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PASSIVE SOLAR-ENERGY AIR-HEATING WALL PANELS

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ABSTRACT

The development of products which enable passive solar-energy air-heating to be integrated into the heating strategies of public, commercial and industrial buildings is described. These buildings are, in general, only occupied significantly during the day; consequently the bulk of heating demand coincides with the period of solar gain. In these circumstances collected solar heat should be delivered with the minimum of delay.

The design and operation of units which are capable of supplying solar heated air in this manner is outlined. These are passive, natural-circulation air-heating collectors, also known as natural-convection air-heaters, or thermosyphoning air panels.

Four methods of retrofitting such solar collectors to non-domestic buildings have been identified, one of which, the overcladding collector, has not been proposed previously. Problems associated with the successful installation and operation of these units have also been considered.

The relative merits of a number of methods of testing passive solar-energy air-heating collectors have been investigated. A method of determining instantaneous collector efficiency based on the measurement of glazing temperature, inlet and outlet air temperature, ambient temperature and insolation has been developed.

Three novel design proposals have been presented: i) a collector constructed with the insulation fitted outside, rather than inside, so that the metal body of the collector may provide more symmetrical heating of the air flow than the conventional arrangement, ii) an absorber which consisted of parallel ducts to increase the rate of heat transfer to the air, heating it symmetrically, (iii) a hinged air-deflector for conversion from the heating to the ventilation mode.

'Stand a little less between me and the Sun'

DIOGENES

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NOTATION

A	Flow Area (m^2)
A, B	Constants in equation (10)
b	Section depth (m)
C	Constant in equation (22)
C_p	Specific heat capacity (J kg^{-1} K^{-1})
D_e	Equivalent hydraulic diameter (m)
D_h	Hydraulic diameter (m)
f	Dynamic friction loss coefficient
F'	Collector efficiency factor
g	Acceleration due to gravity (9.81 ms^{-2})
h	Vertical height (m)
H	Vertical height (m)
I	Total global insolation (Wm^{-2})
L	Length (m)
$\dot{m}$	Mass flow rate (kg s^{-1})
N	Number of cover glazings in equation (7)
P	Pressure (N m^{-2})
q	Rate of Heat gain (W m^{-2})
r	Exponent in equation (12)
Re	Reynolds number ($= \rho v D_h/\mu$)
s	Steady-state limit
T	Temperature (°C or K)
$\bar{T}$	Temperature corresponding to mean density (°C or K)
U_L	Overall loss coefficient (Wm^{-2} K^{-1})
U_t	Top loss coefficient (Wm^{-2} K^{-1})
v	Velocity (ms^{-1})
w	Section width (m)
y	Distance along the collector (m)
Z	Height above Datum (m)

<u>Greek Symbols:</u>

α	Absorbtivity of the plate
ε	Emissivity
μ	Dynamic viscosity (Ns m^{-2})
ρ	Density (kg m^{-3})
$\bar{\rho}$	Mean density (kg m^{-3})
σ	Stefan-Boltzman constant ($5.67x10^{-8}$ W m^{-2} K^{-4})
τ	Transmittance
ν	Kinematic viscosity (m^2 s^{-1})
η	Efficiency

<u>Subscripts</u>

a	Ambient
Aux	Auxiliary
b	Back (i.e. room), base
c	Cover
e	Equivalent
f	Front (i.e. collector)
g	collector glazing
i	Collector inlet, irreversible losses
L	Load
LV	Lower vent
Max	Maximum
Min	Minimum
o	Collector exit
O	Reference loss coefficients : front (collector)
p	Plate
r	Radiative
R	Room or living space
s	Thermosyphonic

S	Storage wall
u	Useful, net, U-tube
UV	Upper vent
x	Bottom of the back section
1-2	Between 1 and 2
1,2,3,4	Location 1, 2, 3, 4.

Chapter 1

LITERATURE SURVEY

1.1 DEFINITION

When solar radiation has passed through the glass cover of a thermo-syphoning air-heating solar collector, it is converted to thermal energy at the absorber plate. This thermal energy raises the temperature of the column of air within the flow channel of the collector which rises under the action of buoyancy forces and enters the building through an upper vent. Replenishing air from the building is drawn into the collector, via a lower vent and replaces the heated air. This process will continue whilst there is a sufficient temperature difference between the air in the building and the air in the flow channel.

1.2 INTRODUCTION

An appreciable amount of information has been published on passive air-heating solar-energy systems. In view of its disparate nature, it has been necessary to rationalise the available information; an attempt has been made to identify those design options which are most appropriate for large-scale commercial exploitation. It is evident that there has been considerable diversity in the design, construction and evaluation of these air-heaters. The optimisation of collector design must take into account not only performance but also materials usage, production techniques, installation costs, and operational considerations such as the subjective "response" of potential users.

During the course of this study it has become apparent that a rigorous method of evaluating the performance of air-heating systems, of this type, has yet to be developed. Furthermore, effective reliable mechanisms for the control of air flow through these devices also need to be developed.

These issues have been addressed, and design solutions proposed, but it is evident that a project of this magnitude demands a significant commitment of resources before the results and conclusions of this study can be validated or confirmed.

1.3 AIR VERSUS LIQUID SOLAR HEATING SYSTEMS

In air-heating solar-energy space-heating systems, the warm air emerging from collectors can be used to heat a building directly. However, although air heaters are considered to be cheaper to buy and to maintain, as converters of solar energy to heat they are inherently less efficient than comparable liquid collectors [Duffie and Beckman, (1980)]. Consequently, large collector areas are required for air heaters than liquid systems to meet the same heat load. This can be a significant disadvantage both economically and aesthetically. Liquid systems usually require heat exchangers, both liquid-to-liquid and liquid-to-air (i.e. wall-mounted "radiators"), between heat collection and delivery. The plumbing costs of connecting up liquid-filled collectors can thus approach the cost of the units themselves, [Smith (1980)]. Air does not freeze or evaporate whilst air leakage is not a serious problem. If a liquid system is allowed to freeze, or run dry, serious damage can occur, whilst leakage can have disastrous consequences for the building and occupants; the building fabric may be damaged and domestic hot water contaminated.

The requirements for an idealised medium for the transfer of heat from a collector are tabulated in Table 1.

o Chemically Unreactive
o Physically Stable
o Inexpensive
o High Heat Capacity
o Good Thermal Capacity
o Non-Flammable
o Non-Toxic
o Non-Viscous
o Non-Corrosive
o Non Energy-Intensive

Table 1. Desirable Properties of a Medium for the Transfer of Heat from a Solar-Energy Collector.

Myers (1984) noted that air does not have a high specific heat capacity or thermal conductivity but does satisfy all of the other broad requirements outlined above.

Two explanations have been offered why liquid solar energy systems are currently more popular than air systems, [Anderson (1977)]: (i) it is easier to carry out the analysis of heat transfer and other related phenomena for liquid systems, and (ii) the ready availability of information on liquid systems means that they are the easiest to develop, so perpetuating the imbalance.

1.4 RELATIVE MERITS OF PASSIVE AND ACTIVE SOLAR-ENERGY AIR-HEATING SYSTEMS

Two distinct approaches to solar space heating can be identified; "passive", which operates exclusively on the energy available from the sun, and "active" which requires additional energy to collect and distribute the solar energy. The pertinent characteristics of the two systems are summarised in Table 2.

Passive air-heating systems have the advantages of simplicity of design, and maintenance. Usually built with standard building materials, they provide a reliable and silent heat input. A classification of passive systems is shown, Fig. (1).

Active air heating systems offer the flexibility of thermostatically-controlled heat delivery and of collectors which are independent of the building. Providing that there is adequate space for the necessary ductwork, this flexibility enables active systems to be more readily adapted to existing structures.

The past popularity of active systems can be attributed to the fact that they represent an identifiable "product" for industry to develop and sell, [Elkington (1984)]. It is also evident that there is a greater margin for error in the design of active, rather than passive, systems. However, passive solar air heating has many benefits. Balcomb et al (1980), in particular, stressed their economic viability:

	DEFINITION	ADVANTAGES	DISADVANTAGES
PASSIVE	OPERATE EXCLUSIVELY ON ENERGY AVAILABLE FROM THE SUN	• SIMPLICITY OF DESIGN • STRAIGHTFORWARD OPERATION • LOW MAINTENANCE REQUIREMENT • RELIABILITY • SILENCE • LOW INITIAL COST • CONSTRUCTED WITH CONVENTIONAL BUILDING MATERIALS	• MAY SIGNIFICANTLY INFLUENCE THE SHAPE, ORIENTATION AND INTERNAL LAYOUT OF BUILT FORM • DESIGN OF PASSIVE COMPONENT MUST TAKE INTO ACCOUNT THERMAL PERFORMANCE OF BUILDING • OFTEN REQUIRES "INTELLIGENT" USE OF SYSTEM BY OCCUPANTS • CONTROL AND DISTRIBUTION OF HEAT OUTPUT NOT ALWAYS ADEQUATE • LOW EFFICIENCY
ACTIVE	REQUIRE ADDITIONAL ENERGY TO ACCOMPLISH THE COLLECTION & DISTRIBUTION OF SOLAR HEAT	• THERMOSTATICALLY CONTROLLED HEAT DELIVERY • COLLECTOR CAN BE TOTALLY INDEPENDENT OF THE BUILDING • READILY ADAPTED TO EXISTING STRUCTURES (APART FROM DUCTWORK) • HIGH EFFICIENCY	• HIGH INITIAL COST • SIGNIFICANT RUNNING AND MAINTENANCE REQUIREMENTS • COMPLEX

TABLE 2. THE RELATIVE MERITS OF ACTIVE AND PASSIVE AIR HEATING

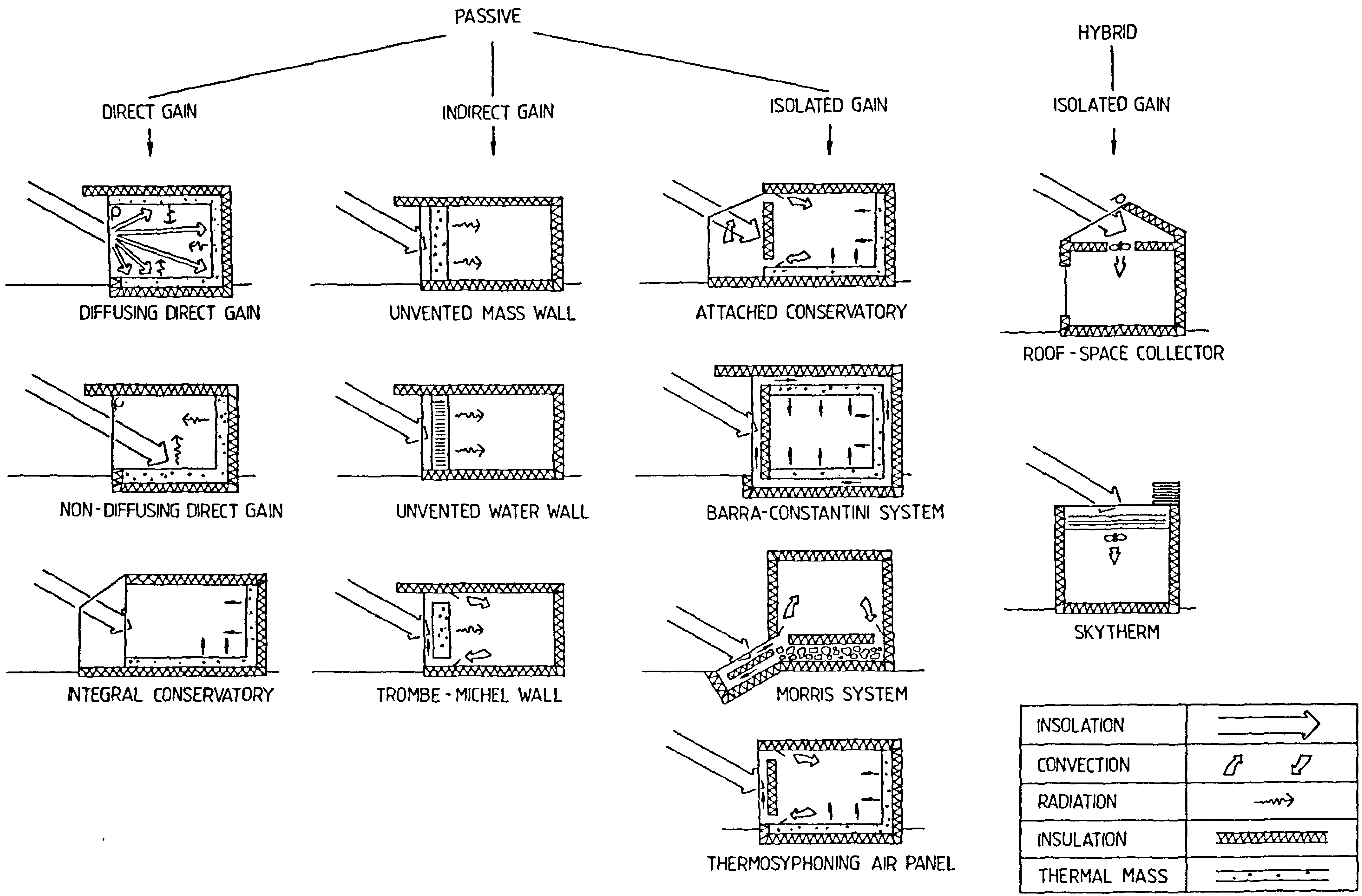


Fig.1 Taxonomy of passive solar systems after Achard and Gicquel (1986) and Holtz and Place (1979)

"For the life of a building a passive system should continually maintain, if not improve its value in at least equal proportion to the rest of the building. It should require no more maintenance than any other wall or roof. This is particularly true when the passive system fulfills the dual role of admitting solar energy and forming an integral part of the building's surface and structure".

Many "Passive Systems" in fact, incorporate insulation or vents which require manual adjustment daily. If the occupants are not motivated sufficiently to complete such tasks these systems do not perform well. A leading "solar architect" has been quoted as saying, : "I'm never putting in moveable insulation again. Even if clients claim they are committed to using it religiously, they don't..." [Bliss (1985)]. Passive systems which incorporate the minimum number of moving parts are preferred.

Unfortunately the reaction of occupants to passive systems is not always favourable. In a report which considered the "cultural" implications of providing low-income families with passive solar space-heating devices, Tukel et al (1978) noted that even the provision of free materials, labour and reduced fuel bills could not overcome aesthetic or other, even less tangible, subjective reservations.

Homeowners who are involved actively in the design and construction of their solar heating systems are more likely to ensure that the systems operate efficiently than residents who are not concerned directly with the introduction of solar heating. Moreover, the occupants of public or commercial buildings may not feel that there are sufficient incentives for them to co-operate in the day to day running of passive heating systems, as they neither see, nor pay directly, the heating bills. This places considerable constraints on the designer of passive solar-energy air-heating components for use in these buildings.

1.5 PASSIVE SOLAR-ENERGY AIR HEATING

This may be achieved by one of three methods:-

1) Direct Gain, in which the living space is directly heated by solar radiation. This is the most commonly adopted passive design, as the use of living space for the collection and storage of heat is an inexpensive and well-tried method. Disadvantages include daytime glare, overheating, loss of privacy, and the degradation of fabrics and materials exposed to ultraviolet radiation for prolonged periods. Direct Gain collection cannot operate at elevated temperatures because of the comfort requirements of the occupants. This limits performance.

2) Indirect Gain, in which solar radiation strikes a thermal mass which converts the absorbed radiation into thermal energy and transfers it into the heated space. Amongst the most popular and widely-researched indirect-gain solar-energy air heater is the thermal storage wall. This is essentially a structural wall which collects and stores heat simultaneously, releasing it into the building during non-gain periods. It consists of south facing glazing behind which a dark masonry surface absorbs solar radiation, heat is distributed by radiation and convection from the inner surface of the masonry.

Convective heat may be gained from a thermal storage wall by installing vents at the top and bottom of a wall. This connects the air-gap between the glazing and the wall to the interior of the building. When the air in this passage has gained sufficient heat it rises and enters the room via the upper vent. Air from the living space is drawn into the lower vent to replace the heated air and this is, in turn, heated. A natural circulation cycle or thermosyphonic loop is then established and it will continue to operate whilst there is an adequate temperature difference between the room air and the air between the glazing and the wall.

Trombe et al (1965) undertook the first major study of this type of wall, and subsequent tests on the Trombe house in Odeillo, in the Pyrenees, France, showed that approximately 36% of the energy incident on the glazing heated the building in Winter, [Mazria (1979)]. Indirect gain systems such as Trombe walls deliver energy without the disadvantages of glare and ultraviolet degradation of furnishing, associated with direct gain systems. However, the Trombe-wall is not suitable for North European climates as heat losses are too high during prolonged non-gain periods, [Lee (1979)]. Moreover, indirect gain systems are not suitable for many applications because of the long delays between the collection of solar energy and its delivery. The occupation of many commercial premises and public buildings occurs during daylight hours whilst Trombe walls release heat mainly at night, [Norton & Probert (1984)].

3) Isolated Gain collectors such as the thermosyphoning air panel, (TAP), overcome some of the disadvantages of indirect gain collectors by dispensing with heat storage and relying totally on convective heat gain. Heat input is almost immediate whilst heat losses during non-gain periods when the collector is isolated from the heated space, are low. This design is ideally suited to the task of providing daytime heat in cool or cold climates. A TAP operates in the same manner as the natural convection mode of a Trombe wall. However, the absorber is often made of metal, usually aluminium or steel, and the unit is insulated to prevent heat loss to, or from, the building. The problem most commonly associated with passive solar-energy systems is control of the heat output. This is not a problem for a TAP as all that is required is for an inlet or exit vent to be closed and the thermosyphoning process ceases. There has been comparatively little commercial development of the thermosyphoning air panel to date, but it has proved to be popular with some homeowners in the USA as it allows the introduction of solar heating on a small scale with the minimum of cost and inconvenience.

1.6 SYSTEM CONFIGURATIONS

In this section five configurations of the thermosyphoning air panel are outlined.

Front-Pass Collectors

The simplest solar-energy air-heaters have the absorber plate attached to the rear panel of the collector so that air passes between the glazing and the absorber Fig. (2a). The disadvantage of this configuration is that air convects over the glazing and heat is lost to the exterior, whilst a dust and smoke film may gradually form on the inner surface of the glass. This reduces the transmittance of the

Fig.2 Types of "standard" thermosyphonic air-heating solar-energy collectors

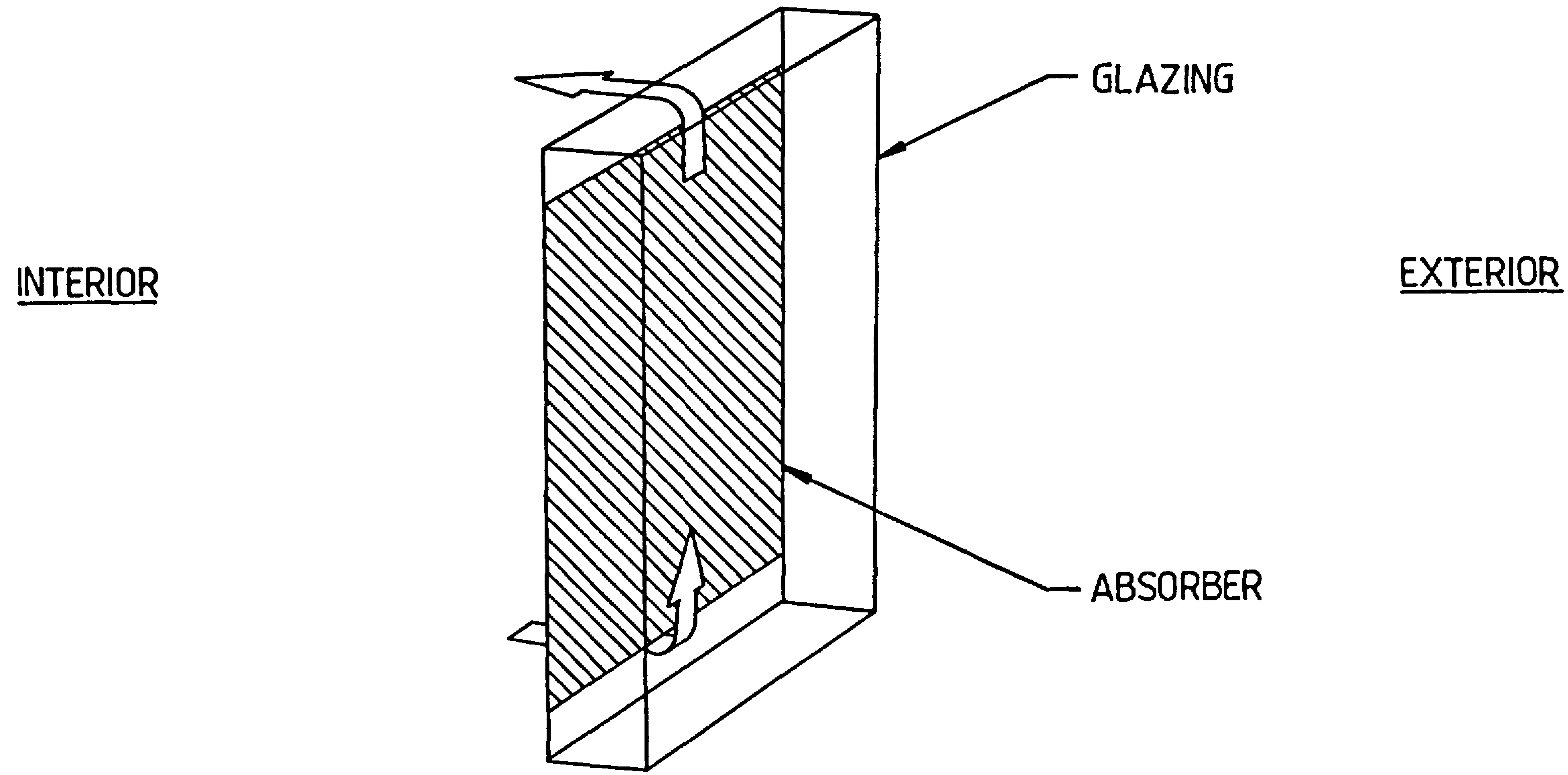
a) Front-Pass collector

b) Back-Pass collector

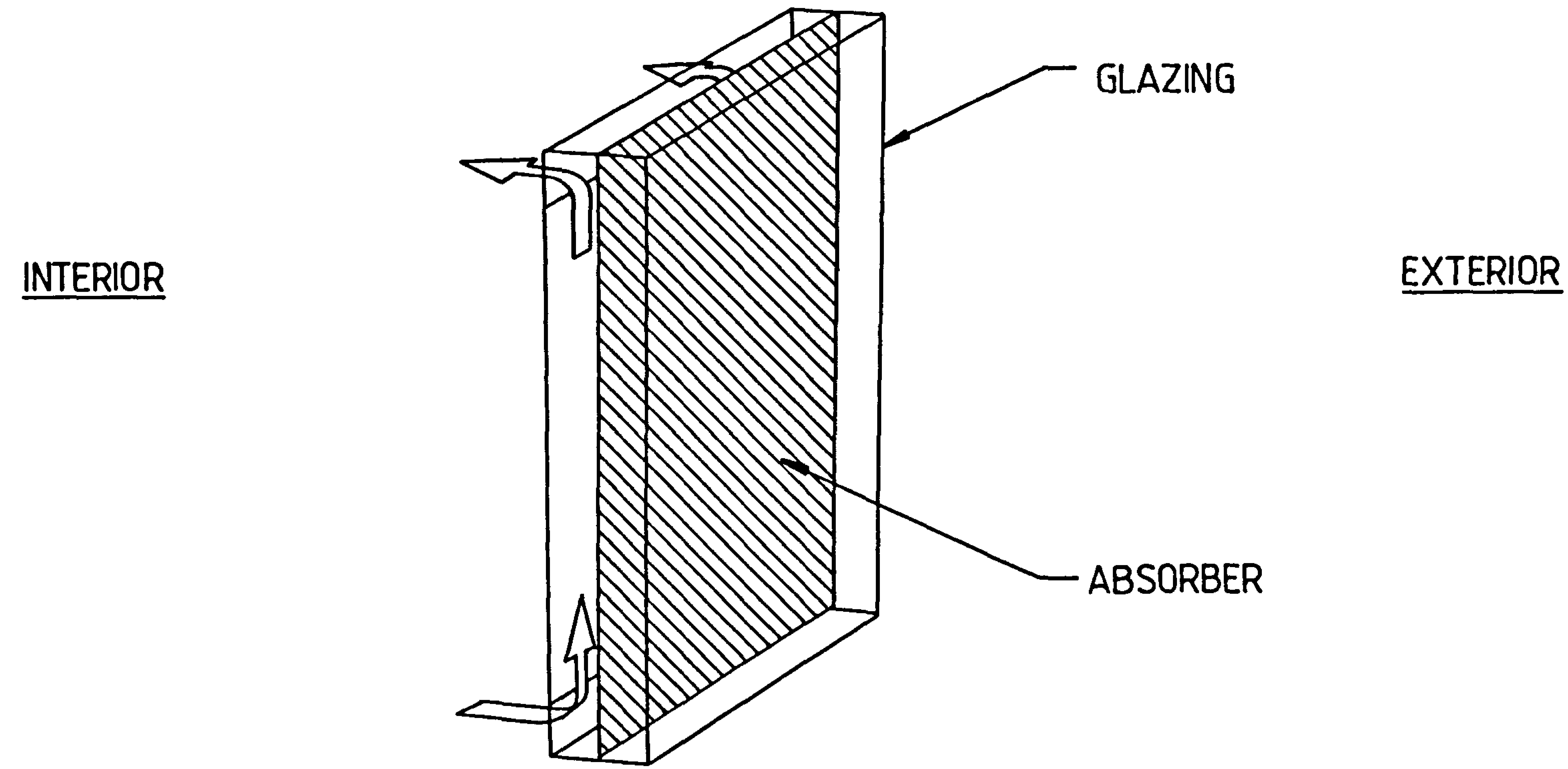
c) Dual-Pass collector

d) Matrix collector

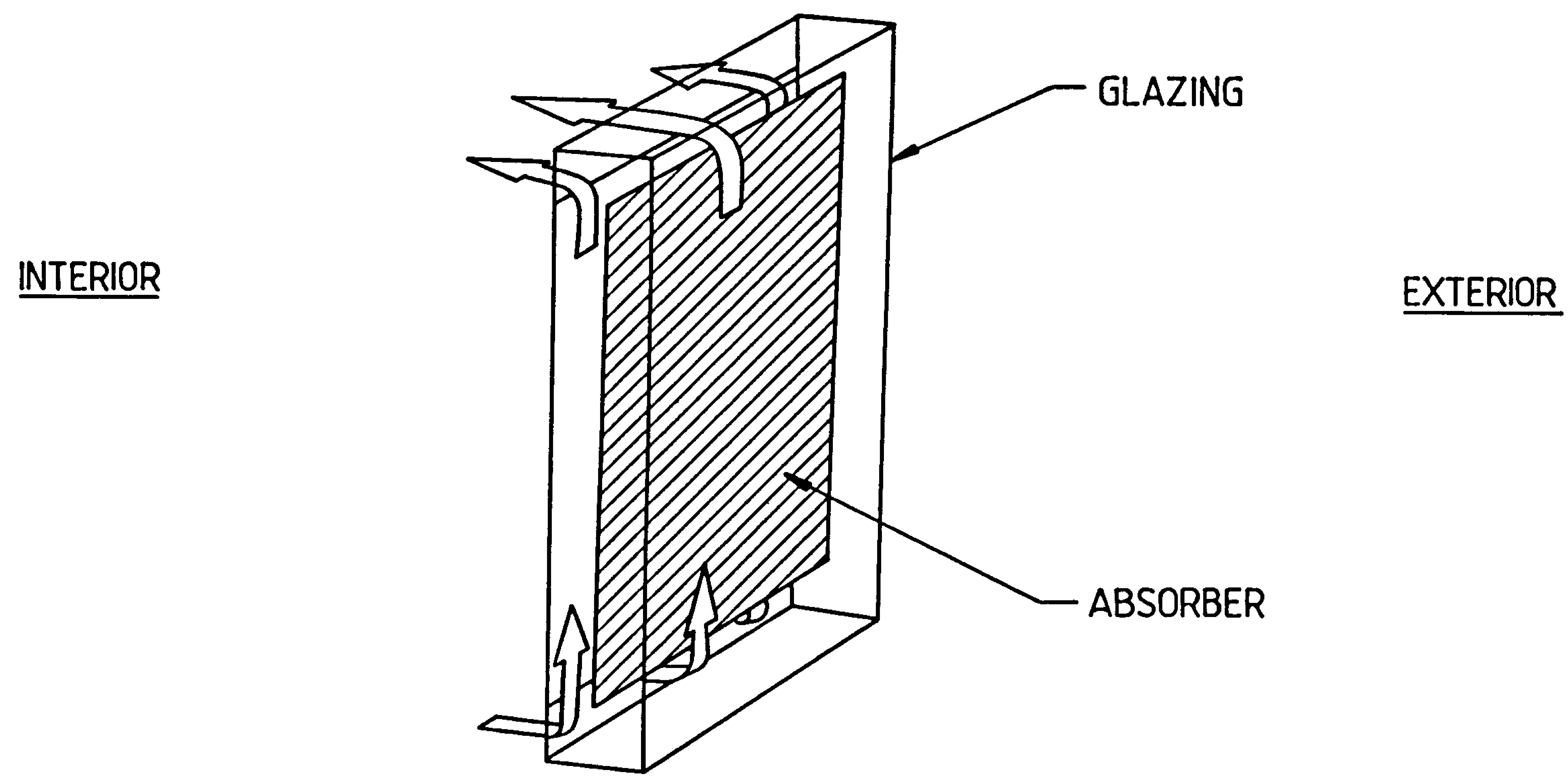
e) Parallel-Duct collector



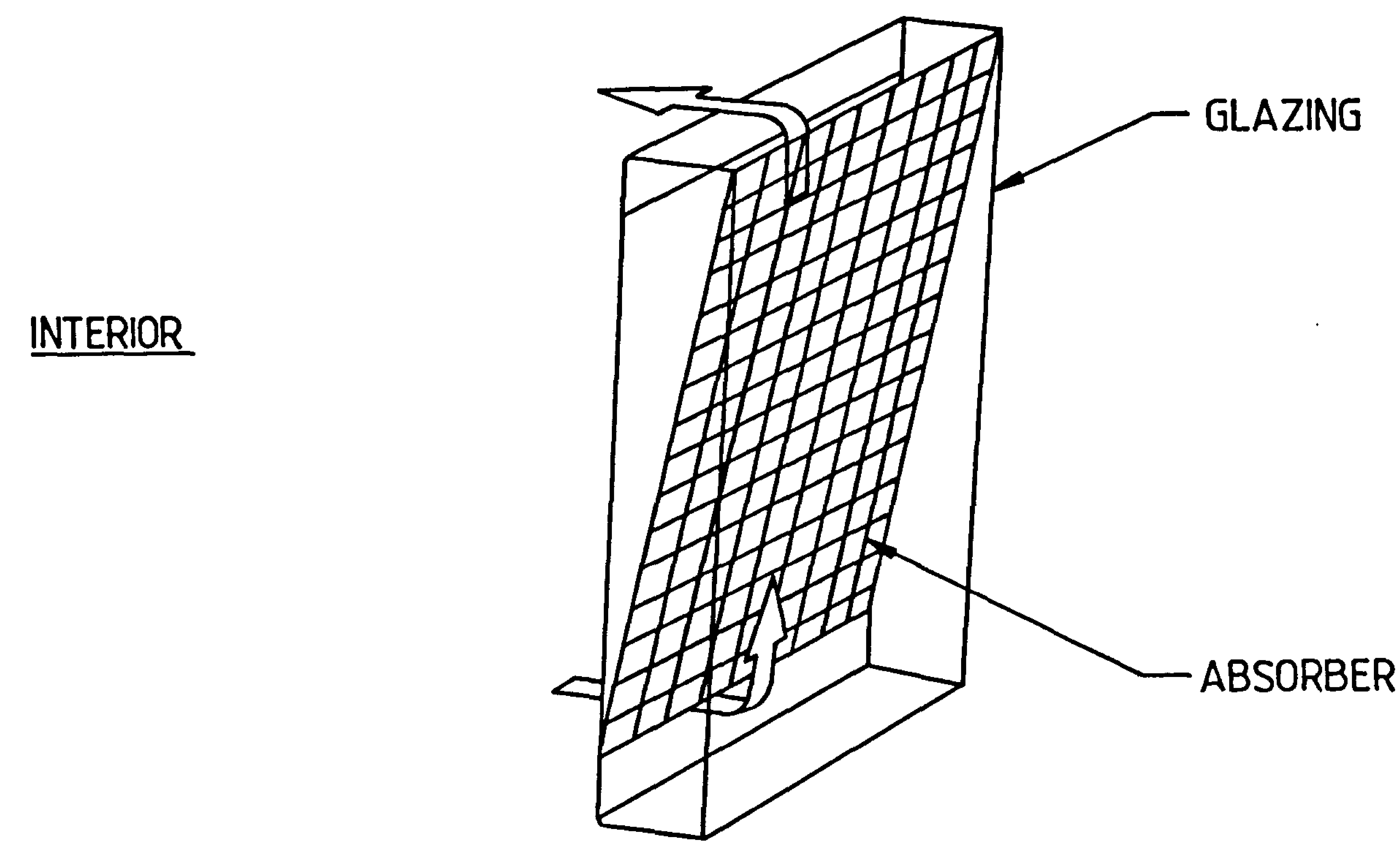
a) Front-Pass collector



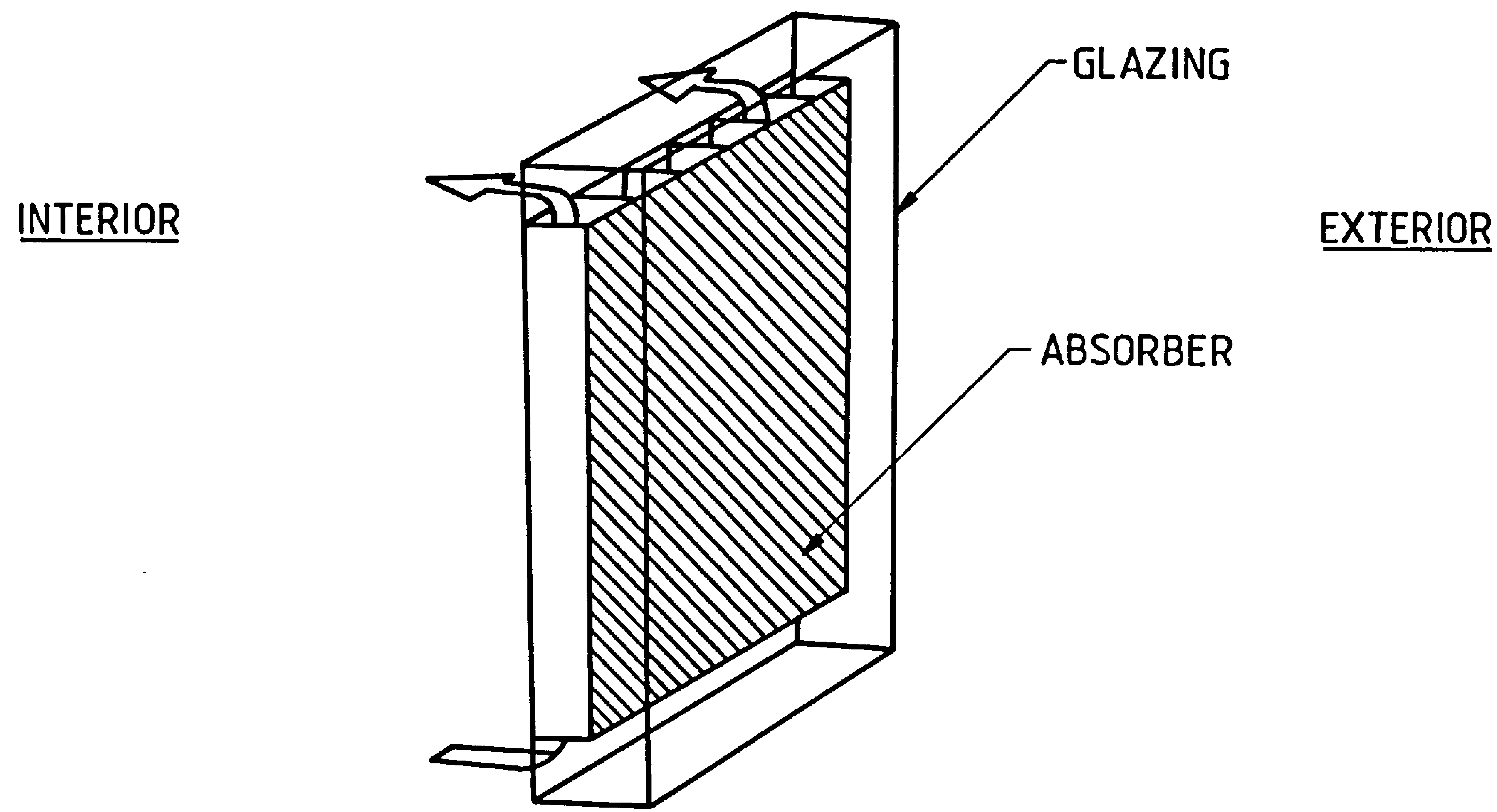
b) Back-Pass collector



c) Dual-Pass collector



d) Matrix collector



e) Parallel-Duct collector

cover and detracts from the appearance of the system. The amount of dust which accummulates will depend on the cleanliness of the room air, and the amount of condensation which forms on the inner surface of the glazing.

Back-Pass Collectors

One advantage of the Back-Pass design is that the moving air-stream is isolated from the glazing Fig. (2b). The layer of air contained between the absorber panel and the glazing provides insulation, whilst the inner surface of the cover is kept free from dust. In the Back-Pass configuration the air-tightness of the cover is not crucial to the performance of the collector. A collector which has the Back-Pass arrangement, and is well sealed, is potentially maintenance-free. However, the air-space between the glazing and the absorber makes the collector deeper than other designs and this may prove to be a disadvantage in certain applications. Insulants and sealants which may give off fumes and leave deposits on the cover should be avoided.

Dual-Pass Collectors

In a Dual-Pass collector air passes on both sides of the absorber, providing twice the heat transfer area of the previous designs Fig. (2c). Some heat from the front air-flow is lost to the glass with this arrangement and it shares the other drawbacks of the Front-Pass design whilst being more difficult to construct.

Matrix Collectors

The Matrix absorbers advocated by Baer (1977) and Morris (1978a) (1978b) consist of metal lath placed diagonally within the collector, creating a mesh through which the air passes Fig. (2d). An attractive feature of this design is that the incoming air passes over the glazing before being heated by the mesh, so less heat is lost through the case with Front-Pass or Dual-Pass collectors. However, Morris (1978b) observed that a mesh was effective only if there was enough air-flow to keep a slight pressure drop at the mesh itself. Slow flows allowed mixing of the hot and cold air on the glazing side, increasing losses and decreasing efficiencies. A solution put forward was to increase flow velocity by decreasing the area of the flow channel or using a slightly less permeable mesh, but this would produce a collector which is better suited to high levels of insolation.

Parallel-Duct Collectors

Collectors with an absorber consisting of a row of ducts has been constructed and evaluated during the course of this study, Fig. (2e). They are discussed later in the text.

Warm, moist air passes between the absorber and cover of Front-Pass, Dual-Pass and, to some extent Matrix designs. As has been stated, this will result in condensation and dirt deposits on the glass which will require cleaning periodically. It would appear that with these designs the glazing will have to be removed more frequently than with a Back-Pass collector, whilst a higher level of air-tightness is required. This may be acceptable in many applications but where large numbers of collectors are installed, the cleaning costs may be unacceptably high for any system other than the Back-Pass design to be considered.

According to Kornher and Zaugg (1984) thermosyphoning air panels should either be constructed with a matrix absorber in a single-glazed collector or with a sealed Back-Pass design in a double-glazed collector. The relative merits of single and double-glazing are reviewed in Table (3).

1.7 THE WORK OF E.S. MORSE

Professor E.S Morse, who was a world-famous late nineteenth-century American zoologist and ethnologist, designed and patented the first passive air heating and ventilating wall-mounted solar-energy collector [Morse (1881)], Fig (3). He had observed that when dark curtains were closed behind glazing they became very hot in strong sunlight, and warm air currents developed between the glass and the curtains, [Butti & Purlin (1981)].

In 1882, Morse's first solar air heater was installed at the Peabody Museum in Salem, Massachusetts described in the Scientific American of the 13th May 1882 as follows:

	Advantages	Disadvantages
Single-Glazing	. Low Cost . High Transmittance . Single-Glazed Collector not subject to high stagnation temperatures . Light weight	. Heat losses via the glazing are significant reducing collector efficiency . Reverse thermocirculation will take place at night therefore an effective form of backdraught damper is required. . Condensation
Double-Glazing	. Reduced heat losses from the collector . Less nocturnal reverse thermocirculation . Backdraught damper does not have to be as effective . Reduced condensation	. High cost . Lower transmittance . Heavy weight if glass is used . High stagnation temperatures . May require venting or shading mechanism

TABLE 3. Relative merits of single and double-glazing for passive air heating solar energy collectors.

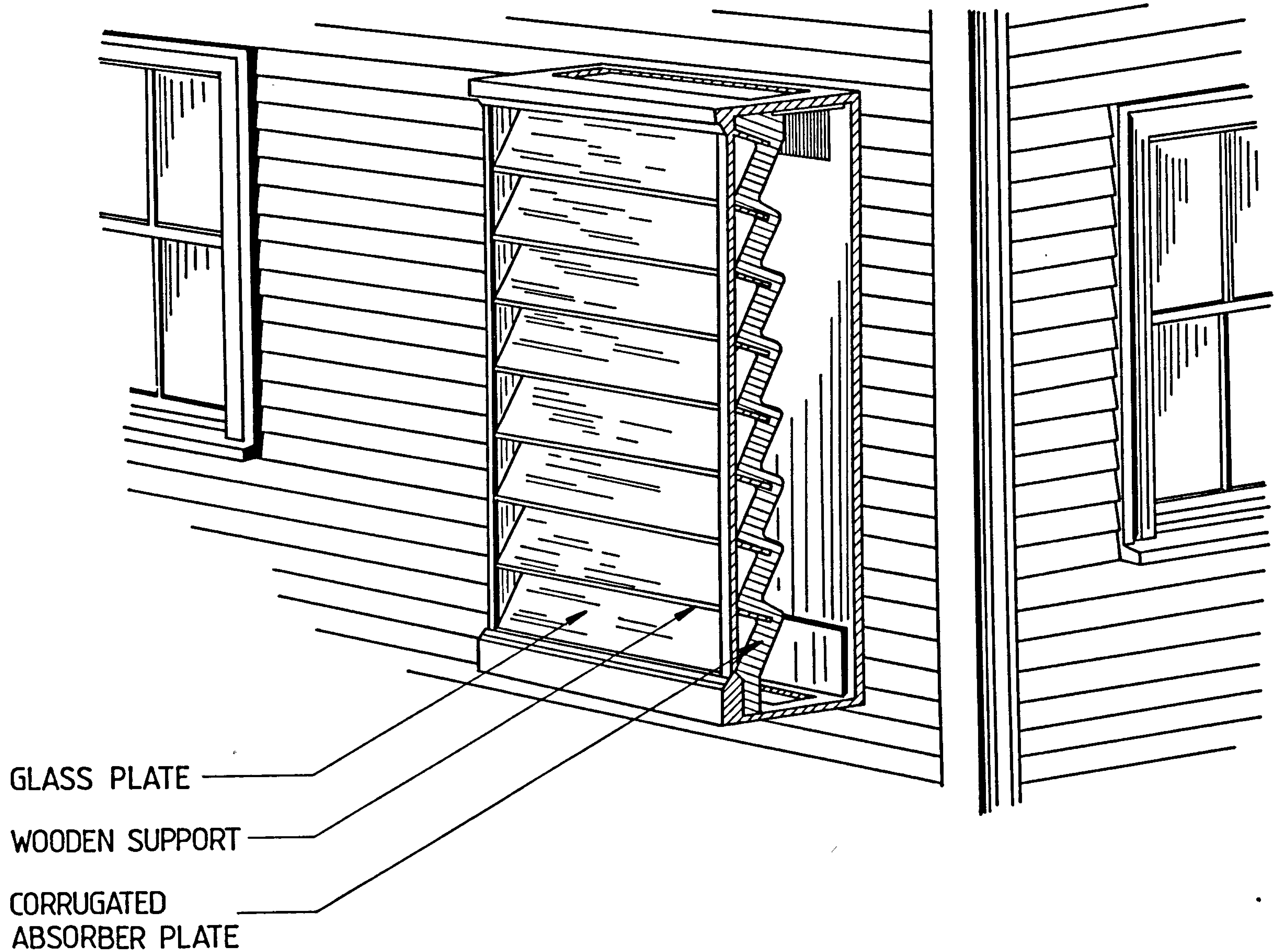


Fig.3 The Morse passive solar-energy air-heating and ventilating unit patented in 1881 (Morse 1881)

"His invention consists of a surface of blackened slate under glass fixed to the sunny side or sides of a house, with vents in the walls so arranged that the cold air of a room is let out at the bottom of the slate, and forced in again at the top by the ascending heated column between the slate and the glass. The outdoor air can be admitted, also, if desirable. The thing is so simple and apparently self-evident that one wonders that it has not always been in use."

The collector had three modes of operation; two for heating and one for ventilating. A damper at the foot of the collector allowed either outside or room air to be drawn up into the collector and a damper at the top allowed the heated air to exhaust into the room or to the exterior Fig (4).

The unit installed at the Peabody Museum, measured 3.96 m by 1.23 m and was fitted to a room 30.43 m long by 12.19 m wide by 6.4 m high. The box was glazed with eight separate sheets of glass each inclined at 30 degrees from the horizontal and the absorber plate was made of blackenned corrugated iron. There were air spaces of "several inches" between the glazing panel and the collector plate and the collector plate and the outside wall of the hall. This collector was designed to preheat ventilating air; outside air entered at the base of the collector, passed between the corrugated iron and the wall as it gained heat, rose to the top of the collector and passed into the room. There was no appreciable rise in the room air temperature when the unit was operating, but temperature rises of 14°C to 17°C were recorded as the air passed through the collector. The heater was capable of completely changing the room air in two days of operation moving 1698m^3 of air, and leaving the room "freshened and improved". [Mokray (1979)].

Temperature rises of over 22°C were recorded in a unit installed at the Boston Atheneum. This measured 1.981 m wide x 12.8 m long with a total area of some 25.3 m^2 and it was reported to be able to move 2264 m^3 of air per hour. [Mokray (1979)]. Although warm air from the heater lost much of its heat through a large skylight in the ceiling, the device helped to save the Athenium 11-22 kg of coal a day during the winter.

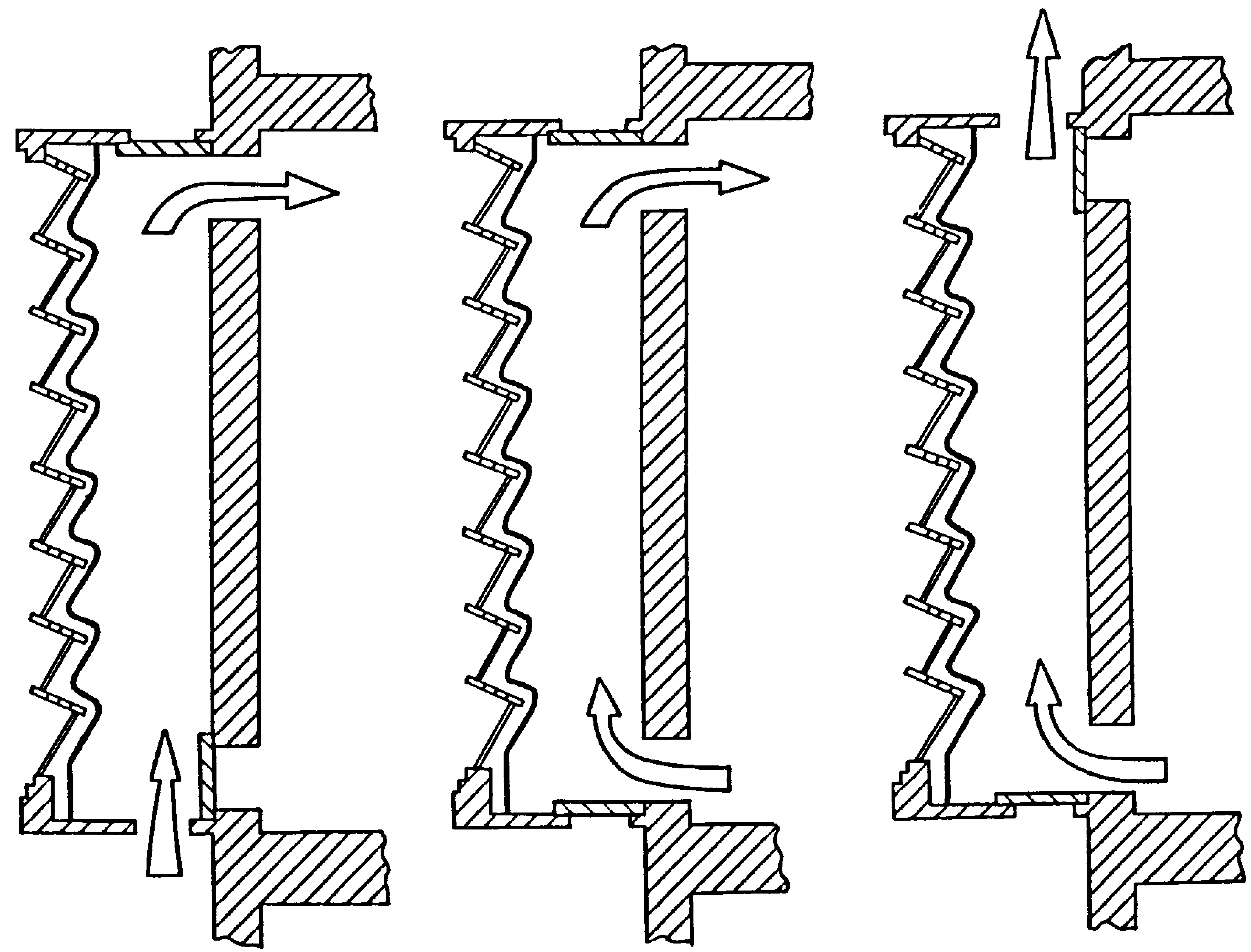


Fig.4 Three modes of operation of the Morse solar collector (Morse 1881)

Morse delivered a paper on April 9th 1884 to the M.I.T Society of Arts which reported on his findings. [Morse (1884)]. In a paper in response the following year, Woodbridge (1885) argued that Morse's heaters were not viable. Nevertheless, according to Butti & Purlin (1980), Morse was certain of the fuel saving potential of such devices.

> "The results were sufficiently satisfactory to show that in regions such as the great (American) Southwest where fuel is scarce, temperatures often low, and (there are) many sunny days, the invention could be of great service".

However, despite the extensive publicity that Morse's work received in his time, it is only in recent years that this type of passive solar air heater has been developed further.

Baer (1977) was amongst the first to propose design recommendations for thermosyphoning air heaters and Morris (1978a) (1978b) tested several designs based on this work and that of Trombe et al (1965). In Europe Barra (1980) developed an air heater which was designed to be an integal part of a building's structure and which overcame many of the shortcomings of the Trombe wall, Figs (5,6). The southern wall of the building contains a Dual-Pass thermosyphoning air panel and heated air from this collector circulates through ducts in the ceiling and floor. It transfers its energy to the thermal mass of these elements before returning to the bottom of the collector. The ceiling and floor supply heat to the living space by convection and radiation and are insulated to prevent heat loss to the exterior. A similar system for new buildings was evaluated by Bilgen & Chaaban (1982). However, in general thermosyphoning solar-energy air-heaters have been developed for retrofit applications.

Opinions differ as to whether the flow regimes generated in thermosyphoning air - heaters are predominantly laminar or turbulent. For example, Biehl (1981) compared theoretical and experimental results for thermosyphon air heaters, assumed turbulent flow, and claimed good agreement between his computer simulation and test data. Conversely Kornher and Zaugg (1984), who based their design recommendations on those of Baer and Morris, stated that laminar flow should be preserved for optimum performance. Mastrullo et al (1983)

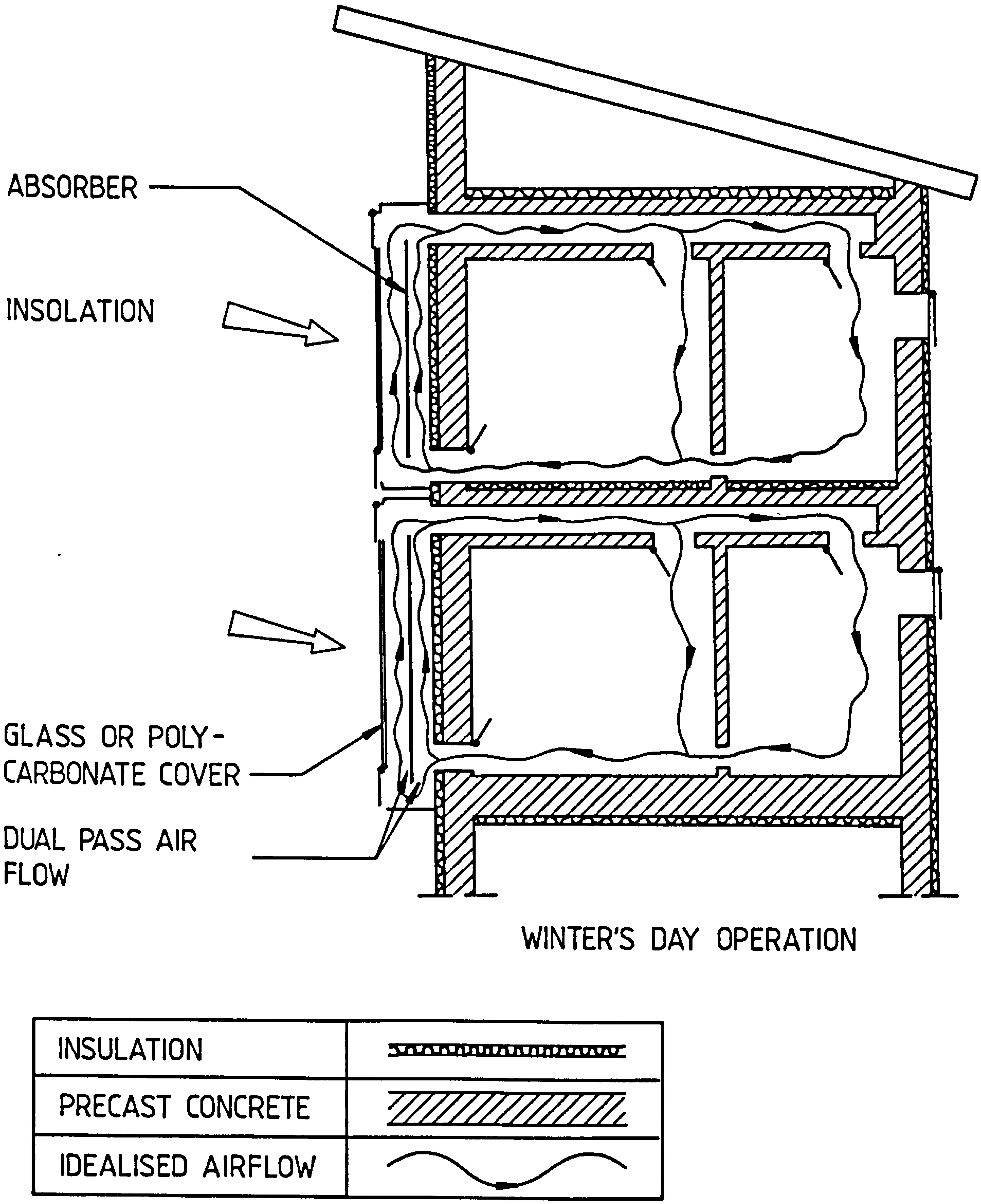


Fig.5 Schematic diagram of Winter's day operation of the Barra-Constantini passive solar system (Barra et al 1981)

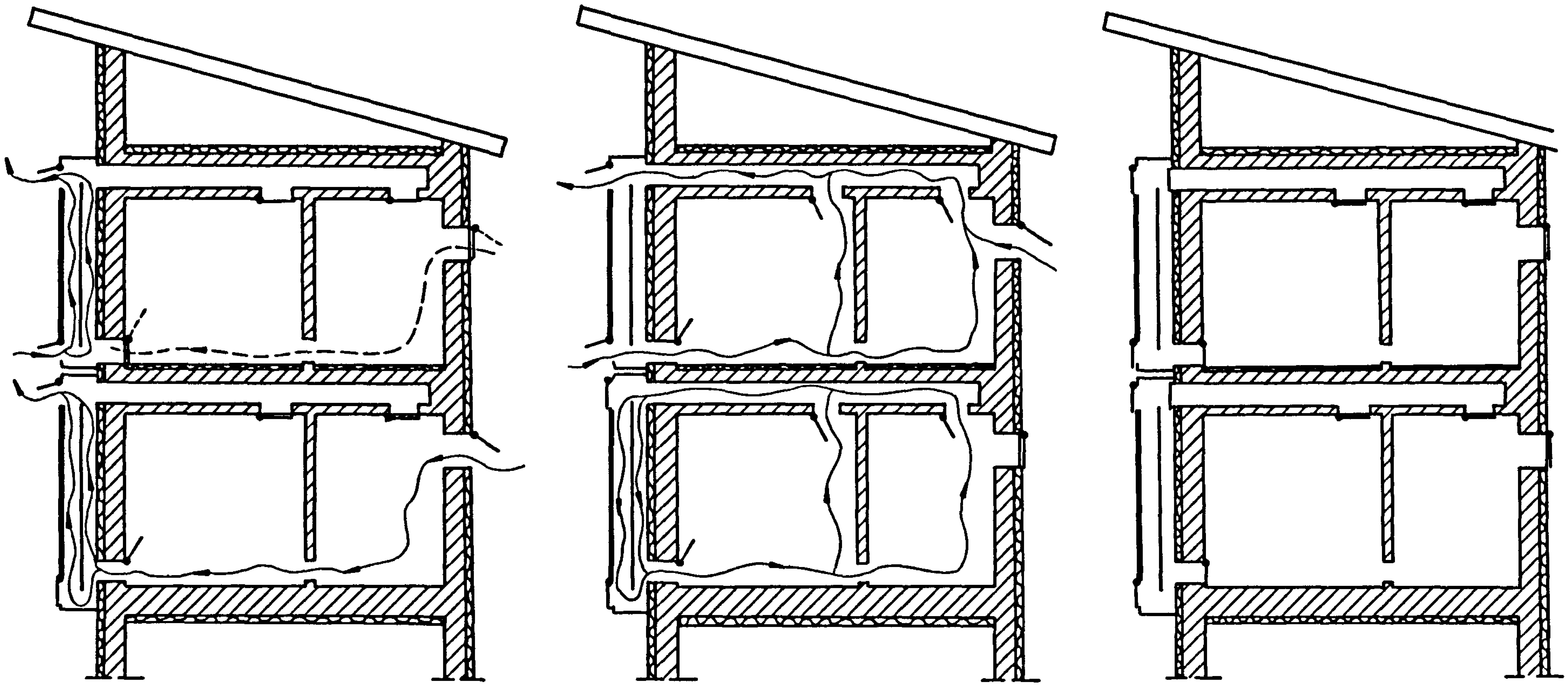


Fig.6 Three modes of operation of the Barra-Constantini Passive solar system (Barra et al 1981)

stated that, since the thermosyphon head is proportional to height, natural convection heaters are essentially concerned with the turbulent regime.

According to Kornher and Zaugg (1984), passive collectors required unrestricted air-flow to maintain an adequate flow rate: the air flow channel must be large, and all openings and changes in direction of the air flow should be very gradual. The best results are achieved by moving a large volume of air through a collector at low velocity. If the air flow path is too short or constricted, the flow rate of solar-heated air is reduced. It was argued that the objective is to design a collector which gets hot enough to develop an adequate flow rate without losing much heat to the ambient environment. The difficulty of measuring air-flow rate in these devices was also noted. It was argued that the best way to determine whether a collector is delivering an adequate amount of air, and performing efficiently is to observe whether the glazing is cool and whether there is an appropriate temperature differential between the incoming and outgoing air. Cool glazing and relatively low delivery temperatures were cited as being indicators of efficient operation.

As Cook & Morris (1978) observed;

> "natural convection collectors are difficult to analyse by conventional mathematical and experimental techniques. The natural self-regulating flow rates are a function of the relative temperatures within a system. The temperatures are a function of flow rates. Thus, it is a situation in which the two most important variables are dependent on each other."

Direct comparisons of the performance figures published for TAPs are difficult because of differences in the design and construction of the collectors tested and, in some instances, lack of sufficient experimental data and details of experimental procedure.

A theoretical study of laminar convection in natural circulation air-heating systems was undertaken by Barra and Carratelli (1979). Results obtained by Barra et al (1980) showed that for low insolation values larger channel depths produced better results because of the lower induced head losses. When insolation levels were higher, small channel depths produced better absolute efficiencies, and in a subsequent paper Barra & Franceshi (1985) suggested that this was because turbulent flow was more likely to take palce under these conditions. Some performance figures for a collector fitted to a house in Maple Cross, Watford, U.K. were also given. It produced an "average efficiency" of approximately 51% for an insolation level of 580W/m^2 in February. The temperature difference across the inlet and outlet was 18°C and the absorber temperature 48°C, with an average external temperature of 4°C.

Morris (1978b) stated that since it is the flow of air that actually transports the heat from the collector to the building, a design that maximises the flow will also maximise the efficiency of the system. A poor flow rate will result in high collector temperatures and increased losses through the glazing.

Mastrullo et al (1983), stated that higher temperature differences and lower flow rates are characteristic for narrower air channels. Tests carried out in Italy on a U-Tube collector 2m high, as shown in Fig. (7), with a channel depth of 0.2m, gave "efficiencies" of between 24% and 40% for a day in March. The seasonal useful collected energy was quoted as being 504 MJ/m^2 which gave an average value of 2.77 MJ/m^2 per day. Monthly efficiencies from October to March were quoted as 43%, 39%, 33%, 32%, 32% and 33%. The seasonal and monthly useful collected energy increased by 10% when the channel depth was increased from 0.05m to 0,2m or 1/20 - 1/10th the height of the collector.

It may be of more value to have collectors which are most efficient at lower levels of insolation, particularly in the early morning and late afternoon. Direct solar gains to buildings during these periods are lower than at midday and thus the contribution made by passive air heaters may be of greater value. It can be argued that the most significant heat input a passive solar air heater could make would be to reduce the amount of energy used in preheating a lightweight structure. If this is the case greater emphasis should be placed on the investigation of laminar convection within natural-circulation air-heating systems.

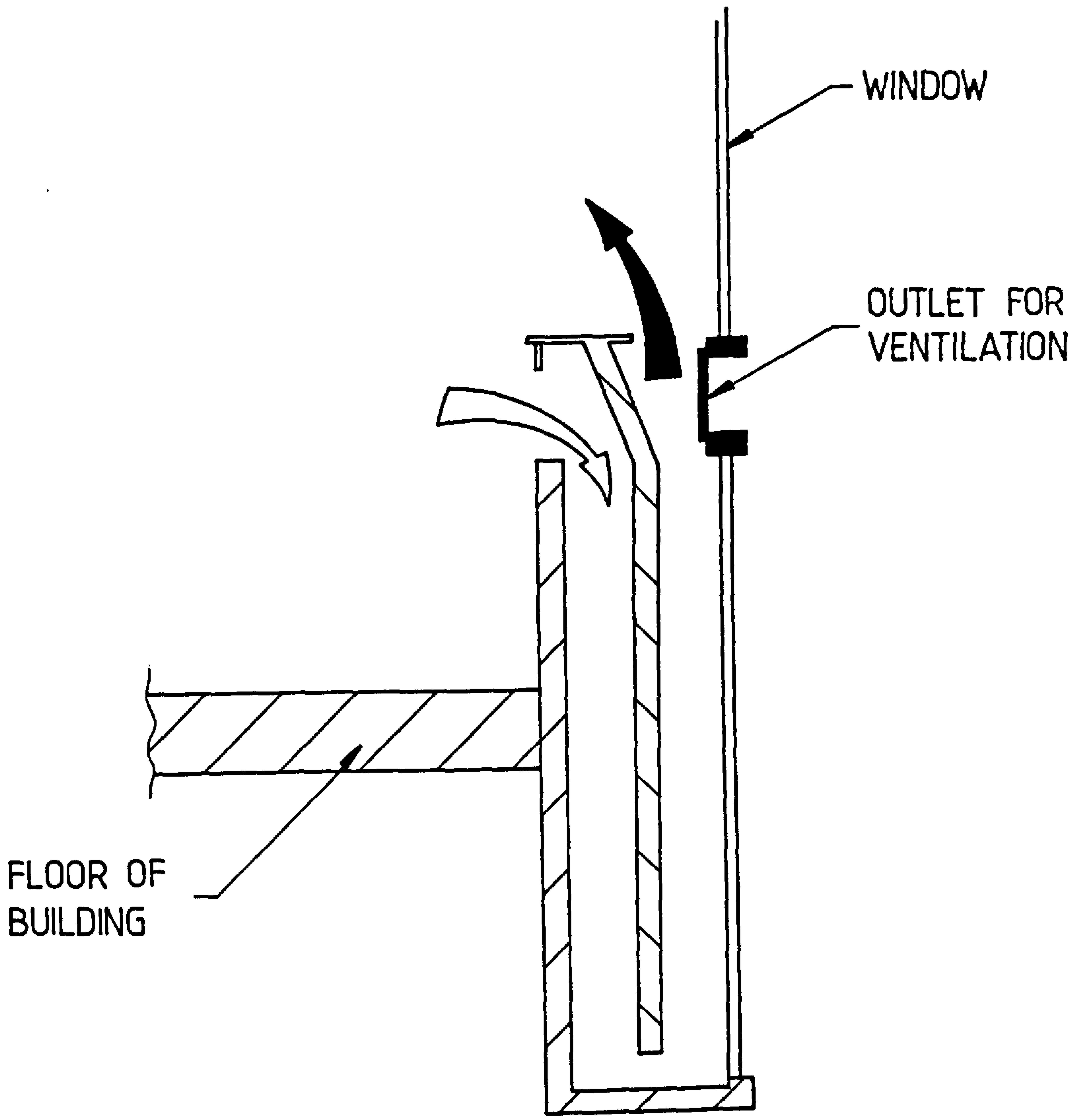


Fig.7 U-Tube thermosyphon solar-energy air-heating collector as installed at the National Scientific Research Centre, Odeillo, France,and as tested by Mastrullo et al (1983)

1.8 THE WORK OF W.S. MORRIS

W.S. Morris has published a number of papers concerned with the design and development of passive solar air-heating systems and matrix absorber U-tube collectors in particular. In general these have been installed in custom-built houses specifically designed to utilise passive solar air heating, located in the favourable semi-arid region of the USA below latitute 40°N, [Jager (1981)]. The collectors are installed below floor-level, to obtain optimum performance, and are coupled to rock-bed thermal storage. An example of this type of system is shown in Fig. (8). The Morris heaters were used usually to heat a thermal store which is usually placed under the floor of south-facing rooms, but in the example shown in Fig. (8), the store has been positioned to provide radiant heat to the south-facing room and to the adjacent north-facing room. Distributing passively-heated air to rooms which are at a distance from the point of collection is a problem often associated with passive systems. It can be resolved by installing fans and ductwork to move the air through the building but the solar heating regime can no longer be considered truly passive. The use of a strategically-positioned rock-store is an ingenious totally-passive solution but is not appropriate for premises only occupied during the day.

It was suggested [Morris (1978b)] that well designed air-cooled natural convection collectors can produce air flow rates and overall efficiencies equal to, or exceeding, those of active air-heating collectors.

A number of single-glazed collector designs were tested which featured 5 layer metal lath absorbers set at a diagonal to the flow channel. The collectors were installed on closed test units and comparisons were made between the different designs. Collectors performed consistently at total daily efficiencies of 40-45% when the collector inlet temperatures averaged 27.7°C above ambient. These results were obtained on "typical" sunny days in New Mexico, USA.

Tests were also conducted to determine the effect of restricting the flow channel area at the outlet of the collectors to simulate space limitations which may occur in real buildings. With unrestricted vents flow rates of 0.0177 - 0.0253 m^3/sec per m^2 of collector were

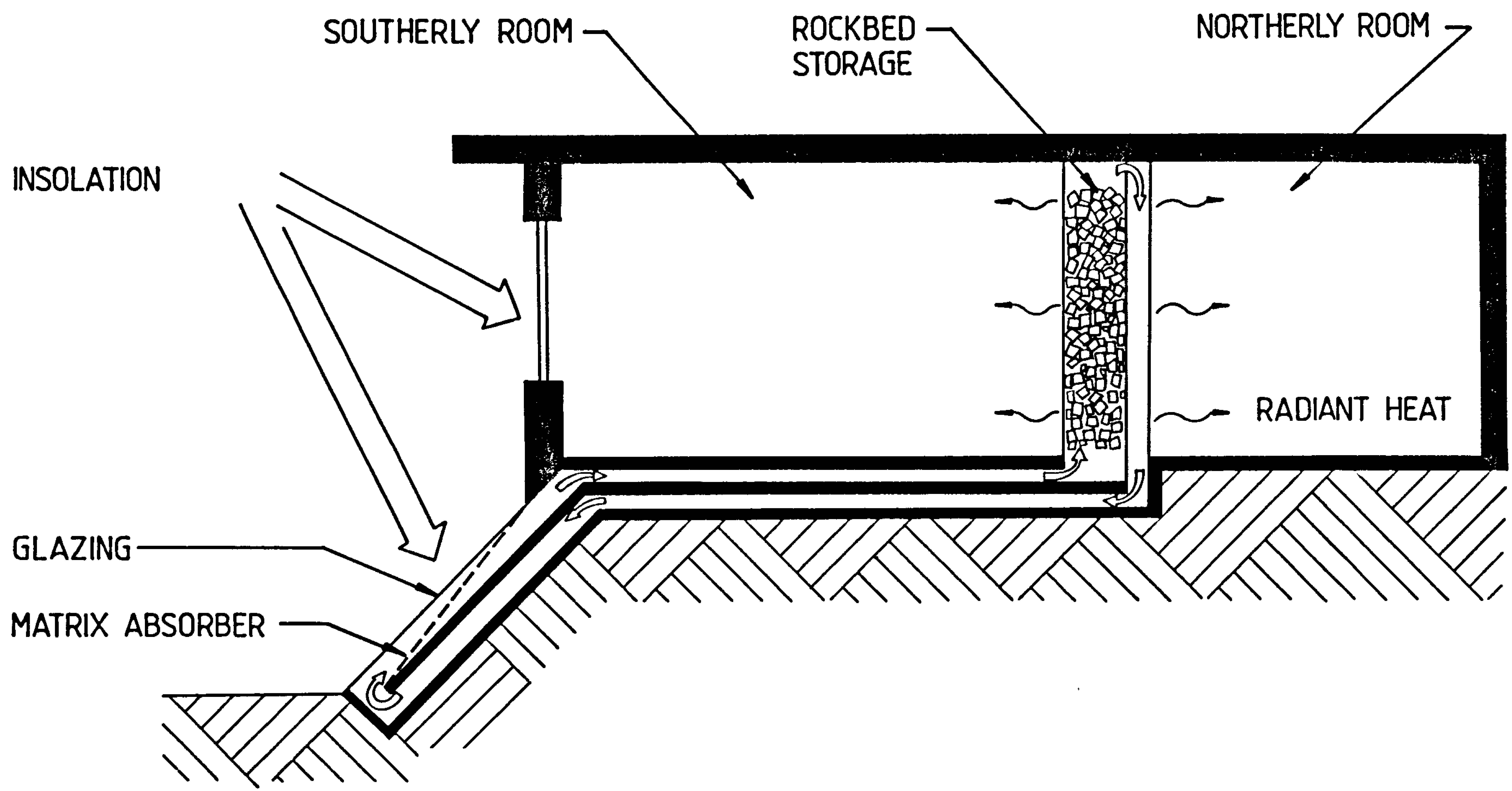


Fig.8 U-Tube thermosyphon solar-energy air-heating collector with rockbed storage (Morris 1980)

common on sunny days. When the flow rate decreased the temperature difference across the collector became greater to perpetuate airflow through the restricted system. In normal flat-plate solar energy collectors overall efficiencies are related directly to collector temperatures and flow rates should be kept high to minimise losses. The diagonal mesh absorber used in these tests minimised these losses by keeping only the coolest air exposed to the glass. Morris noted that the effects of reducing the outlet area were different for the different designs tested. Tests were not carried out on collectors fitted with flat plate absorbers.

A thermosyphoning air panel fitted with dampers was used as a control to assess the performance of an S-Loop collector Fig. (9a), which was designed to eliminate the need for dampers. Cool air is drawn up and down two inner channels before entering the collector channel; the two outer channels create a stagnant cold air pocket during non-gain periods and prevent reverse-circulation. Its measured efficiency was generally 90% of the control collector and it was stressed that the S-Loop collector would only prevent reverse-circulation if cold air is prevented from spilling over into the inner channel. If it is not the system will continue to cycle backwards and heat will be lost. A conventional air collector with dampers performed at 85-90% of the efficiency of collectors placed below the heated space despite the lower temperature differences and flow rates expected with such a geometry.

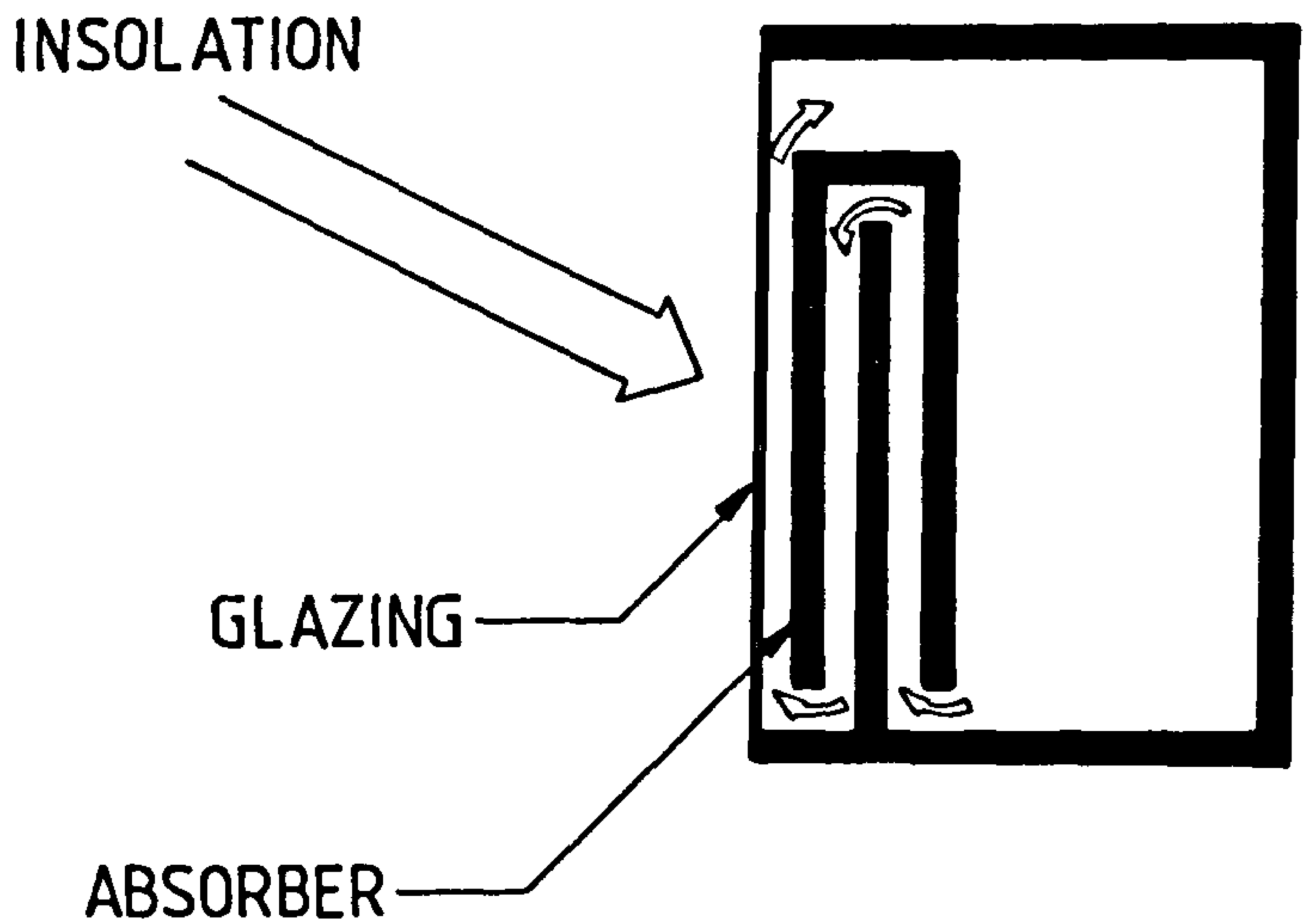
The importance of insulating U-Tube collectors from the building was stressed by Morris (1978b). If heat conducts from the building into the back channel of the collector during the night, it raises the air temperature and creates a temperature difference between the two air columns. It was noted that steady-state reverse circulation depended on the rate at which the entering room air experiences its heat loss through the glazing and how fast heat from the room warms the back column.

The position of vent openings relative to the glazing was also considered. It was suggested that glazing should not come up higher than the bottom vent of U-tube collectors otherwise an extra head of cold air on the glazing side will create a slow but steady reverse convection. Morris (1978a) estimated that losses of up to 20% of daily gains can occur in a poorly designed system.

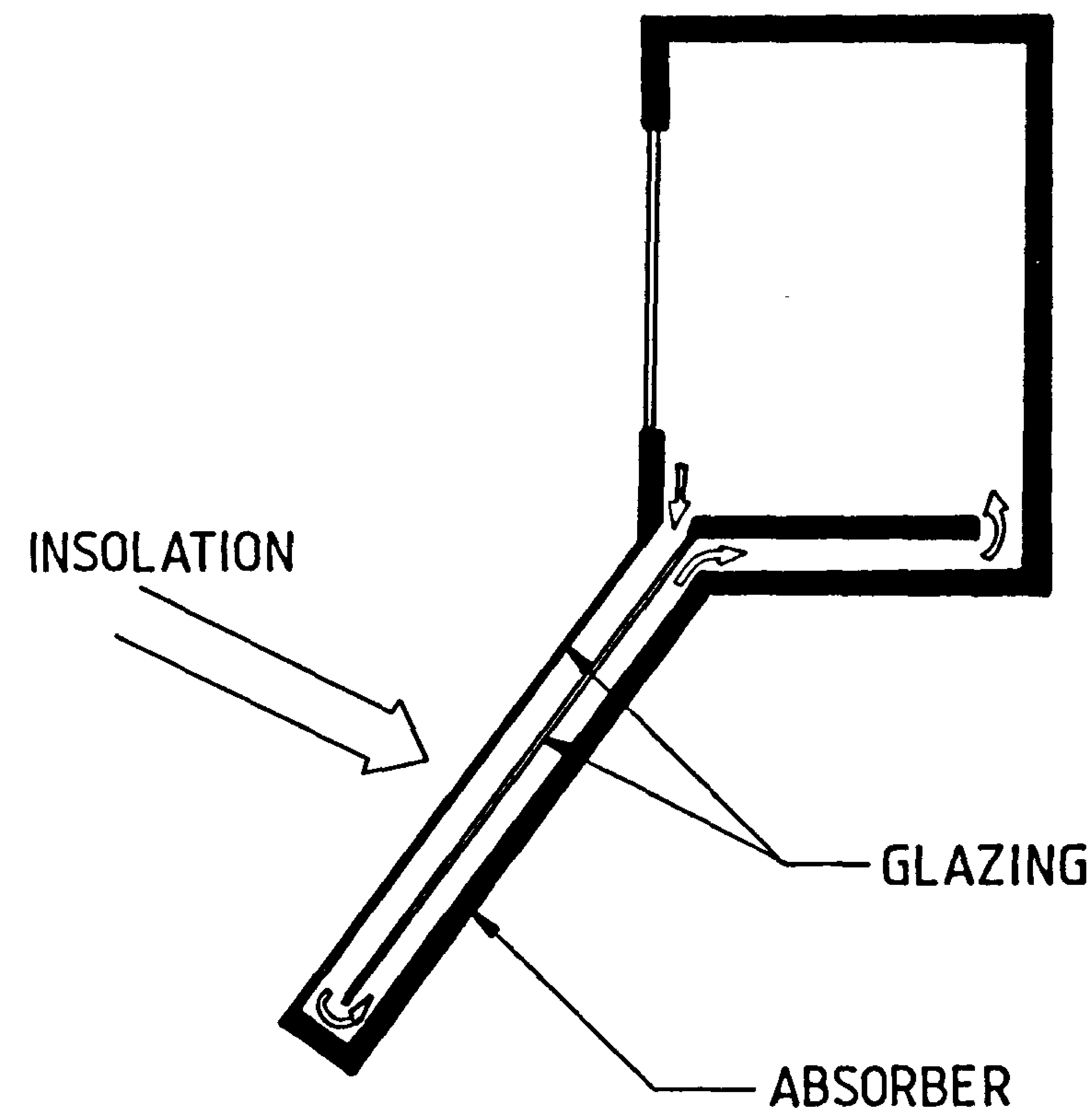
Fig.9 Types of multiple parallel-channel thermosyphon solar-energy air-heating collectors

a) S-Loop (Morris 1979)

b) Centre-Glazed



a) S-Loop (Morris 1979)



b) Centre-Glazed

It was noted that although flap-valve backdraught dampers are inexpensive and easy to install, they are susceptible to static electricity, ripping etc., and should always be placed where they are clearly visible and accessible.

During wintertime in southern New Mexico it would be reasonable to expect an average heat delivery of 39 kJ to 49 kJ per square metre per day from a vertical panel placed at room level, and that a tilted panel placed below the room would do better by some 10 or 20%. For a 1.82 m long collector with a 101 mm deep flow channel, an air velocity of 0.182 metres per second represents a volumetric flow of 0.0101 m^3 of air per minute per square metre of collector, or about the design flow rate for most active systems according to Morris. It was stated that 0.0101 to 0.0152 m^3 of air per minute per square metre would be typical for a room level collector with good sun.

Morris (1978b) gave performance guidelines for a 1.82 - 2.43 m collector during the noon hours of a sunny day, based on inlet and outlet temperatures.

TEMPERATURE DIFFERENCE BETWEEN OUTLET & INLET OF PASSIVE SOLAR ENERGY AIR-HEATING COLLECTOR	COMMENTS
16.6-22.2°C	very good flow - high efficiency
22.2-27.7°C	good flow - good efficiency
27.7-36.1°C	okay flow - reasonable efficiency
36.1-44.4°C	slow flow - collector is getting too hot
44.4°C or more	some problem with the system.

TABLE 4 Performance Guidelines Presented by Morris (1978a)

It was noted that the most common cause of poor flow rates and high temperature differences was restriction in vent openings, although it was possible to obtain about 90% of the unrestricted heat gain potential with vents restricted to 1/2 the cross-sectional area of the flow channel. Morris recommended that all vent openings and ductwork should have the same area as the flow channels, which in turn should equal at least 1/20 the collector area, when a diagonal mesh absorber is used. Some designs benefit by increases in cross-sectional area up to 1/10 the collector area but it was argued that channels larger than this may actually decrease performance by setting up convection currents within the collector itself, preventing a smooth flow of air through it. A flow channel depth of roughly 1/15 of the collector length was recommended for flat plate absorbers.

An alternative collector geometry described was a centre-glazed U-Tube collector that uses a reverse flow pattern, Fig. (9b). The centre divider was glass and the matrix absorber placed in the back channel. Hot air passed under a masonry floor before entering the room. The masonry floor acts as a thermal store and provides radiant floor heat at night. The performance of this centre-glazed collector was almost identical to that of the U-Tube under most operating conditions but, because only the coolest air is exposed to the exterior glazing, it out-performed the normal geometry under more severe conditions. It was suggested that single-glazing should be sufficient for a thermosyphon collector in a relatively-mild climate, double-glazing being less effective in these circumstances because less sunlight reaches the absorber surface. However, it was argued that in almost any climate, double glazing is cost-effective if the collectors are to be operated at high temperatures, because less heat is lost through the glazing to the atmosphere.

The general figures given for performance were that thermosyphon systems should collect 30% of the available insolation in cold climates and 50% in mild climates.

A review of existing natural convection systems of the type described above fitted to housing in New Mexico and Arizona, USA was undertaken by Morris (1979). It was estimated that collectors installed on the Davis house in Corrales, New Mexico, supplied 70% of the heating load, with sufficient thermal storage to provide heating for two successive non-gain winter days.

A house in south-eastern Arizona also used a diagonal mesh absorber and produced peak efficiencies close to 50%. A problem with this system was that hot-spots developed at the top of the collector between the glazing and the mesh. It was suggested that reducing the collector channel depth from 1/12 collector length to 1/20 collector length would increase the air velocity sufficiently to eliminate these hot-spots and take full advantage of the mesh type absorber. Natural circulation systems in the Jones House in Santa Fe, New Mexico and Masterson Studio, La Cienaga, New Mexico, were also described, and basic guidelines for system design proposed. These included a flow channel depth of 1/15 collector length for flat plate absorbers and a depth of 1/20 if diagonal mesh absorbers are used.

Cook & Morris (1978) undertook a brief investigation and analysis of the performance of a site-built solar convective air collector, rock bed store, and house heating delivery system operating with a heating load of 5,000 heating degree days/year in southern Arizona. One of the conclusions drawn from the analysis of this system was that the dimensions and intensity of hot-spots in the collector should be reduced as they are points of maximum thermal loss. Another was the usefulness of employing smoke to gain a qualitative understanding of the natural convection flow in the collectors. A multitude of secondary convective flows were seen to exist within the collectors, often caused by shading on one side, or the cooling effect of the glazing. This technique enabled air streams of various velocities to be followed and areas of air stagnation to be identified.

The performance of a house in Santa Fe, New Mexico which used a thermosyphon air collector system to heat four vertical wall rock-beds which provided heat to the interior was described by Morris (1982a). The absorber was composed of 3 layers of expanded metal mesh suspended 38 mm above a layer of corrugated steel roofing. Evaluation of the test data indicated that this was not a very efficient absorber and a full diagonal mesh absorber was recommended for future designs.

The inferior performance of this system was apparently due to both the collector configuration and undersized ducts. Heat was detected flowing back into the collector from the storage at night. Although the system employed two site-built motorised thermostatically controlled dampers at the exit from the collector, they apparently failed to prevent reverse circulation. Despite these problems the overall performance of the system was encouraging, a solar savings fraction of 90%+ having been estimated after intensive monitoring.

A method of designing thermosyphon solar energy air-heating collectors and estimating their performance, based on empirical data, was outlined by Morris (1982b). The procedure requires the designer to assume two operating characteristics of the collector in question : the typical peak collection efficiencies and the corresponding peak temperature difference across the collector. These two design parameters, plus weather data, then enable peak air flow rates to be calculated and total available pressure head to be estimated. This information allows the designer to size components such that the sum of the pressure losses in the loop does not exceed the total available pressure calculated previously. It was argued that most analytical techniques for predicting the performance of these systems involve lengthy iterative calculations and that a simplified approach based on an empirical analysis would be of more value to designers.

The design method involves estimating a value for the maximum efficiency that a collector would achieve during the noon hours of a sunny winter's day. This value of efficiency is deduced from graphs which show the variation of efficiency of a well-designed passive air-heating solar energy collector system with the prevailing conditions, defined by the term $(T_i - T_a)/I$. The corresponding peak heat gain from the collector is obtained by multiplying efficiency by the peak insolation. It was recommended that a value for the temperature difference across the collector should be assumed, rather than the air flow rate, as the system efficiency is strongly related to collector temperatures, a high temperature difference indicating somewhat lower efficiencies, and a low temperature difference representing a smaller pressure losses and higher flow-rates.

If the cross-section of the flow channel was assumed to be between 2% and 3% of the net glazing area, the appropriate range suggested was 16.6 to 27.7°C, whilst systems with storage would operate at 22.2 to 33.3°C.

Having assumed the peak temperature difference across the collector and the peak collector output has been estimated, values for the flow rate and available pressure head are derived. The final step is to size all system components; channels, ducts etc., so that the sum of the friction, resistance, and velocity pressure losses at the total estimated air flow rate does not exceed the available pressure head. If the original efficiency estimate was reasonable the actual system performance should, it was argued, be very close to the estimate.

This will also depend on the accuracy with which the sum of all friction losses, resistance losses, and velocity losses are derived. The diagonal mesh absorber was again advocated consisting of 5 or 6 layers of slit and expanded metal lath with 9.5 mm diamond shaped holes, as used for plaster reinforcement. The cross-sectional area of the air channel in a collector using this type of absorber was discussed. The author recommended that the air channel should be 5% of the total collector area.

This was compared with the analytical work of Reno (1981) for flat plate absorbers and of Biehl (1981) which indicated that the cross-sectional area may be reduced to approximately 1% with beneficial results and that within limits, reducing the size of the flow channel next to the absorber increases the air velocity, increasing turbulence and, hence, heat transfer. Morris (1982b), on the other hand, suggested a cross-sectional area of no less than 2% of the total glazing area. It is important to note that peak operating conditions were assumed throughout this procedure. Although conditions vary throughout the day, it was argued that it is advantageous to design for peak periods as this is when the majority of heat is delivered. Data from test units and existing installations suggested that the all-day efficiency of these systems on clear days ranged from 75% to 85% of the peak efficiency. A rough guideline of 75% to 80% in cold weather and 75% to 85% in mild weather was given.

It was argued that a collector should be designed to produce an airflow rate of at least $0.0101m^3$ per minute per square metre of collector during peak periods. Efficiencies improve somewhat with increasing flow rates, but flow rates in excess of $0.0152m^3$ per minute per square metre of collector are not usually worthwhile. Gains in efficiency are small and any ductwork needed to distribute the air will be oversized.

Morris (1983) described the performance of a thermosyphon air collector which was designed to heat the masonry floors of a further residence in New Mexico. The passive solar heating system was monitored in February 1983 to determine the typical winter day performance. The system consisted of a simple site-built thermosyphon collector using a single low iron glass cover and five layers of expanded steel mesh placed at a diagonal to the hot air channel in the same manner as in previous systems. Dampers were located at the entrance to the two hot air ducts to prevent reverse thermosyphoning, and to close off the house loop during summer months. These dampers were motorised with thermostatic controls and their performance was reported to have been less than satisfactory.

Midday collector efficiencies of 45% were achieved which is slightly less than the figure quoted for a well-designed thermosyphon collector but several factors may have contributed to this inferior performance including air leaks and obstructions in the air ducts. Reverse circulation was not referred to, but recommendations for future systems include placing the collector glazing entirely below any floor storage mass and ductwork so that the collector will act as a thermal trap for cold air at night. This would obviate the need for a diurnal damping system and would increase the available pressure head and air flow rates.

The design and construction of a further residence in Santa Fe, New Mexico, U.S.A. was reported by Morris (1981). This featured a $37.16m^2$ thermosyphoning air heater and a shallow rock bed under the entire floor of the $171.88m^2$ house. After monitoring of the building had been carried out during January and February 1981, it was concluded that the system had an exceptionally good thermal performance. During a day in which the ambient temperature averaged 1.6^oC and the house averaged 17.2^oC a solar fraction of 97% was calculated based on a design temperature of 18.3^oC.

1.9 FURTHER RESEARCH

A need for retrofit techniques which could drastically reduce space heating fuel consumption in the older existing housing stock of the New England region of the USA was reported by Nisson & Williams

(1980). The authors presented the results of a project which involved the monitoring of four types of site-built solar retrofits, including four thermosyphoning air panels. The TAPs were of wood frame construction and were each $2.1m^2$ in area. They featured absorber plates of blackened metal roofing, and double-glazing with an air-space of 1cm betrween the two layers of glass. The collectors were of the Back-pass configurtion with a 4cm air channel between the absorbert plate and the existing wal. Inlet and outlet vents measured 20cm x 41cm, with a backdraught damper made of "Frisket Paper" mounted on the lower vent. No mention of the level of insulation in these units was made, nor the effectiveness of the backdraught damper.

Three features of the performance of these units were monitored: (1) heating fuel consumption, (2) temperature regimes within the dwellings and the collectors and (3) life-style and thermal comfort of the occupants.

The TAPs produced the least saving of fuel of the different solar retrofit units tested, but their poor performance was attributed to a significant change in the behaviour of the occupants of the dwelling they were heating. On sunny days, the collectors would heat the house to 27°C. The occupants enjoyed these high temperatures and kept their thermostat set at 27°C throughout the winter, negating the contribution made by the TAPs.

This illustrates the difficulty involved in anticipating the reaction of occupants to a solar heating system, and in predicting the performance of such units when installed in a building.

It was acknowledged in this study that the results obtained were not sufficient to identify the proportion of energy savings which were attributable to actual displacement of conventional heating and those which were due to other factors. It was noted that one consequence of the introduction of the solar retrofits was increased energy awareness and energy thrift in the home owners, apart from the above example. Whether the installation of solar heating systems exert the same influence on the occupants of non-domestic buildings remains to be seen.

The thermal performance of a themosyphoning air heating system was evaluated by Marshall et al (1981). This consisted of a double glazed collector, ducting, backdraught dampers to prevent reverse circulation, and thermal storage in the north wall of the test cell. The thermal performance of the system was evaluated by using measured data as input for a simulation model. The simulation made use of an improved Euler technique to solve the transient energy and momentum equations. Design parameters of the sytem were varied and the corresponding effect on the solar fraction was observed using the simulation model. It was concluded that thermosyphon systems exhibit a lower thermal performance than other passive designs such as a Trombe wall or direct gain systems. This was attributed to low flow velocities due to flow resistance in the thermal storage. It was noted that the absorbed solar energy was not effectively transferred to the thermal mass in the north wall.

Backdraught dampers installed in the test cell were deemed to be effective in the prevention of reverse circulation flow. It was also noted that sixty percent of the energy collected by the absorber was lost through the glazing.

The design, retrofit construction, and thermal performance of a TAP located on a house in Iowa. USA. was described by Boast (1984). The TAP was fitted with automatic well-insulated dampers to eliminate reverse-thermocirculation and to reduce infiltrative and conductive heat losses. The dampers were automatically opened by an adjustable master thermal switch. Data was presented for a typical day in December but no attempt was made to extrapolate this data to a monthly or yearly basis.

Wilson & Stickney (1980) compared the performance of two retrofit vertical passive solar air-heaters which were fitted to uninsulated walls with limited storage capacity. One collector was of a Back-Pass configuration with an absorber plate of sheet metal and one layer of glazing. The other was a double-glazed Front-Pass collector in which the wall surface was used as the absorber, in the same manner as a Trombe Wall. The collectors measured 2.39 m by 2.44 m and vented into nearly identical rooms separated by interior walls. The conclusion drawn from a comparison of the performance of the two collectors was that for poorly insulated walls a Front-Pass collector was more effective for heating than a Back-Pass collector. It was suggested that heat-loss through the glazing of the Back-Pass collector was greater than glazing heat loss from the Front-Pass collector because of the higher absorber plate surface temperature of the Back-Pass collector. The influence of the single and double glazing on the performance of the two collectors was not examined. The researchers were not able to determine whether or not a Front-Pass collector would be more effective on insulated walls. They were also unable to make a general recommendation of one system over the other.

In a similar work Hagan et al (1980) published the results of tests carried out on Back-Pass convective air heaters which were fitted to suitably dimensioned test cells. The Back-Pass collector had a black-painted, corrugated aluminium absorber plate and Kalwall plastic double glazing. The Front-Pass collector featured an absorber of blackened aluminium foil stapled to the test cell wall and double acrylic-polyester sandwich "Flexigard" glazing. The collectors also differed in that the vents of the first collector measured 20.3 by 35.6 cm, whilst the Front-Pass collector had continuous width vents 10 cm high.

Plastic film backdraught dampers were fitted to the bottom vents of both collectors. The results showed that in both cases the conductive portion of collector heat input was greater than the convective portion. The Front-Pass collector delivered 2.15 times more convective energy than the Back-Pass collector. The disparity in the delivery of convective heat was attributed to the larger, better positioned vents used on the Front-Pass collector.

In a study which outlined the development of site-built, low-cost, air heating systems for vertical walls, Temple & Kohler (1980) compared the efficiencies of natural circulation and fan-powered collectors and discussed several issues involved in their design. It was stated that the average temperature in an air-cooled solar collector depends on the air flow rate, and that a higher flow rate implies a lower average temperature and greater efficiency. The friction which limits flow through a thermosyphoning collector could be reduced by increasing the depth of the air channel, but it was argued that such an increase in channel depth would reduce velocity. A channel depth of 0.076 m to 0.1 m was recommended for maximum efficiency, however, it is not clear whether this was a general recommendation or applied solely to the collectors tested.

The efficiency of thermosyphoning air panels were found to be considerably lower than that of comparably constructed active collectors. The relative performance of the two designs, both with single and double glazing were compared. The collectors tested were of similar design with the same glazing, insulation, and loss coefficients. The thermosyphoning collector was 2.43m high with a channel depth of 0.1 m (1/24 of the height). Under what were described as typical operating conditions with values for temperature difference between collector inlet and ambient divided by insolation, $(T_i - T_a)/I$, of 0.03 to 0.04m^2 °C/W, the single and double-glazed active collectors were 21% to 35% more efficient than the thermosyphoning collectors. It was noted that, whilst it would be more cost effective to build active collectors rather than thermosyphoning air panels of the same heat output, the active systems would require maintenance and would be more prone to failure. It was stated that double-glazing increases the efficiency of a typical site built collector by approximately 20% in most areas of the U.S.A. and increases cost by a similar amount. A single-glazed collector 20% larger than a double-glazed collector would provide the same amount of energy at the same cost on this basis. Double-glazing was recommended for this application as wall space is often limited. A Back-Pass configuration was preferred to Front-Pass for these collectors because of difficulties experienced in maintaining an air-tight seal around the glazing assemblies.

The performance of a 100% solar-heated house in Arizona which featured natural-circulation solar-energy air-heaters coupled to a "rock-bin" heat store was reported by Cook and Morris (1981). The installation of a backdraught damper in this system reduced night-time losses by an estimated 40%. Manual dampers were also used in the system but leaked considerably. Difficulty was experienced in measuring the manitude of this leakage because of the erratic air-flow patterns generated around these dampers. It was concluded that the quality of dampers on commercially-available registers and grilles was so poor as to challenge their use in passive systems, whilst plastic backdraught dampers appeared to be very effective. A further conclusion was that "radiant comfort" delivered by heat flow through the floor may be more desirable than the comfort achieved by convective air in a passive rock-bed heating system.

The construction and performance of Radiant Panels retrofitted to a large municipal car maintenance facility in Philadelphia U.S.A. was reported by Burnette & Lent (1984). The requirement was for a retrofit that would convert solar gain to radiant heat whilst providing as much glare-free illumination as possible. As 70% of the walls of this building were glazed a high level of window insulation was required. Translucent fibreglass panels backed by an air space and a light-diffusing perforated metal absorber/radiator panel were installed as retrofit windows on the south and southeast facades of the building. The Radiant Panel incorporated a "light shelf" to reflect daylight onto the ceiling of the facility, and a damper for ventilation during the summer.

The components of the retrofit panel are shown in Fig. (10). A standard 69.8mm thick transparent 'Kalwall' panel, U value 0.41 W/m^2K, formed the basis of the unit. The 1.27mm thick aluminium plate was painted black on the absorber side and white (for unstated reasons) on the side radiating into the building. It was perforated with 1.77 mm round holes, 14.2mm on centre (open area = 1.45%) to allow diffused illumination of the building. This plate was mounted and sealed 50mm behind the Kalwall panel, creating a virtually stagnant air space between the two components. The damper at the top of this space was installed to allow venting of excess heat and induce summer ventilation. Above the collector plate and below a panel of insulating glass, a polished aluminium shelf 0.45, deep was set at an angle of 9° below the horizontal to reflect daylight into the interior.

The panels covered almost 80% of the 390.2 m^2 of south/south easterly window area, The largest windows were 4.54 m high and 2.64 m wide and incorporated three of the units described above.

The benefits of the addition of an absorber plate to a window panel in this manner were outlined as follows: (1) the window is converted from

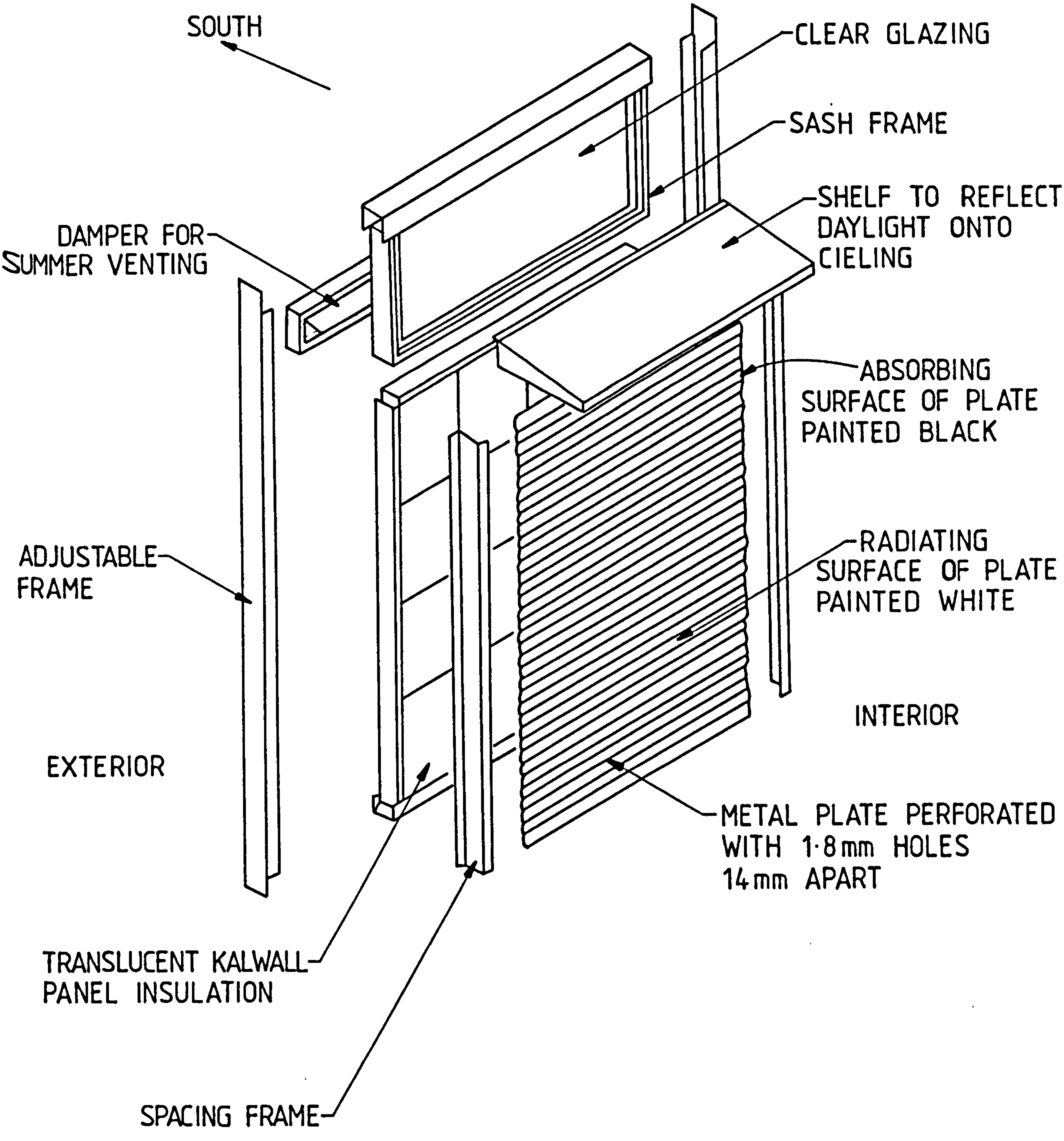


Fig.10 "Radiant Panel" collector for retrofit applications

(Burnette and Lent 1984)

a source of radiant heat loss at night and during cold weather during the day, into a heat source, (2) an efficient and direct thermal distribution device is created, energy is no longer absorbed by the thermal mass of the building by direct gain and released at night when the building is unoccupied, and (3) the radiant heat generated by such a device can provide satisfactory comfort conditions at low air temperatures.

It was estimated that of 1107 kJ absorbed by the panel between 9 a.m. and 2 p.m. on December 26th, 1983, 981.1 kJ were reradiated into the work space as thermal energy.

Ventilation air flow rates were not assessed, but measurement of the air temperatures in the space between the absorber and the glazing during the summer indicated that the unit was capable of inducing such air flow. The pin hole perforations in the absorber plate of the unit reduced the contrast between the light coming through the clear glazing areas and the surrounding wall by producing a diffused "glow" over the panel. Despite being extremely effective on bright days, the permanence of the panel worked against greater illumination on overcast days. It was suggested that a heat transfer surface in the form of an adjustable blind would be more desirable for daylight control and, if properly managed, would probably provide comparable thermal efficiency. The venting damper was considered to be of marginal utility. The economic merits of this retrofit unit, which could be considered to be a non-thermo syphoning air panel, have yet to be determined.

The adverse effects of reverse-thermocirculation were reported by Perry (1980) who monitored the performance of a house heating system which employed natural-convection solar-energy heaters coupled to a rock-bed storage. The "Jones" house in Santa Fe, New Mexico U.S.A. featured a $49.4m^2$ collector which had manual dampers at both the inlet and the exit. The rock bed was fully charged during January 13th, 1980, the blower that circulated air through the house ran briefly between 8.00 and 9.00 a.m. and then not for 24 hours. The manual dampers shut shortly after 4.00 p.m. when the sun was no longer heating the collector. From 5.00 p.m. to 10.00 a.m. the rock bed lost approximately 40% of its thermal charge. This loss was attributed to reverse-thermocirculation, due to incomplete closing of the manual dampers. During the evening the thermal storage lost heat at a rate of 9,706 kJ/hr of which only 1793 kJ/hr, or 18% could be accounted for by conduction losses from the rock bed. 1055 kJ/hr was lost from the top plenum of the collector, through the dampers and wall by conduction. The remaining loss of 7912.5 kJ/hr was attributed to reverse-thermocirculation. It was concluded that manual dampers should be closed tightly and that insulation of the upper dampres and surrounding wall was desirable. Reverse-circulation occurred in the Balcomb House but this was stopped by the installation of a plastic-film backdraught damper on one of the rock-bed stoves.

The development of self-actuating backdraught dampers which featured bi-metallic coils was reported by Lau et al (1985). These were produced as an alternative to the flap-valve backdraught damper in an attempt to overcome the potential problems of low thermal resistance, restriction of air flow, leakage etc., associated with this device. The prototype design was similar in principle to that of valves fitted to automotive carburettors: rotation of the damper blade was about its horizontal axis, thus the only significant resistance that the bi-metallic coils actuating the device had to overcome was friction in the shaft bearings. Two prototype dampers were tested in the upper and lower vents of a "standard" thermosyphoning air panel. The upper damper opened fully on sunny days, but partial opening of the lower damper reduced collector efficiency by 19%. It was noted that the lower damper only opened as much as it did because its bi-metallic coils were directly exposed to sunlight and the warmer collector air. It was argued that a damper in this position would not move significantly, if the coils were located in the inlet air stream and were isolated from solar radiation, as room temperature air would predominate. A damper at the outlet would, on the other hand, function adequately under these circumstances because of the flow of warm air over the damper and coils during collection periods. It was concluded that a single damper at the collector outlet should minimise reverse-thermosyphoning flow, a lower backdraught damper of this type should not be necessary.

Tests were carried out by Lyall (1984) to determine the performance of a device designed for the regulation of air flow through natural circulation air-heating collectors. A prototype which consisted of a temperature sensor/actuator, a control cable and an air damper was evaluated within a simulated solar wall system.

An advantage of having the actuator linked to the damper by a low friction drive cable was that the actuator could be placed in the outlet duct, or at the top of the collector air channel, and the damper installed at the inlet vent. This overcomes some of the problems encountered by Lau et al (1985) during their research on these devices.
Three commercially-available sensor/actuators were tested and rated during the course of this work, so as to determine their thermal response time-constants and force development characteristics. They were i) a wax filled piston activator, ii) a shape memory effect brass alloy spring, iii) a double helical bi-metal coil. The bi-metal coil produced an inadequate extension rate and negligible force development and was rejected. The wax filled activator produced a high degree of hysteresis and potential control problems. The memory-alloy spring produced the best characteristics and was utilised in all subsequent tests.

It was concluded from this work that all of the sensor - actuators tested were subject to the following inherent faults i) hysteresis losses, ii) thermal lag, and iii) short stroke: usable activator stroke (up to 10mm) was too limiting to ensure ease of temperature calibration and operation.

A more responsive sensor/actuator was required to operate at the relatively low ambient temperatures encountered. It was demonstrated that the low level damper operated well and produced a satisfactory air flow distribution over the collector plate, and that such a relatively simple and inexpensive air control device could be set to work automatically within a typical solar wall installation.

Chapter 2

HEAT TRANSFER

The four basic equations which govern the flow of air through a natural convection air heating solar energy collector have been identified by Kohler (1981) and Burek (1984). These are i) the pressure integral equation which enables the forces which move air around a system to be determined, ii) the dynamic friction loss equation: for a closed loop system under constant pressure, these friction losses are equal to the driving forces, and iii) that which expresses the change in heat content of the air passing through a collector, and iv) an energy balance on the collector: heat gained is equal to the heat absorbed minus heat losses.

2.1 THE PRESSURE INTEGRAL EQUATION

The thermosyphon pressure head is related to the driving temperatures and the height of a collector by the equation

$$\Delta P = g\int \rho\ (h).\ dh. \qquad (1)$$

The TAP system under consideration can be divided into front and back sections representing the collector and room respectively Fig. (11) and equation (1) can be rewritten.

$$\Delta P = \left[\left(\int\rho\ (h).dh \right)_b - \left(\int\rho(h).dh \right)_f \right] = P_b - P_f \qquad (2)$$

Now $P_b = g.\bar{\rho}_b.\ h$

$P_f = g.\bar{\rho}_f.\ h$

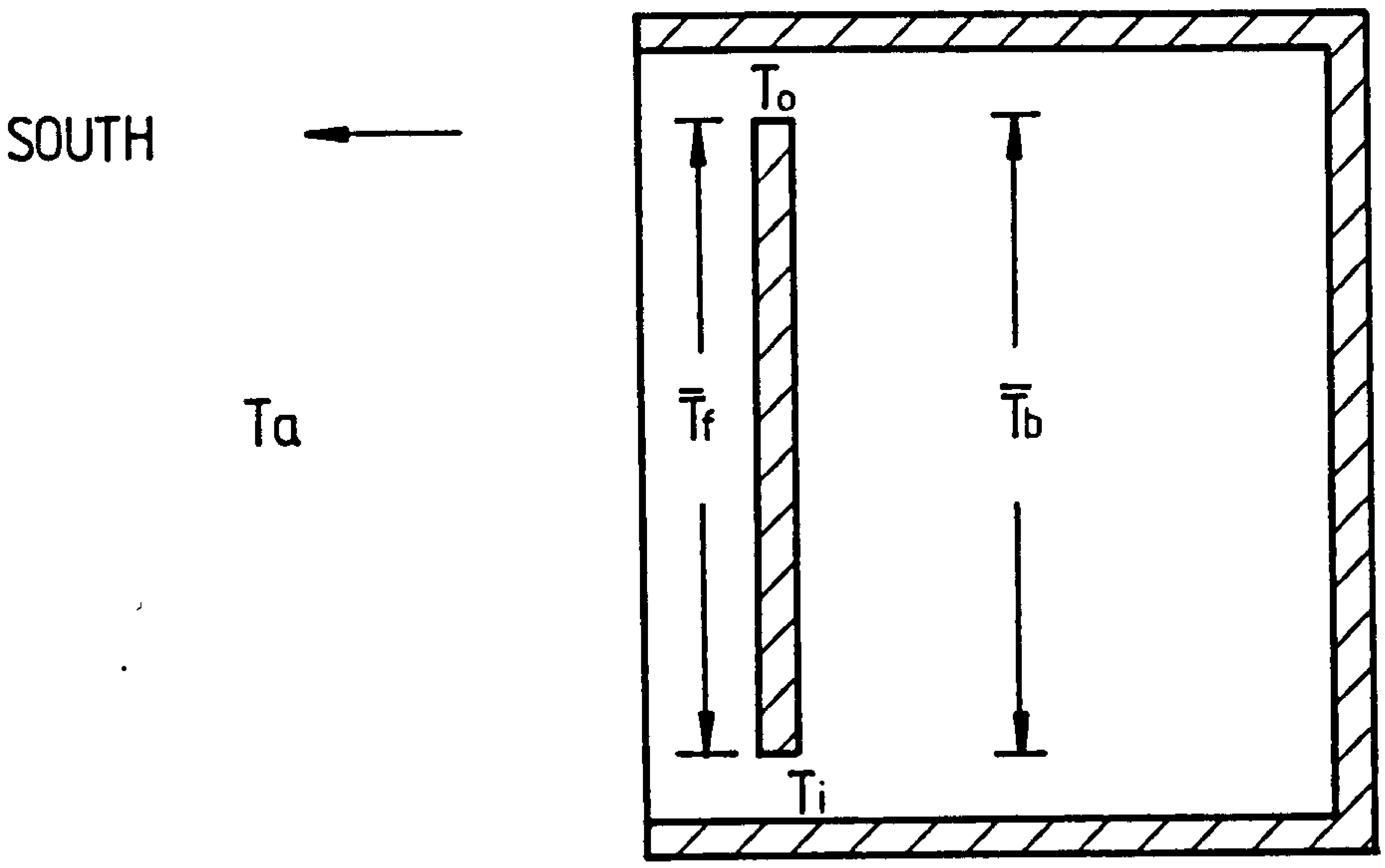
So $\Delta P = g.h.\ \bar{\rho}_b\ (1 - \bar{\rho}_f/\bar{\rho}_b)$ (3)

or $\Delta P = g.h.\ \bar{\rho}_f\ (1 - \bar{\rho}_b/\bar{\rho}_f)$ (4)

If $\bar{T}_f$ and $\bar{T}_b$ are defined as follows:

$\bar{T}_f$ = temperature corresponding to the mean density in the front section (K)

$\bar{T}_b$ = temperature corresponding to the mean density in the back section (K)



$\overline{T}_f$ = TEMPERATURE CORRESPONDING TO THE MEAN DENSITY IN THE FRONT SECTION

$\overline{T}_b$ = TEMPERATURE CORRESPONDING TO THE MEAN DENSITY IN THE BACK SECTION

T_i = TEMPERATURE AT THE COLLECTOR INLET

T_o = TEMPERATURE AT THE COLLECTOR OUTLET

T_A = AMBIENT TEMPERATURE

Fig.11 Temperatures in a TAP and adjacent room

Then, since density is inversely proportional to temperature for a perfect gas:

$$\Delta p = g.h.\ \rho_f\ (1 - \bar{T}_b/\bar{T}_f) \tag{5}$$

A relationship has thus been established between driving pressure ΔP, a collector parameter h, an operating parameter ρ_f and representative fluid temperatures $\bar{T}_f$ and $\bar{T}_b$.

As Burek (1984) stressed, $\bar{T}_f$ and $\bar{T}_b$ are <u>not</u> the mean temperatures in the collector and working sections but are temperatures corresponding to the mean densities, which are not necessarily the same.

2.2 THE FRICTION LOSS EQUATION

This equation describes the dynamic pressure losses in the system due to friction: flow through the collector will increase until the pressure drop due to friction is equal to the pressure head.

$$\Delta P = f.\ (L/D)\ .\ (\rho\ v^2) \tag{6}$$

This is the Darcy-Weisbach equation for friction losses in pipes. The friction factor can be determined from empirical correlations for laminar and turbulent flow as presented in ASHRAE Fundamentals (1977) and will depend on the entrance and exit geometries.

For the case of buoyancy-driven convection being considered it could be argued that the friction factor f should be a function of Grashof number, rather than Reynolds number, the latter is used for forced convection systems. However, as Burek (1984) noted, reports on Grashof number/friction factor relationships are scarce and several workers, [Biehl (1981), Azevov & Babakulov (1982) and Varcolik & Versteegen (1980)] have used Reynolds number correlations for friction coefficients in thermosyphonic heating systems; buoyancy driven convection in a duct can be considered to resemble forced convection flow as axial velocity is zero only at the walls.

In a system comprising N sections, the Darcy-Weisbach equation can be written:

$$\Delta P = \rho V^2 \sum_{i=1}^{N} f_i \tag{7}$$

where V is the characteristic velocity of the system and f_i the dynamic friction loss coefficient, a function of Reynolds number and geometry for each of the N sections.

For the case of a thermosyphoning air heater and room this can be written

$$\Delta P = \rho_0 V_0^{\,2} \left(f_0 + f_1 + f_2\right) \tag{8}$$

where V_0 is the velocity in the back section

ρ_0 is the mean density in the back section

f_0 and f_1, are friction loss coefficients in the back and front sections which are reduced to the reference velocity V_0

f_2 is a friction factor for the inlet and outlet of the collector.

Barra & Carratelli (1980) and Azevov & Babakulov (1982) used the following relationship for laminar flow in ducts:

$$f = 48 / Re \tag{9}$$

which is the friction coefficient for fully-developed laminar flow in an infinitely wide duct for which the hydraulic diameter is twice the duct depth. Hollands & Shewen (1981) and Varkolik & Versteegen (1980) have suggested functions of the form:

$$f = A / Re + B \tag{10}$$

where A and B are constants which depend on the duct dimensions allowing for entrance effects before the flow profile fully develops. Values for these constants may be determined from the work of Olson

(1980). If entrance region effects and edge effects are not significant equation (9) can be reduced to the form of equation (8). Burek (1984) suggested that the validity of the functions chosen for f_0 and f_1 can be tested by the degree of self-consistency of the values for f_2 which can be evaluated by re-arranging equation (7)

$$f_2 = (\Delta P / \rho . V_0) - (f_0 + f_1) \quad (11)$$

It was suggested in the ASHRAE Handbook of Fundamentals (1977) that f_2 is a constant for a given geometry of ductwork, but it may also be considered to be a function of Reynolds number:

$$f_2 = C . Re^r \quad (12)$$

where exponent r will be zero for f_2 independent of flowrate.

As f_0, f_1 and f_2 can be expressed in terms of Reynolds number and collector dimensions it is possible to express ΔP in (7) in terms of velocity, air properties and collector dimensions.

2.3 THE HEAT GAIN EQUATION

This third basic equation is an energy balance which relates the output of the collector to the flow rate. The mass flow rate $\dot{m}$ can be determined from the velocity and temperature measurements. The temperature rise ΔT can be determined from collector inlet and outlet air temperatures, whilst the specific heat capacity C_p is barely temperature sensitive:

$$q = \dot{m} . C_p \ (\Delta T) \quad (13)$$

2.4 THE COMBINED ENERGY BALANCE EQUATION

The linear energy balance on a collector is

$$q_u = F_R \left[I . \tau\alpha - U_L \ (T_i - T_a) \right] \quad (14)$$

where

q_u = useful heat gain (Wm^{-2})
I = global solar radiation (Wm^{-2})
τ = transmittance of cover glazing to incident solar radiation
α = absorbtance of collector plate to incident radiation
T_i = working fluid at collector inlet (K)
T_a = ambient air temperature (K)
U_L = overall loss coefficient ($Wm^{-2}K$)
F_R = heat removal factor

$$F_R \quad \frac{\dot{m}\,C_p}{UA}\left(1 - \exp - \frac{UAF'}{\dot{m}\,C_p}\right) \qquad (15)$$

where F' = collector efficiency factor

This equation was developed from the results of work carried out in the 1950s and 1960s on forced-flow flat-plate collectors. A derivation is presented by Duffie & Beckmann (1980).

The efficiency factor, F_R, and the overall loss coefficient U_L are functions of heat transfer coefficients between the working fluid, absorber plate, the glazing and ambient. They are assumed to be constant for a given collector and operating parameters.

In practice the loss coefficient will not be constant but will be influenced by such phenomena as wind effects. The equation neglects radiative heat transfer between collector components in the range of operating conditions normally encountered, these radiative effects will be small relative to the other heat transfer processes taking place within a collector. However, when high temperature differences exist between the working fluid and ambient such radiative heat transfer must be considered. Fluid flow-rates in thermosyphoning flow systems cannot be considered to be constant, unlike forced-flow systems, which precludes the use of the above equation in their evaluation. A model was developed by Burek (1984) which extends this equation to allow for variation in flow rate.

The Pressure Integral Equation may be combined with the Friction Loss Equation in the following manner to obtain an equation for collector efficiency.

$$\Delta P = g.h.\ \bar{\rho}_f(1 - \bar{\rho}_b/\bar{\rho}_f) = \rho_o V_o^2\ (f_o + f_1 + f_2)$$

There is no velocity or friction loss in the back section in our case so f_1 is the friction loss coefficient in the front section reduced to the reference velocity V_1 and f_2 in the friction factor for the top and bottom corner sections also reduced to the reference velocity V_1.

Therefore we can write:

$$\Delta P = g.h.\ \bar{\rho}_f(1 - \bar{T}_b/\bar{T}_f) = \rho_1 V_1^2\ (f_1 + f_2) \tag{16}$$

Now $\dot{m} = \rho_1\ v_1\ h$

Assuming $\rho_1 = \bar{\rho}_f$

Then $$V_1^2 = \frac{\dot{m}^2}{\rho_1^2\ h^2} \tag{17}$$

Therefore

$$\Delta P = g.h.p_1\ (1 - \bar{T}_b/\bar{T}_f) = \frac{\dot{m}^2}{\rho_1^2 h^2}(f_1 + f_2) \tag{18}$$

$$\therefore \qquad \dot{m} = \sqrt{\frac{g.h^3 \rho_1^2\ (1 - \bar{T}_b/\bar{T}_f)}{(f_1 + f_2)}} \tag{19}$$

Now $\eta = \dfrac{\dot{m}\ C_p\ \Delta T}{IA}$ where $\Delta T = (T_o - T_i)$

$$\therefore\ \eta = \frac{\sqrt{\dfrac{g.h^3\ \rho_1^2\ (1 - \bar{T}_b/\bar{T}_f)}{(f_1 + f_2)}}\ C_p\ (T_o - T_i)}{IA} \tag{20}$$

$$\therefore\ \eta = \frac{h^2\ \rho_1\ C_p\ (T_o - T_i)}{IA} \sqrt{\frac{g\ (1 - T_b/T_f)}{(f_1 + f_2)}} \tag{21}$$

This is an expression for efficiency which does not include a term for mass flow rate.

2.5 BERNOUILLI'S EQUATION

As Duffie & Beckman (1980) noted, flow by natural convection in the air channel of a solar wall is difficult to estimate. Assuming that the density and temperature of the air in the heated air-flow channel varies linearly with height, the following solution of Bernouilli's equation was proposed:

$$\bar{v} = \sqrt{\left[\frac{2gh}{C_1\ A_2 + C_2}\right] \cdot \frac{T_m - T_r}{T_m}} \tag{22}$$

The term $[C_1\ A_2 + C_2]$ represents the combined "pressure drop" of the air channel and the entrance and exit ducts where:

C_1 & C_2 = dimensionless empirical constants
h = height of the solar wall
T_m = mean air temperature in the channel
T_r = room air temperature
A_2 = dimensionless characteristic areas of the system

A method for determining the flow rate of air through the channel of thermal storage walls based on Bernouilli's equation for non-ideal flow was developed by Varcolik & Versteegen (1980). The flow between sections 1 and 2 in fig. (12a) may be expressed as follows:-

$$g\rho_1 z_1 + \rho_1 V_1^{\ 2}/2 = g\rho_2 z_2 + \rho_2 V_1^{\ 2}/2 + \Delta pi \tag{23}$$

This equation was applied to the solar-energy air-heating wall system for which the nomenclature is shown in fig. (12b). Bernouilli's equation was applied to four sections of the flow loop; from 1 to 2, 2

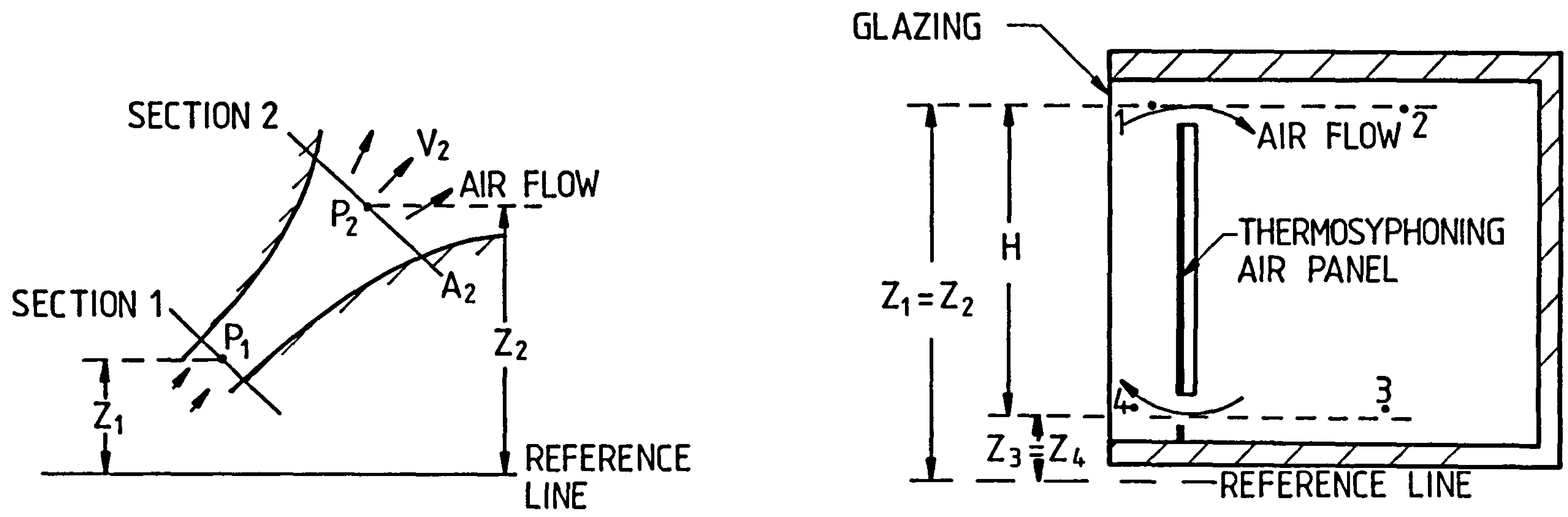


Fig.12 Application of Bernouilli's equation to a TAP and adjacent room

to 3, 3 to 4, and 4 to 1 respectively. For conditions of equilibrium the sum of the pressure changes around the system is zero, therefore :

$$(p_1 - p_2) + (p_2 - p_3) + (p_3 - p_4) + (p_4 - p_1) = 0 \qquad (24)$$

Bernouilli's equation between points 1 and 2 was given as follows:

$$g p_1 Z_1 + p_1 + -p_1 V_1^{\,2}/2 = g\rho_2 Z_2 + p_2 + \rho_2 V_2^{\,2}/2 + K_{uv}\rho_2 V^2_{1-2}/2 \qquad (25)$$

where pressure losses due to irreversibilities have been expressed in terms of a loss coefficient, K_{uv}, and the fluid density and velocity at the smallest flow area. Varcolik & Versteegen (1980) determined the loss coefficients experimentally.

Similar equations may be derived for the other points around the flow loop and these equations were substituted into equation (24). The flow rate and temperatures may then be derived on the basis of the following assumptions:

1. the density ρ_{1-2} is the same as ρ_1
2. the air velocity in the living space is negligible compared to other air velocities, i.e. $V_2 = V_3 = 0$
3. the air in the living space is well mixed and thus at the same temperature throughout so that $\rho_3 = \rho_4 = \rho_{3-4} = \rho_R$ except at ceiling level where $\rho_2 = \rho_1$.

Noting that $Z_3 = Z_4 = Z_{LV}$, $Z_1 = Z_2 = Z_{UV}$ and $Z_2 - Z_3 = H$, the equations were then combined to give:

$$-\rho_1 V_1^{\,2}/2 + K_{UV}\rho_1 V_{1-2}^2/2 - \rho_R\, gH + \rho_R V_4^{\,2}/2 + K_{LV}\rho_R V_{3-4}^2/2 +$$

$$\int_{Z_{LV}}^{Z_{UV}} \rho dz - \rho_R V_4^{\,2}/2 + \int_{Z_{LV}}^{Z_{UV}} f\, D_e^{-1}\, \rho V^2/2 = 0 \qquad (26)$$

It was argued that since the mass flow rate, $\dot{m}$, is a quantity that remains constant through the system, it should be the variable for which a solution should be obtained. Substituting into equation (25) and also substituting the expression for the friction factor

$$f = a\ Re^{-1} + b$$

the following equation was obtained

$$\dot{m}^2/2\ [K_{UV}/\rho_1\ A^2{}_{UV} + K_{LV} + b/2D_e\ A_s{}^2 \int_{Z_{LV}}^{Z_{UV}} 1/\rho dz\ ]$$

$$+ \dot{m}/2\ [a/2\ De^2\ 1/A_s \int_{Z_{LV}}^{Z_{UV}} \gamma\ dz] - \rho_R\ gH + g \int_{Z_{LV}}^{Z_{UV}} \rho\ dz = 0 \quad (27)$$

It was additionally assumed by Varcolik & Versteggen (1980) that

4. air behaves as an ideal gas

5. the integral $\int_{Z_{LV}}^{Z_{UV}} 1/\rho\ dz = 1/2H(1/\rho_2 + 1/\rho_R)$

6. the integral $\int_{Z_{LV}}^{Z_{UV}} \gamma\ dz = H\gamma$

   where γ is the kinematic viscosity evaluated at $1/2\ (T_R + T_1)$

7. the integral $\int_{Z_{LV}}^{Z_{UV}} \rho\ dz = 1/2H(\rho_1 + \rho_R)$

These assumptions were introduced into equation (27) to give the following equation:

$$\dot{m}\,[K_{UV}/A^2UV\,T_d + K_{LV}/A^2LV\,T_R + gH/4D_eA_s^2\,(T_1+T_R)] + \tag{28}$$

$$\dot{m}\,(a\,\gamma\,Hc/2D_e^2A) + gHe^2\,(T_R - T_1/T_1\,T_R) = 0$$

This quadratic equation was reduced by Norton (1986) to the form

$$\dot{m} = \frac{-\,a\gamma Hc/2D_e^2A \pm \sqrt{S}}{2[K_{UV}/A^2UV\,T_d + K_{KV}/A^2LV\,T_R + gH/4D_eA_s^2(T_1 + T_R)]} \tag{29}$$

where S =

$$(a\bar{\gamma}Hc/2D_e^2A)^2 + 4[K_{UV}/A^2UVT_d + K_{LV}/A^2LVT_R + gH/4D_eA_S^2(T_1+T_R)]gH_e^2(T_1-T_R)/T_1T_R]$$

2.6 EFFICIENCY BASED ON GLAZING TEMPERATURE

It was suggested by Kornher & Zaugg (1984) that the efficiency of thermosyphon solar-energy air-heating collectors could be assessed from measurement of the collector glazing temperature and the temperature differential between air entering and leaving the collector, at noon on a sunny day. However, an analysis of this test method was not presented.

An expression describing the instantaneous performance of a collector based on the measurement of glazing temperature, inlet and outlet temperature, ambient temperature and insolation level is presented herein. This expression does not include or require a value for the mass flow-rate of air through the collector and obviates the need for (i) air velocity measurement and (ii) derivation of values for dynamic friction factors.

Four major assumptions are made in the following analysis.

1. The heat storage capacity of the collector is negligible.
2. Collector back and edge losses are negligible.
3. A value of unity for the fin efficiency F′ is assumed.
4. The heat-loss coefficient from the glazing to ambient air remains constant at a particular calculated value.

Furthermore, it must be emphasised that the glazing temperature T_g must be measured and not estimated.

The heat balance of the collector is,

$$\dot{m}\, C_p\, (T_o - T_i) = IA\tau\alpha - UA(T_g - T_a) \tag{30}$$

Thus

$$\dot{m} = \frac{IA\tau\alpha - UA(T_g - T_a)}{C_p\, (T_o - T_i)} \tag{31}$$

Now

$$\eta = F_R\, \tau\alpha - F_R\, U \left(\frac{T_i - T_a}{I}\right) \tag{32}$$

where

$$F_R = \frac{\dot{m}\, C_p}{UA} \left[1 - \exp\left(\frac{UAF'}{\dot{m}\, C_p}\right)\right]$$

Assuming $U = U_L$, then

$$\eta = \frac{\dot{m}\, C_p}{UA} \left[1 - \exp\left(-\frac{UAF'}{\dot{m}\, C_p}\right)\right] \tau\, \alpha$$

$$- \frac{\dot{m}\, C_p}{UA} \left[1 - \exp\left(-\frac{UAF'}{\dot{m}\, C_p}\right)\right] . U \left(\frac{T_i - T_a}{I}\right) \tag{33}$$

But

$$\dot{m} = \frac{IA\, \tau\, \alpha - UA(T_g - T_a)}{C_p \left(T_o - T_i\right)}$$

Substituting

$$\eta = \left(\frac{I\tau\alpha/U - \left(T_g - T_a\right)}{\left(T_o - T_i\right)}\right)\tau\alpha\left[1 - \exp\left(-\frac{F'\left(T_o - T_i\right)}{I\tau\alpha/U - \left(T_g - T_a\right)}\right)\right]$$

$$-U\left(\frac{I\tau\alpha/U - \left(T_g - T_a\right)}{\left(T_o - T_i\right)}\right)\left[1 - \exp\left(-\frac{F'\left(T_o - T_i\right)}{I\tau\alpha/U - \left(T_g - T_a\right)}\right)\right]\left(\frac{T_i - T_a}{I}\right) \quad (34)$$

$$\text{Let } G = \frac{I\tau\alpha/U - \left(T_g - T_a\right)}{\left(T_o - T_i\right)}\left[1 - \exp\left(-\frac{F'\left(T_o - T_i\right)}{I\tau\alpha/U - \left(T_g - T_a\right)}\right)\right]$$

Then

$$\eta = \tau\alpha G - UG\left(\frac{T_i - T_a}{I}\right) \quad (35)$$

or

$$\eta = G\left[\tau\alpha - U\left(\frac{T_i - T_a}{I}\right)\right] \quad (36)$$

2.7 THE SOLAR LOAD RATIO METHOD

The method for predicting the performance of solar heated buildings was developed at the Los Alamos National Laboratory by Balcomb et al (1980). It involves the use of a coefficient, the Solar Load Ratio (SLR), which relates the monthly net solar energy absorbed by a building to the net monthly building heat load.

The terms used in the Solar Load Ratio Method are defined below after Harris et al (1985).

1) Building Load Coefficient (BLC)
This is the building heat loss in W-hr/°C-day caused by heat transmission through the building fabric and air infiltration.

2) Net Load Coefficient (NLC)
This is a term which has replaced BLC to emphasise that the load is a net load, excluding gains and losses through the solar elements of a building. These elements are assumed to be replaced by a "thermally neutral" wall. The units are as above.

3) Auxiliary Heat (AH)
This is the heat supplied to a building by conventional space-heating equipment as a supplement to the solar heat received by the building.

4) Net Reference Thermal Load (NRTL)
This is the degree-day heating load of the non-solar elements of a building assuming constant indoor temperature, and is defined by the equation

$$NRTL = NLC \times DD \tag{37}$$

Alternatively the NRTL can be expressed as the amount of conventional heat energy required for a comparable non-solar building for a stipulated time period such as a month. The NRTL is based on a desired indoor temperature, assuming that the solar collection area is replaced with an adiabatic component, and neglecting any internally generated heat.

5) Degree days (DD)
These represent the differences between a fixed base temperature and a daily mean outdoor temperature over a specified period.

The method for calculating degree days adopted for the UK by the Meteorological Office was outlined by Anderson et al (1985). For each day during the period in question degree days are accumulated as follows:

1. if $T_b > T_{max}$

   $DD = T_b - 0.5\ (T_{max} + T_{min})$

2. if $T_{max} > T_b$ and $T_{min} < T_b$ and $(T_{max} - T_b) < (T_b - T_{min})$.

   $DD = 0.5(T_b - T_{min}) - 0.25(T_{max} - T_b)$

3. if $T_{max} > T_b$ and $T_{min} < T_b$ and $(T_{max} - T_b) > (T_b - T_{min})$

   $DD = 0.025\ (T_b - T_{min})$

Implied but not explicitly stated is

4. if $T_b < T_{min}$

   $DD = 0$

It was noted that for condition 4 the following definition, which is similar to that used outside the UK would be more appropriate.

1. if $T_b > 0.5(T_{max} + T_{min})$

   $DD = T_b - 0.5(T_{max} + T_{min})$

2. if $T_b < 0.5(T_{max} + T_{min})$

   $DD = 0$

The two methods differ in the accummulation mostly at the ends of the heating season, there being no difference on days when the external temperature is less than the base temperature throughout a 24 hour period.

6) Solar Savings

This indicates the reduction in heat demand due to the presence of passive solar heating elements and is defined in terms of the net reference thermal load (NRTL) as shown:

$$SS = NRTL - AH \tag{38}$$

or

$$SS = 1 - (\text{Auxiliary Heat})/(\text{Net Reference Thermal Load}) \tag{39}$$

7) Solar Load Ratio

$$SLR = \frac{\text{Monthly net solar energy absorbed}}{\text{Monthly net reference thermal load}} \tag{40}$$

where the monthly net solar energy absorbed (Q_s) is the total solar radiation energy absorbed by the interior surfaces of the passive building for a given month and the monthly net reference thermal load (MNRTL) is the sum of the heating load for a particular month and is calculated as follows:-

$$MNRTL = NLC \times DD_{MONTH} \tag{41}$$

$$\therefore \quad SLR = \frac{Q_s}{MNRTL} \tag{42}$$

8) Solar Savings Fraction (SSF)

This is the ratio of the solar savings to the net reference thermal load

$$SSF = \frac{SS}{NRTL} \tag{43}$$

The SSF may be defined as that fraction of the thermal load of the non-solar parts of a building that is saved by the solar components.

2.8 FAST SOLAR LOAD RATIO CORRELATIONS

A set of fast solar load ratio (FSLR) correlations were developed by Wray et al, (1984) for passive solar heating systems appropriate for use on metal buildings either as part of the original construction or as retrofits. These performance correlations were based on the theoretical and experimental work of Biehl et al (1983) and Schurr & Wray (1984) reported above.

The FSLR monthly performance correlation is a streamlined version of the SLR correlation originally developed by Balcomb et al (1980). It was used to report work carried out on the Radiant Panels and TAPs described previously. Briefly, the results of this work showed that the primary advantage of the TAP was that solar heat is delivered by a system that has a high resistance to heat loss to the ambient environment. Daytime losses are low and low night-time heat losses can be achieved without the use of expensive moveable insulation. It was concluded that Radiant Panels are less expensive.

The FSLR method is based on the relationship between the monthly solar heating fraction (SHF) and the "monthly scaled" solar load ratio (SLR*)

$$SHF = 1 - \exp(-SLR^*) , \tag{44}$$

where

$$SLR^* = F^* \, SLR , \tag{45}$$

$$SLR = {}^{S}/Q_L , \text{ and} \tag{46}$$

$$Q_L = (LCR + G)^* \, DD \tag{47}$$

F is the scale factor in the above equations and G is the effective conductance of the solar aperture, both quantities being sytem-dependent correlation parameters. DD is the monthly degree days (calculated on an hourly basis) and Q_L the total effective heat load per unit of aperture area. LCR is the ratio of the net load coefficient to the aperture area. S is the solar radiation absorbed monthly per unit area of aperture which, when divided by the effective load, yields SLR, the solar load ratio. Multiplying SLR by the scale factor yields the scaled SLR.

The SHF as defined in the FSLR method is:

$$SHF = 1 - Q_{AUX}/Q_L \tag{48}$$

where Q_{AUX} is the monthly auxiliary heat requirement of the building.

It was stressed that Q_L is a <u>total</u> load per area of aperture as it includes the effective aperture conductance G.

2.9 DAILY EFFICIENCY DERIVED FROM THE FSLR MONTHLY PERFORMANCE CORRELATION

$$SHF = 1 - \exp(-SLR^*) \tag{44}$$

From (46)

$$SLR = S/Q_L$$

And from (45)

$$\begin{aligned} SLR^* &= F^*SLR \\ &= F^*S/Q_L \end{aligned}$$

Now from (47)

$$Q_L = (LCR + G)^* DD$$

$$\therefore \quad SHF = 1 - \exp\left(-\frac{F^*S}{(LCR + G)^*DD}\right) \tag{49}$$

Now

$$F^*S = \eta\, IA$$

$$\therefore \quad SHF = 1 - \exp\left(-\frac{\eta IA}{(LCR + G)^*DD}\right) \tag{50}$$

From (48)

$$\frac{Q_{AUX}}{Q_L} = \exp\left(-\frac{\eta IA}{(LCR + G)^*DD}\right) \tag{51}$$

Efficiency can therefore be expressed as follows:

$$\eta = -\frac{(LCR + G)^*DD}{IA} \ln\left(\frac{Q_{AUX}}{Q_L}\right) \tag{52}$$

2.10 MATHEMATICAL MODELLING

Mathematical models which describe the performance of natural circulation air heating can be divided into those which are based on complex numerical heat transfer relationships which are employed to analyse simplified, idealised geometries, and those which make major assumptions regarding the heat transfer and fluid flow phenomena, but which do not require advanced computational techniques nor equipment.

In the latter category Kohler (1981) developed an empirical model for quantitatively examining the performance of thermosyphoning air panels. This model, entitled "TAPFLOW", allows variation of several parameters to examine their effects on collector performance. Many simplifying assumptions were made to enable it to be programmed on a hand-held calculator: TAPFLOW does not, for example allow for non-linearity of friction loss coefficients in the entrance and exit parts. Summers (1982) undertook validation of this model in which the values of these loss coefficients were originally estimated at 2.4, were revised to 15, and were finally considered variable. The TAPFLOW model predicted efficiencies systematically higher than the measured system produced. Reno (1981) also compared theoretical analysis with experimental data for thermosyphoning air panels using TAPFLOW and also arrived at a friction loss coefficient of 15 for the system tested. According to Varcolik & Versteegen (1980) the pressure loss coefficients for the upper and lower vent configurations of Trombe walls could only be determined experimentally which suggests that the same applies for thermosyphoning air panels.

Several mathematical models have been developed to aid the design of Trombe wall heating systems and have been used to analyse the performance of thermosyphoning air panels.

Ormiston, et al (1985) conducted an analysis of an idealised Trombe wall system which fully accounted for the interaction of the room and the air flow in the channel between the glazing and the wall. This work included predictions for cases in which the glazing temperature was lower than the room temperature. An analysis was presented which showed that, for a given geometry, the heat transfer and flow was defined by two Rayleigh numbers one of which governs the natural convection interaction between the glazing and the wall, and the other which governed the natural convection interaction between the room and the channel. One of the assumptions made in this work was that the wall separating the glazing and the room was adiabatic and, therefore, the heat gains and losses which would normally occur via the storage wall of a Trombe wall system were not considered. The analysis presented could be said to be more representative of a thermosyphoning air panel than a Trombe wall.

Hocevar & Casperson (1979) presented data and analysis of the laminar flow patterns, thermocirculation rates and collector efficiencies for vented Trombe wall systems. The results of their research confirmed the complex nature of the natural convection which occurred in the flow channel between the masonry wall and glazing. Several variables were found to have a significant impact on velocity profiles: ambient air temperature, insolation rate, masonry surface temperature, and the elevation above the Trombe wall inlet duct. The flow channel depth was considered to have a particular influence on the convective flow process. A 2.4 m high Trombe wall was monitored which featured a moveable cover panel. This allowed the flow channel dimension to be varied from about 2.5 cm to 20 cm. It was found that there were distinctly different flow patterns in the narrow and wide channels and that, in general, the maximum velocity value observed decreased as the gap width increased. The results indicated that most of the flow regimes encountered during the experiments were laminar, with a possibility of transitional flow developing in wider flow channels.

A simple model to predict the convective performance of a Trombe wall channel was developed by Akbari & Borgers (1979a). Preliminary comparisons with the experimental work of Casperson and Hocevar (1979) showed satisfactory agreement. Akbari & Borgers (1979a) carried out a detailed numerical solution of the air flow and heat transfer phenomena for laminar free convective flow in the flow channel of a Trombe wall. The momentum and energy equations were solved by the finite-difference method, and the mass flow-rate and convective heat transfer to the air was then calculated. The results obtained were correlated in terms of dimensionless parameters. The correlations were used to furnish estimates of the convective heat gain, mean temperature profile, and the flow-rate of air leaving the vertical air channel, having determined the channel width, height, surface and inlet temperatures. However, this analysis did not take into account the flow resistance which occurs at the inlet and outlet vents of the flow channel.

The "Akbari & Borgers model" was extended by Tichy (1983) to include the effects of inlet and exit friction losses and by Tichy & Quin (1982) to study the importance of heat transfer by convection in Trombe walls. An insulated wall with no thermal storage capacity was found to be superior to concrete walls over the heating season in terms of efficiency, heat delivered and air flow delivered. This was considered to be true regardless of the climate : the more severe the climate the more decisive the superiority. Theoretical analysis using four channel gap depths, four wall heights, four climate locations and walls of 304 mm concrete, 101 mm concrete, and massless insulation was undertaken. This demonstrated the greater efficiency of the massless wall and that a gap width of 0.15 m produced the highest total efficiency, regardless of wall height. Efficiency was not found to be particularly sensitive to gap width and not at all to wall height.

A method for determining the flow rate through the air channel of a Trombe wall based on the solution of Bernouilli's equation for non-ideal flow was presented by Varcolik & Versteegen (1980). Their method accounts for pressure drops using loss coefficients for flow through the inlet and exit vents, and friction factors for flow through the channel. On the basis of theoretical analysis it was concluded that proper design of this type of thermal storage wall can only be achieved if pressure loss coefficients for the upper and lower vent openings are known and that these can only be determined experimentally as noted previously.

Summers (1981) conducted experiments with a back-pass thermosyphoning air heater to validate the TAPFLOW programme. The results of this work indicated that the model gave performance predictions systemmatically higher than results obtained experimentally. This was attributed to difficulty in determining inlet and exit friction coefficients for the programme. However, the dimensions of the Back-Pass collector used to validate the programme may have contributed to this discrepancy. The TAP measured over 4.26 m in height with an air channel of just over 1% of the depth length ratio, whilst inlet and exit ports did not extend over the entire width of the collector. One conclusion drawn from this work was that the value of the dynamic loss coefficient may vary with the flow-rate through the collector.

Biehl (1981) used experimental data and mathematical data to study design parameters for natural convection solar air heaters. The collectors' performance was measured using the closed-loop facility described by Morris (1980) Fig. (13). The convective loop was evaluated by equating the flow pressure losses with the system driving pressure, defined by the area enclosed in a density elevation diagram. Iteration of the air flow-rate through the collector was then undertaken until the numerical difference between flow losses and driving pressure was considered to be acceptable. Turbulent flow was assumed and good agreement claimed between the results obtained experimentally and theoretically. Three collector configurations were tested and simulated numerically to investigate the effects of varying flow channel depth, whilst a fourth was used to examine the influence of collector length and channel tuning resistance. The configurations consisted of channel depths of 266.7, 165.1 and 89 mm for collector lengths of 4.57 m, and a channel depth of 89 mm for a collector length of 2.286 m.

Biehl (1981) stated that his unpublished studies indicated that collector efficiency, mass flow rate and temperature gain were sensitive to the air channel depth behind the absorber plate: flow rates decreased and efficiency increased with decreasing channel depth. However, air flow was not measured directly during this work but was determined from an energy balance on an air-to-water heat exchanger used to cool the heated air. The analytical model was used to show that a collector length of between 1.22 and 6.1 m maximised efficiency. The importance of well designed flow channel corner design and its influence on performance was emphasised, modifications to the bottom flow corners of the collectors tested improved relative performance by 10%. This point was also emphasised by Reno (1981) who stated that vents should run the entire width of the TAP and that the effects of right-angle turns should be minimised. Specific recommendations for tap construction based on a comparison of results obtained

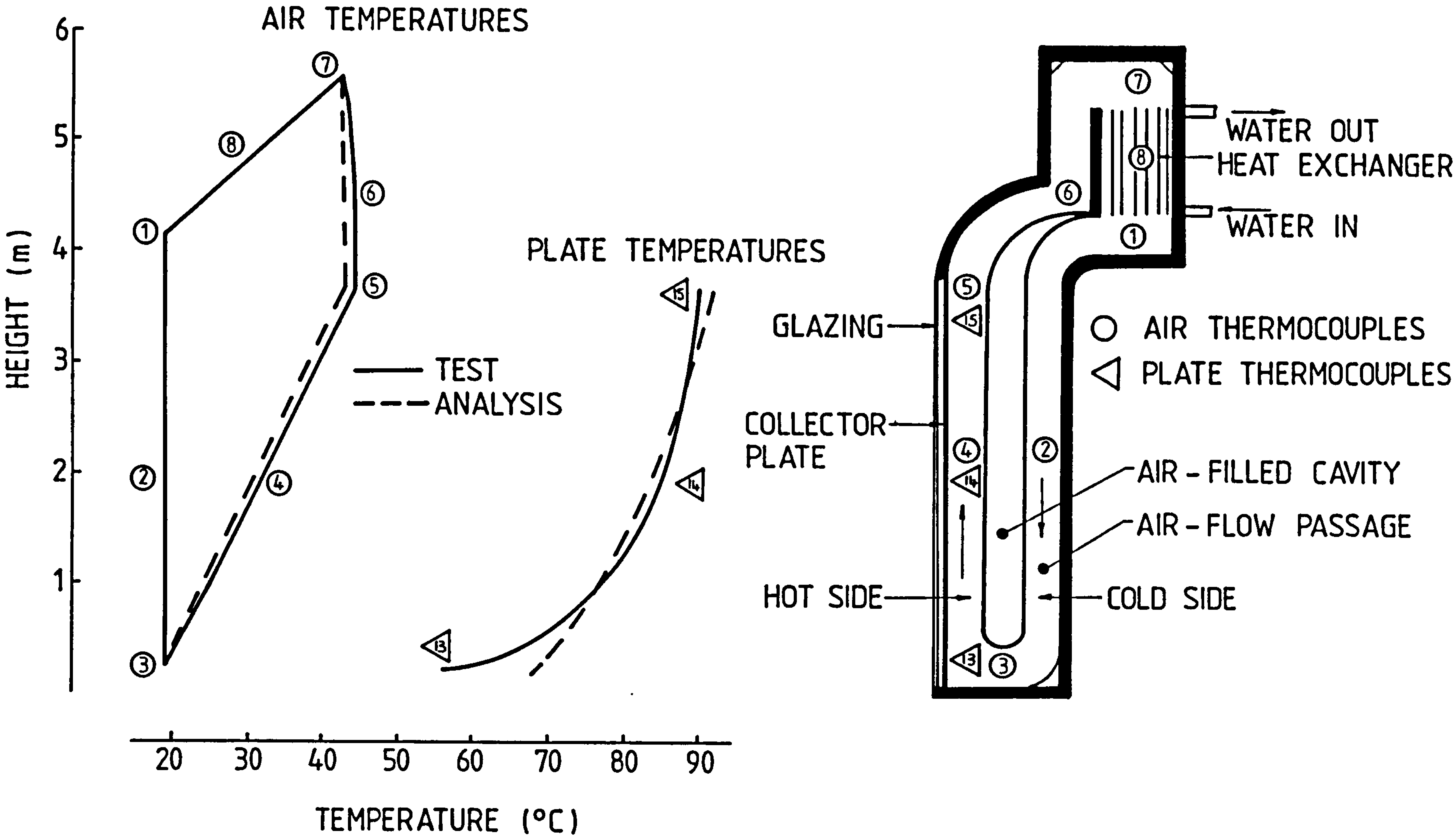


Fig.13 Air and plate temperatures measured using a closed-loop test apparatus (Biehl 1981)

experimentally, and using the TAPFLOW programme were made. Again, difficulty was experienced in determining suitable values for the dynamic loss coefficients. The computer simulation was used to determine the effect of channel depth and inlet and exit vent geometry. One conclusion drawn from this work was that an air channel depth of only 25 mm would give optimum performance: it was suggested that reducing the flow channel cross-sectional area increases velocity more than frictional losses. A further conclusion was that TAP performance increases with height to an upper limit of 4.28 to 4.87 m. It was noted that friction losses at the exit and entrance to a collector should be kept to a minimum and that backdraught dampers fitted to the TAP used to validate the model severely restricted the flow. Tests were carried out on a TAP 2.44 m long by 0.86 m wide with a 76.2 mm flow channel depth for a preliminary validation of the model and showed good agreement.

Sferides & Tsingas (1985) investigated the performance of a Front-Pass thermosyphoning air panel using a theoretical model. It was noted that two different flow patterns are possible in a collector of this type, depending on the prevailing conditions: 1) the formation of one upward stream along the absorber and one upward air stream along the inner face of the glazing and 2) the formation of one upward air stream along the absorber and one downward stream along the glazing. It was considered to be important to design the collector with sufficient absorber-to-glazing distance to keep the boundary layer along the absorber completely separate from the one along the glazing. Mixing of these two layers was considered to reduce efficiency by generating turbulence and reducing the speed of the air within the collector, whilst non-mixing minimised heat exchange between the two layers and leads to improved collector performance.

Under certain circumstances an appropriate method of modelling is identical to that for a thermosyphoning water heater and store. Apart from the properties of the working fluid, data obtained from an air heating system could be used in a passive water collector model to simulate the system.

Chapter 3

THE INSTALLATION AND OPERATION OF PASSIVE SOLAR ENERGY AIR-HEATING COLLECTORS

3.1 POTENTIAL PROBLEMS

There are potential problems associated with the successful installation and operation of these collectors in non-domestic buildings.

1) Loss of glazing: Reduction of glazing area to accommodate solar collectors may antagonize occupants or increase demand for artificial lighting and, hence, energy. Analyses of the potential for passive solar contributions in non-domestic buildings were conducted by Duncan (1982) and Hawkes (1982) who both argued that a high proportion of total primary energy is consumed by artificial lighting in certain types of non-domestic building. The energy contributions arising from daylighting was a benefit of passive solar design stressed in both papers. However, summertime overheating is a major problem in many over-glazed institutional buildings, whilst the large window areas combined with poorly insulated wall panels lead to high heat losses in the winter. As Rostron (1964) noted, a large proportion of unsatisfactory internal environments can be attributed to the irrational use of unnecessarily large areas of glass.

2) Integration of the solar heating system into the heating system of a building: A control strategy must be established which is sufficiently flexible and responsive so that the heat input from these devices can be used to offset demand from the existing heating system.

3) Control of collector output: As there is little incentive for the occupants of institutional buildings to involve themselves in the day-to-day operation of passive heating systems, passive solar collectors should ideally be <u>totally</u> passive and not require any manual operation of dampers, vents or shading devices. Simple manual dampers may be used to control heat output and reverse-thermocirculation, but they demand an appreciable degree of commitment and expertise from the occupants if the maximum benefit is to be obtained from a passive system.

4) Reverse-thermocirculation: At night or during non-gain periods, the column of air adjacent to the absorber may cool below the temperature of the living space. If it is not prevented from entering the building via the lower vent, this cold air falls and warm air is drawn

into the collector to replace it. The warm air is cooled and the process continues, regulated by the temperature difference between the air in the air channel and the air in the room just as in forward circulation. An efficient economic, and reliable method of preventing such reverse circulation is required. It can be inhibited by fitting a one-way valve or damper in the inlet and outlet vents of "standard" natural-circulation air-heaters, Fig. (14), or by using a U-tube collector design Fig. (15). One-way valves often consist of light-weight material suspended from the upper surface of a vent and a mesh or grille Fig. (16). Air flow in the desired direction moves the material away from the grille, but reverse circulation is prevented by the material falling back against the grille and effectively closing the vent.

Construction details for flap-valve backdraught dampers have been presented by Christopher (1979), Reif (1981), and Kornher (1984). There does not appear to have been any rigorous testing to determine the behaviour of flap-valve backdraught dampers. Their effectiveness in preventing reverse-thermocirculation, and the extent to which they impede air flow through collectors has not been quantified. The efficiency of many natural circulation air heating collectors is directly influenced by the performance of these devices and, until they can be shown to perform efficiently and reliably, reservations about their use will continue to be expressed. Tichy (1978) found the flap-valve backdraught damper to be an extremely effective device. Alternative devices such as Freon actuated dampers would, it was suggested, be far more expensive and far less sensitive to the minute pressure variations which determine the flow variation. Conversely Reno (1981) found flap-valve backdraught dampers to be extremely restricting, greatly increasing friction losses. It was argued that a manually-operated slide or hinged damper might be used alone but would require the presence of someone to operate the device during severe changes in levels of insolation. A solution proposed was a thin backdraught damper combined with a more substantial manual damper.
Morris (1978a) was of the opinion that although these dampers are inexpensive and easy to install, they are fragile, susceptible to static electricity and ripping. The opinion of American "solar energy engineers" was quoted by Ward et al (1982) "Universal concern was noted as to the lack of reliability of backdraught dampers: they leak!" Further potential problems with flap-valve backdraught damper

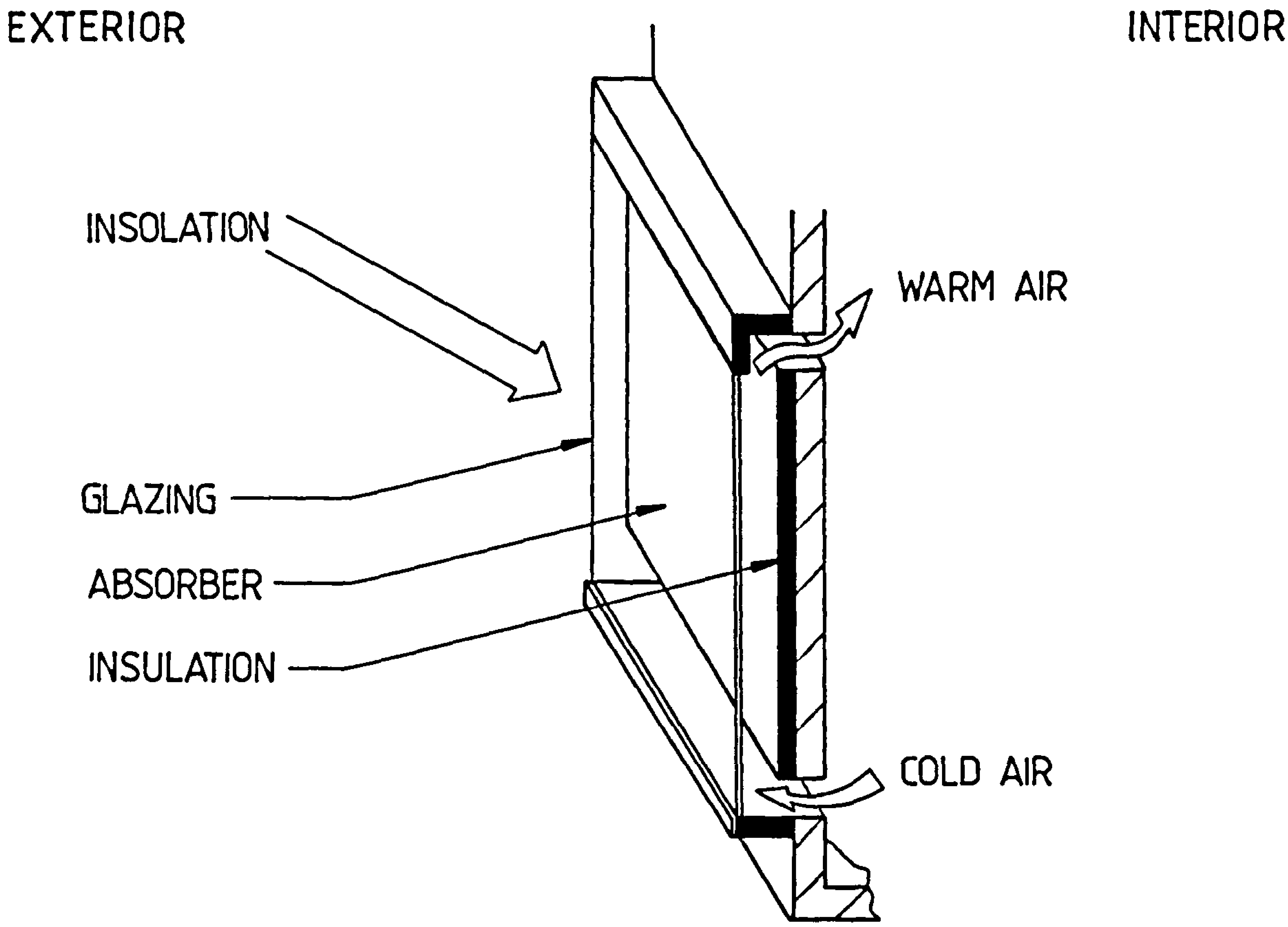


Fig.14 Standard natural-circulation air-heater (ie., the "standard" TAP)

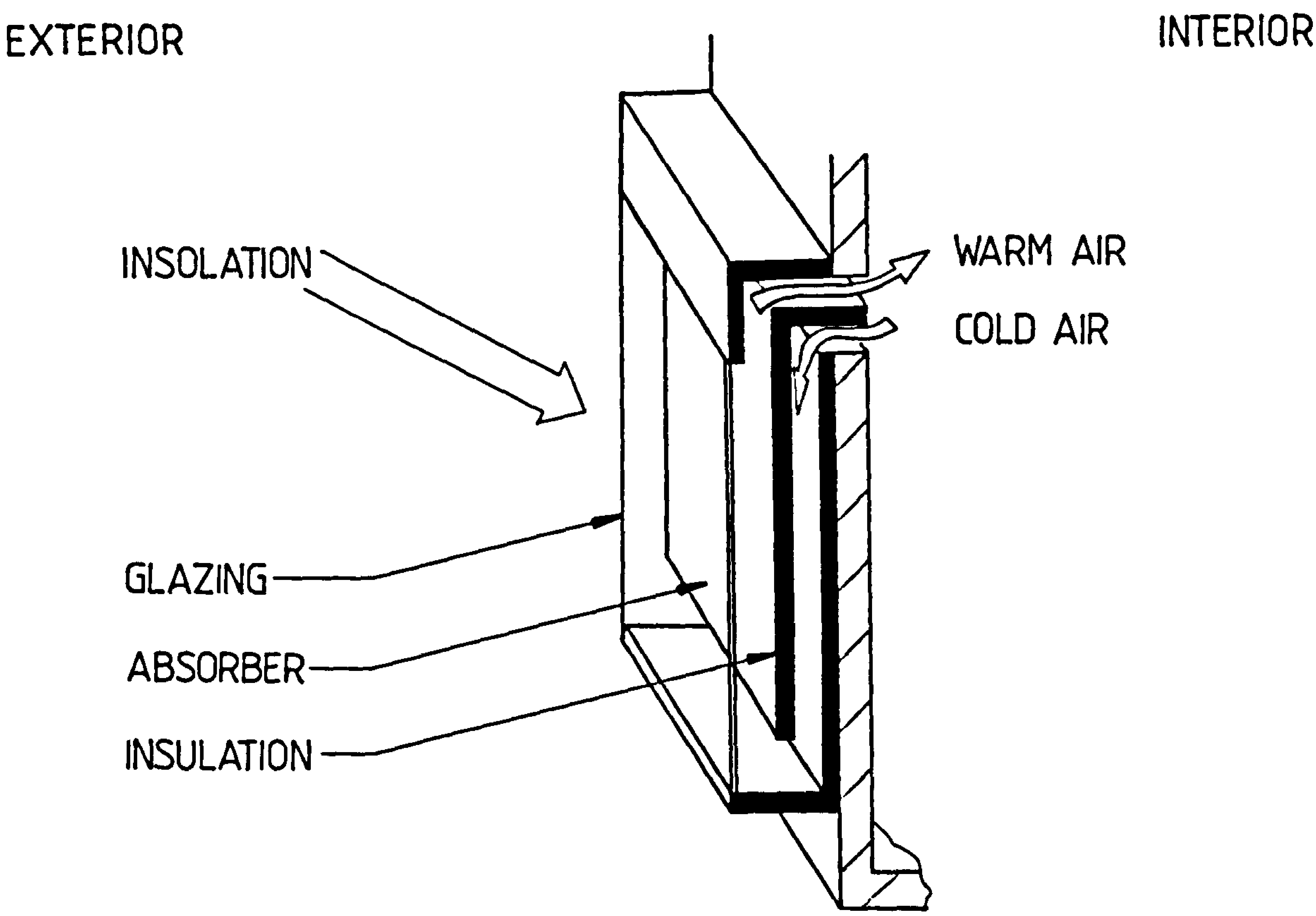


Fig.15 U-Tube natural-circulation air-heater (ie., "U-Tube collector")

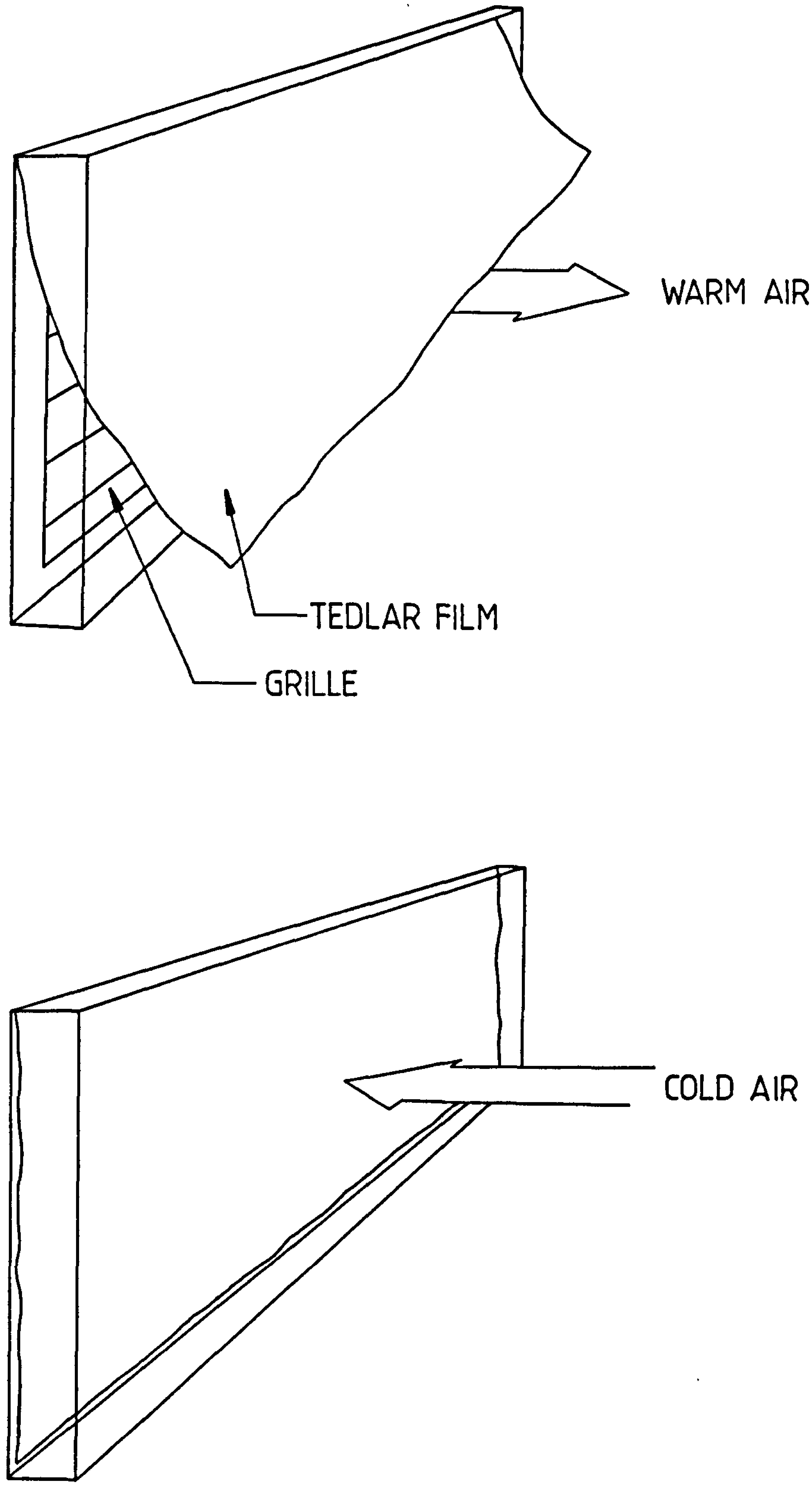


Fig.16 Operation of a flap-valve backdraught damper

were noted by Lau et al (1985) who suggested that leakage might occur as the plastic film used in these dampers aged, whilst the quality of these devices might be inadequate if they were to be assembled on-site.

This implies that factory-built backdraught damper units should be installed so as to guarantee a high standard of manufacture. Such units should be designed so that they may be removed, for repair and maintenance purposes, or replaced in a straightforward manner.

The work of Schnurr & Wray (1984) illustrated that whilst a flap-valve backdraught damper may inhibit convective losses associated with reverse-circulation, the thin film of material used in a backdraught damper will be ineffective in reventing conduction heat losses. A well insulated manual or thermally actuated damper should be effective in preventing both conductive and convective heat loss. Ideally the thermal resistance of a damper should be equal to that of the insulation in the collector to which it is fitted. Passive, thermally-actuated dampers may be powered by refrigerant-charged pistons or memory-metal springs in a similar manner to devices used in greenhouse ventilation systems. However, thermally-actuated dampers of this type do not appear to have been developed commercially for passive solar applications.

The U-tube collector was developed by Trombe et al (1965) and later by Baer (1977). The inlet and outlet vents of this design are at the top of the collector and during periods of solar gain air is drawn around the central partition, passing over or through an absorber in the process. U-Tube collectors are self-damping: during non-gain periods cold air fills the front and back channels of the collector and air movement ceases.

A U-tube collector is approximately twice the depth of an equivalent "standard" TAP which makes is more difficult to accommodate into existing buildings. It does not, however, require a backdraught damper to prevent reverse circulation, which is a significant advantage. The relative merits of the U-Tube and "standard" TAP are summarised in Table 5.

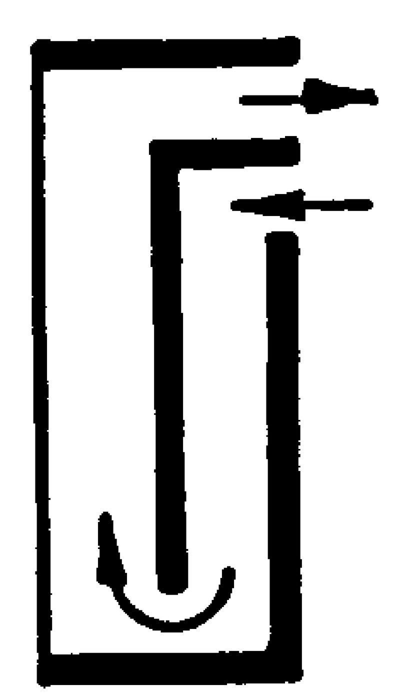
	DESIGN	ADVANTAGES	DISADVANTAGES
STANDARD THERMO-SYPHONING AIR PANEL		• SIMPLE CONSTRUCTION • DEPTH OF COLLECTOR NOT PROBLEMATIC • HEATS COLDER FLOOR-LEVEL AIR • REDUCES THERMAL STRATIFICATION	• REQUIRES DAMPER TO PREVENT NOCTURNAL REVERSE CIRCULATION • LOW LEVEL MOVEMENT OF AIR TOWARDS INLET MAY BE A SOURCE OF DISCOMFORT • ENTRAINS FLOOR-LEVEL DUST AND DIRT
U-TUBE THERMO-SYPHONING AIR PANEL		• DOES NOT REQUIRE A DAMPER TO PREVENT NOCTURNAL REVERSE CIRCULATION • IN VENTING MODE WILL REMOVE AIR FROM THE WARMEST REGION • DOES NOT ENTRAIN FLOOR-LEVEL DUST AND DIRT	• MORE DIFFICULT TO CONSTRUCT THAN STANDARD TAP, USES MORE MATERIAL • DEEPER, PROJECTS FROM, OR INTO, THE BUILDING • PROMOTES THERMAL STRATIFICATION

TABLE 5. RELATIVE MERITS OF STANDARD TAP AND U-TUBE COLLECTORS

A problem referred to by Lebens (1980), Lentz (1980) and Rogers (1974) was that premature heating of incoming air, causing a loss of buoyant pressure and corresponding loss of efficiency, can occur in U-Tube collectors if there is insufficient insulation between the front and back channels. The partition in the collector tested by Lentz (1980) had a total U-value of 1.42 $W.m^{-2}K$ and the temperature of the air in the back channel was found to be higher than at the inlet. Balcomb (1980) descibed a solar collector Fig (17) in which the insulation had a U-value of 0.56 $W.m^{-2}K$ and no mention was made of this problem, although slight reverse thermocirculation was noted. It is important to insulate the back of U-tube collectors from the living space so that, at night, heat is not transferred to the back channel and reverse thermocirculation cannot take place. Morris (1978a) recommended that glazing should not extend above the level of the bottom of the inlet vent so that, at night, the two columns of cold air are of equal height and the damping is effective. 20% of daily gains can be lost because of reverse circulation in systems designed poorly according to Morris (1987b).

5) Summertime operation: An operational strategy must be developed for a natural-circulation air-heater for periods when both solar gain and ambient temperature are high, when the heat output of these devices is no longer required. Collectors may be modified to provide ventilation rather than heating under these circumstances and, whilst this is a desirable feature of these devices, it does present several problems. The conversion from the heating to the ventilating mode must be achieved with a simple reliable and economic mechanism. Ideally, it should be a totally passive feature of the collector but, as in the case of the backdraught damper, passive devices of this kind have not been widely developed or tested. A manually operated mechanism which only allowed for seasonal changes from one function to another might not be adequate. A one-way valve must be installed to prevent gusts of wind from reversing the flow: air in the collector may be blown back down the air channel and into the room being ventilated. This would create a considerable amount of discomfort to occupants and raise the temperature of the room rather than lower it. A shading device could be employed to prevent solar radiation being absorbed and thus inhibit the thermosyphoning air flow. An alternative more simple solution, is to allow the collector to stagnate and high collector temperatures to ensue. This is only

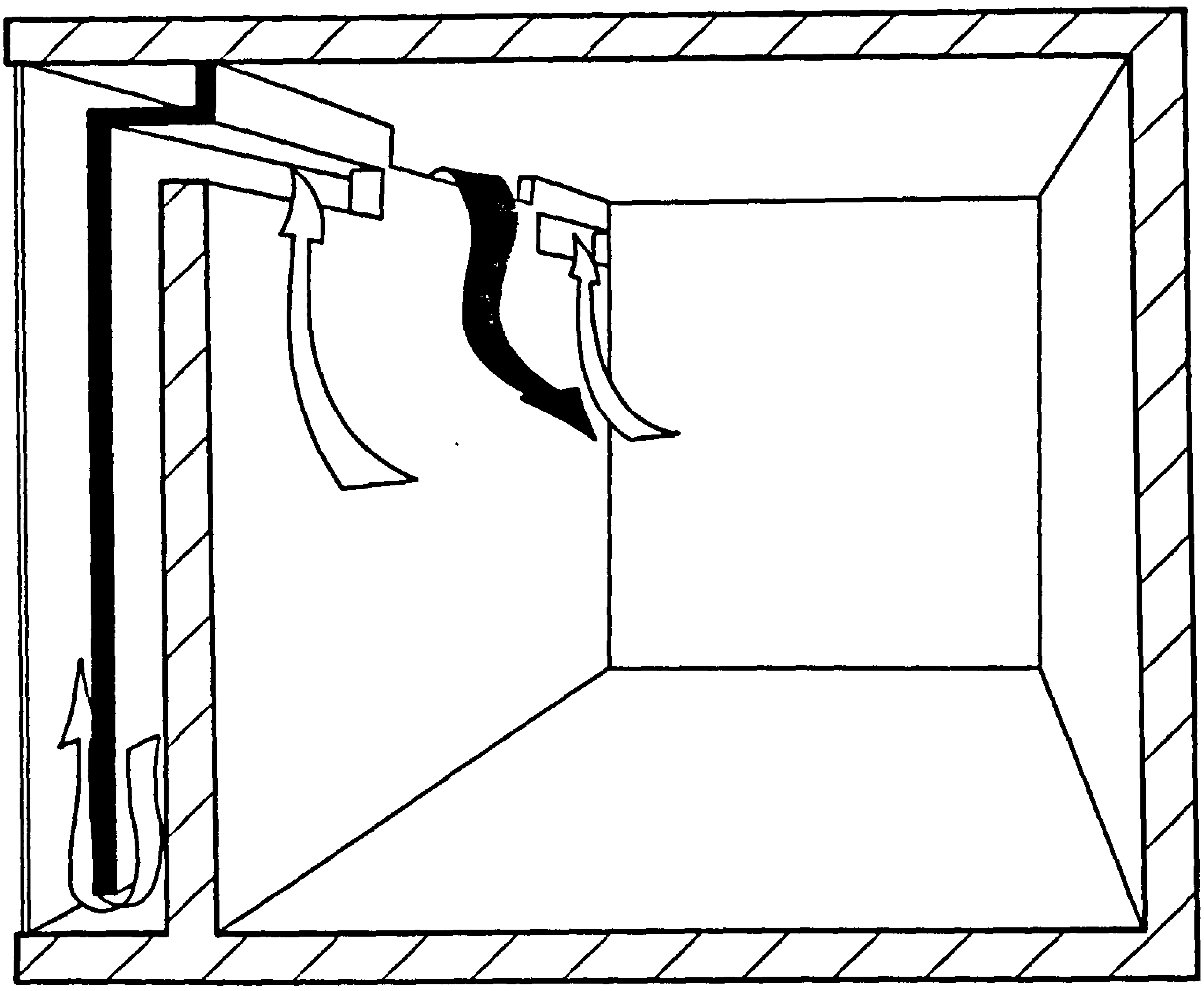


Fig.17 Configuration of a U-Tube air-heating solar-energy collector described by Balcomb et al (1980)

advisable where collector components have been designed to withstand such temperatures for prolonged periods, and have sufficient insulation to prevent heat within the collector being transferred to the building.

The four types of flow which can occur in a "standard" TAP are shown in Fig. (18).

6) Vandalism: Collectors, and in particular, collector controls must be designed to withstand the attention of vandals. This could be hazardous if a collector was made to stagnate by blocking off one or both of the vents when is had not been designed to cope with elevated temperatures. It would also be difficult to retrieve any objects which fell to the bottom of a collector, particularly a U-tube design.

7) Appearance: Solar collector panels may not prove to be aesthetically appealing. They pose a challenge that many architects may avoid. It may be necessary to sacrifice solar-energy collector efficiency to make these units more attractive and compatible with the surrounding building fabric by using coloured absorber plates, tinted glass or some other method.

3.2 VERTICAL ORIENTATION:

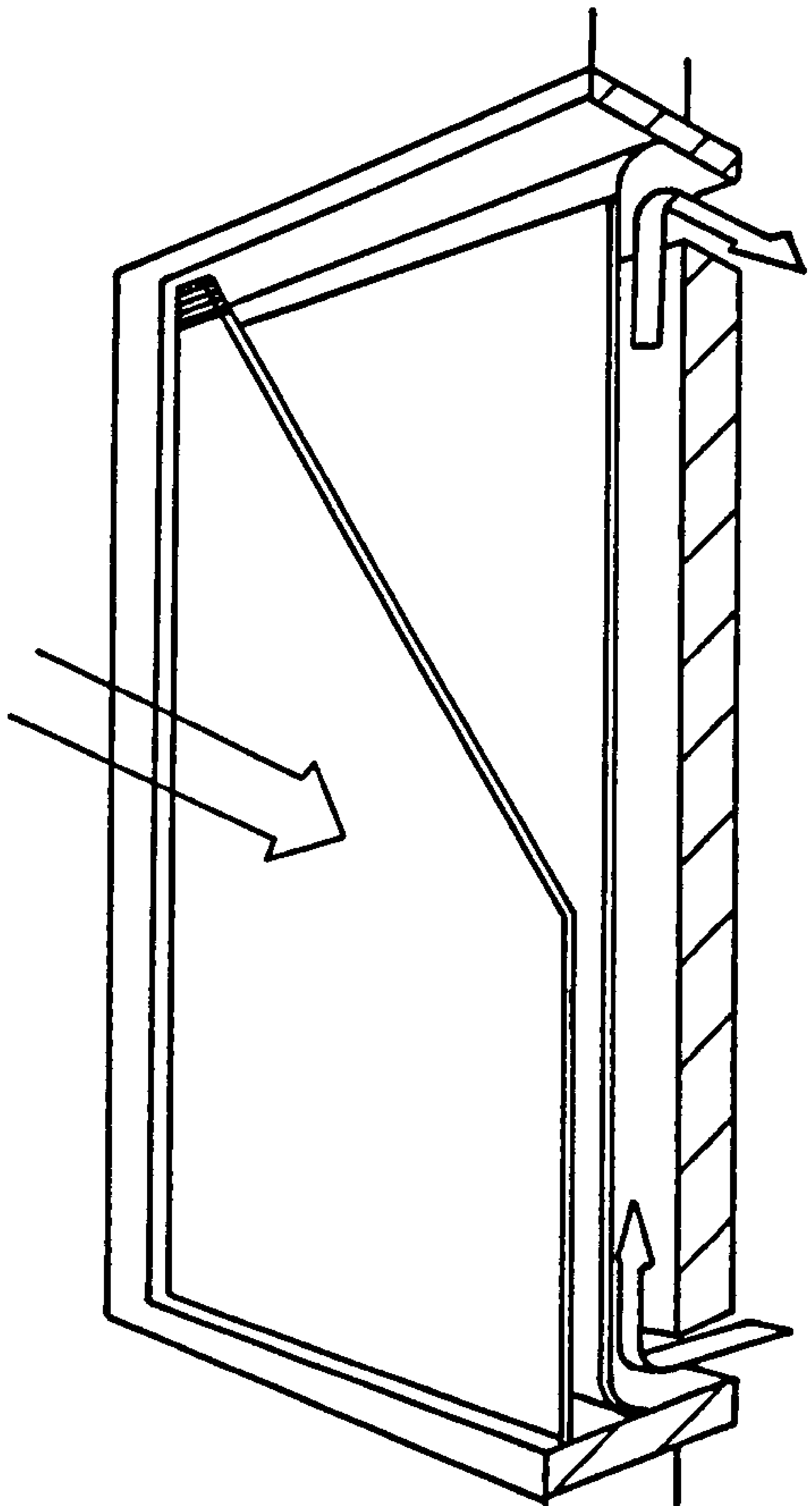
The majority of solar collector panels retrofitted to building walls, or incorporated in them, will be positioned vertically, rather than at the optimum angle. A slight performance penalty is incurred with vertical collector surfaces and this loss of efficiency depends on the latitude of the collector site. Estimates for Ottawa, Canada, 45.3° latitude, show that a vertical surface receives only 7% less incident solar energy than a 60° sloped surface over the entire space-heating season [Sunworld (1986)]. Moreover, it was argued that a vertical collector surface has a number of advantages over a sloped collector.

1. Less incident radiation during the summer months.

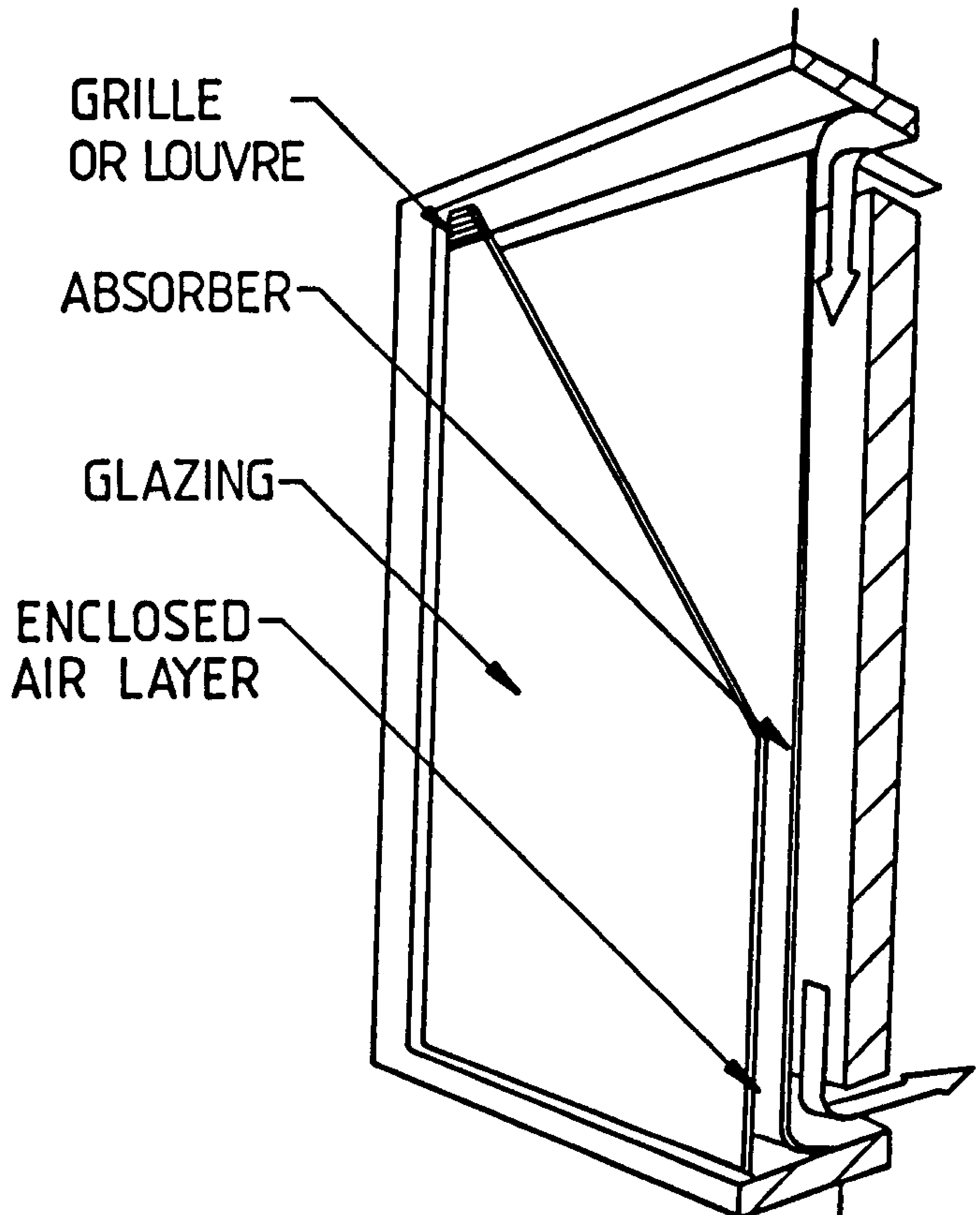
2. Vertical surfaces receive more reflected sunlight from ground snow

3. Structural costs for wall-mounted systems are generally low.

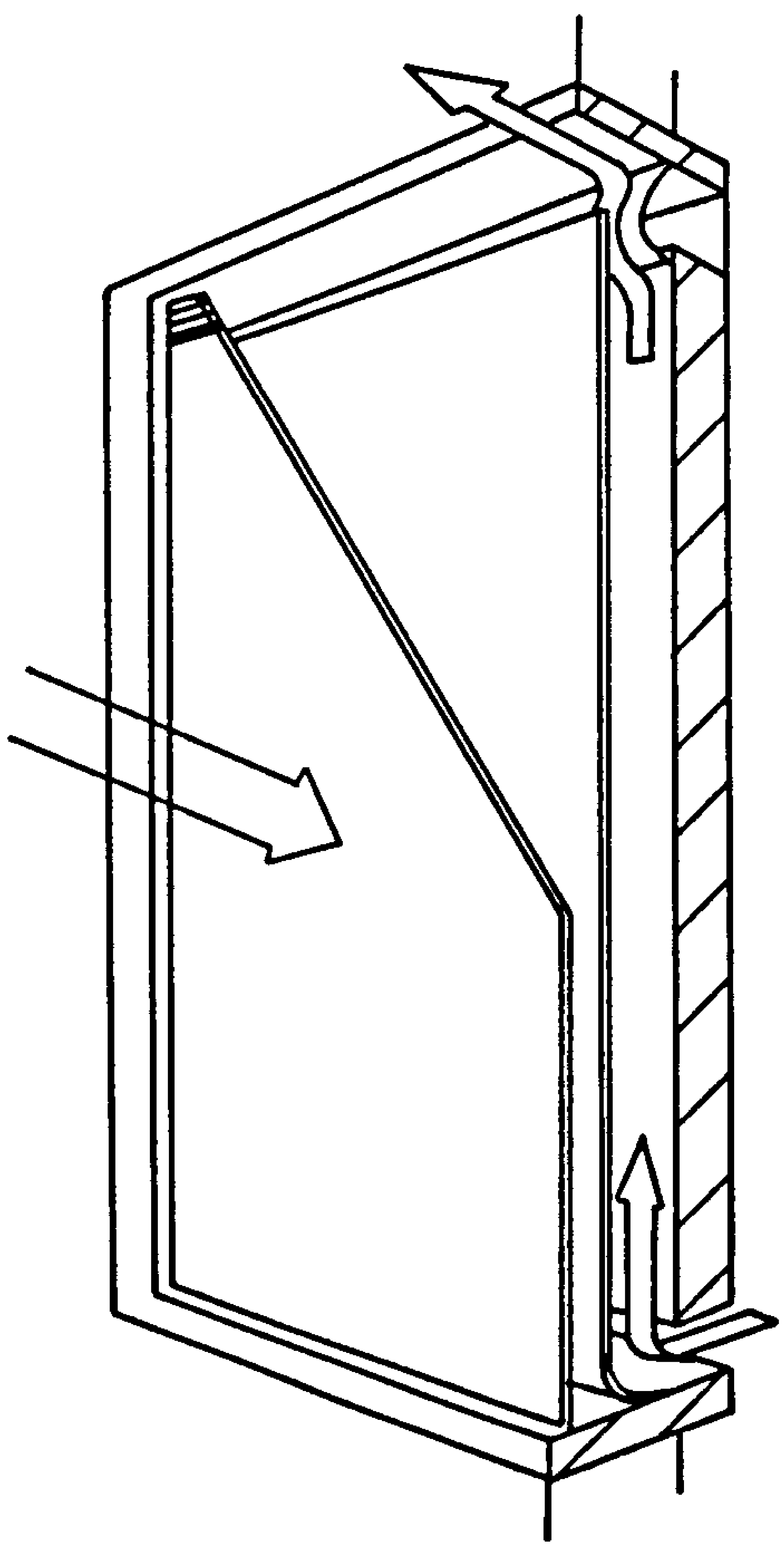
Fig.18 Four types of flow which ensue in a thermosyphon air-heating solar-energy collector



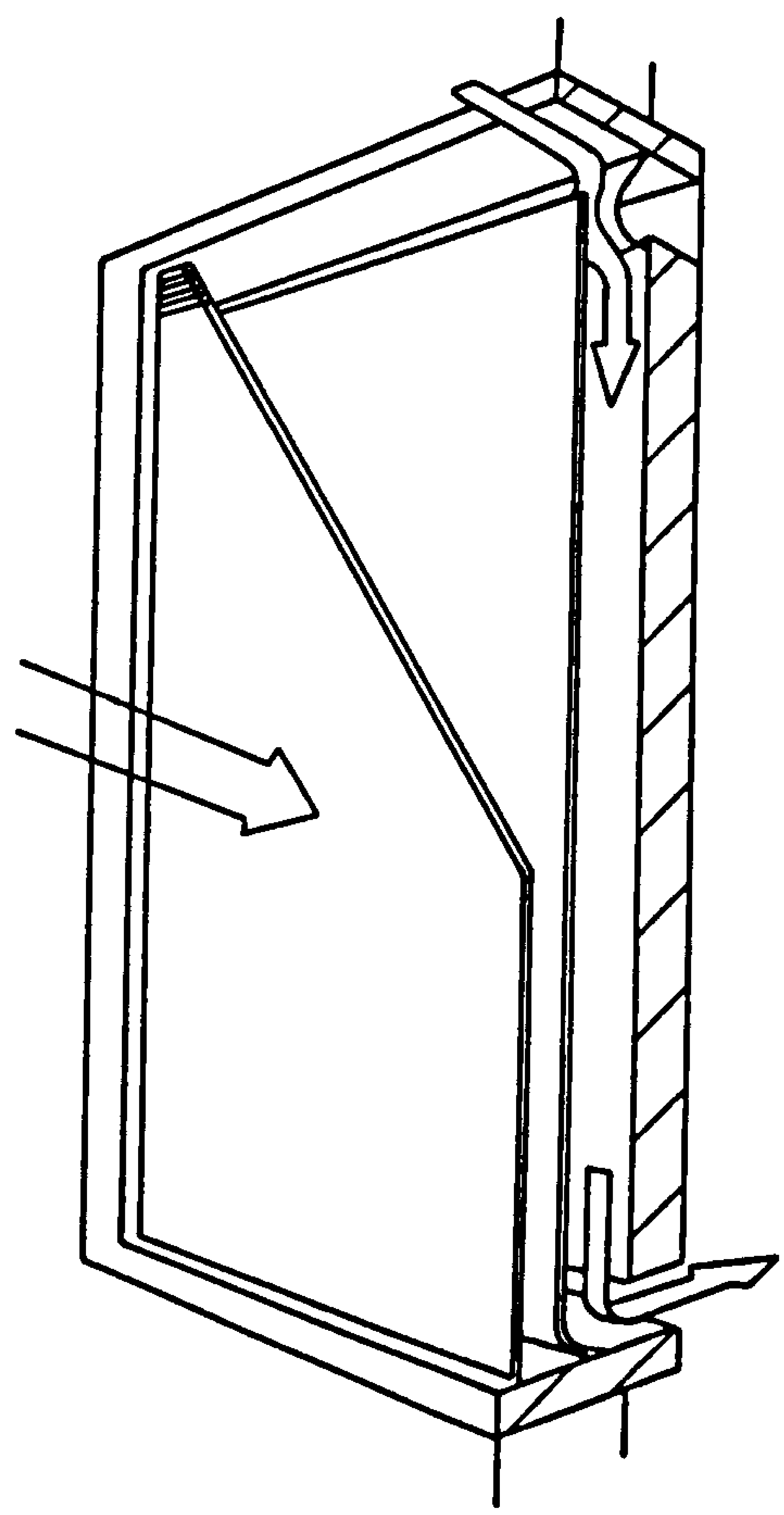
COLLECTOR IN HEATING MODE



REVERSE THERMOCIRCULATION



COLLECTOR IN VENTILATING MODE



WARM AIR BLOWN BACK INTO
COLLECTOR BY PREVAILING WIND

4. Snow-build-up is not a problem on a vertical surface.

5. Vertical panels rarely add wind loads to the building.

3.3 THERMAL COMFORT AND PASSIVE SOLAR HEATING

As Colton (1978) noted, solar dwellings are especially vulnerable to situations unfavourable to thermal comfort, and the first consideration should always be to the individuals who inhabit these dwellings. Alder et al (1984) suggested that the performance of solar houses should be compared on the basis of both thermal efficiency and thermal comfort, as the ultimate thermal performance of passive solar buildings is rarely achieved because of comfort limitations. Ford (1982) stated that it is the occupants of passive solar buildings who define the acceptability of the internal environment, rather than the designers. One consequence of this was that heating controls must be responsive to, and easily understood by the occupants, whilst being sufficiently flexible to take advantage of any useful solar gain.

A phenomenon which is of particular relevance when discussing the comfort problems associated with passive solar heating is the low level movement of cool air. Results from tests conducted by Fishman (1978) on low level air movement showed that the overall assessment of the thermal environment by male and female subjects depends not on the mean air temperature, but on the assessment of the environment around the feet. With cold feet the individuals felt cool overall. Cold air returning from the back of a room to the inlet of collector may constitute an unacceptable source of discomfort. An example, quoted by Ford (1982) was of houses fitted with Trombe walls in Bebington, UK; the elderly tenants blocked off the wall openings to overcome this problem, and in doing so stopped the walls from operating. Rey et al (1982) observed, when testing a Trombe wall system, that regardless of the operating mode, thermocirculation had a negative effect on the comfort condition; and this was attributed to the low inside surface temperatures that the process caused, although precise details were not given.

U-Tube collectors may overcome the potential thermal comfort problem of low level air movement associated with the "standard" thermosyphoning air panel and Trombe wall design as they enable air inlets to be situated at ceiling, rather than floor level. Also the entrainment of dust and dirt into this type of collector may be less of a problem as heavier-than-air particles are less numerous at ceiling level. However, as Mazria (1979) noted, in cold climates convection-type heating systems can lead to unusually large floor-to-ceiling temperature gradients, with low floor temperatures causing thermal discomfort. A U-Tube collector will be less effective in overcoming this problem than a "standard" TAP.

The relationship between passive solar heating and thermal comfort has not been studied extensively. As with many other features of passive solar collectors, thermal comfort depends very much on the geometry of the room in which heaters are installed. It is possible that the only satisfactory way to simulate or predict the thermal comfort performance of a collector is to use a mathematical or physical model which describes both the collector and the room. It will be difficult to determine the impact a collector has on comfort merely by testing the collector. It would be unfortunate if an efficient and economic unit was installed in large numbers only to discover that it constituted a major source of discomfort to occupants.

3.4 THE POTENTIAL MARKET FOR THERMOSYPHONING SOLAR-ENERGY AIR-HEATING WALL PANELS IN THE UK

Many commercial and public organisations are having to improve the working environments within their premises, and the refurbishment of existing buildings is often a less expensive and more convenient method of achieving this than the acquisition of new premises. However, a more pressing reason for the refurbishment of buildings is the failure of ageing glazing and curtain wall cladding systems.

The premature deterioration of many buildings constructed in the years since the last war has been attributed to the widespread use of new materials and techniques in the absence of appropriate standards controls or the necessary expertise [Building Refurbishment & Maintenance (1986)]. The introduction of the curtain wall cladding technique enabled buildings to be constructd with external walls which were merely lightweight, non-load bearing assemblies of components suspended from structural frameworks. The performance of many of these curtain walling systems can depend on the integrity of sealants, none of which had a guaranteed life in excess of twenty years. [Endean (1986)]. The failure of these sealants and any subsequent water penetration can lead to internal rotting and structural damage.

It was estimated that the cost of carrying out the amount of curtain-wall refurbishment needed in Britain could have been as much as £10 - £12 billion in 1984 [Building Refurbishment and Maintenance (1984)]; this sum being divided equally between public and private buildings. This represents a large potential market for passive solar collectors which they can satisfy only if they are designed to perform adequately both as recladding and solar heating components.

In a report on passive solar design in non-domestic buildings, Hawkes & Penz (1984) showed that modifications to existing buildings are likely to provide the major proportion of the benefits that passive solar heating can offer in the short-term as so few new buildings are being currently added to the existing U.K. stock. It was argued that even in times of high economic activitity, new building in the UK represents between 1% and 2% of the building stock in any year. The impact of passive solar design, if it were to be dependent on new building in the prevailing economic climate, would be almost negligible. They noted, for example, that the public sector school building stock in 1976 consisted of a total of 28,310 buildings of which 12,250, 43%, were built before 1945. The number of new school buildings is likely to be small in view of the reduction in school-age population predicted for the UK. However, it was suggested that the effects of reorganisation, maintenance and repair are likely to create opportunities for investment in the existing buildings and some of this investment should be used to increase the passive solar heating capability.

3.5 RETROFIT OPTIONS

Three methods of retrofitting passive solar energy air heating collectors to institutional buildings, and one method for industrial buildings, are outlined. Their relative merits are summarised in Table (6).

	DEFINITION	ADVANTAGES	DISADVANTAGES
CLADDING COLLECTOR	Unit which acts as cladding and solar collector. Replaces conventional cladding panel as part of a refurbishment programme	• No more difficult to install than conventional recladding unit	• Cost of construction • Depth of collector limited by depth of conventional unit • May require development of totally new cladding system
OVERCLADDING COLLECTOR	Unit which fits between overcladding and existing building envelope during refurbishment programme	• Depth of collector only limited by space between overcladding and existing building envelope • Collector does not require extensive weatherproofing	• May be expensive • Inlet and outlet openings must be made in building envelope • Moving parts within collectors, such as dampers, may not be readily accessible for repair or maintenance
GLAZING COLLECTOR	Panel designed to fit behind existing glazing to provide insulation and a convective heat input	• Inexpensive • Easy to install • Does not have to form part of a refurbishment programme	• Occupies interior space • Depends on integrity of existing glazing framework • Glazing area must be reduced
METAL BUILDING COLLECTOR	Part of the existing south-facing metal cladding is glazed and thus forms the absorber for a retrofit tap	• Cost effective • Straightforward installation • Makes use of existing building components	• Large area of building has to glazed • Cannot be implemented on a small scale • May only be suitable for new buildings

Table 6. Relative Merits of Four Options for the Retrofitting of Thermosyphoning Air Panels

SOLAR ENERGY COLLECTOR CLADDING PANELS

By integrating air-heating collectors with the fabric of commercial buildings the marginal costs of making a wall act as a solar collector could be kept very low. An integrated collector design has been proposed based on insulated metal cladding [Gillett (1984)]. Francis et al (1982) identified wall panels as the most promising location for active air heating collectors, and it was suggested that collectors which did not make use of the cladded surface would either dominate the form of a building or provide only a small fraction of the heating load, because of the limited space otherwise available. Hoffman et al (1983) reported on a study of the theoretical and practical aspects of utilising building elements as passive heat collecting and storage systems and made similar recommendations.

An active air heating cladding collector has been developed by Stambolis (1983) Fig. (19). It consisted of a metal absorber plate incorporated in a light, well-insulated non-load bearing wall structure. It was estimated that the addition of the solar collector component produced only a marginal increase in the cost of constructing a light cladding panel; between 10% and 20% of the overall panel cost. Light cladding panels were defined by Rostron (1964) as panels which require a secondary means of support or attachment and which carry no loading beyond their own dead weight and forces such as wind and cleaning loads. A heavier cladding collector was also proposed, Fig. (20). This would function in a similar manner to the modified Trombe-Michel wall developed by Lee (1979) and would be subject to a longer time constant than conventional active air-heating systems.

The development of wall panels which fulfilled the dual functions of providing active solar air heating and protecting the exterior of a building was reported by Turoluv & Khrustov (1981). Three designs were tested: Front-Pass with a flat absorber plate, Back-Pass with a flat absorber plate, and Back-Pass with a corrugated absorber plate. The Front-Pass design was rejected because of its low efficiency and the difficulty of maintaining adequate air-tightness of the flow channel. The efficiency of the collector with a flat absorbing surface was found to be only 4% lower than that of the corrugated surface despite the considerably greater surface area this presented to the air-flow. Units installed as part of the solar heating system

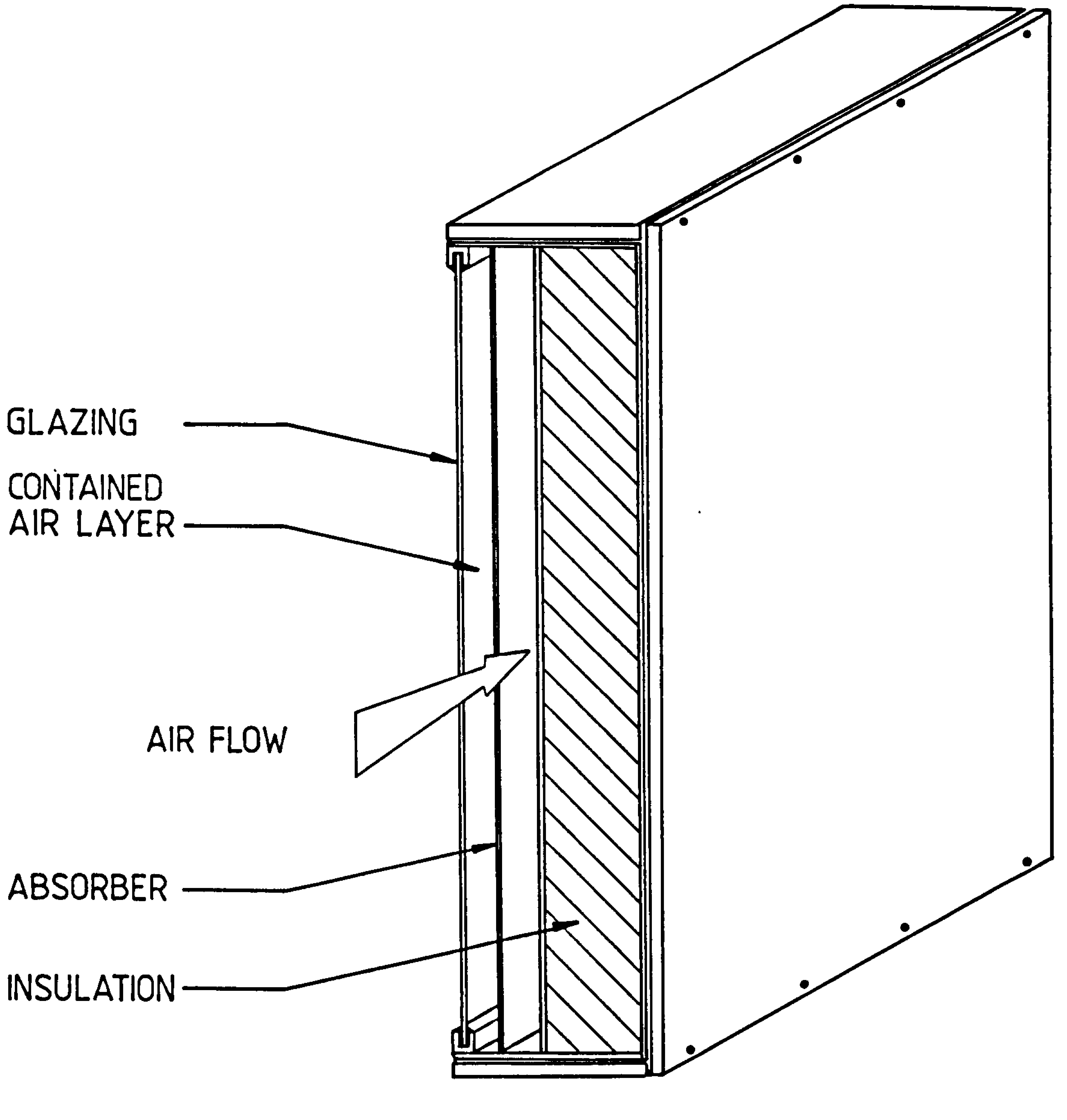


Fig.19 An example of an active air-heating solar-energy collector cladding panel (Stambolis 1983)

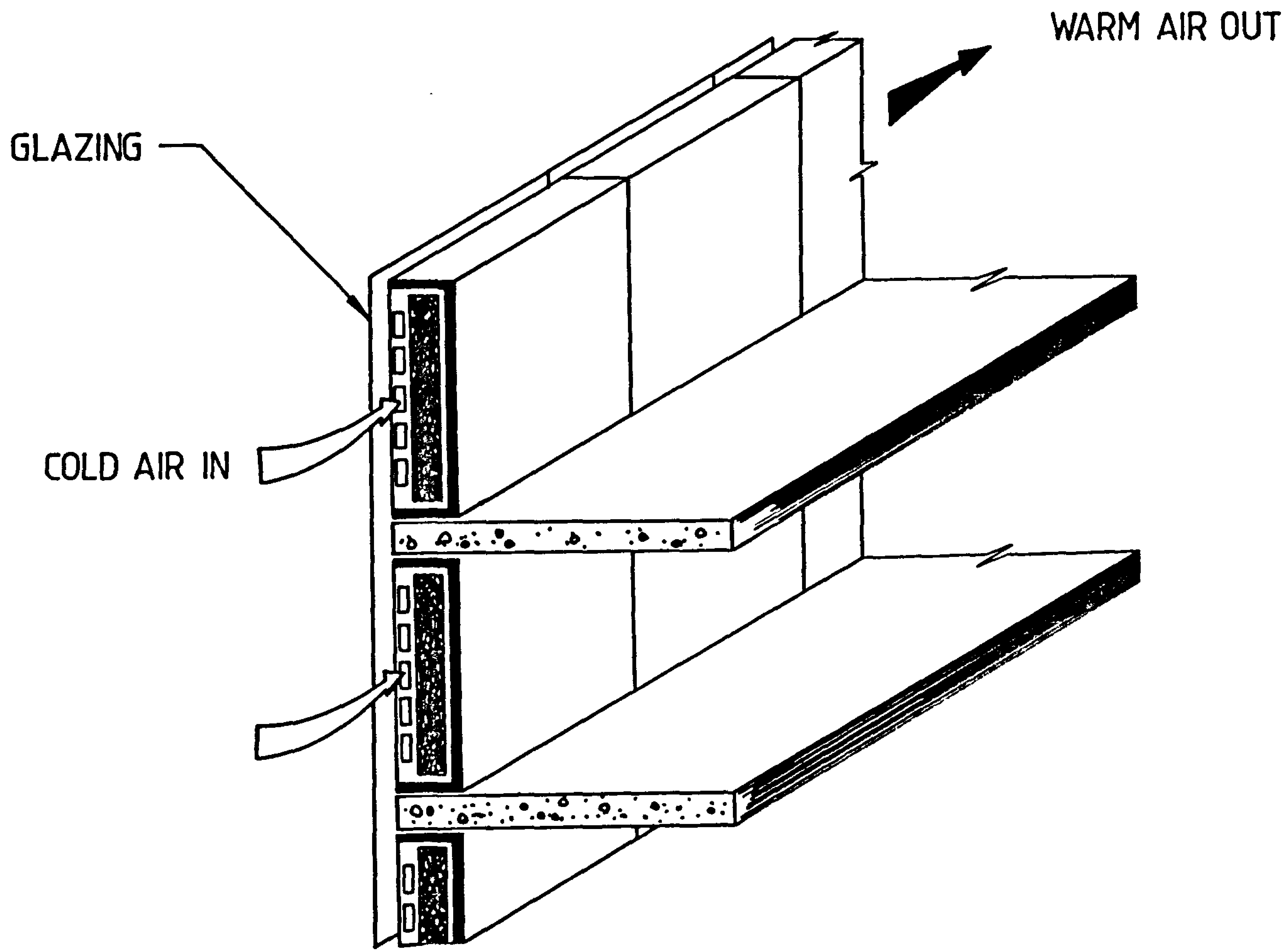


Fig.20 A mounted precast concrete active air-heating solar-energy collector cladding panel (Stambolis 1983)

of the Cardiological Centre in Tashkent, USSR provided 15% of the total heat requirement of the building : 15,161 GJ. per annum.

A curtain wall solar collector which featured latent heat storage and combined the features of collecting and releasing heat at ambient temperature through fan-assisted circulation has been developed by Colombo & Pellegrini (1984). It is not known whether the Stambolis collectors have progressed beyond the development stage. The latent heat storage collector has not [Pellegrini (1986)].

Sferides & Tsingas (1985) reported on the design and development of a natural-circulation solar air-heater, constructed of low-cost materials and intended to form an integral part of building walls. Its performance was investigated using a theoretical model. Two conclusions drawn from this study were that glazing should be made from a low conductivity material, Double Wall Polycarbonate Sheets being a preferred option, and that over-insulation of the collector would lead to high absorber temperatures and correspondingly high radiation losses during periods of high insolation. An extensive list of references was included in this report. An extensive review of the properties of the various materials appropriate for use in the construction of this type of collector was also undertaken and costings of chosen materials were computed.

A combined solar collector and cladding element should enable large areas of southerly facades to be adapted for solar energy collection without altering the overall appearance of a building Fig. (21). However, passive solar collector cladding panels will be more difficult to incorporate into buildings than active panels. Passive collectors have to conform to dimensions which allow thermosyphoning air movement to be generated; collector depth is governed by the height, unlike active collectors. Furthermore, significant passive solar contributions can only be realised by replacing some existing glazing with collector panels.

The problem of collector depth requires careful consideration as it may not be possible to manufacture passive solar collector panels to the same dimensions as standard cladding units. It may be necessary to manufacture non-standard 'deep' cladding panels to combine with the cladding collectors. Weatherproofing costs might render the unit

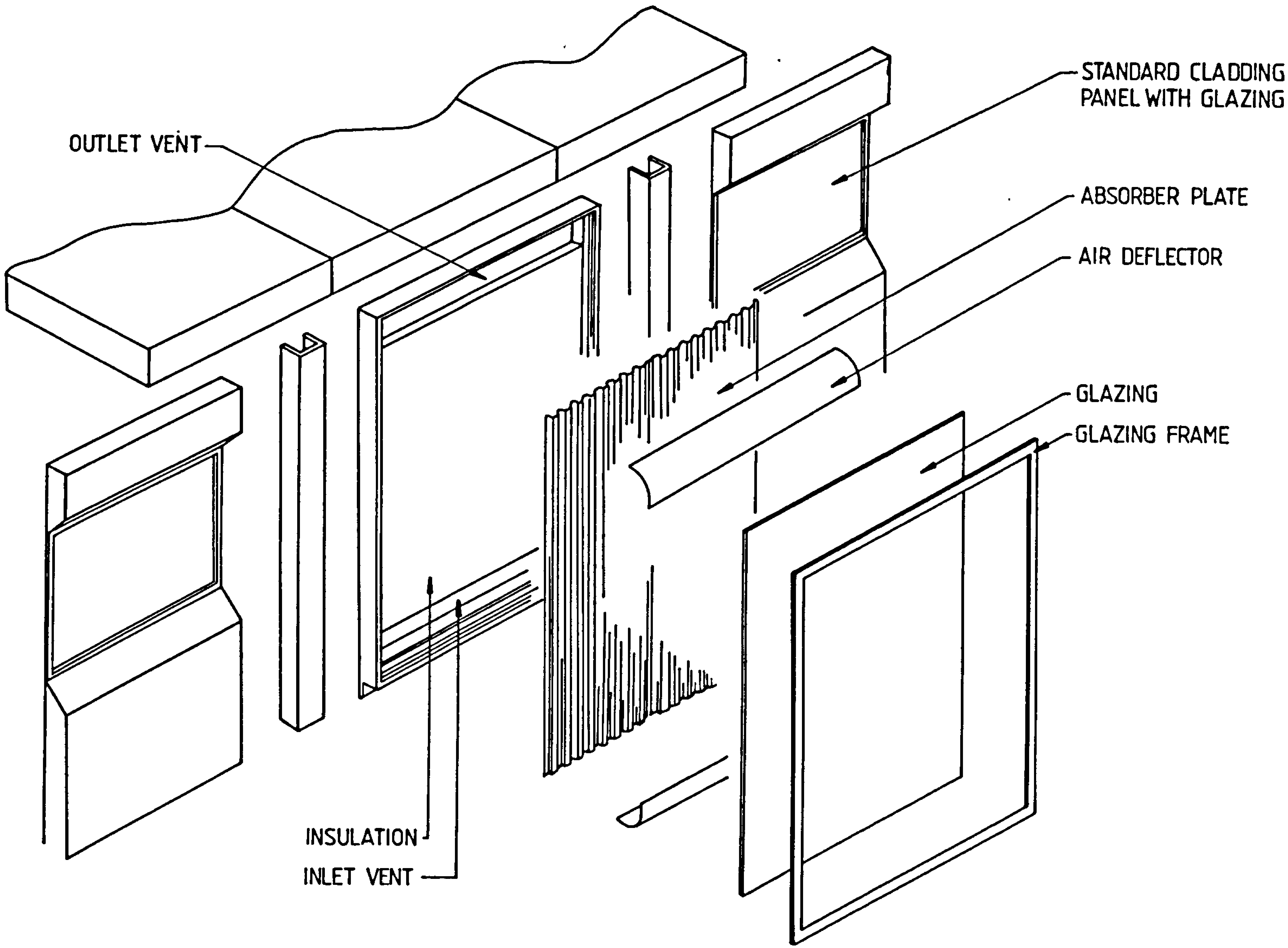


Fig.21 A standard TAP as part of a modular cladding system

uneconomic if a cladding collector projects beyond the building line. Collectors installed within the building line would not be subject to the costs associated with weatherproofing whilst heat losses from the collector would be less significant, those related to wind exposure particularly. A further benefit is that conversion from the heating to the ventilating mode would be more straightforward as the moving components would be within the building and, therefore, more accessible. However, a collector which reduces valuable office or teaching space may prove to be unacceptable in many instances.

By replacing the failing lightweight curtain walling of a building with cladding panels which incorporate thermosyphoning air panels, a passive solar energy air heating system could be created having the same thermal characteristics as the building itself. Neither the building, nor the collectors, have any significant thermal storage capacity and both are subject to rapid heat gains. Lightweight structures such as schools require heating prior to occupation in order to raise the internal temperature to an acceptable level. This level is maintained until the building is vacated at night. As has been stated, this type of building responds quickly to any heat input. Thermosyphoning air heaters are capable of reducing the amount of fuel required to heat these structures because they are equally responsive. In spring and autumn, heat collected during the frequent periods of sunshine at low solar altitudes would quickly be passed to the interior and there should be days when conventional heating is only necessary in the early morning and late afternoon.

Once the daytime use of these buildings has finished, all heating is usually turned off and they cool down virtually completely. According to Diamant (1971) the quantity of heat lost in this way is not large because of the low storage capacity of the walls. For the few hours per day that heating is required, it was argued that it is unlikely that radical temperature changes will take place, therefore the disadvantage of poor thermal storage capacity is not overwhelming Balcomb 1980. However, Hardy & O'Sullivan (1967) stated that, in buildings designed for daylight, the demand for heating and cooling is determined mainly by changes in the external environment, which can vary considerably.

Balcomb et al (1980) stated that for heat savings of less than 30%, solar heating contributes by offsetting daytime heating loads and that little solar gain is available for storage. Thermal mass would reduce temperature swings in a building but would not aid in offsetting demand for heating.

A less disruptive way to improve the performance or halt the deterioration of a building envelope is to overclad it: a lightweight rain-screen can be placed around its walls to protect it from the surrounding environment. Cladding is attached to the existing skin of the building in such a manner that further weather penetration is excluded. Insulation can be incorporated as an integral part of the overcladding so that the overall performance of the refurbished building can equal or exceed that of a new one. Solar collectors could be accommodated in the space between the overcladding and the original building fabric. The collectors would be located on the exterior of the building yet would not require extensive and costly, weatherproofing. A building before and after this type of refurbishment is shown Fig. (22).

An alternative strategy for the installation of TAPs is to position appropriately designed panels behind existing glazing. Insulation levels are increased and glazing area reduced by this method, whilst these collectors are less expensive to construct and install than collector cladding panels and do not have to form part of a refurbishment programme. However, they can only be installed if surplus glazing area is available.

Abrams (1979) developed a combined convective heat-gain and glazing insulation unit which integrated a simple thermosyphon air collector and insulation. The unit was designed to operate as a removable screen so that the daytime privacy of the occupants of direct gain buildings could be achieved without sacrificing passive solar heating: heating which was no longer provided by direct solar gain was replaced with the natural convection air-heating generated by the unit. Fig. (23).

The units consisted of panels of 25.4 mm thick, foil-faced, isocyanurate foam with a 63.5 mm air channel between the panel and the existing glazing. They were held to the glazing mullions by

Fig.22 An example of a building before and after overcladding refurbishment (Andrews 1985)

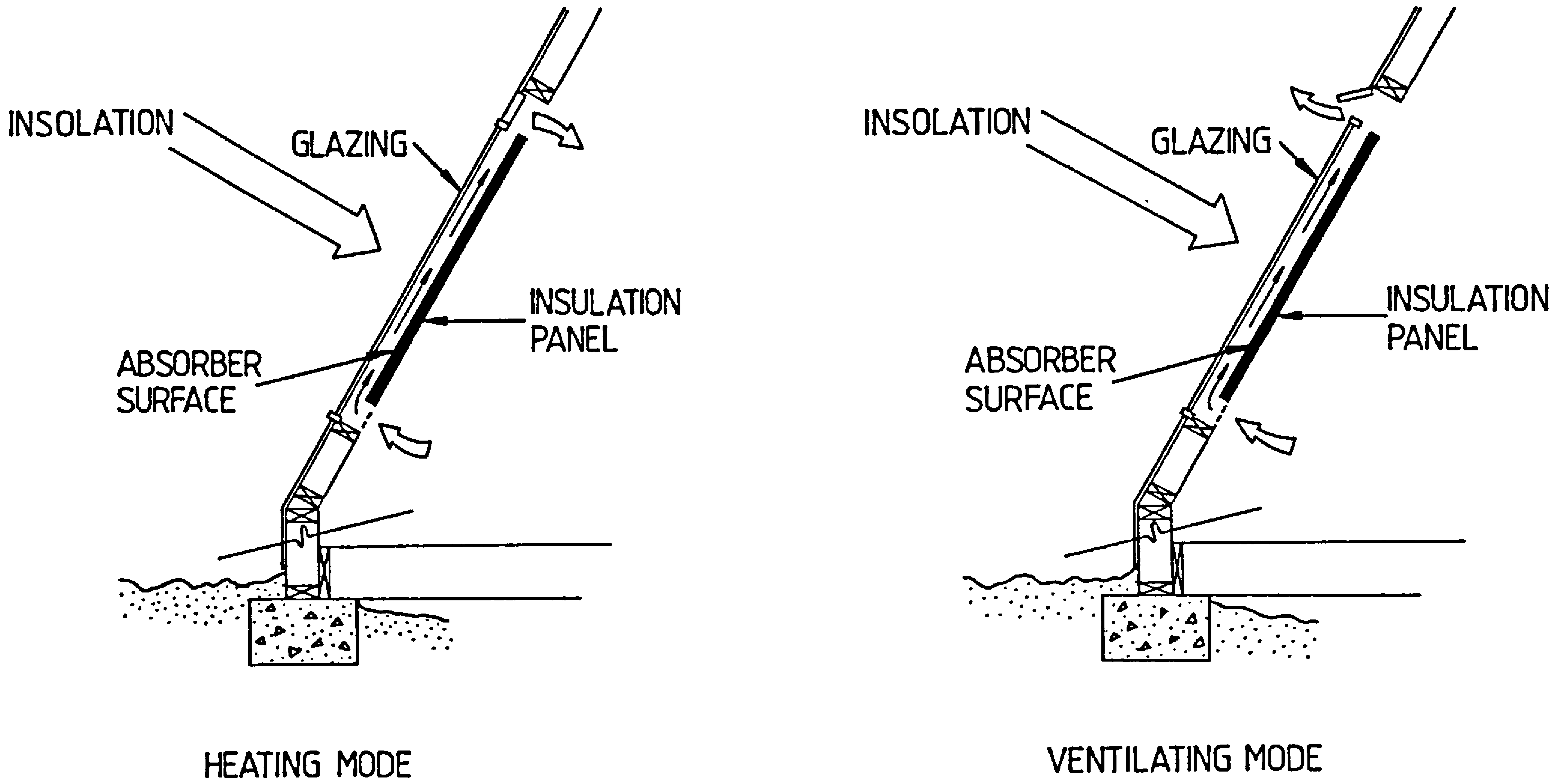


Fig.23 Combined convective heating and ventilating unit developed by Abrams (1979)

compressed foam gaskets. Flat black paint was applied to the foil on the front surface of the insulation, and in one unit a backdraught damper made from thin polythene sheet was mounted at the outlet above the insulation panel. Convection flow was observed at the lower vent on cold nights, indicating that reverse circulation was taking place. Suggestions were put forward to overcome this problem including the movement of the panel downward to block the lower vent at night, or forward to eliminate the air space.

Tests were also carried out to evaluate the performance of these units as solar thermal-chimneys, drawing air from the interior and discharging to ambient. Flow rates of 0.016-0.025 m^3/second were recorded for each 1.58 m^2 collector, but it was noted that the impact of even a slight breeze had an overwhelming effect on the upward flow in the collectors causing it to cease or even reverse. With discharge air temperatures in excess of 48.8°C this was described as being "disconcerting" on warm days. It was noted that the discharge vent configuration used in these experiments aggravated this problem by acting as a scoop for southerly winds, and that attention should be paid to the design of exit vents for this mode of operation.

The construction and performance of similar combined convective heat gain and glazing insulation units which were installed in the Muskegan County Building in Michigan, USA was outlined by Gerrow (1982). The building, a curtain-walled structure built in 1958, was uninsulated and orientated so as to benefit poorly from passive solar heating. Uncontrolled solar gains in the summer months added significantly to the air conditioning load, and heat losses during the winter months were considerable. It was decided to insulate the side wall cavities of the building and reduce the glazing area by approximately 60%. The architect in charge of the refurbishment considered that the potential of the glazing to provide some form of solar heating input should not be lost and, after due consideration of the options available, insulating panels were fitted behind the glazing and vents creased, to allow the movement of heated air to the interior and to the ambient environment.

130 panel collectors, each with a surface area of 2.83 m^2, were fitted to the building and represented 3.5% of the building floor area, whilst direct gain represented 1.8%. The building ventilation system was modified to allow heat collected by the panels on the south-west

side to be distributed throughout the building.

Each panel consisted of drywall bonded to polyisocynanurate, then particle board bonded to a further layer of polyisocynanurate with one foil side anodised black. An air channel of 73 mm was created between the blackened foil and the glass and an absorber of aluminium screen was fabricated and attached to the blackened panel. The aluminium screen had been corrugated to form vertical triangular grooves 63.5 mm deep, a configuration intended to promote laminar flow.

Holes were drilled in the aluminium sills at the top and bottom of the glazing to promote air movement through the air channel when the collectors were not in use, so as to prevent heat stress fracture of the glass. Grilles were used for the interior vents of the collectors and these covered 90% of the collector outlet. Each inlet vent was fitted with a backdraught damper made from nylon sail cloth. It was concluded that these impaired the air flow through the collectors to an unacceptable degree. Problems with excessive turbulence were also encountered at the inlet and outlet of the units. On clear days temperature differentials of 19 to 22°C across the collector produced air velocities of 0.5 m/sec with temperature differentials of 4.4 to 7.2°C between the absorber and the air channel at the same collector locations. Calculated savings based on a heat loss analysis of the building envelope, using 30 year average seasonal heating degree days, showed an improvement of 40% with the increased insulation, whilst measured savings averaged 60%.

Savings during the cooling season were predicted at between 30% and 35% and a total project payback period of under six years was anticipated. The life of the project was estimated to be at least 20 years, and with an 18% per year escalation rate it was suggested that the heating savings alone would provide annual average rate of return of over 50% on the total investment.

Garrett et al (1983) reported on the design and retrofitting of this type of thermosyphonic air heating solar panel to thirteen educational buildings in Pennsylvania, USA. The south, south-east and south-west facades were fitted with a number of insulated panels which were also designed to operate as passive solar air heaters. The advantages of installing TAPs behind existing glazing are outlined in Table.7.

o An expensive collector component, namely the glass, is already installed.
o The collector does not require weatherproofing as it is within the building envelope.
o The existing window sills and mullions provide an ideal framework in which to mount the collectors.
o The facade of the building is not significantly altered.

Table.7 Advantages of Thermosyphoning Glazing Panels

Design criteria were also established for the collectors and are outlined in Table (8). The dimensions of the collectors were derived by using the TAPFLOW computer program. The effectiveness of the collectors was evaluated by Solar Load Ratio method, although performance figures were not presented.

The original design proposed for this project Fig.(24) consisted of a heavy stud panel with a corrugated black metal absorber plate. A temperature actuated mechanical damper was to be fitted at the outlet, and a one-way hinged diaphragm at the inlet to prevent reverse circulation. The collector assembly was to be attached to the window opening so as to allow removal for cleaning of the glazing and absorber surfaces every 2 to 3 years.

This relatively advanced collector design was rejected because of the high cost of materials and labour required in its manufacture and installation. The cost of the sophisticated damper, metal collector surface and stud construction were considered to be prohibitive, as was the cost of the installation of the damper and the finishing work. A less costly-wood-stud construction was also rejected because of the high predicted operating temperatures.

A simplified collector was developed, Fig. (24), which made use of light-weight fibre glass panels manufactured specifically for retrofit window insulation. This panelling formed the basis for all subsequent TAPs installed during this project and it produced a relatively low-cost system which used non-combustible materials capable of withstanding high temperatures and requiring minimal time or experience to install. The top damper was omitted in this second design and a simple backdraught damper of 1/2 mm polythene film was taped to the base of the collector to prevent reverse circulation.

Apart from initial concern about the reduction in glazing area, the main problem encountered by teachers and staff working in buildings retrofitted with these collectors was overheating in the spring and autumn. No provision had been made for controlling heat output from the collectors when the air temperature in the rooms had reached the desired level. The design was modified to include a shading device installed in the flow channel to control absorption of solar radiation

o Low cost. The objective was to develop a system which could be quickly installed with the minimum of finishing work.
o Low maintenance. The limited resources available for the maintenance of these collectors necessitated the retro-fitting of a collector fabricated from durable materias and which required only the minimum of maintenance. A further requirement was that the panels should be demountable for periodic cleaning of the interior surface of the glass, as a front pass absorber configuration was to be used.
o Temperature. It was estimated that temperatures within the collectors could reach 148.8°C and that some form of shading might be required.
o Fire resistance. Materials chosen for the panels had to be constructed from materials which complied with fire regu-lations for educational buildings.
o Simple operation. Manual dampers were required to be simple to operate for the teachers or caretakers responsible for their use.
o It was recognised during the design and construction of the panels that the success of the project was strongly depen-dent upon the acceptance and involvement of the clients, namely teachers, students, and administrators.

Table. 8 Design Criteria defined by Gerrow (1982)

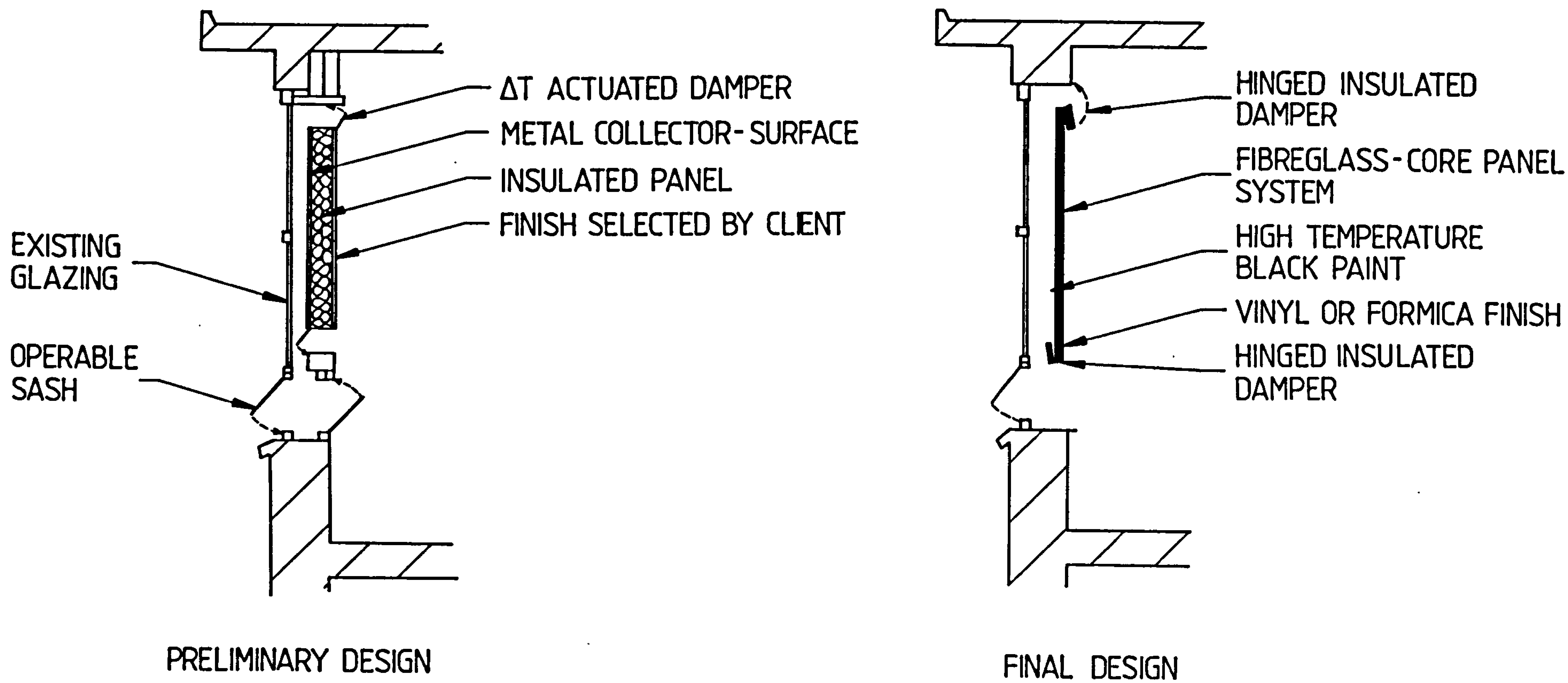


Fig.24 Preliminary and final design of glazing collector panels (Garrett et al 1983)

by the collector surface and eliminate the overheating problem. This device was a roller blind which, when fully extended, covered the absorber plate and which could be operated from within the room. Space limitations at the top of the collectors prevented the blind from being used as a top damper in addition to screening the collector surface. In rooms where teachers were not directly responsible for the operation of the blinds, such as dining halls, the necessary adjustment was performed seasonally by custodial staff.

A further modification to the next group of collectors to be installed was made by dispensing with the shading device and fitting the top and bottom openings with insulated dampers made from the fibre glass panel material. Both dampers were hinged and manually operated in an attempt to give the occupants greater control of the heat output.

It was intended that the top damper would be left open for the duration of the heating season and the bottom damper either opened for air circulation or closed for night time insulation. Summer operation would entail closing the top damper and leaving the bottom damper open to prevent overheating of the collector components. The performance of this final design had not been evaluated, but it was recognised that the collector efficiency depended on the willingness of the users to operate these dampers.

One conclusion drawn from this work was that to maximise the potential of these collectors as passive solar retrofits it would be desirable to develop a thermostat-activated damper system which would respond to the temperature in the room, in the flow channel, and outside. A further recommendation was that existing automatic temperature controls should be suitably modified when TAPs are retrofitted to buildings in this manner to accommodate the effects of increased thermal gain, reduced infiltration rates and added window insulation. No mention was made of the efficacy of the backdraught dampers in preventing reverse thermocirculation, or any influence they may have had on air movement through the collectors.

THE METAL-BUILDING THERMOSYPHONING AIR PANEL

Many industrial buildings, such as warehouse and factory units, are constructed of corrugated sheet metal panels attached to structural frames, Fig. (25) By partially, or fully glazing the southerly walls of such buildings, and using the metal panels as an absorber plate, a passive air-heating system can be created.

If the panels are uninsulated, the inner surface of the absorber radiates heat directly to the interior of the building during periods of solar gain. If insulation is installed behind the radiant panels, a Front-Pass or Back-Pass thermosyphoning air panel can be constructed.

The construction and performance of such a thermosyphoning "solar wall" on the south side of an industrial building was described by Stetzel (1983). The metal-clad building was located in the Ohio River Valley, an area described by Stetzel as being one of the poorest solar climates in the USA; both cold, and exceptionally cloudy. It was built for the garaging and maintenance of two "ready-mix" concrete lorries. The requirement was for a simple, inexpensive 100% solar heated structure.

The Front-Pass solar wall measured 12.2 x 4.5 m and was designed to heat and ventilate the building and provide a supply of preheated ventilation air; the three modes originally tested by Morse (1881). The south-facing wall of the building formed the absorber plate, and a single layer of "Kalwall" glazing was fitted at a distance of 0.187 m Fig.(26). The design recommendations of Morris (1978a) were used in the construction of the collector, apart from the flap-valve back-draught damper which was based on the recommendations of Christopher (1979).

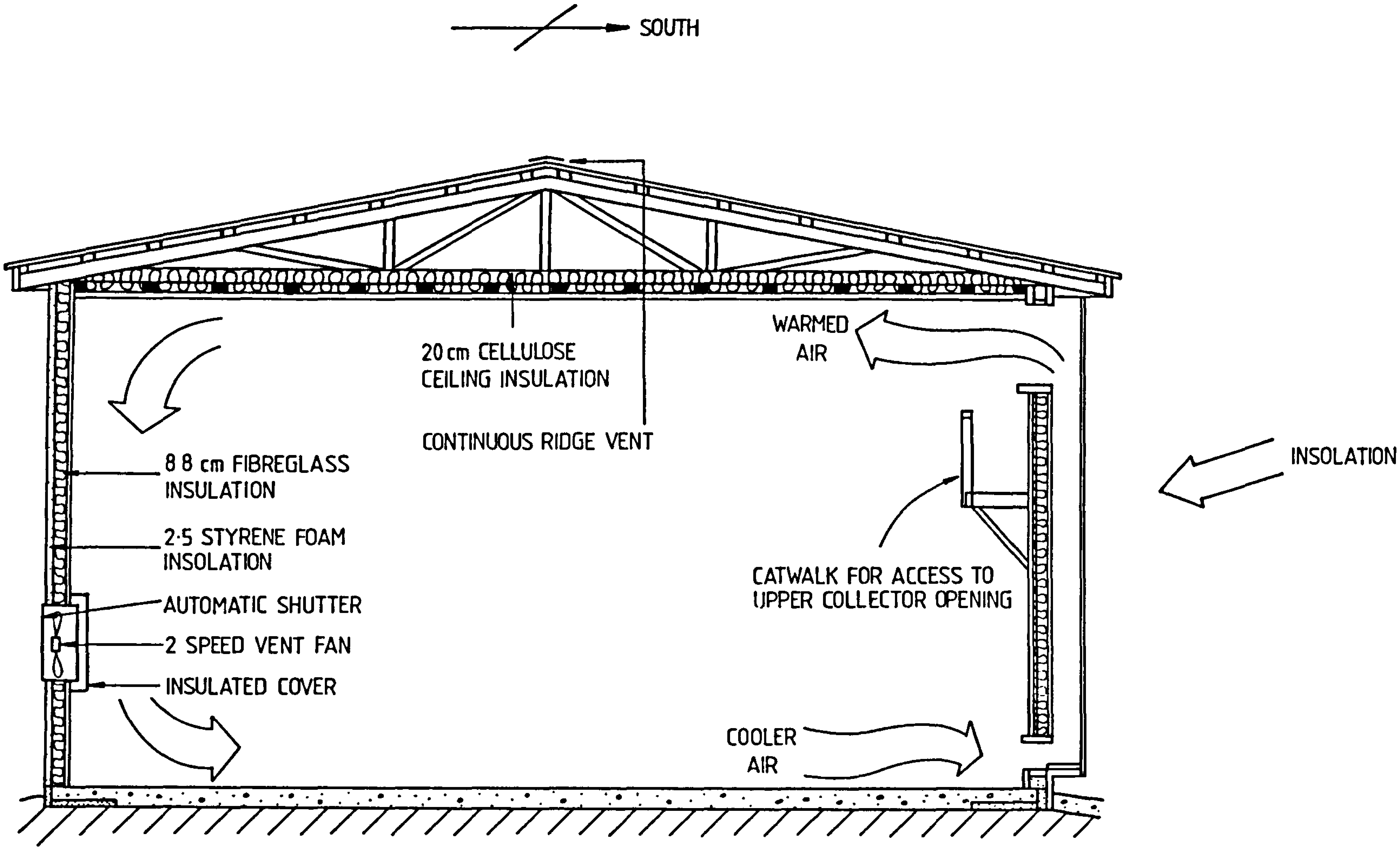
The solar wall had to be dismantled at the end of summer each year to remove the dust layer which built up on the inside of the glazing and the metal surface. Condensation was reported to form on the metal and, to a greater extent on the inner surface of the glazing.
Nevertheless, this passive solar heating system was described as being "an unqualified success"; conventional heating was only required when

Fig.25 A typical (in this case south-facing) metal-clad building wall, interior and exterior views (Phillips 1986)

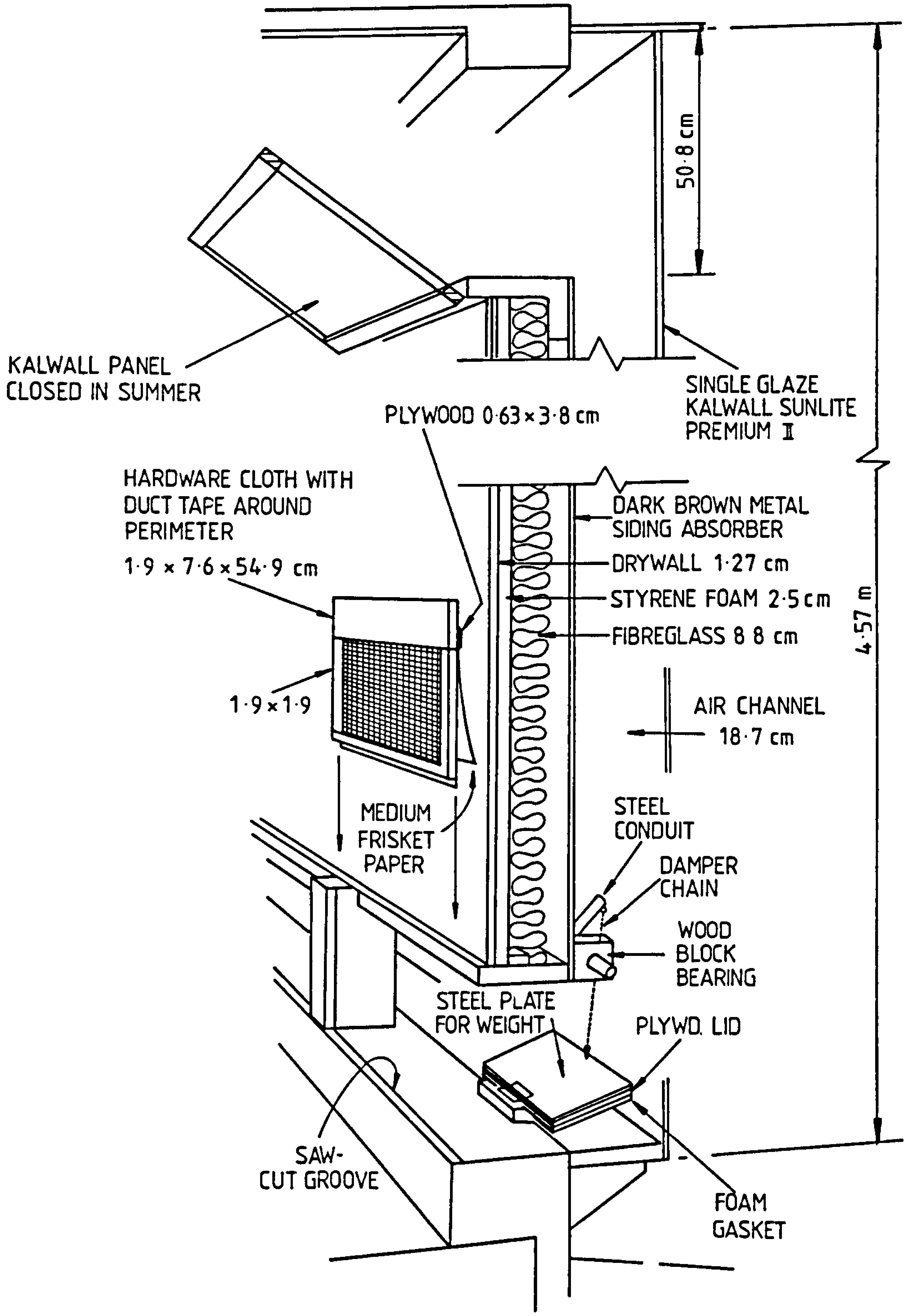
Fig.26 Retrofit "Passive Solar Wall" for a metal-clad building (Stetzel 1983)

a) general schematic

b) detailed cross-section



a) general schematic



b) detailed cross-section

equipment was being painted in the building during the winter. It was reported that the low storage capacity of the building enabled more of the available solar heat to be stored in the lorries and their loads; a significant advantage for this particular business. On many mornings the solar-warmed sand, gravel, cement and water could be used when outdoor stockpiles were too cold. Customers appreciated deliveries under such conditions and also appreciated not being charged for the fossil fuel it would normally have taken to heat up these materials.

One novel feature of this solar wall was that the hinged panel which controlled the movement of warm air into the building was made of the same Kalwall material that was used to glaze the collector. This enabled natural light to contribute to the illumination of the southern end of the building during the summer when the panel was closed. The upper vent was oversized; heat output was sacrificed for the sake of natural lighting.

Two simple passive solar retrofits were evaluated by Biehl, et al (1983) which were suitable for the large number of buildings in the American public and commercial sector which are constructed of corrugated metal. The two retrofits were radiant panels, and backflow thermosyphon panels. It was argued that the advantage of the thermosyphon system over the simple radiant panel is that, provided reverse flow is prevented by backdraught or manually operated dampers, night time losses are smaller because of the presence of the insulated panel.

Data obtained from operating both systems in adjacent test cells was used to provide a basis for validating computer models of the systems under investigation.

Two methods of estimating the flow velocity through the channel were investigated. The first method used the results of Akbari & Borgers (1978b), Borgers & Akbari (1979b), the second method was an application of Bernouilli's equation and followed the method of Varcolik & Versteegen (1980).

Although the Bernouilli method was less accurate for analysing flow within the channel it was found to give more accurate flow rates all because it accounts for pressure drops through the vents which the Akbari & Borgers's method does not.

On the basis of experimental test-cell data it was concluded that radiant panel and thermosyphoning systems were viable retrofits for metal buildings.

In a continuation of the work described above Schnurr & Wray (1984) used numerical analysis to predict the performance of a thermosyphon passive solar air heater, in a test cell at the Los Alamos National Laboratory, New Mexico, USA. Solution of Bernouilli's equation coupled with a thermal analysis using standard empirical convection correlations was employed. Comparisons of experimentally measured collector surface temperatures and auxiliary heating energy with values predicted by the numerical simulations showed good agreement when backdraught dampers were not fitted. The results of tests with a backdraught damper, consisting of a Tedlar flap and wire mesh covering the upper vent, were compared with the results of a numerical simulation. This indicated that the damper was approximately 35% effective in preventing heat loss at night. The numerical simulation assumed an ideal backdraught damper i.e. no reverse circulation occurred and the thermal resistance of the damper was equal to that of the insulation in the collector. These conditions were not achieved by the system tested, and it was reported that the measured night time collector surface temperatures were closer to the interior air temperature than they would have been if the thermal resistance of the damper had been equal to that of the insulated panel used.

The effectiveness of the damper was defined as the saving of auxiliary heat attributable to the damper divided by the reduction in auxiliary heat that would be produced by an ideal damper. The actual savings were estimated by comparing the experimental value of the auxiliary heat to the calculated value for the case where reverse-circulation did not occur. The ideal reduction of the auxiliary heat was defined as the difference between the numerical values for the cases with and without reverse flow. The backdraught damper effectiveness was quoted as 0.28, 0.33 and 0.35 for three days of the simulation.

Tests were carried out to determine the sensitivity of auxiliary heat requirements to thermosyphon flow rate. Computed velocities were decreased by a factor of two and it was found that auxiliary heat consumption changed by only 1%, suggesting that the solar energy transferred to the interior was insensitive to flow rate. It was noted that this conclusion was consistent with the results of TICHY (1983) who stated that increasing vent resistance and decreasing average velocity, acutally caused increased convection.

Auxiliary heat was supplied to the test cells when the temperature dropped below 24°C. It was recognised that the test cell used did not simulate a practical dwelling as the collector surface was too large for the size of the building and there was insufficient mass to store the incoming energy and maintain an inside air temperature within the comfort zone. However, the advantage of this system is that the thermosyphon heater's output is dominant and is not overwhelmed by the thermal mass, infiltration or other effects.

An analysis of the construction, monitoring and performance of a low-cost thermosyphoning air panel retrofitted to a metal building used for cultural and educational purposes was presented by McCarney (1983). In the United States, metal-skinned buildings form a significant proportion of the current building stock and it was stated that in climates with significant heating loads metal buildings can easily be fitted with vertical wall passive solar air heaters to offset daytime heating loads. It was noted that the minimal operation and maintenance costs for TAPs have proven to be a prime asset when compared to active solar systems and that daytime heating loads, usually 25% to 40% of the total heating load could be met with systems designed for daytime use only.

The collector was dimensioned according to the recommendations of Morris (1978a). Backdraught dampers were fitted at both inlet and outlet, and smoke was used to identify any leakage on two cold winter nights. With both backdraught dampers in place no smoke was observed moving in or out of the collector. When the outlet backdraught damper was removed a small amount of smoke was observed to occasionally be drawn into the collector outlet when room air pressure was essentially constant. No details of the construction of these dampers were given.

The TAP under test measured 0.889 m by 1.955 m. Polythene was used as glazing, but was not recommended. The simple payback period for these collectors was estimated at being between two and ten years.

Chapter 4

DESIGN AND TESTING

4.1 ABSORBER PLATE DESIGN

There has been a considerable amount of research devoted to find optimum dimensions and configurations for active solar energy air heating collectors and attempts have been made to use this work in the design of passive systems.

As Wiles (1983) noted, in recent years the thermal efficiency of active air heating collectors has been improved by increasing the effectiveness of the heat transfer between the air and the absorber plate. This has been achieved by extending the surface area of absorber plates by attaching fins, by using a corrugated absorber, or by using a porous absorber or mesh. However, it was evident that the development of mathematical models of air-heating solar-energy collector thermal performance has received more attention then the acquisition of experimental data required to validate these models to evaluate the performance of different designs. A further problem encountered is that work is sometimes carried out in summer months or in conditions of high insulation or high ambient temperature. The results produced are valuable but have little direct bearing on autumn on spring conditions, when heating is required.

Few of the developers of improved absorber plate designs have considered the ensuent manufacturing costs. The addition of fins to a flat plate, for example, is an expensive process demanding a good thermal bond between the fins and the plate. Cole-Appel et al (1978) compared the performance of fourteen different active solar-energy air-heater absorber plate designs and having determined the potential improvements in efficiency, and production costs, concluded that the increased performance achieved by the addition of fins was cost effective.

Flow visualisation tests should give some indication of the nature of the air flow in flat-plate collectors but apart from Summers (1981), Okeke (1979) and Cook and Morris (1978), they do not appear to have been used very widely. It would be advantageous to have some representation of the distribution of the air flow, as in the majority of active and passive collectors it is passing through an assymetrically heated duct of narrow cross-section.

Wiles (1983), reporting on work on active collectors, stated that the degree of asymmetry of heating was found to have a significant effect on convective heat transfer rates, with heat transfer increasing as the degree of asymmetry decreased.

Work by Sorour & Mottaleb (1984) on the design of corrugated absorber plates for active collectors showed that of the designs tested, circular corrugations were the most effective, and triangular the least Fig. (27).

One approach to collector design is to use an absorber of parallel ducts, a configuration which has been used successfully in active air heating systems, and which is essentially a Back-Pass configuration. Peck & Proctor (1983) constructed a low-cost solar-energy air-heater using conventional construction techniques in which the absorber panel consisted of ducts of roughly rectangular section formed by riveting sheets of aluminmum roof cladding face-to-face to create five air-tight channels Fig. (28). Although this was an active system, two features of its performance were noteworthy; the high efficiency achieved by the collector and the even flow distribution through the panel.

The type of parallel duct absorber discussed above does not appear to have been used in passive air heating collectors. The continued emphasis on the use of flat-plate absorbers is understandable in view of the amount of heat transfer data available on air-flow over vertical flat plates, and their low cost. Wile's comments on symmetrical heating and its effect on convective heat transfer rates indicates that the performance of parallel-duct absorbers in thermo-syphoning air heaters might be profitably investigated. The work of Sorour and Mottaleb (1984) suggests that a parallel-duct asborber of similar construction to that reported by Peck and Proctor (1983), but featuring circular corrugations, might prove to be an efficient design.

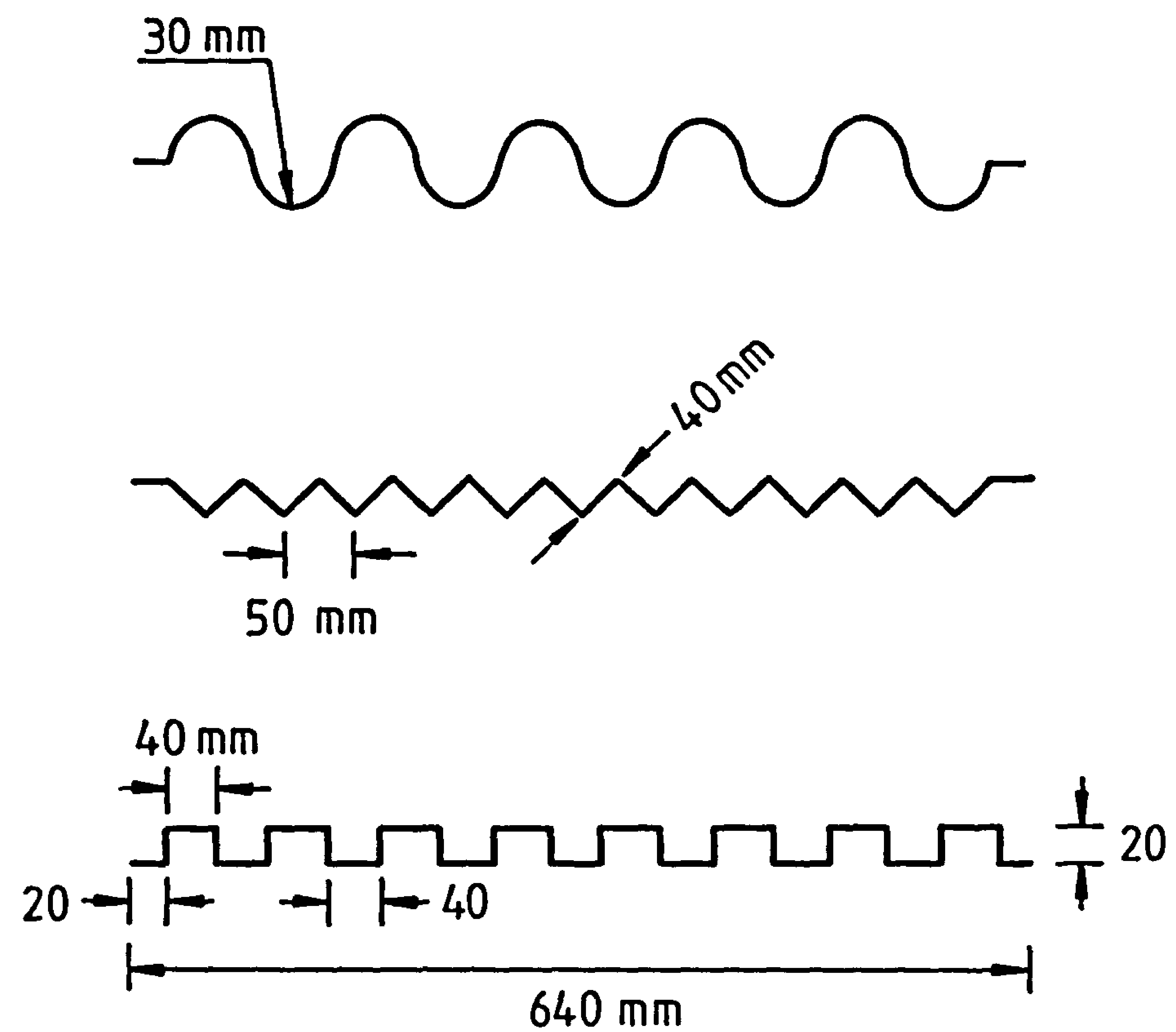


Fig.27 Dimensions and types of absorber-plate corrugations tested by Sorour and Mottaleb (1984)

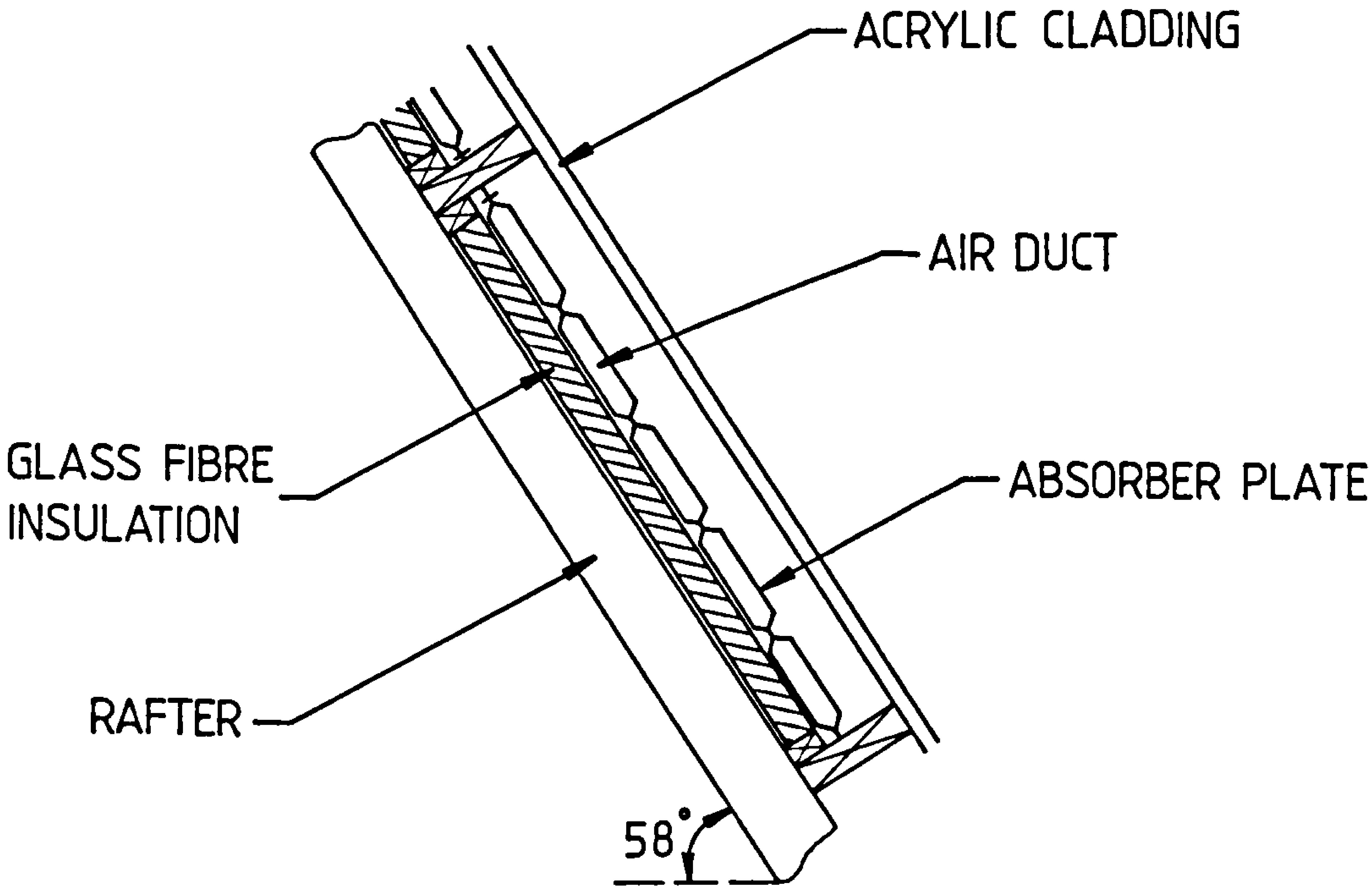


Fig.28 Cross-section of active air-heating solar-collector designed by Peck and Proctor (1983)

4.2 DESIGN OF MECHANISM TO FACILITATE THE TRANSITION FROM HEATING TO VENTILATING

A reliable and economic mechanism is required which will allow a collector to operate both as a heater and as a means of providing ventilation, but which does not impede air-flow, particularly in the heating mode. The majority of collector designs accomplish this conversion by closing off the outlet to the room and opening a flap at the top of the glazing: two separate operations. A design option which has not been investigated is to hinge an air deflector and use it to direct the air-flow in either of the desired directions Fig. (29). The conversion can be completed in one operation with such a device; the collector is either in the heating or ventilating mode so there is no risk of the collector being left to stagnate inadvertently. Ideally, a collector which featured this mechanism would not have to be constructed to withstand stagnation temperatures as a precaution against this eventuality. However, it may be necessary to close the exterior grille or louvre to prevent cold ambient air entering the collector during the heating cycle. If the insulation and air-tightness of the components is not adequate this cold air could reduce significantly the collector efficiency. The louvre could be opened and closed by a direct linkage to the air deflector, or by a thermally actuated device. If, however, either mechanism failed and the louvre remained closed, the collector would stagnate instead of ventilate and high temperatures could ensue. Therefore, a collector which has an exterior vent which opens and closes may have to be constructed to withstand the effects of stagnation if only as a precaution.

The use of a hinged air deflector system with a permanently open exterior louvre or grille could create a situation where a collector is cooling when it should be heating. As soon as the heat output of the collector is no longer required the collector will be converted to the ventilating mode. If the collector is left in this mode overnight it will not be capable of preheating the building the following morning but will, instead, cool it. The combination of a hinged air deflector with a louvre which opens and shuts is a mechanism which is no less complicated than the conventional solution, but which allows the inclusion of a curved air deflector at the top of dual-mode

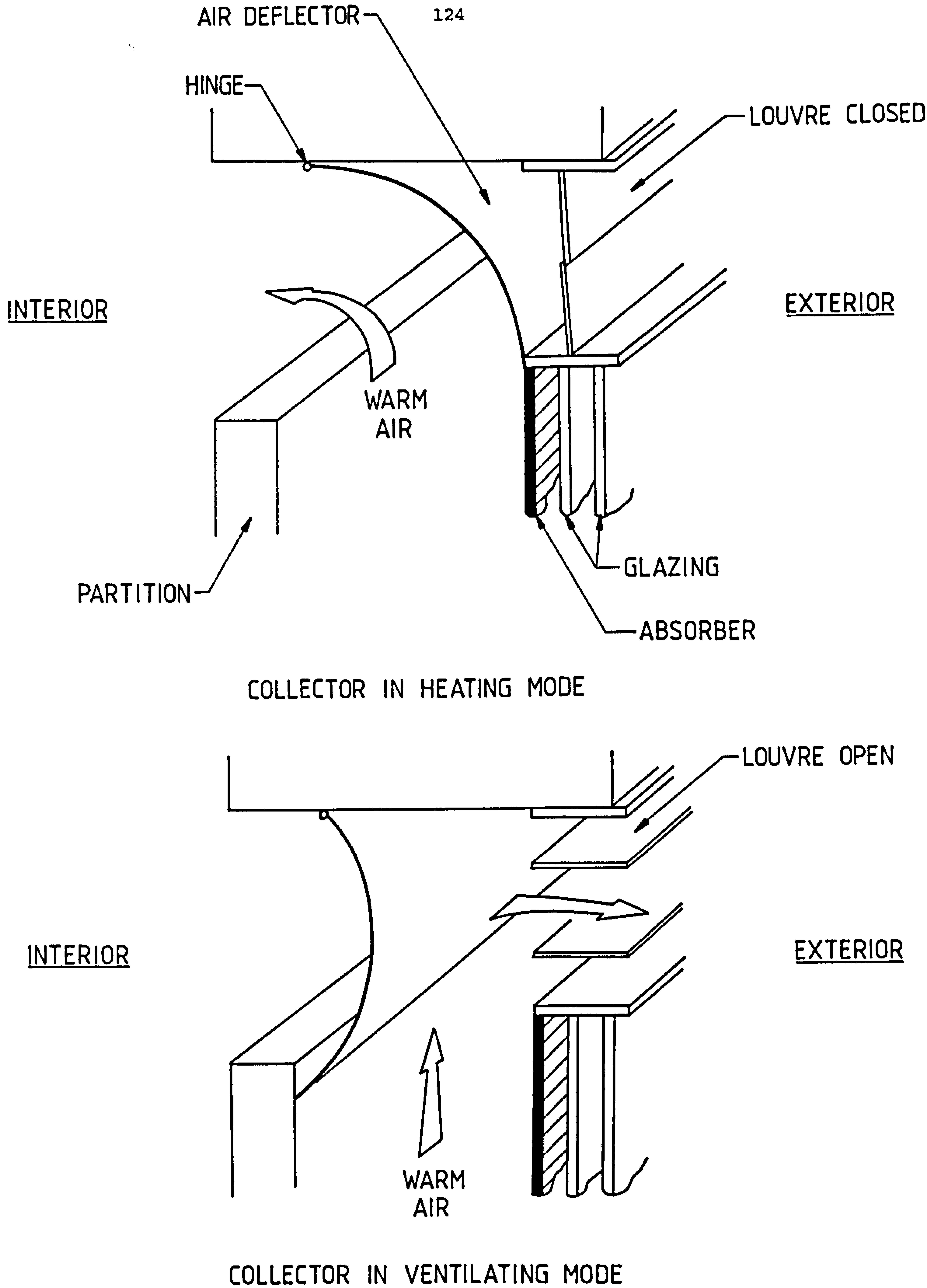


Fig.29 Modes of operation of hinged air-deflector

collectors. Such air deflectors improve the air flow in these collectors during the heating cycle by redirecting the flow path with the minimum of interference. This eliminates some of the friction losses which might otherwise occur at the exit of collectors and hence improves efficiency.

4.3 TESTING OF THERMOSYPHONIC AIR-HEATING SOLAR ENERGY COLLECTORS

An extensive review of the available literature on the testing of active solar energy collectors was presented by Hill (1985).

Five alternative methods can be identified by which the performance of natural circulation air heaters may be measured. These are summarised in Table 9.

These methods have disadvantages which reduce their effectiveness. The direct measurement of air velocity is often inaccurate because of the nature of the air movement through such collectors. Flow-rates are low and fluctuate due to the rapid response of natural circulation air heaters to any variation of insolation level.

Obtaining and integrating density/height profiles is straightforward, but values for friction factors at the entrance and exit of collectors must be derived before values of velocity and mass flow rate can be calculated. These friction factors vary with flow-rate and can only be determined experimentally. Any inaccuracies in the derivation of these friction factors will adversely affect estimates of collector performance. A comparison of energy consumption is valid if the equipment used to carry it out is of a high standard and is correctly located. For example, the position of the thermostat controlling the main heating can have a significant influence on the amount of energy consumed. Even assuming that the generation of long-term performance data obtained from one building to other buildings of different i) construction, ii) heating system, iii) patterns of occupancy iv) located in different climates were possible, considerable care and insight would be required.

The fourth method is based on a solution of Bernouilli's equation for non-ideal flow, and is described earlier in the text.

TEST METHOD	PROCEDURE	COMMENTS
1. HEAT GAIN	MEASURE THERMOSYPHONIC MASS FLOW RATE AND RISE IN TEMPERATURE OF AIR AS IT PASSES THROUGH THE THE COLLECTOR. COMPARE HEAT GAIN BY AIR WITH SOLAR ENERGY INCIDENT ON COLLECTOR.	VALUES OF INSTANTANEOUS EFFICIENCY MAY BE OBTAINED READILY WITH THIS METHOD. HOWEVER, AIR FLOW RATES ARE DIFFICULT TO MEASURE ACCURATELY.
2. TEMPERATURE/ HEIGHT	OBTAIN PROFILES WITHIN THE SYSTEM (i.e. THE COLLECTOR AND TEST CELL) OF VARIATION OF TEMPERATURE WITH HEIGHT. INTEGRATE PROFILES TO OBTAIN VALUES FOR BUOYANCY FORCES WHICH MOVE AIR THROUGH THE SYSTEM. COMBINE RESULTS WITH TEMPERATURE RISE OR AIR AS IT PASSES THROUGH THE COLLECTOR TO OBTAIN EFFICIENCY.	ACCURATE DATA ON THE FRICTIONAL FLOW RESISTANCE IN THE CIRCUIT IS REQUIRED BEFORE EFFICIENCY CAN BE ESTIMATED. THIS CAN ONLY BE ACQUIRED EXPERIMENTALLY.
3. ENERGY CONSUMPTION	MEASURE ENERGY CONSUMED IN HEATING A BUILDING WHICH HAS A PASSIVE SOLAR-ENERGY AIR-HEATER AND ONE WHICH DOES NOT.COMPARE ENERGY CONSUMPTIONS OF THE TWO BUILDINGS AND CORRELATE THE LONG-TERM PERFORMANCE.	RESULTS ARE SPECIFIC TO THE BUILDINGS IN WHICH THE TESTS ARE CARRIED OUT. CONDITIONS WITHIN THE BUILDINGS MUST BE CLOSELY CONTROLLED FOR RESULTS TO BE OF VALUE. ENERGY CONSUMPTION IS INFLUENCED BY MANY FACTORS. THIS METHOD IS NOT SUITABLE FOR SHORT-TERM TESTING.
4. BERNOUILLI	MEAN AIR VELOCITY AT A POINT WITHIN THE SYSTEM MAY BE ESTIMATED BY APPLYING BERNOUILLI'S EQUATION FOR NON-IDEAL FLOW. COMPARE HEAT GAIN BY AIR WITH INCIDENT SOLAR ENERGY.	ACCURATE DATA ON THE FRICTIONAL FLOW RESISTANCE AT A SPECIFIED SECTION IS REQUIRED BEFORE EFFICIENCY CAN BE ESTIMATED. THIS CAN ONLY BE ACQUIRED EXPERIMENTALLY.
5. GLAZING TEMPERATURE	MEASURE GLAZING TEMPERATURE AND RISE IN TEMPERATURE OF AIR PASSING THROUGH COLLECTOR. COMPARE HEAT GAIN BY AIR WITH SOLAR ENERGY INCIDENT ON COLLECTOR.	A SIMPLE METHOD OF OBTAINING INSTANTANEOUS EFFICIENCIES BUT WHICH HAS YET TO BE VALIDATED. MAJOR ASSUMPTIONS ARE NECESSARY AND HEAT TRANSFER COEFFICIENTS MUST BE ESTIMATED.

TABLE 9. SUMMARY OF METHODS FOR EVALUATING SOLAR-ENERGY AIR-HEATING COLLECTORS

The fifth method by which collector efficiency may be measured is that outlined by Kornher & Zaugg (1984): assess the efficiency of a collector on the basis of the temperature differential between the air entering and leaving the collector and on the glazing temperature. In view of the shortcomings of the four methods discussed previously this approach might prove to be a more reliable indicator of collector performance.

4.4 COLLECTOR TESTING PRIOR TO INSTALLATION

It may be advantageous to derive a value of efficiency for a given design determined on standardised test equipment, and then derive a figure which indicates the effect that the building and its occupants will have on this collector efficiency. It would appear however that the interaction between the buildings, the occupants and the collectors will be too complex and unique for comparisons to be of value. In these circumstances, results obtained on test apparatus are valuable in comparing the relative merits of collector designs, but do not form a sound basis for the prediction of collector performance.

Collectors tested in tandem on identical test cells give an indication of the relative performance of various configurations but, unless the test equipment and experimental procedure are of the highest standard the results obtained are of limited value: there are a considerable number of variables which influence the performance of a natural-circulation air-heating system under test, and it would appear that it is only by evaluating collectors under closely controlled conditions that an accurate assessment and comparison of designs can be made. For example, it is evident from the available literature that the ratio of collector air channel depth to height has yet to be satisfactorily defined. Whilst different configurations of collector may be compared side-by-side on outdoor test cells, the results obtained are specific to those test cells and local climate. It is only by exposing designs to predetermined insolation levels and carefully regulated operating conditions that a true basis for analysis can be established.

The solution would appear to be an indoor test facility of the type proposed by Geisheker and Whitaker (1981) and currently under development by the Commission of the European Communities. It must be emphasised that the results obtained from a test facility of this type will enable comparisons of various collector designs but will be inadequate for the prediction of the performance of a building heated with warm air from these collectors. Such consideration as the impact of a certain collector's output on thermal comfort, or the proper use of collector controls by occupants will be beyond the scope of such a test.

Natural-circulation air-heaters are installed to provide heat gain and prevent heat loss from a building. It is, therefore, insufficient to give figures for collector efficiencies which define performance on the basis of heat input alone. Some measure of a collector's effectiveness in preventing reverse-thermocirculation must be included in efficiency tests and form part of any overall assessment of a natural-circulation air-heating collectors performance. However, quantifying the extent to which a collector will prevent reverse thermocirculation will be difficult where manual dampers are fitted, as the test must assume that the dampers will be closed during non-gain periods.

The development of a standard test procedure for active (i.e. forced circulation) air heating collectors has been undertaken by several groups. The Commission of the European Communities collector testing group consists of 15 laboratories throughout Europe which have constructed test facilities. A Belgian forced-circulation air-heating solar-energy collector was selected in 1983 for round robin testing by the participants and, according to Gillett & Moon (1984), the accuracies in efficiency measurement varied greatly between laboratories. This was attributed to variations in collector leakage, poor accuracy of inlet and outlet air temperature measurement and poor accuracy of flow rate measurements. It was suggested that improvements in the accuracy of air flow rate and temperature measurements were required so that air leakage effects could be better defined and efficiencies determined more accurately.

Air leakage has been recognised as a problem in both active and passive collectors and is believed to have reduced the performance of several systems. ASHRAE Standard 93-77 (1978) is concerned with the thermal performance testing of air and liquid collectors. Although this standard does recognise the need to minimise leakage prior to testing, no satisfactory techniques exist for quantifying the effect air leakage has on collector performance.

At the Los Alamos Scientific Laboratory, New Mexico, USA, Morris (1980) designed and constructed a facility for investigating the performance of natural circulation solar air heaters. The unit allows variation of collector length, flow-channel depth, tilt, glazing, absorber, heat transfer to simulated storage, pressure drop through storage, air-flow resistance, and operating temperature. An air-to-liquid heat exchanger was employed in this closed-loop test apparatus - see Fig. (13) - to remove the heat generated by the collectors. Different energy demands could be simulated by altering the temperature of the feed water to the heat exchangers. A test facility based on this design but with an air-to-air heat exchanger was proposed by Sferides & Tsingas (1985).

A proposed test method for thermosyphon solar air heaters was presented by Geisheker & Whitaker (1981). The proposed method consisted of five stages each of which was a separate test. Apart from the free convection test, the four tests are those of ASHRAE Standard 93-77 (1978).

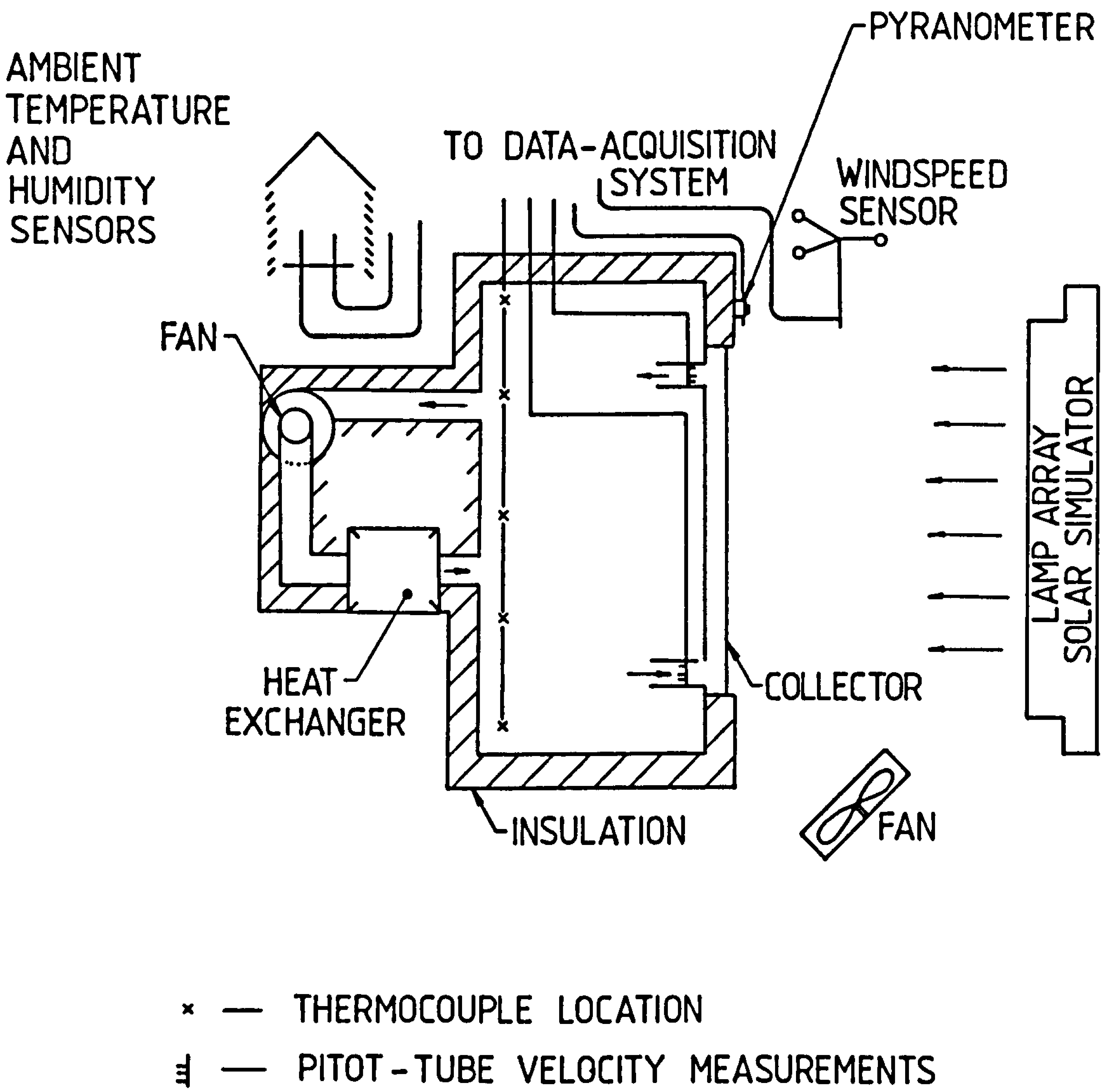
The test method described gave procedures for testing natural circulation heaters indoors under simulated solar irradiation. It was suggested that three major components would be required to conduct the proposed test method: a continuous source solar simulator in an environmentally controlled facility, an open-loop air-flow test facility, and thermosyphon solar air heater test apparatus. During "free convection test" a collector is allowed to reach a steady state condition and generate a flow rate governed by controlled environmental conditions Fig. (30a). The steady-state is achieved for four levels of insolation and for each of these, four levels of corresponding inlet parameters (temperature of inlet minus ambient temperature divided by insolation).

Instantanious efficiency tests are carried out using a forced air flow test facility Fig. (30b). Steady state conditions are reached in the

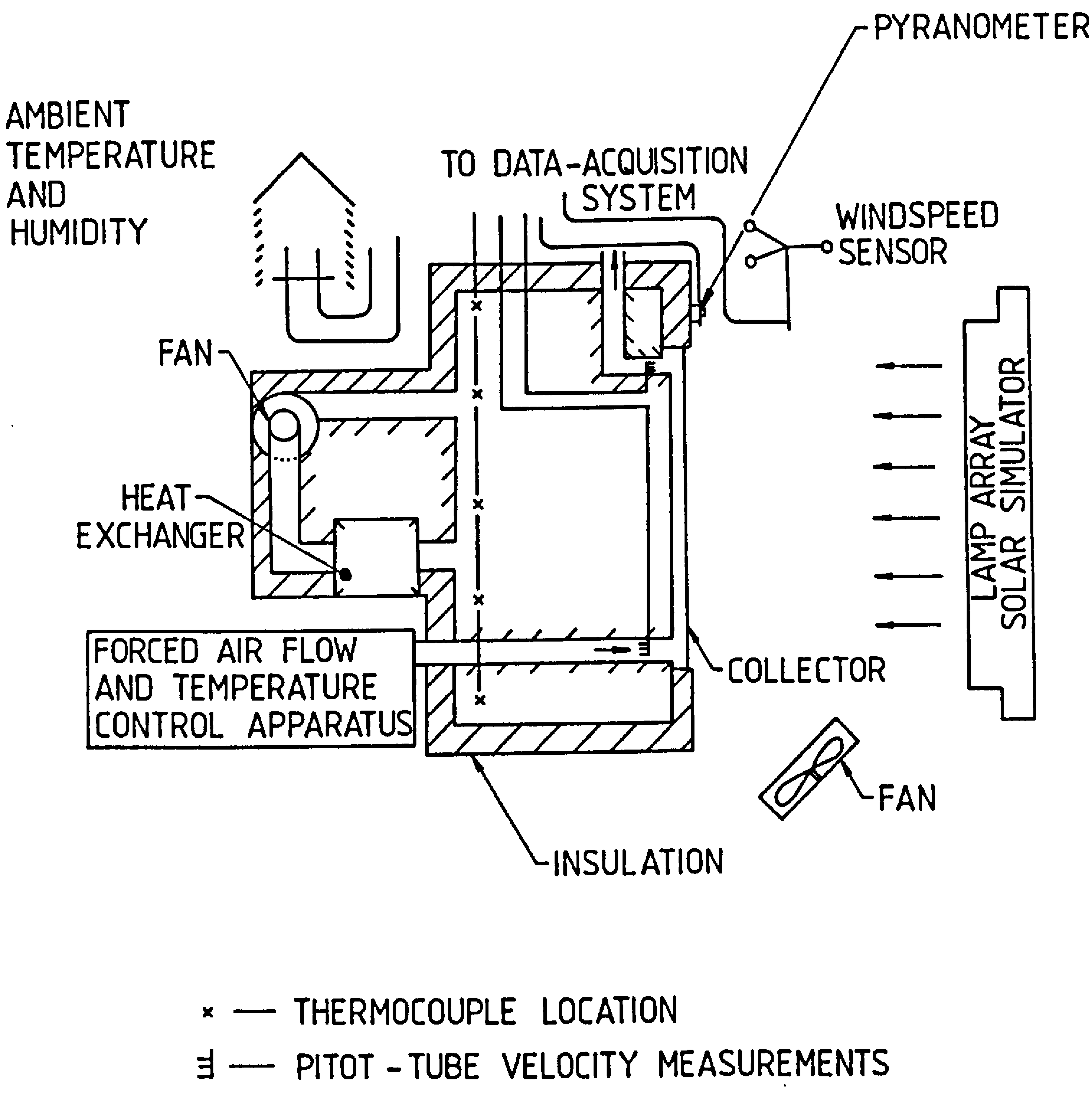
Fig.30 Test apparatus proposed by Geisheker and Whitaker (1981)

a) free convection

b) forced convection



a) free convection



b) forced convection

collector and these conditions are matched to those which existed during free convection in the previous test. This matching is achieved by creating an identical temperature rise across the collector. In each Forced-Flow test the flow rate is adjusted to obtain the same temperature rise as was obtained as was reached in the free convection test, the ΔT match ensuring identical flow rates. As the flow rate is known, collector efficiency may then be determined.

A number of issues regarding passive solar testing were addressed by Baker (1983) in a report which studied the requirement of standardised passive test-cells. The testing of solar components in realistic outdoor test cells was examined in detail. Two categories of physical testing were identified i) individual test rigs with simplified boundary conditions, and ii) outdoor test cells with standardised and realistic conditions. Physical processes which have a crucial influence on the performance of a passive system, e.g. corrective and radiative distribution of thermal energy, were identified, and it was argued that the test environment must reproduce these processes in a realistic way. Two important corrollories of this were that the scaling down of a test cell (which influenced convective heat flow) should be avoided, and that the reflectances and geometry of surfaces should be realistic. It was noted that the increased use of insulation in new buildings has resulted in similar magnitudes for casual gains from occupants, auxiliary heating and solar gains. It was argued that under these circumstances the performance of a component under test is sensitive to the assumed values of such casual gains. Also, as higher levels of insulation have been adopted, the ventilation heat loss has become a greater proportion of the total heat loss. The provision of realistic ventilation during testing was considered to be essential, as the absolute value of the energy saved by a solar component is dependent upon the load of the building. Occupants were recognised as having a major influence on the performance of a building, however occupancy is difficult to specify or simulate.

One conclusion of Baker's work was that steady-state conductive losses through the non-south walls of a (solar) room will always be small relative to both losses through the south wall, (in particular the "solar" component) and those due to ventilation. It was suggested that a simple strategy of using a highly insulated test-cell and allowing the load to be dominated by ventilation and losses through the 'component wall' be adopted. The small losses through the floor and coiling of a "real" room could be replaced by a slightly increased ventilation component in the test cell.

It would appear that such a strategy would not be appropriate for the testing of isolated gain solar-energy air-heating collectors for use in lightweight buildings. Furthermore, the views expressed on the scaling-down of test cells conflicts somewhat with those of Grimmer (1979a) and Grimmer et al (1979b).

Theoretical analyses developed for "test-box" thermal modelling of passive-solar buildings were presented by Grimmer (1979a). The results of these analyses indicated that reasonable quantitative agreement betweeen test-box performance and that of a real building could be obtained. The thermal modelling of (i) multi-room passive-solar buildings, (ii) passive solar buildings with massive walls, (iii) air infiltration effects, (iv) edge effect corrections, (v) microclimate shading effects, and (vi) thermocirculating passive solar designs were discussed. The importance of the "scalability" of various thermal parameters involved in the thermal modelling of such designs was reviewed and an example of such scaling for a thermo-syphoning passive-solar wall collector was presented. The scaling of air infiltration effects for test-box thermal modelling of a room with a south-facing Trombe wall was also presented.

Further thermal modelling of passive-solar buildings was reported by Grimmer et al (1979b). It was argued that small (i.e. $1m^3$) test-boxes could be constructed to perform as thermal models of passive-solar buildings by normalising the appropriate thermal design parameters to one critical parameter, such as south-facing glazing area, or building floor area. Under such conditions the thermal model would not resemble an architectural model of the building, but would be distorted in appearance so as to exhibit thermal behaviour similar to the real building. The accuracy and limitations of using such small test-boxes to predict passive-solar building performance was discussed.

4.5 COLLECTOR STAGNATION & OVERHEATING

Wilson & Sherwood (1981) considered the safety implications of constructing solar energy air heaters from wood in view of the high temperatures which can occur in such collectors when vents are closed during the day. Different collector configurations and operating conditions were simulated using computer analysis. The simulations were designed to give higher temperatures than would have occurred experimentally : 100% absorptivity, and high glazing transmissivity and air-tightness were assumed. With single glazing the effect of stagnation was found to be much less significant than with double glazing. Reported maximum temperatures reached in the different simulations varied from 87°C to 226°C. The effects of high temperatures on wood were investigated and it was concluded that high temperatures can be dangerous in vertical solar collector designs which incorporate unsuitable materials.

4.6 THE USE OF PHOTOVOLTAICS

The use of photovoltaic powered fans to enhance the performance of TAPs was suggested by Reif (1981). This is an attractive idea assuming that the installation of such devices is economic and that there is some form of thermostatic control to prevent the fans moving air when they are not required to do so. This method of increasing the flow-rate of air through a collector would be particularly beneficial in the summer months when venting of room air is required. Whether a unit constructed incorporating such a device could be considered to operate on the basis of natural-circulation is questionable. A possible adaptation of this system would be to use the fan as a one-way valve: reverse-thermocirculation would be prevented by flap valves built into the fan itself. The fan would be installed in a plenum to direct the flow and would have to be accessible for repair and maintenance as would the solar cells. The inlet or outlet of this device would not have to extend the entire width of the collector which may be advantageous. There may also be scope for the distribution of warm air from passive solar collectors using fans of this type driven photovoltaically.

4.7 SELECTIVE SURFACES

A more conventional approach by which collector performance may be enhanced is the use of selective surface materials. When applied to absorber plates they increase the absorbers capacity to absorb and retain solar energy. It is debatable whether the increased cost of using a selective surface is justified by the increased performance which they produce in these collectors. Further investigation of the merits of such materials in this context is required.

Chapter 5

EXPERIMENTAL PROCEDURE

5.1 PRELIMINARY TESTS SUMMARY

A "standard" vertical TAP and a U-tube collector of the same dimensions, apart from depth, were fitted to the south-facing walls of two "identical" test cells as shown in Fig. (31). The temperature distribution in the test cells, and collector temperatures were measured in order to determine the differences in performance of the two systems. Temperatures were measured with copper-constantan thermocouples and output signals recorded at hourly intervals using a FLUKE 2200B Datalogger. They were monitored at the following locations:

1) At the entrance and exit of each collector with two thermocouples positioned in each duct. The thermocouples measuring inlet and outlet air temperatures were positioned 30 cm apart along the horizontal centre-line of the inlet and outlet ducts.

2) At the top of each collector absorber, held in position with adhesive tape.

3) Down the vertical centre-line of each test cell at distances of 5, 60, 120 and 180 cm from the floor of each test cell.

Hourly values of total insolation on the vertical surface were obtained from an integrator which received continuous data from a Moll-Gorczynski pyranometer installed on a south-facing vertical surface close to the test cells. Air velocities were measured with a Prosser Scientific Instruments AVM 501 TC hot-bead anemometer.

The test cells were located at the Solar Energy Technology Centre Annexe at the Cranfield Institute of Technology, Bedfordshire, England, latitude 52°N, and at a height of 111 m, above mean sea level. The site was not overshadowed and was not subject to airborne industrial pollution. The relative positions of the test cells, weather and data acquisition equipment is shown in Fig. (32).

5.2 THE COLLECTORS

The U-Tube and standard TAP collectors were constructed from 20 mm thick plywood sheet and were insulated with a combination of expanded

Fig.31 U-Tube and standard TAP collectors fitted to test cells

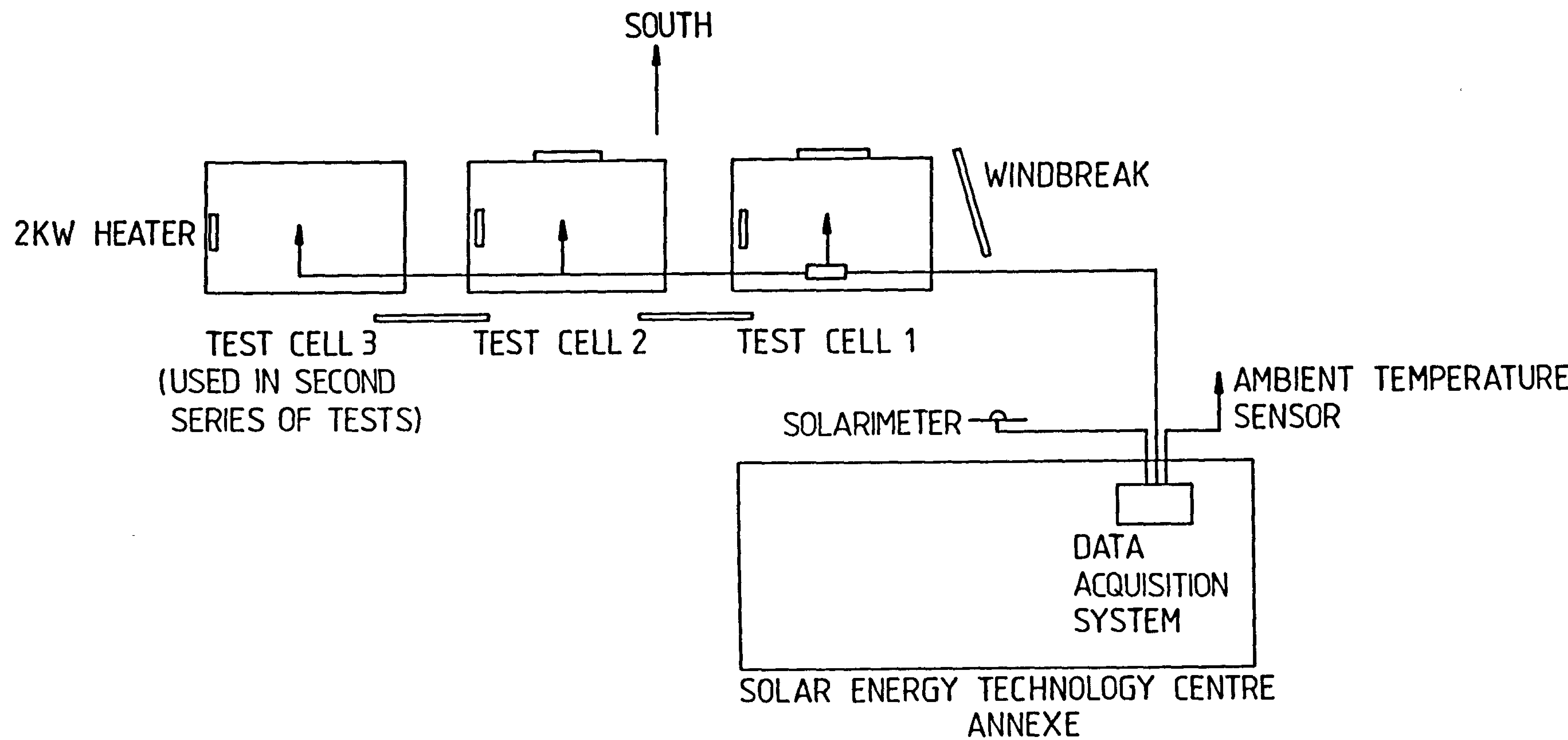


Fig.32 Plan showing relative positions of test cells, weather measurements and data acquisition system.

polystyrene and foil-faced fibreglass. The collector absorbers consisted of six tubes of 16-gauge aluminium of length 1.5 m and 0.1 m diameter, painted matt-black internally and externally. Air deflectors, also of black-painted 16-gauge aluminium were installed at the top and bottom of the absorber-tubes to reduce dynamic head losses, and a cover of 5 mm glass was fitted at a distance of 70 mm from the absorber surface. Details of the U-tube collector are shown in Fig. (33).

5.3 THE TEST CELLS

The test cells consisted of two "Windsor" sheds and their specifications were as follows:

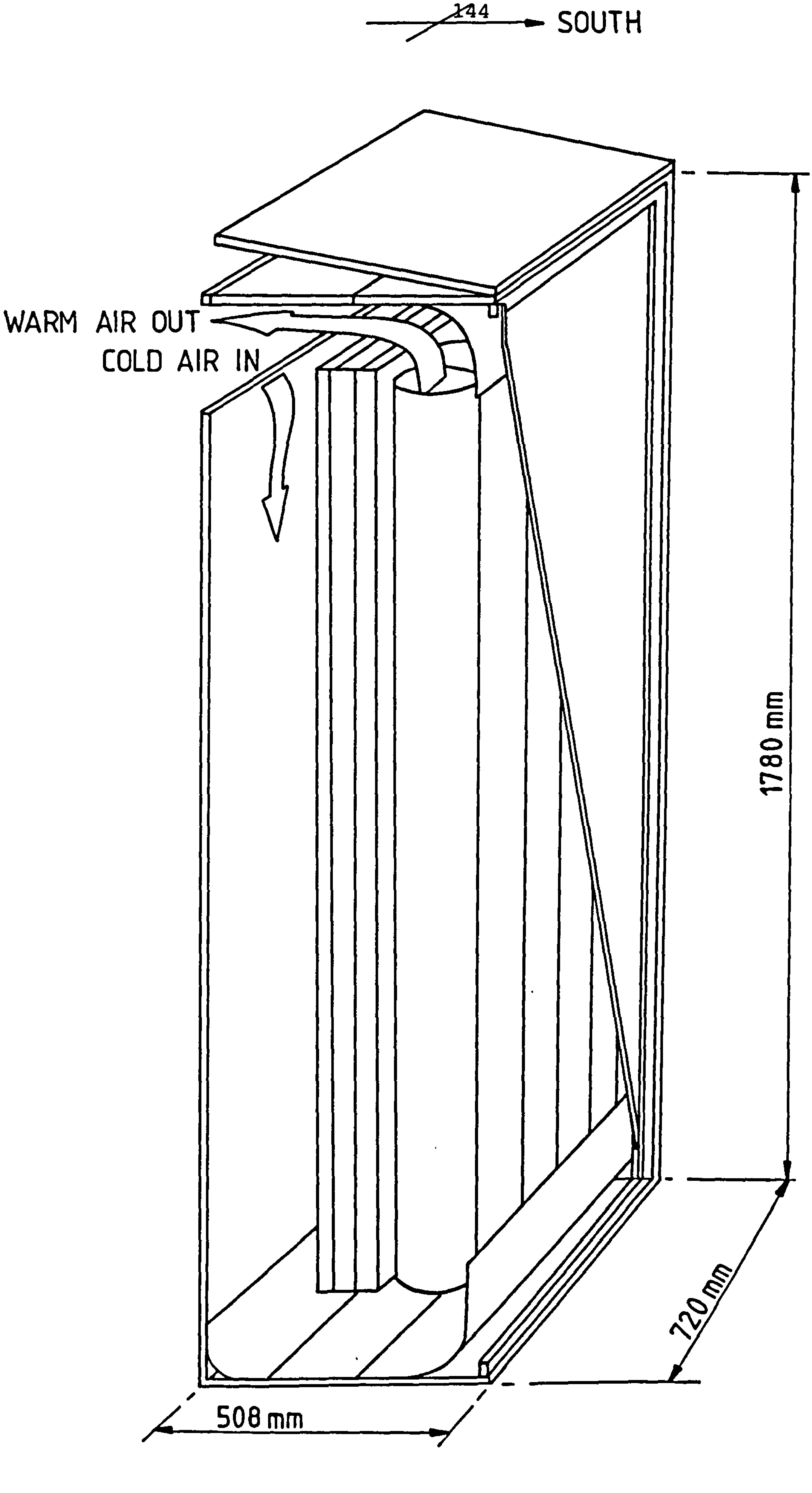
Overall dimensions	:	1.82 x 1.21 x 1.82 m
Glazing	:	1.42 x 0.61 m
Walls	:	0.127 m clad cedar shiplap boards on 0.038 x 0.038 m frame work
Roof	:	Felted 0.015 m tongue and groove machined boards on 0.058 x 0.038m purlins
Floor	:	0.015 m tongue and groove machined boards on 0.058 x 0.038 m joists.

The collectors were attached to the south facing walls of these sheds with 0.038 x 0.038 m softwood batons. The U-Tube collector projected into its test cell a distance of 8 cm. It was accommodated in this manner so that the U-Tube and standard TAP collectors projected the same distance from the test cell wall and thus heat losses from the collectors would be comparable. The collectors were not installed within the test cells so that thermal stratification effects could be assessed for nominally empty sheds. Each test cell was equipped with a standard domestic electricity meter and switching gear. A 2 kW electric convector heater was also installed in each shed and fitted with a time switch and thermostatic control. A cut-away view of a test cell is shown in Fig. (34).

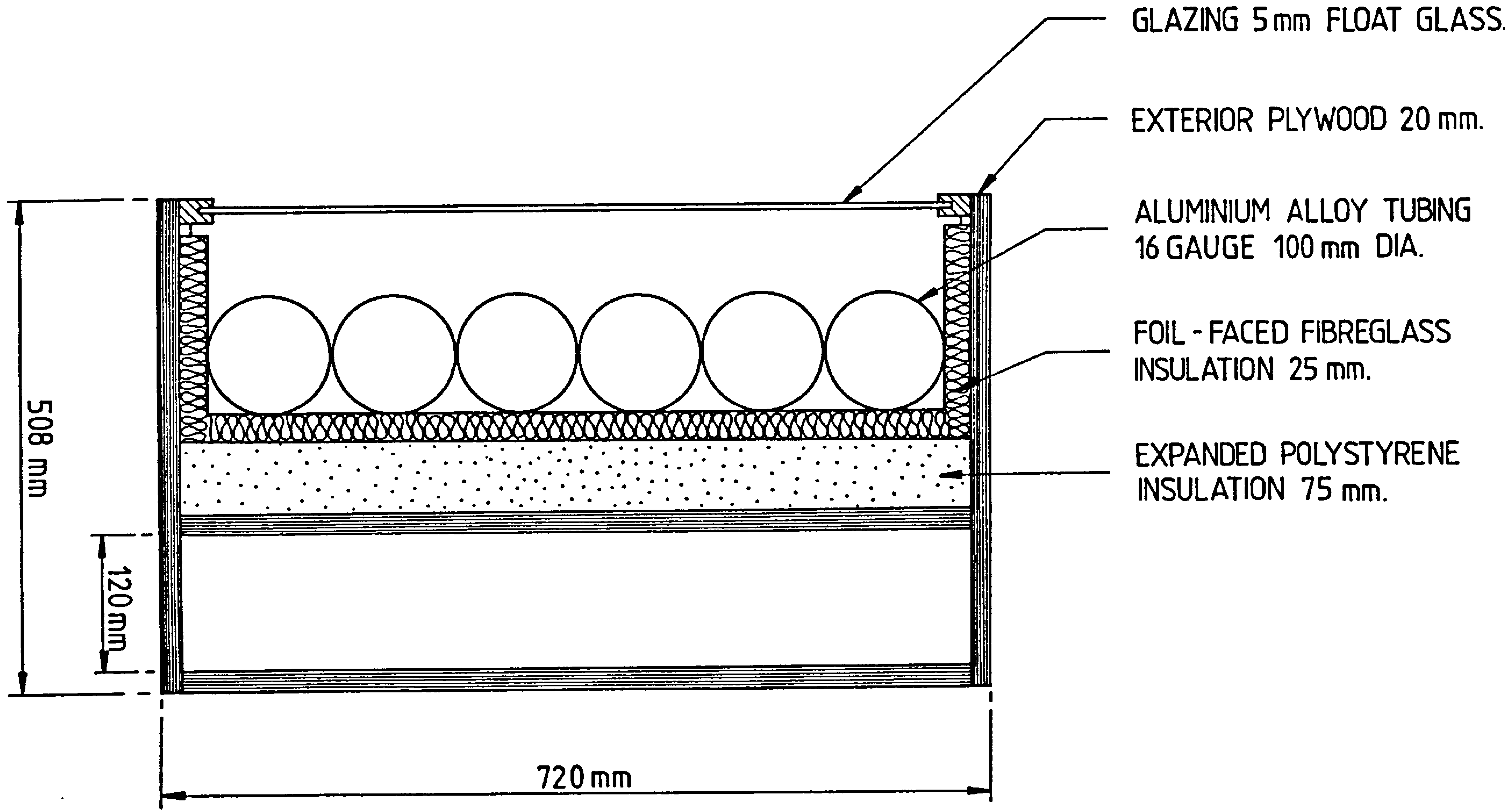
Fig.33 U-Tube collector with tubular absorbers

a) cut-away view

b) section



a) cut-away view


GLAZING 5 mm FLOAT GLASS.
EXTERIOR PLYWOOD 20 mm.
ALUMINIUM ALLOY TUBING 16 GAUGE 100 mm DIA.
FOIL - FACED FIBREGLASS INSULATION 25 mm.
EXPANDED POLYSTYRENE INSULATION 75 mm.
508 mm
120 mm
720 mm

b) section

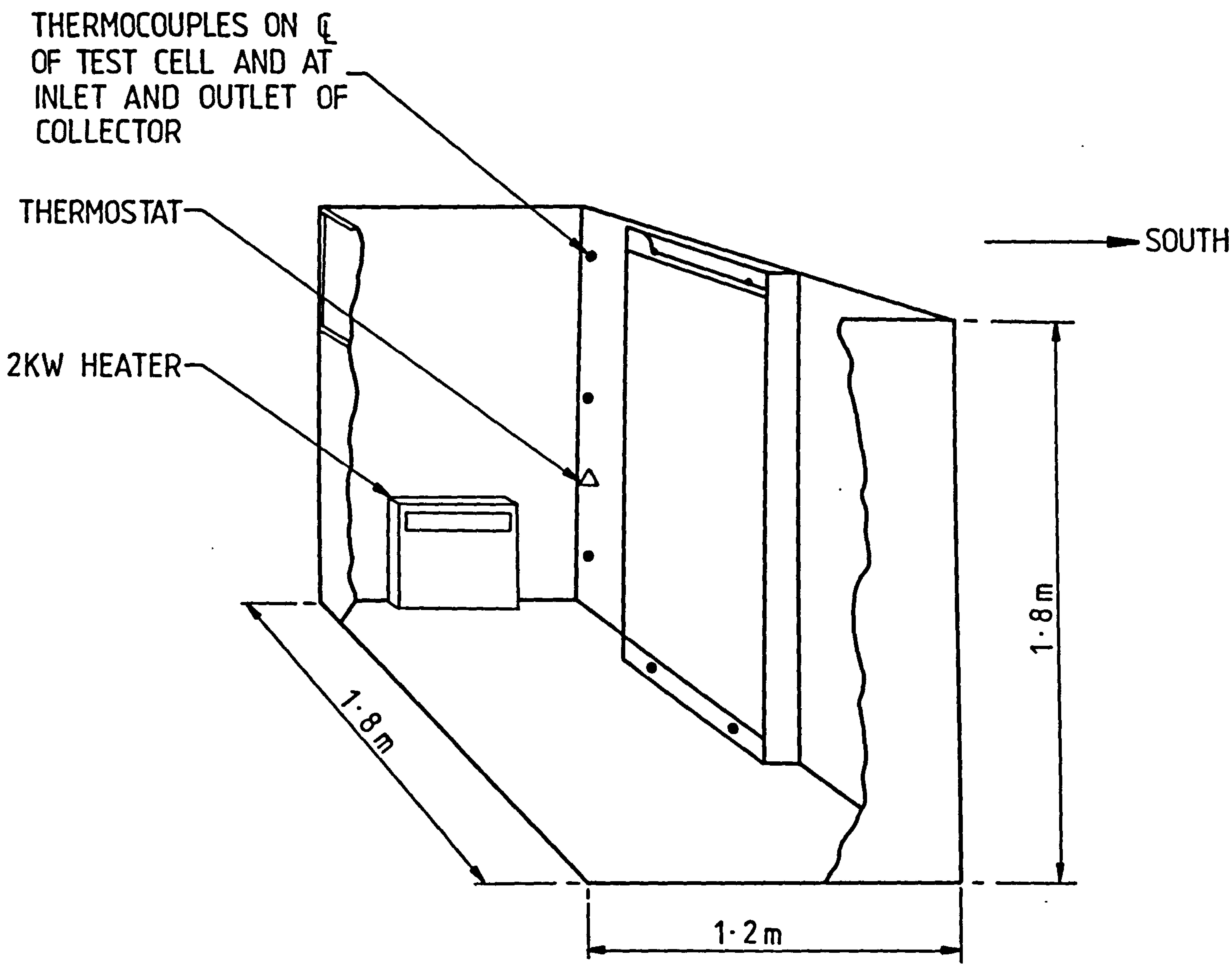


Fig.34 Cut-away view of test cell as used in experiments

5.4 INSTRUMENTATION

All temperatures were measured with copper-constantan (type T) thermojunctions. 0.2 mm fine thermocouple wire was used to ensure rapid response to temperature changes and reduce errors due to radiation effects. Thermocouples were positioned in the centre of the air streams generated by the collectors to be clear of any thermal boundary layers. The themocouple positions are shown in Fig. (34).

The signals from the thermocouples were recorded on a FLUKE 2200B Datalogger at hourly intervals. The thermocouples were calibrated to an accuracy of within ± 0.5 of a degree centigrade.

Ambient air temperature was measured with a further thermojunction, shielded from solar radiation and wind effects located in an enclosure approximately 10 m. from the test cells.

Solar radiation was measured with a Moll-Gorczynski pyranometer which is rated as a "second-class" pyranometer by the World Meteorological Organisation (1971). This thermopile instrument has a receiving surface covered by two concentric ground and polished glass domes.

The World Meteorological Organisation specifications for a second-class pyranometer include a sensitivity of ± 5 W/m^2, a temperature coefficient of ± 2% per °C and a linearity of ± 2%. The Moll-Gorczynski pyranometer is subject to slight zero-depression errors. The calibration curve for the instrument used yielded the relationship

$$\text{Insolation } (W/m^2) = 84034 \times \text{pyranometer output (mV)}$$

The pyranometer was mounted in the same vertical south-facing plane as the collectors, 4 m above them. The device was free from shadows and isolated from conduction sources. Signals from the pyranometer were recorded on the Datalogger after having been averaged with a "Solar Integrator". This latter device, accurate to 12 bits, produced average values of insolation for a range of time periods : i.e. 5, 15, 30, 45 and 60 minutes. The integrator output signal was used to trigger the Datalogger so that all measurements were recorded at time intervals determined by the Solar Integrator. U-Tube and "standard"

TAP efficiencies were estimated using one of two methods:-

1) Measuring the time taken for air to pass through the collector and its heat gain.

2) Estimating the amount of electrical energy consumed in the two test cells during the hours of daylight, and calculating the amount of energy suppplied by the collectors during this period for a given design temperature.

Thermal stratification in each test cell was also investigated so as to determine which collector design produced the more comfortable conditions.

5.5 AIR VELOCITY MEASUREMENT

An instrument was required which was capable of measuring average representative velocity at the entrance or exit of the collector flow channels. Velocities of under 0.3 m/s were anticipated in these regions, a velocity pressure of about 0.05 N/m^2 [BUREK (1984)]. Devices which incur a significant pressure loss such as orifice meters, venturi meters and turbine meters were not suitable. Laser-Doppler anemometry was considered because this technique can be used to measure accurately very low flow rates. However, it is more suited to work on boundary layers where measurement of precise points in the flow are required, rather than average velocities in ducts. Laser-Doppler anemometry equipment is expensive, requires a high degree of specialist skill to operate and is better suited to laboratory conditions rather than outdoor test cell conditions.
A hot-wire anemometer was chosen as being the most practical and least expensive option. BUREK (1984) defined five criteria which a hot-wire device had to satisfy for the tests envisaged.

1) It must be capable of measuring flow rates $\geq$ 0.2 m/sec.

2) It must have a small probe.

3) It should be temperature compensating.

4) It must have a signal output suitable for a Datalogger.

5) It must be capable of operating unattended for prolonged periods.

Only six commercially-available devices met the first criterion. Of these, some did not meet condition 4) whilst others were battery operated and so did not meet condition 5). When used with a mains transformer, a Prosser Scientific Instruments AVM 501 TC hot-bead anemometer satisfied all the above criteria. It was capable of measuring down to 0.2 m/s with an accuracy of $\pm$ 5% [Prosser Scientific Instruments (1976)].

Two of these instruments were used throughout the tests and calibration curves for the hot-bead probes gave the following:-

$$\text{Velocity (m/s)} = A \times (\text{recorder output (V)})^{B}$$

where A and B were constants determined from calibration of the devices in a wind tunnel. Several probes were used during the course of this work.

A method of air velocity measurement which was employed additionally in certain experiments was the determination of the time taken for smoke to travel through the collectors. This method has several disadvantages; it cannot be monitored continuously or recorded on a Datalogger and is extremely unpleasant! It is, however, inexpensive, reliable and direct. A "bee keeper's smoker", which burnt corrugated cardboard, was found to be the most satisfactory equipment for smoke generation for these purposes.

5.6 AIR VELOCITY MEASUREMENT PROCEDURE

Accurate measurement of the slow and fluctuating air flow rates which ensue in natural convection air-heating systems is not straightforward. To record such flow rates, devices such as hot wire anemometers have to operate at the lower limits of their accuracy. Furthermore, it is difficult to ensure that the values recorded by such devices at a particular position within a system are representative of air velocity "as a whole".

Trombe et al (1976) installed a hot wire anemometer in the inlet duct of a collector and calculated average values for flow rate through the duct using the following relationship:

$$\bar{v} = k \bar{v}$$

where k, the relationship between the average velocity $\bar{v}$ and the velocity at the centre of the duct v, was established by measuring the horizontal and vertical velocity profiles in the rectangular duct and then integrating to obtain a value for the total surface area. Values of k were plotted against Reynolds number and it was found that k varied by approximately 0.5 for low speeds (Re < 2000) and 0.7 for higher velocities. The inaccuracy of velocity measurement by this method was such that a figure of ± 28% was quoted for the amount of energy transmitted by thermocirculation in the Trombe wall under test.

An alternative method of directly measuring velocity is to release smoke into the inlet vent of a collector and estimate the time taken for the smoke to pass through the device. This method has been used by Reif (1981) and by Summers (1982) who described the procedure. Air flow was determined by burning incense beneath a metal container and releasing the collected smoke next to the vent. The time taken for the incense smoke to move from the inlet to the exhaust vents was timed with the aid of a stop watch which was started at the time of the release next to the vent, and stopped when the smoke was visible at the exhaust port. It was noted that variations of flow rate of between 10 and 20% were recorded for very similar test conditions. This is to be expected due to dilution of the smoke in the air stream.

A subsequent procedure by Summers (1982) tested the accuracy of the smoke test readings which involved a comparison of air velocity meter and smoke test results. The smoke tests did not give any indication of the difference in flow rate between the top and the bottom of outlet vents, whereas quantitative results were obtained with the velocity meter. The meter measured velocities at three positions on each outlet vent tested: level with the top of the vent; directly in the centre of the vent; level with the bottom of the vent. It was concluded from the results obtained that the volume of air passing through the upper and mid ranges of outlet vents was much greater than that passing through the lower regions. Average values computed from readings taken at the middle and upper vent positions were found to be between 2% and 11% greater than those obtained from smoke tests. The average values of centre and upper air meter readings were considered to be more reliable indicators of TAP performance than average values which included readings taken at the bottom region of the vents. Tests were carried out without grates or backdraught dampers in position in this work.

The work of Trombe et al (1976) and Summers (1982) raises a number of important questions regarding the measurement of air flow rates in natural circulation air heating systems:

1. Should air velocity be measured at the inlet of a collector, or at the outlet?

2. Is it more representative to measure air velocity with backdraught dampers and grilles in position?

3. Should an average value of air flow rates based on velocity profiles for the whole duct be used, or should average values based on a representative portion of a duct be used?

4. The time taken for smoke to pass through a collector gives an indication of the time an equivalent column of air takes to pass through a collector, but is it truly representative, in view of the sensitivity of a thermosyphon system to variations in density and temperature?

An analysis of the smoke test method was undertaken by Hobson (1986). A method by which mass flow rates through natural circulation air heating collectors was developed which accounted for the increase of air velocity with temperature.

Air flow rates have also been computed from velocity profiles by Bilgen & Chaaban (1982), whilst Biehl (1981) determined air flow by conducting an energy balance on an air-to-water heat exchanger which was used to cool air heated by a collector and simulate a "working load".

5.7 FURTHER TESTS

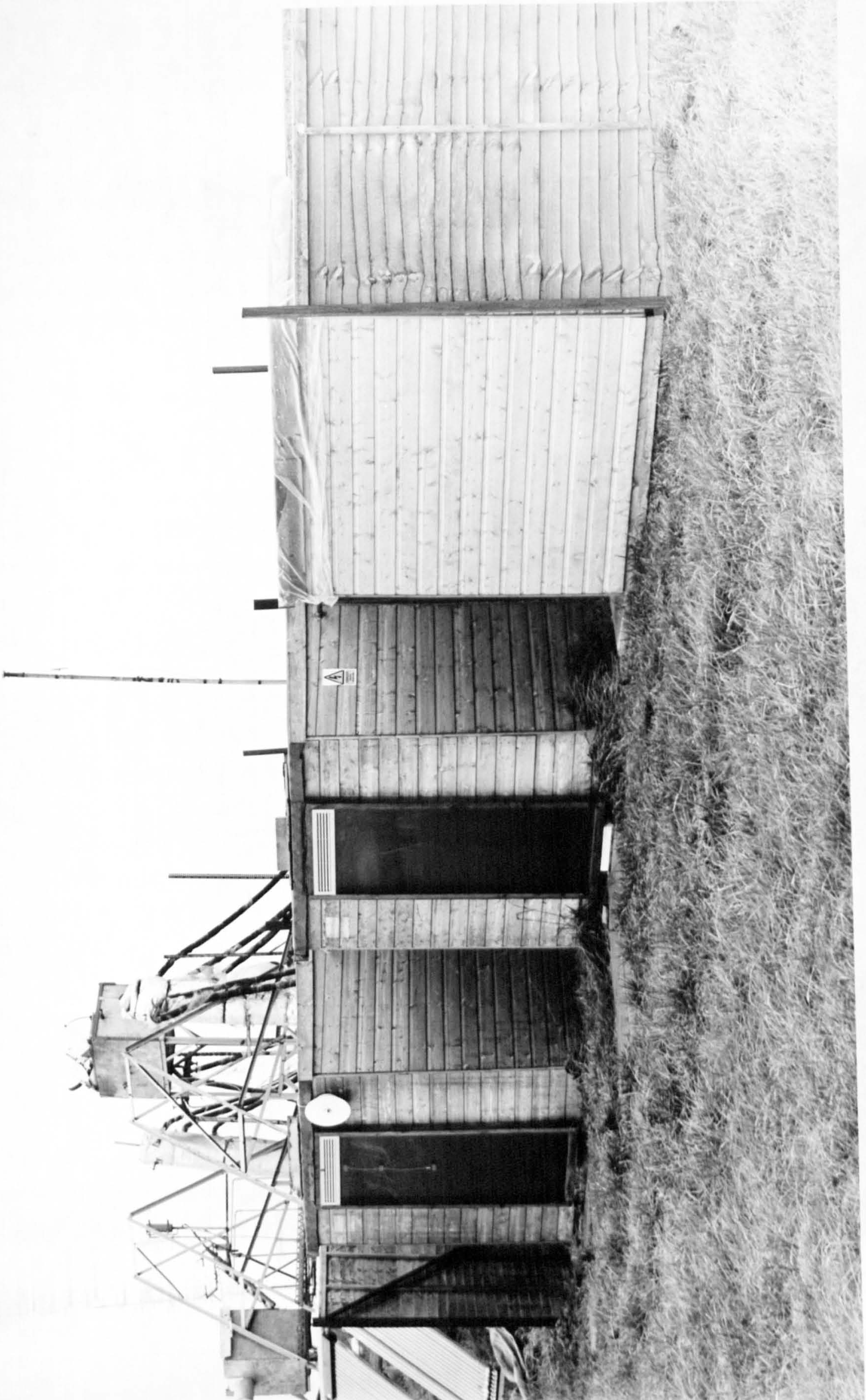
A series of tests were carried out to determine the efficiency of glazing collector panels. Such collectors may be considered to be a variant of standard thermosyphoning air panel designs in that they are installed behind existing glazing.

In view of the difficulties experienced whilst testing two dissimilar systems side-by-side it was decided to use a third test cell as a reference and install identical collectors on the two existing cells. Fig. (35). These were prototypes of glazing collector designs : flat-plate Back-Pass collector boxes fitted behind double-glazed covers and positioned within the test cells. The direct comparison of two identical systems allowed average values of collector temperatures, flow rates, etc. to be derived. Modifications to one or other of the collectors may also be evaluated with this facility, and the effects of such variables as wind speed and direction more acccurately determined.

The glazing panels tested were pre-production prototypes of panels intended for installation in schools. They were, therefore, considered to be more representative of the envisaged retrofit thermosyphoning air panel than the U-Tube and "standard' TAPs tested previously. The glazing panels were of the "standard" TAP configuration, chosen in preference to the U-Tube by the prospective clients because they required a panel which did not project significantly into the school rooms.

The prototype collectors were constructed in an unorthodox manner in that the insulation was placed outside the collector casing rather than inside. There were two reasons for this departure from the conventional practice of placing insulation within the body of the collector; (i) all of the sheet metal fabrication, including the fixing of the absorber plate, could be completed prior to the insulation of the collector, and (ii) the insulation could be changed after the collectors had been installed in the test cells. It was also felt that there might be advantages in having all four sides of the collector casing heating the air as it passed through the flow channel. This is a more symmetrical heating regime than the one which

Fig.35 Glazing collector panels fitted to test cells



exists when air is heated by one side of a thermally-isolated absorber plate. Details of the prototypes are shown Fig. (36)

The relative merits of these two design approaches are outlined in Table 10.

Temperatures were monitored at five minute intervals with the additional measurement of the glazing temperature of the glazing panel fitted to test cell No. 1.

The ceilings and walls of all three test cells were lined with polythene sheet for the remainder of the tests undertaken so that the air infiltration rates of the older sheds could be considered to be the same as that for the new shed. The polythene sheet also advantageously raised the insulation level of the sheds. The interior of a test cell fitted with a glazing collector is shown in Fig. (37).

5.8 DATA ACQUISITION

A series of experiments was conducted in which data was recorded with a FLUKE 2200B Datalogger, which scanned the appropriate channels and printed transducer outputs onto a paper roll. Transferring data from paper into a computer was a time-consuming process and introduced errors into the procedure. A more advanced data-acquisition system was introduced. This had the capacity to transfer data onto magnetic discs where the information could be stored until it could be processed on a main-frame or micro-computer.

The data logging system consisted of a BBC microcomputer, visual display unit, disc drive, a specialist instrumentation peripheral and an IEEE-488 interface.

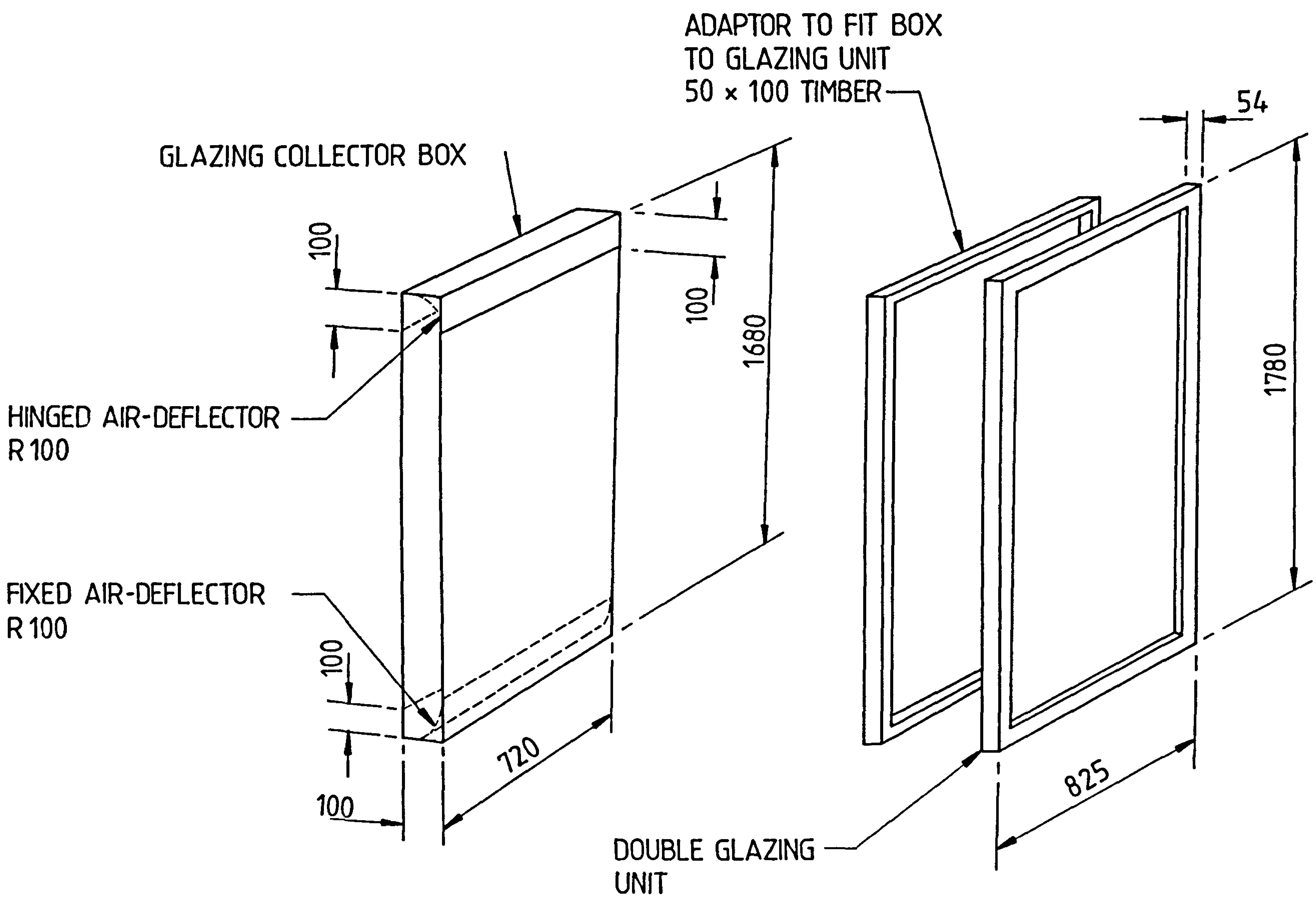
The BBC Model B microcomputer was used in conjunction with a multi-application peripheral system (M.A.P.S.) manufactured by Harlyn Automation. This consisted of two main-frames (type R6B) which accommodated a total of twelve plug-in modules, of which four were utilised. The four modules comprised of two ten channel four-pole scanner modules (type 3700-10a), a four-range bipolar analogue-to-

Fig.36 Glazing collector prototype

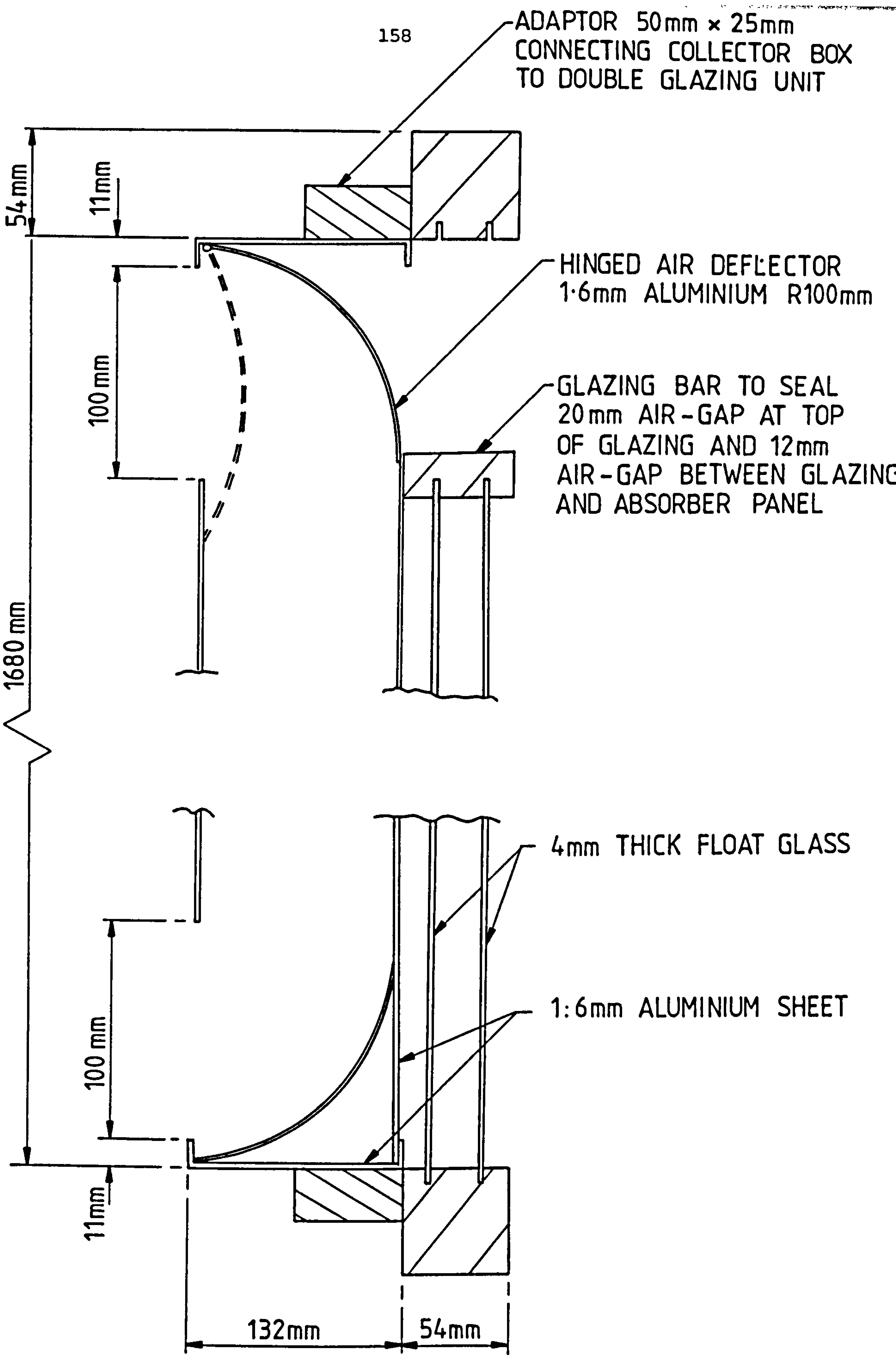
a) exploded view

b) vertical section (uninsulated collector)

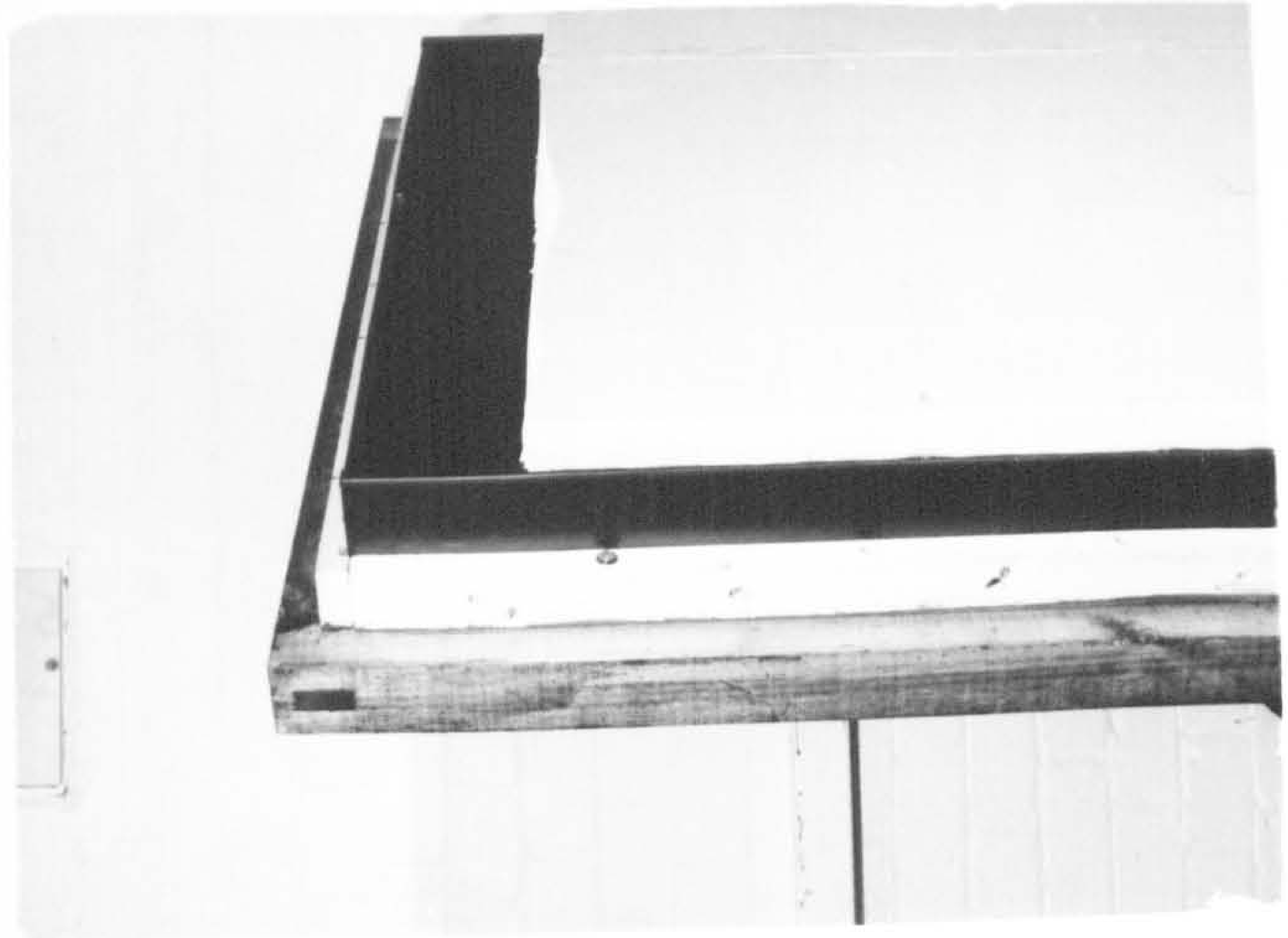
c) details of components

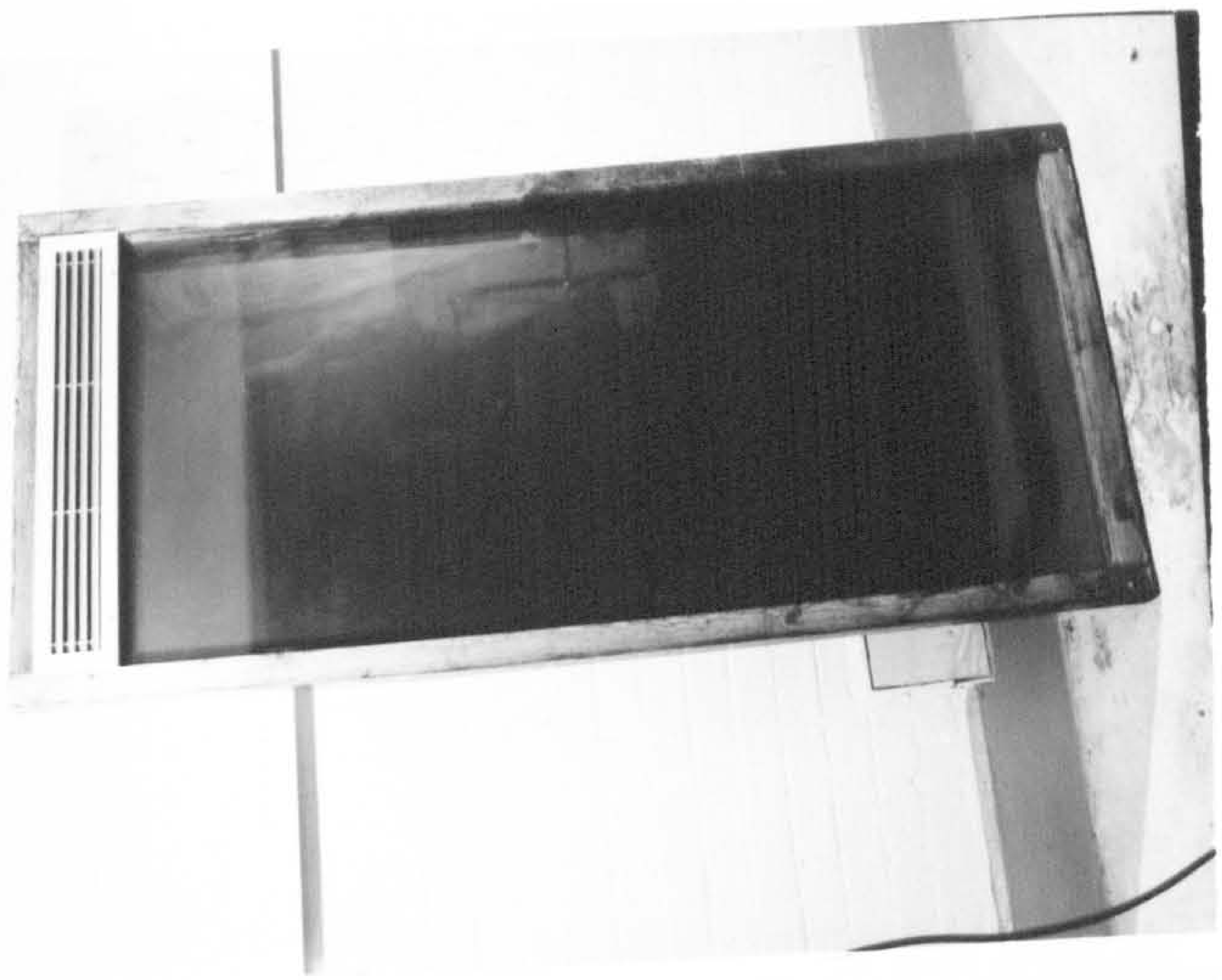


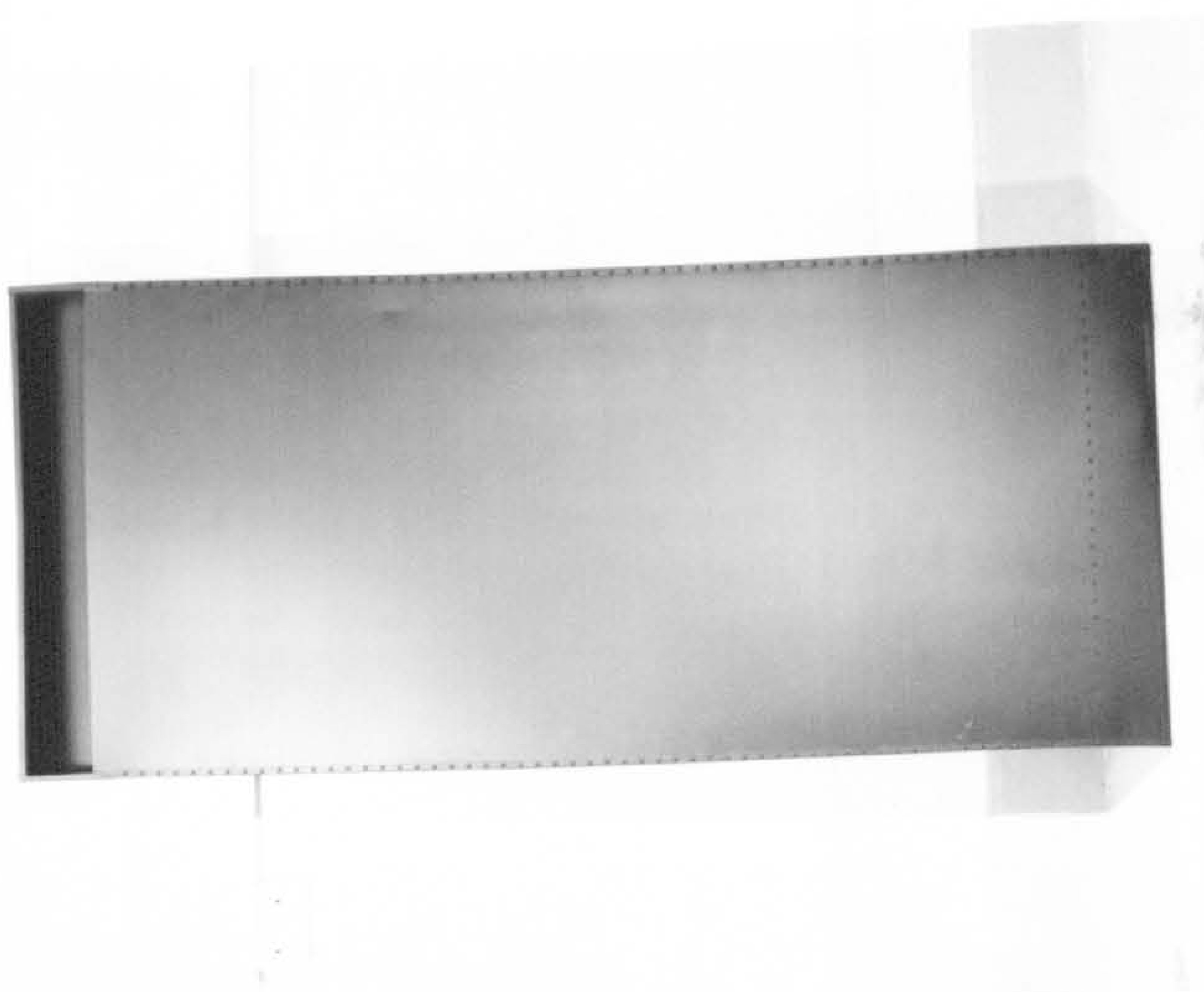
a) Exploded view



b) vertical section (uninsulated collector)







	ADVANTAGES	DISADVANTAGES
INTERNALLY INSULATED COLLECTOR	. TRIED AND TESTED . INSULATION PROTECTED BY COLLECTOR CASING . ISOLATED ABSORBER MAY REACH HIGHER TEMPERATURES	. ABSORBER PLATE MUST BE THERMALLY ISOLATED FROM COLLECTOR CASING . AIR IS IN CONTACT WITH ONE FACE OF THE ABSORBER PLATE ONLY - ASSYMMETRICAL HEATING . INSULATION HAS TO BE FITTED DURING MANUFACTURE OF COLLECTOR CASING
EXTERNALLY INSULATED COLLECTOR	. AIR HEATED BY FOUR WALLS OF COLLECTOR-SYMMETRICAL HEATING . INSULATION MAY BE ADDED AFTER INSTALLATION OF . REQUIRES LESS SHEET METAL TO FABRICATE THIN CONVENTIONAL DESIGN	. GREATER POTENTIAL FOR HEAT LOSS . GREATER AMOUNT OF METAL HAS TO BE HEATED . INSULATION MAY REQUIRE PROTECTION . REQUIRES THERMAL BOND BETWEEN ABSORBER PLATE AND CASING . REQUIRES MORE INSULATION THAN CONVENTIONAL DESIGN

Table 10. Relative Merits of Internally and Externally Insulated Air-Heating Collectors

Fig.37 Interior of test cell fitted with glazing collector panel

digital converter (A.D.C.) (type 3700-11) and a three channel digital-to-analogue converter (D.A.C.) (type 3700-09).

The scanner modules were programmed by transmitting, from the microcomputer, a sequence of ASCII command strings via the IEEE interface to the M.A.P.S. main frame. When a sensor was being read, a series of three basic command strings were transmitted putting into effect the following sequence of events:

1) The appropriate analogue input pair was selected and switched onto the A.D.C. for measurement.

2) The A.D.C. range was set and a measurement taken after an internal delay, set nominally at 120 mS, had transpired. This delay allowed the input amplifiers to settle after a change of range had occurred.

3) After a delay greater than that set in the A.D.C., the reading was requested and then transferred to the disc memory in the form of a twelve bit, three character ASCII string. A choice of four settings were available for the A.D.C. giving ranges of 10, 1, 0.1 and 0.01 volts.

5.9 DATA PROCESSING METHODOLOGY

Data obtained from tests on the U-Tube and "standard" TAP systems was transformed mannually from the FLUKE 2200B Datalogger printout to a data-file on the VAX main-frame computer at the Computer Centre at Cranfield. The program 'GRAP' was used to produce graphs, typically those shown in Fig (41), which indicated temperature distributions through both systems and corresponding diurnal variations of insolation and ambient temperature.

Data recorded using the microcomputer system was processed using a program entitled "VCONR'. This program produced a value of instantaneous collector efficiency for each set of data, based on the equation derived for the Glazing Temperature Method. Further programmes were used to produce graphs of instantaneous efficiency against a number of parameters.

5.10 BACKDRAUGHT DAMPER TESTS

The flap-valve backdraught damper has been installed in many "standard" TAP heating systems as it is the only self-regulating and inexpensive means of preventing reverse circulation. However, a flap-valve may adversely affect the performance of a TAP by obstructing the air flow through the collector to some extent.

In view of the dearth of information on flap-valve backdraught dampers, particularly their influence on air flow through TAPs, a series of experiments were undertaken to investigate the operating characteristics of these devices. Experimental apparatus, which had originally been constructed to simulate conditions commonly found in dwellings: one outside wall containing a window with a parapet above and below and with a radiator positioned in front of the lower parapet Fig. (38), was adapted for this purpose.

The window was represented by a water-cooled steel plate and the radiator by an electrically heated steel plate. The cold-plate was enclosed on one side by hardboard and on the other by Perspex sheet; this allowed an unobstructed view of the flow as it passed over the cold-plate whilst preserving the flow regime. An aluminium air deflector was added to the test rig so that the warm air generated by the hot-plate would be directed through 90° and enter the space in front of the cold plate in a similar manner to air being drawn into the inlet of a TAP. A wire mesh was installed in the space between the top of the air deflector and the lower parapet to represent a grille fitted in the inlet duct of a standard TAP. Tests were conducted with and without a Tedlar flap positioned behind this grille so that the amount of flow resistance generated by the flap could be measured.

Tests were also carried out using the cold-plate to determine the extent to which the presence of a Tedlar flap prevented reverse thermocirculation. The cold-plate generated a cold downdraught similar to that which may ensue in a TAP at night or during non-gain periods. Buoyancy-driven convection and reverse-thermocirculation in the backgraught damper simulation test rig are shown in Fig. (39).

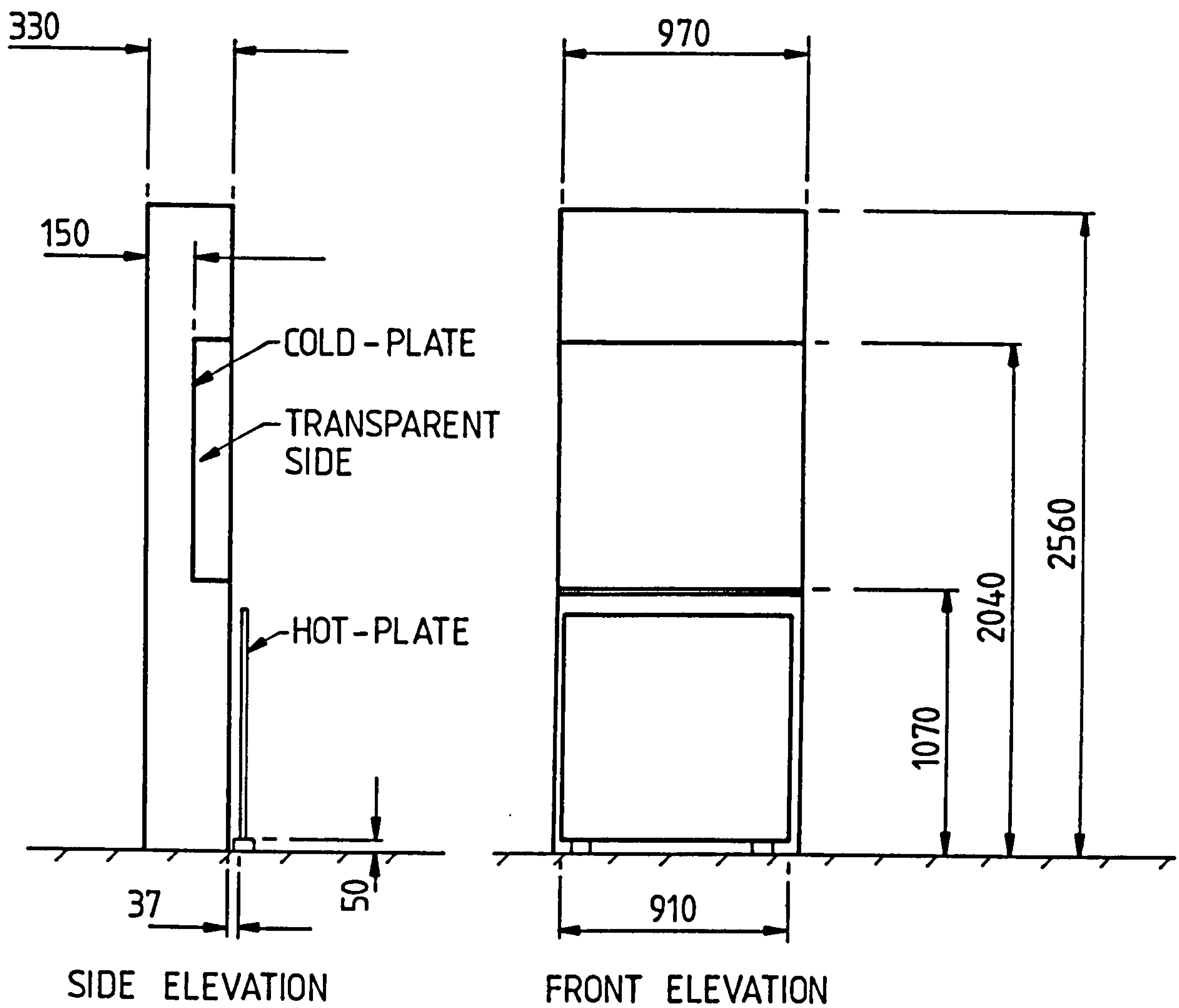


Fig.38 "Backdraught damper" test rig

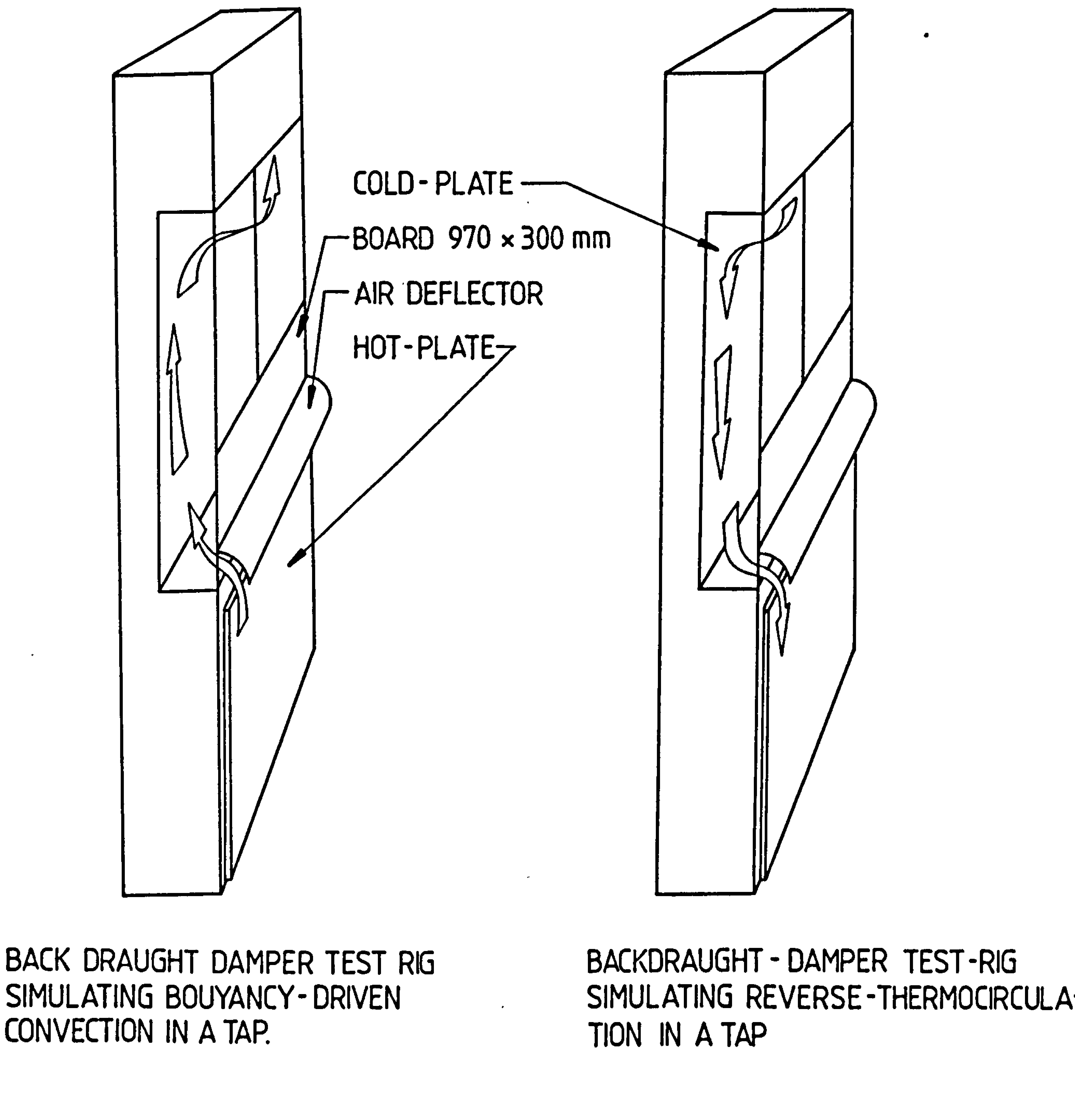


Fig.39 Buoyancy-driven convection and reverse-thermocirculation in backdraught damper simulation test rig

Temperature and air velocity measurements were undertaken at a number of points in the test section to quantify the amount of reverse flow prevented, and the amount of flow resistance generated by this mechanism. A hand-held digital thermometer manufactured by R.S. Components (Model "sfk" No. 610-067) was used to measure temperatures in these experiments. Velocities were measured with a Prosser Scientific Instruments' hot-bead anemometer (Type AVM 501 TC).

Flow visualisation experiments were also conducted so that the extent of the disruption of air flow by a flap valve could be observed. Smoke was generated at the foot of the hot-plate so that it would be entrained into the warm convected air and pass into the test section. Photographs of the resultant flow patterns within this region were taken with a tripod mounted Praktica camera with the aid of electronic flash equipment.

Chapter 6

RESULTS

Attempts were made to measure the instantaneous efficiency of the U-Tube and 'standard'TAP collectors using results obtained from both the smoke-test method and using hand-held air velocity meters. In both cases difficulty was experienced in estimating the degree of accuracy of the few results obtained: the test cells had to be occupied for the duration of each test, which influenced the results to some extent. It was unfortunate that a great deal of the data recorded from the U-tube and 'standard' TAP test cells were of little value as the glazing temperatures of the respective collectors were not monitored. It was only after these collectors had been replaced by glazing collector prototypes, and the Glazing Temperature Method of determining instantaneous efficiency developed, that these temperatures were recorded.

The introduction of the more advanced micro-computer based data-acquisition system allowed the regular monitoring of air flow-rate by installing air velocity probes in the inlet vent of each prototype glazing collector and recording their output at five minute intervals. However this failed consistently to produce results of any value as the probes were affected adversely by condensation which accummulated on them at night. It is questionable whether a method of determining instantaneous collector efficiency using measured air velocities is sufficiently accurate in view of the reservations expressed earlier in the text.

6.1 U-TUBE vs STANDARD TAP

A diurnal adaptation of the SLR/SSF correlation was used to investigate the comparative performance of the standard TAP and U-Tube systems. These correlations apply to the whole system and are thus illustrative of performance differences, and sources of model validation data, rather than design aids. The performance curves of the two systems as shown in Fig. (40) indicate that the U-Tube collector was the more effective auxiliary heater. This conclusion should be treated with some caution in view of the limited amount of data available and the un-investigated influence of several parameters such as the position of the thermostats within the test cells. The superior performance of the U-Tube collector may be attributed to a

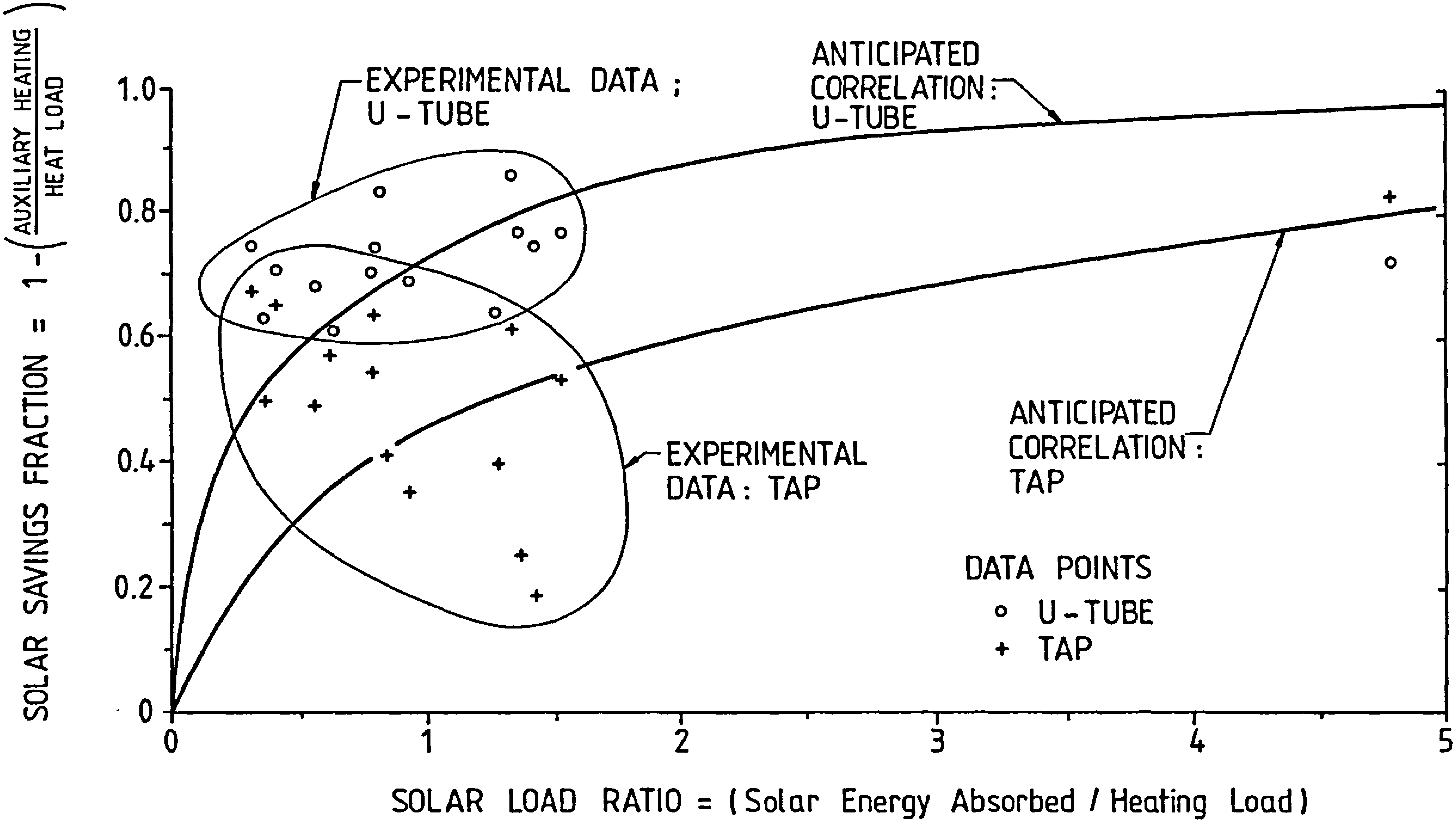


Fig.40 Variation of Solar Savings Fraction with Solar Load Ratio for U-Tube and standard TAP systems

combination of a number of factors, particularly:

1. Thermal stratification and its influence on the thermostatic controllers of the convector heaters.

2. The U-Tube collector entrained warmer, ceiling-level air rather than the cooler floor-level air entrained by the standard TAP.

3. The inhibition of transient reverse-thermocirculation during periods of low insolation by the U-Tube design. The standard TAP was not fitted with a backdraught damper during the course of these tests.

Fig. (41) illustrates the temperature distribution in the U-Tube and standard TAP test cells in the absence of electrical heating. The temperature at the top of the U-Tube test cell is appreciably higher than the equivalent position in the standard TAP test cell. However, there was negligible difference between the average temperatures in the respective test cells. Subjectively the standard TAP system produced a more comfortable environment than the U-Tube because of the more even floor to ceiling temperature distribution.

Fig. (42) shows vertical air-density distributions around the prescribed flow path for both the U-Tube and standard TAP system and illustrates the temporally uniform density variation with height and hence, velocity through both systems during the period of solar gain. Attempts were made to determine friction factors for both systems and develop a computer program which would combine these values with those of bouyancy forces derived from the temperature density profiles.

Collector efficiency could then have been determined from temperature measurement and the measurement of insolation. However, estimates of the values of friction factors, were difficult to determine with any degree of accuracy.

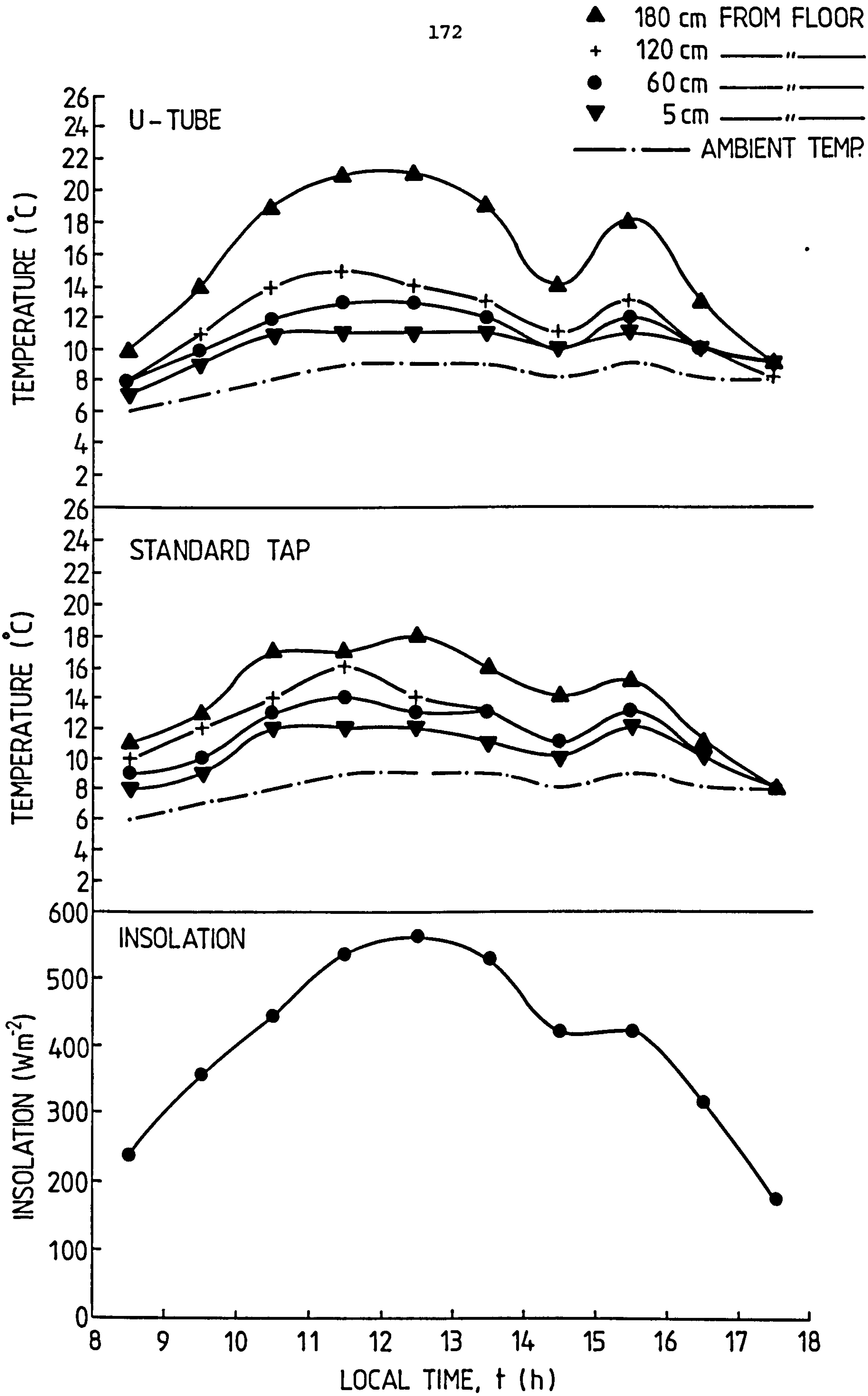


Fig.41 Temperature distribution in U-Tube and standard TAP test cells

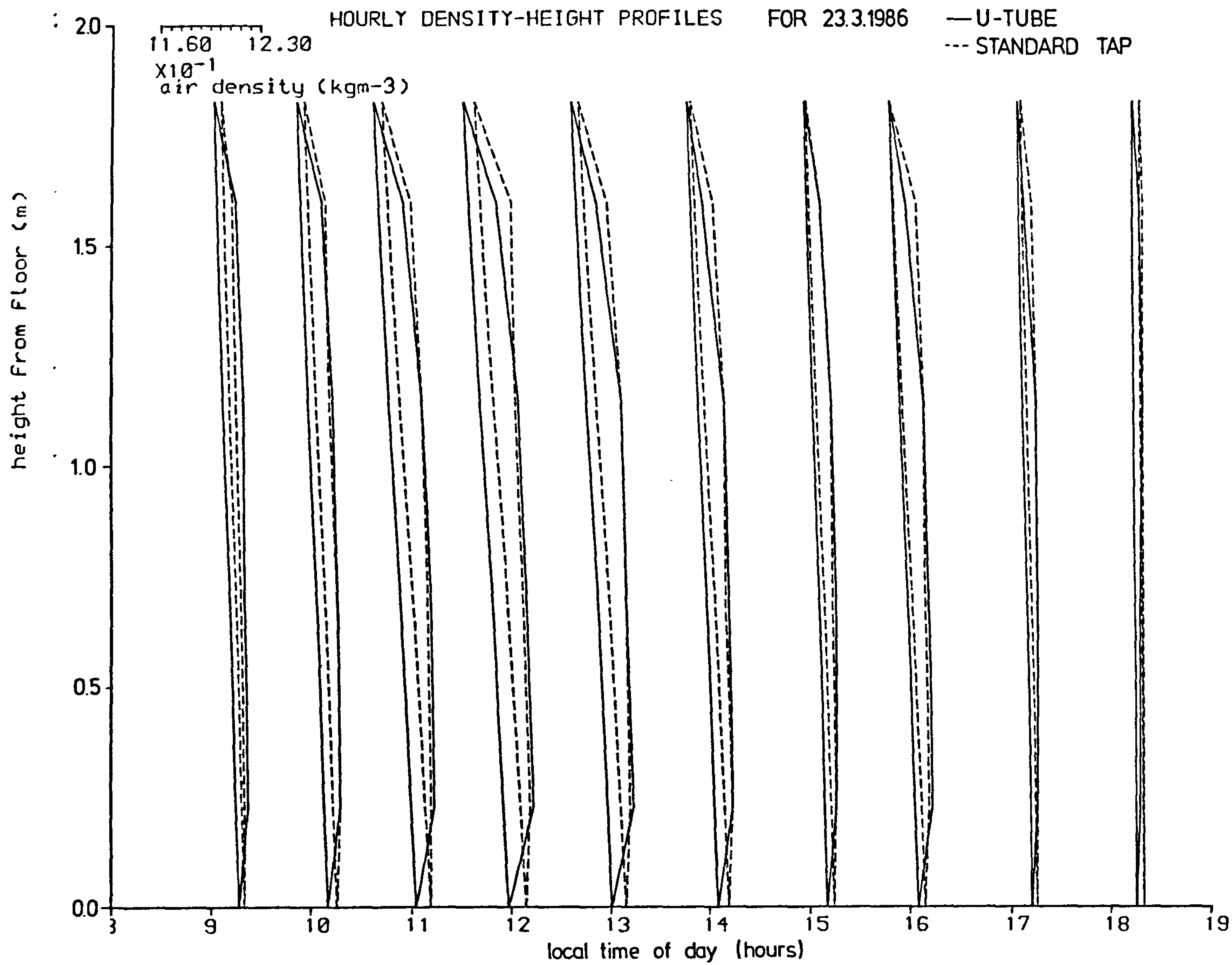


Fig.42 Hourly density-height profiles for U-Tube and standard TAP systems

6.2 GRAPHS OF RESULTS OBTAINED FROM GLAZING COLLECTORS

Larger temperature differences between the inlet and outlet of a collector can be primarily the consequence of the mass flow rate of air through the collector decreasing. This results in higher absorber plate temperatures which, in turn, produce greater losses from the collector. Increased losses from a collector are concomitant with an increase in temperature difference between glazing and ambient. Fig. (43) shows that this is indeed the case: the difference in temperature between glazing and ambient increased as the temperature difference between the collector inlet and outlet increased.

When collector efficiency is plotted against the quantity $(T_i-T_a)/I$ the points should be on a straight line with gradient -U and intercept $\tau\alpha$.

For graphs of instantaneous efficiency against $(T_i-T_a)/I$ for the pre-production glazing collectors (obtained using the Glazing Temperature Method and shown in Figs. (46)(48)(49), this is indeed the case. However, for the graphs of daily efficiency against $(T_i-T_a)/I$ for the U-Tube and standard TAP systems calculated using the adapted "fast" solar load ratio method the gradient is of the "wrong" sign, Figs. (44, 45). This may be the result of thermal storage effects, or may be an effect of fluctuations in infiltration rates within the systems under test.

Plots of hourly efficiencies for an hybrid solar-energy air-heating system evaluated by Clegg et al (1985) also produced gradients with a similarly "wrong" sign, implying that the collector increased in efficiency on dull, cold days: the opposite of behaviour expected from flat-plate collectors. It was suggested initially that this could have been due to the influence of thermal mass in the collector which had not been accounted for. Further investigations revealed that there was sufficient collector mass to store energy during the day and release heat in the evening during lower levels of insolation. Consequently, reduction of insolation levels during the mid-afternoon period were not accompanied by a corresponding drop in efficiency.

A graph of instantaneous efficiency against the difference between the collector inlet temperature and ambient, divided by the insolation,

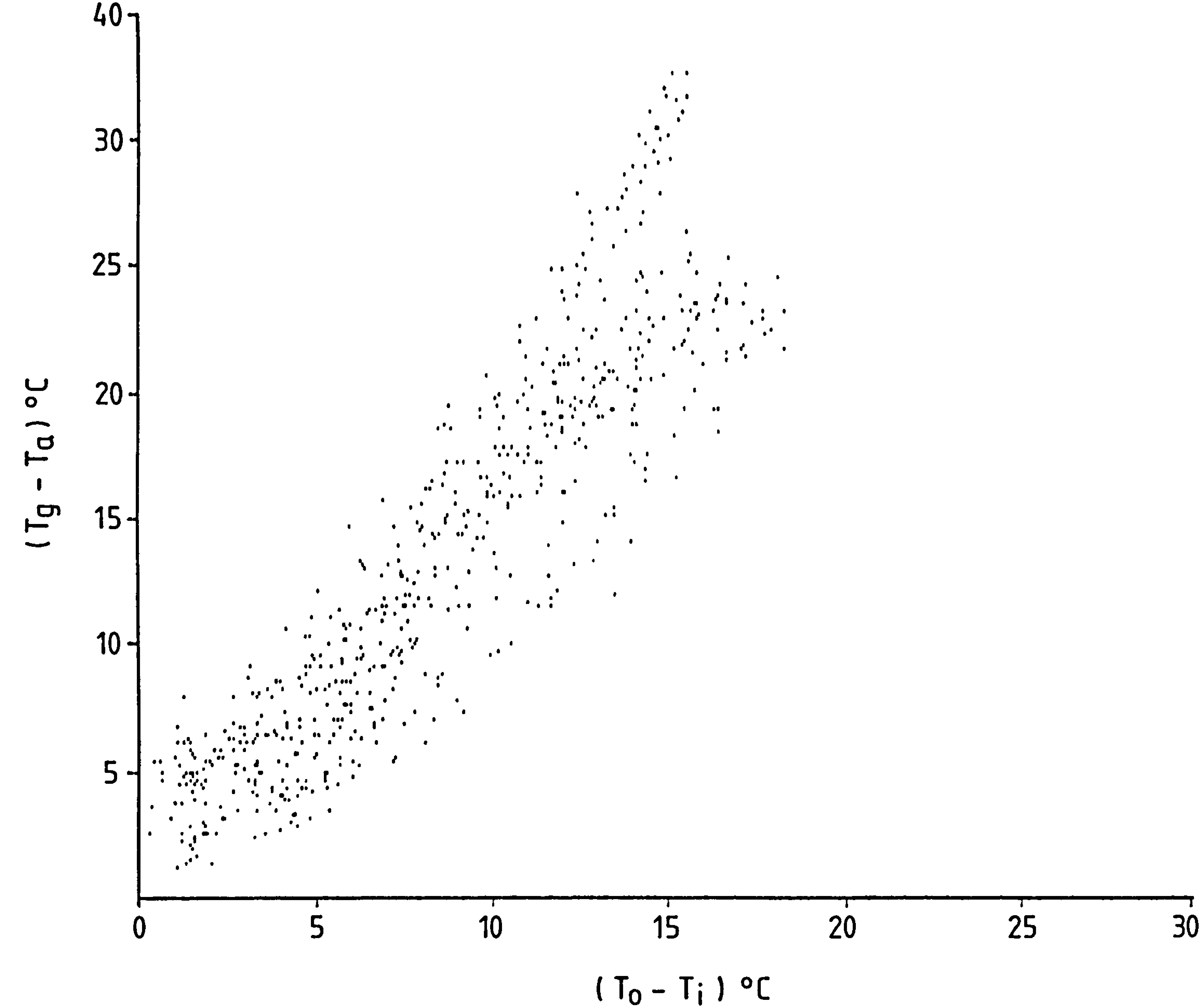


Fig.43 Variation of (T_g-T_a) with (T_o-T_i) 9 days data in the period November/December 1986

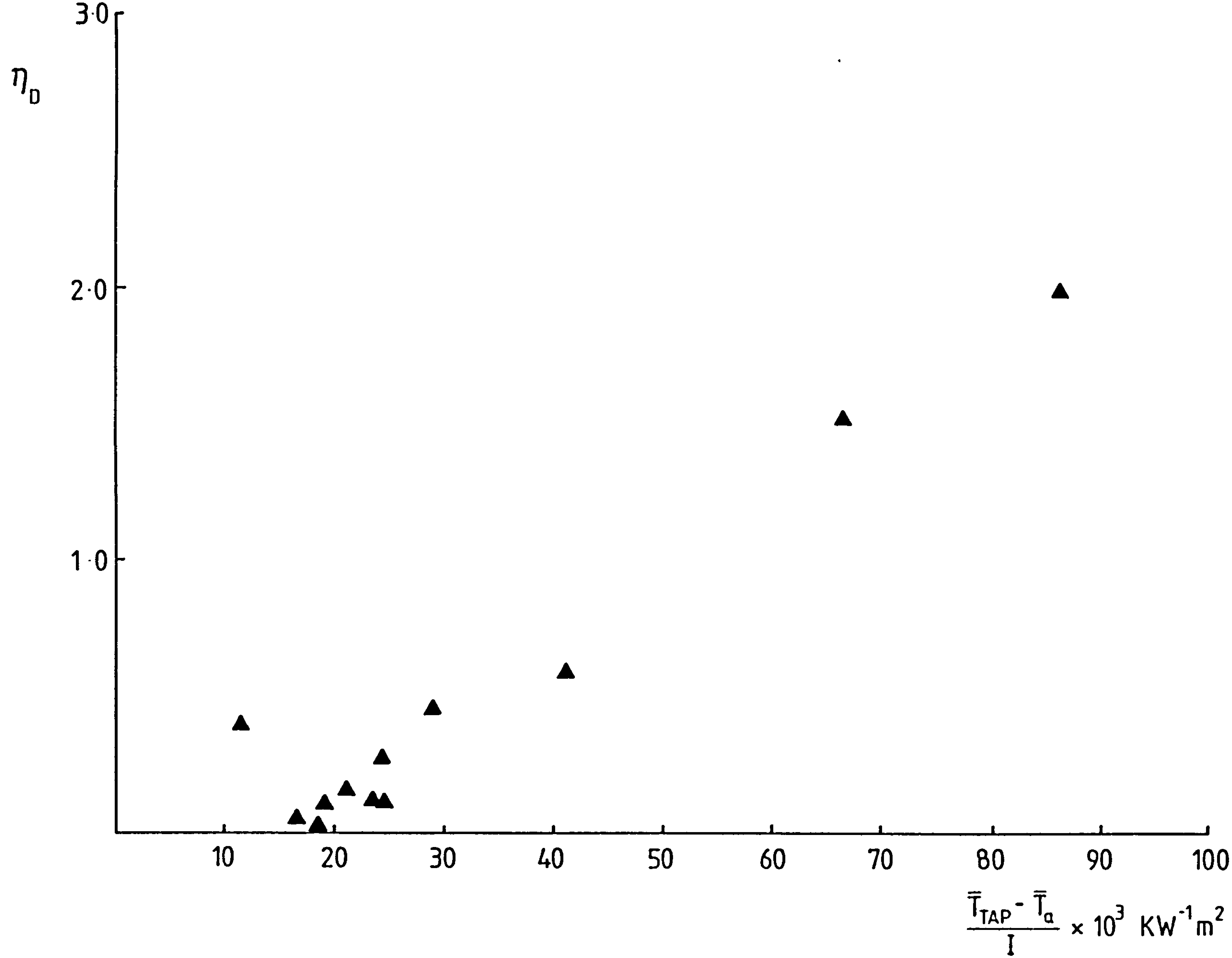


Fig.44 Variation ofddiurnal efficiency of TAP system with $(\bar{T}_{TAP}-\bar{T}_a)/\bar{I}\times 10^3$

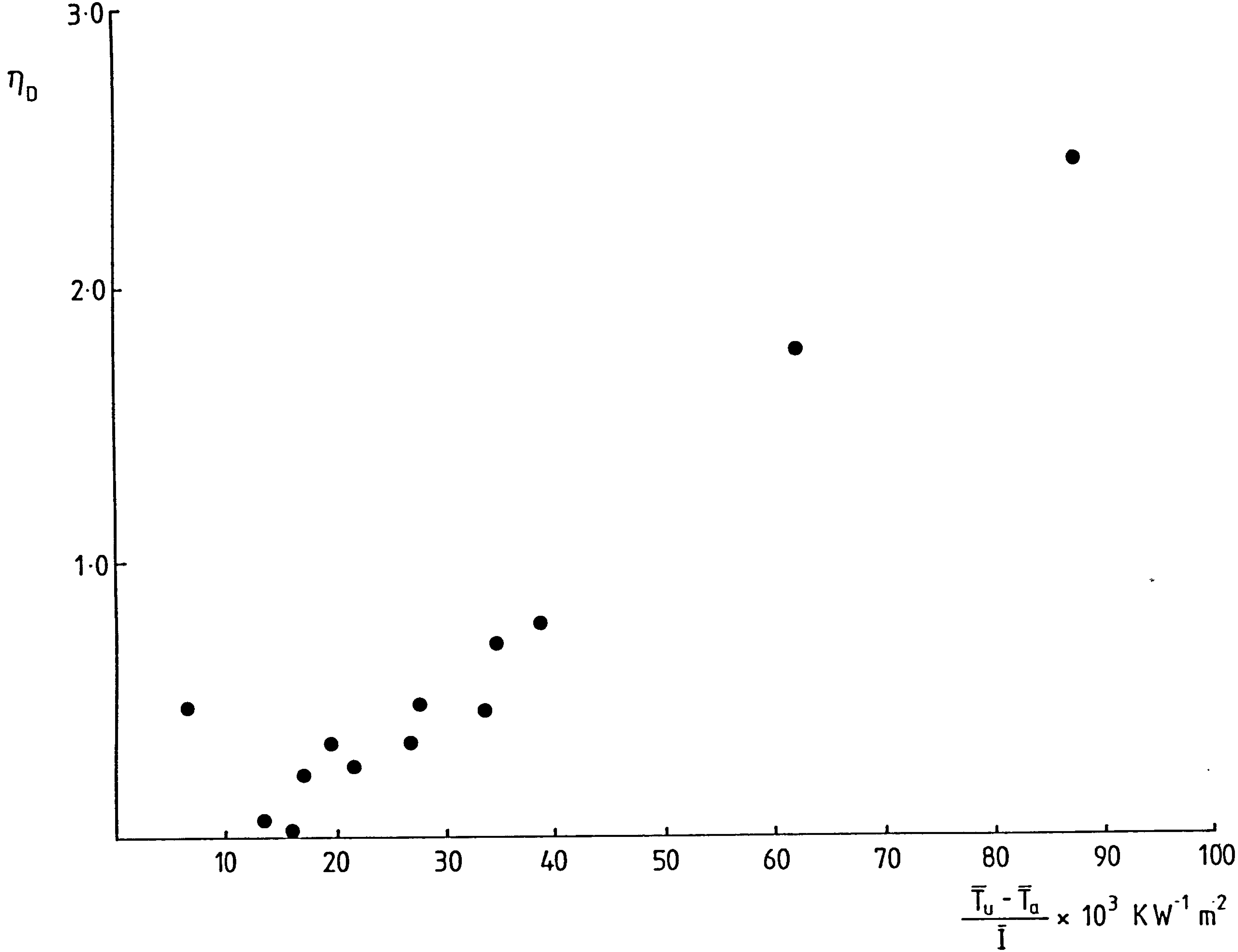


Fig.45 Variation of diurnal efficiency of U-Tube system with $(\bar{T}_U-\bar{T}_a)/\bar{I}\times 10^3$

$(T_i-T_a)/I$, may be compared with an equation of the form:

$$\eta = F_R\, A\, \tau\alpha - F_R\, A\, U_L \left(\frac{T_i - T_a}{I}\right) \qquad (53)$$

where the intercept at the η axis is $F_R\, A_c\, \tau\alpha$

and the slope of the line is $F_R\, A\, U_L$

If the values of collector area A and $\tau\alpha$ are known then

$$F_R = \frac{\text{intercept}}{\tau\alpha} \quad \text{and } U_L = \frac{-\text{slope}}{F_R}$$

F_R is a function of the mass flow rate (among other variables). Thus the mass flow rate may then be determined from the following equation [Duffie & Beckman (1980)], if a value for F′ is known.

$$F_R = \frac{\dot{m}\, C_p}{U_L\, A} \left[1 - \exp - \left(\frac{U_L\, A\, F'}{\dot{m}\, C_p} \right) \right] \qquad (54)$$

From Fig. (46) the values for slope and intercept are -2.57 and 0.394 respectively.

Assuming values of 0.8 for $\tau\alpha$ and 0.5 for F′, as before, the following may be obtained:

$$0.394 = F_R \times 1.2 \times 0.8$$

$$\therefore \quad F_R = 0.41$$

$$2.54 = 0.41 \times 1.2 \times U_L$$

$$\therefore \quad U_L = 5.16$$

$$\therefore \quad 0.41 = \frac{\dot{m} \times 1004}{5.16 \times 1.2} \left[1 - \exp \left(\frac{5.16 \times 1.2 \times 0.5}{\dot{m} \times 1004} \right) \right]$$

This equation is satisfied when,

$$\dot{m} = 0.008 \text{ kg/s}$$

Now $\dot{m} = \rho AV$

$$\therefore \quad V = \frac{0.008}{(0.1 \times 0.72) \times 1.2} = 0.092 \text{ ms}^{-1}$$

The linear distribution of the data points in fig. (46) indicates that the collector was operating with a steady mass flow-rate, in the range $0 < (T_i - T_a)/I < 0.03$. The increase in scatter with increasing $(T_i - T_a)/I$ can be attributed to variation in flow rate through the collector. This indicates that for any given system (i.e. collector and building) there may be a range of $(T_i - T_a)/I$ for which constant mass flow rate can be assumed. This would simplify considerably performance prediction and mathematical modelling of such a system.

It must be emphasised that the results obtained are only representative of the particular system under test and cannot be used in the evaluation of any other combination of collector and building. Furthermore, the accuracy of the Glazing Temperature Method of rating natural-circulation air-heating solar collectors has yet to be compared with that of other methods. Graphs of instantaneous efficiency against $(T_i - T_a)/I$ for a day in November and December 1986 are shown figs (48, 49).

In fig. (47) it can be seen that, as the insolation level increases, the instantaneous efficiency calculated using the glazing temperature method tends to a assymptotic value of F_R $\tau\alpha$ (i.e. 0.39) which is in good agreement with the value of F_R $\tau\alpha$ calculated from fig. (46), and it can also be seen from fig. (47) that a consistently-broad range of efficiencies ensued above a critical insolation level of approximately 200 Wm^{-2}.

Fig. (50) illustrates the temperature distribution in the test cells fitted with the glazing collector prototypes, and the temperature

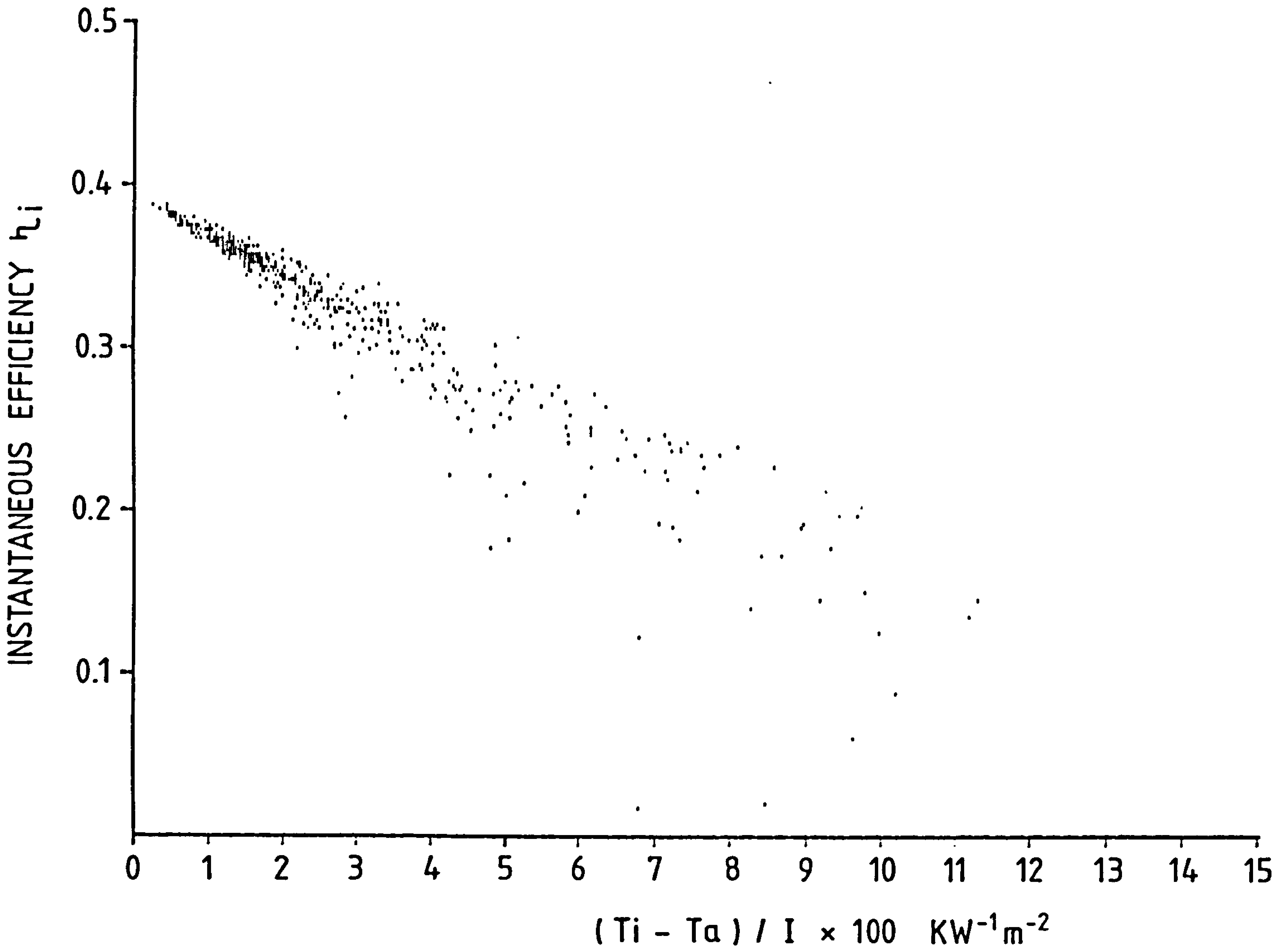


Fig.46 Cumulative graph of instantaneous efficiency against $(T_i-T_a)/I\times100$ for 9 days in November/December 1986

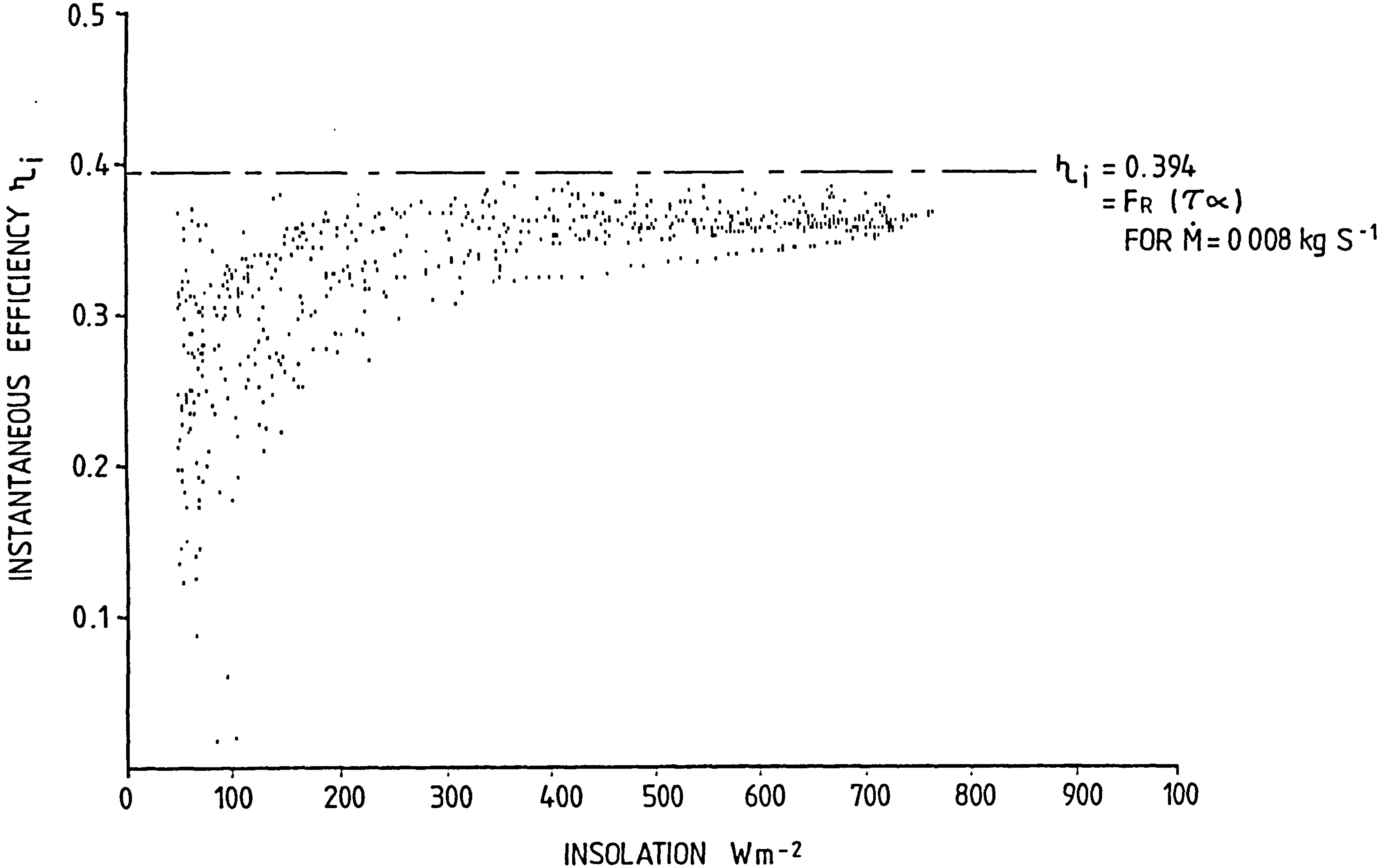


Fig.47 Cumulative graph of instantaneous efficiency against insolation 9 days in the period November/ December 1986

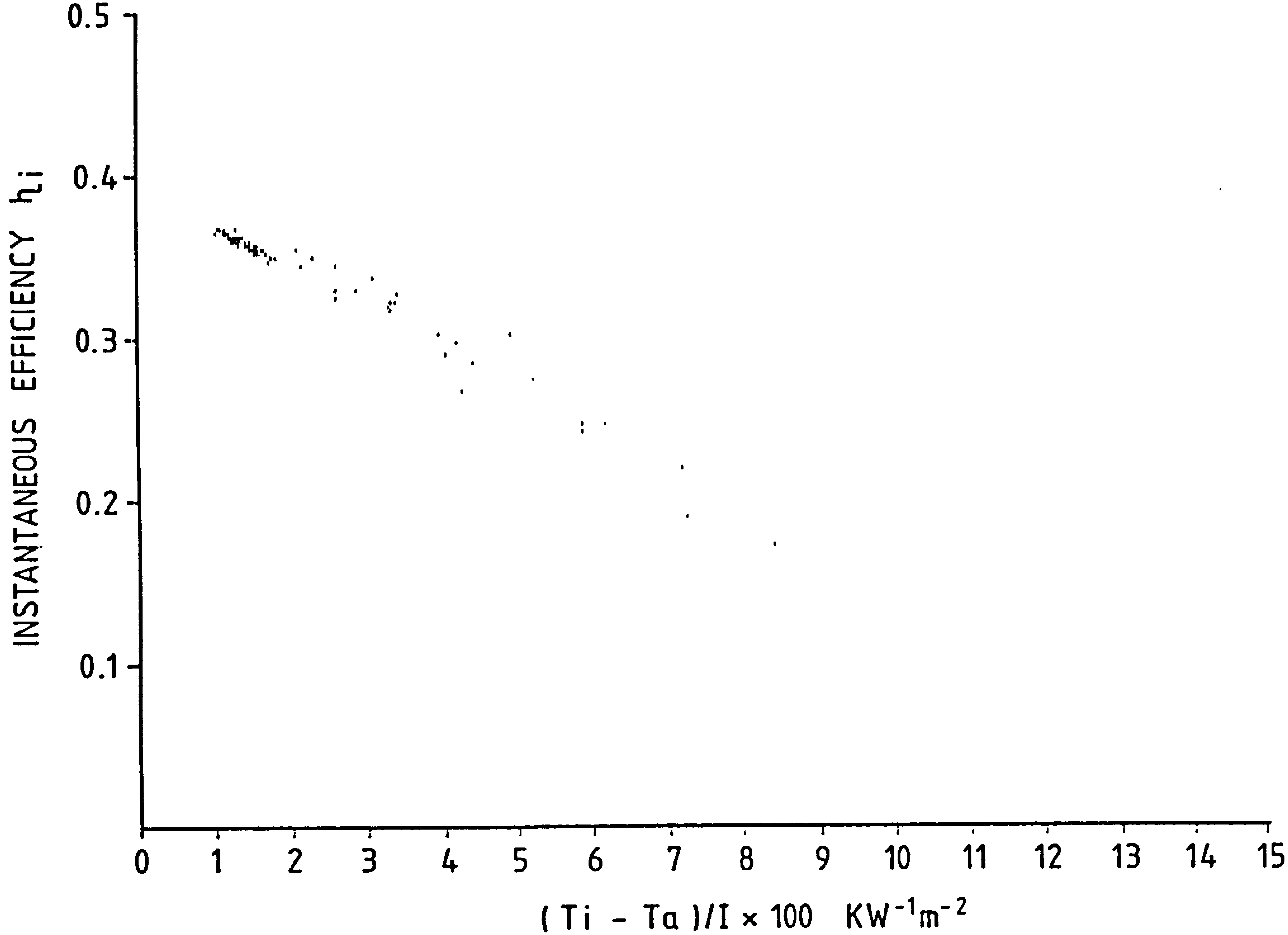


Fig.48 Instantaneous efficiency against $(T_i-T_a)/I\times100$ for glazing collector 27th November 1986

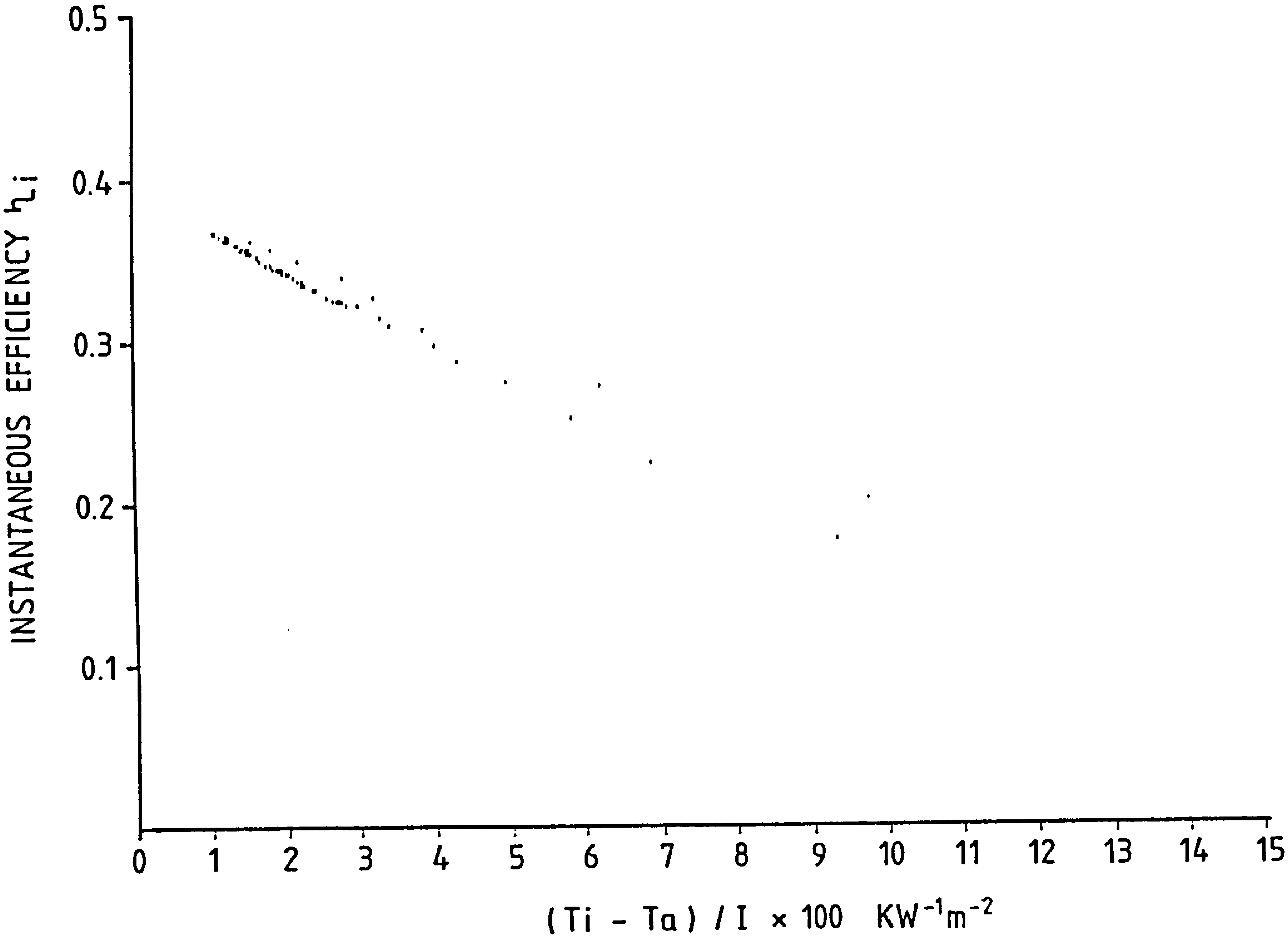
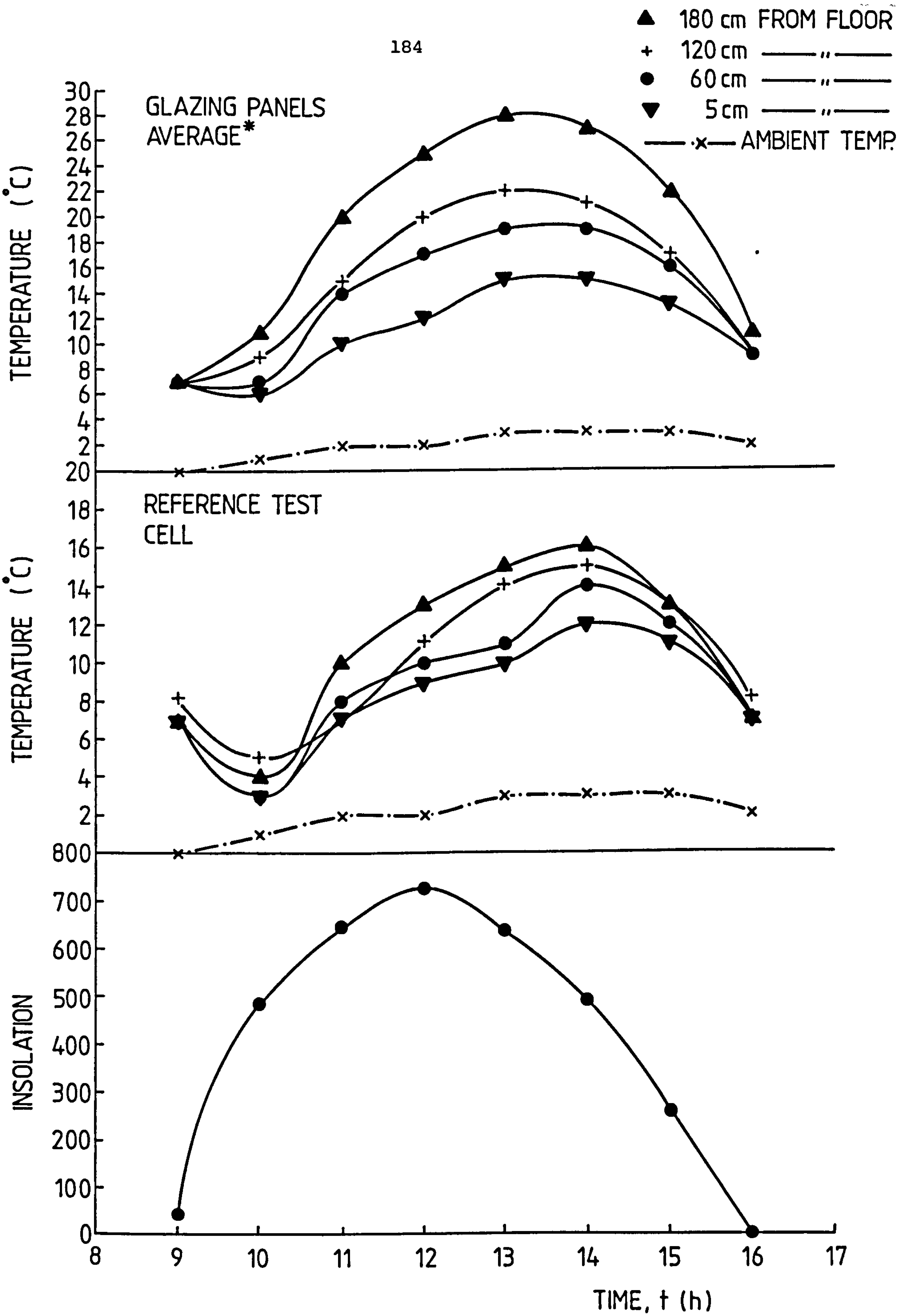


Fig.49 Instantaneous efficiency against $(T_i - T_a)/I \times 100$ for glazing collector 12th December 1986



* Average values of temperatues recorded in test cells 1 & 2

Fig.50 Temperature distribution in glazing collector test cells and distribution in the unheated test cell. It is evident that the temperatures in the solar-heated test cells were appreciably higher than those in the unheated test cell. However a large degree of thermal stratification also ensued in the solar heated test cells. The latter may not occur when thermosyphoning air-heating solar energy collectors are installed in real buildings. This may be attributed to:

i) the far larger ratio of volume to be heated to collector area may mean that the collector is not the dominant determinant of the air-flow pattern in the room,

ii) the effect of occupants,

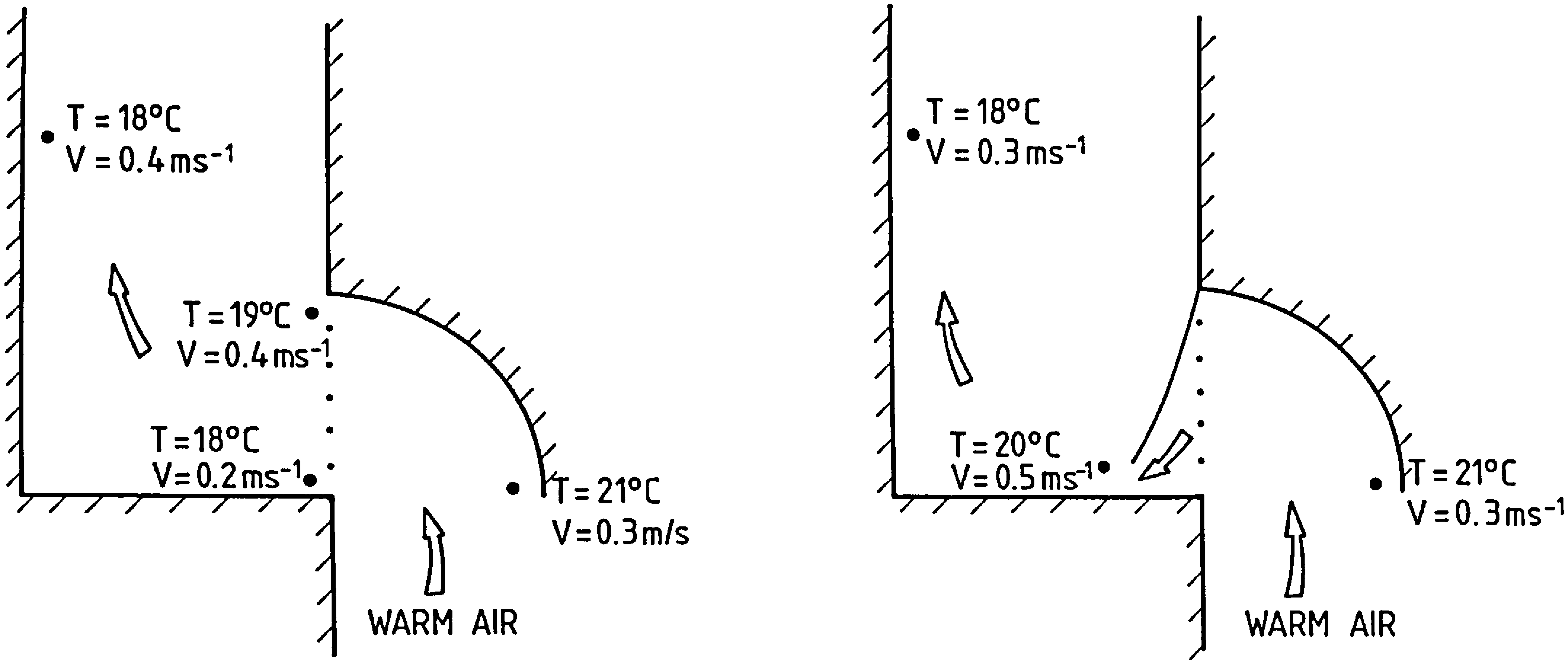
iii) the influence of ventilation and auxiliary heating equipment.

6.3 BACKDRAUGHT DAMPER TEST RIG RESULTS

The results obtained from the backdraught damper test rig were disappointing. The cold-plate temperature was consistently at 0°C, yet the down draught generated was not sufficient to exert any pressure on the Tedlar flap valve. It was not possible, therefore, to conduct tests to determine the extent to which a flap valve inhibits reverse-thermocirculation.

A number of tests were carried out to assess the extent to which a flap valve inhibits the flow of buoyant air. Figs. (51, 52) illustrate typical temperatures and velocities measured at a number of points within the test rig. The presence of a flap valve altered the flow pattern through the apparatus. It is, however, difficult to draw any firm conclusions from the results obtained: the accuracy of the tests was questionable because of the fluctuating flows generated and it is debatable whether warm air converted from a hot-plate can be considered to be a valid simulation of buoyant air movement within a solar energy air-heating collector.

Attempts were made to photograph the passage of smoke through the test rig in order to obtain some visual representation of the affect a flap-valve has an air movement. Unfortunately this too proved less



ROOM TEMPERATURE 17°C
HOT - PLATE TEMPERATURE 27°C

RIG WITHOUT FLAP - VALVE BACKDRAUGHT DAMPER

ROOM TEMPERATURE 17°C
HOT - PLATE TEMPERATURE 27°C

RIG WITH FLAP - VALVE BACKDRAUGHT DAMPER

Fig.51 Velocity and temperature distribution through backdraught damper test rig with and without flap valve

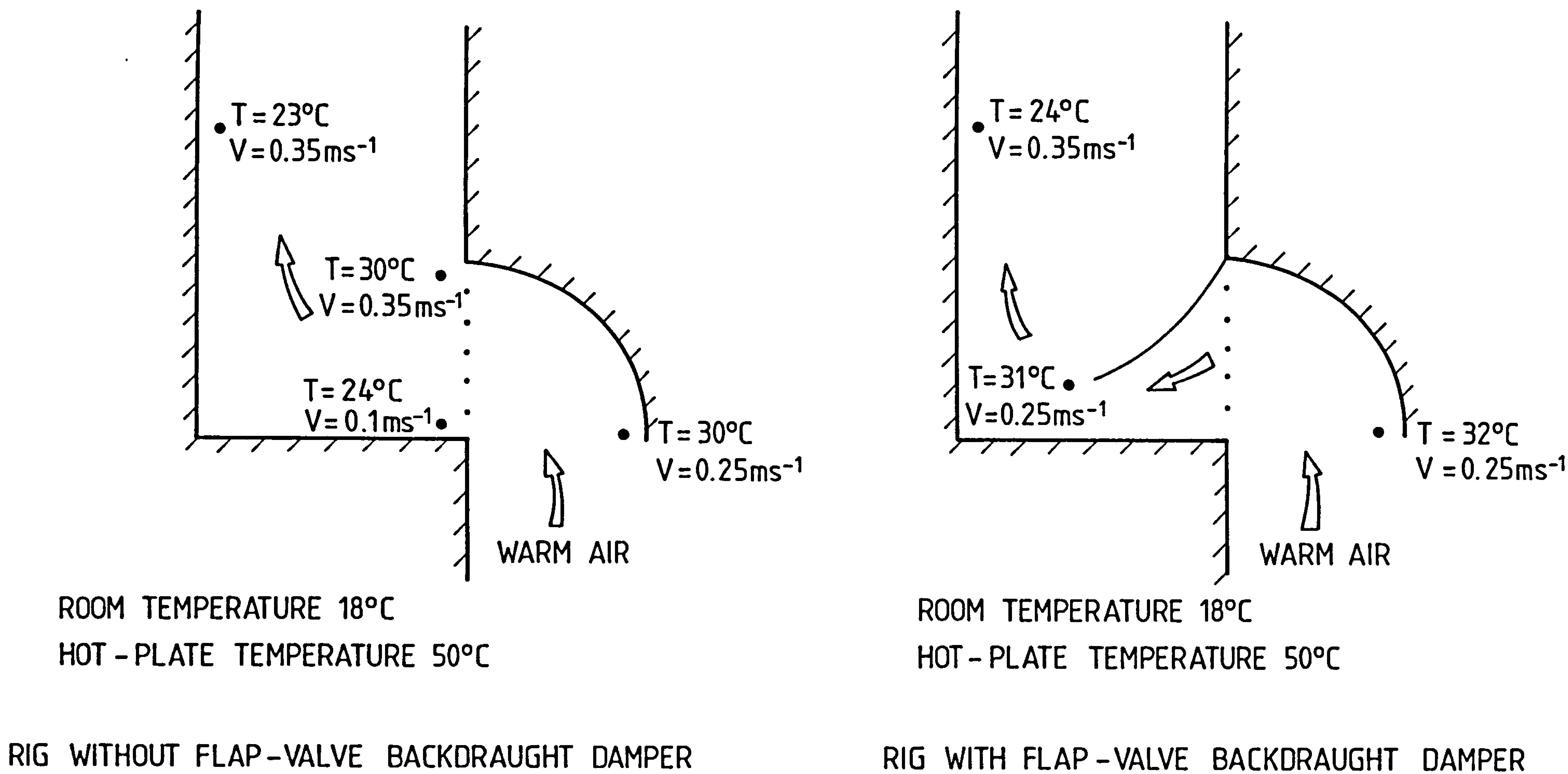


Fig.52 Velocity and temperature distribution through backdraught damper test rig with and without flap valve

than satisfactory Fig. (53). It would appear that a more advanced photographic technique is required to obtain satisfactory results.

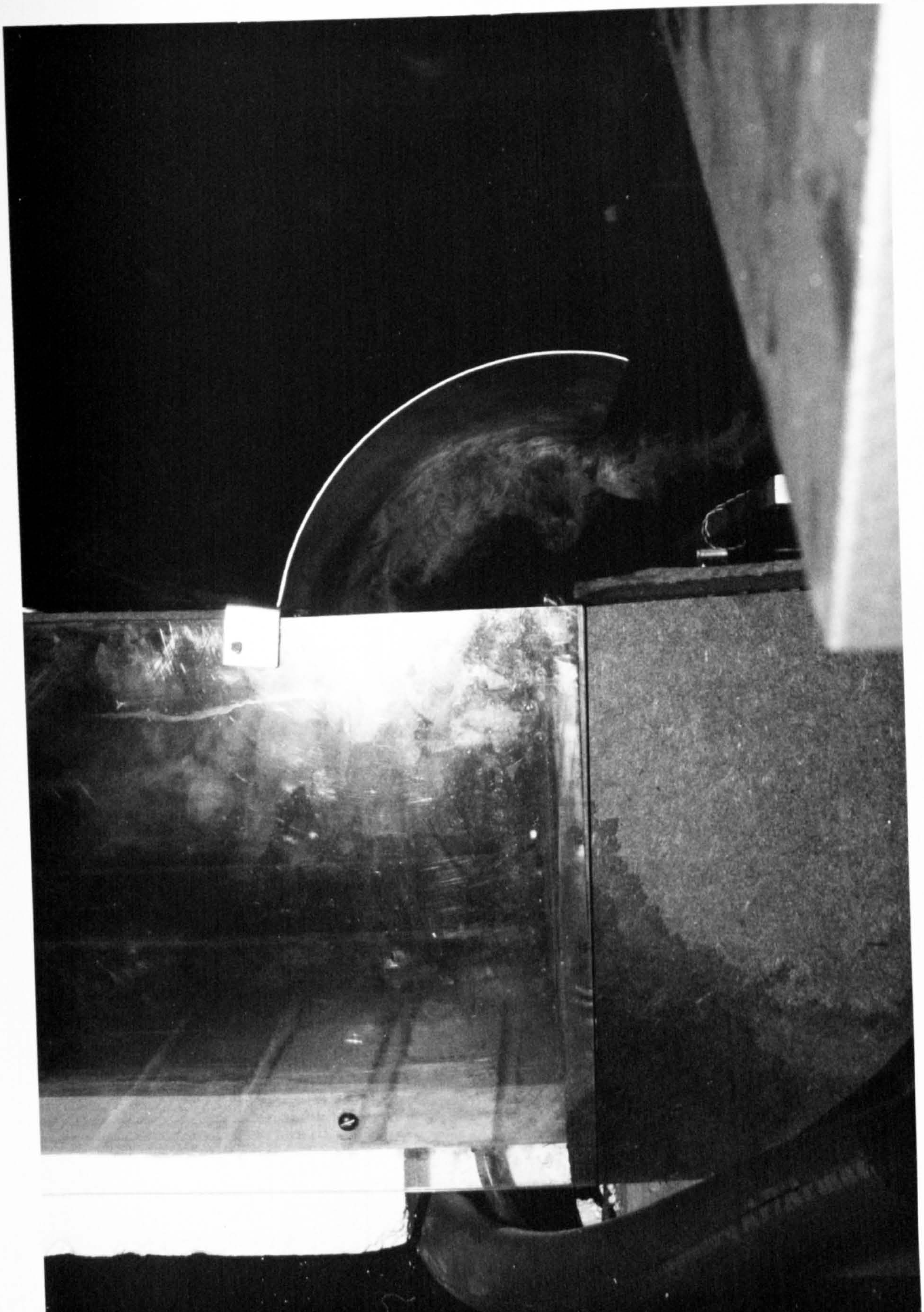
6.4 DISCUSSION

A large potential market for passive air-heating solar-energy collectors has been identified. In the UK, the curtain wall cladding systems used in many non-domestic buildings such as schools and offices are currently being refurbished or replaced. This expanding programme presents an excellent opportunity for the inclusion of passive air-heating solar-energy collectors at low marginal cost. Ageing conventional cladding panels could be replaced with units which incorporate both cladding and solar air heating elements.

Alternatively solar collectors of this type could be accommodated behind the overcladding systems being installed to protect buildings from further deterioration. Another method by which passive solar heating can be added to an existing building is to fit panels behind existing glazing to insulate and deliver a convective heat input. This is potentially the least expensive of the three options to implement as it does not have to be undertaken as part of a refurbishment programme and makes use of existing glazing. Retrofit passive air heating systems for industrial buildings have also been reviewed and would appear to be a very promising method of exploiting solar energy.

The efficiency of natural-circulation air-heaters is directly related to the performance of the buildings which they are heating. Collector efficiency is influenced by many factors: building dimensions orientation, thermal mass, ventilation rates, direct and indirect gains, occupancy patterns, existing heating systems etc., However, it is the manner in which the passive heating system is perceived and used by the people who work in these buildings which determines their success or failure. As has been noted, there will be little incentive for the occupants of non-domestic buildings to make sure that these devices operate efficiently.

Fig.53 Flow visualisation experiment using backdraught damper test rig



The conflicting design recommendations for thermosyphoning air panels have not been addressed satisfactorily. This will not be accomplished until a rigorous method of testing such devices is developed so that the performance of different collector designs can be compared directly. Components such as backdraught dampers will have to be thoroughly examined before any firm conclusions may be drawn as to their efficiency. Furthermore, until the response of the occupants of non-domestic buildings which are heated with thermosyphoning air panels has been elicited and assessed, there can be no guarantee that an efficient, economic device of this type will prove to be commercially viable.

The complex relationship which will exist between a building, its occupants and a natural circulation air heating system may prohibit the generalised accurate prediction of the output or efficiency of these devices once installed.

The collector evaluation carried out using test cells during the course of this work served not only to highlight the difficulties inherent in measuring accurately the performance of these devices, but also the difficulty of devising experiments which were representative. It may be possible to measure collector efficiency using an indoor test facility under closely controlled conditions, but such considerations as thermal comfort, occupant response, and interaction with existing heating would be almost impossible to simulate. It is unlikely that collectors of this type would be installed as the sole form of heating for a non-domestic building, therefore, it could be argued that it is only valid to test them in conjunction with some form of convective or radiant heat source and determine the impact of one on the other. One solution might be to measure the relative efficiencies of different collector designs with indoor test equipment, and progress from there to a more advanced form of the test cells used in this work to gain a better understanding of the interaction between a collector and the building to which it is fitted.

The test cells used in this study were intended to reproduce some of the conditions which may be encountered in light-weight non-domestic buildings: low thermal mass, low levels of insulation, and significant air infiltration. This approach contrasts with that of using test cells which feature high levels of insulation and strictly controlled air-change rates. Ideally, a test cell should have characteristics which place it in the shaded area of Fig. (54). Extrapolated results should be appropriate for use in the thermal design of "real" buildings and yet be sufficiently comprehensive for detailed simulation modelling techniques to be validated. A survey of passive solar test modules has been undertaken by Moore (1982).

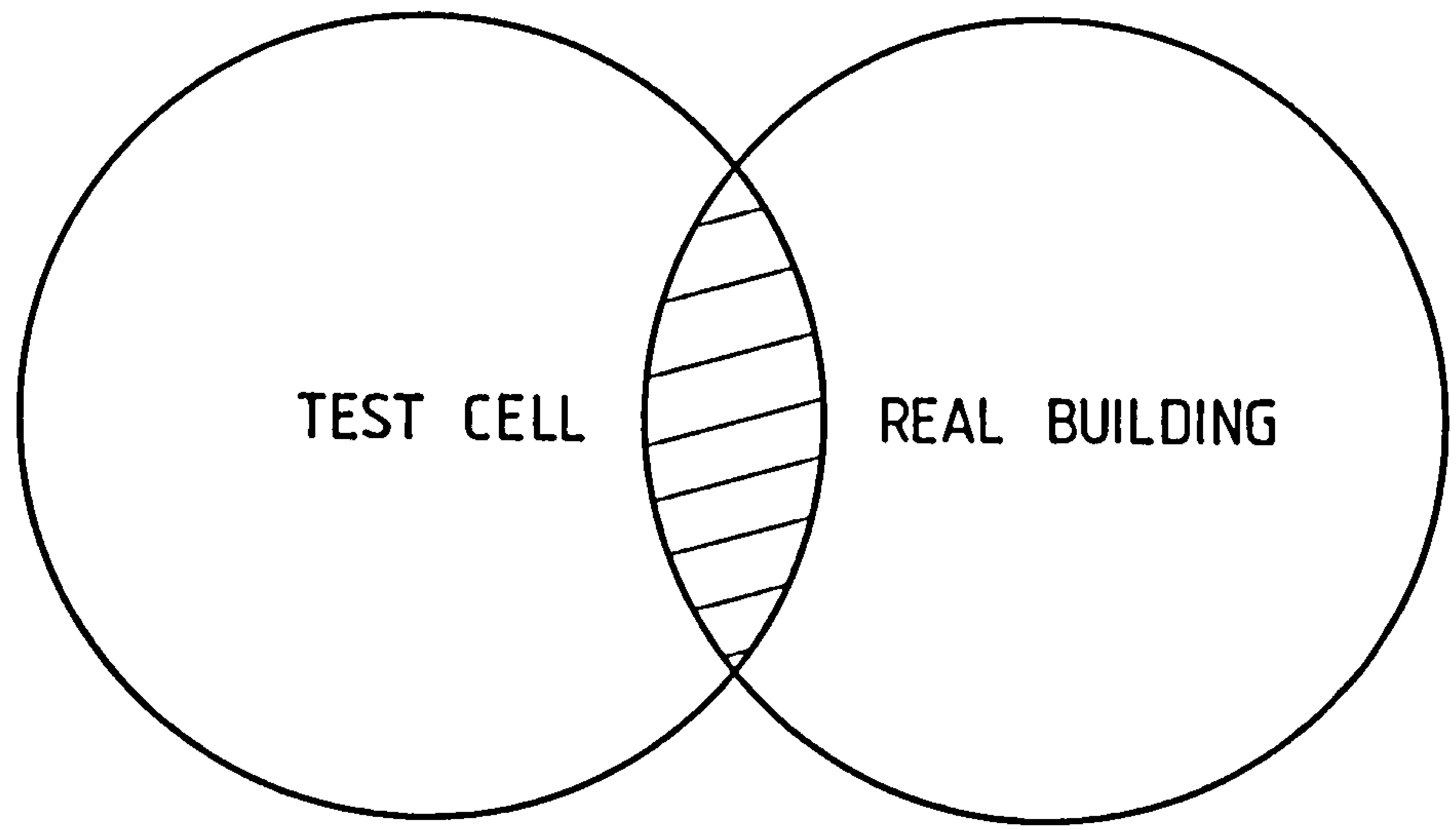
One method of evaluating the performance of solar-energy air-heating collectors of the type studied in this report is to undertake long-term real-time testing of the order of six month's to one year's duration. In view of the number of design considerations which had to be addressed during the course of this project, such testing could not be undertaken. There were a number of alternative methods by which the performance of the various collector designs could be measured. Each of these methods was found to have limitations; consequently the quality and quantity of the results obtained were variable. Analyses have been proposed in an attempt to overcome some of the difficulties encountered during testing, but it remains to be seen whether these will prove to be more satisfactory than the methods previously employed.

The relative merits of the U-Tube and "standard" TAP collector have been reviewed, and tests undertaken indicated that, although the "standard" TAP might be more efficient at generating heat than the U-Tube configuration, the disadvantage of having to rely on some form of damper to prevent reverse thermocirculation may outweight this superiority.

Air leakage around dampers, and the extent to which they impede air movement through a collector were investigated, but no conclusive results were obtained. However, it was apparent that high standards of manufacture, installation and maintenance are essential if they are to operate efficiently for any length of time.

KEY WORDS :

- SHORT - TERM
- HIGH LEVEL MONITORING
- CONTROLLED ENVIRONMENT
- SMALL SCALE
- FAVOURABLE ORIENTATION
- $\frac{\text{COST OF MONITORING}}{\text{COST OF TEST CELL}} \gg 1$



KEY WORDS :

- LONG TERM
- LOW LEVEL MONITORING
- ENVIRONMENT INFLUENCED BY OCCUPANTS
- FULL SIZE
- RANDOM ORIENTATION
- $\frac{\text{COST OF MONITORING}}{\text{COST OF REAL BUILDING}} < 1$

Fig.54 Venn Diagram illustrating the relationship between test cell and real building

The U-Tube collector produced greater floor-to-ceiling temperature gradients then the "standard" TAP but these were recorded in the absence of any other form of heating: thermal stratification of this type may, or may not, be significant when a U-Tube collector is used to supplement a main heating system.

Despite its lower efficiency a single-glazed non-venting U-Tube collector is the most simple and most "passive" of all of the designs of natural circulation air-heating collectors. There is no requirement for moving parts other than those controlling the output of warm air from the collector. A backdraught damper is not required either to stop reverse-thermocirculation or prevent vented air from being blown back down the collector. This overcomes the problems of leakage, reliability and restriction of air-flow which are potential problems when backdraught dampers are installed.

It is important to develop a control strategy which involves the minimum amount of manual operation as there is no guarantee that the occupants of a work place will respond to the demands of a passive programme.

Summertime operation of these devices has proven to be an area which has not received a great deal of attention yet they need to be operated in such a manner that the collectors and the building are not damaged, and a source of discomfort within the building is not created.

The installation of retrofit passive solar-energy air-heating collectors may result in a reduction of the total glazing area of a building and, in consequence, a reduction in the level of natural illumination. It would be unfortunate if the value of the heat supplied by the retrofit collectors was less than the energy required to satisfy an increased demand for artificial illumination. The relationship between glazing area, the energy consumed in providing artificial illumination and the overall energy-balance of a building must be considered if collectors are to replace glazing.

If glazing area is reduced, the requirement for ventilation and cooling of the building may be reduced. The value of any ventilation provided by collectors will be difficult to assess. It may be that in many instances, TAPs are only vented to prevent overheating of the components, rather than overheating of a building. However, it would seem prudent to build collectors to withstand high temperatures so that in the event of the collector being left to stagnate inadvertently, neither the collector nor the building will risk damage.

The glazing collector prototypes were constructed with summer-venting grilles which were permanently open to the ambient environment. In cold weather this arrangement proved to be less than satisfactory: cold ambient air entered the collectors via the grilles and cooled the hinged air deflectors to an unacceptable degree. Gusts of wind forced this cold air around the edges of the air deflector and into the top of the air-flow channel. For the duraction of the heating tests these ventilating grilles were sealed with polythene sheet.

If a practical collector of this type is to be fitted with a permanently open ventilating grille, a high standard of draught-proofing will be required around the edges of the hinged air deflector which will also have to be insulated to prevent cooling of the air flow. A manually or thermally-actuated grille would appear to be a more satisfactory solution.

It was determined that collectors which featured double glazing would require some form of venting or shading to prevent overheating during the periods when the heat input to the building is not required. Design proposals for a manually, or thermally actuated device which changed the collector from heating to a ventilating mode were made.

The use of parallel-duct absorber plates was also proposed to improve heat transfer by creating symmetrical heating of the air flow.

The Back-Pass absorber configuration was considerd to be the most practical of the choices available, obviating the need for regular cleaning of the glazing interior surface whilst reducing the likelihood of air leakage adversely affecting the air flow through the collector.

Chapter 7

CONCLUSIONS

7.1 CONCLUSIONS

- Four passive solar-energy air-heating retrofit options have been identified which are potentially viable economically.

- It is not possible to propose a single design of passive solar-energy air-heating collector that will satisfy the requirements of all four of these options.

- The Back-Pass or rear-duct arrangement is recommended because of the low inherent maintenance requirement, rather than heat transfer considerations.

- The air-space between the absorber and glazing in a Back-Pass collector should be sealed so that neither cold ambient air, nor warm interior air can infiltrate and reduce collector efficiency.

- Ideally duct entrances and exits should have the same cross section as the air-flow channel.

- The optimal depth of the air channel will depend on the levels of insolation and ambient temperature anticipated. It will also depend on the collector height, the type of absorber used, the number of glazing covers, any anticipated restrictions in air-flow, and whether selective surfaces are to be used.

- The internal surfaces of a Back-Pass duct should have a high emittance.

- Curved air-deflectors should be incorporated in collectors to reduce friction losses at the entrance and exit.

- If a collector is to be used to provide summer-venting, some means of redirecting air flow at the top of the collector is required. A vent to ambient and a one-way valve to prevent warm air being blown back down the collector and into the living space are required.

- If summer-venting and collector shading are not practicable options, the collector will have to be built so as to withstand elevated stagnation temperatures.

- Ideally all collector controls should be totally passive and thus not require any commitment from the occupants of non-domestic buildings.

- Where possible collectors should be vandal-proof.

- The 'standard' vertical thermosyphoning air panel is limited by the performance of the dampers used to prevent reverse-thermo-circulation. Manual dampers may not be suitable for many applications and thermally-activated, or flap-valve backdraught dampers have not been developed on a large scale.

- The U-Tube thermosyphoning air panel is potentially self-damping, but may be twice the depth of an equivalent 'standard' TAP and, hence, more difficult to retrofit.

- The evaluation of natural-circulation air-heating systems is not straightforward. Many variables, such as the natural-circulation air velocity, are difficult to measure to the degree of accuracy required for any rigorous analysis.

- These devices interact with the building to which they are fitted and its occupants, thus their performance is difficult to predict accurately.

- A combination of indoor and outdoor testing over a period of months, or even years would appear to be the best method of assessing the merits of various designs.

- Three novel design proposals have been presented:

  i) a hinged air deflector for conversion from the heating to the ventilating mode.

ii) an absorber which consists of parallel ducts to increase heat transfer to the air whilst heating it symmetrically.

iii) a collector constructed with the insulation fitted outside, rather than inside, so that the metal body of the collector may provide more symmetrical heating of the air flow than the conventional arrangement.

- A method of determining instantaneous collector efficiency based on the measurement of glazing temperature, inlet and outlet air temperature, ambient temperature, and insolation has been developed.

- A method of determining diurnal collector efficiency based on the calculation of Solar/Load Ratios has also been developed.

- The reaction of occupants to natural-circulation air-heating systems will largely determine their success or failure as auxiliary heating systems; considerations such as thermal comfort, reduction of glazing area, control of heat output, etc., will only become apparent once the collectors have been installed.

- Factory-built backdraught damper units should be installed so as to guarantee a high standard of manufacture and performance. Such units should be designed so that removal for repair and maintenance is straightforward.

7.2 RECOMMENDATIONS FOR FURTHER WORK

It is evident that further extensive research and development of passive solar-energy air-heating systems is required. Some of the major challenges remaining are:

1. Development and validation of detailed, comprehensive, transient models which describe the performance of these devices.
2. Development of internationally-agreed rating test procedures which can be used to rate these devices.

3. Establishment of clear guidelines for the selection of components, collector dimensions, and construction and installation techniques which will produce an economic and efficient unit.
4. Investigation of the relative merits of "passive" control devices such as flap-valve backdraught dampers, thermally actuated louvres, etc.
5. Assessment of the response of occupants of non-dometic buildings to natural-circulation air-heating systems.
6. Determine whether the provision of a ventilation mode is justified economically, and whether stagnation or shading are better options during the summer months.
7. Determine the merits, or otherwise, of double glazing, selective surface materials, photovoltaics etc., with respect to the enhancement of collector performance.
8. Long-term monitoring of retrofit passive solar energy air-heating systems in non-domestic buildings.
9. Further development and validation of the Glazing Temperature Method of measuring the instantaneous efficiency of passive and active solar-energy air-heating collectors.

7.3 PROPOSED COLLECTOR DESIGN

A single glazed, Back-Pass, U-Tube collector would appear to be the most suitable retrofit design. If left to stagnate, rather than provide ventilation during the summer months, such a unit would be closest to the requirement of a low maintenance, totally "passive" collector. This presupposes that some form of thermally-actuated heat-output control could be developed. As U-tube collectors are self-damping there would be no requirement for backdraught dampers, whilst single glazing precludes elevated stagnation temperatures. This should ensure that collector components will not be subject to severe thermal stresses. The potential disadvantages of such a design include collector depth and thermal stratification, however, the performance of the collector would not be limited by the necessity of fitting a backdraught damper which is the case wth "standard" TAPs. A proposed single glazed, Back-Pass, U-Tube cladding collector is shown Fig.(55).

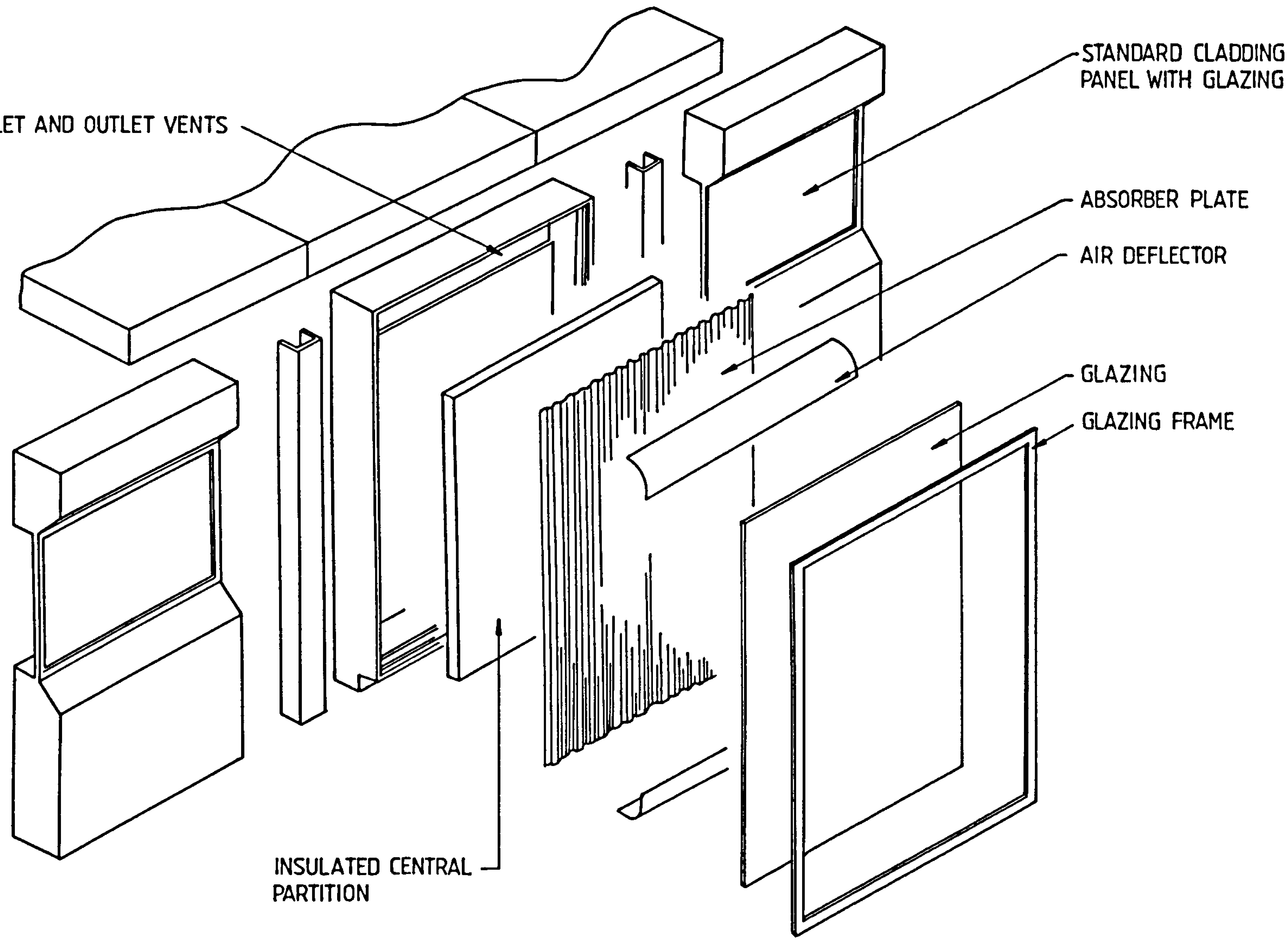


Fig.55 Proposed passive air-heating solar-energy cladding collector design

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APPENDIX A
SAMPLE DIURNAL VARIATIONS OF RECORDED TEMPERATURES

Pages 219 to 223 show sample diurnal variations of recorded temperatures.

THE KEY FOR THESE TABULATED RESULTS IS:

Temperatures

T_1	60 cm from floor of test cell.	TEST CELL 1
T_2	Collector outlet*	
T_3	120 cm from floor of test cell.	
T_4	5 cm from floor of test cell.	
T_5	180 cm from floor of test cell.	
T_6	Top of absorber	
T_7	Collector outlet*	
T_8	Collector inlet*	
T_9	Collector inlet*	
T_{10}	Collector outlet*	TEST CELL 2
T_{11}	180 cm from floor of test cell.	
T_{12}	120 cm from floor of test cell.	
T_{13}	5 cm from floor of test cell.	
T_{14}	Collector inlet*	
T_{15}	Collector outlet*	
T_{16}	60 cm from floor of test cell.	
T_{17}	Top of absorber	
T_{18}	Collector inlet*	
T_{19}	Ambient dry-bulb	

* On horizontal centre-line of duct, 15cm from centre.

Insolation

I, Total insolation on south-facing vertical plane.

TIME (GMT)	I (Wm^{-2})	TEST CELL 1 U-TUBE COLLECTOR T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	TEST CELL 2 STANDARD TAP T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.30	79	17	8	7	10	16	15	10	9	8	15	11	10	8	8	13	9	15	8	6
9.30	117	21	11	9	14	24	21	13	12	10	20	13	12	9	10	18	10	22	10	7
10.30	148	28	14	11	19	35	27	16	16	12	26	17	14	12	12	24	13	32	12	8
11.30	178	31	15	11	21	41	30	18	18	13	28	17	16	12	13	27	14	38	13	9
12.30	186	28	14	11	21	40	28	17	17	13	26	18	14	12	13	26	13	37	13	9
13.30	175	23	13	11	19	33	24	16	14	12	22	16	13	11	12	23	13	31	12	9
14.30	139	19	11	10	14	25	18	13	12	10	18	14	11	10	10	18	11	25	11	8
15.30	139	22	13	11	18	33	23	15	15	12	22	15	13	12	14	23	13	32	12	9
16.30	104	15	10	10	13	18	15	12	11	10	14	11	10	10	10	14	10	17	10	8
17.30	57	11	8	9	9	12	11	10	10	9	9	8	8	8	8	9	8	10	8	8

23rd MARCH 1986

TIME (GMT)	I (Wm^{-2})	TEST CELL 1 U-TUBE COLLECTOR T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	TEST CELL 2 STANDARD TAP T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.00	112	26	27	14	18	16	27	24	20	24	24	18	26	14	13	23	22	18	13	10
9.00	178	20	21	14	18	20	22	21	20	20	20	17	15	14	14	20	15	22	14	11
10.00	203	19	15	12	15	22	19	17	18	15	26	18	26	16	16	25	23	23	16	12
11.00	93	20	19	14	17	22	21	21	20	18	29	19	28	16	16	26	27	24	16	11
12.00	139	20	17	14	17	23	20	18	18	16	23	19	25	16	16	24	23	24	16	11
13.00	216	21	17	14	13	25	21	19	18	16	23	18	17	15	15	23	16	25	16	11
14.00	120	19	15	14	17	21	19	17	16	15	20	17	16	14	15	20	15	22	15	11
15.00	126	19	16	14	15	22	19	18	17	16	27	20	29	18	17	27	24	24	17	11
16.00	123	19	16	14	17	22	19	17	17	16	27	20	28	17	17	26	24	24	17	12
17.00	161	19	15	14	17	24	20	16	15	15	22	17	16	15	15	22	15	25	15	12

7th MAY 1986 THERMOSTAT 20°C

TIME (GMT)	I (Wm^{-2})	TEST CELL 1 U-TUBE COLLECTOR T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	TEST CELL 2 STANDARD TAP T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.00	90	26	26	13	17	15	25	22	19	24	24	14	23	13	12	22	20	16	12	10
9.00	123	17	15	12	15	19	17	16	16	15	19	15	15	14	14	19	15	22	14	11
10.00	131	22	21	14	18	21	23	22	21	20	19	14	14	14	14	19	14	22	14	11
11.00	139	20	17	13	18	23	20	18	18	17	22	17	16	15	16	22	16	25	16	12
12.00	153	20	16	14	18	23	20	17	17	16	23	17	17	16	16	22	17	25	16	12
13.00	362	29	18	15	23	36	29	20	20	18	32	22	19	18	18	31	19	37	18	14
14.00	326	28	19	16	24	36	28	22	20	18	31	22	27	18	18	30	25	36	18	14
15.00	126	24	19	15	21	28	24	21	20	18	24	16	15	15	16	24	15	27	16	12
16.00	408	26	18	15	23	35	26	20	20	17	29	19	19	18	18	31	18	36	18	14
17.00	120	20	15	14	18	22	19	17	17	14	20	17	15	14	15	20	15	23	15	10

8th MAY 1986 THERMOSTAT 20°C

TIME (GMT)	I (Wm^{-2})	TEST CELL 1 U-TUBE COLLECTOR T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	TEST CELL 2 STANDARD TAP T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
7.00	43	9	9	8	9	9	9	9	9	9	9	9	9	9	9	9	9	9	8	9
8.00	65	24	25	12	15	12	23	18	12	22	13	10	14	10	10	12	10	12	10	10
9.00	54	14	15	12	14	14	15	15	14	15	17	15	16	14	13	17	16	19	13	10
10.00	85	16	16	13	15	16	17	16	16	16	18	15	15	14	14	17	15	20	13	11
11.00	87	17	16	13	15	18	17	17	17	16	18	15	15	14	14	18	14	20	14	11
12.00	233	24	18	15	20	28	24	19	19	18	26	18	17	16	16	25	16	29	16	14
13.00	120	20	16	14	19	23	20	18	17	16	21	16	15	15	16	21	15	23	16	13
14.00	222	23	18	15	21	28	23	19	19	17	25	18	17	16	16	25	16	28	16	14
15.00	222	24	18	16	22	31	24	19	19	17	27	21	27	19	18	27	25	32	18	14
16.00	115	22	17	15	20	25	22	19	18	17	24	18	17	17	17	24	17	27	17	14
17.00	161	21	17	15	19	25	21	18	17	16	23	18	17	16	17	23	17	26	17	15

9th MAY 1986 THERMOSTAT 20°C

TIME (GMT)	I (Wm^{-2})	TEST CELL 1 U-TUBE COLLECTOR T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	TEST CELL 2 STANDARD TAP T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
7.00	74	11	9	9	9	12	11	10	10	10	11	9	10	10	10	11	10	12	9	10
8.00	150	29	29	15	20	18	29	27	23	26	25	18	27	16	15	25	24	21	15	11
9.00	189	18	15	13	16	21	18	16	16	15	25	19	26	18	17	25	24	24	16	12
10.00	161	24	20	14	20	27	24	23	22	20	25	18	18	16	17	24	17	28	17	12
11.00	194	25	25	15	21	28	25	24	26	23	27	18	19	17	17	25	18	29	17	13
12.00	139	22	18	15	20	26	23	19	19	18	25	21	26	20	19	26	25	27	19	13
13.00	529	33	20	16	26	42	33	22	24	19	36	21	20	18	19	35	19	42	20	15
14.00	496	30	19	17	26	40	31	22	22	19	33	21	19	18	19	33	19	39	19	15
15.00	326	27	19	16	23	35	28	21	21	18	30	19	21	17	18	29	21	35	19	15
16.00	301	28	21	17	25	36	29	23	22	20	30	21	19	18	19	31	19	36	20	15
17.00	96	20	15	15	18	22	20	18	16	15	20	16	15	15	15	20	15	22	16	13

13th MAY 1986 THERMOSTAT 20°C

TIME (GMT)	I (Wm^{-2})	TEST CELL 1 U-TUBE COLLECTOR T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	TEST CELL 2 STANDARD TAP T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.00	109	27	27	13	17	17	26	25	20	24	24	16	24	14	13	23	22	19	13	11
9.00	244	21	17	13	18	24	21	18	18	17	23	17	16	15	15	22	16	25	15	15
10.00	422	26	22	15	21	27	26	24	23	20	26	19	18	16	17	24	17	27	17	17
11.00	219	27	20	14	20	29	25	21	24	18	27	19	20	18	19	26	19	30	18	14
12.00	436	29	18	15	23	35	29	21	21	18	31	20	19	17	18	31	18	36	19	15
13.00	348	29	19	16	24	38	30	22	20	18	33	22	19	18	19	32	19	38	19	15
14.00	375	29	19	17	24	38	29	22	21	19	32	24	20	18	19	33	19	38	19	15
15.00	433	29	19	17	24	38	29	22	21	19	33	20	19	18	19	33	18	39	20	15
16.00	74	20	16	15	19	23	20	18	19	16	24	20	24	19	18	24	22	24	18	11
17.00	107	16	13	13	14	19	16	14	14	13	16	12	12	12	12	16	12	18	13	11

14th MAY 1986 THERMOSTAT 20°C

TEST CELL 1 U-TUBE COLLECTOR — TEST CELL 2 STANDARD TAP

TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.00	178	24	24	11	15	14	23	22	17	22	20	12	19	10	10	18	17	15	10	11
9.00	263	19	16	12	18	21	19	18	17	16	20	16	14	13	13	20	14	23	13	10
10.00	301	23	17	13	20	26	23	19	19	17	24	18	21	14	15	23	17	27	15	12
11.00	235	23	17	14	20	28	23	19	18	17	26	20	18	16	16	25	17	29	16	11
12.00	87	24	25	14	19	24	26	25	24	23	22	19	21	17	17	22	19	26	17	9
13.00	573	32	19	15	27	42	33	22	23	18	36	26	18	16	18	35	18	41	18	12
14.00	392	26	18	14	21	33	26	20	20	17	29	21	20	16	16	28	17	33	17	9
15.00	507	30	21	16	27	41	31	23	22	20	35	27	20	18	18	35	19	41	19	12
16.00	219	23	17	15	21	29	24	19	19	17	27	21	24	16	17	26	21	30	17	11
17.00	301	22	16	14	22	28	23	19	18	16	26	22	18	16	17	27	17	31	17	12

15th MAY 1986 THERMOSTAT 20°C

TEST CELL 1 U-TUBE COLLECTOR — TEST CELL 2 STANDARD TAP

TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.00	186	24	24	10	15	13	22	19	16	21	12	13	12	10	9	13	12	15	9	20
9.00	310	23	23	12	23	22	25	23	22	20	22	24	20	16	15	22	18	25	15	21
10.00	425	29	20	14	28	30	29	23	24	18	29	29	23	18	18	27	20	30	18	20
11.00	370	35	22	16	32	39	35	27	29	20	36	33	24	20	20	34	22	38	20	15
12.00	655	41	26	18	38	49	42	31	33	23	44	39	28	22	23	42	25	48	23	16
13.00	222	36	24	18	34	42	36	29	30	22	36	35	25	21	21	36	23	40	21	15
14.00	644	44	27	20	41	54	44	33	35	25	45	43	29	23	24	45	26	52	24	18
15.00	540	42	27	20	40	53	43	33	33	25	45	41	29	24	25	45	27	52	25	17
16.00	192	33	23	19	32	40	34	29	29	22	35	32	24	21	22	35	23	39	22	16
17.00	452	31	23	19	32	36	31	28	28	21	32	32	24	21	21	32	22	36	21	17

16th MAY 1986 THERMOSTAT 20°C

TEST CELL 1 U-TUBE COLLECTOR — TEST CELL 2 STANDARD TAP

TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.00	63	23	24	14	16	14	25	15	13	22	19	14	23	13	12	20	18	16	11	11
9.00	74	13	12	12	12	14	13	13	13	12	15	13	13	13	12	14	14	17	12	11
10.00	76	14	12	12	12	16	14	13	13	13	17	15	18	16	14	18	17	19	13	11
11.00	96	14	13	12	13	16	15	14	13	13	16	12	12	12	12	15	12	17	12	11
12.00	87	14	14	13	13	16	15	15	14	14	16	13	13	13	13	15	13	17	12	11
13.00	104	16	13	13	14	18	15	13	13	13	17	13	12	13	12	17	13	19	12	12
14.00	107	16	13	13	13	18	16	13	14	13	17	12	12	13	13	16	13	19	13	12
15.00	142	16	13	12	14	19	16	13	13	13	21	15	23	15	14	21	21	20	14	11
16.00	310	23	15	13	20	31	24	17	17	15	28	18	16	15	15	28	15	34	15	14
17.00	389	21	15	13	20	26	21	17	17	14	23	16	15	14	15	24	14	29	15	13
18.00	241	18	14	13	17	20	18	15	15	13	19	14	13	13	13	19	13	21	13	13

21st MAY 1986 THERMOSTAT 15°C

		TEST CELL 1 U-TUBE COLLECTOR									TEST CELL 2 STANDARD TAP									
TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
9.00	109	24	25	17	22	22	25	25	24	23	31	24	30	20	17	31	28	29	17	10
10.00	216	34	25	19	28	34	33	27	27	24	28	30	28	22	22	28	26	27	23	12
11.00	277	38	27	21	32	44	39	29	29	25	30	32	29	24	25	30	28	29	25	14
12.00	384	50	38	24	37	55	47	37	41	33	31	36	30	25	26	31	28	29	26	16
13.00	373	47	30	24	41	59	47	34	36	28	34	40	33	27	28	34	30	32	28	17
14.00	425	49	32	25	4	61	48	37	37	30	32	39	31	27	28	32	29	30	27	18
15.00	414	46	29	24	40	59	46	34	36	27	30	37	30	27	28	31	929	29	28	18
16.00	384	42	28	24	38	55	42	32	32	27	33	35	32	28	29	32	31	32	29	18
17.00	340	36	25	22	33	45	36	29	29	24	28	31	27	26	26	28	27	27	26	18
18.00	246	29	22	20	28	33	29	25	24	22	24	26	23	22	23	24	23	24	23	17

4th SEPTEMBER 1986 THERMOSTAT 30°C

		TEST CELL 1 U-TUBE COLLECTOR									TEST CELL 2 STANDARD TAP									
TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
9.00	104	28	29	19	24	22	29	28	26	27	31	24	29	20	18	31	27	29	18	11
10.00	214	33	24	19	27	33	33	26	26	23	24	26	24	21	22	24	23	23	21	13
11.00	323	42	30	22	35	47	41	33	33	28	28	32	27	24	25	28	26	27	24	15
12.00	386	47	34	25	40	56	47	38	37	32	36	35	36	25	27	35	33	37	27	16
13.00	389	46	28	23	40	58	46	32	33	27	33	37	31	27	28	32	30	31	27	18
14.00	403	45	28	24	39	56	44	32	34	27	31	36	30	27	28	31	29	30	28	17
15.00	389	46	31	25	40	60	45	35	35	30	33	36	32	28	28	32	31	31	29	19
16.00	301	40	28	24	35	50	39	30	30	27	35	36	34	29	29	35	33	34	29	19
17.00	301	30	24	22	29	32	30	27	27	24	26	29	25	25	25	26	25	25	25	17

5th SEPTEMBER 1986 THERMOSTAT 30°C

		TEST CELL 1 U-TUBE COLLECTOR									TEST CELL 2 STANDARD TAP									
TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.30	65	32	31	19	22	16	31	14	13	29	34	20	34	14	11	33	29	34	11	8
9.30	137	33	32	19	26	27	31	31	32	29	32	25	33	18	20	31	27	32	20	10
10.30	246	40	36	22	33	42	39	35	38	32	29	34	28	22	23	29	27	27	23	12
11.30	329	44	28	22	38	50	43	32	34	27	32	38	30	24	24	31	28	29	25	13
12.30	225	39	28	23	35	46	39	32	32	27	28	36	28	25	24	28	27	27	25	14
13.30	109	31	26	22	28	33	31	29	28	25	27	28	26	24	23	27	26	26	23	14
14.30	145	35	33	24	32	40	36	35	35	30	29	33	28	25	24	29	27	29	24	15
15.30	90	28	25	21	26	23	29	28	27	24	27	26	27	23	22	27	26	26	22	14
16.30	107	29	25	21	26	31	29	27	26	24	25	26	25	23	22	25	25	25	22	14
17.30	74	23	19	18	21	25	23	21	21	19	20	21	19	19	19	20	19	19	19	14

10th SEPTEMBER 1986 THERMOSTAT 30°C

		TEST CELL 1	U-TUBE COLLECTOR								TEST CELL 2	STANDARD TAP								
TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
9.00	40	7	7	7	7	10	7	8	7	7	8	7	8	7	6	8	7	8	8	0
10.00	483	14	9	6	11	35	14	7	5	7	15	4	5	3	6	15	3	33	6	1
11.00	642	24	15	10	20	53	24	11	10	14	23	10	7	7	12	22	8	52	8	2
12.00	725	30	20	12	25	60	29	15	14	17	28	13	11	9	16	29	10	61	13	2
13.00	633	32	22	15	28	63	31	16	16	19	31	15	14	10	18	31	11	63	16	3
14.00	491	29	21	15	27	55	28	15	15	19	30	16	15	12	17	30	14	56	16	3
15.00	257	23	17	13	22	39	21	14	13	16	22	13	13	11	15	22	12	49	13	3
16.00	0	10	9	9	11	13	10	9	8	9	11	7	8	7	9	10	7	14	9	2

12th DECEMBER 1986

		TEST CELL 1	U-TUBE COLLECTOR								TEST CELL 2	STANDARD TAP								
TIME (GMT)	I (Wm^{-2})	T_1	T_2	T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{14}	T_{15}	T_{16}	T_{17}	T_{18}	T_{19}
8.00	60	31	31	19	22	17	31	13	13	29	32	19	33	13	9	31	28	32	9	6
9.00	79	27	27	18	22	19	29	19	18	26	30	19	29	16	14	29	25	29	15	8
10.00	109	32	32	20	25	26	31	30	28	30	36	24	36	19	18	36	31	36	18	9
11.00	307	38	24	19	32	45	37	27	27	23	36	31	35	22	22	36	32	35	23	11
12.00	359	43	31	21	33	49	41	33	37	29	27	28	25	21	21	27	24	25	21	12
13.00	318	41	26	21	36	52	41	30	31	25	40	33	41	23	24	40	35	48	25	12
14.00	389	46	28	22	39	59	45	33	35	27	42	36	42	25	25	42	37	42	26	13
15.00	343	42	30	22	35	51	41	32	36	26	26	32	26	24	23	26	25	26	23	13
16.00	216	32	23	20	28	37	32	27	26	23	26	26	25	23	22	26	24	25	22	13
17.00	145	25	18	17	22	29	25	21	20	18	18	20	18	18	18	18	18	18	18	13

17th SEPTEMBER 1986 THERMOSTAT 30°C

APPENDIX B

DATE	THERMOSTAT SETTING °C	DAILY AV. TEMPERATURE U-TUBE °C	HEATING kWh	DAILY AV. TEMPERATURE TAP °C	HEATING kWh	DAILY AV. TEMPERATURE AMB. °C	DAILY AV. INSOLATION I Wm^{-2}	DEGREE DAY	NET.REF.LOAD kWh/DAY
7.5.86	20	16.27	2.89	19.20	3.18	11.20	151	8.80	7.44
8.5.86	20	17.42	2.1	17.05	4.39	12.00	204	8.00	6.80
9.5.86	20	15.9	2.1	15.5	3.36	12.3	130	7.70	6.54
13.5.86	20	18.16	0.81	18.52	2.28	13.09	257	6.91	5.88
14.5.86	20	18.1	1.38	18.7	2.72	13.09	294	6.91	5.88
15.5.86	20	18.0	2.86	18.08	4.50	10.9	307	9.10	7.48
16.5.86	20	23.42	0.68	24.33	0.5	17.05	392	2.95	2.50
21.5.86	15	14.02	1.59	14.5	0.93	11.82	160.6	3.18	2.70
4.9.86	30	26.67	3.65	25.0	5.6	15.80	317.21	14.2	12.07
5.9.86	30	28.14	3.1	28.94	4.3	16.11	312.82	13.89	11.8
10.9.86	30	26.17	3.8	26.08	4.9	12.8	153.12	17.2	14.62
17.9.86	30	25.48	4.8	26.52	5.7	11.00	232.97	19.0	16.15

SOLAR kJ/DAY	AUX/NET REF LOAD U-TUBE	AUX/NET REF LOAD TAP	SSF U-TUBE	SSF TAP	SLR	$\frac{\bar{T}_u-\bar{T}_a}{\bar{I}}$	$\frac{T_{TAP}-T_a}{\bar{I}}$	η_u	η_{TAP}
4620	0.387	0.427	0.613	0.573	621	0.0335	0.0113	0.45	0.4
6242	0.309	0.646	0.691	0.354	918	0.0265	0.0247	0.34	0.12
3978	0.321	0.514	0.679	0.486	562	0.0276	0.0246	0.48	0.28
7864	0.14	0.388	0.86	0.612	1337	0.0197	0.0211	0.34	0.16
8996	0.23	0.463	0.77	0.537	1530	0.0170	0.0190	0.22	0.11
9394	0.36	0.602	0.64	0.398	1256	0.0231	0.0233	0.25	0.12
11995	0.27	0.169	0.73	0.831	4798	0.0162	0.0185	0.02	0.03
4914	0.60	0.344	0.4	0.656	1820	0.0136	0.0166	0.05	0.06
9706	0.302	0.463	0.698	0.537	804	0.0342	0.0290	0.71	0.462
9572	0.262	0.364	0.738	0.636	811	0.0384	0.0410	0.77	0.58
4685	0.259	0.335	0.741	0.665	320	0.873	0.0867	2.45	1.99
7128	0.297	0.352	0.703	0.648	441	0.0621	0.0666	1.772	1.52

Sample Calculation Table for Calculation of η via SLR/SSF Correlation

APPENDIX C

Calculation of Mass Flow Rate Taking into Account the Increasing Velocity with Temperature [Hobson (1986)]

Three major assumptions are made in the following analysis:

1. Mass flow rate constant through collector.
2. Linear temperature profile along collector.
3. Air behaves as ideal gas.

Using assumption 1. there is no variation of mass flow rate with distance, so

$$\frac{\partial}{\partial x}(\rho A \nu) = 0$$

where ρ = density of air
A = Cross sectional area of duct
ν = velocity

thus

$$\rho \frac{\partial \nu}{\partial x} + \nu \frac{\partial \rho}{\partial x} = 0$$

and

$$\rightarrow \quad \rho \frac{\partial \nu}{\partial x} + \nu \frac{\partial \rho}{\partial T} = 0 \qquad \ldots(1)$$

From assumption 3., $\rho = \frac{m}{V} = \frac{P}{RT}$

$$\text{So} \quad \frac{\partial \rho}{\partial T} = \frac{-P}{RT^2} = \frac{-\rho}{T} \qquad \ldots(2)$$

Also using assumption 2.,

$$\frac{\partial T}{\partial x} = \frac{\Delta T}{L} \qquad \ldots(3)$$

where ΔT = Temperature drop across collector and
L = length of collector

Substituting equations 2. and 3. into equation 1. gives

$$\rho\frac{\partial \nu}{\partial x} - \nu\frac{\rho}{T} \cdot \frac{\Delta T}{L} = 0$$

thus

$$\frac{\partial \nu}{\partial x} = \frac{\nu \Delta T}{TL}$$

or

$$\int \frac{1}{\nu}\, d\nu = \frac{\Delta T}{L}\int \frac{1}{T}\, dx \qquad \ldots(4)$$

T is required as a function of x, employing assumption 2. gives

$$T = T_i + \frac{\Delta T}{L}x \qquad \ldots(5)$$

where T_i = collector inlet temperature

Substituting equation 5. into equation 4. gives

$$\int \frac{1}{\nu}\, d\nu = \frac{\Delta T}{L}\int \frac{L}{(T_i L + \Delta T x)}\, dx$$

Integrating both sides gives

$$\ln(\nu) = \frac{\Delta T}{\Delta T}\ln(T_i L + \delta T x) + \theta \qquad \ldots(6)$$

where θ is a constant

To find θ let $\nu = \nu_i$ when $x=0$ where ν_i is the (as yet unknown) velocity of air at the inlet to the collector.

So equation 6. becomes

$$\ln(\nu_i) = \ln(T_i L) + \theta$$

thus

$$\theta = \ln\left(\frac{\nu_i}{T_i L}\right)$$

So 6. becomes, after substituting θ and rearranging,

$$\ln(\nu) = \ln\left[(T_i L + \Delta T x)\left(\frac{\nu_i}{T_i L}\right)\right]$$

or $$\nu = \left(\frac{\nu_i}{T_i L}\right)(T_i L + \Delta T x) \qquad \ldots(7)$$

Equation (7) is the velocity profile as a function of distance along the collector.

By definition, $\nu = \dfrac{dx}{dt}$

thus

$$t = \int \frac{1}{\nu}\, dx \qquad \ldots(8)$$

Substituting (7) into (8) gives:

$$t = \left(\frac{T_i L}{\nu_i}\right) \int \frac{1}{(T_i L+\Delta T x)} \, dx$$

$$t = \frac{T_i L}{\nu_i \Delta T} \ln(T_i L+\Delta T x) + \theta \qquad \ldots(9)$$

With the boundary conditions $t = 0, \quad x = 0$

$$\frac{T_i L}{\nu_i \Delta T} \ln(T_i L) + \theta = 0$$

thus

$$\theta = - \frac{Ti_o L . \ln(T_i L)}{\nu_i \Delta T} \qquad \ldots(10)$$

Substituting (10) into (9) gives

$$t = \frac{T_i L}{\nu_i \Delta T} \ln\left(1 + \frac{\Delta T x}{T_i L}\right) \qquad \ldots(11)$$

The time ΔT for the smoke to travel the length of the collector L will therefore be, from equation (11)

$$\Delta t = \frac{T_i L}{\nu_i \Delta T} \ln\left(1 + \frac{\Delta T}{T_i}\right) \qquad \ldots(12)$$

Dividing both sides of (12) by $\frac{PA}{R}$

gives

$$\frac{\Delta t R}{PA} = \frac{RT_i}{PA\nu_i} \frac{L}{\Delta T} \ln\left(1 + \frac{\Delta T}{T_i}\right) \qquad \ldots(13)$$

But

$$\frac{PA\nu_i}{RT_i} = \dot{m}$$

mass flow rate $\dot{m}$ at the inlet which from assumption 1. is the mass flow throughout the collector, so (13) becomes

$$\frac{\Delta tR}{PA} = \frac{1}{\dot{m}} \frac{L}{\Delta T} \ln\left(1 + \frac{\Delta T}{T_i}\right) \qquad \ldots(14)$$

therefore,

$$\boxed{\dot{m} = \frac{PAL}{\Delta tR\Delta T} \ln\left(1 + \frac{\Delta T}{T_i}\right)} \qquad \ldots(15)$$

<u>SAMPLE CALCULATION</u>

$A = 0.0471\ m^2$

$P = 101{,}325\ N/m^2$

$R = 287.1\ J\ kg^{-1}K$

$T_i = 25$

$T_o = 39$

$\Delta T = 6°C$

$\Delta T = 14°C$

$\nu = 0.25\ ms^{-1}$

If the equation, $\dot{m} = \rho A\nu$ – (16) is employed, then if

$\rho \approx 1.1\ kg\ m^{-3}$ (from tables for the mean temperature in the system), then

$$\dot{m} = 1.1 \times 0.0471 \times 0.25 = 0.01295\ kgs^{-1}$$

$$\dot{m} = \frac{101325 \times 0.0471 \times 1.5}{6 \times 287.1 \times 14} \ln\left(1 + \frac{14}{298}\right)$$

$$= 0.0136276\ kg\ s^{-1}$$

Equation 15 provides a predicted flow rate which is typically 5% greater than predicted using empirical relations -i.e. equation 16. Unfortunately air-flow velocity measurement techniques available for such small flow rates are not accurate enough to demonstrate that the veracity of the result provided by equation 15 is greater.

APPENDIX D

ECONOMIC BENEFITS OF INSTALLING CLADDING COLLECTORS

The economic benefits of installing collectors which are integrated into the building fabric were investigated by Francis et al (1982). General costing of a number of panel options were presented which included materials; manufacture and installation costs based on discussions with manufacturers:

	Solar Collector Overcost £/m^2 Collector (1982)
Separate Panels	90 - 120
Cladding Collector System with no Store	40 - 60
Integral Cladding Collectors	20 - 50

The following performance ranges were used in this analysis, and were taken from the reported performance of water and space heating systems:

	GJ/m^2/Year Saved
Separate Panel System and Store (Space Heating and Hot Water)	0.8-1.2
Separate Panel and Store (Space Heating only)	0.6-1.0
Cladding Supplementary Collector No Store (Space Heating only)	0.5-0.6

One retrofit system proposed in this report consisted of active air-heating cladding panels which would replace the existing timber cladding panels of a four storey office block situated in Central London. The payback period for this system was estimated at 9 years.

A retrofit passive solar-energy air-heating system was proposed for a school in Essex, UK, as a demonstration project for the Commission of the European Communities [Norton (1986)]. During the refurbishment of an existing school building 160 m^2 of vertical solar panels were to be installed. Of the total solar panel area 140 m^2 was to be on the south side and 20 m^2 on the east of the building constituting 28% of the total area to be reclad.

It was estimated that on average each unit would supply 500 MJ/m^2 per annum of useful heat. A further 600 MJ/m^2 per annum would be saved by each installed unit, as the solar panels were designed to improve the level of thermal insulation of the building in comparison with the pre-existing plaster-board single glazed wall elements. It was argued that the solar collector cladding panels would produce a total substitution of fossil fuels of approximately 176×10^3 MJ each year. This is equivalent to 48888 kWh/year which, assuming oil or solid fuel at 2.1 pence/kWh is replaced, has a monetary value of £1027 per annum.

The current cost of standard cladding panel was of the order of £250/m^2. The anticipated cost of mass-produced solar-energy cladding collectors was £269/m^2. The overcost of the solar collector components of the proposed system was calculated as follows:

$$160 \times 19 = £3,100$$

The simple payback period for the cladding collector system was then:

$$\frac{3,100}{1,027} = 3.1 \text{ years}$$

A discounted payback period of 9.6 years was estimated using the formula

$$P = \left(\frac{n \times C}{(A_s - m_c)\left(\frac{(1+i)^n - 1}{i(1+i)^n}\right)} \right)$$

The following assumptions were made:

Annual Saving, A_s, = £1.027

Capital Cost, C, = £3,100

Discount Rate, i, = 5%

Maintenance Cost, m_c, = 0

Minimum anticipated lifetime cost, n, = 60 years.

It was estimated that at the demonstration phase of the project the capital cost of the installation would be recovered in 42 years. This was because of additional costs of prototype manufacture and provision of means by which the unit's performance could be monitored readily.

FACTORS INFLUENCING THE INSTALLATION OF RETROFIT PASSIVE AIR-HEATING COLLECTORS

The installation of passive isolated gain solar-energy air-heating systems isolated gain, has been limited to date to one non-domestic building in the UK. Consequently the factors influencing the adoption of this form of heating in this type of building have yet to be determined. Such considerations as the type and orientation of the building, and the needs of the occupants will be paramount. This presupposes that such a system can be shown to make a useful contribution to the heating regime of a building. Less tangible factors such as the appearance of the passive solar energy components, and the perception of the occupants of non-domestic buildings of the benefits of passive solar energy air heating may prove to have a considerable influence on the successful implemetation and operation of such units. Until there is more information available on the installation and operation of these, and other passive solar energy air-heating devices in non-domestic buildings, commercial development and production will not be undertaken on a large scale.

THE MANUFACTURE AND MARKETING OF PASSIVE AIR HEATING COLLECTORS

Each of the retrofit options reviewed in this study require a specific marketing strategy. The installation techniques for each option differ significantly and would be undertaken by a variety of commercial concerns. These considerations are reviewed as follows:

1) Cladding Collector

The installation of an air-heating system consisting of cladding collector units would be identical to that of standard cladding panels. The mass-produced cladding collectors would be marketed as one of a range of cladding panels produced by a manufacturer and would be recommended for use where solar heat could contribute significantly to the heating regime of a building. The installation of such panels would be undertaken during a building refurbishment programme and, as such it would be necessary to inform clients and contractors of the benefits of these systems.

2) Overcladding Collector

Overcladding is a more specialised refurbishment technique than the recladding of curtain wall buildings, which would be applicable to a wider range of building types. This specialisation may limit the scope for mass-production - without flexible manufacturing - of overcladding collectors, as the techniques and materials involved in the overcladding process may be specific to each building.

There may be a greater requirement for site-built or custom-built solar collectors with this retrofit option than the cladding collector discussed previously. However, upon further investigation it may transpire that the technique of overcladding provides greater scope for the incorporation of passive solar energy air-heating units than conventional recladding refurbishment. If this is the case it may be possible to market overcladding as a method of refurbishment which both protects a building and provides passive solar air heating.

3) The Glazing Collector Panel

Suitably-designed glazing collector panels could be retrofitted to almost any non-domestic or domestic building with the advantage that this would not <u>have</u> to be undertaken as part of an expensive refurbishment programme. Such units could be marketed, manufactured, and installed in a similar manner to the products of double glazing companies. Indeed, glazing collector panels could form part of the product range of a company which is concerned with double glazing and replacement windows. The advantage of these particular collectors is that their costs should be low compared to those of water or air-heating collectors added to a building in the conventional manner.

4) Metal Building Collector

The creation of a passive solar wall on an existing metal building would appear to be a straightforward procedure. This could be undertaken by the owners of the building, a building contractor, or a company specialising in replacement glazing as mentioned above. This retrofit option represents less of a marketable "product" than cladding, overcladding, and glazing collectors, as all that is being added to the building envelope is insulation and one, or maybe two, layers of glazing. The extent of implementation of this retrofit option, as with those discussed previously, is limited by the orientation and overshading of the non-domestic building stock. Buildings such as schools are often favourably positioned to take advantage of available sunshine whereas metal buildings such as those on industrial estates are built in close proximity to one another because of the limited space available. The metal building collector would appear to be the cheapest retrofit solar energy air-heating system to construct per square meter of collector. As with the glazing collector panel, the introduction of this type of heating system does not have to form part of a refurbishment project.

ENERGY CONSUMPTION AND GLAZING AREA

The relationship betwen glazing area, illumination and the overall energy consumption of a building is an important consideration when retrofit passive solar energy air-heating collectors are to be fitted. It has been shown that for single-storey buildings of various sizes and aspect ratio (the ratio of building length to width) of 2:1 that when glazing constitutes over 40% of the total wall area the energy requirement of the building rises significantly. Below 20% there is no corresponding saving in energy and where glazing is on average between 20% and 40% of the external wall of the building as a whole, the energy requirement varies by only 5% (Anon (Undated)). It was recommended that windows should normally be provided within limits. This indicates that glazing area should be reduced to accommodate passive solar collectors only when it constitutes more than 40% of the total wall area.

As the ratio of glazing to external wall area increases, the overall energy balance of a building will be influenced by the following factors in winter:

i) heat loss from glazing will increase

ii) energy consumed by lighting decreases

iii) heat gain from lighting decreases

iv) direct solar gains increase.

If glazing is replaced with panels which both insulate and provide solar energy air heating the following will apply:-

i) reduced heat loss from glazing

ii) more energy consumed by lighting

iii) more heat gain from lighting

iv) reduced direct solar gains

v) increasing indirect solar gains.

The relative merits of reducing the glazing area of non-domestic buildings are summarised below. The relationship between heat loss through glazing and energy consumed by lighting as glazing area is reduced is shown in figure (D1). The relationship between reduction of glazing area and heat gain in shown in figure (D2).

ADVANTAGES
o Reduced heat loss via glazing o Less overheating in summer o Reduced glare o Less ultra violet degradation of internal materials o Greater potential for incidental heat gains and heat from passive solar-energy air-heating wall panels o Greater privacy
DISADVANTAGES
o Increased energy consumption from artificial lighting needed to replace natural light excluded from the building o Reduced exterior views o Reduced heating from direct solar gains

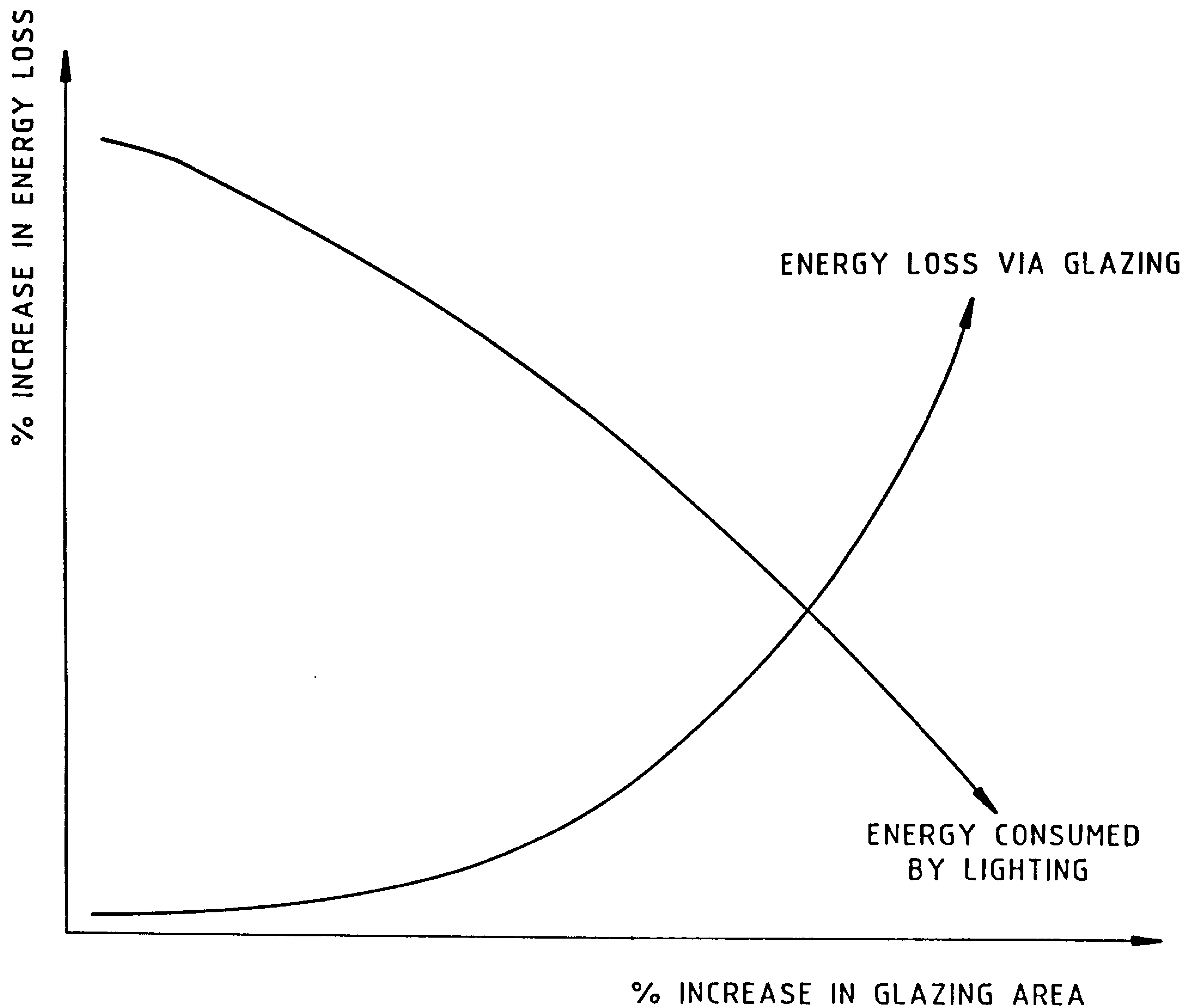

Fig.D1 Relationship between energy lost via glazing and energy consumed by lighting as glazing area is increased.

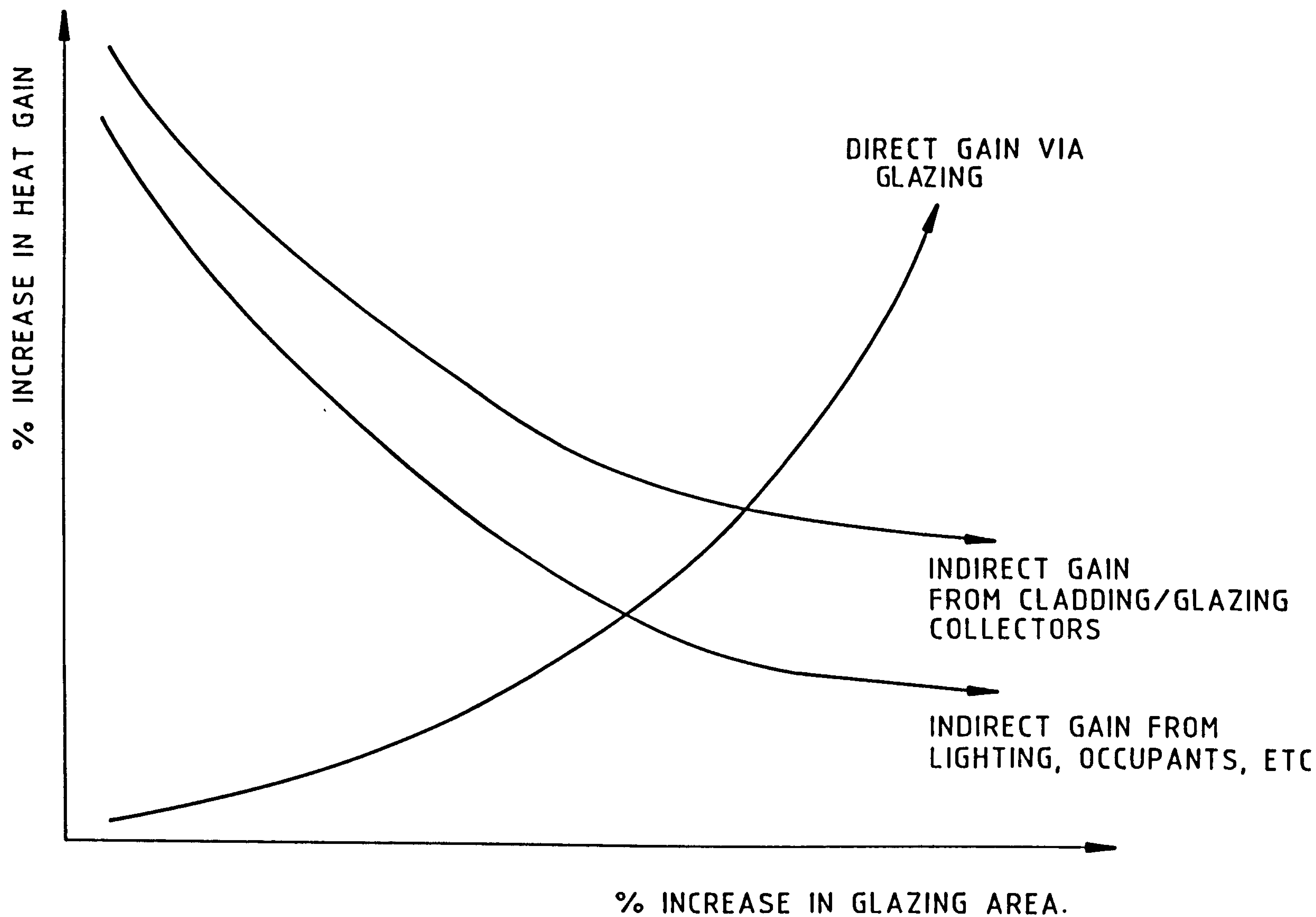

Fig.D2 Relationship between direct and indirect heat gain as glazing area is increased.