Diagnosis of Effectiveness of HVAC System and Energy Performance of Osaka-Gas Building through Retro-Commissioning
Part 1 Outline of HVAC Systems and Diagnosis of Energy Efficiency of Air Systems.

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1.ABSTRACT

The south part of Osaka-Gas Building was built in 1933 and the north annex was built in 1966, both of which were equipped with complete HVAC systems using centrifugal chillers. The original HVAC system of the latter was induction unit system for the perimeter and dual-duct system for the interior. The present energy plant consists of gas-fired absorption and cogeneration systems. Both parts of HVAC as well as energy plant system experienced several times of retrofits for energy conservation. Present three papers introduce some of the results of retro-commissioning process to identify actual figures of operation, controls and maintenance for raising energy efficiency using actual measurements, theoretical system analyses and HVAC system simulations.

2. INTRODUCTION

In case of the initial commissioning, verification items are specified based on the Owner's Project Requirement (OPR), the performance is verified in each process, and finally results are checked whether the performance defined in the OPR is achieved or not through functional performance tests.

In contrast, the retro-commissioning includes a process for commissioning the performance of the existing building systems. After the performance was evaluated in light of the verification purposes, improvement plans are developed and implemented as needed, and the results are checked.

In the retro-commissioning, items and scope to be evaluated shall be defined beforehand, as a part of the OPR, because the purposes of retro-commissioning will vary widely such as: improvement of control or operational methods of equipment and sub-systems, improvement of the system itself, and renewal or renovation of equipment, which seek for direct effects, and/or such as; introducing or upgrading continuous monitoring system, favorably with fault detection and diagnosis system to identify system and equipment performance, which seeks for broader effects. Thus the area to be evaluated should be determined in accordance with the commissioning purposes.

The guideline how to proceed commissioning process and how to specify equipment performance have been established elsewhere as in SHASE, however, standardizing how to define and how to proceed the verification process for existing system performance are difficult, because whichever system has its own design philosophy despite of the lack of detailed documentations.

The present paper describes, the outline of the retro-commissioning process of a memorial kind of building, situated in Osaka, with complicated system combination coming from its historical character, which had been brought up with the growth of HVAC technology itself since 1930s.

3. OUTLINE OF BUILDING AND EQUIPMENT

3-1. Outline of Osaka-Gas Building

(1)Outline of building

Address : Hirano-machi, Chuo-ku, Osaka City
Use : Office building
Structure: Steel-reinforced concrete, reinforced concrete
Number of floors: Eight floors above ground and two floors below ground
Total floor area : Total of south and north parts: about 46,987 m^2
South part: 18,422 m^2 (built in 1933)
North part: 28,475 m^2 (extended in 1966)

(2)Outline of HVAC system. The south part of the building was originally equipped with complete air conditioning system installed by TOYO Carrier Corporation, despite the fact that it was built as early as in 1933. The entire building, which comprises offices, an auditorium, dining rooms, and a showroom, was air-conditioned with temperature and humidity control devices. Also installed were the heating radiators for the perimeter area. The energy plant consisted of the turbo refrigerating machine and gas-fired boilers, both of which were made in USA.

In 1966, the north annex building was completed, the air conditioning systems of which were the state-of-the art in those times, that is, induction unit systems for the perimeter zone and dual-duct systems for the interior zone. The energy plant was again the turbo refrigerators and gas-fired boilers, but with cold water thermal storage tank. After the oil crises from 1973, energy un-efficiency of the north annex HVAC system was clarified and various kind of renovations for energy saving were implemented by stages.

Also promoted were changeover of energy source to gas from electricity, natural as gas supplier, such as; introduction of CGS, introduction of absorption type cooling and heating machines, integration of energy plant of the north and south, removal of heat storage tanks.

Radiators and induction units in the perimeter zone were changed to four-pipe fan-coil units with primary air, dual-duct system for interior zones were renovated as VAV system, ventilation demand control based on CO_2 concentration were also introduced.

4. APPLICATION OF RETRO-COMMISSIONING TO OSAKA-GAS BUILDING

4-1. Outline of retro-commissioning processes

The retro- commissioning consists of six phases to be followed as shown in Figure 4.1.

4-2. Retro-commissioning plan

(1)Set-up the project and define objectives

Osaka-Gas Building is historical architecture, the south part of which was built 80 years ago. The systems have been changed and the apparatuses have been renewed many times. Few people can understand the design concept of the building systems at the time of construction or system retrofits, which is typical of old architecture.

The objective of the project is to confirm the validity of retro-commissioning, to clarify problems concerning equipment management and operation and to identify performance of equipment and subsystems by analyzing the actual operation status, in order for the building owner to make decision for further actions.

(2) Study and development of working plan

i) Scope of application of the process. The process includes a part of the *Planning phase* and the *Inspection phase* in Figure 4.1, but does not include on-site detailed measurement. It also does not include an implementation phase and subsequent phases.

The process should include air side system, water distribution system and energy plant as the object of commissioning, however, due to the insufficient data, detailed measurements and analyses on energy plant and water system was postponed to the 2nd stage.

ii) Commissioning team and work sharing. The commissioning management team, CMT, independent

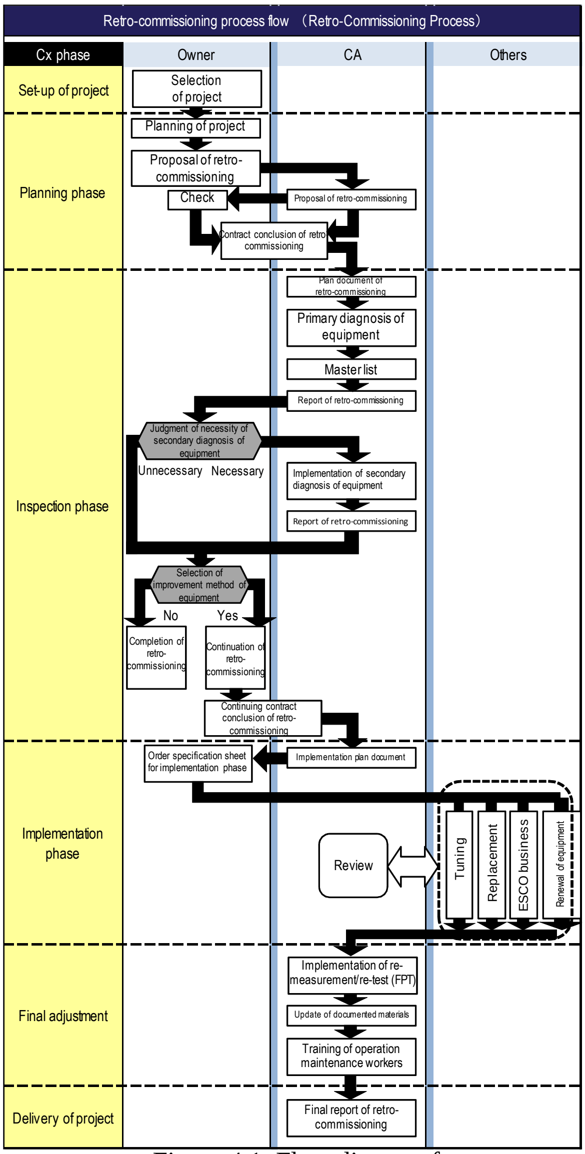

Figure 4.1 Flow diagram for retro-commissioning process

from building management and O&M, was set up mainly by BSCA. The CMT drew up commissioning documents for the project, reviewed materials relevant to the system and equipment stored by the building manager, and clarifies the characteristics of the system constitution and operational problems. The CMT also proposes performance improvement plans and makes reports based on analyses of provided BEMS data and actual measurement data by its own.

(3)Performance verification process. The performance verification process is shown in Figure 4.2. Energy conservation measures were extracted based on ⅰ) macro analyses using BEMS data, ⅱ) additional actual measurement data analyses and ⅲ) HVAC system simulations using LCEM tool. The detailed air-conditioned state analyses and the mixing energy loss analyses for a representative HVAC system, AC-22, were carried out and details are given in the present paper.

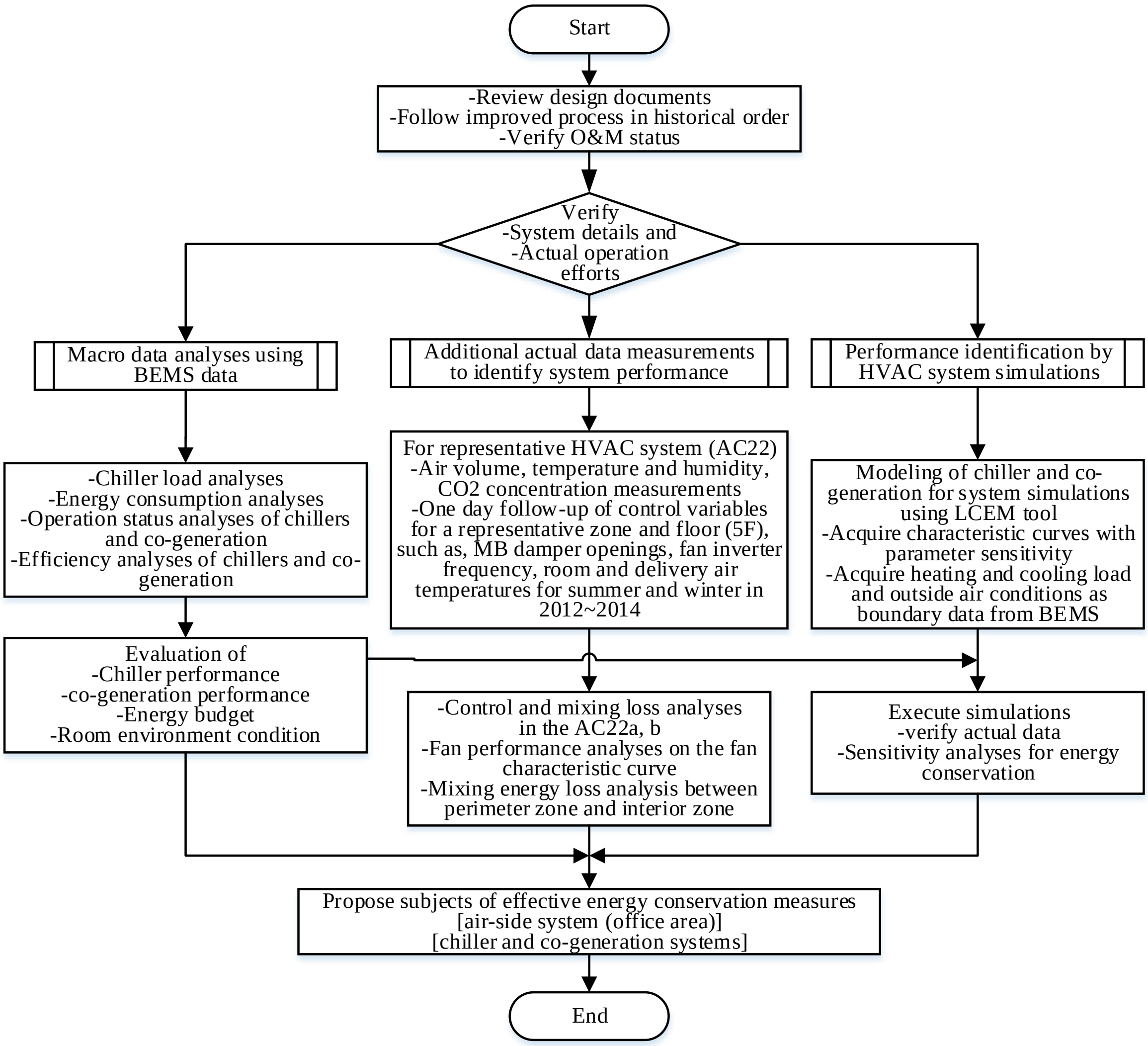

Figure 4.2 Performance verification process (1)

5. IDENTIFYING SYSTEM OPERATION

5-1.Detail of Air-Conditioning System and Controls for the Office Area of the North Building

Main outside air for the entire parts of the building is drawn through outside air shafts situated in the north part, treated by coils and air washers before being introduced to each air conditioning unit. Recirculated air from the interior zone of standard floors return via return air louvers on each floor and each shaft situated in the south and north parts to each air conditioning unit. The demand control by CO_2 concentration is incorporated into the return air system to minimize the amount of outside air. An exhaust air fan is installed to enable total outside air operation as the economizer at the intermediate seasons, which is currently not in use.

As with the south part, the perimeter zone in the office area adopts four-pipe fan coil units together with the primary conditioned air, rich in the outside air, using the primary-air ducts for the past induction units. In the interior zone, the past dual-duct system with mixing boxes are in use as VAV system. Fans were originally suction vane-controlled, which is currently fixed and fans are inverter-controlled. .

Flow diagrams of AC-22 for the north part office area are shown in Figure 5.1

5-2. System operation in summer

The AC-22a, the cold deck, is in operation, while AC-22b, the hot deck, is off operation. Part of the return air to AC-22b is drawn to AC-22a through the bypass damper between AC-22a and AC-22b, mixed with the outside air supplied from AC-24 after primary processing and cooled by the cooling coil before supplied to the office interior zones. The air volume is controlled with the reference point temperature in the interior zone using the cold air damper before the mixing box and supplied from the breeze-line outlets.

5-3. System operation in winter

The outside air taken into AC-24 is humidified with the air washer, mixed with the return air through VD1, 50% open, and further humidified and cooled by water spray in AC-22a before supplied to the office interior zones. The AC-22b heats and humidifies (with steam when in use) the return air and supplies warm air. The VD2 shall be shut but is stick open to 100%.

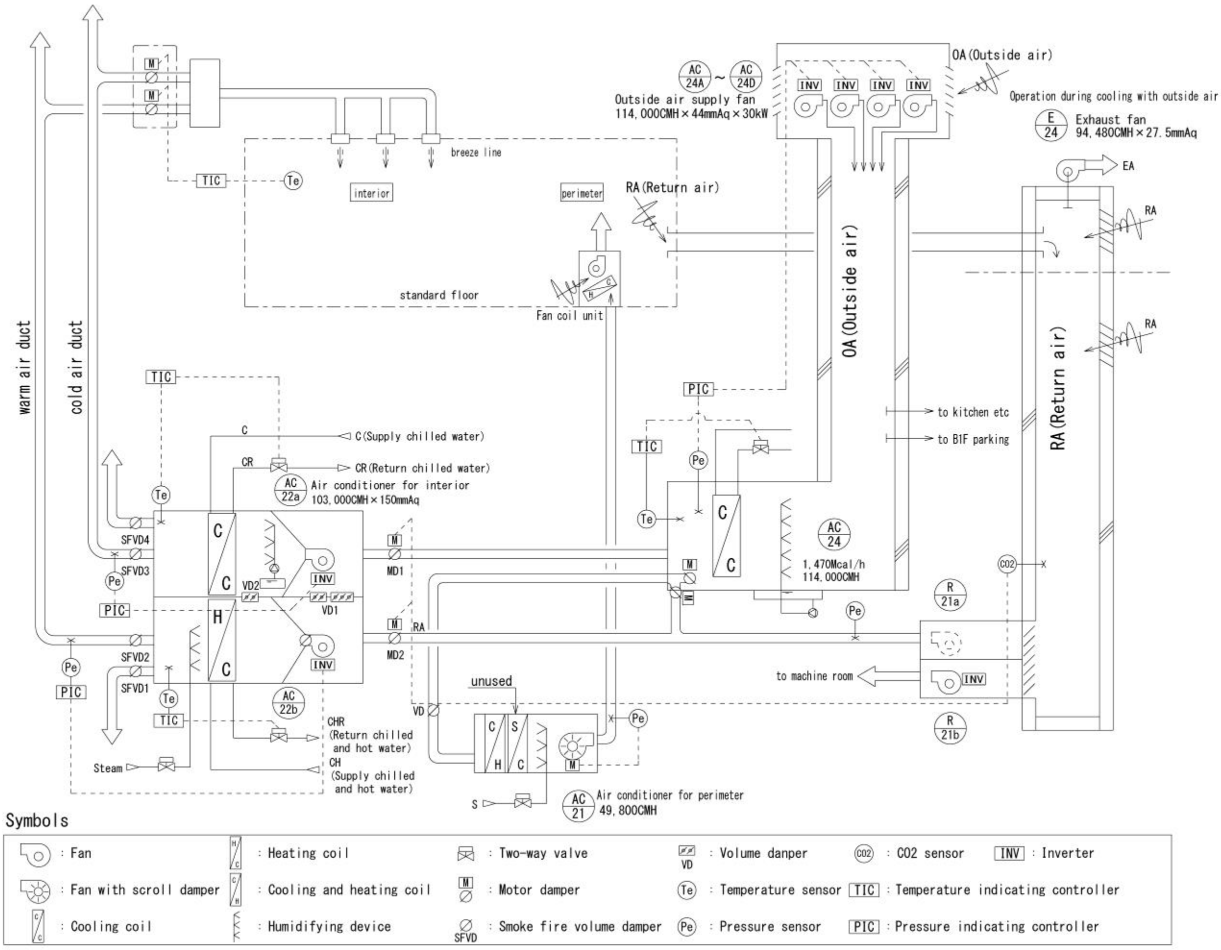

Figure 5.1 Flow diagram of duct system in north part.

6. EVALUATION OF ENERGY LOSS BY MIXING IN AC-22 BY ACTUAL MEASUREMENTS AND PROCESS ANALYSIS

6-1 Measurement data in summer and winter in FY 2013

Table 6.1 shows the measurement data of AC-22a and AC-22b. Both fans have the same specification with the performance curve as shown in Figure 6.1 with plotting for the estimated operating points.

Table 6.1 AC-22 simple measurement data in FY 2013

	At 11:00 on Tuesday, August 27, 2013 (in summer)		At 11:00 on Friday, January 17, 2014 (in winter)	
	AC-22a	AC-22b	AC-22a	AC-22b
INV Output	97% (58.2Hz)	Stopped	84% (50.4Hz)	70% (40.8Hz)
Air volume at the outlet duct	87,848 m³/h		56,968 m³/h	42,635 m³/h
Power consumption	70.0kW		46.8kW	20.6kW
Outlet static pressure	854Pa		798Pa	802Pa
5F MD cold air damper position	9% open		100% open	0%
Outlet temperature	12.4℃		13.4℃	36℃

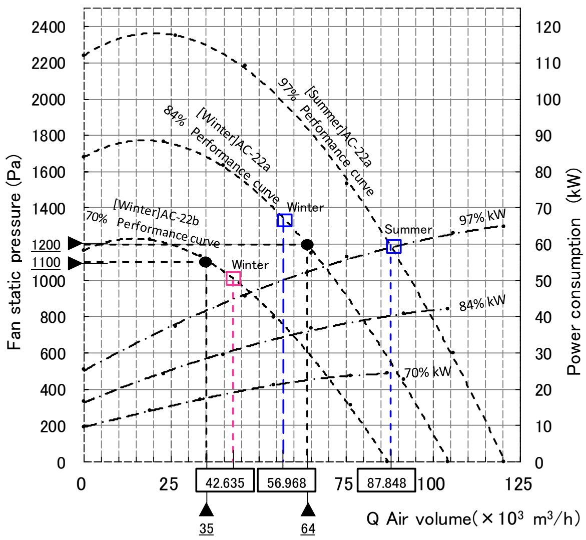

Figure 6.1 Operating points of AC-22a and AC-22b in FY 2013

Note 1: Circles (●) in Figure 6.1 represent operating points of the fans alone. Triangles (▲) represent operating air volumes calculated based on the fan's static pressure.

Note 2: Numbers in boxes ○○○ represent air volume measurement values at the outlet of AC-22.

6.2 Some measured process variables around AC-22 in winter

Figure 6.2 shows one day record of the delivery temperatures and fan inverter outputs of AC-22a and AC-22b, and supply air temperatures from breeze-line outlet on the 5th floor on Friday, January 17, 2014.

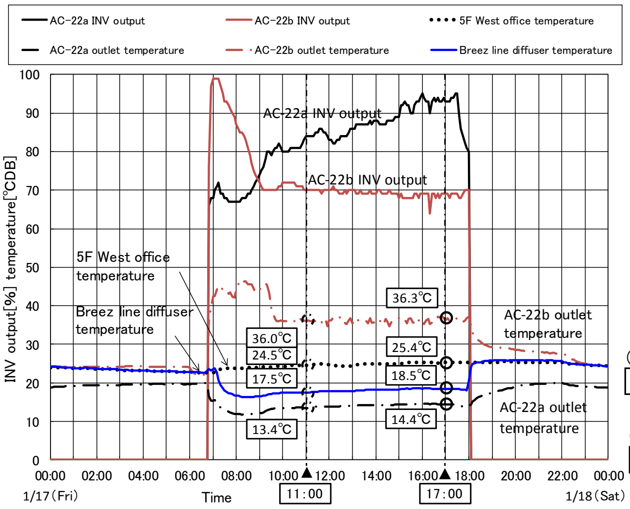

Figure 6.2 Delivery temperatures and inverter outputs of AC-22a, b and breeze-line outlet temperature on 5F

(1) AC-22 inverter output. The AC-22a inverter output is 70% just after start-up, and constantly increases to approximately 93% in the evening. The AC-22a outlet air temperature also increases gradually. On the contrary the AC-22b inverter output is stable at 70% throughout the day after 9:00.

(2) AC-22 outlet static pressure. The outlet static pressure of AC-22a and AC-22b that was recorded otherwise was stable at 800 Pa throughout the day.

(3) AC-22 power consumption. The inverter output almost matches the power consumption

Table 6.2 AC-22 operation status on January 17, 2014

	11 : 00		17 : 00	
	INV Output %	Power consumption kW	INV Output %	Power consumption kW
AC-22a	84	46.8	93	66.0
AC-22b	70	20.6	69	12.7

(4) AC-22 outlet temperature. The AC-22a outlet temperature is approximately 12–15°C during the daytime and approximately 19–20°C during the nighttime (fans are off operation.). The AC-22b outlet temperature is 43–46°C at the time of start-up, and is stable at approximately 36°C from 10:00.

(5) Cold air and warm air mixing damper (MD). The cold air MD opening at the time of start-up is unknown, but is almost 100% throughout the day. The warm air MD opening at the time of start-up is open 40% during 6:00–7:00, and is closed (opening: 0%) for the rest of the day. (Here, the damper opening refers to the controller output. Actually, it was disclosed that the minimal opening had been preset.)

6-3. Mixing inside AC-22 due to bypass dampers

The air volumes through the by-pass dampers VD1 and VD2 in AC-22 are estimated by calculating the air volumes of fans alone (derived from fan operating points based on the operating static pressure at the respective parts). Circles (●) shown in Figure6.1 represent operating points of AC-22a and AC-22b alone.

- [AC-22a] 64,000 m^3/h × 1,200 Pa
- [AC-22b] 35,000 m^3/h × 1,100 Pa

Figure 6.3 shows the AC-22 internal bypass air volume.

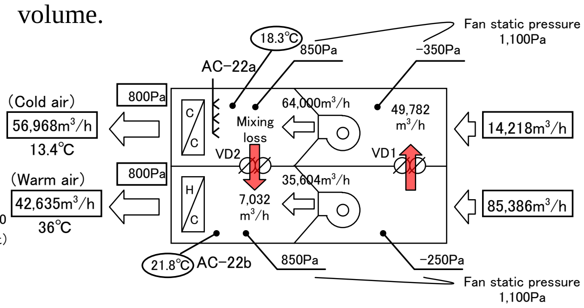

Figure 6.3 Air volumes at 11:00 on January 17, 2014

6-4. Evaluation of the energy loss by mixing

(1) Presumption of the balance of air volumes by measurement. In winter, the interior zones of office space of the typical floor as of the 5th floor need to be cooled, while the cafeteria on B2F and the bank on 1F need to be heated. As the minimal opening was preset at 10% for the control sequence of mixing dampers, the mixing energy loss between cold air and warm air stably exists, which causes excessive air to be fed to both AC-22a and AC-22b and excess heat and fan power are consumed.

Figure 6.4 shows the identified temperatures and air volumes around AC-22 at 11:00 on January 17, 2014.

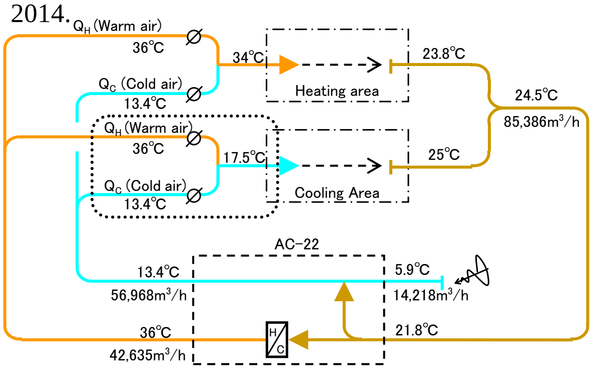

Figure 6.4 Temperatures and air volumes in the AC-22 system at 11:00 on January 17, 2014

Based on Fig. 6.4, the cold air/warm air ratio in the cooling/heating areas is as follows:

Cooling supply area: $Q_H = 0.22Q_C$

Heating supply area: $Q_C = 0.10Q_H$

Figure 6.5 shows the overall air volume balance calculated based on the above relational expression and cold/warm air fan outlet volume.

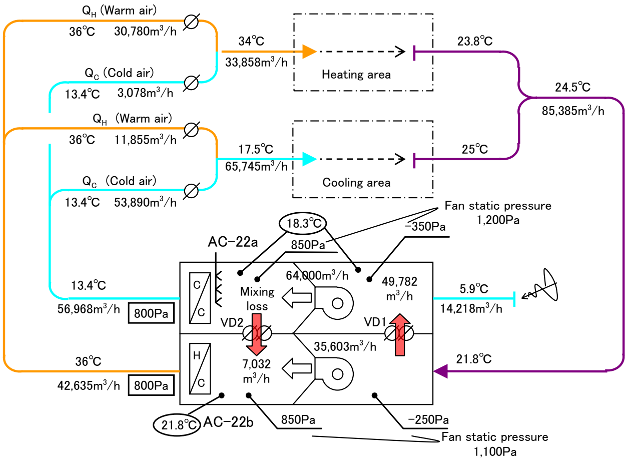

Figure 6.5 Air volume balance of cold air and warm air at 11:00 on January 17, 2014

(2) Presumption of air and heat balance in case of no mixing of cold and warm air. Based on the actual heat and air balance as shown in Figure 6.5, the air volume necessary to treat the net cooling and heating load when the breeze-line supply air temperature were assumed to be kept at the AC22 delivery temperature without any mixing is as follows:

• Cooling area:

65,745 × (25 − 17.5)/(25 − 13.4) = 42,508 m^3/h

Excess air volume: 56,968 − 42,508 = 14,460 m^3/h.

• Heating area:

33,858 × (34 − 23.8)/(36 − 23.8) = 28,308 m^3/h

Excess air volume: 42,635 − 28,308 = 14,327 m^3/h.

Furthermore, if the by-pass air through the VD2 could be completely eliminated, an ideal heat and air balance without mixing loss is estimated as figure 6.6

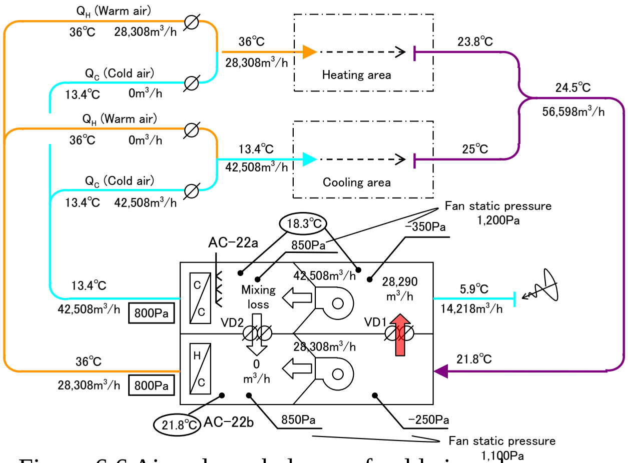

Figure 6.6 Air volume balance of cold air and warm air in case of no mixing

(3) Thermal energy loss by mixing. Because the economizer operation was applied, no energy was originally consumed for the cooling side, so that the thermal energy loss occurs only on the heating side. The difference of the heating energy between Figure6.5 and Figure 6.6 serves as the thermal energy loss due to the mixing of the cold air and warm air in this system as a whole.

• Amount of present heating:

(36°C – 21.8°C) × 42,635 m^3/h ×1.2 kg/m^3 × 1.006 kJ/kg ÷ 3,600 s/h = 203.0 kW

• Amount of heating when there is not mixture:

(36°C – 21.8°C) × 28,308 m^3/h ×1.2 kg/m^3 × 1.006 kJ/kg ÷ 3,600 s/h = 134.8 kW

• Thermal energy loss by the mixture
= Difference of the quantity of heating = 68.2kW

(4) Fan power energy loss by mixing. Amount of power consumption equivalent to the amount of air volume differences becomes the excessive energy loss. Supposing the fan efficiency × motor efficiency is 0.5,

• AC-22a fan :

14,460 m^3/h/3600 × 1,200 Pa/0.5 = 9.6 kW

• AC-22b fan :

14,327 m^3/h/3600 × 1,100 Pa/0.5 = 8.8 kW

• The total : 18.4kW

(5) Primary energy loss by mixing. Assuming the comprehensive primary system COP, which is defined as the heating and cooling load versus total primary energy consumed for HVAC system, is 0.7, primary energy mixing loss is estimated as follows.

• Primary thermal energy loss :
68.2kW/0.7=97.4kW×3.6MJ/kW=350.7MJ/h

• Primary energy loss of fan power consumption :
18.4kW×9.97MJ/kW=183.4MJ/h

• The total : 534.1MJ/h

However, it should be noted that this calculation is based on the heat/air balance on the representative day and time in winter (at 11: 00 on January 17, 2014). Although the energy loss during a specific period cannot be directly calculated, this analysis shall offer a valuable information concerning specific character of mixing energy loss of the dual-duct system for retrofit.

7. ENERGY MIXING LOSS BETWEEN INTERIOR AND PERIMETER ZONES AT SIMULTANEOUS HEATING AND COOLING

In the above-mentioned air-conditioning system, although the office area in the north part was air-conditioned, it was pointed out at the first stage of commissioning in FY 2012 that extensive mixing loss might be incurred in this area in winter between the interior zone and perimeter zone. This phenomenon is very popular but not well recognized, then authors are to analyze and evaluate the mixing loss hereafter.

7-1. Temperature measurement in the interior zone and the air-conditioning system on the typical floor

To analyze and evaluate the mixing loss in the interior zone and perimeter zone on the typical office floor, the temperature was measured in the west office area on 5F of the north part building for a week, i.e., from Tuesday, January 14 to Monday, January 20, 2014. Figure 7.1 shows the measurement data on Friday, January 17, 2014 (the representative day for measurement). Figure 7.2 shows the measurement points.

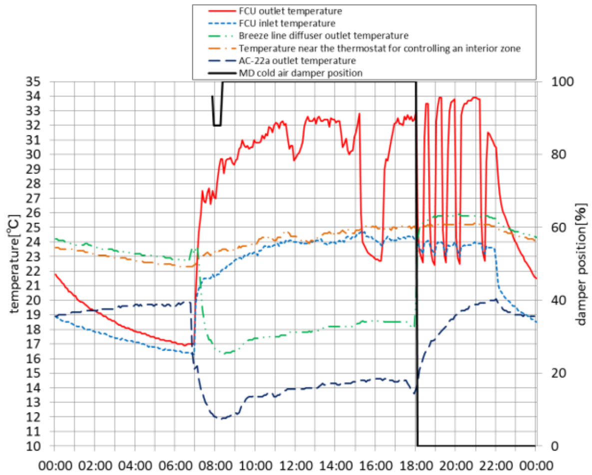

Figure 7.1 Temperature measurement on Friday, January 17, 2014 in the west office area on 5F of the north part

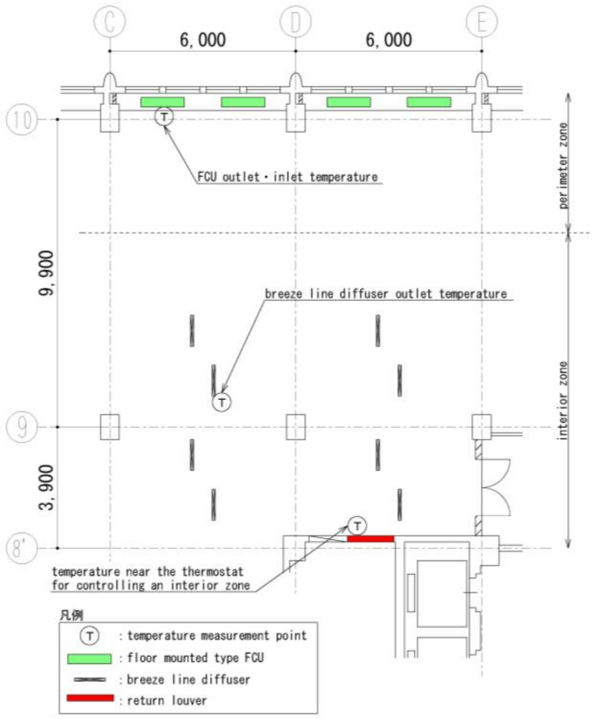

Figure 7.2 Temperature measurement points in the west office area on 5F of the north part

7-2. Definition of the mixing energy loss

When cold air is mixed with warm air at the boundary of the interior and perimeter zones, the load processing capacity is mutually offset and lost. This phenomenon is referred to as the mixing energy loss between the interior zone and perimeter zone.

Equation (1) shows the definition of the mixing loss ratio (MLR). Figure 7.3 shows the schematic diagram of the mixing energy loss.

$$MLR = \{(|Q_P| - |H_P|) + (Q_I - H_I)\}/(|H_P| + H_I), \quad (1)$$

H: Real heat load (subscript I: interior cooling load (+), P: perimeter heating load (-))

Q: Actual heat supply/removal (subscript I: interior cold heat (+), P: perimeter warm heat (-))

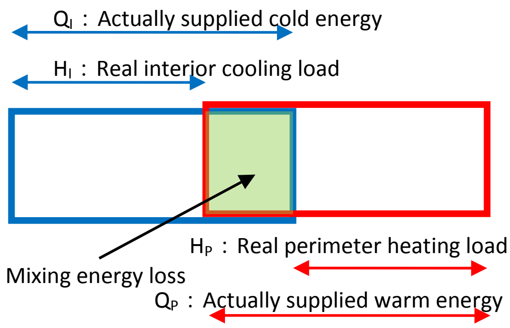

Figure 7.3 Mixing energy loss

7.3 Estimation of the status of mixing loss

Figure 7.1 shows that, during the measurement period, the perimeter zone is heated by FCU and AC-21, while the interior zone is cooled by AC-22a. The MD cold air is open 100% except during the start-up of the air-conditioning system. As discussed below, the interior room temperature was higher than the SP value and cooling capacity arrived at the high limit, while the perimeter zone temperature was closer to the SP value and abundant heating capacity even heated up the interior zone. Thus, it is evident that the mixing energy loss between the interior cold air and perimeter warm air took place and further, as the result of saturated cooling capacity, warmed up the interior zone above SP. Meanwhile, it should be noted that this is another story than the mixing phenomenon as introduced in the previous chapter.

7-4. Temperature control in the interior and perimeter zones

It has been reported in literature that the preset temperature in the interior and perimeter zones serves as a very important factor in calculating the mixing loss.[1) 2)] Table 7.1 below shows the preset temperatures for the interior and perimeter zones that were used for the study.

Table 7.1 Preset temperatures in each zone

	Set value	Remarks
Preset temperature in the interior zone	22.5 °C	Check the set value of MD (mixing dampers) from the central monitor.
Preset temperature in the perimeter zone	24 °C	Visual check of the set value of the northwestern FCU, the temperature of which were measured.

As shown in Figure 7.1, the inlet temperature of the FCU during operation is almost 24°C in the perimeter zone, which is almost the same as the preset temperature in the perimeter zone. Meanwhile, the temperature in the interior zone (in the vicinity of the thermostat for mixing damper control) is 24–25°C, which is approximately 1.5–2.5°C higher than the preset temperature. On the same day, the MD in this area was open 100% on the cold air side and 0% on the warm air side, which means the cooling capacity was at the maximum determined by the air volume and outlet air temperature difference. The heating preset temperature prevailed and raised the temperature in the interior zone, too, due to the excessive capacity in the perimeter zone by FCU and primary air. Thus, the cooling capacity limit prevented further mixing loss.

The temperature in the interior zone (which is used in the mixing loss estimation equation as described below) is the preset temperature that determines the cooling output (i.e., a mixing loss factor). Thus, if the preset temperature were 24°C, there would not be cooling needs to this extent, and mixing loss must have been prevented. With this in mind, 22.5°C was set as the preset temperature for the interior zone in the estimation equation.

7-5. Estimation of the mixing loss ratio (MLR)

The mixing loss is basically determined by the preset temperature difference between the interior and perimeter zones. There are some other relevant factors including the zone air change rate. According to a study conducted by Nakahara et al., [1)2)] the mixing loss ratio can be estimated using various significant factors (P: Depth-length, P: Outlet direction, I: Outlet air volume (air change rate), I-P: Preset temperature difference, Depth of hanging wall, Outlet Ar. number, I Thermostat position, wall or ceiling). The conditions of the full-scale experiment model used in the study by Nakahara et al. are not fully equivalent to those of the objective floor of the present building, but it has sufficient similarity except that the cooling capacity reached the limit in the present case. As no other applicable reference is found, this estimation table has been used.

Table 7.2 shows the substitution results using factor effect values for respective significant factors (based on the effect estimation table of the mixing loss ratio). According to Table 7.2, the mixing loss ratio (MLR) in winter is as follows:

MLR = +23 (−2) = 21% (± 27%)

where (−2 ± 27%) represents the average value ± confidence limit.

8. IMPROVEMENT MEASURES FOR OPTIMIZATION

8-1. Improvement measures for optimization around AC-22

(1) Preventing mixing thermal energy loss. By preventing mixture at the dual duct mixing dampers, and at the bypass-damper for the cold and warm air in AC-22, mixing thermal energy loss will be reduced by 68.2kw. However, outside air demand control should be carefully examined with this measure.

(2) Reducing excessive fan power consumption due to mixing. The minimum opening policy of mixing dampers to allow mixing of warm and cold air shall be examined to reduce bypassed air. Inside AC-22 VD2 bypass damper shall be completely closed to reduce air volume and fan power and to reduce excessive outside-air heating load. The power consumption can be reduced by 18.4 kW (total of AC-22a and AC-22b) in winter.

(3) AC-22 internal mixing loss. As shown in Figure 6.3, AC-22b internal mixing loss is attributed to the mixing of AC-22a fan outlet air (18.3°C) and AC-22b outlet air (21.3°C). The internal mixing loss can be reduced by 7.1 kW by closing VD2. (This effect is, however, included in the 68.2kW of item (1))

(4) Effective utilization of outside air (cold air) in the intermediate season as economizer. The cold/cool outside air supplied to the office floors is balanced with the exhaust air volume from the restrooms and kitchenettes, etc. Because these exhaust air volume are constant, it was difficult to increase the outside air volume. However, the E-24 fan is installed perhaps for this purpose, shall be then operated in the intermediate

Table 7.2 Factor effect of the mixing loss ratio (value substitution)

Factor		Factor level	Mixing loss ratio
A	P : Depth	Estimated at about 4.5 m	15
C	P : Outlet direction	P Outlet direction : 0º	4
F	I : Outlet air volume (air change rate)	Air volume in the interior is estimated at the rated value (8,230 m^3/h), as MD opening on the cold air side is 100%. Air change rate in the interior is about 5 times/h.	-8
G	I-P : Preset temperatures difference	-1.5ºC (I: 22.5ºC, P: 24ºC)	44
B × D Length of hanging wall × P Outlet Ar number		Hanging wall: None, Ar number: middle	-18
B × H Length of hanging wall × I thermostat position		Hanging wall: None I thermostat position: Wall	-22
C × D P Outlet direction × P outlet Ar number		P Outlet direction: 0ºC P Outlet Ar number: middle	8
		Total factor effect	23 (%)

season as the economizer, depending on the cold air requirements. When there are needs for cooling and outside air has less enthalpy than recirculated air, the outside air volume shall be increased. The control algorithm is available elsewhere as outside air cooling. An inverter is already installed to the E-24 fan to ensure this control.

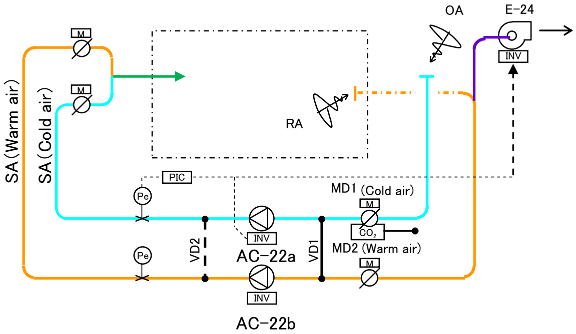

Figure 8.1 Conceptual image of control

(5) Re-installing the return fan and reducing the outlet static pressure settings for AC-22a and AC-22b.

It should be noted that the return fan became out of order several years ago and removed as shown in Figure 5.1, while the outside air is sent by plural OA fans. In winter, the operating pressure of AC-22 system is shown in Fig. 8-2. The suction pressure of AC-22b is -250Pa, higher than that of AC22a by 100 Pa, the resistance of the damper VD1, in order to pass the return air from AC-22b to AC-22a. The negative pressure was caused by removal of the return fan, R-21a (145,590m^3×300Pa×30kW), and resistance of MD1 becomes 380Pa to perform the pressure balance, resulting in the useless energy loss. The effect of energy saving by re-installing the return fan R-21a is calculated as follows. As corrected values are shown in Figure 8.2, pressure balance is corrected as for the resistance of MD1 to become as small as 80Pa. Supposing the same efficiency for the reinstalled fan and motor as AC22 fans, the fan power consumption before and after reinstallation is as follows.

Before (present condition)

- AC-22a fan :
  64,000 m^3/h/3600 × 1,200 Pa/0.5 = 42.7 kW
- AC-22b fan :
  35,603 m^3/h/3600 × 1,100 Pa/0.5 = 21.8 kW
- The total : 64.5kW

After (future condition)

- AC-22a fan :
  64,000 m^3/h/3600 × 900 Pa/0.5 = 32.0 kW
- AC-22b fan :
  35,603 m^3/h/3600 × 800 Pa/0.5 = 15.8 kW
- R-21a fan :
  85,385 m^3/h/3600 × 300 Pa/0.5 = 14.2 kW
- The total : 62.0kW

Reduction of the fan power consumption becomes 2.5kw. (= 64.5 – 62.0) However, this value changes with the amounts of introduced outdoor air and becomes larger if high efficient fan and motor are used.

In addition to energy saving effect, pressure balance of the relevant air-conditioning systems connected in parallel to the common outside air shaft and return air shaft is improved and controllability of each duct-fan system, will be stabilized as the byproduct.

Moreover, although the outlet static pressure is preset at constant value of 800 Pa for now, if optimal pressure setting control is introduced depending on the air- conditioning load change and resulting air volume change, far more energy saving will be realized. Thus, the reduction of fan power consumption of AC-22 systems is aimed at by carrying out these two methods.

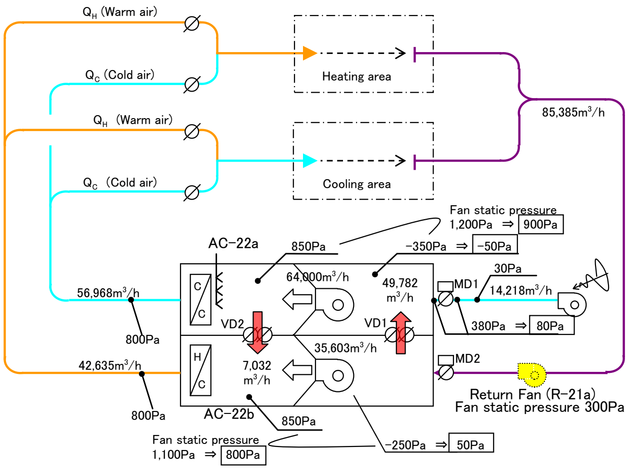

Figure 8.2 Static pressure and air volumes in the AC-22

Note 1: Numbers in boxes $\boxed{\circ\circ\circ}$ numerical value after re-installing the return fan.

8-2. Improvement measures to minimize the mixing energy loss between the interior and perimeter zones.

In Chapter 7, the mixing loss ratio was estimated as 21%. To reduce the mixing loss ratio, it is necessary to either reduce the difference in room temperature settings between the interior and perimeter zones or reverse the settings (i.e., lower the preset temperature in the perimeter zone than that of the interior zone), as is evident from Table 2 and its original estimation table[1) 2)].

Apart from the estimation table, it was observed that the interior cooling capacity reached to the upper limit with fully opened cooling damper and resulted raised room temperature. This suggest that limiting the excessive heating and cooling capacity is an effective measure to limit the mixing energy loss. To limit the interior cooling capacity optimal supply temperature setting control is effective. To reduce the perimeter heating capacity, most effective way in this case will be reduce primary air supply by AC-21, which has too excessive heating capacity and uncontrollable at the terminal.

Another most effective solution to exclude mixing energy loss would be to eliminate the need for simultaneous cooling and heating with the use of perimeter-less system. In realistic viewpoint, however, it is difficult to switch to the perimeter-less system in the existing Osaka-Gas Building. Described below are feasible solutions.

1) Reducing the excessive air conditioning capacity in the perimeter zone. a) Minimizing the primary air volume in the AC-21 system. The primary air of the AC-21 system not only introduces outside air and serves as a heat source but also humidifies rooms. An investigation must be conducted before stopping the primary air. b) Reducing the maximum capacity of FCU and AC-21 through the cascade control of the hot water supply temperature based on the outside air temperature. This solution can curb the excess mixing loss.

2) Lowering the preset temperature in the perimeter zone to the extent that the occupied zone is not affected, and raising the preset temperature in the interior zone taking the same care for comfort.

3) Change the mixing damper control schedule to eliminate minimum opening preset to exclude stable mixing of warm and cold air.

4) Introducing optimal delivery air temperature setting control for the AC-22a and b to limit excessive cooling effect and limit the mixing loss capacity.

5) Reducing the outlet volume of cold air in the interior zone in the vicinity of the perimeter zone, and preventing direct mixing loss with the perimeter zone

6) Lowering the cold air ventilation temperature from the current level to reduce the air change rate

7) Setting up control sensors at appropriate locations so that they can correctly detect the room temperature.

9. DISCUSSION

The commissioning process has been applied for the existing Osaka-Gas Building as a case study of research subject how to proceed retro-commissioning for the old and memorial kind of building with long history of building annex and several times of retrofits for energy conservation. The project was performed for two and a half years up to the former half of the inspection phase of retro-commissioning process. One of the principal works were to determine the performance of the current system, identify problems, review solutions, present the possibilities of optimizing the operation method, and to help achieve energy conservation in the future.

The inspection phase started in August 2012. After conducting hearings, scrutinizing the drawings, and performing on-site inspections, authors identified the performance based on the BEMS data as shown in the Part-2, analyzed the operating status of air and heat source systems based on added actual measurements, and then examined energy-saving effect of several ways of retrofit based on simulation study as shown in the Part 3.

Figure 9.1 shows the performance verification process flow in this study. Figure 9.1 shows the long-term improvement strategy for the air system and the basic concept of grading-up the energy plant now under consideration. Some energy-saving measures require extensive modification. Continuous renovation plans will be formulated in collaboration with the operation staff as the continuous commissioning process, that is, the combination of on-going commissioning by O&M +FM and re-commissioning by the third party.

This study is one of the result of research works by (NPO) Building Services Commissioning Association on the useful application of the commissioning process. Authors acknowledge the building owner and O&M staff of the Osaka Gas Building and member of the research committee for their assistance.

ABBREVIATION

- BEMS : Building and Energy Management System
- BSCA : Building Services Commissioning Association
- CGS : Cogeneration System
- CMT : Commissioning Managing Team
- FCU : Fan Coil Unit
- FM : Facility Management
- LCEM : Life Cycle Energy Management
- O&M : Operation & Maintenance
- SHASE : The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan
- VAV : Variable air volume

REFERENCES

1)-Nakahara, N. and Ito, H.: PREDICTION OF MIXING ENERGY LOSS IN A SIMULTANEOUSLY HEATED AND COOLED ROOM: PART 1- EXPERIMENTAL ANALYSES OF FACTORIAL EFFECTS, ASHRAE TRANSACTIONS, 1993. V. 99, Pt. 1

2)-Nakahara, N. and Ito, H.: PREDICTION OF MIXING ENERGY LOSS IN A SIMULTANEOUSLY HEATED AND COOLED ROOM: PART 2- SIMULATION ANALYSES ON SEASONAL EFFECTS, ASHRAE TRANSACTIONS, 1993. V. 99, Pt. 2

3)-Matsuda,N. Hatanaka,T. Aoki,K. Tanaka,H. et al. :Performance verification of an existing building which has the cogeneration system : Part.1 The way of application and evaluation of the Commissioning Process for existing system, Part.2 Performance verification of the energy system by BEMS data, Part.3 Evaluation of Operating Performance and Energy Efficiency for Air-conditioning System, Part.4Performance evaluation with system simulation by using BEMS measured data, The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, 2013.9

4) -Hatanaka.T. Aoki,K. et al., : Performance verification of an existing building which has the cogeneration system : Part.5 The evaluation and the measure against mixing loss between the zones of a north building standard floor, Part.6 The Mixed Energy Loss of the AC-22 Perimeter Air in the HVAC system for a typical Office Floor and the Countermeasures, The Society of Heating, Air-Conditioning and Sanitary Engineers of Japan, 2014.9

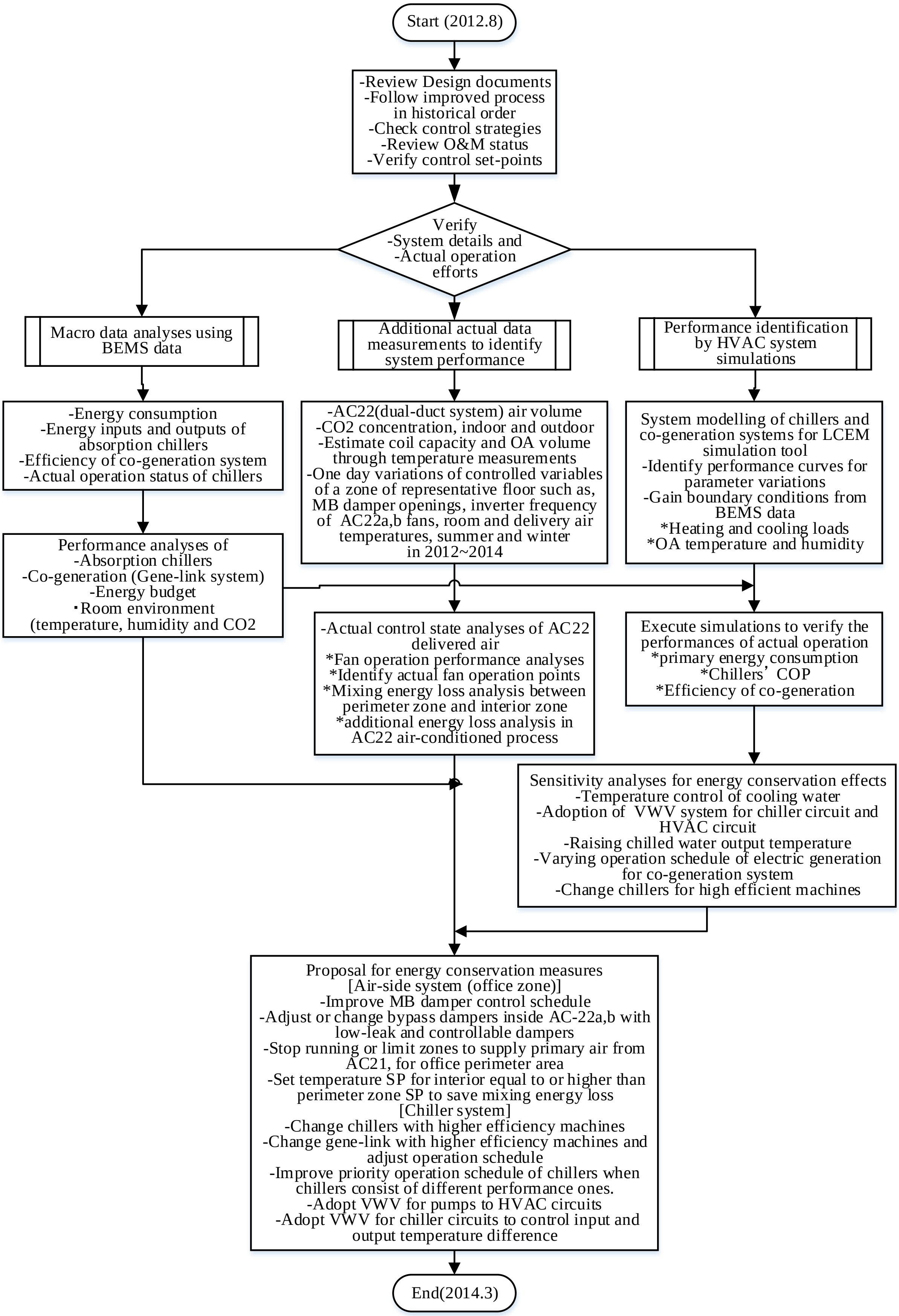

Figure 9.1 Performance verification process(2)