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# The Dual Air Vented Thermal Box: a laboratory apparatus to test air permeable building envelope technologies

Andrea Alongi[a,*], Livio Mazzarella[a]

[a]*Politecnico di Milano, Dipartimento di Energia, via Lambruschini 4, Milano 20156, Italy*

**Abstract**

Dynamic insulation technology has been investigated in detail in the context of a research program conducted by the Building Physic group of the Department of Energy in the Politecnico di Milano. In order to assess the energy performance of this kind of building envelope, either in steady-state or in dynamic conditions, a specific test apparatus has been designed and built.

It consists of two thermally controlled chambers, separated by a metal frame hosting the air-permeable wall sample, one fed by a temperature controlled air flow, and the other discharged of the air flowing through the sample. The air flow is realized through a close air loop between the two chambers. The temperature control is achieved with three parallel water circuits (two for the boxes, one for the air loop) mixing cold (~11°C) and hot (~60°C) water. This work deals with the description of the whole apparatus and its measurement and control system, along with the procedure of characterization and calibration of the temperature control system in steady-state conditions.


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*Keywords:* laboratory apparatus; vented double chamber; dynamic insulation; porous material.

## 1. Introduction

Among the building envelope technologies, dynamic insulation seems to be a promising solution to reduce thermal energy needs. In order to perform experimental studies about this technology, a suitable apparatus has been developed. In the following sections, a brief overview about similar set-ups and technical standard is reported; then,

* Corresponding author. Tel.: +39 02 2399 3867; fax: +39 02 2399 3868.
*E-mail address:* andrea.alongi@polimi.it


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the apparatus itself is described. Finally, the control algorithm for steady state thermal boundary conditions simulation is presented, along with the investigation of its effectiveness in several scenarios. Strengths and weaknesses of the algorithm are discussed.

## 2. The research framework

Dynamic insulation is a building envelope technology which has been developed in northern Europe in the Nineties of the last century [1]. It is based on air permeable layers embedded in walls: ventilation air for the enclosed environment is forced to pass slowly through these porous layers, integrating them as heat recovery unit and filter into the ventilation plant. This technology has been investigated in detail within a research program [2] conducted by the Building Physic group of the Department of Energy in Politecnico di Milano.

As far as the experimental analysis is concerned, the Dual Air Vented Thermal Box (DAVTB) has been designed and built. Due to its peculiar features, no useful examples was found in literature during the design phase: the main technical standard available as a reference is the ISO 8990 [3], which deals only with guarded (GHB) and calibrated (CHB) hot-box, as a tool to investigate steady-state properties of building envelope components (some examples of both systems can be found in [4]-[6]). Even when dynamic insulation walls are investigated [7], only a simple set-up is used, being able to partially control thermal boundary conditions.

For all these reasons, the DAVTB system has been developed: it is able to control temperature conditions on both side of sample, apply an air flow through porous materials, and has been designed to simulate both steady-state and dynamic boundary conditions. Therefore, this apparatus is also suitable to test other wall technologies, such as cavity wall or PCM-integrated walls, and investigate their behaviour under unsteady thermal conditions. In the next sections, a description of the apparatus is presented, along with the control algorithm developed for steady-state thermal conditions, within the thesis work presented in [8].

## 3. The DAVTB apparatus

The experimental set-up under discussion (Fig. 1) is mainly composed by two insulated chambers divided by the sample and connected by the ventilation system. Temperature can be controlled in each chamber separately, through a heating and cooling plant, and the air flow can be controlled both in velocity and direction.

Fig. 1: picture of the DAVTB apparatus. Chambers, sample frame and part of the air plant are visible in the foreground. The external section of the secondary plant is in the background.

### *3.1. Chambers*

External dimension of both chambers are 1.5 m x 1.5 m x 1.29 m, and wall section is a sandwich made of internal and external laminated coating (4 mm each) with a polystyrene layer interposed, for a total thickness of 140 mm. Its purpose is to make the envelope as much adiabatic as possible. Each chamber has two openings: the first is a missing wall on the side with external dimension of 1.5 m x 1.5 m, the second is a hole on the opposite side, with a diameter of 20 cm, which is used to connect both chambers to the air system. The wall sample under testing is hosted by a dedicated metal frame, which is placed in between the chambers missing walls, which are then in thermal (convective and conductive) contact through the sample. Because of its transversal size, frontal area of samples can be around 1 m x 1 m, which allows testing single or multilayer walls within a thickness of 33 cm.

### *3.2. Heating and cooling plant*

As stated before, temperature control is achieved through a combined heating and cooling plant. Going more in detail, it is linked to the central supply of the Energy Department building (*primary plant*) through two water tanks, which have to provide hot and cold water to the apparatus through the *secondary plant*. Moreover, they allow decoupling primary and secondary plants for any maintenance operation. Through a preliminary test, loading time of both tanks has been assessed around 3 h, after which about 70°C and 11°C are reached in the hot and cold tank respectively. It is important to emphasize that temperature inside tanks should be fairly stable, in order to achieve effective thermal control inside chambers. During the mentioned test, a fluctuating behaviour of the primary plant has been identified; moreover, it has been discovered that it is disabled during night time regardless the agreed management of 24 h a day. This last factor may affect the chance to achieve, in the near future, a reliable simulation of realistic dynamic boundary conditions, which are usually 24 h time period based.

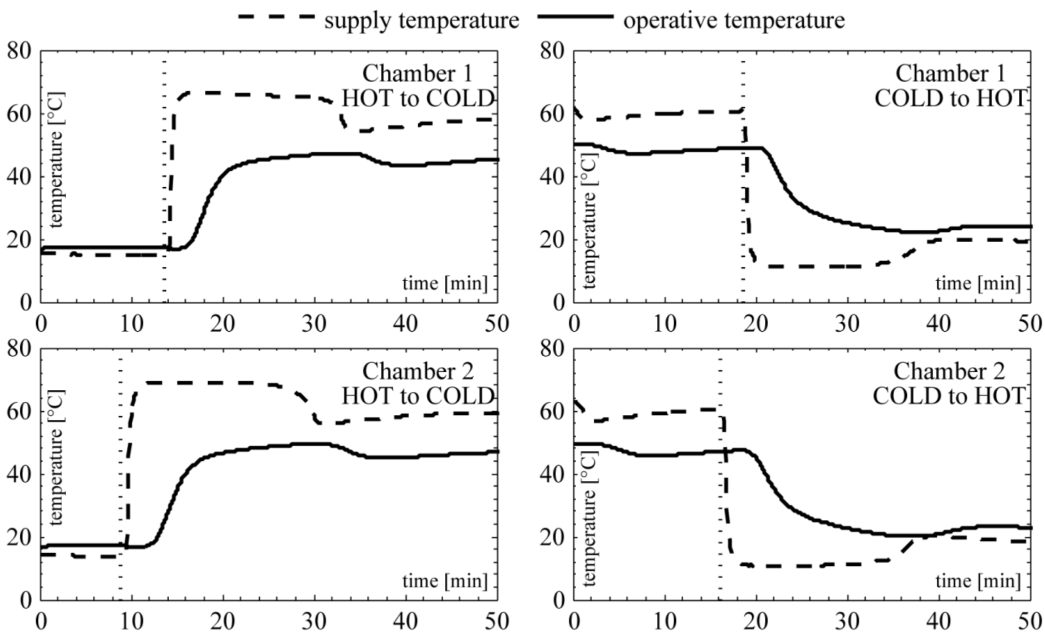


Fig. 2: characterization of the inertial behaviour of Chamber 1 (top) and 2 (bottom). For both boxes, the transitions from cold to hot (left) and from hot to cold (right) have been considered.

The secondary plant consists of three parallel circuits: two are used to provide thermally treated water to chambers; the third is connected to a heat exchanger in the air system. In each of them, supply temperature to terminals is controlled independently: hot and cold water coming from tanks are mixed to obtain a given supply temperature by acting on two servo valves for each circuit (a mixing valve and a diverter). The behaviour of these valves has been investigated, and a direct correlation between the control signal and the valve opening has been found. As far as delivery terminals are concerned, five parallel radiant panels are located directly into both chambers. Every panel has been made with copper strips for solar collectors. In order to promote the radiative heat exchange, every inward surface of each panel has been painted with a matte black varnish. The new thermal emissivity has been estimated through thermographic measurements, and is around 0.92÷0:94.

Furthermore, system inertial behaviour has been assessed (Fig. 2) measuring the time interval needed by both chambers to go from hot to cold state, and vice versa. This phenomenon is caused by the pipe length in the delivery system, by the thermal inertia of radiant panels and, finally, by the time needed by each valve to switch from a status to another (almost 42 seconds are needed to go from a fully cold to a fully hot water flow rate). A characteristic delay has been defined as the time required changing operative temperature by ±0.1°C. Considering Fig. 2, it is possible to claim that such delay is around 2 minutes: more precisely, it is 110 seconds for Chamber 1 and 140 seconds for Chamber 2, due to the additional pipe length needed being the most distant from the tanks (i.e. the hot/cold source), as can be seen from Fig. 1.

### *3.3. Air plant*

Due to the main function of the apparatus under discussion, an air system connecting the two chambers has been introduced. Going more in detail, it is divided into three main parts: first, the heat exchanger section; second, the fan section, which has been designed to allow the control of the air flow in terms of velocity and direction. It consists of two parallel fans, facing in opposite direction (centrifugal devices with a PWM velocity control, 48 Pa÷560 Pa at 30.6 $m^3/h$÷106.2 $m^3/h$), and two butterfly damper for each fan (to regulate air flow rate, or insulate one of the circuit branches). The third section of the air plant consists of a 2 m long tube (diameter 50 mm) and is used to measure air flow rate, thanks to a small fan anemometer located in the middle.

### *3.4. Probes and control devices*

The whole apparatus is automatically driven by a multifunctional switch unit, which is remotely controlled by a LabVIEW® algorithm. The multifunctional switch is an Agilent 34980A, equipped with modules with three main purposes: first, voltage and current measurement; secondly, proportional control and waveform generation; finally, switches for ON/OFF control. As far as the measurement of environmental parameters is concerned, following probes have been adopted: a differential micro-manometer, a bidirectional fan anemometer, a series of thermocouple for air, sample and water temperature measurement, and finally two globe thermometers. All temperature probes have been calibrated before installation.

## 4. Development of the control algorithm for steady-state conditions

At the very beginning, a plain PI controller was used, which had to adapt the control signal to valves according to the difference between the actual operative temperature inside chambers and their set-point values. This approach was ineffective: even after a careful analysis of PI gains, this algorithm was not able to modulate properly the control signal, leading to significant temperature errors (up to ±5°C from set point values) and fast discharge of both tanks, maybe as a result of the aforementioned inertial behavior. For these reasons, a mixed approach has been introduced. First of all, circulation pumps are controlled with an ON/OFF system: temperature variations over time are monitored and a prediction is made on the time needed to reach the set point value; if it is greater than the chamber delay (which has been discussed in the previous section) pumps are left active, otherwise they are disabled. Secondly, according to the desired operative temperature, supply water temperature is defined at the beginning of every experiment: it is calculated as the sum between the operative temperature and a positive or negative (for heating and cooling regime respectively) temperature difference ($\Delta T_{supp}$); then, a PI controller is used to achieve the desired supply temperature throughout the test. It is important to notice that the range for the control signal is limited: the cold water fraction needed is pre-calculated according to hot and cold water instantaneous temperatures; then the corresponding voltage signal is evaluated using the above mentioned correlation, and is used as the central value for a ±0.5V range. Once this mixed algorithm has been introduced, some preliminary tests have been performed at various combinations of set-point temperature in the two chambers, with the assumption that winter boundary conditions need bigger temperature differences between Chamber 1 and 2 (which can represent indoor and outdoor environments alternatively), while smaller difference can represent summer Mediterranean climate.

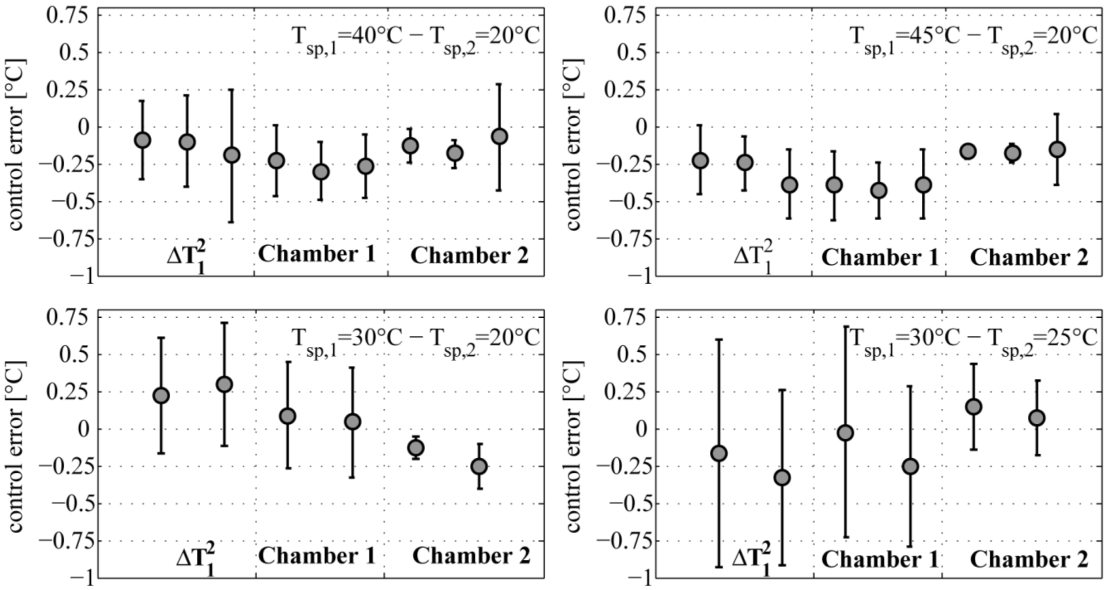


Fig. 3: control error (defined as the difference between actual and set-point values) is represented for each of the four test groups, both in terms of average (dots) and standard deviation (whiskers). For every set errors related to Chamber 1 and 2 are reported, along with the error of the temperature difference between chambers ( $\Delta T_1^2$ ) referred to the expected value.

This phase was aimed at the detection of possible issues, along with the identification of the most adequate values for $\Delta T_{supp}$ in each thermal condition. In the heated chamber temperatures from 30°C to 50°C (with 5°C increments) were investigated, with $\Delta T_{supp}$ =+1°C ÷5°C; at the same time, in the cooled chamber set point temperature was changed in a range from 17.5°C to 25°C (with 2.5°C increments), using $\Delta T_{supp}$ =-3°C ÷-1°C These tests have shown that set-point temperature lower than 20°C are unlikely obtained; moreover , it has been found that control is more difficult when difference between chamber operative temperature is smaller than 10°C. Finally, it has been shown that one of the most important parameters to achieve good temperature stability is $\Delta T_{supp}$. At this stage, no air flow through the sample has been considered.

After this first set of tests, four couples of representative temperature combinations have been chosen:40°C-20°C and 45°C-20°C for winter, 30°C-20°C and 30°C-25°C for summer. A second series of tests has been performed: their average duration was around 7 hours, with a temperature of the air in laboratory room around 25°C. For every test, the instantaneous operative temperature error (i.e. the difference between the actual and the set-point values) has been investigated. Results are reported in Fig. 3.

### *4.1. Winter conditions*

Generally, winter temperature conditions can be reasonably simulated with steady-state conditions, and differences around 20°C-25°C between indoor and outdoor. Moreover, if phase change phenomena are not involved in the investigated sample, absolute values for temperature are not important. For these reasons, this kind of boundary conditions has been simulated considering 40°C-20°C and 45°C-20°C as set-point for both chambers.

First of all, it has been noticed that pumps are turned on for approximately 50% of the time, which means that there is a good balance between ON/OFF and PI controllers. Then, the average error observed is between -0.43°C and -0.23°C for the hot chamber (with bigger absolute values when the set-point is 45°C), and from -0.2°C to +0.23°C for the cold chamber. Moreover, errors are scarcely dispersed around the average, as shown by their standard deviations (which is around 0.19°C÷0.24°C for the hot chamber and 0.06°C÷0.12°C for the cold one). These findings show a good reliability of the control algorithm, represented by both the ability of the system to obtain the required conditions and the repeatability of the tests.

### *4.2. Summer conditions*

Due to significant changes of outdoor temperature during summer season, a dynamic variation of operative temperature in chambers would be more representative than a steady state approach. Actually, the whole DAVTB apparatus has been designed to be able reproduce this kind of boundary condition. However, as explained before, at this stage all efforts have been focused on steady state conditions with small temperature differences between the two chambers (which are typical for summer days). In this case, it has been observed that while hot chamber pump is active for more than the 50% of time, the other one is switched on less frequently (from 2.5% to 36% of the time). This means that cooling power is too high: temperature drops well below the desired value, then the pump is deactivated and set-point temperature reached thanks to the inertial behaviour.

Two combinations have been considered (30°C-20°C and 30°C-25°C), and average and standard deviation of errors have been investigated. As far as the hot chamber is concerned, the average error goes from -0.26°C to 0.09°C, with a standard deviation from 0.36°C to 0.7°C, which means that oscillations around set point temperature can be relevant. Considering now the cold chamber, average errors go from -0.26°C to 0.14°C, with standard deviation from 0.07°C to 0.76°C. Even if oscillations around the set point temperature are similar to that obtained for the hot chamber, in this case they are even more significant in percentage terms.

These tests have shown a greater complexity in the control at temperatures closer to that of the air in the laboratory room and for small temperature differences between the chambers. Therefore, more effort is needed to obtain a reliable tuning for the control algorithm, even if results obtained up to now will be useful for the future development of a control strategy for time-dependent thermal conditions.

## 5. Conclusions

The main purpose of this work was the definition of a reliable algorithm to control operative temperature in steady-state conditions inside the two chambers of the novel DAVTB apparatus. After some preliminary tests, aimed at the characterization of the set-up itself, a mixed ON/OFF and PI controller has been implemented. Then, thermal conditions suitable for both winter and summer seasons have been evaluated, along with the control inefficiencies (i.e. the difference between actual and expected temperature values). All analyses show that stable winter conditions can be obtained consistently; while summer conditions (smaller temperature difference between chambers, and values closer to laboratory room temperature) are more fluctuating.

Future development will deal with various aspects of the apparatus: first, thermal control for summer conditions will be improved working both on PI gains and supply water temperature; second, air velocity measurement system will be calibrated, and temperature control in steady-state condition will be updated including air flow across the sample; finally, a control strategy for sinusoidal (dynamic) thermal conditions will be implemented and tested.

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