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Article

Analysis and Assessments of Combined Cooling, Heating and Power Systems in Various Operation Modes for a Building in China, Dalian

Miao Li, Hailin Mu * and Huanan Li

Laboratory of Ocean Energy Utilization and Energy Conservation of Ministry of Education, Dalian University of Technology, Dalian 116024, China; E-Mails: limiaolaile@163.com (M.L.); huanan3721@163.com (H.L.)

* Author to whom correspondence should be addressed; E-Mail: hailinmu@dlut.edu.cn; Tel.: +86-411-8470-8095; Fax: +86-411-8470-8095.

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Abstract: Combined Cooling, Heating and Power (CCHP) systems have been widely used in different kinds of buildings to make better use of fuels because of their high overall efficiency. This paper presents a mathematical analysis of a CCHP system in comparison to a Heating, Ventilation and Air Conditioning (HVAC) system. The operation strategies following electric load (FEL), thermal load (FTL) and a hybrid electric-thermal load (FHL) are proposed and investigated in this study. Criteria, namely primary energy saving (PES), exergy efficiency (η_{exergy}), and CO_2 emission reduction (CER) are defined to evaluate the performances of CCHP systems for a hypothetical building located in Dalian (China). The results indicate that: (1) a new mathematical foundation is established to find whether the recovered thermal energy and the amount of electricity generated by the power generation unit (PGU) are enough to provide the energy required; (2) the CCHP system does not always perform better than a HVAC system from an instantaneous perspective, especially in FTL mode; (3) the CCHP system in FEL operation mode can be seen as a suitable energy system in Dalian from the annual performance perspective. Furthermore, a sensitivity analysis is presented in order to show how the performances vary due to the changes of various technical variables.

Keywords: combined cooling heating and power (CCHP) system; evaluation criteria; various operation modes; China

Nomenclature

CCHP	Combined cooling heating and power
HVAC	Heating, ventilating, and air conditioning
PGU	Power generation unit
PEC	Primary energy consumption
PES	Primary energy saving
FESR	primary energy saving ratio
CE	CO_2 emissions
CER	CO_2 emissions reduction ratio
FEL	Following electric load
FTL	Following thermal load
FHL	Following hybrid electric-thermal load

Symbols

E	Electricity consumption
EX	Exergy
F	Fuel
Q	Heat consumption
T	Temperature
A	exergy coefficient
η	efficiency
α	Conversion factor
γ	the coefficient of the supplementary fuel
δ	the coefficient of the additional electricity
ρ	Electricity-heat constant

Subscripts

e	Electricity
grid	Electricity grid
pgu	Power generation unit
building	Reference building
grid	Electricity grid
p	pump
b	Boiler
c	Cool
h	heat
ch	Absorption chiller
hc	Heating coil
rec	Waste heat recovery system
r	Recovery heat
req	require
exergy	The exergy of each type energy
f	fuel

1. Introduction

Combined cooling heating and power (CCHP) systems, also called trigeneration systems, are broadly defined as an energy supply system that generates electricity near the load. The main difference between CCHP systems and the typical methods of electric generation is the utilization of the waste heat rejected from the PGU. CCHP systems have the potential to utilize fuels with an overall efficiency of 70%–85% [1,2].

The performances of CCHP systems are closely related to their operation mode strategies [3,4]. Some researchers [5,6] have investigated the operation modes of CCHP systems under two basic modes: following the electric load (FEL) and following the thermal load (FTL). However, these two basic modes do not provide optimal operation of a CCHP system. A new operation mode following a hybrid electric-thermal load (FHL) was proposed and investigated by Mago, whose results showed that CCHP systems operated following the hybrid electric-thermal load have better performance than CCHP-FEL and CCHP-FTL [7].

In addition to the operation modes, it is necessary to apply evaluation methods to guarantee the better performance of CCHP systems over conventional technologies. In the past research, a significant amount of effort has been made by researchers using minimization of economic cost as a criterion to evaluate CCHP systems [8–12]. However, if a CCHP system project is not economically feasible, other benefits from CCHP technology such as primary energy savings and emissions reduction could overcome the economic weakness. Many studies have been reported on this topic. A new generalized performance indicator—the trigeneration primary energy saving (TPES)—is introduced and discussed by Gianfranco and Pierluigi [13] with the aim of effectively evaluating the primary energy savings from different CCHP alternatives. Li *et al.* [14] used fuel energy savings ratio (FESR) to evaluate the operation of CCHP systems. Nelson *et al.* [15] defined Building Primary Energy Ratio (BPER) as a parameter to guarantee better energy performance of a CCHP system. Other researchers have investigated CO_2 emissions reduction as a criterion to evaluate the performance of CCHP system, like Kari and Arto [16], Nelson *et al.* [17], Gianfranco and Pierluigi [18], Monica *et al.* [19], Lund *et al.* [20], Minciuc *et al.* [21], Pierluigi and Gianfranco [22], Neil and Alexander [23].

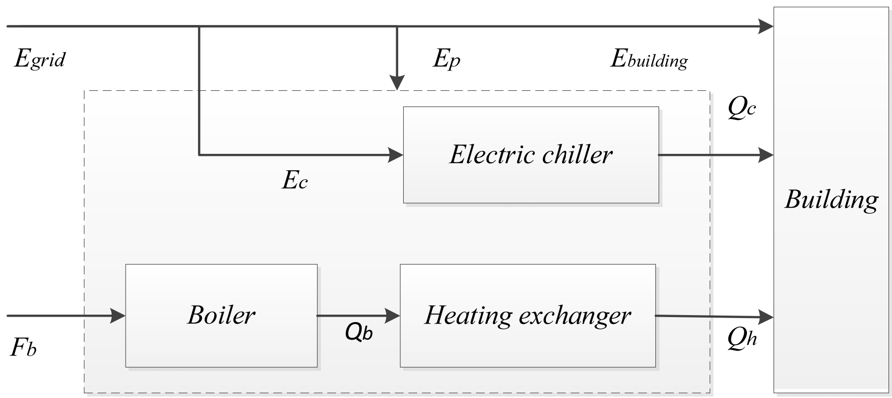
In this paper, a commercial building, located in Dalian (China), is chosen to analyze the integrated performances of a CCHP system, such as analyzing instantaneous and monthly performances. Different from previous Chinese cases, the energetic and environmental performances of the CCHP system are analyzed not only under FEL and FTL mode, but also under FHL operation mode. Moreover, a new mathematical foundation is established, through calculating the ratio of heat/cool to electricity required, to reveal whether the recovered thermal energy and the amount of electricity generated by the PGU are enough to provide the energy required. Additionally, generation efficiency of CCHP system η_e, efficiency of HVAC system η_e^{HVAC} and the exhaust gas temperature (T_h) of boiler export are introduced as a changing variable to generate a sensitivity analysis to show how the optimal operation strategy would vary.

2. Descriptions of HVAC System and CCHP System

2.1. HVAC System

Figure 1 illustrates the **HVAC** system used as a reference system.

Figure 1. General separated production system layout for building and energy flows.



The total electrical energy from the grid, E_{grid}^{HVAC} is:

$$E_{grid}^{HVAC} = E_{building} + E_p + E_c \tag{1}$$

where $E_{building}$ is the electricity for lights and equipment in building; E_p is parasitic electricity such as the energy consumption of pumps and fans; and E_c is the electricity supplied to the chiller, which is employed to produce space cooling. The electricity needed by the chiller can be replaced by:

$$E_c = \frac{Q_c}{COP_e} \tag{2}$$

where Q_c is the thermal energy for space cooling; and COP_e is the coefficient of performance (COP) of the electrical chiller.

Considering the energy loss in electricity transmission, the total electricity from grid is converted to the fuel energy consumption as follows:

$$F_e^{HVAC} = \frac{E_{grid}^{HVAC}}{\eta_e^{HVAC}\eta_{grid}} \tag{3}$$

where η_e^{HVAC} and η_{grid} are the efficiency of electricity generation and the transmission efficiency of grid, respectively.

Q_h is the thermal energy for space heating and domestic hot water, which comes from gas boiler and is distributed to user through heating exchanger. The fuel energy consumption for the heating system is computed as:

$$F_b^{HVAC} = \frac{Q_b}{\eta_b^{HVAC}} = \frac{Q_h}{\eta_b^{HVAC}\eta_h^{HVAC}} \tag{4}$$

where Q_b is the output heat from the boiler, and η_b^{HVAC} and η_h^{HVAC} are the boiler efficiency and the heating coil efficiency, respectively.

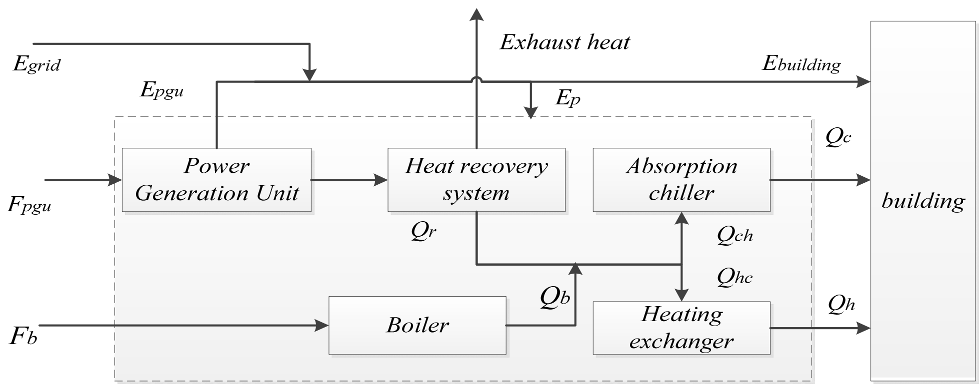
Therefore, the total fuel energy consumption can be calculated as:

$$F^{HVAC} = \frac{E_{building} + E_p}{\eta_e^{HVAC} \eta_{grid}} + \frac{Q_c}{COP_e \eta_e^{HVAC} \eta_{grid}} + \frac{Q_h}{\eta_b^{HVAC} \eta_h^{HVAC}} \tag{5}$$

2.2. CCHP Model

The schematics of the CCHP model are shown in Figure 2. The CCHP system consists of a power generation unit (PGU), waste recovery system, back-up boiler, absorption chiller and heating exchanger. In various studies, the absorption chiller is considered as a "bad" energy converter due to its low coefficient. Therefore, the absorption chiller only works when waste heat is available and conventional electric chiller is preferred to operate when waste heat is not available and absorption chiller operation is uneconomical.

Figure 2. General CCHP system layout for building and energy flows.



Recently, the increasing number of cooling facilities has increased the already strong daily variations in electricity demand. The utilization of heat-driven absorption chillers instead of electricity-driven compression cooling machines is decreasing the power used for cooling. Furthermore, Hondeman [24] illustrated that for electricity-optimised CHP systems based on natural gas, compression chiller technology is more favorable than absorption chiller technology from an energy perspective; however, Poredos and Kitanovski [25] claim that from an exergy perspective, absorption chillers are almost 10% more efficient than compression chillers considering a COP of 6.6. Also the CO_2 emission reductions due to avoided electricity use in compression cooling chillers is larger than the increase due to DH (industrial excess heat associated with zero emissions) and electricity used in AC chillers [26]. Therefore in order to simplify the analysis, it is possible to calculate the Q_{ch} values basing on the assumption to cover the entire cooling load by the absorption chiller [27,28]. This section describes the model used to calculate the energy consumption for the CCHP system [2,29–31].

The balance of the electric energy is expressed as:

$$E_{grid} + E_{pgu} = E_{building} + E_p = E_{demand} \tag{6}$$

where E_{grid} is the electricity from grid; E_{pgu} is the generated power by PGU; and E is the electricity consumption of lights and equipment, $E_{building}$, plus the electricity consumption of pumps and fans, E_p.

The PGU fuel consumption, E_{pgu}, can be estimated as:

$$F_{pgu} = \frac{E_{pgu}}{\eta_e} \tag{7}$$

where η_e is the fuel to electricity efficiency of the PGU.

The recovered waste heat from the prime mover, Q_r, can be calculated to:

$$Q_r = \frac{E_{pgu}}{\eta_e}\eta_{rec}(1-\eta_e) \tag{8}$$

The heat supplied to the cooling system and heating coil is:

$$Q_r + Q_b = Q_{ch} + Q_{hc} \tag{9}$$

where Q_h is the supplementary heat from boiler. While Q_{ch} and Q_{hc} are the heat supplied to cooling system for space cooling and heating coil for space heating, respectively.

The heat required by the cooling system and heating coil are estimated, respectively, as:

$$Q_{ch} = \frac{Q_c}{COP_{ch}} \tag{10}$$

and:

$$Q_{hc} = \frac{Q_h}{\eta_h} \tag{11}$$

where COP_{ch} represent the coefficient of performance of the absorption chiller; and η_h is the efficiency of heating coil.

When the recovered thermal energy does not satisfy the requirements of the absorption chiller, additional heat is provided by the boiler of the CCHP system. The supplementary fuel energy consumption from the boiler can be estimated as:

$$F_b = \frac{Q_b}{\eta_b} = \frac{Q_{ch} + Q_{hc} - Q_r}{\eta_b} \tag{12}$$

where η_b is the auxiliary boiler efficiency.

Therefore, the on-site fuel energy consumption:

$$F = F_{pgu} + F_b \tag{13}$$

During analysis and application of CCHP system, some important assumptions are followed [6]:

(1) The CCHP equipment can operate anywhere between 0% and 100% of its rated capacity, and ramping rate for load adjustment is not included;
(2) The CCHP system is assumed to be 100% reliable;
(3) The efficiency drops of CCHP equipments at part load operation are neglected to simplify the analysis and calculations.

Equations (6)–(13) shows the energy flows of the CCHP system. Based on these equations, the fuel energy consumptions are calculated in two simple operation modes: following electrical load and following thermal load (FEL and FTL), respectively [32]. In FEL mode, the capacities of the CCHP system are decided according to the electricity demand of buildings and the maximum output of PGU is related to the capacities of the CCHP system. When the electricity generated by the PGU is not enough to satisfy the demand, the additional electricity comes from grid. However, in order to simplify the model, the generated electricity of CCHP system is assumed equal to the electrical load at any moment in time, $E_{grid} = 0$. The total fuel energy consumption is expressed to [3,6]:

$$F^{FEL} = \frac{E}{\eta_e} + \frac{\frac{Q_c}{COP_{ch}} + \frac{Q_h}{\eta_h} - \frac{E}{\eta_e}\eta_{rec}(1-\eta_e)}{\eta_b} \cdot \gamma \tag{14}$$

From the above equation, the second term determines whether the supplementary fuel is necessary and γ is the coefficient of the supplementary fuel. The auxiliary boiler provides the additional heat if the heat demand of building is larger than the recovered heat, γ = 1. If the recovered heat is enough to satisfy the heat requirement of building, the excess heat could be exhausted, γ = 0. Consequently, the Equation (14) should be rewritten as [6]:

$$F^{FEL} = \begin{cases} F_{pgu}^{FEL} + F_{boiler}^{FEL} = \frac{E}{\eta_e} + \frac{\frac{Q_c}{COP_{ch}} + \frac{Q_h}{\eta_h} - \frac{E}{\eta_e}\eta_{rec}(1-\eta_e)}{\eta_b} & \gamma = 1 \\ F_{pgu}^{FEL} = \frac{E + E_p}{\eta_e} & \gamma = 0 \end{cases} \tag{15}$$

Similarly, in FTL mode, the capacities of equipment in the CCHP system are determined according to the energy demands of the building. After the equipment is selected, the useful thermal output is limited. When the waste heat recovered from the PGU is not enough to supply the cooling system and heating coil, the supplemental boiler provides the additional heat. However, in order to simplify the model, the useful thermal output at any moment in time is assumed equal to the thermal load. Therefore, the total fuel energy consumption is calculated as:

$$F^{FTL} = \frac{\frac{Q_c}{COP_{ch}} + \frac{Q_h}{\eta_h}}{\eta_{rec}(1-\eta_e)} + \frac{E - \frac{\frac{Q_c}{COP_{ch}} + \frac{Q_h}{\eta_h}}{\eta_{rec}(1-\eta_e)}\eta_e}{\eta_e^{'}\eta_{grid}}\delta \tag{16}$$

In the above equations, the second part of above equations also gives the meaning of whether the demand electricity is from the grid and δ is the coefficient of the additional electricity. When δ = 0, the electricity generated by the system is enough to supply the building; Otherwise, when δ = 1, the additional electricity must be provided by the local grid. Therefore, Equation (16) should be rewritten as:

$$F^{FTL} = \begin{cases} F_{pgu}^{FTL} + E_{grid}^{FTL} = \frac{\frac{Q_c}{COP_{ch}} + \frac{Q_h}{\eta_h}}{\eta_{rec}(1-\eta_e)} + \frac{E - \frac{\frac{Q_c}{COP_{ch}} + \frac{Q_h}{\eta_h}}{\eta_{rec}(1-\eta_e)}\eta_e}{\eta_e^{'}\eta_{grid}} & \delta = 1 \\ F_{pgu}^{FTL} = \frac{\frac{Q_c}{COP_{ch}} + \frac{Q_h}{\eta_h}}{\eta_{rec}(1-\eta_e)} & \delta = 0 \end{cases} \tag{17}$$

However, the above two operating strategies (FEL and FTL) will produce large amounts of excess heat and electricity, respectively, at certain points. The excess electricity is not allowed to be sold back to grid in China and the excess heat is discharged to the environment directly in most cases. Therefore, an optimized operation (FHL) that can eliminate the excess energy should be followed and CCHP systems can then operate at higher efficiency than other situations [4,6].

The electricity generated by the CCHP system can be found as a function of the heat recovered as follows, which is advanced by Mago [7]:

$$E_{pgu} = F_{pgu}\eta_e = \frac{Q_r}{\eta_{rec}(1-\eta_e)}\eta_e = \frac{\eta_e}{\eta_{rec}(1-\eta_e)}Q_r \tag{18}$$

where $\frac{\eta_e}{\eta_{rec}(1-\eta_e)}$ as a new constant ρ,

$$\rho = \frac{\eta_e}{\eta_{rec}(1-\eta_e)} \tag{19}$$

and Equation (18) can be rewritten as:

$$E_{pgu} = \rho Q_r \tag{20}$$

Using this function, the prime mover can be setup to produce only what is required by the system, thus producing no excess electricity or heat. The mode operates as follows:

If $E_{req} < \rho Q_{req}$, the system operates in FEL mode and the electricity generated and the heat recovered by the system can be estimated by:

$$E_{pgu} = E_{req} \tag{21}$$

$$Q_r = \frac{E_{pgu}}{\rho} = \frac{E_{req}}{\rho} \tag{22}$$

The supplementary heat needs to be added by the auxiliary boiler and expressed as:

$$Q_{boiler} = Q_{req} - \frac{E_{req}}{\rho} \tag{23}$$

If $E_{req} > \rho Q_{req}$, the system operates in FTL mode and the electricity generated and the heat recovered by the system can be determined by:

$$E_{pgu} = \rho Q_{req} \tag{24}$$

$$Q_r = Q_{req} \tag{25}$$

The supplementary electricity needs to be added by the local grid and expressed as:

$$E_{grid} = E_{req} - \rho Q_{req} \tag{26}$$

The total amount of fuel energy required by the operation mode can be determined as:

$$F^{FHL} = \begin{cases} F_{pgu}^{FHL} + F_{boiler}^{FHL} = \dfrac{E_{req}}{\eta_e} + \dfrac{Q_{req} - \dfrac{E_{req}}{\rho}}{\eta_b} & E_{req} < \rho Q_r \\ F_{pgu}^{FHL} + E_{grid}^{FHL} = \dfrac{\rho Q_{req}}{\eta_e} + \dfrac{E_{req} - \rho Q_{req}}{\eta_e^{HVAC}\eta_{grid}} & E_{req} > \rho Q_r \end{cases} \tag{27}$$

and the above equation can be rewritten as follows:

$$F^{FHL} = \begin{cases} F_{pgu}^{FHL} + F_{boiler}^{FHL} = \dfrac{E}{\eta_e} + \dfrac{\dfrac{Q_c}{COP_{ch}} + \dfrac{Q_h}{\eta_h} - \dfrac{E}{\rho}}{\eta_b} & E_{req} < \rho Q_r \\ F_{pgu}^{FHL} + E_{grid}^{FHL} = \dfrac{\rho(\dfrac{Q_c}{COP_{ch}} + \dfrac{Q_h}{\eta_h})}{\eta_e} + \dfrac{E - \rho(\dfrac{Q_c}{COP_{ch}} + \dfrac{Q_h}{\eta_h})}{\eta_e^{HVAC}\eta_{grid}} & E_{req} > \rho Q_r \end{cases} \tag{28}$$

3. Optimization Criteria

The selection of the optimization criteria depends on the main goal of the CCHP system. The criteria used in this paper are introduced in the following section.

3.1. Primary Energy Saving

PES is defined as the ratio of the saving energy of CCHP system in comparison to the HVAC system [18]. PES is used to evaluate the primary energy saving achieved by CCHP system with respect to the reference system [33], which can be written as:

$$PESR = \frac{F^{HVAC} - F}{F^{HVAC}} = 1 - \frac{F}{F^{HVAC}} \tag{29}$$

3.2. Exergy Efficiency

The first law of thermodynamics is merely a reflection of the quantity relationship of energy conversion, and does not distinguish the forms and quality grade of heating, cooling, and electricity energy. The exergy analysis method can overcome the limitations of the first law of thermodynamics, since exergy efficiency weighs energy flows by accounting for each in terms of exergy. Specifically, exergy analysis is an effective method and tool in practice for improving the efficiency of energy use and revealing whether or not and by how much it is possible to design more efficient systems by reducing the inefficiencies in existing systems. An exergy analysis is accomplished to calculate the exergy of the flows based on the following assumptions: the reference state is set as the atmospheric environment at 0.1 MPa pressure and 298.15 K; The gain and loss of heat, pressure and exergy in the pipe connections have been neglected [3,12,34].

Exergy efficiency calculates the efficiencies of CCHP system and HVAC system taking into account the different thermodynamic values of different energy forms and quantities and used 'Straight heat input or output' exergy (not Stream-carried exergy) which can be written to [35]:

$$\eta_{exergy} = \frac{EX_e + EX_h + EX_c}{EX_f} = \frac{A_e E + A_c Q_C + A_h Q_h}{A_f F} \tag{30}$$

where η_{exergy} is the exergy efficiency, A_e, A_c, A_h and A_f are the exergy coefficient of electricity, cooling , heating and fuel respectively, and:

$$A_e \approx 1,\ A_f \approx 1$$
$$A_c = \frac{T_0}{T_C} - 1 \qquad (31)$$
$$A_h = 1 - \frac{T_0}{T_h}$$

where T_0 is the ambient temperature; T_c and T_h are the cold water temperature and the heat thermal temperature respectively; T_c and T_h are assumed to be constant and 280 K (7 °C) and 433 K (160 °C), respectively. T_0 is not constant and it changes with time.

3.3. CO_2 Emission Reduction

The amount of CO_2 emission (CE) from a CCHP system depends strongly on the site energy consumption and the emission conversion factors for electricity and natural gas. It can be determined as follows [6]:

$$CE^{CCHP-FEL} = F^{CCHP-FEL}\alpha_f \qquad (32)$$

$$CE^{CCHP-FTL} = F^{CCHP-FTL}\alpha_f + E^{CCHP-FTL}\alpha_e \qquad (33)$$

where α_f and α_e are the emission conversion factors of natural gas and the electricity from grid.

CER is defined as the ratio of the amount of carbon emissions of the CCHP system in comparison to a HVAC system, which can be calculated by [6]:

$$CER = \frac{CE^{HVAC} - CE}{CE^{HVAC}} = 1 - \frac{CE}{CE^{HVAC}} \qquad (34)$$

4. Analysis and Discussion

4.1. Building Description and Energy Demand

This section presents the analysis and discussion of a CCHP system located in Dalian (China). Dalian is a city with a typical maritime climate. The annual average temperature is about 10 °C. The hottest month occurs generally in August, with a monthly average temperature of about 24 °C, and the coldest month is January, with a monthly average temperature of about −6 °C. The baseline building under consideration is a hypothetical hotel building. The hotel has a floor area of 3467 m^2 and an average main ceiling height of 3.6 m. The hourly energy demands of the building are estimated by the DeST software. General information of this building is presented in Table 1.

The hourly heating and cooling demand of this hotel are shown in Figure 3. Figure 4 displays the representative daily energy demands in winter and summer. The values of the variables used to obtain the results for FEL, FTL and FHL operation strategies are presented in Table 2, and the values usually represent the mean value of the Chinese case [3,6,31].

Table 1. General descriptions of the simulated building using DeST software.

Location		**Dalian China**
Building type		Hotel
Total area		3467 m^2
People		2 for everyday
Room	Occupancy schedule	Until [a] (fraction) [b]: 7(1), 9(0.5), 11(0.3), 12(0.1), 14(0.5), 16(0.3), 18(0.1), 21(0.3), 23(1)
	Electric equipment	20 W/m^2
	Equipment schedule	Until [a] (fraction) [b]: 6(0.1), 8(0.3), 11(0), 13(0.3), 17(0), 22(1), 23(0.3)
	Lights	15 W/m^2
	Lights schedule	Until [a] (fraction) [b]: 5(0), 6(0.2), 15(0.1), 17(0.3), 21(1), 22(0.7), 23(0.5)
	Thermostat set point	Winter [c]:18–22 °C ; Summer [d]: 24–26 °C; Spring and autumn [e]: 21–24 °C

Notes: [a] Indicates the hour of the day until the specified fraction is considered; [b] indicates the fraction of the total value of the variable that is considered in the calculation for that specific period of time; [c] Winter: November 15—April 5; [d] Summer: June 1—August 31; [e] Spring and autumn: April 6—May 31, and 1 September—14 November.

Figure 3. The simulated hourly heating and cooling demand of this hotel based on DeST software.

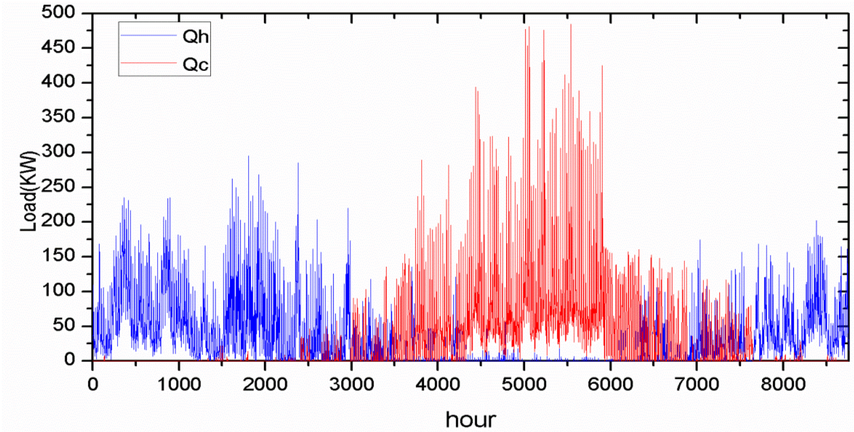


Figure 4. The simulated daily energy demands based on DeST software in representative (**a**) winter and (**b**) summer days.

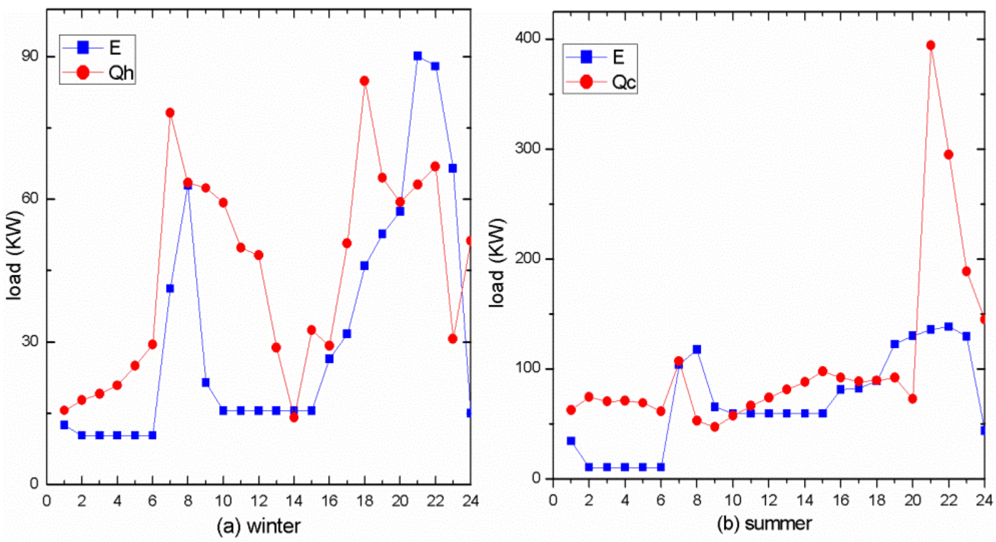


Table 2. Parameters values for CCHP system and HVAC system energy used calculation.

System	Variable	Symbol	Value
CCHP system	PGU efficiency	η_e	32%
	Heat recovery system efficiency	η_{rec}	80%
	Heating coil efficiency	η_h	80%
	Absorption chiller coefficient of performance	COP_{ch}	0.7
	Boiler efficiency	η_b	80%
HVAC system	PGU efficiency	η_e^{HVAC}	25%
	Heating coil efficiency	η_h^{HVAC}	80%
	Vapor compression coefficient of performance	COP_e	3
	Boiler efficiency	η_b^{HVAC}	80%
	Grid transmission efficiency	η_{grid}	95%
CO_2 emission conversion factor (g/KWh)	Electricity from the grid	α_e	923
	Gas	α_f	220

4.2. Instantaneous Performance Analysis

Whether the recovered thermal energy and the amount of electricity generated by the PGU are enough to provide the energy required by the hotel at a certain time can be determined through calculating the ratio of heat/cool to electricity required of representative daily as follows:

Following electric operation, the total electric energy needed by the building has to be supplied by the power generation unit (PGU):

$$E_{req} = E_{pgu} \tag{35}$$

The heat required to satisfy the thermal load can be estimated using Equations (10) and (11) for cooling and heating, respectively. The recovered waste heat from the PGU can be estimated using Equations (7) and (8), respectively. Therefore:

$$\frac{Q_{req}}{E_{pgu}} = \begin{cases} \dfrac{\frac{Q_h}{\eta_h}}{E_{pgu}} = \dfrac{Q_h}{E_{req}\eta_h} & winter \\ \dfrac{\frac{Q_c}{COP_{ch}}}{E_{pgu}} = \dfrac{Q_c}{E_{req}COP_{ch}} & summer \end{cases} \tag{36}$$

$$\frac{Q_{rec}}{E_{pgu}} = \frac{\frac{E_{pgu}}{\eta_e}\eta_{rec}(1-\eta_e)}{E_{pgu}} = \frac{\eta_{rec}(1-\eta_e)}{\eta_e} \tag{37}$$

From above Equations, if $\frac{Q_{req}}{E_{req}} > \frac{Q_{rec}}{E_{pgu}}\left(Q_{req} > Q_{rec}\right)$, then:

$$\frac{Q_h}{E_{req}} > \frac{\eta_{rec}(1-\eta_e)}{\eta_e}\eta_h = \omega_{winter} \text{ or } \frac{Q_c}{E_{req}} > \frac{\eta_{rec}(1-\eta_e)}{\eta_e}COP_{ch} = \omega_{summer} \tag{38}$$

which means that the recovered thermal energy is not enough to handle the thermal load (heating or cooling), supplementary heat needs to be added by the auxiliary boiler.

For the thermal load operation (FTL), the total heat recovered from the PGU has to match the thermal energy load (heating or cooling). Therefore:

$$Q_{req} = Q_{rec} = \begin{cases} \dfrac{Q_h}{\eta_h} & winter \\ \dfrac{Q_c}{COP_{ch}} & summer \end{cases} \tag{39}$$

Since the recovered waste heat from the prime mover is known, the fuel energy can be estimated from Equation (8), while the total electric energy that is supplied by the power generation unit can be determined from Equation (7). Therefore:

$$\frac{Q_{rec}}{E_{pgu}} = \begin{cases} \dfrac{\dfrac{Q_h}{\eta_h}}{\dfrac{Q_h \eta_e}{\eta_h \eta_{rec}(1-\eta_e)}} = \dfrac{\eta_{rec}(1-\eta_e)}{\eta_e} & winter \\ \dfrac{\dfrac{Q_c}{COP_{ch}}}{\dfrac{Q_c \eta_e}{COP_{ch} \eta_{rec}(1-\eta_e)}} = \dfrac{\eta_{rec}(1-\eta_e)}{\eta_e} & summer \end{cases} \tag{40}$$

$$\frac{Q_{req}}{E_{req}} = \begin{cases} \dfrac{Q_h}{E_{req}\eta_h} & winter \\ \dfrac{Q_c}{E_{req}COP_{ch}} & summer \end{cases} \tag{41}$$

From the above Equations: if $\frac{Q_{req}}{E_{req}} < \frac{Q_{rec}}{E_{pgu}} (E_{req} > E_{pgu})$, then

$$\frac{Q_h}{E_{req}} < \frac{\eta_{rec}(1-\eta_e)}{\eta_e}\eta_h = \omega_{winter} \text{ or } \frac{Q_c}{E_{req}} < \frac{\eta_{rec}(1-\eta_e)}{\eta_e} COP_{ch} = \omega_{summer} \tag{42}$$

Refering to the analysis above, $\omega_{winter} = \eta_h \eta_{rec}(1-\eta_e)/\eta_e$ and $\omega_{summer} = COP_{ch}\eta_{rec}(1-\eta_e)/\eta_e$. From Table 2, ω_{winter} is approximately equal and about 1.36. Following electric operation, the supplemental heat from the auxiliary boiler is needed when the ratio of heating load to electricity load is larger than 1.36. While in FTL mode, additional electricity from the grid is needed when the ratio of heating load to electricity load is less than 1.36. In summer, the variable, ω_{summer}, is about 1.19.

Figure 5 shows the primary energy saving of three operation modes compared to a HVAC system. The ratio of thermal demand to electricity required in winter and summer representative day is also presented in Figure 5.

From Figure 5a, it can be seen that the trends of the PES curves are reversed with the trends of ratio curve at certain times (see Table 3). For following FEL mode, the PES decreases with the increases of the ratio of heating load to electric demand. This can be explained by the fact that the ratio increase means that the system needs large amounts of supplemental heat from the boiler; therefore, the primary energy saving achieved from recovering waste heat is decreased. Contrary to FEL operation mode, the trends of the PES curves are reversed with the ratio curve when the generation electricity from the PGU is enough to satisfy the electrical demand in FTL mode. The reason is that if the ratio is increased, a large demand of heat is recovered and excess electricity may be generated since it is already enough for the building demand (see Table 3). However the surplus electricity cannot to be sold back to the grid in China, so this will lead to a waste of energy and the PES will be decreased. In the other times of the day, the PES curves are in accordance with the ratio curve.

Figure 5. Ratio of thermal demand to electricity and PES of different operation mode in representative (**a**) winter and (**b**) summer days based on simulated data.

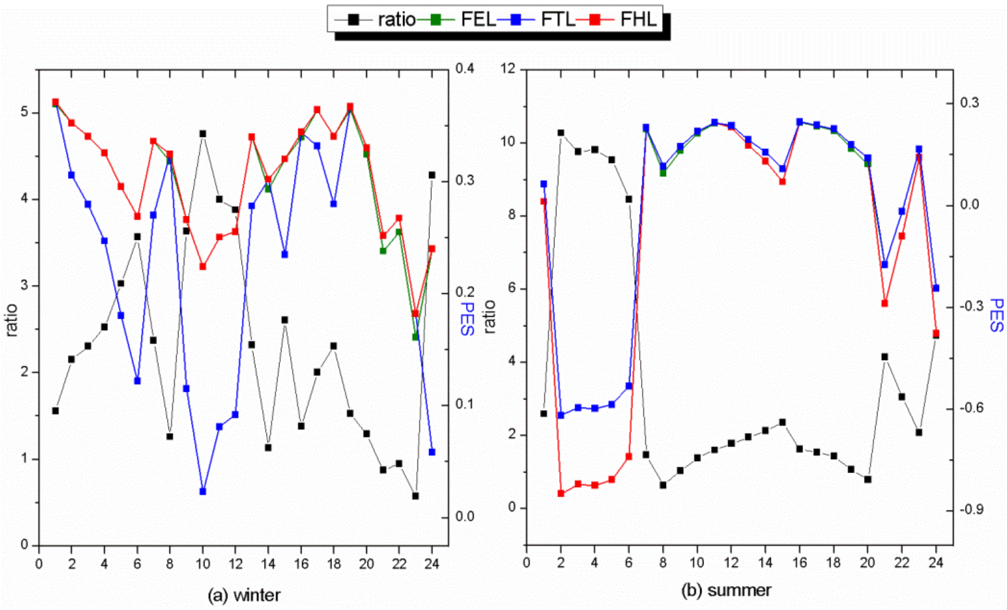


Table 3. Energy demand times for the hotel (hours).

Operation mode		Needing supplemental heat	Heat enough	Needing additional electricity	Electricity enough
Winter	FEL	2–7, 9–13, 15, 17–18, 24	1, 8, 14, 16, 19–23	-	-
	FTL	-	-	1, 8, 14, 16, 19–23	2–7, 9–13, 15, 17–18, 24
Summer	FEL	1–6, 12–15, 21–24	7–11, 16–20	-	-
	FTL	-	-	7–11, 16–20	1–6, 12–15, 21–24

Similarly, in summer, the trends of the PES curves are reversed with the ratio curve trends when the system needs supplemental heat from the boiler in FEL operation mode and the electricity generation from the PGU is enough to satisfy the electrical demand of the building in FTL operation mode. While the PES curves are in accordance with the ratio curve, the recovered heat is enough to provide to the building in FEL operation mode and part of the electricity is needed from the grid in FTL operation mode (see Table 3 and Figure 5b). Figure 6 shows the hourly exergy efficiencies of HVAC system and different operation modes of CCHP system on respective winter and summer days. In winter, the exergy efficiencies of HVAC system are all around 30%, while the range of fluctuation of the exergy efficiencies of the CCHP system in the three operation modes is great, especially in FTL mode (see Figure 6a). In summer, the exergy efficiency of the CCHP system is lower than the exergy efficiency of the HVAC system from 2–6 o'clock. However, during the rest of the time, the CCHP system presents better performance than the HVAC system in exergy efficiency (see Figure 6b).

The hourly CER variations of CCHP systems in FEL, FTL and FHL operation are shown in Figure 7. The trends of the CER curves are similar to those of the PES curves. The potential of CER is higher when the primary energy saving is higher. In representative winter days, the range of fluctuation of the CER variation is [0.364, 0.491], [0.137,0.477] and [0.220, 0.488] in the three operation modes, respectively. It can be found that the range of fluctuation is higher in FTL mode than that in FEL and FHL modes (see Figure 7a). Figure 7b shows that, on a representative summer day, the system in FEL mode can achieve the largest reduction of CO_2 emissions during most of the day, which implies FHL is not a better operation mode than FEL and FTL in CO_2 emissions reduction, although it offers the best performance in terms of primary energy savings. Furthermore, there are even some negative values in FTL mode, which indicates that the application of a CCHP system does not always reduce carbon dioxide emissions compared to a HVAC system from an instantaneous performance view point.

Figure 6. Exergy efficiencies of HVAC system and different operation modes of CCHP system in representative (**a**) winter and (**b**) summer days based on simulated data.

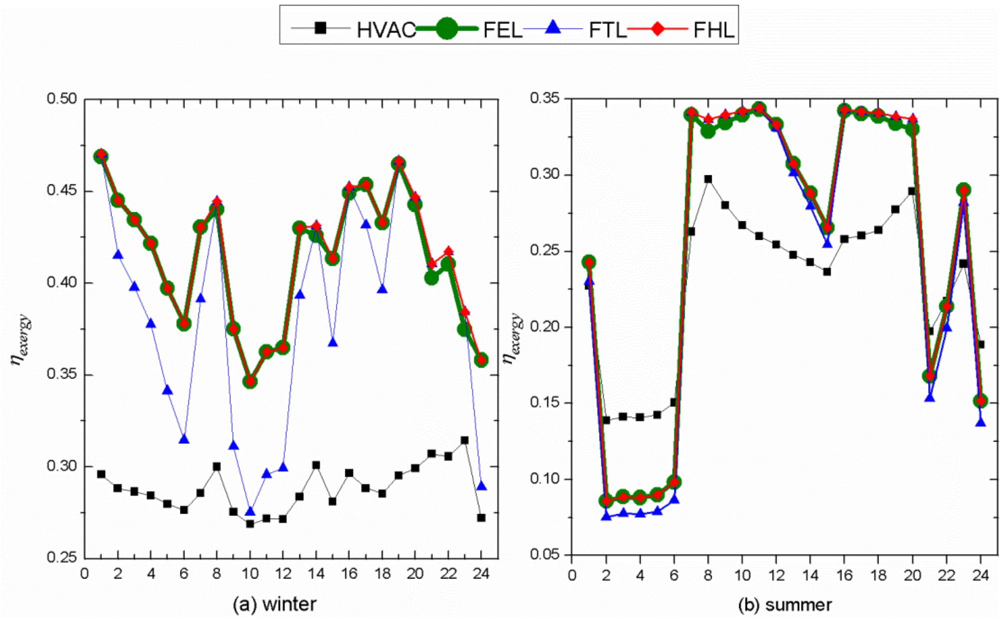
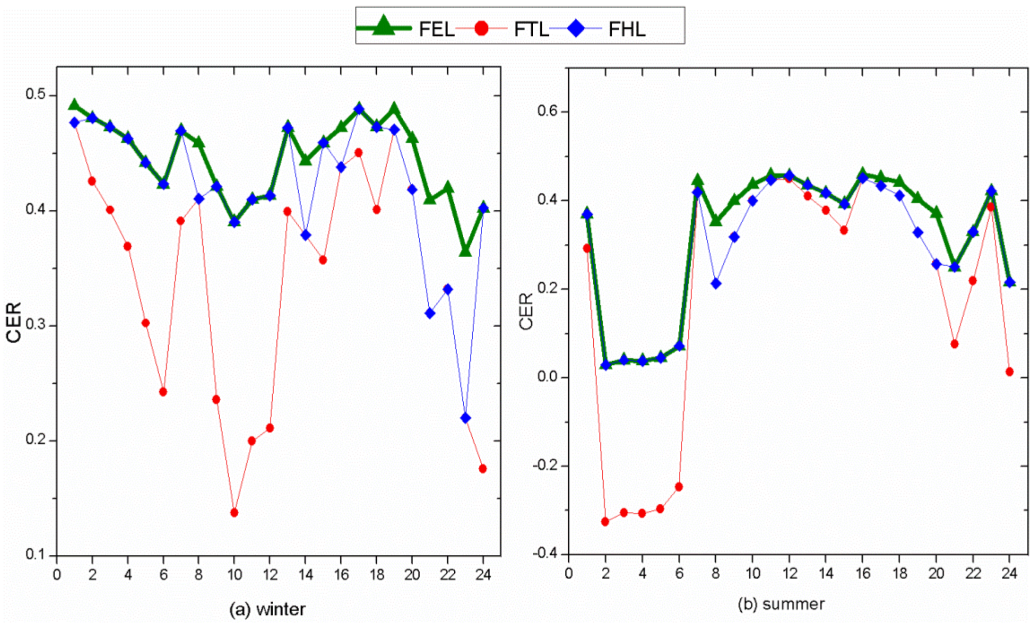


Figure 7. CER of different operation mode in representative (**a**) winter and (**b**) summer days based on simulated data.



4.3. Monthly and Annually Performance Analysis

The building cooling, heating, and electric loads are illustrated in Figure 8.

Figure 9 shows the variation of the primary energy savings for three basic CCHP operation strategies: FEL, FTL, and FHL. Figure 9 demonstrates how the FHL follows the operation strategy, either electric or thermal, which consumes less primary energy. It is observed that the FHL operation strategy approximately follows the electric load (FEL) during heating and cooling season and it follows the thermal load (FTL) during the transition season. This can be explained by the fact that during the heating season and cooling season, the building thermal and cooling load is high as compared with the electric load (see Figure 8) for the weather in Dalian. Therefore, following the electric load and supplementing the heat required with the auxiliary boiler is more beneficial than following the thermal load and producing a great amount of excess electricity that cannot be sold back to utility companies. Similarly, for the transition season, since the thermal and cooling load is not high compared with the electric, following the thermal load and buying electricity from the grid when needed is more beneficial than following the electric load and producing more excess heat.

The monthly exergy efficiencies of CCHP system operating in FEL, FTL and FHL mode are shown in Figure 10. Figure 10 also shows that the entire trends of the exergy efficiency of the two systems are similar. However, the efficiencies of the HVAC system every month are lower than those of the CCHP system. It can be obtained that the monthly exergy efficiencies of the CCHP system and HVAC system are 38.65% in FEL mode, 38.27% in FTL mode, 38.94% in FHL mode and 29.73% in HVAC system. The CCHP system following FHL mode has highest exergy efficiency during the whole year. From Figure 10, the CCHP system achieves the lowest exergy efficiency in the summer during the entire year. The reason is that T_0 is close to T_c, which could lead to low cooling exergy. It is also observed that the exergy efficiency in FHL operation mode approximately follows the electric load during heating and cooling season and follows the thermal load during the transition season, which is in agreement with the above PES research results.

Figure 8. The simulated electric, cooling, and heating loads based on DeST software for the reference building in Dalian, China.

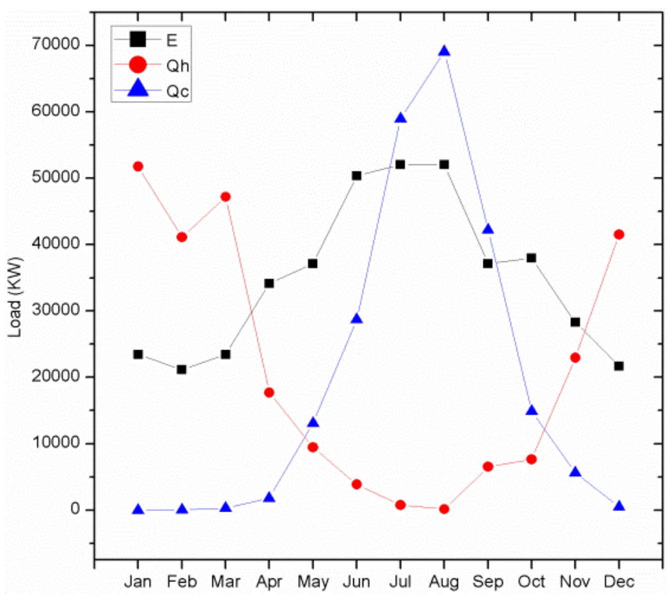


Figure 9. Variation of the primary energy saving based on simulated data for three basic CCHP operation strategies: FEL, FTL, and FHL.

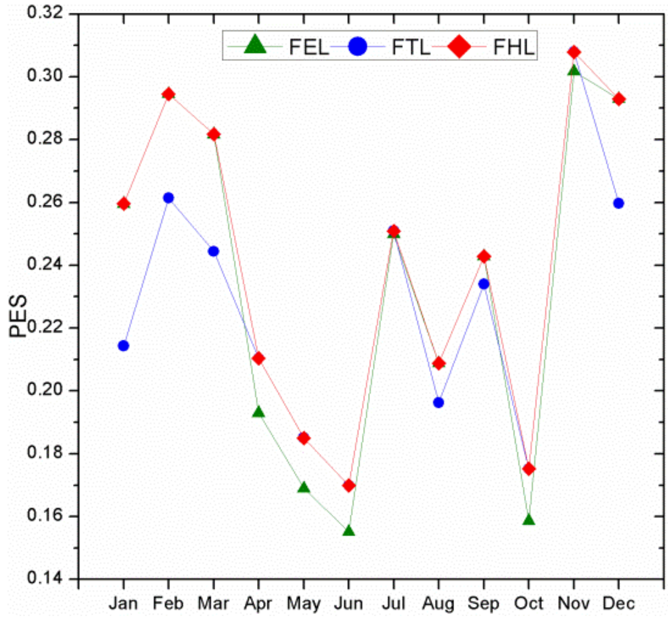
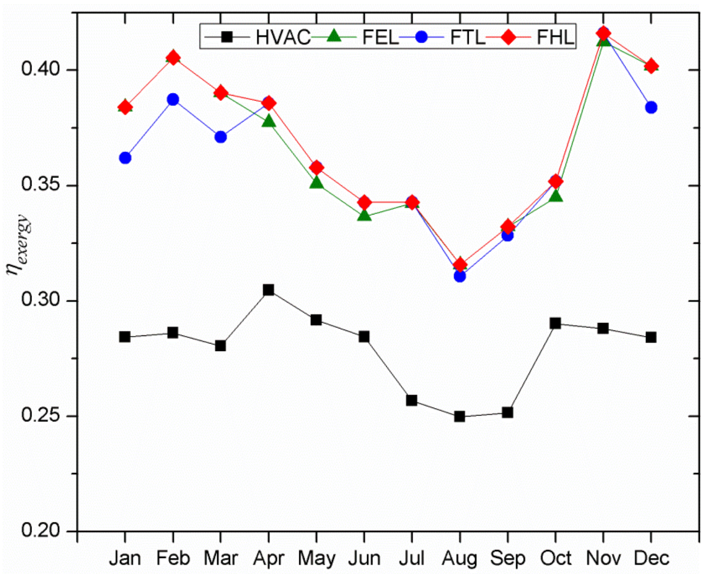


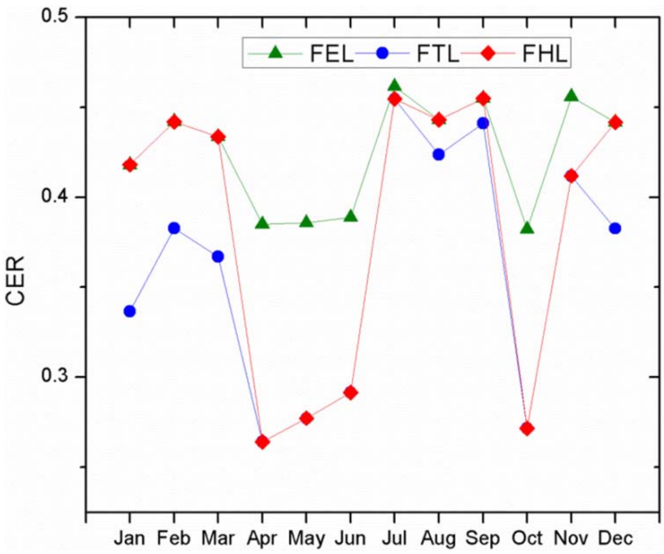
Figure 10. The exergy efficiency of HVAC system and CCHP system in FEL, FTL, and FHL mode based on simulated data.



The monthly CER variations of CCHP systems in FEL, FTL and FHL operation compared to a HVAC system are shown in Figure 11. From Figure 11, the trends of CER curves are similar in the three operation strategies. Furthermore, it is found that all CER values are positive, which implies the application of CCHP systems should be helpful to reduce CO_2 emissions in China.

Conversely to the PES results, FEL operation mode presents the highest reduction (42.6%), while FTL and FHL give reductions of 36.5% and 38.7%. Hence the CCHP system in FEL can be seen as an environmental-friendly energy system in Dalian. The reason is that most of electricity in China is produced from coal and reducing the amount of electricity from a center power plant can help reducing the carbon dioxide emissions. In FTL and FHL mode, additional electricity will be purchased from the grid when the generated electricity cannot satisfy the building's needs, while in FEL operation mode only natural gas is consumed, which is a low carbon-content type of energy.

Figure 11. The CER variations of CCHP system in FEL, FTL, and FHL mode based on simulated data.



From the above analysis, it can be seen that CCHP system in FEL mode saves 23.1% primary energy, reduces CO_2 emissions by 42.6% and the exergy efficiency is 38.65% (see Table 4). Although FEL operation mode might not offer the best results in terms of primary energy consumption, it is really valuable for buildings that must address environmental concerns. Therefore, a CCHP system in FEL operation mode can be seen as a suitable energy system for Dalian from a annual performance perspective. The result is contrary to the result presented by Mago [4]. In Mago's paper, the excess electricity generated by PGU can be translated into additional primary energy saving for being sold back to grid in FTL operation mode. However, usually the sale of surplus generated electricity from the CCHP system back to grid in China is not allowed when the CCHP system operates FTL mode. Therefore the excess electricity in China cannot contribute to the primary energy savings, and the results of this paper are in agreement with the results obtained by Wang, who focused on the Chinese case [6].

Table 4. Annually operation parameters of CCHP system (unit: %).

Operation mode	PES	η_{exergy}	CER
FEL	23.1	38.65	42.6
FTL	22.3	38.27	36.5
FHL	23.7	38.94	38.7

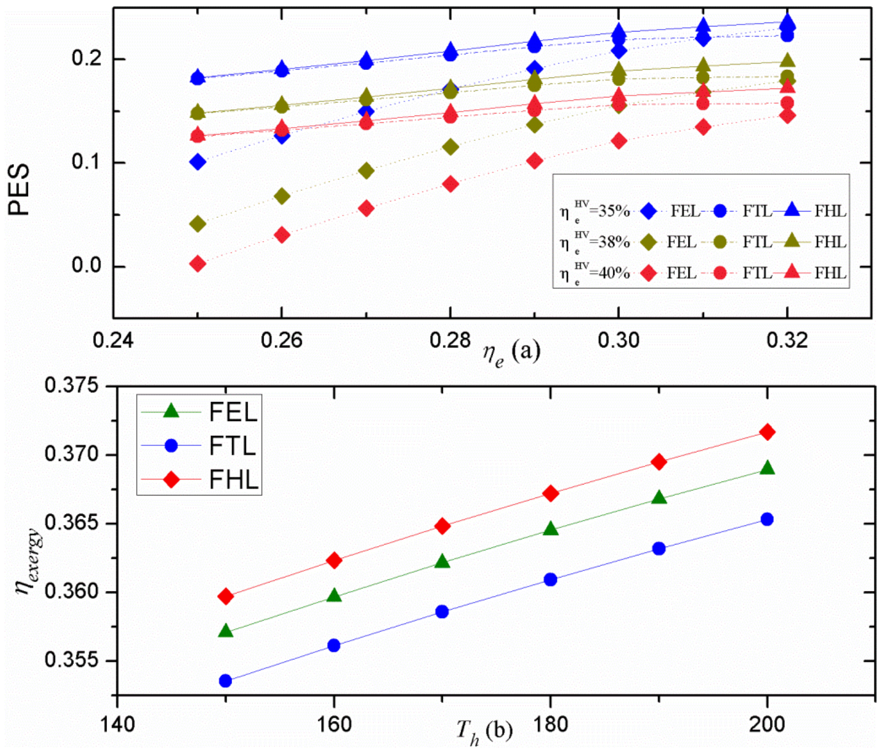
4.4. Sensitivity Analysis

The sensitivity analysis is helpful to guide the optimal design of the CCHP system and improve the optimal results. The integrated and accurate parameters included generation efficiency (η_e) of CCHP system, efficiency of HVAC system η_e^{HVAC} and the exhaust gas temperature (T_h) of boiler export.

To find out the improvement potential of CCHP system, η_e and η_e^{HVAC}, are used to analyze its primary energy saving ratio compared to the HVAC system. As shown in Figure 12a, the current generation efficiency range of the micro gas turbine is about 25%–32% and comparable HVAC systems with different efficiencies (35%, 38% and 40%) are provided as baseline. The above values are taken into the original calculation and the performance of CCHP system will change with the different values of the variables used. It can be integrally found that the primary energy saving increases with the increasing of η_e in all operation modes and higher energy savings are obtained from FHL operation mode, mainly due to the low PGU efficiencies of the HVAC system. Moreover, with the increasing of PGU generation efficiency (η_e), the PES values of the three modes become more and more close. η_e^{HVAC}, as a reference value, is another key factor when calculating primary energy savings. From Figure 12a, the PES decreases with the increase of η_e^{HVAC}. Additionally, it can be seen that when the PGU generation efficiency is 25% and the electricity efficiencies of HVAC system is 40%, the CCHP system can save at least 0.3% primary energy. However, if the generation efficiency of the HVAC system can reach above 40%, the PES of CCHP system will be negative in FEL mode and the CCHP system doesn't really save energy when other parameters are kept constant.

Similarly, exergy efficiency (η_{exergy}) is related with the exhaust gas temperature (T_h) of boiler effluent. It can be found that in Figure 12b that η_{exergy} increases with the increasing of Th in all three operation modes. From Section 2, there is $A_h = 1 - T_0/T_h$, and the increase of T_h can make A_h increase as T_0 is constant. Consequently the heating exergy and exergy efficiency are directly increased. Figure 12b shows that the range of fluctuation of η_{exergy} is [35.36%, 37.17%] with the exhaust gas temperature changing from 150 °C (423 K) to 200 °C (473 K) in the three operation modes and FHL operation mode has the higher exergy efficiency than the other two operation modes.

Figure 12. The sensitivity analysis of CCHP system based on simulated data in FEL, FTL and FHL mode.



5. Conclusions

This paper presents an analysis of CCHP systems operated following FEL, FTL and FHL mode. All the operation modes are compared based on three optimization criterion, namely energy savings (PES), exergy efficiency (η_{exergy}) and CO_2 emissions reductions (CER). As an illustrative example, a hypothetical hotel building located in Dalian, China, has been examined for a case study. According to the simulation results, the following conclusions can be deduced.

The instantaneous performances of the CCHP system with the instantaneous building loads are analyzed for representative winter and summer days. The performance of the CCHP system is closely related to the hourly ratio of heating/cooling load to electricity demand. It can be identified when the CCHP system needs supplemental heat or whether the generated electricity is enough for the building. Moreover, the exergy efficiency of the CCHP system is lower than the exergy efficiency of a HVAC system at certain points in summer and the application of CCHP system does not always reduce carbon dioxide emissions referring to HVAC system, especially in FTL mode.

From the viewpoints of monthly performance of the CCHP system, FHL operation mode achieves a better performance than FEL and FTL modes in the PES and exergy efficiency aspects. However, FEL operation mode presents a rather high CO_2 emissions reduction compared to FHL mode, which is really valuable for buildings in Dalian that are required to address environmental concerns. Therefore a CCHP system in FEL operation mode can be seen as a suitable energy system for Dalian from a monthly performance perspective. Additionally since the excess electricity from the CCHP system cannot be sold back to the grid, the operation of a CCHP system following electricity load is a better mode.

The sensitivity analysis shows that primary energy saving increases with the increasing of η_{exergy} in all operation modes (FEL, FTL, FHL) and higher energy savings are obtained from operation in FHL mode, mainly due to the low PGU efficiencies of the HVAC system (η_e^{HVAC}). The improvement of exhaust gas temperature T_h of boiler export in CCHP system can also obviously make exergy efficiency increase.

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