2022

Development of an Emulator-Based Assessment Method for Representative Evaluation of the Dynamic Performance of Air Conditioners

Niccolo Giannetti

Hifni Ariyadi

Yoichi Miyaoka

Jongsoo Jeong

Kiyoshi Saito

Development of an Emulator-Based Assessment Method for Representative Evaluation of the Dynamic Performance of Air Conditioners

Niccolo GIANNETTI[1]*, Hifni ARIYADI[2], Yoichi MIYAOKA[3], Jongsoo JEONG[4], Kiyoshi SAITO[5]

[1]Waseda University, Waseda Institute for Advanced Study,
Tokyo, Japan
Contact Information (niccolo@aoni.waseda.jp)

[2]Department of Mechanical and Industrial Engineering, Universitas Gadjah Mada,
Yogyakarta, Indonesia
Contact Information (hifni.m.a@ugm.ac.id)

[3]Research Organization for Open Innovation Strategy, Waseda University,
Tokyo, Japan
Contact Information (miyaoka@aoni.waseda.jp)

[4]Research Institute for Science and Engineering, Waseda University,
Tokyo, Japan
Contact Information (jeong@aoni.waseda.jp)

[5]Waseda University, Department of Applied Mechanics and Aerospace Engineering,
Tokyo, Japan
Contact Information (saito@waseda.jp)

* Corresponding Author

ABSTRACT

The systematic assessment of representative performance of air-conditioners poses a challenge in the testing reproducibility owing to the necessity of eliminating the influence of the characteristics of the testing facilities while dynamically reproducing both the air-conditioner side and the room-side air conditions. To overcome this challenge, this study presents the development of a hybrid testing facility for the assessment of the actual dynamic performance of air conditioners. The assessment method features a virtual room emulator and the testing equipment of the actual air conditioner, which can measure up to 14kW-capacity and reproduce outdoor air temperature between -7°C and 46°C. The virtual room emulator simulates the return indoor air condition to reproduce the dynamic interactions between specific conditioned environments, load condition and the cooling capacity of the air conditioner. Consequently, the air conditioner responds to the simulated indoor air conditions by supplying cooling capacity according to the response guided by its control system. The hardware and the software sides are interfaced through a measuring chamber and a condition generator. The measuring chamber measures the supplied cooling capacity by using the air-enthalpy method and provides the digital signal of the supply air condition to the virtual room emulator, whereas the condition generator converts the digital signal from the emulator into precisely controlled experimental conditions of the return air to the air conditioner.

The virtual room emulator and the technical solutions adopted to achieve sound repeatability and reproducibility of the testing results are presented and discussed to support the reliability of the proposed assessment technique. Specifically, the effect of the thermal capacity of the measuring chamber on the dynamic measurement of the air conditioner cooling capacity is quantified and design guidelines are provided to minimize this source of bias.

1. INTRODUCTION

Efficient energy management is an indispensable requirement across different technological fields as a measure against global warming. Air conditioning is a widespread technology that has become essential in industry, transportation, and commercial sectors, as well as for maintaining people's safety and comfort on the user-end, with a particularly high coefficient of performance. As a result, air conditioners are installed in a wide range of environmental and operational conditions, including cold, hot, and humid climates, different demand requirements and dynamically changing loads interfacing the conditioned and external spaces. Accordingly, the modulated parameters (such as compressor speed, fan speed, and valve opening) dynamically respond to these disturbances based on the native control system of the air conditioner. However, Japan Industrial Standards (JIS C9612, 2018) and other international standards (AHRI:2019) regulate the performance rating with reference to a set of steady-state data collected at constant compressor speed, where the system control modulating the output capacity is overridden.
It has been lengthily pointed out that such approach in the performance rating may be not representative of the actual operational performance of ACs (Watanabe et al., 2007) and misleading for guiding the performance improvement in commercialized units. This underlying divergence between the actual and rated performance represents the biggest challenge of the HVAC sector in achieving carbon neutrality.
Consequently, the study of representative performance evaluation techniques for air-conditioners operating under their built-in control systems has been targeted by various research efforts in different countries (Ban et al., 2016; 2017; Palkowski et al., 2019; Chen et al., 2021), but a standardized methodology has not yet been fully established and the challenges arising in the reproducibility of such control-sensitive assessment procedures in different testing facilities are still to be clearly evaluated. In this regard, the development of a solid methodology requires in-depth analysis of the sources of error and measurement delay in the dynamic performance assessment technique.
This study presents the development of a hybrid testing facility for the assessment of the actual dynamic performance of air conditioners. The assessment technique combines the opportunities provided by software and hardware testing equipment. A virtual room emulator simulates the return indoor air condition to reproduce the dynamic interactions between specific conditioned environments, load condition, and the cooling capacity of the air conditioner. The hardware and the software sides are interfaced through a measuring chamber and a condition generator. The methodology and the technical solutions adopted are analyzed and discussed in relation to accuracy, repeatability, and reproducibility of the proposed assessment technique. Specifically, the effect of the thermal capacity of the measuring chamber on the dynamic measurement of the air conditioner cooling capacity is quantified and design guidelines are provided to minimize this source of bias.

2. ROOM EMULATOR AND TESTING FACILITY

The hybrid assessment method (conceptually presented in Figure 1) features a virtual room emulator, which flexibly replicate the interactions between the cooling capacity of the air conditioner and the room heat load, and a testing facility, which can measure up to 14 kW-capacity and reproduce outdoor air temperature between -7°C and 46°C.

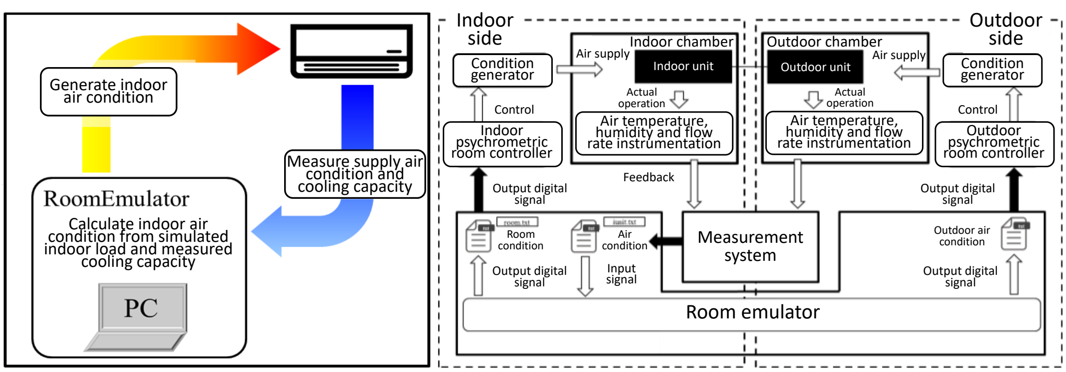


Figure 1: Conceptual representation of the hybrid performance assessment method

The virtual room emulator simulates the return indoor air condition to reproduce the dynamic interactions between specific conditioned environments, load condition, and the cooling capacity of the air conditioner. Consequently, the air conditioner responds to the simulated indoor air conditions by supplying cooling capacity according to the response guided by its control system. The hardware and the software sides are interfaced through a measuring chamber and a condition generator. The chamber measures the supplied cooling capacity by using the air-enthalpy method and provides the digital signal of the supply air condition to the virtual room emulator. The condition generator converts the signal from the emulator into experimental conditions of the return air to the air conditioner.

Table 1: Facility specifications

Measurable capacity range	Settable outdoor temperature range	Explosion-proof	Cross wind speed control	Size
Up to 14kW	-7℃ ~ 46℃	For flammable refrigerants	0.2m/s±0.1m/s	7m(D)×8m(W)×3m(H)

2.1 Room Emulator

The virtual room emulator represents the main tool for addressing the characterization of representative and time-dependent performance of air conditioners while accounting for the interaction between the measured cooling capacity and simulated heat load from the occupants and the building. It embodies the interface between the three main parts of the testing facility, namely the indoor and outdoor condition generators, and the measuring chamber.

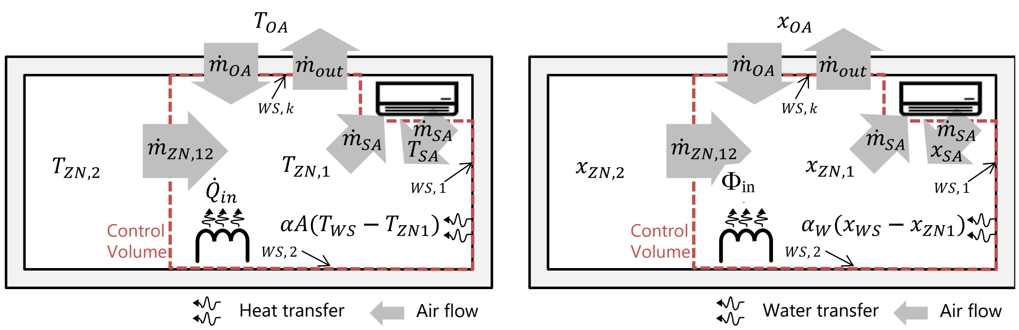


Figure 2: Schematics of the mathematical model of the virtual room emulator

The room (schematically represented in Figure 2) mainly experiences the following physical phenomena: I. heat infiltration from the external environment; II. ventilation and heat transfer from solar radiation through windows; III. internal generation of sensible and latent heat from room occupants, lighting, and indoor equipment; IV. heat and moisture accumulation due to heat and mass capacities of the volume of indoor air, fixtures, walls, and floors.

$$\dot{m}_{out} = \dot{m}_{OA} + \dot{m}_{ZN12} \tag{1}$$

$$C_{S,ZN1}\frac{dT_{ZN1}}{dt} = c_{p,SA}\,\dot{m}_{SA}\left(T_{SA}-T_{ZN1}\right)+Q_{in} \tag{2}$$

$$M_{L,ZN1}\frac{d\varphi_{ZN1}}{dt} = \dot{m}_{OA}\,\varphi_{OA} + \dot{m}_{SA}\left(\varphi_{SA}-\varphi_{ZN1}\right)+\Phi_{in} \tag{3}$$

In the present study, the mathematical model is based on the continuity of indoor air (Equation 1), the energy equation (Equation 2), moisture balance (Equation 3), and the auxiliary relations for heat and mass transfer of walls and fixtures considering thermal and moisture capacity. However, the model formulation may be freely modified to capture different characteristics with different levels of accuracy and complexity. In fact, the representativeness and reproducibility of the results require dedicated studies for the development an advanced and standardized emulator.

2.2 Testing Facility

According to the elementary formulation represented by Equations (1-3), first it is necessary to measure the time-varying cooling capacity and power consumption of the air conditioner, which are to be continuously fed into the room emulator and, contemporarily, recreate reproducible outdoor and indoor return air conditions by dedicated condition generators in accordance with the virtual room emulator. A schematic diagram of the correspondingly required equipment is shown in Figure 3. In this study, the air-enthalpy measurement technique is adopted for the dynamic measurement of the cooling capacity of the air conditioner (Ban et al, 2017) by the instantaneous measurement of the air flow rate and the specific air enthalpy difference between suction and discharge of the indoor unit (based on temperature and humidity measurements).

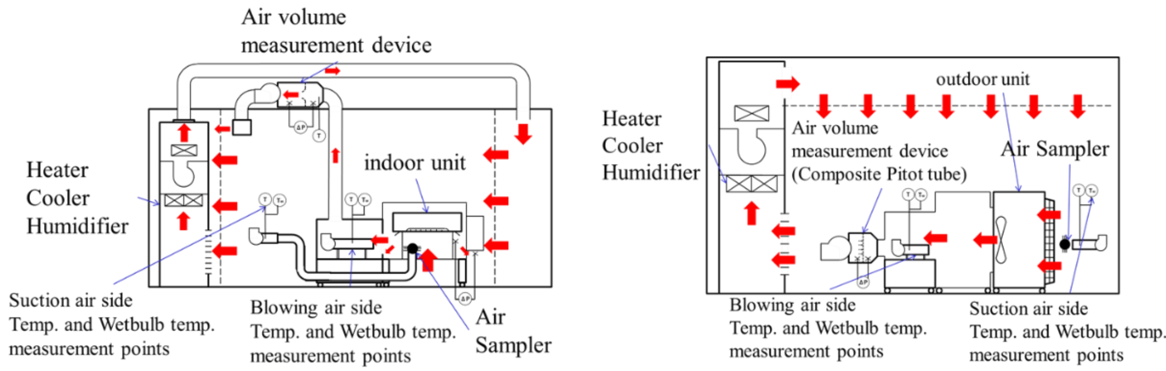


Figure 3: Schematic air flow diagrams of (a) the indoor unit room and (b) outdoor unit room

As prescribed by ANSI/ASHRAE (2006) guidelines, common practices in the measurement of the average temperature and flow rate of an airstream requires suitable precautions to deal with I. nonuniformity of the temperature across the airstream; II. nonuniformity of the air velocity; and III. nonuniformity of the humidity ratio of the air. Additionally, when temperature measurements are required in connection with the estimation of heat transfer rates, suitable precautions should be taken between air inlet and outlet measurement locations to minimize the effect of I. heat gains or losses to test apparatus ductwork, for both steady state and transient conditions; II. Heat gains or losses due to air leakage from or into the test apparatus; III. unsuitable instrument response time; and IV. thermal capacity of the structural and sensing components. Accordingly, a suitably designed measuring chamber is essential for ensuring reliability in the measurement of temperature, humidity, and flow rate of the airstream from the air conditioner. When dynamic capacity measurement procedures are to be carried out, the design of this equipment and the data reduction methodology should unavoidably account for the measurement delay due to sensors displacement and the thermal capacity of the chamber. Figure 3 shows the air flow diagrams of the indoor and outdoor unit rooms. In the indoor unit room, an air sampler is placed in front of the indoor unit and an inlet air temperature and humidity measuring devices are connected through a duct. The air drawn in by the indoor unit is supplied to the measuring chamber after heat exchange with the refrigerant in the indoor unit. The design of the chamber adopted in this study is constructed to gather and mix the air flow supplied by ceiling-mounted cassette-type air conditioners, as shown in Figure 3(a). The blown air is fed into a blown-air greenhouse temperature meter, which measures the dry- and wet-bulb temperatures and transported to an airflow measuring device, which measures the air flow rate. The air flow measuring device is equipped with nozzles of four different diameters providing opening and closing optimal combinations in relation to the magnitude of the air flow rate. In these processes, dry bulb and wet bulb temperatures and airflow are measured at a certain physical distance from the outlet. Finally, the air is circulated to the indoor unit room. The heat load in the indoor unit room is reproduced in terms of temperature and humidity by the condition generator, featuring a cooler, a heater, and a boiler. Temperature, humidity, and air flow rate in the outdoor unit room are similarly measured with the only difference that the outdoor air condition is controlled by supplying a conditioned air stream from the ceiling (Figure 3b).

Table 2: Static test conditions for the accuracy certification of the testing facility

Test condition	Dry-bulb (wet bulb) indoor temperature	Dry-bulb (wet bulb) outdoor temperature	Cooling (heating) capacity (JATL)	Cooling (heating) capacity (Waseda)	Error
Cooling	27 ℃ (19 ℃)	35 ℃ (24 ℃)	7038 W	6926 W	-1.6%
Heating	20 ℃ (14.5 ℃)	7 ℃ (6 ℃)	(7845 W)	(7730 W)	-1.5%
Heating*low T	20 ℃ (14.5 ℃)	2 ℃ (1 ℃)	(8927 W)	(8715 W)	-2.4%

The accuracy of the testing facility was firstly verified in accordance with JIS annual performance tests (JIS B 8615, 2013) for a 7.1 kW air conditioner owned by the Japan Air Conditioning and Refrigeration Testing Laboratory (JATL). Experimental conditions and relative deviations in the measured cooling and heating capacity between

JATL and the present facility are summarized in Table 2. It is shown that results were collected with high accuracy and a maximum error of 2.4%, which resulted in the quasi-certification of the present facility as a -satellite laboratory- of JATL on 1 October 2020.

3. SOURCES OF MEASUREMENT DELAY IN DYNAMIC TESTS

The development of a dynamic performance testing method, representing the system response according to its native control method, requires an assessment of the sources of measurement error and delay. Specifically, the factors to be considered are those affecting the modulation of the indoor air temperature and humidity through the input signals to the virtual room emulator. Consequently, errors and delay in the modulations of the indoor air condition due to the testing methodology interfere with the repeatability and the reproducibility of the dynamic test. Among these factors, I. the calculation time delay of the emulator; II. the air flow rate, temperature and humidity tracking at the condition generator; III. the time delay of the signal from various sensors; and IV. the air condition tracking, heat transfer and thermal capacity of the structure and instrumentation of the measuring chamber are hereby evaluated. A first assessment of these factors is conducted for a 10-kW system installed in a virtual 147 m^3 room, which, at standard airflow rate, has a thermal time constant of approximately 500 s. Note that the influence of the thermostat location (in this case, placed at the suction) is hereby disregarded, and it would require a dedicated study (Cheng et al., 2021).

3.1 Calculation Time Delay of the Emulator (I.)

The thermal response of the indoor space is to be calculated by the virtual room emulator, which discretizes the differential equations representing the room model in a first order forward finite difference approximation. As the calculation time for a time step of 1 s is approximately 0.5 s, the calculation time delay of the emulator may be disregarded (provided that time step and calculation power of the computer are properly selected).

3.2 Air Flow Rate, Temperature and Humidity Tracking at the Condition Generator (II.)

The digital signal generated by the emulator defines the temperature and humidity of the return indoor air to the suction of the indoor unit. Accordingly, during a dynamic test, the signal is to be converted to actual air stream conditions, which should track the digital signal with minimal time delay and maximal accuracy. Corresponding statements refer also to the mirroring of the temperature and humidity to the outdoor unit.
Tests were conducted for start-up cooling and heating (Figure 4), as well as intermittent cooling conditions (Figure 5). Temperature and humidity at the outdoor unit (Figures 4b and 4d) were stably reproduced by the condition generator (continuous lines) according to the target values from the emulator (dashed lines), and a delay of approximately 20 s was demonstrated in tracking indoor temperature and humidity (Figures 4a and 4c).

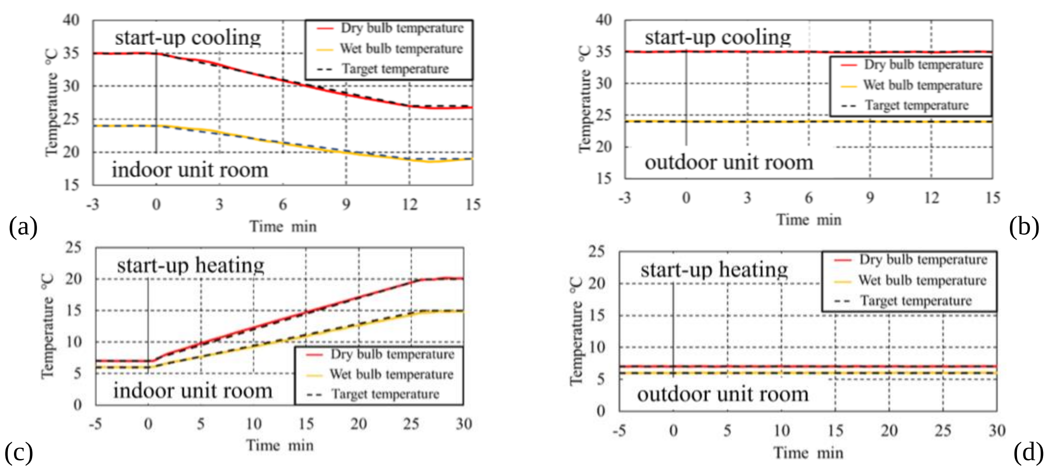


Figure 4: Tracking of target temperature and humidity of the condition generator at the indoor unit room and outdoor unit room for start-up cooling, start-up heating conditions

Figure 5 shows that dry-bulb and wet-bulb temperatures from the condition generator at the indoor unit room are able to closely follow the signals from the emulator during intermittent operation. In this case, the maximum delay between the target and the actual temperature and humidity was 64 seconds, and about 20 seconds in average. This means that the maximum (and average) delay is 1.2% (and 0.4%) of the time constant of the room.

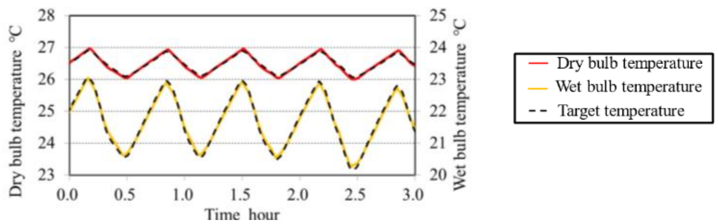


Figure 5: Tracking of target temperature and humidity of the condition generator at the indoor unit room for intermittent cooling condition

Additionally, the volumetric air flow rate is estimated from the nozzle differential pressure of the air flow measuring device (Figure 3a), but sudden variations of the air flow rate according to the control of the air-conditioner fan momentarily create a pressure differential which interferes with an accurate measurement. Accordingly, a dedicated control system has been introduced to minimize the differential pressure between the inner and outer sides of the chamber. Figure 6 exemplifies the response of this control system to a stepwise change of the fan operation. It was confirmed that the corresponding delay is contained within 25 sec (approximately 0.5% of the room time constant).

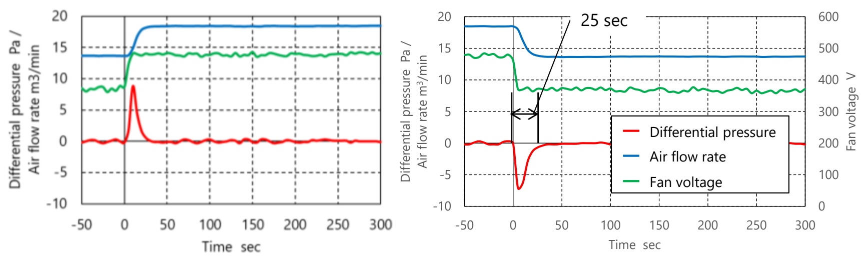


Figure 6: Time delay time of the differential pressure control for flow rate measurement

3.3 Time Delay of the Sensor Signal (III.)

Several sensors, including thermoresistors, thermocouples and hygrometers, were installed in the testing facility. These exhibited a measurement delay which stays below 15 seconds.

3.4 Air Condition Tracking and Heat Transfer of the Measuring Chamber (IV.)

The measuring chamber consists of the equipment and sensors installation sections, as shown in Figure 7. The insulation wall is a 50-mm thick urethane board sandwiched between 0.4-mm thick structural steel plates.

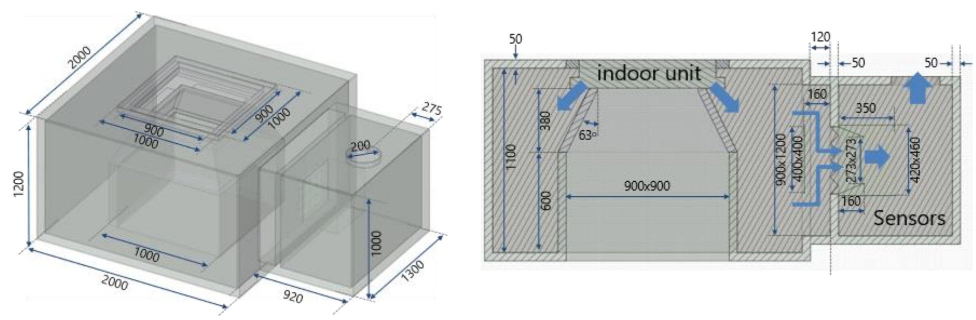


Figure 7: Schematic configuration of measuring chamber for a ceiling type indoor unit

The air conditioning units are installed in the upper center of the equipment installation area. Air drawn in from the center is blown out against the ventilation channels. The blown air passes through a rectifier, after which the temperature and humidity are measured. The displacement of the instrumentation for temperature and humidity measurement results in a minimum measurement delay due to the time required for the supply air to cross the chamber and the rectifier (approximately 4.9 s, 7.6 s, and 13.7 s for high, medium, and low flow rates, respectively). Additionally, the wall area and thermal capacity of the chamber structure are responsible for some heat losses/ infiltrations and a thermal response which results in a measurement delay during dynamic tests.
Static tests were conducted to evaluate the heat transfer through the wall when the room temperature (external to the chamber) is kept at 25 °C while the air temperature is supplied in the camber by a spot cooler. At steady state

conditions, the amount of heat infiltration from the measuring chamber was approximately 100 W, which is 1% of the rated capacity of a 10-kW system. Lower heat transfer through the wall may be achieved with lower transfer area, larger thickness or lower thermal conductivity of the walls.
The thermophysical properties of the insulation material and the size of the chamber would also affect the thermal response of the measuring equipment during dynamic tests. The data gathered for a step-wise heating test, from 25 ℃ to 35 ℃, are plotted in Figure 8. The room temperature is set to a constant value of 25 ℃, which corresponds to the initial temperature condition of the walls. With the present design of the measuring chamber a maximum heat transfer of almost 1200 W and a time delay of approximately 195 s are encountered. This demonstrates the need for a careful design of the measuring chamber in both static and dynamic performance tests.

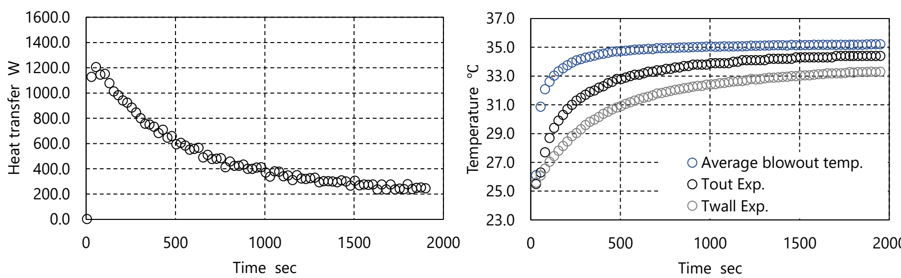


Figure 8: Air heat transfer and temperature response in the measuring chamber for a 25-35 step-wise heating test

4. ANALYSIS OF THE MEASURING CHAMBER

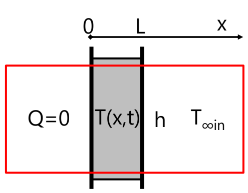
According to the investigation presented above, the time delay, heat infiltration, and thermal capacity of the measuring chamber appear to be the major challenges in achieving sound repeatability and reproducibility of the hybrid assessment method for dynamic testing of air conditioners. The following analysis clarifies the effect of different design features of the measuring chamber on the dynamic measurement of the air conditioner cooling capacity to provide design guidelines and minimize this source of bias.

4.1 Analytical Study Influential Factors

The mentioned factors affecting the measurement delay may be approximated as a 1-dimensional transient heat transfer problem between the wall and the air stream within the measuring chamber as in Equation (4).

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{a}\frac{\partial T}{\partial t} \quad , \quad a = \frac{k}{\rho c} \tag{4}$$

As a first approximation, valid for relatively short time windows, the transient phenomenon may be represented by the boundary and initial conditions illustrated in Figure 9. In these circumstances, the analytical solution of the transport equation provides a general description of the temperature field within the wall (Equation 5) and heat transfer to the air stream (Equation 6), along with the characteristic equation and Fourier coefficients (Equation 7), where the eigenvalues are represented by λ_n.



-Boundary conditions $\quad \frac{\partial T(0,t)}{\partial x} = 0 \quad , \quad -k\frac{\partial T(L,t)}{\partial x} = h\left(T(L,t) - T_{\infty in}\right)$

-Initial conditions $\quad T(x,0) = T_i$

Figure 9: Representation and boundary conditions for the analytical solution of the 1-D energy transport equation

The analysis of this solution highlights that the proper tuning of physical features, such as wall thickness *L*, transfer area *S*, thermal conductivity *k*, density ρ, specific heat *c*, and the air-side heat transfer coefficient *h*, is to be considered for a suitable design of the measuring chamber. Specifically, an appropriate design should balance the conflicting requirements of minimizing heat transfer through the wall and minimal thermal inertia, while producing homogeneous temperature and velocity fields for accurate tracking of the air stream conditions.

$$T\left(x,t\right)=T_{\infty in}+\left(T_{i}-T_{\infty in}\right)\sum_{n=1}^{\infty}A_{n}e^{-\lambda_{n}^{2}\frac{kt}{\rho cL^{2}}}\cos\left(\frac{\lambda_{n}}{L}x\right) \tag{5}$$

$$Q=kS\left(T_{i}-T_{\infty in}\right)\frac{\lambda_{n}}{L}\sum_{n=1}^{\infty}A_{n}e^{-\lambda_{n}\frac{kt}{\rho cL^{2}}}\sin(\lambda_{n}) \tag{6}$$

$$\lambda_{n}\tan\left(\lambda_{n}\right)=\frac{hL}{k}\quad ,\quad A_{n}=\frac{4\sin\left(\lambda_{n}\right)}{2\lambda_{n}+\sin\left(2\lambda_{n}\right)} \tag{7}$$

Figure 10 depicts the temperature field across the wall thickness for different values of the specific heat, thermal conductivity, and thickness of the wall in comparison to the baseline results, which refer to the properties of 50-mm thick urethan board material. While the corresponding heat transfer to the air stream is plotted in Figure 11.

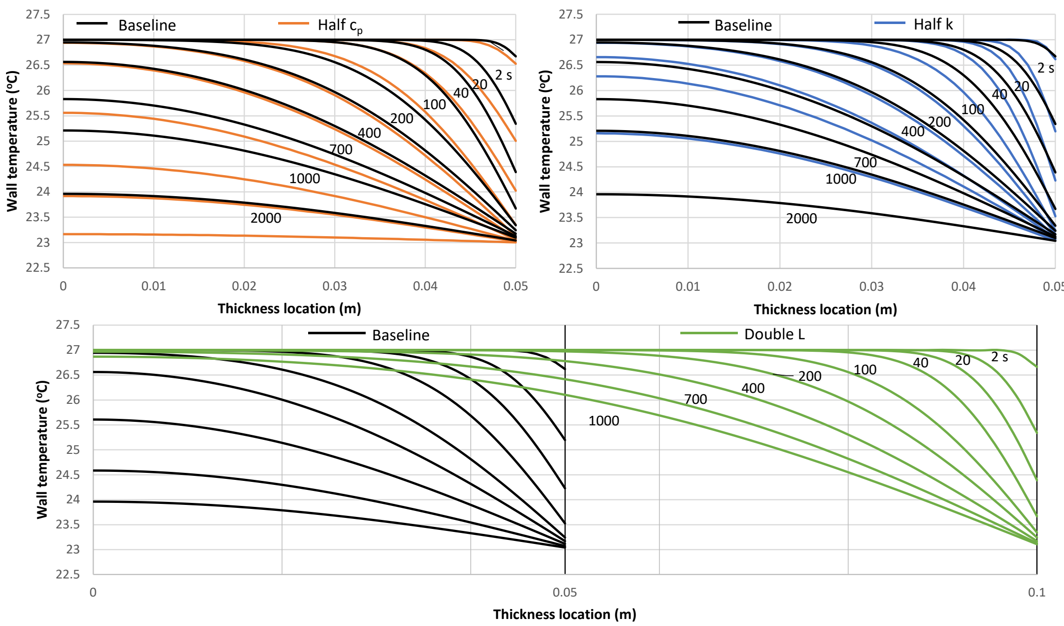


Figure 10: Wall temperature field for different material properties in comparison with the 500-mm urethan foam

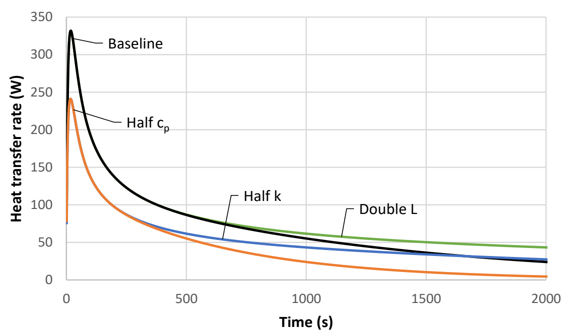


Figure 11: Chamber wall heat transfer response for different material properties

These results demonstrate that effect of the transient heat transfer due to the thermal capacity of the chamber structure is minimized using wall materials with lower specific heat, thickness, and thermal conductivity.

4.2 CFD Analysis

Given the complexity of the actual boundary conditions and the 3-dimensional fluid dynamics of the air stream in the measuring chamber during dynamic tests, thermal and flow characteristics are further investigated by means of a CFD study, which considers wall and chamber air volume as the 3-dimensional calculation domain (Figure 12).

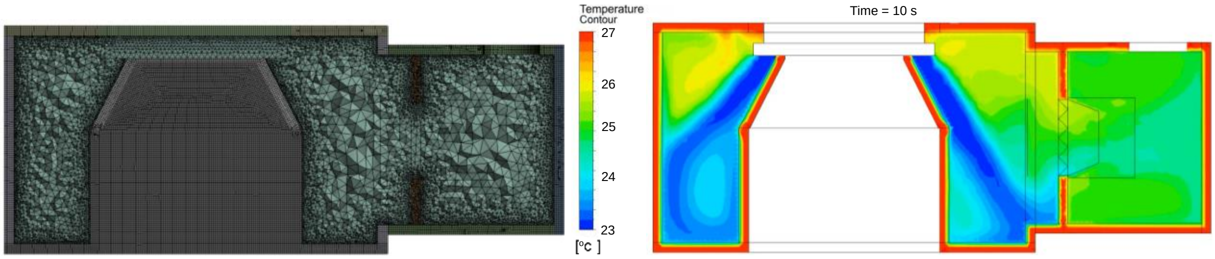


Figure 12: Calculation domain and temperature profile within the simulated chamber

A k-ε realizable turbulence model is adopted for the air side, and a coupled heat transfer model with shell conduction for the walls. The mesh sensitivity analysis resulted in the use of a total element number of approximately 1.1 million elements and the model was validated against the reference step-wise heating test in Figure 8. When the material properties are set as for the present design of the chamber and instrumentation equipment, the comparison with the experimental data shows that the model can accurately represent the transient thermal interactions between the chamber and the air stream from the ceiling type indoor unit (Figure 13).

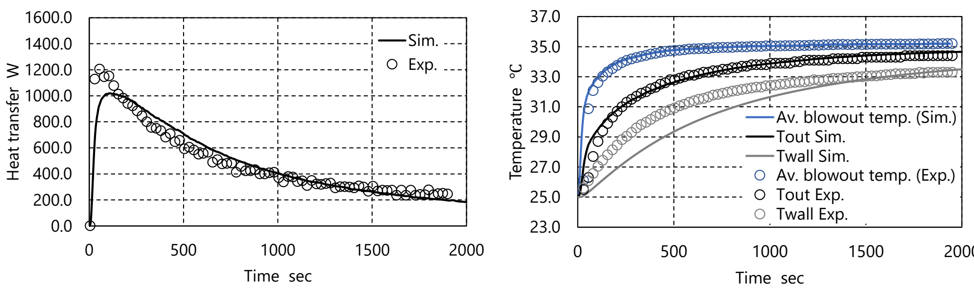


Figure 13: Validation of the air heat transfer and temperature response in the measuring chamber

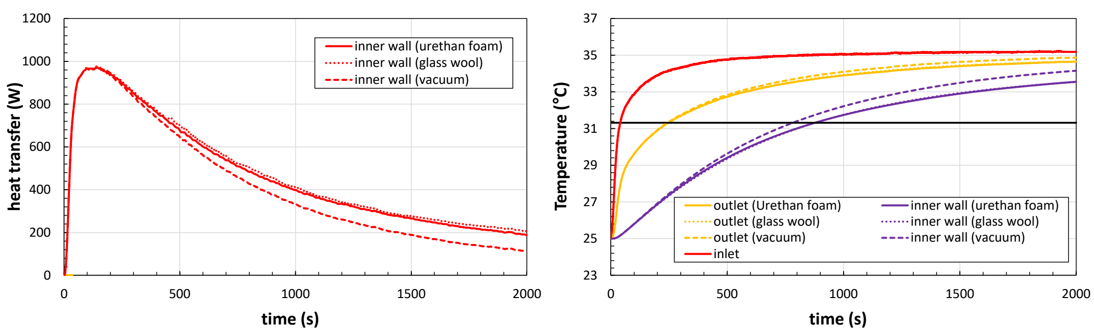


Figure 14: Air heat transfer and temperature response for the materials in Table 3 (with metallic plates)

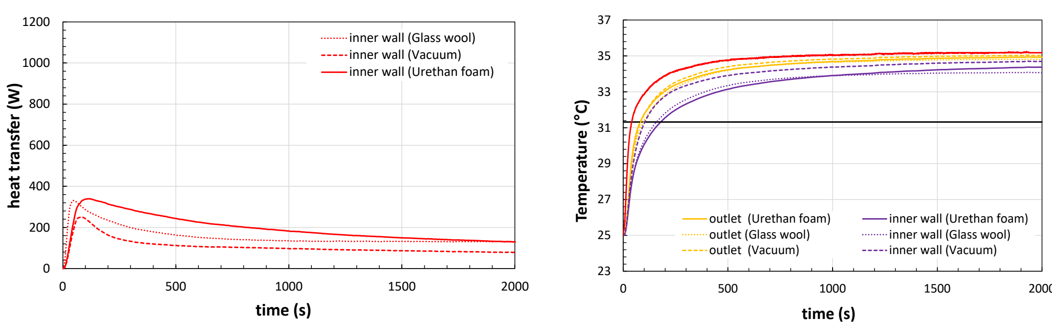


Figure 15: Air heat transfer and temperature response for the materials in Table 3 (without metallic plates)

Finally, the model is used to investigate possible alternative materials for the measuring chamber (Table 3). The corresponding results are summarized in Figure 14, where it is shown that, if the structural steel plates are maintained, alternative internal insulating materials affect the sole heat infiltration through the chamber walls.

Conversely, if the structural steel plates are removed, the same set of alternative internal insulating materials substantially reduces the measurement delay (~56 s) and the thermal inertia of the chamber (maximum ~340 W).

Table 3: Properties and thickness of alternative insulation materials for the measuring chamber

Material	Thickness	Density	Thermal conductivity	Specific heat
SUS 304	0.4 mm	7930 kg/m^3	45 W/m·K	590 J/kg·K
Urethan foam	50 mm	35 kg/m^3	0.021 W/m·K	1700 J/kg·K
Glass wool	50 mm	24 kg/m^3	0.038 W/m·K	840 J/kg·K
Vacuumed insulation	50 mm	210 kg/m^3	0.002 W/m·K	840 J/kg·K

5. CONCLUSIONS

The developed testing technique exhibits (1) a delay in the temperature and humidity sensors of the intake air (about 15 seconds), (2) a delay in the passage through the measuring chamber (about 15 seconds, in parallel to the measurement delay), (3) a delay in the sensor of the temperature and humidity of the supply air (about 15 seconds) and (4) a delay in the condition generator (about 20 seconds), which totals to about 50 seconds. As the thermal time constant of the target room is about 5000 s (humidity time constant ~500 s), the various thermal delays in the testing facility represent approximately 1% (10%), in total.

The measuring chamber enables accurate measurement of temperature, humidity, and flow rate, but its thermal inertia introduces a noteworthy time delay and heat transfer to the air stream from the structure of the chamber during dynamic measurements unless a proper structural redesign is considered. In the simulated cases the maximum heat transfer and time delay were kept below 340 W (~3% of the nominal capacity) and 56 s (~1% of the room thermal time constant), respectively. Alternatively, dynamic tests may have to rely on a direct measurement of the supplied air temperature before mixing in the chamber, which poses experimental challenges in the accuracy of the estimation of air stream conditions for a ceiling type indoor unit. Additionally, the estimation of the chamber characteristics may be used to estimate the heat infiltration through the walls for correcting the static operation measurement and improve reproducibility of the testing methodology.

REFERENCES

AHRI Standard 340/360-2019. (2019). Performance Rating of Commercial and Industrial Unitary Air Conditioning Condensing Units. *Air-Conditioning, Heating, and Refrigeration Institute (AHRI)*.

ANSI/ASHRAE Standard 41.1-1986 (RA 2006). Standard Method for Temperature Measurement, American Society of Heating, *Refrigerating and Air-Conditioning Engineers, NE, Atlanta, GA 30329*.

Ban, T., Ohno, K., Yamaguchi, S., Saito, K., Yamaguchi, H. (2016). Evaluation Method of Compression type Heat Pump for Actual Driving Performance ―Construction of evaluation device and proposition of unsteady evaluation method―(in Japanese). *Proceedings of 2016 JSRAE Annual Conference, Kobe, Japan*.

Ban, T., Nakano, D., Ohno, K., Yamaguchi, S., Saito, K., Yamaguchi, H. (2017). Testing facility for unsteady performance of air conditioning System. *Proceedings of 12th IEA Heat Pump Conference 2017, Rotterdam.*

Cheng, L., Dhillon, P., Horton, W.T., Braun, J.E. (2021). Automated laboratory load-based testing and performance rating of residential cooling equipment. *International Journal of Refrigeration*, 123, 124-137.

JIS C8615: 2013. (2013). Room Air Conditioners; Japanese Industrial Standards Committee: Tokyo, Japan.

JIS C9612: 2018. (2018). Room Air Conditioners; Japanese Industrial Standards Committee: Tokyo, Japan.

Palkowski, C., Zottl, A., Malenkovic, I. and Simo, A., (2019). Fixing Efficiency Values by Unfixing Compressor Speed: Dynamic Test Method for Heat Pumps, *Energies*, 12(6), 1045.

Watanabe, C., Nagamatsu, K., Nakayama, H., Hirota, M., Ohashi, E. (2007). Performance characteristics of multi-type air-conditioners for building under partial thermal load operations. *Int. Symposium EcoTopia Science*.

ACKNOWLEDGEMENT

This paper is based on results obtained from a project commissioned by the New Energy and Industrial Technology Development Organization (NEDO).