EVALUATION OF ANNUAL ENERGY CONSUMPTION OF MULTI-SPLIT TYPE EHP AIR-CONDITIONER FOR BUILDINGS

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Abstract: Part-load performance tests were made on recently developed multi-split type electric heat pump air-conditioners for buildings with rated capacities of 45 kW (cooling) and 50 kW (heating). For the cooling performance test, the outdoor air temperature was changed from 20 ºC to 45 ºC, and the indoor thermal load was changed from 12.5% to 100% of the rated cooling capacity of the air conditioner. Cooling COPs were measured at 36 test points. For the heating performance test, COPs were measured at 31 test points. Based on the measured COPs and several thermal load models for business buildings, the annual energy consumption and annual average COP of A/C were evaluated. Moreover, we proposed a new method to predict the annual average COP. In this method, variations of COP with the outdoor air temperature and the indoor thermal load were modeled based on COPs measured at eight test points. The error of the annual average COP predicted by this method was smaller than 3% for thermal load models of a home appliance mass merchandiser but increased for a food supermarket.

Key Words: multi-split type air conditioner, part-load performance, annual energy consumption, annual average COP, business-related building

1 INTRODUCTION

In a business-related building, the energy used for air conditioning amounts to 30 % - 40 % of its total energy consumption. Development of air conditioners with lower energy consumption is therefore crucial for energy saving in buildings. Over the year, air conditioners are mostly operated with indoor thermal loads lower than their rated capacities; this situation is called part-load operation. In Japan, the calculation method of the annual energy consumptions of air conditioners that took into account their part-load performances was specified in Japanese Industrial Standards (JIS B 8616: 2006) for EHP-type packaged air conditioners with rated cooling capacities less than 28 kW. For an accurate prediction of the annual energy consumption of an air conditioner, we need to know its detailed part-load performance. In general, however, the COP of an air conditioner operating under part load changes, depending on the indoor thermal load and the outdoor air temperature, in quite a complex manner. In particular, the detailed part-load performance of a multi-split type air conditioner for buildings with relatively large capacities has not been clarified yet. Since the use of this type of air conditioner is now spreading to many kinds of business-relating buildings, accurate knowledge of its detailed part-load performance would be helpful in achieving effective energy savings in those buildings. With these points as background, we have been carrying out the part-load performance tests of multi-split type air conditioners for buildings with rated cooling capacities of 45 kW - 56 kW by using the air-enthalpy-method testing apparatus that can control the indoor thermal load and the outdoor air temperature arbitrarily. In addition, based on the results of these tests, we have been examining the applicability of JIS B 8616: 2006 to the prediction of the annual energy consumption of air conditioners with larger capacities (Watanabe et al., 2008, 2009).

Recently new multi-split type EHP air-conditioners for buildings were developed in Japan, and higher energy-saving effects were stated in them. In this paper, we present COPs measured in the part-load performance tests of those new multi-split type EHP air conditioners for buildings with rated capacities of 45 kW (cooling) and 50 kW (heating). For the cooling performance test, the outdoor air temperature was changed from 20 ºC to 45 ºC, and the indoor thermal load was changed from 12.5% to 100% of the rated cooling capacity of the air conditioner. Cooling COPs were measured at 36 test points. For the heating performance test, COPs were measured at 31 test points. By combining these measured COPs and indoor thermal load models of business-related buildings, we calculated the annual average COPs of those air conditioners. Then, their energy-saving performances were evaluated by comparing their annual average COPs with those obtained for the conventional air conditioners made in 2005 - 2008. Moreover, we proposed a new simple method to predict the annual average COPs of the air conditioners. In this method, the variation of COP with the outdoor air temperature and the indoor thermal load was reproduced based on COPs measured at eight test points in the part-load performance test. It was confirmed that the error included in the annual average COPs predicted by the proposed method was within 3% for the thermal load model of a home appliance mass merchandiser but somewhat increased for a food supermarket.

2 TEST CONDITIONS AND METHOD

2.1 Specifications of Tested Air Conditioners

The part-load performance tests were conducted in the test facility of Chubu Electric Power Inc., in which the performance of an air conditioner with a rated cooling capacity up to 168 kW and a rated heating capacity up to 200 kW can be measured by the air-enthalpy method (Nakayama et al., 2011). Table 1 shows specifications (nominal values listed in brochures) of the tested air conditioners. We tested five kinds of multi-split type EHP air conditioners for buildings manufactured by different manufacturers in Japan. A1, B1 and C1 are conventional air conditioners marketed in 2005 - 2008 by different manufacturers. The rated cooling and heating capacities are 56 kW and 63 kW, respectively. A2 and B2 are new air conditioners with high energy-saving effects marketed in 2011 by the same manufacturers as A1 and B1, respectively. The rated capacities are 45 kW (cooling) and 50 kW (heating). In all the tested air conditioners, an inverter was used to control the revolutions of the compressors, and one outdoor unit was connected with four indoor units of ceiling-mounted cassette type.

Table 1: Specifications of Tested Air Conditioners

Name of air conditioners	A1	A2	B1	B2	C1
Type of air conditioners	conventional	new	conventional	new	conventional
Rated cooling capacity	56 kW	45 kW	56 kW	45 kW	56 kW
Rated heating capacity	63 kW	50 kW	63 kW	50 kW	63 kW
Marketed year	2005	2011	2008	2011	2005

2.2 Conditions of Part-load Performance Tests

The conditions of the part-load performance tests made for the new type air conditioners A2 and B2 are shown in Table 2. In the cooling performance test, the dry-bulb temperature of outdoor air t_j was changes from 20 °C to 45 °C at 5 °C intervals, and the indoor thermal load BL_c was changed from 12.5 % to 100 % of the rated cooling capacity of the tested air conditioner. The sensible heat fraction (SHF) was set at 0.85. The temperature in the indoor test room was controlled at 27 °C by the tested air conditioner itself to simulate its actual operating situation. Totally cooling COPs were measured at 36 test points. In the heating performance test, the dry-bulb temperature / wet-bulb temperature of outdoor air were changed from -7 °C / -8 °C to 12 °C / 11 °C, and the indoor thermal load BL_h was changed from 12.5 % to 100 % of the rated heating capacity of the tested air conditioner. The temperature in the indoor test room was controlled at 20 °C by the tested air conditioner. Heating COPs were measured at 31 test points. For the conventional air conditioners A1, B1 and C1, t_j was changed from 20 °C to 35 °C and BL_c was changed from 25 % to 100 % of the rated cooling capacity in the part-load cooling performance test. Cooling COPs were measured at 16 test points. In the heating performance test, COPs of the air conditioner were measured at 12 test points (Watanabe et al., 2008, 2009).

The values of thermal loads in these part-load performance testes were determined based on the measured values of the rated cooling and heating capacities of each air conditioner, not on the nominal rated values described in the brochure. Those rated capacities were measured by the method prescribed in JIS B 8615-1:1999 under the condition shown in Table 3. In these tests, the performances of the air conditioner were measured keeping the compressor revolutions at rated values and the indoor air temperature was controlled by the air conditioner installed in the test apparatus not by the tested air conditioner.

In the part-load performance tests, we measured the pressure and temperature of the refrigerant, revolutions of the compressors, etc., as well at every 10 seconds. COPs of the air conditioner were calculated based on the data obtained under the steady-state operation. In low thermal load conditions, however, intermittent operations of the compressors were observed (Hirota et al., 2007). In this case, we confirmed that a periodicity appeared in the operations of the compressor and COP was calculated based on the average in one or more cycles. COPs shown in this paper are defined based on the electricity consumption of both the indoor and outdoor units, namely electricity consumed by compressors, indoor and outdoor fans and auxiliaries. In the heating performance test, the electricity consumption in the defrosting mode was included in calculating COPs.

Table 2: Conditions of Part-load Performance Tests for A2 and B2

Part-load cooling performance test		Part-load heating performance test	
Outdoor air temperature DBT/WBT [°C]	Cooling load ratio	Outdoor air temperature DBT/WBT [°C]	Heating load ratio
20 / -	12.5, 25, 50, 65, 75, 85, 100	-7 / -8	12.5, 65, 75, 85, 100
25 / -	12.5, 25, 50, 65, 75, 85, 100	-3 / -4	25, 50, 65, 75, 85, 100
30 / -	25, 50, 65, 75, 85, 100	2 / 1	25, 50, 65, 75, 85, 100
35 / -	25, 50, 65, 75, 85, 100	7 / 6	12.5, 25, 50, 65, 75, 85, 100
40 / -	12.5, 25, 50, 65, 75, 85, 100	12 / 11	12.5, 25, 50, 65, 75, 85, 100
45 / -	50, 75, 100		

Table 3: Conditions of Rated Capacity Tests (JIS B 8615-1:1999)

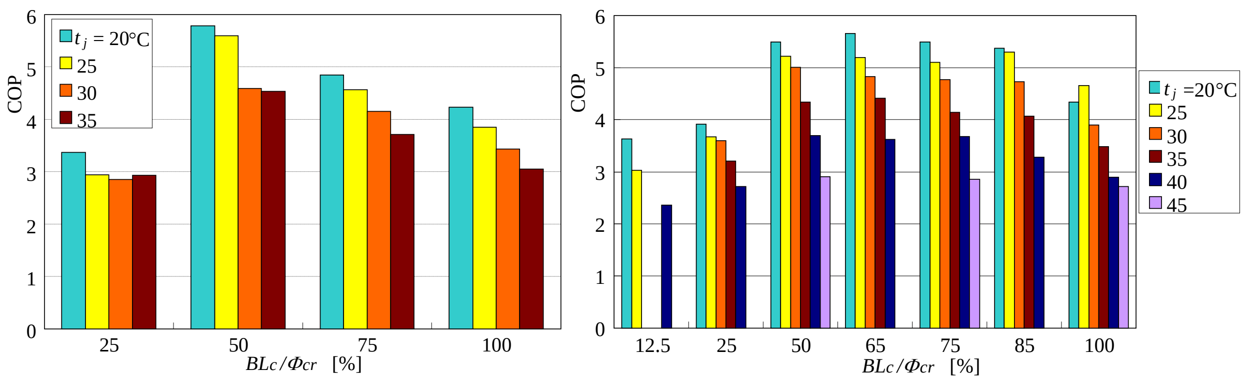
Type of Test	Outdoor Air		Indoor Air	
	DBT	WBT	DBT	WBT
Cooling	35 °C	-	27 °C	19 °C
Heating	7 °C	6 °C	20 °C	15 °C (max)

3 RESULTS OF PART-LOAD PERFORMANCE TESTS

The results of COPs measured in the part-load performance tests of A2, one of the new type air conditioners with high energy-saving effects, are compared with those of the conventional machine A1. Figure 1 shows cooling COPs of A1 and A2. The abscissa shows the thermal load ratio BL_c/Φ_{cr}, where Φ_{cr} denote the measured value of the rated cooling capacity of the air conditioner. The parameter is t_j. In both air conditioners, at a constant cooling load ratio, COP decreases as t_j rises. This decrease of COP is caused by the pressure rise of the refrigerant in the condenser with the increase of t_j. Under constant t_j, COP of A1 increases gradually as the thermal load is decreased from 100 % and attains the maximum at BL_c/Φ_{cr} = 50 %, but it decreases to the minimum at BL_c/Φ_{cr} = 25 %. This deterioration of COP at low thermal load is caused by the intermittent on-off operations of the compressors (Hirota et al., 2007). In the new type air conditioner A2, high values of COP are maintained over a wide range of BL_c/Φ_{cr}= 85 % - 50 %. COP decreases as BL_c/Φ_{cr} is further decreased, but COP of A2 is higher than that of A1 at BL_c/Φ_{cr} = 25 %.

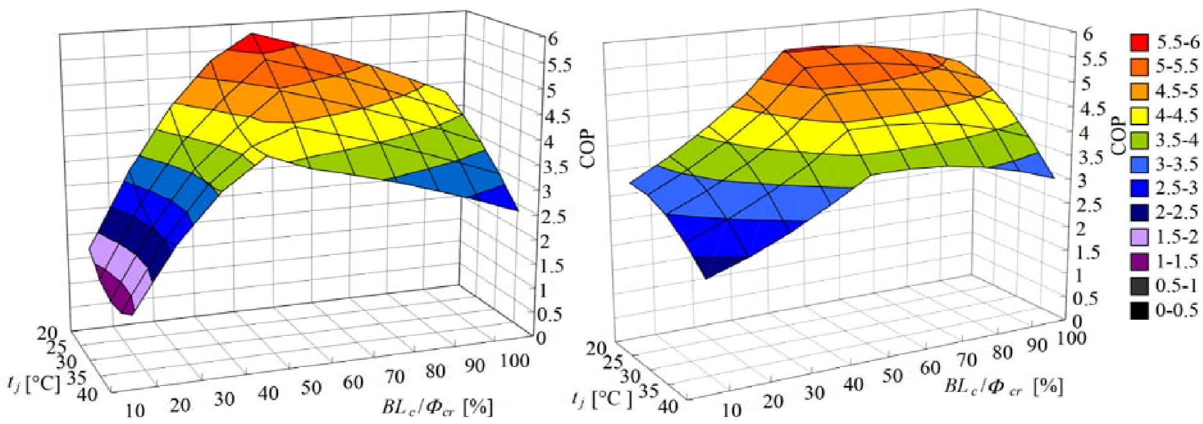
Based on these COP data obtained in the part-load cooling performance tests, we can plot the cooling COP surfaces of these air conditioners as shown in Fig. 2 by expressing COPs as a function of t_j and BL_c/Φ_{cr}. These COP surfaces are quite helpful for instinctively grasping the dependence of COP on the outdoor air temperature and the indoor thermal load. We can see that the new air conditioner can maintain high COP over quite a wider range of BL_c/Φ_{cr}, and that deterioration of COP in lower cooling load is suppressed in comparison with the conventional machine A1.

Heating COPs measured in the part-load heating performance tests of A1 and A2 are shown in Fig. 3, and their heating COP surfaces are shown in Fig. 4. Similar to the cooling COPs,



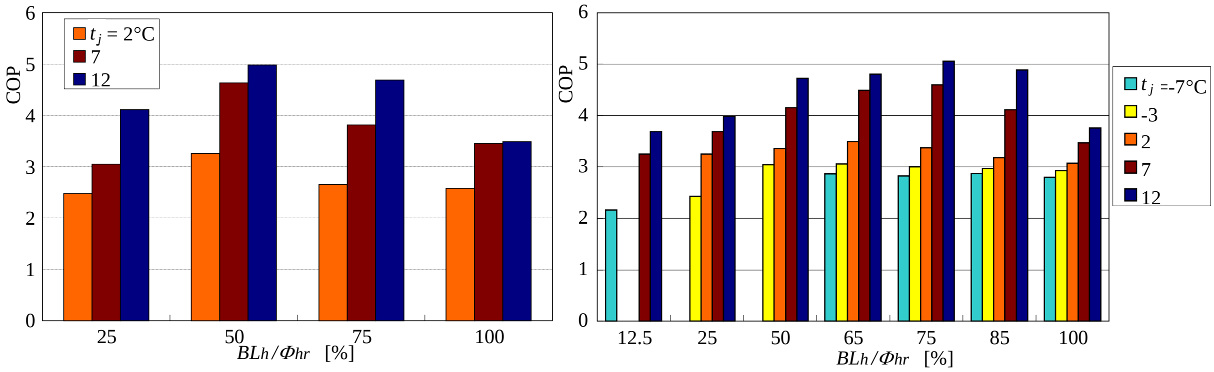
(a) A1 (conventional machine) (b) A2 (new machine)

Figure 1: COPs in Cooling Mode Measured in Part-load Performance Test



(a) A1 (conventional machine) (b) A2 (new machine)

Figure 2: COP Surfaces in Cooling Mode



(a) A1 (conventional machine) **(b) A2 (new machine)**

Figure 3: COPs in Heating Mode Measured in Part-load Performance Test

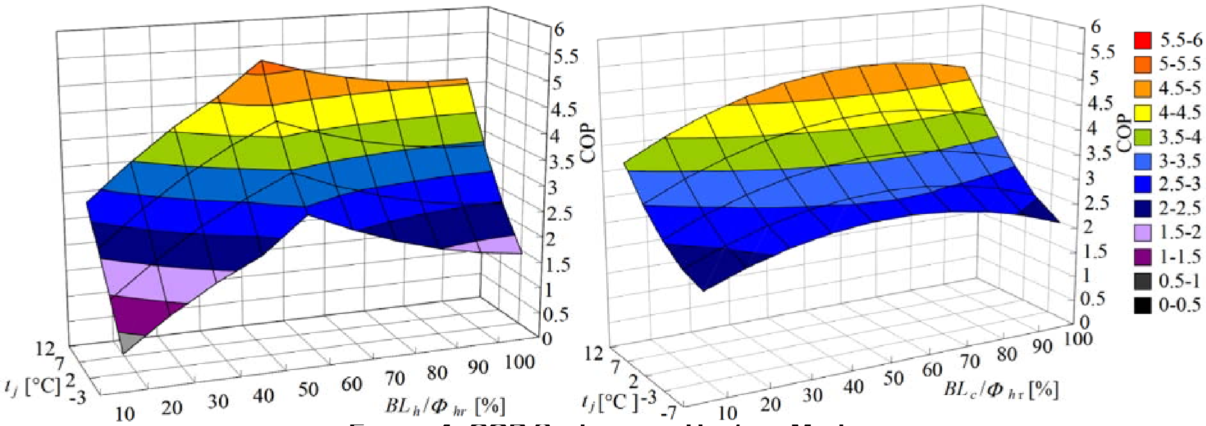


Figure 4: COP Surfaces in Heating Mode

relatively high COP is maintained over a wider range of BL_h/Φ_{hr} and the deterioration of COP in low BL_h/Φ_{hr} is suppressed in the new type air conditioner A2.

Next, the cooling COP surfaces obtained in the part-load performance tests of B1 and B2 are shown in Fig. 5. In the conventional machine B1, the cooling COP surface shows quite flat characteristics in comparison with that of A1. In the new machine B2, COPs are increased in a high cooling load region of BL_c/Φ_{cr} > 50 % and the dependence of COP on t_j is increased than that of B1. The minimum COP of B2 in a very low cooling load shows almost the same value as that of B1. The heating COP surfaces of B1 and B2 are shown in Fig. 6. Similar to the cooling COPs, the heating COPs of B2 are remarkably increased in a high heating load region, while COPs in a low load region are not improved so much in comparison with B1.

4 EVALUATION OF ANNUAL ENERGY CONSUMPTIONS AND ANNUAL AVERAGE COPS OF TESTED AIR CONDITIONERS

4.1 Indoor Thermal Load Model

As shown above, in the new type air conditioners, COPs are partly improved in comparison with the conventional machines and the improved portions on the COP surfaces differ depending on the manufacturers. In order to evaluate quantitatively the substantial energy-saving effects of the new air conditioners, we calculated the annual energy consumption and annual average COP of each air conditioner by combining the COP surfaces, indoor thermal load models of buildings and time (hours) of appearance of t_j in a year. We adopted two contrastive indoor thermal load models, home appliance mass merchandiser and food supermarket, which were measured by the authors in Nagoya city located in central Japan.

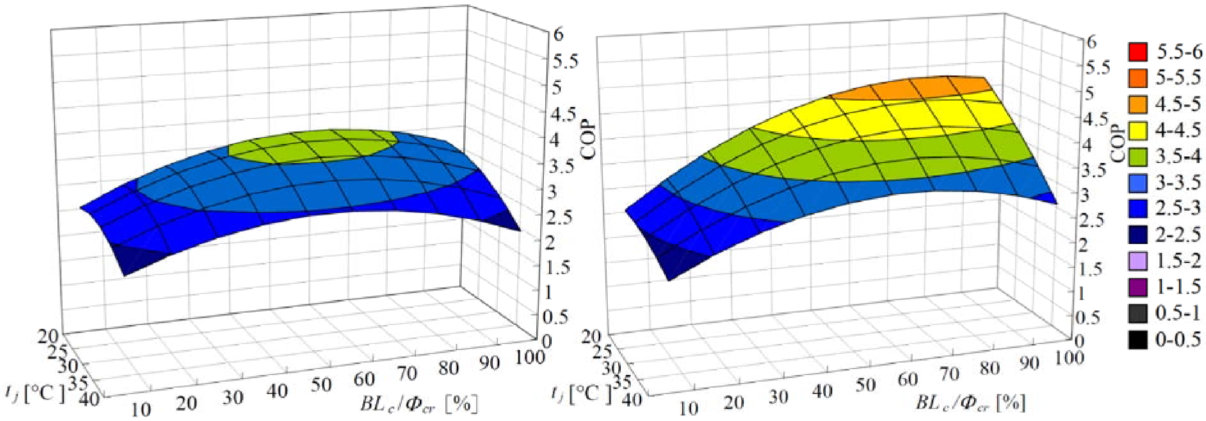


Figure 5: COP Surfaces in Cooling Mode

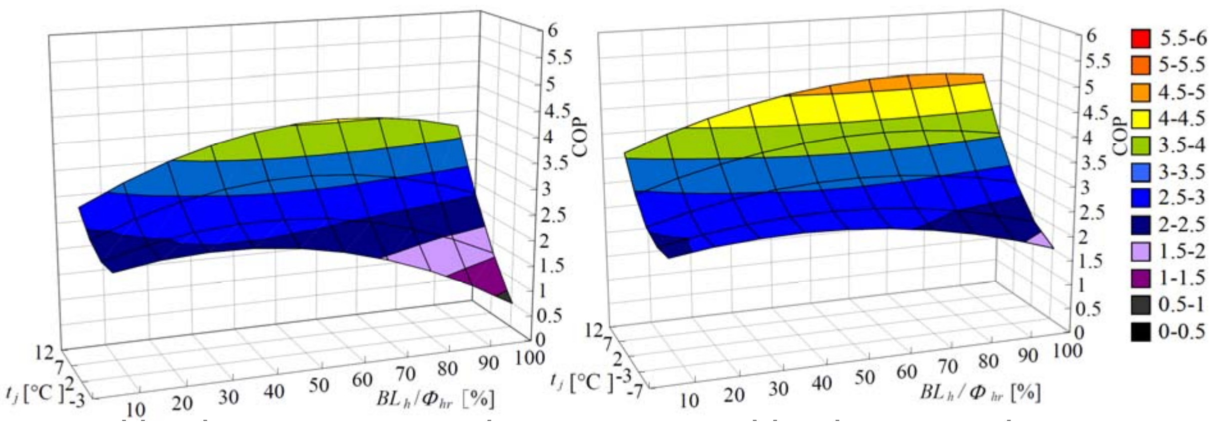
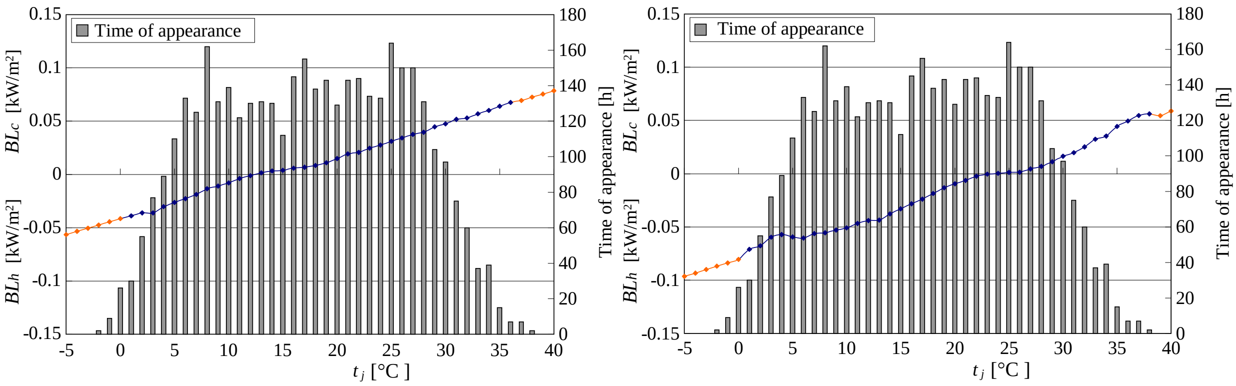


Figure 6: COP Surfaces in Heating Mode

The thermal loads in those stores were measured assuming that they were in equilibrium with the sum of capacities of air conditioners operating in them. This assumption would be reasonable because the floor spaces of those stores were so large (about 2000 m^2) that the air conditioners could follow the relatively slow temporal variations of the indoor thermal loads. As advance preparation to measure the thermal loads in the stores, we conducted the part-load performance tests of the air conditioners that were the same types as those installed in the stores. Based on measured data, the cooling / heating capacities of the air conditioners were formulated as functions of dry-bulb temperature of outdoor air t_j and energy consumption ratio (energy consumption of air conditioner under the part-load operation / that under the rated operation). Then, we measured t_j around the stores and the energy consumption ratio of each air conditioner installed in the stores at every 10 minutes through a year. We calculated the capacity of each air conditioner from those field data, and the thermal load in the store was obtained by summing up capacities of all the air conditioners operating simultaneously in it. By this method we could measure the temporal variations of the indoor thermal loads in the stores at every 10 minutes through the year. Details of this method are explained in the reference (Hirota et al., 2011).

Figure 7(a) shows the indoor thermal load model of a home appliance mass merchandiser (solid line) and time (hours) of appearance of t_j in a year (histogram). The indoor thermal loads per unit floorage BL_{cu} (cooling) and BL_{hu} (heating) are expressed as a function of t_j. This result was obtained by ensemble averaging the measured thermal loads at every 1 °C. Therefore, in this figure, the local maximum thermal loads such as experienced in opening time in winter are leveled off by steady thermal loads that occur in the same t_j range. In higher and lower t_j regions, the thermal load was estimated by the extrapolations of the measured data (orange lines). The heating load is shown by negative values. Here it should



(a) Home appliance mass merchandiser **(b) Food supermarket**

Figure 7: Indoor Thermal Load Models for Business-related Buildings

be noted that we tried to correlate the measured indoor thermal load with other parameters such as wet-bulb temperature of outdoor air, temperature difference between outdoor air and indoor air, etc. It was, however, found that the thermal load in the store was best correlated with t_j. The cooling load BL_{cu} arises from t_j = 13 °C and it increases linearly with t_j. Since there are large heat sources inside the home appliance shop, the cooling load is larger than the heating load in a year. As t_j drops below 12 °C, the operation of the air conditioners is switched from the cooling mode to the heating mode continuously. Figure 7(b) shows the indoor thermal load model of a food supermarket. The cooling and heating loads switch around t_j = 25 °C, and the heating load is much larger than the cooling load in a year. This large heating load is caused by the leakage of cold air from cold showcases in the shop.

4.2 How to Calculate Annual Energy Consumption and Annual Average COP of Air Conditioner

The rated capacities of air conditioners measured in this study Φ_{cr} and Φ_{hr} differed depending on the tested machines even if their nominal values were the same. In order to evaluate the annual energy consumptions and annual average COPs of air conditioners with different capacities under an equal condition, the thermal load ratio of the air conditioner was determined as follows. As observed in Fig. 7, the maximum thermal load of a home appliance mass merchandiser appeared in the cooling season while that of a food supermarket occurred in the heating season. Hence, in calculating the energy consumption with the model of a home appliance mass merchandiser, its floorage was determined so that the cooling load at t_j = 40 °C agreed with the minimum rated cooling capacity of the air conditioners measured in this study. In the calculation with the food supermarket model, the floorage was determined so that the heating load at t_j = -5 °C agreed with the measured minimum rated heating capacity of the air conditioners. The thermal load rations BL_c/Φ_{cr} and BL_h/Φ_{hr} were calculated based on the thermal loads in buildings with these floorages and measured rated capacities of each air conditioner.

The annual energy consumption and the annual average COP of the air conditioner were calculated by the following procedure.

(1) At arbitrary t_j, the thermal load ratio BL_c/Φ_{cr} or BL_h/Φ_{hr} was determined by combining the thermal load model shown in Fig. 7, the floorage described above and the measured rated capacity of the air conditioner Φ_{cr} or Φ_{hr}.

(2) COP of the air conditioner at this t_j and corresponding BL_c/Φ_{cr} or BL_h/Φ_{hr} was determined using the COP surface.

(3) The annual energy consumption at t_j was calculated by combining COP, indoor thermal load BL_c or BL_h, and time (hours) of appearance of t_j in a year.

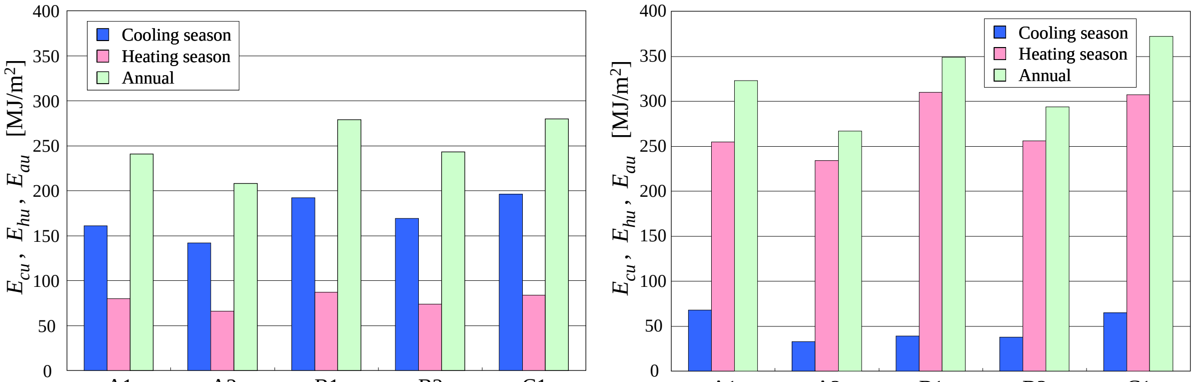
(4) The energy consumption and indoor thermal load at t_j were summed up from t_j = -5 °C to 40 °C to obtain the annual energy consumption and the annual average COP.

4.3 Annual Energy Consumptions and Annual Average COPs of Air Conditioners

Figure 8(a) shows the comparisons of the seasonal energy consumptions E_{cu} (cooling) and E_{hu} (heating) and the annual energy consumption E_{au} of the tested air conditioners evaluated with the indoor thermal load model of a home appliance mass merchandiser. The abscissa shows the type of the tested air conditioner listed in Table 1, and the primary energy consumptions per unit floorage are presented in this figure. The conversion factor from electricity into the primary energy is 9.97 MJ/kWh. The ratio of the energy consumption in the cooling season to that in the heating season is almost constant in all the air conditioners. The annual energy consumption of A2, one of the new type air conditioners, is the smallest among the tested air conditioners, and it is about 13 % smaller than that of A1. As for the other new type air conditioner B2, the annual energy consumption is also decreased by 13 % in comparison with B1. Therefore, the energy-saving effects are improved in both new type air conditioners tested here, but E_{au} of B2 is almost the same as that of A1.

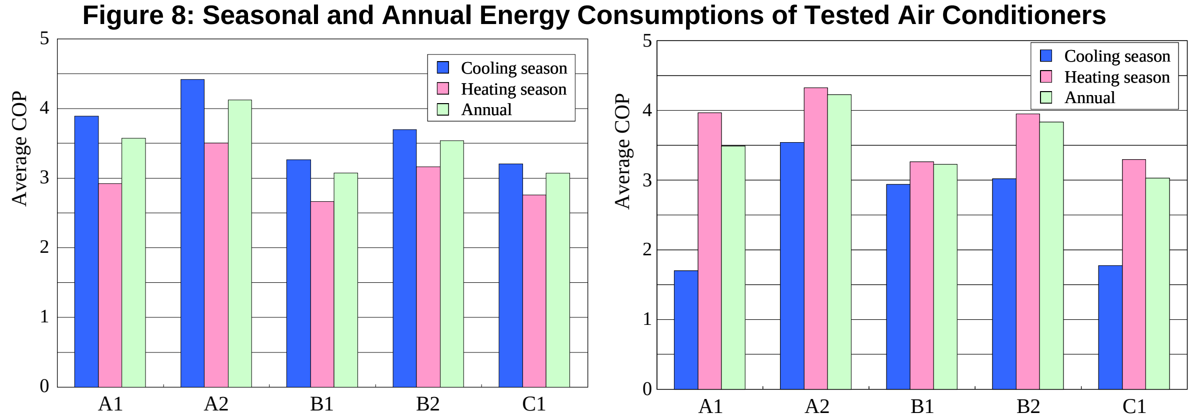
The results for the food supermarket are shown in Fig. 8(b). As expected from the indoor thermal load model shown in Fig. 7(b), the energy consumption in the heating season is several times larger than that in the cooling season. The annual energy consumptions of the new type air conditioners are decreased by 17 % than those of conventional machines of the same manufacturer. Thus, the energy-saving effects of the new type air conditioners can be also confirmed for the thermal load model of the food supermarket.

The seasonal and annual average COPs of the tested air conditioners are shown in Fig. 9 ((a) home appliance mass merchandiser, (b) food supermarket). The annual average COPs of the new type air conditioners are increased by 15 % and 20 % for the indoor thermal load models of the home appliance mass merchandiser and the food supermarket, respectively. In the home appliance mass merchandiser, the heating season average COPs are lower than the cooling season average COPs in all the tested air conditioners. As understood from



(a) Home appliance mass merchandiser **(b) Food supermarket**

Figure 8: Seasonal and Annual Energy Consumptions of Tested Air Conditioners



(a) Home appliance mass merchandiser **(b) Food supermarket**

Figure 9: Seasonal and Annual Average COPs of Tested Air Conditioners

the indoor thermal load model shown in Fig. 7(a), the air conditioner is operated under lower thermal load ratios in the heating season in this shop. Since the heating COPs of air conditioners deteriorate under the low load ratio conditions, the heating season average COPs show lower values than the cooling COPs. For the same reason, the cooling season average COPs are lower than the heating season average COPs in the food supermarket.

5 NEW PREDICTION METHOD OF ANNUAL AVERAGE COPS OF AIR CONDITIONERS

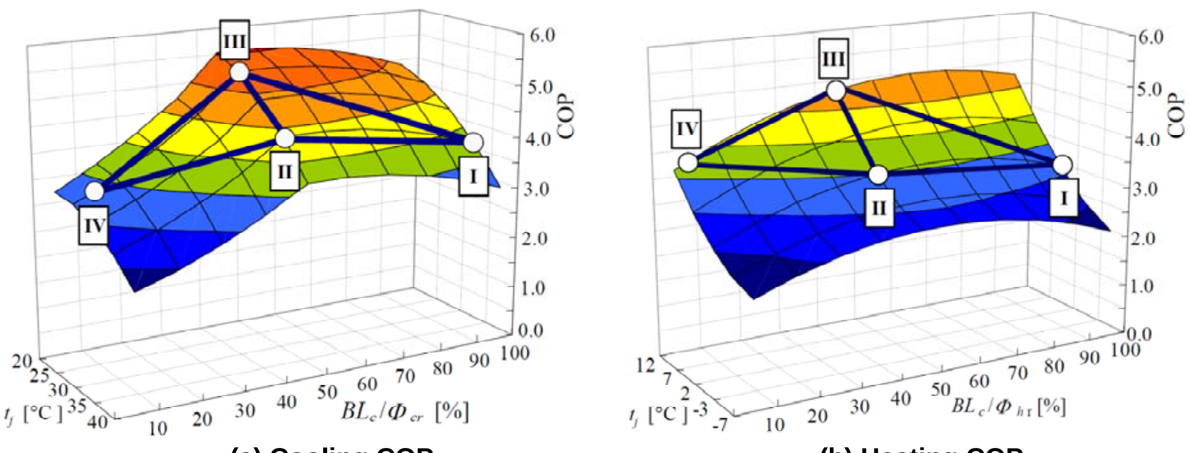
5.1 Concept of the New Prediction Method of Annual Average COP

As described above, COPs of the tested air conditioners show high values in the mid thermal load region and deteriorate in lower thermal load, and they change depending on t_j as well. For an accurate prediction of the annual energy consumption and annual average COP of an air conditioner, it is necessary to reproduce such dependence of COP on the thermal load and outdoor air temperature. We propose the method to express the COP characteristics of the air conditioners based on the COPs measured at the smaller number of test points.

Figure 10(a) shows the concept of this method for the cooling COP. The numbers [I] - [IV] on the cooling COP surface designate COPs measured under corresponding t_j and BL_c/Φ_{cr} in the part-load cooling performance test. Considering that COP of the air conditioner becomes the maximum in intermediate thermal load ratios, we divide the COP surface into two regions setting the boundary in the mid thermal load region (the line connecting [II] and [III] on the COP surface). The COP surface in the higher thermal load region is approximated by the plane that contains [I], [II] and [III], and that in the lower thermal load region is approximated by the plane containing [II], [III] and [IV]. By reconstructing the COP surface with these two planes, the deterioration of COP in the low thermal load operation and the dependence of COP on outdoor air temperature can be reasonably reproduced. The concept for the heating COP is shown in Fig. 10(b). Similar to the cooling COP, the heating COP surface is approximated by two planes that contains [I], [II], [III] and [II], [III], [IV]. Therefore, it follows that the proposed method expresses the variation of COP of the air conditioner in a year by COPs measured at eight test points of the part-load performance test.

5.2 Selection of Test Points

We selected the test points considering that (1) differences between COPs on the original COP surface and on two planes reconstructed by [I], [II], [III] and [II], [III], [IV] become smaller, and that (2) they coordinate with the thermal load models of buildings. At first, we calculated



(a) Cooling COP **(b) Heating COP**

Figure 10: Concept of Proposed Reconstruction Method of COP Surfaces

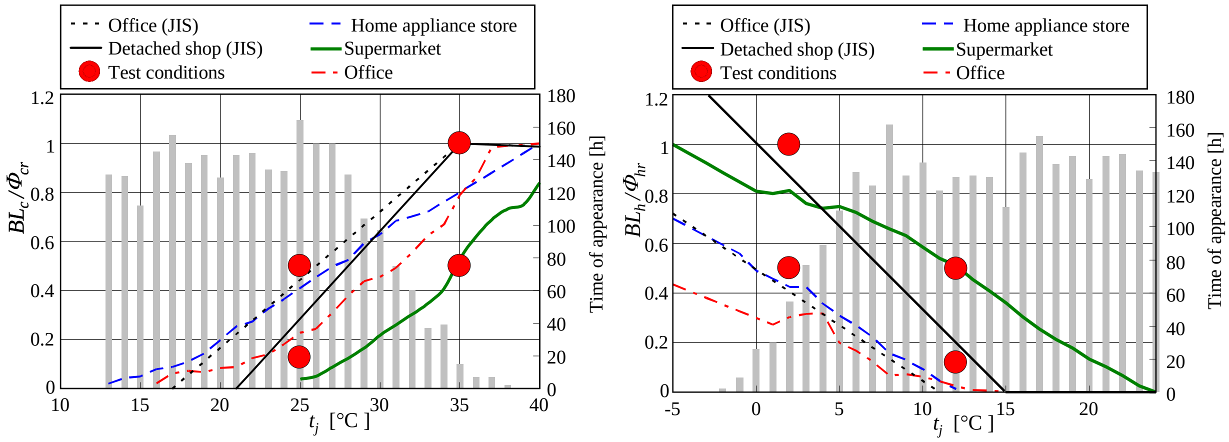
the differences between COPs on the original COP surface and those on two planes reconstructed by several combinations of test points for air conditioners A2 and B2 to find the appropriate selection of the test points. A2 and B2 were selected because the measured points in the part-load performance tests were dense enough to produce accurate COP surfaces. Table 4 shows the test points selected in this study. In this case, the mean square errors of cooling COPs calculated by the proposed method were 2 % of the mean cooling COP of A2 and 4 % of the mean cooling COP of B2. The mean square errors of heating COPs were 5 % for A2 and 8 % for B2. The prediction errors increase in the heating COPs in both air conditioners, but the COP characteristics of the air conditioners are approximated reasonably well by the proposed method. It should be, however, noted that the errors shown here are different from those included in the seasonal / annual average COPs predicted by the proposed method, because COPs under specific conditions corresponding to the indoor thermal load and time of appearance of t_j are weighted in calculating those average COPs.

Next, the relationship between the selected test points and the indoor thermal load models is shown in Fig. 11. Variations of the indoor thermal loads with t_j are shown by solid and broken lines for a home appliance mass merchandiser, food supermarket and office measured by the authors, and office and detached shop prescribed in JIS B 8616: 2006. These thermal load models were selected because the multi-split type air conditioners with relatively large capacities are often installed in these buildings. The maximum thermal load was assumed that, in a home appliance mass merchandiser and an office (measured model), the cooling load ratio BL_c/Φ_{cr} = 100 % at t_j = 40 °C. In a food supermarket, the maximum thermal load was assumed to appear in the heating season and BL_h/Φ_{hr} = 100 % at t_j = -5 °C. The red symbols correspond to the test points shown in Table 4. The regions enclosed by selected test points can cover the major part of the indoor thermal load models, meaning that the test points selected in this study coordinate with the thermal loads in buildings.

We calculated the annual average COP of the air conditioner based on the COP surfaces reconstructed by the proposed method, and compared the results with those obtained by the original COP surfaces (results shown in 4.3) to evaluate the accuracy of this method.

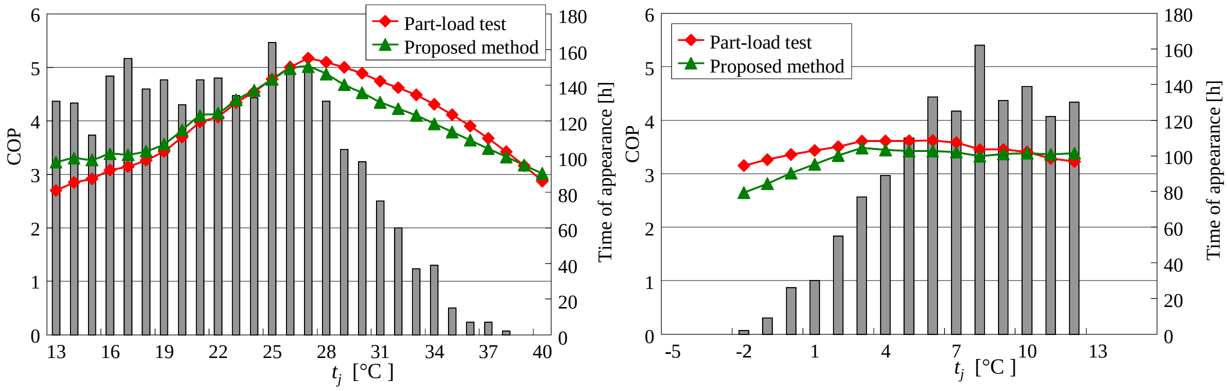
Table 4: Test Points Selected in the Proposed Prediction Method of Annual Average COP

	Cooling tests		Heating tests	
No.	Outdoor air temp.	Thermal load ratio	Outdoor air temp.	Thermal load ratio
I	35 °C	100 %	2 °C	100 %
II	35 °C	50 %	2 °C	50 %
III	25 °C	50 %	12 °C	50 %
IV	25 °C	12.5 %	12 °C	12.5 %



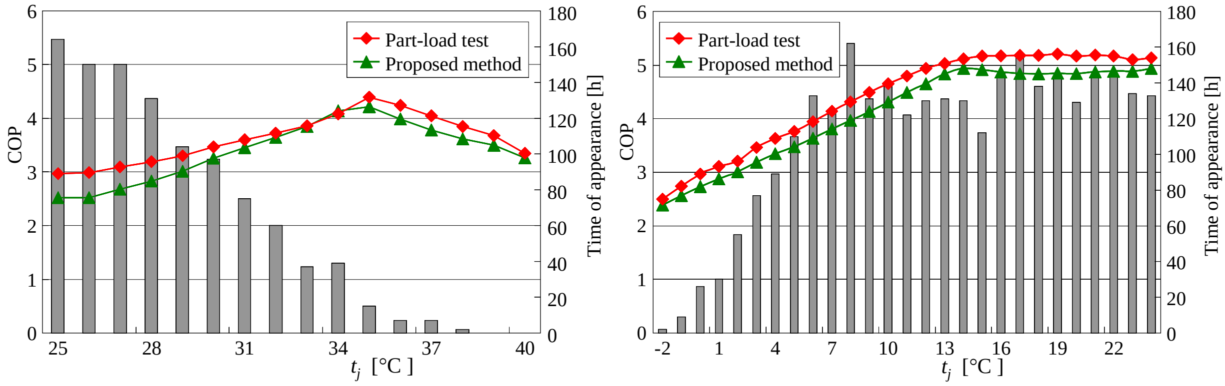
(a) Cooling COP **(b) Heating COP**

Figure 11: Relationship between Indoor Thermal Load Models and Selected Test Points



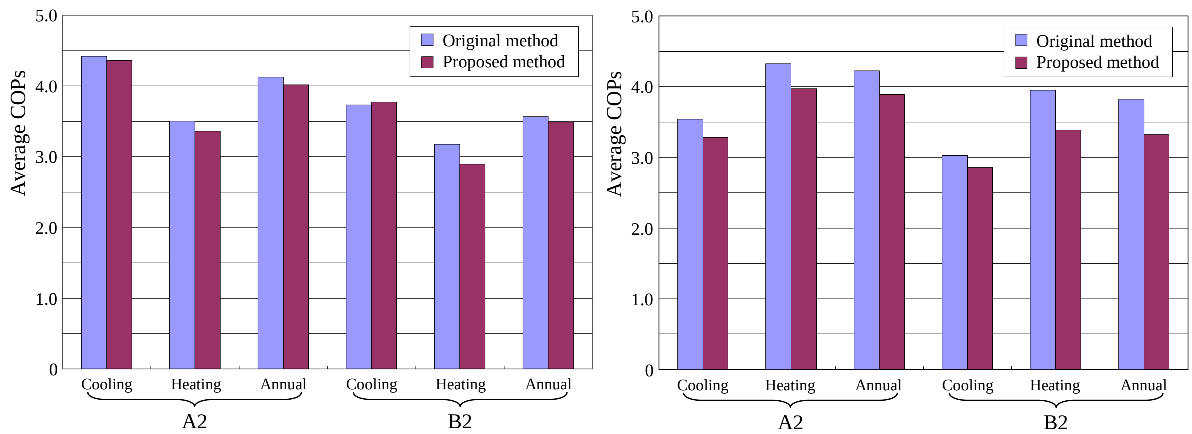
(a) Cooling COPs **(b) Heating COPs**

Figure 12: Comparison of Measured and Predicted COPs (Home Appliance Mass Merchandiser)



(a) Cooling COPs **(b) Heating COPs**

Figure 13: Comparison of Measured and Predicted COPs (Food Supermarket)



(a) Home appliance mass merchandiser **(b) Food supermarket**

Figure 14: Comparison of Measured and Predicted Average COPs of the Air Conditioners

5.3 Results of COP Predictions

Figures 12 and 13 show the variations of COPs with t_j calculated based on the original COP surfaces (red lines) and on reconstructed one by the proposed method (green lines). The air conditioner is A2 and the building uses are a home appliance mass merchandiser (Fig. 12) and food supermarket (Fig. 13). Time (hours) of appearance of t_j in a year is also shown by histograms. Here it should be noted that since the indoor thermal load ratio is given as a function of t_j, COPs shown in these figures reflect the influences of both t_j and corresponding thermal load ratio BL_c/Φ_{cr} or BL_h/Φ_{hr}. Although the proposed method slightly underestimates COPs for both indoor thermal load models, COPs predicted by the new method agree quantitatively well with COPs calculated from the original COP surfaces, i.e., COPs measured at 67 test points in the part-load performance test.

In Fig. 14, we compare the seasonal and annual average COPs of new type air conditioners A2 and B2 obtained by the proposed method with those calculated from the original COP surfaces. For the indoor thermal load model of a home appliance mass merchandiser shown in Fig. 14(a), the annual average COPs obtained by the proposed method agree with those calculated from the original COP surfaces within errors of 3 %. For the food supermarket shown in Fig. 14(b), the errors of the annual average COPs predicted by the new method are increased to 8 % in A2 and 13 % in B2. As observed in Fig. 11(b), the heating load arises in a relatively high t_j region in the supermarket. Since the heating load in this region is out of the area enclosed by the selected test points, the heating COP in this region is predicted by the extrapolation of the measured data. Moreover, the time of appearance of t_j in this region is quite large as found in Figs. 11(b) and 13(b). These factors increased the prediction errors of the annual average COPs in the food supermarket.

6 CONCLUDING REMARKS

In this study, part-load performance tests were made on recently developed multi-split type electric heat pump air-conditioners for buildings with rated capacities of 45 kW (cooling) and 50 kW (heating). Cooling COPs and heating COPs were measured at 36 test points and 31 test points, respectively. Based on the measured COPs and thermal load models for business buildings, the annual energy consumptions and annual average COPs of the tested air conditioners were evaluated. Moreover, we proposed a new method to predict the annual average COP, in which the variations of COP with the outdoor air temperature and the indoor thermal load were modelled based on COPs measured at eight test points. The error of the annual average COP predicted by this method was smaller than 3% for the thermal load model of a home appliance mass merchandiser but increased for a food supermarket.

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