Building Simulation 2009

Eleventh International IBPSA Conference
Glasgow, Scotland
July 27-30, 2009

POTENTIAL FOR THE ENERGY EFFICIENCY ADVANCEMENT IN DISTRICT HEATING AND COOLING PLANT BY RENOVATION

Yoshiyuki Shimoda[1], Yoshitaka Uno[1], Yoshie Togano[1], Ryo Sunagawa[1], and Yutaka Shoji[2]
[1]Division of Sustainable Energy and Environmental Engineering,
Graduate School of Environment, Osaka University.
2-1, Yamadaoka, Suita, Osaka, Japan. e-mail shimoda@see.eng.osaka-u.ac.jp
[2]Energy Advance CO., Ltd., Japan
Kaigan 1-5-20, Minato-ku, Tokyo, Japan

ABSTRACT

Energy efficiency improvements of renovated district heating and cooling (DHC) plants were evaluated by simulation. In this paper, the simulation models for the original and renovated plants are developed based on the equipment specifications of the original plant. Accuracy of this model is examined based on the comparison of the measurement data from the operations of the original plant. From the result of this comparison, few parameters related to the chiller operational control and chiller efficiency were modified. In the final part of this paper, the model quantifies the expected total annual energy efficiency improvement and the contribution of each piece of replaced equipment.

INTRODUCTION

In Japan, district heating and cooling systems (DHC) have been in use for approximately 40 years and about 150 plants have been constructed as high efficiency heat supply systems at central business districts. DHC plants are classified based on the energy source into three categories: electric heat pump driven which runs on electricity; absorption chiller and boiler which uses natural gas; and systems that are combinations of these two types. Even within a single category, measured results show that energy efficiencies of DHC plants vary widely due to differences in the heat demand profile, efficiency of the heat source machines, system design, operation, and so on. However, the simulation study proved that DHC plants usually show higher energy efficiency than the conventional heat source systems in individual buildings because of the concentration effect of heat demand and the grade of operation (Shimoda et. al. 2008). In addition, in absorption chiller and boiler DHC systems, the introduction of combined heat and power (CHP) has a unique advantage for energy efficiency improvement.

In recent years, the energy efficiency of chillers, electricity generators, and pumps related to the DHC systems have seen remarkable progress. This means that there is a potential for remarkable improvements in the total energy efficiency of DHC systems by renovating the plant. For plants with an absorption chiller and boiler with CHP: 1) introduction of large-scale, high efficiency electricity generators, such as gas engines; and 2) high efficiency turbo refrigerators which enables the CHP system to operate for longer hours, are expected to increase the energy efficiency of the DHC system significantly (Kubara et. al. 2007). In addition, improvement of the operation also affects the efficiency of the DHC plant (Wang et. al. 2007, Ono et. al. 2007).

A plant referred to as A-plant, which has an absorption chiller and boiler with CHP, was chosen for a case study in this paper. The plant that was originally constructed in 1992 was renovated in 2008. In this paper, the simulation models for both the original and the renovated plant are developed based on the equipment specifications of original plant. The accuracy of this model was examined based on the comparison with data measured during the operation of the original plant. From this comparison, some parameters related to chiller operation sequence control and chiller performance were modified. Using this simulation model, the expected total annual energy efficiency improvement and the contribution of each piece of equipment replaced were quantified.

OUTLINE OF THE DHC PLANT AND RENOVATION WORKS

A-plant is located in central Tokyo, Japan. It was constructed in 1992 and has 22.6 MW cooling capacity and 30.1 MW heating capacity for supplying heat to four buildings including a hospital, an office and an apartment.

Initially, this plant had two gas-engine CHP systems. They supplied electricity to the hospital, and waste heat, which consists of low-pressure (0.09 MPa) steam and high-pressure (0.78 MPa) steam, to the DHC plant. Three gas-fired boilers produced high-pressure steam. Cooling heat was produced by two single- and double-effect absorption chillers, which consumed both high- and low-pressure steam, and four double-effect absorption chillers, which consumed high-pressure steam.

On renovation, the CHP system was replaced with a highly efficient gas engine generator. It supplied electricity to the hospital, and waste heat, which consists of high-pressure steam and hot water (88°C), to the DHC plant. The single- and double-effect absorption chillers were replaced with one high-efficiency waste heat utilization absorption chiller, which consumes hot water and/or high-pressure steam, and one high-efficiency variable speed turbo chiller. Two of the four double-effect absorption chillers were replaced with brand-new high efficiency models. Table 1 lists the heat source equipments used before and after renovation. In addition, two cold water pumps and two cooling water pumps were replaced with variable speed inverter driven pumps.

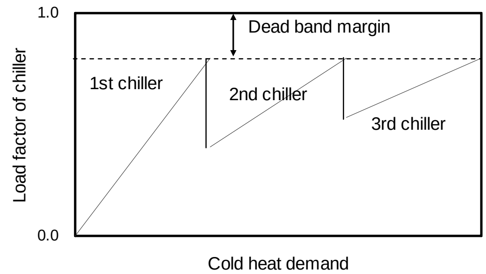


Figure 2. Dead band margin

Table 1. List of heat source equipments

	BEFORE RENOVATION	**AFTER RENOVATION**
Gas engine generator	480 kW × 2 η = 29.0% (elect.) η = 20.7% (l-p.s.) η = 15.7% (h-p.s.)	930 kW η = 36.2% (elect.) η = 16.6% (hot.w) η = 13.4% (h-p.s.)
Boiler	11,280 kW × 2 7,520 kW × 1 (η = 0.83)	Not replaced
Chiller 1	Double-effect absorption chiller 4,747 MW × 4 COP = 1.23	Double-effect absorption chiller 4,220 MW COP = 1.51 (steam base)
Chiller 2		Not replaced
Chiller 3		Same as chiller 1
Chiller 4		Not replaced
Chiller 5	Single & double-effect absorption chiller 1,758 kW × 2 COP = 1.23	Waste heat utilization absorption chiller COP = 1.43
Chiller 6		Variable speed turbo chiller COP = 5.5

Efficiencies are expressed as HHV base.

SIMULATION MODEL

Numerical models were developed for simulating the energy consumption of the A-plant before and after renovation. Figure 1 shows the flowchart of the model. The time step of this model is one hour.

The chiller sequence control model determines the chillers to be operated to meet both the required heat load and flow rate. When the gas engine is operated, at least one chiller that uses waste heat (hot water or low-pressure steam) is set to be in operation. In the determination of the chiller operation from the cooling demand, there are two constraint parameters: The *dead band margin* (Johnson, 1985) as shown in Fig 2 and the *minimum flow rate of the bypass pipe* to provide for the sudden increase in the demand (Shimoda et. al. 2008). These two constraints are not related, since the temperature difference between the supply and return water usually differ from the designed fixed value. When the total cooling load is small, the temperature difference becomes small and the flow rate constraint prevails over the heat demand constraint. The chiller operating order is set as in the original plan of the A-plant.

The model for gas engine performance calculates the consumption of city gas and waste heat (hot water and steam) generation from the electrical demand of the hospital. The model considers part load efficiency.

The chiller performance model calculates the steam and electricity consumption of each chiller from the part load factor and cooling water temperature. The

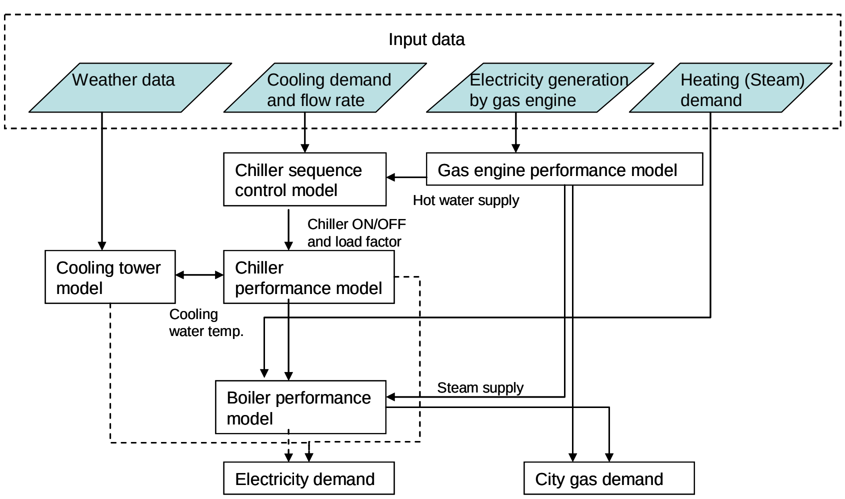


Figure 1. Flowchart of the simulation model.

function to determine the chiller coefficient of performance (COP) from the part load factor and cooling water temperature is set from the catalogue provided by the manufacturer. Electricity consumed by the cold-water pump and the chiller's accessories is also calculated. Figure 3 shows COP variation by part load factor and cooling water temperature for double-effect absorption chiller installed at renovation.

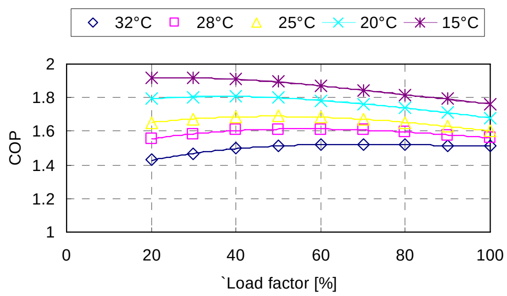


Figure 3. COP variation of double-effect absorption chiller (Brand-new model installed at renovation)

Iterative calculation between the chiller performance model and cooling tower model determines the cooling water temperature. The cooling tower model simulates mass and heat transfer between cooling water and outdoor air. Electricity consumption of the fan and cooling water pump is also calculated.

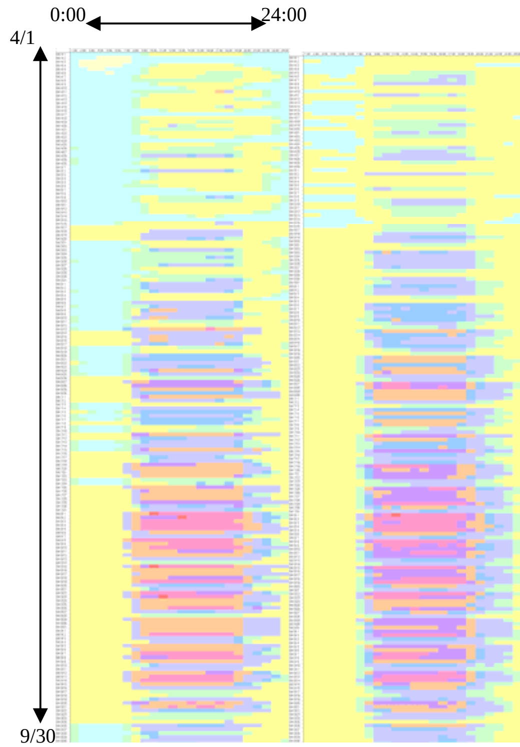
The boiler performance model calculates the city gas and the accessories' electrical consumption. The boiler part load efficiency was derived from plant measurements.

SETTING PARAMETERS WITH THE OLD SYSTEM SIMULATION

To determine the parameters used in the simulation model, actual operating data from April 2005 to March 2006, was compared with simulation results for the plant before the renovation.

Four types of simulation were demonstrated as follows:

- Run-1: Base case simulation
- Run-2: The chiller sequence is not simulated, but a measured chiller sequence is used as as the input data.
- Run-3: Considers the degradation of double-effect absorption chillers.
- Run-4: Change the dead band margin and minimum flow rate of the bypass.



No.of chillers	500RT*1	500RT*2
1350*0	1	2
1350*1	3	4
1350*2	5	6
1350*3	7	8
1350*4	9	10

Figure 5. Chiller operation status
(Left: Measured, Right: run-1)

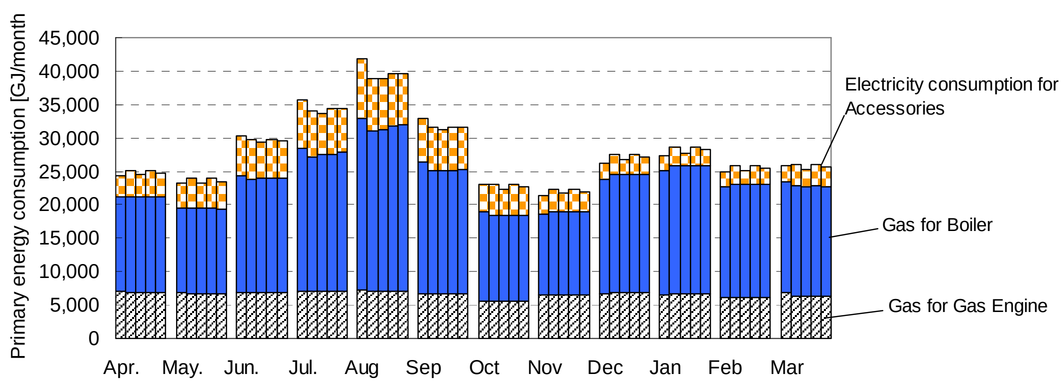


Figure 4. Comparison of monthly variation of energy consumption between measured and simulation result
(From left to right, Measured data, Run-1, Run-2, Run-3, Run-4)

Evaluation of the base case result

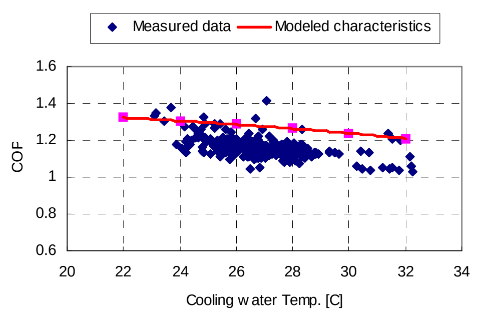
By setting the dead band margin at 5% and the minimum flow rate of the bypass pipe at 86.4 m^3/h, which is the value measured at the other plant, a simulation was carried out as the base case (Run-1). Figure 4 shows a comparison of the measured energy consumption and simulated values. Run-1 showed good agreement with the measured values. For example, the difference in monthly boiler gas consumption is less than 7%.

Figure 5 shows the comparison of chiller operation status between the measured data and the simulation. As shown, the number of chillers in actual operation is often smaller than the simulated result.

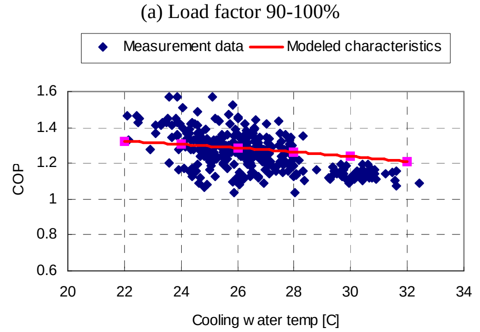
The difference in electrical consumption for accessories is larger than that for gas consumption, since the measured electricity consumption contains various kinds of electricity consumption in the DHC plant, such as plant ventilation, that are not considered in the simulation. The simulation considers only the cold-water pump, cooling water pump, cooling tower fan and the accessories of the absorption chiller and boiler.

Degradation of the double-effect absorption chiller.

In Run-2, the chiller operation status is the same as the actual condition shown in Fig. 5. Therefore, the difference in the gas consumption of the boiler and gas engine represents the difference in the efficiency of the boiler, absorption chillers, and the gas engine. The difference in the gas consumption between the

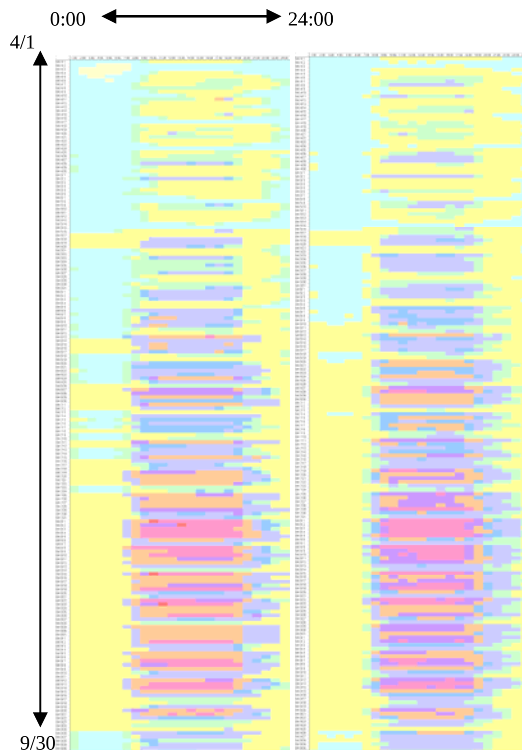


(a) Load factor 90-100%



(b) Load factor 60-70%

Figure 6. Comparison between modelled COP and Measured value.



No.of chillers	500RT*1	500RT*2
1350*0	1	2
1350*1	3	4
1350*2	5	6
1350*3	7	8
1350*4	9	10

Figure 7. Chiller operation status
(Left: Measured, Right: run-4)

measured value and simulated value for the boiler is large in summer. This is thought to be due to the degradation of the double-effect absorption chiller, which is operated more frequently in summer.

Figure 6 compares the modelled and measured COP of the double-effect absorption chiller. The data was selected from two partial load ranges. Agreement between the measured and simulated COP is not good, especially in low-load factor, due to the difficulty of steam flow-rate measurement. However, the COP decreased for the high-load factor, therefore, the COP for the 80-100% part load factor is decreased by 10% from the original value. Figure 4 shows that the errors of Run-3 become smaller than Run-2 during the summer.

The chiller control sequence.

To improve the discrepancy in chiller operation status, as shown in Fig. 5, in Run-4 the dead band margin are set to 10% and the minimum flow rate of the bypass is set to 0 m^3/h. Figure 7 shows a comparison of the chiller operational status. The simulated result comes closer to the measured condition. However, small differences still exists at midnight. At this time, under actual conditions, only one small single-double effect absorption chiller is in operation, even if both the flow rate and cooling load exceed the rated capacity. This is because the chiller capacity increases as the cooling water temperature decreases. The pump can also operate at a larger flow rate than the designed volume, since the pump's actual capacity is usually larger than the designed value. This energy saving operation is based on the decisions made by an experienced operator, and is not considered in our model.

Figure 4 also shows the results for the energy consumption. A comparison of measured energy consumption and that simulated from Run-4 is shown in Table 2. The data shows good agreement between the simulation and measured values.

Table 2.
Comparison of measured and simulated data.

[GJ/YEAR]	MEASURED	RUN-4	ERROR
Gas for gas engine	80,126	79,478	−0.8%
Gas for boiler	205,428	204,008	−0.7%
Electricity for Accessories	51,604	50,907	−1.4%
Total	334,393	337,158	−0.8%

SIMULATING THE RENOVATED PLANT

Comparison between the actual data and simulation

Using the renovated plant configuration and the revised parameters derived in the previous section, simulated energy consumption and chiller operation status were compared with the actual operating data from December 2008. Energy consumption of variable speed pump was modelled as the function of the flow rate from the catalogue. The chiller operating order was planned to switch among three patterns:

1. When the gas engine is in operation (daytime on weekdays and Saturday)

   Chiller 5 → 6 → 1 , 3 → 2 , 4
2. When the gas engine is suspended

   Chiller 6 → 1, 3, 5 → 2 , 4
3. Daytime on weekdays in July and August (peak-cut mode)

   Chiller 5→1, 3 → 2, 4 → 6

The daily energy consumption from the measured data and simulation are compared in Fig. 8 and Table 3.

Table 3. Comparison between measured and simulated data.(Dec. 2008)

[GJ/MONTH]	MEAS-URED	SIMUL-ATION	ERROR [%]	RMSE [%]
Gas for gas engine	3,426	3,496	−2.1	0.2
Gas for boiler	13,145	12,580	4.3	0.9
Electricity for Accessories	3,137	2,059	34.4	46.7
Electricity for turbo chiller	876	779	11.1	77.4
Total	20,583	18,914	8.1	1.7

Overall, the simulated energy consumption is slightly smaller than the measured value. In particular, the difference for the accessories and the turbo chiller are large. One reason is that the chiller operating status differs slightly from the plan since this period was immediately after the completion of the renovation. If the chiller operation status was set for the actual conditions, the turbo chiller difference becomes smaller, as shown in Table 4. The reason for the difference in electricity consumption of the accessories is thought to be the same as that before renovation

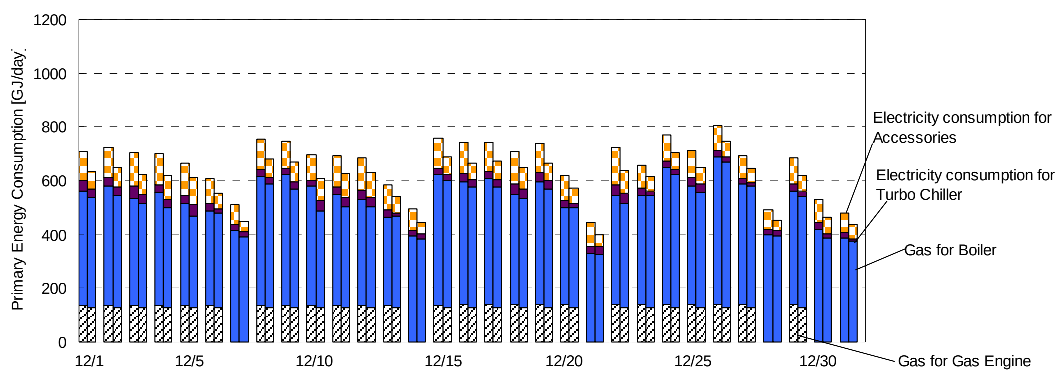


Figure 8. Comparison of daily variation in energy consumption between measured data and simulation results.
(Left: measured data, right: simulated results)

Table 4. Comparison between measured and simulated data (chiller operating status is the same as actual conditions).

[GJ/MONTH]	MEAS-URED	SIMUL-ATION	ERROR [%]	RMSE [%]
Gas for gas engine	3,426	3,496	−2.1	0.2
Gas for boiler	13,145	12,700	3.4	0.6
Electricity for Accessories	3,137	1,904	39.3	67.8
Electricity for turbo chiller	876	847	3.3	11.1
Total	20,583	18,947	8.0	1.6

Prediction of the annual energy consumption of the renovated plant.

Using the model that simulates the plant after renovation, the annual plant performance was predicted, based on the heat demand profile and weather conditions from April 2005 to March 2006.

Energy efficiency of the plant is expressed as an *EER* (Energy efficiency ratio), defined as the following equations:

$$EER = \frac{Q}{G_b + E \cdot e_p + G_w} \qquad (1)$$

$$G_w = \frac{Q_w}{Q_w + E_g \cdot e_p} \cdot G_g \qquad (2)$$

Figure 9 shows the simulated results of the monthly EER before and after renovation. The annual total EER increased from 0.68 to 0.89 from the renovation. The difference between before and after renovation is large in the intermediate seasons (spring and autumn) because the ratio of cooling heat produced by the replaced high-efficiency chiller becomes large in these seasons, while the total cooling demand is small.

Annually averaged hourly variation of EER is shown in Fig. 10. Before renovation, the nighttime EER was significantly lower than daytime, since the gas engine is off in this period. On the other hand, in the renovated plant, the early morning and midnight EER did not decrease much. One reason is that the COP of the variable speed turbo chiller increases, since both the cooling water temperature and load factor are low in this period. Figure 11 shows the COP characteristics of the variable speed turbo chiller.

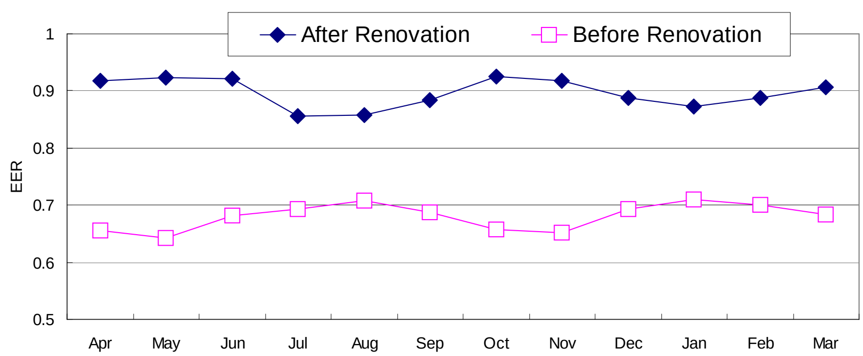


Figure 9. Simulated monthly EER at before/after renovation

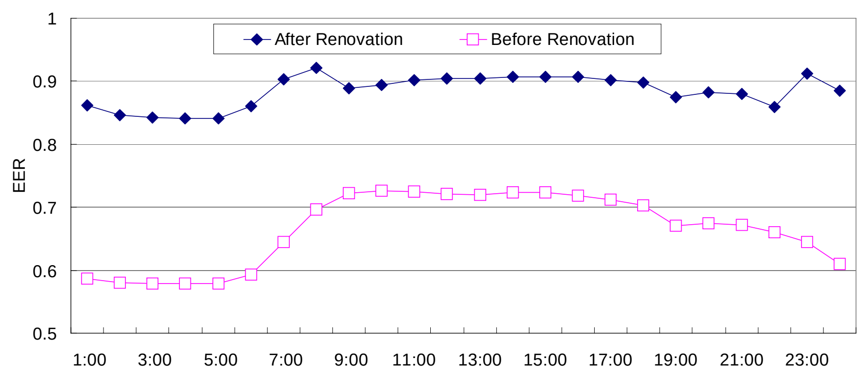


Figure 10. Simulated hourly change of annually-averaged EER at before/after renovation

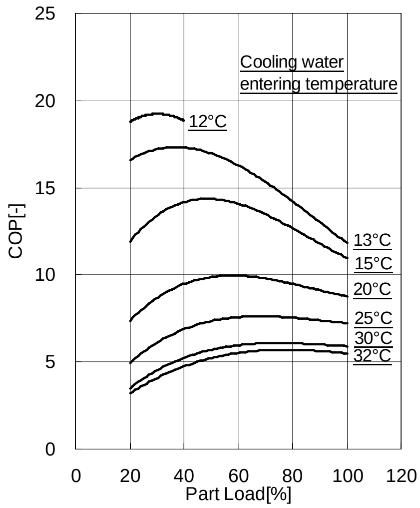


Figure 11. COP characteristics of variable speed turbo chiller

Equipment specific energy savings

To clarify the energy efficiency improvements for the specific replaced equipment, the energy balance before and after renovation are compared for each segment.

- Gas engine

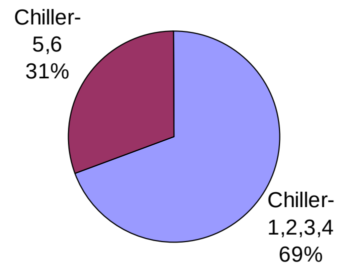
In this paper, the efficiency of the gas engine is expressed as the CHP exhaust heat coefficient (Kubara et. al, 2007). This is defined as:

$$\eta_{CHP} = \frac{Q_w}{G_g - E_g \cdot e_p} \qquad (3)$$

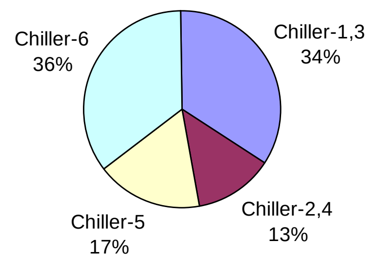
Table 5 shows a comparison of the energy balance before and after renovation. When η_{CHP} is larger than the boiler efficiency (= 0.83), the energy efficiency of CHP is superior to the conventional boiler and power generation plant. Since the power generation efficiency of the new gas engine is close to that of a conventional power generation plant (= 0.369), η_{CHP} increases drastically (14.7 times larger than before renovation).

Table 5.
Comparison of gas engine performance.

	BEFORE	AFTER	DIFFER-ENCE
Gas for gas engine [GJ/year]	79,478	40,527	51.0%
Electricity generated [GJ/year]	20,909	14,670	70.2%
Waste heat utilization [GJ/year]	25,275	12,521	49.5%
CHP exhaust heat coefficient	1.108	16.287	1470%



(a) before renovation



(b) after renovation

Figure 12. Share of cold heat production

- Chiller performance

Figure 12 shows the share of cold heat production. Table 6 shows the comparison of chiller performance for before and after renovation. Before renovation, single and double-effect absorption chillers (seasonal COP = 1.28) produced 31% of the cold heat, and double-effect absorption chiller driven by steam (seasonal COP = 1.20) produced the other 69%. After renovation, a variable-speed turbo chiller produces 36% of the cold heat and its seasonal COP is 9.5 in the secondary energy base (3.51 in the primary energy base). High-efficiency, double-effect absorption chillers (seasonal COP = 1.59) produced 34% of the cold heat.

- Boiler efficiency

Table 7 shows the comparison of boiler efficiency for before and after renovation. Since the turbo chiller, which uses electricity, produce 36% of the cold heat, boiler steam demand is decreased by 25% after renovation; even the use of waste heat from gas engine decreased by 50%.

Table 6.
Chiller performance before and after renovation

	Steam Consumption	Cold Heat Production	COP
Before renovation			
Chiller-1, 2, 3, 4 Double effect absorption	70,705 [GJ]	84,696 [GJ]	1.20
Chiller-5, 6 Single-Double effect absorption	29,275 [GJ]	37,588 [GJ]	1.28
After renovation			
Chiller-1, 3 High efficiency double effect	26,242 [GJ]	41,749 [GJ]	1.59
Chiller-2, 4 not replaced	12,994 [GJ]	15,506 [GJ]	1.19
Chiller-5 waste heat utilization	16,044 [GJ]	21,289 [GJ]	1.33
Chiller-6 Variable speed chiller	4,596 [GJ] (electricity)	43,740 [GJ]	9.5

Table 7. Boiler efficiency

	BEFORE	AFTER	DIFFER-ENCE
Gas consumption [GJ/year]	212,964	159,522	74.9%
Steam generation [GJ/year]	176,718	129,652	73.4%
Efficiency	0.830	0.813	97.9%

- Electricity consumption of accessories

Table 8 shows the electrical consumption of accessories, which also decreased by 26%. This is because of the introduction of variable speed pumps and a decrease in cooling water volume from using a turbo chiller.

Table 8. Electricity consumption of accessories

	BEFORE	AFTER	DIFFERENCE
Electricity consumption	6,465 [MWh]	4,803 [MWh]	74.3%

CONCLUSION

This study predicted the energy savings from renovation of a DHC plant. Simulation results showed energy efficiency of the plant increased by 31%. The next step is further validation of the model, using annual performance data of the renovated plant, and operational optimization.

NOMENCLATURE

E: Total electricity consumption of the plant [kWh]

E_g: Generated electricity by gas engine [kWh]

e_p: Conversion factor to primary energy [= 9,760 kJ/kwh]

G_b: Gas consumption of boiler [GJ]

G_g: Gas consumption of gas engine [GJ]

G_w: Gas consumption of gas engine for producing waste heat [GJ]

Q: Total heat supply (cold water and steam) [GJ]

Q_w: Waste heat used in DHC plant [GJ]

η_{CHP}: CHP exhaust heat coefficient

ACKNOWLEDGEMENT

This work is collaboration research between Energy Advance Co. Ltd. and Osaka University. In addition, a part of this work is also supported by a Grant-in-Aid for Exploratory Research awarded by the Japan Society for the Promotion of Science, No. 20656088

REFERENCES

Johnson GA. 1985. Optimization Techniques for a Centrifugal Chiller Plant Using a Programmable Controller. ASHRAE Transactions, 91(2), pp.835-847

Kubara, R. Shimoda, Y. Nagota, T. Isayama, N. and Mizuno, M. 2007. Prediction about Progress in Performance of District Heating and Cooling System Using Combined Heat and Power, Proceedings of Tenth International IBPSA Conference, Beijing China, pp.569-574

Ono, E. Yoshida, H. Wang, F. and Shingu H. 2007. Study on Optimizing the Operation of Heat Source Equipments in an Actual Heating/Cooling Plant Using Simulation, Proceedings of Tenth International IBPSA Conference, Beijing China, pp.580-587

Shimoda, Y. Nagota, T. Isayama, N. and Mizuno M. 2008. Verification of Energy Efficiency of District Heating and Cooling System by Simulation Considering Design and Operation Parameters, Building and Environment, 43, pp.569-577

Wang, F. Yoshida, H. Ono, E. and Shingu H. 2007. Methodology for Optimizing the Operation of Heating/Cooling Plants with Multi-Heat Source Equipments Using Simulation, Proceedings of Tenth International IBPSA Conference, Beijing China, pp.529-536