PRACTICAL STUDY ON ENERGY SAVING METHOD APPLIED TO EXISTING HVAC SYSTEM -CASE OF CEPCO GIFU BRANCH OFFICE- Part.1 : HVAC SYSTEM OUTLINE AND RESULTS OF THREE YEARS INITIAL COMMISSIONING AFTER COMPLETION

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Abstract: As our environmental problem such as climate change has become significant issue, we should reduce carbon dioxide emission from both public and private section including not only industry but also office buildings. As to office buildings, considering HVAC system consume a lot of energy and it reaches almost half of total energy of whole building, we should take measure to control them appropriately to reduce CO_2 emission. Generally speaking, it is said that installing high efficiency HVAC system into office buildings is one of the most effective method to achieve this goal. However, when we take a look at actual operation of HVAC systems, some of them couldn't meet expected performance and results in energy waste, mainly due to the way to handle with HVAC system after completion. With a view to establish a high quality control method of HVAC system, we applied to commissioning process to an existing HVAC system of our Gifu branch office with high efficiency HVAC system. This paper, Part.1, shows heat source system outline and results of initial commissioning which was carried out during 2001 to 2003. It was found that initial commission is greatly contributed to optimization of HVAC system operation.

Key Words: Highly efficient HVAC system, Energy saving, Initial Commissioning

1 INTRODUCTION

It is said that one of the most effective way to reduce CO_2 emission from office buildings is adopting high efficiency HVAC system, because annual energy consumption of HVAC system holds almost 50%[1] of total energy use of office buildings. As guide line of academic society such as SHASE[2] goes, quality control or life cycle commissioning process should have been applied to HVAC system throughout planning phase to operational phase to meet expected performance which is prescribed by owner at a planning phase, called owner's requirements. However, when we take a look at actual operation of newly completed HVAC system, we could see some of them couldn't meet them and results in energy wasting despite of adopting high efficiency HVAC system. Considering the background mentioned

above, we think as a one of owner, we need more business oriented quality control method for HVAC system to make them environmental friendly during their endurance periods. Aiming to reduce CO_2 emission from our office building after completion, commissioning process has been applied to our Gifu branch office. Our Gifu branch office was designed under the concept of “energy saving” or ”consideration for environment” and was completed March, in 2001. As for its heat source system, ultra highly efficient heat pump unit whose designed COP during chilled water mode is 7.59 and thermal stratified storage tank (shallow and multiple type) were installed in. We carried out initial commissioning during 2001 to 2003 and re-commissioning during 2008 to 2009 to optimize their energy saving performance. Graph format for daily operational management using BEMS were also developed with a view to making operational fault detect easy. In this paper, part.1, heat source system outline and results of initial commissioning utilizing data which was stuck at central monitoring and control panel are presented. It is found that initial commissioning is greatly contributed to optimization of heat source system operation.

2 HEAT SOURCE OUTLINE

Table 1 shows building and heat source outline of target buildings. Our Gifu branch office building is located in the downtown of Gifu city, central Japan with mild climate. Its total floor area is 24,097m^2 with 11 floors above the ground and 1 basement. It is mainly used for office except 11th floor for space of welfare and meeting and basement for machinery room. Figure 1 shows schematic diagram of heat source. Figure 2 shows plan drawing of thermal storage tank and its layout of header pipe, diffuser or perforated pipe which is installed into double slab space under the basement. Water thermal energy storage system is adopted as main HVAC system for office space, 1st floor to 10th floor of this building. Heat source unit, listed in table 1, consist of two heat pump units. The one is well water cooled type heat pump unit, called ultra highly efficient heat pump unit, R1.The other is air cooled type heat pump unit, R2. R1 is newly developed heat pump unit with a view to achieve not only further energy

Table 1 Building and heat source outline

Building outline	Total floor area : 24,097m^2, 1 basement floor , 11 floors above ground	
Component	Symbol	Outline
Heat source unit	R1	Well water cooled type ultra highly efficient heat pump unit Chilled water mode : Cooling capacity 668kW COP 7.59 Heat recovery mode : Cooling capacity 545kW COP 7.93 Heating capacity 700kW Electrical power input : 88kW (Chilled water mode) Electrical power input : 157kW (Heat recovery mode) Cooling water temperature 15 deg
	R2	Air cooled type heat pump unit Cooling capacity : 300kW, Heating capacity : 375kW Electrical power input : 153.9kW(chilled water mode)
Water thermal storage tank	A-C Tank	Shallow and multi- tank parallel type thermal stratified storage tank Water Depth 1.5m, Utilized temperature difference 10 deg. A Tank : Chilled / hot water thermal storage tank 259m^3 B Tank : Chilled water thermal storage tank 309m^3 C Tank : Chilled water thermal storage tank 191m^3
Heat exchangers (primary)	HEX C1	Exhaust cooling* (Inlet temperature compensation for R1 unit)
	HEX D1	Heat source for chilled water mode (R1 unit) Exhaust heating for heat recovery mode (R1 unit)
	HEX H1	Warm water supply for heat recovery mode(R1 unit)
	HEX 1	Chilled and warm water supply (R2 unit)
Pumps (primary)	PC 01	Chilled water pump for R1, evaporator side
	PH 01	Warm water pump for R1, condenser side
	PCH 01	Primary chilled / warm water pump for R2
	PCH 02	Secondary chilled / warm water pump for R2

Note *Inlet water is warmed until 15 deg. by well water to stabilize heat source operation.

saving performance but also reducing environmental load. As it adopt Lorentz cycle for refrigerating cycle, high efficiency plate type counter flow heat exchanger and semi closed type screw compressor from the point of preventing refrigerant leakage like that, its energy saving performance is greatly improved comparing with conventional heat pump unit which had developed so far. Therefore, it could achieve high COP both of chilled water mode and heat recovery mode. These two heat pumps and water thermal storage tanks are operated in chilled water mode during summer season. When it comes to winter season, R1 shifts its operation mode into heat recovery mode and supplies both warm water and chilled water. R2 turns its operation into warm water mode. Water thermal storage tank is also divided into warm water tank, A tank, and chilled water tank, B and C tank, during winter season. Cooling and heating load predictive control using Kalman filter is applied to daily operation of heat source system including water thermal storage tank. R1 is designated priority operation unit all the year round under this control from the point of further energy saving. R2 is used as auxiliary unit to deal with peak load during summer season and operational suspension of R1 such as maintenance or breakdown.

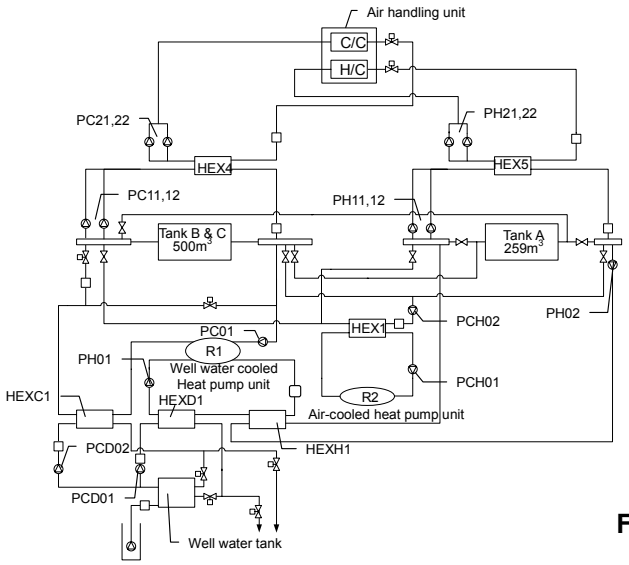
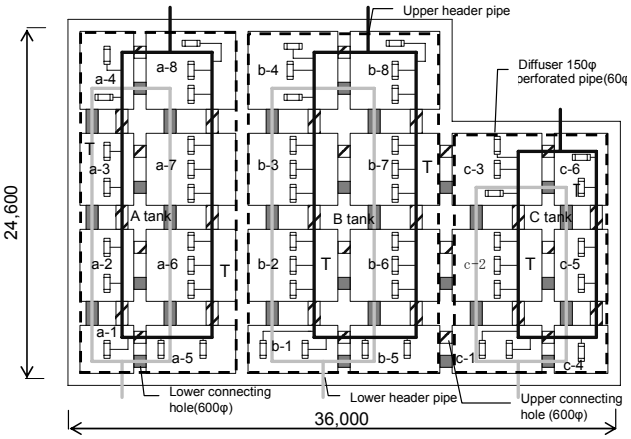


Figure 1 Schematic diagram of HVAC system



T: Measurement points of water temperature
Set at 300,600,900 and 1,200mm from the bottom of thermal storage tank

Figure 2 Plan drawing of thermal storage tank and layout of header pipe, diffuser and connecting hole

3 RESULTS OF THREE YEARS INITIAL COMMISSIONING AFTER COMPLETION

3.1 Three years commissioning outline

Table 2 shows three years initial commissioning outline for heat source system. A series of performance verification of heat source had been carried out through fiscal 2001 to 2003 to confirm whether heat source system demonstrates expected performance which was prescribed by owner at the planning phase or not. Operational optimization, which based on verification results and aims to further energy saving, had been carried out beside performance verification mentioned above. Measurement points such as electrical energy, flow quantity and water temperature were set at each unit of heat source system not only for initial commissioning but also for continuous operational optimization during endurance

Table 2 Three years initial commissioning outline for heat source system

Component	Verification items	Implementation year		
		2001	2002	2003
Heat pump unit	Operational state analysis of R1 and R2 Performance assessment using COP index	○	○	○
Hear source control	Heat load predictability	○	○	-
	Reduction of exhaust heating during winter	○	-	-
	Optimization of cooling water temperature	○	-	-
Thermal storage tank	Efficiency on thermal storage tank Stored heat balance between divided tank	○	-	-

period of this system. As for water thermal storage tank, detailed measurement points of water temperature were set at a-1 tank and a-8 tank every 100mm from the bottom for further verification of stored heat balance during 3 years initial commissioning in addition to measurement points shown in Figure 2. These data were measured every one minute except electrical energy, collected every 1 hour.

3.2 Performance verification of ultra highly efficient heat pump unit (R1)

Results of performance verification during chilled water mode of R1 are listed in Table 3. As for maker test value, as electric power input became higher than designed value while cooling capacity inferior to designed value, COP resulted in 6.59, rather smaller value comparing with designed value 7.59. It is mainly due to cooling water temperature which was three degree higher than designed value owing to the restriction of test device of laboratory. COP of measured value resulted in 6.50, 86% of designed value. Although operational condition such as electric power input and cooling water temperature were almost similar to maker test value, cooling capacity was inferior to maker test value. It was attribute to larger flow quantity of heat exchanger, HEXD1 shown in Figure 1, than designed value during verification term. Unstable cooling water inlet temperature also influenced this result. Same performance verification was carried out during heat recovery mode and COP resulted in 7.77, slightly lower than designed value 7.93.

Table 3 Performance verification results of R1 during chilled water mode(Representative day)

Items	Unit	Designed	Maker test	Measured *1	Evaluation *2
Electric power input	kW	88.0	100	96.8	110
Chilled water inlet temperature	degree	15.0	15.1	14.8	99
Chilled water outlet temperature	degree	5.0	5.25	5.64	113
Cooling water inlet temperature	degree	17.0	20.1	19.0	112
Cooling water outlet temperature	degree	27.0	30.0	28.0	104
Chilled water quantity	m^3/h	57.5	57.5	59.4	103
Cooling water quantity	m^3/h	65.1	65.3	70.6	108
Cooling capacity	kW	674	661	628	93
COP	-	7.59	6.59	6.50	86

Note *1 Average value, obtained August 1st, 2001

*2 Relative value of measured value to designed value

3.3 Results of operational improvement aim to reduce exhaust heating during winter (R1)

Result of operational improvement of exhaust heating during winter is shown in Figure 2 and operational status of R1 during fiscal 2001 and 2002 are shown in Figure 3. Inlet water temperature of R1 is cooled by well water at heat exchanger, HEXD1, to keep warm water inlet temperature constant. Actually, inlet water temperature of R1 is controlled to be 38 degree during winter season. However, it is equivalent to discharge or exhaust heating because generated warm water is cooled substantially using well water. Despite of installing high efficiency heat pump unit, efficiency deterioration which was caused by a large amount of exhaust heating became clear from the operational status in fiscal 2001, the first year after

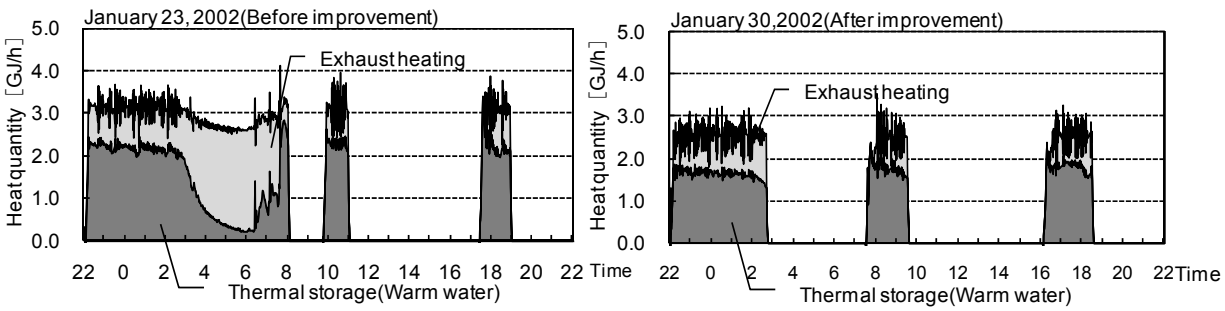


Figure 2 Operational improvement of exhaust heating (Representative day)

completion, shown in Figure 3. As R1 was only operated based on the predicted values of heat load predictive control device, thermal storage operation during night time had been continued until it met predicted value or target thermal storage quantity before improvement. However, inlet water temperature tends to rise in accordance with thermal storage quantity and began to meet target value. Especially, exhaust heating increased sharply after 3 AM, shown in the left of Figure 2, because thermal storage quantity almost met target value at this time. Therefore, another control index which control operation of R1 was installed to reduce exhaust heating in the second half of thermal storage time in addition to heat load predictive control. When water temperature sensor, which was set 300mm above from the bottom of thermal storage tank, met 44 degree while utilized temperature difference of water thermal storage tank were 47 degree to 37 degree, it judged that thermal storage quantity met target value and stopped R1. Although load shifting ratio, an important index of water thermal storage system, were slightly down owing to this improvement, exhaust heating after 3 AM were greatly improved in comparison with before improvement. (See in the right of Figure 2) Annual exhaust heating during fiscal 2002, shown in the below of Figure 3, was also decreased comparing with fiscal 2001, shown in the above of Figure 3.

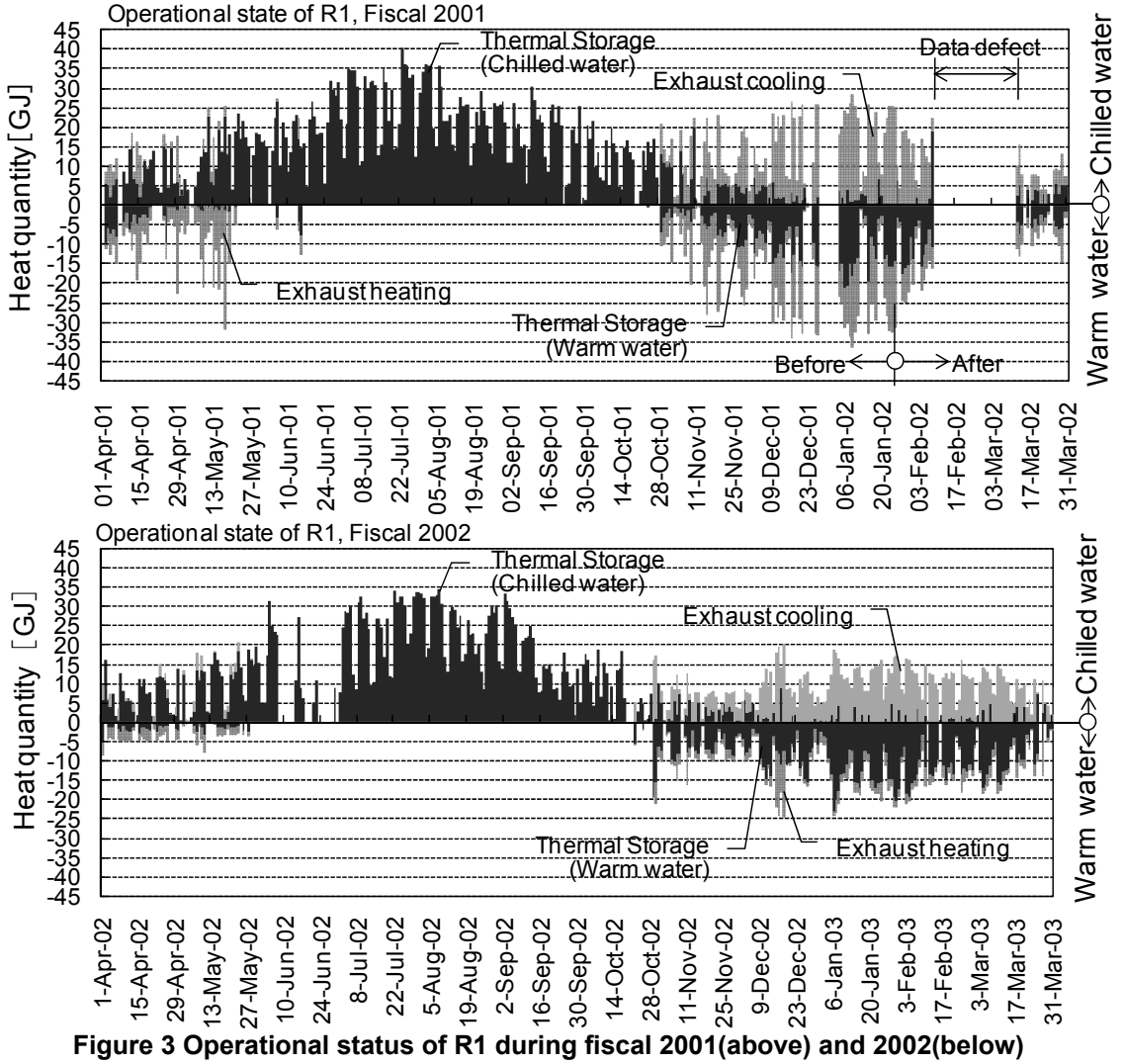


Figure 3 Operational status of R1 during fiscal 2001(above) and 2002(below)

3.4 Results of annual performance verification with COP during 3 years after completion (R1)

Results of annual performance verification using COP of R1 and R2 during 3 years initial commissioning are listed in Table 3. To take fully advantage of strong points of water thermal storage system with highly efficient heat source, effective COP of R1 and system COP of R2 should be kept high. To achieve this goal, several operational improvement which was

reflected the results of daily operation analysis including exhaust heating, as stated above, had been taken continuously. COP and system COP of R1 tended to decline in each year. It is attributed to operating hour of heat recovery mode with high COP comparing with chilled water mode. Operating hour of heat recovery mode in fiscal 2001 became longer than any other years because it was before improvement of exhaust heating (see Figure 2), while that of fiscal 2003 became shorter owing to warm winter with small heat load. On the other hand, effective COP including system COP tended to improve gradually in each year owing to operational improvement of exhaust heating. Although operational condition which aim to decrease exhaust heating were the same past two years in fiscal 2003, effective COP was superior to fiscal 2002 on the grounds that operating hour of R1 during winter became shorter because of warm winter and resulted in small amount of exhaust heating comparing with past two years. As for R2, although we didn't take measure to improve operation for further energy saving, COP and system COP were gradually improved. It is mainly due to decrease discharge mode operating hour during summer time with strict air temperature condition for air cooled type heat pump unit in 2001 and 2002. Cool summer and warm winter in 2003 with moderate air temperature condition comparing with past two years also influenced this result.

Table 3 Results of annual performance verification with COP

Heat pump	COP Classification	Fiscal year		
		2001	2002	2003
R1	Compressor COP *1	7.05	6.97	6.80
	System COP*2	5.90	5.75	5.63
	Effective compressor COP*3	4.66	5.04	5.27
	Effective system COP*3	3.90	4.16	4.36
R2	COP*1	2.23	2.42	2.38
	System COP*2	2.04	2.23	2.20

Note *1 Compressor COP Includes electrical energy consumption of compressor
*2 System COP includes electrical energy consumption of compressor and auxiliary pumps
*3 Effective COP is evaluated excluding exhaust heating or cooling.

3.5 Verification of stored heat balance in different thermal storage tank

Water thermal storage tank are consisted of three tanks with several divided tanks and each divided tanks are connected by header pipe and connecting hole, listed in Table 1 and shown in Figure 2. As for flow rate balance or stored heat balance in each divided tanks are adjusted equally by the length of branch pipe which is connected header pipe and connecting hole. Even if flow rate of chilled/hot water unbalanced, heat balance could be kept by connecting hole. Figure 4 shows temperature response at 4 points of 300mm, 600mm,

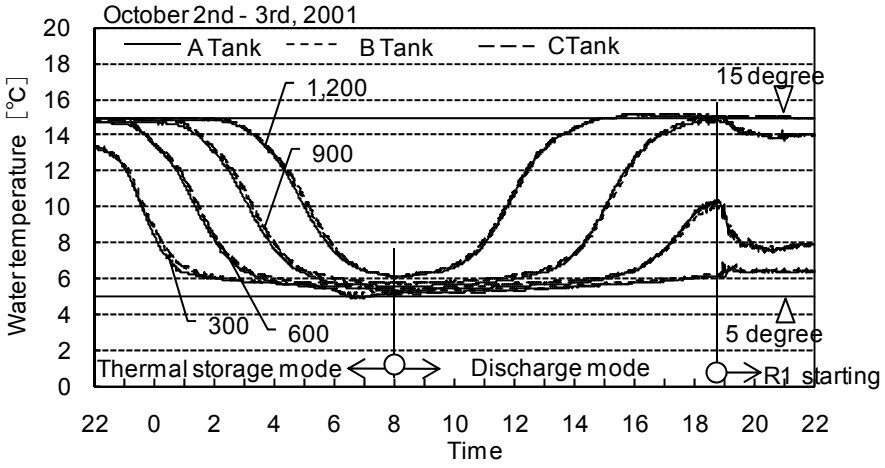


Figure 4 Temperature responses in each tank (Stored heat balance, representative day)

900mm and 1,200mm height from the bottom of each tank. It can be seen that any difference was hardly observed in temperature response at the various height from these results. Figure 5 compares the transition of temperature profile of a-1 tank and a-8 tank, divided tank of A tank. Although a-1 tank and a-8 tank are the most distance tank each other in A tank, their temperature profiles roughly show the same movement and stored heat balance of two end tank was also maintained by the same pipe.

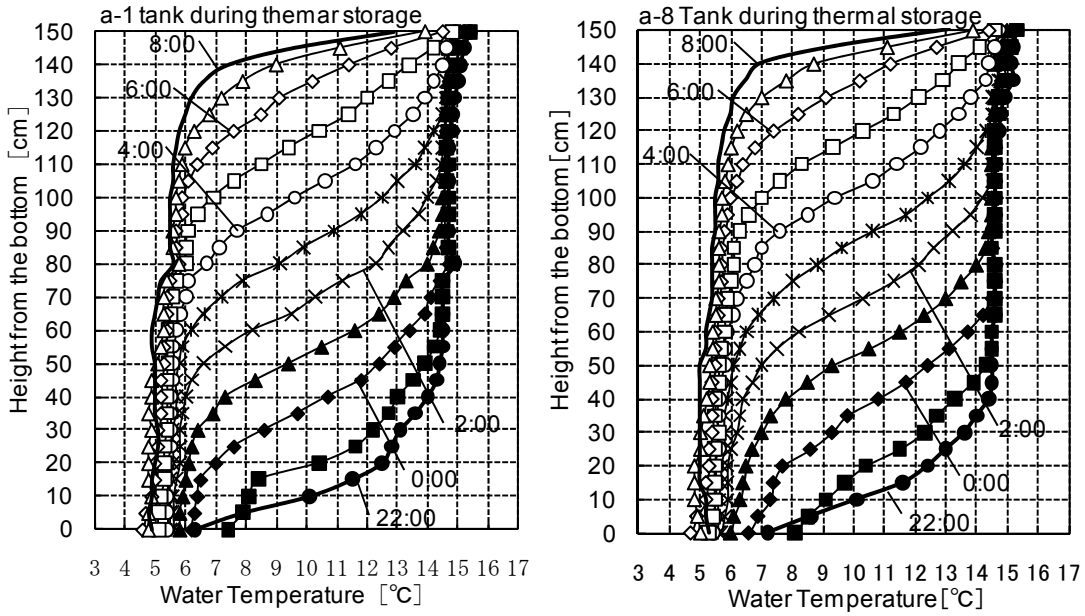


Figure 5 Temperature transition of divided tank (a-1 tank and a-8 tank, October 2 to 3, 2001)

3.6 Load shift ratio of water thermal storage system

R1 is operated based on the predicted values of heat load predictive control device, as stated above. As it predicts heat load using results of daily operation, it can roughly follow the annual change of load. Figure 6 shows correlation between measured value and predicted value during three years initial commissioning. We can say that heat load predictive control could follow the change of heat load precisely from these results because correlation coefficient or R^2 value became 0.93 during summer and 0.88 during winter. Figure 7 shows results of monthly load shift ratio of water thermal storage system including both heat pump units and its auxiliary pumps. Although designed annual load shift ratio including auxiliary pumps were 80%, results under actual operation resulted in 66.9% in fiscal 2001, 72.1% in 2002 and 78.1% in 2003. As for fiscal 2001, operational error of heat load predictive control could be one reason because it couldn't be in operation during April to June owing to control disorder. After it began to operate, it failed to follow the change of heat load because

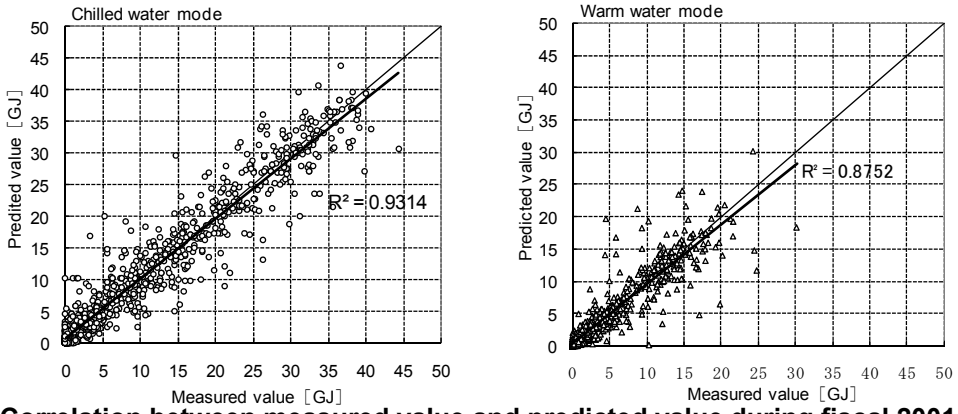
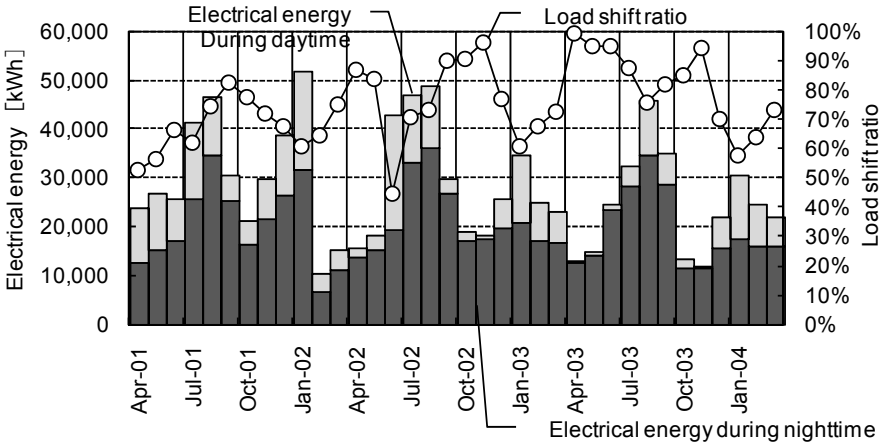


Figure 6 Correlation between measured value and predicted value during fiscal 2001 to 2003

of inadequate operational setting during mid season. Suffering from breakdown of R1, main heat pump of water thermal storage system, in fiscal 2002 could be another reason of the low load shift ratio. On the other hand, this index was rather improved in fiscal 2003, mainly due to cool summer and warm winter with light heat load comparing past two years.



Note Load shift ratio is the index which stands for ratio for electrical energy during night time (22:00-8:00, 10hours) to daily electrical energy (24hours).

Figure 7 Monthly load shift ratio of R1 and R2 during fiscal 2001 to 2003

4 CONCLUSIONS

Through 3 years initial commissioning which aimed to confirm the performance of highly efficient heat source system with newly developed heat pump and realize further energy saving, the following results were obtained.

Measured COP of ultra highly efficient heat pump during chilled water mode resulted in 6.50, 86% of the designed value 7.59. It was mainly due to large flow quantity and unstable cooling water inlet temperature during verification term, while COP during heat recovery mode or winter season became 7.77, slightly lower designed value 7.93.

Despite of installing high efficiency heat pump unit, efficiency deterioration could be seen in fiscal 2001, soon after completion, owing to a large quantity of exhaust heating during heat recovery mode. After installing another control index in addition to heat load predictive control for heat source, exhaust heating was greatly decreased in fiscal 2002.

After operational improvement of exhaust heating which lead to decreasing operating hour of heat recovery mode with high COP was put into practice, annual basis compressor COP and system COP of R1 tended to decline in each year. On the contrary, effective COP and effective system COP gradually improved same reason mentioned above.

Flow rate balance and heat balance of shallow and multi-tank parallel type thermal stratified tank were confirmed. As any difference was hardly observed in temperature response at vertical direction in each tank, we could say that satisfactory results were obtained.

Annual based load shift ratio resulted in 66.9% in fiscal 2001, 72.1% in 2002 and 78.1% in 2003, lower than designed value 80%. As for fiscal 2001, operational error and inadequate setting during mid season influenced these results.

5 REFERENCES

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