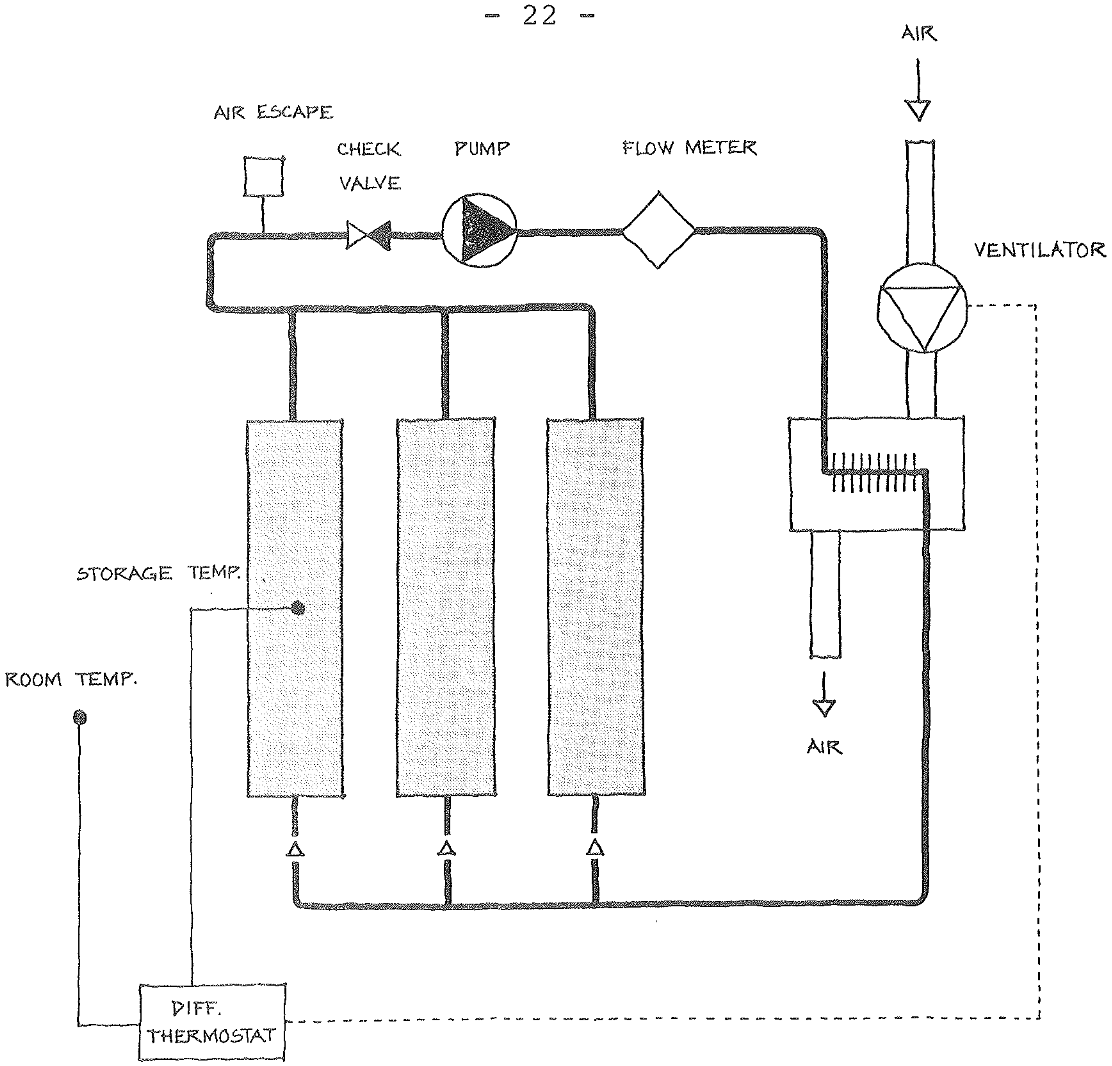


Fig. 3.3 (above)
Principle of storage unit with separate containers and heat exchanger to the air

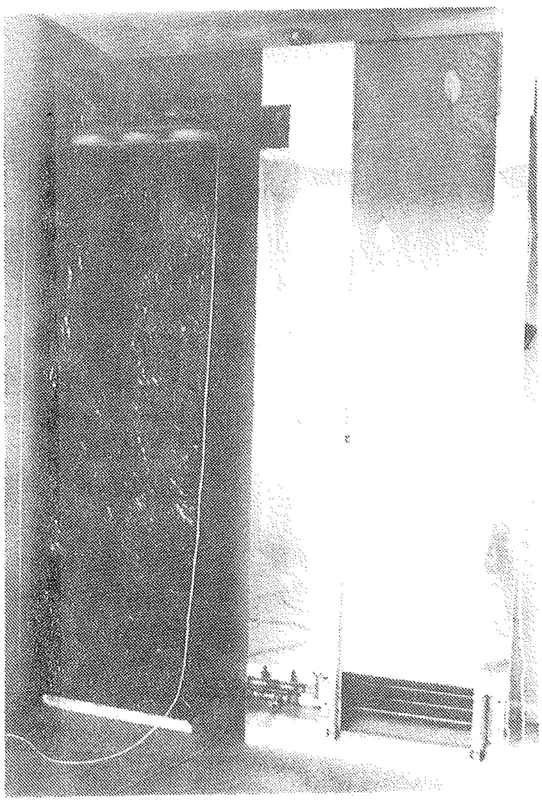


Fig. 3.4 (left)
Photograph showing the storage unit with separate heat exchanger

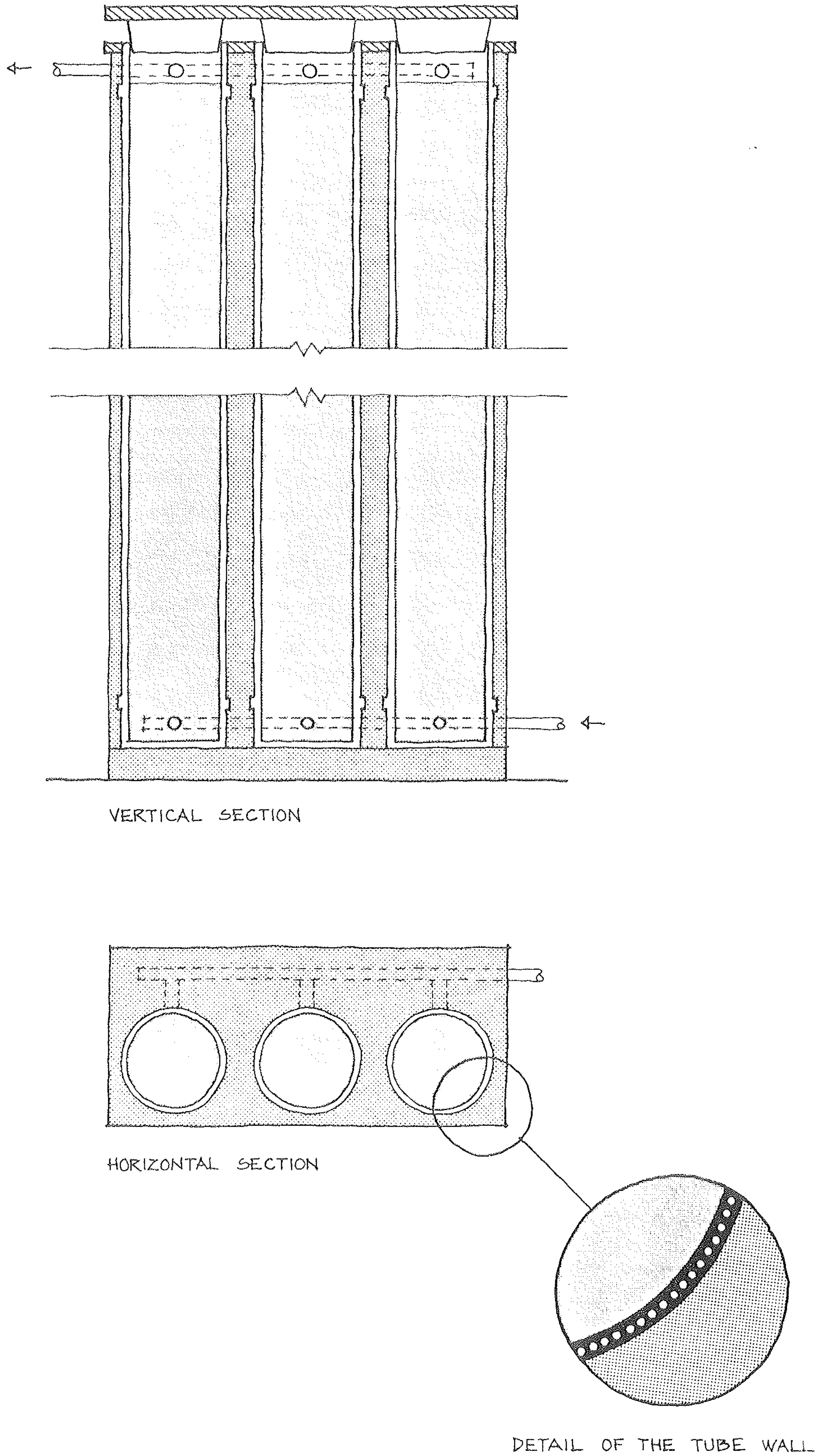


Fig. 3.5 Storage containers made of plastic tubes with water channels in the wall

Thermal characteristics of the storage units

In table 3.1 are listed values for the heat content of the storage unit in the temperature interval 20°C to 26°C, the overall heat loss coefficient from the storage medium to the room air, and the overall heat transfer coefficient between the storage medium and the room air in case of charging and in case of discharging of the unit. The heat content between 20°C and 26°C was calculated from the theory explained in chapter 2. The heat loss and the heat transfer were measured in different set-ups. The measurement of the overall heat transfer coefficient was carried out with some uncertainty, so the numbers stated should be regarded only as approximate values.

The separate heat exchanger did not provide as good a heat transfer as was expected which has to do with the dimensions of the heat exchanger not being optimized for the actual purpose.

	Combined Container and heat exchanger	Separate Container and heat exchanger
Calculated storage capacity 20-26°C	1.7 kWh	1.5 kWh
Overall heat loss coefficient from storage to room	1.7 W/°C	6 W/°C
Overall heat transfer coefficient from room to storage	50 W/°C	75 W/°C
Overall heat transfer coefficient from storage to room	50 W/°C	50 W/°C

Table 3.1 The thermal characteristics of the two storage units used for testing under realistic conditions

3.2 The test rooms and the measuring system

The experiments were carried out in an experimental house situated on the test area of the laboratory. Fig. 3.6 shows a picture of the house.

Three identical test rooms were used; the storage units were placed in two of the rooms and the third was used as a reference room. A plan of the rooms is shown in fig. 3.7. Each of the rooms has a floor area of 10.4 m^2 and a window with a glazing area of 2.12 m^2. The outer wall, the ceiling and the floor are insulated with approx. 200 mm rockwool, and 80 mm polyurethane foam is used in the partition walls. The heat capacity of the rooms is very small as only gypsum boards, plywood boards and chipboards are used on the inside of the insulation. During the test period there was no furniture in the rooms.

The rooms were heated by electric radiators (supplying approx. 950 W) controlled by the standard thermostat built into the radiator.

Quite a detailed measuring system was set up; this is described in the following.

The energy savings due to the local storage units were measured with ordinary electricity meters (kWh meters) placed on each radiator. The meters were read manually once a day on all working days. The distribution of the electricity consumption in each room was measured, throughout all 24 hours, as the running time for the radiators. This was done in such a way that the on-signal was transformed into pulses which were then counted by means of three full-time counters (totalizers) being part of the data acquisition system.

The solar radiation entering the room was measured with a Kipp & Zonen CM11 pyranometer placed vertically behind the window in the reference room. The pyranometer was scanned every minute.

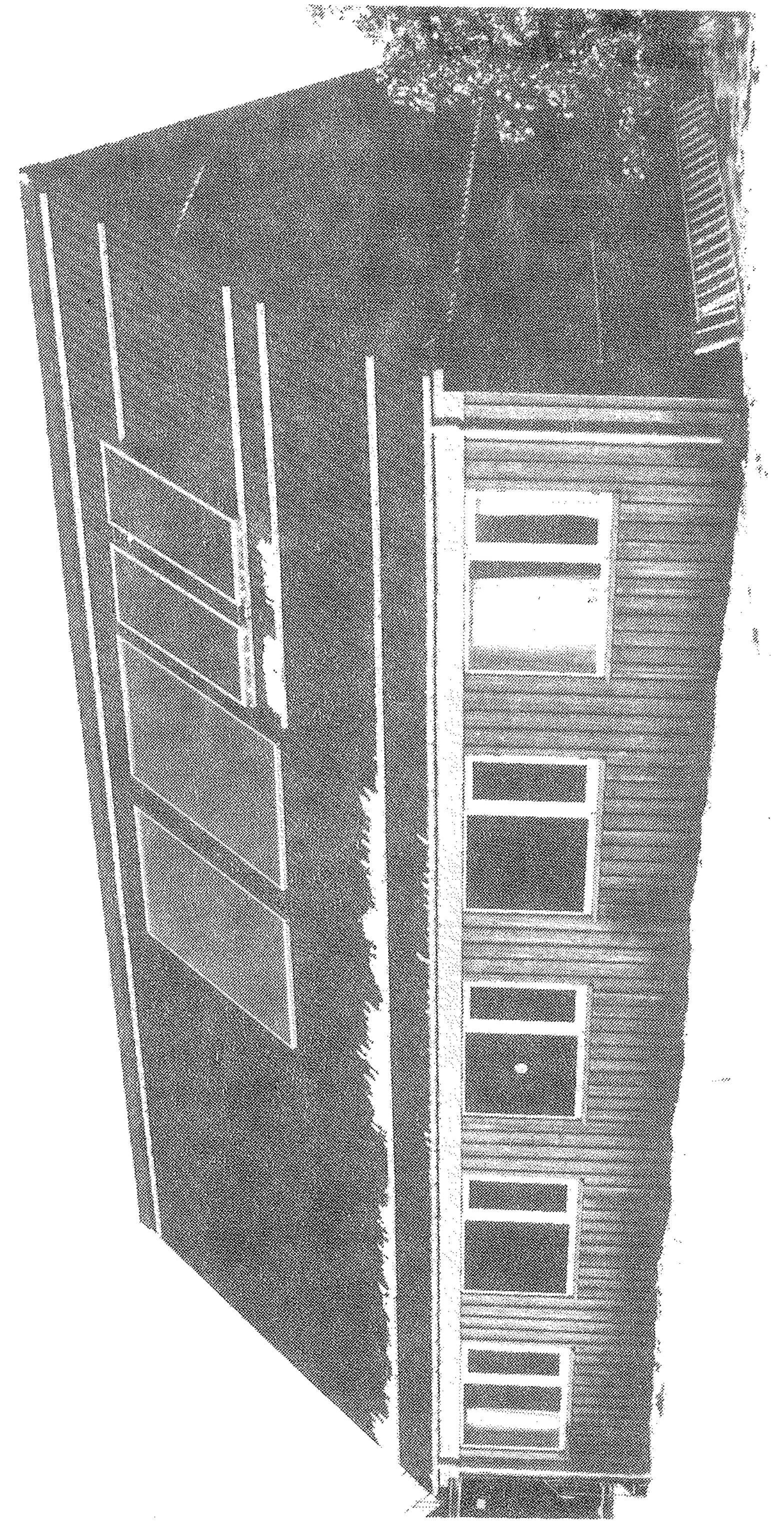


Fig. 3.6 The test house

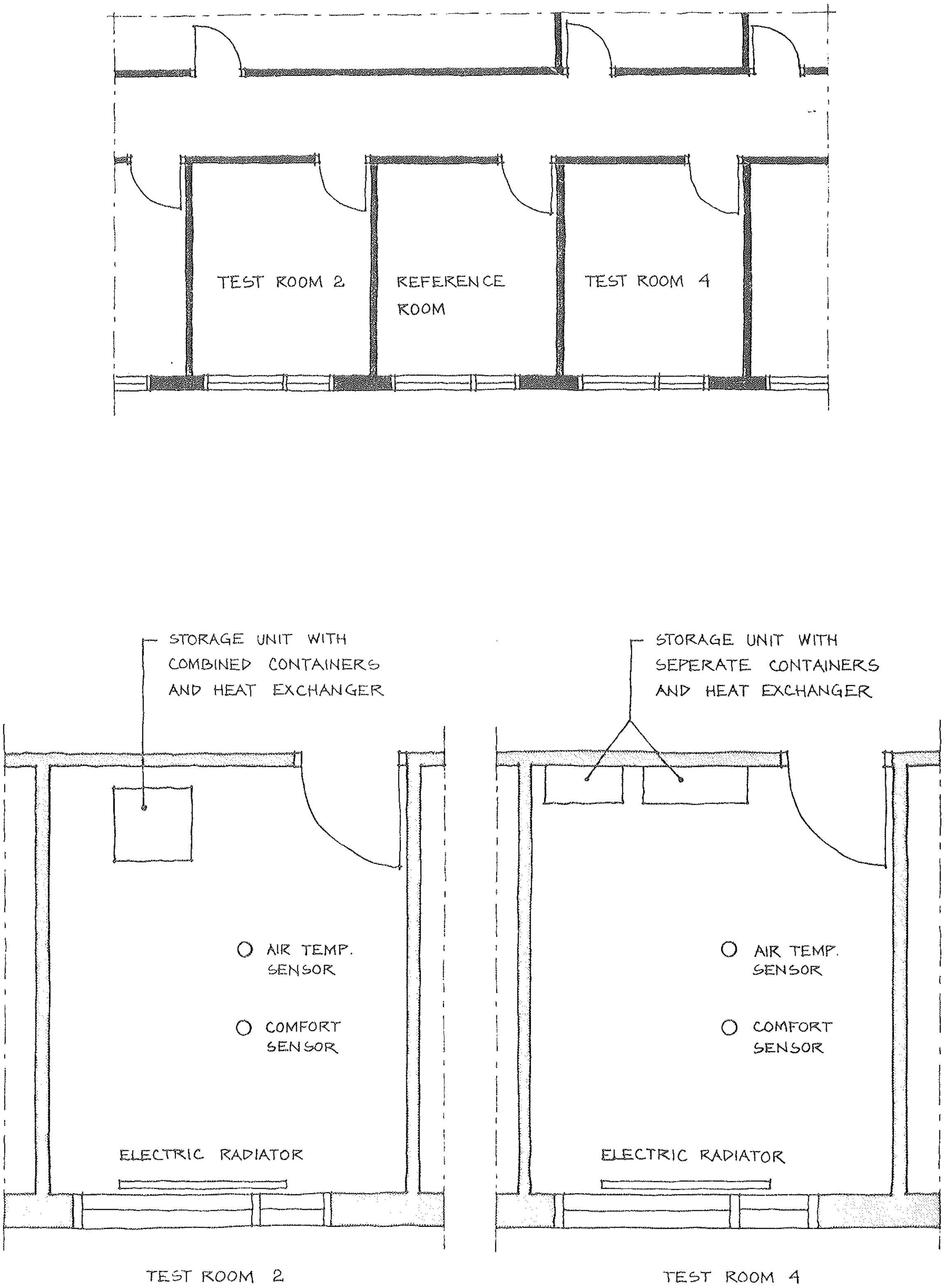


Fig. 3.7 The test rooms

The outdoor air temperature was measured with a thermocouple type T placed in a radiation shield on the north side of the house.

The air temperature in the rooms was measured with type T thermocouples placed in radiation shields. The temperature was measured at ten different levels as shown in fig. 3.8.

The temperatures in the storage units were also measured with type T thermocouples. In the storage with the many plastic containers, the temperature was measured in one container on each shelf. In the storage with three plastic tubes, the temperature was measured at five levels in one of the tubes. The temperature of the air entering and leaving the units was measured in both storage units.

The running time of the storage units, when charging or discharging, was measured as the on-off signal was scanned every minute.

The comfort of the rooms was measured by means of a comfort meter (Brüel & Kjaer, type 1212) with a sensor placed in each room. The location of the sensors is shown in fig 3.7 and 3.8. The comfort meter is normally equipped with one sensor only, but as the sensors can be interchanged without any calibration problems, it was fairly simple to make the comfort meter shift between the sensors controlled by the data acquisition system. Each sensor was used once every hour. The comfort meter measured the PMV-value (Predicted Mean Vote) according to an activity level of 1.2 met, a clothing level of 1.0 clo and a vapor pressure of 1.8 % kPa. This corresponds to a comfort temperature of 21°C.

The data acquisition system used was a Hewlett Packard A3052. All the temperatures, solar radiation and running condition were measured every minute and averaged over a ten-minute period. The ten-minute values were printed out and stored on magnetic tape cassettes for further treatment. The system is shown on Fig. 3.9.

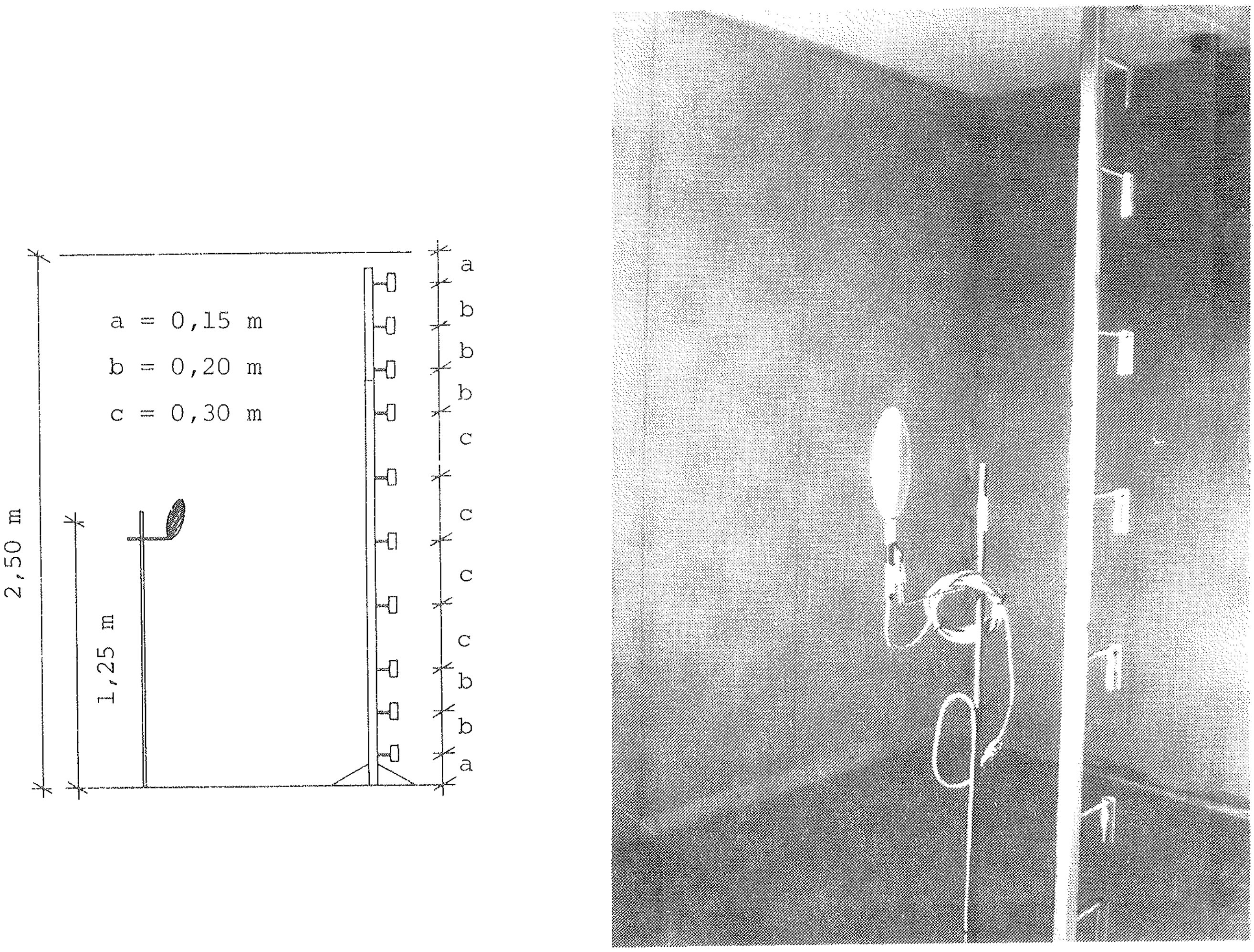


Fig. 3.8 Location and photo of sensors for air temperature and comfort meter

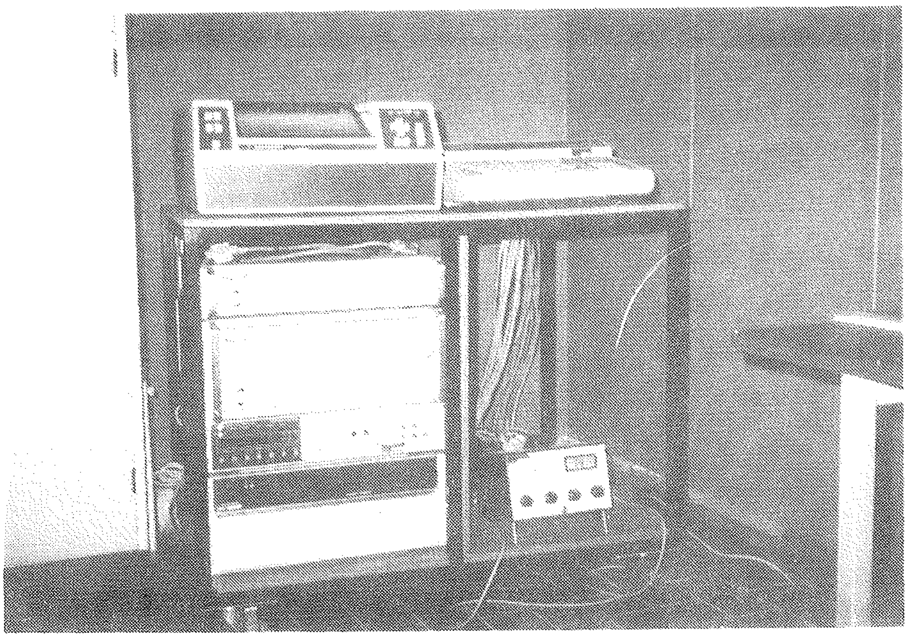


Fig. 3.9 The data acquisition system

3.3 The results of the experiments

The realistic testing took place during part of the spring and in the autumn of 1983. Due to some irregularities during the first time of measuring (amongst other, the control system of one of the storage units did not function well) results only from the autumn are presented.

Results of the measurements for two selected periods of 48 hours are shown in figs. 3.10 - 3.17. Fig. 3.10 (first period) and fig. 3.14 (second period) are showing the measured values of the ambient temperature and the solar radiation transmitted through the windows during the two periods. Figs. 3.11 - 3.13 (first period) and figs. 3.15 - 3.16 (second period) are showing the variations of the mean room temperature, the mean storage temperature, the PMV-value and the running time of the radiators and of the storage units for the different test rooms.

The PMV-value is an index describing the predicted mean vote of a large group of people concerned about the thermal comfort with regard to clothing, activity level, water vapour pressure, air velocity, air temperature and mean radiant temperature. The following values are defined: + 3 = hot, + 2 = warm, + 1 = slightly warm, 0 = neutral, - 1 = slightly cool, - 2 = cool, and - 3 = cold.

Figs. 3.12 - 3.13 and figs. 3.16 - 3.17 give a clear impression of the principal function of the storage unit. When the solar radiation is transmitted to the rooms, the room air temperature will rise almost immediately. Shortly after the charging of the storage unit begins and will continue until the mean storage temperature equals the room temperature. Later in the afternoon, when the room temperature has fallen to about 21°C, the ventilator will start again resulting in heat being transferred from the

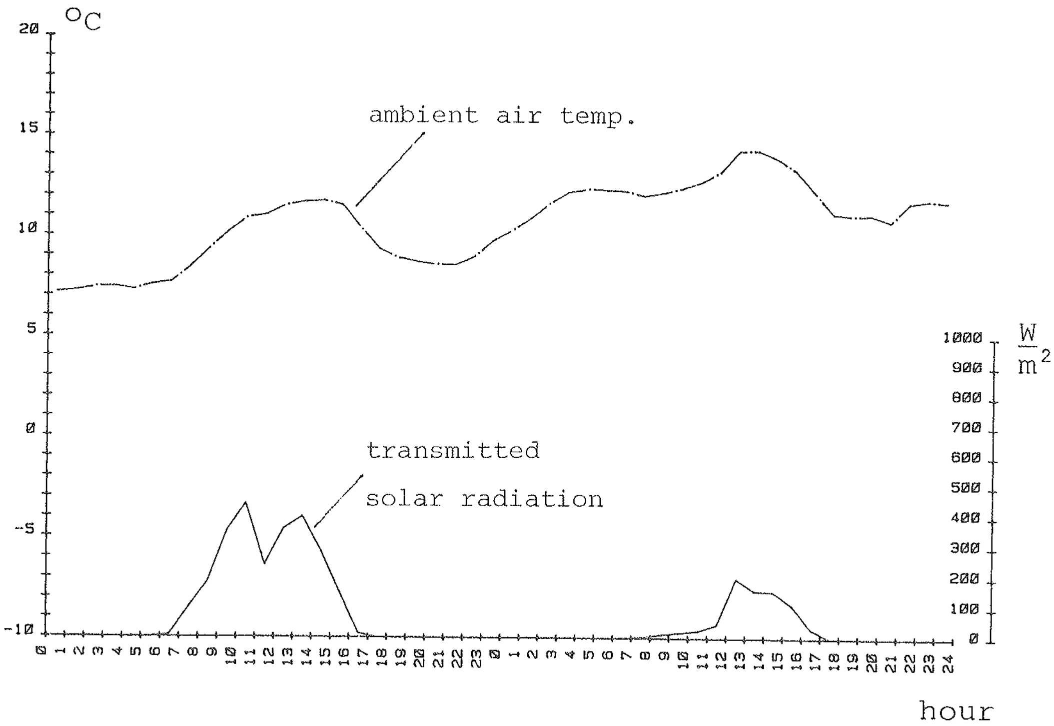


Fig. 3.10 The outdoor temperature and the solar radiation transmitted through the windows as measured on October 12-13.

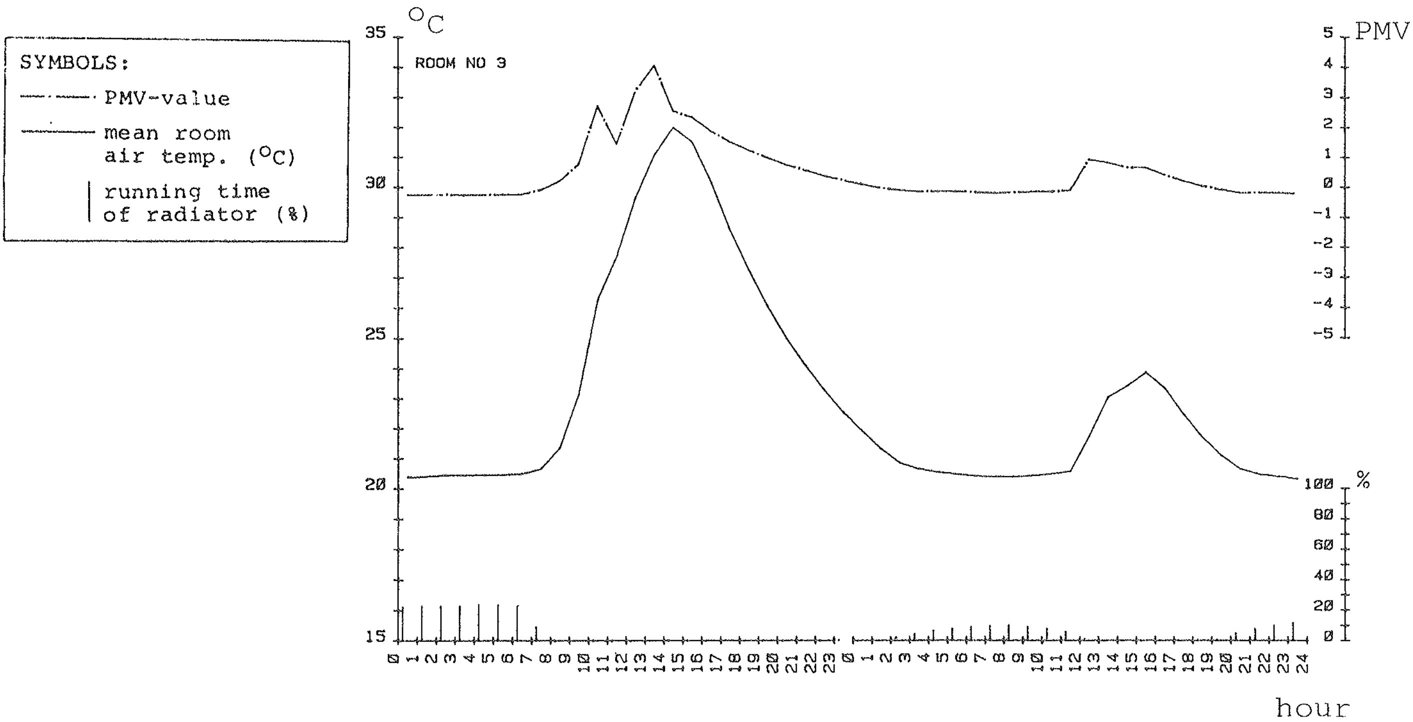


Fig. 3.11 The mean room temperature, the PMV-value and the running time (in % of the hour) of the radiator for the reference room on October 12-13.

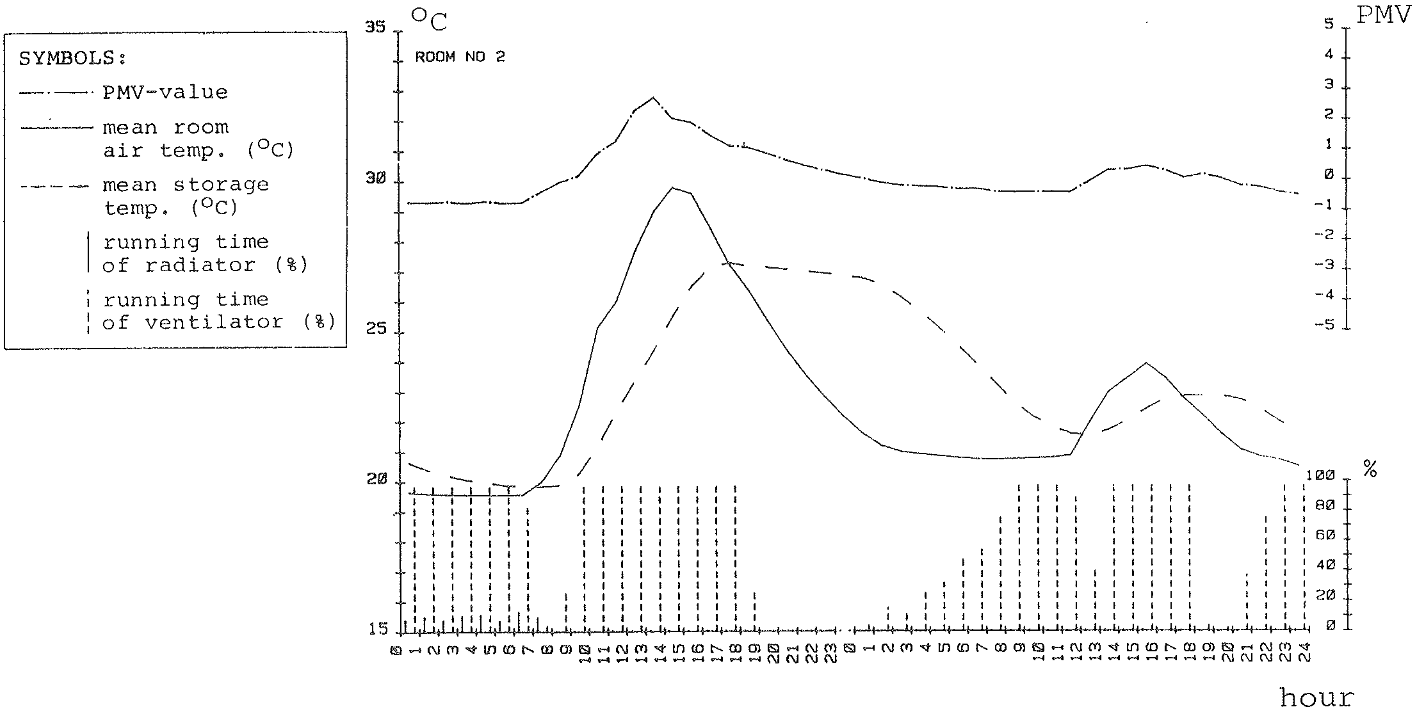


Fig. 3.12 The mean room temperature, the PMV-value, the mean storage temperature and the running time (in % of the hour) of the storage unit and the radiator for room 2 (room in which the heat exchanger of the storage unit is the surface of the containers) on October 12-13.

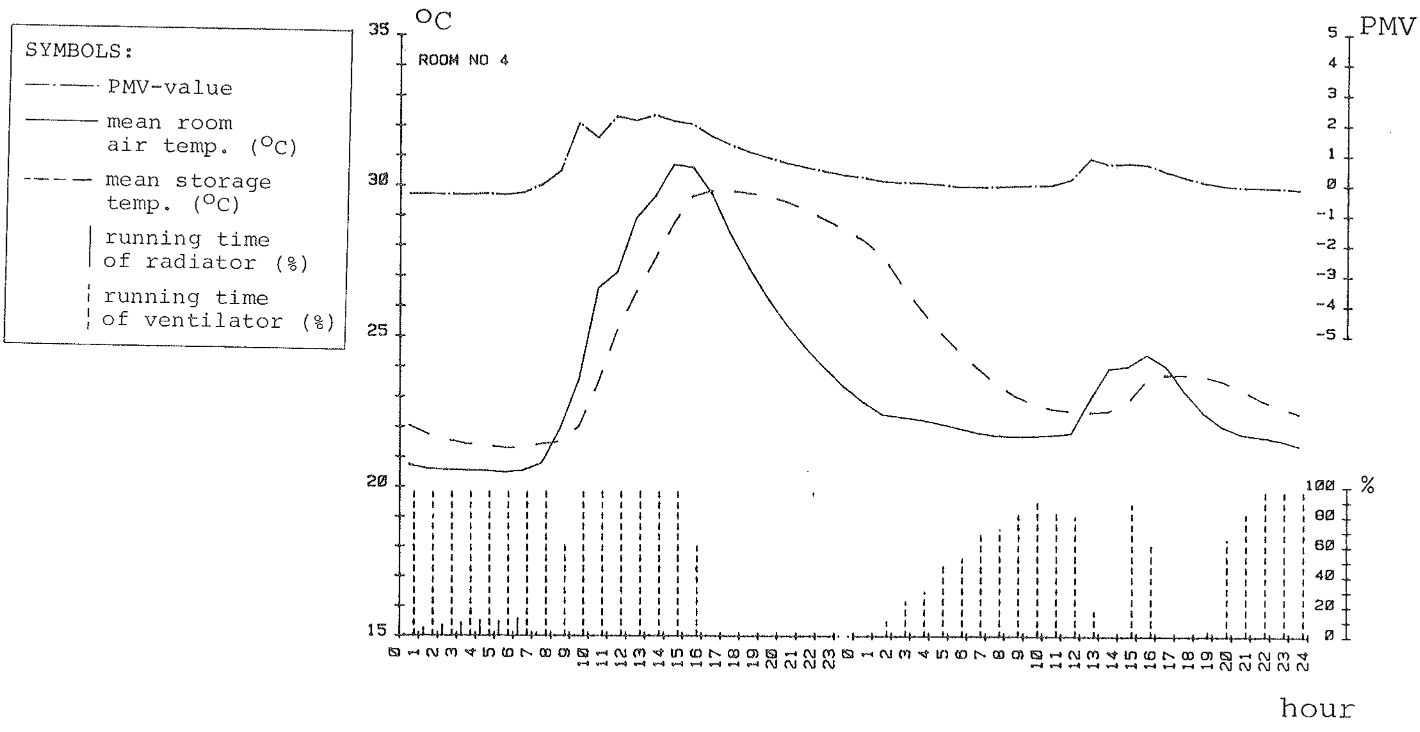


Fig. 3.13 The mean room temperature, the PMV-value, the mean storage temperature and the running time (in % of the hour) of the storage unit and the radiator for room 4 (room in which the storage unit has separate heat exchanger) on October 12-13.

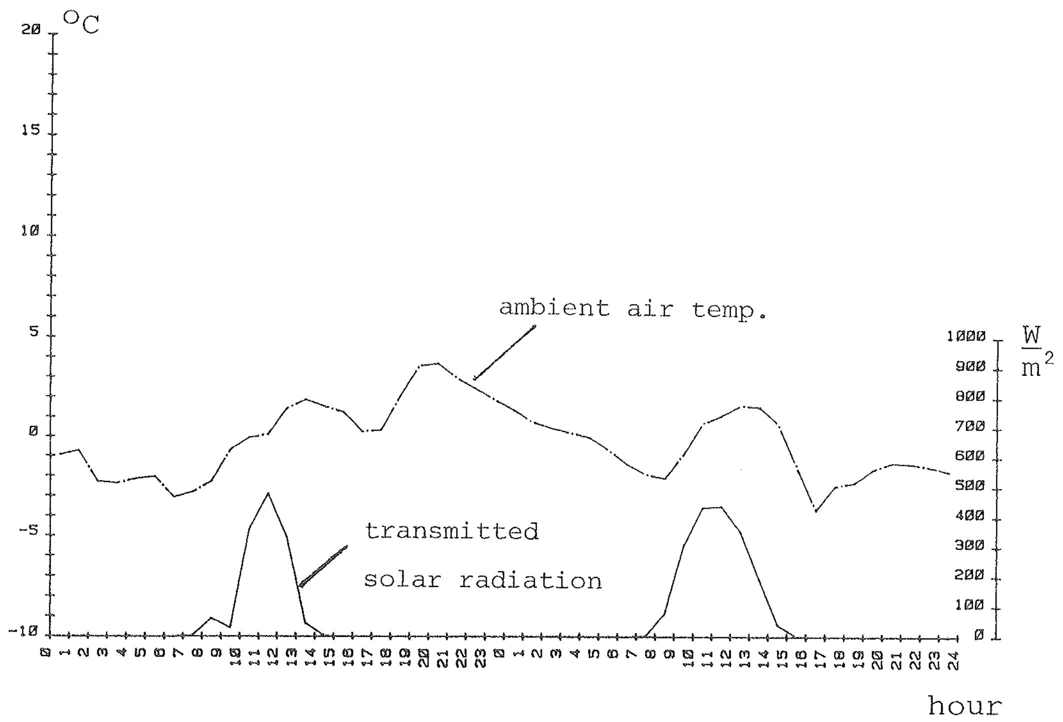


Fig. 3.14 The outdoor temperature and the solar radiation transmitted through the windows as measured on November 22-23.

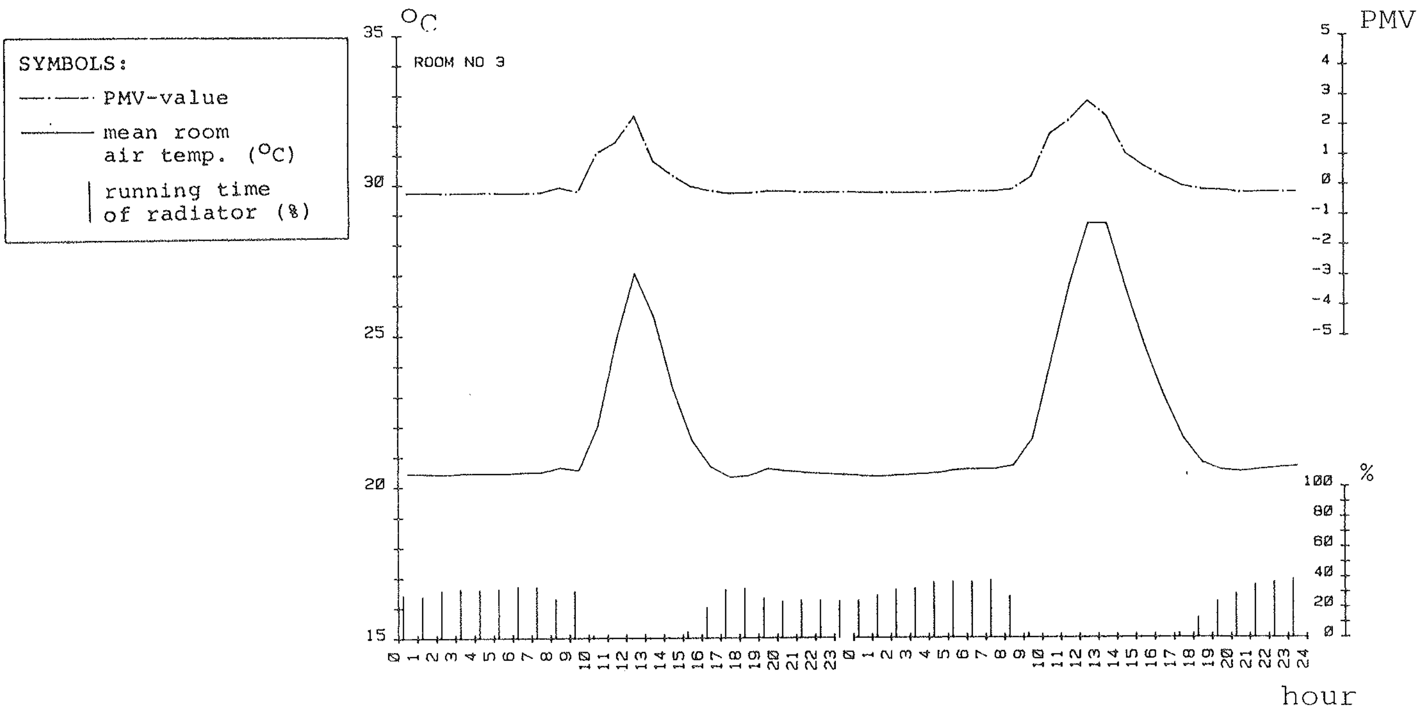


Fig. 3.15 The mean room temperature, the PMV-value and the running time (in % of the hour) of the radiator for the reference room on November 22-23.

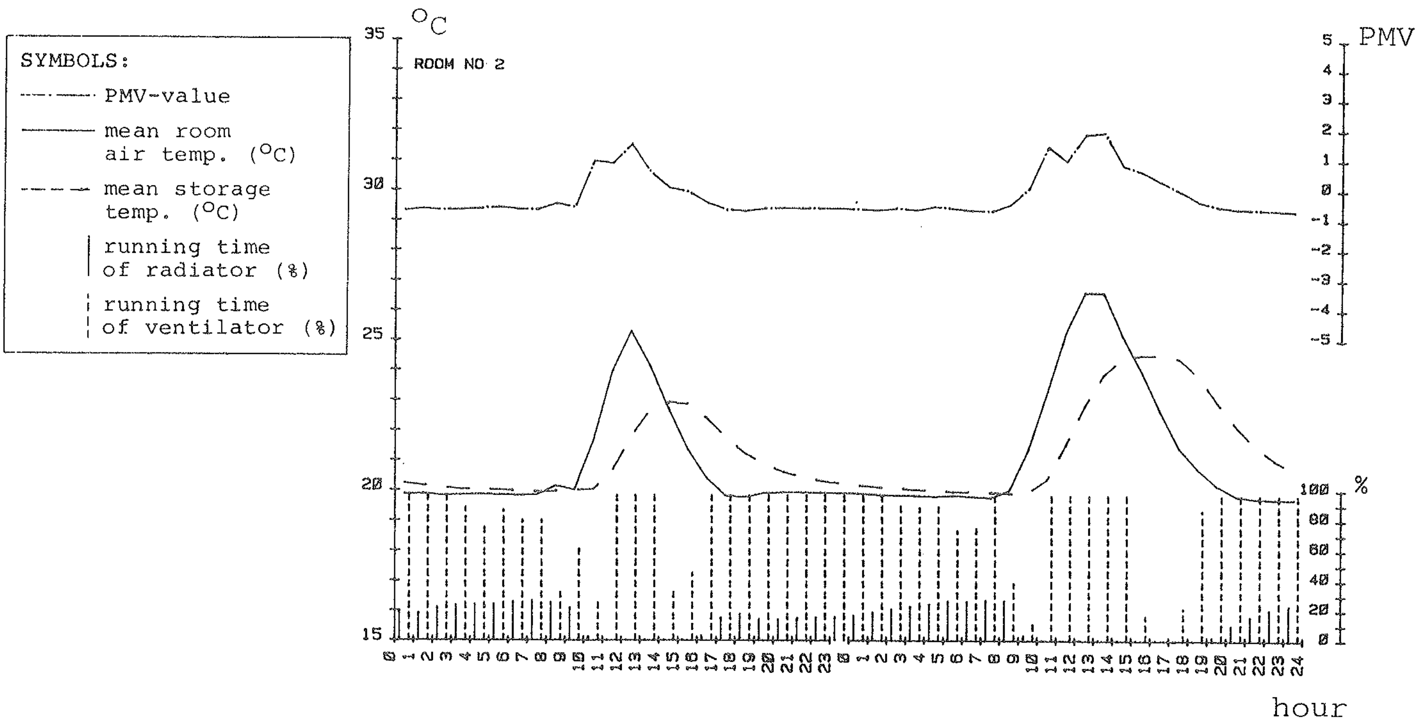


Fig. 3.16 The mean room temperature, the PMV-value, the mean storage temperature and the running time (in % of the hour) of the storage unit and the radiator for room 2 (room in which the heat exchanger of the storage unit is the surface of the containers) on November 22-23.

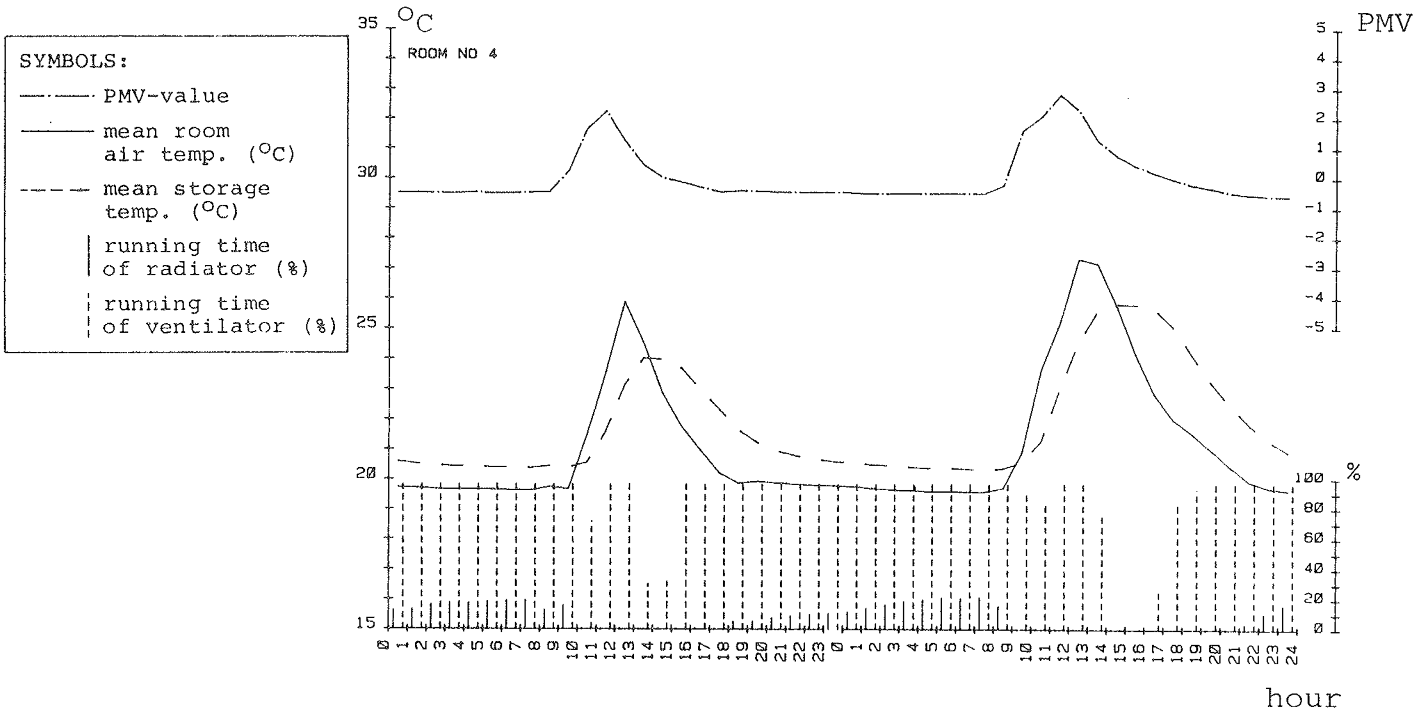


Fig. 3.17 The mean room temperature, the PMV-value, the mean storage temperature and the running time (in % of the hour) of the storage unit and the radiator for room 4 (room in which the storage unit has separate heat exchanger) on November 22-23.

storage to the room. If the heat requirement is large (as for the second of the two periods) or if the heat supply from the storage becomes too small as the storage temperature decreases, the room temperature will drop to about 20°C and the radiators will cover the remainder of the need for energy.

During the hours with much sunshine, the air temperature is lower in the two rooms with storage units than in the reference room. During the second day of the first period, where the amount of solar radiation is moderate, the air temperature in room 2 (the room in which the surfaces of the containers act as the heat exchanger of the storage unit) and room 3 (the reference room) are practically equal, whereas the air temperature in room 4 (the room in which the storage unit has a separate heat exchanger) is a little higher. The main reason for the temperatures in room 2 and room 4 not being lower in this case is that the storages have not been discharged during the preceeding morning, and with a modest difference between the respective room air temperature and storage temperature, only a small amount of heat is transferred to the storage. Regarding room 4, the room air temperature, at which the storage delivers heat, is almost 22°C during the morning. This is about 1°C higher than the temperature in the other rooms during the morning, and in connection with the heat capacity of the rooms, this explains the reason why the air temperature in room 4 becomes higher.

The PMV-value reflects the variations of the room temperature. The maximum value on the first day in the first period is significantly higher for the reference room than for the other rooms, but during the hours of sunshine in the second period, the thermal comfort appears to be about the same in room 3 and room 4. In room 2 the thermal comfort appears to be better; this is partly due to the room temperature being a little higher and probably partly because the ventilator of the unit in room 2 gives rise to higher air velocities in this room.

In table I in the appendix are shown the results on daily basis for the mean ambient temperature, the total transmitted solar radiation, the minimum and maximum value of the room air temperature and the storage temperature for the day (mean values of temperatures measured at different locations in the room or in the storage unit), the running time of the ventilators, the running time of the radiators and the maximum PMV-values. The results cover the period from October 10 to November 24; a few days during which the number of missing data were considerably large, are not included; for days with only a few missing data, these have been estimated. For a rather long period (October 16 - November 01) the power consumption of the radiator in room 2 was not registered by the data acquisition system, but the energy consumption during the whole period could be found by means of manual readings.

The maximum room temperature and the maximum PMV value are usually kept on a lower level for the rooms with storage units than for the reference room on days with much solar radiation. For a few days with more moderate solar radiation, the situation is reversed due to the reasons explained previously. The minimum air temperature is generally about 0.6^{o}C higher for the reference room, but this is compensated somewhat by the fact that the temperatures in the rooms with storage units are correspondingly higher during periods when the need for heating is covered exclusively by the storage units. It is noticed that the ventilators are switched on most of the time, which is related to the ventilators usually being switched on during periods at night and in the morning when the storage temperature is only slightly higher than the room temperature. In this situation the ventilator works because of the temperature difference, but due to the poor heat transfer, the storage temperature remains almost constant.

The total supply of auxiliary energy can be found by adding up the electricity consumption of the radiators, ventilators and the pump (for the unit with separate heat exchanger). The results for the whole period are stated in table 3.2. The total amount of solar radiation through the windows of each room during the days was 98.2 kWh.

The energy consumption of the two rooms with storage units appears to be about 12% (room 2) and 20% (room 4) less than that of the reference room.

	reference room	room where storage has combined storage container and heat exch.	room where storage has separate heat exchanger
energy consumption by radiator (kWh)	151.3	94.6	45.4
energy consumption by ventilator (and pump) (kWh)	-	38.0	75.9
total need for auxiliary energy (kWh)	-	132.6	121.3
energy savings compared to the reference room (kWh)	-	18.7	30.0

Table 3.2 The consumption of energy during the period between October 10 and November 24 (a few days with missing data are not included).

It should be mentioned that the results for the differences of energy consumption are very sensible to modest variations concerning the minimum temperature and the overall heat loss coefficient of the individual room and the temperature of the adjoining rooms. The facts, that the minimum air temperature for the reference room was about 0.6^{o}C higher than for the other test rooms and that there was a difference of about 3^{o}C in the temperature of the two office rooms next to test room 2 and 4 meant that the results should be corrected. By that the energy savings became 8% for room 2 and 14% for room 4.

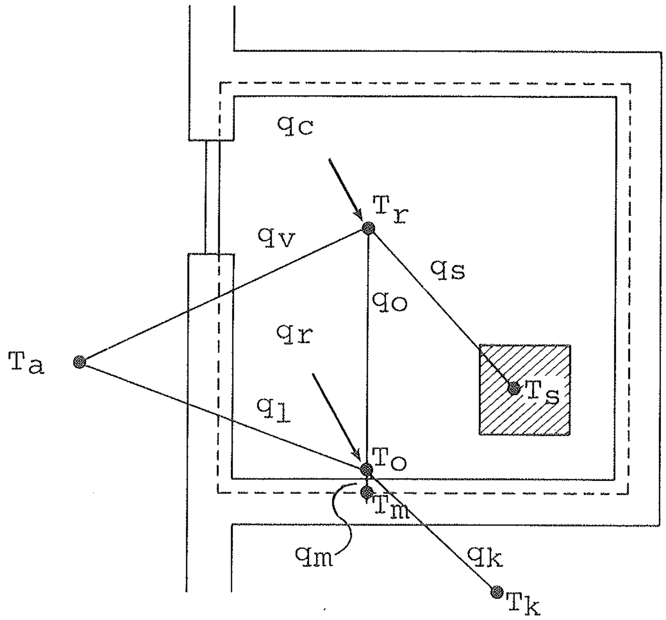
4. CALCULATION OF THE ENERGY SAVINGS

4.1 The computer program

In order to calculate the savings due to the local storage unit on a yearly basis, a model of the heat transfer to and from the storage unit was prepared and built into an existing computer program. The theory behind the program is explained in [6], and the structure of the program is described in detail in [7]. In the following is given a brief description in consideration of the additions made in connection with the storage unit. The program is capable of calculating the heat flows and temperatures in a given room at every half hour during the year. The program is using a rather simple calculation method. The different temperatures and ways of heat transmission taken into account are shown in fig. 4.1.

The main simplifications are that all inside surfaces (except the windows) are assumed to have the same temperature, and that the full heat capacity of the building is imagined to be concentrated in one fictive heat accumulating layer of insignificant thickness inside the walls. Changes of the stored heat takes place by heat transfer from the interior surfaces of the room. Further, the heat transfer coefficients between the air and the different surfaces are assumed to be constant.

With these assumptions, the different ways of heat transmission concerning the room can be combined by the following three heat balances.



Symbols used:

T_a = outdoor temperature

T_r = room air temperature

T_o = temperature of the interior surfaces

T_s = temperature of the storage unit

T_k = temperature of the adjoining rooms

T_m = temperature of the heat accumulating mass in the building structure

q_c, q_r = heat supplied by convection to the room air and by radiation to the room surfaces due to persons, electricity consumption, solar radiation, radiators and the ventilator of the storage unit

q_v = heat transfer due to air change with the ambient air

q_ℓ = heat loss of the room calculated from the interior surfaces

q_o = heat transfer between the room air and the interior room surfaces

q_s = heat transfer between the room air and the storage medium in the local storage unit

q_k = heat transmission to the adjoining rooms

q_m = heat transmission to the heat accumulating layer

Fig. 4.1 Thermal model used by the calculation program

For the heat accumulating layer we have:

$$(UA)_m \cdot (T_o - T_m) = S \cdot \frac{\delta T_m}{\delta \tau} \tag{4.1}$$

for the room air:

$$q_c + (UA)_o \cdot (T_o - T_r) + (UA)_s \cdot (T_s - T_r) + (G \cdot C) \cdot (T_a - T_r) = 0 \tag{4.2}$$

and for the interior surfaces of the room:

$$q_s - (UA)_\ell \cdot (T_a - T_o) - (UA)_k \cdot (T_k - T_a) - (UA)_o \cdot (T_r - T_o)$$

$$-(UA)_m \cdot (T_m - T_o) = 0 \tag{4.3}$$

where

T_o = the temperature of the inside surfaces of the room

T_m = the temperature of the heat accumulating mass in the building structure

T_r = the room air temperature

T_s = the temperature of the local storage unit

T_a = the ambient air temperature

T_k = the temperature of the adjoining room

$(UA)_m$ = the heat transfer between the room surfaces and the fictive heat accumulating layer per ^oC in temperature difference

$(UA)_o$ = the heat transfer from the room air to the interior room surfaces per ^oC

$(UA)_s$ = the heat transfer from the room air to the salt-water mixture in the heat storage unit; the quantity depends on whether there is a forced heat transfer by means of the ventilator or if only heat loss takes place

$(G \cdot C)$ = the heat transferred from the room air to the ventilation air (ambient air) per ^oC due to the ordinary air change of the room

$(UA)_\ell$ = the heat loss from the interior room surfaces to the ambient air per ^oC

S = the total heat capacity of the materials in the walls, ceiling and floor. If the time constant of one of the layers (i.e. the ratio between the heat capacity and the heat transmission coefficent between the surface and the layer) is larger than 3.8 hours, the thickness accounted for is reduced

τ = the time

q_c = the heat added to the room air by convection from persons, lighting, electric devices, radiators and the ventilator of the heat storage unit. Also part of the solar radiation through the windows (normally 50%) is encountered in this term, because a large part of the radiation will hit lightweight materials for example furniture, which will be heated very fast and thereby give off the absorbed radiation to the room air. If there should be a need for cooling, q_c also includes the amount of heat removed from the room by extraordinary ventilation.

q_s = the heat added to the room surfaces by radiation from persons, lighting, electric devices and because of solar radiation.

For each new half-hour, the program calculates the temperature of the heat accumulating layer from the change in heat content during the previous half hour. This is done by transforming (4.1) into the difference equation:

$$(UA)_m \cdot (T_{o,n-1} - T_{m,n-1}) = S \cdot \frac{T_{m,n} - T_{m,n-1}}{\Delta\tau} \qquad (4.4)$$

where indices n and $n-1$ refer to the present and the previous half-hour, and $\Delta\tau$ is one half-hour.

Next the room air temperature and the temperature of the room surfaces are found by solution of the two linear equations (4.2) and (4.3). In the preliminary calculation of each half-hour, these temperatures are calculated without the consideration of any supply of heat by radiators and removal of heat by extraordinary ventilation. Neither is any heat transfer by forced convection to and from the local storage unit considered; only the heat loss from the storage unit is taken into account. From these temperatures is then calculated how much additional heating or cooling is needed in order to keep the room air temperature at the minimum or maximum acceptable value.

If a heating demand exists, it is investigated whether the temperature of the storage unit is higher than the room temperature, and if this is the case, the heat delivered from the storage unit is calculated. The part of the heating demand that cannot be covered by the storage unit is supplied by the radiators.

In case the room temperature is higher than both the minimum acceptеble value and the temperature of the storage unit, the heat transferred to the unit is calculated by setting $(UA)_s$ equal to the overall heat transfer coefficient to the storage unit instead of the heat loss coefficient. After this a new calculation of the room air temperature, the surface temperature and the need for cooling (if any) is carried out.

After the calculation of the heat flows and temperatures for each half-hour, the heat content of the storage unit is found, and from that a new storage temperature is found. This storage temperature is calculated from an expression that approximates the relation between the heat content and the storage temperature (corresponding to the curves in fig. 2.2). If this storage temperature turns out to be higher than the room temperature in case of the storage unit being charged,or the storage temperature turns out to be lower than the room temperature after a period of discharging, the concerned half-hour is regarded as if the ventilator of the unit has not been switched on.

The program uses weather data from the Danish Test Reference Year [8], which is composed in such a way that it represents a typical Danish climate based on measurements during 15 years.

4.2 Calculation of the heat transfer to and from the storage unit

This section explains the different assumptions used in the calculation program concerning the design of the storage unit and the heat transfer to and from the salt-water mixture. As the heat transfer coefficient depends on the velocity of the air, this has to be estimated in advance from a calculation of the pressure drop.

In the calculations is assumed that the salt-water mixture in the heat storage unit is placed in a number of box-shaped one litre containers with the outer measures 215x120x45 mm. They are placed in an upright position, vertically directly above each other in seven layers and horizontally in rows with four containers in each. The number of rows is normally nine, but the total volume of the salt-water mixture can be changed by choosing another number. The distance between the containers is set to be 15 mm in horizontal direction and 35 mm in vertical direction.

This way, the outer measures of the unit will be about: height 1.85 m, width 0.60 m and depth (with 9 rows of containers) 0.60 m.

Determination of the air velocity

In order to obtain a large heat transfer coefficient, it was found desirable to obtain a mean air velocity of about 5 m/s. It is assumed that a ventilator with a power consumption of 70 W and an efficiency of 32% at the actual pressure drop and air quantity, is used. The air velocity is determined by the equation:

$$P \cdot \eta = \Delta H \cdot v \cdot A_d \qquad (4.5)$$

where

P = the power consumption of the ventilator

η = the efficiency of the ventilator

ΔH = the pressure drop between the inlet and outlet of the unit

v = the mean air velocity

A_d = the total area of the cross section of the vertical ducts between the containers

The pressure drop is the sum of the friction loss and the minor losses at the inlet and outlet. When calculating the pressure drop it is assumed that there exists a fully developped turbulent air flow in the vertical ducts between the containers. Since the thickness of the air space between layers of containers is relatively small, the air stream is considered to pass by undisturbed, i.e. there is not accounted for the minor losses at these locations.

The friction loss ΔH_f is found from:

$$\Delta H_f = f \cdot \frac{L}{d_n} \cdot \frac{\rho \cdot v^2}{2}, \qquad f = \frac{0.316}{Re^{1/4}}$$

where

f = the friction factor

L = the length of the duct

d_h = the hydraulic diameter

ρ = the density of the air

Re = the Reynold number $(= v \cdot d_h/\nu$, where ν is the kinematic viscosity of the air)

The formula (stated in e.g. [9], page 597), is derived from experiments with circular smooth pipes, but it is expected to be usable in the actual case, when the hydraulic diameter

$$d_h = \frac{4 \text{ x area of the duct}}{\text{perimeter of the duct}}$$

is inserted. Further the formula is valid only when $3000 < Re < 100000$ which showed to be fulfilled for the intended designs of the storage unit.

The minor losses are calculated from:

$$\Delta H_m = (\Sigma k) \cdot \frac{\rho \cdot v^2}{2}$$

where the loss coefficient k is set to o.5 at the inlet and 1.0 at the outlet.

When the pressure drop is calculated as mentioned, the velocity found from equation (4.5) will be between approx. 4 m/s for the largest storage unit considered and aprox. 6 m/s for the smallest, and so the size of the ventilator seems to be appropriate.

Calculation of the heat transfer coefficient between the room air and the salt-water mixture

The rate of heat transfer between the salt-water mixture and the circulating air in the storage unit is found from the expression:

$$q = h \cdot A_c \cdot (T_{s,m} - T_{a,m}) \tag{4.6}$$

where

h = heat transfer coefficient by forced convection

A_c = the area of the containers exposed to the air flow

$T_{a,m}$ = the mean temperature of the circulating air

$T_{s,m}$ = the mean temperature of the surfaces of the containers. All the salt-water mixture in a container is assumed to have practically the same temperature as the surface

The change in heat content of the circulating air per unit time between the inlet and outlet of the storage unit can be calculated from:

$$q = \dot{m} \cdot c_p \cdot (T_{a,o} - T_{a,i}) \tag{4.7}$$

where

$\dot{m}$ = the mass flow through the storage unit

c_p = the specific heat of the air

$T_{a,o}$ = the outlet temperature of the circulating air

$T_{a,i}$ = the inlet temperature of the circulating air

By assuming that $T_{a,m} = 1/2 \cdot (T_{a,i} + T_{a,o})$, (4.6) and (4.7) can be combined so that the heat transfer is related directly to the difference between the room air temperature T_r (which is assumed to be equal to the inlet air temperature) and the mean temperature of the salt-water mixture. This way the following expression is obtained:

$$q = (UA)_s \cdot (T_{s,m} - T_r) \qquad (4.8)$$

where the overall heat transfer coefficient $(UA)_s$ is calculated from:

$$(UA)_s = (1/(h \cdot A_c) + 1/(2 \cdot \dot{m} \cdot c_p))^{-1} \qquad (4.9)$$

In order to find the heat transfer coefficient h, the space in the vertical plane between the containers was regarded as if it was the gap between two parallel plates. In case of a fully developed turbulent air flow, the following formula from [10], page 135, can be used for the Nusselt number, Nu:

$$Nu = 0.0158 \cdot (Re)^{0.8} \qquad (4.10)$$

where Re is the Reynold number. In accordance with [10], the effect of the entrance region on the heat transfer should be taken into account in case of small values of L/d_h, where L is the length of the plate in the direction of the heat flow, and d_h is the hydraulic diameter which in this case is twice the plate spacing. If (as it was assumed at the calculation of the pressure drop) the effect of the horizontal air space between the layers is not considered, the dimension L is chosen as the total height of the containers, and by that the mentioned ratio is about 50. From the discussion in [10] it was decided to rise the value of Nu with 5% and by this we obtain the corrected formula:

$$Nu = 0.0166 \cdot (Re)^{0.8} \tag{4.11}$$

From $h = Nu \cdot k/d_h$, where k is the conductivity of air, the heat transfer coefficient in the storage unit is calculated, and it is plotted as a function of the air velocity on fig. 4.2 (as the air flow is turbulent for Reynold numbers larger than 2200 corresponding to velocities larger than aprox. 1.1 m/s).

When alternatively a formula for pipes were used, with the hydraulic diameter of the ducts instead of the pipe diameter, the heat transfer coefficient showed to be aprox. 25% larger than calculated from (4.11). Thereby it was supposed that formula (4.11) does not overestimate the heat transfer coefficient.

It is noticed that the temperature of the surface of the individual container and all the salt-water mixture inside are assumed to be equal in so far as only the convective heat transfer coefficient between the air and the container wall is used in the calculation of the heat transfer. This simplification is based on the assumption that the movements in the melted phase, due to natural convection, in a very efficient way can convey the heat to the place where melting or crystallization take place. In reality, the total heat transfer coefficient will probably be decreased slightly.

The heat transferring area A_c in formula (4.2) is set to the area of the vertical surfaces of the containers, ignoring the air movement in the horizontal space between the containers.

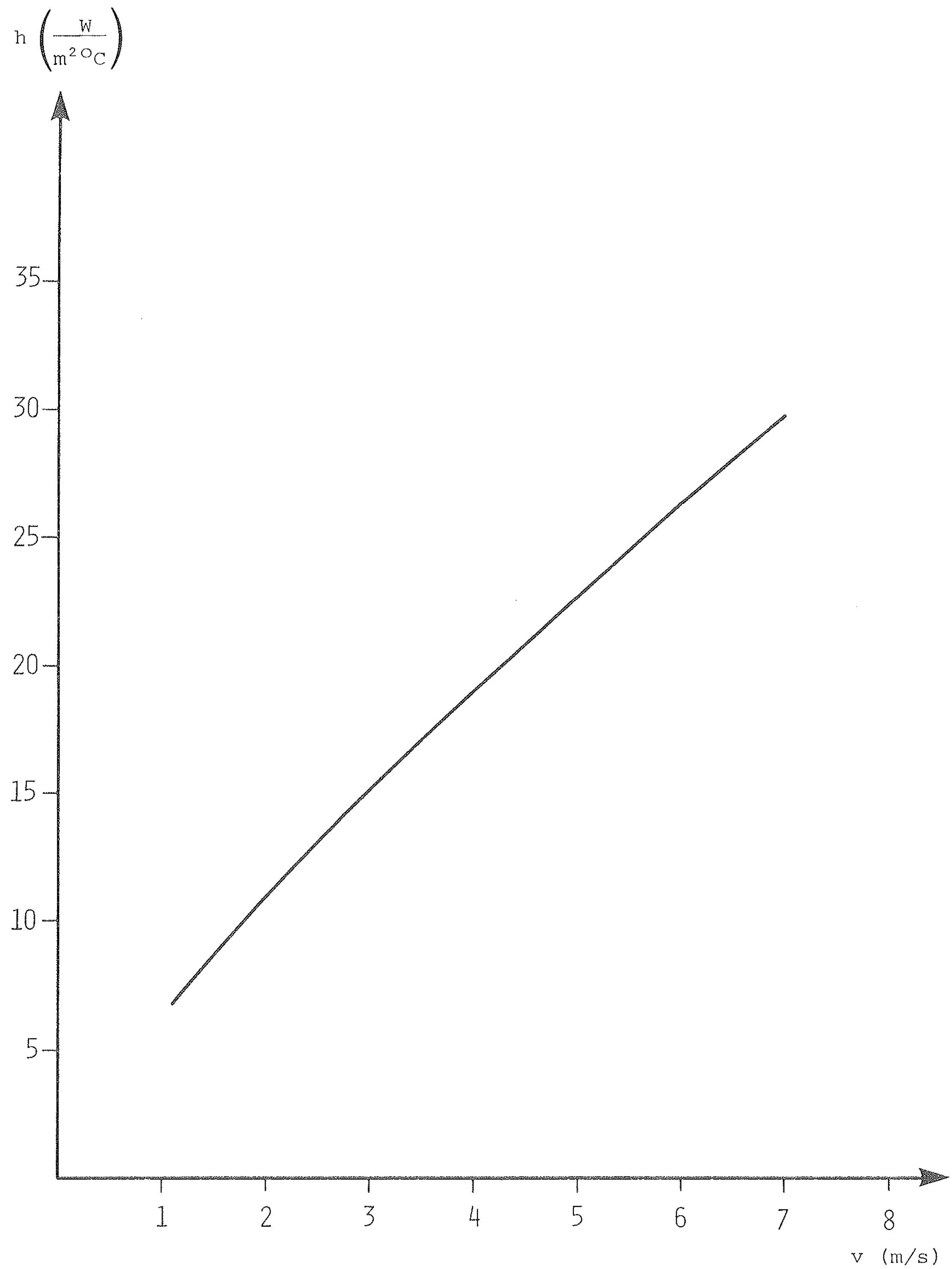


Fig. 4.2 The convective heat transfer coefficient between the circulating air and the containers in the heat storage unit as a function of the air velocity.

4.3 Comparison of the results of the calculation program with the results from the realistic testing

The measured and the calculated values for the room temperature, the storage temperature and the need for auxiliary energy for the three test rooms were compared for a period of two days, by using the measured outdoor temperature and the transmitted solar radiation in the computer program.

The heat loss coefficient of the rooms was calculated in accordance with [11]. By heating the rooms to approx. 30^{o}C during a night period with stable ambient temperature at approx. 0^{o}C, measured values for the heat loss coefficient were obtained as the ratio of the electricity consumption to the difference between the indoor and the ambient temperature, and these showed to be in good agreement with the calculated values. The heat capacity of the rooms was calculated as the heat capacity of the plywood board, gypsum boards and the chip boards on the inside of the insulation.

The measured outdoor temperature and transmitted solar radiation are shown in fig. 4.3, and the measured and calculated room temperatures and storage temperatures are shown in figs. 4.4 - 4.6. The measured and calculated needs for auxiliary energy and the corresponding energy savings compared to the reference room are listed in table 4.1.

It appears that the calculated total consumptions of auxiliary energy agree with the measurements as the deviations are less than 5%. The calculated room temperatures for the two rooms with a storage unit are up to 2-3^{o}C lower than the measured values during the hours with insolation. The reason for this may be that the heat losses from the rooms decreased during these periods due to a low wind velocity. This has influence especially on the losses through the windows and the losses caused by air infiltration. Further-

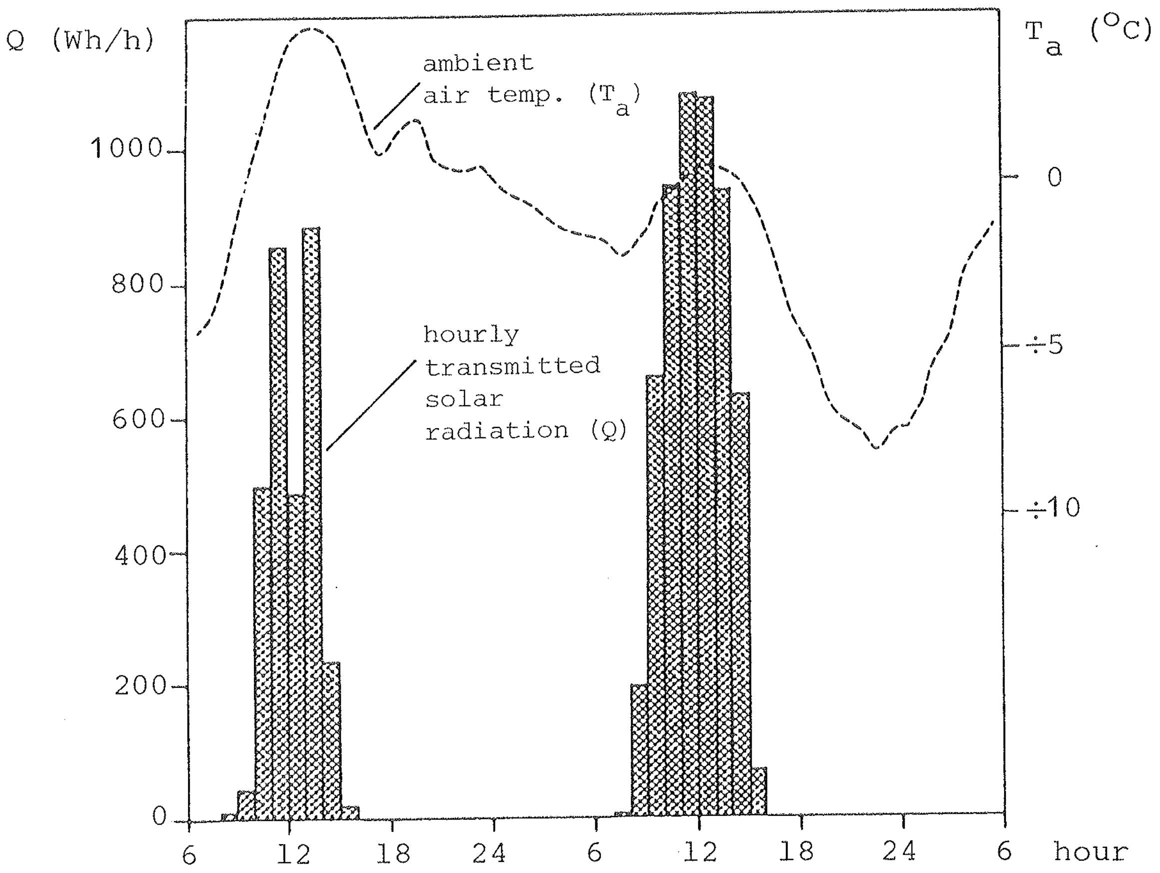


Fig. 4.3 Measured weather data for the period 12 nov 6 am - 14 nov 6 am

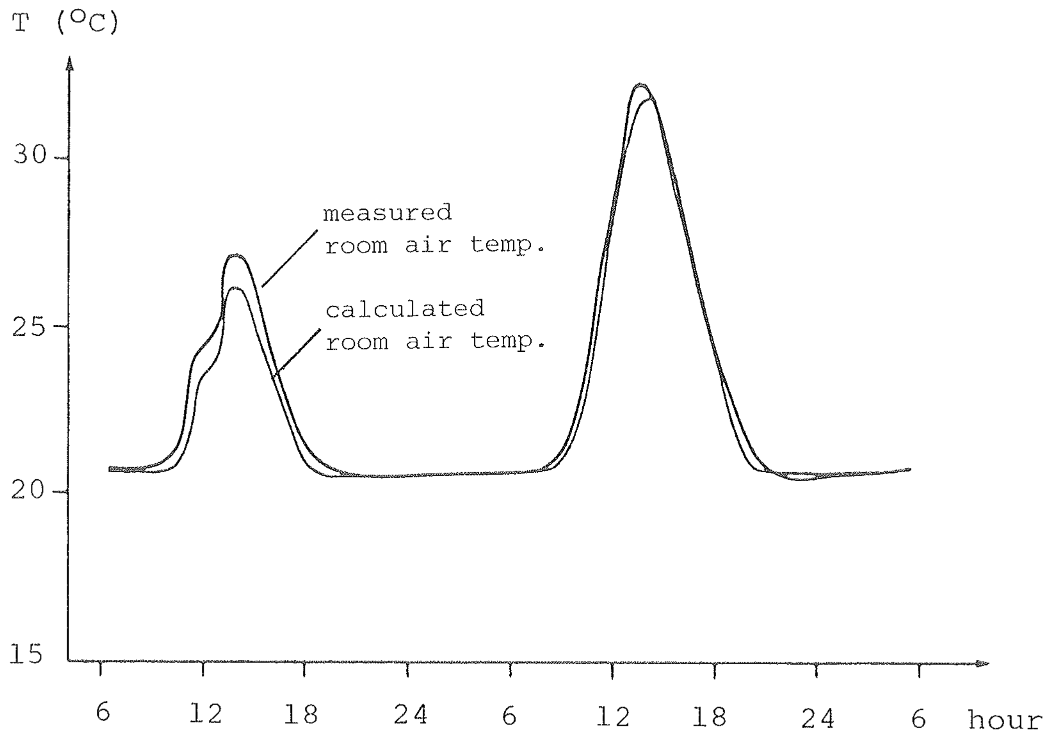


Fig. 4.4 Measured and calculated room air temperature for the reference room for the period 12 nov 6 am - 14 nov 6 am

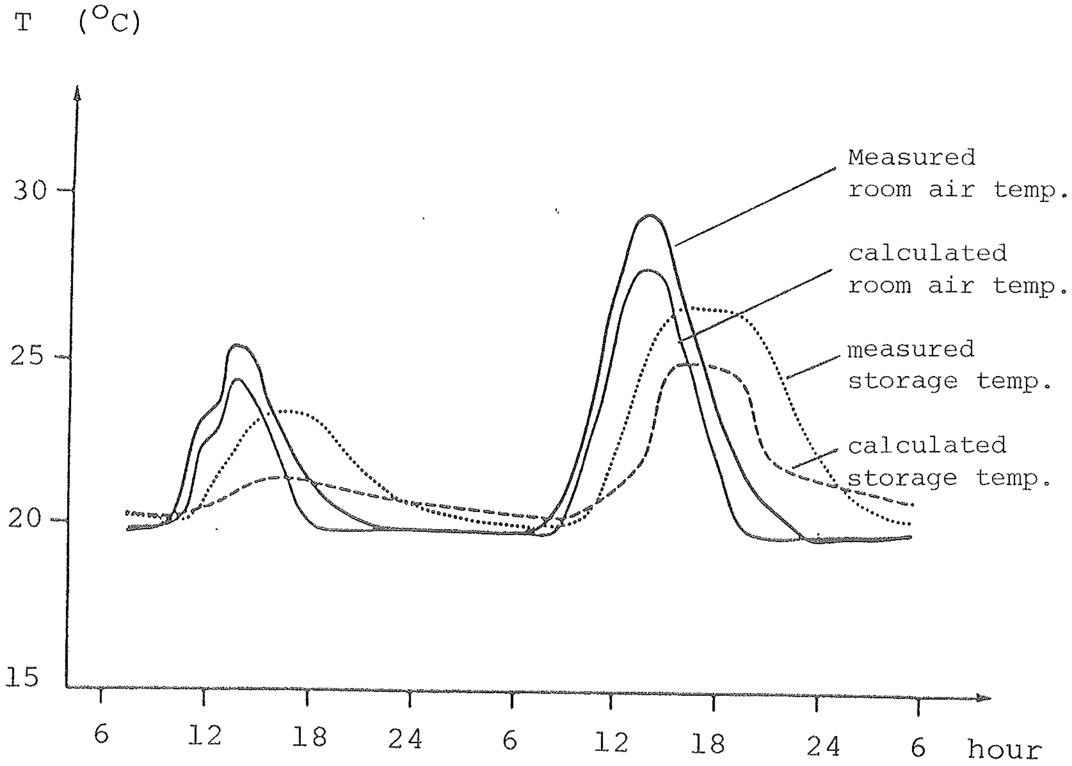


Fig. 4.5 Measured and calculated temperatures for the room with the storage unit having combined heat exchanger and storage containers for the period 12 nov 6 am - 14 nov 6 am

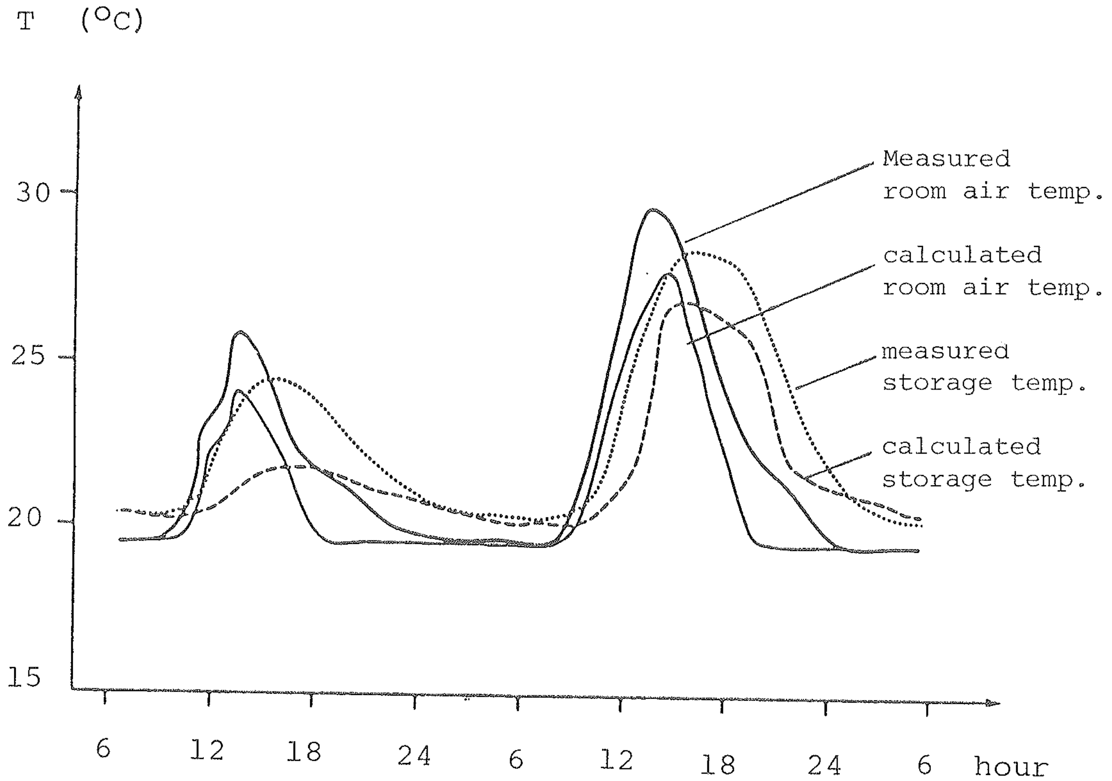


Fig. 4.6 Measured and calculated temperatures for the room with the storage unit having separate heat exchanger for the period 12 nov 6 am - 14 nov 6 am

	reference room		room with storage unit with combined heat exchanger and storage containers		room with storage unit with separate heat exchanger and storage containers	
	meas.	calc.	meas.	calc.	meas.	calc.
energy consumption of radiator (Wh)	9702	9246	5900	5385	3726	3765
energy consumption of ventilator (and pump) (Wh)	-	-	2542	2742	3814	3619
total consumption of electric energy (Wh)	9702	9246	8442	8127	7540	7384
energy savings compared to the reference room (Wh)	-	-	1260	1119	2162	1862
relative energy savings compared to the reference room	-	-	13.0%	12.1%	22.3%	20.1%

Table 4.1 Measured and calculated values for the energy consumption of the three test rooms during a period of two days (12/11 6 a.m. - 14/11 6.a.m.)

more, a low wind velocity will lead to that a lot of heat, originating from solar radiation absorbed by the roof, will be transmitted to the loft above the test rooms, thus affecting the heat loss through the ceiling. However, as the calculation model is very simple, especially with regard to the treatment of the heat capacity of the rooms, close agreement between the measured and the calculated temperatures cannot be expected.

The calculated storage temperatures are also up to 2-3^{o}C lower than the measured values. This seems rather much compared to the total fluctuation of the storage temperature. The shapes of the curves, showing the measured and calculated storage temperature, are different. The main reason is that the measured storage temperature represents the mean temperature of the liquid phase, whereas the program calculates the temperature in the zone where melting or crystallization takes place. Due to the stirring inside the containers not being very efficient, the temperature in the middle of the liquid phase will increase more rapidly than the temperature of the melting crystals at the bottom when the storage is charged. It should also be emphasized that the heat content of the storage is increased only slightly when all the salt has melted and therefore the heat contents, corresponding to the calculated and the measured maximum storage temperatures, do not deviate very much from each other.

As the calculated and measured energy consumptions of the test room correspond rather well, it is concluded that the calculation model is sufficiently suited for the simulations on yearly basis. The simulations are presented in the following.

4.4 The results of the calculations

The various calculations which have been carried out, are discussed in this section. To begin with a "reference case" was defined, in which reference values were fixed for the properties of the heat storage unit, the acceptable range of the room temperature and the characteristics of the room where the unit is placed.

The reference case

In the reference case a storage volume V of 252 litre (volume, at the temperature at which the salt hydrate will begin to solidify) was used. According to the assumptions in the previous section, a value of 290 W/^{o}C was calculated for the overall heat transfer coefficient between the salt-water mixture and the room air in case of the air being ventilated through the storage unit. The heat loss from the unit was assumed to be 1.7 W/^{o}C and the concentration of the $CaC\ell_2$ was set to 46%.

The minimum allowable room temperature was chosen to 20^{o}C, except for the night hours where 17^{o}C was specified. The maximum room temperature was set to 26^{o}C assuming that it is possible to keep this value by increased ventilation with ambient air (i.e. by opening a window). At least, this should be realistic within the heating season, which is from the beginning of September (11/9) to the end of May (29/5). Heat is not supplied by the radiators or the storage unit during the summer period.

The living room in a typical Danish single-family house from the sixties or early seventies, used in the calculations, is shown in fig. 4.7.

The floor area of the room is 32.7 m^2 and the ceiling height is 2.35 m. The facade facing south has a window area of 10.9 m^2 (transparent area 9.8 m^2) and the facade facing west has a window area of 1.3 m^2 (transparent area 1.2 m^2). The windows are furnished with two layers of glass. It is assumed that the insulation of the roof is up to todays

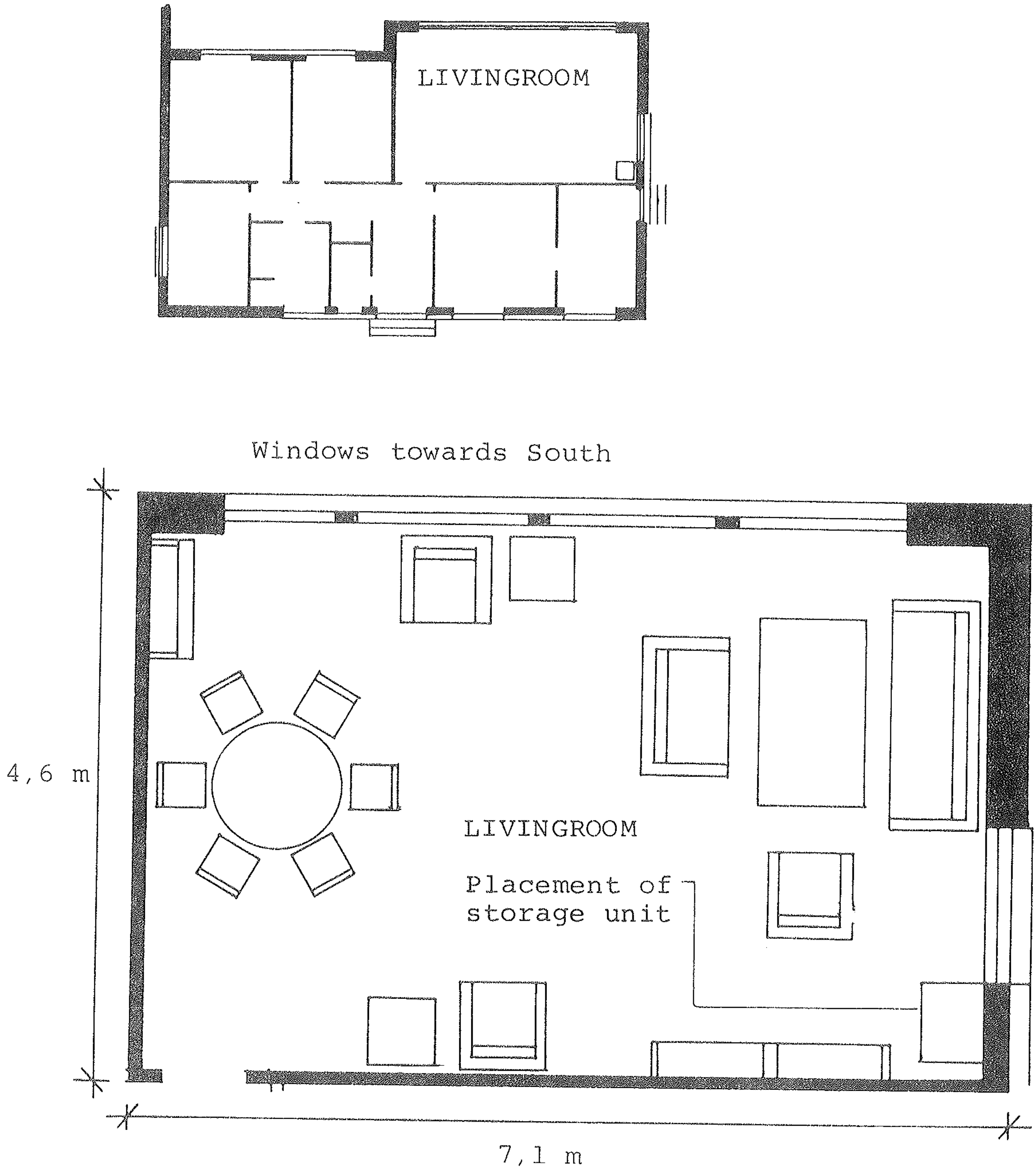


Fig. 4.7 Plans of the single-family house and the livingroom used in the calculations

standard, as an improvement of this insulation compared to the original standard, is a typical arrangement for this type of houses. The heat loss (in relation to the room air and the ambient air) for the room is 81 W/^{O}C, including the heat loss by ordinary air change which is 16 W/^{O}C.

The partition walls and the part of the outer walls which is on the inside of the insulation, are made from light weight concrete, the ceiling consists of gypsum boards and the flooring is parquet with insulation immediately beneath. The heat capacity of the materials used in the building structure is therefore very modest.

The emission of heat from lighting, electric devices and persons is assumed to be between 2500 and 2950 Wh depending on the time period for the light to be switched on. (The light is switched on when the total solar radiation through the windows is less than a certain value). All this heat is supplied during the late afternoon and the evening when the room is supposed to be used by the occupants.

For the reference room with and without any storage unit, the calculated annual sums of the different flows of energy are listed in table 4.2, and in fig. 4.8 the monthly distribution of the need for auxiliary energy is shown. In fig. 4.9 the total amount of solar energy transmitted through the windows and the utilized part of this energy, (i.e. the amount collected by the storage unit) are compared on monthly basis.

Room with storage unit		
energy to the storage unit	696	kWh/year
energy from the storage unit	689	-"-
net heat loss from the storage unit	7	-"-
energy used by the ventilator	169	-"-
energy supplied by the radiators	2630	-"-
Room without storage unit		
energy used by the radiators	3389	kWh/year

Table 4.2 Yearly sums of the energy flows with regard to the reference room with and without storage unit

The energy savings Q_s for the room are calculated from:

$$Q_s = Q'_r - Q_r - Q_v$$

and the relative savings p are found from:

$$p = \frac{Q_s}{Q'_r}$$

where

Q'_r = the need for energy from the radiators for the room without the storage unit

Q_r = the need for energy from the radiators for the room with the storage unit

Q_v = the energy consumption of the ventilator

The energy savings in the reference case is

$$\underline{Q_s = 590 \text{ kWh/year}}$$

corresponding to

$$\underline{P = 17.4\%}$$

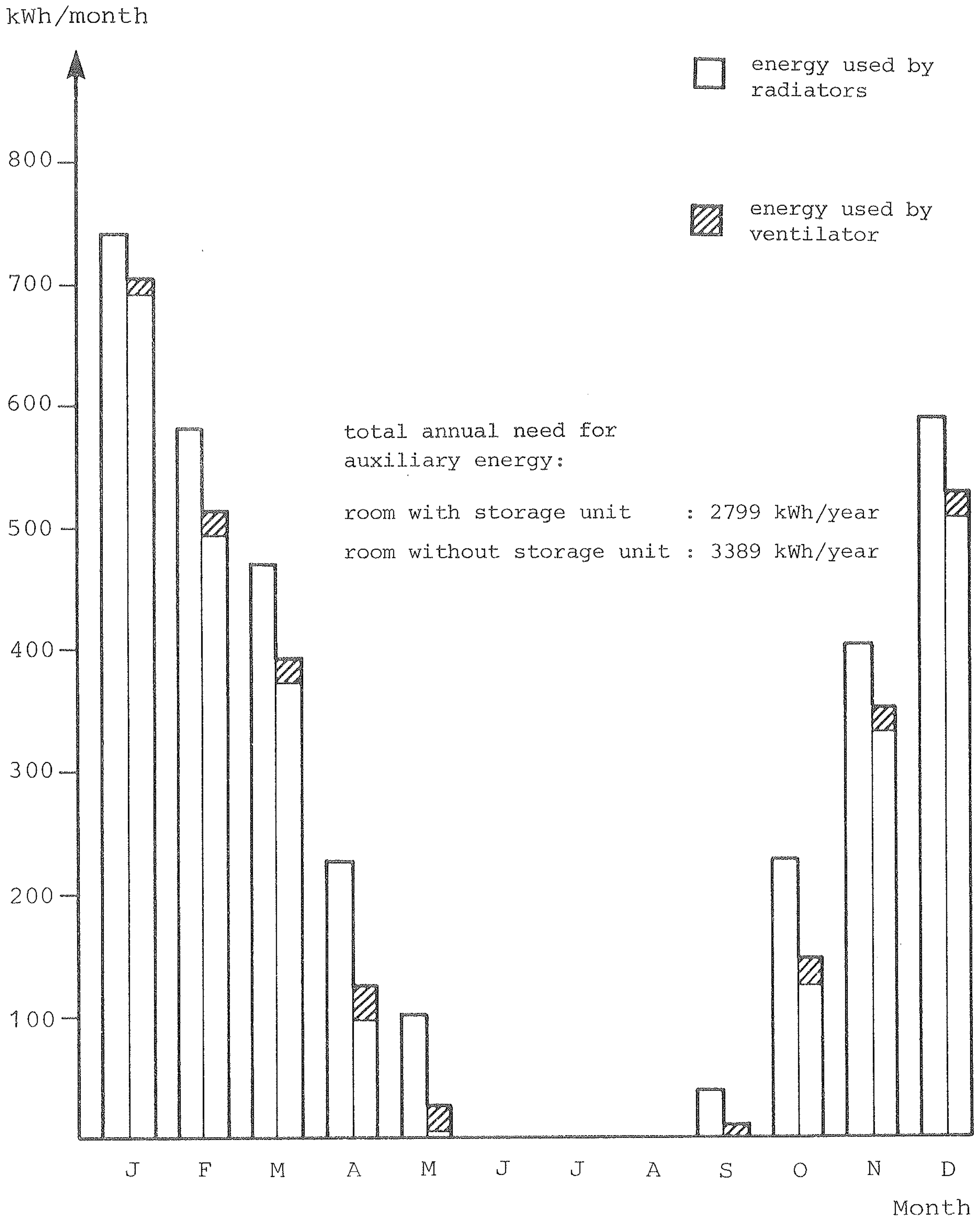


Fig. 4.8 Need for auxiliary energy of the room with and without the storage unit (the left and the right column respectively) on monhtly basis

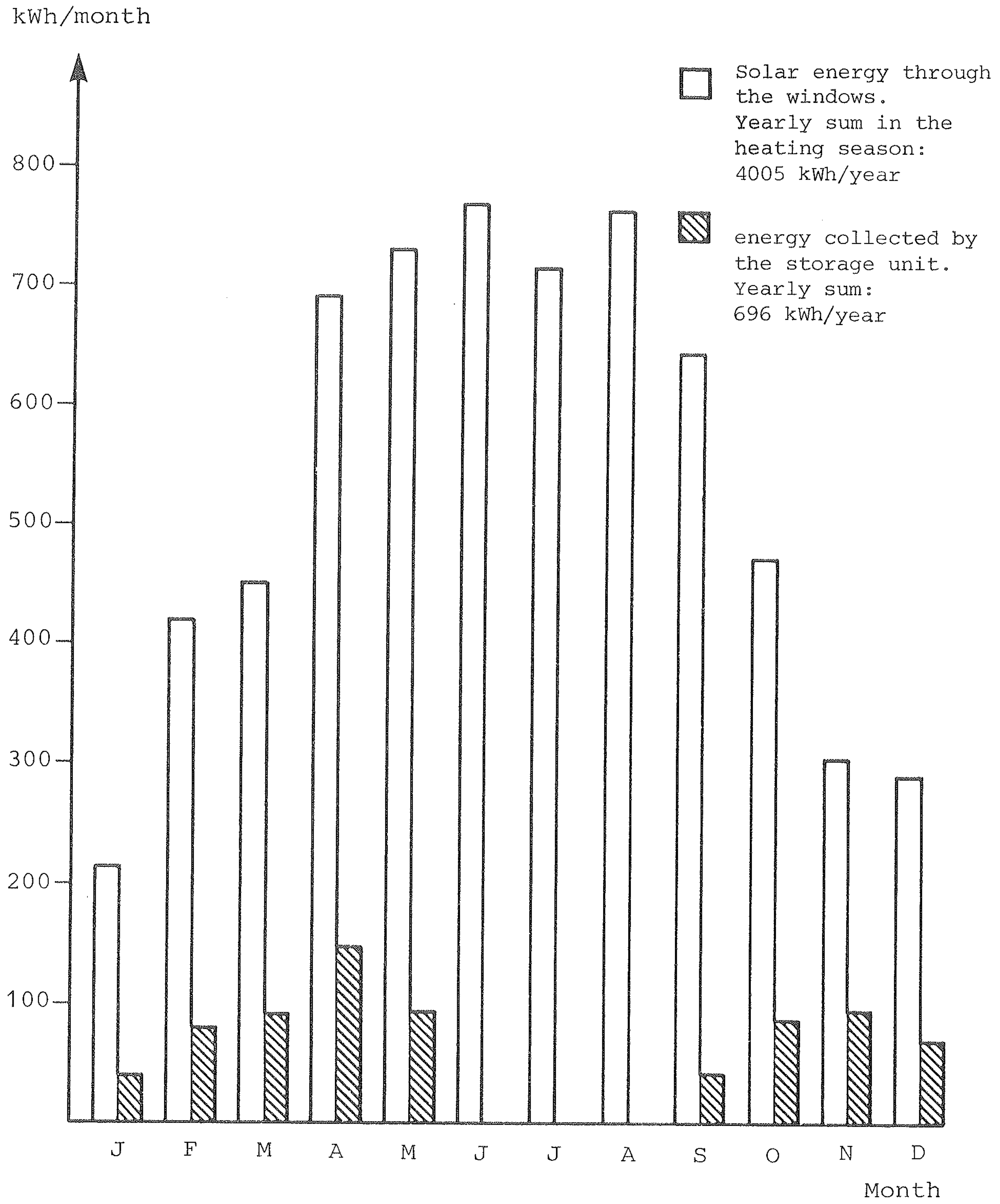


Fig. 4.9 Comparison of the total amount of solar energy transmitted through the windows and the energy collected by the storage unit on monthly basis

From table 4.2 it appears that the net heat loss from the storage unit is neglegible. The reason for this is that the heat transmitted through the sides of the unit during periods where the room temperature is higher than the storage temperature, is taken into account. When the half-hour values of the different heat flows were considered, the heat loss to and from the storage unit generally showed to be of very little importance because of the small heat loss coefficient of the storage unit and the rather small differences between the temperature of the storage unit and the temperature of the room air.

The annual amount of energy delivered to the room from the storage unit, the radiator and the ventilator is a little larger than the energy delivered from the radiators to rooms without a storage unit. This is partly because some of the energy from the ventilator is delivered during periods where the storage unit is being charged, and as there is no heating demand, part of this energy will not be utilized (if the room temperature is at its maximum allowable value, it will all be wasted). Another reason is that the room temperature will reach high values more frequently in a room without storage unit and therefore a little more heat will be stored in the building elements when comparing with a room with a storage unit; i.e. the thermal capacity of the room is utilized better.

From fig. 4.9 it appears that 17% of the transmitted solar radiation is transferred to the storage unit when the radiation transmitted during the summer period is not considered.

The parameter variations

A survey of the parameter variations carried out is given below:

A) The influence of the maximum allowable room temperature and of the lowering of the room temperature at night is investigated.

B) The volume of the storage medium is varied.

C) The overall heat transfer coefficient to and from the storage unit is varied.

D) A salt-water mixture with 43.6% $CaCl_2$ is used instead of the 46% solution.

E) The window area of the room is varied.

F) The heat loss from the room is varied.

G) The influence of the thermal mass of the room is investigated.

Usually the results are compared with the need for auxiliary heat in a room without a local storage unit and with the same maximum and minimum allowable temperature, the same window area, heat loss and heat capacity. The energy savings Q_s and p are calculated as mentioned before.

Tables showing the yearly sums of energy consumed by the ventilator and radiators, the heat supplied to and from the storage unit and the heat loss from the unit are included as appendix II. Each parameter variation is shown and discussed in the following. The results are usually shown by curves on which the reference case is marked by a small circle.

A) The influence of the limits of the room temperatures

The amount of heat transferred to the storage unit during the hours where the unit is being charged, is increasing if the difference between the temperature of the salt-water mixture and the room temperature increases. On the other hand, temperatures should be kept below approx 26 °C because of the thermal comfort for the occupants of the room. However, the room temperature will not be limited to the specified upper limit during periods when no occupants are at home or in cases where the extraordinary air change by opening of windows is not enough to provide the necessary cooling effect. In order to investigate these uncertain factors connected to the fixing of the upper limit of the room temperature $T_{r,max}$, this was varied between 25°C and 30°C.

The savings of auxiliary energy for different values of $T_{r,max}$ are illustrated in fig. 4.10. It is noticed that the savings are almost the same when $T_{r,max}$ is between 26°C and 30°C; when $T_{r,max}$ is higher than approx. 28°C the savings tend to drop slightly, the reason being that the air temperature only very seldom will be above 28°C in the room with the storage unit, whereas this will be the case more frequently in the room without storage unit. Thereby the extra heat stored in the thermal mass of the building in the latter case will exceed the additional heat transferred to the unit in the first case. When $T_{r,max}$ is lowered to below 25-26°C, the energy savings will clearly decrease.

It is concluded that a room temperature of at least 26°C must be accepted in the room during periods when the storage unit is charged, and that the savings are practically unaffected by the room temperature sometimes being higher.

The consequences if the minimum room temperature during the night is different from 17°C was investigated by varying this value between 11°C and 20°C. The results are shown in fig. 4.11. It appears that when the minimum room temperature is not lowered during the night, the energy savings will be somewhat smaller which has to do with the storage capacity between 17°C and 20°C not being utilized. Also when the temperature is lowered further than to approx. 17°C, the savings will decrease even though the storage capacity is increased. In this case the heating requirement during the period between two chargings of the storage on two consecutive days will often be so limited that a large part of the energy collected during the first day cannot be used during the night.

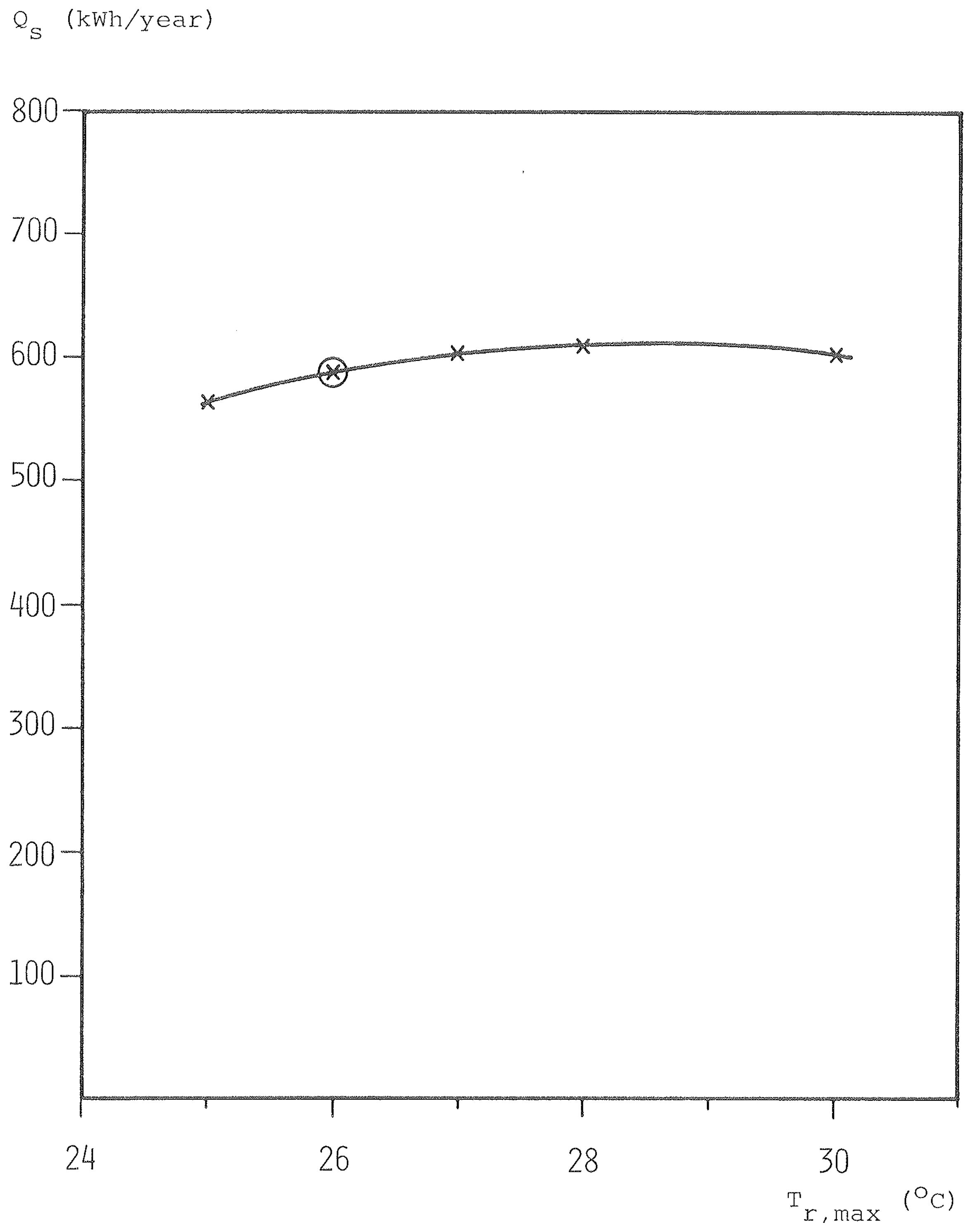


Fig. 4.10 The influence of the maximum room temperature on the yearly savings of energy

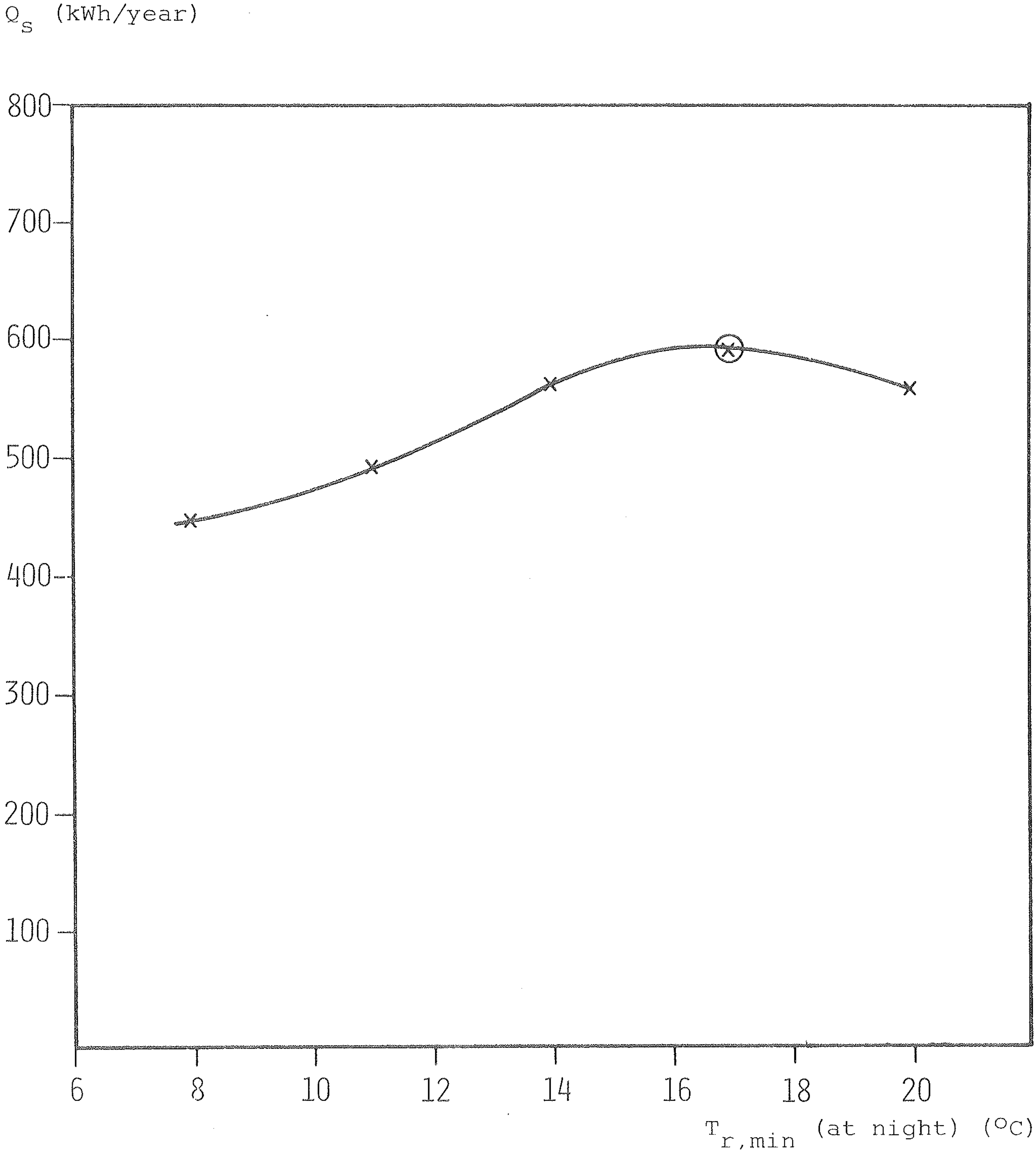


Fig. 4.11 The influence of the minimum room temperature during the night on the energy savings

B) Variation of the volume of the storage medium

An increase of the volume of the storage medium (by increasing the number of containers) has several advantages. The increased container surface area will cause an improved heat transfer between the room air and the unit. The larger heat capacity will result in a slower rise of the storage temperature during charging providing a further improvement of the heat transfer to the unit. Also the number of days, during which the storage temperature equals the room temperature while there is still an incoming solar radiation that could be utilized, is reduced.

The influence of the storage volume on the savings is illustrated in fig. 4.12. The influence of the heat transfer on the result is also shown by varying the storage volume and keeping the overall heat transfer coefficient on the reference value.

If the modification of the heat transfer coefficient is taken into account, it shows that the energy savings caused by the storage unit will increase significantly in all the intervals from the smallest to the largest of the considered volumes. If, for example, a volume of 378 litre (54 containers in each layer) is used instead of the reference value of 252 litre, the savings will increase from 590 kWh/year to 707 kWh/year. A small unit with 126 litre (18 containers in each layer) will result in an energy saving of only 393 kWh/year. If the heat transfer coefficient is fixed, the variations are more modest, especially for storage volumes larger than in the reference case. For storage volumes of 393 litre and 126 litre, the savings will be 638 kWh/year and 471 kWh/year respectively.

C) Variation of the heat transfer coefficient between the storage unit and the room air

The heat transfer coefficient h between the air in the storage unit and the storage medium is determined with some

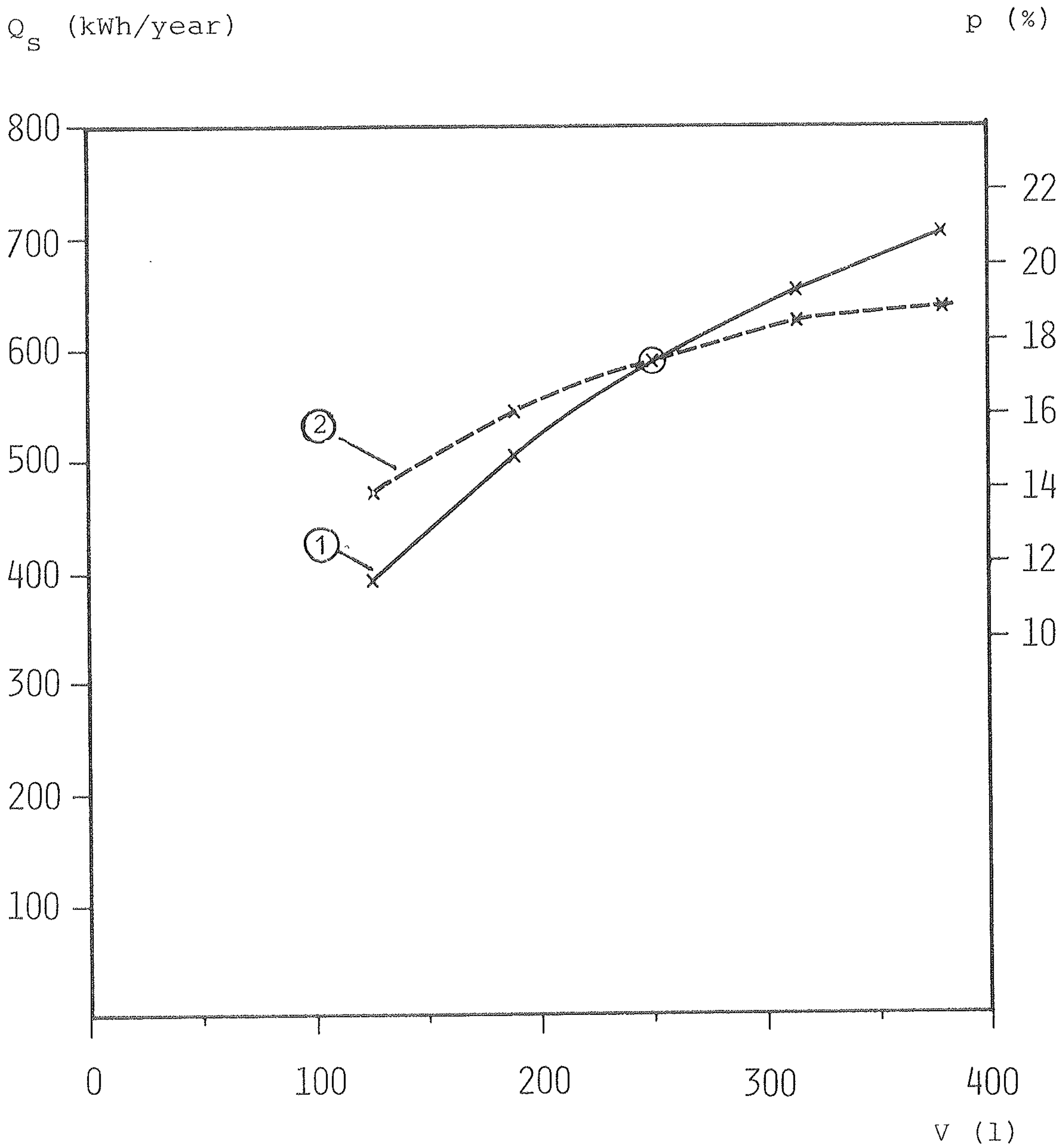


Fig. 4.12 The influence of the storage volume on the savings of auxiliary energy. In case (1) the overall heat transfer coefficient is also changed. In case (2) it is fixed at the reference value

uncertainty. The formula used in order to find h from the velocity v is based on the air flow between two parallel plates, which is not strictly the case, and the calculation of v includes some assumptions regarding the pressure drop. Furthermore, it is assumed that the heat transfer inside the containers is so considerable that temperature gradients within the salt-mixture can be ignored.

In fig. 4.13 is shown the savings' dependence on the overall heat transfer coefficient $(UA)_s$ between the room air and the storage medium.

If, for example, h is 25% larger, $(UA)_s$ will increase with 17% (according to equation (5.5)) resulting in an increase of the energy gain from 590 kWh/year to 624 kWh/year. If h is reduced 25%, $(UA)_s$ will decrease with 20% resulting in a decrease of the gain to 539 kWh/year. Some additional calculations showed that a variation of the heat transfer coefficient, in case of charging, has a much stronger influence on the results than when it is changed only in case of discharging.

It is evident that the energy savings are strongly influenced by a reduction or increase of h, and consequently efforts must be made to ensure as high a value of h as possible.

D) A variation in concentration

The 43.6% $CaCl_2$ was used in the storage units constructed for the experiments. When using the data for the relation between the temperature and the heat content for this concentration, the savings show to be 548 kWh/year, which is 42 kWh/year lower than for the 46% concentration. The reduction seems rather moderate considering that the heat content per unit volume (as mentioned in chapter 2) is more than twice as big for the 46% concentration in the temperature interval 20-26°C. On the other hand it must be considered that also the heat capacity between 17°C and 20°C is utilized during the night, and between 17°C and

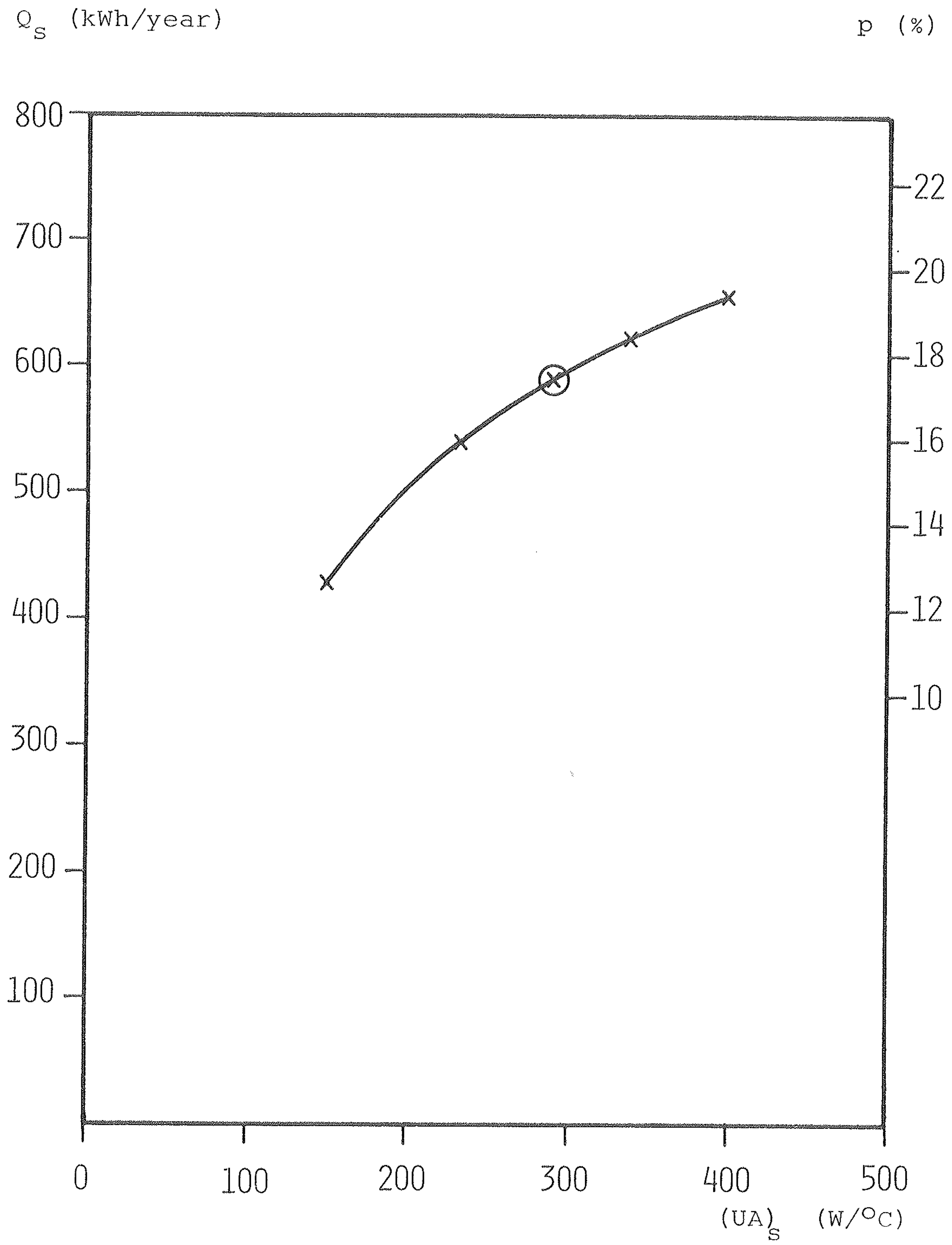


Fig. 4.13 The savings of auxiliary energy for different values of the overall heat transfer coefficient to and from the storage unit

$26^{\circ}C$ the 46% concentration contains only 1½ times as much heat as the 43.6% concentration.

Furthermore, the temperature of the storage will often only reach 22-23$^{\circ}C$ after one day with heat supplied to the storage, and the heat content between 17$^{\circ}C$ and 23$^{\circ}C$ is larger for the 43.6% $CaCl_2$ than for the 46% $CaCl_2$.

E) Variation of the window area

The window area of the reference room (12.2 m^2) is typical for the building design of the sixties, but at the same time many houses from the sixties and especially from the seventies are constructed with a smaller window area.
In fig. 4.14 the window area in the wall facing south is varied between the value in the reference case and what corresponds to half of this. At the same time the heat loss from the house is modified, corresponding to the changed areas of the window and the wall.

It shows that the savings of energy due to the storage unit decreases with 225 kWh/year down to 282 kWh/year in case of the smallest window area. On the whole it does not look as if the window area should be much smaller than in the reference case if the installation of a storage unit is to be worthwhile.

F) Variation of the heat loss from the room

The influence of the heat loss from the room on the ambient air is investigated. Since the sum of the heat loss through the windows and the heat loss due to the natural air change amounts to 66% of the total heat loss, and these contributions are kept constant during the calculations, the changes of the heat loss due to different thicknesses of the insulation in the walls, ceiling and floor, is relatively small.
In fig. 4.15 and the heat loss per $^{\circ}C$ between the room air and the ambient air is varied between what is assumed to be the lowest and the highest realistic value for the considered type of house.

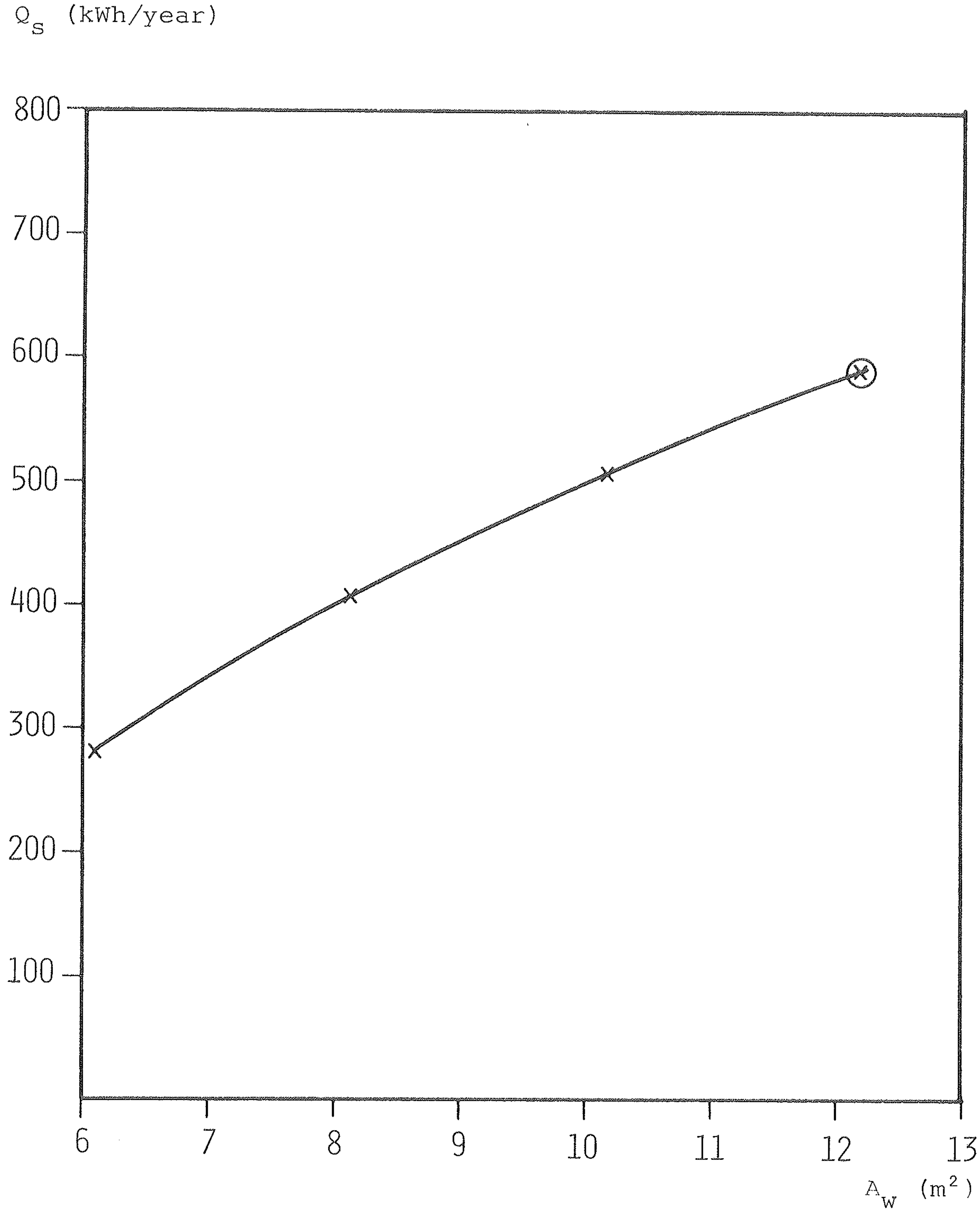


Fig.4.14 The influence of the total window area on the savings of auxiliary energy

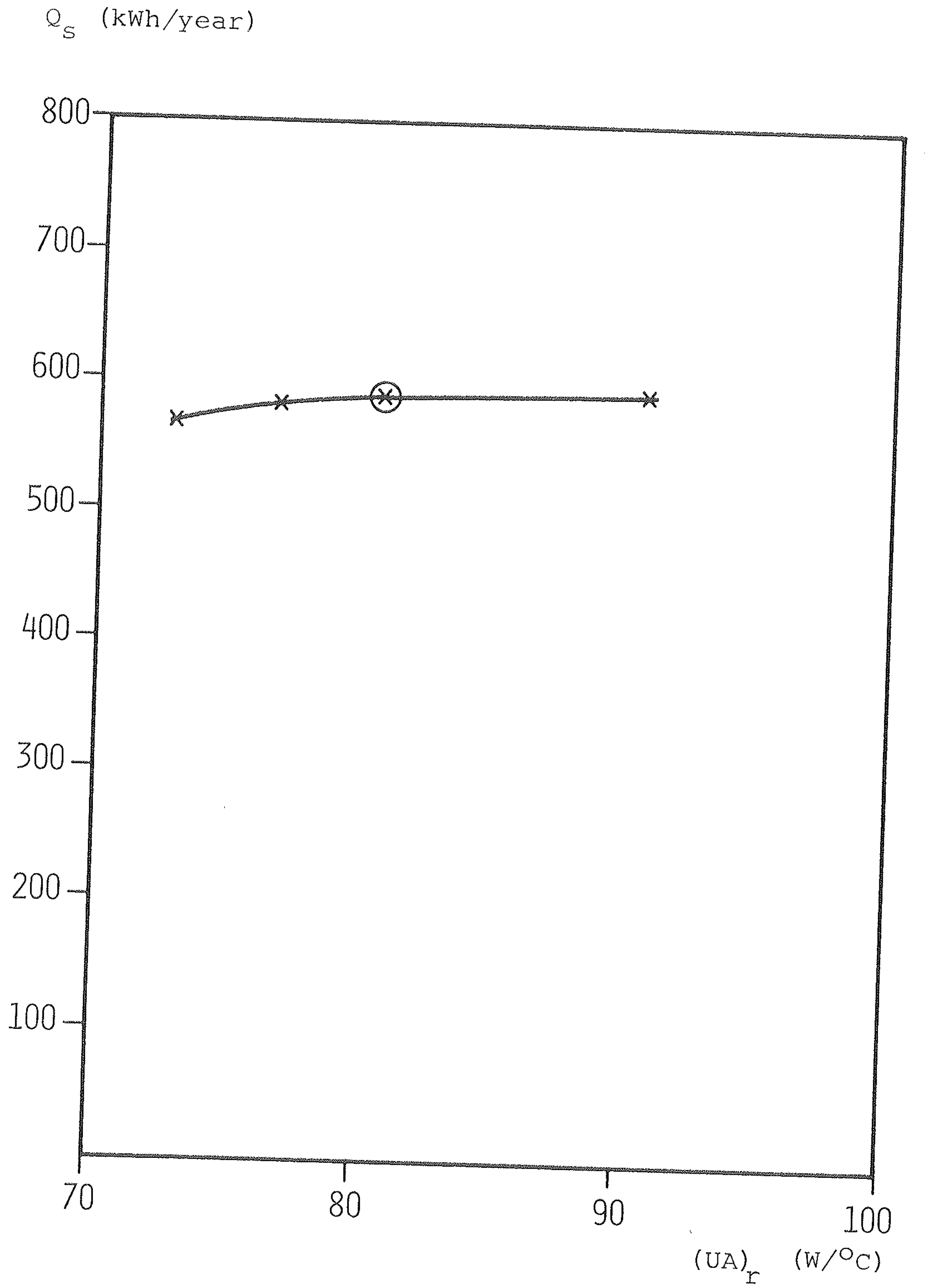


Fig.4.15 The influence of the heat loss from the room on the savings of auxiliary energy

It must be concluded that within this range of heat loss, the savings of energy is reduced only slightly if the insulation of the room is less than in the reference case. An improvement of the insulation leads to a negligible increase of the savings. With a reduction of the heat loss, the storage unit will not always be discharged during the night between two periods (on consecutive days) with charging of the storage and consequently all the heat stored during the first day is not utilized. On the other hand, the reduced heat loss will cause a larger surplus of heat on days with a large quantity of solar radiation and this will increase the rate of heat transfer to the storage unit. Therefore, the two mentioned consequences will tend to counteract each other with the result that the savings are almost unaffected by the insulation level.

G) The influence of the thermal mass of the room

It is assumed that the heat capacity of the walls, ceiling and floor in the reference room is small (totally 667 Wh/^{o}C). The consequences, if this condition is not fulfilled, was investigated by specifying other building materials. Calculations were carried out in the below mentioned four cases; one room has an extremely small heat capacity (case 1) and the other rooms have a larger heat capacity than the reference room.

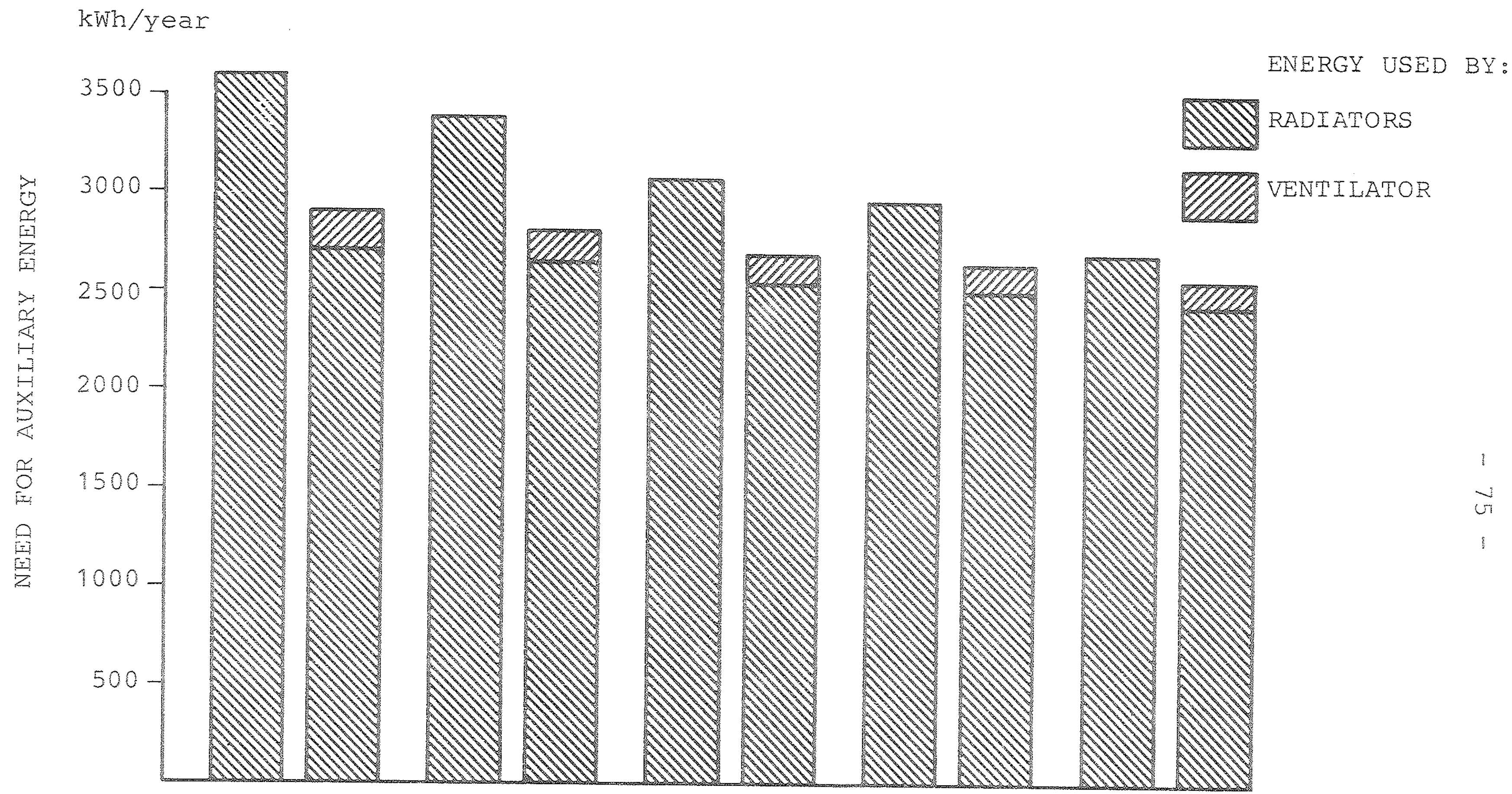
1) Boards of gypsum are fitted to the outer walls (on the inside of the insulation), to the partition walls and the ceiling. Insulation is placed directly under the parquet flooring.

2) Lightweight concrete is used for the outer walls and the partition walls. The ceiling is made from boards of gypsum. Immediatly under the parquet flooring is a concrete slab.

3) The outer walls and the partition walls consist of bricks. Boards of gypsum are fitted to the ceiling. Immediately under the flooring is a concrete slab.

4) Concrete is used for the outer walls, the partition walls and the ceiling. A concrete slab is immediately under the flooring.

The energy savings in the different cases are shown in fig. 4.16, where also the quantity of heat capacity is stated. From the figure it is evident, that the room has to be very "light" if the storage unit is to be of any advantage. If, for example, a heavy concrete slab is placed directly under the parquet flooring, the savings caused by the storage unit will decrease with 196 kWh/year to only 394 kWh/year, mainly due to the decreased energy consumption of the room without a storage unit. For an extremely heavy room (case 4) the yearly savings are only 138 kWh or about one fourth of the savings in the reference case.



	CASE I	REFERENCE CASE	CASE II	CASE III	CASE IV
THERMAL MASS (Wh/°C)	450	667	1390	1930	4170
ENERGY SAVED (kWh/year)	697	590	394	309	138

Fig. 4.16 The demand for auxiliary energy for different building designs. The left column in each case is for a room without and the right column is for a room with a local storage unit

5. DEVELOPMENT OF THE STORAGE UNIT

The computer calculation in chapter 4 showed that the overall heat transfer coefficient between the room air and the heat storage medium is a very critical parameter; and the heat transfer coefficients obtained for the two storage units, constructed for the realistic testing, showed not to be sufficiently large. It was therefore proposed to improve the heat transfer for that type of storage where the room air is ventilated along the storage containers, by using smaller containers and in this way increase the heat transferring area. In the computer calculations it was assumed that small one-litre containers were used, and the heat transfer coefficient was determined according to some theoretical assumptions.

A third storage unit was constructed for testing at the Laboratory in order to demonstrate that the acceptable heat transfer, used in the reference case, could be obtained in reality. Further, it was decided to investigate the dependence of the heat transfer coefficient on some important parameters by theoretic calculations.

5.1 Theoretical investigations concerning the overall heat transfer coeffcient

The heat transfer coefficient $(UA)_s$ from the room air to the storage during charging and discharging, in the case where the heat exchanger is the surface of the storage container, were calculated in the previous chapter. The horizontal distance between the containers (d), and thereby the hydraulic diameter, the loss coefficient for the minore pressure losses (k) and the performance of the ventilator ($P_1 = P \cdot \eta$, where P is the energy consumption and η is the efficiency) were fixed at certain values. In connection with the realization of a storage unit for which $(UA)_s$ is sufficiently high, it is of interest to investigate the influence of these parameters. In this section $(UA)_s$ is calculated by the method explained in section 4.2, and it is assumed that 252 box-shaped one-litre containers are arranged in the same way as described in section 4.2 (with the exception that d can be varied).

The calculations are carried out for two different assumptions concerning the minor pressure loss in the horizontal air space between two layers of containers. In the first case (I) it is assumed that the air flow takes place with negligible minor losses at this location, i.e. they are put to zero as in the previous calculation. In the second case (II) it is assumed that half of the kinetic energy of the air is lost at each passage between two layers of containers (giving $k = 0.5$ at each passage) which must be considered as an overestimation of the minor losses. The different values obtained for the Reynold number were generally high enough to provide turbulent flow, but as the ratio of the hydraulic diameter to the length of the duct is rather small (which is the case especially for the second assumption concerning the pressure drop), attention must be paid to the influence of the entrance region. This is done by multiplying the Nusselt number from equation (4.10) with the following factor c (according to page 133 in [10]:

$$c = 1 + (d_h/L)^{0.7}$$

where d_h is the hydraulic diameter and L is the height of the seven layers of containers in case I and the height of the individual container in case II.

The dependence of $(UA)_s$ on P_1 is illustrated in fig. 5.1. The dependence of $(UA)_s$ on d is shown in fig. 5.2 (pressure drop in accordance to case I), and in fig. 5.3 (case II) for some realistic values of P_1. The dependence on P_1 is of course very significant. If, for example, P_1 is only half as big as in the reference case, where P_1 is set to 22 W, and the pressure drop is calculated

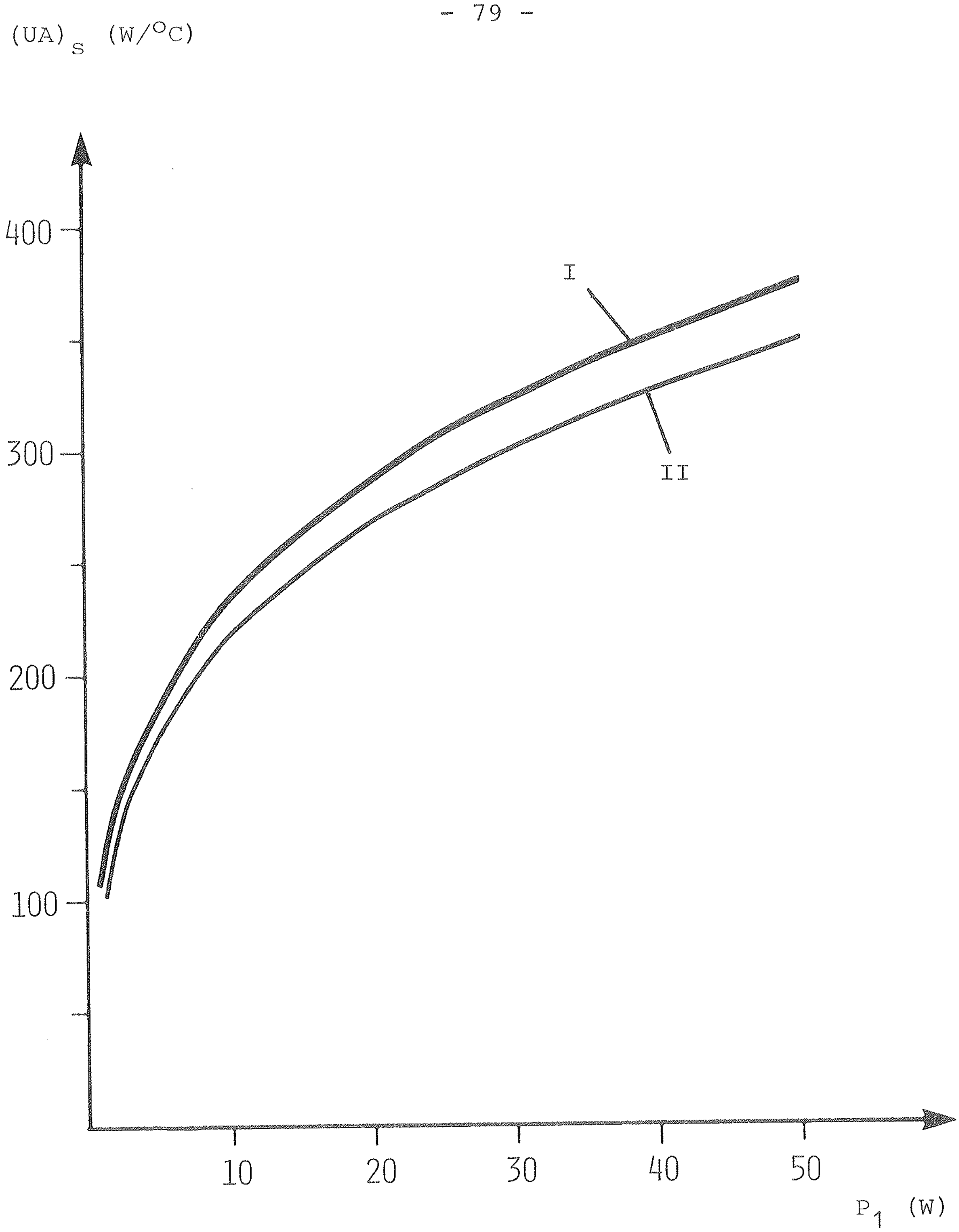


Fig. 5.1 The overall heat transfer coefficient as a function of the performance of the ventilator in case that minor losses are rather small (I) and in case they are rather large (II). The distance between the containers are 0.015 m. ($P_1=P\cdot\eta$ where P = power supply to the ventilator and η = efficiency of the ventilator).

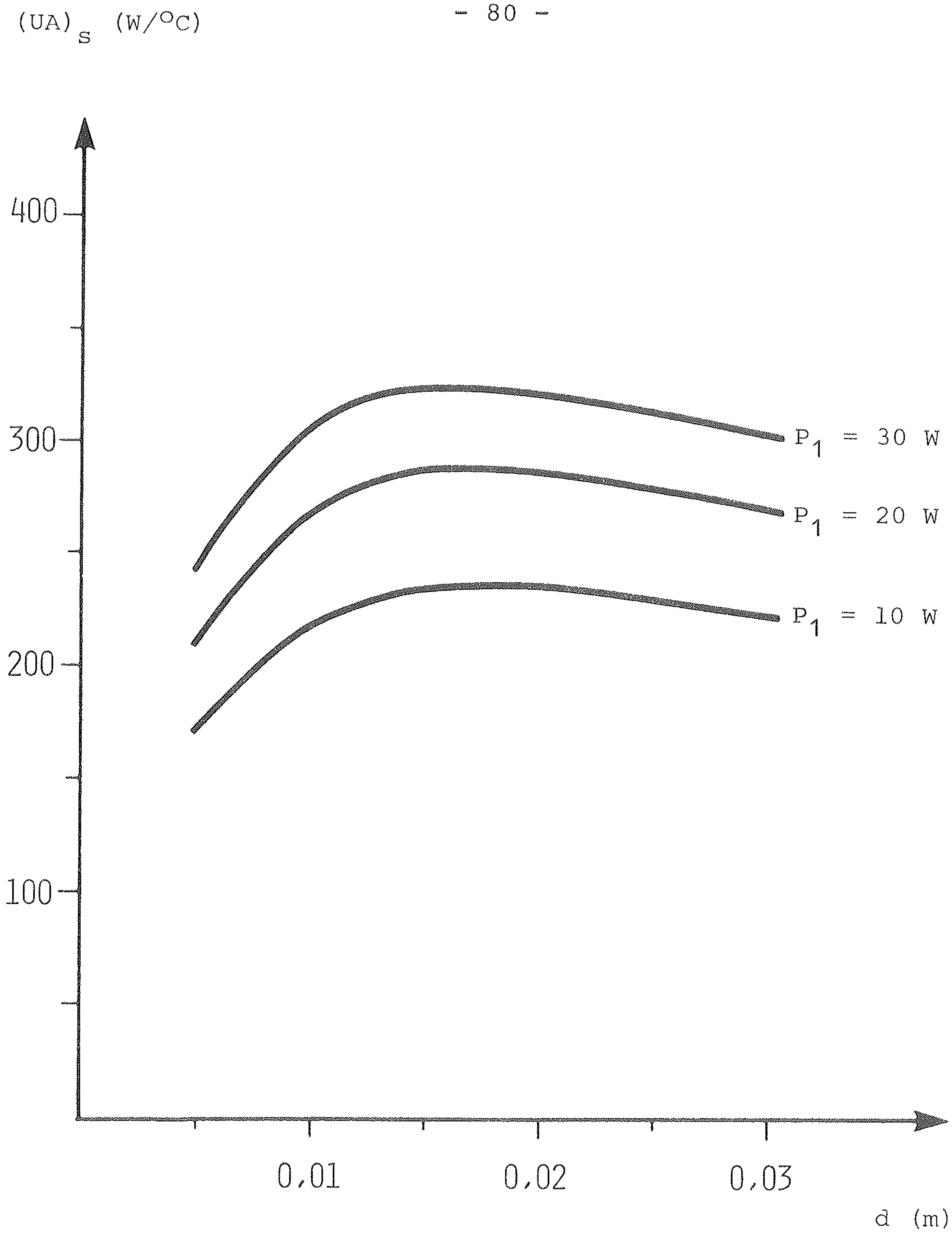


Fig. 5.2 The overall heat transfer coefficient as a function of the distance between the containers (d) for different values of the ventilator performance (P_1) in case that minor losses are rather small. ($P_1 = P \cdot \eta$, where P = power supply to the ventilator and η = efficiency of the ventilator).

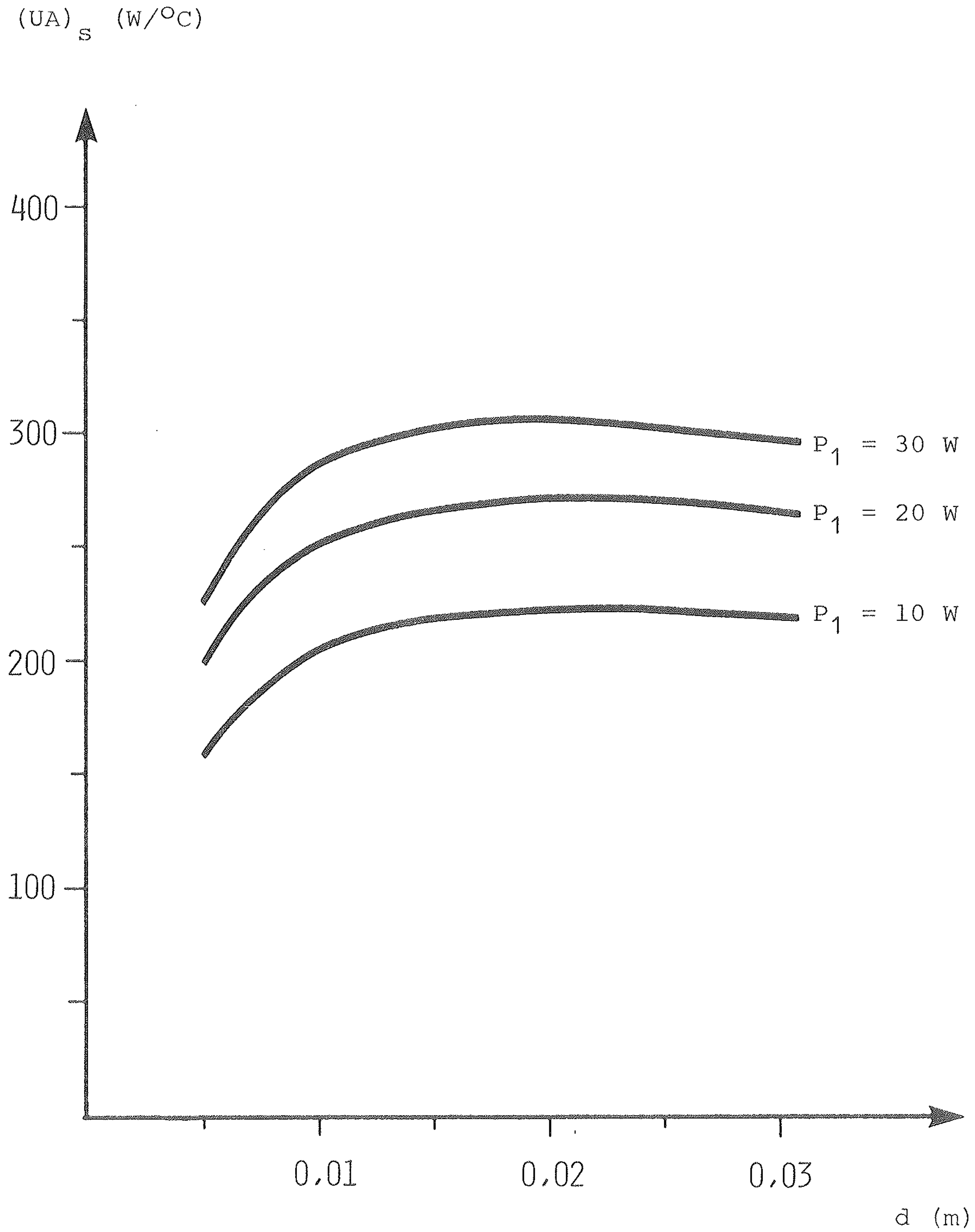


Fig. 5.3 The overall heat transfer coefficient as a function of the distance between the containers (d) for different values of the ventilator performance (P_1) in case that minor losses are rather large. ($P_1 = P \cdot \eta$, where P = power supply to the ventilator and η = efficiency of the ventilator).

according to case I, $(UA)_s$ will decrease with about 20%. Generally, the values of $(UA)_s$ is decreased only slightly if the pressure drop is calculated according to case II instead of case I. This is because the effect of the decreased velocity in case II will be partially compensated by an increase of the correction factor c on the Nusselt number. It does not seem possible to increase $(UA)_s$ by choosing another distance than 0.015 m between the containers since the value of $(UA)_s$ obtained in case of this distance in general is very close to the optimum. If d is less than approx. 0.01 m, the obtained mass flow rate will be too small giving a significant decrease in $(UA)_s$. When d is between 0.015 m and 0.03 m, the variation of $(UA)_s$ is modest. In this interval the decrease of the velocity, due to an increase of d, will cause a reduction of the convective heat transfer number between the surface of the containers and the air in the duct, but at the same time the mass flow rate will be increased.

5.2 Construction of an improved storage unit

The design of the improved storage unit corresponds to a large degree to the reference unit described in chapter 4. The unit is illustrated by drawings in figs. 5.4 and 5.5 and a photograph is shown in fig. 5.6. 252 one-litre containers were utilized, but as a small air space had to be left for expansion, only 235 litre (volume at 26°C where all the salt hydrate has melted) were filled into the containers. The outside measurements of the containers are 45 mm x 120 mm x 215 mm. The containers were placed in seven layers with 36 containers in each layer. They were arranged on "shelves" of thin aluminum plates with a distance of 15 mm between the containers, in both horizontal and vertical direction. The sides of the boxes are made of insulating plates consisting of approx. 25 mm polyurethane foam, laminated between two thin layers of plywood. The ventilator is a propeller type and according to the manufacturer, the power consumption is 70 W. The ventilator was placed at the top of the unit and was equipped with a louver damper which opens automatically when the ventilator is starting and closes when the ventilator is stopping. This way reverse flow should be prevented during periods when there is no heat demand and the storage is charged. The outside measurements of the storage is 0.60 m x 0.60 m x 2.0 m.

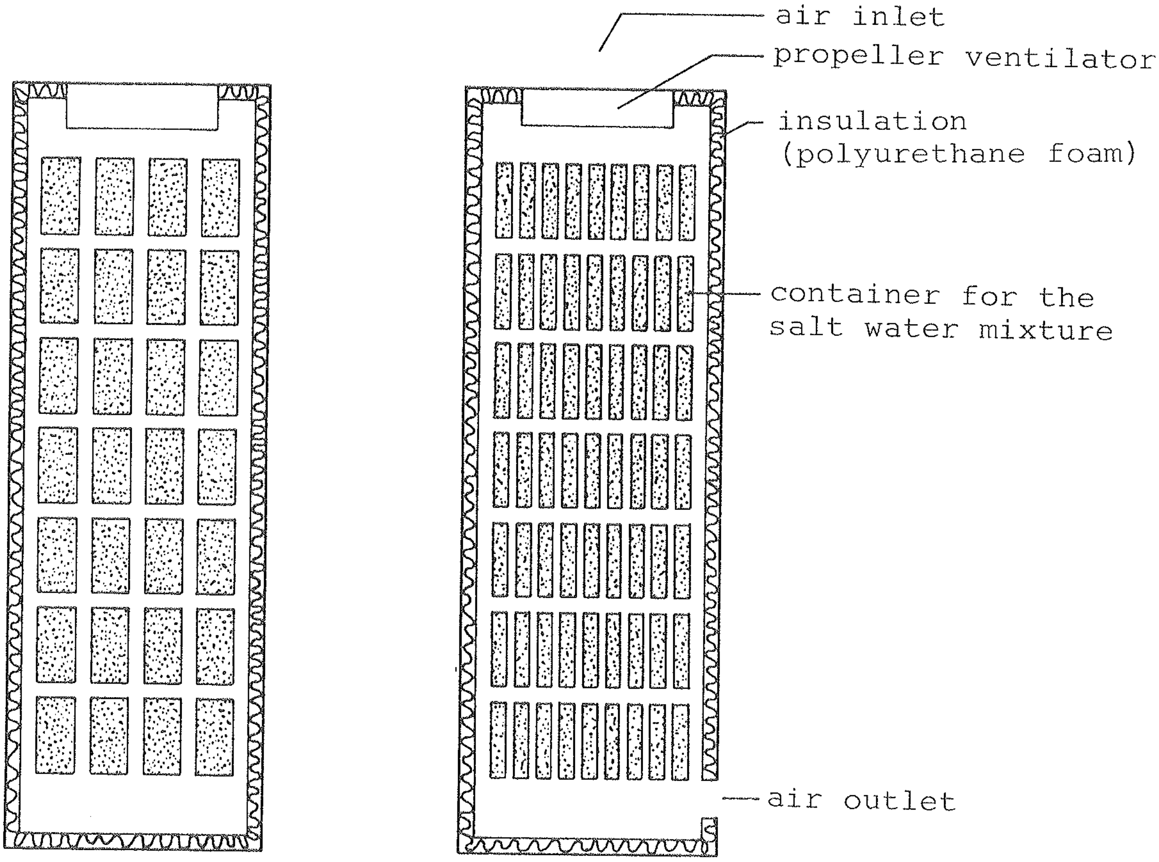


Fig. 5.4 Vertical sections of the improved storage unit.

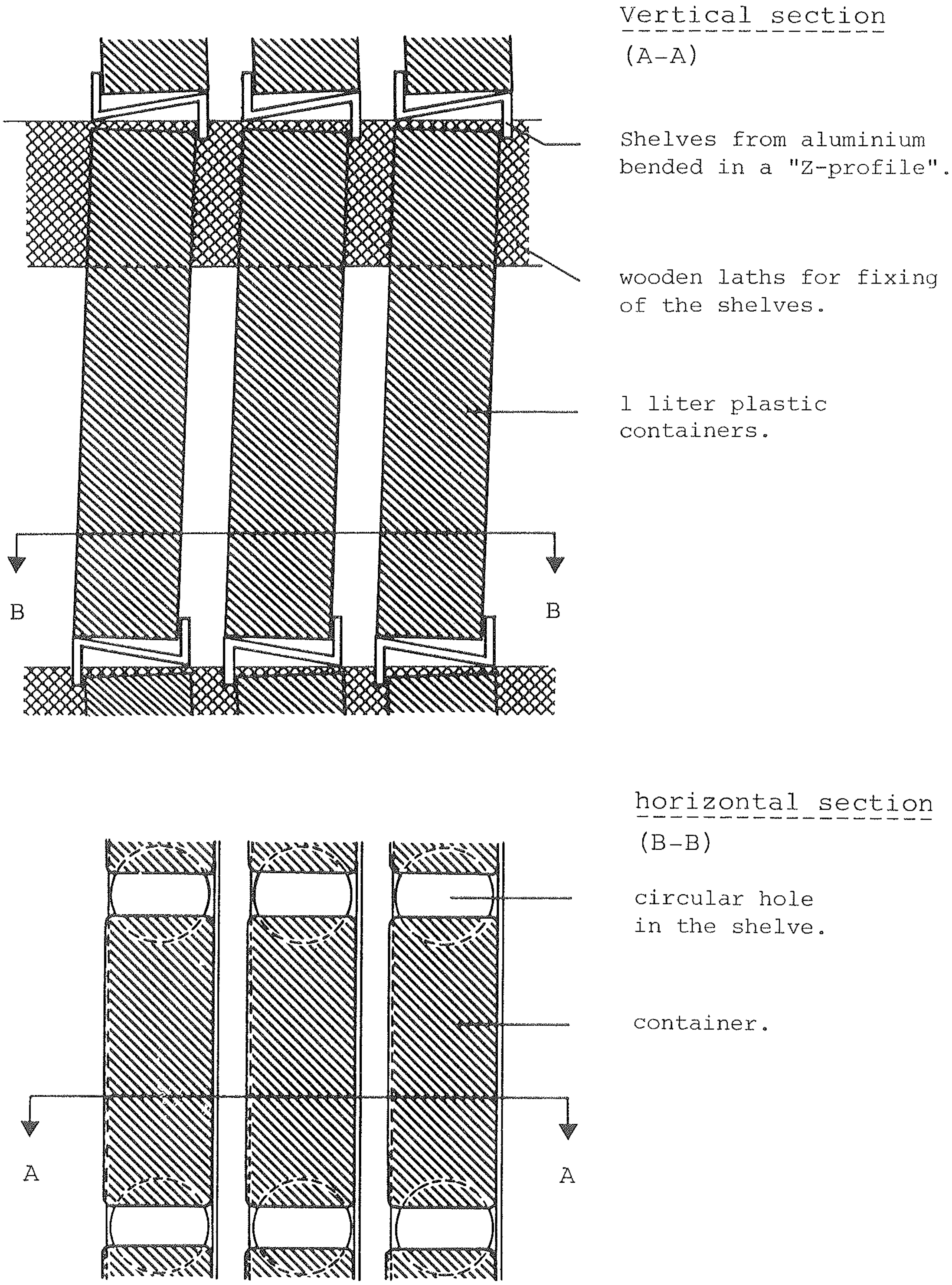


Fig. 5.5 The fixing of the containers for the improved storage unit.

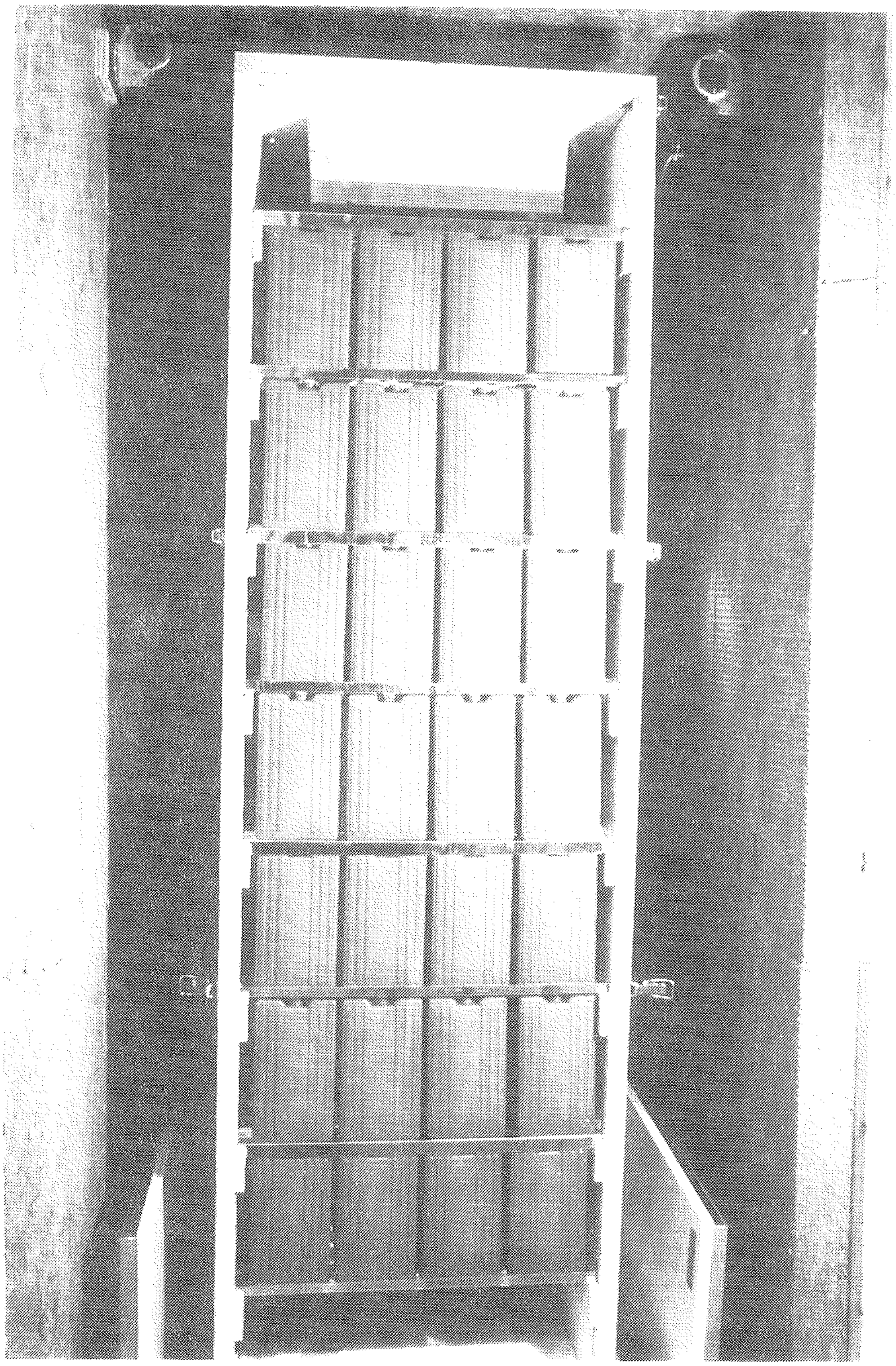


Fig 5.6 The improved storage unit (the front board has been removed)

5.3 Measurement of the thermal characteristics

The heat flow to and from the storage unit was measured by means of the "hot box principle". The set-up is shown in fig. 5.7. The storage unit and a source of heat (two electric radiators) were placed inside an insulated box (the hot box) and the heat transfer to the unit can be found by subtracting the heat transferred through the insulation of the hot box from the heat supplied by the ventilator of the unit and the radiators.

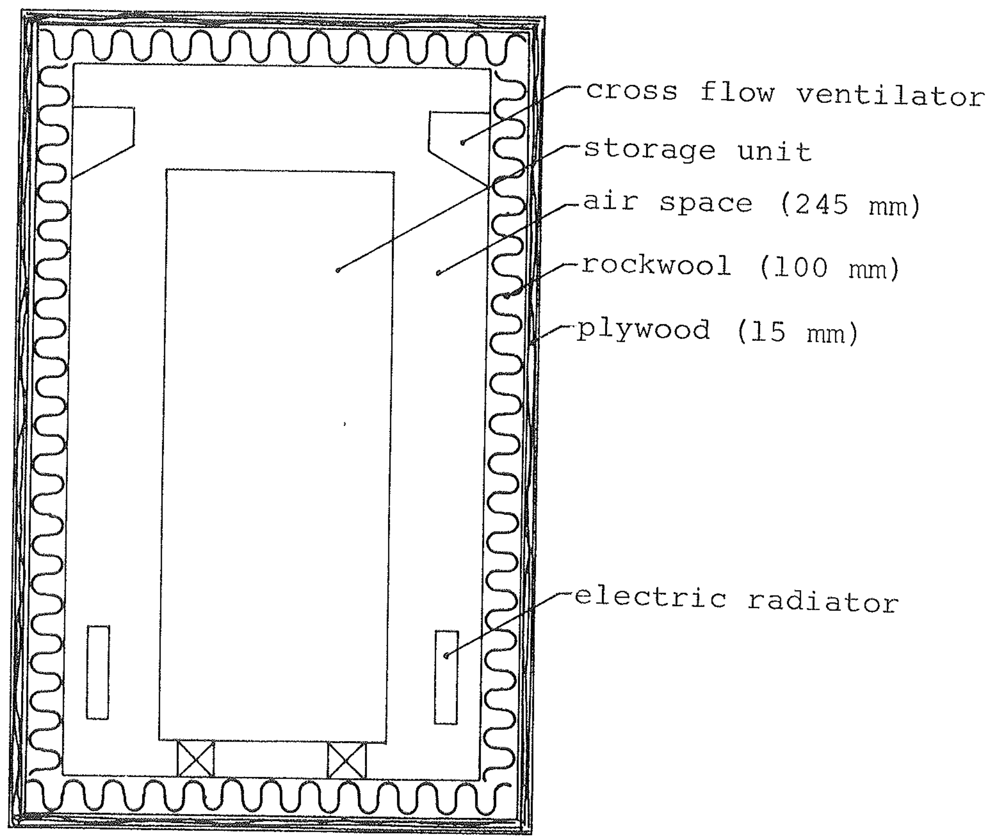


Fig. 5.7 Set-up for the measurement of the heat transfer characteristics for the storage unit by means of a "hot box". The cross flow ventilators in the hot box ensure fully mixing of the air when measuring the heat loss from the storage.

The total heat supply was measured by means of a watt-meter (manually read). All temperatures were measured by type T thermocouples connected to a multipoint recorder (Philips PM 8237A). The temperature was measured at the following locations:

- In the center of one container in each of the seven layers.
- At the top and at the bottom of one container in the central layer.
- At the inlet and outlet of the storage unit.
- At the top, the middle and the bottom of the air space in the hot box.
- The temperature of the ambient air was measured just outside the hot box.

<u>The heat loss due to heat transfer through mineral wool of the hot box</u>

During the experiments with charging of the storage unit, the air temperature in the hot box will increase and because of this non-stationary progress, the heat capacity of the hot box must be considered. It was assumed that the heat loss ΔQ_ℓ from the air space through the mineral wool in the hot box could be approximated by the expression:

$$\Delta Q_\ell = (UA)_h \cdot (T_h - T_o) \cdot \Delta\tau + K \cdot \Delta T_m$$

where

$(UA)_h$ = the overall heat loss coefficient for the storage unit under stationary conditions

K = the heat capacity of the hot box

T_h = the mean temperature of the air in the hot box during the considered period

T_o = the mean ambient air temperature during the period

ΔT_m = the change of the mean temperature of the mineral wool, taken as the mean value between T_h and T_o during the period.

$\Delta\tau$ = the length of the period (about one hour)

The heat loss coefficient was measured by removing the storage unit and heating the hot box until stationary conditions were obtained, leading to $(UA)_h = 4.5\ W/^oC$. When this value and the measured temperatures (during the period when the hot box was being heated) were inserted in the expression for ΔQ_ℓ, it was found that $K \approx 30\ Wh/^oC$. Since ΔQ_ℓ is only 5-10% of the total heat supply, this way of determining ΔQ_ℓ must be considered satisfactory.

Measurement of the overall heat transfer coefficient to the storage unit

The overall heat transfer coefficient to the storage unit related to the inlet air temperature is found from:

$$(UA)_s = \frac{\Delta P - \Delta Q_\ell}{(T_{a,i} - T_s)\cdot\Delta\tau}$$

where

ΔP = the total heat supplied to the hot box during the considered period

T_s = the mean value of the storage temperature (i.e. the average of the temperatures, measured in the middle of one container in each layer) during the period

$T_{a,i}$ = the mean temperature of the air at the inlet during the period

Further, the heat transfer coefficient h from the air in the storage unit to the surface of the containers was calculated:

$$h = \frac{\Delta P - \Delta Q_{\ell}}{(T_{a,m} - T_s)\cdot \Delta\tau \cdot A_c}$$

where

$T_{a,m}$ = the mean value between the temperature of the air at the inlet and at the outlet

A_c = the area of the vertical surfaces of the containers (= 17.9 m^2)

The heat transfer coefficients $(UA)_s$ and h were measured in case that the containers were filled with water and subsequently in case they were filled with a 46% salt-water mixture. In the two experiments, heat was supplied to the hot box for approx. five hours and the electricity consumption was read at approx. one-hour periods. The results from the experiments are shown in table III in appendix III. $T_{f,i}$ is some degrees higher than T_h, the reason being that only one of the radiators in the hot box was used, whereas T_h was measured at the opposite side of the storage unit where the air is not heated. The increased air temperature between the radiator and the inlet of the storage has minor influence on the heat loss from the hot box, since T_h is representing most of the air space. The mean values of $(UA)_s$ and h are listed in table 5.1

Storage medium	$(UA)_s$	h
Water	114 W/^oC	11.1 W/(m^2 ^oC)
46% $CaCl_2$, 54%H_2O	116 W/^oC	12.8 W/(m^2 ^oC)

Table 5.1 Mean values of the measured heat transfer coefficients to the storage unit $(UA)_s$ (related to the inlet air temperature) and h (related to the mean of the air temperature at the inlet and the outlet).

The value of $(UA)_s$ is improved compared to the values obtained for the units constructed for the realistic testing. They are, however, not as high as was exptected from the calculations. A measurement of the air flow and the pressure drop gave values much lower than expected, leading to the conclusion that the performance of the ventilator was not sufficiently good.

The air flow measurement showed that the velocity of the air in the ducts between the storage containers was about 1.4 m/s. From the theoretical expression used in the calculation, this corresponds to h being between 8.2 and 9.7 W/(m^2 ^{o}C) depending on the size of the minor pressure losses. This is somewhat lower than the measured values, which must be due partly to the uncertainty connected to the fixing of the mean air temperature and partly to the theoretical formular giving too low values for the actual shape of the ducts. The heat transfer coefficients are a little smaller in case of water than in case of salt-water mixture. Since the thermal resistance from the surface to the heat storage medium is larger for the salt-water mixture (mainly due to the higher viscosity), it was predicted that h was slightly smaller in this case. The inaccuracy connected to the calculation of the mean temperature of the heat storage medium must be the reason why this was not observed. Since the temperature variations from the top to the bottom of the storage unit, and within the individual container, turned out to be largest for salt-water mixture, the measurements for water must be regarded as being the most exact.

Other results

The overall heat loss coefficient was measured by comparing the heat loss from the containers with water, with the difference between the mean storage temperature and the temperature of the surrounding air, after having heated the water to about 10^{o}C above the ambient air temperature. The heat loss coefficient was measured by comparing the change in the heat content of the water (according to the change in the mean water temperature) with the difference between the mean water temperature and the temperature of the air outside the storage. Unfortunately, the heat loss coefficient appeared to be several times larger than predicted (about 35 W/^{o}C instead of 5 W/^{o}C). It was concluded that this was due to the louver damper of the ventilator not being sufficiently airtight, thus leading to a significant heat flow by reverse air flow through the unit. As, however, airtightness can be obtained rather easily (as demonstrated with the storage unit with the many containers constructed for the realistic testing) and as the heat loss coefficient is considered to be of less importance than the heat transfer coefficient, no attempts were made to improve the conditions at this point.

As mentioned earlier, the temperature variations of the salt-water mixture between the top and bottom of the unit, and within the individual container, showed to be considerable. The difference in temperature between the top and the bottom of the unit (measured in the middle of the containers) was about 6^{o}C; the difference between the top and the bottom of the container in the central layer was up to 4^{o}C. Since the temperature of the lower part of each container, where the melting takes place, was less than the mean temperature of the solution, the mean storage temperature had to be rather high before all the salt hydrates were melted. Consequently, the heat content of the storage unit, at a given mean storage temperature within the interval of the room temperature (20-26^{o}C), was less than if the stirring had been efficient enough to ensure fully mixing of the solution. The aim must therefore be to obtain a uniform distribution of the temperature all over the storage. With regard to the temperature difference between the inlet and outlet (for an equal rate of heat transfer) this will be reduced considerably if the mass flow through the unit is increased corresponding to the ventilator being as efficient as assumed in section 4.2. Regarding the temperature differences inside the container, these can be counteracted by using containers with a lower height. The containers used for the experiments could, for example, be turned round, so the height would be only 0.12 m.

6. ECONOMIC EVALUATIONS

6.1 Calculation of the economic savings

Based on the calculation of the energy savings described in chapter 4, the private economic benefits of a local storage unit has been calculated. The calculations have been carried out by using the present value method described in [12].

The profitability of an investment is determined by converting the expected future savings (for example during the lifetime of the investment object) into an amount corresponding to the total gain being realized immediately. This amount is called the present value and it has to be positive if the project is to be profitable at all. If it is assumed that no extra costs for maintenance are involved, the present value PR can be calculated by means of the equations:

$$PR = s_o \cdot f_s - I_o \ , \qquad f_s = \frac{1 - (1 + r_{rs})^{-n}}{r_{rs}}$$

$$r_{rs} = \frac{r_s - i_e}{1 + i_e} \ , \qquad r_s = r \cdot (1 - t)$$

where

s_o = the yearly economic savings according to the energy price of today

f_s = the present value factor

I_o = the invested amount, i.e. the cost of the storage unit

n = the considered number of years

t = the marginal rate of taxation

i_e = the relative yearly increase of the energy prices

r = the calculation rate corresponding to the rate of interest in case the money has to be borrowed from the bank, or if the money could have been invested in for instance bonds

r_s = the calculation rate, reduced due to the energy savings not being taxed

r_{rs} = the real calculation rate for the savings when accounting of the expected rise of the energy prices

Several of the parameters used in the calculations are subject to some uncertainty. The analysis is therefore carried out for a number of different assumptions concerning the lifetime and the manufacturing cost of the storage unit, the rate of interest and the development of the energy prices. The marginal rate of taxation t is assumed to be 60% being the percentage generally paid by that category of people who might want to invest in a storage unit.

The real rate of interest is calculated for the following cases:

1) The yearly increase of the energy prices will be rather low, about 2% higher than the inflation rate. The yearly inflation rate is set to 5% (case 1.a) and 8% (case 1.b).

2) The increase of the energy prices will be 5% higher than the inflation rate which is set to 5% (case 2.a), 8% (case 2.b) and 11% (case 2.c).

3) A rather large yearly increase of the energy prices on 8% in excess of the inflation rate is assumed. The inflation rate is set to 8% (case 3.a) and 11% (case 3.b).

An increase of energy prices of approx. 5% (case 2) corresponds to the predictions in [13], page 64, concerning the most probable development of the oil prices. In the present calculations it is assumed that the prices on oil and electricity will increase with the same rate. The rate of interest r is fixed according to what would be a typical rate if the money was borrowed and the inflation is taken into account. In table 6.1 is shown the different assumed values for i_e and r and the corresponding calculated values for r_s and r_{rs}.

CASE	yearly increase of energy prices in excess of the inflation	infla-tion rate	i_e	r	r_s	r_{rs}
1.a	2%	5%	7%	14%	5.6%	-1.3%
1.b		8%	10%	17%	6.8%	-2.9%
2.a	5%	5%	10%	14%	5.6%	-4.0%
2.b		8%	13%	17%	6.8%	-5.5%
2.c		11%	16%	20%	8.0%	-6.9%
3.a	8%	8%	16%	17%	6.8%	-7.9%
3.b		11%	19%	20%	8.0%	-9.2%

Table 6.1 The real calculation rate r_{rs} for different values of the yearly increase in energy prices i_e and the interest rate r. The calculation rate r_s corresponds to a marginal rate of taxation on 60%

For the different values of r_s, the present value factor f_s can be determined. This is done in case of n = 10, 15 and 20 years, and the results are shown in table 6.2.

CASE	r_{rs}	f_s		
		n = 10	n = 15	n = 20
1.a	-1.3%	10.8	16.7	23.0
1.b	-2.9%	11.8	19.2	27.7
2.a	-4.0%	12.6	21.1	31.6
2.b	-5.5%	13.8	24.3	38.1
2.c	-6.9%	15.1	27.9	46.1
3.a	-7.9%	16.2	30.9	53.2
3.b	-9.2	17.7	35.5	64.4

Table 6.2 The present value factor f_s for different values of the real calculation rate r_{rs} and the lifetime of the storage unit n (in years)

The auxiliary heating is normally due to consumption of oil or electricity. The following prices of today have been informed:

> The oil price is 3.46 Danish Kroner (dkr) per litre. If the efficiency of the boiler is set to 0.81 and the heating value of the oil is 9.84 kWh/l, the energy price will be 0.434 dkr/kWh.
>
> The electricity price is 0.659 dkr/kWh.

The yearly economic savings s_o are calculated as the product of the energy prices and the yearly energy savings (according to the reference case in chapter 5) on 590 kWh/year. In case of oil and electricity, this gives:

s_o = 256 dkr and s_o = 389 dkr

Regarding the manufacturing cost of the storage unit, it is also possible to choose from different assumptions. One possibility is to consider the expenses paid in connection with the storage unit described in chapter 5. The cost of materials and components were approximately:

salt-water mixture:	350 dkr
plastic containers:	1250 dkr
insulating boards for the box:	2900 dkr
ventilator and control system:	1500 dkr
minor items:	300 dkr
Total:	6300 dkr

If a mass-production of the storage unit could be established, the labour cost for the construction including the profit to the manufacturer would probably come to about 1000 dkr, and the total cost of the unit would be I_O = 7300 dkr. Furthermore, the box could be made from cheaper materials than the boards originally used. The cost of 7300 dkr must therefore be considered as an upper limit. If it is assumed that the box is made from chipboard with about 2 cm polyurethane foam glued on to the inside, the cost of the box can be reduced to approx. 500 dkr. If the cost of salt-water mixture, the containers, the ventilator and the control system is reduced with 20% because of a mass-production, the cost of all the materials and components will be approx. 3300 dkr. When adding 1000 dkr (labour cost and profit) a total cost of I_o = 4300 dkr is obtained. As this amount depends on optimistic assumptions, it is regarded as a minimum amount. It is likely to consider the mean value between 7300 dkr and 4300 dkr as the expected cost which gives I_o = 5800 dkr.

In tables 6.3 and 6.4 the results for the present value are shown in case the auxiliary heating is provided by oil and electricity respectively. It is obvious that apart from the influence of the cost and the lifetime of the storage unit, there is a strong dependence on the assumptions concerning the real calculation rate. If , for instance, the manufacturing cost of the unit is 5800 dkr, the lifetime is 15 years and the house is heated by electricity, it shows from table 6.4 that PR = 200 dkr when the lowest calculation rate is used (moderate increase of energy prices and moderate rate of interest), and 8000 dkr for the highest rate (large increase of oil prices and high rate of interest). If we consider case 2.b (13% yearly increase of oil prices and an interest rate of 17%) we obtain PR = 3700 dkr. If oil is the source of auxiliary energy, PR will decrease to 400 dkr (table 6.3).

When evaluating whether it is advantageous to invest in a storage unit, it is reasonable to demand that PR is positive with the lowest of the proposed real calculation rates in order to make sure that capital is not wasted. Thus, when considering case 1.a in table 6.3, it shows that with a lifetime of the storage unit of 15 years and in case that oil is the source of auxiliary energy, I_o must not exceed 4300 dkr as PR = 0 for this price. If electricity is used instead of oil, the conditions turn out

CASE	I_o = 4300 dkr			I_o = 5800 dkr			I_o = 7300 dkr		
	n (years)			n (years)			n (years)		
	10	15	20	10	15	20	10	15	20
1.a	-1.5	0	1.6	-3.0	-1.5	0.1	-4.5	-3.0	-1.4
1.b	-1.3	0.6	2.8	-2.8	-0.9	1.3	-4.3	-2.4	-0.2
2.a	-1.1	1.1	3.8	-2.7	-0.4	2.3	-4.1	-1.9	0.8
2.b	-0.8	1.9	5.5	-2.3	0.4	4.0	-3.8	-1.1	2.5
2.c	-0.4	2.8	7.5	-1.9	1.3	6.0	-3.4	-0.2	4.5
3.a	-0.2	3.6	9.3	-1.7	2.1	7.8	-3.2	0.6	6.3
3.b	0.2	4.8	12.2	-1.3	3.3	10.7	-2.8	1.8	9.2

Table 6.3 The present value PR in 10^3 dkr of the investment in a storage unit (in the reference case) for different assumptions in regard to the energy prices and the calculation rate (case 1.a - 3.b), the cost of the unit I_o and the lifetime in years n, in the case where the auxiliary energy is provided by oil.

CASE	I_o = 4300 dkr			I_o = 5800 dkr			I_o = 7300 dkr		
	n(years) 10	15	20	n(years) 10	15	20	n(years) 10	15	20
1.a	-0.1	1.7	4.6	-1.6	0.2	3.1	-2.9	-1.3	1.6
1.b	0.3	3.2	6.5	-1.2	1.7	5.0	-2.7	0.2	3.5
2.a	0.6	3.9	8.0	-0.9	2.4	6.5	-2.4	0.9	5.0
2.b	1.1	5.2	10.5	-0.4	3.7	9.0	-1.9	2.2	7.5
2.c	1.6	6.6	13.6	0.1	5.1	12.1	-1.4	3.6	10.6
3.a	2.0	7.7	16.4	0.5	6.2	14.9	-1.0	4.7	13.4
3.b	2.6	9.5	20.8	1.1	8.0	19.3	-0.4	6.5	17.8

Table 6.4 The present value PR in 10^3 dkr of the investment in a storage unit (in the reference case) for different assumptions in regard to the energy prices and the calculation rate (case 1.a - 3.b), the cost of the unit I_o and the lifetime in years n, in the case where the auxiliary energy is provided by electricity.

to be more favourable. Here we obtain PR = 1700 dkr if it is possible to produce the storage unit for 4300 dkr and PR = 200 dkr when the cost is 5800 dkr.

6.2 Comparison with other energy saving arrangements

The savings obtained because of the local storage unit are compared with savings due to some realistic alternative ways of lowering the energy consumption of the room used in the reference case. The different designations and assumptions concerning the economic savings correspond to what has been informed in section 6.1.

(I) The windows are supplemented with one extra layer of glass

One possibility is to use three layers of glass for the windows instead of two. In this case the computer calculations showed that the need for auxiliary energy (for the reference room described in chapter 4) will decrease from 3389 kWh/year to 2871 kWh/year. Thereby the yearly savings will be 518 kWh or 15.3%. This corresponds to the yearly economic savings of s_o = 225 dkr in case of oil being used as the auxiliary energy source and s_o = 341 dkr in case of electricity.

It is assumed that the extra layer of glass can be fitted to the existing window frame without any problems. A glazier was consulted and he estimated that the total cost of the work would be approx. I_o = 4700 dkr. The lifetime of the windows is set to 20 years. The present value is calculated by using the same calculation rates as in section 6.1 and the results are shown in table 6.5.

(II) Additional insulation of the ceiling

Another method of reducing the heat consumption of the considered room is to add more insulation, for instance to the ceiling. By increasing the thickness of this insulation from 200 mm to 400 mm, the computer calculations showed that the need for auxiliary energy will decrease from 3389 kWh/year to 3211 kWh/year. This way the savings are only 178 kWh/year or 5.3% resulting in 77 dkr/year when oil is the auxiliary energy source and 117 dkr/year in case of electricity. When the insulation is carried out by professionals the total cost will be approx. 3800 dkr. By setting the lifetime of this arrangement to 20 years, the present values for the different assumptions concerning the real calculation rate are calculated and listed in table 6.5.

From table 6.5 it appears that PR is positive in general when adding an extra layer of glass to the window. For the most "pessimistic" calculation (case 1.a) PR is 900 dkr when the room is heated by oil and 3500 dkr when electricity is used. The economic savings obtained from additional insulation to the ceiling are not very promising; in case 1.a present values of minus 2000 dkr and minus 1100 dkr are obtained for the two types of auxiliary energy.

CASE	f_s	(I) Extra layer of glass to the window		(II) Additional insulation to the ceiling	
		oil	el.	oil	el.
1.a	23.0	0.9	3.5	-2.0	-1.1
1.b	27.7	1.9	4.7	-1.7	-0.6
2.a	31.6	2.4	6.1	-1.4	-0.1
2.b	38.1	3.9	8.3	-0.9	0.7
2.c	46.1	5.7	11.0	-0.3	1.6
3.a	53.2	7.3	13.4	0.3	2.4
3.b	64.4	9.8	17.3	1.2	3.7

Table 6.5 The present value in 10^3 dkr for two energy saving methods for different values of the present value factor f_s, when the auxiliary energy is provided by oil and electricity respectively

It must be concluded that of the two arrangements, only the method of fitting an extra layer of glass to the window can compete with the cost of the storage unit. It should be pointed out that the mounting of a third layer of glass is not always possible, whereas there will be no problems with the installation of a storage unit. If the entire window has to be renewed in order to obtain three layers of glass, the cost will be much higher (about 10.900 dkr according to the glazier) and thereby this possibility will not be attractive.

7. CONCLUSION

From the results of this work it is concluded that the installation of the proposed "local storage unit" is a realistic energy saving arrangement for use in existing single-family houses. The computer calculations showed that for a typical living room having large south facing windows and a small heat capacity and with a storage volume of 252 litre, the yearly energy savings will be about 590 kWh or 17% of the energy required for heating the room when weather data of the "Danish Test Reference Year" are used. The power consumption of the ventilator of the storage unit is included in the result. The purchase of a storage unit can be economic advantageous after 10 years even if (as a conservative assumption) the rate of interest and the increase of the energy price will be low, provided that electricity is used for auxiliary heating. This is on the condition that the unit can be produced for about 4,300 dkr which is an optimistic, but not unrealistic, assumption. The economic evaluation showed that the storage unit saves more energy than if an extra layer of glass is supplied to the windows (corresponding to an increase from two to three layers of glass). The supply of one extra layer of glass to the windows will often be a difficult and expensive arrangement and it may be necessary to change the entire windows and in these cases the use of a storage unit is a more attractive way of saving energy.

The mentioned benefits of the local storage unit implies that some conditions are fulfilled. It is important that the south facing window area is very large and the existing heat capacity of the room is small, otherwise the energy savings will be smaller. Anyhow the number of houses from the 1960'ies and early-70'ies is very large, so although many houses do not fulfil the requirements there will still be a considerable amount of houses for which the storage unit is fitted. The room temperature should further be allowed to be increased to at least 26°C during the day and during the night the required room temperature should be lowered to about 17°C.

It was demonstrated that by the addition of extra water to the salt hydrate $CaCl_2 \cdot 6H_2O$ it is possible to obtain a stable heat storage medium (no phase separation). It was therefore concluded that a mixture with 46% $CaCl_2$ and 54% water was suitable for the storage unit. The calculations showed that this concentration gives slightly larger energy savings than the mixture with 43.6% $CaCl_2$ used for the realistic testing of the two storage units that were constructed in the first phase of the project. The realistic testing demonstrated that due to the presence of the units it is possible to decrease considerably the extent of the periods for which it is necessary to provide a strong ventilation with the ambient air if an acceptable level of the thermal comfort shall be obtained.

The heat transfer coefficient between the room air and the storage containers turned out to be a very important parameter. A satisfactorily large heat transfer coefficient can be obtained by placing the salt-water in one-litre containers which was done for the improved storage unit constructed in the last phase of the project. Nevertheless, the heat transfer coefficient for this storage unit was somewhat smaller than assumed from the calculations which was due to the fact that the ventilator did not produce the necessary air flow. The heat transfer coefficient corresponding to the obtained air flow was a little higher than the theoretical value for an equivalent air flow. It is not considered a major problem to procure a ventilator with a higher performance at the air flow and pressure drop in question. By using that it should be possible to obtain a heat transfer coefficient which corresponds to what was used in the computer calculations.

The experiments showed, further, that with a storage container height of about 0.20 m there will be a considerable temperature difference between the top and the bottom of the container while the storage is being charged. The temperature at the bottom becomes lower than the mean temperature of the container which gives the result that the heat content for a certain mean storage temperature, within the assumed interval of the room temperature, is smaller than what corresponds to the phase diagram. In order to ensure that heat is supplied more efficiently to the zone where melting takes place the one liter containers should be placed horizontally instead of vertically.

LIST OF SYMBOLS

A_c area of the vertical surfaces of the containers, m^2

A_d total area of the cross section of the ducts, m^2

A_w window area, m^2

c concentration of anhydrous salt in salt-water mixture, g/g

c correction factor for the effect of the entrance region on the Nusselt number

c_p specific heat, $J/(kg \cdot {}^oC)$

C_{lh} specific heat of liquid phase salt hydrate, $J/(g \cdot K)$

C_{mh} the specific heat of the salt-water mixture, when all of the salt hydrate is melted, $J/(g \cdot K)$

C_{ml} the specific heat of the salt-water mixture, when some of the salt hydrate is in the solid phase, $J/(g \cdot K)$

C_{sh} specific heat of solid phase salt hydrate, $J/(g \cdot K)$

C_w specific heat of water, $J/(g \cdot K)$

d horizontal distance between containers, m

d_h hydraulic diameter, m

f the friction factor

f_s present value factor

F_{as} weight fraction of anhydrous salt in the salt hydrate

F_{am} weight fraction of anhydrous salt in the salt-water mixture

F_{sol} weight fraction of the salt hydrate that is in the solid phase

(Gc) heat transfer per ^oC due to air change with the ambient air, $W/{}^oC$

h heat transfer coefficient between the container wall and the air in the ducts, $W/(m^2\,{}^oC)$

ΔH pressure drop, Pa

ΔH_f pressure drop due to friction loss, Pa

ΔH_m pressure drop due to minor losses, Pa

i_e relative yearly increase of energy prices

I_o invested amount, dkr

k loss coefficient for minor pressure loss

k conductivity, $W/(m \cdot k)$

K heat capacity of the hot box, $Wh/^oC$

L heat of fusion of salt hydrate, J/g

L length of vertical ducts (= total height of the layers of containers), m

$\dot{m}$ mass flow through the storage unit, kg/s

n number of years

Nu Nusselt number

p relative energy savings

P power consumption of ventilator, W

ΔP energy supplied to the hot box, Wh

P_1 performance of ventilator, W

PMV Predicted Mean Vote

PR present value, dkr

q heat flow, W

q_c heat supplied by convection to the room air due to persons, electric devices, radiators and solar radiation, W

q_k heat transmission from the considered room to the adjoining rooms, W

q_l heat transfer from the room surfaces to the ambient air, W

q_m heat transmission from the room air to the heat accumulating layer, W

q_o heat transfer between the room air and the interior room surfaces, W

q_r heat supplied by radiation to the room surfaces due to persons, electric devices and solar radiation, W

q_s heat transfer between the room air and the storage medium in the storage unit, W

q_v heat transfer due to air change with the ambient air, W

Q energy, Wh

ΔQ_l energy loss from the hot box to the ambient air, Wh

Q_r yearly need for energy from the radiators for the room with storage unit, kWh

Q'_r yearly need for energy from the radiators for the room without storage unit, kWh

Q_s yearly energy savings, kWh

Q_s heat content of the salt-water mixture, J/g

Q_v yearly energy consumption of ventilator, kWh

r calculation rate

r_s calculation rate, reduced due to the energy savings not being taxed

r_{rs} the real calculation rate

Re Reynold number

S_o the yearly economic savings according to the energy price of today, dkr

S the total heat capacity of the materials in the walls, ceiling and floor within the insulation of the house, Wh/^oC

S_a the weight fraction of anhydrous salt in a saturated solution

t marginal rate of taxation

t temperature of salt-water mixture, K

t_o temperature at which the heat contant of the salt-water mixture is put to zero, K

t_s the lower unit of the temperature interval in which all the salt hydrate is in the liquid phase, K

T temperature, ^oC

T_a outdoor temperature, ^oC

$T_{a,i}$ temperature of circulating air at the inlet, ^oC

$T_{a,m}$ mean temperature of circulating air, ^oC

$T_{a,o}$ temperature of circulating air at the outlet, ^oC

T_h mean temperature of the air in the hot box, ^oC

T_k temperature of adjoining room, ^oC

T_m temperature of the heat accumulating mass of the building structure, ^oC

ΔT_m change of the mean temperature of the mineral wool of the hot box, ^oC

T_o temperature of the interior surfaces of room, ^oC

T_o mean temperature of air outside the hot box, ^oC

T_r room air temperature, ^oC

$T_{r,max}$ maximum room air temperature, ^oC

$T_{r,min}$ minimum room air temperature, ^oC

T_s temperature of the local storage unit, ^oC

$T_{s,m}$ mean temperature of the surfaces of the containers, ^oC

$(UA)_l$ heat transfer per ^oC from the interior surfaces to the ambient air, $W/^oC$

$(UA)_m$ heat transfer between the room sufaces and the heat accumulating mass per ^oC, $W/^oC$

$(UA)_o$ heat transfer between the room air and the interior room surfaces per ^oC, $W/^oC$

$(UA)_r$ overall heat loss coefficient from room air to ambient air, $W/^oC$

$(UA)_s$ heat transfer between the room air and the salt-water mixture of the storage unit, $W/^oC$. (If nothing else is stated it is identical with the overall heat transfer coefficient obtained by forced air circulation. In the description of the computer program the symbol is a general name for the overall heat transfer coefficient and the overall heat loss coefficient).

v mean air velocity, m/s

V volume of salt-water mixture, l

ρ density, kg/m^3

τ the time, hours

η efficiency of ventilator

υ kinematic viscosity, m^2/s

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DATE	mean ambient temperature °C	transmitted solar radiation kWh	max/min of air temperature in room 2 °C	max/min of air temperature in room 3 °C	max/min of air temperature in room 4 °C	max/min of storage temperature in room 2 °C	max/min of storage temperature in room 4 °C	running time of ventilator in room 2 % of the 24 hours	running time of ventilator and pump in room 4 % of the 24 hours	running time of radiator in room 2 % of the 24 hours	running time of radiator in room 3 % of the 24 hours	running time of radiator in room 4 % of the 24 hours	max PMV value in room 2	max PMV value in room 3	max PMV value in room 4
Okt. 12	9.2	5.3	29.8/ 19.5	32.0/ 20.4	30.7/ 20.5	27.3/ 19.8	29.8/ 21.3	68	63	3	7	2	2.8	4.0	2.3
Okt. 13	12.2	1.6	23.9/ 20.5	23.8/ 20.3	24.4/ 21.4	26.7/ 21.5	28.2/ 22.4	63	52	0	4	0	0.4	0.9	0.9
Okt. 14	-	3.0	-	-	-	-	-	75	65	-	-	-	-	-	-
Okt. 15	10.5	0.6	21.1/ 20.1	20.9/ 20.3	21.5/ 20.1	21.8/ 20.3	22.7/ 21.1	60	100	5	13	0	0.0	0.1	0.0
Okt. 16	11.4	1.5	22.8/ 20.0	23.2/ 20.3	22.8/ 19.9	21.5/ 20.2	22.3/ 20.7	78	93	-	15	3	0.6	0.6	2.1
Okt. 17	-	0.9	-	-	-	-	-	73	100	-	-	-	-	-	-
Okt. 20	8.2	2.9	24.8/ 20.1	25.3/ 20.3	25.3/ 20.5	23.9/ 20.5	24.7/ 21.5	78	68	-	6	0	1.0	1.9	1.7
Okt. 21	-	6.5	-	-	-	-	-	73	67	-	-	-	-	-	-
Okt. 22	8.3	2.3	25.6/ 20.6	26.1/ 20.4	26.3/ 21.6	29.4/ 23.3	30.1/ 22.8	33	52	-	4	0	1.5	1.9	1.9
Okt. 23	8.0	2.5	24.6/ 20.0	25.2/ 20.4	24.8/ 20.1	22.9/ 20.4	24.1/ 20.9	88	90	-	11	0	1.5	1.8	1.7
Okt. 24	-	5.0	-	-	-	-	-	63	70	-	-	-	-	-	-

Table I. Results from the realistic testing of the two storage units, part 1. according to section 3.3.

DATE	mean ambient temperature °C	transmitted solar radiation kWh	max/min of air temperature in room 2 °C	max/min of air temperature in room 3 °C	max/min of air temperature in room 4 °C	max/min of storage temperature in room 2 °C	max/min of storage temperature in room 4 °C	running time of ventilator in room 2 % of the 24 hours	running time of ventilator and pump in room 4 % of the 24 hours	running time of radiator in room 2 % of the 24 hours	running time of radiator in room 3 % of the 24 hours	running time of radiator in room 4 % of the 24 hours	max PMV value in room 2	max PMV value in room 3	max PMV value in room 4
Okt. 25	3.7	0.5	20.6/ 19.8	20.7/ 20.3	21.6/ 19.6	25.2/ 20.0	24.7/ 20.4	93	98	-	30	10	-0.4	-0.2	-0.1
Okt. 26	11.4	0.7	22.1/ 20.0	21.9/ 20.4	22.2/ 19.8	21.1/ 20.0	22.1/ 20.5	50	97	-	11	2	0.0	0.2	0.3
Okt. 27	11.0	0.6	21.0/ 20.1	20.8/ 20.3	21.3/ 20.1	20.6/ 20.3	22.0/ 21.0	45	100	-	12	0	-0.1	-0.0	0.0
Okt. 29	1.6	6.7	32.2/ 19.8	35.4/ 20.4	32.4/ 19.6	29.2/ 20.1	31.4/ 20.5	68	65	-	11	3	4.2	5.4	4.4
Okt. 30	5.8	1.7	21.7/ 19.7	22.2/ 20.2	22.3/ 19.7	27.5/ 20.0	27.0/ 20.5	88	97	-	24	5	0.5	0.3	1.5
Okt. 31	8.7	0.7	21.0/ 19.9	20.9/ 20.2	20.9/ 19.7	20.3/ 20.0	21.6/ 20.5	25	100	-	17	4	0.0	-0.0	0.8
Nov. 01	9.3	0.3	20.7/ 20.0	20.5/ 20.3	20.2/ 19.9	20.3/ 20.2	21.0/ 20.6	32	100	-	17	4	-0.1	-0.2	-0.4
Nov. 02	7.0	5.4	31.2/ 20.3	33.8/ 20.4	31.4/ 20.0	28.0/ 20.2	30.1/ 20.9	38	63	5	5	1	3.7	4.5	4.1
Nov. 03	6.5	2.0	23.0/ 19.9	23.4/ 20.4	23.3/ 19.9	26.5/ 20.3	25.9/ 20.8	88	85	4	14	1	0.5	0.8	1.4
Nov. 06	8.3	1.3	22.7/ 20.0	23.3/ 20.3	22.7/ 19.8	21.3/ 20.2	22.1/ 20.6	68	97	10	15	4	0.6	1.4	1.2

Table I. Results from the realistic testing of the two storage units, part 2. according to section 3.3.

DATE	mean ambient temperature °C	transmitted solar radiation kWh	max/min of air temperature in room 2 °C	max/min of air temperature in room 3 °C	max/min of air temperature in room 4 °C	max/min of storage temperature in room 2 °C	max/min of storage temperature in room 4 °C	running time of ventilator in room 2 % of the 24 hours	running time of ventilator and pump in room 4 % of the 24 hours	running time of radiator in room 2 % of the 24 hours	running time of radiator in room 3 % of the 24 hours	running time of radiator in room 4 % of the 24 hours	max PMV value in room 2	max PMV value in room 3	max PMV value in room 4
Nov. 07	8.8	0.1	20.3/ 20.0	20.5/ 20.4	20.1/ 19.9	20.3/ 20.2	20.8/ 20.7	48	100	14	19	6	-0.3	-0.2	-0.4
Nov. 08	7.1	0.1	20.2/ 20.0	20.6/ 20.4	19.9/ 19.7	20.3/ 20.1	20.7/ 20.5	55	100	17	23	9	-0.4	-0.2	-0.4
Nov. 09	-	1.3	-	-	-	-	-	65	95	-	-	-	-	-	-
Nov. 10	7.3	0.1	20.3/ 20.0	20.5/ 20.4	20.1/ 19.8	20.4/ 20.2	21.0/ 20.7	53	100	13	18	5	-0.3	-0.2	-0.4
Nov. 11	-	6.2	-	-	-	-	-	62	72	-	-	-	-	-	-
Nov. 12	-0.3	3.1	25.4/ 19.7	27.2/ 20.5	25.9/ 19.5	24.3/ 20.0	24.5/ 20.3	88	90	11	21	6	2.2	1.8	2.3
Nov. 13	-2.5	5.6	29.3/ 19.6	32.3/ 20.5	29.8/ 19.5	26.7/ 19.9	28.5/ 20.3	75	80	11	18	7	3.2	3.7	3.4
Nov. 14	1.8	1.9	23.6/ 19.6	24.8/ 20.4	23.2/ 19.5	21.7/ 19.9	22.3/ 20.3	85	95	17	25	11	1.6	1.0	1.8
Nov. 15	5.6	5.2	30.3/ 19.9	33.1/ 20.4	30.7/ 19.8	27.5/ 20.2	29.4/ 20.6	55	65	6	8	3	3.1	4.0	3.7
Nov. 16	1.6	3.0	26.6/ 19.8	28.5/ 20.4	26.9/ 19.7	26.2/ 20.4	25.5/ 20.6	82	85	6	16	2	2.6	2.6	2.5
Nov. 17	-2.4	5.2	29.2/ 19.7	32.1/ 20.5	29.6/ 19.4	26.5/ 19.8	28.1/ 20.2	75	78	11	18	9	3.2	3.6	3.4

Table I. Results from the realistic testing of the two storage units, part 3. according to section 3.3.

DATE	mean ambient temperature °C	transmitted solar radiation kWh	max/min of air temperature in room 2 °C	max/min of air temperature in room 3 °C	max/min of air temperature in room 4 °C	max/min of storage temperature in room 2 °C	max/min of storage temperature in room 4 °C	running time of ventilator in room 2 % of the 24 hours	running time of ventilator and pump in room 4 % of the 24 hours	running time of radiator in room 2 % of the 24 hours	running time of radiator in room 3 % of the 24 hours	running time of radiator in room 4 % of the 24 hours	max PMV value in room 2	max PMV value in room 3	max PMV value in room 4
Nov. 18	3.0	1.1	22.6/ 19.7	23.1/ 20.4	22.4/ 19.6	21.9/ 20.0	22.1/ 20.4	90	98	16	25	9	1.3	1.4	1.1
Nov. 19	-	0.4	-	-	-	-	-	72	100	-	-	-	-	-	-
Nov. 20	5.9	3.1	25.8/ 19.9	27.5/ 20.3	26.1/ 19.7	24.0/ 20.0	25.2/ 20.4	63	83	10	14	5	2.0	2.4	2.1
Nov. 21	1.6	2.4	24.4/ 19.8	25.8/ 20.4	24.6/ 19.7	22.7/ 20.0	23.4/ 20.5	87	90	10	18	5	1.4	1.5	2.3
Nov. 22	0.2	2.7	25.3/ 19.8	27.1/ 20.3	25.9/ 19.6	22.9/ 19.9	24.0/ 20.4	83	93	14	22	9	1.5	2.3	2.2
Nov. 23	-0.7	3.9	26.6/ 19.7	28.7/ 20.3	27.3/ 19.6	24.5/ 19.9	25.8/ 20.3	78	85	12	20	8	1.9	2.8	2.8
Nov. 24	1.7	0.8	21.8/ 19.7	21.8 17.6	21.2/ 19.5	20.4/ 19.9	21.1/ 20.3	72	100	24	32	17	0.4	0.1	0.4

Table I. Results from the realistic testing of the two storage units, part 4. according to section 3.3.

	$Q_{s,i}$	$Q_{s,o}$	$Q_{s,\ell}$	Q_{ven}	Q_{rad}	Q'_{rad}	Q_{sav}	P
	kWh/year							%
Reference Case	696	689	7	169	2630	3389	590	17.4
$T_{r,max} = 25^oC$	662	658	4	175	2673	3411	563	16.5
$T_{r,max} = 27^oC$	718	709	10	165	2601	3369	603	17.9
$T_{r,max} = 28^oC$	731	731	11	163	2580	3353	610	18.2
$T_{r,max} = 30^oC$	742	742	13	159	2560	3325	606	18.2
$T_{r,min} = 20^oC$ at night	650	641	8	191	3071	3821	559	14.6
$T_{r,min} = 14^oC$ at night	673	662	12	143	2363	3066	560	18.3
$T_{r,min} = 11^oC$ at night	604	582	23	131	2272	2895	492	17.0
$T_{r,min} = 8^oC$ at night	558	524	34	131	2263	2840	446	15.7
C = 43.6%	658	656	3	181	2660	3389	548	16.2

Table II. Part 1. Parameter variations in accordance with point (A) and (D) in section 4. $T_{r,max}$ = maximum room temperature, $T_{r,min}$ = minimum room temperature, C = consentration of salt in salt-water mixture, $Q_{s,i}$ = heat supplied to the storage, $Q_{s,o}$ = heat delivered by the storage, $Q_{s,\ell}$ = net heat loss from the storage, Q_{ven} = energy consumption by the ventilator, Q_{rad} = energy supplied by the radiators for the room with storage unit, Q'_{rad} = energy supplied by the radiators for the room without storage unit, Q_{sav} = energy savings, P = relative energy savings.

	$Q_{s,i}$	$Q_{s,o}$	$Q_{s,\ell}$	Q_{ven}	Q_{rad}	Q'_{rad}	Q_{sav}	P
	kWh/year							%
Reference Case	696	689	7	169	2630	3389	590	17.4
V_s = 126 ℓ, $(UA)_s$ = 179 W/°C	481	479	3	178	2818	3389	393	11.6
V_s = 189 ℓ, $(UA)_s$ = 236 W/°C	602	591	5	174	2711	3389	504	14.9
V_s = 315 ℓ, $(UA)_s$ = 337 W/°C	767	759	9	165	2571	3389	653	19.3
V_s = 378 ℓ, $(UA)_s$ = 382 W/°C	826	817	10	161	2521	3389	707	20.9
V_s = 126 ℓ, $(UA)_s$ = 290 W/°C	555	550	6	136	2782	3389	471	13.9
V_s = 189 ℓ, $(UA)_s$ = 290 W/°C	643	637	7	159	2686	3389	544	16.1
V_s = 315 ℓ, $(UA)_s$ = 290 W/°C	729	723	7	174	2597	3389	626	18.5
V_s = 378 ℓ, $(UA)_s$ = 290 W/°C	753	746	8	177	2574	3389	638	18.8
$(UA)_s$ = 150 W/°C	524	522	2	213	2749	3389	427	12.6
$(UA)_s$ = 232 W/°C	642	637	5	183	2667	3389	539	15.9
$(UA)_s$ = 340 W/°C	731	723	9	158	2609	3389	624	18.4
$(UA)_s$ = 400 W/°C	764	755	10	147	2587	3389	655	19.3

Table II Part 2. Parameter variations in accordance with point (B) and (C) in section 4. V_s = storage volume, $(UA)_s$ = overall heat transfer coefficient, $Q_{s,i}$ = heat supplied to the storage, $Q_{s,o}$ = heat delivered by the storage, $Q_{s,l}$ = net heat loss from the storage, Q_{ven} = energy consumption of the ventilator, q_{rad} = energy supplied by the radiators for the room with storage unit, Q'_{rad} = energy supplied by the radiators for the room without storage unit, Q_{sav} = energy savings , P = relative energy savings.

	$Q_{s,i}$	$Q_{s,o}$	$Q_{s,\ell}$	Q_{ven}	Q_{rad}	Q'_{rad}	Q_{sav}	P
	kWh/year							%
Reference Case	696	689	7	169	2630	3389	590	17.4
A_w = 6.1 m^2, $(UA)_r$ = 66 W/^oC	394	399	-3	140	2681	3103	282	9.1
A_w = 8.1 m^2, $(UA)_r$ = 71 W/^oC	525	523	3	153	2590	3151	408	12.9
A_w = 10.2 m^2, $(UA)_r$ = 76 W/^oC	619	614	6	160	2576	3243	507	15.6
$(UA)_r$ = 73 W/^oC	678	669	10	165	2201	2936	570	19.4
$(UA)_r$ = 77 W/^oC	688	681	8	167	2417	3166	582	18.4
$(UA)_r$ = 91 W/^oC	709	705	5	173	3133	3900	594	15.2
S = 447 Wh/^oC	814	809	5	187	2711	3595	697	19.4
S = 1392 Wh/^oC	500	490	10	143	2529	3066	394	12.9
S = 1928 Wh/^oC	431	421	10	132	2492	2936	309	10.5
S = 4167 Wh/^oC	271	262	10	105	2435	2678	138	5.2

Table II Part 3. Parameter variations in accordance with point (E), (F) and (G) in section 4.4. A_w = window area, $(UA)_r$ = overall heat loss coefficient for the room, S = heat capacity of the building structure, $Q_{s,i}$ = heat supplied to the storage, $Q_{s,o}$ = heat delivered by the storage, $Q_{s,l}$ = net heat loss from the storage, Q_{ven} = energy consumption of the ventilator, Q_{rad} = energy supplied by the radiators for the room with storage unit, Q'_{rad} = energy supplied by the radiators for the room without storage unit, Q_{sav} = energy savings, P = relative energy savings.

	$\Delta\tau$	T_o	T_h	ΔT_m	$T_{a,i}$	$T_{a,m}$	T_s	ΔP	ΔQ_ℓ	$(UA)_s$	h
	hours	^oC						Wh		$W/^oC$	$\frac{W}{m^2\ ^oC}$
A	1.017	23.4	27.2	1.4	32.0	28.9	24.6	940	60	117	11.2
	1.567	23.6	30.4	2.0	35.4	32.2	27.8	1470	110	114	11.0
	0.717	23.7	33.3	1.0	38.3	35.1	30.7	660	60	110	10.6
	1.617	23.4	36.3	1.8	41.3	38.1	33.8	1530	150	114	11.1
B	0.833	23.0	26.5	1.1	31.3	28.1	24.1	750	50	118	11.7
	0.917	23.3	28.4	1.0	33.5	29.9	25.9	830	50	112	11.9
	1.750	23.9	30.5	2.0	35.8	32.0	28.4	1720	110	124	14.3
	0.750	24.5	32.2	0.6	37.7	33.7	30.4	630	40	108	13.3

Table III Mean values of different temperatures and heat flows during the period τ and the corresponding heat transfer coefficients $(UA)_s$ and h, in accordance with section 5.3.

Experiment A: the contents of the containers is water.

Experiment B: the contents of the containers is a 46% salt-water mixture.

$\Delta\tau$ = length of the period, T_o = temperature of the air outside the hot box, T_h = temperature of the air space in the hot box, ΔT_m = increase of the mean temperature of the insulation of the hot box, $T_{a,i}$ = temperature of the air at inlet of the unit, $T_{a,m}$ = mean temperature of the air in the unit, T_s = mean temperature of the salt-water mixture, ΔP = energy supplied by ratiator and ventilator, ΔQ_ℓ = energy transferred from the air space in the hot box to the insulation of the hot box.