Synchronized coupling of thermal mass and buoyancy ventilation: wood versus concrete

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Abstract. This study describes an experiment that validates scaling rules for the design of thermal mass, coupled with buoyancy ventilation, suggesting that wood can perform as well as concrete if these rules are respected. The scaling rules potentially offer a shortcut for early design, showing how to tune the interior temperature and rate of buoyancy ventilation by adjusting the thickness and surface area of an internal thermal mass. A pair of test chambers (H~1m), comparing wood and concrete internal thermal masses, were located in Alabama, USA and Montreal, Canada, and left outside in sun- and wind-sheltered environments for consecutive months. The thermal mass thicknesses were optimized so the chambers would maintain similar interior temperatures and airflow rates. The scaling rules predicted the behavior of the chambers with reasonable accuracy and both the concrete and wood thermal masses performed equivalently. For instance, the test chambers in Alabama were both designed to damp the maximum exterior temperature by a factor $1-1/A_i \approx 0.7$ and produce a maximum ventilation flow rate of $Q \approx 0.37$ l/s. The measured damping was $1-1/A_i = 0.81\pm0.1$ and $1-1/A_i = 0.81\pm0.13$ for the concrete and wood chambers, respectively, while the maximum flow rates were $0.374\pm0.03$ and $0.36\pm0.04$ l/s, respectively.

1. Introduction

A study was published in 2007, defining mathematical ratios ($F$, $\lambda$, and $\Omega$) which characterize the coupling between buoyancy ventilation and an internal thermal mass [1]. The ratios assume the thermal mass and buoyancy ventilation are coupled in a natural feedback cycle: updraft at night, downdraft during the day. As such, the building harnesses diurnal temperature oscillations for ventilation. Similar thermal behavior has been observed in termite mounds [2].

More recently, a study was published that defines the optimal balance of $F/\lambda$ and $\Omega$ for a targeted interior temperature damping, $1-1/A_i$ [3,4]. Henceforth, the optimal relationship between these ratios ($F/\lambda$, $\Omega$, and $1-1/A_i$) is referred to as “scaling rules.” They are illustrated in Figure 1 and described in §2. These scaling rules offer a theoretical shortcut in early design. They show how to optimally tune the interior temperature and rate of buoyancy ventilation by adjusting the thickness and surface area of an internal thermal mass, no matter the material.
This study reports the results of an experiment designed to test if the scaling rules work. Small test chambers were fabricated to compare wood and concrete thermal masses and the experiment was replicated in two climates. This experiment is the first stage in a larger research project exploring the potential of these scaling rules. How reliable are they as a shortcut for designing climate-resilient buildings? Can they provide a fair functional unit for comparative life-cycle analysis in low-carbon building design?

2. Theory

The scaling rules for optimizing an internal thermal mass coupled with buoyancy ventilation are briefly outlined in this section. For a more in-depth review, see [3].

![Figure 1. Scaling rules for synchronizing an internal thermal mass with buoyancy ventilation. (a) Contour map. (b) Temperature definitions. [3]](image_url)

Figure 1a shows a contour map defining the optimal balance of ratios $F/\lambda$, $\Omega$, and $1-1/Ai$. The parameter $1-1/Ai$ is the damping coefficient chosen by the designer. As shown in Figure 1b, this factor limits the interior temperature, $\theta_i$, relative to the exterior temperature, $\theta_e$ (and therefore the thermal mass surface temperature, $\theta_s$, and mean thermal mass temperature, $\theta_m$, too). Bigger values of $1-1/Ai$ mean more interior temperature damping. Knowing the likely exterior diurnal swing, $\Delta T_e = (T_{e_{max}} - T_{e_{min}})/2$, a designer may choose a damping coefficient that corresponds with an acceptable range of interior temperature conditions.

For every increment of $1-1/Ai$ there is an optimal pairing of $F/\lambda$ and $\Omega$. The parameter $F/\lambda$ determines the rate of heat exchange between the thermal mass and the interior air. $F$ is a ratio that compares the rate of heat loss or gain from ventilation exchanges to the rate of heat loss or gain by surface convection:

$$F = \frac{Q \rho C_i}{S h}$$

(1)

Where $Q$ is ventilation rate, $S$ is thermal mass surface area, $\rho C_i$ is volumetric heat capacity of air, and $h$ is the surface heat transfer coefficient (the rate of convection between the thermal mass and the interior air). The factor $\lambda$ is between 0 and 1; it measures the divergence between the thermal mass surface temperature and the average thermal mass temperature:

$$\lambda = \frac{1}{1 + \frac{\eta (\sinh(2\pi) - \sin(2\pi))}{(cosh(2\pi) - \cos(2\pi))}}$$

(2)

Where $\omega$ is angular frequency, $\omega = 2\pi / 86400$, $\rho C$ is the volumetric heat capacity of the thermal mass material, $l$ is the thickness of the mass, $\eta = l \sqrt{\omega / 2\alpha}$, and $\alpha$ is the thermal diffusivity of the material. The ideal thermal mass has $\lambda = 1$ (the surface temperature and the average temperature are the same).
Hence, $\lambda$ accounts for the fact that some materials have better thermal mass properties. However, equation (1) and the contour map in Figure 1a show it is possible to compensate for inferior thermal properties by increasing $S$ to maintain an optimal value of $F/\lambda$. The parameter $\Omega$ is a tuning ratio that compares the rate of heat storage to the rate of surface heat transfer:

$$\Omega = \frac{\omega \rho_c l_p}{h \lambda}$$

(3)

Where $l_p$ is the effective thickness of the thermal mass and a function of $\eta$ (see equation 2.23 in [3]). Contour lines of $1-1/A_i$ as a function of $F/\lambda$ and $\Omega$ are shown in Figure 1a according to the following approximation [3]:

$$1 - \frac{1}{A_i} = 1 - \frac{\sqrt{1+\Omega^2}}{\sqrt{1+(\eta+1.07)(\frac{\lambda^2}{F\Omega})^{1/3}}}$$

(4)

Where the blue curve in Figure 1a shows the ideal tuning, for which $F/\lambda$ is maximized:

$$\left(\frac{F}{\lambda}\right)_{\text{max}} = \tan \left(0.535 \Omega^{3/4}\right) - 1$$

(5)

How can these thermal mass scaling rules be used in early design? By way of example, to achieve $1-1/A_i = 0.5$, Figure 1a and equations (4) and (5) suggest that one should optimize the thermal mass such that $\Omega = 1.62$; this will maximize the ventilation parameter such that $F/\lambda = 0.61$. Knowing the design value for $\Omega$, one can use equation (3) to find the optimal thickness $l$ of any thermal mass material, while using equation (2) to find the corresponding value for $\lambda$. Equation (1) can then be rearranged to find the required surface area of the thermal mass, after choosing a target ventilation rate, $Q$. Recall that the ventilation is powered by buoyancy. A reference rate for buoyancy ventilation, $Q_{\text{ref}}$, can be defined as:

$$Q_{\text{ref}} = A^* \sqrt{\beta g H \Delta T_e}$$

(6)

Where $A^*$ is the effective area of ventilation openings [5], $\beta$ is the thermal expansion coefficient of air, $g$ is gravitational acceleration, $H$ is the stack height, and $\Delta T_e = (T_{e\text{ max}} - T_{e\text{ min}})/2$, as shown in Figure 1b. An app which solves equations (1-6) is free to download online [4]. The following sections describe an experiment to test these scaling rules and to see if wood can perform as well as concrete.

3. Methodology

Four thermal mass chambers were constructed to test the scaling rules while comparing the performance of wood and concrete. The test chambers’ design is shown in Figure 2. One pair of chambers was installed in Hale County, Alabama (AL), US. [ASHRAE Climate Zone 3A], The other pair was installed in Montreal, Quebec, Canada (MTL), CA. [ASHRAE Climate Zone 7]. All test chambers in Alabama were designed to produce damping coefficients of $1-1/A_i \approx 0.7$. This target corresponds with optimal design values of $F/\lambda \approx 0.18$ and $\Omega \approx 1.5$. The target design values in Montreal were similar.

To define optimal thickness of the mass, the thermal properties – that is, thermal conductivity ($k$), the volumetric heat capacity ($\rho c$), and the thermal diffusivity ($\alpha$) – must be estimated using typical values from literature or measured specifically. Thermal property testing was done using a C-Therm TCi thermal conductivity analyser, model 1-A. 10 samples each of Quikrete Profinish 5000, Southern Yellow Pine harvested in Alabama, and Eastern White Pine were measured, with thermal paste applied before measurement, to ensure full sensor contact. The results are shown in Table 1. Note that Eastern White Pine is typically half as dense as Southern Yellow Pine, which explains the difference in volumetric heat capacity. The thickness for the panels is optimized for a chosen damping coefficient ($1-1/A_i$), using the scaling rules illustrated in Figure 1 and described in §2, assuming a surface heat transfer coefficient of $h \approx 2$ W/m$^2$/K (i.e. natural convection only; note that the optimal thickness increases almost linearly with $h$).
Table 1. Measured properties of thermal mass panels

| Thermal Mass Panels                  | $k$ (W/m/K) | $\rho c$ (J/m$^3$/K * 10$^6$) | $\alpha$ (m$^2$/s* 10$^{-7}$) | $l$ (cm) |
|-------------------------------------|-------------|-------------------------------|-------------------------------|----------|
| AL Concrete (Quikrete Profinish)    | 1.36 ± 0.02 | 1.89 ± 0.03                   | 7.15 ± 0.16                   | 2        |
| AL Pine (Southern Yellow Pine)      | 0.208 ± 0.003 | 1.213 ± 0.019                | 1.71 ± 0.04                   | 2.85     |
| MTL Concrete (Quikrete Profinish)   | 1.36 ± 0.02 | 1.89 ± 0.03                   | 7.15 ± 0.16                   | 2.1      |
| MTL Pine (Eastern White Pine)       | 0.197 ± 0.003 | 0.535 ± 0.08                 | 3.67 ± 0.08                   | 5        |

The typical design of the test chambers is shown in Figure 2a. The chambers in Alabama were located outdoors under full shade. The chambers in Montreal were located in an open garage. The Alabama test chambers interior dimensions were 30.5 cm x 30.5 cm x 91.5 cm with thermal mass surface area ($S$) of 1.2 m$^2$. The Montreal chambers were 30 cm x 30 cm x 90 cm, with an $S$ of 1.08m$^2$. The size of the ventilation openings was determined by the effective opening area, $A^*$, necessary to maintain a ventilation rate close to the design value for $F$ and $Q_{ref}$. Discharge coefficients were estimated with [6]:

$$c_d = \frac{1}{1 + 0.707 \sqrt{\frac{F_0}{F_1}} + \left(\frac{5}{\sqrt{Re}}\right)}$$  \hspace{1cm} (7)

Where $F_0$ is area of the orifice, $F_1$ is area of chamber, and Reynolds number ($Re = (\rho Hu)/v$). The values for $A$, $c_a$, and $A^*$ were: AL concrete and pine = (0.00176 m$^2$, 0.63, 0.00111 m$^2$), MTL concrete and pine = (0.00382 m$^2$, 0.63, 0.00224 m$^2$) and (0.00305 m$^2$, 0.64, 0.00194 m$^2$)

Typical sensor locations are represented in Figure 2a. Differential pressure for updraft ($\Delta P_u$) and downdraft ($\Delta P_d$) were recorded by Sensirion SDP810 sensors, fitted with pitot tubes. Exterior air temperature ($T_e$) and chamber interior air temperature ($T_i$) were recorded via type K thermocouple or GreenTeg ambient air temperature sensors. Thermal mass front surface temperature ($T_{si}$), back surface temperature ($T_{se}$) and heat flux ($HF$) were measured via FluxTeq Ultra-09 or GreenTeg surface and heat flux sensors. The reported error incorporates the reading error and standard deviation using built-in capabilities of Wolfram Mathematica.

Figure 2. Concrete test chamber (a): chamber section (b): example chamber data
4. Results and Discussion

Data were collected for all chambers from July to September 2020 resulting in a sample of 27 days for Alabama, and 32 days for Montreal. The typical evolution of temperature and ventilation over one day is shown in Figure 2b. There is an updraft during the night, when the mass is warmer than the exterior, and downdraft during the day, when the mass is cooler than the exterior. All data were taken at 10 minute intervals and smoothed with a rolling average of 10 measurements to reduce noise due to local air turbulence. The heat transfer coefficient \( h \) was recorded by using the measured Heat Flux \( HF \) and the temperature difference between the interior air and mass surface: \( HF / (T_i - T_s) \). Measurements were discarded when \( T_i = T_s \) and \( HF > 0 \). The parameter \( F \) was calculated using equation (1) and as a function of \( Q_{\text{mean max}} \), which most closely reflects the predicted \( Q_{\text{ref}} \).

**Table 2.** Measured heat exchange parameters

| Panel Name     | \( F \)     | \( \lambda \) | \( \Omega \) | \( h \) (W/m\(^2\)/K) |
|----------------|-------------|---------------|--------------|------------------------|
| AL Concrete    | 0.19 ± 0.05 | 0.989 ± 0.003 | 1.3 ± 0.4    | 2.3 ± 0.6              |
| AL Pine        | 0.27 ± 0.35 | 0.91 ± 0.03   | 1.9 ± 2.5    | 2.2 ± 0.7              |
| MTL Concrete   | 0.31 ± 0.11 | 0.986 ± 0.004 | 1.12 ± 0.35  | 2.8 ± 0.8              |
| MTL Pine       | 0.18 ± 0.10 | 0.76 ± 0.06   | 0.8 ± 0.05   | 3.7 ± 1.1              |

The measurements for interior air temperature damping \((1-1/A_i)\) and ventilation flow rate \((Q)\) are reported in table 3, in comparison to predicted values. The ventilation results are reported in terms of the mean and mean maximum ventilation rate during updraft flow, i.e. during nighttime periods. Temperature damping measurements were analyzed during daytime periods. Damping coefficient measurements were excluded if \(0 < 1-1/A_i < 1\). Values outside this range corresponded with significant deviations from quasi-periodic daily temperature swings (for instance, due to a storm or sudden shift to a hotter or cooler week).

**Table 3.** Temperature damping \((1-1/A_i)\) and ventilation flow rate \((Q)\)

| Panel Name     | \( Q_{\text{ref}} \) (l/s) | \( Q_{\text{mean max}} \) (l/s) | \( Q_{\text{mean}} \) (l/s) | \( I-1/A_i \) (predicted) | \( I-1/A_i \) (measured) | \( \Delta T_e \) (measured) |
|----------------|-----------------------------|----------------------------------|-----------------------------|----------------------------|----------------------------|-----------------------------|
| AL Concrete    | 0.377 ± 0.03                | 0.374 ± 0.03                     | 0.303 ± 0.028              | 0.68 ± 0.04                | 0.81 ± 0.1                 | 3.9 ± 0.7                   |
| AL Pine        | 0.374 ± 0.03                | 0.36 ± 0.4                       | 0.28 ± 0.05                | 0.65 ± 0.12                | 0.81 ± 0.13                | 3.8 ± 0.7                   |
| MTL Concrete   | 0.91 ± 0.12                 | 0.71 ± 0.11                      | 0.54 ± 0.09                | 0.56 ± 0.03                | 0.58 ± 0.23                | 4.9 ± 1.3                   |
| MTL Pine       | 0.73 ± 0.1                  | 0.52 ± 0.11                      | 0.37 ± 0.09                | 0.55 ± 0.05                | 0.61 ± 0.21                | 4.8 ± 1.3                   |

How well did the measurements match with predictions? Figure 3a shows the thermal masses behaved predictably, despite the ambient temperature ‘noise’ shown in Figure 3b. In Alabama, the concrete and wood thermal masses performed equivalently: the air temperature damping and ventilation rate were practically identical each day. The scaling rules did slightly underpredict the temperature damping in Alabama. However, in Montreal, the scaling rules predicted the temperature damping accurately in both chambers. The ventilation was slightly lower than expected, though still in range.
5. Conclusion

A simple experiment was conducted to test the accuracy of a set of approximate equations, which suggest how to optimize the coupling of an internal thermal mass with buoyancy ventilation, regardless of the material or the scale of the building. Small test chambers were installed outside in Alabama and Montreal to compare wood and concrete thermal masses. The chambers were 1m tall, insulated on the exterior, and had openings top and bottom to exchange buoyancy ventilation with the exterior environment in a daily cycle. As predicted, the wood and concrete thermal masses performed equivalently (or as designed). The results suggest the thermal mass ‘scaling rules’ may be valid in the early stages of the design process, and useful in two ways. First, to understand the optimal quantities of thermal mass material required for a chosen temperature damping and ventilation production rate. And second, to compare the life-cycle environmental impacts of different thermal mass materials with a fair functional unit. The results also suggest that biomaterials can perform as well as conventional thermal mass materials. In theory, one can compensate for inferior thermal properties in optimal thermal mass design by adjusting the thickness and the surface area. For example, in a true-scale building, the surface area of a wood thermal mass would have to be larger than a concrete thermal mass (as well as thicker) if it was to perform the same thermodynamic work. A larger comparative experiment is under construction in Alabama, to investigate the reliability of the scaling rules at full scale and to characterize the interior climate dynamics with and without internal loads.

Acknowledgements

This research was supported by the McGill Sustainability Systems Initiative (MSSI)

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