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LARGE SCALE TEST OF A NOVEL BACK-PASS NON-PERFORATED UNGLAZED SOLAR AIR COLLECTOR.

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Abstract

This paper describes large scale tests conducted on a novel unglazed solar air collector system. The proposed system, referred to as a back-pass solar collector (BPSC), has on-site installation and aesthetic advantages over conventional unglazed transpired solar collectors (UTSC) as it is fully integrated within a standard insulated wall panel. This paper presents the results obtained from monitoring a BPSC wall panel over one year. Measurements of temperature, wind velocity and solar irradiance were taken at multiple air mass flow rates. It is shown that the length of the collector cavities has a direct impact on the efficiency of the system. It is also shown that beyond a height-to-flow ratio of 0.023 $m/m^3/hr/m^2$, no additional heat output is obtained by increasing the collector height for the experimental setup in this study, but these numbers would obviously be different if the experimental setup or test environment (e.g. location and climate) change. An equation for predicting the temperature rise of the BPSC is proposed.

Keywords: solar air heaters, back pass non-perforated unglazed solar collector, thermal efficiency, and renewable energy.

Nomenclature

A_o = orifice area (m^2)

ΔP = orifice differential pressure (Pa)

β = orifice diameter / duct diameter (d/D)

C_p = heat of air *(Kj/kg °C)*

d = orifice diameter (m)

D = duct diameter (m)

E_{dir} = Direct radiance incident on tilted surface (W/m^2)

E_{dir} = Diffuse radiance incident on tilted surface (W/m^2)

E_{ref} = Reflected irradiance incident on tilted surface (W/m^2)

E_L =Incident long wave radiation on the collector in W/m^2

F_R= Collector efficiency factor

f = enhancement factor

G_{solar} = Global solar radiance incident on surface (W/m^2)

G_{tot} =Total solar radiance incident on surface (W/m^2)

M = molar mass ($kg/kmol$)

m_{air} = Air flow rate (kg/s)

NO = Universal gas constant ($kJ/kmol.K$)

P_a = atmospheric pressure (kPa)

P_s = saturated vapour pressure over water (kPa)

P_{sc} = corrected saturated vapour pressure (kPa)

P_u = orifice upstream pressure (Pa)

P_v = partial vapour pressure (kPa)

R = specific gas constant for moist air (kJ/KgK)

RH = relative humidity (%)

t = temperature ($°C$)

t_o = Outlet temperature ($°C$)

t_a = Ambient temperature ($°C$)

U_{L}= Heat loss coefficient (W/m²K)

ε = expansibility factor

σ = Stefan-Boltzmann constant (=5.67 x 10^{-8} W/m²K⁴)

$\tau\alpha$= Effective transmittance-absorptance factor

μ = instantaneous efficiency (%)

ν = kinematic viscosity (m^2/s)

ρ_u = density of moist air (kg/m^3)

α = flow rate coefficient

γ = isentropic exponent for air(~1.4)

1. Introduction

The operational energy of non-residential buildings accounted for 40% of the European Union's energy consumption and carbon emissions in 2010 [1]. Of this, 50% was used for heating, ventilation and air conditioning services (HVAC). In the UK, the Government target for carbon emissions is to achieve an 80% reduction by 2050 [2]. One of the means of achieving this reduction is through the use of renewable energy systems. However, only 1.8% of operational energy is supplied from renewable energy sources in the UK at present [3].

Energy efficiency measures have been included in part L of the UK Building Regulations [4], which are aimed at reducing energy consumption and therefore reducing CO_2 emissions. One of the major challenges for the UK construction industry is to develop more efficient and effective technologies based on renewable sources of energy, such as solar energy. Additionally, effective energy storage systems must be developed to satisfy the energy demands of end users, as and when it is required, because most renewable energy sources are transient in nature.

Solar energy can potentially be absorbed and converted by using solar collectors to provide space heating in commercial buildings and in large enclosures, such as warehouses and superstores. Technologies, such as solar air collectors (SACs), can therefore result in the building envelope becoming a producer of energy for space heating [5].

SACs are a special type of heat exchangers that absorb incident solar radiation, and convert it to useful thermal energy via a photothermal process (see Fig. 1). In a SAC, the absorber transfers the energy from the solar irradiance to the air flowing through the collector by forced or natural convection, depending on the collector configuration. This heated air inside the collector is then transported as circulating air directly into the building. SACs were first described by Hollick and Peter [6] who used solar radiation to preheat air for ventilation. However, it was in the last three decades that effective solar air collector technologies have been developed [7]. Since then more than one thousand SACs have been installed in over 30 countries [8].

Fig.1 Wall integrated solar air collector

SACs can be classified as glazed and unglazed depending on the material of the absorber plate. Glazed SACs recirculate the internal air of the building through a solar air glazed panel in which the air is heated and then directed back into the building. Unglazed SACs consist of a bolt-on dark-coloured metal absorber plate, through which ambient air outside the building is passed, before being drawn into the building to provide pre-heated fresh air for both ventilation and heating purposes. The most common applications of this technology are the transpired solar air collectors (TSC).

A TSC consists of an unglazed solar air system with a perforated absorber layer. Unglazed transpired solar air collectors (UTSC) use solar energy to heat the absorber surface, which transmits thermal energy to the ambient air (Fig. 2). The absorber surface is generally a metal sheet (usually steel or aluminium), which can be attached to the building facade. The contact surface between the metal skin and air is increased by drawing air through multiple small perforations in the solar absorbing sheet into the cavity between the skin and facade. The heated air is then drawn into the building to provide space heating.

Fig. 2 Unglazed transpired solar collectors (UTSC)

A number of studies on the layout of UTSC perforations in the solar absorbing sheet have been conducted to evaluate heat transfer, efficiency, airflow distribution, and pressure drop. Leon and Kumar [9] developed a mathematical model to predict the thermal efficiency of a “bolt-on” UTSC over a range of different operating conditions. It was reported that the main factors affecting the heat exchanger effectiveness and air temperature rise ($\Delta T°C$) were: (i) air flow rate (ms^{-1}), (ii) solar radiation (Wm^{-2}), and (iii) solar absorptivity (α) by the collector. Efficiencies of up to 65% were reported in this work. Gunnewiek [10] studied the flow distribution in UTSC using CFD simulations. Gawlik [11] studied the performance of low-conductivity unglazed, transpired solar collectors numerically and experimentally.

As an alternative to the UTSC, Othman [12] developed a “bolt-on” prototype solar drying system using back-pass solar collector (BPSC) technology and found that the controlled air flow could maintain the output temperature from the collector constant even if the solar radiation intensity varies to certain degree.

The integration of a BPSC into an insulated wall panel (see Fig. 3) has both on-site installation and aesthetic advantages over conventional UTSC in that it is fully integrated within a standard insulated wall panel, avoiding the negative impact of the perforations on the building’s appearance, and matching aesthetically the rest of the building’s envelope. However, to the best knowledge of the authors, no large scale study on BPSC systems, similar to that on an UTSC, has been conducted. This paper presents the results of a study of the thermal efficiency of such an SAC system.

Fig.3 Back pass non-perforated solar collectors (*BPSC)*

For the BPSC described in this paper, an existing composite panel consisting of five crowns was modified (see Fig. 4) in order to integrate the BPSC through which fresh external air is taken from the base of the profiled voids under the crowns of the panel. By utilizing the outer steel skin of the panel as a solar collector, incident solar radiation is absorbed, resulting in an increased temperature of the air within the crown (see Fig.5).

Existing studies have indicated that the BPSC system may result in savings of up to 20% of the energy required for heating, with a pay-back period of 2.5 years [13].

Fig.4 Photograph of BPSC

Fig.5 Drawing of the test BPSC system

2. Experimental setup and instrumentation

A South facing test rig [14] was constructed at Kingspan R&D facilities in Kingscourt, Ireland (Fig. 6). Kingscourt has a longitude of 6.8 degrees west and a latitude of 53 degrees North. The BPSC dimensions of the test rig are 7.04 m x 4 m. The rear plenum was connected a fan outlet using ducts which were insulated.

Fig.6 Photograph of BPSC test-rig

2.1. Global solar radiation measurement

The global solar radiation was measured using two Kipp and Zonen CMP11 pyranometers. A Kipp and Zonen CM121 shadow ring was used to shade one of the pyranometers, allowing the ground-reflected solar radiation to be measured. The unshaded pyranometer included a white body shading cone to minimise body heating. The pyranometers were installed to one side of the test panel at around its mid-height. Both were installed vertically and aligned with the test panel. None of the sensors or mountings shaded the panel. The CM121 shadow ring was used with the pyranometer also in the vertical position, aligned with the test panel. The shadow ring was periodically adjusted to ensure the sensor remained shaded through the test period.

The long wave radiation from the sky was also measured by a Kipp and Zonen CGR4 pyrgeometer. This was installed in the vertical plane alongside the test panel at approximately mid-height. A body shading cone was also used. The integral thermistor output was used for the calculation of net long wave radiation.

2.2. Air temperature measurement

The air temperature was measured using class 1/10th DIN, 4-wire PT100 probes at 5 air inlets equally spaced across the base of the panel, and at 5 positions arranged in an array in the collector chamber or plenum immediately before the air outlet. A single sensor suspended at approximately the mid-height of and behind the panel was used to measure the ambient temperature. All the exposed air temperature sensors (at inlets and behind the panel) were housed in double skin radiation shields to minimise effects of incident solar radiation. These shields had a tube-in-tube construction. The cylindrical body of the outer tube was wrapped by reflective foil to reflect solar radiation. The ends of the tubes were open and ventilation holes were drilled in both tubes prior to assembly. These holes were offset between the inner and outer tubes to prevent direct ingress of radiation at any angle. Figure 7 shows the air temperature sensor shield at the air inlets. Four channel cavities of the BPSC collector were instrumented with one air temperature sensor each at a height of 3.5 m above the air inlets to measure the air temperature there. All holes for cable passage were well sealed with duct sealant and visually inspected. All sensors were checked using a PT100 simulator across the range of expected operation and corrected for any offsets.

Fig.7 Air temperature sensor shield and positioning at air inlet point of BPSC

2.3. Air flow measurements

Air flow was measured via two ISO 5167 [15] orifice plates with corner taps, mounted in separate parallel duct sections downstream from the panel plenum. Flow rate could therefore be calculated using the methods in ISO 5801[16], by measuring differential pressure across the plates as well as the variables such relative humidity, atmospheric pressure and duct air temperature. Only one orifice plate was used at a time. The use of two plates enabled a greater flow range to be tested whilst ensuring that the pressure difference did not either drop too low to enable accurate measurements to be taken, or for the overall pressure drop to be too high for the driving fan. The two orifice plates were designed and manufactured by Poddymeter, and the specification was 200 Pa differential pressure at 500 m^3/hr (for an orifice diameter of 0.12532 m) and 900 Pa at 2200 m^3/hr (for an orifice diameter of 0.17264 m). The installation was checked for correct flow directions on the plates and that the plates were suitably sealed with gaskets on the flange faces. The housing ducts were 250 mm diameter and the straight sections joining the plates were greater than 3 m upstream and 1.5 m downstream, thus achieving greater than 12 and 6 duct diameters of straight section, respectively (see Fig.8).

Fig.8 Orifice plates with corner taps, mounted in separate parallel duct sections

Differential pressure across the orifice plates was measured using Sontay PA267 transmitters (with an optional higher accuracy specification), in 0 - 500 Pa or 0 - 1000 Pa range depending on the orifice plate and flow rate used. Atmospheric pressure was measured using a Pi605 atmospheric pressure transmitter from Omni Instruments, with the higher accuracy specification option (< 0.1 % combined error). Relative humidity inside the duct was measured by a Rotronic HC2-S transmitter, with the head completely inserted into the duct at the same location as the temperature sensor array. Calibration was performed by Industrial Temperature Sensors Ltd, and was within the specification of the manufacturer in the region of interest, typically 20 – 50 % RH after heating.

2.4. Winds peed measurements

Commented [BS1]: Wind speed measurements

Wind speed was measured at either side of the panel at mid-height by Vector Instruments A100 series anemometers with a reduced full-scale range of 0 - 25.74 m/s. These were mounted with leading / trailing edge boards extending from the panel edge behind the anemometers to give a continuous surface as per the requirements of EN12975 [17].

3. Data collection

Test data were collected over a one year period, during which a total of 250 tests were conducted, mostly in conditions with relatively high solar radiation (400 to 900 W/m^2). A feature of the test is the variation in energy collection under apparently similar conditions.

The general principle for the measurements was to record a stable panel pre-conditioning period of 15 minutes followed by a steady state data collection period for performance evaluation of 10 minutes [18]. During these periods, the parameters of solar and long wave irradiance, air temperature, fluid mass flow rate, collector fluid inlet temperature and surrounding air speed were within specified tolerances in EN12975-2:2006. Other parameters were not necessarily within the specified ranges but they were beyond our control. Those data outside the specified ranges were The BPSC fluid inlet temperature in this configuration was the same as the ambient air temperature. The air mass flow was generally within tolerance although for individual readings, there were temporary deviations outside of the ±1% average. In general, the average over the measurement period was stable with almost no drifting.

3.1. Measurement processing

The global solar radiation (G_{solar}) on a tilted surface consists of the direct irradiance(E_{dir}), diffuse irradiance (E_{diff}) and the reflected radiation or albedo (E_{ref}):

$$G_{solar} = E_{dir} + E_{diff} + E_{ref} \tag{1}$$

The unshaded CMP11 reading provides the direct solar radiation on the panel. The CGR4 reading provides the diffuse or incident long wave radiation from the sky. The CMP11 mounted with the shadow ring provides the ground-reflected solar radiation. However, the global solar radiance per m² area was calculated in accordance to EN12975- 2 [17].

For subsequent solar radiation to collected heat conversion efficiencies, the total radiation available for collection was determined by the useful area of the panel. This is defined as the proportion of the panel where air circulates through the channels, including the “finned” elements between the channels. It does not include the panel area above the top of the connecting holes through the panel insulation to the plenum, since the air here is stagnant. For the test BPSC, the effective area was taken as 4 m x 6.5 m. However, the 0.5 m above the plenum holes were considered ineffective. The effective area of the test panel was therefore 26 m².

3.2. Air inlet and outlet temperatures

The five inlet temperatures were averaged to give an air inlet reference temperature: their variation was small, generally within 0.5 °C. The readings of the five outlet sensors were also averaged. They were used to determine the reference temperature difference in the heat output calculations.

3.3. Air flow rate

The calculation of the airflow through the orifice plate was conducted in accordance with ISO 5801 [16]. The method required the following measured parameters:

- Differential pressure across the plate
- In-duct relative humidity
- In-duct air temperature
- Atmospheric pressure

Specific heat capacity for moist air was calculated as a function of the air temperature and moisture content, with the reference temperature being the average of the panel inlet and outlet temperatures [19].

4. Data analysis

The amount of heat delivered by the BPSC is given by:

$$Q = m_{air} C_p (t_o\text{-}t_a) \qquad (2)$$

where C_p is the specific heat of air (kj/kg °C), m_{air} is the mass flow rate though the BPSC (kg/s), t_o is the outlet air temperature (°C), t_a is the inlet (ambient) air temperature (°C).

The instantaneous efficiency of the BPSC is defined by the Hottel-Whillier-Bliss equation [20]

$$\mu = \frac{Q}{AG_{tot}} = F_R \tau\alpha - \frac{F_R U_L (t_o - t_a)}{G_{tot}} \qquad (3)$$

where A is the effective area of the collector (m²), G_{tot} the total solar irradiance on the collector surface (W/m²), F_R the heat removal factor, $\tau\alpha$ the effective transmittance-absorptance product which depends on the angle of solar incidence, and U_L the overall heat loss coefficient (W/m²K). In Equation (3) the effective transmittance-absorptance product, $F_R\tau\alpha$, and the overall heat transfer coefficient for losses from the collector $F_R U_L$ are constant over the entire collector plane.

The total solar radiation is dependent on the global solar radiation and the ratio ε/α based on the ambient temperature as follows:

$$G_{tot} = G_{solar} \pm (\varepsilon/\alpha)(E_L\text{-}\sigma t_a{}^4) \qquad (4)$$

where σ is the Stefan-Boltzmann constant (=5.67 x 10^{-8} W/m²K⁴), ε is the emittance of the collector surface, α is the solar absorptance of the collector surface, E_L in W/m² is the incident long wave radiation mainly from the sky but also from terrestrial surroundings of the collector, and t_{amb} is the ambient temperature.

The rise of air temperature in the collector varies with the distance from the air inlet, which is dependent on (i) the absorptivity of the BPSC, (ii) the heat transfer coefficient between the collector surface and the environment, and hence (iii) the local wind speed, (iv) the heat transfer coefficient between the BPSC surface and the air in the collector, and thus potentially on (v) flow rate and (vi) the back losses from the collector to the environment behind it, usually the building interior. The basic model in Fig. 9 shows that the highest efficiency between 35% and 39% was obtained with flow rates between 90 and 120 m³/hr/m² at inlet temperatures between -6 °C and 8 °C. For a given total air flow, defined by the room ventilation needs, splitting the flow between a series of low collectors has two effects: a low collector is more efficient but has a reduced flow rate per collector.

Fig.9 Effect of air flow rate on efficiency

4.1.1. Effect of wind speed on temperature rise

Figure 10 shows that the temperature rise per unit of solar radiation is not affected by the wind speed for the tested BPSC within the test range of 0.3 and 4 m/s.

Fig.10 Effect of wind speed on temperature rise per unit radiation

4.1.2. Effect of flow rate on temperature rise per unit of solar radiation

The majority of the tests were conducted with a flow rate between 40 and 140 m³/hr/m². Within this range, there is a consistent linear relationship where an increase of the flow rate reduces the temperature rise.

Fig.11 Effect of air flow rate on temperature rise

4.1.3. Stabilised temperature rise

Based on the results on the effect of air flow rate on efficiency, Fig. 12 presents the results of temperature rise for different levels of solar radiation at airflow rates between 90 and 120 $m^3/hr/m^2$ at inlet temperatures between -6 °C and 8 °C.

Fig.12 Effect of solar radiation on temperature rise

The results for temperature rise per unit radiation (Fig.13) show very little variation with flow rate, suggesting that the combination of flow rate and height is sufficient for the observed temperature increases close to saturation value. This relationship can be used to estimate the exit air temperature as a function of flow rate, radiation level and height, and thus to estimate how the performance of the BPSC would vary with height. Qualitatively, if the collector is relatively large compared to the characteristic length, the exit temperature will be close to the saturation temperature. However, if the height is low, the exit temperature will be lower, although the efficiency will be higher.

Fig.13Variation of temperature rise per unit radiation against flow rate

The results show that the temperature increases with the height/flow ratio up to a certain point, after that the temperature is constant and the collector height is effectively infinite.

5. Discussion of results

The BPSC on the south elevation was tested in detail to characterise the performance of the panel and develop an empirical model to be used to estimate the panel temperature discharge and efficiencies.

The exit temperature from the BPSC is a function of solar radiation, height of the collector and air flow rate in the collector:

$$T_{out} = f(G_{tot}, h, m_{air}) \quad (5)$$

The performance of the BPSC is determined by the temperature rise of the exit temperature above to ambient temperature. From the experimental data of the collector, the exit temperature, relative to ambient, in terms of rise per unit incident radiation can be calculated, which defines the basic thermal model of the collector:

$$\Delta T_{rad} = \Delta \dot{T}_{rad}\left(1 - e^{-\frac{hE}{m_{air}}}\right) \quad (6)$$

Where E is the air flow rate coefficient, m_{air} is the air flow rate ratio

At the maximum height or zero flow rate: $\Delta T_{rad} = \Delta \dot{T}_{rad}$

The useful thermal energy delivered to the building is simply calculated by:

$$Q = m_{air}\rho Cp\, \Delta T_{rad} G_{tot} \quad (7)$$

Where ρ is the density of the air, 1.22 kg/m^3, G_{tot} is given by equation (16).

ΔT_s can be estimated from these values by using the data point from the highest value of the height/flow ratio, in this case 0.0134 (at a ratio of 0.223). Re-arranging equation [6], the following equation is proposed:

$$Ln\left(\frac{1\text{-}\Delta T}{\Delta Ts}\right) = E(h/m_{air}) \quad (8)$$

The estimation of E is -8.07/0.223=-36.19. This illustrates that beyond a certain height (h) to air flow rate (m_{air}) ratio, no additional heat is collected.

6. Conclusions

A novel wall-integrated and unglazed solar air collector has been developed. A large scale rig having a panel surface area of 26m² was tested to determine the physical behaviour of the back-pass non-transpired solar collectors (BPSC) prototype, and to identify the governing factors in the collector performance.

It was found that wind speeds of up to 4 m/s across the collector metal plate had no impact on the performance of BPSC. The results also showed that the estimation of the exit air temperature of the solar air collector would depend on the intensity of the solar radiation, air flow rate through the collector crowns and the solar collector height ($T_{out} = f(G_{tot}, h, m_{air})$). For the panels monitored, it was shown that beyond a 0.023 m/m³/hr/m² height-to-flow ratio, there was no additional heat collection. It was also observed that the temperature increases with the height/flow ratio up to a certain height where there is no additional heat collection. However, efficiencies up to 39% can be achieved with a combination of collector lengths and effective air flow rates in the range of 90-120 m³/hr/m².

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Appendix A. Mass flowrate of air

Mass flowrate of air [17]:

$$m_{air} = \alpha\varepsilon\pi\frac{d^2}{4}\sqrt{2\rho_u\ \Delta_p} \quad (1A)$$

Where α is the flow rate coefficient [14]:

$$\alpha = (1-\beta^4)^{-0.5}[0.5959+0.0312\beta^{2,1}- 0.184\beta^8+0.0029\beta^{2.5}((\frac{10^6}{R_D})^{0.75}] \quad (2A)$$

The Reynolds number [14] is:

$$R_D = \frac{\alpha\varepsilon\beta d}{v}\sqrt{\frac{2\Delta_p}{\rho_u}} \quad (3A)$$

The expansibility factor [14] is:

$$\varepsilon = 1 - (0.41 + 0.35\beta^4)\frac{\Delta_p}{kp_u} \quad (4A)$$

The kinematic viscosity [14] is:

$$v = \frac{(17.1+0.0048t_{out})^{-6}}{\rho u} \quad (5A)$$

The saturated vapour pressure [18]:

$$P_s = 10^{(30.590521-8.2\log(\theta+273.15)+0,0024804(\theta+273.15)-(\frac{3142.31}{\theta+273.15}))} \quad (6A)$$

The enhanced factor [19]:

$$f = 1 + A + P_a[B+C\,(t_{out} + D + EP)^2] \quad (7A)$$

The partial vapour pressure [18]:

$$P_v = (\frac{R_H}{100})\rho_s \qquad (8A)$$

The Molar ratio [19]:

$$\alpha_i = \frac{P_v}{P_a} \qquad (9A)$$

Molar mass moist air [19]:

$$\alpha_a = \left(\frac{P_v}{P_a}\mu_{wat}\right) + (1\text{-}\frac{P_v}{P_a})\,\mu_{air} \qquad (10A)$$

Equivalent gas constant [19]:

$$R = \frac{\bar{R}}{\alpha_a} \qquad (11A)$$

Density moist air [19]:

$$\rho_u = \frac{P_a}{R(t_{out}+273.15)} \qquad (12A)$$