Measurement data analysis for heat balance of air-conditioning system in actual office space

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Abstract. Research on net-zero energy buildings (nZEB), is currently increasing. To realize a ZEB, it is important to develop a control system for air conditioners and ventilators. This research aims to construct a simulator for office air conditioning to evaluate system control technology.

Generally, more than two air conditioning systems are installed in offices, and the cooling operations of each are influenced by the other system's operation status. To realize a simulator that can capture this effect, we take into account the building envelope, fluid dynamics, and actual system controls. The cooling load and capacity from every source should be balanced in a steady state; thus, obtaining experimental data within a certain error is critical to simulate the power consumption of office cooling precisely.

The purpose of this experiment is to analyze the heat balance between the measured cooling load and cooling capacity evaluated from the change in a) inlet and outlet air enthalpy of the air conditioners, and b) enthalpy of the refrigerant.

A measurement in an actual office is carried out to study space cooling considering the building environment, office layout (e.g., desks), electricity consumption by computers and lighting, and cooling operation status of the air-conditioning systems. Temperature/humidity sensors, electric-power meters, and several other types of sensors are installed. The office is located on the top floor of a four-story building completed in 1997. The air-conditioning system consists of two VRF outdoor units with a cooling capacity of 28 kW, with six indoor units for perimeter and interior, respectively, and a total of 1600m³/h ventilation flow with the ventilator units.

Good heat balance is observed with errors within 10% between the measured cooling load and a), and 15% between measured cooling load and b).

Keywords. cooling, measurement data, heat balance, japanese office, load

1. Introduction

Research on net-zero energy buildings (nZEB) is currently increasing. To realize nZEBs, the Japanese government is accelerating nZEB technology development by providing subsidies to office builders that introducing equipment with high energy performance to their buildings. Air conditioners are one of the main systems that consume electric power in buildings, and thus it is important to construct simulators for office air conditioning to evaluate the system control technology for reducing electric-power consumption. Generally, as shown in Figure 1, when one of two air conditioners operates at a low cooling capacity, the other will operate at a high cooling capacity. This is because the total cooling capacity for all air conditioners installed in an office is equal to the cooling loads from all sources. To capture this, we are developing an integrated simulation of computational fluid dynamics (CFD) and refrigerant cycle simulation with actual system control software. Calculating the capacities of air conditioners and air conditions with ventilation in an office space simultaneously allows us to evaluate multiple air-conditioning control for energy performance. The data balancing between the load and capacity is fundamental data that is compared with the simulation results to develop highly accurate calculation models. Additionally, the conditioned air is cooled by the refrigerant flowing into
the air conditioner, and thus it is also important to confirm the heat balance between the refrigerant and air.

**Figure 1.** Heat balance of air-conditioning system.

Recently, researchers have studied building energy performance. For example, the characteristics of energy consumption in various building types were analyzed (Xiaoyu LIU et al., 2016[1]). In another case, the potential for nZEB in existing buildings was analyzed for assumed building specifications (Jun HARATA et al., 2014[2]); however, there is insufficient detail regarding the energy balance of the actual measurement results of the air-conditioned space. In this research, we validate the energy balance between the air-conditioning load and capacity from measurement data.

**2. Measurement conditions**

**2.1. Objective office space**

The building containing the objective office space used in this study is in Kamakura city, Kanagawa, Japan, and was completed in 1997. The target office is approximately 12.7 × 27.9 m with a height of 2.7 m, and is located on the top floor of the four-story building. The south and west walls are 225-mm thick external walls with an approximately 0.3-W/m²K thermal transmittance, consisting of 22-mm plasterboard, 45-mm air layer, 25-mm rock wool, and 133-mm ALC panels. In contrast, the north and east walls separating this office space and other rooms/corridors are internal walls with an approximately 0.2-W/m²K thermal transmittance. The thermal transmittance values of the ceiling and floor are approximately 4.0 and 0.2 W/m²K, respectively. The office also has a south-facing window of 1.7 m in height and 5.8 W/m²K thermal transmittance, with window eaves that block direct solar radiation during summer. Figure 2 illustrates the system structure and unit layout of the ductless air conditioners and ventilator systems. The air conditioners are two variable-refrigerant flow (VRF) outdoor units providing 28-kW rated cooling capacity with six four-way ceiling cassette indoor units for the perimeter and interior, respectively. Two Lossnay units perform a heat-recovery function from the exhaust air to the supply air with a 67% enthalpy effectiveness for both temperature and humidity. The rated total ventilation flow outputted from the two Lossnay units is 1600 m³/h; they are supplied by six supply ports and the exhaust ports exhaust to the ceiling.

**Figure 2.** System structure of air conditioners and ventilators installed in measured office area.
2.2. Heat-balance evaluation

The heat balance of the air conditioners in terms of cooling capacity and load are described in Figure 3 with the measurement points.

![Figure 3. Schematic diagram of heat balance and measurement points of air conditioners.](Image)

The heat balances of the air conditioners in terms of load are evaluated using Eq. (1) for the refrigerant, and Eq. (2) to Eq. (4) for the total, sensible, and latent heat, respectively, for air.

\[
\begin{align*}
    r_e &= \frac{\sum (Q_r - L_i)}{\sum L_t} \times 100 \\
    r_{a,t} &= \frac{\sum (Q_{a,t} - L_i)}{\sum L_t} \times 100 \\
    r_{a,s} &= \frac{\sum (Q_{a,s} - L_i)}{\sum L_s} \times 100 \\
    r_{a,l} &= \frac{\sum (Q_{a,l} - L_i)}{\sum L_l} \times 100
\end{align*}
\]

Eq. (1), Eq. (2), Eq. (3), Eq. (4)

2.3. Cooling capacity

Refrigerant capacity \( Q_r \) can be represented as Eq. (5), where the refrigerant enthalpies at inlet \( h_{in} \) and outlet \( h_{out} \) for the indoor units are determined from the temperature and pressure sensors installed inside the outdoor units. In addition, refrigerant mass flow rate \( G_r \) was determined from the mass flowmeter (precision ± 0.5%) installed in the refrigerant pipe connecting the outdoor unit and indoor units. The right side of the formula is the summation of the capacities of the two refrigerant circuits, i.e., interior and perimeter.

\[
Q_r = \sum_{j=1}^{iN} \left\{ \frac{G_r(j)}{3600} \times [h_{out}(j) - h_{in}(j)] \right\}
\]

Eq. (5)

The air capacity of the total, sensible, and latent heat can be represented by the summation of the capacities of all indoor units as Eq. (6) to Eq. (8), respectively, where the air enthalpy for the inlet \( I_{in} \) and outlet \( I_{out} \) and air density \( \rho_a \) are determined from the temperature and humidity sensors (precision ± 0.3 °C; ±5% RH) installed in each indoor unit. The air-flow volumes \( V_{fic} \) of all indoor units are measured (precision ± 3%) independently before taking measurements for the experiment.

\[
\begin{align*}
    Q_{a,t} &= \sum_{i=1}^{IN} \left\{ \frac{V_{fic}(i)}{3600} \times \rho_a(i) \times [I_{in}(i) - I_{out}(i)] \right\} \\
    Q_{a,s} &= \sum_{i=1}^{IN} \left\{ \frac{V_{fic}(i)}{3600} \times \rho_a(i) \times c_p a(i) \times [T_{in}(i) - T_{out}(i)] \right\} \\
    Q_{a,l} &= Q_{a,t} - Q_{a,s}
\end{align*}
\]

Eq. (6), Eq. (7), Eq. (8)

2.4. Cooling Load

The cooling loads from various sources in the office space are as shown in Figure 4 and are given by

\[
\begin{align*}
    L_t &= L_{light} + L_{plug} + L_{occ,t} + L_{wall} + L_{wind} + L_{floor} + L_{ceiling} + L_{vt} + L_{inf,t} \\
    L_s &= L_{light} + L_{plug} + L_{occ,s} + L_{wall} + L_{wind} + L_{floor} + L_{ceiling} + L_{vs} + L_{inf,s} \\
    L_l &= L_{occ,l} + L_{vl} + L_{inf,l}
\end{align*}
\]

Eq. (9), Eq. (10), Eq. (11)
Figure 4. Schematic diagram of cooling-load structure in office space.

where the calculation of each heat source element, namely total heat \( L_t \), sensible \( L_s \), and latent \( L_l \), is described as follows. The heat gains from luminaire and office automation (OA) plugs are determined as

\[
L_{\text{light}} = W_{\text{light}} \times 0.5 \\
L_{\text{plug}} = W_{\text{plug}}
\]

respectively, where \( W_{\text{light}} \) and \( W_{\text{plug}} \) are measured by electric-power meters (precision \( W_{\text{light}} \pm 0.1 \text{ kW}; W_{\text{plug}} \pm 0.4 \text{ kW} \)). We assume that half of the lighting heat gain was in the office area, and the remaining gain was to the ceiling plenum\(^3\). The occupant loads given by Eq. (14) to Eq. (16) can be determined by the number of occupants \( N_{\text{occ}} \), based on occupant counting by a PC typing-detection tool, and assuming \( L_{\text{occm,s}} \) and \( L_{\text{occm,l}} \) as 65 and 55 W per employee, respectively \(^3\).

\[
L_{\text{occ,s}} = N_{\text{occ}} \times L_{\text{occm,s}} \quad \text{Eq. (14)} \\
L_{\text{occm,l}} = N_{\text{occ}} \times L_{\text{occm,l}} \quad \text{Eq. (15)} \\
L_{\text{occ,t}} = L_{\text{occm,s}} + L_{\text{occm,l}} \quad \text{Eq. (16)}
\]

Heat flux sensors (precision ±3\%) are installed on the walls in each direction, as well as on the ceiling and floor. The loads can be determined as

\[
L_{\text{wall}} = \left( \frac{A_{\text{wall}} \times q_{\text{wall}}}{1000} \right) \\
L_{\text{floor}} = \left( \frac{A_{\text{floor}} \times q_{\text{floor}}}{1000} \right) \\
L_{\text{ceiling}} = \left( \frac{A_{\text{ceiling}} \times q_{\text{ceiling}}}{1000} \right)
\]

The heat-flux values include solar radiation because the external wall and floor surface temperature rise with the absorption of radiation energy. We determined the window load as shown in Eq. (20) by utilizing the blind surface temperature for the covered window, since the heat-flux sensor errors are influenced directly by the solar-radiation heating. A convection heat transfer coefficient \( \alpha_{\text{wind}} \) of 12 W/m\(^2\)K is assumed. Furthermore, the effective window area rate \( \Phi_{\text{wind}} \) is considered as 85\% owing to the blind-opening rate.

\[
L_{\text{wind}} = \left( \frac{A_{\text{wind}} \times \Phi_{\text{wind}} \times \alpha_{\text{wind}}}{1000} \right) \times (T_b - T_{\text{ro}}) \quad \text{Eq. (20)}
\]

The ventilation loads of the total, sensible, and latent heat are calculated as

\[
L_{v,t} = \frac{V_{f,v}}{3600} \times \rho_a \times (I_{\text{ro}} - I_{\text{sa}}) \quad \text{Eq. (21)}
\]


\[ L_{\text{vent,s}} = \frac{V f_{\text{vent}}}{3600} \times \rho_a \times C_p, a \times (T_{\text{ro}} - T_{sa}) \]

\[ L_{\text{vent,t}} = L_{\text{vent,s}} + L_{\text{vent,t}}, \]

Eq. (22)

\[ L_{\text{inf,s}} = \frac{N_{\text{inf}}}{3600} \times V_{\text{ro}} \times \rho_a \times (I_{oa} - I_{ro}) \]

\[ L_{\text{inf,t}} = L_{\text{inf,s}} - L_{\text{inf,t}}, \]

Eq. (23)

respectively, where the supply air temperature \( T_{sa} \) and enthalpy \( I_{oa} \) are determined from the average air temperature and humidity detected by the temperature and humidity sensors installed in the supply ports; thus, the \( T_{sa} \) value is smaller than that of the outdoor temperature owing to the heat-recovery function of the Lossnay. The return air (room air) temperature \( T_{ro} \) and enthalpy \( I_{ro} \) are determined by the temperature and humidity sensors installed in the office space. The ventilation volume \( V f_{\text{vent}} \) was determined from the total flow volume for each supply port measured prior to operation. Under 0.5 the number of infiltration loads \( N_{\text{inf}} \) is determined as

\[ L_{\text{inf,s}} = \frac{N_{\text{inf}}}{3600} \times V_{\text{ro}} \times \rho_a \times (I_{oa} - I_{ro}) \]

\[ L_{\text{inf,t}} = L_{\text{inf,s}} - L_{\text{inf,t}}, \]

Eq. (24)

\[ L_{\text{inf,s}} = \frac{N_{\text{inf}}}{3600} \times V_{\text{ro}} \times \rho_a \times (T_{oa} - T_{ro}) \]

\[ L_{\text{inf,t}} = L_{\text{inf,s}} - L_{\text{inf,t}}, \]

Eq. (25)

\[ L_{\text{inf,s}} = L_{\text{inf,s}} - L_{\text{inf,t}}, \]

Eq. (26)

where the outdoor temperature \( T_{oa} \) and enthalpy \( I_{oa} \) are determined from the temperature and humidity sensors installed in the Stevenson screen on the rooftop. Minimal solar radiation was detected by the sensors located near the south windows, and thus it was not considered in the heating load.

### 2.5. Operating conditions

The weather on the day of measurement was sunny with occasional clouds. The average outdoor temperature during office hours from 08:00 to 17:00 was 32 °C, with minimum and maximum temperatures of 30 and 33 °C, respectively, and an average relative humidity of 67%. All indoor units operated in cooling mode from 06:30 with setpoint of 26 °C, and both Lossnays operated in circulation mode with the heat recovery function. The average number of occupants during office hours was 642 per cm². To eliminate a highly unsteady state due to the starting duration of the air conditioners and cooling of the office equipment and building envelope, the heat balance evaluation was conducted from 08:00.

### 3. Results

Table 1 demonstrates the results of the energy balances over the day (08:00 to 20:00). The refrigerant (Ref.) cooling capacity error compared to the load is +3.6%, whereas the air errors of the total, sensible, and latent heat compared to the load are -10.4%, -7.2%, and -16.0%, respectively. Eventually, good total heat balances are observed, with errors within 10% between the measured cooling load and refrigerant capacity, and 15% between the measured cooling load and air capacity.

| Items                  | Total            | Sensible       | Latent          |
|------------------------|------------------|----------------|-----------------|
| Load                   | \( \sum L_t = 289 \text{ kWh} \) | \( \sum L_s = 181 \text{ kWh} \) | \( \sum L_1 = 108 \text{ kWh} \) |
| Ref. capacity          | \( \sum Q_r = 300 \text{ kWh} \) | -              | -               |
| Air capacity           | \( \sum Q_{a,t} = 259 \text{ kWh} \) | \( \sum Q_{a,s} = 168 \text{ kWh} \) | \( \sum Q_{a,l} = 91 \text{ kWh} \) |
| ERR compared to load (Ref.) | \( r_r = +3.6\% \) | -              | -               |
| ERR compared to load (Air) | \( r_{a,t} = -10.4\% \) | \( r_{a,s} = -7.2\% \) | \( r_{a,l} = -16.0\% \) |

Figure 5 illustrates the energy balances for every hour. The air capacity error remains negative because it is made up of both the inlet air temperature detection and flow volume measurement. In contrast, the refrigerant capacity error remains positive because it includes not only office cooling, but also heat loss from the outdoor unit due to heat exchange between the outdoor air and the circuit components. The wall and floor loads are small compared to other loads because of the small temperature differences between the outdoor and indoor air. The reason for the high ceiling loads is the solar radiation on the rooftop. The cooling capacities for the interior [OC66] and perimeter [OC72] are almost equal. Over the hour with the highest cooling load (14:00 to 15:00), the total cooling load value is approximately 81 W/m², with a 31-W/m² internal heat gain consisting of lighting, OA plug, and
occupant loads, and a 50-W/m² heat-transfer load consisting of wall, floor, ceiling, ventilation, and infiltration loads. The heat-transfer load is bigger than the heat-gain load.

Figure 5. Cooling energy balance between loads, ref. and air from 08:00 to 20:00.

4. Conclusions & Future Work
We measured the conditions for cooling operation in an actual office space and evaluated the heating balance between the loads and the cooling capacity of the air and refrigerant. It is found that good heat balance is observed, where the errors are within 10% between the measured cooling load and REF capacity, and 15% between the measured cooling load and AIR capacity. This work could be extended to develop a highly accurate simulator for comparison with the simulation results.

Nomenclature

\(A\) area \([\text{m}^2]\)  
\(C_p\) specific heat at constant 
pressure \([\text{kJ/kgK}]\)  
\(G\) mass flow rate \([\text{kg/s}]\)  
\(h\) refrigerant specific enthalpy \([\text{kJ/kg}]\)  
\(I\) air specific enthalpy \([\text{kJ/kg}]\)  
\(L\) cooling load \([\text{kW}]\)  
\(N\) number \([-\text{-}]\)  
\(Q\) cooling capacity \([\text{kW}]\)  
\(q\) heat flux \([\text{W/m}^2]\)  

Relative humidity \([\%]\)  
Residual rate \([\%]\)  
Temperature \(\circ\text{C}\)  
Volume \([\text{m}^3]\)  
Air flow volume \([\text{m}^3/\text{h}]\)  
Electric power consumption \([\text{kW}]\)  
Heat transfer coefficient \([\text{W/m}^2\circ\text{C}]\)  
Density \([\text{kg/m}^3]\)  
Effective rate \([\%]\)

Subscripts

\(a\) air  
\(b\) blind  
\(ceiling\) ceiling  
\(floor\) floor  
\(i\) indoor  
\(ic\) indoor unit  
\(in\) inlet  
\(l\) latent  
\(oa\) outdoor  
\(occ\) occupant  
\(occm\) per occupant  
\(of\) number  
\(oN\) number  
\(out\) outlet  
(\(\) plug  
(\(\) OA plug  
(\(\) refrigerant  
(\(\) room  
(\(\) sensible  
(\(\) total  
(\(\) ventilation  
(\(\) wall  
(\(\) wind

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