

Breathing walls: The design of porous materials for heat exchange and decentralized ventilation



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ABSTRACT

This study demonstrates how to design pores in building materials so that incoming fresh air can be efficiently tempered with low-grade heat while conduction losses are kept to a minimum. Any base material can be used in principle, so long as it can be manufactured with millimeter-scale air channels. The channel-pores are optimized according to the thermal conductivity of the base material, the dimensions of the panel, and the suction pressure sustained by a given fan or a chimney. A water circuit is integrated at the interior surface to ensure direct thermal contact and prevent radiant discomfort. Correlations from the thermal sciences literature were used to optimize the size and distribution of channel-pores in wood, glass, and concrete test panels. The measurements showed good agreement with theory and were presented in a general form so that designers can predict the steady-state performance of any optimal design in sensible heat-transfer mode. Schlieren imaging was used to characterize the different regimes of mixed convection at the interior and exterior surface. The data explain the discrepancy between prediction and measurement in the dynamic insulation literature, and how the integrated water circuit overcomes these problems. Surface heat-flux measurements were correlated in a general form so that designers can account for convection at the interior and exterior surface.

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1. Introduction

Modern enclosures are designed as insulators and made from layers of different materials. But it might be better to design them as heat exchangers, using one base material that can perform several functions. This paper describes a method for designing building materials as heat exchangers, so that incoming fresh air can be efficiently tempered with low-grade heat while conduction losses are kept to a minimum (Fig. 1). Any base material can be used in principle, so long as it can be manufactured with millimeter-scale air channels.

The results of two experiments are reported. The first experiment measures the convection at the surface of a porous heated plate as air is sucked or blown through the pores. The results are correlated to characterize the different kinds of convection between a room and a porous wall. This is important because, in order to transfer heat to the air flowing through it, a porous material must

first receive heat at one face. Precedents in the building science literature underestimate the importance and complexity of the first transfer stage.

The second experiment measures the heat exchange inside glass, concrete and wood samples designed with parallel channels. The results validate a materials design method from the thermal sciences literature, which can be used to design high-tech porous materials, such as turbine blades that are cooled through their pores to withstand extreme heat. Taken together, the results of our two experiments show how to translate that method to the context of building design, while resolving problems associated with *dynamic insulation*.

1.1. Dynamic insulation

In the 1960s, '70s, and '80s, engineers in Norway, Canada, Sweden, Denmark, Finland, and Switzerland developed a new ventilation system for livestock buildings, based on a novel idea for recovering heat [1]. A layer of porous insulation divided the shed into a room and an attic. The attic space extended half way down the walls to form a wrap-around cavity. With a fan, fresh air from the cavity was sucked into the room through the porous insula-

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Subscript

A	area [m^2]
c_p	specific heat capacity [$\text{J kg}^{-1} \text{K}^{-1}$]
D	diameter of channel [m]
H	height of unit (channel spacing) [m]
h	heat transfer coefficient [$\text{W m}^{-2} \text{K}^{-1}$]
k	thermal conductivity [$\text{W m}^{-1} \text{K}^{-1}$]
L	length of channel (depth of panel) [m]
T	temperature [K , $^\circ\text{C}$]
Q	volumetric flow rate [$\text{m}^3 \text{s}^{-1}$]
q''	heat flux [W m^{-2}]
u	area-averaged velocity [m s^{-1}]
x	length/height at point of measurement [m]
α	thermal diffusivity [$\text{m}^2 \text{s}^{-1}$]
β	thermal expansion coefficient [K^{-1}]
ΔT	temperature difference [K , $^\circ\text{C}$]
ΔP	pressure difference [Pa]
ΔZ	height difference
ϵ	Emissivity
ε	heat-exchange efficiency
μ	dynamic viscosity [$\text{kg s}^{-1} \text{m}^{-1}$]
ν	kinematic viscosity [$\text{m}^2 \text{s}^{-1}$]
ρ	density [kg m^{-3}]
ϕ	void fraction (porosity)
Be	Bejan number
NTU	number of Transfer Units
Nu	Nusselt number
Pe	Péclet number
Pr	Prandtl number
Ra	Rayleigh number
Nu_m/Nu_n	ratio of mixed convection to natural convection
$\sqrt{Pe}/Nu_n$	ratio of forced convection to natural convection

Subscript

a	air
ae	air, exterior
ai	air, interior
c	convection
m	mixed (measured)
max	maximized
min	minimized
n	natural
opt	optimized
r	radiation
s	surface
se	surface, exterior
si	surface, interior
x	local, at point of measurement

tion. On its way through the pores, the fresh air was warmed by heat conducting out through the material in the opposite direction. The Norwegian phrase *Motstrømstak* highlighted the novelty of counter-current exchange [2], while, in German, *Porenlüftung* alluded to the idea of ‘breathing’ with ventilation through pores [3]. British researchers later settled on the term *dynamic insulation* [4].

In the 1980s, '90s, and '00s, engineers more carefully examined the heat-recovery traits of open-pore and fibrous materials [5,6], and worked on transferring the *dynamic insulation* concept from industrial cattle-sheds to homes [7], offices [8,9], and sports facilities [10]. The goal was to save energy while providing more than the minimum fresh air. However, while initial experiments showed good agreement with theory [1,4], later experiments gave

inconclusive results and suggested some serious limitations. With increasing fresh air, the temperature of the interior surface tended to fall, and so, therefore, did the portion of recovered heat and the radiant temperature of the interior [7,11,12]. Researchers turned their attention to the air-filtration characteristics [13], before moving on to the study of heat-recovery concepts that do not use porous materials [14].

A new series of studies were conducted 2015 and 2016. Researchers examined micro-scale [15], latent [16], and transient [17] heat transfer; the filtration of particulates [18]; and the use of buoyancy to replace fans [19]; while one group developed custom apparatus to test some of these aspects in combination [20]. But despite the renewed interest, today's designs share two common features worth questioning. First, they presume to use stock insulation materials, so additional materials are needed to complete a given envelope design. Second, they incorporate an air-mixing cavity to cover the interior surface. However, this only seems to lessen the impact of the wayward surface temperature rather than resolve the issue conclusively.

This study revisits the basic premise of *dynamic insulation*, but takes a different approach, using a method from the thermal sciences literature [21]. The method shows how to optimize parallel channels in any material, to counteract exterior heating by forcing a coolant through the channels at a given pressure. Kim, Lorente, and Bejan [21] used analytical methods and numerical simulations to develop a set of design correlations that apply to a wide range of scenarios and base materials. Our experiment validates their results, which we present in standard notation for heat exchanger design, so they can be used to design building envelopes that exchange heat efficiently (rather than things like turbine blades for which hotspot temperatures are the primary concern).

1.2. Multifunctional materials

Technically speaking, *dynamic insulation* is a kind of heat exchanger. The laws of thermodynamics suggest a possible advantage to a heat exchanger enveloping a room: it might make it easier to exploit low-grade and renewable sources of heat. To achieve the same power from a larger exchange surface, one does not need an especially hot or cold control fluid—a tepid fluid will suffice [22,23].

Modern buildings are made up of many different materials and technologies, and it is rare for one member of the design team to have a complete picture of the functional interdependencies between all parts [24]. The process of design and construction is highly decentralized, and the activities, relationships, and conditions are often particular to the given project and site. General methods for materials design are needed, to show how concepts for multifunctional materials can be applied to any particular context, using local skills and materials, in a variety of ways.

Surveying the surfeit of bulk materials now available, some materials scientists advocate a method of combining existing materials with geometric features to create multifunctional hybrid materials [25,26]. Studying natural materials, some biologists argue we should improve the performance of simple constituent materials by manipulating their internal geometry at different length scales [27–29].

In a similar vein, Adrian Bejan and Sylvie Lorente propose an approach to materials design for cases where flows of heat, mass, and mechanical stress need reconciling. Based on the idea that natural and artificial systems reconfigure over time by adjusting to the currents that flow through them, *Constructal Theory* [30] has influenced a growing number of engineers worldwide. The methods have been used to design things such as heat exchangers, vascular materials, and transport networks, and to study natural flow systems such as lungs, rivers, and trees [31–34].

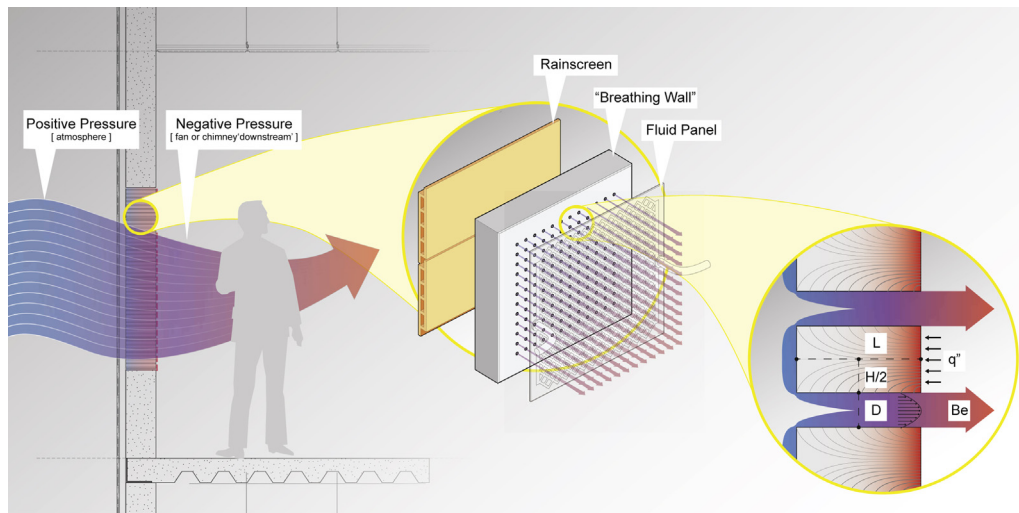


Fig. 1. Building envelope with porous material optimized to exchange heat to the incoming air (sucked in by a fan or a chimney) with minimal conduction losses. A water circuit integrated at the interior surface of the panel controls the temperature. Weatherproofing and wind-buffering can be done by an external rain-screen.

The *intersection of asymptotes* method is one of the most widely used in the *Constructal Theory* oeuvre [35]. Kim, Lorente, and Bejan [21] used that method to develop a formula for optimizing porous materials in counter-flux heat exchange, which they then calibrated numerically. We base our work on the results of that study.

1.3. Outline

Before translating the results of Kim, Lorente, and Bejan [21] to the context of building design, the boundary conditions needed to be carefully evaluated. It was important to understand how room-side heat transfers to a porous wall, because this ultimately determines how much heat can be passed to the incoming air. We characterized the heat transfer by taking convection measurements at the surface of a bespoke milli-fluidic panel with air holes while controlling the surface temperature and the air flow in suction and blowing. We also took schlieren images to better understand the physical relationship between natural and forced convection at the surface. Among other things, the results suggest why dynamic insulation suffers from lower than expected heat recovery rates, and how direct contact heating would solve this problem (Fig. 1).

The second experiment shows how to design a porous material following the method of Kim, Lorente, and Bejan [21]. Experimental results for three test panels are shown, made from concrete, wood, and glass. A milli-fluidic panel was used to control the room-side surface temperature, and a proprietary fan controlled the air-flow rate. The data are correlated and presented in standard heat exchanger notation, so designers can evaluate different material designs, in a variety of conditions, subject to different constraints.

The two experimental studies are reported separately (in Sections 2 and 3, respectively) because the results of each are more clearly discussed in isolation. Section 4 gives an outlook of the challenges and opportunities for future research, making clear how the results of both experiments can be used in concert.

2. Experiment I: heat exchange at the surface of porous materials

Before it can pass heat to the air flowing through it, a porous material must first receive heat at one face. The importance and complexity of this first stage seems to have been underestimated

in the *dynamic insulation* literature, as the experiment reported in this section shows.

Heat transfer science is still working to improve methods for analyzing convection in porous media. Numerical techniques are increasingly used, but it can be computationally expensive to solve the full differential equations for convection, particularly if the porous material resides in an unbounded region. Commentators have noted the lack of experimentally validated solutions, in particular for mixed convection on the surface of transpiring surfaces [36,p. 367]. This kind of mixed convection can take on a number of complex forms, depending on the strength of blowing or suction; the strength of buoyancy from the heated or cooled surface; and the inclination of the surface. Numerical solutions exist [37–41], but only for laminar flow, and implementing them is not straightforward.

The dynamic U-value of a porous insulation material is defined as the heat loss coefficient at zero air flow minus the coefficient of recovered heat with increasing air-flow. In early models, the temperature of the interior and exterior surface was assumed to be equal to, respectively, the bulk temperature of the interior and exterior environment [4]. In a later model, Taylor and Imbabi tried to incorporate the effects of surface convection to get a more realistic estimate of surface temperature, and hence a more accurate prediction of heat-recovery [5]. But they did so while making one important simplifying assumption: that the convection boundary layers on both the interior and exterior surface would behave practically the same as in the normal, non-transpiring case.

To test this assumption, and to evaluate its impact on subsequent studies in the dynamic insulation literature, we took Schlieren images of a porous, vertical, heated plate, subjected it to controlled blowing and suction, and measured the convective heat transfer on both the interior and exterior surface.

2.1. Materials and methods

Fig. 2 shows the porous heating panel that was fabricated for both experiments presented in this paper. The water is dyed to highlight the circuit, which was designed to ensure an even temperature distribution, following a method described by [42]. The panel is 15 cm × 15 cm and the fluid channels are 2 mm wide and 1 mm deep. There are 144 holes, spaced evenly at 1 cm centers, each 2.52 mm in diameter. The panel is made from two sheets of optically clear acrylic: one sheet (2 mm thick) was CNC machined to

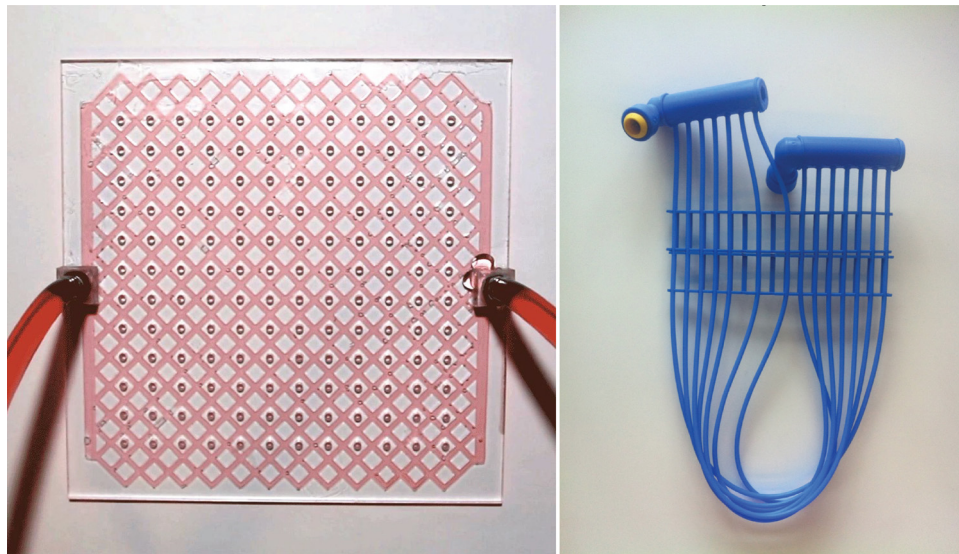
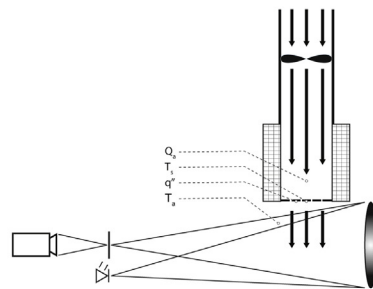
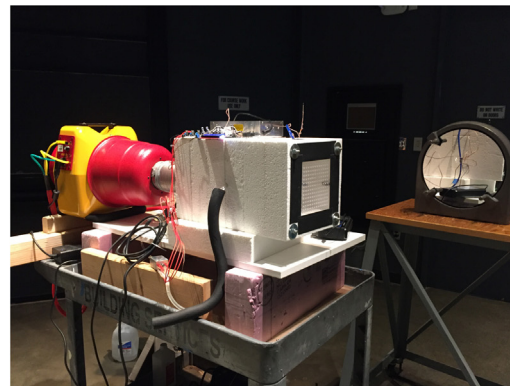


Fig. 2. Left: porous milli-fluidic panel fabricated for both experiments (15 cm × 15 cm, 2 mm wide channels). Right: proprietary capillary tubing could also be used (sample with 4 mm diameter tubing shown).



(a) Plan view schematic, schlieren



(b) In-situ, schlieren

Fig. 3. Experimental set-up for Schlieren and surface heat transfer measurements.

create the diamond-grid water circuit, while a solvent welder was used to bond it to the other sheet (1 mm thick).

Schlieren imaging is a technique where lenses and mirrors are used to visualize density differences in gases. Fig. 3 shows the set-up. A digital camera, pin-light and ‘cut-off plane’ were aligned with a concave mirror of 300 mm diameter and 6m spherical radius. The porous plate was connected to a heated water circuit to control the surface temperature. And it was fitted to an open-face enclosure and fan blower (RetroTec 300 DucTester) so that the air-flow rate could be incrementally increased in suction and blowing. Air that came in contact with the heated plate became warmer and therefore less dense and the optical apparatus captured the consequent refraction of light.

Using the same set-up, the local mixed convection heat transfer coefficient was measured as a function of increasing air-flow. Two heat-flux sensors were attached to the middle of one surface of the fluid panel, 30 mm apart, and the average heat-flux was recorded. Thermocouples were used to measure the ambient air temperature and surface temperature. An infrared camera was used to monitor the panel during the experiment to check it stayed isothermal.

To obtain the convective heat transfer, the radiative heat transfer had to be estimated and subtracted from the data. The radiant temperature of the environment was unknown, and the thermal emissivity of the heat-flux sensor (greenTEG GSkin) had not been

tested by the suppliers (they are made of aluminum are powder-coated white, suggesting an emissivity of approximately 0.9). So to calibrate our estimate we measured the total heat transfer coefficient at zero air flow, and compared it to the value predicted by the Squire-Eckert formulation [43,sec. 8.3] for the local heat transfer coefficient in laminar natural convection on a vertical, isothermal plate:

$$Nu_n = 0.508 Ra_x^{1/4} \left(\frac{Pr}{0.952 + Pr} \right)^{1/4} \quad (1)$$

where the local Rayleigh number, which gives the ratio of buoyancy force to dissipation forces, is defined as:

$$Ra_x = \frac{g\beta\Delta T x^3}{\alpha\nu} \quad (2)$$

and the local convective heat transfer coefficient is normalized to limit of pure conduction, giving the local Nusselt number:

$$Nu = \frac{h_c x}{k_a} \quad (3)$$

The data were correlated to the local rather than the area-averaged heat transfer coefficient because of the small size of the heat-flux sensor (1 cm × 1 cm), which was chosen so as not to block the pores. Once the measurements were taken, the radiative heat

Table 1
Accompanying data for Schlieren images in Fig. 4.

Image	Q (l/s)	u (m/s)	ΔT	$\sqrt{Pe}$	$Ra^{1/4}$	Nu_n	$\frac{\sqrt{Pe}}{Nu_n}$
a.	–	–	20.5	–	30.96	12.72	0.
b.	0.1	0.007	20.4	5.	30.92	12.71	0.39
c.	0.15	0.01	20.4	6.13	30.92	12.71	0.48
d.	0.25	0.017	20.6	7.91	31.	12.74	0.62
e.	0.35	0.024	20.5	9.36	30.96	12.72	0.74
f.	0.45	0.031	20.5	10.62	30.96	12.72	0.83
g.	0.55	0.038	20.3	11.74	30.88	12.69	0.92
h.	0.7	0.049	20.5	13.24	30.96	12.72	1.04

transfer coefficient was estimated and subtracted from the data using

$$h_r = \sigma \epsilon (T_s^2 + T_r^2) (T_s + T_r) \quad (4)$$

while assuming the radiant temperature of the environment T_r to be the same as the ambient temperature T_a , and by setting the emissivity at $\epsilon = 1$. This value is unrealistic in physical terms: it should be interpreted as an ‘effective’ value that accounted for uncertainties (such as the unknown radiant temperature and actual emissivity). The calibration was successful, as shown in Fig. 5 where the data at zero air-flow are in good agreement with the prediction for pure natural convection given by Eq. (1). That is, $Nu_m/Nu_n \approx 1$ when $\sqrt{Pe}/Nu_n = 0$. These dimensionless numbers and ratios are explained below.

2.2. Results

A montage of the Schlieren images is shown in Fig. 4. Fig. 4 should be read in conjunction with Table 1, which gives the flow rates and temperature differences for each image. With suction, the natural convection boundary layer grows thinner with increasing airflow. With blowing, it grows steadily thicker, until turbulence ensues.

Normalized measurements for convective heat transfer are shown in Fig. 5. This should be read in conjunction with Table 2, which gives the correlating equations plotted on the graph, and a verbal description of each regime.

The x-axis of Fig. 5 measures the heat carrying capacity of the forced air flow ($\sqrt{Pe}$), relative to the strength of laminar, undisturbed natural convection (Nu_n) defined by Eq. (1). The y-axis measures the strength of mixed convection (Nu_m) relative to the same baseline (Nu_n). The subscript ‘m’ in Nu_m conveniently denotes both ‘measured’ and ‘mixed’. The Péclet number is defined as

$$Pe = \frac{ux}{\alpha} \quad (5)$$

where u is the area-averaged velocity, that is, the bulk airflow Q (m^3/s) divided by the area of the porous panel or wall. Note that the Péclet number is also the product of the Prandtl number multiplied

by the Reynolds number. The ratios $\sqrt{Pe}/Nu_n$ and Nu_m/Nu_n were chosen after consulting [44,sec. 4.10], [36,chap. 8] and [39].

In suction mode, the total convective heat transfer on the surface is enhanced ($Nu_m/Nu_n > 1$) with increasing air flow ($\sqrt{Pe}/Nu_n > 0$). In blowing mode, the heat transfer on the surface is mostly reduced ($Nu_m/Nu_n < 1$). At the saddle-point, when the normalized flow rate is ($\sqrt{Pe}/Nu_n \approx 0.5$), the convection heat transfer at the surface reaches practically zero.

The correlating equations shown in Table 2 were made following the method of Churchill and Usagi [45]. These correlations can be used to estimate the local heat transfer coefficient in a wide range of scenarios. For instance, if u , x , T_{ai} and T_{si} are known, then the local heat transfer coefficient h_c at the interior surface can be calculated by as follows:

- Evaluate β at $T = T_{ai}$ and (Pr, α, ν, k_a) at $T = (T_{si} + T_{ai})/2$
- Calculate Ra_x , Nu_n , Pe , Eqs. (2), (1) and (5), then $\sqrt{Pe}/Nu_n$
- Calculate Nu_m/Nu_n , Table 2, or read directly from Fig. 5
- Calculate h_c from Nu_m , Eq. (3)
- Estimate h_r , Eq. (4)

The first two steps follow the procedure for calculating Eq. (1). That is, the Squire-Eckert prediction for natural convection on a vertical, isothermal plate, as described by Lienhard and Lienhard [43,section 8.3]. Note that these results apply in heating or cooling, that is, for $T_s > T_a$ or $T_s < T_a$.

2.3. Discussion

The results show several distinct modes of convection heat transfer. What changes physically from one mode to the other? And how generally can the correlations be applied?

2.3.1. Suction mode (exterior surface)

Boundary layer theory [46] predicts that thinner boundary layers have lower thermal resistance. Fig. 4 shows the boundary layer getting thinner with increased suction, while Fig. 5 confirms this corresponds with an increase in heat transfer, by up to factor 5. Fig. 4 also shows that as suction increases, the surface film gets redirected into the channels. This has two implications for design and the results presented in Section 3.

First, any heat that is not recovered inside the porous material may be recovered at the exterior with the incoming air. Second, the increased rate of surface heat transfer can be exploited for pre-heating or pre-cooling the incoming air. This can be done, for instance, by putting the exterior surface in radiant communication with the sun, a technique known as transpiration solar heating [47–49]. And in principle, it is possible to pre-cool the air by putting the exterior surface in radiant communication with the sky-dome [50–52]. Radiative cooling is most easily done at night, but it can be

Table 2
Correlating equations for the mixed convection measurements. In accordance with the axes of Fig. 5, $x = \sqrt{Pe}/Nu_n$ and $y = Nu_m/Nu_n$. The subscripts ‘s’ and ‘b’ mean ‘suction’ and ‘blowing’ respectively. BL means ‘natural convection boundary layer’.

Range	Correlating equation	Description
$0 \leq x \leq 0.66$	$y_{s1} = 4.66x^2$	BL made thinner by suction, natural convection augmented. Forced convection dominates.
$0.66 \leq x \leq 1.2$	$y_{s2} = 1 + 3x$	
	$y_s = (y_{s1}^{-100} + y_{s2}^{-100})^{-1/100}$	
$0 \leq x \leq 0.5$	$y_{b1} = \frac{e^6}{e^{50x/3} + e^6}$	Incoming air mixes with BL, hence $Nu_m \rightarrow 0$ as $\Delta T \rightarrow 0$. Interior air entrained into BL, hence $Nu_m/Nu_n \rightarrow 1$. Transition (unsteady). Forced convection dominates, turbulence entrains interior air.
$0.5 \leq x \leq 0.66$	$y_{b2} = 28x^8$	
$0.66 \leq x \leq 0.9$	$y_{b3} = 1$	
$0.9 \leq x \leq 1.1$	$y_{b4} = 1.25x^{7/2}$	
	$y_{b1 \rightarrow 2} = (y_{b1}^3 + y_{b2}^3)^{1/3}$	
	$y_{b1 \rightarrow 3} = (y_{b1 \rightarrow 2}^{-100} + 1)^{-1/100}$	
	$y_b = (y_{b1 \rightarrow 3}^{10} + y_{b4}^{10})^{1/10}$	

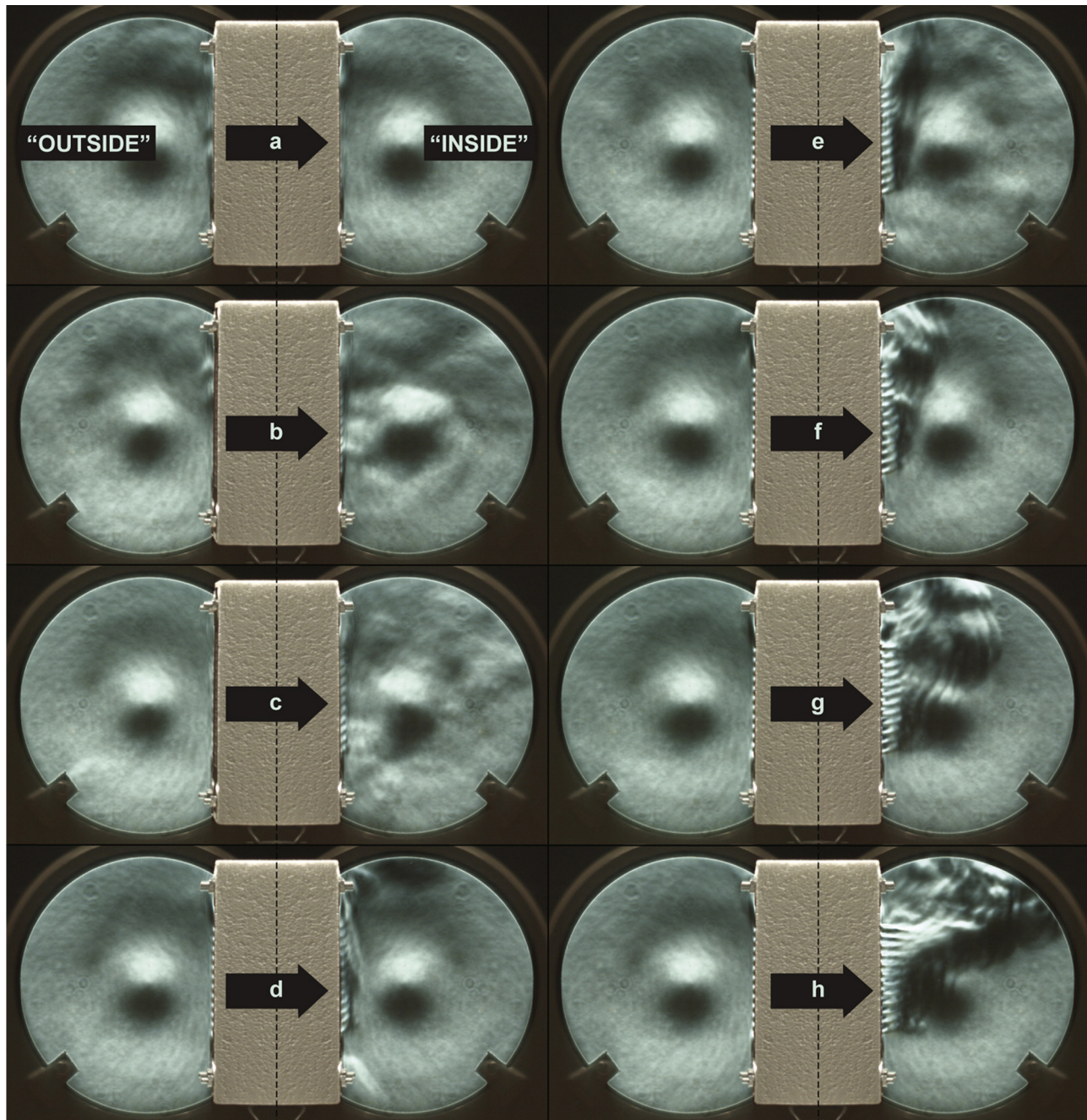


Fig. 4. Schlieren images of mixed convection on the surface of a heated porous panel with increasing suction ('exterior') and blowing ('interior'). The images for blowing and suction were taken separately, then combined into a series of mirrored images to suggest what happens across a single wall at different flow rates; there was only one Schlieren mirror. See Table 1.

also be done during the day, by using a spectrally-selective surface [53,54], or by using compound parabolas to enhance the radiator's view of the cold part of the sky-dome while blocking direct sunshine [50].

2.3.2. Blowing mode (interior surface)

See the right hand side of Fig. 4c, the third row of Table 5, the saddle-point in Fig. 5 and consider the limiting case when Nu_m/Nu_n approaches zero as $\sqrt{Pe}/Nu$ approaches one half. The surface is flooded by incoming air jets that have been heated on their way through the panel. These jets have displaced the natural convection boundary layer, creating an intermediate air-film. The temperature difference between the surface and this air-film approaches zero.

So too, therefore, does the convective heat transfer. Past the saddle-point ($\sqrt{Pe}/Nu_n > 0.5$), interior air gets entrained to the surface and convection heat transfer between the surface and the room ensues.

The same saddle-point is possible without active control of the surface temperature. A natural convection boundary layer will form so long as there is a temperature difference between the interior surface and the interior air (if $T_s < T_a$, the boundary layer falls instead of rising). And incoming jets may still flood the surface and overwhelm that boundary layer. The only difference is that the air jets do not approach the temperature of the surface; instead, the surface temperature tends to fall to the exterior temperature. Either way, the end result is the same: the convective coupling between surface and interior is partially or significantly reduced.

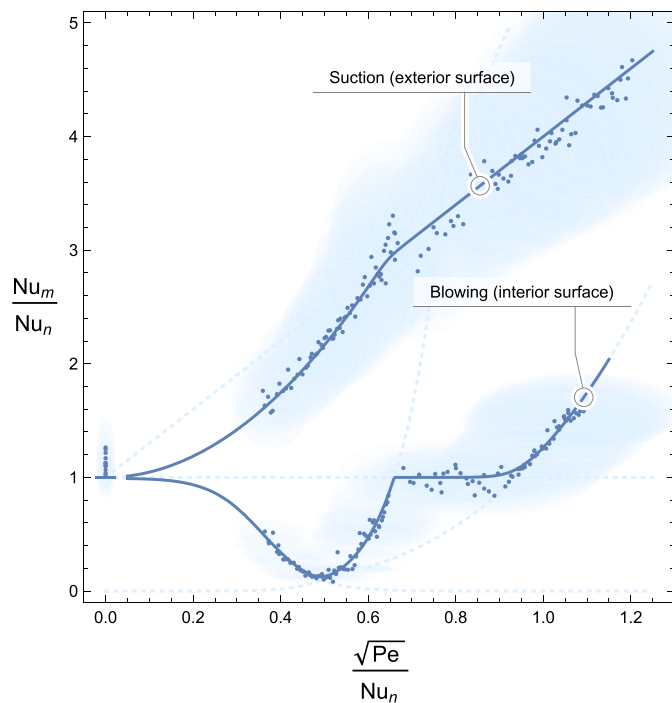


Fig. 5. Measurements of mixed convection (y-axis) on a heated porous surface as a function of increasing air-flow (x-axis). The clouds show the standard error. See Table 2.

2.3.3. Dynamic insulation

Do these results predict the heat transfer on the interior surface of *dynamic insulation*? Dynamic insulation does not have an actively heated surface, and the pores are much smaller and more closely packed than in the test plate. However, if our results have been properly correlated, the predictions should translate, so it is worth comparing with published data.

One research group [7] took dynamic insulation measurements in a small residential installation in Japan. The dynamic insulation wall was 1.66 m high. They sucked air through the dynamic wall at an average rate of $1.5 \text{ m}^3/\text{m}^2 \text{ h}$ ($u = 0.042 \text{ m/s}$) over a period of 24 h. During this time, they measured the average interior surface temperature and the average air temperature. The surface was cooler by an average of 2.5°C . This compares to their prediction of 1.3°C , which was made, presumably, by applying the Taylor and Imbabi model [5]. What does this discrepancy indicate? That less heat was transferred from the room to the surface than expected.

The predicted decrement of 1.3°C corresponds to the baseline condition: $Nu_n/Nu_m = 1$. (Note that the numerator and denominator are flipped, because the surface is not actively heated and so cannot exceed the temperature of the room.) Viewed upside down, the Schlieren images of Fig. 4 show what happens when the surface is colder than the interior air. The surface temperature falls below the interior temperature by a combination of effects. The Taylor and Imbabi model [5] accounts for the effects associated with heat exchange inside the material. But it does not account for any of the effects the incoming air has on the boundary layer, as discussed in Section 2.3.2.

The heat transfer coefficient was not directly measured in the Japan study. However, the relative change can be estimated, by setting Eq. (1) as both numerator and denominator, and noting that, aside from the temperature decrements, all terms cancel. Hence $Nu_n/Nu_m \sim (1.3/2.5)^{1/4} = 0.85$. This indicates that the thermal contact between the room and the surface was reduced, on average, by approximately 15%. How well does this estimate correspond to a prediction made by the correlations presented in Table 2? Take the

properties of air (Pr, β, α, ν) at $T = 20^\circ\text{C}$, the measured average flow rate ($u = 0.042$), and the height of the middle seventh of the wall ($0.71 < x < 0.95$). The resulting values for the normalized flow rate range between $0.62 < \sqrt{Pe}/Nu_n < 0.65$. This corresponds to the right hand side of the saddle-point in Fig. 5, where the normalized heat transfer approaches the baseline sharply, rising by the eighth power. Using Table 2, the calculated range is $0.49 < Nu_n/Nu_m < 0.88$. This falls in line with the earlier estimate of $Nu_n/Nu_m \sim 0.85$.

In summary: a single calculation, based on our correlation, corresponds well with a single estimate, based on available data. This is far from conclusive evidence, though it does give some indication that our results can be used generally—whatever the type of porous material, and whether the surface temperature is controlled actively or not.

A thorough comparison would require measurements of the local heat transfer coefficient with and without suction. But this may be impossible to measure directly on materials with small and tightly spaced pores. Even a small heat flux sensor would block the pores and hence skew the results. Furthermore, the Japan study appears to be the only study in the literature that reports the interior surface temperature. Since that study, installations and test cells have tended to use plasterboard in front of the interior surface to create an air-mixing cavity [11,12,16–19]. This intervention only seems to lessen the impact of the wayward surface temperature rather than resolve the issue conclusively.

2.4. Conclusion

We characterized the heat transfer by taking convection measurements at the surface of a bespoke milli-fluidic panel with air holes while controlling the surface temperature and the air flow in suction and blowing. We also took Schlieren images to understand the physical relationship between natural and forced convection at the surface. On the interior face, natural and forced convection are in competition, and at a certain point, they cancel each other out, so there is no convective heat transfer between the room and the surface.

Before it can pass heat to the air flowing through it, a porous material must first receive heat at one face. So the performance of heat exchanging walls is contingent on the method of room heating, and how that heat reaches the interior surface. Remote radiant heating is one way of controlling the surface temperature. Another way is to integrate a closed-loop water circuit into the surface. This ensures direct thermal contact between the source of heating (or cooling), the interior environment, and the incoming air supply. The heated surface can temper the incoming fresh air, and, by radiation, the occupants. (In both experiments reported in this paper, we used a bespoke milli-fluidic panel and a fan. But in a real building, proprietary capillary tube mats [55,56] could control the surface temperature, and buoyancy pressure up the height of the building could pull the air through the pores.) The next section shows how to optimize porous materials so they can efficiently exchange heat from a tempered surface to incoming air.

3. Experiment II: heat exchange inside designed porous materials

In the previous section, measurements were taken to characterize the mixed convection heat transfer on the surface of porous materials. Among other things, the results showed that, for dynamic insulation to work properly, the interior surface temperature must be carefully controlled. One way to improve the overall performance is to integrate a hydronic circuit into the surface while designing the porous material to perform explicitly as a heat exchanger. This section details an experiment that validates

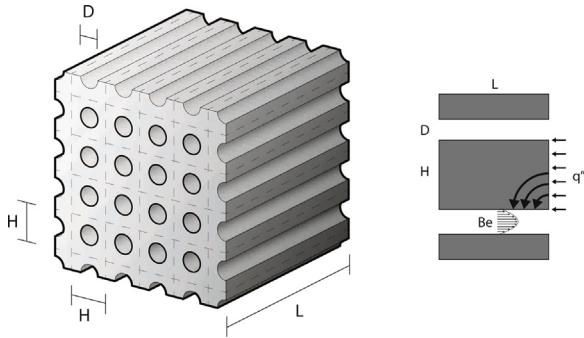


Fig. 6. Geometry definition.

a method for designing porous materials as heat exchangers. Any base material can be used in principle, so long as it can be manufactured with millimeter-scale air channels.

3.1. Theory

Fig. 6 defines the geometry for a simple kind of ‘designed porous material’. Kim, Lorente and Bejan [21] showed how to optimize this geometry for the purpose of ‘vascular’ cooling. For example, to optimize the porosity of a turbine blade, so that fluid from inside the blade could be pushed through the channels at a fixed design pressure, to stop the blade from overheating and so avoid catastrophic creep.

Their aim was to optimize this geometry for ‘contraflux’ heat-exchange. That is, to minimize the hotspot temperature for a fixed pressure drop (ΔP) and a given thickness (L), thermal conductivity (k_s) and void fraction (ϕ). The exterior surface was subjected to a constant heat flux, so the hotspots occurred on that same surface, equidistant from the pores, where coolant—any gas or fluid—was discharged. First they used scale analysis and the intersection of asymptotes method [35] to define the important parameters and their relations. Then they undertook a series of numerical simulations to test and calibrate the results of their initial analysis. They concluded by presenting a numerically validated equation linking the driving forces and thermal properties to the optimal geometry:

$$\frac{H_{opt}}{L} = 3.22Be^{-1/3}\phi^{-0.85}\left(\frac{k_s}{k_a}\right)^{0.17} \quad (6)$$

and an equation that predicts the total thermal conductance of an optimal panel:

$$\frac{q''L}{\Delta T_{min}k_s} = 0.41Be^{1/3}\phi^{0.6}\left(\frac{k_s}{k_a}\right)^{-0.65} \quad (7)$$

where the void fraction (i.e. porosity) is defined as:

$$\phi = \frac{\pi D^2}{4H^2} \quad (8)$$

and Be is the Bejan number, a dimensionless pressure drop number that accounts for the heat carrying capacity of the fluid:

$$Be = \frac{\Delta PL^2}{\alpha\mu} \quad (9)$$

These results are valid so long as the panel is be thicker than the half spacing of the pores ($L > H/2$). Note that the left hand side of Eq. (7) gives the ratio of the total conduction coefficient ($q''/\Delta T$) normalized to (i.e. divided by) the baseline conduction coefficient at zero mass flow (k_s/L). They refer to this ratio as the ‘global’ thermal conductance (or it is inverse, the ‘global’ thermal resistance). The subscript ‘min’ beside ΔT in Eq. (7) refers to the fact that by optimizing the geometry, the hotspot temperature difference has been minimized. That is, the difference in temperature between the

coldest spot in the material (at the fluid inlet) and the hottest part (at the outer surface). In order to switch contexts and apply these results for the design of porous building envelopes, we note that Eq. (7) may also be written as:

$$\left(\frac{q''L}{\Delta T k_s}\right)_{max} = 0.41Be^{1/3}\phi^{0.6}\left(\frac{k_s}{k_a}\right)^{-0.65} \quad (10)$$

where

$$\Delta T = T_{si} - T_{ae} \quad (11)$$

The heat-flux q'' is no longer fixed. Instead, T_{si} is fixed (by the integrated water circuit) while the total heat-flux q'' varies (it increases with the air flow). The subscript ‘max’ denotes that the total heat transfer is maximized (since the panel is optimized). It is worth emphasizing a basic principal of heat exchanger design here: To recover more heat in proportion, more heat must be transferred in total.

The left hand side of Eq. (10) has the same form as the Nusselt number (which gives the ratio of total heat transfer normalized to the small limit of pure conduction). It also has the same form as the Number of Transfer Units, NTU , used in heat exchanger theory (which gives the total heat transfer normalized to the heat carrying capacity of the weakest heat flow). Therefore:

$$NTU_{max} = 0.41Be^{1/3}\phi^{0.6}\left(\frac{k_s}{k_a}\right)^{-0.65} \quad (12)$$

The theory of heat exchanger design is well established and solutions describing the performance of different types are widely available [43,57,58]. The assumption of a fixed and uniform T_{si} implies that the heat capacity rate of the water flowing through the interior surface is significantly larger than that of the air passing through the channel-pores. In this limit, the fraction of exchanged (i.e. recovered) heat, ε , increases asymptotically to unity as a function of the Number of Thermal Units, NTU :

$$\varepsilon = 1 - \exp(-NTU) \quad (13)$$

If, however, the heat-capacity rates of the two flows are similar, then T_{si} will vary across the surface, since the outlet temperature of the water-circuit will be cooler than the inlet temperature. In this case, a different ε - NTU solution to Eq. (13) should be used—one that describes the performance of a cross-flow heat exchanger. Eq. (12) will still hold.

Knowing how ε varies with NTU , the coefficient of conduction losses to the exterior can be described as an effective U -value:

$$U^* = \frac{k_s}{L}(1 - \varepsilon) \quad (14)$$

And the temperature of the incoming air-jets (T_{ai}) may be calculated as:

$$T_{ai} - T_{ae} = \frac{\varepsilon q''}{u\rho c_p} \quad (15)$$

where u is bulk air flow rate per unit area, as defined in Eq. (5). Note that as $\varepsilon \rightarrow 1$, $U^* \rightarrow 0$ and $T_{ai} \rightarrow T_{si}$. By design, one should ensure that the conduction losses are less than the losses through a standard and well insulated envelope. Furthermore, the heat exchanging wall should be sized according to the ventilation requirements of the space it serves. The air flow rate through one channel, Q_n , can be calculated using the Hagen-Poiseuille equation [59]:

$$Q_n = \frac{\pi D^4 \Delta P}{128\mu L} \quad (16)$$

There are $n = 1/H^2$ channels per unit area of wall. So the air flow rate per unit area of wall can be calculated as:

$$u = \frac{\pi D^4 \Delta P}{128\mu L H^2} = \frac{D^2 \phi \Delta P}{32\mu L} \quad (17)$$

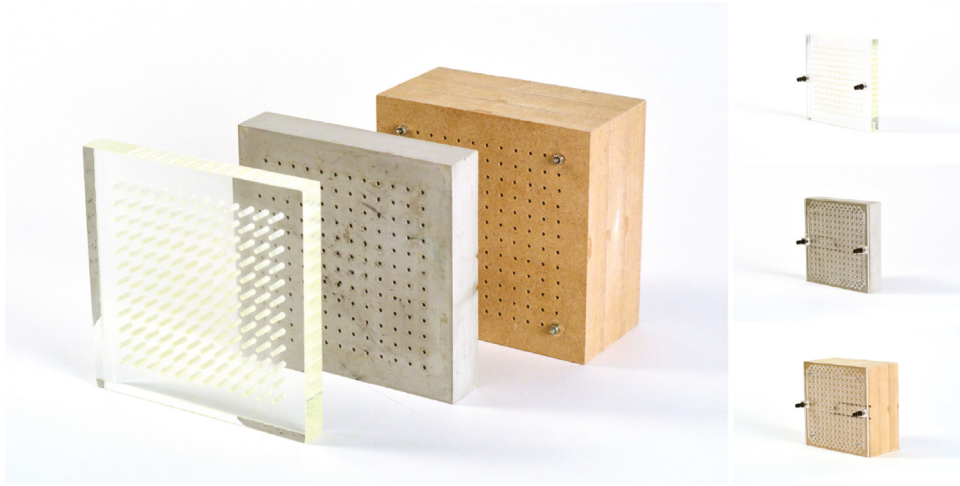


Fig. 7. Test panels, with and without the milli-fluidic panel of Fig. 2 attached. See Table 3 for specifications.

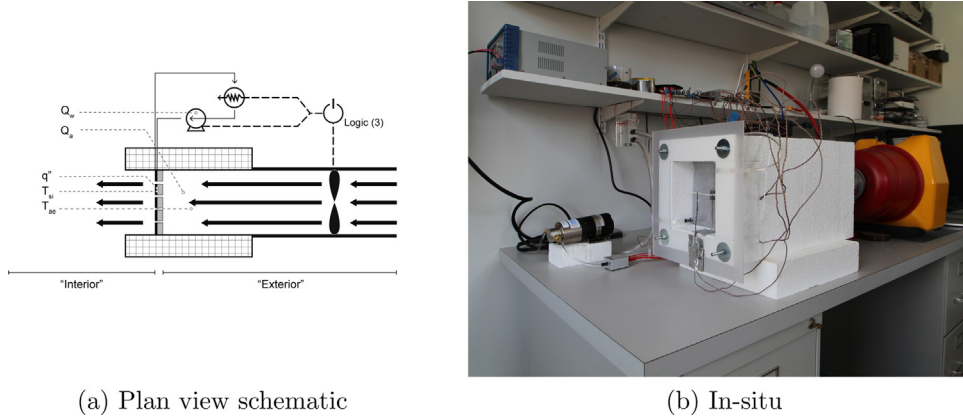


Fig. 8. Experimental set-up for heat transfer and total heat recovery measurements.

The size of the heat exchanger can then be tied to the required ventilation rate according to:

$$A_{req} = \frac{pQ_p}{u} \quad (18)$$

where A_{req} , p , and Q_p are, respectively, the necessary size of the heat exchanger, the number of people, and the ventilation rate per person. Finally, the suction pressure ΔP can be sustained mechanically, by a fan, or naturally, by the stack effect. Assuming the resistance of the exhaust is much smaller than the resistance of the heat exchanger, the stack pressure sustained by a chimney or tall room (with the heat exchanger at low level) is:

$$\Delta P = g\beta\rho\Delta T\Delta Z \quad (19)$$

where ΔZ is the height between the middle of the heat exchanger and the middle of the exhaust, and ΔT is the temperature difference between inside and outside, assuming the interior air is well mixed. This can be linked directly to the performance of the heat exchanging wall by substituting Eq. (15):

$$\Delta P = g\beta \frac{\varepsilon q''}{uc_p} \Delta Z \quad (20)$$

It is straightforward to add extra terms to the heat balance (interior heat loads, convective and radiant heating from heated surface, conduction losses from the non-permeable portions of the envelope, etc) and the pressure balance (wind pressure, exhaust resistance, etc). However, this simple formulation is enough to

Table 3

Specifications for test panels. The thermal conductivities k_s were estimated using data tables [60] and the uncertainty was taken into account in the measurements.

Material	ΔP_{opt}	k_s	H (mm)	ϕ	D (mm)	L (mm)
Cement	7	2.0 ± 0.4	10	0.05	2.52	29
Glass	4	1.15 ± 0.1	10	0.05	2.52	20
MDF	7	0.3 ± 0.1	10	0.05	2.52	75

show that, in principle, the entire system could be regulated by the heat of the hydronic circuit.

3.2. Materials and methods

To formulate Eqs. (6) and (7), Kim et al. [21] used the *intersection of asymptotes* method [35], before carrying out numerical simulations to calibrate the exponents. We undertook an experiment to validate the results and evaluate if they could be used for building envelope (or floor or ceiling) design.

Fig. 7 shows the three test panels which were designed to test the predictions of Kim et al. [21] while fitting the milli-fluidic panel shown in Fig. 2. Table 3 shows the design specifications. The glass panel was fabricated using a pre-cut pane of borosilicate glass (20 mm thick) and a CNC water jet was used to cut the channels. The ‘concrete’ panel was made using Rockite mortar cement poured in a bespoke, 3D printed (Stratys Connex 500) mold. The ‘wood’ panel was made from three panels (25 mm thick) of medium density

Table 4

Example test data. The middle test runs are at the design pressures.

	Test run	q''	ΔT	$q''/\Delta T$	ΔP	u	$Be^{1/3}$
Cement	1.	293.82	8.87	33.12	1.89	0.03	174.84
	3.	295.55	5.48	53.98	6.88	0.08	224.07
	8.	298.33	4.51	66.11	29.88	0.23	438.71
Glass	1.	312.06	6.57	47.49	1.97	0.04	138.41
	2.	312.64	5.25	59.53	4.12	0.07	176.99
	8.	312.22	3.94	79.3	30.01	0.25	342.94
MDF	1.	281.27	20.18	13.94	1.85	0.01	327.24
	3.	286.24	13.39	21.38	6.91	0.05	507.36
	8.	293.97	8.64	34.01	30.03	0.15	828.

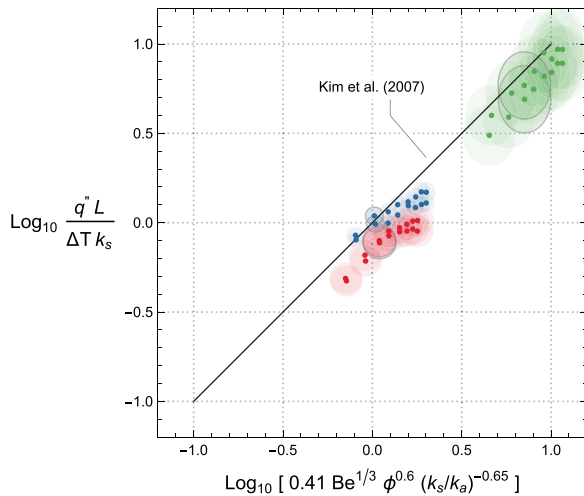


Fig. 9. Normalized measurements for total heat exchange as a function of increasing air-flow (Be). The red, blue and green data points are for the concrete, glass and wood test panels, respectively. The colored clouds show the standard error. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of the article.)

fibre-board (MDF), which were individually drilled and fastened together mechanically.

The experimental set-up is shown in Fig. 8. The milli-fluidic panel (Fig. 2) was mechanically fastened to each test panel in turn (see Fig. 7) and housed in an open-face enclosure made of insulating styrofoam (60 mm thick). The side with the milli-fluidic panel was exposed to the room (which represented the ‘interior’, while the enclosure represented the ‘exterior’). The same fan-blower (RetroTec 300 DucTester) was fixed to the back of the enclosure to incrementally increase the air-flow while a pitot-tube assembly was fitted to either side of the panel to monitor the pressure difference so it could be measured, logged and controlled with a flow-gauge (RetroTec DM32). To measure the heat-flux entering the material, a heat-flux sensor (greenTeg GSkin) and thermocouple (Omega, k-type) were fitted between the milli-fluidic panel and the test panel. They were placed in notches that were CNC machined to ensure a flush fit and thermal paste was applied to ensure thermal contact. The heat-flux sensor (1 cm × 1 cm) was arranged diagonally to cover a unit area while not blocking the airflow. Another thermocouple was installed inside the enclosure to measure the temperature of the ‘exterior’.

The data-logger (Graphtec GL220) was connected in a feedback loop with the heat-flux sensor, a gear pump (Micropump, DC306A drive, GB-P23.PDS.A pump-head), a bespoke water immersion heater, and a micro-controller (Arduino, ATmega32u4). An algorithm was written to control the temperature and flow-rate of the water so that a constant heating flux was maintained during each incremental measurement. The heat capacity rate of the

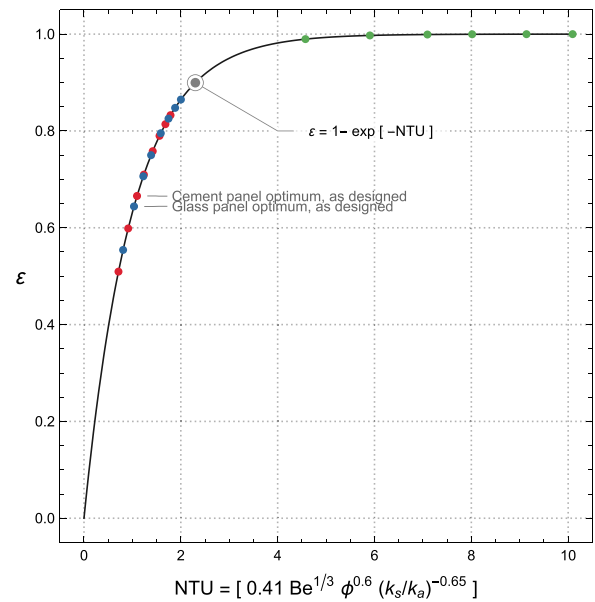


Fig. 10. Normalized predictions for heat recovery (ϵ) as a function of increasing air-flow (Be). The red, blue and green data points are for the concrete, glass and wood test panels, respectively. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of the article.)

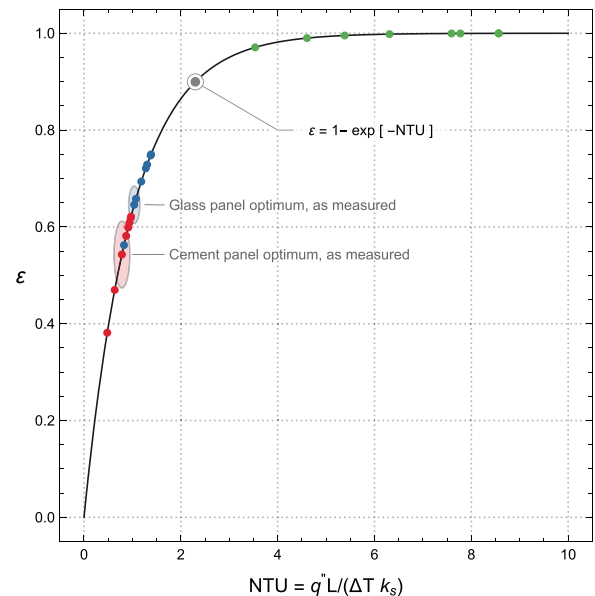


Fig. 11. Normalized measurements for heat recovery (ϵ) as a function of increasing air-flow (Be). The red, blue and green data points are for the concrete, glass and wood test panels, respectively. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of the article.)

water-flow rate was set so that it was at least double the heat capacity rate of the air-flow. This ensured that the milli-fluidic panel was isothermal, as confirmed by an infrared camera. The pressure was incrementally increased and heat-flux measurements were taken after steady-state was achieved. Two rounds per test panel were undertaken. Table 4 gives a sample of the operating conditions for some of the tests.

3.3. Results

Fig. 9 shows the normalized measurements for total heat transfer. The x- and y-axis correspond to the logarithm of each side

of Eq. (10), which is also plotted on the graph. As the pressure is increased, Be increases, and so therefore does the total heat transfer coefficient, $q''/\Delta T$. The red, blue and green data points are for the concrete, glass and wood test panels, respectively. The glass and concrete samples have a similar conduction coefficient (k_s/L) while the wood sample is higher up the slope because its conduction coefficient is lower by an order of magnitude. The colored clouds show the standard error. The framed clouds highlight the measurements taken at the pressure for which the panels were optimized for (see Table 3). In theory, these data points should have the closest fit with Eq. (10).

The center of the graph (0,0) corresponds with $NTU = 10^0 = 1$. At this point the heat capacity rate of the air-flow matches the conductance of the test sample. The concrete and glass samples were optimized at this tipping point. Beyond this point, the concrete and glass data begin to plateau with respect to the slope. This is because the thermal boundary layers inside the air channels no longer perfectly merge, so the panel is now too thin to recover all the heat it can.

The data in Fig. 9 correlate well with the predictions of Eq. (10). The data for the concrete sample do not meet the slope but they are still within the bounds of error that Kim et al. [21] found when they correlated a wide range of virtual test samples to arrive at Eq. (10).

Fig. 10 shows the heat-recovery rates as predicted according to Eq. (13). Compare this with Fig. 11, which shows the results as measured. For clarity, the average results over both rounds of testing are shown. The predicted and measured heat recovery at the design operation point are highlighted for the concrete and glass panels (the design operation point for the wood sample is the third data point from the left). The data at this point match reasonably well, and less so at higher air-flow rates. This simply shows a different view of the data in Fig. 9 and the impact of the plateau in performance beyond the optimum operating point. This discrepancy is negligible so long as the target heat-recovery at the optimum operating point is sufficiently high.

3.4. Discussion

The experimental data agree well with the correlation defined in Eq. (10), as formulated by Kim et al. [21]. This suggests that Eqs. (6), (12) and (13) can be used with confidence to optimize porous materials for a target heat exchange efficiency, ε . This is true only for sensible, steady state heat transfer. And only if the interior surface temperature is controlled (see Section 2). The wood panel achieved heat recovery rates close to 100% while the concrete and glass samples achieve rates closer to 60%. But that is besides the point. In principle, the concrete and glass panels could have been designed to achieve any NTU and therefore any heat exchange efficiency. We designed the panels to validate a correlation, not to achieve high efficiency across the board. (The size of the heat-flux sensor determined H and ϕ , and therefore the size of the milli-fluidic panel. Following that, we decided to keep the design pressure approximately the same for each panel. This determined the thickness of the panels and constrained the NTU .)

At any time during operation, the air flow rate may be lower or higher than the design value. This will affect the heat exchange efficiency, because the NTU varies with the air flow rate. However, if a sufficiently high NTU is chosen for the design value, the variations in heat exchange efficiency will be negligible. This is true even though Eq. (12) fails to predict the NTU above the design value.

3.5. Conclusion

The analysis and numerical correlations of Kim et al. [21] have been experimentally validated, and interpreted so they can be

applied in a building design context, suggesting how building envelopes can be designed as heat exchangers. Wood, glass and concrete test samples were made and the measurements showed good agreement with theory. The results were presented according to the conventions of the ε - NTU method, so that designers can predict the steady-state heat-recovery of any optimal design in sensible heat-transfer mode. Practically any base material can be used, so long as it can incorporate millimeter-size channels.

Any concept designs based on these results should incorporate a means of delivering heat to the interior surface, as demonstrated by the experimental results reported in Section 2. The channel-pores are optimized to recover heat according to the thermal conductivity of the base material, the dimensions of the panel, and the pressure difference across the panel. Fresh air is sucked through the channel-pores by a fan or a chimney, while a closed-loop water circuit integrated at the interior surface of the panel can control the temperature. The interior space is tempered radiantly while the incoming fresh air is pre-heated or pre-cooled. The envelope heat losses are close to zero when the heat-recovery is close to unity.

4. Outline of further research

There are a number of outstanding thermal and practical issues which must be evaluated before full-scale installations can commence or design guidance can be issued. These concerns and opportunities are dependent on the choice of material and the particulars of the site, project, program and climate. In this section we outline the most important ones while highlighting avenues for further research.

4.1. Weather- and wind-proofing

In most cases, an exterior screen or rainscreen facade would be necessary for weatherproofing. The rainscreen could be designed following the principles of pressure-equalization [61,62] to reduce or eliminate the deleterious effects of buffeting or downwind suction on the overall heat-recovery. Having the exterior screen or rainscreen panels made of glass would allow for solar-transpiration heating while reducing radiant losses. In some cases, an external screen may not be necessary—for instance, if the walls faced a covered courtyard.

4.2. Filtration, cleaning and anti-fouling

Air filters could be incorporated into: the rain-screen system (between panels); at the bottom of a vertical cavity made by an exterior screen; or integrated into the exterior surface of the porous material. The channels could be tilted slightly to resist water ingress and the channel diameter and spacing could be tailored to accommodate a multi-pin cleaning tool. Anti-fouling surface technologies (such as 'omniphobic' SLIPS [63,64]) could be applied inside the channels to resist microbial adhesion and repel liquids. For instance, researchers have demonstrated that wax-based treatments on wood can repel water while allowing vapor to pass, so that wood products can be protected from degradation while still retaining their moisture-buffering properties [65].

4.3. Vapor transfer and latent heat exchange

The possible modes of enthalpy transfer are an important area for further research. For instance, condensation and moisture-related problems can occur when building materials and envelopes are not sufficiently ventilated. In contrast, the porous materials proposed in this study are, by default, well ventilated. They also have an actively heated interior surface. Together, these two features may facilitate faster and more effective drying after periods of

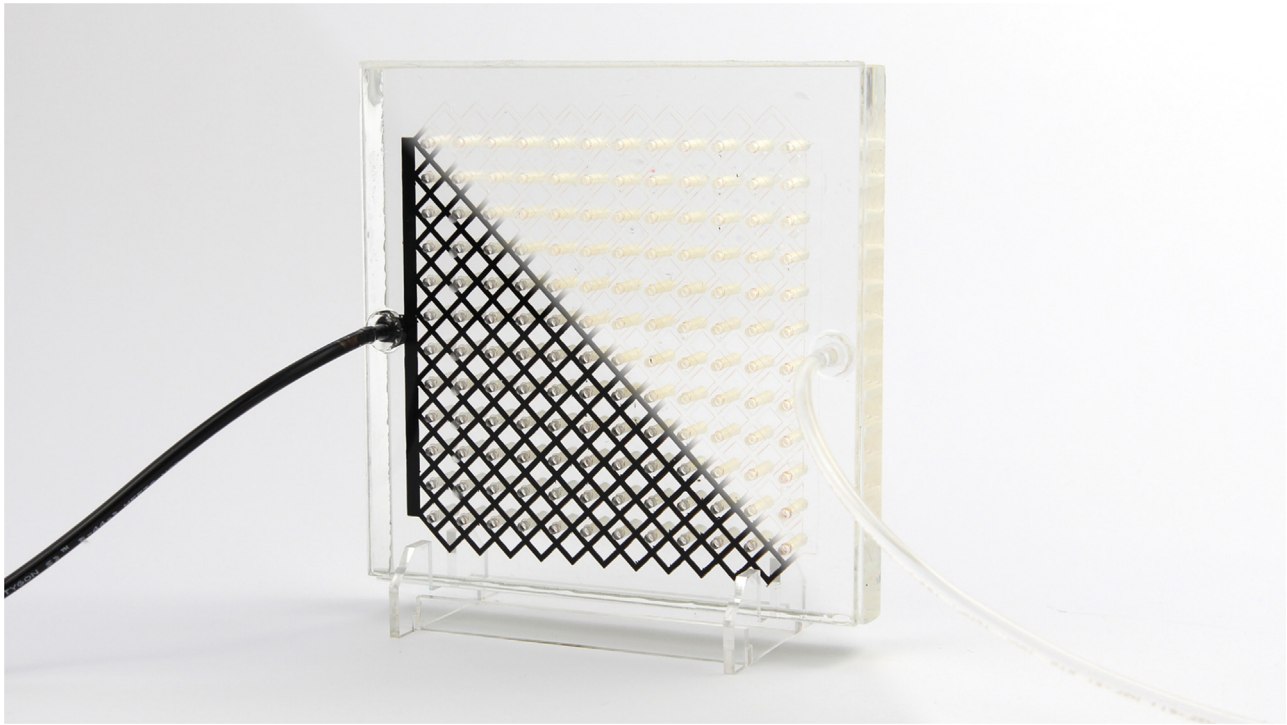


Fig. 12. A composite image of the glass test panel, with the hydronic circuit directly laminated on to the surface, using a foamed acrylic adhesive interlayer and an additional pane of glass. In one half of the image, black ink fills the hydronic circuit, which is otherwise barely visible. Glasses and plastics offer transparency and the possibility of additional solar heating of the air-stream from within the material.

high humidity. Another possible application is moisture-buffering. Moisture-buffering occurs when a porous material, in net effect, absorbs moisture, leading to a stabilized material temperature and reduced relative humidity inside a room [66–68]. Once sufficiently dried, and depending on the type of base material used, moisture from the interior might be passively absorbed into the heat-exchanger wall. If the drying cycles of two or more walls were synchronized at alternate periods, the moisture-buffering effects could be modulated to improve occupant comfort and reduce energy use. Phase-change materials [69–71] might also be incorporated and run in a similar manner—that is, in alternate cycles of charging and discharging. The performance of heat-exchangers with phase-change can be analyzed using the ε -NTU method and ‘pinch-point’ analysis [57].

4.4. Transient heat transfer (and other geometries)

After their 2007 study [21], Kim et al. went on to evaluate the performance of tree-like, ‘dendritic’ configurations [72]. They concluded that the additional complexity was worthwhile so long as the available driving pressure was sufficiently high (too high for buildings, in our reading). They characterized the transient response of this more complex design in a follow-up paper [73]. The same method might be applied to the simpler channel configuration studied in this paper, in order to characterize the transient response and evaluate the possibility for short term thermal storage (e.g. pre-cooling of a massive envelope with night ventilation). Alternatively, the standard ε -NTU method may be used with some adjustments to characterize the effects of thermal storage (i.e. ‘passive regenerators’) [57].

4.5. Buoyancy ventilation and ventilation heat recovery

Most examples of dynamic insulation have been installed as part of a mechanical air-supply system, which in principle offers reliable

control of the interior pressure and the opportunity for ventilation heat-recovery at the exhaust. Etheridge and Zhang [74] evaluated the use of dynamic insulation with natural ventilation, focussing mainly on the wind-driven kind. More recently, there have been important advances in the understanding and application of buoyancy ventilation [75], where interior heat loads provide the driving force instead of wind (which can be unreliable). Researchers have solved buoyancy design problems such as how to pre-heat incoming air in winter [76], how to pre-cool incoming air in summer [77], and how to vary openings in multi-story buildings so that all occupants receive the same amount of fresh air [78].

Eqs. (17)–(20) show how the design of the heat exchanging wall can be tied to the ventilation requirements of the building, and the motive force of buoyancy. During operation of the building, the ventilation flow rate can in principle be controlled by the varying heating power of the water circuit or by varying the opening at the exhaust. Heat could be recovered at the exhaust, but probably not without a reduction in the ventilation flow rate; an optimal balance can be found. It is worth noting that exhaust heat recovery may not be necessary in temperate climates and seasons. If the difference in temperature between the water and the exterior is sufficiently small, the enthalpy losses from the chimney may be acceptable.

4.6. Low-grade heating and cooling

Designing building envelopes as heat-exchangers raises prospects for better exploitation of low-grade heat. Heat is defined as ‘low grade’ if it is marginally hotter or colder than the surrounding environment [79]. If more of the world’s heating and cooling was done using low-quality energy, there would be less need for combusting fossil fuels. The larger the exchange surface compared to the room, the smaller the temperature lift needed to heat the room and perform thermodynamic work (such as ventilation by buoyancy) [22]. The interior surface could be heated or cooled using a low temperature-lift heat-pump connected to the ground [80]. Or

a buffer space could be built that follows the ground temperature instead of the exterior air temperature [81]. The opportunities for integrating solar transpiration heating and radiative cooling were discussed in Section 2.3.1 (Fig. 12).

5. Conclusion

This paper described a material design method to optimize the porosity of building envelopes so they can be designed as heat exchangers. In principal, any heat exchange efficiency can be designed for, and any base material can be used (so long as it can be manufactured with millimeter scale channels). Extra functions, such as transmitting daylight or transferring structural load, can be achieved by selecting the appropriate base material. Weather-proofing, wind-buffering and air-filtering can be done by a specially designed external rain-screen (made from the same material, perhaps).

Using the numerical correlations of Kim et al. [21], the channel-pores are optimized to recover heat according to the thermal conductivity of the base material, the dimensions of the panel, and the pressure difference across the panel. Fresh air is sucked through the channel-pores by a fan or a chimney, while a water circuit integrated at the interior surface of the panel controls the temperature. The interior space is tempered radiantly while the incoming fresh air is pre-heated or pre-cooled. The envelope heat losses are close to zero when the heat exchange efficiency is close to unity. The larger the exchange surface compared to the room, the smaller the working temperature gradient—and with careful design, this may translate, overall, to lower energy, enthalpy and exergy losses.

An experiment was undertaken to empirically validate the results of Kim et al. [21] and show they could be used for building envelope design. Wood, glass and concrete test samples were made and the measurements showed good agreement with theory. The results were presented according to the conventions of the ε - NTU method, so that designers can predict the steady-state heat exchange efficiency of any optimal design in sensible heat transfer mode.

The mixed convection at the interior and exterior surface can take on a number of complex forms, depending on the strength of blowing or suction, and the strength of buoyancy from the heated or cooled surface. It was mischaracterized by dynamic insulation researchers and this seems to explain the low interior surface temperatures found in full-scale installations. An integrated water circuit solves this problem. Schlieren imaging was used to characterize the different regimes of mixed convection, and surface heat-flux measurements were correlated in a general form so that designers can accurately estimate the convection heat transfer coefficient at the interior and exterior surface. These are needed to size the hydronic circuit and to calculate, for instance, the potential for preheating by solar transpiration.

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