Evaluation of the intrinsic thermal performance of an envelope in the summer period

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Abstract. Ensuring the proper thermal performance of a building’s envelope upon reception is an important stage in the life cycle of the building. Several methods already exist for this purpose, and continue to be improved, such as co-heating, ISABELE, EPILOG, QUB and SEREINE. All these methods follow the common protocol consisting of heating the measured building. These measurement protocols quantify the dynamic evolution of indoor and outdoor temperatures, and the thermal power injected into the building and these data are used in calibration algorithms to determine, by an inverse method to deduce a heat loss value. These methods require a difference of a few degrees between the interior and the exterior which can cause in summer periods a risk of damaging the building, as the outside temperature may already be high. The objective of this work is to explore the possibility of determining the intrinsic thermal performance of a building’s envelope in the summer period using a cooling system. This work leans on an experiment of a square meter scale cell and explore the capacities and limitations of the method at this scale by varying several stress parameters of the enclosure. Results in cooling mode are also compared to heating mode.

1. Introduction

Ensuring the proper thermal performance of a building’s envelope, whether new or renovated, upon its reception is an important step in the life cycle of the building. It allows to ensure the good consistency between the design phase and the implementation during the construction work, in order to consider possible corrective actions to reduce its environmental impact. Several methods exist for this purpose and continue to be improved such as co-heating, ISABELE [1] [2] [3] [4], EPILOG, QUB [5] or SEREINE methods. The objective is to determine the thermal performance of an envelope which can be quantified by the Heat Loss Coefficient (HLC in [W/K]), which is the sum of the heat loss due to infiltration (Hinf) and the heat loss due to conduction through the walls (Htr), (towards the exterior and the ground). These values will partly depend on the test conditions including the indoor and outdoor pressure difference and the operative temperature inside, the outside air temperature for Hinf and the convective heat transfer coefficient under real conditions for Htr. All these methods have the common point to stress the building over a certain period of time using a heating system. These protocols measure the dynamic evolution of the indoor temperatures, the thermal power injected into the building and the outdoor conditions. For the majority of these methods, these data are then used in a calibration algorithm using a RC model to determine, by an inverse method, the model parameters and thus deduce HLC.
(Heat Loss Coefficient). These methods require an indoor temperature a few degrees higher than the outside temperature, a gap that is potentially no longer acceptable in summer periods without the risk of damaging the measured building. The aim of the work is consequently to develop a test methodology with a cooling system in order to be able to measure a HLC coefficient even in summer periods. A first step was to design and implement an experiment on the scale of a square meter cell using a heat exchanger. The test consists of cooling the air inside the cell and measuring the dynamic evolution of the injected cooling power and the temperature inside and outside the cell. The power required to compensate the heat losses with the exterior is then estimated leading to HLC coefficient of the cell using the SEREINE calibration method. Heating experiments using electrical resistances were also conducted in order to be compared with the cooling experiments.

2. Experimental setup

2.1. Experimental cell description
In order to study the feasibility of a method for assessing the intrinsic performance by cooling, a first experiment with a cell of approximately one square meter was conducted. The experimental chamber Figure 1 has a rectangular base with sides \( L = 1.2 \, \text{m} \) and \( l = 0.8 \, \text{m} \) and a height \( h = 1.2 \, \text{m} \). Its walls are made from film-coated plywood of 1.5 cm thickness. The lower wall is constituted of an additional outer layer of polystyrene of 4 cm thickness. This choice of a non-insulated envelope on 5 sides aims to have a significant order of magnitude in the injected power compared to measurement uncertainties. The front wall is removable to allow access to the interior at all times. The cell lies on a metal structure, and is raised from the ground by 50 cm, to help convective air / wall exchanges. Airtightness is provided by silicone seals at the intersection of 5 sides, and by adhesive tape on the removable side. Some gaz tracer measurements were carried on to measure the infiltration impact. The tests revealed an infiltration loss (\( H_{\text{inf}} \)) below 0.1W, which will be neglected for the different scenario runs. The theoretical coefficient \( H_{\text{tr}} \) of the envelope is estimated at 23.2 W/K thanks to thermal conductivities and thicknesses of film-coated plywood and polystyrene (\( \lambda_{\text{cp}} = 0.13 \, \text{W.m}^{-1}.\text{K}^{-1} \) and \( \lambda_{\text{pol}} = 0.035 \, \text{W.m}^{-1}.\text{K}^{-1} \)) and surfaces of each face. The thermal conductivities of the envelope materials were also measured using a ct-meter and gives a large uncertainty with values varying from 0.08 and 0.12 W.m\(^{-1}.\text{K}^{-1} \). With these values of thermal conductivity, the range of estimated \( H_{\text{tr}} \) is between 18 and 21 W/K.
The cooling/heating transmitter is an air/water exchanger placed inside the cell Figure 2. The water circulating in this exchanger is cooled or heated by a thermal controlled bath Figure 3 with a variable flow pump to set the water flow running through the hydraulic circuit. The heat exchanger is paired to an electric fan, its purpose is to stir the air that runs through the exchanger and to homogenize the interior atmosphere of the cell Figure 2. The power consumption of the fan is calculated upstream, and is taken into account in the overall thermal power that is injected into the experimental cell, considering that the electrical power consumed by the fan is dissipated into heating power.

Figure 3. Thermal and hydraulic equipment outside the cell.

2.2. Sensors
The physical quantities measured are air, surfaces and water temperatures, the heat fluxes through certain walls, the relative humidity of the air, as well as the water flow rate. The setup and position of the sensors is summarized in Table 1. The temperature probes were calibrated upstream of the tests with their complete acquisition chain. The data from the sensors are pre-processed to feed the algorithm presented in the following paragraph. An average indoor air temperature is calculated from the 4 sensors installed. The outside temperature is given directly by the air temperature sensor.

| Type of sensors | Unit   | Position       | Setup                | Number |
|-----------------|--------|----------------|----------------------|--------|
| Temperature     | °C     | Inside Cell    | At different points  | 4      |
|                 |        | Outside Cell   | Adjacent Room        | 1      |
|                 |        | Hydraulic loop | Cell Inlet           | 1      |
|                 |        |                | Cell Outlet          | 1      |
| Flowmeter       | kg/h   | Hydraulic loop | Cell Inlet           | 1      |
| Hygrometer      | %      | Air            | Outlet of heat exchanger | 1 |
|                 |        |                | Adjacent Room        | 1      |
| Heat-flux       | W/m²   | Indoor walls   | Top                  | 1      |
|                 |        |                | Bottom               | 1      |
|                 |        |                | Right                | 1      |

In the usual methods, electrical resistors are used as heat sources. The dissipated power is then measured directly by monitoring the electrical consumption. One of the challenges in cooling mode is to be able to measure with a sufficient accuracy the thermal power injected into the enclosure. In our case, it is calculated by adding the calorific power supplied by the hydraulic system to the electrical power supplied by the fan. The calorific power is calculated using the water flow rate and the water
temperature measurements of the hydraulic circuit taken at the inlet and outlet of the cell [eq (1)]. The formula is therefore:

\[ P_{\text{in}} = P_{\text{Heat/Cool}} + P_{\text{fan}}; \text{ with } P_{\text{Heat/Cool}} = \dot{m}_{\text{water}} \cdot C_p \cdot (T_{\text{water-out}} - T_{\text{water-in}}) \] (1)

The power of the fan was calculated before the presented tests by measuring the intensity of the current running through the fan using a multimeter for various voltage values.

2.3. Scenarios setup

Tests were conducted between the end of August and mid November 2020. The cell is placed in a room of a non-air-conditioned office building. The temperature of the room which serves as the outer boundary condition of the cell is mainly constant during a same test phase.

Firstly, an alternation of cooling tests during the day and heating ones at night were run with the hydraulic system, in order to be able to compare the evaluation of the heat loss coefficient for both configurations. The parameters set for these scenarios are the water temperature of the thermostatic bath, the water flow rate of the hydraulic circuit pump, and the air mixing rate of the fan inside the cell.

Secondly, some tests with an electrical heater were run in order to compare the impact of the thermal system on the result. For these tests, short tests (~12h each) were first run to act as closely as possible as the hydraulic tests. Then a one-week test with an imposed constant temperature inside the cell was run to act as a co-heating test in order to get a reference value of HLC. As illustrated in Table 2, the same scenario was implemented several times in order to verify the repeatability of the experiment.

| Table 2. The run scenarios. |
|----------------------------|
|                            |
| Scenarios                  | Duration per test | Nb of tests per scenario | Thermal Bath Set Up | Fan |
|                           |                  |                        | T\text{in} water °C | Water Flow Rate kg/h | Air Mixing Rate m3/h |
| Hydraulic Cooling          | 1C               | ~8h                     | 9                   | 13 | 70 | 370 |
|                           | 2C               | ~8h                     | 5                   | 13 | 70 | 210 |
|                           | 3C               | ~8h                     | 5                   | 15 | 70 | 210 |
|                           | 4C               | ~8h                     | 7                   | 17 | 70 | 370 |
| Hydraulic Heating          | 1H               | ~15h                    | 9                   | 42 | 70 | 370 |
|                           | 2H               | ~15h                    | 6                   | 42 | 70 | 210 |
|                           | 3H               | ~15h                    | 1                   | 38 | 70 | 210 |
|                           | 4H               | ~15h                    | 5                   | 40 | 70 | 210 |
|                           | 5H               | ~15h                    | 5                   | 38 | 70 | 370 |
| Electrical Heating         | 5E               | ~6h                     | 3                   | 100|    | 210 |
|                           | 6E               | ~8h                     | 3                   | 100|    | 370 |
|                           | 7E               | ~1 week                 | 1                   | T cell setup at 27°C | 370 |

2.4. Presentation of the calibration method

The purpose of the optimization algorithm for the calibration is to find the set of parameters (of the RC model retained) which minimizes the difference between measured interior temperature and the simulated one based on the experimental data (injected power inside and the outside temperature). The algorithm used is taken from the numerical code used in SEREINE method, currently under development. This algorithm is an evolution of the versions used in previous methods (ISABELE [1] [2] [3] [4] and EPILOG), and is partly based on the pySIP uncertainty propagation algorithm [7].
The present analysis consists in calculating by inverse method the HLC of an envelope, from a RC model [6] [8] from the SERENE tool named TWTM (or M2_TmTmiTi model in [1] and [6] without taking into account the indoor superficial resistance due to convective heat transfer coefficient, equivalent to having \( R_{si} = 0 \)). Figure 4, where TW indicates the presence of a capacitance associated with two transmittances modelling a wall and TM indicates the presence of a capacitance modelling the internal mass. In our case, this internal mass is represented by the heat exchanger. It requires as input parameters only the power supplied and the equivalent interior and exterior temperatures of the enclosure.

![Figure 4. RC model used for the calibration.](image)

The uncertainty of the estimated coefficient is only based on the propagation of the random type errors via the pySIP algorithm and does not currently consider the systematic uncertainties of experimental origin (for example, the bias on the estimation of the cooling power, indoor temperatures, etc.), even though we might consider that the uncertainties related to the temperatures measurements are minimized with the sensor’s calibration.

3. Results

3.1. Results with a hydraulic cooling scenario

![Figure 5. Temperatures for a cooling scenario, dt=5min.](image)

![Figure 6. Heat flux for a cooling scenario, dt=5min.](image)
Figure 7. Comparison between the predicted and measured temperature.

Figure 5 and Figure 6 illustrate the measurements of test 1 and represents, on the top graph, the evolution of the interior air temperatures of the four interior sensors (Ta$_{int 1}$ to Ta$_{int 4}$), the temperatures of the water entering (Twater$_{in}$) and leaving (Twater$_{out}$) the cell and the exterior temperature (Ta$_{ext}$). At the bottom is plotted the evolution of the heat fluxes on the upper, lower and right walls of the cell. For indoor air temperatures, we note an exponential decrease from 26°C to a stabilization around 19°C. The transient regime is about an hour. Ta$_{int 1}$, 3 and 4 temperatures are homogeneous around 18.5°C, as for Taint2, it is slightly below 17.8°C since the sensor is located immediately at the outlet of the exchanger where the air blows. The outside temperature varies between 21.5°C and 25.2°C. Under steady state conditions, the air temperature difference between the inside and the outside of the cell is on an average of 6.4°C, and the water temperature difference between the inlet and the outlet of the cell is 2.2°C. Fluxes on the non-insulated faces (top and side faces) are steady-state around 20 W/m², while the flux on the insulated lower face is unsurprisingly lower around 5 W/m². Figure 7 compares the inside air temperature measured with the one simulated with the parameters retained after calibration of the TWTM model for test 2. The blue band corresponds to the uncertainty propagated on this output. The agreement between the measured and simulated values is one of the criteria to ensure the correct convergence of the method. Another indicator used in the optimization process is the likelihood function. At the end of the analysis, we obtain a value of the coefficient HLC as well as the associated random uncertainty. It should be noted that no adaptation of the calibration algorithm was necessary for the cooling calibration.

3.2. Results overview

Figure 8 represents the estimation of HLC coefficient for the different tests described in Table 2. Air mixing rate of the fan located inside the cell influenced the HLC estimation with an average difference of 3W/K. Indeed, most of the tests run with a mixing rate of 210m³/h give a HLC between 16 and 18 W/K, while the tests with a mixing rate of 370 m³/h range from 18 to 20 W/K. This difference probably comes from the inside convective heat transfer coefficient in the cell, the infiltration tests demonstrate that this phenomenon is negligible in this experiment. For a fixed mixing rate, majority of the tests results is in a range of +/- 1 W/K which represent +/-6% and +/-5% for a mixing rate of respectively 210 m³/h and 370m³/h. At this stage of the study, no clear difference appears between the hydraulic tests either in cooling or heating mode and the electric heating ones. Retaining the larger mixing rate (370m³/h), a longer test of one week behaving as a co-heating test was run and gives HLC value of 19.3 W/K which will be considered as a reference value for this set of scenarios. In parallel, the 2 tests with an electric resistance gave a value of 18.2 and 18.4 W/K on the lower bound of the values.
Different inlet water temperatures were tested for both cooling and heating hydraulic tests as described in the different scenario presented in Table 2, but no clear impact of this parameter appears as shown in Figure 8. The air temperature difference from inside / outside the cell which varies from -6°C to 16°C in the different tests run seems not to impact the HLC evaluation as shown in Figure 9. Figure 10 shows the HLC as a function of the absolute value of the difference between the inlet water temperature and the average air temperature in the cell for a test, and no direct relation was found between these values either.

4. Conclusion and discussion
This article aims to develop a methodology to assess the overall heat loss coefficient of the building envelope in the summer, by cooling it. For this, a small-scale experimentation was carried out and made the methodology feasible from the experimentation up to the calibration process of the RC model to obtain an overall heat loss coefficient. Almost 60 tests were run varying different parameters, among them hydraulic cooling/heating system or electric thermal resistance, the air mixing rate inside the measured cell, and even the water temperatures for the hydraulic system.
The first tests show that the configuration adopted (dimensions of the cell and the system sizing of generating and emitting cold / heat coupled to a fan) allows to create a homogeneous and stable interior atmosphere, with a sufficient difference between the temperatures of outside and inside air.

Regarding the calibration process, no adaptation of the calibration algorithm developed for heating methods was necessary in this first phase to determine the HLC under cooling demand.

The results obtained in this first stage are promising. The air mixing rate inside the cells impacts clearly the HLC value estimation. Increasing it from 210 to 370 m$^3$/h changes the averaged HLC from 17.1 to 19.3 W/K (~10%). At a fixed air mixing rate, the calculated values of the HLC coefficient for both heating and cooling loads and for both hydraulic and electric heating are mostly in a range of +/- 5%.

To extend this study, some uncertainty analyses needs to be investigated, before moving to a larger scale. Indeed, the hydraulic thermal system produced results similar to the electric heating one but it has a larger uncertainty which still needs to be quantified. In parallel the test cell scale of 1m$^3$ is quite small and some tests at a larger scale need to be performed to confirm these first results. In addition, the issues of potential condensation should be studied. Therefore, lower water temperatures instructions are ought to be tested to allow condensation at the heat exchanger’s level.

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