Energy Efficient Operation Strategy Design for the Combined Cooling, Heating and Power System

by

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B. Eng., Harbin Institute of Technology, 2010

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ABSTRACT

Combined cooling, heating and power (CCHP) systems are known as trigeneration systems, designed to provide electricity, cooling and heating simultaneously. The CCHP system has become a hot topic for its high system efficiency, high economic efficiency and less greenhouse gas (GHG) emissions in recent years. The efficiency of the CCHP system depends on the appropriate system configuration, operation strategy and facility size. Due to the inherent and inevitable energy waste of the traditional operation strategies, i.e., following the electric load (FEL) and following the thermal load (FTL), more efficient operation strategy should be designed. To achieve the highest system efficiency, facilities in the system should be sized to match with the corresponding operation strategy. In order to reduce the energy waste in traditional operation strategies and improve the system efficiency, two operation strategy design methods and sizing problems are studied (In Chapter 2 and Chapter 3).

Most of the improved operation strategies in the literature are based on the "balance" plane, which implies the match of the electric demands and thermal demands.

However, in more than 95% energy demand patterns, the demands cannot match with each other at this exact "balance" plane. To continuously use the "balance" concept, in Chapter 2, the system configuration is modified from the one with single absorption chiller to be the one with hybrid chillers and expand the "balance" plane to be a "balance" space by tuning the electric cooling to cool load ratio. With this new "balance" space, an operation strategy is designed and the power generation unit (PGU) capacity is optimized according to the proposed operation strategy to reduce the energy waste and improve the system efficiency. A case study is conducted to verify the feasibility and effectiveness of the proposed operation strategy.

In Chapter 3, a more mathematical approach to schedule the energy input and power flow is proposed. By using the concept of *energy hub*, the CCHP system is modelled in a matrix form. As a result, the whole CCHP system is an input-output model. Setting the objective function to be a weighted summation of primary energy savings (PESs), hourly total cost savings (HTCs) and carbon dioxide emissions reduction (CDER), the optimization problem, constrained by equality and inequality constraints, is solved by the sequential quadratic programming (SQP). The PGU capacity is also sized under the proposed optimal operation strategy. In the case study, compared to FEL and FTL, the proposed optimal operation strategy saves more primary energy and annual total cost, and can be more environmental friendly.

Finally, the conclusions of this thesis is summarized and some future work is discussed.

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Nomenclature

Abbreviations

ATC	Annual Total Cost
CCHP	Combined Cooling, Heating and Power
CDE	Carbon Dioxide Emissions
CDER	Carbon Dioxide Emissions Reduction
CHP	Combined Heating and Power
COP	Coefficient of Performance
EC	Evaluation Criteria
FEL	Following the Electric Load
FTL	Following the Thermal Load
GHG	Greenhouse Gas
HTC	Hourly Total Cost
HTCS	Hourly Total Cost Saving
HVAV	Heating, Ventilation and Air Conditioning
KKT	Karush-Kuhn-Tucker
PEC	Primary Energy Consumption
PES	Primary Energy Saving
PGU	Power Generation Unit
ppm	Parts Per Million
ppmvd	Parts Per Million, Volumetric Dry
SP	Separation Production

Symbols

C	Cost
E	Electricity
F	Fuel
L	Facility Life
N	Installed Capacity
Q	Thermal Energy
x	Electric Cooling to Cool Load Ratio
η	Efficiency
μ	Carbon Conversion Factor

Superscripts

$CCHP$	CCHP System
SP	SP System
PGU	PGU

Subscripts

ac	Absorption Chiller
$annual$	Annual Value
b	Boiler
c	Cooling
ca	Carbon
e	Electricity
ec	Electric Chiller
f	Fuel
gap	Energy Gap
$grid$	Local Grid
h	Heating
$hour$	Hourly Value
hrc	Recovered Heat for Cooling
hrh	Recovered Heat for Heating
hrs	Heat Recovery System
m	Total Consumption
p	Parasitic
pgu	Power Generation Unit
$pgum$	PGU Capacity
r	Recovered
red	Redundant
$user$	User
$userl$	Lower Bound of User
$useru$	Upper Bound of User

Chapter 1

Introduction

1.1 Combined Cooling, Heating and Power Systems

With the rapid development of distributed energy supply systems [7–10], combined heating and power (CHP) systems and combined cooling, heating and power (CCHP) systems have become the core solutions to improve the energy efficiencies and to reduce greenhouse gas (GHG) emissions [11–15]. The CCHP system is an extended concept of the CHP system – a proven and reliable technology with more than 100 years history, which is utilized mainly in large-scale centralized power plants and industrial applications [1]. CHP systems are developed to conquer the problem of low energy efficiency of conventional separation production (SP) systems. In SP systems, electric demands, which include daily electricity usage and electric chiller usage, and heating demands are provided by the purchased electricity and fuel, respectively. Since no self-generation exists in SP systems, they must be of low efficiency, however, in CHP systems, most of the electric and heating demands are provided simultaneously by a prime mover together with a heat recovery system, a heat storage system, etc. Energy demands beyond the system capacity can be supplied by the local grid and an auxiliary boiler. If introducing some thermally activated technologies, e.g., absorption and adsorption chillers, into the CHP to provide the cooling energy, the original CHP system evolves to be the CCHP system [16], which can also be referred to as the *trigeneration* system and building cooling heating and power (BCHP) system. Since there is no need for cooling energy from cooling system generally in winter, a CHP system can be regarded as a special case of the CCHP system. A CCHP system can achieve up to 50% greater system efficiency than a CHP plant of the same size does [17].

A typical CCHP system is shown in Figure 1.1. The power generation unit (PGU)

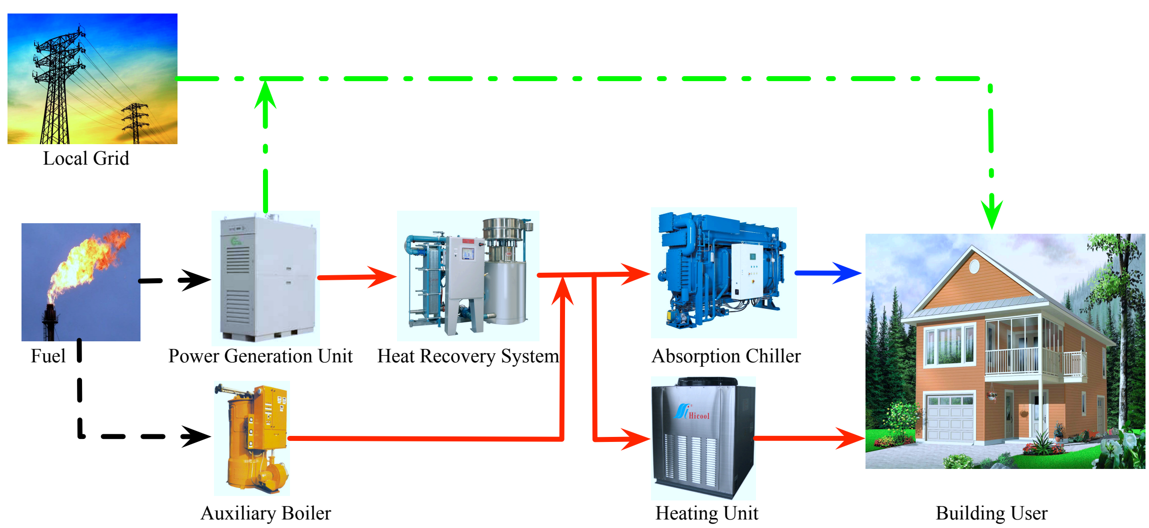

Figure 1.1: *A typical CCHP system.*

provides electricity for the user. Heat, produced as a by-product, is collected to meet cooling and heating demands via the absorption chiller and heating unit. If the PGU cannot provide enough electricity or by-product heat, additional electricity and fuel need to be purchased to compensate the electric gap and feed the auxiliary boiler, respectively. In this way, three types of energy, i.e., cooling, heating and electricity, can be supplied simultaneously.

Compared with conventional generation plants, the advantages of a CCHP system appears in three-fold: High efficiency, low GHG emissions and high reliability.

First, the high overall efficiency of a CCHP system implies that less primary fuel is consumed in this system to obtain the same amount of electric and thermal energy. In [1], the authors give an example to show that, compared with the traditional energy supply mode, the CCHP system can improve the overall efficiency from 59% to 88%. This improvement owes to the cascade utilization of different energy carriers and the adoption of the thermally activated technologies. As the main electricity source, the PGU has an electric efficiency as low as 30%. By implementing the heat recovery system, the CCHP system can collect the by-product heat to feed the absorption/adsorption chiller and heating unit to provide cooling and heating energy, respectively. By adopting the absorption chiller, no additional electricity needs to be purchased from the local grid to drive the electric chiller in summer, but only the recovered heat is used. In winter, a CCHP system degenerates to be a CHP system. The high efficiency of the CHP system is investigated in [18–25]. In a nutshell, a CCHP system can dramatically reduce the primary consumption and improve the energy efficiency.

The second advantage involved in the CCHP system is the low GHG emissions. On the one hand, the trigeneration structure of the CCHP system contributes to this reduction. Compared with SP systems, if within the capacity limitation of the prime mover, no additional electricity needs to be purchased from the local grid, which is supplied by fossil-fired power plants. It is well known that, even though the penetration of some types of renewable energy, e.g., the wind, tide and solar energy, increases significantly [26–28], because of their intermittency, the main electricity producer is the fossil-fired power plant. By reducing the consumption of electricity from the local grid, GHG emissions from fossil-fired power plants can be decreased.

Moreover, adopting the thermally activated technologies can also reduce the electricity consumption by the electric chiller, which will result in less consumptions of fossil fuel in the grid power plant. On the other hand, new technologies in the prime mover also contribute to the GHG emissions reduction. Incorporating fuel cells , which are one of the hottest topics in recent years, in he CCHP system can increase the system efficiency up to $85 - 90\%$ [29]. Compared with some conventional prime movers, such as the internal combustion (IC) engine and combustion turbine, the new-tech prime movers can provide the same amount of electricity with less fuel supply and less GHG emissions. In recent years, aiming to reduce the GHG emissions, an increasing number of countries begin to run the carbon tax act [30–34]. As a result of the act, reducing GHG emissions can not only reduce the contaminate of the air, but also can improve the system economic efficiency.

The other benefit brought by the CCHP system is the reliability. Reliability can be regarded as the ability of an energy system to secure the energy supply at a reasonable price [35]. Recent cases have demonstrated that the centralized power plants are vulnerable to natural disasters and unexpected phenomenon [36]. Changes in climate, terrorism, customer needs and electricity market are all fatal threats to the centralized power plants [1]. The CCHP system, which adopts the distributed energy technologies, can be resistant to external risks and has no electricity blackouts, for its independency on electricity distribution. A comparison of the reliability between the distributed and centralized energy systems in Finland and Sweden can be found in [35].

A typical CCHP system consists of a PGU, a heat recovery system, thermally activated chillers and a heating unit. Normally, the PGU is the combination of a prime mover and an electricity generator. The rotary motion generated by the prime mover can be used to drive the electricity generator. There are various options for the prime mover, e.g., steam turbines, stirling engines, reciprocating IC engines, combustion turbines, micro turbines and fuel cells. The selection of the prime mover depends on current local resources, system size, budget limitation and GHG emissions policy. The heat recovery system plays a role in collecting the by-product heat from the prime mover. The most frequently used thermally activated technology in the CHP/CCHP system is the absorption chiller. Some novel solutions, such as the adsorption chiller, and the hybrid chiller, are also adopted in CCHP systems [37–40]. The selection of heating unit depends on the design of the heating, ventilation and air conditioning (HVAC) components.

With the benefits of high system and economic efficiency, and less GHG emissions, CCHP systems have been widely installed in hospitals, universities, office buildings, hotels, parks, supermarkets, etc. [41–45]. For example, in China, the CCHP project at Shanghai Pudong International Airport generates combined cooling, heating and electricity for the airport's terminals at peak demand times. It is fuelled by natural gas from offshore in the East China Sea [46]. This system is equipped with one 4 MW natural gas turbine, one 11 t/h waste heat boiler, cooling units of four YORK OM 14,067 kW, two YORK 4,220 kW, four 5,275 kW steam LiBr/water chillers, three 30 t/h gas boilers and one 20 t/h as standby for heat supply [47]. In the last decade, the installation of CCHP systems grows flatly. Especially, the development is much slower in developing countries than that in developed countries due to following barriers: Less public awareness, insufficient incentive policies and instruments, nonuniform design standards, incomplete connections with power grid, high price and supply pressure of natural gas, and difficulties in manufacturing equipment [47]. According to a survey provided by the World Alliance for Decentralized Energy (WADE), the penetration of CCHP systems can be enlarged by introducing of European Union Emissions Trading Scheme (EUETS) and increasing carbon tax.

1.2 Prime Movers

A prime mover, defined to be a machine that transforms energy from thermal, electrical or pressure form to mechanical form, typically an engine or turbine, is the heart of an energy system. Normally, the output of a prime mover is the rotary motion, so it is always being used coupling with an electric generator. In recent years, the mostly installed prime movers are gas engines and gas turbines [48]. These two types of prime movers belong to the reciprocating IC engine and the combustion turbine/micro-turbine, respectively. Some other types of prime movers, such as steam turbines, micro-turbines, stirling engines and fuel cells, are also being used in CCHP systems in some particular cases. In this section, emphasis will be put on reciprocating IC engines and combustion turbines/micro-turbines; other types will also be discussed.

1.2.1 Reciprocating internal combustion engines

A reciprocating engine, also known as a piston engine, is a heat engine that uses one or more reciprocating pistons to convert pressure into a rotary motion [49]. A classical reciprocating engine can be shown in Figure 1.2. There exist two types of reciprocating engines, i.e., spark ignition, which uses the natural gas as the preferred fuel and can also be fed by the propane, gasoline or landfill gas, and compress ignition, which can operate on diesel fuel or heavy oil [50]. The size of the reciprocating engines can range from 10 kW to over 5 MW.

As stated in [51, 52], with the advantages of low capital cost, quick starting, well load following, relatively high partial load efficiency and generally high reliability, the reciprocating engines have been widely used in many distributed generation applications, such as the industrial, commercial and institutional facilities for power

Figure 1.2: *A reciprocating internal combustion engine.*

generation, and CCHP systems. The waste heat of the reciprocating engine, consisting of exhaust gas, engine jacket water, lube oil cooling water and turbocharger cooling [50], can be used in thermally activated facilities in the CCHP system. However, a reciprocating engine does need regular maintenance and service to ensure its availability [53]. Since the rising level of GHG emissions has become a big concern, applications of diesel fueled engines are restricted for the high emission level of NO_x. Current natural gas ignition engines have relatively low emissions profiles and are widely installed. One example is that HONDA has developed a new cogenerator, which is a natural gas-powered engine, powered by "GF160V". This cogeneration unit can reduce the CO_2 emissions up to approximately 20% [54]. Another classical application of the reciprocating engine example is the CHP plant with IC engines installed at the Faculty of Engineering of the University of Perugia since 1994 [55].

Reciprocating IC engines are quite popular in some applications when working together with an electric or absorption chiller. High temperature exhaust gas from the engine can be used to provide heating and cooling, or to drive the desiccant dehumidifier. Mentioned in [1], the configuration of a large percentage of CCHP systems using reciprocating engines can be shown in Figure 1.3. Maidment *et al.* [44] analyze a CCHP system for a typical supermarket using a gas turbine and a LiBr/water absorption chiller. They discuss the methodology for choosing the prime mover between a gas engine and a gas turbine. The result shows that this CCHP system offers significant primary energy consumption savings and CO_2 savings compared to conventional heat and power schemes. In [56], the authors assess a CCHP system, which is driven by a reciprocating IC engine, combined with a desiccant cooling system. This system incorporates a desiccant dehumidifier, a heat exchanger, a direct evaporative cooler and a direct evaporative cooler. Longo *et al.* [57] discuss a CCHP system equipped with an Otto engine and an absorption chiller. The exhaust thermal energy is recovered to drive a double-effect LiBr/water cycle, and the heat recovered

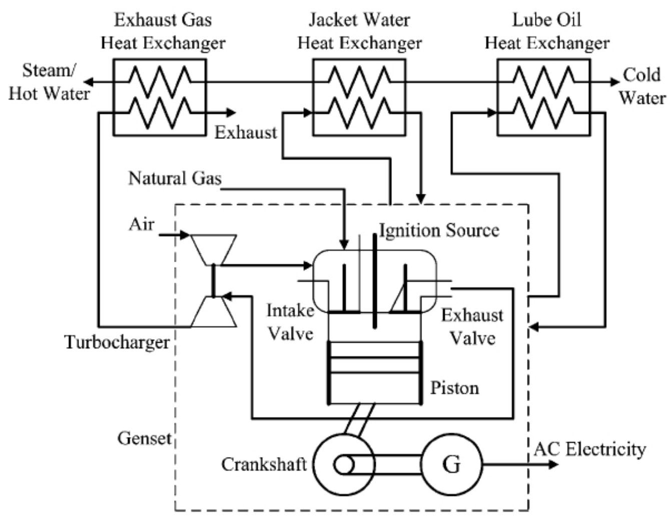

Figure 1.3: *Reciprocating engine heat recovery [1].*

from the cooling jacket is used to drive a single-effect LiBr/water cycle. In [58], Talbi *et al.* explore the theoretical performance of four different configurations of a CCHP system equipped with a turbocharger diesel engine and an absorption refrigeration unit. The situation of CO_2 emissions of CCHP systems with gas engines is discussed in [59].

1.2.2 Combustion turbines

The combustion turbine, which is shown in Figure 1.4, also known as the gas turbine, is an engine in which the combustion of a fuel, usually the gas, occurs with an oxidizer in a combustion chamber [60]. Combustion turbines have been used for the purpose of

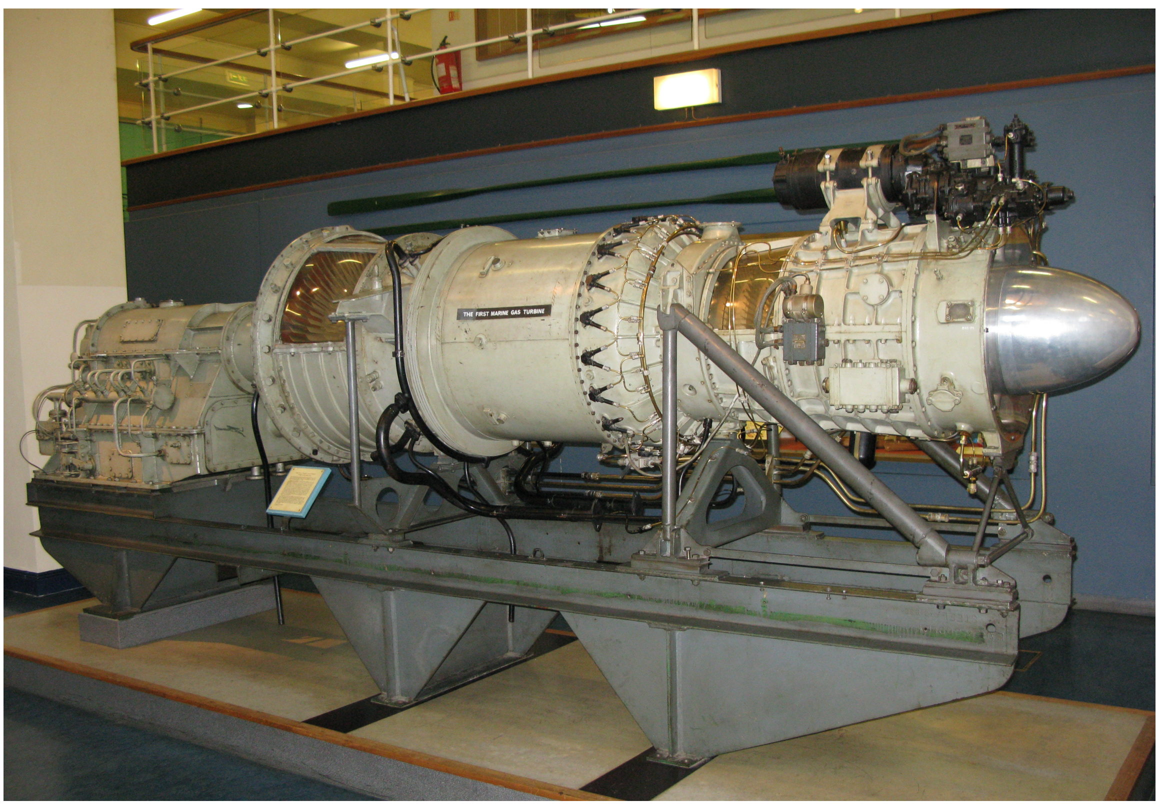

Figure 1.4: *A gas turbine from Metrovick Gatric.*

electricity generating since 1930s. The size of gas turbines ranges from 500 kW to 250 MW, which makes it suitable for large-scale cogeneration or trigeneration systems.

At partial load, the efficiency of the gas turbine can be unacceptable lower than full-capacity efficiency. As a result, generation sets smaller than 1 MW are proven to be uneconomical [1]. Gas turbines also produce high-quality (high-temperature around 482°C) exhaust heat that can be used by thermally activated processes in CCHP systems to produce cooling, heating or drying, and to raise the overall system efficiency to approximately 70–80% [61]. Adopting some cycle integration technologies, such as steam injection gas turbines [62] and humid air turbines [63], can improve the performance of the simple-cycle gas turbine by integrating the bottoming water/steam cycle into the gas turbine cycle in the form of water or steam injection [64].

For GHG emissions, because of the use of natural gas, when compared with other liquid or solid fuel-fired prime movers, gas turbines can dramatically reduce CO_2 emissions per kilowatt-hour [65]. Emissions of NO_x can be below 25 ppm and CO emissions can be in the range of 10 to 50 ppm. Some emission control approaches, such as the diluent injection, lean premixed combustion, selective catalytic reduction, carbon monoxide oxidation catalysts, catalytic combustion and catalytic absorption systems can also help to reduce NO_x emissions.

One typical application of gas turbine-based cogeneration or trigeneration systems is for colleges or university campuses, where the produced steam is used to provide space heating in winter and cooling in summer. Another typical application is for the supermarket. In the U.S., CCHP systems have been widely installed in supermarkets to improve the system efficiency. Produced steam and heat from the gas turbine is used to drive the food-refrigeration system, which requires a huge amount of cooling energy, and to provide the basic space heating [66]. CCHP systems using gas turbines have attracted a certain amount of attentions. Exergy analyses for a combustion gas turbine based power generation system are addressed in [67]; investigations of CCHP systems using gas turbines can be found in [13, 68];

1.2.3 Steam turbines

A steam turbine is a mechanical device that extracts thermal energy from pressurized steam, and converts it into rotary motion [69]. An example can be shown in Figure 1.5. Compared with reciprocating steam engines, the higher efficiency and lower cost

Figure 1.5: *A steam turbine that has served for more than fifty years.*

make steam turbines being used for about 100 years. The size of steam turbines can range from 50 kW to several hundred MWs for large utility power plants [70]. Because of the low partial load electric efficiency, steam turbines are not suitable for small-scale power plants. In the U.S. and some European countries, steam turbines have been widely installed in large-scale CHP/CCHP systems. If given well maintenance, the life of the steam turbine can be extremely long, which can be counted in years.

The working principle of the steam turbine is different from those of reciprocating IC engines and combustion turbines. For the latter two, electricity is the product and the heat is generated as a by-product. However, for the steam turbine, electricity is the one generated as the by-product. When equipped with a boiler, the steam turbine can operate with various fuels including clean fuels, such as natural gas, and other fossil fuels. This character dramatically improves the flexibility of the steam turbine. In CHP/CCHP applications, the low pressure steam can be directly used for space heating or for driving thermally activated facilities.

GHG emissions of the steam turbine depend on the fuel it uses. If using some clean fuel, i.e., the natural gas, and adopting some effective emission control approaches, GHG emissions can be relative low. However, the low electric efficiency and long start-up time restrict the installation of steam turbines in small-scale CCHP systems and distributed energy applications [1].

1.2.4 Micro-turbines

Micro-turbines are extensions of combustion turbines. A micro-turbine manufactured by Capstone Turbine Corporation can be shown in Figure 1.6. The size of micro-turbines ranges from several kWs to hundreds kWs. They can operate on various fuels, e.g., natural gas, gasoline, diesel, etc. One important character of the micro-turbine is that it can provide an extremely high rotation speed, which can be used to efficiently drive the electricity generator. Because of the small size, micro-turbines are suitable for distributed energy systems, especially for CHP and CCHP systems. In the CCHP system, by-product heat of the micro-turbine is used to drive the sorption chillers and desiccant dehumidification equipment in summer, and to provide space heating in winter. The designed life of micro-turbines ranges from 40,000 to 80,000 hours [71, 72].

Figure 1.6: *Capstone C200 micro-turbine with power output of 190 kW.*

Another key advantage of the micro-turbine is the low level of GHG emissions, thanks to the *gaseous fuels feature lean premixed combustor* technology. In addition, low inlet temperature and high fuel-to-air ratios also contribute to emissions of NO_x of less than 10 ppm. According to the data in [71], despite of the stringent standard of less than 4–5 ppmvd of NO_x, almost all of the example commercial units have been certified to meet it.

Even though with the drawbacks of higher capital costs than reciprocating engines, low electrical efficiency, and sensitivity of efficiency to changes in ambient conditions, the compact size and low-weight per unit power, a smaller number of moving parts, lower noise, multi-fuel capability [73] and low GHG emissions still make the micro-turbine an arisen prime mover in distributed energy systems. Analyses of CHP/CCHP systems installing micro gas turbines can be found in [73, 74].

In distributed energy systems, small-scale CCHP systems have been proven to be an efficient one. Due to the advantages, installing micro-turbine becomes the best choice for a small-scale CCHP system. Much work has been done to investigate the performance of using micro-turbines in CCHP systems. Tassou *et al.* [2] validate the feasibility of the application of a micro-turbine based trigeneration system in a supermarket. The scheme of this trigeneration system is shown in Figure 1.7. Karellas

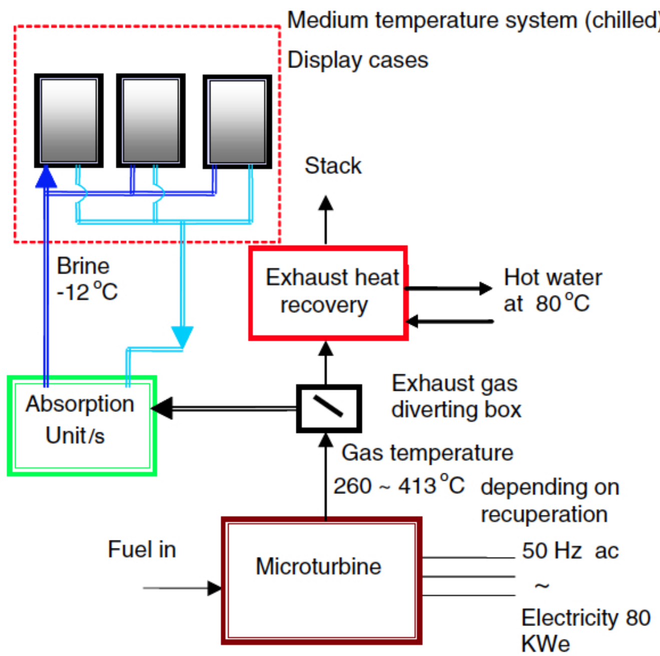

Figure 1.7: *Scheme of a CCHP system with micro-turbine [2].*

et al. [75] propose an innovative biomass process and use it to drive a micro-turbine and a fuel cell in a CHP system. In 2002, the Oak Ridge National Laboratory (ORNL) presented their work of testing a micro-turbined CCHP system. The testing facility consists of a 30 kW micro-turbine for a distributed energy resource, whose exhaust is used to feed thermally activated facilities, including an indirect-fired desiccant dehumidifier and a 10-ton indirect-fired single-effect absorption chiller [76]. Bruno *et al.* [77] conduct a case study of a sewage treatment plant, which is a trigeneration system. The prime mover selected in this system is a biogas-fired micro gas turbine. Hwang [78] in his work investigates potential energy benefits of a CCHP system with micro-turbine installed. Velumani *et al.* [79] propose and mathematically model a CCHP system with an integration of a solid oxide fuel cell (SOFC) and a micro-turbine installed. This plant uses natural gas as the primary fuel and the SOFC is fed with gas fuel. Other evaluations, analyses and control strategy designs for CCHP systems running with micro-turbines can be found in [80–84], to name a few.

1.2.5 Stirling engines

In contrast to the IC engine, the stirling engine is an external combustion engine, which is based on a closed cycle, where the working fluid is alternatively compressed in a cold cylinder volume and expanded in a hot cylinder volume [85]. Two basic categories of stirling engines exist: Kinematic stirling engines and free-piston stirling engines. Also, the engine can fall into three configurations: Alpha type, beta type and gamma type [86, 87].

A stirling engine can operate on almost any fuel, e.g., gasoline, natural gas and solar energy. Compared to IC engines, stirling engines operate with a continuous and controlled combustion process, which results in lower GHG emissions and less pollution [87, 88]. According to the data in [89], implementing a same capacity of 25 MW, NO_x emissions of the stirling engine is 0.63 kg/MWh, compared with 0.99 kg/MWh of the IC engine. It is worth mentioning that, since the working fluid is sealed inside the engine, there is no need to install valves or other mechanisms, which makes the stirling engine simpler than an IC engine. As a result, stirling engines can be relatively more safe and silent when running.

However, some challenges arise when using stirling engines in CHP/CCHP systems. The first one is the low specific power output compared with a same sized IC engine. High capital cost is also a key factor that restricts its development. Another aspect is the working environment in CHP/CCHP systems. Unlike IC engines, the efficiency of a stirling engine drops when the working temperature increases. The last one but equally important is that the power output of the stirling engine is not easy to tune. Despite the above drawbacks, stirling engines have been put into some CHP/CCHP applications because of the flexibility in fuel source, long service time and low level of emissions. A small-scale CHP plant with a 35 kW hermetic four cylinder stirling engine for biomass fuels is designed, created and tested by the Technical University of Denmark, MAWERA Holzfeuerungsanlagen GesmbH and BIOS BIOENERGIESYSTEME GmbH in Austria [90]. Moreover, SIEMENS collaborated with some European boiler manufactures, such as Remeha and Baxi, to conduct a large field test in 2009 and market introduction in 2010 of micro-CHP systems with stirling engines. A novel stirling solar engine can be shown in Figure 1.8.

Figure 1.8: *A stirling solar engine in Sandia National Laboratories, Albuquerque, New Mexico.*

The most promising aspect of the stirling engine must be that it can be solar driven. Because of the increasing rate of carbon tax and more attention paid on the GHG emissions, the using of solar energy in CHP/CCHP systems gives more chances for the stirling engine.

Some theoretical work has also been done to investigate stirling engines installed in CCHP systems. Kong *et al.* [89] propose a trigeneration system with a stirling engine installed and claim that this system could save more than 33% primary energy compared to the conventional SP system. Aliabadi *et al.* [91] discuss the efficiency and GHG emissions of a stirling engine-based residential micro-CHP system fueled by diesel and biodiesel. According to the market assessment, stirling engines have not been widely applied in the CCHP market. To be further penetrated in CHP/CCHP systems, solutions to high capital cost, long warming up time and short durability of certain parts should be found [92].

1.2.6 Fuel cells

Another environment concerned type of prime mover is the fuel cell. Fuel cells convert chemical energy from a fuel into electricity through a chemical reaction with oxygen or other oxidizing agents, and produce water as a by-product [93–95]. Compared to other fossil fuel based prime movers, fuel cells use hydrogen and oxygen to generate electricity. As a result, the fuel cell can operate quietly and extremely environmental friendly. Since it contains few moving parts, the fuel cell system has a higher reliability than the combustion turbine or the IC engine [96]. The efficiency of fuel cells can range from 50% to 90% [97]. There exists various types of fuel cells, i.e., proton exchange membrane fuel cell (PEMFC), alkaline fuel cell (AFC), phosphoric acid fuel cell (PAFC), molten carbonate fuel cell (MCFC) and the previously mentioned SOFC.

In recent years, much work has been done to investigate fuel cells in CCHP systems. The most widely choice is the SOFC. A 5 kW SOFC demonstration unit can be shown in Figure 1.9. Tse *et al.* [98] investigate a trigeneration system, which is jointly driven by a SOFC and a gas turbine, for marine applications. In [99], Kazempoor *et*

Figure 1.9: *A 5 kW SOFC demonstration unit at VTT.*

al. develop a detailed SOFC model, and study and optimize different SOFC system configurations. They also assess the performance of a building integrated with a trigeneration system, which comprises a SOFC and a thermally driven chiller. In [100], a SOFC with the capacity of 215 kW is combined with a recovery cycle for the sake of simultaneously meeting cooling load, domestic hot water demand and electric load of a hotel with 4600 m^2 area. An economic comparison between the trigeneration and SP systems indicates that, due to the lower heating value of the fuel, the maximum efficiency of 83% for energy trigeneration and heat recovery cycle can be achieved. Verda *et al.* [101] model a distributed power generation and a cogeneration system incorporated with the SOFC. The authors also compare three configurations for this system, based on different choices of refrigeration systems, i.e., single-effect absorption chiller, double-effect absorption chiller and vapor compression chiller, from both technical and economic points of view. Other work on the environmental, economical and energetic analyses of CCHP systems equipped with SOFCs can be found in [102–108], to name a few. In [109], the authors model the CCHP system with stationary fuel cell systems from thermodynamic and chemical engineering aspects; and optimize the operation for that. Margalef *et al.* [110] compare two strategies of operating a CCHP system equipped with a magnesium-air fuel cell (MAFC). The first strategy is to blend the exhaust gas with the ambient air; while the other one is to use the exhaust gas to drive an absorption chiller. The result shows that the second strategy is preferred, for the overall estimated efficiency is as high as 71.7%. Bizzarri [111] discusses the size effect of a PAFC system incorporated into a trigeneration system. Investigations reveal that the more the proper sizing is carried out for the highest environmental and energy benefits, the higher the financial returns will be.

1.3 Thermally Activated Technologies

The most efficient solution to providing cooling is to utilize the rejected heat instead of electricity. This solution is called the thermally activated technology, which is dominated by the sorption cooling. The difference between the sorption cooling and the conventional refrigeration is that the former one uses the absorption and adsorption processes to generate thermal compression rather than the mechanical compression. One important reason for the CCHP system be efficient and of low GHG emissions is because space cooling and heating can be provided by using the rejected heat from the prime mover along with the electricity generating. This cascade utilization of heat owes to the thermally activated technology. In conventional SP systems, approximately two-thirds of the fuel used to generate electricity is wasted in the form of rejected heat. By introducing thermally activated technologies, the electric load for cooling is shifted to the thermal load, which can be fully or partially achieved by absorbing or adsorbing the discard heat from the prime mover. The main application of the sorption refrigeration is for CCHP systems in residential buildings, hospitals, supermarkets, office buildings and district cooling systems [112].

Mainly, three types of the thermally activated technologies exist, i.e., absorption chiller, adsorption chiller and desiccant dehumidifier. Since the temperature of the discard heat from prime movers can lie in different ranges, thermally activated facilities should be chosen to couple with prime movers. For example, if the heat source temperature is around 540°C, then the suitable choice is a double-effect/triple-effect absorption chiller.

1.3.1 Absorption chillers

Investigations of the absorption cycle began in 1700's when it was found that ice could be produced by an evaporation of pure water from a vessel contained within an evacuated container in the presence of sulfuric acid [113]. The absorption chiller is one of the most commonly used and commercialized thermally activated technologies in CCHP systems. The difference between an absorption chiller and a vapor compression chiller is the process of compression. Since absorption chillers use heat to compress the refringent vapor instead of mechanically using rotating devices, they can be driven by the steam, hot water or high temperature exhaust gas. As a result, electricity needed for the conventional refrigeration can be dramatically reduced, and the noise of the cooling process can be lowered significantly.

The working process of an absorption chiller can be divided into two processes: The absorption process and the separation process. The absorption process can be shown in Figure 1.10. The left vessel contains the refrigerant and the right vessel is filled with a mixture of refrigerant and adsorbent. Caused by the absorption process of the refrigerant vapor in the right vessel, pressure and temperature in the left vessel will drop. The temperature reduction in the left vessel is the refrigeration process. At the same time, as a result of the absorption process in the right vessel, heat must be rejected to the surroundings.

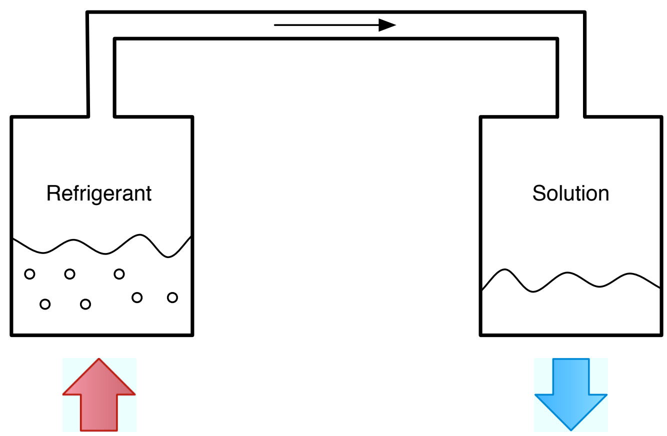

Figure 1.10: *Absorption process.*

As the absorption process continuing, the solution in the right vessel gradually becoming saturated. To keep the capability of absorbing, refrigerant must be separated from the solution. Figure 1.11 shows the separation process, which can be regarded as a reverse of the absorption process. Heat from the heat source is used to dry the refringent from the saturated or almost saturated solution. The refrigerant vapor is then condensed by a heat exchanger to act in the next cycle of absorption process.

Chemical and thermodynamic properties of the working fluid determine the performance of an absorption chiller. The working fluid should be chemically stable, non-toxic and non-explosive. Moreover, in liquid phase, it must has a margin of miscibility within the operating temperature range of the cycle [114]. According to [115], there are around 40 refrigerant compounds and 200 absorbent compounds available for the absorption chiller working fluid. However, the most commonly used two are the

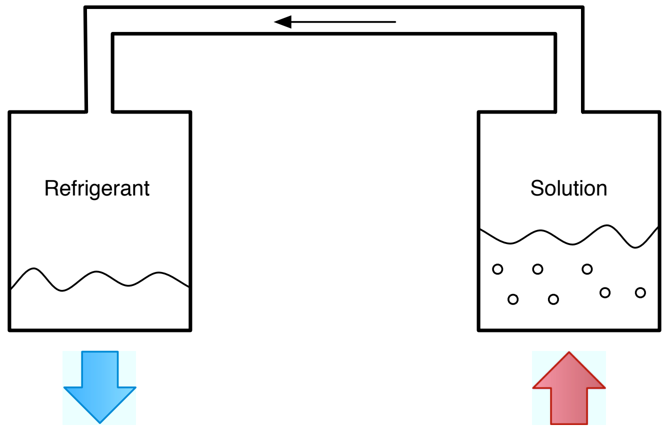

Figure 1.11: *Separation process.*

lithium-bromide/water (liBr/water) and the water/ammonia (NH_3/water). Usually, LiBr/water absorption chiller are used in air cooling applications with evaporation temperature in the range of 5-10°C; while NH_3/water absorption chillers are used in small-scale air conditioning and large industrial applications with evaporation temperature below 0°C [112].

In the literature, absorption chillers have been widely installed in CCHP systems. In 2002, the U.S. government awarded Burns & McDonnell Engineering Co. a development contract of building an integrated gas turbine energy system based on improved CHP/CCHP technology. This plant is powered by a 4.6 MW Solar Turbines Centaur 50 gas turbine and two-stage indirect fired Broad Co. absorption chillers [116]. A small-scale CCHP system, installing a micro-turbine and an absorption chiller is demonstrated in the University of Maryland [117]. In [108], a decentralized system with the integration of an SOFC and a double-effect LiBr/water absorption chiller is investigated. In [118], the authors introduce a CCHP system, with three engines and a total electrical power production of 9 MW, which supplies the thermal energy to drive an NH_3/water absorption chiller ARP-M10 by Colibri.

1.3.2 Adsorption chillers

The development of the adsorption cooling began when the phenomenon of adsorption refrigeration caused by ammonia adsorption on AgCl was discovered by Faraday in 1848 [119]. Similar as absorption chillers, adsorption chillers make use of the discard heat from the prime mover to provide space air conditioning. One important difference of an adsorption chiller from an absorption chiller is that the former one can be driven by low temperature heat source. Further more, the noiselessness, solution pump free, corrosion and crystallization trouble free, and small volume make adsorption chillers suitable for CCHP systems, especially small-scale ones [120–122].

Different from the absorption chiller, in which a fluid permeates or is dissolved by a liquid or solid, the adsorption chiller provides cooling by using solid adsorbent beds to adsorb and desorb a refrigerant. Similar to the two processes in the absorption chiller, temperature of the adsorbent changes according to the refrigerant vapor adsorbed and desorbed by adsorbent beds. A simple adsorption refrigeration circuit consists of a solid adsorbent bed, a condenser, an expansion valve and an evaporator [39]. The refrigeration process of the adsorption chiller can also be divided into two processes, i.e., absorbent heating and desorption process, and the adsorption process. In the first one, the adsorbent bed is connected with a condenser first. Driven by a low temperature heat source, the refrigerant is condensed in the condenser and heat is released to the surroundings. Following that, in the adsorption process, the adsorbent bed is connected to an evaporator, at the same time disconnected form the condenser. Then cooling is generated from evaporation and absorption processes of the refrigerant. However, this simple adsorption chiller provides cooling in an intermittent way. To continuously provide cooling, two absorbent beds should be installed in the system together, in which one bed is heated during the desorption process and the other one is cooled during the adsorption process.

The same as absorption chillers, adsorption chillers have no internal mechanical moving parts either. As a result, they can not only run quietly, but also need no lubrication and less maintenance. In addition, adsorption chillers are always made modularity, which makes them suitable for the cooling capacity expansion. Moreover, as mentioned before, since no electricity and fuel is needed to drive the chiller, low level of GHG emissions is guaranteed. Because of the advantages of the adsorption chiller, research and demonstrations of this type of chiller installed in the CCHP system have been developed widely. A theoretical research of a silica gel-water adsorption chiller in a micro-scale CCHP system can be found in [123]. In 2000, a CCHP system, equipped with a fuel cell, a solar collector and an integration of a mechanical compression chiller and an adsorption chiller, was installed in the St. Jphannes Hospital [1]. In the same year, a CCHP system with an adsorption chiller began to operate in the Malteser's Hospital, Germany. Shanghai Jiaotong University (SJTU) has been investigating the applications of adsorption chillers in CCHP systems for many years. In 2004, SJTU set up a gas-fired micro-CCHP system consisting of a small-scale power generator set and a novel silica gel-water adsorption chiller [112].

1.3.3 Desiccant dehumidifier

A desiccant dehumidifier removes the humidity from the air by using materials that attract and hold moisture. To achieve comfort cooling, sensible cooling, aiming to lower the air temperature, and latent cooling, which means reducing humidity, should be achieved simultaneously. Since, by introducing desiccant dehumidifiers, the control of humidity independent of the temperature is allowed; potentially wasted thermal energy can be used to reduce the latent cooling load; and bacteria and virus can be scrubbed out, desiccant dehumidifiers always operate with chillers or conventional air-conditioning systems to provide comfort cooling and to increase overall system efficiency [1, 124].

Mainly, there exist two commercialized types of desiccant dehumidifiers, which are distinguished by desiccant types, i.e., solid desiccant dehumidifier and liquid desiccant dehumidifier. Solid desiccant dehumidifiers are usually used for dehumidifying air for commercial HVAC systems; while liquid desiccant dehumidifiers are popular in industrial or residential applications. Desiccant dehumidifiers are suitable for CHP/CCHP systems, because the regeneration process in the desiccant system provides an excellent use of waste heat [125]. In [126], the authors introduce a CCHP system utilizing the solid desiccant cooling technology. Researchers in Tsinghua University, China, also carry out a laboratory research to assess the operational performance and energy efficiency of a CCHP system installing a liquid desiccant dehumidifier [127]. In [56], the authors assess a desiccant dehumidifier system in a CHP application incorporating an IC engine. Badami *et al.* [128] analyze the performance of a trigeneration plant with liquid desiccant cooling system installed.

1.4 System Configuration

An economical, efficient and of low emissions CCHP system should be designed with fully consideration of energy demands in a specific area, prime mover and other facilities' types and capacities, power flow and operation strategy, and the level of GHG emissions. The selection of facility types belongs to the design of the system configuration, which emphasizes on the selection of prime movers according to current available technologies and the system scale. It is well known that different climate conditions in different areas lead to different patterns of energy demands. For example, in conventional CHP systems, steam turbine based plants are always used as heat plants with electricity generated as a by-product in some cold areas. While in the temperate zone, in summer, electricity needed by the air conditioning could be a huge amount. Thus, in this kind of area, combustion turbine based CHP systems are popular. Some CHP/CCHP applications based on prime mover selections have been mentioned in Section 1.2. The existing CHP/CCHP sites in the market sorted by prime movers can be shown in Figure 1.12. With a selected CCHP system config-

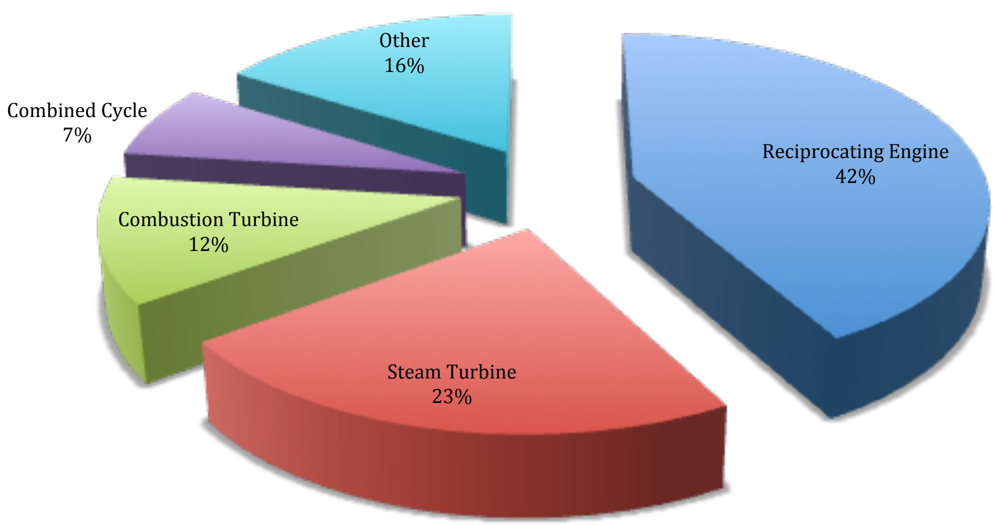

Figure 1.12: *Existing CHP/CCHP sites classified by prime movers.*

uration, operation strategy is the key to achieve the most efficient way for the CCHP to operate. The operation strategy determines how much electricity or fuel should be input to the system according to the demands; which facility should be shut down to keep the efficiency; how the energy carriers flow between facilities; and how much is the power one facility should operate at. With a designated configuration and an appropriate operation strategy, suitable sizing and optimization can make the system operate in an optimal way. As we know, in the configuration part, one chooses prime movers in a vague way, i.e., no accurate rated power is chosen. For instance, when designing a small-scale CCHP system, whose capacity is in the range of 20 kW–1 MW, but what is the specific value on earth? Too small capacity can cause more purchasing on electricity from the local grid; while too large capacity costs more in the capital cost. In addition, the optimal size is also affected by the operation strategy. Thus, to design an efficient and green CCHP system, the above mentioned points should be taken full consideration. Here, CCHP applications categorized by the plant size will be mainly discussed.

Categorized by the rated electricity generation capacity, the CCHP system can land in micro-scale (under 20 kW), small-scale (20 kW–1 MW), medium-scale (1 MW–10 MW) and large-scale (above 10 MW).

1.4.1 Micro-scale CCHP systems

Micro-scale CCHP systems are the ones with rated size under 20 kW. Recently, much work has been done to investigate and analyze micro-scale CCHP systems, for they are suitable for distributed energy systems. In the literature, Easow *et al.* [129] discuss the potential of the micro trigeneration system being applied in the decentralized cooling, heating and power. The authors verify this concept by an experimental plant, which is a micro trigeneration system with a liquefied petroleum gas driven Bajaj 4-stroke IC engine. In North Carolina Solar Center, Raleigh, North Carolina, 2010, an integrated micro-CCHP and solar system was installed to demonstrate technical and economic feasibilities of incorporating photovoltaic (PV), solar thermal, and propane-fired CHP systems into an integrated distributed generation system [130]. The rated output of the CHP plant incorporated, PV, and solar thermal is 4.7 kW plus 13.8 kW, 5.4 kW, and 4.1 kW, respectively. Thermal energy produced by this system can be used for space heating, domestic hot water, process heating, dehumidification and absorption cooling. With this solar based CCHP system, CO and NO_x emissions can be reduced to below 250 ppm and below 30 ppm, respectively. Other experimental and test results of micro-CCHP systems can be found in [131–133], to name a few. In addition, various work on energetic, economic and thermodynamic analyses has been done in recent years. In the most recent work, in [134], the authors provide an analysis of matching prime mover heat sources to thermally driven devices in a micro-scale trigeneration system. The T-Q analysis of the prime mover waste heat in this literature also indicates the promise of incorporating micro-turbines, SOFCs and HT-PEMFCs into the trigeneration system. Other analyses can be referred to [135–140]. Some researchers also focus on the optimization of micro-CCHP systems. Arosio *et al.* [141] model a micro-scale CCHP system based on the linear optimization and incorporate the Italian tariff policy into this model. The proposed model allows to evaluate the influence of each parameter on the system performance. Other optimization research can be found in [133, 142]. Recently, some renewable energy, such as the solar energy, has been implemented in the CCHP system to further reduce GHG emissions. In [143], a micro trigeneration system equipped with a solar system is studied. This system is integrated by a micro-turbine with output power of 5 kW and a LiBr/water absorption chiller. The heat source for the absorption chiller and the micro CHP system is a solar storage tank. Immovilli *et al.* [144] compare a conventional CCHP system with the one based on solar energy. Besides the PV, they also propose two technical solutions for the solar CCHP to access residential applications, i.e., concentrated sunlight all-thermoacoustic and hybrid thermo-PV systems. Beyond the above, some novel micro-scale CCHP structures are also proposed. Henning *et al.* [126] investigate a micro trigeneration system, whose air conditioning facilities integrate a vapor compression chiller and a desiccant wheel, for the indoor air conditioning in mediterranean climate. The research results show that, compared to conventional technologies, an electricity saving of 30% can be achieved. In [145], the authors investigate the performance of an absorption chiller, which is installed in a micro-scale BCHP system, under varying heating conditions. Huangfu *et al.* [11] introduce a novel micro-scale CCHP system which can be applied in domestic and light commercial applications. Evaluations and analyses in this literature show that this micro CCHP system enjoys good economic efficiency with a payback period of 2.97 years; also the electric load conditions determine the electric efficiency, which means that, compared to half load, the system can perform better when operating with full load.

1.4.2 Small-scale CCHP systems

Small-scale CCHP systems are the ones with rated size ranging between 20 kW and 1 MW. They have been widely used in the supermarkets, retail stores, hospitals, office buildings and university campuses. Different types of prime movers and refrigeration systems can be combined freely according to energy demands. Around the world, small-scale CCHP plants have been installed in many applications. A 500 kW biomass CCHP plant is installed in the Cooley Dickinson Hospital, Northampton, Massachusetts, which is a 55,742 m^2 hospital with 140 beds [146]. In 1984, the first boiler installed in this system was a Zurn-550 HP biomass boiler, which was fired by virgin wood chips. Then in 2006 and 2009, due to increased energy demands, an AFS-600 HP water/fire tube high pressure boiler, two 250 kW Carrier Energent micro steam turbines and a 2,391 kW absorption chiller were installed consecutively. This CCHP system has brought a lot of benefits to this hospital, especially the 99.5% particulate removal accomplished by the Multiclone separator and Baghouse. The Easr Bay Municipal Utility District (EBMUD), which was a publicly owned utility that provided water service to portions of two countries in the San Francisco bay area, began to use a 600 kW micro-turbine CHP/chiller system at its downtown Oakland administration building in 2003 [147]. This system is composed by ten 60 kW Capstone micro-turbines and one 633 kW YORK absorption chiller. The total project costs $2,510,000, whose payback period is estimated to be 6–8 years. Another small-scale CCHP application is in the Smithfield Gardens, which is a 56-unit affordable assisted-living facility in Seymour, Connecticut [148]. This system includes a 75 kW Aegen 75LE CHP module, an American Yazaki absorption chiller, and a Baltimore Air Coil cooling tower. With this system installed, the Smithfield Gardens can save 22% on its annual energy costs. With the pollution controlled by the Non-Selective 3-way Catalytic Reduction System, CO_2 and NO_x reductions can achieve 32% and 74%, respectively. Vineyard 29, a winery located in St. Helena, California, installs a 120 kW micro-turbine/chiller system to reduce GHG emissions as well as toxins into the environment [149]. Two 60 kW Capstone C60 micro-turbine systems are installed to provide electricity and thermal energy. Through the heat recovery system, hot water produced is for the wine processing, and the other part of thermal energy is adsorbed by a 70 kW Nishiyodo adsorption chiller to provide space cooling. In addition, a Dolphine pulsed power system is used in the EvapCo cooling tower. With a total capital cost of $210,000, the estimated payback period is 6–8 years, which means $25,000–$38,000 per year. In [150], the authors provide experimental results for a real small-scale CCHP system operating at full load and partial load. This test plant, consisting of a 100 kW natural gas-powered micro-turbine and a liquid desiccant system, is installed at the Politecnico di Torino, Turin, Italy. Comparisons of primary energy savings (PESs) among different prime mover load situations is made. The data shows that adopting the partial load strategy can cause an energetic performance decrease. Katsigiannis *et al.* [151] conduct two case studies in the indoor Swimming Pool Building, and the Law School Building in the Democritos University of Thrace, Greece, to investigate their systematic computational procedure for assessing a small-scale trigeneration system. The procedure includes an indirect estimation of pertinent loads, indoor swimming pool heating, CCHP facility selection, system sizing and economic evaluation. Other applications and test work can be found in [152–154].

In the theoretical work, Chicco *et al.* [155] summarize some key issues and challenges of the planning and design problems for a small-scale trigeneration system. Time domain simulations are conducted to assess each energy vector production within the system by introducing new performance indicators, i.e., trigeneration primary energy saving (TPES), electrical-side incremental trigeneration heat rate (EITHR), thermal-side incremental trigeneration heat rate (TITHR) and cooling-side incremental trigeneration heat rate (CITHR). Other energetic, economic and thermodynamic analyses can be referred to [128, 156, 157]. In [128], the authors investigate an innovative natural gas based CCHP system, whose electricity, heating and cooling capacities are 126 kW, 220 kW and 210 kW, respectively. The gas-fired IC engine works in pairs with a liquid LiCl/water desiccant cooling system. The authors also give energetic and economic analyses, including the influence of the fuel and electric price can make, and indices variations due to the plant cost, of this system. Some researchers focus on optimization problems involved in the small-scale CCHP system design. Abdollahi *et al.* [158] propose a multi-objective optimization method for a small-scale distributed CCHP system design. The environmental impact objective function is defined to be the cost. An economic analysis is conducted using the total revenue requirement (TRR) method. They adopt the genetic algorithm (GA) to find a set of Pareto optimal solutions; and apply the risk analysis to complete the decision-making to find the optimal solution from the obtained set. In [159], an optimization problem of the energy management in the CCHP system is solved by the mixed integer linear programming (MILP). The solution aims to control the on/off status of system components. A comparison, whose data is collected from a 985 kW plant, is made between the proposed energy management and conventional management. Hossain *et al.* [160] present a design and a construction of a novel small-scale trigeneration system driven by neat non-edible plant oils, including jatropha, jojoba oil, etc. The use of local available non-edible oil can leave this plant run without depending on imported petroleum fuel, which results in a high economical efficiency. Moreover, GHG emissions can be dramatically reduced by using the rejected heat from the prime mover to provide cooling and heating.

1.4.3 Medium-scale CCHP systems

As mentioned, medium-scale CCHP systems are those with rated power ranges of 1 MW–10 MW. From this level of rated size, CCHP systems begin to operate in large factories, hospitals, schools etc. A 4.3 MW CCHP plant has been severing the Elgin Community College, Elgin, Illinois, since 1997 [3]. The first phase of this plant, which was a 3.2 MW CCHP plant, was installed in 1997 to provide electricity, low pressure steam and absorption cooling to main campus buildings. In 2005, due to the campus expansion, the generation set and the absorption chiller were both expanded in the second phase. The prime mover in this system is a combination of four 800 kW Waukesha reciprocating engines and one 900 kW Waukesha reciprocating engine. Cooling is provided by one YORK 1,934 kW absorption chiller and one Trane 2,813 kW absorption chiller. The heat recovery equipment includes five Beaird heat recovery silencers and five Beaird exhaust silencers. The two phases cost $2,500,000 and $1,200,000, respectively. The scheme of this system can be shown in Figure 1.13. Another medium-scale CCHP application is the one with a capacity of 3.2 MW installed in Mountain Home VA Medical Center, Mountain Home, Tennessee, in 2011 [161]. This medical center serves for 170,000 military veterans in surrounding countries. The whole plant consists of one 3.2 MW dual-fuel engine generator set,

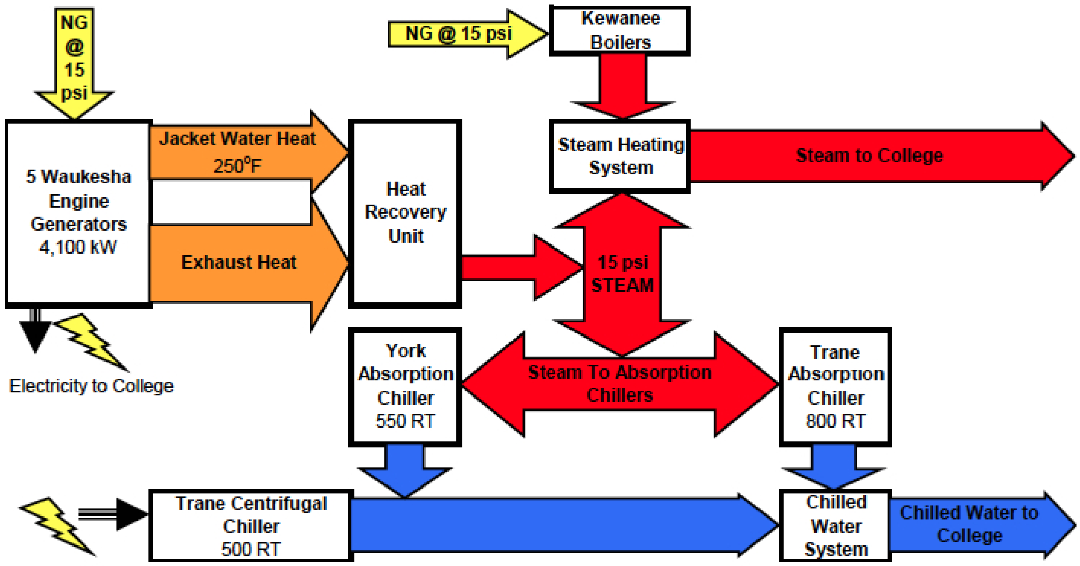

Figure 1.13: *The 4.3 MW CCHP system installed in the Elgin Community College [3].*

fired by landfill bio-gas, two 1.8 MW back up diesel-fired engine generator sets, a heat recovery steam generator, and a 3.5 MW absorption chiller. With this plant, the estimated cost savings over 35 years can be \$5–15 million. In the University of Florida, a 4.3 MW CHP plant began to serve the Shands HealthCare Cancer Hospital since 2008 in order to solve the problem of increasing electricity and fuel prices, to reduce budget, and to reduce GHG emissions. The total installation costs \$45 million. This system, designed by Gainsville Regional Utilities (GRU), consists of one 4.3 MW combustion turbine, one 6.5 t/h heat recovery steam generator, one 4.2 MW steam turbine centrifugal chiller, two 5.3 MW electric centrifugal chillers and one 13.6 t/h packaged boiler. The 4.3 MW natural gas turbine provides 100% of the hospitals electric and thermal needs. Moreover, a total thermal efficiency of 75% can be achieved. Other applications of medium-scale CCHP systems can be found in [162–164], to name a few.

Moreover, researchers also concern on the theoretical research on medium-scale CHP/CCHP systems. In [165], in order to raise the energy efficiency, the authors propose a trigeneration scheme for a natural gas processing plant by installing a turbine exhaust gas waste heat utilization. This trigeneration system makes use of the rejected heat from the gas turbine to generate process steam in a waste heat recovery steam generator. A double-effect LiBr/water absorption chiller is driven by the process steam to provide space cooling; while another part of the process steam is used to meet furnace heating load and to supply plant electricity in a combined regenerative Rankine cycle. The measured CCHP power output is 7.9 MW. In [166], the authors design a CCHP system for a business building in Madrid, Spain. Basic demands of this building are 1.7 MW of electricity, 1.3 MW of heating and 2 MW of cooling. By designing the operation strategy and optimizing facility capacities, the final design of the configuration is given to be an integration of three 730 kW IC engines, one 3 MW double-effect absorption chiller, one conventional chiller of 4 MW, and one 200 kW boilers for back up. Because of the incorporating of a thermal solar plant, the capital cost is 3.32 M€, which is expected to be paid back in 11.6 years. Compared to conventional trigeneration systems, PESs in this plant increase a lot due to the incorporation of the thermal solar plant into the trigeneration plant. In [167, 168], the authors propose a methodology for thermodynamic and thermoeconomic analyses of a trigeneration system equipped with a Wartsila 18V32GD model 6.5 MW gas-diesel engine. This system is installed in the Eskisehir Industry Estate Zone, Turkey. Efficiencies of energy, exergy, Public Utility Regulatory Policies Act and equivalent electrical of the trigeneration system are determined to be 58.94%, 36.13%, 45.7% and 48.53%. This CCHP system can also be transplanted to an airport to provide cooling, heating and electricity. Other theoretical work can be referred to [169, 170].

1.4.4 Large-scale CCHP systems

Large-scale CCHP systems are categorized to be the ones with output power of above 10 MW. This type of CCHP system can provide substantial electricity for industry use, and vast heating and cooling for universities and residential districts, which with a high density of people. So far, as the problem of GHG emissions and increased price of electricity and fuel, an increasing number of large-scale CCHP systems have been installed to serve. The University of Michigan, Ann Arbor, Michigan, began to adopt the cogeneration system in 1914 [171]. Combined with absorption chillers, this 45.2 MW CHP plant consists of six conventional gas-/oil-fired boilers from Combustion Engineering, Wickes, Murray and Foster Wheeler (in total of 453.6 t/h of steam capacity); three Worthington back-pressure/extraction steam turbine generators (rated at 12.5 MW each); two gas/oil solar combustion turbines (3.7 and 4 MW, respectively); and two Zurn heat recovery steam generators (HRSG) with supplemental gas firing (29.5 t/h each). The electricity production of this plant can rarely reach the maximum capacity, for the plant has to provide steam for other use, such as, in summer, the absorption chiller. The system installed saved the university $5.3 million in 2004. In San Diego, California, the University of California at San Diego installed a 30 MW polygeneration plant in 2001 [172]. The 30 MW combined cycle is composed by two 13.5 MW Solar Turbines Titan 130 gas turbine gen-sets and a 3 MW Dresser-Rand steam turbine. The rejected heat is used to run a steam driven centrifugal chiller; to provide domestic hot water for campus use; and to run the steam turbine for additional electricity production. The whole system can achieve 70% gross thermal efficiency. Annually, by installing this set of system, $8–10 million can be saved. In this site, an emission control system, i.e., SoLoNOx$^{\mathrm{TM}}$, is adopted to control the level of NO_x emissions to 1.2 ppm, which is much lower than the permitted 2.5 ppm. Another classic large-scale CCHP application is the plant installed in the

University of Illinois at Chicago [173]. This plant, established from 1993 to 2002, is separated into two parts: The east campus system and the west campus system. In the east campus CCHP plant, two 6.3 MW Cooper-Bessemer dual-fuel reciprocating engine generators and two 3.8 MW gas reciprocating engine generators are installed as the prime mover. Cooling is provided by one 3.5 MW Trane two-stage absorption chiller, two 7 MW YORK electrical centrifugal chillers and several remote building absorption chillers. The capital cost of $25.7 million is estimated to be paid back in 10 years. PESs, CO_2 reductions, NO_x reductions and SO_2 reductions can achieve 14.2%, 28.5%, 52.8% and 89.1%, respectively. In the west campus, because of the large energy demand in the hospital and several buildings, an additional 37.2 MW generation set, composed by three 5.4 MW Wärtsilä gas engines and three 7 MW Solar Taurus turbines, is added. Besides the prime mover, an additional 7 MW absorption chiller is also installed in the west campus CCHP plant. With the capital cost of $36 million, the payback period is estimated to be 5.1 years. Other applications of large-scale CCHP systems can be found in [174–176].

1.5 Development & Barriers of CHP/CCHP Systems in Representative Countries

1.5.1 The United States

The U.S. government began to develop CHP/CCHP plants since 1978, when the Public Utility Regulatory Policy Act (PURPA) was proposed. In the PURPA, utilities are required to interconnect with and purchase electricity from cogeneration systems, in order to give industrial and institutional users access to the grid and allow excess electricity to be sold back. With the help of the PURPA and the federal tax credit for CHP investment, the installed capacity of CHP/CCHP systems grew to 45 GW in 1995 from 12 GW in 1980. Due to the intense competition and instability in the electricity market, the development of CHP/CCHP plants slowed down in 1990s. Only 1 GW installed capacity increased from 1995 to 1998. To boost the development, together with the Environmental Protection Agency (EPA), the U.S. Department of Energy (DOE) proposed the "Combined cooling heating & power for buildings 2020 vision", which aimed to double the installed capacity in 2010. Following the proposed document, the installed capacity grew significantly to 56 GW in 2001. Then in 2004, with a total installed capacity of 80 GW, the goal of 92 GW has been almost achieved. In 2009, after the Energy Policy Act in 2005, the installed capacity has achieved 91 GW. The development trend of the U.S. CHP/CCHP installed capacity from 1970 can be shown in Figure 1.14. Till 2011, as shown in Figure 1.15, 30% CHP/CCHP

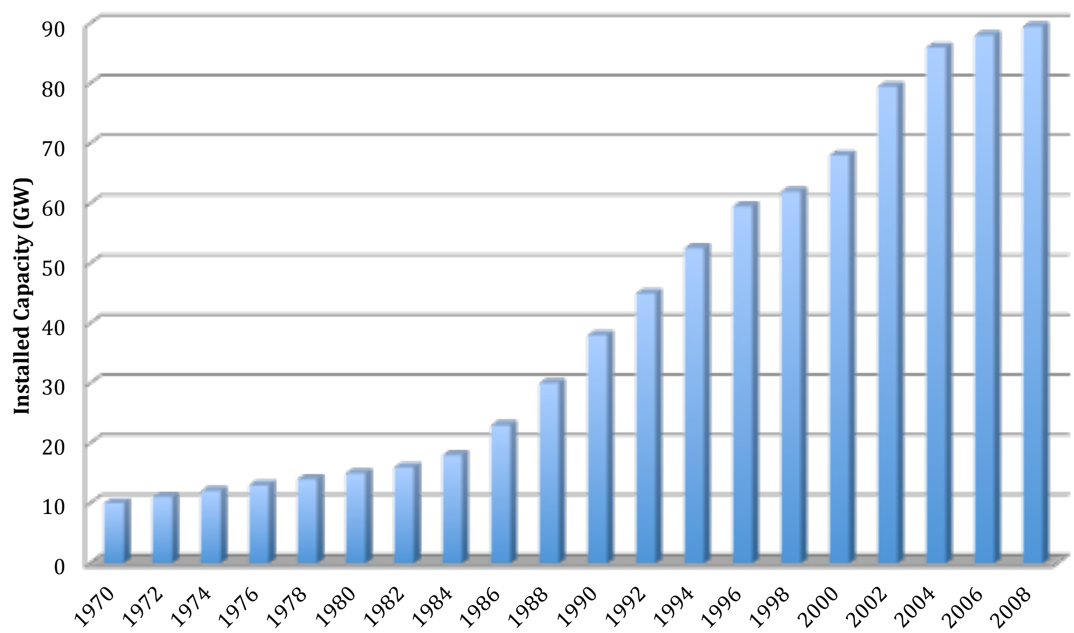

Figure 1.14: *U.S. CHP/CCHP development from 1970 [4].*

installed capacity is used in chemical industries, 17% is used for petroleum refining, 14% for paper industries, 12% for commercial or institutional buildings, 8% for food manufacturing, 8% for other manufacturing, 5% for primary metals industries and 6% for other industries. According to "the White Paper on CHP in a Clean Energy

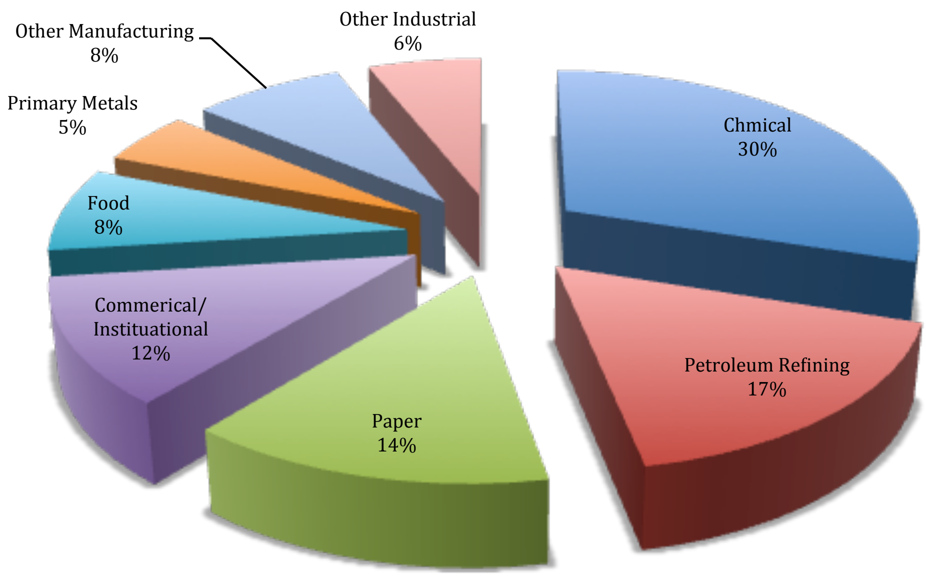

Figure 1.15: *The installed capacity of CHP/CCHP plants classified by applications in the U.S..*

Standard" [177], the U.S. DOE aims to have an 11% increase, from current 9%, of CHP share of the U.S. electric power by 2030. By doing so, 60% projected increase in U.S. carbon emissions can be avoided; over 1 million new, highly skilled jobs can be created; and $234 billion in new investment can be generated.

However, there still exist some barriers to further develop CHP/CCHP plants in the U.S.. The first one is the high capital investment of CHP/CCHP plants. A firm may not have enough budgets to invest in such a high capital cost plant; or, if not sure about the payback of such a plant, it still cannot invest in it. Second, to keep a connection with the utility grid to supply power needs beyond the self generation capacity, extra charges will be made for this connection. This will with no doubt reduce the money-saving potential of CHP/CCHP plants. Third, nonuniform interconnection standards make it difficult for manufacturers to provide CHP/CCHP components. In addition, some policies, such as the Clean Air Act's New Source Review, only consider short-term carbon emissions instead of a long-term and overall vision. Because "the CHP/CCHP can increase onsite air emissions even as it reduces total emissions associated with the facilities heat and electricity consumption" [178], the development of CHP plants can be restricted by such regulations. Finally, further theoretical research, development and demonstrations should be conducted to find out more efficient operation modes, system configurations and system components.

1.5.2 The United Kingdom

In the United Kingdom, the number and installed capacity of CHP/CCHP plants increased dramatically from 1999 to 2000, during which the UK government took methods of fiscal incentives, grant support, regulatory framework, promotion of innovation, and government leadership and partnership to support the development of CHP/CCHP. Before 2000, the installed capacity kept around 3.5 GW, while in 2000, it increased to 4.5 GW. From then on, the UK government continuously drafted a series of policies to target at achieving 10 GW *of good quality* installed CHP plants. In the end of 2010, the total installed capacity in the UK reached 6 GW, which can be shown in Figure 1.16. Classified by applications, as shown in Fig. 1.17, 38% of the installed capacity is used for oil and gas terminals and refineries, another 30% is used for chemical industries and only 4% is used for community usage, etc. In [179], it is also pointed out that "the *of good quality* CHP will be a key technology in helping to deliver our carbon budgets while the grid decarbonises, and will still play a pivotal role in providing secure and cost-effective energy supplies, particularly for industry. The government will continue to promote the development of *of good quality* CHP in

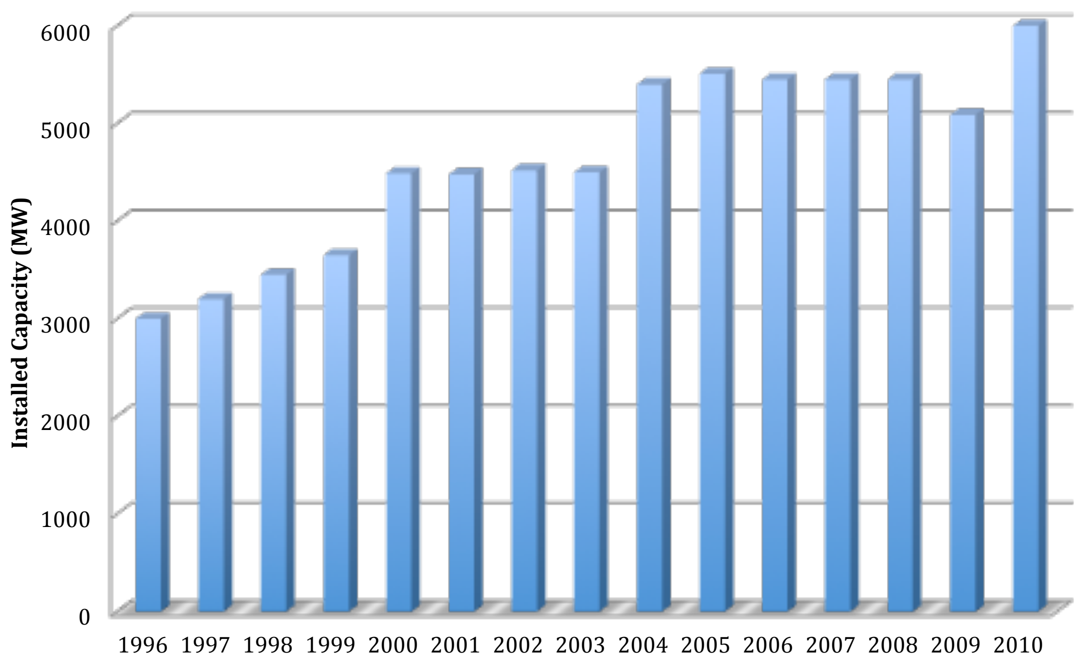

Figure 1.16: *The CHP/CCHP installed capacity in the UK [5].*

the UK."

Meanwhile, there are still some obstacles for CHP/CCHP to be further developed in the UK. The first one is the inconsistency between incentive frameworks and market signals. One significant character of the UK market is the price volatility. The differential between electricity and gas prices put the investment of CHP/CCHP in an uncertain situation. This issue may be addressed by the Climate Change Levy. Second, in theory, the establishment of the carbon market should directly support the expansion of CHP/CCHP capacity. However, due to the unstable carbon price and uncertain allocation arrangements of CHP/CCHP plants, this theory has not yet been verified. Only with a robust price signal and a stable carbon market, the direct relationship between the CHP/CCHP expansion and carbon market can be established. In addition, lacking of locational signals for heat utilization also shield the development of CHP/CCHP plants. Moreover, to achieve the peak efficiency,

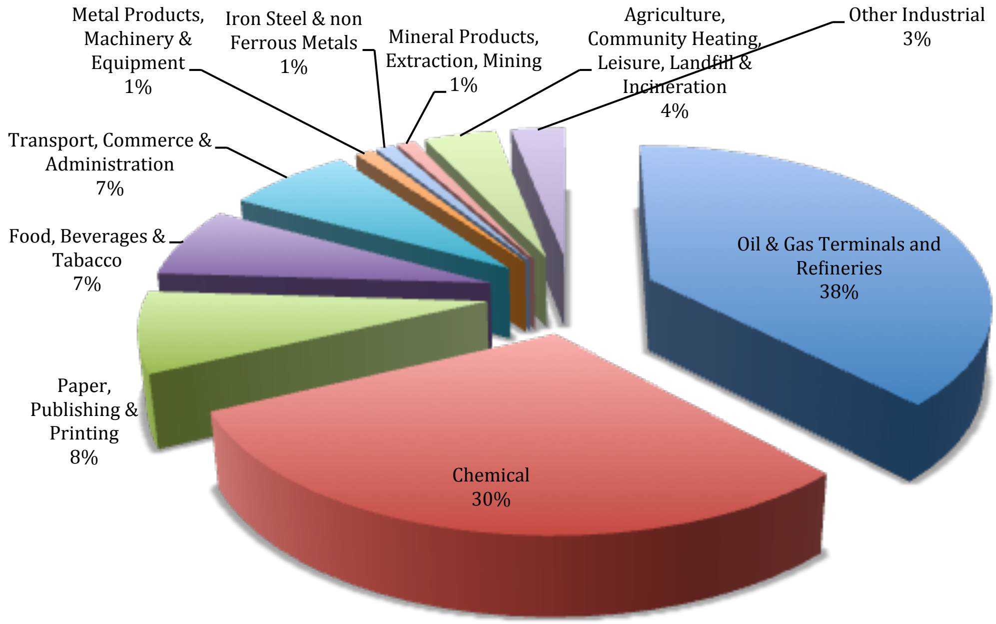

Figure 1.17: *The installed capacity of CHP plants classified by applications in UK [5].*

heat transmission and distribution network should be completed. Finally, insufficient incentive to investment in heat distribution infrastructures also slows down the pace of the CHP/CCHP development in the UK.

1.5.3 The People's Republic of China

Due to the Reform and Open Up to the Outside World Policy, the rapid development of economics, technology and industry leads China to be the well known second-largest energy consumer and carbon emitter in the world. To solve the problem of the increasing demand for primary energy, China has issued a series of policies, including the Energy Saving Law, the Renewable Energy Law, the Air Pollution Prevention Law and the Environment Protection Law, to support the development of CHP/CCHP plants since 1980s. In addition, accompanying with those laws, some standards, e.g., Energy Efficiency Standards for Buildings, Energy Efficiency Standards for Appliances, etc.; and some dedicated funds, subsidies and discounted loans for energy efficiency investments are also carried out by the Chinese government. These steps make China become the second-largest country in terms of installed CHP capacity. In 1986, the Notice on the Report Regarding the Work on Strengthening Urban District Heat Supply Management enhanced the urban district heating supply management. The China Energy Conservation Law, drafted in 1997, listed CHP as a key national energy conservation technology that should be encouraged. The 1998 Some Regulations for CHP Development considered the ratio between heat and electricity as an important indicator to define and approve new CHP. In 2004, the China Medium- and Long-Term Energy Development Plan considered CHP/district heating and cooling (CHP/DHC) as an encouraging technology and named CHP as one of the 10 key national energy conservation programmes. In 2006, the NDRC's China Energy Conservation Technology Policy Outline recommended that CHP should take place of small heating boilers; and they should be developed in large- and medium-sized cities in north heating areas. The 2007 Implementation Scheme of the National 10 Key Energy Conservation Projects further specified important applications and supporting policies for CHP. Another important policy that could boost the development of CHP/CCHP plants in China is the Industrial Guidance Catalogue for Foreign Investments drafted in 2007. This policy encouraged foreign investments and operations of CHP/CCHP power stations in China.

In 1990, the total installed CHP capacity in China was only 10 GW. After ten years' construction and development, with an annual growth rate of 11.6%, a goal of 30 GW installed capacity was achieved in 2000. By 2005, almost 70 GW of capacity had been installed. Till 2006, over 2,600 CHP plants with over 80 GW capacity were installed in China. The development trend can be shown in Figure 1.18. The share of CHP capacity in thermal power generation in China is shown in Figure 1.19.

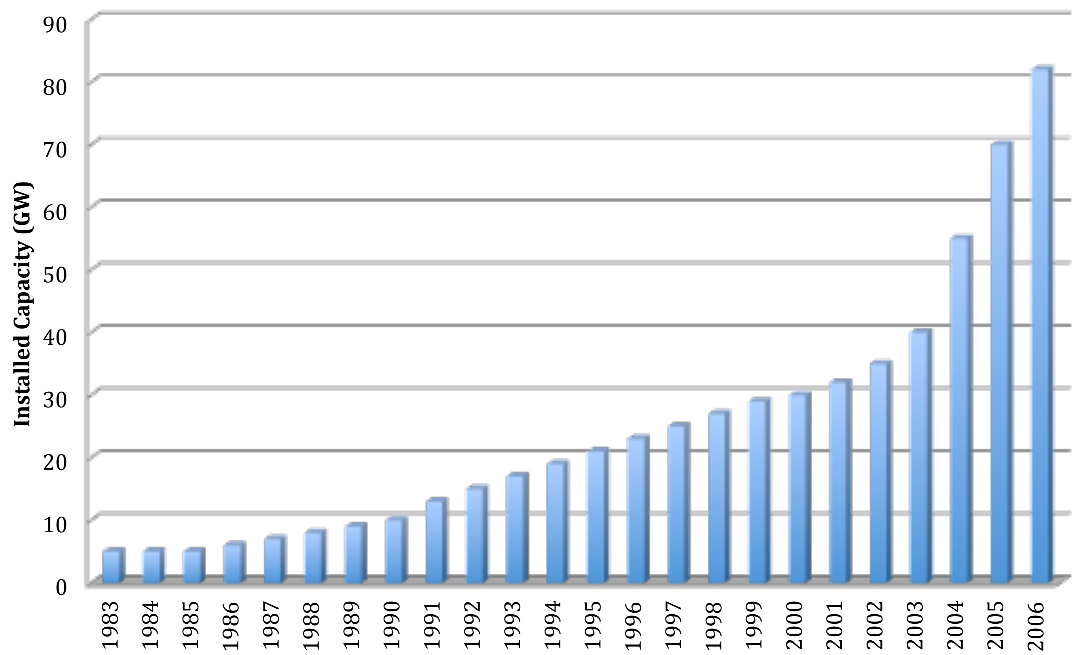

Figure 1.18: *The installed capacity of CHP in China [6].*

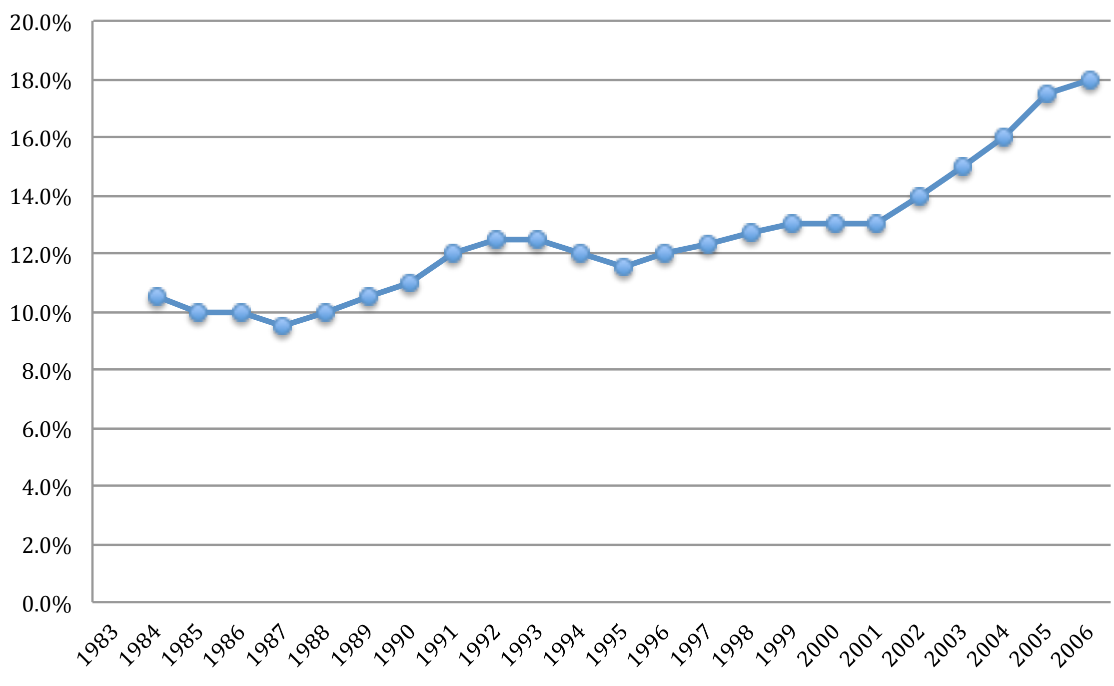

Figure 1.19: *Share of CHP capacity in thermal power generation [6].*

With no doubt, CHP/CCHP is a promising solution for the energy short and air pollution problem, however, there are still some barriers to further develop CHP/CCHP plants in China. The first one to be solved in urgent is the reform of energy price policies. In China, even though the coal price, which increases dramatically, is based on the market, the price of electricity, which slightly increased, is decided by the government. Unbalance increasing rates between the prices of coal and electricity severely restrict the development of CHP/CCHP in China. Besides the reform of energy price policies, heating and power sector reforms also need to be further developed. Not only the economic and price aspect, but also some favorable fiscal and tax incentives should be proposed to support the construction of CHP/CCHP. In addition, since some newly-built CHP projects are operating only in thermal generation mode after established, energy efficiencies of these plants are significantly reduced. Thus, the monitoring and enforcement of the government should be enhanced. Finally, due to an increasing number of large and more efficient CHP plants, some old, small but quite efficient CHP plants are forced to shut down. Hence, some policies that suitable for the small but efficient units should be drafted to keep the whole efficiency.

1.6 Contributions

Though there are numerous results on CCHP systems in the literature, the minimum energy wasted or matrix modelled optimized operation strategies have not been fully studied. The motivations of the thesis are two-fold:

- **"Balance" space based operation strategy design & PGU sizing**. As mentioned, once the system configuration is selected, the performance of the whole CCHP system depends on the operation strategy and facility size. Two classical operation strategies, i.e., FEL and FTL, will inherently waste a certain amount of energy. Though some literature, aiming to solve this problem, adopt the combined FEL and FTL strategy or FEL/FTL switching strategy, as long as the demand point away from the "balance" plane, energy waste is inevitable and vast. Thus, aiming to obtain a larger space for electric demands, cooling demands and heating demands to match, the influence of the varying electric cooling to cool load ratio on the "balance" plane is studied. By tuning that ratio, the "balance" plane can be expanded to be a "balance" space. Thus, the energy waste space can be significantly reduced. Since the partial load efficiency and facility capital cost are both concerned in the objective function and modelling process, respectively, the PGU capacity indirectly affects the system performance. Hence, the PGU capacity should be sized to corporate with the proposed operation strategy.

- **Optimal operation strategy design using a matrix approach & PGU sizing**. Matrix, as a strong mathematic tool, has been widely used in various areas, e.g., economic analysis, modern control theory, signal processing, etc. So far, limited work has been done to investigate the operation strategy design of CCHP systems using matrix approaches. One of the advantages of the matrix approach is that matrix can deal with multi-input multi-output problems with multi-input single-output, single-input multi-output and single-input single-output as special cases. Numerous optimization algorithms can be adopted to tackle a matrix modelled optimization problem. Thus, presented in a matrix form, the system model can be compact, comprehensive, and easier to be optimized. Power flow inside the system and the electric cooling to cool load ratio are inherently included in the system model. In addition, with the objective function set to be a linear function of the optimizer, the optimization problem constrained by equality and inequality constraints, can be solved by an appropriate algorithm.

1.7 Thesis Organization

This thesis is organized as follows. Chapter 1 introduces advantages and the working principle of CCHP systems. In addition, as equally important, some core components, various system configurations and applications, and development and barriers in representative countries of CCHP systems are also introduced in Chapter 1.

In Chapter 2, aiming to reduce the inherent energy waste and to improve system efficiency, the classical absorption chiller only based CCHP system is modified to be the one with hybrid chiller, i.e., an integration of an electric chiller and an absorption chiller. As a result, by tuning the electric cooling to cool load ratio between the electric chiller and the absorption chiller, an operation strategy, aiming to expand the "balance" plane, meshed by the electric demands, cooling demands and heating demands, to be a "balance" space, in which the mentioned three demands can match with each other in more energy consumption patterns, is proposed. Then, with the proposed operation strategy, the PGU is sized to achieve the maximum value of the objective function, defined as a weighted summation of PESs, hourly total cost savings (HTCSs) and carbon dioxide emissions reduction (CDER). Finally, a case study is presented to illustrate the feasibility and effectiveness of the proposed operation strategy and PGU size.

Although also designing the operation strategy, different from the method in Chapter 2, matrix modelling approach is adopted to model the CCHP system in Chapter 3. The concept of *energy hub* is used to transform the system into an input-output model. With the matrix form, the operation strategy, power flow and energy input can be optimized by various optimization algorithms. With a linear objective function set using a similar idea as in Chapter 2, the optimization problem with equality and inequality constraints is solved by the sequential quadratic programming (SQP) approach. Moreover, the PGU is sized to annually optimize the system performance. Using the same energy consumption data as in Chapter 2, a case study is conducted to compare the proposed optimal operation strategy with FEL and FTL.

Chapter 4 summarizes the work in this thesis, and presents some possible future work.

Chapter 2

"Balance" Space Based Operation Strategy Design and PGU Sizing

2.1 Introduction

Performance and efficiencies of CCHP systems mainly depend on system structures, operation strategies and choices of facility capacity. This chapter proposes an improved structural configuration of a CCHP system installed a hybrid chiller, consisting of a combined electric and absorption chiller, whose electric cooling to cool load ratio varies according to different electric and thermal loads in every hour. Energy waste reduced operation strategies, based on the variational electric cooling to cool load ratio, for CCHP systems with unlimited and limited PGU capacity is investigated. Under the proposed operation strategy, PGU capacity is optimized to achieve the optimal annual performance. In addition, a case study of a hypothetical building in Victoria, B.C., Canada is conducted to verify the feasibility of the proposed CCHP system structure and the corresponding energy waste reduced operation strategy.

2.1.1 Background and Related Work

CCHP systems, which can also be referred to as trigeneration systems, are broadly employed in small-/medium-scale power plants in order to achieve economical efficiency and less contamination [1, 112, 180]. The main idea of the CCHP system is to make use of the excess heat rejected from the PGU to regenerate thermal energy that can be used to compensate for the user's energy demand gap [181–183]. CCHP systems have been widely introduced into various kinds of applications, such as office buildings, hotels and hospitals [42, 43, 184].

Classical CCHP systems adopt PGUs to generate electricity, and the rejected heat from PGUs is recovered by a heat recovery system to provide thermal energy for cooling and heating loads. Thermally activated technology is adopted in the CCHP system. As shown in Figure 1.1, the sorption chiller is installed to absorb or adsorb the recovered heat to meet cooling demands [123, 185]; the heating unit reheats the recovered heat to meet heating demands. A CCHP system can be regarded as a generalized model of a CHP system – a proven and reliable technology [186, 187]. When operating in winter, with no chiller running, a CCHP system degrades to a CHP system. The goal of designing CCHP systems is to reduce primary energy consumptions, annual total cost and carbon dioxide emissions as much as possible. To achieve this, the following issues should be incorporated into the design consideration: The system operation strategy, the individual component efficiencies, and the user's demands for cooling, heating and power [188].

Typically, there are two basic operation strategies, i.e., FTL and FEL [189, 190]. They can also be referred to as the thermal demand management (TDM) and the electric demand management (EDM) [191]. When operating at FTL mode, the CCHP system satisfies the user's thermal load first; if the by-product electricity cannot meet electric demands, additional electricity should be purchased directly from the local grid. When operating at FEL mode, it provides sufficient electricity for the user first, and then, if the by-product heat cannot meet thermal demands, an auxiliary boiler, combusting the fuel, will be activated. However, both FEL and FTL strategies will inherently waste a certain amount of energy. This is because, for instance, when the CCHP system runs under the FEL strategy to provide enough electricity for the user, if the thermal demand is less than the thermal energy PGU provides, excess thermal energy will be wasted. It is a similar case for the FTL strategy. Comparisons and analyses of the two strategies are investigated in [38, 180, 191–198], to name a few.

In order to reduce the energy waste and to reduce primary energy consumptions, annual total cost and GHG emissions, it is necessary to design an optimal operation strategy. Due to different definitions of "optimal", the operation strategies designed are different. In [199], the author proposes an optimal operation strategy for an offline nonlinear model, i.e., TOOCS-off, of a CCHP system. This optimization model considers the electric and thermal load in each time interval, prices of electricity sold to costumers or purchased from utility, and prices of heating and cooling. In the cost function of the TOOCS-off model, the total economic benefit of this system is maximized during total daily operation time. In the constraints, facilities' thresholds and output upper bounds are considered simultaneously. An CCHP system with a capacity of 143 kW, equipped with a 450 kW auxiliary boiler, a 600 kW absorption chiller and a 800 kWh content heat storage tank, is used to verify the feasibility of this offline model and the optimal operation strategy. Based on source primary energy consumption (PEC), Fumo *et al.* [200] analyze four CCHP system operation conditions, including power and cooling without requiring boiler operation (in spring/fall), power and cooling requiring boiler operation (in summer), power and heating without requiring boiler operation (in spring/fall) and power and heating requiring boiler operation (in winter). The results of this study can contribute to the design of operation strategy to reduce undesired increase of energy consumptions. In [180], the authors design an optimal operation scheme for a CCHP system by considering the PEC and emissions of pollutants besides the energy cost. The operation is optimized by an optimal energy dispatch algorithm. The evaluation of the performance of a CCHP system, operating under the proposed strategy, is conducted using five cities' realistic climate data. Cardona *et al.* [201] proposed a profit-oriented optimal operation strategy, considering both of the articulated energy tariff system and the technical characteristics of components. In [202], instead of the profit-oriented strategy and the primary energy-oriented strategy, the authors adopt an emission-oriented strategy in order to reduce GHG emissions. The control scheme in the proposed strategy is an on-off control, i.e., if the level of GHG emissions is greater than a specific value, then the PGU should stop; otherwise, the PGU runs to meet the energy demand. A comparison of GHG emissions of the proposed strategy, profit-oriented strategy and primary energy-oriented strategy is made to show the effectiveness of the proposed operation strategy. In [203], the authors propose an FEL/FTL switching operation strategy for the CCHP system. An integrated performance criterion, including PEC, carbon dioxide emissions (CDE) and cost (COST), is used to determine the switching action between FEL and FTL strategies. However, the inherent energy waste still exists. In [204], by considering the uncertainties of the price of the purchased electricity, the delivered demand for electricity, and the marginal cost of self generation, the authors propose an operation strategy design method using a risk management approach. By using the risk metrics, the steam and gas turbine generated electric power, the benefits and costs can be forecasted. Moreover, an optimal control tool, i.e., the model predictive control (MPC), is also used to schedule the operation strategy. Mago *et al.* [189] propose an optimized operation strategy, which can be referred to as following the hybrid electric-thermal load (HETS). The analyses and evaluations show that, when operating under the HETS, a CCHP system can perform better in the aspects of PEC, operational cost and CDE, when compared to FEL and FTL strategies. In [205], the optimization of the operation strategy is formulated to be a linear programming (LP) problem with the objective function set to be the operation variable cost. This problem is constrained by capacity limits, equipment efficiencies, energy balance equations and demand constraints. The obtained optimal operation strategy is classified in nine operational modes due to the price of electricity from grid, electricity sold back price, auxiliary heat and waste heat. A thermoeconomic analysis, based on the marginal cost, is also conducted to investigate the relationship between the optimal operation mode and energy demands, as well as the prices of consumed resources. Aiming to maintain the system autonomy to ensure the grid reliability and to minimize excess power production, Nosrat *et al.* [206] propose a dispatch strategy for a PV-CCHP system, in which the thermal energy waste can be significantly reduced. Decision-making of this dispatch strategy depends on the output of the PV array and is separated into four steps. In each step, several operation conditions are analyzed to choose the strategy between FEL and FTL. The results show that an improvement of 50% can be achieved by using this dispatch strategy. Because the strategy is chosen from FEL and FTL directly, the inherent energy waste still cannot be avoided.

2.1.2 Motivations

As stated in [207], CCHP systems operate at peak efficiency when thermal and electric loads are well-matched. Thus, one of the objectives in this chapter is to design a high efficiency operation strategy to meet the thermal and electric loads with less energy consumption and carbon dioxide emissions, and to match the loads in a relatively larger space. The strategy design depends on the user's energy consumptions, and also electricity and fuel prices.

In [37, 38], the authors propose a new CCHP system structure different from conventional CCHP system structures, whose cooling load all lands on absorption chillers. The new structure adopts a combination of an absorption chiller and an electric chiller. An electric chiller has a high COP to be around 3, which leads to a high cooling efficiency. However, taking the high rates of electricity into account, an operation strategy needs to be designed to determine the optimal effort of the electric chiller. A parameter called electric cooling to cool load ratio is the one to determine this effort. In the literature, this ratio is chosen to be fixed [37,38]. However, in this chapter, despite the same hybrid chiller is chosen, this ratio is optimized according to variant energy consumptions and energy rates, which has not been investigated in other literature.

Another most concerned problem in designing CCHP systems is the sizing problem, i.e., determining the capacities of facilities. Facilities' capacities in the SP system and chillers, heating unit and boiler in the CCHP system are easy to choose, for their sizes depend on the corresponding energy loads. However, as a result of the complexity of the operation strategy, the PGU capacity in the CCHP system is hard to determine. Considering the prices of facilities, partial load efficiency, external electricity and fuel rates, and facilities' lives, some optimization approaches have been adopted to obtain the optimal PGU capacity, such as the particle swarm optimization [37], the genetic algorithm optimization [38,208] and the mixed integer nonlinear programming (MINLP) algorithm [209, 210]. Other sizing optimization approaches can be referred to [42, 169, 180, 211–224]. In this chapter, since the sizing problem is an off-line one parameter optimization, only the enumeration algorithm is adopted to determine the optimal value of PGU capacity.

2.1.3 Objective and Chapter Organization

In this chapter, an energy waste reduced operation strategy, aiming to expand the "balance" plane, will be designed. Moreover, PGU will be sized to match the proposed operation strategy. This chapter is organized as follows. A description of energy flow in the CCHP system, and the energy efficient operation strategies for CCHP systems with unlimited and limited PGU capacities are presented in Section 2.2. Evaluation criteria used in choosing strategies from candidates are introduced in Section 2.3. The last Subsection of Section 2.3 gives a mathematical model of the optimization problem. A case study in Section 2.4 verifies the feasibility of the proposed energy waste reduced operation strategy and the optimal PGU capacity. Finally, Section 2.5 concludes this chapter.

2.2 Operation strategy design

The system diagram of the CCHP system with a hybrid chiller installed is shown in Figure 2.1. The solid line, dashed line and dot dashed line represent the thermal energy flow, fuel flow and electricity flow, respectively.

The system equations and operation strategy design methods for CCHP systems are separated into the following two parts.

2.2.1 CCHP systems with unlimited PGU capacity

In this section, the scenario in which the PGU has no capacity limitation will be discussed. The idea of the energy waste reduced strategy is to make the electric demands and thermal demands match with each other [207]. The term '*match*' here means that all of the electric and thermal demands of the building is provided by the PGU, namely $F_b = 0$, $E_{grid} = 0$ and electric cooling to cool load ratio $x \in [0, 1]$.

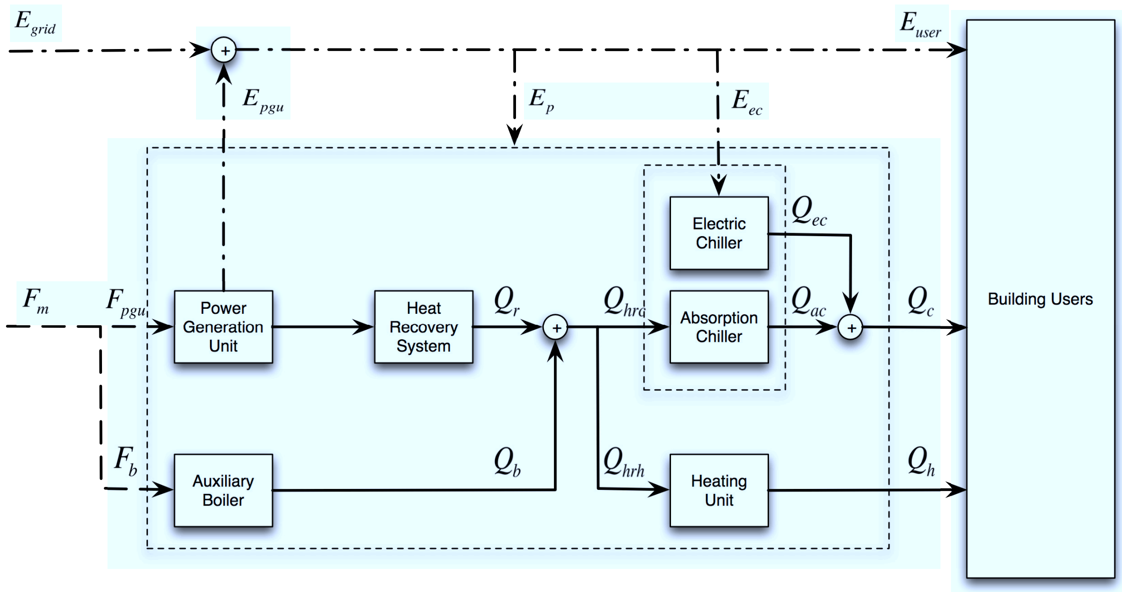

Figure 2.1: *The CCHP system with a hybrid chiller installed.*

Most of the time, according to different energy consumption in different seasons and different hours in one day, the PGU cannot exactly provide the electricity and thermal energy for the building use. For example, following the FEL strategy, if the thermal load is less than that the PGU provides, energy waste is inevitable; if the thermal load is larger than the amount PGU provides, additional fuel should be purchased for the auxiliary boiler. For a CCHP system with fixed electric cooling to cool load ratio, electric demands, cooling demands and heating demands can only match at a fixed plane. However, by introducing the variational electric cooling to cool load ratio, the range for the energy demands to match with each other can be expanded. By tuning the electric cooling to cool load ratio, the electric load and thermal load can be accordingly tuned in a certain range. Since x can only vary in $[0, 1]$, there exists an appropriate range for the E_{user} to vary to match with Q_c, Q_h and E_p. This range can be obtained from two limiting situations.

First, when $x = 0$, the absorption chiller takes all the cooling load. Thus we have

$$Q_{ac} = Q_c \tag{2.1}$$

and

$$Q_{hrc} = \frac{Q_{ac}}{COP_{ac}}, \tag{2.2}$$

where COP_{ac} is the absorption chiller's COP. At the heating unit node, we have

$$Q_{hrh} = \frac{Q_h}{\eta_h}, \tag{2.3}$$

where η_h is the heating efficiency of the heating unit. Then it can be readily derived as

$$\begin{aligned} Q_r &= Q_{hrc} + Q_{hrh} \\ &= \frac{Q_c}{COP_{ac}} + \frac{Q_h}{\eta_h}. \end{aligned} \tag{2.4}$$

Having the heat recovery system efficiency η_{hrs} and the PGU thermal efficiency η_{pgu}, the fuel consumption F_{pgu} can be calculated as

$$F_{pgu} = \frac{Q_r}{(1 - \eta_{pgu})\eta_{hrs}}, \tag{2.5}$$

and electricity provided by the PGU is

$$E_{pgu} = \frac{Q_r}{(1 - \eta_{pgu})\eta_{hrs}}\eta_{pgu}. \tag{2.6}$$

The PGU efficiency varies depending on the PGU load. Usually, low PGU load results in low PGU efficiency. Represented in a second order polynomial, η_{pgu} is

$$\eta_{pgu} = af^2 + bf + c, \tag{2.7}$$

where f is the fraction of the PGU electric load. If the maximum electric load the PGU can take is E_{max}, then we have

$$f = \frac{E_{user} + E_p + xQ_{ec}/COP_{ec}}{E_{max}}. \tag{2.8}$$

To avoid the low system efficiency caused by the low PGU efficiency, a threshold α of the electric load fraction should be set to be an on-off coefficient of the PGU, i.e., if $f < \alpha$, the PGU will be shut down to keep a high system efficiency.

If the electric and thermal demands match, E_{pgu} in (2.6) can meet the electric demand. E_{pgu} can be represented as

$$\begin{aligned} E_{pgu} &= E_p + E_{user} \\ &= \frac{Q_c/COP_{ac} + Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu}. \end{aligned} \tag{2.9}$$

Thus, when $x = 0$, we have

$$E_{user} = \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} + \frac{Q_c/COP_{ac}}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} - E_p. \tag{2.10}$$

Define E_{useru} to be (2.10), we have

$$E_{useru} = \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} + \frac{Q_c/COP_{ac}}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} - E_p. \tag{2.11}$$

In the next step, x is set to be 1, implying that the electric chiller takes all the cooling load and the absorption chiller is left idle. Hence

$$Q_r = \frac{Q_h}{\eta_h}, \tag{2.12}$$

and the fuel consumption F_{pgu} can be readily given by

$$F_{pgu} = \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}. \tag{2.13}$$

At the cooling node, the electric chiller fully provides the cooling demand of the user. Thus $Q_{ec} = Q_c$ and

$$E_{ec} = \frac{Q_c}{COP_{ec}}. \tag{2.14}$$

Since electric and thermal demands match, E_{pgu} provided by F_{pgu} can meet the electric demand including the part required by the electric chiller. Thus we have

$$\begin{aligned} E_{pgu} &= E_p + E_{ec} + E_{user} && \text{(2.15a)} \\ &= \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu}. && \text{(2.15b)} \end{aligned}$$

Then E_{user} can be represented as

$$E_{user} = \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} - \frac{Q_c}{COP_{ec}} - E_p. \tag{2.16}$$

Define E_{userl} to be (2.16), then we have

$$E_{userl} = \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} - \frac{Q_c}{COP_{ec}} - E_p. \tag{2.17}$$

Obviously, there exist three cases to be discussed, i.e., given a set of Q_c, Q_h, E_p and E_{user}, $E_{userl} \leq E_{user} \leq E_{useru}$, $E_{user} > E_{useru}$ and $E_{user} < E_{userl}$. In the rest of the thesis, if no specific statement, E_p is assumed to be a constant. Without losing generality, we set $E_p = 0$.

$E_{userl} \leq E_{user} \leq E_{useru}$

Under this situation, no electricity from the grid ($E_{grid} = 0$) and extra fuel need to be purchased. What need to be done is tuning x to make the electric and thermal demands match. The space for the point (Q_c, Q_h, E_{user}) to vary in this case is shown in Figure 2.2, like point α, which is the one between the two planes E_{useru} and E_{userl}. This space can be called the "balance" space. If without the expansion, the energy demands can only match with each other on one fixed "balance" plane, which is contained in the "balance" space. The angle between the two planes may vary according to different η_{pgu} under different electric loads.

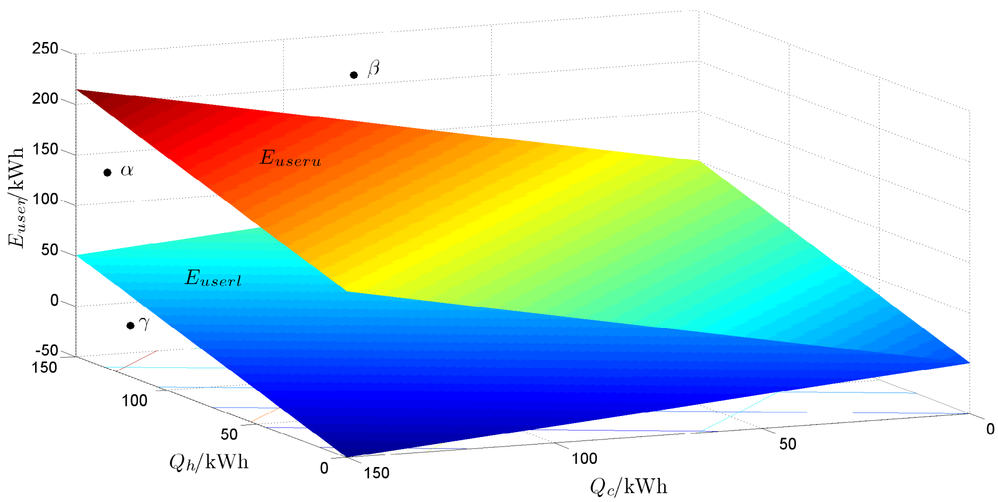

Figure 2.2: *Space of Q_c, Q_h and E_{user}.*

$E_{user} > E_{useru}$

For the second case, $E_{user} > E_{useru}$ implies that, even though x is tuned to 0 to reduce the load of electricity, PGU still cannot meet the electric and thermal demands. The demand point is above the E_{useru} plane in Figure 2.2, e.g. the point β. If adopting the FEL, once the electric demands are met, the thermal energy provided excess the thermal demands. Thus, in order to avoid this energy waste, the FTL strategy is adopted with x set to be 0. From (2.1)-(2.9), we have

$$E_{grid} = E_{user} + E_p - \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} - \frac{Q_c/COP_{ac}}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu}. \tag{2.18}$$

$E_{user} < E_{userl}$

$E_{user} < E_{userl}$ implies that even though electric chiller runs at full capacity, the electricity provided by the PGU still excesses the demand. The demand point, like point γ, is below the plane E_{userl} in Figure 2.2. Because of the much high COP of the electric chiller, x is set to be 1 to make the electric chiller run in full capacity, then reduce the corresponding F_{pgu}; the thermal gap is then compensated by the auxiliary boiler.

The redundant electricity E_{red} is expressed as

$$E_{red} = \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu} - \left(\frac{Q_c}{COP_{ec}} + E_{user}\right), \tag{2.19}$$

and the corresponding redundant fuel F_{red} is

$$F_{red} = \frac{E_{red}}{\eta_{pgu}}, \tag{2.20}$$

which reveals the thermal gap Q_{gap} to be

$$Q_{gap} = F_{red}(1-\eta_{pgu})\eta_{hrs}. \tag{2.21}$$

This thermal gap will be compensated by the auxiliary boiler. Fuel needed to feed the auxiliary boiler is

$$F_b = \frac{Q_{gap}}{\eta_b}, \tag{2.22}$$

where η_b is the efficiency of the auxiliary boiler. From (2.19)-(2.22), we have

$$F_b = \frac{Q_h/\eta_h}{\eta_b} - \left(\frac{Q_c}{COP_{ec}} + E_{user}\right)\frac{(1-\eta_{pgu})\eta_{hrs}}{\eta_{pgu}\eta_b}. \tag{2.23}$$

F_{pgu} will be reduced by F_{red}, which means

$$\begin{aligned} F_{pgu} =& \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}} - F_{red} \\ =& \frac{Q_c/COP_{ec} + E_{user}}{\eta_{pgu}}. \end{aligned} \tag{2.24}$$

Under this situation, the strategy can be regarded as the FEL strategy.

In the above design procedure, even though the FEL and FTL are still in use, the energy waste space has been significantly shrunk. Compared to the fixed "balance" plane operation strategy, the proposed one can significantly improve the system performance.

2.2.2 CCHP systems with limited PGU capacity

In this section, the PGU being discussed can only provide limited electricity and thermal energy, which means that we cannot simply search for the optimal x to make demands match, but need to take the installed capacity of PGU into further consideration. This is a more practical case. Given that the PGU runs at full capacity, eight situations are to be discussed. In the following parts, Q_{pro} and E_{pro} represent the maximum thermal energy and electricity provided by PGU, respectively; Q_{req} and E_{req} represent the thermal and electric demands of the building and the electric chiller, respectively. The optimal matching operation strategies for each of the eight cases are stated in the following eight parts.

$Q_{pro} < Q_{req}$ **and** $E_{pro} < E_{req}$**,** $\forall x \in [0, 1]$

In this scenario, no matter what x is, the PGU can meet neither electric nor thermal demand. Thus PGU should run at full capacity to provide as much energy as it can. Since the hourly evaluation criteria (EC_{hour}) function, which will be discussed in Section 2.3, is a monotonous function of x, according to different primary energy prices and carbon dioxide emissions, two restricted situations need to be compared, i.e., $x = 0$ and $x = 1$.

When $x = 0$, the absorption chiller runs at full capacity. PGU will provide the maximum electricity and thermal energy as

$$E_{pgu} = F_{pgum}\eta_{pgu}, \tag{2.25}$$

$$Q_r = F_{pgum}(1 - \eta_{pgu})\eta_{hrs}, \tag{2.26}$$

where F_{pgum} is the capacity of PGU. Then the boiler will be used to compensate the thermal gap. Fuel consumed by the auxiliary boiler is

$$F_b = \frac{Q_h/\eta_h + Q_c/COP_{ac} - F_{pgum}(1 - \eta_{pgu})\eta_{hrs}}{\eta_b}. \tag{2.27}$$

The electricity shortage should be purchased from local grid which reveals that

$$E_{grid} = E_{user} - F_{pgum}\eta_{pgu}. \tag{2.28}$$

When $x = 1$, the electric chiller runs in full capacity. PGU provides the maximum electricity and thermal energy as shown in (2.25) and (2.26). The thermal gap will be compensated by the heat provided by the auxiliary boiler. Fuel consumption of the auxiliary boiler can be calculated as

$$F_b = \frac{Q_h/\eta_h - F_{pgum}(1-\eta_{pgu})\eta_{hrs}}{\eta_b}. \tag{2.29}$$

Electricity consumed by electric chiller E_{ec} is Q_c/COP_{ec}. Then the purchased electricity is

$$E_{grid} = E_{user} + Q_c/COP_{ec} - F_{pgum}\eta_{pgu}. \tag{2.30}$$

After calculating the EC_{hour} function for the two configurations, the one with the larger result will be chosen to be the optimal solution for the corresponding hour.

$Q_{pro} \geq Q_{req}$ **and** $E_{pro} < E_{req}$**, when** $x = 0$

If $x = 0$, $Q_{pro} \geq Q_{req}$ and $E_{pro} < E_{req}$, as long as we raise the value of x, Q_{req} keeps descending and E_{req} keeps rising. Thus, $\forall x \in [0, 1]$, we have $Q_{pro} \geq Q_{req}$ and $E_{pro} < E_{req}$. The same as the first scenario, the strategy choosing should also be determined by electricity and fuel prices and carbon dioxide emissions. This time, the strategy should be chosen from three candidates.

First, let PGU run at full capacity with $x = 0$. Under this configuration, heat provided by PGU exceeds the thermal demand, but electricity provided is still not enough. Thus, there is no need for the auxiliary boiler to run, but additional electricity must be purchased from local grid, which reveals that

$$E_{grid} = E_{user} - F_{pgum}\eta_{pgu}, \tag{2.31}$$

$$F_{pgu} = F_{pgum}. \tag{2.32}$$

Second, set $x = 0$ and let PGU exactly cover the thermal load, which means that

$F_{pgu} < F_{pgum}$ and $F_b = 0$. F_{pgu} can be calculated from thermal load by

$$F_{pgu} = \frac{Q_c/COP_{ac} + Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}. \tag{2.33}$$

The electricity gap will be compensated by electricity purchased from local grid with

$$\begin{aligned} E_{grid} &= E_{user} - E_{pgu} \\ &= E_{user} - \frac{Q_c/COP_{ac} + Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu}. \end{aligned} \tag{2.34}$$

Finally, set $x = 1$ and let PGU exactly cover the thermal load, which means that $F_{pgu} \ll F_{pgum}$ and $F_b = 0$. F_{pgu} can be calculated from heating load by

$$F_{pgu} = \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}. \tag{2.35}$$

The electricity gap will be compensated by electricity purchased from local grid with

$$\begin{aligned} E_{grid} &= E_{user} + Q_{ec} - E_{pgu} \\ &= E_{user} + \frac{Q_c}{COP_{ec}} - \frac{Q_h/\eta_h}{(1-\eta_{pgu})\eta_{hrs}}\eta_{pgu}. \end{aligned} \tag{2.36}$$

After calculating the values of EC_{hour} function of these three cases, the largest result will be chosen as the optimal operation strategy for the corresponding hour.

$Q_{pro} < Q_{req}$ and $E_{pro} \geq E_{req}$, when $x = 1$

If $x = 1$, $Q_{pro} < Q_{req}$ and $E_{pro} \geq E_{req}$, as long as we reduce the value of x, Q_{req} keeps rising and E_{req} keeps descending. Thus, $\forall x \in [0, 1]$, we have $Q_{pro} < Q_{req}$ and $E_{pro} \geq E_{req}$. In this scenario, if PGU runs at full capacity, no matter what x is, the electricity provided by PGU exceeds the electric demand, and the thermal energy provided is still not enough. As a result, the auxiliary boiler needs to be started to compensate the thermal gap. Here $x = 1$ is adopted to purchase less fuel, and raise the cooling efficiency by using the electric chiller. Fuel required to be purchased is

$$\begin{aligned} F_b &= \frac{Q_{hrh} - Q_r}{\eta_b} \\ &= \frac{Q_h/\eta_h - F_{pgum}(1-\eta_{pgu})\eta_{hrs}}{\eta_b}. \end{aligned} \tag{2.37}$$

Obviously, we have $E_{grid} = 0$ and $F_{pgu} = F_{pgum}$ for this strategy.

$Q_{pro} \geq Q_{req}$ and $E_{pro} \geq E_{req}$, when $x = 0$

If the condition is satisfied $\forall x \in [0, 1]$, then this scenario means that when PGU runs at full capacity, both of the electricity and thermal energy provided by PGU exceed the demands, respectively. Thus we can simply refer to the strategy choosing method in Section 2.2.1. However, this scenario can not only serve this much strong condition, i.e., $\forall x \in [0, 1]$, but also the released condition, only $x = 0$. This is because that, in the worst situation, if when $x = 0$, $Q_{pro} \geq Q_{req}$ and $E_{pro} \geq E_{req}$, and $x = 1$, $Q_{pro} \geq Q_{req}$ and $E_{pro} < E_{req}$, we can always raise x to reduce F_{pgu} by an amount respective to the increase of x. At the same time, electricity provided decreases and electric demands increase. There must exists a point for x, at which $Q_{pro} \geq Q_{req}$ and $E_{pro} = E_{req}$. Then, by reducing x and reducing F_{pgu}, either $Q_{pro} = Q_{req}$ and $E_{pro} = E_{req}$, or $Q_{pro} \geq Q_{req}$ and $E_{pro} = E_{req}$ will be achieved. Hence, in this situation, F_{pgum} can also be regarded as an unlimited capacity. The following procedure will follow the similar line in Section 2.2.1.

$Q_{pro} = Q_{req}$ and $E_{pro} = E_{req}$, $\exists x \in [0, 1]$

This scenario can be expanded as that when $x = 0$, $Q_{pro} < Q_{req}$ and $E_{pro} \geq E_{req}$ and when $x = 1$, $Q_{pro} \geq Q_{req}$ and $E_{pro} < E_{req}$. Since a particular $x \in [0, 1]$ can make electric and thermal demands match, when PGU runs at full capacity, it satisfies our optimal matching conditions stated before. Thus, the calculated x and $F_{pgu} = F_{pgum}$ are chosen as the optimal strategy.

$Q_{pro} < Q_{req}$ and $E_{pro} < E_{req}$, when $x = 1$, and $Q_{pro} < Q_{req}$ and $E_{pro} \geq E_{req}$, when $x = 0$

In this scenario, two candidates need to be compared. The first one is setting $x = 1$ and purchasing both of the electricity from local grid and the fuel. The second one is to find an appropriate $x \in [0, 1]$, which makes the system meet the electric demand with only fuel purchased. The optimal decision is made depending on the EC_{hour} function value.

F_b and E_{grid} in the first option can be calculated by (2.29) and (2.30).

With the second option, no additional electricity from local grid needs to be purchased. But the electric cooling to cool load x should be adjusted to

$$x = \frac{(F_{pgum}\eta_{pgu} - E_{user})\, COP_{ec}}{Q_c}, \tag{2.38}$$

and the purchased fuel is

$$\begin{aligned} F_b = &\frac{\left[Q_c - (F_{pgum}\eta_{pgu} - E_{user})COP_{ec}\right]/COP_{ac} + Q_h/\eta_h}{\eta_b} \\ &- \frac{F_{pgum}(1-\eta_{pgu})\eta_{hrs}}{\eta_b}. \end{aligned} \tag{2.39}$$

By calculating the EC_{hour} function values of the two options, the one with the larger result will be chosen as the optimal solution for the corresponding hour.

$Q_{pro} \geq Q_{req}$ **and** $E_{pro} < E_{req}$**, when** $x = 1$**, and** $Q_{pro} < Q_{req}$ **and** $E_{pro} < E_{req}$**, when** $x = 0$

In this scenario, no matter what $x \in [0, 1]$ is, PGU cannot provide enough electricity for the user and facilities use. However, thermal energy provided by the PGU can exceed the demand when x is set to be 1. Thus, x is tuned to make the PGU, when operating at full capacity, provide exact thermal demand and compensate the electricity gap by purchasing electricity from local grid. x can be calculated as

$$x = 1 - \frac{[F_{pgum}\eta_{hrs}(1 - \eta_{pgu}) - Q_h/\eta_h]\, COP_{ac}}{Q_c}, \tag{2.40}$$

and electricity purchased from the local grid is

$$E_{grid} = \frac{xQ_c}{COP_{ec}} + E_{user} - F_{pgum}\eta_{pgu}. \tag{2.41}$$

$Q_{pro} \geq Q_{req}$ **and** $E_{pro} \geq E_{req}$**, when** $x = 1$**, and** $Q_{pro} < Q_{req}$ **and** $E_{pro} \geq E_{req}$**, when** $x = 0$

When PGU runs at full capacity, electricity provided by the PGU always exceeds the electric demand, but the relationship between the heat provided and the thermal demand varies from smaller to greater as x increases from 0 to 1. The strategy adopted is, with x set to be 1, checking the condition

$$\frac{Q_h}{(1 - \eta_{pgu})\eta_{hrs}\eta_h}\eta_{pgu} \geq \frac{Q_c}{COP_{ec}} + E_{user}. \tag{2.42}$$

If the condition can be satisfied, x is set to be 1 and

$$F_{pgu} = \frac{Q_h}{(1 - \eta_{pgu})\eta_{hrs}\eta_h}; \tag{2.43}$$

if not, we can tell that the situation is the same as the fourth scenario. By a similar procedure of calculating $x \in [0, 1]$ and selecting an appropriate F_{pgu}, which is smaller than F_{pgum}, we have the optimal strategy chosen.

2.3 Evaluation criteria function

To evaluate an energy system, what we concern most are the primary energy consumption, total cost of the system and how much the system would affect the environment. Thus, the most commonly used evaluation criterion is to compare with the SP system. The criterion is divided into three parts, including PES, HTCS and CDER. The diagram of the conventional SP system is as shown in Figure 2.3.

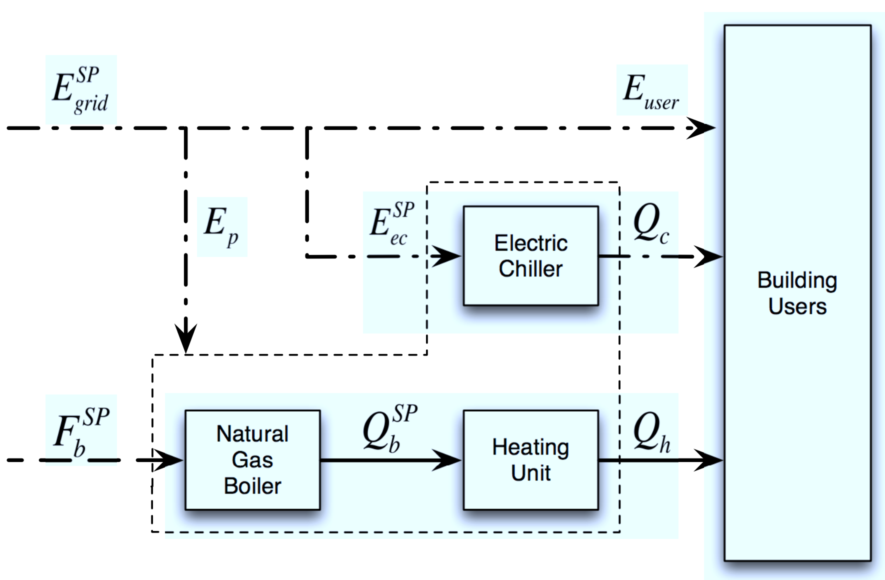

Figure 2.3: Conventional SP system

The SP system runs in the following way: The electric demand of users and facilities is provided by electricity purchased from the local grid; electricity required to drive the electric chiller, which is used to provide the cooling load for users, is also purchased from the local grid; heating load of users is provided by the boiler and heating unit, which are fed by purchased fuel. Thus, the primary energy consumption of the SP system is calculated as

$$F^{SP} = \frac{E_{user} + E_p^{SP}}{\eta_e^{SP}\eta_{grid}} + \frac{Q_c/COP_{ec}}{\eta_e^{SP}\eta_{grid}} + \frac{Q_h/\eta_h}{\eta_b}, \tag{2.44}$$

where E_p^{SP} is the parasitic electricity of the SP system, η_e^{SP} and η_{grid} is the generation efficiency of the grid power plant and efficiency of grid, respectively.

All of the above three criteria are hourly based. In addition, the capital recovery factor is assumed to be 1, and that all facilities have the equal long lives.

2.3.1 Primary energy savings (PES)

PES is the relative primary energy savings of the proposed CCHP system compared with the SP system. PES is defined as

$$PES \triangleq 1 - \frac{F^{CCHP}}{F^{SP}}. \tag{2.45}$$

2.3.2 Hourly total cost savings (HTCS)

Normally, in the literature, annual total cost savings is used to optimize the PGU capacity, but here, in order to obtain the optimal operation strategy for every hour, the hourly total cost savings is adopted. HTC can be written as

$$HTC \triangleq E_{grid}C_e + E_{grid}\mu_e C_{ca} + F_m C_f + F_m \mu_f C_{ca} + \frac{\sum_{k=1}^{l} N_k C_k}{8760 \cdot L}, \tag{2.46}$$

where N_k and C_k are the installed capacity of facilities and the capital cost of each facility, respectively; l is the number of facilities being used and L is the life of an facility; C_e and C_f are the unit prices of electricity and the fuel, respectively; C_{ca} is the carbon tax rate, and μ_e and μ_f are the carbon conversion factor of the electricity and fuel, respectively. Then the $HTCS$ can be defined as

$$HTCS \triangleq 1 - \frac{HTC^{CCHP}}{HTC^{SP}}. \tag{2.47}$$

2.3.3 Carbon dioxide emissions reduction (CDER)

The total carbon dioxide emissions of an energy system can be calculated as

$$CDE = E_{grid}\mu_e + F_m\mu_f, \tag{2.48}$$

where μ_e and μ_f are the carbon conversion factor of electricity and the fuel respectively. Then the $CDER$ can be defined as

$$CDER \triangleq 1 - \frac{CDE^{CCHP}}{CDE^{SP}}. \tag{2.49}$$

2.3.4 Evaluation criteria (EC) function

To measure the performance of the CCHP system, all of these three criteria need to be considered. Thus, the weighted summation of PES, $HTCS$ and $CDER$ as the hourly EC function is adopted and defined as

$$EC_{hour} \triangleq \omega_1 PES + \omega_2 HTCS + \omega_3 CDER, \tag{2.50}$$

where ω_1, ω_2 and ω_3 are the weights of PES, $HTCS$ and $CDER$, respectively. The boundary condition is $0 \leq \omega_1, \omega_2, \omega_3 \leq 1$ and $\omega_1 + \omega_2 + \omega_3 = 1$.

The annual EC function is defined as

$$EC_{annual} \triangleq \sum_{i=1}^{365} \sum_{j=1}^{24} EC_{hourij}, \tag{2.51}$$

where EC_{hourij} is the EC function value of day i, hour j.

2.3.5 Optimal PGU capacity

Capacity of the PGU in CCHP systems is a key factor. As mentioned in the last subsection, all the strategies are based on the PGU capacity. PGU capacity should not be too small, which will not be advantageous; meanwhile, it should not be too large either, for high capital cost and low partial efficiency. In [38], the authors use the genetic algorithm to obtain the global optimal PGU capacity and electric cooling to cool load ratio x.

The main contribution in this chapter is to tune x hourly according to different electric and thermal demands. Thus, no global optimal x exists, every x calculated hourly is the optimal one under the electric and thermal demands in the corresponding hour and a certain PGU capacity. Our goal here is to find an optimal PGU capacity in order to obtain the largest annual EC_{annual} function value, say

$$\underbrace{\max}_{F_{pgum}} EC_{annual}. \tag{2.52}$$

Here, in this chapter, enumeration algorithm is adopted, i.e., searching for every reasonable PGU capacity value, and then choose the one with the largest EC_{annual} function value.

2.4 Case study

2.4.1 Building configuration

In this section, EnergyPlus [225] is chosen to analyze energy consumption of a building in Victoria, B.C., Canada. The building model, shown in Figure 2.4 is a hypothetical

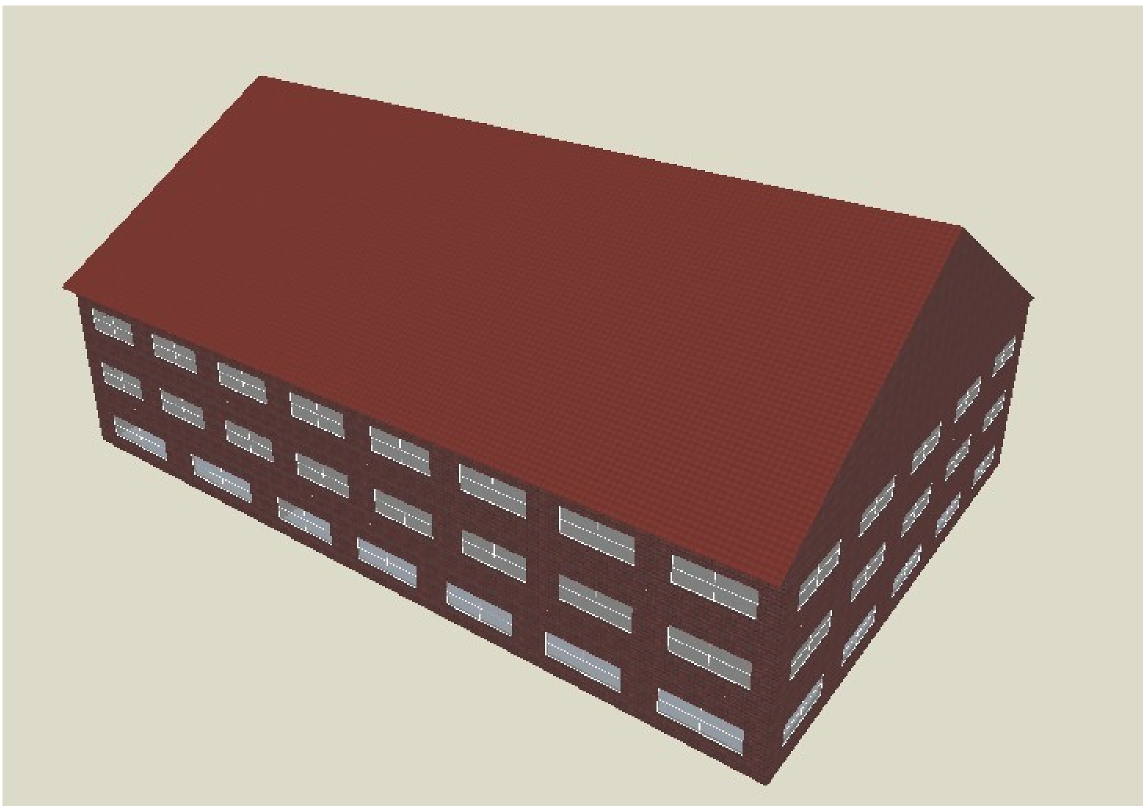

Figure 2.4: *A hypothetical building drawn in Google SketchUp.*

building drawn in Google SketchUp [226]. The building, which operates all year round, is assumed to have four floors with a total construction area of 4500 m^2. The first floor consists of 300 m^2 dining halls and 825 m^2 office rooms; the other floors are guest rooms. Construction parameters of the hypothetical building is listed in Table 2.1. Figure 2.5 shows the energy consumption in one year of the hypothetical building.

Table 2.1: Construction parameters of the hypothetical building

Orientation	Aligned with North
Latitude	48.469°N
Longitude	123.33°W
Location	Victoria, B.C., Canada
Each floor area	$30\text{m} \times 37.5\text{m}$
Each floor height	3.3m
Glass area	40% in each wall
Glazing heat transfer coefficient	$4.247\text{W}/(\text{m}^2 \cdot \text{K})$
Exterior wall heat transfer coefficient	$0.442\text{W}/(\text{m}^2 \cdot \text{K})$
Interior wall heat transfer coefficient	$0.718\text{W}/(\text{m}^2 \cdot \text{K})$
Floor heat transfer coefficient	$2.930\text{W}/(\text{m}^2 \cdot \text{K})$
Roof heat transfer coefficient	$0.368\text{W}/(\text{m}^2 \cdot \text{K})$
Electric equipment, lights and people densities	According to the public building energy-saving design standard

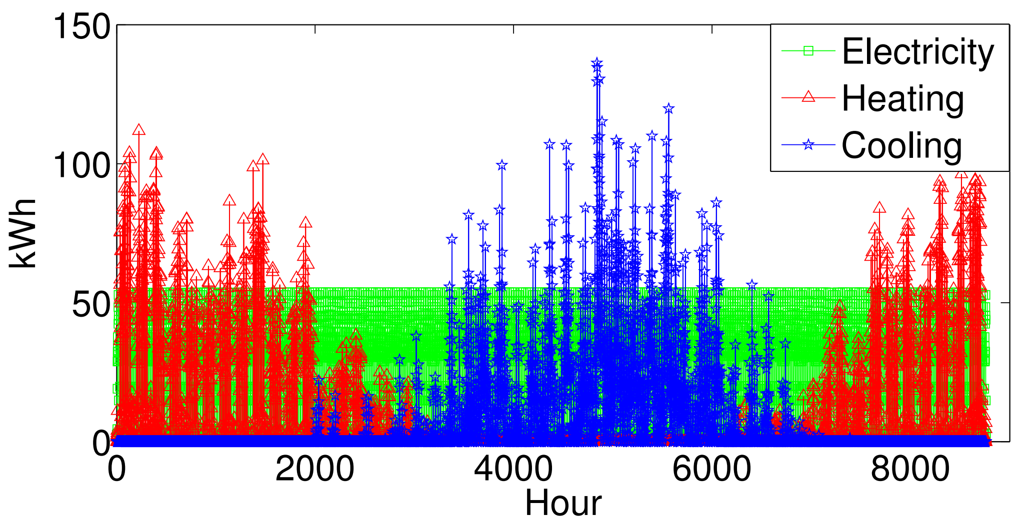

Figure 2.5: *One year consumption of a hypothetical building in Victoria, B.C., Canada.*

2.4.2 Simulation parameters

Given the proposed operation strategy and other facilities' capacities, the other goal of this chapter is to find an optimal PGU capacity. The detailed operation strategy has been discussed in the last section. Table 3.1 shows related coefficients in the CCHP system. The fuel chosen here is the natural gas which has been widely used

Table 2.2: System coefficients

Symbol	Variable	Value
a	First coefficient of η_{pgu}	-0.2
b	Second coefficient of η_{pgu}	0.4
c	Third coefficient of η_{pgu}	0.1
α	Threshold of the electric load fraction	25%
η_e^{SP}	Generation efficiency SP system	0.35
η_h	Efficiency of heating unit	0.8
η_b	Efficiency of boiler	0.8
η_{hrs}	Efficiency of heat recovery system	0.8
COP_{ac}	Coefficient of performance of absorption chiller	0.7
COP_{ec}	Coefficient of performance of electric chiller	3
η_{grid}	Transmission efficiency of local grid	0.92
μ_e	CO_2 emissions conversion factor of electricity (g/kWh)	968
μ_f	CO_2 emissions conversion factor of natural gas (g/kWh)	220
C_e	Electricity rates ($/kWh)	0.0667
C_f	Natural gas rates ($/kWh)	0.0516
C_c	Carbon tax rates ($/g)	0.000003
C_{pgu}	Unit price of PGU ($/kWh)	1046
C_b	Unit price of boiler ($/kWh)	46
C_h	Unit price of heating unit ($/kWh)	30
C_{ac}	Unit price of absorption chiller ($/kWh)	185
C_{ec}	Unit price of electric chiller ($/kWh)	149
L	Facilities' lives (year)	10
ω_1	Coefficient of PES	1/6
ω_2	Coefficient of HTCS	1/6
ω_3	Coefficient of CDER	2/3

in North America.

The initial facilities' capacities for optimization, except the PGU capacity, which is the one to be optimized, are chosen according to the simulation data. The initial boilers in the SP system and CCHP system are chosen to be 200 kW and 120 kW, respectively; the initial heating unit is chosen to be 150 kW; the initial absorption and electric chillers are both chosen to be 150 kW.

2.4.3 Test results and discussions

The EC_{hour} function value in a whole year for this strategy is calculated to verify the feasibility of the proposed optimal operation strategy for the system with unlimited PGU capacity. As shown in Figure 2.6, in a whole year, all of the EC_{hour} function

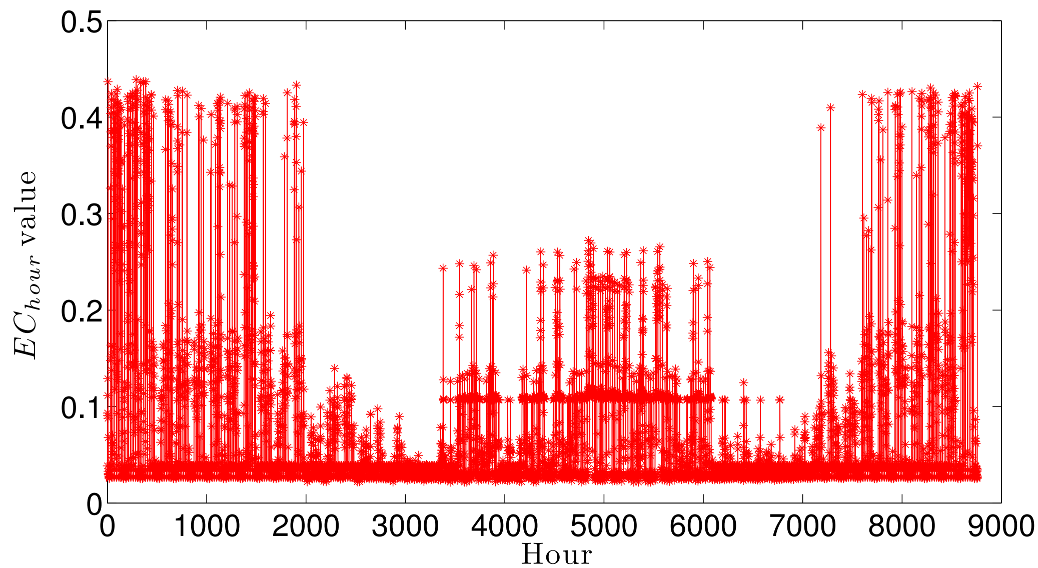

Figure 2.6: *EC_{hour} function value of CCHP system without PGU capacity limit.*

values are greater than 0, which means that, with the proposed matching operation strategy, CCHP systems with unlimited PGU capacity perform better than the SP system, and can lead to economical efficiency. Especially, the highest value can achieve around 0.45, which implies a 45% better integrated performance compared with the SP system.

Another case study, aiming to obtain the optimal PGU capacity under the proposed operation strategy, is conducted for the CCHP system with limited PGU capacity. Because of the complexity of the proposed operation strategy, we cannot simply use the classical linear method to obtain the optimal value. In this chapter, adopting the enumeration algorithm, the optimal PGU capacity is searched in $[1, 500]$ with the whole year (8,760 hours) data. The result for searching is shown in Figure 2.7.

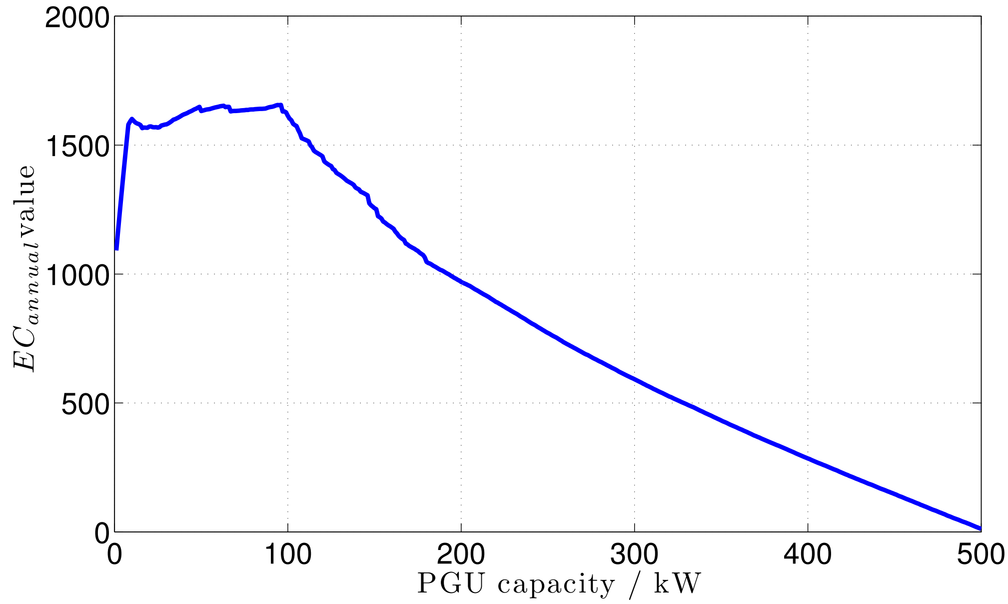

Figure 2.7: *EC_{annual} function value of different PGU capacities from 1 kW to 500 kW.*

From Figure 2.7, we can readily read the optimal point:

$$F_{pgumopt} = 96 \text{ kW}. \tag{2.53}$$

Before the PGU capacity reaches 96 kW, the EC_{annual} function value oscillates as capacity rises, which is the result of the complicated operation strategy. After reaching the optimal capacity, EC_{annual} function value begins descending when the PGU capacity keeps rising. The reason for the descending is that, with the same or even worse performance, the larger the PGU capacity is, the more need to be paid for the extra PGU installed capacity. Moreover, as the PGU capacity keeps increasing, because of the low electric load, the PGU efficiency can be relative low or even the PGU will be cut down. This can also leads to low system efficiency. In a nutshell, the PGU capacity cannot be too small, which makes CCHP system lose advantages; neither be too large to keep the EC_{annual} function value and system efficiency. According to the optimal operation strategy and evaluation criteria, this optimization approach helps to choose the optimal PGU capacity.

With the optimal PGU capacity, EC_{annual} function value is 1,663.57, which implies that the performance of the system can be dramatically improved. EC_{hour} function value of a whole year is shown in Figure 2.8. It is observed that, in Figure 2.8, most

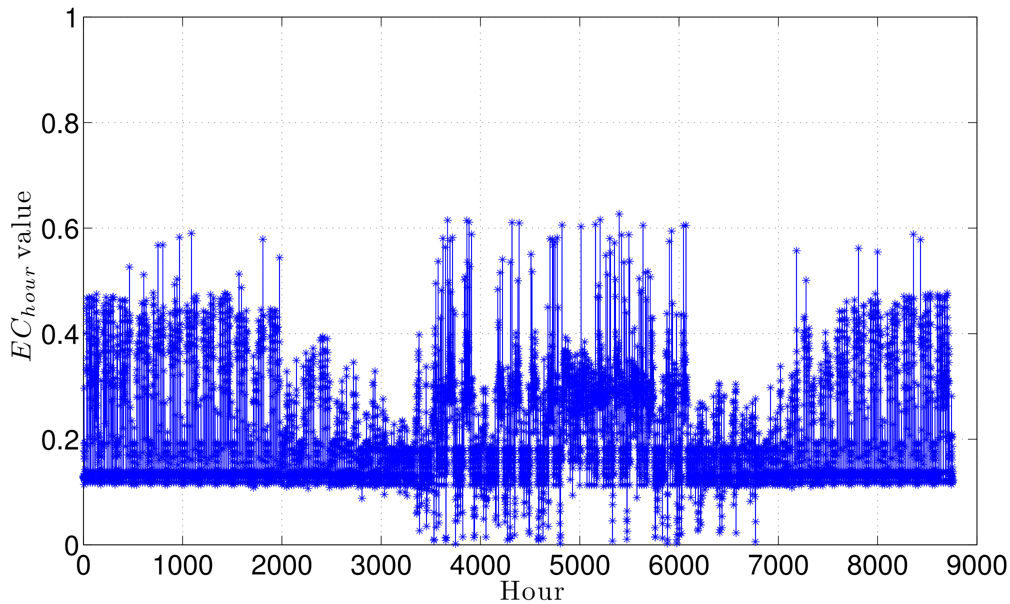

Figure 2.8: *EC_{hour} function value with 96 kW PGU.*

of the points are between 0.16 to 0.61, which implies that the proposed CCHP system can improve the integrated performance up to 61% compared with the SP system; few EC_{hour} values are pretty close to 0, which do not affect the performance of the CCHP system much.

As mentioned before, the electric cooling to cool load ratio varies every hour according to different electric and thermal load. The variation of x, with the optimal PGU capacity and under the proposed operation strategy, is shown in Figure 2.9. In most of the time in winter, spring and autumn, the value of x switches between 0 and 1; however, in summer, the optimal x varies a lot between 0 and 1, according to the

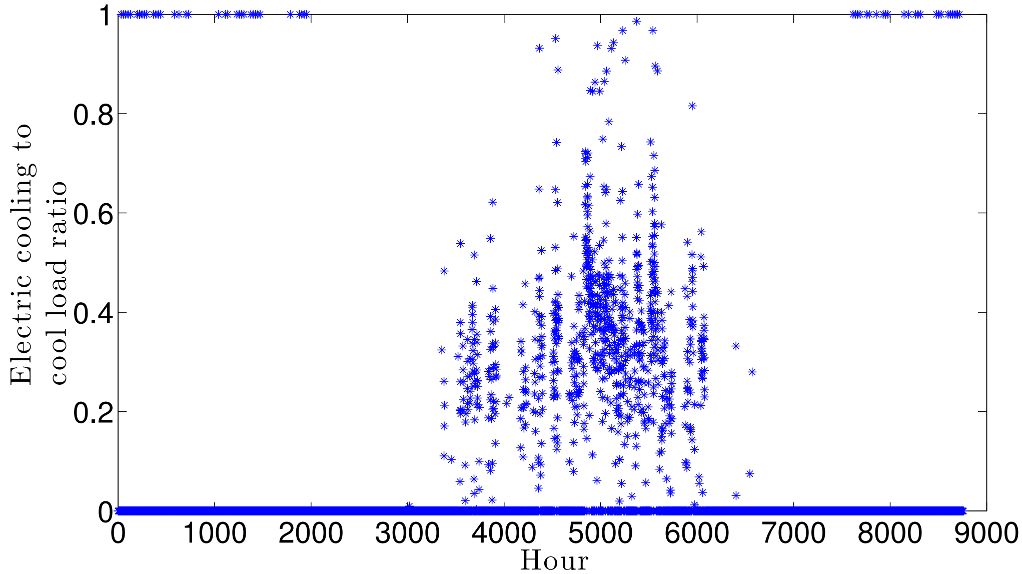

Figure 2.9: *Variation of electric cooling to cool load ratio in a whole year.*

optimal strategy, to obtain the optimal system performance.

The annual performance criteria of the CCHP system under proposed optimal operation strategy, FEL strategy, FTL strategy, and SP systems are shown in Table 2.3. From Table 2.3, it can be readily verified that the CCHP system with the

Table 2.3: Evaluation Criteria of SP and CCHP systems.

System	Primary energy (kWh)	Annual cost ($)	Carbon emissions (g)
SP	776860	75305	234130000
CCHP	714989	47213	175821242
FEL	838652	70773	184505056
FTL	760576	50715	216413613

proposed operation strategy outperforms the SP system, and CCHP system under FEL and FTL strategies in all performance criteria. In addition, compared with the SP system, the FEL strategy results in less annual cost and carbon dioxide emissions, while more primary energy consumption. When operating in the FTL strategy, the CCHP system performs better than the SP system; except for the carbon dioxide emissions, the primary energy consumption and annual cost are less than those of the

FEL strategy.

2.5 Conclusions

A modified structure of the CCHP system, whose cooling part consists of a hybrid chiller, has been proposed in this chapter. The electric cooling to cool load ratio is tuned hourly according to different electric and thermal demands. Energy flow of this CCHP system is investigated as the foundation for the further operation strategy design. First, the operation strategy for a system with unlimited PGU capacity is proposed. By expanding the "balance" plane to be the "balance" space, the matching methodology is adopted to provide a relatively larger space for electric and thermal demands to be well-matched.

To be more practical, an optimal operation strategy, which is based on the relationship between full capacity output of PGU and energy loads, is designed for a CCHP system with limited PGU capacity. Enumeration algorithm is adopted to obtain the optimal PGU capacity.

A case study is conducted to show the feasibilities of the proposed operation strategies. Test results show that, with the proposed operation strategy and corresponding optimal PGU capacity, the modified structured CCHP system performs much better than the conventional SP system, and CCHP systems under FEL and FTL, in all the three evaluation criteria. Especially, the annual total cost of the proposed system is reduced more than half of that of the SP system.

Chapter 3

Optimal Operation Strategy Design and PGU Sizing Using a Matrix Modelling Approach

3.1 Introduction

Performance of CCHP systems depends on system structures, power flow strategies and choices of facility capacity. This chapter presents a matrix modelling approach to optimize the CCHP system. Modelled in a matrix form, a CCHP system can be viewed as an input-output model. Energy conversion and power flow from the system input to the output is governed by a conversion matrix, consisting of dispatch factors and components efficiencies. By designing an objective function and determining constraints, the optimization problem of minimizing the objective function is solved. Furthermore, PGU capacity is also optimized to achieve the optimal performance of the CCHP system. An illustrative case study is conducted to present the effectiveness and economic efficiency of the proposed optimal operation strategy.

3.1.1 Background and Related Work

As mentioned, CCHP systems have been broadly employed in small-/medium-scale power systems to improve the economic efficiency and to reduce GHG emissions [1, 112, 180]. A CCHP system can provide energy users electricity, cooling energy and heating energy by installing the PGU and other facilities. Compared to SP systems, CCHP systems make use of the high temperature exhaust from PGUs to generate thermal energy for cooling and heating demands. GHG emissions, fuel consumptions and even the electricity directly purchased from local grid can be significantly reduced by collecting the excess heat [181–183, 227, 228].

In classical CCHP systems, PGUs are adopted to generate electricity to meet users' electric demands. Rejected heat from the PGU is recovered by a heat recovery system to provide thermal energy for cooling and heating demands. A sorption chiller is installed in the system to absorb or adsorb the recovered heat to provide cooling; a heating unit makes use of the recovered heat to provide space heating. Designing a CCHP system includes the design of the system structure, optimization of the power flow and operation strategy, and the sizing of facilities' capacities.

To optimize the system performance, a mathematical model should be constructed first to make use of the various optimization approaches. In the literature, much work on the optimization has been done to investigate the CCHP optimization problem. Among these approaches, due to the on-off character of the components, mixed integer programming is the most widely used one. Based on the concept of *superstructure*, the authors in [216] propose a systematic method to optimize the size of a CCHP system powered by natural gas, solar energy and gasified biomass. Modelled by the MINLP model, PESs, GHG emissions and economic feasibility are optimized. They also point out that the trade-off between the economical and environmental concerns should be taken into consideration when designing a CCHP system. Following the previous work, which only concerns the monthly average requirement, Rubio *et al.* [229] take the hourly data, analysis and energy storage system into consideration. This NP problem is solved by a generalized reduced gradient (GRG)-based algorithm. The results of the case study show the advantages of the later work. In [230], Buoro *et al.* model a trigeneration system to be a MINLP model. They propose a scheme of several buildings connected with each other. Thus, the optimal solution of this problem contains prime mover's type and size, positions of district heating and cooling pipelines and the operation of each system component. Besides considering the thermodynamic of each system component, the objective function also takes the facilities' cost, district heating and cooling network cost into consideration. Moreover, the influence of various amortization periods on the optimal solution is also discussed. Li *et al.* [231] model and optimize a system by an MINLP model. Analyses in this literature show that the optimal facility size and the economical performance of the whole system mainly depend on the average energy demand. In [232], MILP is used to model and optimize a CCHP system with a thermal storage system installed and to minimize the annual total cost (ATC). In this literature, the effect of legal constraints and different operation modes on the optimal design is also discussed. In [210], the authors construct an MINLP model for increasing the power production in a small-scale CHP plant. This CHP plant is driven by a steam Rankine process fired by biomass fuel. Using the MINLP, due to the complicated decision-making process in the system, the optimization problem is modelled to be a nonconvex problem. This problem is solved several times to filter out suboptimal solutions and to find out the most likely global optimal solution. This model is also tested on four existing CHP plants, in which the result shows that, by adding a two-stage district heat exchanger, a preheater, a steam reheater and a fuel dryer, the electric efficiency and power to heat ratio can be increased. Arcuri *et al.* [42] carry out a mixed integer programming model for the optimization of a CCHP system in a hospital. The optimization results, including short-term optimization and long-term optimization, give the optimal plant design, i.e., facility sizes and running conditions. With the proposed optimization approach, the case study result shows that, by utilizing size optimized heat pumps, the trigeneration system can be improved from economic, energetic and environmental aspects. Li *et al.* [233] thermoeconomically optimize a distributed trigeneration system by considering thermodynamic, economic and GHG emissions aspects. This optimization includes the system configuration and operation strategy. With the objective function set to be the system net present value (SNPV), MINLP is used to model the system and the GA is adopted to solve it. An optimal solution is found under different economic and environmental legislation contexts in Beijing, China. Rong *et al.* [224] model a trigeneration system by the LP model with three components' characteristics. The objective function is set to be a combination of production and purchase cost, and the carbon cost. This problem is solved by the Tri-commodity algorithm, which is 36 to 58 times faster than an efficient Simplex code. Using the PGU capacity as the decision variable, Wang *et al.* [234] optimize a CCHP system by the GA. The objective function is set to be a weighted summation of PEC, ATC and CDE. Wang *et al.* [37] design an optimal operation strategy by optimizing the capacity of PGU, the capacity of heat storage tank, the on-off coefficient of PGU and the ratio of electric cooling to cool load using the particle swarm algorithm. All of the four decision variables are globally optimized, i.e., fixed once determined. The authors of this literature also compare the result of [37] to that of another literature [38]. In [38], only the PGU capacity and electric cooling to cool load ratio are considered to be decision variables. The objective function, which includes PESs, annual total cost savings (ATCSs) and CDER, is minimized by the GA. When compared to the GA, particle swarm algorithm converges faster and the result is better. Sheikhi *et al.* [235] conduct a cost-benefit analysis of a CHP system aiming to maximize the benefit-to-cost ratio. With the benefit-to-cost ratio incorporated into the objective function, using the concept of *energy hub*, the size and efficiency of this CHP system is optimized using the evolutionary-algorithmic (EA) approach. Kavvadias *et al.* [236] set up a multi-objective optimization problem for a trigeneration system, in which facilities sizes, pricing tariff schemes and the operation strategy are to be optimized according to realistic conditions. Pointing out the drawbacks of traditional load following strategies, the authors propose a new load following strategy, i.e., electric/heat equivalent load follow, which includes the continuous operation, peak shaving, electricity equivalent demand following and heat equivalent demand following. The optimization problem is solved by the multi-objective EA approach. Wang *et al.* [237] analyze the energy consumption and construct an environmental impact model, consisting of the global warming, acid precipitation and stratospheric ozone depletion, for an SP system and a CCHP system. The system capacity is optimized by the GA following the FEL strategy. In [238], the authors model a trigeneration system using a fuzzy multi-criteria decision-making model. Different configurations of trigeneration systems are compared with an SP system under this model. This fuzzy multi-criteria model can help to choose the optimal trigeneration configuration from technical, economical and some external (like the environmental) aspects. Piacentio *et al.* [239] propose a robust optimization method for a CCHP system based on energetic analyses. They point out and verify that, by considering the energetic behavior, instead of improving the efficiency of an optimization algorithm, the optimization result can be significantly improved. Moran *et al.* [139] propose a thermoeconomic modelling approach, including the monthly operation cost, monthly fuel consumption, overall system efficiency, etc., for micro-scale CCHP systems in residential use. This model helps to choose the optimal prime mover type and capacity by taking the ratio of required heating and cooling loads to the required electric loads into consideration.

In recent years, a matrix modelling approach based on the *energy hub* begins to be used to model and optimize CCHP systems. Chicco *et al.* [223] propose a matrix modelling approach for a small-scale trigeneration system and optimize the operation strategy for it. Concepts of efficiency matrices, dispatch factors, interconnection matrices and input-to-output connectivity matrix are introduced in the modelling process. A depth-first manner is adopted to construct the overall plant efficiency matrix. NP techniques are required for solving the trigeneration plant optimization problem. In 2005, Geidl *et al.* [240] proposed a general matrix modelling and optimization approach for an energy system with various energy carriers. The modelled problem is a nonlinear, multi-variable and inequality-constrained problem and is solved by the NP algorithm. In a further work of Geidl *et al.* [241], using the concept of *energy hub* [242], the authors model the system by introducing the dispatch factors and coupling matrix, and optimize the dispatch and power flow using the Karush-Kuhn-Tucker (KKT) conditions in order to minimize the total energy cost. The marginal cost is used to solve the KKT conditions. Using the same modelling approach, the optimization problem is solved by MATLAB *fmincon.m* in [243]. Ghaebi *et al.* [244] model a CCHP system using the TRR model to exergoecnomically optimize the cost of the total system production. Design parameters in this model are quite diverse, which include the air compressor pressure ratio, gas turbine inlet temperature, temperature in the heat recovery system, steam pressure, etc. The GA is adopted to solve for the optimal solution. Effects of the decision variables on different objective functions are also discussed in their work.

3.1.2 Motivations

The CCHP system is the connection between the energy input (i.e., electricity and fuel) and users' demands (i.e., cooling, heating and power). From such a mapping relationship perspective, a CCHP system can be viewed as an *energy hub* with multiple-variable energy vectors at the input and output terminals [242]. The *energy hub* represents an interface between different energy infrastructures and/or loads. The coupling of different energy carriers is established by conversions among them. For instance, the PGU in a CCHP system can generate electricity and thermal energy simultaneously by combusting fuel. Electricity and thermal energy provided by the PGU will affect the purchasing of electricity from local grid and additional fuel for the auxiliary boiler [240, 243]. By using the input-output mapping [245], the CCHP system can be described by a connection matrix. With a constructed matrix model of the CCHP system, various efficient optimization algorithms can be adopted to optimize the operation strategy and power flow. In [241], the authors model the system by introducing the concepts of dispatch factors and coupling matrix, and optimize the power flow and operation strategy using the KKT conditions. In [223], the authors model the system by using the concept of junctions, bifurcations and the backtracking. However, the modelling processes of the above two are too complicated. In this chapter, a more comprehensive and intuitive matrix modelling approach to model a CCHP system is proposed. The matrix form of the CCHP system in this chapter is derived from the basic system equations, constructed efficiency matrices and dispatch matrices. Having the matrix modelled system, sequential quadratic programming is adopted to optimize the hourly linear objective function, which is a weighted summation of PECS, HTCS and CDER, and is constrained by equality and inequality constraints. The result of the optimization problem is the optimal power flow and operation strategy for the CCHP system.

In [37, 38], the authors propose a modified CCHP system structure, in which an absorption chiller and an electric chiller are simultaneously installed. An electric chiller has a much higher COP than that of the absorption chiller. By introducing the electric chiller into the CCHP system, the operation strategy is designed to determine the electric cooling to cool load ratio, in order to optimize the performance. In [37,38], this ratio is set to be *fixed*. However, in practice, by appropriately tuning and setting this ratio, better performance can be achieved. Motivated by this observation, the first objective of this chapter lies in: To optimize the power flow and energy input, according to the energy demands and facilities capacity limitations, to obtain the optimal performance, while the electric cooling to cool load ratio is optimized simultaneously. To my best knowledge, the CCHP system with hourly optimized electric cooling to cool load ratio has been rarely investigated in literature.

Another equally important problem involved in the CCHP system design is the facility sizing problem, i.e., determining the capacities of facilities; here, emphasis is put on the PGU capacity. As mentioned in Chapter 2, since the complexity of the operation strategy and the power flow process, the PGU capacity is hard to determine. By considering the facility capital cost, PEC, ATC and GHG emissions, the optimal PGU capacity can be obtained by solving the optimization problem of maximizing the annual objective function.

3.1.3 Objective and Chapter Organization

In this chapter, the CCHP system is firstly modelled to be an input-output model using the matrix modelling approach. System input, inside power flow, and electric cooling to cool load ratio are optimized hourly by the SQP algorithm. Then, the PGU is sized to achieve the annual optimal system performance under the proposed optimal operation strategy. This chapter is organized as follows. The matrix modelling of the

CCHP system is described in Section 3.2, which includes the components efficiency matrices modelling, dispatch matrices modelling and system conversion matrix modelling. Section 3.3 presents the definition of the optimization problem and constraints of this problem. A case study in Section 3.4 shows the effectiveness and economic efficiency of the proposed optimal power flow and operation strategy. Finally, this chapter is concluded in Section 3.5.

3.2 System matrix modelling

In this section, a comprehensive and intuitive matrix modelling approach for the CCHP system will be introduced. The configuration of the CCHP system with hybrid chillers can be referred to Figure 2.1.

3.2.1 Efficiency matrices of system components

The efficiency matrices, also known as coupling matrices [241], are the description of the energy conversion of the system components. In this chapter, the input and output vectors of the ℓ^{th} component are defined $\mathcal{V}_i^\ell = [F_i^\ell\ E_i^\ell\ Q_{ci}^\ell\ Q_{hi}^\ell]^{\mathrm{T}}$ and $\mathcal{V}_o^\ell = [F_o^\ell\ E_o^\ell\ Q_{co}^\ell\ Q_{ho}^\ell]^{\mathrm{T}}$, respectively. In the rest part of this chapter, if no specific notes, the elements of the input and output vectors have the same order as $\mathcal{V}_i^\ell$ and $\mathcal{V}_o^\ell$. In the above definition, $F_\cdot$, $E_\cdot$, $Q_{c\cdot}$ and $Q_{h\cdot}$ denote the fuel, electricity, cooling energy and heating energy in the "$\cdot$" node, respectively. Then the input-output relation of the ℓ^{th} component can be represented as

$$\mathcal{V}_o^\ell = \mathcal{H}^\ell \mathcal{V}_i^\ell, \tag{3.1}$$

where $\mathcal{H}^\ell$ is the efficiency matrix of the ℓ^{th} component.

The input-output relation of PGU, whose electric efficiency is η_{pgu}, can be represented as the following matrix form

$$\begin{bmatrix} 0 \\ E_o^{PGU} \\ 0 \\ Q_{ho}^{PGU} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ \eta_{pgu} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 1-\eta_{pgu} & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} F_i^{PGU} \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{3.2}$$
$$= \mathcal{H}^{PGU}\mathcal{V}_i^{PGU}.$$

Note that, in this chapter, η_{pgu} is assumed to be a constant, i.e., the PGU efficiency will not vary according to the partial load, for the sake of a better interpretation to the proposed method. A model incorporated the second order ploynominal PGU efficiency will be further discussed in the future work.

Following the same procedure, the efficiency matrices for the auxiliary boiler $\mathcal{H}^b$, heat recovery system $\mathcal{H}^{hrs}$, electric chiller $\mathcal{H}^{ec}$, heating unit $\mathcal{H}^h$ and absorption chiller $\mathcal{H}^{ac}$ can be obtained from

$$\begin{bmatrix} 0 \\ 0 \\ 0 \\ Q_{ho}^{b} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ \eta_b & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} F_i^{b} \\ 0 \\ 0 \\ 0 \end{bmatrix}, \begin{bmatrix} 0 \\ 0 \\ 0 \\ Q_{ho}^{hrs} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \eta_{hrs} \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ 0 \\ Q_{hi}^{hrs} \end{bmatrix},$$

$$\begin{bmatrix} 0 \\ 0 \\ Q_{co}^{ec} \\ 0 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & COP_{ec} & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} 0 \\ E_{ci}^{ec} \\ 0 \\ 0 \end{bmatrix}, \begin{bmatrix} 0 \\ 0 \\ Q_{co}^{ac} \\ 0 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & COP_{ac} \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ 0 \\ Q_{hi}^{ac} \end{bmatrix},$$

$$\begin{bmatrix} 0 \\ 0 \\ 0 \\ Q_{ho}^{h} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \eta_h \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ 0 \\ Q_{hi}^{h} \end{bmatrix},$$

respectively.

3.2.2 Dispatch matrices

The efficiency matrices characterize the performance of the energy conversion inside the components, whereas the dispatch matrices describe the power flow between them. In addition, the dispatch factors only exist at the bifurcations of the system. For example, in Figure 2.1, the fuel supply F_m will be separated into two parts: One for the PGU and the other one for the auxiliary boiler. Let α_{pgu} and α_b denote the dispatch factors for the PGU and auxiliary boiler respectively. Then we have

$$F_{pgu} = \alpha_{pgu} F_m, \tag{3.3}$$

$$F_b = \alpha_b F_m, \tag{3.4}$$

subject to

$$\alpha_{pgu} + \alpha_b = 1. \tag{3.5}$$

For sake of the later optimization, the input vectors of the components can be modified to be the functions of the system input. At the PGU side, we have

$$\begin{aligned}\mathcal{V}_i^{PGU} &= \begin{bmatrix} \alpha_{pgu} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} F_m \\ E_{grid} \\ 0 \\ 0 \end{bmatrix} \\ &= \Gamma_{pgu}\mathcal{V}_i,\end{aligned} \tag{3.6}$$

where Γ_{pgu} is the dispatch matrix for the PGU.

Next, the dispatch matrices for the auxiliary boiler, heat recovery system, electric chiller, absorption chiller and heating unit can be generated as

$$\Gamma_b = \begin{bmatrix} \alpha_b & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix},$$

$$\Gamma_{hrs} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ \alpha_{pgu}(1-\eta_{pgu}) & 0 & 0 & 0 \end{bmatrix}, \Gamma_{ec} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & \alpha_{ec} & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix},$$

$$\Gamma_{ac} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ [\alpha_{pgu}(1-\eta_{pgu})\eta_{hrs} + (1-\alpha_{pgu})\eta_b]\,\alpha_{ac} & 0 & 0 & 0 \end{bmatrix}$$

$$\Gamma_h = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ [\alpha_{pgu}(1-\eta_{pgu})\eta_{hrs} + (1-\alpha_{pgu})\eta_b]\,\alpha_h & 0 & 0 & 0 \end{bmatrix}$$

Similarly, in addition to (3.5), we have

$$\begin{aligned} \alpha_{user} + \alpha_{ec} &= 1, \\ \alpha_{ac} + \alpha_{h} &= 1. \end{aligned} \tag{3.7}$$

3.2.3 Conversion matrix of the CCHP system

The conversion matrix of the whole CCHP system describes the efficiencies of the components and the whole procedure of the power flow. The operation strategy of the system can also be included inherently in the conversion matrix. From Figure 2.1, the system input vector is defined to be

$$\begin{aligned} \mathcal{V}_i &= \begin{bmatrix} F_m & E_{grid} & Q_{ci} & Q_{hi} \end{bmatrix}^{\mathrm{T}} \\ &= \begin{bmatrix} F_m & E_{grid} & 0 & 0 \end{bmatrix}^{\mathrm{T}}. \end{aligned} \tag{3.8}$$

The second equality in (3.8) holds since neither cooling energy nor heating energy is the input to the system. Even though no Q_{ci} and Q_{hi} exist in the system input, their positions are reserved to make all the input vectors, including components and the system, in a same dimension and unified form. The output of the system is defined to be

$$\begin{aligned} \mathcal{V}_o &= \begin{bmatrix} F_o & E_{user} & Q_c & Q_h \end{bmatrix}^{\mathrm{T}} \\ &= \begin{bmatrix} 0 & E_{user} & Q_c & Q_h \end{bmatrix}^{\mathrm{T}}. \end{aligned} \tag{3.9}$$

The second equality in (3.9) holds since no fuel is output from the system. Then the conversion matrix of the CCHP system, $\mathcal{H}$, can be defined to be

$$\mathcal{V}_o = \mathcal{H}\mathcal{V}_i. \tag{3.10}$$

Without losing generality, we can assume that the parasitic electricity $E_p = 0$. For the output element E_{user}, we have

$$\begin{aligned} E_{user} &= (E_{pgu} + E_{grid})\alpha_{user} \\ &= (F_{pgu}\eta_{pgu} + E_{grid})\alpha_{user} \\ &= (F_m\alpha_{pgu}\eta_{pgu} + E_{grid})\alpha_{user} \\ &= \alpha_{user}\alpha_{pgu}\eta_{pgu}F_m + \alpha_{user}E_{grid}. \end{aligned} \tag{3.11}$$

For the cooling part, by further considering the high COP but high unit price of the electricity, there must exists an optimal point, at which the system can achieve the optimal performance. Thus, the hybrid chillers are introduced in the system instead of the absorption chiller only in the conventional CCHP system configuration. Optimization should be conducted to obtain the optimal electric cooling to cool load ratio, which is inherently included in the modelling procedure. From Figure 2.1, for the cooling part, we have

$$\begin{aligned} Q_c =& Q_{ec} + Q_{ac} \\ =& E_{ec}COP_{ec} + Q_{hrc}COP_{ac} \\ =& (E_{grid} + E_{pgu})\alpha_{ec}COP_{ec} + [F_{pgu}(1 - \eta_{pgu})\eta_{hrs} + F_b\eta_b]\alpha_{ac}COP_{ac} \\ =& \alpha_{pgu}\eta_{pgu}\alpha_{ec}COP_{ec}F_m + \alpha_{ec}COP_{ec}E_{grid} + [\alpha_{pgu}(1 - \eta_{pgu})\eta_{hrs} + \alpha_b\eta_b]\alpha_{ac}COP_{ac}F_m. \end{aligned} \tag{3.12}$$

Note that, the electric cooling to cool load ratio is calculated by Q_{ec}/Q_c.

The last output of the CCHP system is the heating demand which is all provided by the heating unit as

$$
\begin{aligned}
Q_h =&(Q_b + Q_r)\alpha_h\eta_h \\
=&[F_b\eta_b + F_{pgu}(1-\eta_{pgu})\eta_{hrs}]\alpha_h\eta_h \\
=&[F_m\alpha_b\eta_b + F_m\alpha_{pgu}(1-\eta_{pgu})\eta_{hrs}]\alpha_h\eta_h \\
=&[\alpha_{pgu}(1-\eta_{pgu})\eta_{hrs} + \alpha_b\eta_b]\alpha_h\eta_h F_m.
\end{aligned}
\tag{3.13}
$$

From (3.11) to (3.13), we can readily obtain the conversion matrix in the form as

$$
\mathcal{H} = \begin{bmatrix}
0 & 0 & 0 & 0 \\
\alpha_{user}\alpha_{pgu}\eta_{pgu} & \alpha_{user} & 0 & 0 \\
\alpha_{pgu}\eta_{pgu}\alpha_{ec}COP_{ec} + [\alpha_{pgu}(1-\eta_{pgu})\eta_{hrs} + \alpha_b\eta_b]\alpha_{ac}COP_{ac} & \alpha_{ec}COP_{ec} & 0 & 0 \\
[\alpha_{pgu}(1-\eta_{pgu})\eta_{hrs} + \alpha_b\eta_b]\,\alpha_h\eta_h & 0 & 0 & 0
\end{bmatrix}
\tag{3.14}
$$

The main objective of this chapter is to determine the dispatch factors and the system input to minimize the evaluation criteria objective function. By doing so, the power flow and the electric cooling to cool load ratio is optimized.

3.3 Optimization

In this section, the operation strategy design is formulated as an optimization problem with appropriate evaluation criteria and the corresponding equality and inequality constraints.

3.3.1 Evaluation criteria

The evaluation criteria and annual EC function are defined as same as the ones defined in Chapter 2. The definitions of PES, HTCS, CDER and annual EC function can be referred to (2.45), (2.47), (2.49) and (2.51), respectively. The hourly EC function is redefined to be

$$EC_{hour} \triangleq 1 - (\omega_1 PES + \omega_2 HTCS + \omega_3 CDER), \tag{3.15}$$

for the sake of the latter optimization.

3.3.2 Optimization

The optimization of the overall CCHP system needs to fulfill the following three aspects: (1) Optimization of the dispatch factors; (2) optimization of the input energy; (3) optimization of the PGU capacity. The first part of the optimization is to coordinate the power flow of the system so as to minimize the objective function. Optimizing the energy consumption is to purchase a reasonable amount of electricity and fuel to run the CCHP system, in order to meet the users' demands and to minimize the objective function. Capacity of the PGU in CCHP systems is also a key factor, for the dispatch factors and energy consumption are to be optimized based on the PGU capacity. PGU capacity should not be too small, which will make the CCHP system lose advantages; meanwhile, it should not be too large either, for larger installed capacity costs more.

Since the to be optimized dispatch factors $\alpha_{pgu}, \alpha_b, \alpha_{user}, \alpha_{ec}, \alpha_{ac}$ and α_h are dependent variables, the dispatch factors can be reduced to be $\alpha_{pgu}, \alpha_{user}$ and α_{ac}, which are independent with each other. Thus, the dispatch factor vector is defined to be

$$\alpha = \begin{bmatrix} \alpha_{pgu} \\ \alpha_{user} \\ \alpha_{ac} \end{bmatrix}. \tag{3.16}$$

In addition, the input energy vector is defined to be

$$\beta = \begin{bmatrix} F_m \\ E_{grid} \end{bmatrix}. \tag{3.17}$$

Both α and β are needed to be optimized, so α and β can be augmented to be the optimizer

$$\mathfrak{X} = \begin{bmatrix} \alpha \\ \beta \end{bmatrix}. \tag{3.18}$$

Objective function

The EC_{hour} function can be rewritten to be a linear function of the optimizer $\mathfrak{X}$ as

$$EC_{hour}(\mathfrak{X}) = \omega_1 \frac{\mathcal{P}\mathfrak{X}}{F^{SP}} + \omega_2 \frac{\mathcal{C}\mathfrak{X} + \mathcal{L}}{HTC^{SP}} + \omega_3 \frac{\mathcal{D}\mathfrak{X}}{CDE^{SP}}, \tag{3.19}$$

where

$$\begin{aligned}
\mathcal{P} &= \begin{bmatrix} 0 & 0 & 0 & 1 & 1/(\eta_e^{SP}\eta_{grid}) \end{bmatrix}, \\
\mathcal{C} &= \begin{bmatrix} 0 & 0 & 0 & C_f + \mu_f C_{ca} & C_e + \mu_e C_{ca} \end{bmatrix}, \\
\mathcal{D} &= \begin{bmatrix} 0 & 0 & 0 & \mu_f & \mu_e \end{bmatrix}, \\
\mathcal{L} &= \frac{\sum_{k=1}^{l} N_k C_k}{8760 \cdot L}.
\end{aligned}$$

In (3.19), $\mathcal{P}$ can be called the primary energy vector; $\mathcal{C}$ can be called the cost vector; $\mathcal{D}$ can be named as the carbon dioxide vector and $\mathcal{L}$ can be called the life cost vector.

Then, if given a specific PGU capacity, the hourly optimization problem becomes

$$\begin{aligned}
&\underbrace{\min}_{\mathfrak{X}} \{EC_{hour}(\mathfrak{X})\} \\
=&\underbrace{\min}_{\mathfrak{X}} \left\{ \omega_1 \frac{\mathcal{P}\mathfrak{X}}{F^{SP}} + \omega_2 \frac{\mathcal{C}\mathfrak{X} + \mathcal{L}}{HTC^{SP}} + \omega_3 \frac{\mathcal{D}\mathfrak{X}}{CDE^{SP}} \right\},
\end{aligned} \tag{3.20}$$

and is subject to equality and inequality constraints.

Equality constraint

The equality constraint represents the balance between the supply side and demand side, including the fuel, electricity, cooling energy and heating energy. With this equality constraint, the energy waste can be avoided. The only one equality constraint for the optimization problem is

$$\mathcal{H}\mathcal{V}_i - \mathcal{V}_o = 0. \tag{3.21}$$

There is no need to constrain the supply and demand relation of each component, for this type of constraints are already included in the conversion matrix $\mathcal{H}$.

Since the elements of the conversion matrix $\mathcal{H}$ is represented by the elements of the optimizer $\mathfrak{X}$, $\mathcal{H}$ can be rewritten to be a function of $\mathfrak{X}$. Then we have

$$\begin{aligned}\mathcal{H} =&(h_{11}W_{22} + h_{311}W_{33} + h_{313}W_{32} - h_{411}W_{43})\mathfrak{X}\mathfrak{X}^{\mathrm{T}}U_{11} \\ &+ (h_{312}W_{33} + h_{314}W_{31} + h_{411}W_{41} - h_{412}W_{43})\mathfrak{X}Q_{11} \\ &+ (h_{221}W_{22} + h_{321}W_{32})\mathfrak{X}Q_{12} \\ &+ (CON_{COP_{ec}} + COP_{\eta_h\eta_b}),\end{aligned} \tag{3.22}$$

where

$$W_{22} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}, W_{31} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix},$$

$$W_{32} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}, W_{33} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix},$$

$$W_{41} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 \end{bmatrix}, W_{43} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \end{bmatrix},$$

$$Q_{11} = \begin{bmatrix} 1 & 0 & 0 & 0 \end{bmatrix}, \qquad Q_{12} = \begin{bmatrix} 0 & 1 & 0 & 0 \end{bmatrix},$$

$$U_{11} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}, CON_{\eta_h \eta_b} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ \eta_h \eta_b & 0 & 0 & 0 \end{bmatrix},$$

$$CON_{COP_{ec}} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & COP_{ec} & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix},$$

and

$$\begin{aligned} h_{211} &= \eta_{pgu}, & h_{221} &= 1, \\ h_{312} &= \eta_b COP_{ac}, & h_{313} &= -\eta_{pgu} COP_{ec}, \\ h_{321} &= -COP_{ec}, & h_{314} &= \eta_{pgu} COP_{ec}, \\ h_{311} &= (\eta_{hrs} - \eta_{pgu}\eta_{hrs} - \eta_b) COP_{ac}, & h_{412} &= \eta_h \eta_b, \\ h_{411} &= \eta_h(\eta_{hrs} - \eta_{pgu}\eta_{hrs} - \eta_b), \end{aligned}$$

In addition, the system input $\mathcal{V}_i$ should also be represented by the optimizer $\mathfrak{X}$ as

$$\mathcal{V}_i = P\mathfrak{X}, \tag{3.23}$$

where

$$P = \begin{bmatrix} 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}.$$

Then the equality constraint represented by the optimizer becomes a nonlinear equality constraint as

$$\begin{aligned} &(h_{11}W_{22} + h_{311}W_{33} + h_{313}W_{32} - h_{411}W_{43})\mathfrak{X}\mathfrak{X}^{\mathrm{T}}U_{11}P\mathfrak{X} \\ &+(h_{312}W_{33} + h_{314}W_{31} + h_{411}W_{41} - h_{412}W_{43})\mathfrak{X}Q_{11}P\mathfrak{X} \\ &+(h_{221}W_{22} + h_{321}W_{32})\mathfrak{X}Q_{12}P\mathfrak{X} \\ &+(CON_{COP_{ec}} + COP_{\eta_h\eta_b})P\mathfrak{X} - \mathcal{V}_o = 0. \end{aligned} \tag{3.24}$$

Inequality constraints

The inequality constraints come form the characters of parameters, capacities of components and thresholds of the components output. The dispatch factors, i.e., the elements in α should be no less then zero and no more than one. The reason is that if one power flow is divided into several parts at a bifurcation, each part can only has a certain percent of the total power flow; and the sum of all divided power flow should be identical to the original power flow. For the input energy, since it is assumed that no energy can be sold back, the F_m and E_{grid} should be no less than zero. The restrictions for the decision variables in the optimizer generate

$$-\alpha \leq 0, \tag{3.25a}$$

$$\alpha - 1 \leq 0, \tag{3.25b}$$

$$-\beta \leq 0. \tag{3.25c}$$

Inequalities in (3.25) are the linear inequality constraints, however, they can be dealt with as the special case in nonlinear inequality constraints, or the lower and upper bounds for the optimizer.

The capacities of components constrain the upper bounds of the components output. Since, except for the PGU capacity that is to be optimized, the capacities of other components have specific limits, here only the upper bound of the PGU is addressed. The capacity of PGU is denoted by F_{pgum}, thus the upper bound of the PGU output is represented as

$$\overline{\mathcal{V}}_o^{pgu} = \begin{bmatrix} 0 \\ \overline{E}_o^{pgu} \\ 0 \\ \overline{Q}_{ho}^{pgu} \end{bmatrix} = \begin{bmatrix} 0 \\ F_{pgum}\eta_{pgu} \\ 0 \\ F_{pgum}(1-\eta_{pgu}) \end{bmatrix}. \tag{3.26}$$

The output upper bounds for other components are $\overline{\mathcal{V}}_o^{b}$, $\overline{\mathcal{V}}_o^{hrs}$, $\overline{\mathcal{V}}_o^{ec}$, $\overline{\mathcal{V}}_o^{ac}$ and $\overline{\mathcal{V}}_o^{h}$.

Some components have thresholds, which imply that if the expected output of one component is lower than its threshold, this component should be shut down to keep the efficiency. The lower bound for the PGU, auxiliary boiler, heating recovery system, electric chiller, absorption chiller and heating unit are $\underline{\mathcal{V}}_o^{pgu}$, $\underline{\mathcal{V}}_o^{b}$, $\underline{\mathcal{V}}_o^{hrs}$, $\underline{\mathcal{V}}_o^{ec}$, $\underline{\mathcal{V}}_o^{ac}$ and $\underline{\mathcal{V}}_o^{h}$. Without losing generality, we can set those lower bounds to be zero.

With the components output upper and lower bounds constraints, we can readily have the inequality constraints as

$$\mathcal{H}^\ell \mathcal{V}_i^\ell - \overline{\mathcal{V}}_o^\ell \leq 0, \tag{3.27a}$$

$$\underline{\mathcal{V}}_o^\ell - \mathcal{H}^\ell \mathcal{V}_i^\ell \leq 0. \tag{3.27b}$$

Then from (3.6), (3.7) and (3.23), (3.27) becomes

$$\mathcal{H}^\ell \Gamma_\ell P \mathfrak{X} - \overline{\mathcal{V}}^\ell \leq 0, \tag{3.28a}$$

$$\underline{\mathcal{V}}^\ell - \mathcal{H}^\ell \Gamma_\ell P \mathfrak{X} \leq 0. \tag{3.28b}$$

By following the similar procedure of deriving (3.22), Γ_ℓ can also be represented by the function of $\mathfrak{X}$ as

$$\begin{aligned}
\Gamma_{pgu} =& W_{11}\mathfrak{X}Q_{11}, \qquad \Gamma_b = I_{11} - W_{11}\mathfrak{X}Q_{11},\\
\Gamma_{hrs} =& \gamma_1 W_{41}\mathfrak{X}Q_{11}, \qquad \Gamma_{ec} = I_{22} - W_{22}\mathfrak{X}Q_{12},\\
\Gamma_{ac} =& \gamma_2 W_{43}\mathfrak{X}\mathfrak{X}^{\mathrm{T}}U_{11} + \gamma_3 W_{43}\mathfrak{X}Q_{11},\\
\Gamma_h =& -\gamma_2 W_{43}\mathfrak{X}\mathfrak{X}^{\mathrm{T}}U_{11} + (\gamma_2 W_{41} - \gamma_3 W_{43})\mathfrak{X}Q_{11} + CON_{\eta_b},
\end{aligned} \tag{3.29}$$

where

$$W_{11} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix}, I_{11} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix},$$

$$I_{22} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}, CON_{\eta_b} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ \eta_b & 0 & 0 & 0 \end{bmatrix},$$

and

$$\gamma_1 = 1 - \eta_{pgu},$$

$$\gamma_2 = h_{411}/\eta_h,$$

$$\gamma_3 = h_{412}/\eta_h.$$

Then (3.25) and (3.27) construct the nonlinear inequality constraints of the optimizations problem in (3.20).

Optimization algorithm

In the optimization problem, the hourly objective function is a linear one, however, nonlinear equality, linear inequality and nonlinear inequality constrains are involved. This type of problem can be solved by using a variety of methods, e.g., penalty and barrier function methods, gradient projection methods, and SQP methods. Among these methods, SQP algorithms have been proved highly effective for solving general constrained problems with smooth objective and constraint functions [246]. In this chapter, the SQP algorithm is adopted to obtain the optimal solution with arbitrary initial points; moreover, the Hessians of the Lagrangian is approximated using the Broyden-Fletcher-Goldfarb-Shanno (BFGS) formula. Even though the algorithm allows arbitrary initial points, to accelerate the convergence, some feasible initial points, calculated from FEL, FTL, and other data combinations, are manually designated. By doing so, time for convergence is significantly reduced and the local optimal solution can be avoided. The comparison of different algorithms is beyond the scope of this chapter.

To obtain the optimal PGU capacity, another step is to minimize the EC_{annual} function. Naturally, since the optimizer $\mathfrak{X}$ is optimized hourly, under a specific PGU capacity, the system is annually optimized. Thus, we only need to sum a whole year EC_{hour} value to obtain the hourly based optimal EC_{annual} and then search for the optimal PGU capacity $F_{pgumopt}$, which minimizes the EC_{annual} function.

3.4 Case study

3.4.1 Building configuration

In this section, EnergyPlus is chosen to analyze energy consumption of a building in Victoria, B.C., Canada. The building model is the same as the one in Chapter 2. The construction parameters of the hypothetical building can be referred to Table 2.1.

3.4.2 Simulation parameters

Given the proposed power flow and operation strategy, and other facilities' capacities, the rest work is to find an optimal PGU capacity. Table 3.1 shows related coefficients in the CCHP system. The fuel chosen here is the natural gas which has been widely used in North America. The weights of the evaluation criteria are chosen to satisfy different requirements, which implies that we can increase ω_3 to reduce the GHG emissions or increase ω_1 to reduce primary energy consumption. In this case study, weights are set as stated in Table 3.1 to firstly save the primary energy and then secondly reduce the hourly total cost. The GHG emissions is placed at last.

Table 3.1: System coefficients

Symbol	Variable	Value
η_{pgu}	Efficiency of PGU of CCHP system	0.3
η_e^{SP}	Generation efficiency SP system	0.35
η_h	Efficiency of heating unit	0.8
η_b	Efficiency of boiler	0.8
η_{hrs}	Efficiency of heat recovery system	0.8
COP_{ac}	Coefficient of performance of absorption chiller	0.7
COP_{ec}	Coefficient of performance of electric chiller	3
η_{grid}	Transmission efficiency of local grid	0.92
μ_e	CO_2 emissions conversion factor of electricity (g/kWh)	968
μ_f	CO_2 emissions conversion factor of natural gas (g/kWh)	220
C_e	Electricity rates ($/kWh)	0.0987
C_f	Natural gas rates ($/kWh)	0.0577
C_{ca}	Carbon tax rates ($/g)	0.00003
C_{pgu}	Unit price of PGU ($/kWh)	1046
C_b	Unit price of boiler ($/kWh)	46
C_h	Unit price of heating unit ($/kWh)	30
C_{ac}	Unit price of absorption chiller ($/kWh)	185
C_{ec}	Unit price of electric chiller ($/kWh)	149
L	Facilities' lives (year)	10
ω_1	Coefficient of PES	0.5
ω_2	Coefficient of HTCS	0.4
ω_3	Coefficient of CDER	0.1

The initial facilities' capacities for optimization, except the PGU capacity, which is the one to be optimized, are chosen according to the simulation data. The initial boilers in the SP system and CCHP system are chosen to be 200 kW and 120 kW, respectively; the initial heating unit is chosen to be 120 kW; the initial absorption and electric chillers are both chosen to be 120 kW.

3.4.3 Test results and discussions

The comparisons of FEL, FTL and the proposed optimal operation strategy are shown in Figure 3.1, 3.2 and 3.3, which are in summer, winter and spring, respectively.

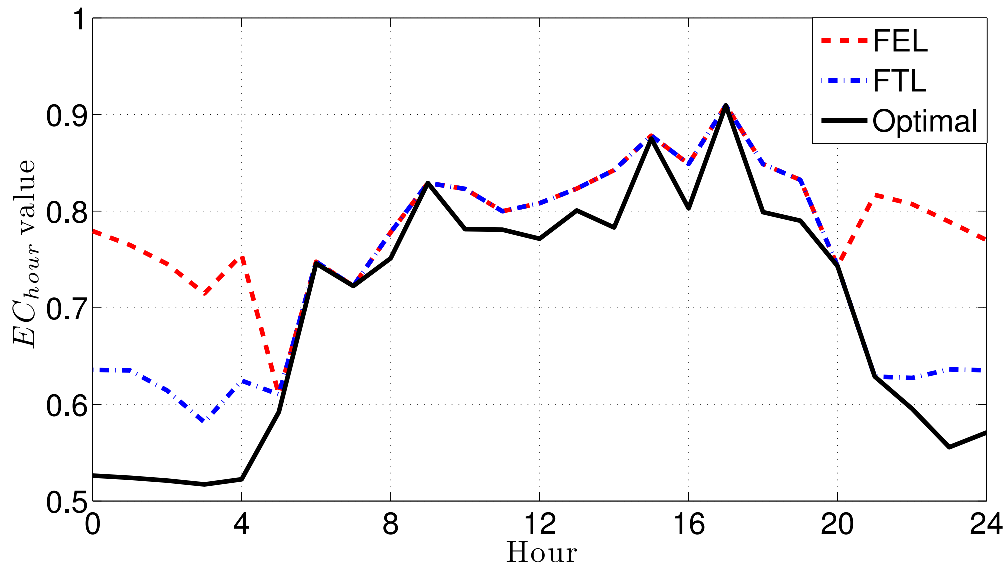

Figure 3.1: *Comparison of FEL, FTL and the optimal operation strategy in a summer day.*

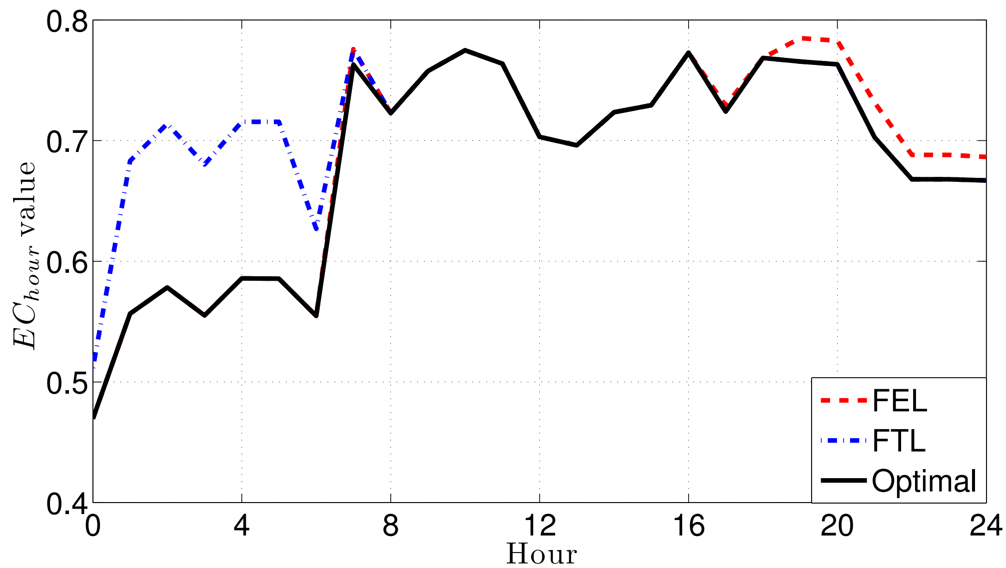

Figure 3.2: *Comparison of FEL, FTL and the optimal operation strategy in a winter day.*

It can be noticed that, from Figure 3.1, 3.2 and 3.3, the EC_{hour} value of the proposed optimal power flow and operation strategy is no more than those of the FEL and FTL strategies. In summer and winter, especially in the early morning

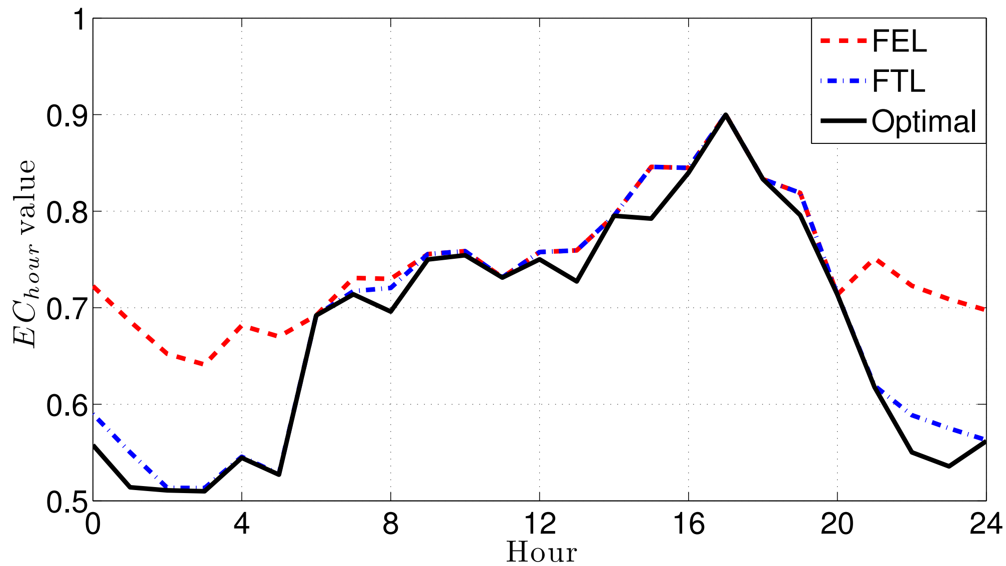

Figure 3.3: *Comparison of FEL, FTL and the optimal operation strategy in a spring day.*

and late night, compared to the FEL and FTL strategies, the values of the proposed optimal power flow or strategy are much lower. While in the middle of a day, the proposed optimal strategy can perform a little bit better than those two. In spring, the similar case as in autumn, the performance of the proposed optimal strategy is a little bit better than that of the FTL strategy, which is much better than that of the FEL strategy.

We can tell from the above analyses that the proposed optimal solution can manage the energy input and power flow reasonably to eliminate the waste of energy. As presented in (3.21), the equality constrains the output of the CCHP system to exactly match the users' demands. While, because of the inherent waste of the FEL and FTL strategy, with no doubt, the proposed optimal strategy can perform better. It is reasonable to tell that part of the distance between the curve of the proposed operation strategy and FEL/FTL is the waste energy which is eliminated. On the other hand, through optimizing the power flow inside the CCHP system, the electric cooling to cool load ratio can be tuned hourly to minimize the objective function. The above two advantages make the proposed power flow and operation strategy to be optimal.

The searching result for the optimal PGU capacity is shown in Figure 3.4. It can

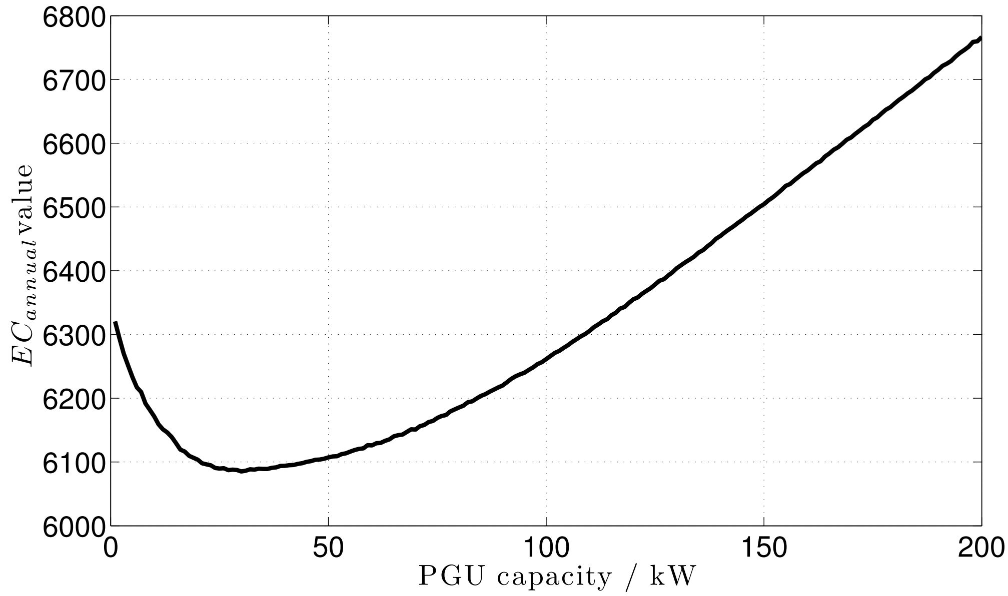

Figure 3.4: *EC_{annual} of PGU capacity from 0 kW to 200 kW.*

be readily obtained from Figure 3.4 that the optimal PGU capacity is 30 kW. One of the reasons for the EC_{annual} blowing up after 30 kW is the capital cost of the PGU facility. Since the weight for the hourly total cost, say ω_2 is set to be 0.4, the price of the PGU is a much strong factor in the final objective function value. If we decrease the weight for the HTCS to 0.2 and increase the weight for CDER to 0.3, the optimal PGU capacity can be larger, which can be shown in Figure 3.5. In a nutshell, the designed optimal operation strategy is dependent on different requirements. The more attention paid on the economical aspect than on the environmental one, the smaller the PGU capacity will be. The trade-off between the cost and the environment should be taken into consideration when designing the operation strategy.

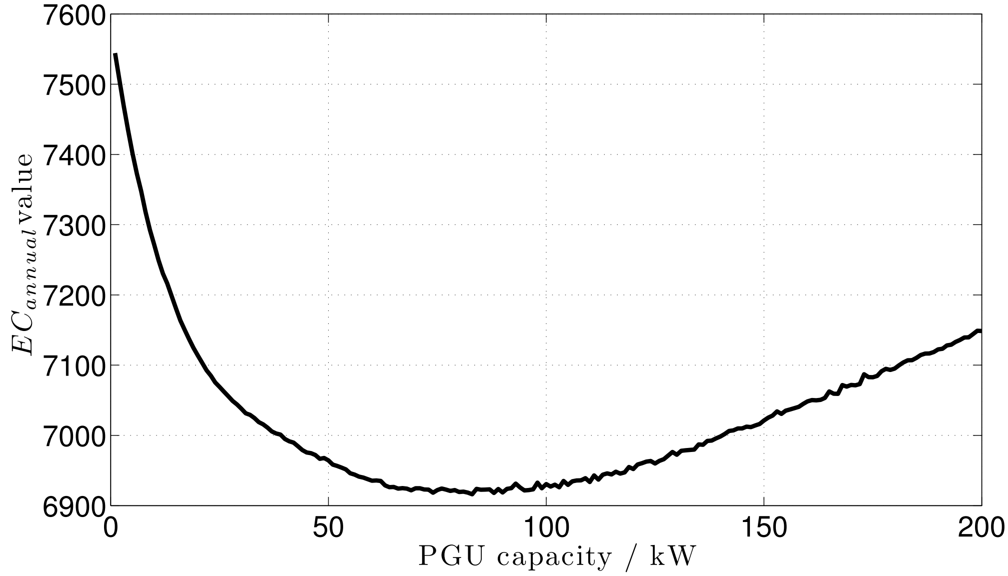

Figure 3.5: *EC_{annual} of PGU capacity from 0 kW to 200 kW.*

As mentioned above, in order to minimize the objective function hourly, the electric cooling to cool load ratio must vary according to the energy demands. Since this ratio, which can be represented by the dispatch factors and component efficiencies, is inherently included in the modelling approach, then in the optimization process, the electric cooling to cool load ratio is optimized automatically. Figure 3.6 shows the optimal electric cooling to cool load ratio in every hour in one year. We can see from this figure that, during one year, most of the ratio values vary between 0 and 1 to keep an optimal system performance; some values stay in 0 to make full use of the absorption chiller except for the winter cases; only a few values lie in 1 to take advantage of the high COP of the electric chiller. We can also draw the conclusion that, despite of the high COP, the high rate of electricity is still a restriction for the electric chiller.

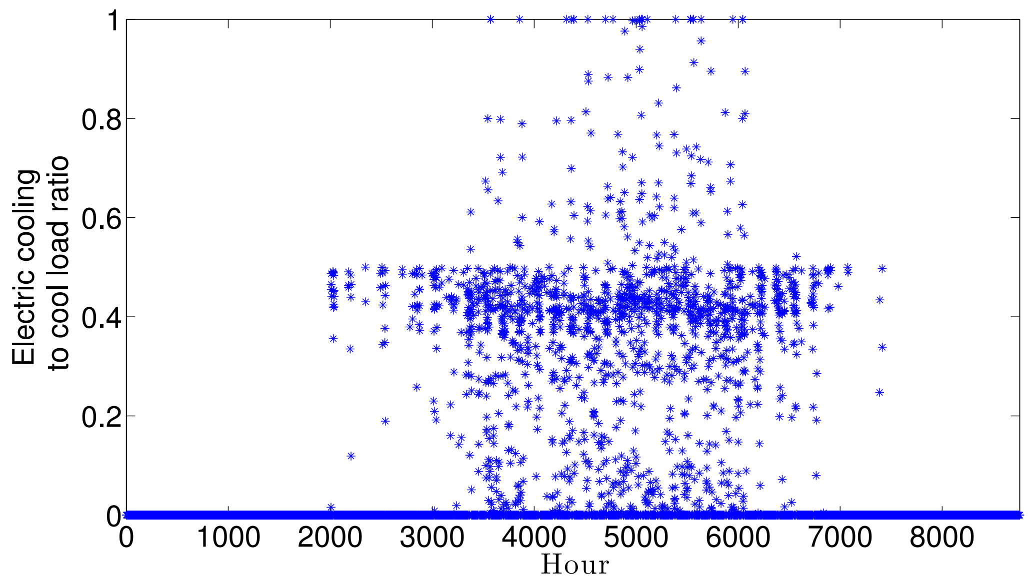

Figure 3.6: *Variation of the electric cooling to cool load ratio.*

3.5 Conclusions

A comprehensive yet intuitive approach of matrix modelling for a CCHP system with a hybrid chiller installed has been proposed in this chapter. The modelling procedure is much easier to be understood than other modelling approaches in the literature. Efficiency matrices of system components, conversion matrix of the whole system and dispatch matrices are incorporated into this model. The formulation of the CCHP system optimization consists of two parts: (1) A linear evaluation criteria objective function is proposed; (2) the supply-demand balance is formulated as a nonlinear equality constraint, and the output upper and lower bounds of the system components are formulated as the nonlinear inequality constraints. The SQP algorithm has been employed to optimize the PGU capacity for a specified system configuration and pre-set weights of the criteria. In addition, the influence on the optimal PGU capacity of different criteria weights selections is also discussed. The electric cooling to cool load ratio varies between 0 and 1 to minimize the objective function in every hour. It is worthwhile mentioning that the primary energy consumptions, hourly and annually total cost, and GHG emissions are all considered in the objective function of the optimization problem. An illustrative case study is conducted to show the effectiveness and the economic efficiency of the proposed optimal power flow and operation strategy. Compared to FEL and FTL, the proposed operation strategy performs much better during the one-year run testing.

Chapter 4

Conclusions and Future Work

In this thesis, the operation strategy designs and PGU sizing problems for a CCHP system are investigated. With a configured CCHP system, the operation strategy determines the system performance. Since FEL, FTL and other FEL/FTL based operation strategies inherently waste a certain amount of energy, newly energy waste reduced operation strategy should be designed. I adopt two ideas, i.e., the "balance" space approach and the matrix modelling approach, in designing operation strategies for the CCHP system in this thesis. Besides the operation strategy design, the PGU capacity is sized according to the respective proposed operation strategy to achieve the optimal system performance.

4.1 Conclusions

The operation strategy design based on the concept of "balance" space is investigated in Chapter 2. First, the design procedure for CCHP systems with unlimited PGU capacity is discussed. In this case, by tuning the electric cooling to cool load ratio, the "balance" plane in a single absorption chillered CCHP system is expanded to a "balance" space, which means that cooling demands, heating demands and electric demands can match with each other in a larger *space* rather than a fixed *plane*, which is contained in the space. Aiming to maximize the objective function, which is a weighted summation of PESs, HTCs and CDER, three energy demand patterns are studied in deciding the decision-making on the electric cooling to cool load ratio. Based on the preliminary results of the system with unlimited PGU capacity, a more practical case, considering the limited PGU capacity, is studied. Different from the one without PGU capacity limitation, eight energy demands patterns are discussed. In each pattern, decision-making process determines the electric cooling to cool load ratio, electricity needed to be purchased from local grid and fuel to be purchased to compensate the thermal gap, in order to obtain the maximum objective function value in each hour. Thus, different from other literature, in which the electric cooling to cool load ratio does not exist or set to be fixed, the one in this thesis is optimized hourly according to different energy consumption pattern and realistic electricity and fuel rates. In the second case, since the decision-making depends on the PGU capacity, the PGU capacity will directly affect the system performance. Moreover, the low partial load efficiency of the PGU also has a nonnegligible impact on the final operation strategy. Considering the complexity of the proposed operation strategy, enumeration algorithm is adopted to obtain the optimal PGU capacity in a reasonable range. Finally, a case study, using the data of a hypothetical building in Victoria, B.C., Canada, is conducted to verify the feasibility and effectiveness of the proposed operation strategy. By comparing with the FEL, FTL and SP system, the CCHP system with the proposed operation strategy shows a significant improvement in all of the three evaluation criteria.

In Chapter 3, a more mathematical way to model the CCHP system with hybrid chillers is proposed. Using the concept of *energy hub*, the CCHP system is modelled as an input-output model, in which the electricity purchased from local grid and fuel needed are the system input; the cooling, heating and electricity provided are the system output. Based on this model, efficiency matrices of system components, dispatch matrices describing the energy transmission between components, and conversion matrix of the whole system are established to describe the CCHP system. The objective function is redefined however in the same idea in Chapter 2. To accomplish the optimization task, dispatch factors at the PGU thermal, PGU electric and cooling nodes together with two energy inputs are selected to be the optimizer. In the modelling process, the electric cooling to cool load ratio is inherently included. Thus, as long as the optimization runs, the ratio would be optimized along with other decision variables. Considering the energy and mass balance, and upper and lower bounds of components' outputs, the optimization problem is constrained by nonlinear equality, linear inequality and nonlinear inequality constraints. By adopting the SQP, the optimal operation strategy, power flow and energy inputs are obtained for a fixed PGU capacity. Similar as in Chapter 2, the enumeration algorithm is used to search for the optimal PGU capacity in a reasonable range. In the case study, compared with the FEL and FTL, with the same PGU capacity, the proposed operation strategy outperforms. An analysis of the influence of the evaluation criteria weight on the optimal PGU capacity is also presented.

4.2 Future Work

Data Forecast. As we can see, the same as other literature, all of the works in this thesis are based on the known weather and energy demand data. However, in practical applications, the energy demand data in the upcoming hour or day cannot be crystally obtained. Thus, accurate weather and energy consumption prediction model should be included when designing the operation strategy [247, 248]. Moreover, the inaccurate weather and energy consumption data can be considered as the parameter uncertainty [249]. Some approaches in the automatic control area can be used to solve the problem of uncertainty.

Renewable Energy. Although, compared with conventional power plants, CCHP systems have much less impact on the environment, the huge amount of primary energy consumption is still a big problem. Even with a reduced rate, GHG emissions of CCHP systems are still increasing year after year. Therefore, there is an urgent call for renewable energy, including wind, solar and tide.

The wind energy, because of the high capital cost and large volume, must be installed to provide electricity for a large district. In addition, a classical fossil-fired power plant has to work in pairs with the wind power station in order to solve the problem of intermittent electricity generating. Another method that can deal with the grid fluctuation caused by the intermittent wind energy is the demand-side management [250]. If given a constructed wind power plant, the power flow between different CCHP systems in a district should be controlled to achieve the optimal performance of the entirety. Some distributed control theories [251,252] maybe helpful for designing such a CCHP system group.

Different from the wind energy, the solar energy can be implemented in a single micro-/small-scale CCHP system. The PV technology can provide a certain amount of electricity for a CCHP system [206,253–255]. However, same as wind energy applications, the problem of intermittent electricity generating also exists in applications with PV arrays installed. Thus, solutions to the intermittent electricity generating, such as the battery, should be carried out for the use of PV array in CCHP systems. Another solar solution is the thermal solar storage technology [256–258]. By applying this technique, solar energy collected can be stored during daytime, then provide hot water or drive thermally activated facilities during nighttime.

Appendix A

Publications

- **Refereed journal papers that have been published**

J1. **M. Liu**, Y. Shi, and F. Fang, "A new operation strategy for CCHP systems with hybrid chillers," *Applied Energy*, vol. 95, pp. 164-173, 2012.

- **Refereed journal papers that have been accepted with minor revisions**

J2. H. Zhang, Y. Shi, and **M. Liu**, "H_∞ step tracking control for networked discrete-time nonlinear systems with integral and predictive actions," *IEEE Transactions on Industrial Informatics*, 2011.

- **Refereed journal papers that are under review**

J3. **M. Liu**, Y. Shi, and F. Fang, "Optimal power flow and PGU capacity of CCHP systems using a matrix modelling approach," *Applied Energy*, 2012.

J4. **M. Liu** and Y. Shi "T-S fuzzy-model-based H_2 and H_∞ filtering for networked control systems with two-channel Markovian random delays," *IEEE Transactions on Systems, Man and Cybernetics, Part B: Cybernetics*, 2012.

- **Refereed conference papers that have appeared or been accepted**

C1. **M. Liu** and Y. Shi, “An energy efficient optimal operation strategy design for CCHP systems,” *Canadian Society of Mechanical Engineers (CSME) International Congress*, Winnipeg, Manitoba, Canada, June 4 – June 6, 2012.

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