Optimal design of mass timber panels as dynamic insulation: simulations of steady and transient heat exchange

Anna Halepaska, Salmaan Craig
Peter Guo-hua Fu
School of Architecture, Faculty of Engineering, McGill University, Montreal, QC, Canada
anna.halepaska@mail.mcgill.ca

Abstract. Mass timber panels could be designed as heat exchangers for use in building envelopes. Fresh air, drawn through geometrically optimized channels in the panel, is pre-tempered with building heat that would otherwise be lost to the exterior via conduction. Recent experiments have shown that timber heat exchanging panels can approach \( U \approx 0.1 \text{ W/m}^2\text{K} \) – but there are potential limitations. The sizing correlations which predict panel geometry and steady heat exchange must be numerically calibrated for building-scale contexts, the heat-exchange efficiency must be verified virtually, and practical thresholds for transient response time must be determined. This study uses numerical simulations to investigate these factors for one design ‘case’ of timber panels, and establishes a methodology for studies of further cases.

1. Introduction
There has been renewed interest in Dynamic Insulation (DI), as integrating thermoregulation and ventilation functions into porous building materials may provide a pathway to reducing embodied and operational carbon \([1,2,3]\). Recent research suggest that standard construction materials can be optimized for envelope heat-exchange, replacing external insulation with channels through the material at the millimeter scale so that outgoing conduction preheats the incoming ventilation \([1,4]\). Structural timber is highly suited to this kind of heat-exchange, as its low thermal conductivity makes it possible to balance outgoing conduction with relatively low ventilation rates. Recent experiments suggest that the dynamic \( U \)-value of mass timber panels optimized for steady heat-exchange can approach \( U \approx 0.1 \text{ W/m}^2\text{K} \) with ventilation rates under 10 l/s/m\(^2\)\([2]\).

However, there are potential limitations in achieving this performance. First, the sizing correlations which predict panel geometry and total heat-exchange are composed of dimensionless parameters. Their exponents likely need to be calibrated for the relevant range of parameters via numerical simulations. Second, while the correlations have been experimentally validated for both steady and transient total heat exchange, these experiments were not able to measure heat-exchange effectiveness directly due to practical limitations \([2,4]\). Therefore it is necessary to verify the heat-exchange virtually. Finally, the transient response of said panels may be a limiting factor, to avoid impractical thermal ‘lag’ in the system. Transient response is a function of thickness and thermal diffusivity; practical limits for both will need to be established, beyond which panels take longer than three hours to reach steady state.

This study consists of three sets of simulations. The first shows how numerical simulations of specific panel designs may be used to calibrate predictions for optimal geometry of and steady heat exchange through the panels. The second virtually measures the efficiency of steady heat-exchange with
the incoming ventilation, comparing simulations to predictions from standard theory. The third examines the transient response by virtually measuring the time-to-steady-state for panels designed at 10, 15, and 20 cm thicknesses and simulated using the minimum and maximum values of diffusivity for timber ($1 \times 10^{-7} < \alpha < 4.4 \times 10^{-7}$).

![Figure 1. Principle of optimization for heat exchange. a) geometry definition. b) balancing outgoing conduction with incoming convection.](image)

2. Theory

2.1. Design Correlations: maximizing heat exchange

Mass timber panels can, in principle, be designed as a form of DI. Fresh air, drawn through geometrically optimized channels in the envelope, is pre-tempered with thermal energy that would otherwise be lost to the exterior through conduction. Figure 1 illustrates the principle. Kim et al. developed two analytical correlations to describe the optimal geometry of a vascularized material heated from one side with coolant forced from the other [5]. The first correlation gives the optimal spacing of the channels ($H_{opt}$):

$$H_{opt} = \frac{3.22 Be^{-1/3}}{3 \Phi (k/k_f)^b}$$  \hspace{1cm} (1)

where $L$ is the panel thickness, and $k/k_f$ is the ratio of solid and fluid conductivities. The dimensionless pressure drop $Be$, is defined as

$$Be = \frac{\Delta P L^2}{\mu \alpha_a}$$  \hspace{1cm} (2)

where $\Delta P$ is the pressure drop along the channel, $\mu$ is the dynamic viscosity of the fluid, and $\alpha_a$ is its thermal diffusivity. The void fraction of the panel is:

$$\Phi = \frac{\pi D^2}{4H^2}$$  \hspace{1cm} (3)

where $D$ is the channel diameter. Geometry definitions are given in Figure 1. The second correlation gives the total heat transfer through the panel, given as the Number of Transfer Units (NTU):

$$NTU = 0.41 Be^{1/3} \Phi^a (k/k_f)^b$$  \hspace{1cm} (4)

Solving Eq.s 1-4 for given values of the global parameters $Be$, $\Phi$, and $k/k_f$ gives the optimal channel diameter and spacing so that heat exchange is maximized. Both correlations are robust with respect to $Be$ (consistently varying with $Be^{1/3}$) and qualitative trends for $\Phi$ and $k/k_f$. Kim et al. showed that more robust predictions for $\Phi$ and $k/k_f$ could be achieved by numerically fitting the exponents $a$ and $b$ for a relevant range of global parameters. However, universally fit exponents may not be achievable for the full range of building applications. The most robust predictions would be achieved
by numerically fitting the exponents for individual design cases: a unique combination of $\Phi$ and $k/k_f$ and a relevant range of Be. Since Be includes both pressure drop and thickness, a range of panels in each case may then be designed to achieve any range of NTUs by varying $\Delta P$ or $L$. Example design cases are given in Table 1, and calibration is described in Sections 3.1 and 4.1.

2.2. Heat Exchange Efficiency

Equation 4 may also be expressed in terms of the total heat transfer coefficient $q''/\Delta T$ normalized to the baseline conduction coefficient $k/L$:

$$NTU = \frac{q''}{k/L} \frac{(T_s - T_e)}{(T_s - T_e)}$$ (5)

where $q''_1$ is the fixed value for heat flux on the interior surface and $T_s - T_e$ is the temperature difference between the hottest point in the system (at the corner of $H \times H$ on the interior surface, c.f. Figure 1) and the coldest point (the fluid inlet). The sizing correlations maximize NTU by finding the optimal geometry where $T_s - T_e$ is minimized. Conduction losses are reduced as a portion of $q''_2$ is rerouted to the incoming air ($q''_2$), while the remainder is lost to the exterior ($q''_3$):

$$q''_1 = q''_2 + q''_3$$ (6)

The effectiveness of this heat-exchange with the air stream may theoretically be understood as the efficiency of a single-fluid heat-exchanger, given as a function of the overall heat transfer:

$$\varepsilon = 1 - e^{-NTU}$$ (7)

Alternatively, efficiency may be defined as the ratio of the temperature uplift of the incoming air to the overall temperature difference across the system:

$$\varepsilon = \frac{T_i - T_e}{T_s - T_e}$$ (8)

where $T_i$ is the temperature of the air as it enters the interior. Note that Equation 8 is an assumption for efficiency based on heat exchanger theory and has not yet been validated experimentally in the literature. Virtual steady-state measurements are described in sections 3.2 and 4.2.

2.3. Transient Response

The transient thermal response of a panel with parallel channels subjected to a step change in surface heating can be approximated as a function of the Fourier number $Fo$ [2]:

$$Fo = \frac{\alpha}{t \cdot L_c^2}$$ (9)

where $\alpha$ is the thermal diffusivity of the solid, $t$ the time in seconds, and $L_c$ the characteristic length.

$$L_c = \frac{\left(\frac{H^2 - \frac{\pi D^2}{4}}{L}\right) \cdot L}{2 \left(\frac{H^2 - \frac{\pi D^2}{4}}{L} + \frac{\pi D}{L}\right)}$$ (10)

The total heat transfer evolves similar to a plane wall, with correction factors $c_1$ and $c_2$ for shape effects from the vascular geometry:

$$NTU(t) = \left(c_1 \cdot NTU + \frac{c_2}{\sqrt{Fo}}\right) \frac{L}{L_c}$$ (11)

where NTU is the steady-state total heat exchange predicted by Eq. 4 for a given design case. The time to steady-state (where NTU'(t) is less than some reasonable percentage of the steady-state value) is therefore highly dependent on the thermal diffusivity and thickness of the panel material.

3. Methodology

Each set of simulations uses a 2D, axisymmetric finite element model simulated in a commercially-available FEM solver [6]. A fixed heat flux boundary ($q''_1$) is applied at the interior surface. A fixed-temperature boundary condition is applied at the exterior surface, set to the free-stream temperature.
The choice of boundary conditions was validated by comparing simulation results with experimental data for total heat exchange [2]. As the geometry is periodic, behaviour may be modelled as an axisymmetric volume consisting of the half-channel and -diagonal of the solid (Figure 1). Required mesh refinement is determined iteratively by increasing the number of elements in steps of 100 until simulated measurements for a selected spot temperature differ by <.5% between steps, replicating the method presented in [5].

Table 1: Example design cases for timber heat exchangers.

| Case | Φ   | $k/k_f$ | Be       |
|------|-----|--------|----------|
| 1    | .001| 5      | $10^8 \leq Be \leq 10^9$ |
| 2    | .001| 25     | $10^8 \leq Be \leq 10^9$ |
| 3    | .01  | 5      | $10^8 \leq Be \leq 10^9$ |
| 4    | .01  | 25     | $10^8 \leq Be \leq 10^9$ |

Figure 2. Optimization of Case 4, showing how D (and thus H) is varied for fixed values of Φ, $k/k_f$, and a range of Be to find the optimum configuration.

3.1. Calibration of sizing correlations

As established in Section 2.1, a design case consists of a unique combination of Φ, $k/k_f$, and a range of Be. Example cases are shown in Table 1: for the purposes of demonstration in this paper, we will focus on Case 4 (representative of conductivities for hardwoods, $k \sim 0.625 \, W/mK$ and relatively low pressure drops $1 < \Delta P < 15$). Numerically calibrated fits for the exponents are found by conducting simulations with varying values of D, to identify the geometric configuration where the temperature difference is minimized (Figure 2). The optimal variations are then plotted against the analytical solutions (Eq.s 1 and 4), and a and b varied until the data resolves towards the analytical solution.

3.2. Heat-exchange efficiency

Steady-state heat exchange is measured for a series of panel designs in Case 4 as a function of the incoming air temperature as it enters the interior (Eq. 8). Geometries are optimized using the numerically calibrated exponents to achieve a range of values for target efficiency. This is achieved by varying L (and thus Be) along a range of standard thicknesses for CLT ($5 < L < 25cm$) while holding Φ and $k/k_f$ fixed. Panel geometries and predicted efficiencies are given in Table 2.

3.3. Transient Response

Transient heat exchange is modelled for Case 4. A series of panel geometries of different thicknesses ($5 < L < 25cm$) are optimized to achieve the same steady-state NTU using the numerically calibrated exponents. Simulations are then performed using the minimum and maximum values of thermal diffusivity for timber ($1 \times 10^{-7} < \alpha < 4.4 \times 10^{-7}$) to determine the specific heat capacity of the
simulated element. Time to steady state is determined when the ΔNTU/hr reaches less than 10% of the steady-state total heat exchange.

Figure 3. Optimization of sizing correlations for Case 4.

4. Results

4.1. Calibration of sizing correlations
The optimization of Case 4 is shown in Figure 2, plotting the simulated temperature difference \((T_s - T_e)\) against varying values for D (and thus H/L). Optimal cases are highlighted in yellow. Figure 3 then compares the optimal geometry found by the simulations with the sizing correlations determined by Kim et al [5]. Fig. 3(a) shows the results plotted with the original analytical solutions \(a = \frac{2}{3}\) and \(b = \frac{1}{3}\), while 3(b) uses the newly-fitted exponents \(a = 0.67, 0.88\) and \(b = 0.21, 0.33\) for equations 1 and 4 respectively. As anticipated, optimizing these coefficients for the individual case provides significantly more robust alignment with the analytical solution.

4.2. Heat-exchange efficiency
Figure 4(a) plots simulated values of heat exchange efficiency (Eq. 8) for five panel geometries against the predicted \(\varepsilon\)-NTU curve for a single-fluid heat exchanger (Eq. 7). Results are also given in Table 2, including the simulated values for total heat exchange. Both show a close agreement between simulations and the theory. This suggests that the sizing correlations may adequately predict the temperature uplift of the ventilation stream. Efficiency may be increased by varying \(L\) and/or \(\Delta P\) within a design case, or by selecting an alternate design case by varying \(\Phi\) and \(k/k_f\).

| L  | NTU | \(\varepsilon = 1 - e^{-NTU}\) | \(\varepsilon = (T_s - T_e)/(T_s - T_e)\) |
|----|-----|-------------------------------|---------------------------------|
| 5  | 1.23 | .71                           | .72                             |
| 10 | 1.43 | .76                           | .78                             |
| 15 | 1.60 | .79                           | .82                             |
| 20 | 1.77 | .82                           | .83                             |
| 25 | 1.92 | .85                           | .84                             |

4.3. Transient response
Figure 4(b) plots the evolution of total heat exchange over time for 10, 15, and 20cm panels from Case 4, with \(\Phi = 0.01\), \(k/k_f = 25\) and \(Be = 6.82 \times 10^9\) (determined by panel thickness and a 5Pa pressure drop). The variations simulated with lower diffusivity take longer to reach steady state as anticipated,
as do the panels of increasing thickness. However, four of the simulations reach the threshold of $\Delta NTU/hr < 10\%$ of steady state in less than three hours - i.e., within a reasonable time frame considering diurnal temperature swings or similar building-scale temperature variations. And all simulations reach steady state in less than six hours.

![Figure 4](image)

**Figure 4.** Simulated results for (a) steady-state efficiency and (b) transient response time.

5. Conclusion

This study used numerical simulations to investigate the optimal sizing, heat exchange efficiency, and transient response time of one case of timber heat exchanging panels. Exponents for predicting channel sizing, channel spacing, and total heat exchange were numerically calibrated by varying panel geometries until a configuration produced the minimum temperature difference across the system. Then these exponents were used to design a range of panels for different steady-state heat exchange efficiencies. Simulated efficiencies were calculated as a function of interior air temperature lift and compared to predictions for a single-fluid heat exchanger. Results showed close agreement with predictions, suggesting the sizing correlations may adequately predict – and be used to design panels for – the interior temperature lift. Finally, a range of panels was designed with different thicknesses and thermal diffusivities using the same exponents. Transient simulations of total heat exchange showed that several designs ($L= 10, 15, 20cm$ with $\alpha < 4.4 \times 10^{-7}$) reached steady state in less than 3 hours, while all designs reached steady state in less than 6. This suggests that timber species with higher thermal diffusivities (given by a higher density and/or conductivity) may be preferable. It also suggests that timber panels may easily be optimized for steady heat-exchange within a range of structural thicknesses.

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