Energy-Saving Technology for Multi Split-Type Air-Conditioning Systems for Buildings

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Abstract: Reduction of power consumption in air-conditioning systems is a key factor to solving global warming. This paper presents two energy saving technologies that maintain thermal comfort using multi split-type air-conditioning systems for buildings. The first technology is a control for the evaporation temperature in cooling operations. In this technology, the lower the air-conditioning load is, the higher the evaporation temperature is set, which leads to high energy efficiency operation of the air-conditioning system. The second technology is a compressor frequency control applied to an air-conditioning system made up of multiple outdoor units. The system controls the compressor frequency in the peak efficiency zone by optimizing the number of operating outdoor units corresponding to the air-conditioning load. In addition, this system is equipped with Air-conducting fans for indoor air circulation, which are used to reduce the temperature distribution in the room.

Key Words: Outdoor air-processing unit, evaporation temperature, compressor efficiency

1 INTRODUCTION

In Japan, 30% of total energy consumption in buildings goes to air-conditioning. In office buildings, annual cooling hours are higher than annual heating hours, with many cooling hours at low air-conditioning load. Therefore, it is important to reduce power consumption when the air-conditioning load is low in cooling operations. We have developed two energy-saving technologies for the multi split-type air-conditioning systems for buildings that are being used in many office buildings recently. These two technologies can save energy and maintain thermal comfort at the same time.

The first technology is control of the evaporation temperature in cooling operations. In conventional controls, the evaporation temperature in cooling operations is always controlled so that it is lower than necessary to avoid a deterioration in comfort due to cooling capacity shortages. However, when the evaporation temperature is low, the energy efficiency of the system is also low because latent heat capacity (the amount of removed latent heat load) increases higher than necessary. As a result, power consumption increases. In the control that we have developed, the air-conditioning load is estimated based on the temperature and humidity measured at the outdoor air-processing unit to optimize the operation of the air-conditioning system. The lower the air-conditioning load is, the higher the evaporation temperature is set, which leads to high energy efficiency operation of the air-conditioning system. The excessive latent heat capacity is also reduced using this control. This paper reports on how the developed control functions and on the calculated results of the energy-saving effect.

The second technology is applied to an air-conditioning system composed of multiple outdoor units. Usually, low air-conditioning loads cause low energy efficiency operations or short cycle operations (on – off operation) of the compressor, which increases power consumption (Shinagawa K et al. 2010). This technology allows the system to maintain the compressor frequency in the peak energy efficiency zone by optimizing the number of operating outdoor units corresponding to the air-conditioning load (Arai H, Nobe T 2010). When the air-conditioning load is low and the compressors are running at low frequency, exhibiting low energy efficiency, the number of operating outdoor units is reduced. With this reduction in operating outdoor units, the compressor frequency increases, exhibiting peak energy efficiency. This system is equipped with air-conducting fans for indoor air circulation. These fans are used to reduce the temperature distribution in the room by mixing the air of the zones that are covered by the operating and non-operating outdoor units (Onishi S, Hamada M 2012). This paper reports on the experimental evaluation results of the energy-saving effects and the temperature distribution. In addition, the results of an analysis of the effects on room temperature distribution by the Air-conducting fans placement are described.

2 Control of evaporation temperature

2.1 Air-conditioning system

The air-conditioning system schematic diagram and the outdoor air-processing unit schematic diagram are shown in Figure 1. The system is composed of multiple indoor units, one outdoor air-processing unit, and one outdoor unit. The indoor units and the outdoor air-processing unit are connected to the outdoor unit by the same refrigerant pipes. The outdoor air-processing unit is equipped with an energy recovery core and a direct expansion coil. Outside air (OA) exchanges heat with return air (RA) using the energy recovery core. Then OA is cooled by the direct expansion coil and supplied to theindoors as supply air (SA). RA is exhausted to the outdoors as exhaust air (EA).The outdoor air-processing unit is also equipped with two thermo-hygrometers to measure OA and RA.

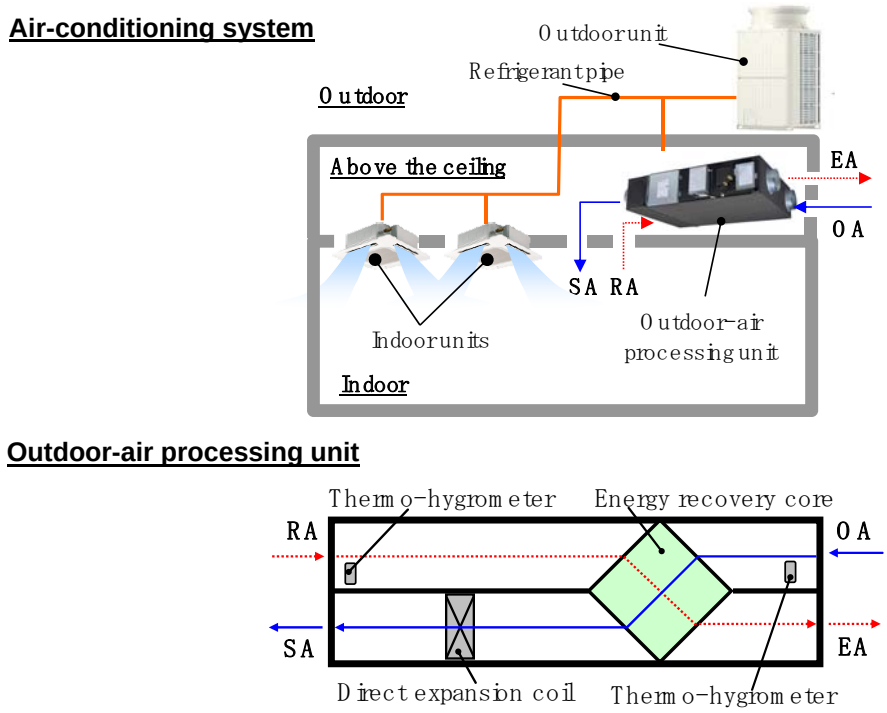


Figure 1: Schematic of the air-conditioning system and the outdoor air-processing unit

2.2 Comparison of conventional control and developed control

2.2.1 Conventional control

Multiple indoor units of various types are connected to one outdoor unit in multi split-type air-conditioning systems for buildings. The air-conditioning load for each indoor unit is different. Therefore the evaporation temperature is always maintained at the temperature at which the maximum load can be removed in order to avoid sensible heat or latent heat shortages in the indoor unit. The relationships between cooling capacity and evaporation temperature and COP and evaporation temperature are shown in Figure 2. The cooling capacity is high at the low evaporation temperature point, which shows low COP. In actual use, operating hours at high air-conditioning loads are short. Therefore, the evaporation temperature and the COP are lower than necessary for many hours during the cooling season.

The room temperature is maintained at the target temperature by adjusting the operating number of the indoor units, where sensible heat load is equal to sensible heat capacity. On the other hand, the latent heat is not controlled at all. When the evaporation temperature is low, the sensible heat factor (SHF) of the cooling capacity of both the outdoor air-processing unit and the indoor units is low, which means that the latent heat capacity exceeds the latent heat load. Thus the total heat capacity exceeds the total heat load.

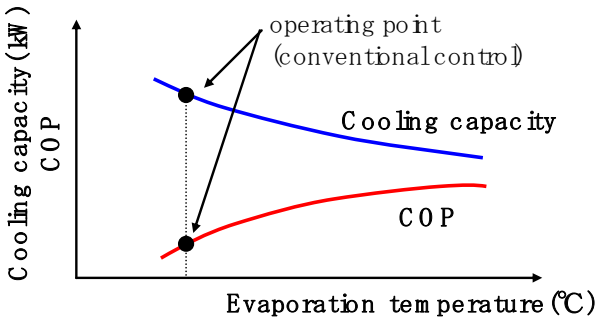


Figure 2: Relationship between cooling capacity and evaporation temperature, COP and evaporation temperature

2.2.2 Developed control

In the developed control, the evaporation temperature is maintained at the highest temperature at which both the sensible heat load and the latent heat load can be removed. Outside air (OA) exchanges heat with return air (RA) using the energy recovery core. Then, OA is cooled by the direct expansion coil and supplied to the indoors as supply air (SA). OA that passes through the energy recovery core has a higher temperature and humidity than RA. Thus the SHF of the capacity of the outdoor air-processing unit is smaller than that of the indoor unit when evaporation temperatures are equal. The reduction of both latent and sensible heat using the energy recovery core is taken into account to determine the evaporation temperature.

The method of determining the evaporation temperature is shown in Figure 3. The horizontal axis indicates the difference between the room temperature and the room set temperature (ΔT). The vertical axis indicates the evaporation temperature. The evaporation temperature (hereafter ET) is determined to be between ETmax and ETmin, depending on ΔT. If ΔT is greater than T1, the ET is set to ETmin. If ΔT is between 0 and T1, the ET is set closer to ETmax as ΔT diminishes. T1 is set to the temperature difference at which thermal comfort is satisfied. ETmax is set to the temperature at which the latent heat load can be

removed, while ETmin is set to that at which the sensible heat capacity equals the sensible heat load. In actual operation, the maximum sensible heat load and the minimum SHF of the load is estimated from the temperature and the humidity sensor in the outdoor air-processing unit. Then ETmin is set to the temperature at which the maximum sensible heat can be removed, while ETmax is the temperature at which the SHF of the capacity equals the minimum SHF of the load. When the actual sensible load is smaller than the maximum sensible load, ΔT is smaller than T1, which leads to a rise in the ET. On the other hand, the ET cannot exceed ETmax, so a latent heat capacity shortage is prevented.

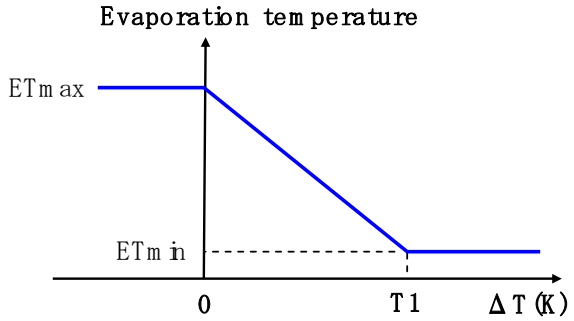


Figure 3: Evaporation temperature control

2.3 Calculation of the energy-saving effect

2.3.1 Air-conditioning space and air-conditioning system

The calculation model for the air-conditioning system is shown in Figure 4.The floor area of the air-conditioning space is 364m^2. The system has two outdoor units, and each outdoor unit is connected to four indoor units and one outdoor air-processing unit by the same refrigerant pipes. The specifications for the equipment are shown in Table 1.

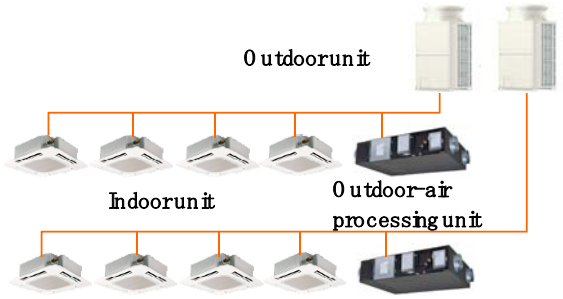


Figure 4: Schematic of the air-conditioning system

Table 1: Specifications of air-conditioning systems

	Specification	Unit number
Indoor unit	・Cooling capacity = 7.1 kW ・Power consumption = 50 W ・Air flow rate = 21 m^3/min	8
Outdoor unit	・Cooling capacity = 28 kW ・Power consumption = 8.16 kW	2
Outdoor air-processing unit	・Air flow rate = 1000 m^3/hour ・Power consumption = 620 W	2

2.3.2 Load and calculation conditions

When calculating the air-conditioning load, human heat generation, internal heat generation, ventilation, natural ventilation, and heat transfer from walls and windows are all taken into account. The air-conditioning load is calculated using the set value shown in Table 2. The indoor air is 26°C/60%, and the outdoor air is determined using the outdoor air temperature and humidity that are stipulated in JIS B 8616 (Tokyo). The heat transfer from the wall is calculated using the equivalent temperature difference (ETD).

Table 2: Load condition

		Unit	Value
Human heat generation	Sensible	W/person	69
	Latent	W/person	53
Occupant density		m^2/person	5
Internal heat generation		W/m^2	60
Coefficient of heat transmission	Wall	$W/(m^2 \cdot K)$	3.7
	Window	$W/(m^2 \cdot K)$	6.4
Ventilation air flow		m^3/hour	2000
Natural ventilation		times/hour	0.5

When the outdoor air temperature is 35°C during the summer season, the air-conditioning load shows the maximum value, which is 48kW. The air-conditioning load, which the system has to remove, is reduced to 43kW using the energy recovery core in the outdoor air-processing unit. The load factor (air-conditioning load / rated capacity) of the outdoor units (56kW) is 77%. Therefore, it is possible to raise the evaporation temperature even when the outdoor temperature is 35°C.

Throughout the calculation of energy saving effect, it is assumed that the ET is a constant 0°C in the conventional control. For the developed control, the ET is set to the highest temperature at which both the sensible heat capacity and latent heat capacity shortage are prevented. However, the upper limit of the ET is set to 14°C. Further, it is supposed that the energy recovery core is not used under conditions in which the outdoor air temperature is lower than the room air temperature.

2.4 Calculated results

The air-conditioning load and cooling capacity when the outdoor air temperature is 30°C are shown in Figure 5. The sensible heat capacity (34kW) is equal to the sensible heat load (34kW) in both the conventional control and developed control. On the other hand, the latent heat capacity (11kW) is 5.5kW larger than the latent heat load (5.5kW) in the conventional control. However, in the developed control the latent heat capacity is 5.5kW, which equals the latent heat load.

Under these conditions, the ET is about 13°C at an outdoor temperature of 35°C. Furthermore, it rises as the outdoor temperature lowers. The ET reaches the maximum value (14°C) when the outdoor air temperature is 30°C.

Power consumption for each season (cooling, heating) and annually is shown in Figure 6. Power consumption for the heating season was calculated, as well as that for the cooling season, using the duration of the outdoor air temperature, which is stipulated in JIS B 8616 (Tokyo). Since the conventional control continues to be used during the heating season, energy consumption of the developed control is equal to that of the conventional control. The savings ratios for seasonal power consumption for the cooling season and annually are 32% and 26%, respectively.

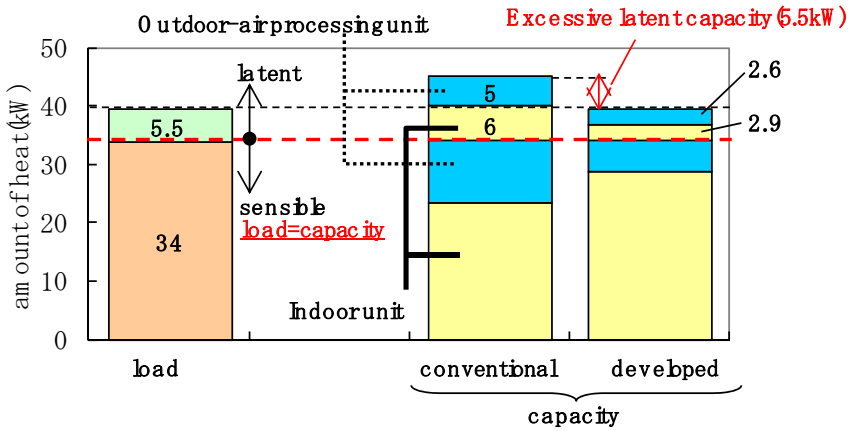


Figure 5: Load and cooling capacity (outdoor air temperature=30°C)

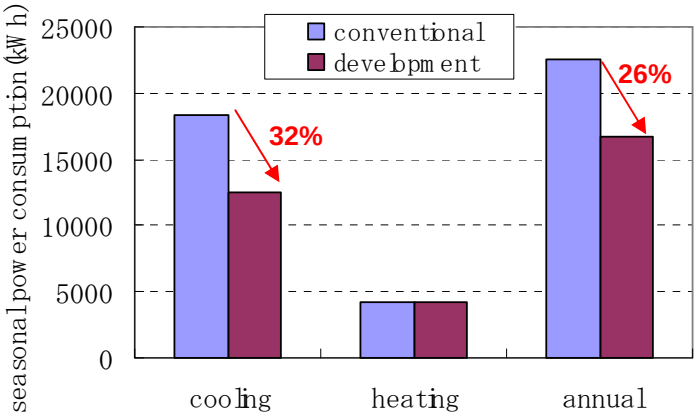


Figure 6: Seasonal power consumption

3 Optimization of operating number of outdoor units

3.1 Air-conditioning system

An air-conditioning system schematic diagram is shown in Figure 7. The system is composed of multiple outdoor units. Multiple indoor units are connected to each outdoor unit by the same refrigerant pipes.

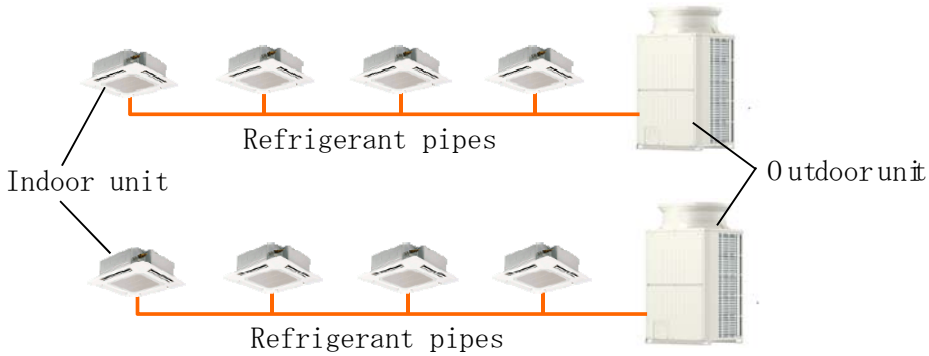


Figure 7: Schematic of the air-conditioning system

3.2 Comparison of conventional control and developed control

A comparison of the conventional and developed controls is shown in Figure 8. In the conventional control, both compressors of the two air conditioners operate at low frequency F1 when the air-conditioning load is low. In the developed control, the compressor of the air conditioner operates at frequency F2, which is higher than F1, due to the reduction in the number of operating outdoor units. The relationship between compressor frequency and COP is shown in Figure 9. The COP at compressor frequency F2 is higher than the COP at compressor frequency F1. Thus the power consumption of the developed control is lower than that of the conventional control. Since the characteristic of COP is a parabola, the lower the compressor frequency becomes, the higher the improvement rate of COP becomes. In addition, Air-conducting fans are used to reduce temperature distribution in the room by mixing the air in the zones that are covered by operating and non-operating outdoor units.

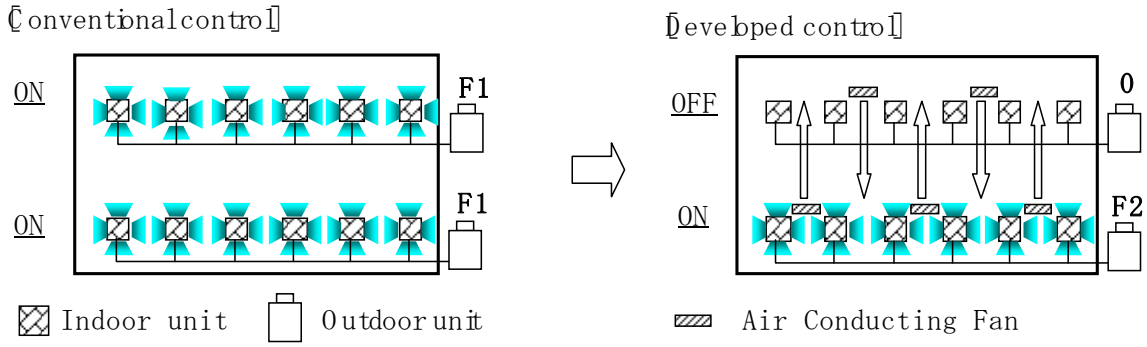


Figure 8: Comparison of conventional control and developed control

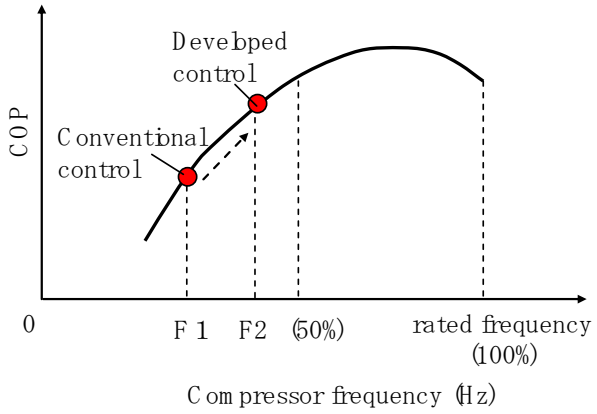


Figure 9: Relationship between compressor frequency and COP

3.3 Experimental evaluation

3.3.1 Air-conditioning space and system

An overhead view of the air-conditioning space is shown in Figure 10, and the specifications for the equipment are provided in Table 3. The floor area of the air-conditioning space is 364m^2 and has two air-conditioners, which are composed of four indoor units (four-way cassette type) and one outdoor unit. One air conditioner is placed on the perimeter side and the other on the interior side of this space. In addition, seven Air-conducting fans, which are shown in Figure 11, are placed on the ceiling in this area. The height of the ceiling is 2.7m. Two Air-conducting fans are placed on the interior side, and move the air from the interior

side to the perimeter side. The other five Air-conducting fans are installed on the perimeter side and move the air from the perimeter side to the interior side. As shown in Figure 10, fourteen temperature and humidity sensors are installed to measure the distribution of the room temperature.

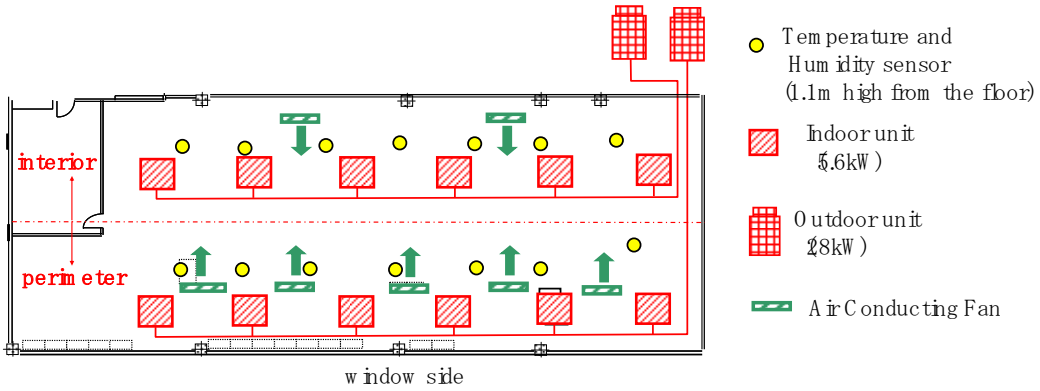


Figure 10: Overhead view of air-conditioning space



Figure 11: Air-conducting fan

Table 3: Specifications of air-conditioning systems

	Specification	Unit number
Indoor unit	・Cooling capacity = 5.6 kW ・Power consumption = 40 W ・Air flow rate = 16 m^3/min	12
Outdoor unit	・Cooling capacity = 28 kW ・Power consumption = 8.16 kW	2
Air-conducting fan	・Air flow rate = 740 m^3/hour (Air velocity = 4.2 m/s) ・Power consumption = 30 W	7

3.3.2 Evaluation methods and conditions

The evaluation was performed under the following four conditions. The room temperature was fixed at 26°C. Condition 1 is the conventional control. As shown in Figure 12, the two air conditioners are operated, and the Air-conducting fans are stopped. Condition 2 is the developed control. In condition 2, the number of operating outdoor units was reduced, and only the air conditioner on the perimeter side was operated. Two Air-conducting fans on the interior side and three Air-conducting fans on the perimeter side were operated. In condition 3 and condition 4, only the operation of the Air-conducting fans was changed compared to condition 2. In condition 3, the air conditioner on the perimeter side was operated in the same way as in condition 2, and only five Air-conducting fans on the perimeter side were operated. In condition 4, the air conditioner on the perimeter side was operated in the same way as in condition 2, and the Air-conducting fans were stopped. Under each condition, the room temperature and humidity, outdoor air temperature and humidity, power consumption of

the air-conditioners, compressor frequency, refrigerant temperature, and refrigerant pressure were measured at one-minute intervals.

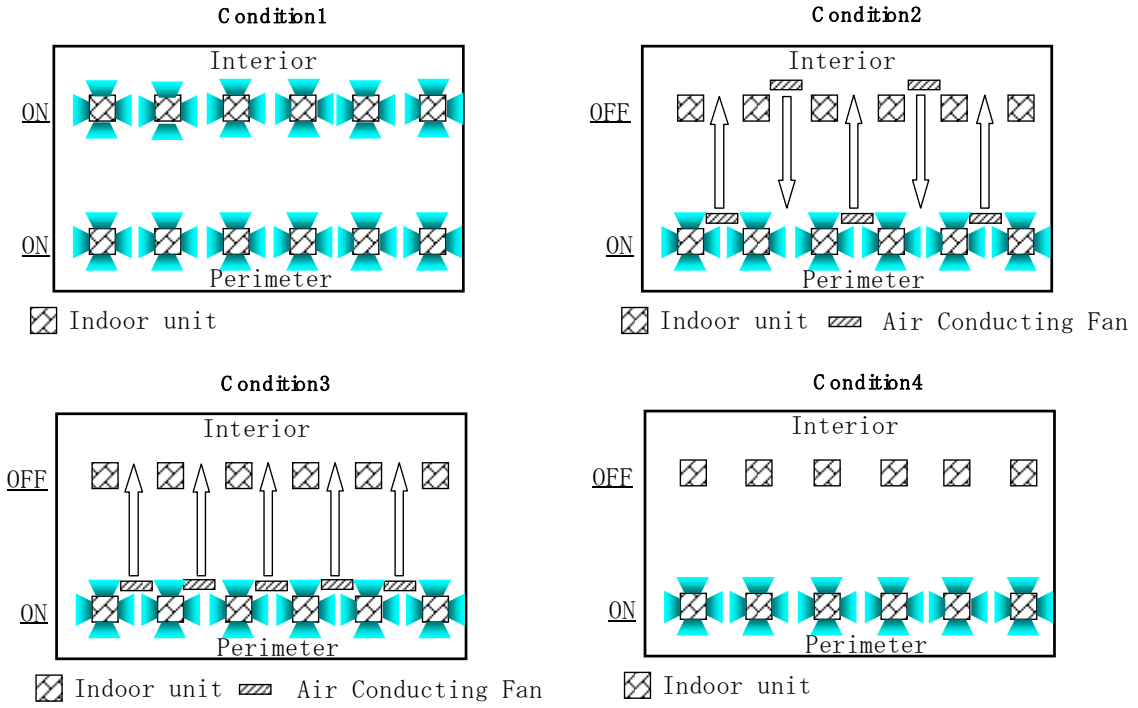


Figure 12: Operating mode (Condition1-Condition4)

3.3.3 Results of the experiment

1) Energy-saving effect

The relationships between the cooling load factor (capacity / rated capacity of the outdoor units) and power consumption are shown in Figure 13 with respect to condition 1 and condition 2. When the cooling load factor is 100%, the cooling capacity is 56kW. In Figure 13, the broken line indicates the approximate line of condition 1, and the solid line indicates that of condition 2.

Under each condition, the measurement duration was 8 hours per day (11:00 ~ 19:00), over a period of three days. In condition 1, power consumption is the sum of the outdoor units and the indoor units. In condition 2, power consumption is the sum of the outdoor unit, indoor units, and the Air-conducting fans. The cooling capacity was calculated using the refrigerant flow rate, enthalpy at the inlet of the evaporator, and enthalpy at the outlet of the evaporator. When the cooling load factor is 50% (cooling capacity = 28kW), power consumption under condition 1 and condition 2 is almost the same. The cooling load factor of 50% is the upper limit for reducing the number of operating outdoor units. When the cooling load factor is 21% (cooling capacity = 12kW), power consumption for condition 2 is 12% lower than that for condition 1. As the cooling load factor decreases, the energy-saving effect increases.

Seasonal power consumption was calculated using the experimental results for condition 1 and condition 2. The outdoor air condition was determined from the duration of outdoor air temperature stipulated in JIS B 8616 (Tokyo). As a result, condition 2 decreases cooling season power consumption by 7% against condition 1. Heating season power consumption was also calculated using the experimental results, as with the cooling season power

consumption. The room temperature was set at 24°C, and the outdoor air conditions were chosen from JIS B 8616 (Tokyo). The result showed that condition 2 decreased heating season power consumption by 21% against condition 1. The annual power consumption was decreased by 12%. The relationship between the average load factor and the energy-saving effect is shown in Figure 14. The average load factor is obtained by dividing the seasonal average load by the rated load. As the average load diminishes, the energy-saving effect increases during both the cooling season and heating season.

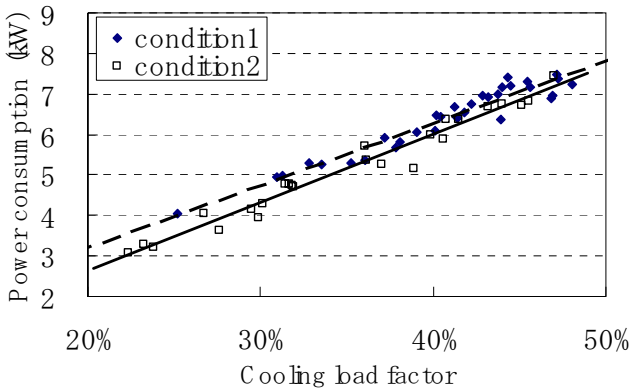


Figure 13: Relationship between cooling load factor and power consumption

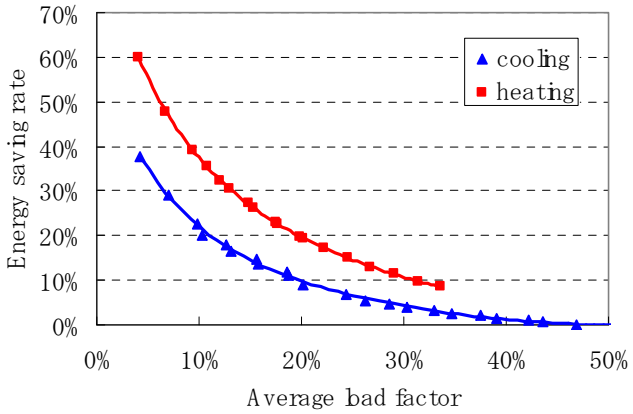


Figure 14: Relationship between average load factor and energy-saving rate

2) Temperature distribution improvement

The room temperatures, the interior side room temperatures, the perimeter side room temperatures and the average temperature differences between the interior side and the perimeter side from 11:00 to 12:00 are shown in Table 4. The room temperature is an average temperature of 14 measurement points (1.1 m above the floor). The temperature of the interior side is an average value of seven measurement points (1.1 m above the floor of the interior side), and the temperature of the perimeter side is an average temperature seven measurement points (1.1 m above the floor of the interior side).

In all conditions, the room temperatures are within 0.5 °C of the set room temperature. However, the temperature differences vary depending on the conditions. In condition 4, in which the Air-conducting fans are stopped, the interior side room temperature rose to 27°C, the largest temperature difference (1.0 K). In condition 3, the temperature difference is smaller than that of condition 4, which means that the temperature of the room is more uniform. In condition 2, the temperature difference is smaller than that of condition 3, which indicates that the room temperature distribution has narrowed.

Table 4: Specifications of air-conditioning systems

Condition	Temperature (°C)			Temperature difference (K)
	Average	Interior	Perimeter	
1	25.9	25.9	26.0	-0.1
2	26.3	26.5	26.1	0.4
3	26.3	26.7	25.9	0.8
4	26.5	27.0	26.0	1.0

Fluctuations in the temperature differences between the interior side and perimeter side are shown in Figure 15. This shows that the temperature distribution of condition 2 is smaller than that of both condition 3 and condition 4.

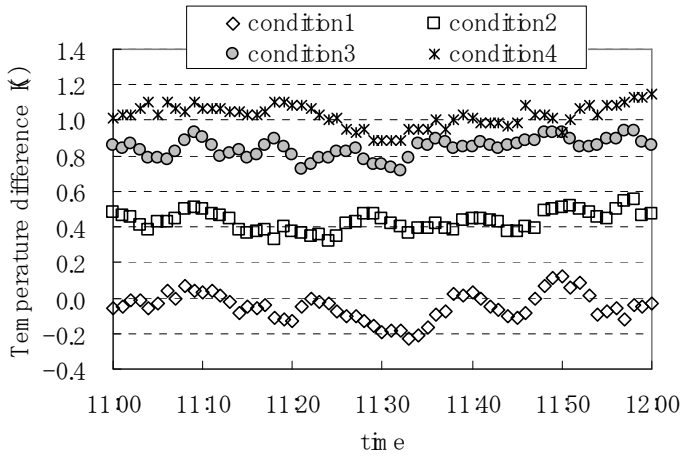


Figure 15: Temperature differences between interior side and perimeter side

3.4 Analytical evaluation

In order to understand the effect of the Air-conducting fans, which decreased the room temperature difference, an analysis that modeled the placement of the Air-conducting fans shown in Figure 10 was conducted. For the analysis, the CFD using the k-ε model is adopted. The air-conditioning space is W×D×H=28×13×2.7 (m), the internal load is 23 W/m^2, and the outdoor air temperature is raised at a constant rate from 30°C to 32°C over 1 hour. Furthermore, solar radiation from 11:00 to 12:00 on August 8 at Yokohama, Kanagawa, is taken into consideration, while the transient analysis. The analysis was conducted for condition 2 and condition 3, which are shown in Figure 12. The analysis results for temperature distribution and velocity distribution are shown in Figure 16. These are the results of measurements taken in an area 1.1 m above the floor at 11:30 a.m.

In condition 2, the average temperature is 26.1°C, and the temperature difference between the interior side and the perimeter side is 0.2K. In condition 3, the average temperature is 26.4°C, and the temperature difference between the interior side and the perimeter side is 0.4K. The temperature difference for condition 2 is smaller than that for condition 3, which indicates agreement with the experimental result. Figure 16 shows that the velocity of condition 2 is faster than that of condition 3, where the swirl flow is generated in condition 2. From this result, we can conclude that the air mixing effect is higher when Air-conducting fans are installed in a zigzag position (condition 2), which results in a small temperature difference.

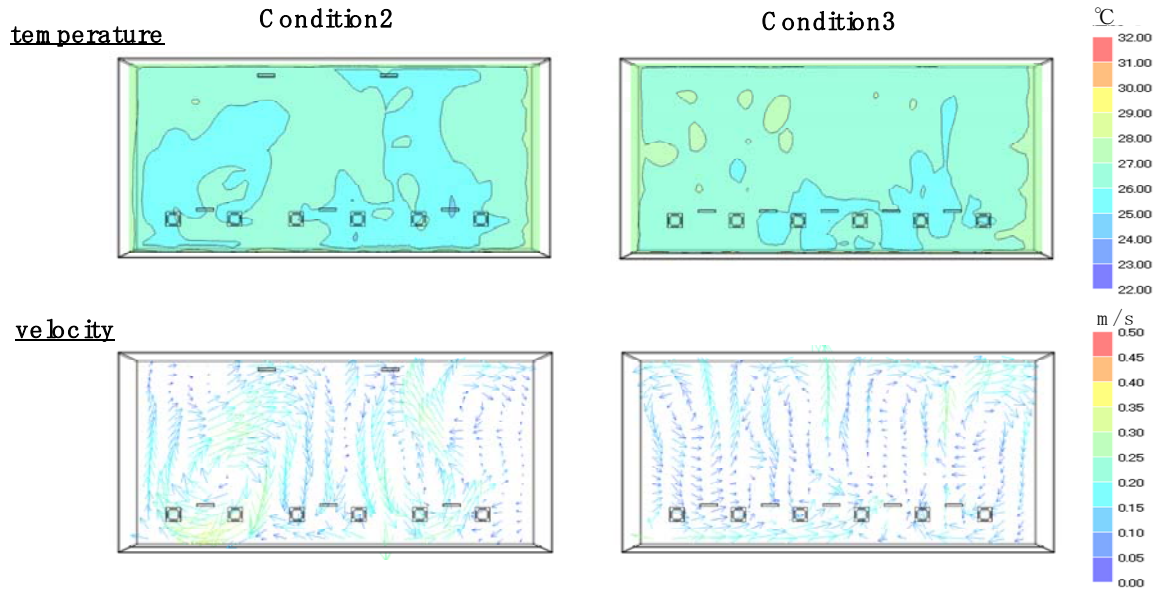


Figure 16: Room temperature distribution and velocity distribution (11:30 a.m.)

4 Conclusions

Two energy saving technologies for multi split-type air-conditioning system for buildings were developed. The first technology optimizes ET, depending on the temperature and the humidity that is measured at outdoor air-processing units, and this achieves both energy savings and thermal comfort. Power consumption is reduced by improving the COP and decreasing the excessive latent capacity. The saving rates for seasonal power consumption (cooling and annually) are estimated to be 32% and 26%, respectively. The second technology is optimization of the operating outdoor unit number applied to air-conditioning systems composed of multiple outdoor units in order to improve the compressor energy efficiency. In addition, Air-conducting fans are used to reduce temperature distribution in the room. Energy-saving effects during the cooling season and heating season were demonstrated, and the saving rates of the seasonal power consumption of cooling and heating were confirmed to be 7% and 21%, respectively. The annual power consumption saving rate was 12%. The maximization of temperature distribution in the room caused by Air-conducting fans was confirmed. The reduction effect of the temperature distribution varies depending on the placement of the Air-conducting fans, with a zigzag placement being much more effective than a one-way placement. In this paper, the control method of the machine under static condition was described. Because the evaluations were performed only under the static indoor temperature, we will evaluate the effect of the new control method under transient condition in the future as well.

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