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IEA District Heating and Cooling

Programme of Research, Development and Demonstration on District Heating and Cooling

INTEGRATING DISTRICT COOLING WITH COMBINED HEAT AND POWER

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Integrating District Cooling With Combined Heat and Power

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Preface

The International Energy Agency (IEA) was established to strengthen the cooperation between the member countries in the energy field. One element of these cooperative activities is to undertake energy research, development and demonstration (RD&D).

District Heating and Cooling is seen by the IEA as a means by which countries may reduce their dependence on oil and improve their energy efficiency. It involved increased use of indigenous or abundant fuels, the utilization of waste energy and Combined Heat and Power (CHP). IEA's "Program of Research, Development and Demonstration on District Heating and Cooling" was established at the end of 1983.

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The work which is the subject of this report was prepared under Annex IV of the program, which was implemented in 1993 with the partication of nine countries: Canada, Denmark, Finland, Germany, The Netherlands, Norway, Sweden, United Kingdom and the USA. The Republic of Korea is now also participating in this Annex.

An Experts Group provided valuable comment on drafts of this report. The members of the Experts Group were:

- H.C. Mortensen (Denmark)
- Arno Sijben (The Netherlands)
- Zoltan Korenyi (Germany)
- Marc Rosen (Canada)
- Kim, Dong Joon (Korea)
- Jorma Kotakorpi (Finland)
- Atle Norstebo (Norway)
- William Rowe (United Kingdom)

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Executive Summary

SCOPE AND PURPOSE

This report describes the energy efficiency, economic and environmental implications of alternatives for integrating district cooling with Combined Heat and Power (CHP). The purpose of the report is to provide guidance to designers of district cooling systems to identify the best options for integrating district cooling with CHP in new plant facilities.

Each case will have its own particular technical and economic parameters, and this report is intended to aid in structuring the essential case-specific analysis, rather than substituting for such an analysis. Capital and operating costs for CHP and chiller technologies are presented, but significant variations in costs can occur due to currency values and other case-specific factors.

For the purposes of this report, district cooling is defined as any system which provides building cooling through the distribution of chilled water, hot water or steam from a central plant. Thus, cooling achieved through distribution of district hot water or steam to drive absorption chillers located in buildings is also considered district cooling.

The report addresses:

- the thermodynamic fundamentals of CHP and cooling, providing a conceptual foundation for later quantification of the efficiency of alternative cooling/CHP options;
- the efficiency, air emissions and economics of alternative CHP technologies (gas turbine, reciprocating engine, steam turbine and gas turbine combined cycle);
- the efficiency, refrigerant environmental impacts and economics of alternative cooling technologies (electric centrifugal, steam turbine centrifugal, one-stage steam absorption, two-stage steam absorption and hot water absorption);
- review of fundamental aspects of district heating and cooling systems which are relevant to integrating district cooling with CHP;
- the efficiency and economics of integrated cooling/CHP technology alternatives, including presentation of economic formulas, discussion of key economic variables and calculation of cooling costs for illustrative hypothetical scenarios; and
- case study examples of integrating district cooling/CHP.

ENERGY EFFICIENCY

Basis for Efficiency Comparisons

A consistent "figure of merit" for comparing the energy efficiencies of different options for combining CHP and cooling is problematic because each option, employed in a given circumstance, will produce different annual quantities of electricity, heating and cooling. Efficiency comparisons based on summing these three types of energy outputs will be misleading because they ignore the differing exergy qualities of electricity, heating and cooling.

Consequently, comparisons of the efficiencies of alternative CHP/chiller options were made on the basis of maximizing chilled water production. Heat-driven chillers were supplemented with electric-drive chillers using available electric output from CHP.

Findings Regarding Efficiency

1. If the goal is maximum cooling output per unit of fuel used, the CHP technologies rank as follows, from highest to lowest output:

    - Gas turbine combined cycle
    - Diesel engine
    - Gas turbine
    - Steam turbine

    This ranking holds true regardless of the chiller technologies employed, although the extent of differences between the CHP types varied depending on the chiller technologies.

2. With a simple cycle gas turbine, the higher-temperature heat-driven chillers (supplemented by electric drive chillers) provide more cooling output than the lower-temperature options, with the electric-chiller-only option providing the lowest cooling output. This is also roughly true with a diesel engine, although the lower-temperature heat-driven options compare more favorably because the temperature of the useful thermal output of diesel engines is more limited compared to the gas turbine.

3. With steam turbine and gas turbine combined cycle CHP, the electric drive chiller provides the highest cooling output, followed by hot water absorption and other heat-driven options, roughly in order of increasing driving temperature. The differences between chiller types with gas turbine combined cycle are less than those for steam turbine CHP.

4. When combining cooling with CHP in new gas turbine combined cycle facilities, there are only small differences in overall efficiency between maximizing electric production and using electric drive chillers compared to extracting some of the thermal energy and

using it to operate absorption chillers. The differences in practical efficiencies are within the range where specific equipment selection and design conditions will determine which alternative is most efficient.

5. Simple cycle gas turbine CHP can appear attractive from an efficiency standpoint when the thermal output is viewed as "waste heat." However, it can be argued that this is because, from the standpoint of new plant design, total efficiency has not really been optimized with a simple cycle, i.e., generally there is the capability to generate additional electricity in a combined cycle.

6. For a new CHP facility, there is not a compelling argument for using heat generated through CHP to drive chillers as opposed to installing a condensing tail to drive electric chillers. However, this argument does not hold for the smaller end of the scale of CHP facilities (e.g., 5 MW_e), where due to economies of scale it is generally not cost-effective to install a steam turbine to drive a generator in a combined cycle.

ECONOMICS

This report addresses the costs of generating cooling energy using CHP. However, distribution costs can be a significant part of the total cost of district cooling. Where a district heating system is well developed, distribution of "cooling energy" via the district heating loop for conversion with absorption chillers has the potential to be the most cost-effective option considering both plant and distribution costs.

The economics of integrated cooling/CHP options are highly dependent on many case-specific factors. The following discussion summarizes the results of the illustrative scenarios presented in the report for new CHP systems in the 20-25 MW_e size range under stated load and economic assumptions.

CHP options

1. In the illustrative scenarios, simple cycle gas turbine CHP provides the lowest cooling cost at low values of electricity (3 cents/kWh_e), due in large part to its low investment cost.

2. Combined cycle gas turbine CHP provides the lowest cooling cost at higher electricity values (above 5 cents/kWH_e) as a result of its high electric efficiency. As electricity value rises, the competitiveness of the gas turbine combined cycle increases faster than the other CHP options.

3. With the potential for steam turbine CHP to be fired with lower-cost fuel, this CHP option has the potential to be the most cost-effective option depending on specific fuel costs.

4. In CHP plants under 20 MW_e, reciprocating engine CHP can become more competitive than indicated in the illustrative scenarios, and in CHP plants above 50 MW_e, steam turbine CHP has the potential to be more competitive than indicated.

5. Sensitivity of cooling costs to changes in fuel cost, heat value and electricity value is lowest in the warm climate because net CHP costs are spread over a relatively large number of cooling utilization hours. Conversely, sensitivity of cooling costs to these factors is highest in the cold climate because net CHP costs are spread over a relatively small number of cooling utilization hours.

Chiller options

1. Based on the illustrative scenarios, electric drive chillers combined with gas turbine CHP (at low electric values) and gas turbine combined cycle CHP (at high electric values) provided the lowest cooling costs for centralized chilled water district cooling. However, in many scenarios the cost differences between electric drive cooling and heat-driven options (supplemented with electric drive) were quite small and can be considered insignificant in view of the many case-specific variables which can affect the calculations. In general, the costs of the CHP are more significant than the costs of the chiller equipment.

2. Generally, cost differences between the cooling technologies combined with simple cycle gas turbine and diesel engine CHP are very small because the electric output of these CHP technologies is not affected by thermal extraction. In contrast, with steam turbine CHP and to a lesser extent gas turbine combined cycle CHP, cost differences between chiller technologies are more significant because with the steam cycle the electric output decreases when thermal energy is extracted, and this derate increases with increasing thermal extraction temperature.

3. Aside from direct economic considerations, the value of flexibility and reliability may lead the system designer to install heat-driven chillers. For example, heat-driven cooling can help protect against penalties associated with a loss of power generation capacity at peak, since with heat-driven chillers the system operator can fire up relatively inexpensive standby boiler capacity.

4. For all CHP types, the economic differences between the heat-driven chiller options were relatively small, with costs slightly higher for chillers requiring higher-temperature driving energy. In essence, the higher investment costs for higher-temperature heat-driven options was to a large extent offset by their higher efficiencies.

Chapter 1
Introduction

1.1 PURPOSE

The purpose of this report is to provide guidance to designers of district cooling systems to identify the best options for integrating district cooling with Combined Heat and Power (CHP) by describing the energy efficiency, economics and environmental implications of alternatives for integrating district cooling with CHP. Each case will have its own particular technical and economic parameters, and this report is intended to aid in structuring the essential case-specific analysis rather than substituting for such an analysis. Capital and operating costs for CHP and chiller technologies are presented, but significant variations in costs can occur due to currency values and other case-specific factors.

For the purposes of this report, district cooling is defined as any system which provides building cooling through the distribution of chilled water, hot water or steam from a central plant. Thus, cooling achieved through distribution of district hot water or steam to drive absorption chillers located in buildings is also considered district cooling.

It is important to note that this report addresses comparison of technologies for *district* cooling, as opposed to comparison of district cooling with individual building chillers. There are many advantages to district cooling generally, regardless of the specific district cooling technology employed. Although not the focus of this report, it is worth noting these advantages, which include:

1. improved opportunities for reducing peak electric demand through non-electric cooling systems and through thermal energy storage;

2. cost avoidance due to reduced requirements for electricity distribution infrastructure serving downtown areas;

3. improved ability to produce chilled water from plant capacity which is underutilized during summer;

4. improved energy efficiency resulting from optimal loading of chiller (and, as applicable, CHP) equipment;

5. improved ability to use low-temperature heat sinks for condenser cooling;

6. lower costs due to economies of scale in plant investments and operating costs; and

7. improved economics resulting from load diversity.

1.2 APPROACHES TO DISTRIBUTING COOLING ENERGY

This report focuses on CHP and chiller technologies for *generating* cooling energy. *Distribution* of cooling energy can be accomplished through distribution of chilled water (or some other cooling medium such as ice slurry) or district heat in the form of hot water or steam. Although the analysis of distribution alternatives is not the focus of this report, the distribution approach can have a major impact on the economics of alternatives for integrating district cooling with CHP. From the standpoint of cooling energy distribution, the major approaches to integrated district cooling/CHP can be categorized as follows:

- Centralized chilled water
- Decentralized chilled water
- Dispersed district-heat-driven
- Decentralized district-heat-driven

Centralized chilled water systems serve a large area through one integrated chilled water distribution system fed by one or several larger CHP plants using any type of chiller technology. (See **Figure 1.1**.)

Decentralized chilled water systems employ multiple independent chilled water distribution systems, fed by smaller CHP plants using any type of chiller technology. This approach may be used during the early stages of development, with eventual interconnection of the multiple systems in the long term. (See **Figure 1.2**.)

Dispersed district-heat-driven absorption approaches use small absorption chillers, located in user buildings, which are driven with district hot water or steam. This approach eliminates the need for investment in chilled water distribution. (See **Figure 1.3**.)

Decentralized district-heat-driven systems use small absorption plants driven with district hot water or steam to produce chilled water for distribution. This reduces the investment in chilled water distribution, and allows the use of district hot water while avoiding some of the potential problems relating to: 1) installation of chillers and cooling towers in each building; and 2) potential hot water service pipe constraints. (See **Figure 1.4**.)

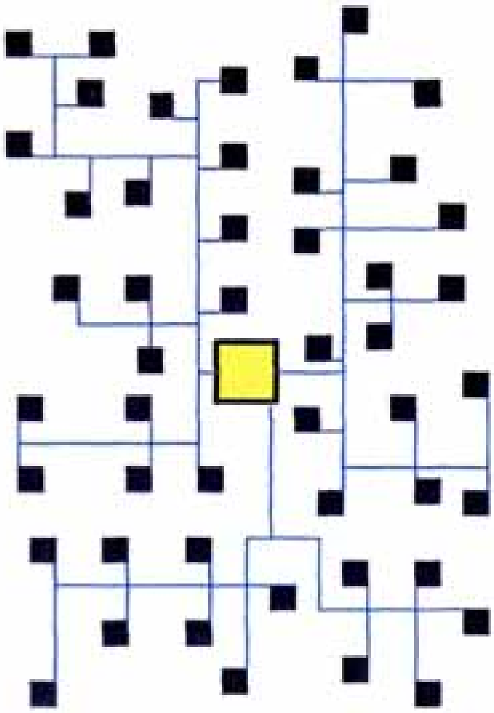

Figure 1.1 Centralized chilled water system

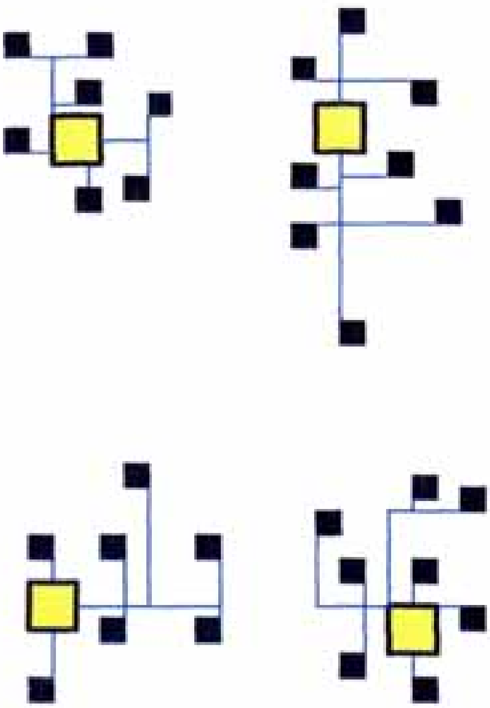

Figure 1.2 Decentralized chilled water system

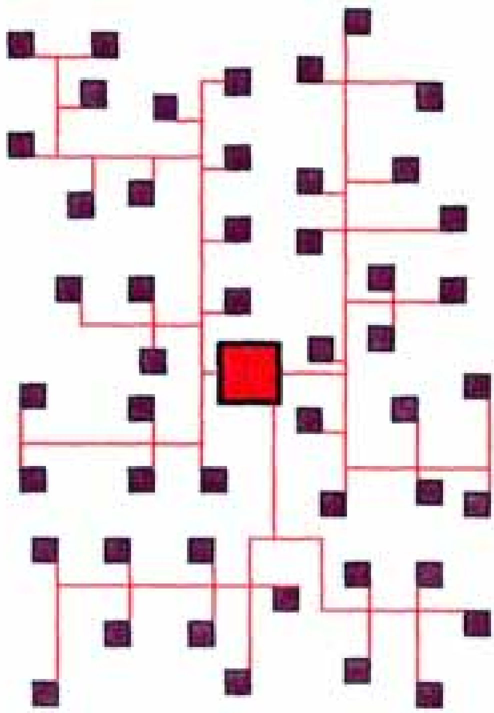

Figure 1.3 Dispersed district-heat-driven system

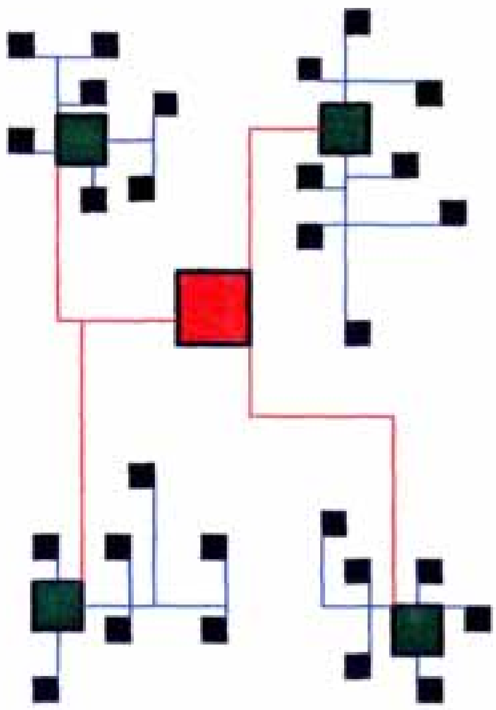

Figure 1.4 Decentralized district-heat-driven system

LEGEND FOR FIGURES 1.1 -- 1.4

- Building receiving district chilled water service
- Building converting district heat to cooling using absorption chiller
- Chilled water distribution
- District heat distribution
- CHP/district chilled water plant
- CHP/district heating plant
- Decentralized district-heat-driven absorption plant

1.3 STRUCTURE OF THE REPORT

The balance of the report is organized as follows:

- Chapter 2 discusses the thermodynamic fundamentals of CHP and cooling, providing a conceptual foundation for later quantification of the efficiency of alternative CHP/cooling configurations.
- Chapter 3 describes the efficiency, air emissions and economics of alternative CHP technologies (gas turbine, reciprocating engine, steam turbine and gas turbine combined cycle);
- Chapter 4 describes the efficiency, refrigerant environmental impacts and economics of alternative cooling technologies (electric centrifugal, steam turbine centrifugal, one-stage steam absorption, two-stage steam absorption and hot water absorption);
- Chapter 5 briefly reviews fundamentals of district heating and cooling systems which are relevant to integrating district cooling with CHP.
- Chapter 6 addresses the efficiency and economics of integrated cooling/CHP technology alternatives, including presentation of economic formulas, discussion of key economic variables and calculation of cooling costs for illustrative hypothetical scenarios.
- Chapter 7 describes case study examples of integrated district cooling/CHP.

Appendices are provided on unit conversion factors and currency exchange rates.

1.4 UNITS

The following units are used in this report unless otherwise noted:

• Temperature	°C
• Pressure	bar (absolute)
• Heat	kWh_{th} or MWh_{th}
• Electricity	kWh_e or MWh_e
• Cooling energy	kWh_c or MWh_c
• Fuel heat content	Lower Heating Value (LHV)
• Cost	U.S. dollars ($) or U.S. cents
• Volume	liters
• Air emissions	mg/MJ fuel or g/kWh_e

Appendix A provides conversion factors which may be useful in adapting data into other units. Appendix B provides currency conversion factors at the time of publication.

1.5 ABBREVIATIONS

The following abbreviations are used in the report:

• CHP	Combined Heat and Power
• LHV	Lower Heating Value
• kWh	kiloWatt hour
• MWh	MegaWatt hour
• C	Degree Celsius
• $	U.S. dollar
• EFLH	Equivalent Full Load Hours
• mm	millimeters
• m	meters
• m^3	cubic meters
• NOx	Nitrogen Oxides
• SO_2	Sulfur dioxide
• CO	Carbon Monoxide
• HC	Hydrocarbons
• CO_2	Carbon dioxide
• SCR	Selective Catalytic Reduction
• ISO	International Standards Organization
• mg	milligrams
• MJ	MegaJoule
• g/kWh_e	grams per kWh electricity
• CFC	Chlorofluorocarbon
• HCFC	Hydrochlorofluorocarbon
• HFC	Hydrofluorocarbon
• COP	Coefficient of Performance
• mbar	millibar
• FBC	Fluidized Bed Combustion
• PFBC	Pressurized Fluidized Bed Combustion

Chapter 2
Thermodynamic Issues in Integrating District Cooling and CHP

2.1 INTRODUCTION

The basis for choosing between mechanical, typically electric drive, chillers and absorption chillers is not straightforward when chilling is integrated with Combined Heat and Power (CHP). Often, utilities are advised that absorption chillers should be used when CHP is adopted because this will provide a summer load for the heat rejected from the electricity generation process. On the other hand, operators of steam turbines know that electric output will decrease if the condensing or back pressure on the turbine is increased above its design condensing pressure and temperature.

A fundamental thermodynamic issue is whether it is more efficient to:

1. produce as much electricity as possible from a condensing power plant and produce cooling with electric drive chillers; or

2. extract steam from the turbine or increase the turbine back pressure enough to operate an absorption chiller.

2.2 TERMS AND SUBSCRIPTS

The terms and subscripts used in the following discussion are summarized below:

Terms

η	Efficiency
COP	Coefficient of Performance -- the quantity of desired energy output (heating or cooling) divided by the energy input to the process
T	absolute temperature (degrees Kelvin)

Subscripts

C	Ideal process (Carnot)
ABS	absorption cycle
O	Overall (combining power generation with refrigeration cycle)
E	Evaporator
G	Generator
AC	Absorber/condenser
EX	Extraction

2.3 IDEAL EFFICIENCY

The best way to understand the differences between the two alternatives noted in **Section 2.1** is to start with the Carnot equation for the ideal theoretical efficiency for a heat driven engine:

$$\eta_c = \frac{T_1 - T_2}{T_1} \qquad (1)$$

where T_1 is the absolute temperature of the heat input and T_2 is the absolute temperature of the heat rejected from the process, i.e., for a steam turbine the condensing temperature. (The absolute temperature is the temperature above absolute zero (0°K) and is equal to the temperature in °C plus 273.) Generally, the condensing temperature can be assumed to be close to ambient temperatures. In the following discussion the condensing temperature will be assumed to be 35°C, a conventional summer design point for cooling towers used to reject heat in power plants and cooling plants.

If the condensing temperature is assumed to be fixed, the principal variable is therefore the temperature of the heat supply. It follows that to get the most "usefulness" out of fuel it is important to utilize the high temperatures that are available when fuels burn. It is equally clear from the Carnot equation (1) that if the condensing temperature (T_2) increases, then the efficiency will decrease.

Figure 2.1 shows the relationship of η_c with the condensing temperature (T_2) when T_1 equals 550°C. Once a heat supply temperature has been established, then η_c moves from the maximum at the lowest value of T_2 to zero as T_2 approaches T_1. Thus, as the condensing temperature T_2 increases, the quantity of electricity decreases.

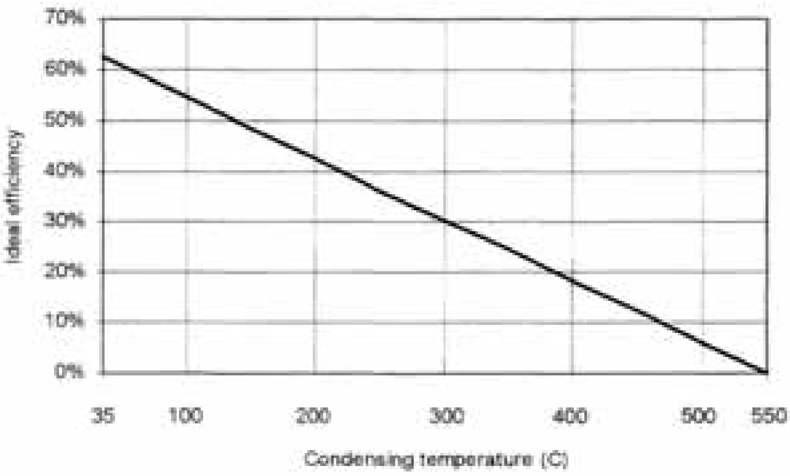

Figure 2.1 Relationship of ideal efficiency to condensing temperature when heat supply temperature is 550°C

For a compression refrigeration cycle the ideal performance can be described by the reverse Carnot cycle:

$$COP_C = \frac{T_2}{T_1 - T_2} \qquad (2)$$

where T_2 is the temperature of the heat input (evaporator), T_1 is the temperature of the heat rejected (condenser) and COP_c is the Coefficient Of Performance of the cooling cycle. As is the case for the heat engine discussed above, the condenser temperature may vary according to the available heat sink, but can generally be assumed to be close to ambient. Different condenser temperatures will affect the performance of refrigeration cycles but won't have a significant impact on the comparative efficiencies of alternative cooling/CHP options. The evaporator temperature T_2 is the temperature of chilled water supply and is fixed according to the design of the district cooling system, typically about 5°C.

Combining the ideal efficiency of power generation (equation 1) with the ideal efficiency of the refrigeration cycle (equation 2), the overall efficiency of the ideal compression chiller cooling η_o is:

$$\eta_{co} = \eta_c * COP_C \qquad (3)$$

The absorption cycle differs from the compression cycle in that the environment for the required evaporation temperature is created through a physio-chemical process with some minor mechanical (pumping) inputs, rather than a purely mechanical process. The formula for the COP_C for an absorption refrigeration cycle is equal to the formula of the COP_C for an ideal refrigeration cycle working between T_E and T_C multiplied by η_c for an ideal heat engine working between T_G and T_A. This can be simplified as shown in equation 4 assuming the temperature rise over the absorber and condenser is quite small (in the range of 5-7°C) and can thus be set as one temperature, T_{AC}, in the simplified expression shown.

$$COP_c = \frac{T_E}{(T_{AC}-T_E)} * \frac{(T_G-T_{AC})}{T_G} = \frac{T_E*(T_G-T_{AC})}{T_G*(T_{AC}-T_E)} \qquad (4)$$

where

T = evaporator temperature (chilled water temperature)
T_G = generator temperature (steam temperature)
T_{AC} = absorber/condenser temperature (condenser water temperature)

COP will increase with increasing T_G. However, it is important to note that T_G is limited by the working fluids and the design of the absorption chiller. Representative values for T_G in lithium bromide absorption are 120°C/2 bar for one-stage and 170°C/8 bar for two-stage. Although high heat supply temperatures could be achieved through combustion of fossil fuel or high pressure steam, commercially available absorption chillers cannot take advantage of the available "usefulness" of high temperature energy.

For a refrigeration cycle using electric drive chillers, the temperature of the heat supply to the engine is equal to the temperature at which heat is supplied in an electric power plant. This temperature can range from 1100°C (if a gas turbine topping cycle is used) to about 550°C (if only a steam cycle is used). Optimizing the electric output from the power plant usually requires a steam turbine bottoming cycle with the temperature of the rejected heat at about ambient temperature.

The overall ideal efficiency η_{CO} (which accounts for the efficiency of both power generation and refrigeration) of electric drive compression, one-stage and two-stage absorption chillers can be derived as shown in **Table 2.1** assuming:

- chillers and power plants both reject heat at 35°C (T_2 from equation 1, T_1 from equation 2 or T_{AC} from equation 4);

- evaporator temperatures for the chillers are 5°C (T_2 from equation 2 or T_E from equation 4); and

- energy supply temperature to the power plant turbine is 550°C (T_2 from equation 2 or T_1 from equation 1);

- energy supply temperature to the 1-stage absorption chiller is 120°C and to the 2-stage absorption chiller is 170°C (T_G from equation 4)

Electric drive compression cooling

$$\frac{550-35}{550+273} \cdot \frac{5+273}{35-5} = 0.626 \cdot 9.27 = 5.80$$

One-stage absorption cooling

$$\frac{120-35}{120+273} \cdot \frac{5+273}{35-5} = 0.216 \cdot 9.27 = 2.00$$

Two-stage absorption cooling

$$\frac{170-35}{170+273} \cdot \frac{5+273}{35-5} = 0.305 \cdot 9.27 = 2.82$$

Table 2.1 Overall ideal efficiencies of cooling technologies (η_{CO})

These idealized values for the overall efficiency (kW_c/kW_{th}) according to the Carnot formulas indicate that electric cooling utilizing a steam turbine power plant is two to three times as efficient as the absorption alternatives if the heat for absorption is taken from a boiler.

The Carnot formulas have been arranged above so as to illustrate that the key difference in the overall efficiencies of the electric drive and absorption chillers is the limited ability of absorption equipment to use higher generator temperatures. The overall efficiency of the electric drive chiller, under the assumption that the temperature of the heat supplied to the power plant is limited to $T_1 = 170°C$, will be equal to that of the two-stage absorption chiller shown in **Table 2.1**. This reinforces the concept that cooling requires work and, based on Carnot's principles, the efficiency of accomplishing this work is the same regardless of the refrigeration cycle if we assume identical temperatures and operating conditions.

Conceptually, a steam turbine drive chiller is the same as the steam turbine power plant/electric drive chiller combination, except that the temperature of the heat supply to the steam turbine drive chiller is significantly lower (e.g., 185°C) than for the steam turbine power plant, thereby resulting in an overall efficiency close that of the absorption processes.

2.4 ACTUAL EFFICIENCIES WITHOUT CHP

The above equations describe ideal reversible processes and provide a useful perspective on the differences between the technologies. However, the efficiency of real processes is lower and can vary considerably depending on boiler efficiencies, turbine and chiller design and other case-specific parameters. **Table 2.2** shows typical values for actual efficiencies (see **Chapters 3 and 4** for further information on actual efficiencies).

Power plant efficiency (LHV):	0.38
Steam boiler efficiency (LHV):	0.90
COP electric drive chillers (excluding auxiliaries):	5.87
COP one-stage absorption chillers (excluding auxiliaries):	0.67
COP two-stage absorption chillers (excluding auxiliaries):	1.20

Table 2.2 Typical actual efficiencies for chillers and related energy conversion equipment

By combining the efficiencies in **Table 2.2** for power plants/boilers with the refrigeration cycle efficiencies, the overall efficiencies of chiller technologies can be calculated as shown in **Table 2.3**. These values show that the real process efficiencies are 30-38% of the ideal efficiencies. The overall actual efficiency ($kW_{cooling}/kW_{fuel}$) of electric drive chillers will be almost 4 times that of one-stage absorption chillers (about 3 times using the Carnot formula) and 2 times that of the two-stage alternative (about equal using the Carnot formula).

		Actual Overall Efficiency (η_O)	Actual / Ideal (η_O / η_{CO})
Electric drive cooling			
0.38 * 5.87	=	2.23	38%
One-stage absorption cooling			
0.90 * 0.67	=	0.60	30%
Two-stage absorption cooling			
0.90 * 1.20	=	1.08	38%

Table 2.3 Overall efficiency ($kW_{cooling}/kW_{fuel}$) for different cooling options, comparing actual and ideal efficiencies

2.5 ACTUAL EFFICIENCIES WITH CHP

As illustrated above, the key source of inefficiency of absorption processes is their inability to effectively use the high temperature of the primary energy, which is typically made available through fuel combustion. The exhaust gas from a gas turbine is sometimes seen as "waste heat." However, the temperature is still about 450-550°C.

Based on the discussion above, the heat available in the exhaust gas or extracted from a steam turbine cannot be considered "waste" heat if it is taken at temperatures greater than the design condensing temperature (typically about 35°C). Above that temperature the heat can be used for electric production (via a bottoming steam cycle) or to provide steam to absorption chillers. However, if the energy in the exhaust gas is used directly in an absorption cycle, which only utilizes a temperature of 120-170°C, the overall actual efficiencies will be low compared to the electric cooling option, as shown in **Table 2.3**.

Therefore, to make a comparison between the different chiller options, "waste heat" should be defined as heat below the assumed ambient heat rejection temperature of 35°C. A CHP plant is therefore assumed to include a steam turbine in order to maximize efficiency. By integrating CHP and absorption technology, the advantage is not that "waste heat" is used, but that the steam turbine converts the higher temperature energy available from fuel combustion into mechanical energy or electricity, and brings the temperature of the energy down to the level where it can be effectively used in the absorption process. The potential to "produce work" that is squandered in a boiler/absorption chiller combination is captured in a CHP/absorption chiller combination.

For a CHP plant the electric efficiency of a gas turbine/generator or a high pressure steam turbine/generator will not be affected if low pressure steam is extracted for purposes other than electric production. For example:

- In a simple cycle gas turbine CHP plant, the electric efficiency is not affected if steam is produced using the hot flue gas.
- In a steam turbine CHP plant, the electric efficiency of the low pressure steam turbine/generator is affected by extraction of low pressure steam but the efficiency of the high pressure steam turbine/generator is not affected.

Thus, to compare the efficiencies of electric drive chillers utilizing electricity from a condensing power plant and absorption chillers extracting steam from a CHP plant, only the conditions from the extraction point down to condensing pressure need to be compared.

As the backpressure or condensing temperature of a CHP turbine is increased, it loses electric output. However, the quality (temperature), quantity and usefulness of the rejected heat increases. If the ratio of useful thermal energy extracted is divided by the electricity lost compared to a full condensing power plant, then the resulting number is the extraction COP (COP_{EX}):

$$COP_{EX} = \frac{\text{Thermal energy extracted}}{\text{Electricity lost}} \qquad (5)$$

Representative values for COP_{EX} at different extraction pressure levels and saturation temperatures for different steam turbine thermodynamic efficiencies are illustrated in **Figure 2.2** and **2.3**. The two different steam turbines with thermodynamic efficiencies of 0.85 and 0.75 are shown in the enthalpy/entropy diagram in **Figure 2.4**. The inlet steam conditions, 540°C/140 bar, were chosen to reflect a modern small scale steam turbine plant. COP_{EX} will vary somewhat depending on chosen inlet conditions but the variations are not significant for this conceptual discussion. High COP_{EX} can be obtained with low extraction pressures, with COP_{EX} going to about 16 for an extraction temperature of 70°C, as required for low temperature heating loads and for domestic hot water heating.

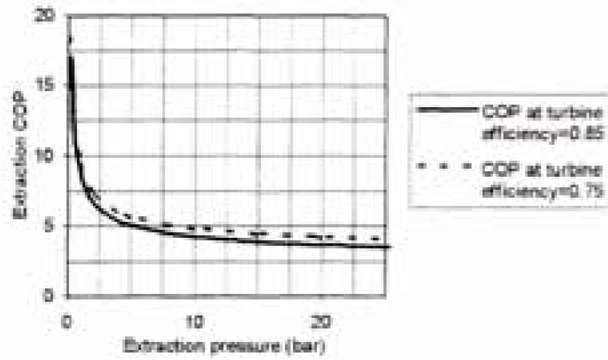

Figure 2.2 Extraction COP(kW_{th}/kW_e) as a function of steam extraction pressure

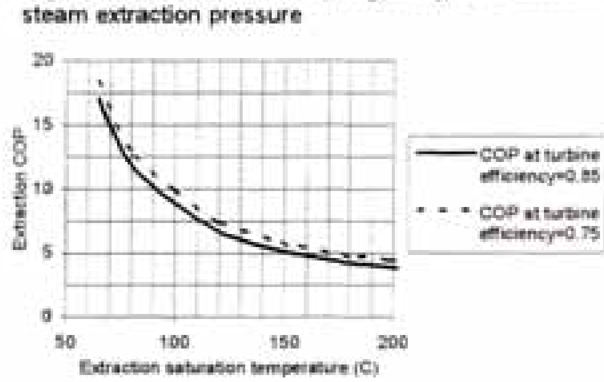

Figure 2.3 Extraction COP(kW_{th}/kW_e) as a function of steam extraction saturation temperature

COP_{EX} can then be combined with the COP of the absorption process (COP_{ABS}) to derive a COP comparable to the COP for an electric drive chiller:

If the extraction COP_{EX} =

$$\frac{\text{Quantity of useful energy at } T_2 \ (kWh_{th})}{\text{Electricity penalty } (kWh_e)} \qquad (6)$$

and, for an absorption chiller,

COP_{ABS} =

$$\frac{\text{Quantity of cooling (heat absorbed) } (kWh_{th})}{\text{Quantity of useful energy to operate the absorption heat pump } (kWh_{th})} \qquad (7)$$

and the quantity of useful energy at T_2 equals the quantity of useful energy to operate the absorption chiller (i.e., the numerator in equation 6 is the same as the denominator in equation 7), then the simplified equation is

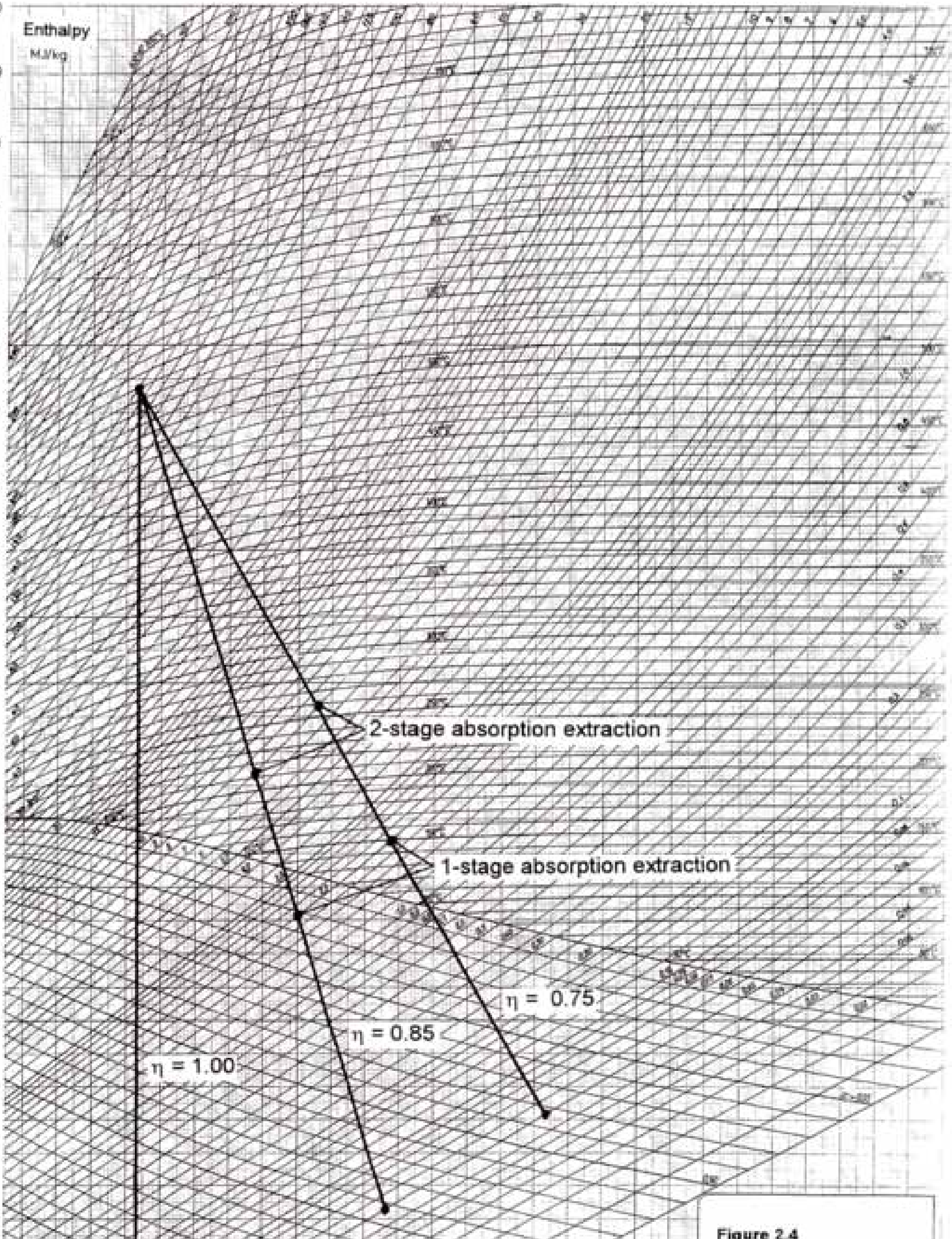

Figure 2.4

Enthalpy/entropy diagram

$$COP_{EX} * COP_{ABS} = \frac{\text{Quantity of cooling } (kWh_{th})}{\text{Quantity of electricity lost or used } (kWh_e)} \quad (8)$$

This expression is the definition of the COP of an electric drive chiller.

Calculations of actual overall COP_O for chiller/CHP combinations are shown in **Tables 2.4 and 2.5** using the chiller COPs from **Table 2.2** and extraction COPs as shown in **Figure 2.2** for the two steam turbine efficiencies.

Electric drive chillers		5.87
One-stage absorption chiller (2 bar steam)	6.67 * 0.67 =	4.47
One-stage absorption chiller (1 bar steam)	8.92 * 0.67 =	5.97
Two-stage absorption chiller (8 bar steam)	4.47 * 1.20 =	5.36
Two-stage absorption chiller (4 bar steam)	5.37 * 1.20 =	6.44

Table 2.4 COP_O ($kW_{cooling}/kW_{electricity}$) for different chiller options with a steam turbine thermodynamic efficiency of 0.85

Electric drive chillers		5.87
One-stage absorption chiller (2 bar steam)	7.39 * 0.67 =	4.95
One-stage absorption chiller, (1 bar steam)	9.90 * 0.67 =	6.63
Two-stage absorption chiller (8 bar steam)	4.96 * 1.20 =	5.95
Two-stage absorption chiller, (4 bar steam)	5.98 * 1.20 =	7.18

Table 2.5 COP_O ($kW_{cooling}/kW_{electricity}$) for different chiller options with a steam turbine thermodynamic efficiency of 0.75

These tables show that the actual overall COP is higher for absorption chillers if the actual temperature/pressure of the steam used to drive the absorption chiller is lower than design value. The drawback with lower extraction pressure is a lower output from the chiller (capacity decrease of about 30-35% at 50% of design pressure for both one-stage and two-stage absorption chillers), which will increase the investment per ton of capacity. On the other hand, although driving absorption chillers with very low temperature thermal energy, e.g., district hot water, involves relatively high capital costs for chiller capacity located in buildings or in decentralized chilled water production plants, substantial capital savings are possible due to reduced investment in distribution piping.

The tables also show that the COP of the absorption alternatives increases with lower steam turbine efficiency. However, this does not mean that installing a less efficient turbine will increase the feasibility of absorption cooling. A less efficient steam turbine produces a higher ratio of thermal to electric output, and is producing the thermal output at a relatively higher cost. Essentially, this excess thermal output truly is "waste heat" because the best available equipment is not being used.

2.6 SUMMARY OF THERMODYNAMIC OBSERVATIONS

To compare the efficiencies of electric drive chillers utilizing electricity from a condensing power plant and absorption chillers using extracted steam from a CHP plant, the extraction of low pressure steam in a steam turbine is the appropriate process to examine. Although the exhaust gas from a gas turbine is sometimes seen as "waste heat," the temperature is still in the range of 450-550°C and is therefore capable of generating additional electricity in a combined cycle. Only the conditions from the low pressure steam extraction point down to condensing pressure need to be compared.

Driving absorption chillers with thermal energy directly from boilers results in low theoretical and actual efficiencies compared to electric drive chillers using electricity from a condensing power plant. When combining cooling with CHP, there are only small differences in overall efficiency between maximizing electric production and using electric drive chillers compared to extracting some of the thermal energy and using it to operate absorption chillers. The differences in actual efficiencies are within the range where specific equipment selection and design conditions will determine which alternative is most efficient.

These conclusions relate to design considerations for new facilities. The availability of existing equipment may substantially alter the conclusions and the optimal design may in many cases include absorption chillers. For example, an existing CHP plant (either steam turbine or simple cycle gas turbine) without an condensing tail may be available for electricity generation but there is no corresponding heat load during the cooling season. In this case, the summertime thermal production truly is "waste heat" from an economic optimization standpoint, unless it is determined that it is cost-effective to install a steam turbine condensing tail and electric drive chillers.

2.7 COMPARING THE EFFICIENCIES OF CHP/DISTRICT HEATING AND COOLING OPTIONS [2-1]

2.7.1 Energy Analysis

Conventional thermodynamic analysis is based primarily on the first law of thermodynamics, which states the principle of conservation of energy. An energy analysis of an energy conversion system is essentially an accounting of the energies entering, exiting and stored within the system. Efficiencies, normally expressed as ratios of energy quantities, are often used to assess and compare systems.

However, energy analysis has several shortcomings:

- It is the usefulness or quality of an energy quantity, rather than simply the energy quantity itself, that is of value. For example, the heat rejected from the condensers of a power plant, although large in quantity, is of little usefulness since its temperature is close to that of the surrounding air or water (i.e., the thermal energy is of low quality).

- The thermodynamic losses which occur within a system are often not accurately identified and assessed with energy analysis. Although energy is conserved, and therefore energy losses are tracked as waste energy emissions from a system, energy quality is not conserved and can be degraded during a process (even if there are no energy losses). For example, the energy efficiency of electric resistance heating is almost 100% because there are almost no energy losses; however, the quality of the energy is greatly degraded in converting it from electricity to relatively low temperature heat, and this degradation is not accounted for with energy analysis.

These shortcomings make it problematic to compare the "energy efficiencies" of different options for combining CHP and cooling, because each option, employed in a given site-specific circumstance, will produce different annual quantities of electricity, heating and cooling. Therefore, efficiency comparisons based on the sums of the quantities of these three types of energy outputs will be misleading because they ignore the differing qualities of electricity, heating and cooling.

2.7.2 Exergy Analysis

Exergy analysis permits many of the shortcomings of energy analysis to be overcome. Exergy is the maximum work obtainable as a system comes to equilibrium with a reference environment, or, more simply, exergy is that portion of energy which is available for performing useful tasks. Exergy analysis identifies the causes, locations and magnitudes of process inefficiencies, and is founded upon the second law of thermodynamics. The second law states that, although energy cannot be created or destroyed, it can be degraded in quality, eventually reaching a state in which it is in complete equilibrium with the surroundings and hence of no further use for performing tasks.

Numerous investigations and applications have been reported for energy analysis, since it is the conventional method of thermodynamic analysis. Recently, exergy analysis has received increasing recognition by researchers in industry, government and academia, and the number of reported applications has grown considerably. Despite its potential usefulness relative to evaluating options for integrating CHP and cooling, exergy analysis is not used in this report because exergy analysis is not commonly understood, accepted or applied, and because application of exergy analysis would add a layer of complexity which could constrain the usefulness of the report. In the "real world" energy, not exergy, is still what is metered and paid for. In addition, since the focus of the report is on cooling, an appropriate efficiency comparison using energy analysis can be made by assuming that all CHP outputs (electricity and thermal energy) are used to produce cooling (the "maximum chilled water output" calculations presented in **Chapter 6**).

However, the efficiencies of CHP/cooling technology options are not compared on the basis of total energy outputs (electricity, heating and cooling) which would be produced on an annual basis, for the reasons discussed above. Similarly, environmental performance of CHP/cooling combinations is not expressed as emissions per unit of total energy output.

REFERENCES

2-1. This discussion is based on an unpublished paper, "Thermodynamics of CHP/District Cooling," by Dr. Marc A. Rosen, January 1995.

Chapter 3
CHP Technologies

3.1 INTRODUCTION

This chapter describes and quantifies the efficiency, emissions and economics of Combined Heat and Power (CHP) technologies. Following a discussion of key assumptions in **Section 3.2**, **Sections 3.3 - 3.6** provide detailed discussions of combustion turbines, reciprocating engines, steam turbines and combined cycles, respectively. **Section 3.7** briefly addresses fuel cells. **Section 3.8** compares the efficiency, emissions and economics of the major CHP technologies.

3.2 ASSUMPTIONS

In describing the CHP technologies, a medium temperature hot water system with peak supply/return temperatures of 120/75°C is assumed. However, because the economic optimum for CHP units (see **Chapter 5**) usually is less than 100% of the thermal peak load, base CHP supply/return temperatures of 100/75°C are used to present the basic characteristics of each CHP technology. As described in **Chapter 5**, the CHP unit can then produce about 55% of the peak heating demand and over 90% of the heating energy in a 120/75°C system, with the temperature of the thermal output increased with a boiler under peak conditions. Installation of relatively expensive CHP units larger than 50-55% of the peak heating demand is usually not economical unless the priority is producing electricity, with heat as a byproduct.

CHP performance is also described at the higher temperatures often used for absorption chillers: 2 bar/120°C for conventional one-stage absorption chillers and 8 bar/170°C for two-stage absorption chillers.

All efficiency calculations are based on the Lower Heating Value (LHV) of fuels. LHV does not include the latent heat of vaporization of water vapor in fuel combustion products.

In the discussions of simple cycle and combined cycle gas turbine technologies, reference is made to "ISO" conditions, which are: 15°C outdoor temperature, 60% relative humidity and 1013 mbar barometric pressure. Pressure drops in the intake and in the outlet were each assumed to be 10 mbar.

Detailed assumptions behind the economic calculations for each CHP technology are presented. Labor cost estimates were based on the assumption that CHP system labor represented an incremental addition to already-existing base staffing for a district energy plant. On a project-specific basis, labor costs are highly variable depending on site-specific circumstances.

3.3 GAS TURBINES

3.3.1 Description of Technology

Combustion turbines, often called gas turbines, are available in a range of sizes, from 1 to over 150 MW_e, and can be briefly described as follows (see **Figure 3.1**):

- The conventional gas turbine is an open process, with the intake air and exhaust gas respectively being taken from and released to the surroundings at atmospheric pressure.
- Air is compressed in a compressor, thereby increasing both the pressure and temperature.
- The compressed air is delivered to a combustion chamber where it is mixed with gaseous or liquid fuel and burned. The combustion takes place at a constant pressure and occurs with large quantities of excess air. The turbine exhaust contains oxygen (about 15% O_2) and is therefore capable of supporting additional combustion.
- The high-temperature, high-pressure gaseous combustion products enter the turbine, where the expanding gases perform mechanical work by rotating the turbine shaft. A portion of the produced work is used to drive the compressor and overcome losses, and the remainder is available for power production.
- In CHP applications the heat in the hot exhaust gas, with a temperature of 450-550°C, is recovered in a heat recovery boiler.

Natural gas and light to heavy fuel oil can be used as fuel for combustion turbines. While natural gas is a "clean" fuel and is relatively problem-free to use in a gas turbine, heavier fuel oils must usually be cleaned to reduce the level of substances that can cause high temperature corrosion or surface deposits in the hot gas path of the turbine. One potentially problematic aspect of using natural gas is the pressure level of the natural gas. With the high pressure ratio (pressure in the combustor after the compressor divided by intake air pressure) of modern gas turbines, the pressure of the natural gas from low pressure pipelines must be boosted to be able to use the gas in the gas turbine.

Research and development for gas turbines is intensive due to the large and expanding market. R&D efforts are primarily focused on increasing efficiency and/or reducing emissions (primarily NO_x). All major manufacturers of gas turbines 20 MW_e and larger now have combustors available or on the drawing board for NO_x emissions below 50 mg/MJ for natural gas without external cleaning.

or steam/water injection. Increased turbine inlet temperature is the main alternative for increasing the efficiency. R&D is therefore focused on advanced cooling of turbine blades and materials that can sustain turbine inlet temperatures of 1200 to 1400°C. Efficiencies above 40% are now attained by commercial aeroderivative gas turbines, with the latest industrial gas turbines having typical efficiencies of 37-38%.

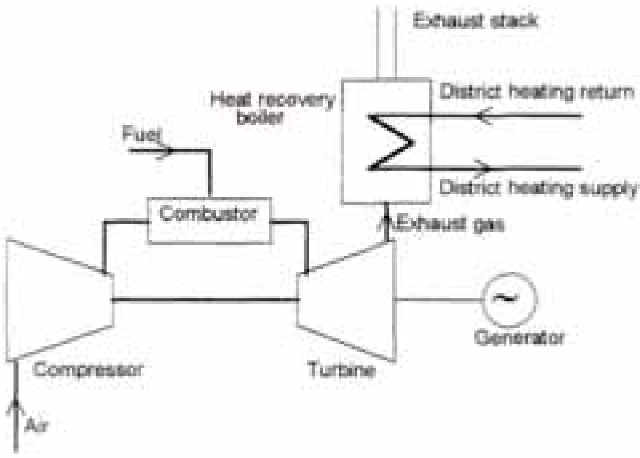

Figure 3.1 Schematic for gas turbine CHP

3.3.2 Performance

Electric and Thermal Efficiency

As noted above, in presentation of turbine performance data throughout this chapter, reference is made to ISO conditions, which are: 15°C outdoor temperature, 60% relative humidity and 1013 mbar barometric pressure. Pressure drops in the intake and in the outlet were each assumed to be 10 mbar. **Figure 3.2** summarizes the electric and thermal efficiency of a representative gas turbine under ISO conditions. As with other Sankey diagrams presented in this chapter, **Figure 3.2** is based on heat recovery at 100/75°C supply/return.

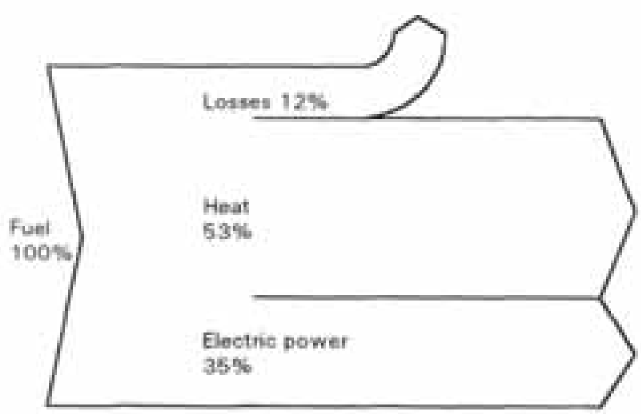

Figure 3.2 Sankey diagram (LHV) for CHP with gas turbine (size range 20 MW_e)

Electric efficiencies of a variety of gas turbines, at ISO conditions, are shown in **Figure 3.3**. Efficiency is generally higher in the larger turbines, ranging from 25% for very small turbines (1-2 MW_e) to 35-40% for larger turbines (20 MW_e and up). Efficiencies in the 20-40 MW_e interval are relatively high because many aeroderivative gas turbines, which generally have higher efficiencies, are available in that size range.

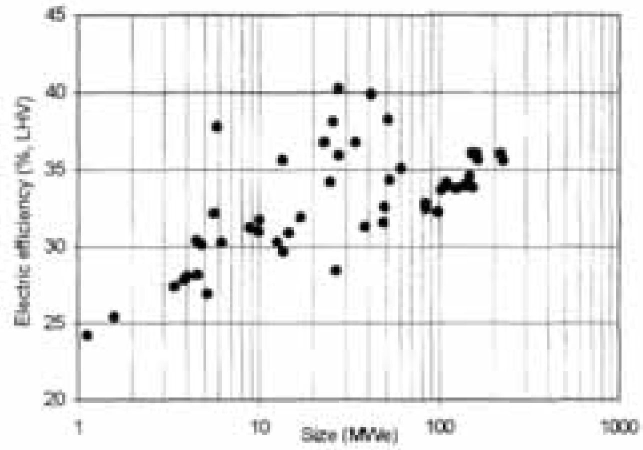

Figure 3.3 Gas turbine efficiency (LHV) at ISO conditions [3-1]

Figure 3.4 illustrates the impact of variations in the temperature of intake air on the *power output* of a variety of gas turbines, expressed as a percent of the output at the ISO outdoor temperature condition (15°C). Although there are significant variations between units, for most turbines power output increases by about 10% for every 15 °C drop in outdoor temperature, and conversely output decreases by about 10% for every 15°C increase in outdoor temperature. Extreme ambient temperature/power output behaviors can be found for some gas turbines, as shown in **Figure 3.4**. The extreme values are usually associated with aeroderivative gas turbines. Originally designed for other purposes, aeroderivatives can have "bottlenecks" that are normally not found in industrial gas turbines.

In an economic evaluation of a CHP plant it is important to consider performance at different ambient temperatures depending on the climate conditions during which electric power is most valuable. Power output can be boosted by chilling inlet air, either cooling directly on a baseload basis or indirectly through a thermal storage system.

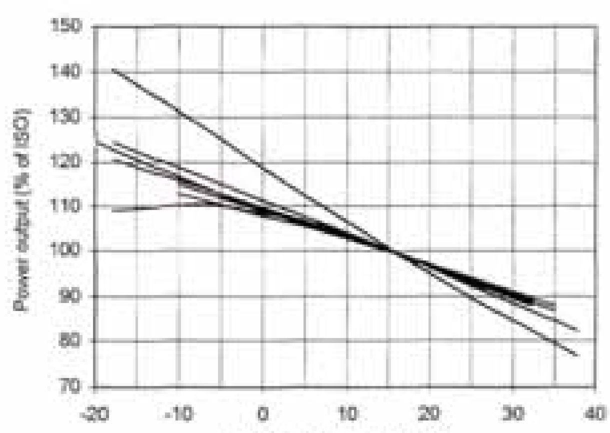

Figure 3.4 Gas turbine power output at different outdoor temperatures for selected gas turbines [3-2]

Figure 3.5 illustrates the impact of variations in the temperature of intake air on the *efficiency* of conversion of fuel to electricity. As with the power output, the relationship between outdoor temperature and efficiency varies between different gas turbines, but the variations are somewhat smaller. A representative impact, based on the data in **Figure 3.5**, is a 1% drop in electric efficiency for every 15°C increase in intake air temperature.

Figure 3.6 illustrates the range of exhaust gas temperatures at ISO conditions for various gas turbines, plotted according to facility size. Most equipment falls into the range of 450-550°C. This range of values will be used in later comparative analysis of gas turbine equipment. As discussed further below, the exhaust temperature has an impact on the heat recovery efficiency, especially if higher pressure/higher temperature steam is recovered.

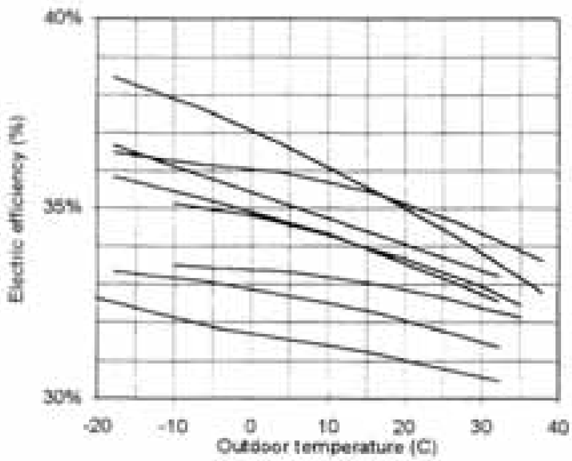

Figure 3.5 Electric efficiency (LHV) at different outdoor temperatures for selected gas turbines [3-2]

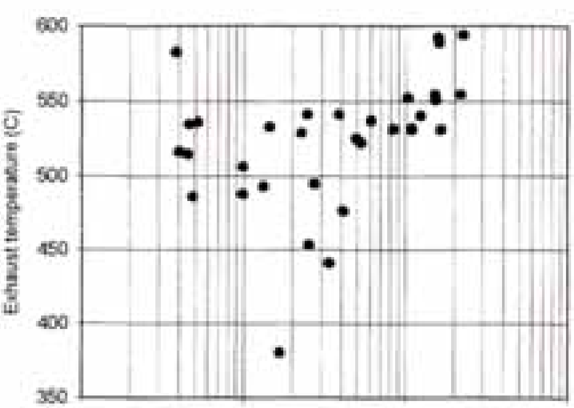

Figure 3.6 Gas turbine exhaust temperature at ISO conditions [3-1]

The electric efficiency of gas turbines can be increased by increasing the turbine inlet temperature and/or by increasing the pressure ratio. Generally, a higher pressure ratio results in a lower exhaust temperature. However, lower exhaust temperatures also reduce the potential for thermal recovery, thereby decreasing total energy efficiency. Higher electric efficiencies in gas turbine combined cycles can be obtained for turbines which have higher exhaust temperatures in simple cycle mode.

Figure 3.7 illustrates the relationship between exhaust gas temperature and intake air temperature for a variety of gas turbines. The exhaust gas temperature decreases by about 0.5-1.0°C per 1°C drop in the temperature of the intake (outdoor) air. Although the outdoor temperature has a relatively small impact on the exhaust gas temperature, variations in the temperature of intake air must be considered in designing the heat recovery boiler.

Figure 3.8 illustrates gas turbine electric efficiency as a function of load at ISO conditions. For economic optimization, it is critical that plants be sized to maintain a high utilization.

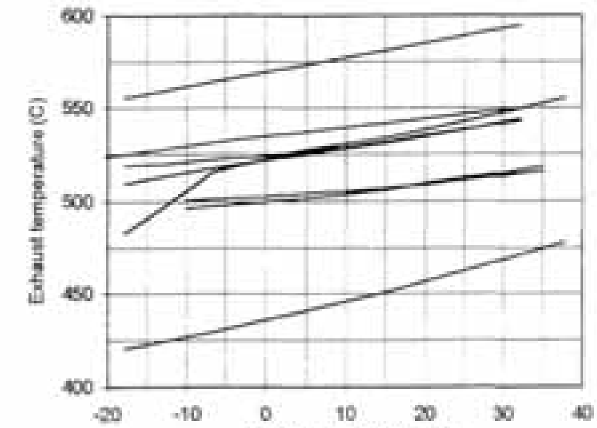

Figure 3.7 Gas turbine exhaust temperature at different outdoor temperatures for selected gas turbines [3-2]

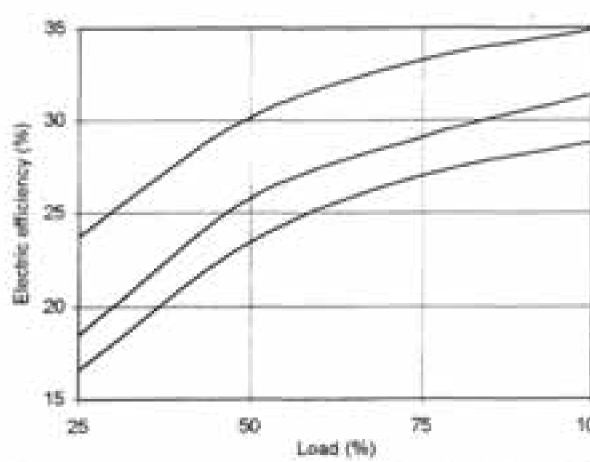

Figure 3.8 Gas turbine part load electric efficiency (LHV) at ISO conditions for selected gas turbines [3-2]

Figure 3.9 illustrates exhaust gas temperatures as a function of load at ISO conditions for a variety of gas turbines. Significant drops in temperature occur as load is decreased. Higher exhaust temperatures can be maintained at part load through the use of variable nozzles for inlet air. This is illustrated by the dashed lines, showing, for a particular gas turbine, the improved performance in the upper line (with variable nozzles) compared to the lower line (same turbine without variable nozzles).

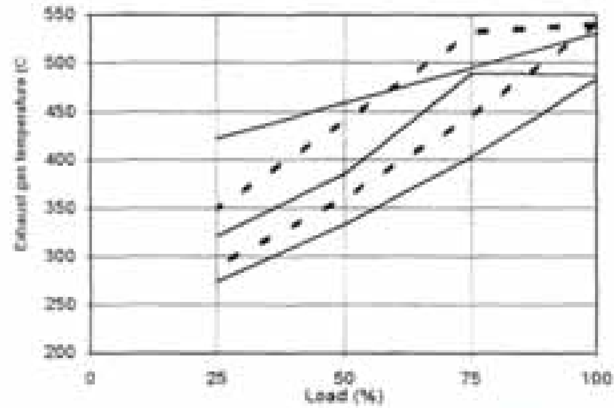

Figure 3.9 Exhaust gas temperature for different gas turbines [3-2]

Figure 3.10 illustrates the recovery of heat from a simple cycle gas turbine. The horizontal axis is the percentage transfer of the energy contained in the exhaust gas. The vertical axis is the temperature of the exhaust gas (two solid lines for 450°C and 530°C) and the temperature of the recovered hot water (dashed line). The exhaust gas enters the heat recovery boiler, where it transfers heat to the district energy loop returning at 75°C and exiting the heat recovery boiler at 100°C. The "pinch point," or lowest temperature difference, of 10°C occurs at the point where the temperature of the exhaust gas has been reduced to 85°C and the incoming district energy water is 75°C. (The pinch point will vary based on facility-specific conditions; however, the 10°C pinch point assumed here is a reasonable generalized assumption.)

The heat recovery efficiency can be calculated by following the exhaust gas lines down to the temperature of the intake air to the gas turbine, i.e., 15°C at ISO conditions. The amount of unrecovered energy (shown as negative energy transfer) is indicated on the horizontal axis by the interval between zero energy transfer and the point of intersection of this line and horizontal axis. This interval is about 20% (for 450°C exhaust gas) and 15% (for 530°C exhaust gas). The efficiency can then be calculated by dividing the total heat transferred (100%) by the sum of the total heat transfer (100%) and the energy loss, resulting in heat recovery efficiencies of 84% (for 450°C) and 86% (for 530 °C).

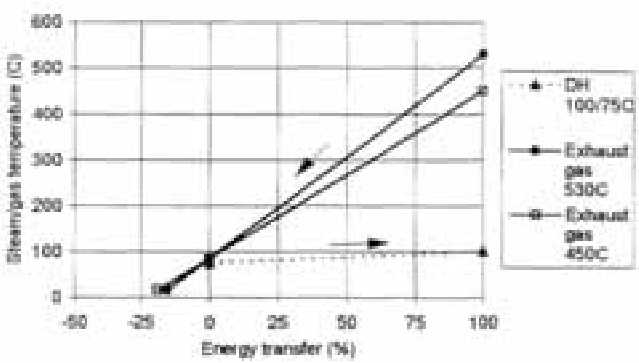

Figure 3.10 Heat recovery with hot water 100/75°C

Figure 3.11 illustrates the heat transfer if 170°C/8 bar steam is recovered instead of 100°C hot water. (This heat recovery temperature is used in this report as representative for driving for two-stage absorption chillers, although it is possible to drive such chillers with higher or lower temperature thermal energy.) In this case, the heat recovery efficiency varies between 71% and 78% depending on exhaust gas temperature. The exhaust gas enters the heat recovery boiler, where it transfers heat to steam (at 170°C in the hot end) and condensate (returning at 100°C in the cold end), and exits the heat recovery boiler at 130-140°C. In this case, the "pinch point" occurs at the end of the economizer. The exhaust gas temperature is more critical if higher temperature steam is used in the heat recovery boiler compared to hot water heat recovery.

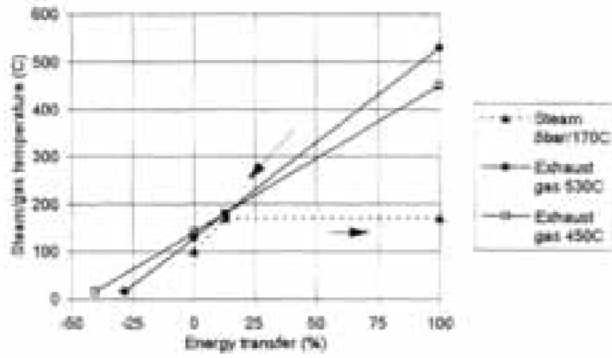

Figure 3.11 Heat recovery with steam 170°C/8 bar

Part-load performance for a representative gas turbine with heat recovery boiler is shown in **Figure 3.12**. The electric efficiency part-load performance is the same whether hot water (100/75°C) or steam (170°C/8 bar for two-stage absorption chillers) is recovered, but the overall efficiency differs. With recovery of 170°C/8 bar steam, overall efficiency is lower at 100% load and decreases faster at lower loads due to the more significant impact of lower exhaust temperatures.

Figure 3.12 and the associated calculations were made based on the following assumptions:

- 35% electric efficiency at ISO base load conditions
- 530°C exhaust temperature at base load conditions
- 97% generator efficiency
- 97.5% gearbox efficiency
- 2.5% heat/gas losses and boiler blowdown
- 15 mbar backpressure from the heat recovery boiler
- Representative values from previous figures for part load and outdoor temperature characteristics
- The back pressure from the heat recovery boiler will decrease the performance of the gas turbine, with electric efficiency reduced by 0.7% for every 10 mbar back pressure

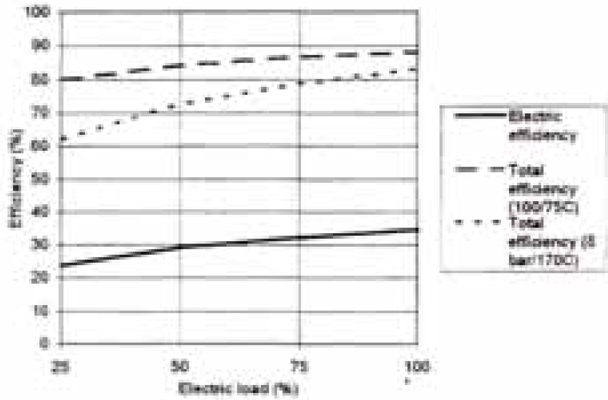

Figure 3.12 Part load efficiency (LHV) for gas turbine CHP with hot water heat recovery (100/75°C) and with steam heat recovery (170°C/8 bar)

Figure 3.13 illustrates representative gas turbine CHP efficiency at different outdoor temperatures. Although electric output increases with lower outdoor temperatures (about 3% per 15°C), overall efficiency declines (about 1.5% per 15°C) due to declining exhaust gas temperatures. Electric efficiency is equal for both hot water (100/75°C) and steam (170°C/8 bar) heat recovery. Overall efficiency at 100% load is lower for steam recovery compared to hot water recovery, but as temperature declines, efficiency declines at the same rate as with hot water recovery.

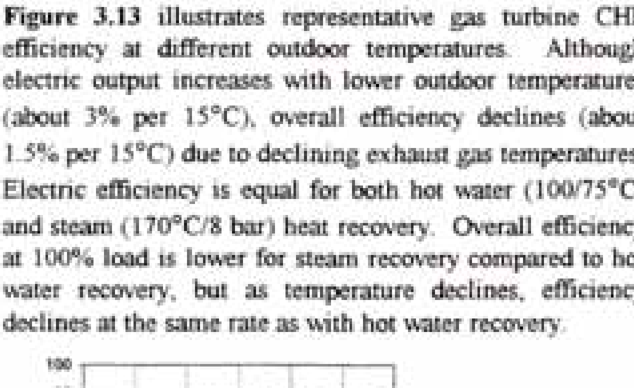

Figure 3.13 Gas turbine CHP efficiency (LHV) at different outdoor temperatures with hot water heat recovery (100/75 °C) and with steam heat recovery (170°C/8 bar)

Figure 3.14 shows the electric and total efficiencies as a function of heat recovery temperature. The electric efficiency is unchanged regardless of district heat supply temperature as long the backpressure from the heat recovery boiler is assumed to be equal regardless of the heat recovery temperature (10°C pinch point used for all temperatures). Heat recovery up to 100°C heat supply temperature is based on hot water with a 25°C temperature increase in the heat recovery boiler. Above 120°C, saturated steam with a condensate return temperature of 100°C is assumed.

Although the total efficiency decreases with increasing heat supply temperature, it is important to note that if there is a use for lower temperature hot water, an additional hot water heat recovery boiler can be installed to raise the total efficiency up to the same level achievable when recovering only hot water.

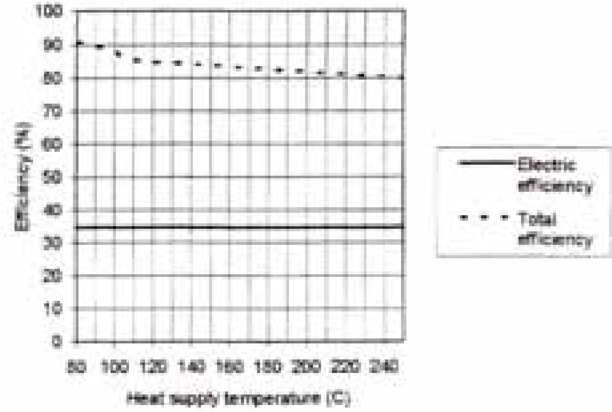

Figure 3.14 Efficiency (LHV) for a generalized gas turbine CHP at different heat supply temperatures

3.3.3 Emissions

The main environmental concern regarding gas turbines is the nitrogen oxides (NO_x) emission. Gas turbine plants can reach NO_x emissions below 50 mg/MJ without any external flue gas cleaning. Low NO_x emissions were previously achieved by injecting steam or water into the combustion chamber, which decreases the efficiency and increases the operating cost. Most manufacturers of larger (> 20 MW_e) gas turbines can now meet emission limits with dry low-NO_x combustors.

Carbon dioxide emissions, also a concern for fuel combustion facilities, are related directly to the amount of fuel burned. Natural gas combustion results in CO_2 emissions of about 56 g/MJ of gas burned, although this can vary somewhat depending on the chemical properties of the natural gas.

Emissions can vary based on the particular gas turbine equipment, fuels used and flue gas cleaning equipment. Actual emissions for a facility can only be determined based on facility-specific factors and are strongly affected by regulatory requirements in effect in a particular country. **Table 3.1** summarizes generalized emissions from gas turbine CHP, calculated per unit of fuel and per unit of electricity. The calculations assume dry/low NO_x combustion or steam/water injection but no selective catalytic reduction (SCR).

	Low	High
Per unit of fuel (mg/MJ)		
NO_x	25	75
SO_2	0.3	0.3
CO	15	50
HC	0	0
Particulates	1	1
CO_2	56,000	56,000
Per unit of electricity (g/kWh$_e$)		
NO_x	0.25	0.75
SO_2	0.00	0.00
CO	0.15	0.50
HC	0.00	0.00
Particulates	0.01	0.02
CO_2	570	570

Assumes dry/low NOx combustion or steam or water injection, but without SCR.

Table 3.1 Emissions from gas turbine CHP [3-3]

3.3.4 Economics

Capital Cost

Equipment costs for gas turbines are illustrated in **Figure 3.15** according to size. Gas turbine costs are extremely sensitive to size, and range from over \$600/kW$_e$ for very small turbines (1-2 MW$_e$) to \$300-500/kW$_e$ for mid-sized turbines (5-25 MW$_e$) to \$200/kW$_e$ for large turbines (over 100 MW$_e$). Total gas turbine CHP installed costs, including installation, heat recovery boiler and building, are illustrated in **Figure 3.16**.

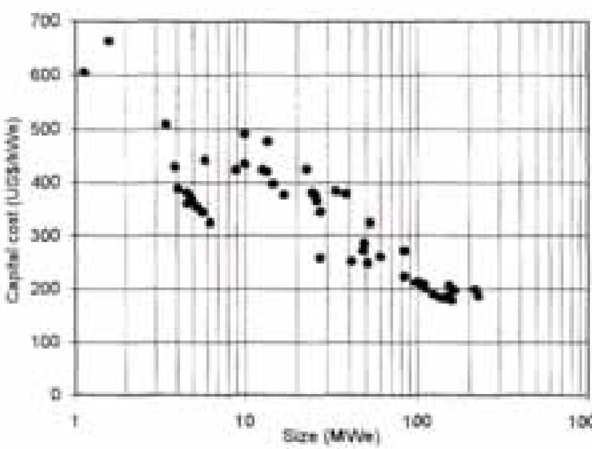

Figure 3.15 Capital costs for gas turbine only, not including installation [3-1]

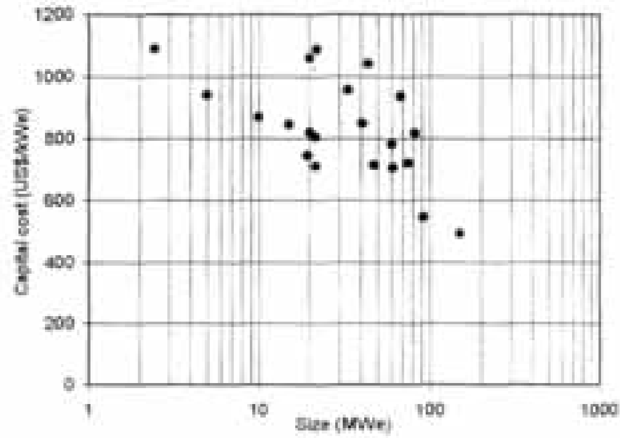

Figure 3.16 Capital costs for gas turbine CHP with heat recovery boiler, installation and auxiliary equipment [3-1, 3-4, 3-5, 3-6, 3-7, 3-8, 3-9]

Operation and Maintenance Cost

Gas turbine operation and maintenance (O&M) costs include: 1) monthly maintenance which can be accomplished without equipment shutdown; 2) periodic maintenance (approximately every 4,000 hours of operation) including borescope inspection for blade erosion and checkout of fuel systems, sensors and controls, burner cleaning; and 3) major overhaul at intervals of 30,000 to 40,000 hours. Different estimates of gas turbine O&M show a cost of about 0.25 cents/kWh$_e$ [3-9, 3-10]. For a CHP plant about 0.05 cents/kWh$_e$ should be added for the heat recovery boiler, based on an estimated O&M cost equal to 2% of the capital cost.

As noted in **Section 3.2**, labor cost estimates are based on the assumption that CHP system labor represents an incremental addition to already-existing base staffing for a district energy plant. Additional staffing assumed for these calculations is: one day-time staff person for a smaller gas turbine and one person per shift (total of 4) for a larger gas turbine (25 MW$_e$).

Overall Economics

Figures 3.17 and 3.18 illustrate the overall economics of gas turbine CHP. The total cost per kWh of electricity is calculated as a function of the value of recovered thermal energy, based on fuel costs of 0.0, 1.0 and 2.0 cents/kWh and equivalent full load hours (EFLH) of 2500, 5000 and 8000 hours per year. EFLH expresses the facility utilization as a ratio of the total annual production divided by facility output at rated capacity. Other key assumptions are summarized in **Table 3.2**.

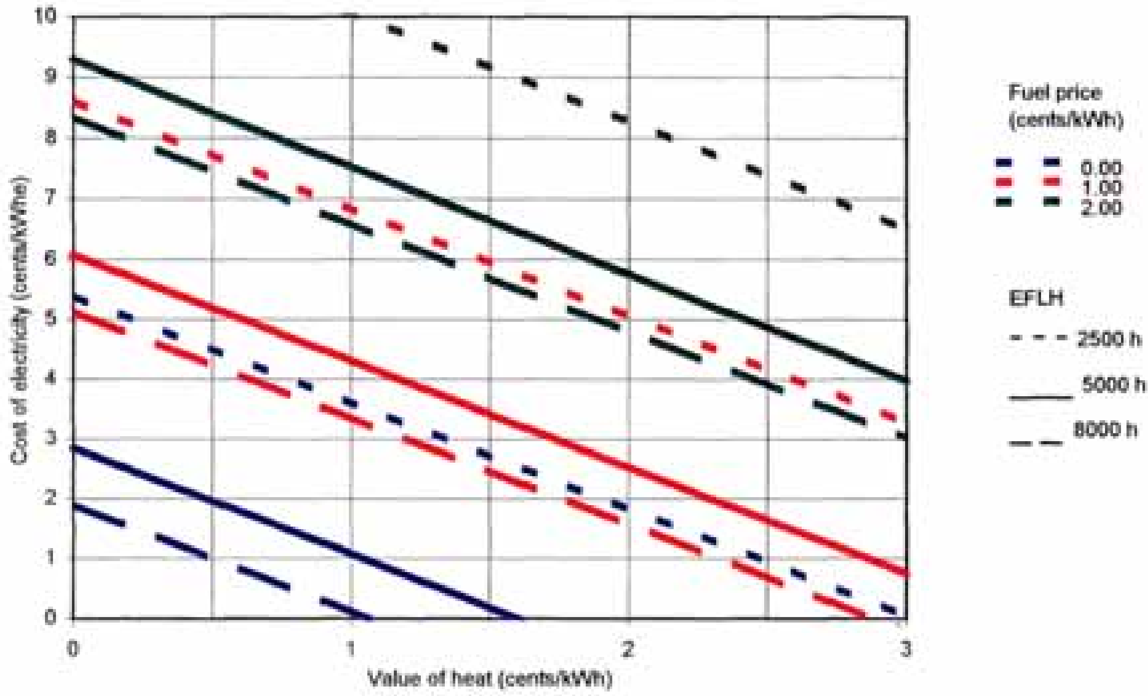

Figure 3.17 Overall economics for a gas turbine with heat recovery boiler (size 5 MW_e)

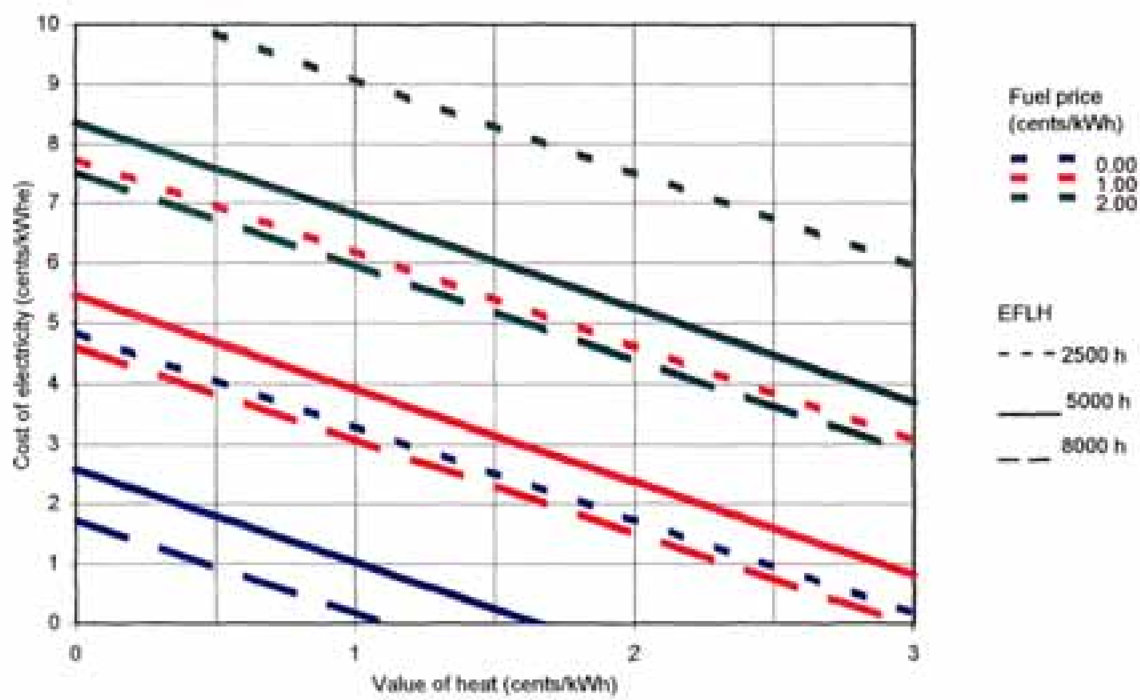

Figure 3.18 Overall economics for a gas turbine with heat recovery boiler (size 25 MW_e)

		5 MWe	25 MWe
A	Size (MW_e)	5	25
B	Capital cost ($/kW$_e$)	1000	900
C	Real interest rate (%)	8	8
D	Capitalization period (years)	15	15
E	Capital recovery factor	0.1168	0.1168
F	O & M costs (cents/kWh$_e$)	0.30	0.30
G	Labor ($/year at $50,000/person)	50,000	200,000
H	Fuel price (cents/kWh fuel)	0 to 2	0 to 2
J	Equivalent Full Load Hours	2500-8000	2500-8000
K	Electric efficiency	31%	35%
L	Thermal efficiency	55%	53%
M	Overall efficiency	86%	88%
N	Value of heat (cents/kWh$_{th}$)	0 to 3	0 to 3

Formulas: Electric price (cents/kWh) =

$\{(B \times E \times 100) + [(G \times 100/A)/1000] + [F \times J] + [H \times J/K] - [(N \times J)/(K/L)]\} / J$

Capital recovery factor (E) = $[C \times (1 + C)^D] / [(1 + C)^D - 1]$

Table 3.2 Assumptions for cost calculations in Figures 3.17 and 3.18

3.4 RECIPROCATING ENGINES

3.4.1 Description of Technology

Two types of reciprocating engines are common for CHP usage: otto engines and diesel engines. These types are primarily defined by the method of ignition, not by the fuels used. Combustion in the otto engine is initiated through spark ignition and the engine works with a moderate pressure to avoid self-ignition and knocking. In the diesel engine the compression is high enough that the fuel sprayed into the engine self -ignites. The fuel in either type of engine may be liquid or gaseous.

The diesel engine is dominant in sizes above 1-2 MW_e. Both the diesel engine and the otto engine can be found in a number of different applications and designs, including 4 and 2 stroke, with 1 to 20 cylinders. Turbochargers are common on both otto engines and diesel engines to increase the efficiency and power output. Diesel engines are available in sizes up to 50 MW_e. Otto engines are usually limited to below 2 MW_e, although some manufacturers are developing larger (5-10 MW_e) otto engines because it is increasingly difficult to meet nitrogen oxide emission limits with diesel engines without expensive catalytic converters. These engines are sometimes called "spark-ignited diesel engines" or "gas engines."

Reciprocating engine CHP is illustrated in **Figure 3.19** and can be briefly described as follows:

- A generator attached to the engine shaft generates electricity.
- Heat is recovered when the hot exhaust gas, with a temperature around 350-450°C, is cooled in a heat recovery boiler.
- Heat can also be recovered from the engine cooling water (at about 90°C) and oil lubrication system (at about 50°C).
- In addition, heat can be recovered from the turbocharger and intercooler. Intercooler recovery temperatures can vary, but 50°C is representative for one-stage intercoolers and 50°C and 90°C are representative for two-stage intercoolers.

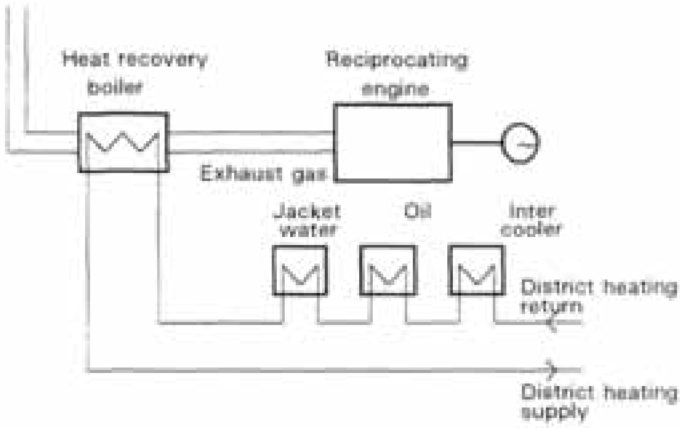

Figure 3.19 Schematic for reciprocating engine CHP

Multiple-stage intercoolers as well as exhaust gas turbines producing additional electricity can be used for larger engines if economical. A multi-stage intercooler provides the possibility of making some of the heat rejected from the cooling of compressed air available at a higher and more usable temperature. An exhaust gas turbine converts some of the high temperature "waste" heat to electricity. Many variations are possible for the design of specific equipment for CHP, depending on site-specific conditions.

Both gaseous and liquid fuels can be used in reciprocating engines. However, fuel ignition in diesel engines presents a challenge when using natural gas (with an ignition temperature of about 650°C as opposed to about 250°C for fuel oil). Conversion of reciprocating engines to use gaseous fuels is achieved in two ways:

1. *Injection of oil as a "pilot fuel," using about 5% oil at full load and up to about 10% at part loads.* This can be achieved by mixing air with gas fuel outside the engine. However, in modern larger diesel engines converted to gas combustion the gas fuel is compressed in an external compressor up to a pressure of about 250 bar. The compressed gas is then injected into the engine, where

air already has been compressed, just before the ignition point. With this method, the power output is usually not affected by conversion to gaseous fuels, and the engine can be switched between gaseous and liquid fuels.

2. *Conversion to spark ignition (otto engine) in combination with "lean burn" (high air/fuel ratio) designs.* This is generally the approach taken with smaller (under 6 MW_e) engines, although R&D is continuing to increase the size of engines employing this approach due to its environmental benefits. One disadvantage is the lack of ability to switch fuels. This modified engine has a higher compression ratio than a normal otto engine but low enough not to self-ignite. The electric efficiency of this modified engine is higher than a conventional otto engine.

Since the beginning of 1970s, intensive diesel engine R&D has been performed, especially regarding diesel engines for ships due to rapidly increasing oil prices during that time. During the 1970s and 1980s the efficiency was increased from 40% to over 50% for the most efficient two-stroke engines. Substantial increases in efficiency are not expected in the near future. Instead, R&D is concentrated on reducing emissions and maintenance requirements and, to a lesser extent, use of alternative fuels.

3.4.2 Performance

Electric and Thermal Efficiency

Electric efficiencies for diesel engines are usually in the range of 40-45% (LHV). Efficiencies over 50% can be achieved with slow-speed two-stroke engines. However, these engines are larger in size (about 15 MW_e and above), are expensive and have higher emissions relative to gas turbines, with which they will be competing in this size range. The higher efficiency slow-speed two-stroke engines are not addressed in this report because gas turbines (simple cycle or combined cycle) are usually a better choice from the standpoints of both economy and emissions.

For a CHP plant the ratio of electric output to thermal output (sometimes called "alpha value") will be slightly above 1.0, and the total efficiency will be about 80%, assuming recovery of thermal energy at supply/return temperatures of 100/75°C (see **Figure 3.20**) The diesel engine has a good part-load performance, as shown in **Figure 3.21** for a representative diesel engine.

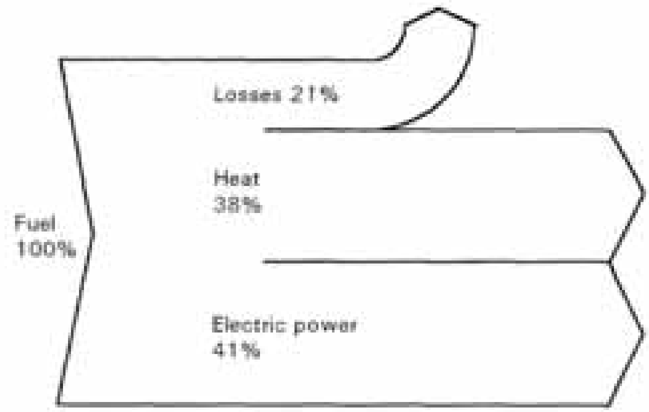

Figure 3.20 Sankey diagram (LHV) for CHP with diesel engine (4 stroke, size range 5-15 MW_e)

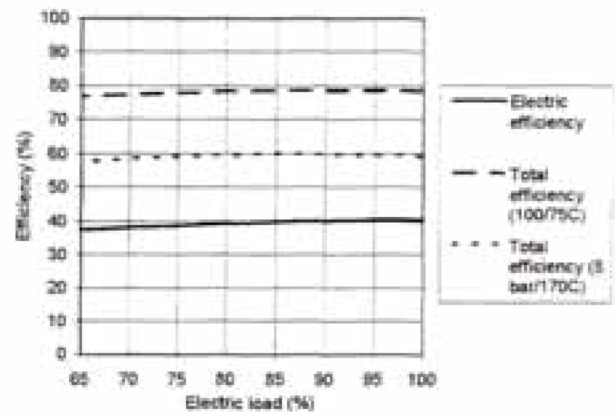

Figure 3.21 Part load performance for a diesel engine (size range 5-15 MW_e)

Table 3.3 summarizes the outputs from a diesel engine expressed as a percentage of input fuel, assuming recovery of 100/75°C heat. If thermal energy down to 50°C can be used, the total efficiency at 100% load can be increased to 90% from about 80% (with the ratio of electric and thermal outputs decreasing from about 1.1 to about 0.8) by also recovering additional heat from the lubricating oil and low temperature air charge.

	Total Engine Output	Recoverable Energy (100/75°C)	Recoverable Energy (170°C)
Electricity	40.6	40.6	40.6
Thermal			
Exhaust gas	29.6	23.8	18.8
Jacket water	10.8	10.8	0.0
Charge air	8.3	3.3	0.0
Lubricating oil	5.5	0.0	0.0
Radiation	5.3	0.0	0.0
Total	100.0	78.5	59.4

Table 3.3 Heat and electric output from a diesel engine (percentage of fuel input LHV) assuming 75°C hot water heat recovery temperature and 170°C (8 bar) steam heat recovery 3-11

The electric efficiency is unchanged regardless of heat supply temperature as long as the intercooler or jacket water temperatures are not raised to accommodate higher heat supply temperatures. Heat recovery up to 100°C heat supply temperature is based on hot water with a 25°C temperature increase from the engine. Above 120°C, saturated steam with a condensate return temperature of 100°C is assumed. The total efficiency decreases with increasing heat supply temperatures. However, it is important to note that if there is a use for lower temperature hot water, an additional hot water heat recovery boiler can be installed to raise the total efficiency up to the same level as for hot water heat recovery only.

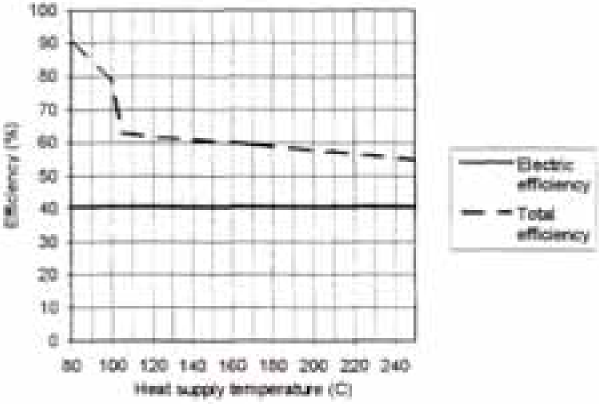

Figure 3.22 Efficiency for a diesel engine at different heat supply temperatures

For otto engines the electric efficiency ranges from 30-40%, with 35% as a representative value for engines up to 2 MW_e, as shown in **Figure 3.23**. A total efficiency of around 85%, with an electric/thermal output ratio in the range of 0.55-0.90, can be reached for a CHP otto engine assuming 100/75°C thermal energy recovery. For larger otto engines or lean-burn gas engines the performance is similar to the performance for a diesel engine. While the gross electric efficiency can be higher for the diesel engine, this can be offset by the electric consumption for compressing gas to the required high pressure in situations where a low pressure gas pipeline supplies the fuel. Part-load performance of the otto engine is comparable to the diesel engine (see **Figure 3.24**).

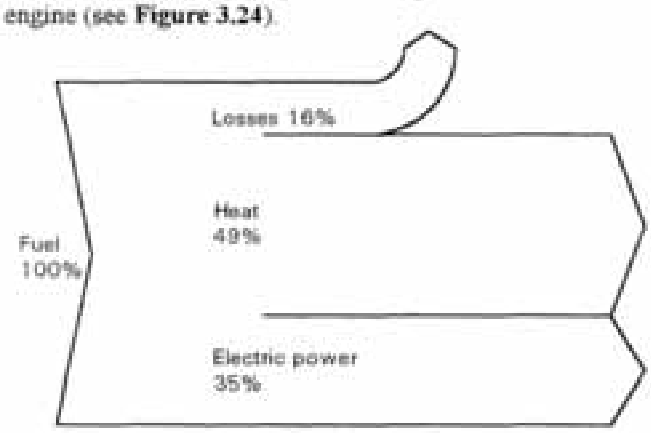

Figure 3.23 Sankey diagram (LHV) for CHP with otto engine (size range 1-2 MW_e)

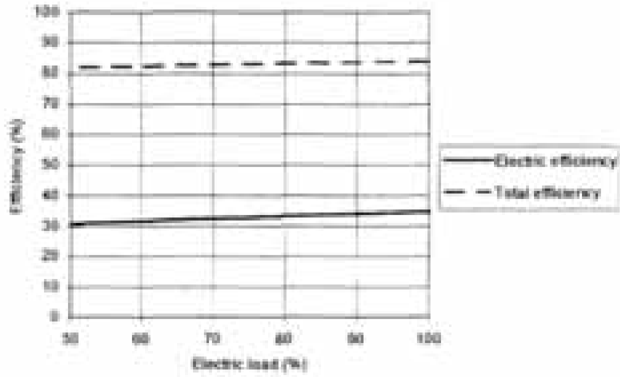

Figure 3.24 Part load performance for an otto engine (size range 1-2 MW_e) [3-11]

3.4.3 Emissions

Emissions can vary based on the particular engine, fuels used and flue gas cleaning equipment. Actual emissions for a facility can only be determined based on facility-specific factors and are strongly affected by regulatory requirements in effect in a particular country.

NO_x emissions from reciprocating engines are relatively high compared to other energy conversion equipment. For a diesel engine the NO_x emissions are around 1000-1500 mg NO_x per MJ fuel without cleaning equipment. Selective catalytic reduction (SCR) is usually used, with a possible emission reduction around 90-95%. SCR is normally not used for otto engines. Instead, two other methods can be used: 1) three-way catalytic converters (non-selective catalytic reduction) with lambda control; and 2) lean-burn

Figure 3.25 shows emissions of NO_x, CO and HC as a function of the lambda-value or excess air ratio, i.e., the ratio of the actual air/fuel mixture compared to the stoichiometric air/fuel mixture (the amount which is theoretically required to combust the fuel). Standard otto engines with three-way catalytic converters are typical for smaller engines and are operated at a lambda value of 1.0. Without the catalytic converter the emissions are high at that air-to-fuel ratio. With the catalytic converter the emissions are lower compared to a lean-burn engine as long as the converter functions.

Lean burn is the typical approach in most engines. A lambda value of about 1.6 is used in lean-burn engines, resulting in low NO_x emissions. To further reduce the HC and CO emissions an oxidation catalytic converter can be used.

Diesel engines operate at lambda values of about 2.5. Extrapolation from **Figure 3.25** would imply that NO_x emissions would be low. However, due to the high combustion temperature in the engine, diesel engine NO_x emissions are high unless SCR is used.

Emissions for basic types of reciprocating engine CHP burning natural gas are summarized in **Table 3.4**. Emissions of sulfur dioxide (SO_2) are essentially zero when burning natural gas.

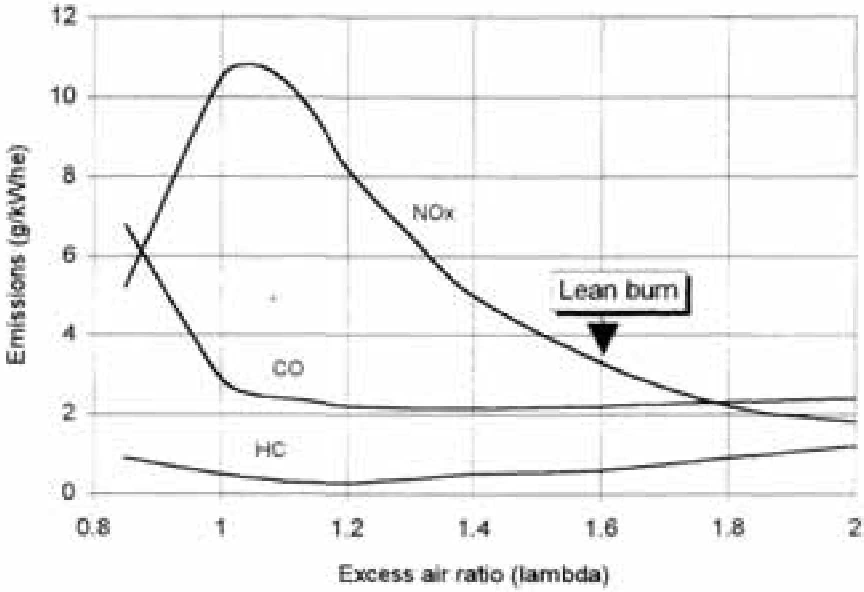

Figure 3.25 Emissions from otto engines at different fuel/air ratios

	Standard Otto [1]		Lean Burn [2]		Diesel [3]	
	Low	High	Low	High	Low	High
Per unit of fuel (mg/MJ)						
NO_x	25	75	100	200	50	
SO_2	0.3	0.3	0.3	0.3	0.3	0.3
CO	200	300	200	350	50	150
HC	200	300	100	200	50	150
Particulates	10	20	10	20	10	20
CO_2	56,000	56,000	56,000	56,000	56,000	56,000
Per unit of electricity (g/kWh_e)						
NO_x	0.30	0.85	1.05	2.05	0.45	
SO_2	0.00	0.00	0.00	0.00	0.00	0.00
CO	2.25	3.40	2.05	3.60	0.45	1.35
HC	2.25	3.40	1.05	2.05	0.45	1.35
Particulates	0.10	0.25	0.10	0.20	0.10	0.20
CO_2	625	625	575	575	495	495

[1] With three-way catalytic converter.

[2] CO and HC emissions in lean burn engines can be reduced with oxidation converter.

[3] With selective catalytic reduction (SCR).

Table 3.4. Emissions from reciprocating engine CHP burning natural gas [3-11]

3.4.4 Economics

Capital Costs

Capital costs for CHP plants based on reciprocating engines are shown in **Figure 3.26**. The values are based on a variety of sources and represent the total investment for equipment and installation, including construction of building space. The three highest investments are based on two-stroke diesel engines with normally higher electric efficiency and electric/thermal output ratios.

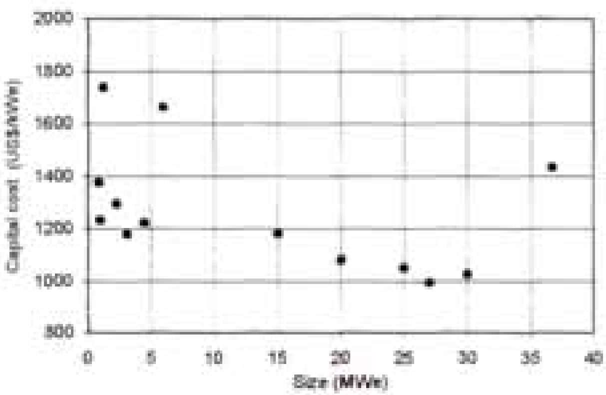

Figure 3.26 Installed capital costs for reciprocating engines with heat recovery [3-5, 3-6, 3-7]

Operation and Maintenance Cost

The operation and maintenance cost for reciprocating engines includes oil consumption (about 1 g/kWh$_e$), oil changes, replacement of components such as filters, gaskets and spark plugs, and major overhauls at an interval of approximately 50,000 hours. For small otto engines, below 1 MW$_e$, the operation and maintenance cost is in the range of 1.0-2.0 cents/kWh$_e$, and for larger otto and diesel engines 0.5-1.0 cent/kWh$_e$. With SCR, 0.25-0.5 cent/kWh$_e$ should be added. [3-7, 3-11, 3-12, 3-13]

The smaller otto engine is assumed to be operated mainly through remote monitoring and control, with additional operational staffing of 50% of a full time equivalent. For a larger diesel engine, with an increased emphasis on reliable electric output likely, one person per shift is assumed for additional staffing.

Overall Economics

Figures 3.27 and 3.28 illustrate the overall economics of otto engines and diesel engines, respectively. The total cost per kWh of electricity is calculated as a function of the value of recovered thermal energy, based on fuel costs of 0.0, 1.0 and 2.0 cents/kWh and equivalent full load hours (EFLH) of 2500, 5000 and 8000 hours per year. Other key assumptions, and the formula for the economic calculation, are described in **Table 3.5**. As would be expected, the larger and more efficient diesel engine produces more favorable economics and is less sensitive to changes in fuel price and utilization.

		Otto	Diesel
A	Size (MW$_e$)	2	20
B	Capital cost ($/kW$_e$)	1300	1100
C	Real interest rate (%)	8	8
D	Capitalization period (years)	15	15
E	Capital recovery factor	0.1168	0.1168
F	O & M costs (cents/kWh$_e$)	1.5	1.0
G	Labor ($/year at $50,000/person)	25,000	200,000
H	Fuel price (cents/kWh fuel)	0 to 2	0 to 2
J	Equivalent Full Load Hours	2500-8000	2500-8000
K	Electric efficiency	35%	41%
L	Thermal efficiency	49%	38%
M	Overall efficiency	84%	79%
N	Value of heat (cents/kWh$_{th}$)	0 to 3	0 to 3

Formulas: Electric price (cents/kWh) =

$$\{(B \times E \times 100) + [(G \times 100/A)/1000] + [F \times J] + [H \times J/K] - [(N \times J)/(K/L)]\} / J$$

Capital recovery factor (E) = $[C \times (1 + C)^D] / [(1 + C)^D - 1]$

Table 3.5 Assumptions for cost calculations in Figures 3.27 and 3.28

3.5 STEAM TURBINES

3.5.1 Description of Technology

Independent steam turbine power plants (i.e., steam turbines which are not just a component of a larger plant) are available in sizes ranging from 5 MW$_e$ to over 1000 MW$_e$, and are the most common type of power plant in use worldwide. (As a component in a larger plant, steam turbines are available in sizes of under 1 MW$_e$.) One of the strengths of this technology is the ability to use a wide variety of fuels, including solid fuels and waste materials. The basic elements of steam turbine CHP are illustrated in **Figure 3.29**, and can be briefly described as follows:

- Fuel and air are combusted in a boiler, generating steam. To increase the efficiency of the steam turbine cycle the steam is normally superheated.
- The steam exits the boiler and is directed to the steam turbine, where the steam expands through the turbine, turning the turbine blades which are connected to the electric generator shaft.
- In a backpressure turbine, the steam is exhausted to a heat exchanger where thermal energy is transferred at a relatively low pressure to the district heating loop or steam-driven chiller.
- If higher pressure steam is required, some steam is extracted through ports in the turbine prior to full expansion at the turbine exhaust.

Figure 3.43 illustrates the total efficiency of a variety of combined cycle gas turbine configurations. These calculations are based on the following assumptions:

- 35% electric efficiency at ISO base load conditions
- 530°C exhaust temperature at base load conditions
- 97% generator efficiency
- 97.5% gearbox efficiency
- 2.5% heat/gas losses and boiler blowdown
- 15 mbar backpressure from heat recovery boiler
- Representative values from previous figures for part load and outdoor temperature characteristics
- The back pressure from the heat recovery boiler will decrease the performance of the gas turbine, with electric efficiency reduced by 0.7% for every 10 mbar back pressure.

In condensing mode, the total efficiency is 50%, with 35% contributed by the gas turbine and 15% by the steam turbine. The electric efficiency for a CHP plant decreases from 45% to under 40% as the steam extraction temperature increases from 100/75°C hot water to 8 bar steam, as shown in **Figure 3.43**.

The overall efficiency of all of the CHP configurations are 87% if 100/75°C hot water from a hot water district heating economizer can be used. With the temperature in the economizer varying with the overall temperature for the different applications, the overall efficiency will decrease to about 82% when 8 bar steam is the only form of thermal energy recovered, as shown in **Figure 3.44**. Heat recovery can be increased if hot water with temperatures below 100/75°C can be used. However, recovery of additional heat in the economizer will increase the gas turbine back pressure and decrease the electric output. The flue gas temperature will also be lowered to a point where condensation of sulfuric acid is a potential concern. (Sulfur is usually added to natural gas in very small concentrations for leak detection purposes.)

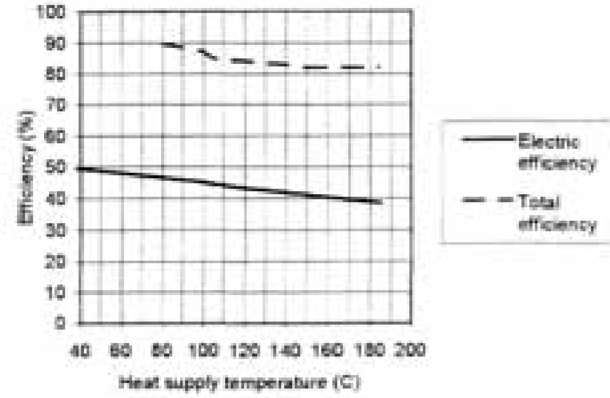

Figure 3.44 Gas turbine combined cycle CHP efficiency (LHV) at different heat supply temperatures (two-stage hot water heating)

As shown in **Table 3.9**, the electricity lost due to heat extraction is similar to that which occurs in a steam turbine CHP plant, and varies from 0.25 kW_e per kW_{th} with extraction of 8 bar steam down to 0.13 kW_e per kW_{th} with extraction for 100/75°C hot water, assuming no economizer. If an economizer can be used, slightly lower values for lost electricity will result.

Heat extraction pressure and temperature	Electric efficiency	Reduced electric output kWe/kWth	
	%	Steam extraction only	Including economizer
8 bar/170C CHP CC	39.7	0.24	0.24
2 bar/120C CHP CC	43.3	0.17	0.16
100/75C CHP CC	45.1	0.13	0.12

Table 3.9 Impact of extraction temperatures on combined cycle electric efficiencies and outputs

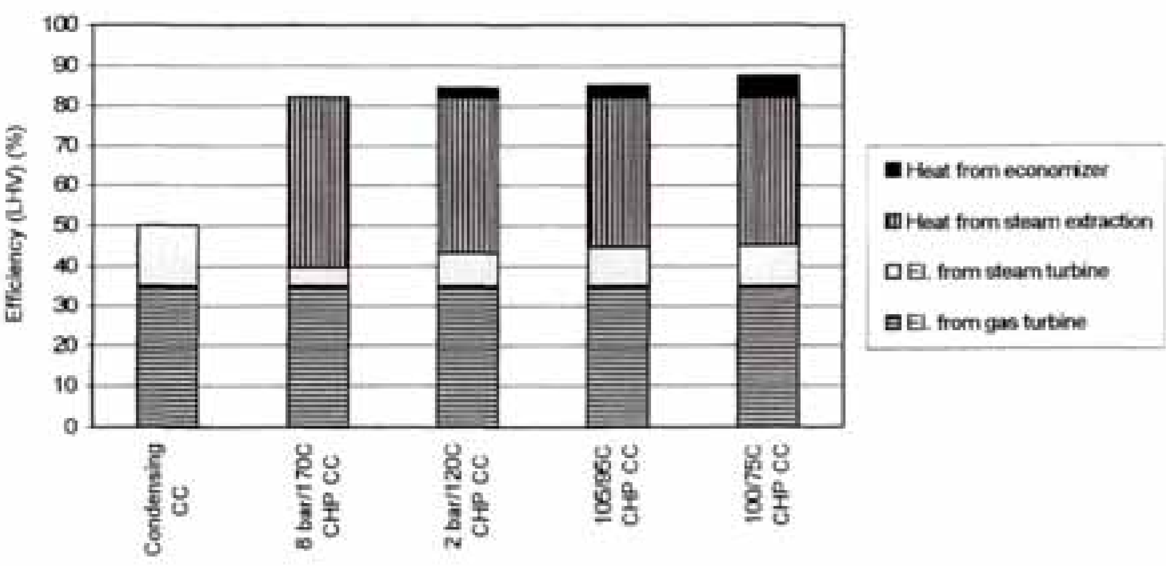

Figure 3.43 Gas turbine combined cycle efficiency at different extraction temperatures

3.6.1.3 Emissions

Generalized gas turbine combined cycle emissions are shown in **Table 3.10**. Emissions can vary based on the particular gas turbine equipment, fuels used and flue gas cleaning equipment. Actual emissions for a facility can only be determined based on facility-specific factors and are strongly affected by regulatory requirements in effect in a particular country.

The emissions per unit of fuel input are comparable for a gas turbine simple cycle and a gas turbine combined cycle. However, the combined cycle will have lower emissions per unit of electricity due to the higher electric efficiency.

	Low	High
Per unit of fuel (mg/MJ)		
NO_x	25	75
SO_2	0.3	0.3
CO	12.5	50
HC	0	0
Particulates	1	1
CO_2	56,000	56,000
Per unit of electricity (g/kWh$_e$)		
NO_x	0.20	0.55
SO_2	0.00	0.00
CO	0.10	0.35
HC	0.00	0.00
Particulates	0.01	0.01
CO_2	400	400

Assumes either: dry/low NOx combustion; or steam or wat injection but without SCR

Table 3.10 Representative emissions from gas turbine combined cycle CHP [3-4]

3.6.1.4 Economics

Capital Cost

Figure 3.45 illustrates the installed capital cost for gas turbine combined cycle CHP and condensing plants. Costs for condensing combined cycle plants would be approximately 10% lower, based on the power output derate which occurs when shifting from condensing to extraction of 100 °C hot water. However, as with steam turbine plants, there is no significant difference in capital cost shown in the figure between the condensing plants and the CHP plants, which could be explained by the relatively large influence of site-specific conditions.

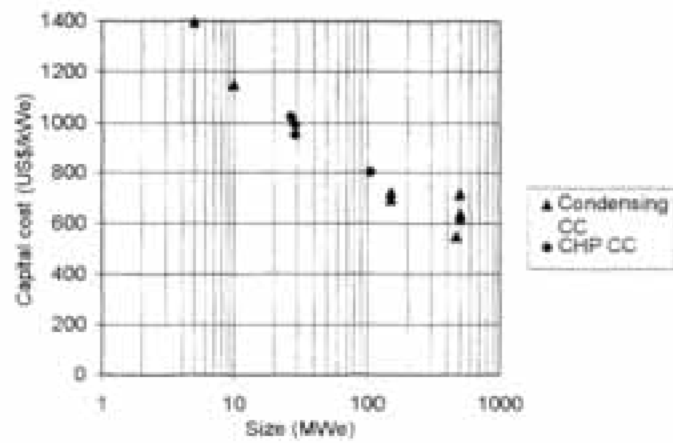

Figure 3.45 Installed capital cost for gas turbine combined cycle CHP plants [3-4, 3-5, 3-6, 3-7, 3-9, 3-10]

Operation and Maintenance Cost

Estimates of O&M costs for condensing combined cycle CHP indicate a total per kWh$_e$ of about twice that of a gas turbine simple cycle, or 0.55 cents/kWh$_e$. [3-4, 3-9, 3-10]

Staffing costs were based on the following assumptions for additional staff for the two different sizes of CHP plants.

	25 MW$_e$	50 MW$_e$
Control room	1 per shift (4 total)	2 per shift (8 total)
Misc.	3 day shift (3 total)	3 day shift (3 total)
Total workers	7	11

Overall Economics

Figures 3.46 and 3.47 illustrate the overall economics of combined cycle gas turbine CHP. The total cost per kWh of electricity is calculated as a function of the value of recovered thermal energy, based on fuel costs of 0.0, 1.0 and 2.0 cents/kWh and equivalent full load hours (EFLH) of 2500, 5000 and 8000 hours per year. Key assumptions for the calculations are shown in **Table 3.11**.

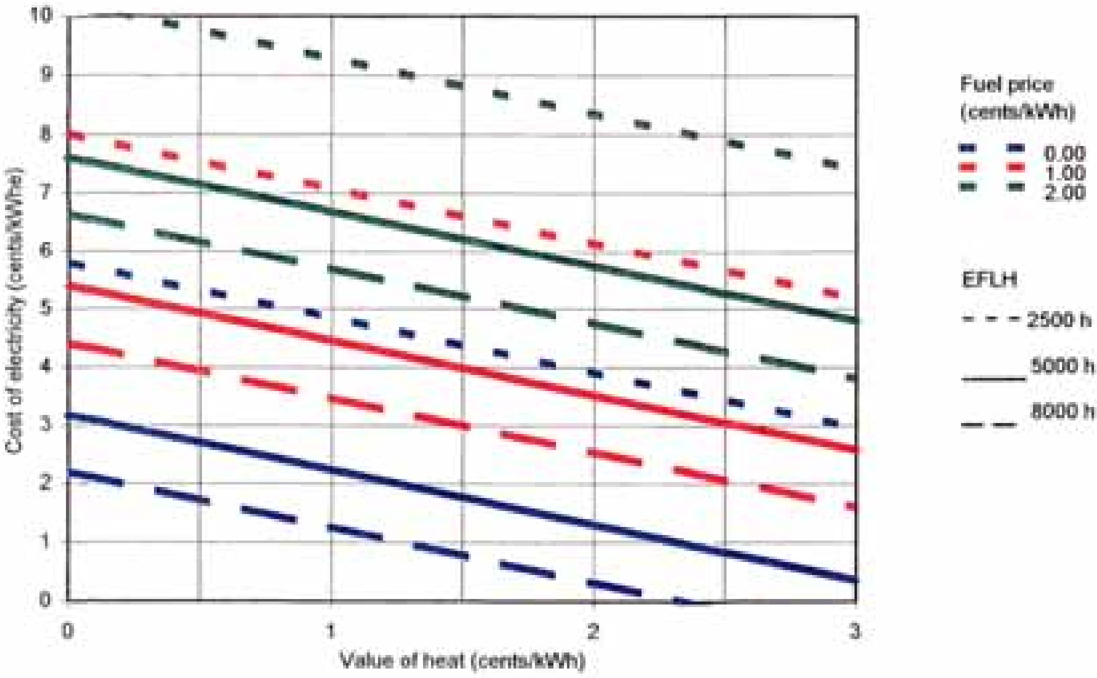

Figure 3.46 Overall economics for gas turbine combined cycle CHP (size range 25 MW_e)

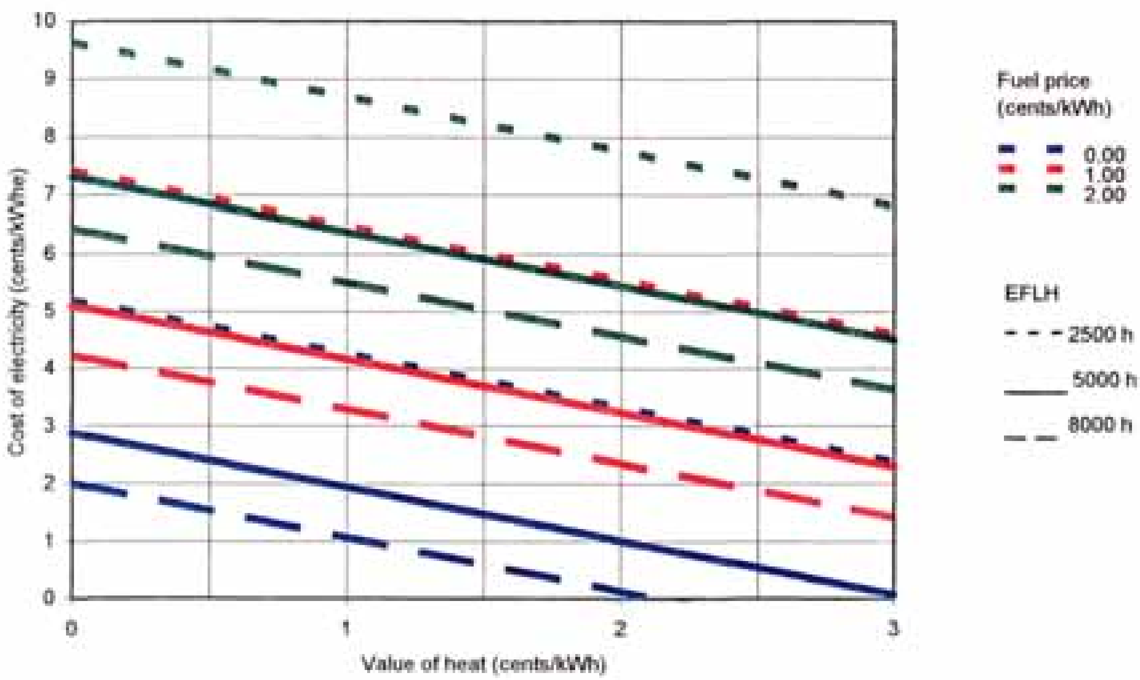

Figure 3.47 Overall economics for gas turbine combined cycle CHP (size range 50 MW_e)

		25 MWe	50 MWe
A	Size (MW_e)	25	50
B	Capital cost ($/$kW_e$)	1000	900
C	Real interest rate (%)	8	8
D	Capitalization period (years)	15	15
E	Capital recovery factor	0.1168	0.1168
F	O & M costs (cents/kWh_e)	0.55	0.55
G	Labor ($/year at $50,000/person)	350,000	550,000
H	Fuel price (cents/kWh fuel)	0 to 2	0 to 2
J	Equivalent Full Load Hours	2500-8000	2500-8000
K	Electric efficiency	45%	45%
L	Thermal efficiency	42%	42%
M	Overall efficiency	87%	87%
N	Value of heat (cents/kWh_{th})	0 to 3	0 to 3

Formulas: Electric price (cents/kWh) =

((B x E x 100) + [(G x 100/A)/1000] + [F x J] + [H x J/K] - [(N x J)/(K/L)]) / J

Capital recovery factor (E) = $[C \times (1 + C)^D] / [(1 + C)^D - 1]$

Table 3.11 Assumptions for cost calculations in Figures 3.46 and 3.47

3.6.2 Reciprocating Engine Combined Cycle

It is possible, although unusual, to generate additional electricity in a steam turbine with steam produced from reciprocating engine exhaust gas. Combined cycles are much more common with gas turbines because:

- Gas turbines are available in larger sizes than reciprocating engines and can therefore more frequently support the economies of scale needed for the addition of a steam cycle.

- Less heat is available for steam generation in reciprocating engines compared to gas turbines. The energy in the exhaust gas is only about 30% of the energy input for a reciprocating engine while the exhaust gas energy for a gas turbine is about 65% of the energy input.

- The temperature of the exhaust gas is usually lower (about 400°C) compared to the exhaust temperature from a gas turbine of (about 500°C). With a lower exhaust gas temperature, the generated steam will have a lower pressure and temperature, thereby reducing the steam cycle efficiency.

Gas turbines have been extensively developed and marketed for combined cycle applications, whereas diesel engines have not. In addition, concerns about emissions, noise, maintenance and extra staffing requirements have limited the number of diesel engines installed and may thereby also have contributed to the lack of implementation of diesel combined cycles.

Because reciprocating engine combined cycles are uncommon, heat from a reciprocating engine is usually perceived as "waste heat," unusable for additional electric generation. Electricity lost by producing higher pressure steam for absorption chillers is thus often perceived as zero. However, as discussed below, there is an argument that reciprocating engine combined cycles can make sense from both an economic and efficiency standpoint. Reciprocating engine combined cycles may be increasingly used, particularly as fuel and electricity prices rise.

To the extent that recoverable heat is viewed as a potential resource for additional electric generation rather than simply "waste," some of the same thermodynamic and economic issues discussed relative to gas turbine combined cycles can also be raised, such as whether it is more appropriate to generate chilled water with absorption as opposed to using recovered heat to generate steam for combined cycle operation. Reciprocating engine combined cycles are suggested by various engine manufacturers, and installations have been successfully implemented.[3-19]

3.6.2.1 Description of technology

Similar to the gas turbine combined cycle, a steam cycle with a steam turbine, condenser and auxiliary equipment is added to the simple cycle reciprocating engine (see **Figure 3.48**). Depending on the temperature level in the heating system, heat can be recovered from the intercooler, oil cooler, jacket water, steam condenser and an exhaust gas economizer. In what order the jacket water, condenser and economizer should be connected to the system depends on the price of electricity compared to the price of heat.

3.5.2.2 Performance

Electric efficiency in the representative simple cycle diesel engine depicted in **Figure 3.20** was shown as 41%, a rounding of 40.5%. For the base case CHP conditions (100/75°C), the diesel engine combined cycle increases the electric efficiency to 43%, providing an increase in electric output of about 6% compared to simple cycle electric output. The total efficiency for this base case condition is nearly 80%, as illustrated in **Figure 3.49**.

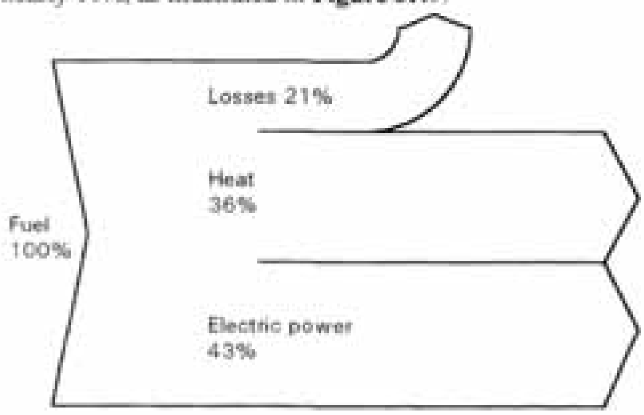

Figure 3.49 Sankey diagram (LHV) for a diesel engine combined cycle CHP (4 stroke, size range 5-15 MW_e)

With a diesel combined cycle in condensing mode, electric efficiency can reach 44.5%, a 10% increase in electric output compared to the simple cycle. (Assumptions for this calculation included a steam cycle with 11 bar 350°C inlet conditions, 75% steam turbine efficiency and a condensing temperature of 40°C. The 11 bar steam might result in additional staffing requirements in some countries.). This increase in output is illustrated in **Figure 3.50**. If 11 bar saturated steam is used rather than 350°C superheated steam, electric efficiency can be increased to 43.5% compared to simple cycle efficiency of 40.5%. Saturated steam equipment is less costly and easier to operate compared to superheated steam equipment and can therefore be a better option despite a 1% lower electric efficiency.

With lower-temperature heat (80/55°C) recovered for district heating/cooling, total efficiency can reach 90%. As the temperature recovered for district heating/cooling increases, total efficiency drops sharply to about 60% at heat supply temperatures of 120°C and above, due to the loss of the ability to recover heat from the jacket, intercooler and oil, and the added electric output resulting from combined cycle operation diminishes to nearly zero.

The added steam turbine electric production can be obtained for a relatively low additional capital cost compared to the cost per kW of simple cycle diesel capacity, as discussed below in **Section 3.6.2.4**.

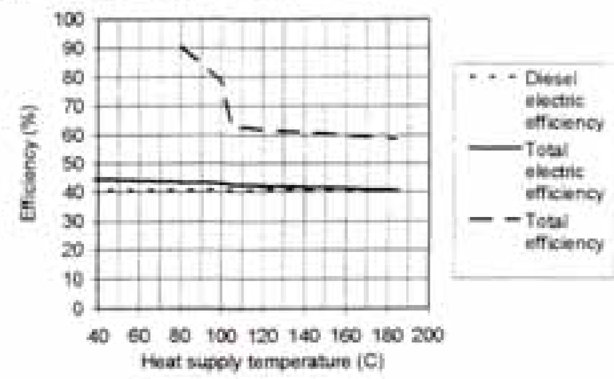

Figure 3.50 Representative diesel combined cycle efficiency at different heat supply temperatures

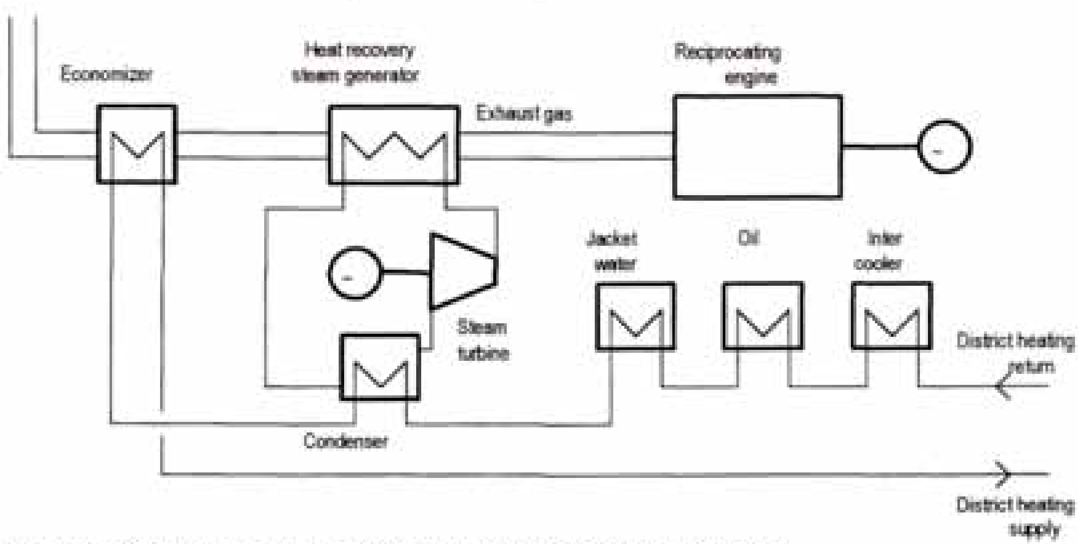

Figure 3.48 Schematic for a reciprocating engine combined cycle plant

Because the power from the steam turbine is relatively small compared to the total output, the part-load performance will be close to that of the simple cycle diesel engine. The exhaust gas from the diesel engine has a high oxygen level (12-13% O_2), so the gas could be used for additional firing to increase the steam output.

3.6.2.3 Emissions

Emissions for reciprocating engine combined cycles will be similar to simple cycle diesel engines, as presented in **Table 3.4** and **Section 3.4.3**. However, because of the higher electric efficiency (about 6% more electricity output), the specific emissions per unit of electric output will be correspondingly lower.

3.6.2.4 Economics

As shown previously in **Figure 3.26**, installed capital costs for reciprocating engines are about \$1000-1200/kW_e for larger engines and \$1200-1400/kW_e for smaller engines. In contrast, the total installed cost of a small steam turbine (including deaerator, etc.) is about \$1000/kW_e. Thus, it is potentially cost-effective, and more efficient, to purchase additional electric generation capacity by using a reciprocating engine for a combined cycle, as opposed to investing in additional conventional reciprocating engines.

However, for the reasons discussed above, applications of diesel combined cycles are rare. Economic analysis of reciprocating engines combined cycle CHP is not presented here due to the lack of applications data.

3.6.3 Solid Fuel Combined Cycle

3.6.3.1 Description of Technology

Pressurized Fluidized Bed Combustion (PFBC) and Integrated Gasification Combined Cycle (IGCC) technologies have been implemented to increase the efficiency of power production from solid fuels through a combined cycle.
The basic layout of a PFBC is close to a natural gas combined cycle (see **Figure 3.51**). The main difference is the combustor, which in a PFBC plant is substantially larger and is a fluidized bed boiler. The gas turbine provides compressed air to the boiler and, because of the pressurization of the boiler (12-16 bar), the size can be considerably smaller compared to what would be required for a normal solid fuel boiler.

Solid fuel is injected into the boiler and combustion takes place in the fluidized bed at a low temperature (about 850° C). The low combustion temperature reduces the formation of NO_x but is also essential to avoiding ash agglomeration. The bed is cooled by the steam that is distributed to the steam turbine. Limestone or dolomite is injected into the bed to capture sulfur during combustion. Particulates from the hot flue gas are cleaned with cyclones before entering the gas turbine. In addition to supplying the boiler with compressed air, the gas turbine also provides about 25% of the electric output.

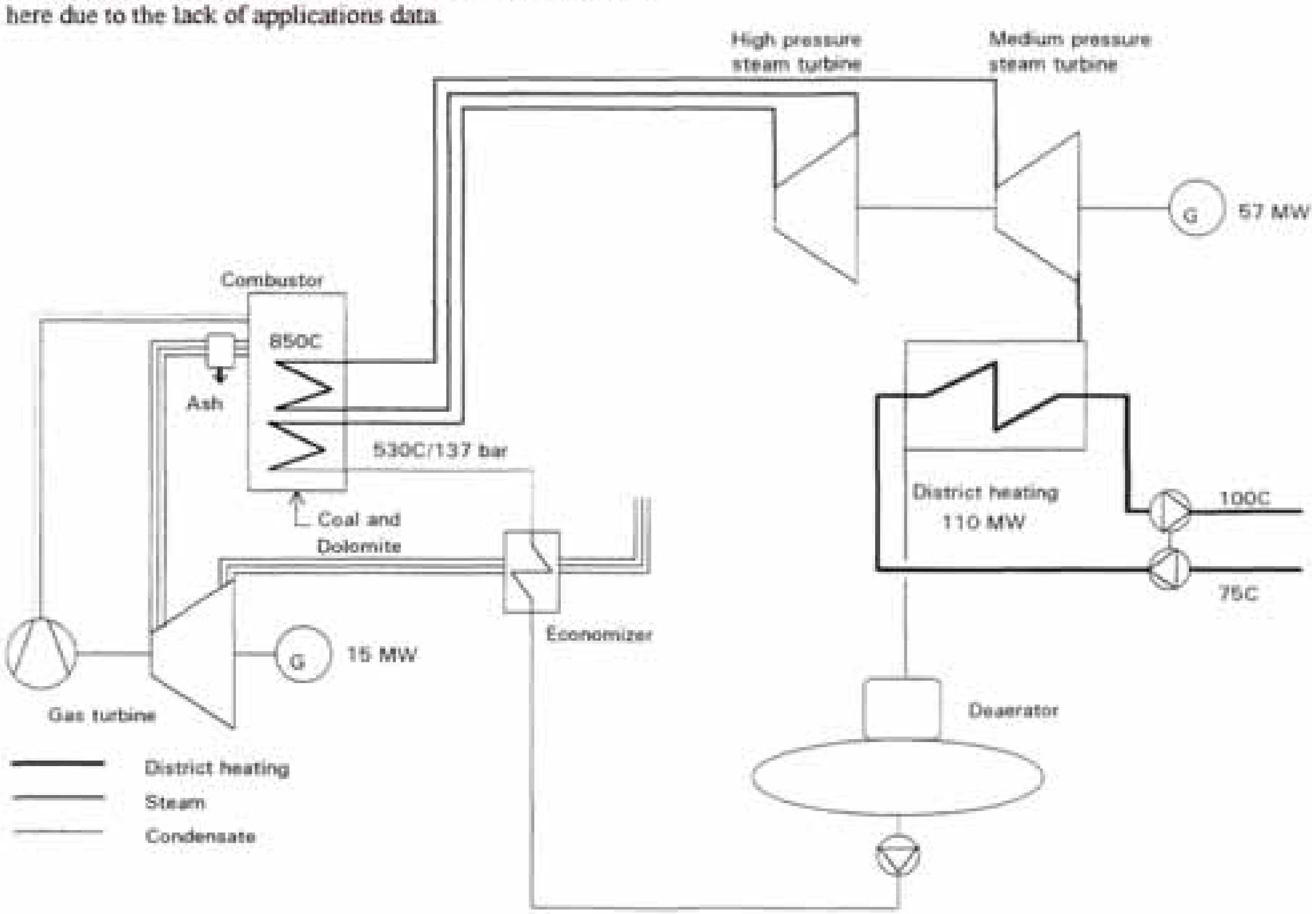

Figure 3.51 Schematic for a solid fuel-fired combined cycle CHP plant (PFBC)

Only 3-4 PFBC power plants have been built to date, and the technology is still under development. PFBC plants are competing with coal gasification for high technical and environmental performance utilizing coal. Because of the limited market, the price for a PFBC plant under commercial conditions is uncertain. However, the technical and environmental performance should make PFBC an important future option for use of coal for district energy.

3.6.3.2 Performance

Electric and Thermal Output

A higher electric efficiency as well as improved emissions can be reached with a PFBC compared to conventional solid fuel power plants. An electric efficiency of 44-46% LHV can be reached in condensing mode. In CHP mode, electric efficiency can reach 34%, with an overall efficiency of 89% LHV. [3-20]

3.6.3.3 Emissions

The environmental performance from existing PFBC plants is almost equivalent to gas-fired plants. The absorption of sulfur in the bed can reduce the SO_2 emissions by over 95%. NO_x emissions around 10 mg/MJ have been obtained with ammonia injection and a small catalytic aid in the flue gas duct. Measured levels of CO and N_2O are less than 200 parts per million by weight (ppm) and 10 ppm, respectively. With a textile baghouse, the particulate emissions are around 2 mg/MJ.

3.7 FUEL CELLS

A number of new technologies are under development for advanced CHP, including supercritical steam cycles, various technologies for gasification of coal and/or biomass, and fuel cells. Of these, the fuel cell is perhaps of greatest interest due to its environmental advantages. For this reason, fuel cells will be briefly addressed here, although not in the depth of the CHP technologies presented earlier in this chapter.

3.7.1 Description of technology

Fuel cells generate electricity and heat through an electrochemical conversion process which has long been applied in automobile batteries. Chemical energy is converted to electricity when hydrogen is combined with oxygen to make water. Hydrogen gas can be provided directly to the fuel cell. The hydrogen can be extracted from anything that contains hydrocarbons, including natural gas, biomass, landfill gas, methanol, ethanol, methane and coal-based gas.

Different types of fuel cells are named according to the type of medium used to combine the hydrogen and oxygen. Three types of fuel cells are usually considered for CHP applications:

- Phosphorous acid cells, now operating in various sites providing CHP.
- Molten carbonate systems, now in the demonstration phase for baseload power.
- Solid oxide cells, with a small-scale unit now in the demonstration phase.

Several other types of fuel cells are in use or being developed for various other applications:

- Alkaline -- used in space applications since the 1960s.
- Proton exchange membrane -- for transportation and small-power applications.

Full-scale field tests of molten carbonate fuel cells were initiated in 1994, and developers of this technology expect it to reach commercial deployment by the end of the 1990s. [3-21]

3.7.2 Performance

Fuel cells are highly efficient because they convert chemical energy directly into electricity without going through an intermediate combustion step. Total efficiencies exceeding 80% can be achieved when both heat and electricity are used. Fuel cells efficiency is maintained under a wide range of unit output. Electric efficiencies and operating temperatures for the various types of fuel cells are summarized in **Table 3.12**. [3-21]

Fuel Cell Type	Electric Efficiency (HHV)	Operating Temperature
Phosphorous acid	36-45%	200°C
Molten carbonate	43-55%	650°C
Solid oxide	43-55%	1000°C

Table 3.12 Fuel cell electric efficiencies and operating temperatures, assuming natural gas fuel

3.7.3 Emissions

Virtually no emissions are produced in this process (zero emissions if pure hydrogen is used).

3.7.4 Economics

Currently, fuel cell CHP plants have a capital cost of about $3000/kW$_e$. [3-22] Intensive R&D and government-sponsored commercialization programs are expected to bring prices down.

3.8 COMPARATIVE ANALYSIS OF CHP TECHNOLOGIES

Choice of CHP technology is based on many case-specific variables, including:

- climate conditions, which affect the heating peak demand and utilization hours as well as the operating performance of certain types of equipment such as gas turbines;
- temperatures of the district heating supply and return;
- cost and availability of fuels;
- value of electricity;
- value of district heat; and
- environmental restrictions.

The following comparative discussion of CHP technologies and the related discussion in **Chapter 6** were developed with the above variables in mind.

Unless otherwise indicated, the following comparisons between technologies are based on representative units summarized in **Table 3.13**. The 20-25 MW_e size range was chosen because this range is likely to be applicable to many district heating and cooling situations, and because all of the major technology types are available in this size range. Additional information on these representative units may be found in the previous sections on each technology type.

Technology Type	Capacity (MW_e)	Fuel
Simple cycle gas turbine	25	Natural gas
Simple cycle diesel engine	20	Natural gas
Combined cycle gas turbine	25	Natural gas
Combined cycle diesel engine	20	Natural gas
Steam turbine	25	Coal

Table 3.13 Unit sizes and fuels used for comparisons

3.8.1 Unit Sizes

Figure 3.52 summarizes the ranges of sizes of CHP technologies.

- Otto engines/lean-burn engines are most common in the range up to 2 MW_e, although larger (5-10 MW_e) otto engines are being developed because it is increasingly difficult to meet NO_x emission limits in diesel engines without expensive catalytic converters. Thus, the top end of the range of otto engines sizes is being pushed upward.
- Diesel engines have historically been the dominant type of reciprocating engine in sizes above 1 MW_e. Four stroke diesel engines are available in the 5-20 MW_e range, which is relevant to many district energy applications. Although larger diesels (up to 50 MW_e) are made, these are expensive relative to the gas turbines.
- Gas turbines are available in sizes ranging from 1 to over 150 MW_e.
- Steam turbines are available in sizes from under 5 MW_e to over 1000 MW_e.

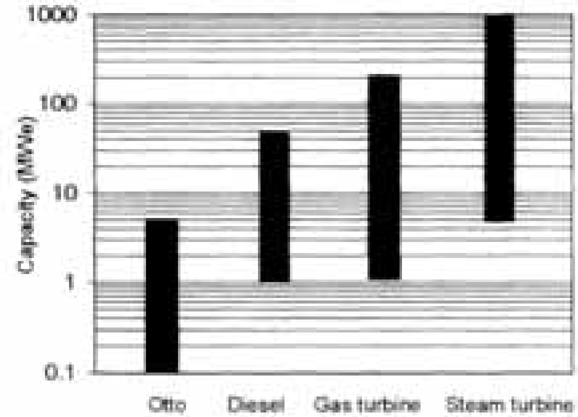

Figure 3.52 Size ranges of CHP technologies

3.8.2 Efficiency

Figure 3.53 summarizes the electric efficiency and total efficiency (electric plus thermal efficiency) for a variety of CHP technologies in the 20-25 MW_e range under the base case assumption of 100/75°C thermal energy recovery. The solid fuel-fired steam turbine CHP plant has a high total efficiency but also has the lowest electric efficiency (under 30%). The gas turbine, either simple cycle or combined cycle, provides total efficiencies of about 87%, compared to a total efficiency of about 80% for diesel engines. Electric efficiency in the gas turbine is the lowest of the options (35%) with a simple cycle, but is significantly increased (to 45%) with a combined cycle. The diesel provides a higher simple cycle electric efficiency compared to the gas turbine (41%) but shows less improvement with a combined cycle, increasing to 43%.

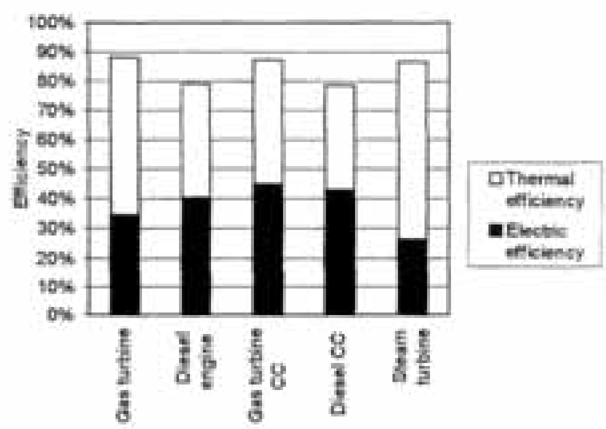

Figure 3.53 Comparative efficiencies of representative CHP technologies (20-25 MW_e, 100/75°C thermal supply/return)

CHP technologies differ relative to the impact of variations in heat recovery temperature on efficiency. These impacts are illustrated in **Figure 3.54** and can be summarized as follows:

- The thermal efficiency of simple cycles (gas turbines and reciprocating engines) decreases with increasing heat supply temperature, while the electric efficiency is constant.
- For combined cycle gas turbines both the electric efficiency and total efficiency decline with increasing heat supply temperature.
- In combined cycle diesel systems the electric efficiency and total efficiency also both decline with increasing heat supply temperature, but the electric efficiency declines less and the total efficiency declines more than for combined cycle gas turbines.
- The electric efficiency of steam turbines decreases with increasing heat supply temperature while the total efficiency remains constant.

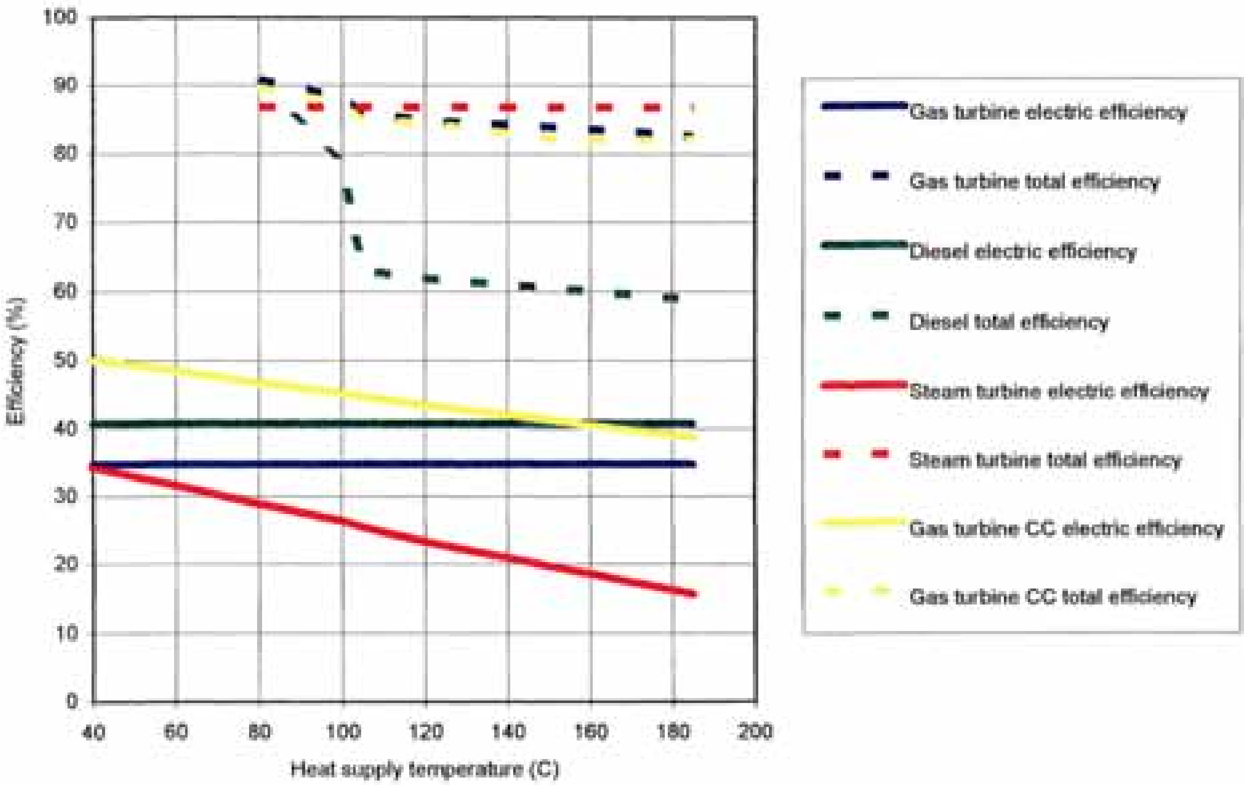

Figure 3.54 CHP efficiencies at different heat supply temperatures

The part-load efficiencies of the CHP technologies are compared in **Figure 3.55**. The chart was developed based on manufacturers' data, including guaranteed performance data. Although diesel engines have a reputation for superior part-load performance, data for the 65-100% range indicate no particular advantage for diesels. Data for gas turbines show a drop of about 10% in both electric and total efficiency as load drops from 100% to 25%.

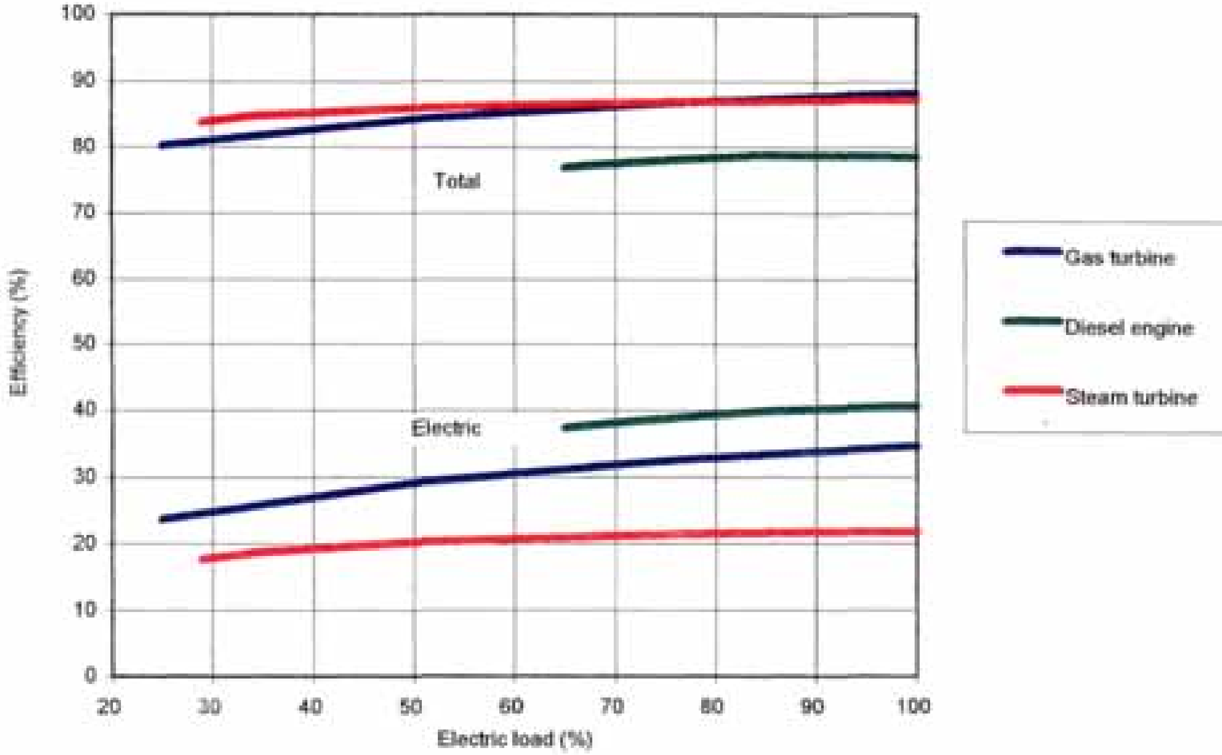

Figure 3.55 Comparative part-load efficiencies of CHP technologies

One other performance parameter deserves note: the impact of ambient temperature on the efficiency and output of gas turbines. At higher outdoor temperatures, when cooling demand will be highest, efficiency and output decline. At 30 °C, output is 10% lower than at ISO conditions (15°C) and efficiency is about 1% lower. The drop in power output of gas turbines at high ambient temperatures should be kept in mind in the technical and economic evaluation of alternatives.

3.8.3 Emissions

Figure 3.56 compares CHP emissions per kWh_e, summarizing the low end of the ranges previously presented. As noted above, emissions can vary significantly depending on the specific technology configuration. Not surprisingly, gas turbine (simple or combined cycle) technologies provide the lowest emissions, except for the potential, which must be analyzed on a case-specific basis, to provide low net CO_2 emissions when burning biomass in steam turbine CHP plants (see footnote for **Table 3.7**). However, like coal, biomass results in relatively high CO emissions.

3.8.4 Economics

Figure 3.57 illustrates the comparative capital costs of various types of CHP technologies, including installation, building and auxiliaries. While there is some "scatter" to these data, generalizations can be made, as summarized in **Table 3.14**.

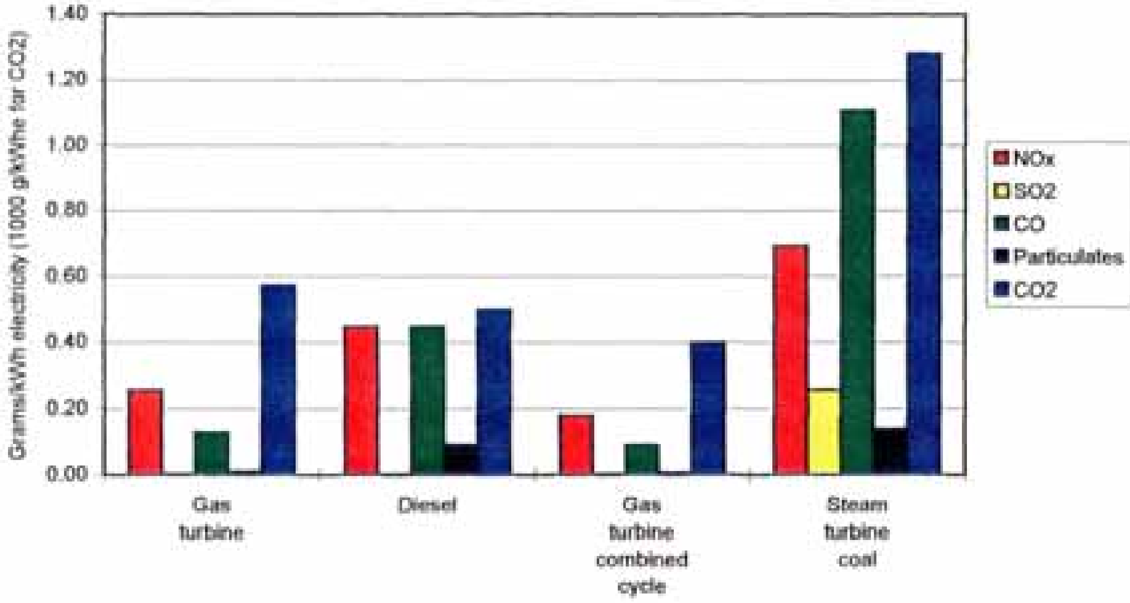

Figure 3.56 Comparison of representative emissions per total kWh_e for different CHP technologies, using lowest values from estimated range of emissions previously presented

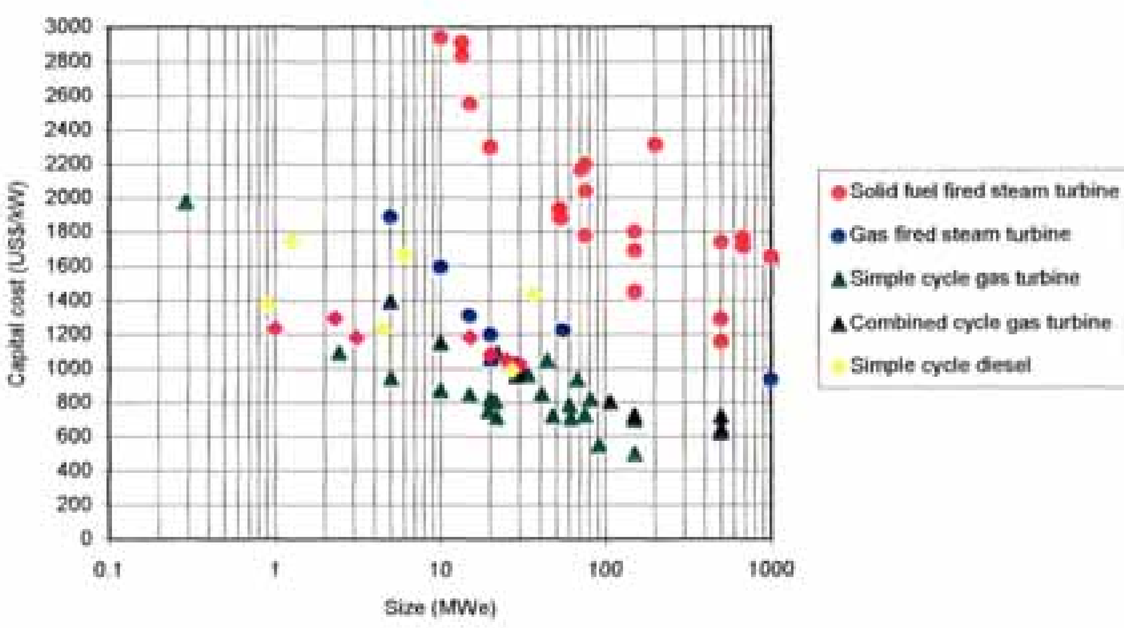

Figure 3.57 Comparative installed capital costs of CHP technologies

	SIZE RANGE (MWe)							
	1 -	5	5 -	50	50 -	150	1000 -	150
Simple cycle gas turbine	1000 -	900	1000 -	700	1000 -	500		
Combined cycle gas turbine			1400 -	1000	1000 -	700	700 -	600
Simple cycle diesel	1400 -	1200	1200 -	1000				
Solid fuel steam turbine			3000 -	1900	2200 -	1400	1800 -	1200
Gas/oil steam turbine			1900 -	1200				

Table 3.14 Summary of installed capital costs of CHP technologies ($/kW$_e$)

Most notable, particularly for the 5-50 MW$_e$ size range that is frequently applicable in CHP district heating and cooling, is the significantly higher capital cost for steam turbine facilities. Solid fuel-fired steam turbine plants in this size range are over twice (and in some cases nearly three times) the capital cost of gas turbine facilities. Gas/oil fired steam turbine plants are considerably less expensive, costing about half the cost of the solid fuel fired steam turbine facilities.

The impact of the higher capital costs for solid fuel-fired steam turbine plants can be seen in **Figure 3.58**, which illustrates the component costs of the CHP technologies in the 20-25 MW$_e$ range, under a consistent assumption of 1.0 cents/kWh fuel cost and 2.0 cents/kWh$_{th}$ thermal value. The component costs are shown for 2500, 5000 and 8000 Equivalent Full Load Hours (EFLH). Note that the net cost is derived by subtracting the thermal credit from the total of the component costs.

The figure shows the significant impact of utilization on the cost of produced electricity. At low levels of utilization (2500 EFLH), capital is the most significant cost, with fuel the next most significant cost. This relationship is reversed as utilization is increased to baseload (8000 EFLH) levels. Labor is an insignificant cost, and non-fuel operation and maintenance costs are relatively minor.

Figure 3.58 compares costs at static assumptions regarding fuel cost and the value of heat. Although solid fuel-fired steam turbine plants have a high capital cost, this is partly compensated by the ability to use lower-cost fuels. The cost-effectiveness of various CHP technologies depends on fuel costs, the value of thermal energy and many other site-specific factors. **Figures 3.59, 3.60 and 3.61** compare the total costs of CHP technologies in the 20-25 MW$_e$ range (expressed as cost per kWh of electricity produced) at a range of fuel costs and a range of thermal values, for 2500, 5000 and 8000 EFLH, respectively.

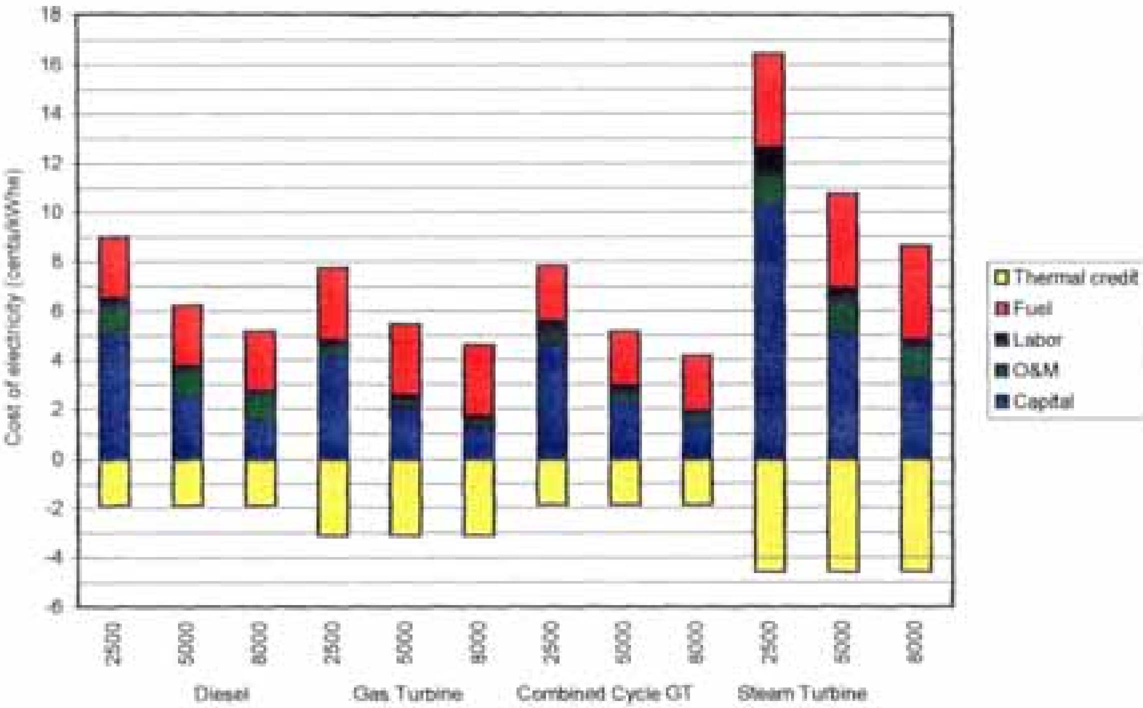

Figure 3.58 Comparative component costs of CHP technologies at various EFLH, size range 20-25 MW$_e$, fuel cost 1.0 cents/kWh and thermal value 2.0 cents/kWh$_{th}$

The following discussion summarizes results under the conditions assumed in the previous calculations. However, different conclusions could result depending on case-specific circumstances.

Under the stated assumptions, simple cycle gas turbines provide the lowest cost technology at a utilization of 2500 EFLH, throughout the ranges of fuel costs and thermal values, as illustrated in **Figure 3.59**. At a fuel cost of 2.0 cents/kWh and zero value for thermal energy, simple cycle and combined cycle gas turbines are equal in cost, but as thermal value increases the simple cycle obtains an increasing cost advantage. The diesel engine is somewhat more expensive than the gas turbine combined cycle by 0.5 cent/kWh$_e$ (at zero thermal value) to about 1.0 cent/kWh$_e$ (at 3.0 cents/kWh$_{th}$ thermal value). At this level of utilization, the steam turbine plant is significantly more expensive than any of the alternatives.

At 5000 EFLH (see **Figure 3.60**), the simple cycle and combined cycle gas turbines are equivalent in cost at a fuel cost of 1.0 cent/kWh and zero value for thermal energy, but again, as thermal value increases the simple cycle obtains an increasing cost advantage. However, at 2.0 cents/kWh fuel, the combined cycle is lower cost at thermal values under 1.0 cent/kWh$_{th}$. Diesel engines and gas turbines are equivalent in cost at a fuel cost of 2.0 cents/kWh and zero thermal value.

Solid fuel-fired steam turbine plants are still relatively expensive at this level of utilization but some feasible scenarios emerge. If the solid fuel has a zero cost (e.g., possible with municipal solid waste) and the fuel cost for the other alternatives is 1.0 cent/kWh, the steam turbine cost becomes lower than the diesel, gas turbine combined cycle and gas turbine simple cycle at thermal values of 0.7, 1.2 and 1.8 cents/kWh$_{th}$ thermal value, respectively. The solid fuel-fired steam turbine has a lower cost than any of the alternatives, at any thermal value, if the solid fuel has zero cost and the fuel for the other alternatives is about 1.5 cents/kWh or greater. At 1.0 cent/kWh fuel cost for solid fuel and 2.0 cents/kWh fuel cost for the alternatives, the steam turbine cost becomes lower than the diesel, gas turbine combined cycle and gas turbine simple cycle at thermal values of 1.5, 2.2 and 3.0 cents/kWh$_{th}$ thermal value, respectively.

At 8000 EFLH (see **Figure 3.61**), the solid fuel-fired steam turbine shows more significant potential advantages. At zero cost of solid fuel, the steam turbine plant is less costly than any of the alternatives if the cost for fuel for those alternatives is 1.0 cents/kWh or greater. If the solid fuel costs 1.0 cent/kWh and the fuel cost for the alternatives is 2.0 cents/kWh, the steam turbine cost becomes lower than the simple cycle diesel or gas turbine at a heat value of 0.6 cents/kWh$_{th}$ and the combined cycle gas turbine at 1.1 cents/kWh$_{th}$ heat value.

At this level of utilization and at zero thermal value, the cost advantage of the simple cycle over the combined cycle gas turbine disappears at low heat values if the price of fuel is over 1.0 cent/kWh. The combined cycle shows a cost advantage over the simple cycle if the fuel cost is 2.0 cents/kWh and the thermal value is under 1.5 cents/kWh$_{th}$.

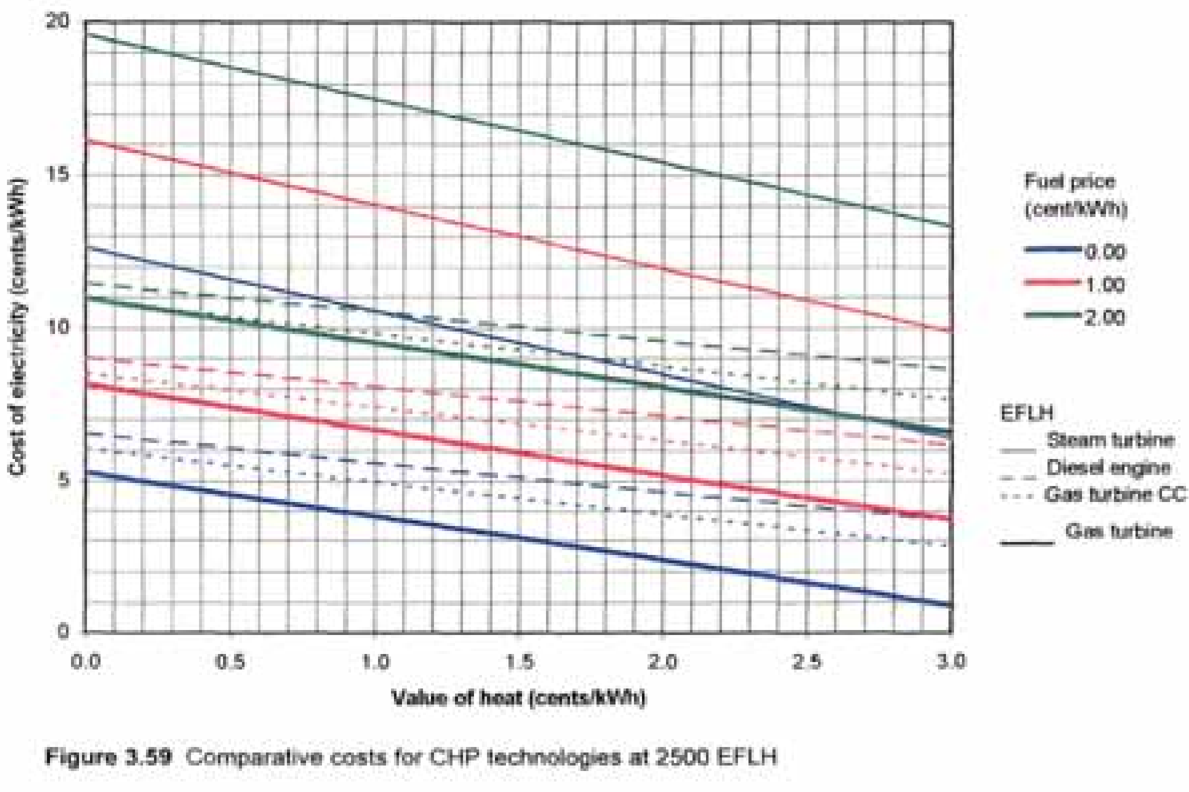

Figure 3.59 Comparative costs for CHP technologies at 2500 EFLH

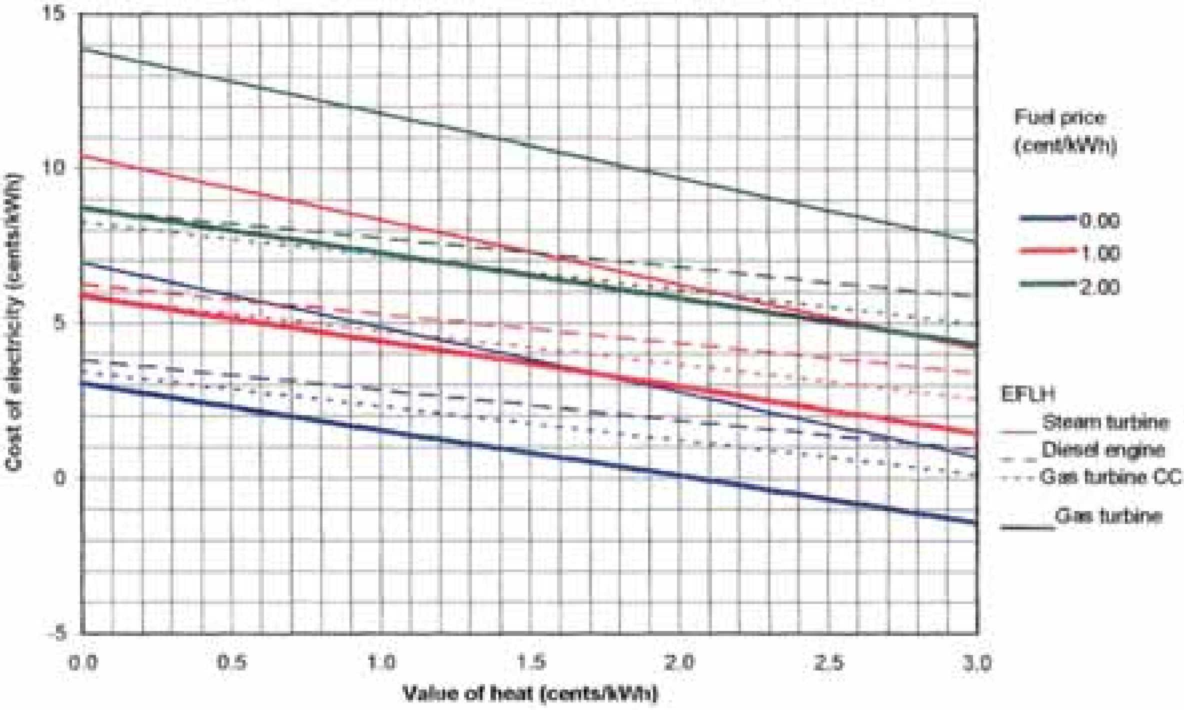

Figure 3.60 Comparative costs for CHP technologies at 5000 EFLH

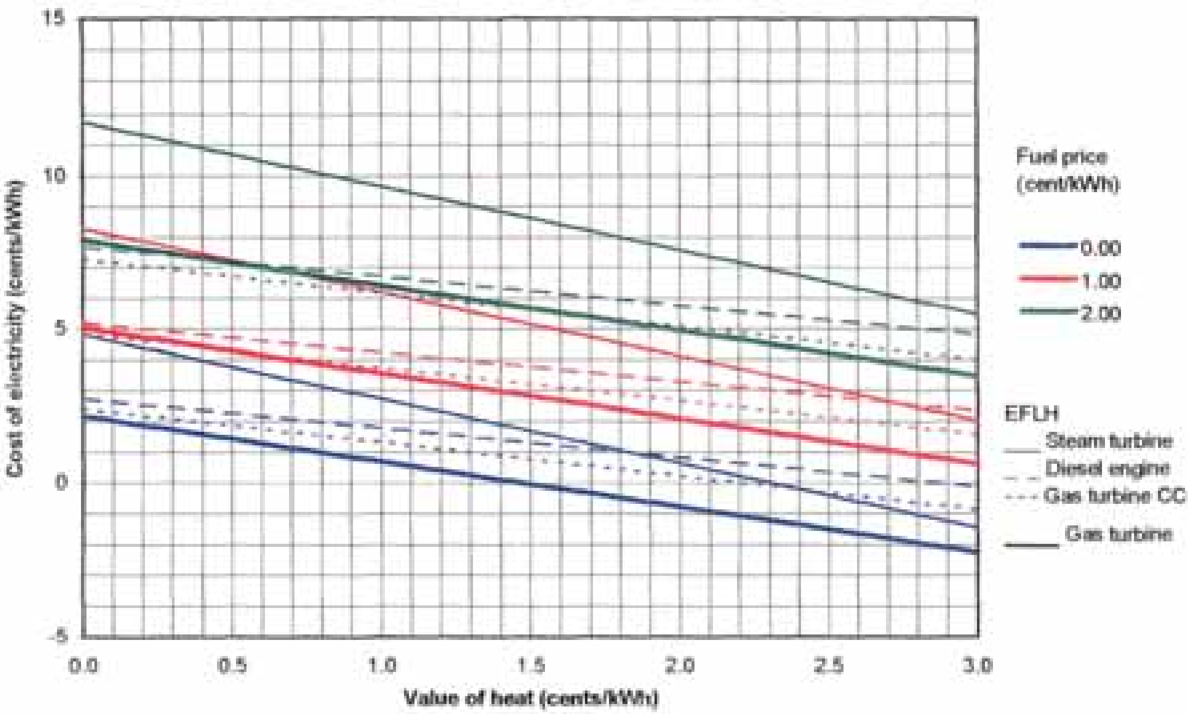

Figure 3.61 Comparative costs for CHP technologies at 8000 EFLH

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Chapter 4
Chiller Technologies

4.1 INTRODUCTION

This chapter describes and quantifies the efficiency, refrigerant environmental impacts and economics of chiller technologies. Following a discussion of key assumptions in **Section 4.2**, **Sections 4.3 - 4.5** focus on chiller technologies likely to be installed for central chilled water production. As discussed in **Section 1.2**, some approaches to district cooling are based on using a district heating system to distribute hot water or steam to drive absorption chillers located in individual buildings (*dispersed absorption*) or in multiple small chilled water production plants (*decentralized chilled water plants*). Because these approaches eliminate or significantly reduce chilled water distribution costs, assessment of their overall economics requires consideration of distribution costs, which is beyond the scope of this report. Therefore, hot water absorption chillers, which will most likely only be employed in district heating-based absorption schemes, are discussed separately in **Section 4.6**.

4.2 ASSUMPTIONS

All performance data for the different chiller technologies are presented at the following conditions unless stated otherwise:

- 6.7°C leaving chilled water temperature
- 29.5°C entering condenser water temperature

These values represent the conditions for which many manufacturers provide performance data. (The precise numbers for °C result from conversion from original data in °F.) However, chilled water temperatures lower than 6.7°C are frequently used in district cooling systems to satisfy customer temperature demands and to minimize pipe sizes in the distribution network. Condenser water temperatures will vary depending on outdoor temperature conditions and cooling tower design considerations.

While different chilled water and condenser water design temperatures will affect chiller performance, variations in these parameters are not shown in **Sections 4.3 - 4.5** because such variations will not substantially affect the *comparative* efficiencies and economics of centralized chiller plant technologies, particularly in the context of combining cooling with CHP. Although the efficiencies of absorption chillers driven with higher temperature energy are relatively unaffected by condenser water temperatures, efficiencies are significantly affected by the condenser water temperature if chillers are driven with lower temperature hot water or steam. Therefore, the impact of varying condenser water temperatures for hot water absorbers is addressed in **Section 4.6**.

Part load performance characteristics for central plant chillers are provided for both a fixed condenser water temperature of 29.5°C and according to the North American standard ARI/IPVL (Air-Conditioning & Refrigeration Institute/Integrated Part-Load Value), where the condenser water temperature is decreased by 1.4°C (2.5°F) for every 10% decrease in chiller output.[4-1]

The economic calculations are based on chiller efficiencies and auxiliary electric usage under peak conditions. Annual performance will in most cases be lower than the efficiency at peak conditions. At part load conditions the overall chiller efficiency will be affected not only by the part load performance of the chiller but also by the auxiliary electric usage for evaporator and condenser water pumps, which is normally fixed. While relatively low at peak conditions, the auxiliary equipment can add substantially to the electricity consumption per unit of cooling output during off-peak conditions.

The annual average chiller performance including auxiliaries will depend on design aspects such as chiller sizing relative to peak and base loads and the availability of chilled water storage, and will be significantly affected by site-specific operation and maintenance factors including heat exchanger fouling, and sub-optimal refrigerant charge. Therefore, average annual performance is extremely difficult to quantify in a general way.

Basing the economic calculations on chiller performance under peak conditions has conflicting impacts on the relative competitiveness of absorption chillers compared to electric centrifugals. Absorption chillers tend to have better part-load efficiencies, but are less flexible operationally -- they take longer to start up and shut down and are slow to respond to changes in inlet water temperature. As noted above, under part load conditions auxiliaries can add substantially to chiller electric requirements per unit of cooling output. Because absorption chillers require more heat rejection in the condenser loop, there is a greater detrimental impact on overall efficiency due to auxiliaries under part-load conditions.

In the economic analyses, operating labor costs are not accounted for because labor cost differences between the different chiller options would not be significant and because labor costs for cooling depend on the extent to which the same labor pool can be used to operate and maintain CHP, district heating and district cooling.

As with all economic calculations in this report, chiller capital costs are amortized over 15 years. However, as is the case with CHP systems, the technical lifetime of chiller equipment may be different than this standardized assumption. For example, the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) considers the technical life of both electric centrifugal and absorption chillers to be 23 years.[4-2]

4.3 COMPRESSION CHILLERS

4.3.1 Description of Technology

The compression cooling cycle can be seen as a reversed steam cycle for power generation. Instead of gaining work, work has to be added to lift the energy in the system from a lower temperature to a higher temperature. **Figure 4.1** illustrates a generalized compression cooling cycle, which can be summarized as follows:

- Refrigerant at low pressure and at a dry saturated condition is compressed to a higher pressure.
- Due to the increased pressure, the refrigerant vapor is condensed and releases heat to the surroundings (condenser water) at a constant condensing temperature (30-40°C).
- The refrigerant condensate is expanded through a valve to a lower pressure.
- At the lower pressure the wet refrigerant vapor picks up heat from the surroundings (evaporator water) at a low temperature (0.5-10°C), thereby evaporating and returning to dry saturated conditions at constant temperature.

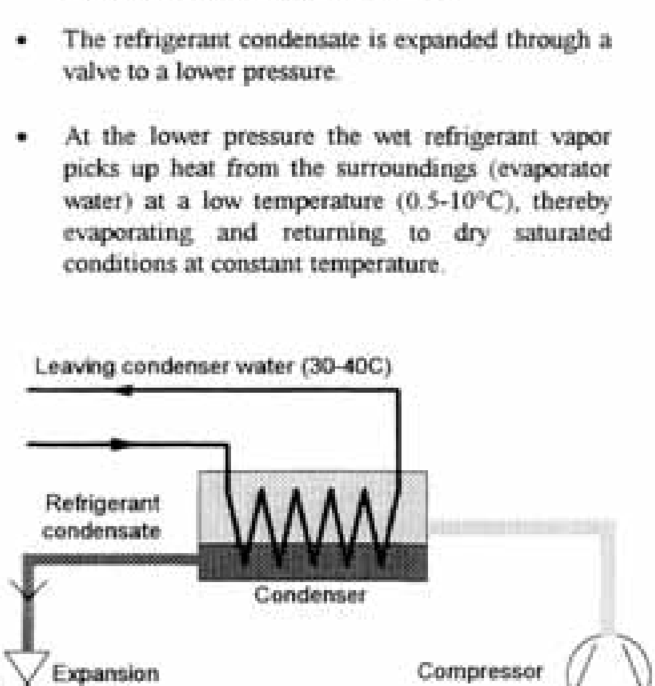

Figure 4.1 Schematic of a compression cooling cycle

4.3.2 Compressor Drives

Reciprocating, screw or centrifugal drives can be used in compression chillers. Reciprocating chillers are available in relatively small sizes (under 1.5 MW_c) and have lower COPs compared to other types of drives. Screw compressors are becoming more common and are available in sizes more suited to district cooling applications. Interest in screw compressors is growing, particularly for use with ammonia as a refrigerant. The screw compressor has a high efficiency and is well suited for the high pressure ratios of ammonia, although noise is a potential problem compared to centrifugal compressors. While the technical and economic differences between centrifugal and screw compressors should be evaluated on a project-specific basis, the differences are not significant for the purposes of this report. This report will focus on the centrifugal type of compression chillers because they are available in a wide range of sizes applicable to district cooling -- from less than 25 kW_c for hermetic packaged units to 24 MW_c for open-drive units.[4-3]

Due to their easy operation and low capital cost, electric drives have been a widespread choice for driving centrifugal chillers. However, a variety of drives can be used in centrifugal chillers. Steam turbine drives are employed by a number of district cooling systems in the U.S. With recent increases in electric prices and the availability of relatively cheap natural gas during the summer, direct combustion engine drives and gas turbine drives are also being applied. Steam turbine drives and combustion engine drives are sometimes promoted as providing redundancy in case of power failure. However, even if the chiller itself is not driven electrically, electricity is still required for the chiller, boiler and plant auxiliaries, and the plant must have its own power source and frequency control to be able to operate during a power failure, thereby adding to the investment cost.

While the combustion engine has a higher efficiency than the steam turbine, the steam turbine drive offers better redundancy because it is possible to feed the chiller from more than one boiler and therefore from multiple fuel sources. Reciprocating engine or gas turbine driven chillers are not further considered in this report because they do not relate to integration of district cooling with CHP, which is the focus of this report.

4.3.3 Performance

Table 4.1 summarizes energy consumption for various types of electric drive chillers, including the chiller and associated auxiliaries (evaporator water pump, condenser water pump and cooling tower fans). **Table 4.2** summarizes energy consumption for steam turbine drive centrifugal chillers, including the chiller and auxiliaries. The chiller steam turbine was assumed to be driven with 11 bar (185°C) steam. The values in **Tables 4.1 and 4.2** are based on the following assumptions:

- 6.7°C leaving chilled water temperature with a 8.3°C temperature difference
- Water cooled condenser with a 29.5°C inlet condenser water temperature and a 5.6°C temperature difference
- 2.1 bar condenser water pump head and 0.8 bar chilled water pump head
- 85% pump efficiency
- 0.011 kW_e/kW_{th} cooling tower fan electric usage

Chiller type	Consumption		Total COP
	Chiller (kWe/kWc)	Auxiliaries (kWe/kWc)	(kWc/kWe)
Reciprocating	0.22 - 0.24	0.03	3.85
Screw	0.18 - 0.21	0.03	4.53
Centrifugal			
- high efficiency	0.16 - 0.18	0.03	5.06
- moderate efficiency	0.18 - 0.20	0.03	4.63

Table 4.1 Electric drive centrifugal chiller and auxiliary performance [4-3, 4-4]

Chiller type	Consumption		Total COP
	Chiller (kWth/kWc)	Auxiliaries (kWe/kWc)	(kWc/kWe)
Steam turbine centrifugal	0.92	0.04	1.04

Table 4.2 Steam turbine drive centrifugal chiller and auxiliary performance [4-3]

Part-load performance of electric centrifugal chillers (chiller only -- no auxiliaries) is illustrated in **Figure 4.2**. These data indicate that with a constant condenser temperature the electricity consumption at 30% load is 40% of the consumption at 100% load, i.e., the consumption per unit of cooling output is 33% higher. However, with variable condenser temperature the performance is closer to linear, i.e., the COP of the chiller is nearly constant throughout the range of loads.

Increasing electricity costs has been a driving force for improving electric chiller efficiency and, as shown in **Table 4.1**, centrifugal chillers with electricity consumption (chiller only) as low as 0.16 kW_e/kW_c (COP about 6.1) are commercially available. While the efficiency of the compressors cannot be improved much further, overall efficiency can be improved by increasing or enhancing the heat transfer surfaces, thereby lowering the temperature difference between the refrigerant and chilled/condenser water. To increase efficiency, some manufacturers are using plate heat exchangers while others are enhancing the standard tube heat exchangers.

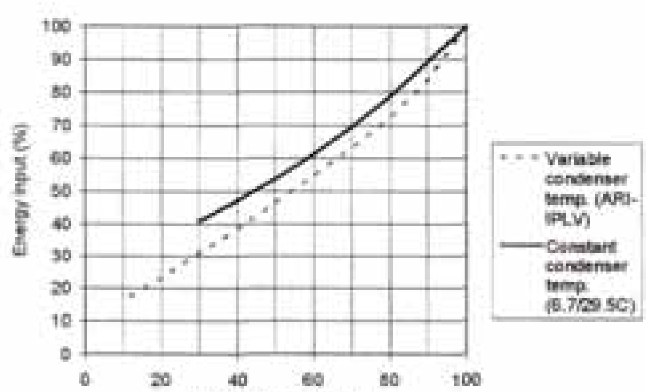

Figure 4.2 Representative part load performance for electric centrifugal chillers [4-5]

4.3.4 Refrigerants

In 1987, twenty three countries signed the Montreal Protocol on Substances That Deplete the Ozone Layer. The Protocol established schedules for curtailing and eventually ceasing production and consumption of chlorofluorocarbons (CFCs) used as refrigerants and for other purposes because these compounds contain chlorine, which destroys the ozone layer in the stratosphere.

The destruction of stratospheric ozone is of serious international concern because this layer protects the earth from harmful ultraviolet radiation which can cause health and environmental damage including increased incidence of skin cancer, cataracts, suppression of the immune system, damage to crops and other impacts. [4-6] CFCs and some other refrigerants also act as "greenhouse gases," i.e., they trap heat entering as solar radiation and prevent the heat from escaping into space -- an issue which is also the focus of international concern and joint action.

The phaseout schedule was tightened at the Second Meeting of the Protocol Parties held in London in 1990, and was again amended in 1992 in the Copenhagen Revisions to the Montreal Protocol. Key provisions of the Copenhagen Revisions are:

- phaseout of CFC production and consumption was accelerated to January 1, 1996;
- hydrochlorofluorocarbons (HCFCs) were added to the list of chemicals to be controlled;
- a schedule was established which completely phases out production and consumption of HCFCs by 2030, and freezes consumption of HCFCs beginning in 1996 to a baseline ceiling of 100% of the consumption of HCFCs in 1989, weighted by ozone depletion potential (ODP), plus 3.1% of the CFCs consumed in 1989, also weighted by ODP.

Table 4.3 summarizes the refrigerants used in compression chillers, and their potential for impact on ozone depletion and global warming. Hydrofluorocarbons (HFCs) and ammonia are not restricted by international protocols.

	Ozone Depletion Potential [4-7]	Global Warming Potential [4-8]
Chlorofluorocarbons		
CFC-11	1.00	1,500
CFC-12	0.93	4,500
CFC-113	0.83	2,100
CFC-114	0.71	5,500
CFC-115	0.38	7,400
Hydrochlorofluorocarbons (interim replacement)		
HCFC-22	0.05	510
HCFC-123	0.02	29
Hydrofluorocarbons (long term replacement)		
HFC-134a	0.00	420
HFC-152a	0.00	47
Other refrigerants		
Ammonia	0.00	0

Table 4.3 Refrigerants and their environmental impact

4.3.5 Economics

Capital Cost

Table 4.4 summarizes generalized capital costs per kW_c for electric drive and steam turbine drive compression chiller plants. These estimates are based on a 35 MW_c chiller plant, assuming seven 5 MW_c chillers plus auxiliaries, building space and project overhead (engineering and project management). Key assumptions are the same as summarized for **Tables 4.1** and **4.2**. No costs were included for chilled water distribution. Capital costs for a particular installation will depend on many site-specific variables, including the extent of upgrades to the plant electric service which may be required to handle the power requirements of the chillers. No major electric service upgrade was assumed in **Table 4.4**.

In the generalized costs presented in **Table 4.4**, costs for the chiller and auxiliary equipment is only 40% of the total installed cost, with installation, piping, electrical and building adding another 40%. The remaining 20% of the total installed cost is for engineering, project management and contingency. Total costs for steam turbine drive chiller plants are over 40% higher than electric-drive, with significant differences in the cost of the chiller drive, cooling tower and installation.

	Electric	Steam turbine
Bare chiller	57	57
Chiller drive	14	45
Chiller pumps (primary)	4	4
Cooling tower, condenser pumps	20	30
Mechanical installation and piping	50	87
Instrumentation and control	8	14
Electrical	25	17
Building and foundation	14	21
Subtotal	194	277
Project management (5%)	10	14
Engineering (10%)	19	28
Contingency (10%)	19	28
TOTAL	242	346

Table 4.4 Generalized compression chiller system capital costs (\$ per kW_c) [4-3, 4-9]

Operation and Maintenance Cost

Chiller plant operating costs include: cooling tower make-up water; chemical treatment for make-up water; maintenance parts and labor; and operating labor. As discussed in **Section 4.2**, operating labor costs are too site-specific to usefully address in this general analysis.

Make-up water is required to replace water lost in the cooling towers due to evaporation, bleed-off and drift. In addition to costs for purchase of water, chemical treatment is applied to control corrosion and biological growth. Make-up water requirements and costs for a particular installation will vary depending on chiller design, climate, the cost and hardness of water available for condenser cooling, and the specific water treatment program.

Representative costs for make-up water and chemical costs are summarized in **Table 4.5** based on heat rejection requirements calculated from the COPs and the cost factors as indicated. Costs are higher for steam turbine drive chillers because greater quantities of heat must be rejected compared to electric centrifugal chillers.

	Electric	Steam Turbine
Make-up water consumption (liters/kWhth)	3.50	3.50
Make-up water consumption (liters/kWhc)	4.08	6.71
Water cost (\$/1000 liters)	0.79	0.79
Chemical cost (\$/1000 liters)	0.26	0.26
Total water/chemical cost (\$/1000 liters)	1.06	1.06
Total water and chemical cost (cent/kWhc)	0.43	0.71

Table 4.5 Costs for makeup water and chemicals for compression chiller cooling towers [4-4]

Maintenance requirements include regular monthly and annual maintenance, periodic major maintenance (overhauls, etc.) and unscheduled repairs. Estimated chiller maintenance costs are summarized in **Figure 4.3**. For larger electric centrifugal chillers (over 2.5 MW_c) the estimated annual cost is about \$5.00/kW_c.

Maintenance costs for steam turbine drive chillers are assumed to be higher (about \$7.00/kW_c) due to the additional maintenance requirements associated with the steam turbine drive itself. The additional \$2.00/kW_c annual costs were estimated to equal 2% of the difference in capital cost between the two types of chillers.

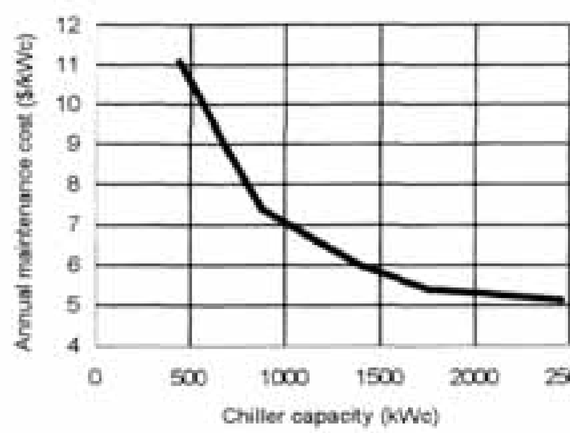

Figure 4.3 Electric centrifugal chiller maintenance cost [4-4]

Overall Economics

Figures 4.4 and 4.5 illustrate the overall economics of electric centrifugal and steam turbine drive compression chillers, respectively, at various electricity prices and Equivalent Full Load Hours (EFLH). These calculations represent generalized unit costs for 5 MW_c chillers in a multiple-unit plant. Key assumptions are detailed in **Table 4.6**. Operating labor costs were not included, as discussed in **Section 4.2**. Consistent with all economic analyses in this report, capital costs are amortized over 15 years.

The cost of cooling is quite sensitive to utilization because of the large role that capital costs play in the overall economics. Based on this generalized analysis, electric centrifugal chillers providing base load service (2000-5000 EFLH) would provide a kWh of cooling for a cost of 1.5-4.0 cents/kWh$_c$ based on electricity costs ranging from 2-10 cents/kWh$_e$. Electric centrifugal chillers used for peaking (500 EFLH) provide cooling at a significantly higher cost: 7.5-9.0 cents/kWh$_c$ based on electricity costs ranging from 2-10 cents/kWh$_e$.

Steam turbine chiller economics are even more sensitive to utilization because of their higher capital costs. Based on this analysis, steam turbine drive centrifugal chillers providing base load service (2000-5000 EFLH) would provide a kWh$_c$ of cooling for a cost of 2-7 cents/kWh$_c$ based on steam costs ranging from 0-4 cents/kWh$_{th}$. Steam turbine drive centrifugal chillers used for peaking (500 EFLH) would provide cooling at a significantly higher cost: 10.5 cents/kWh$_c$ at zero cost of steam to over 14 cents/kWh$_c$ at a steam cost of 4 cents/kWh$_{th}$. Although not shown in **Figure 4.5**, steam turbine drive chillers used for peaking display a minor sensitivity to electric costs for auxiliaries. If steam turbine drive chillers are used for base load service their economics are not sensitive to the cost of electricity for auxiliaries.

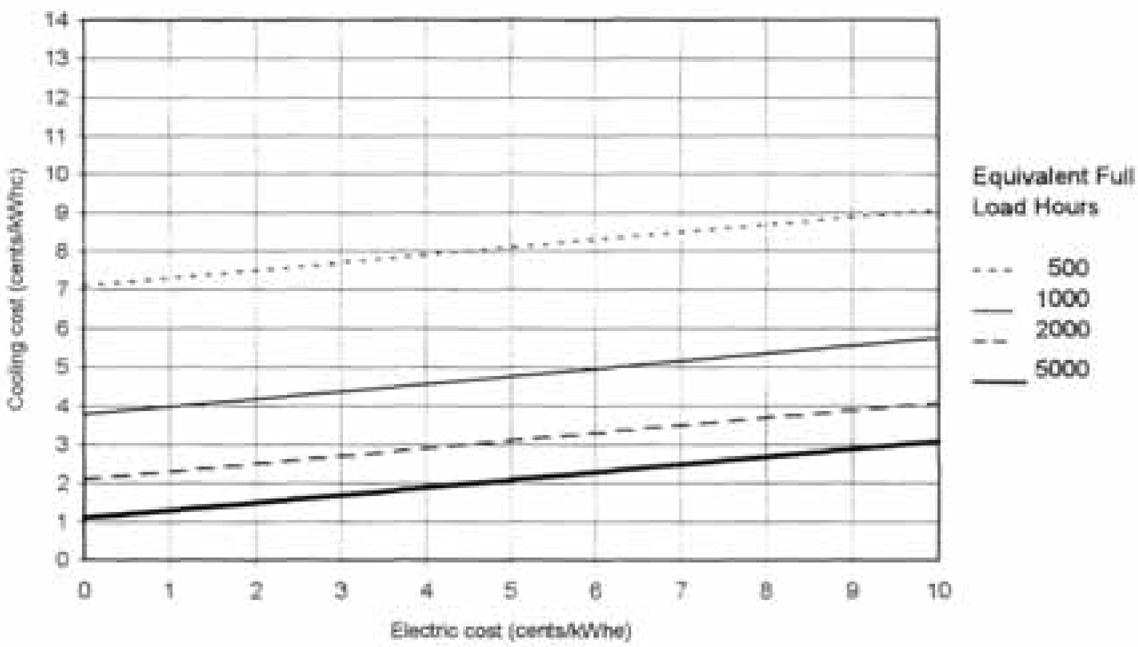

Figure 4.4 Representative economics for electric centrifugal chiller (5 MW$_c$ chillers in a 35 MW$_c$ plant)

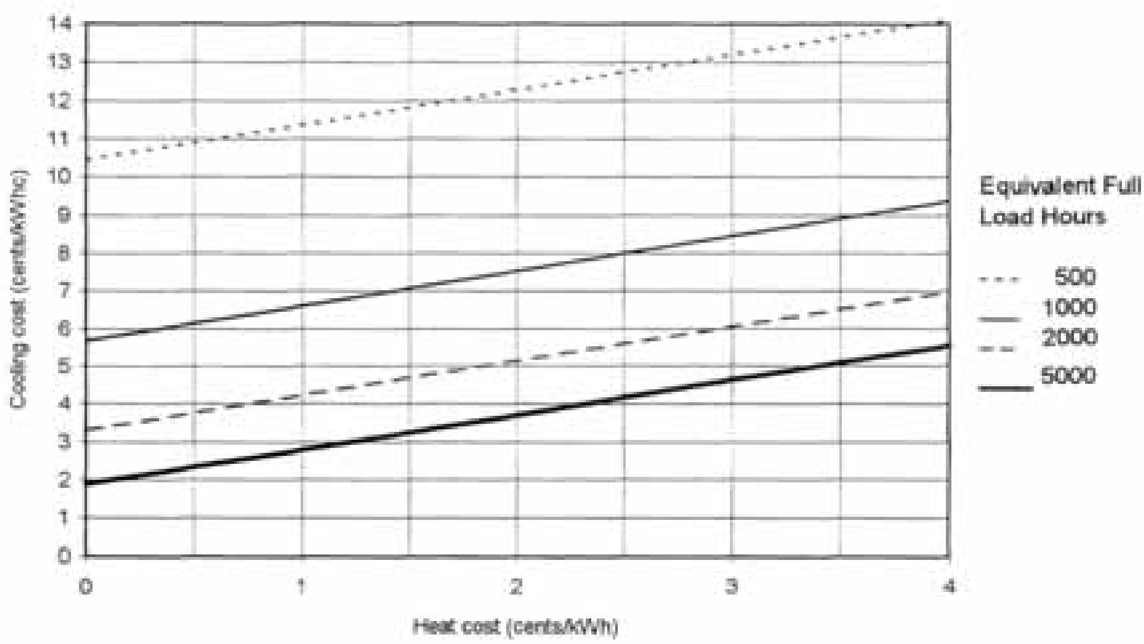

Figure 4.5 Representative economics for steam turbine drive centrifugal chiller (5 MW$_c$ chillers in a 35 MW$_c$ plant)

		Electric Centrifugal	Steam Turbine Drive
A	Capital cost ($/kWc)	242	346
B	Real interest rate (%)	8.00%	8.00%
C	Capitalization period (years)	15	15
D	Capital recovery factor (%)	11.68%	11.68%
E	Water & chemical costs (cents/kWhc)	0.431	0.709
F	Maintenance cost ($/kWc/year)	5.12	7.20
G	Electricity to chiller (kWhe/kWhc)	0.17	
H	Electricity to auxilliaries (kWhe/kWhc)	0.036	0.043
I	Cost of electricity (cents/kWhe)	0-10	5
J	Heat to chiller (kWhth/kWhc)		0.92
K	Cost of heat (cents/kWhth)		0-4
L	Equivalent Full Load Hours	500-5000	500-5000

Formulas: Cooling price (cents/kWh$_c$) = (A x D x 100/L) + E + (F x 100/L) + [(G+H) * I] + (J * K)

Capital recovery factor (D) = $[B \times (1 + B)^C] / [(1 + B)^C - 1]$

Table 4.6 Assumptions for compression chiller economics (5 MW$_c$ chillers in a 35 MW$_c$ plant)

4.4 ABSORPTION CHILLERS

4.4.1 Description of Technology

The absorption cycle uses two media: a refrigerant and an absorbent. Water/lithium bromide and ammonia/water are the most common refrigerant/absorbent media pairs, but other pairs can be used. (See **Section 4.4.3** for further discussion of refrigerants.) A schematic of one-stage absorption is shown in **Figure 4.6**.

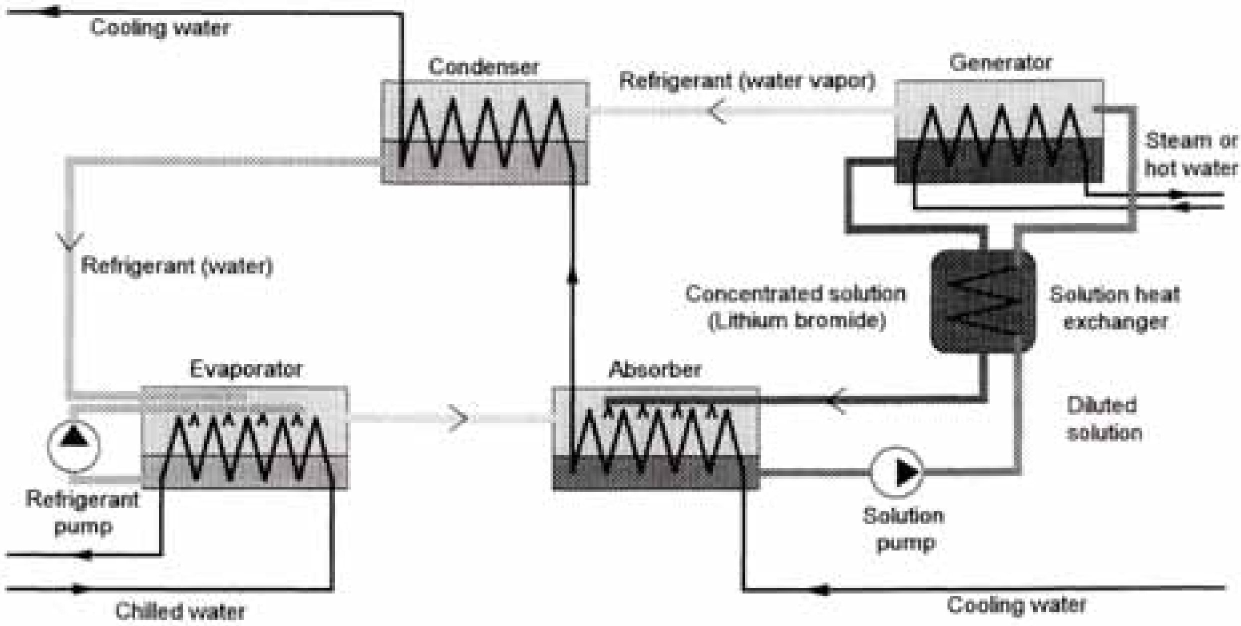

Figure 4.6 Schematic of one-stage absorption cycle (water/lithium bromide)

The absorption cycle can be summarized as follows:

- Generator - Steam or hot water is used to boil a solution of refrigerant/absorbent (water/lithium bromide or ammonia/water). Refrigerant vapor is released and the absorbent solution is concentrated.

- Condenser - The refrigerant vapor released in the concentrator is drawn into the condenser. Cooling water cools and condenses the refrigerant.

- Evaporator - Liquid refrigerant flows through an orifice into the evaporator. Due to the lower pressure in the evaporator, flashing takes place. The flashing cools the remaining liquid refrigerant down to the saturation temperature of the refrigerant at the pressure present within the evaporator (approximately 4°C for a water/lithium bromide chiller). Heat is transferred from the chilled water to the refrigerant, thereby cooling the chilled water and vaporizing the refrigerant.

- Absorber - Refrigerant vapor from the evaporator is drawn to the absorber section by the low pressure resulting from absorption of the refrigerant into the absorbent. Cooling water removes the heat released when the refrigerant vapor returns to the liquid state in the absorption process. The diluted solution is circulated back to the generator.

- Heat exchanger - The heat exchanger transfers heat from the relatively warm concentrated solution being returned from the generator to the absorber and the dilute solution being transferred back to the generator. Transferring heat between the solutions reduces the amount of heat that has to be added in the generator and reduces the amount of heat that has to be rejected from the absorber.

In two-stage absorption cycles, heat derived from refrigerant vapor boiled from solution in the first stage generator is used to boil out additional refrigerant in a second generator, as illustrated in **Figure 4.7**. Two-stage absorption requires a higher quality thermal source. Two-stage absorption chillers are typically driven with 8-9 bar steam (170-175°C), but lower pressures can be used if the capacity derate (as discussed in Section 4.4.2) is acceptable. In later calculations, a steam pressure of 8 bar (170°C) will be assumed for driving two-stage absorption chillers.

Packaged absorption chillers are available in relatively small sizes, up to 5.8 MW_c for one-stage and up to 5.3 MW_c for two-stage.[4-3]

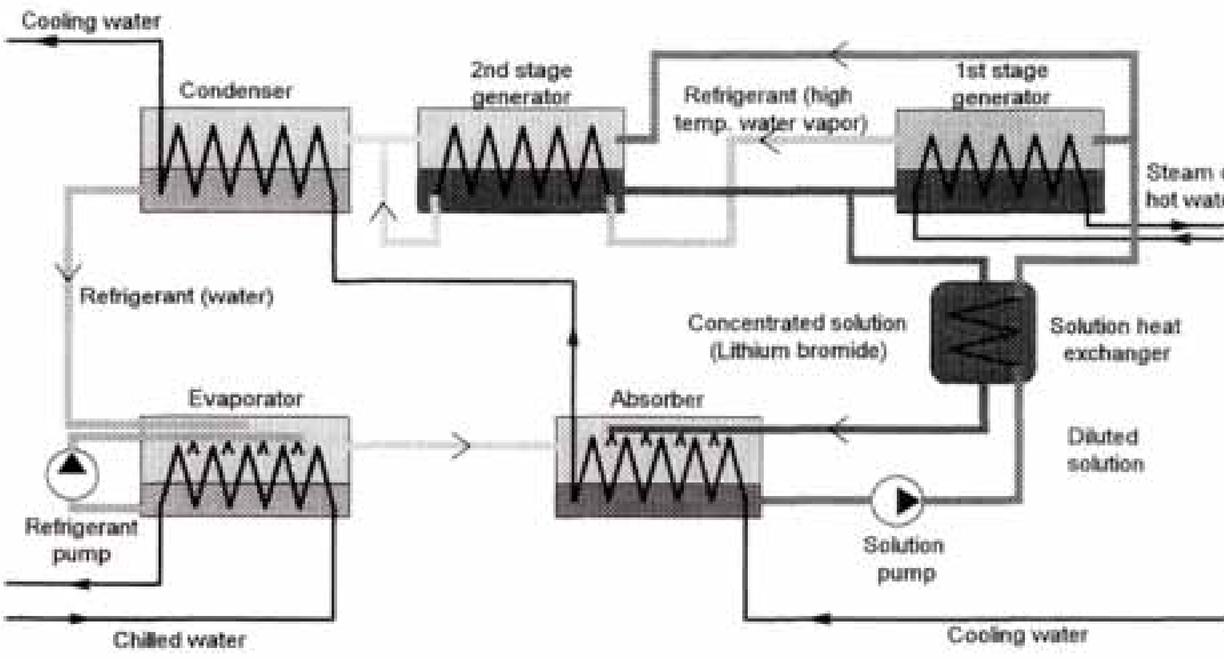

Figure 4.7 Schematic of two-stage absorption cycle (water/lithium bromide)

4.4.2 Performance

Table 4.7 summarizes energy consumption for water/lithium bromide steam absorption chillers, including the chiller and associated auxiliaries (evaporator water pump, condenser water pump and cooling tower fans). The values are based on the steam pressures shown and the following assumptions:

- 6.7°C leaving chilled water temperature with a 8.3°C temperature difference
- Water cooled condenser with a 29.5°C inlet condenser water temperature and a 5.6°C temperature difference
- 2.1 bar condenser water pump head and 0.8 bar chilled water pump head
- 85% pump efficiency
- 0.011 kW_e/kW_{th} cooling tower fan electric usage

Due to the lower efficiency of one-stage steam absorption compared to two-stage, more heat must be rejected in the cooling tower. Therefore, electric consumption for auxiliaries is higher for one-stage absorption.

Representative part-load performance of a water/lithium bromide absorption chiller (chiller only -- no auxiliaries) is illustrated in **Figure 4.8**. The solid line illustrates performance with a constant condensing temperature (29.5°C entering condenser temperature for a 6.7°C chilled water leaving temperature). The improved energy efficiency with variable condenser temperature based on ARI-IPLV, as described in **Section 4.3**, is illustrated by the dashed line. The part-load performance of absorption chillers is generally better than electric centrifugal chillers under the same condenser temperature assumptions. In fact, the absorption chiller performance shown in **Figure 4.8** for constant condenser temperature is only slightly worse than the electric centrifugal chiller performance shown in **Figure 4.2** for variable condenser temperature.

The capacity, efficiency and economics of chillers depend on many case-specific variables, including the design of the particular chiller. For absorption chillers a key consideration is the temperature of the driving thermal energy. The capacity decline resulting from declining steam pressure will vary somewhat depending on the design of the chiller. Chillers with more heat transfer surface area in the generator would be able to provide higher capacities with low temperature driving energy. However, the additional surface area adds to the capital cost. The impacts of the driving energy temperature on capacity and COP are further discussed in **Section 4.6**.

Figure 4.9 shows the decline in capacity for one-stage steam absorption chillers as the steam pressure (and therefore temperature) of the driving energy decreases. Capacity with 1 bar steam (100°C) is about 65% of the capacity with 2 bar steam (120°C), and with 0.5 bar steam (about 80°C), this percentage drops to less than 40%. **Figure 4.10** shows the decline in capacity for two-stage steam absorption chillers. Capacity with 4 bar steam (145 °C) is about 50% of the capacity with 9 bar steam (175°C).

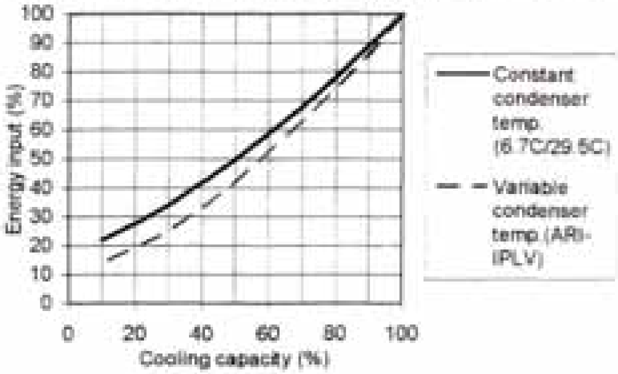

Figure 4.8 Representative part load performance for absorption chillers [4-3]

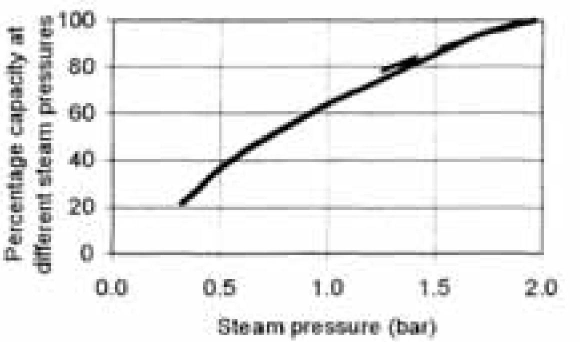

Figure 4.9 Impact of steam pressure on one-stage absorption chiller capacity for two different chillers [4-3]

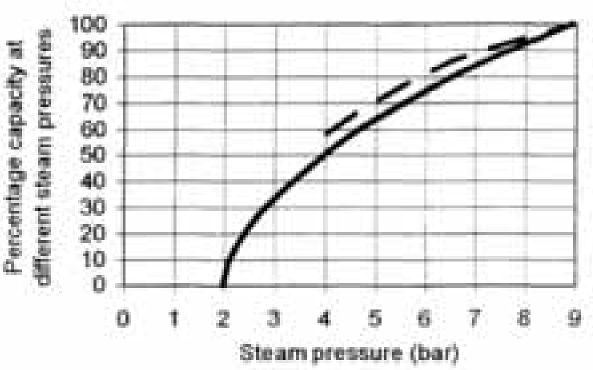

Figure 4.10 Impact of steam pressure on two-stage absorption chiller capacity for two different chillers [4-3]

Chiller type	Steam pressure (temperature)	Consumption: Chiller (kWth/kWc)	Consumption: Auxiliaries (kWe/kWc)	Total COP (kWc/kW)
1-stage steam	2 bar (120C)	1.50	0.06	0.64
2-stage steam	8 bar (170C)	0.83	0.04	1.14

Table 4.7 Steam absorption chiller and auxiliary performance [4-3]

4.4.3 Refrigerants

The most common media pairs used in absorption cycles are: 1) water (refrigerant) with lithium bromide (absorbent); and 2) ammonia (refrigerant) with water (absorbent). Other media pairs are under development for absorption chillers. Advantages (+) and disadvantages (-) of the common media pairs are:

Water/lithium bromide

- \+ No toxicity
- \+ Low pressures in the process
- \+ No losses comparable with the water separating losses in the ammonia/water process
- \- The lithium bromide easily crystallizes during shutdown periods or malfunctions
- \- Normally limited to a minimum chilled water output temperature of about 4° C to avoid freezing of the refrigerant (water)
- \- Large vapor volumes
- \- Air can leak into the process due to the low pressures
- \- The temperature lift is limited to 30-35°C

Ammonia/water

- \+ Ammonia is an inexpensive refrigerant
- \+ Ammonia is stable during shutdowns
- \+ Can produce ice or lower temperature chilled water
- \+ Can be used for relatively high temperature lifts
- \+ Leaks are easily detected due to distinctive and strong odor
- \- Ammonia is toxic
- \- Lower COP than lithium bromide chillers
- \- Water separating losses occur in the generator
- \- High pressures are needed in the process
- \- Higher capital costs

Research on other media pairs is focused on increasing the generator temperature in multi-stage absorbers and thereby increasing the efficiency. However, media pairs which enable use of lower generator temperatures with the same or better COP are of greater relevance for combining district cooling with CHP. Examples of such chemicals, which are currently under study, are listed below, with notes on their advantages:

Water/Lithium chloride (LiCl)

- \+ Higher COP than $H_2O/LiBr$
- \+ Better heat transfer than $H_2O/LiBr$
- \+ Possible to use lower generator temperatures compared to $H_2O/LiBr$

Water/Lithium bromide/Lithium cyanat (LiSCN)

- \+ Higher COP than $H_2O/LiBr$ and $H_2O/LiCl$
- \+ Possible to use lower generator temperatures compared to $H_2O/LiBr$ but higher than $H_2O/LiCl$

4.4.4 Economics

Capital Costs

Table 4.8 summarizes generalized capital costs per kW_c for one-stage and two-stage water/lithium bromide steam absorption plants. These estimates are based on a 35 MW_c chiller plant, assuming seven 5 MW_c chillers and including auxiliaries, building space and project overhead. Key assumptions are the same as summarized for **Tables 4.1** and **4.2**. No costs are included for chilled water distribution. Total unit costs for two-stage absorption are about 20% higher than for one-stage chillers.

	1-stage steam	2-stage steam
Bare chiller	74	128
Chiller pumps (primary)	4	4
Cooling tower, condenser pumps	38	29
Mechanical installation and piping	66	66
Instrumentation and control	11	11
Electrical	13	13
Building and foundation	23	28
Subtotal	229	279
Project development/management (5%)	11	14
Engineering (10%)	23	28
Contingency (10%)	23	28
TOTAL	286	349

Table 4.8 Generalized steam absorption chiller system capital costs ($/kWc) [4-3, 4-9]

Operation and Maintenance

Chiller system operation and maintenance costs are discussed generally in the **Section 4.3**. Opinions vary regarding the relative maintenance costs of absorption and compression chillers.* Estimated maintenance costs per kW_c from several sources are shown in **Table 4.9**.

	Reference		
	4-4	4-10	4-11
Electric centrifugal compression	5.12	4.10	6.97
Absorption	6.26	4.72	4.66
Absorption/compression cost ratio	1.22	1.15	0.67

Table 4.9 Estimates of relative maintenance costs of compression and absorption chillers ($\$/kW_c$/year)

Given the wide range of estimates and opinions regarding the relative costs of maintenance, in the following estimates it is assumed that the cost per kW_c of maintaining steam absorption and electric centrifugal compression chillers are equal at $5.12 per kW_c per year.

One potential operational concern for water/lithium bromide absorption chillers is the possibility of crystallization of concentrated solution. This is now a relatively uncommon and manageable problem, because microprocessor-based control systems in newer absorption chillers monitor crystallization parameters closely, allowing the machine to operate at lower temperatures yet maintain proper flow and provide an orderly shutdown.

Make-up water requirements and costs for a particular installation will varying depending on chiller design, climate, the cost and hardness of water available for condenser cooling, and the specific water treatment program. Greater quantities of heat must be rejected in the cooling towers for absorption chillers compared to electric centrifugal chillers, resulting in higher make-up water costs and higher cooling tower electricity costs. The one-stage absorption chiller has higher make-up water requirements, compared to two-stage, because it is less efficient in converting thermal energy to cooling.

Representative costs for absorption chiller make-up water and chemical costs are summarized in **Table 4.10** based on heat rejection requirements calculated from the COPs and the cost factors as indicated.

	1-stage steam	2-stage steam
Make-up water consumption (liters/kWhth)	3.50	3.50
Make-up water consumption (liters/kWhc)	8.76	6.42
Water cost ($/1000 liters)	0.79	0.79
Chemical cost ($/1000 liters)	0.26	0.26
Total water and chemical cost ($/1000 liters)	1.06	1.06
Total water and chemical cost (cent/kWhc)	0.93	0.68

Table 4.10 Costs for makeup water and chemicals for absorption chiller cooling towers [4-4]

Overall Economics

Figures 4.11 and 4.12 illustrate the overall economics of one-stage and two-stage steam absorption chiller systems, respectively, at various electricity prices and Equivalent Full Load Hours (EFLH). The one-stage two-stage chillers are assumed to be driven with 2 bar and 8 bar steam, respectively. These calculations represent generalized unit costs for 5 MW_c chillers in a multiple-unit plant.

Key assumptions are detailed in **Table 4.11**. Auxiliaries were assumed to be powered by electricity at a cost of 5.0 cents/kWh_e. Total costs per unit of cooling were not significantly affected by variations in the cost of electricity. As with all economic calculations in this report, chiller capital costs were amortized over 15 years.

The cost of absorption cooling is quite sensitive to utilization. Based on this analysis, one-stage steam and two-stage steam absorption chillers providing base load service (2000-5000 EFLH) would cost 2.0 - 9.0 and 2.0 - 6.5 cents/kWh_c, respectively, based on heat costs ranging from 0-4 cents/kWh_{th}. Absorption chillers used for peaking (500 EFLH) provide cooling at a significantly higher cost: 5.5 - 13.5 and 5.0 - 15.0 cents/kWh_c, respectively, based on heat costs ranging from 0-4 cents/kWh_{th}.

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* Several references addressed relative maintenance costs qualitatively. One reference notes that one of the advantages of absorption chillers is "low cost maintenance because there are few moving parts to service and refrigerant is inexpensive and readily available." [4-12] Another source states "Absorption chillers have few moving parts, and a properly cared for absorption chiller has no greater chance of failure than a mechanical chiller, nor does it cost more to maintain." [4-13]

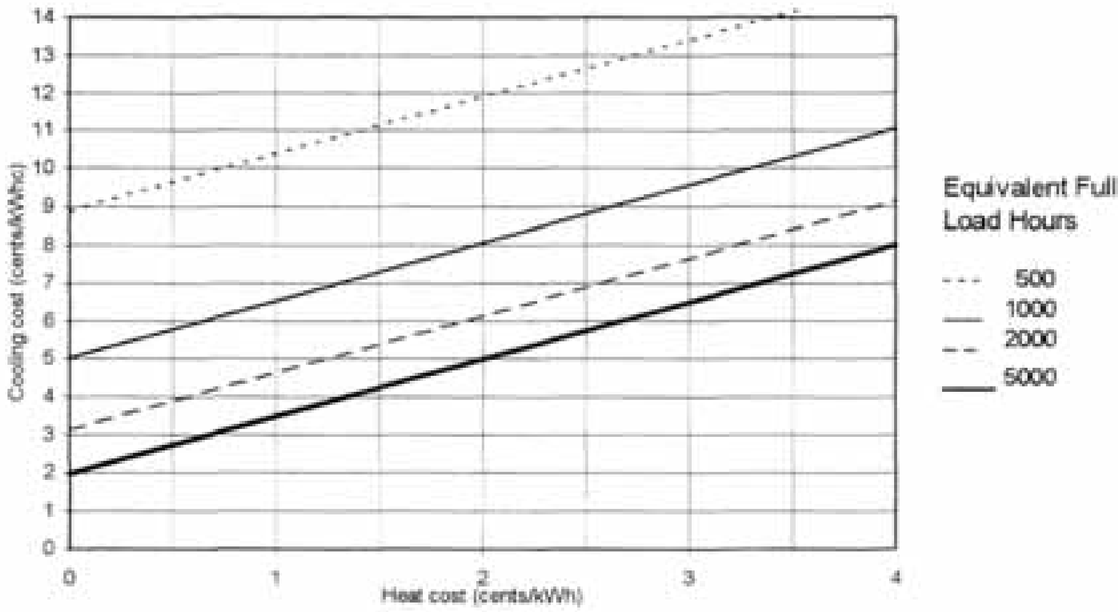

Figure 4.11 Representative economics for one-stage steam absorption chiller

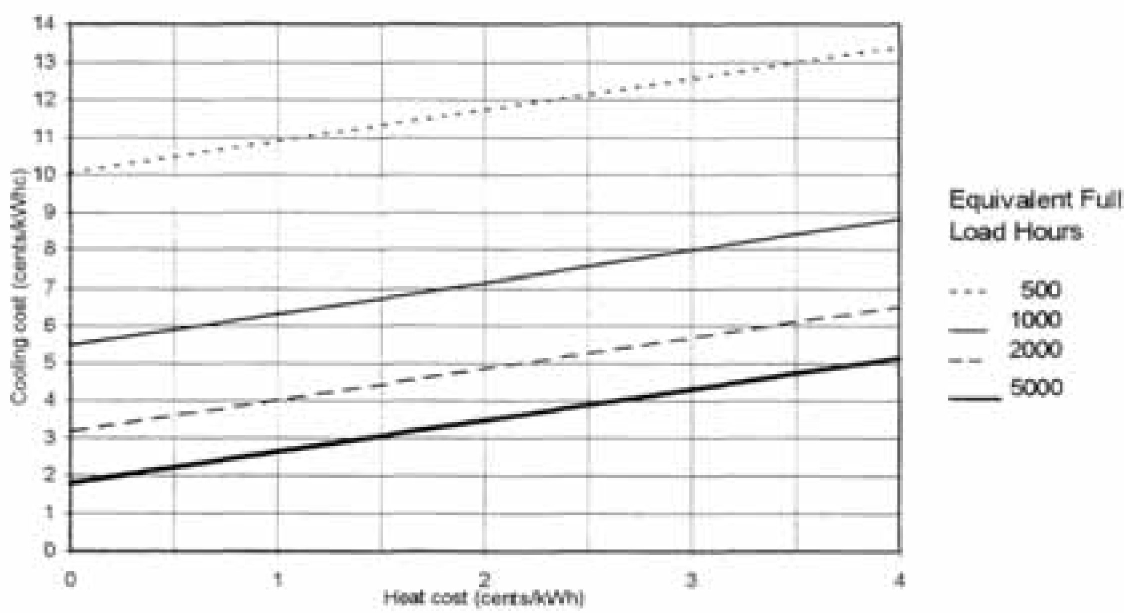

Figure 4.12 Representative economics for two-stage steam absorption chiller

		One-Stage Absorption	Two-Stage Absorption
A	Capital cost ($/kWc)	285	349
B	Real interest rate (%)	8%	8%
C	Capitalization period (years)	15	15
D	Capital recovery factor (%)	11.68%	11.68%
E	Water & chemical costs (cents/kWhc	0.93	0.68
F	Maintenance cost ($/kWc/year)	5.12	5.12
G	Electricity to chiller (kWhe/kWhc)		
H	Electricity to auxilliaries (kWhe/kWhc	0.056	0.042
I	Cost of electricity (cents/kWhe)	5	5
J	Heat to chiller (kWhth/kWhc)	1.50	0.83
K	Cost of heat (cents/kWhth)	0-4	0-4
L	Equivalent Full Load Hours	500-5000	500-5000

Formulas: Cooling price (cents/kWh$_C$) = (A x D x 100/L) + E + (F x 100/L) + [(G+H) * I] + (J * K)

Capital recovery factor (D) = $[B \times (1 + B)^C] / [(1 + B)^C - 1]$

Table 4.11 Assumptions for absorption chiller economics (5 MW$_C$ chillers in a 35 MW$_C$ plant)

4.5 COMPARATIVE ANALYSIS OF CENTRAL CHILLER TECHNOLOGIES

4.5.1 Efficiency

As discussed in **Section 4.2**, the efficiencies presented in this report reflect peak conditions rather than annual average conditions. **Figure 4.13** compares the input energy per kW$_C$ produced using the various chiller technologies, based on the following driving energy temperatures:

- Steam turbine centrifugal drive 11 bar steam (185 °C)
- One-stage steam absorption 2 bar steam (120°C)
- Two-stage steam absorption 8 bar steam (170°C)

The chiller system efficiency (including auxiliaries) can also be expressed as a Coefficient of Performance (COP), or kW$_C$ per total kW input energy (see **Figure 4.14**).

Although the heat-driven chillers appear inefficient, these chillers can use low-temperature thermal energy from CHP and can therefore provide overall system efficiencies comparable to electric drive chillers, as discussed in **Chapter 2**. The two-stage steam absorption chiller is significantly more efficient than one-stage steam absorption, requiring about 45% less input energy. However, because two-stage steam absorption requires higher temperature steam, it has a greater detrimental impact on electricity production when combined with CHP. The steam turbine drive chiller is slightly less efficient than the two-stage chiller and also requires higher temperature steam, so it has a low overall efficiency when combined with CHP. However, as discussed in **Chapter 6**, under certain circumstances, such as if low-cost non-CHP heat is available, steam turbine drive chillers can be attractive.

In addition to COP, operating flexibility and part-load performance are important performance considerations. Centrifugal chillers are generally easier to operate and provide more operating flexibility because they can be started up and shut down more quickly. Regarding part-load performance, the absorption chiller itself is better compared to the centrifugal chiller. Part-load performance of absorption chillers with a constant condenser temperature is basically linear down to 30% of capacity. This is superior to the part-load performance of electric centrifugal chillers with a constant condenser temperature, which at 30% of capacity has a chiller energy input per kW cooling output over 35% higher than at full capacity. With a variable condenser temperature the part-load performance of either type of chiller is improved.

However, chiller auxiliaries can significantly affect total chiller system efficiency under part-load conditions because these auxiliaries are generally run with one-speed motors. Poor part-load performance of chiller auxiliaries is more detrimental for absorption chillers because they require more heat rejection and therefore greater overall electric requirements for cooling tower operation. With absorption chillers there is the additional concern about avoiding condenser temperatures low enough to cause crystallization.

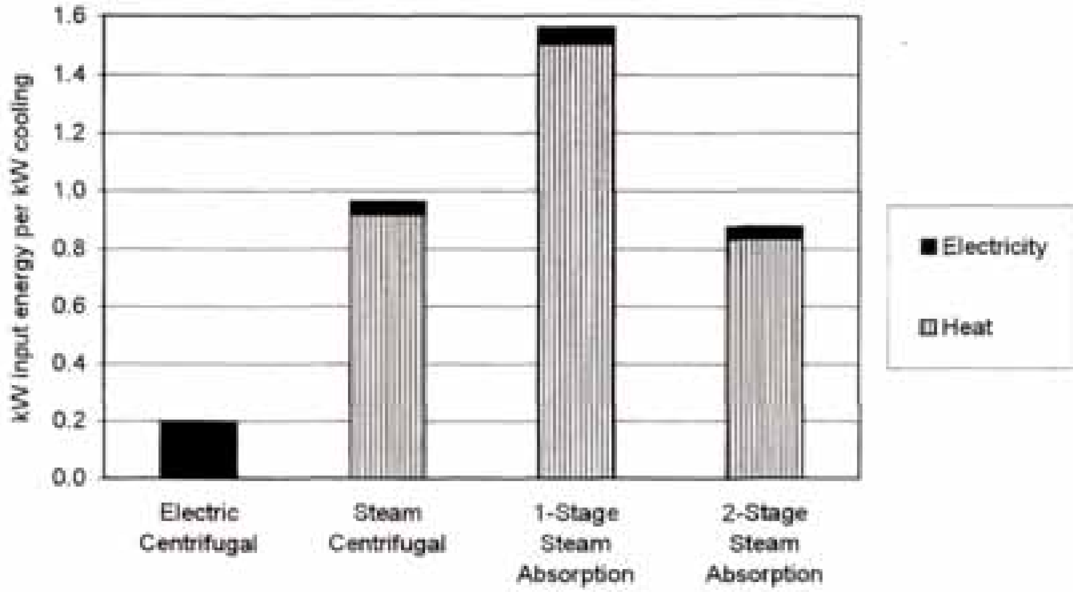

Figure 4.13 Representative input energy required per kW_c produced

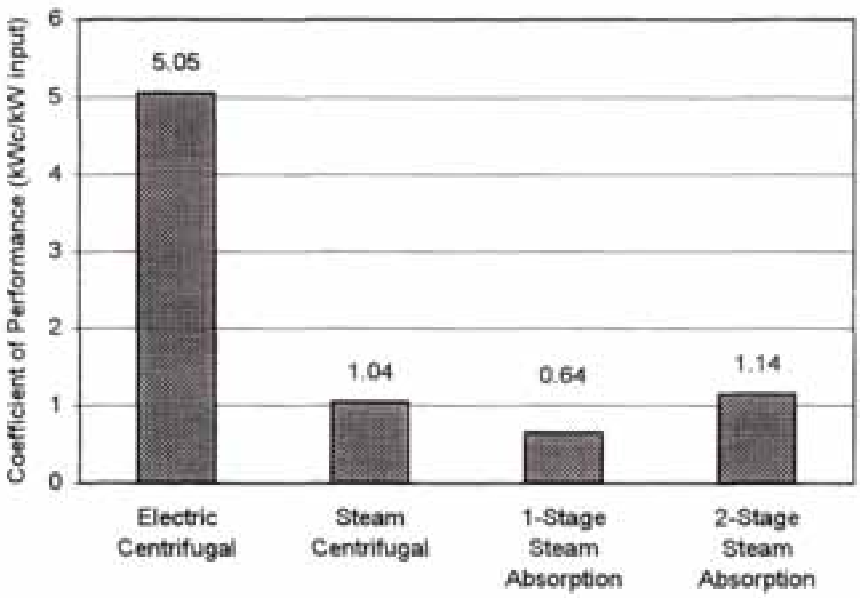

Figure 4.14 Representative chiller system COPs (includes auxiliaries)

4.5.2 Economics

Figure 4.15 illustrates the comparative capital costs of various types of chiller technologies, including installation, auxiliaries and building. The electric centrifugal chiller has a distinct capital cost advantage over the thermal drive technologies in this generalized comparison. Site-specific factors, such as additional costs to upgrade electrical service to power electric drive chillers, can change the comparative capital costs.

The impact of the higher capital costs for thermal drive chillers can be seen in **Figure 4.16** on the following page, which illustrates the component costs of the central chiller technologies for 500, 1000 and 5000 Equivalent Full Load Hours (EFLH) under a consistent assumption of 1 cent/kWh_{th} heat cost and 5 cents/kWh electricity cost.

The figure shows the significant impact of utilization on the cost of cooling. At low levels of utilization (500 EFLH), capital is by far the most significant cost. The cost of driving energy (electricity or heat) is the next most significant cost. This relationship is reversed as utilization is increased to baseload (5000 EFLH) levels, with the driving energy becoming the most significant cost. At 5000 EFLH total costs per kW_c drop to 25-30% of the value at 500 EFLH. Maintenance costs per ton of capacity are higher for steam turbine drive chillers compared to other technologies due to the unique maintenance requirements of the steam turbine equipment. Water and chemical costs for cooling tower make-up are higher for thermal drive technologies, although this cost component is relatively minor.

Different assumptions regarding the cost of capital and/or the amortizable life of the chillers could also have a significant impact on the results of a site-specific comparative analysis.

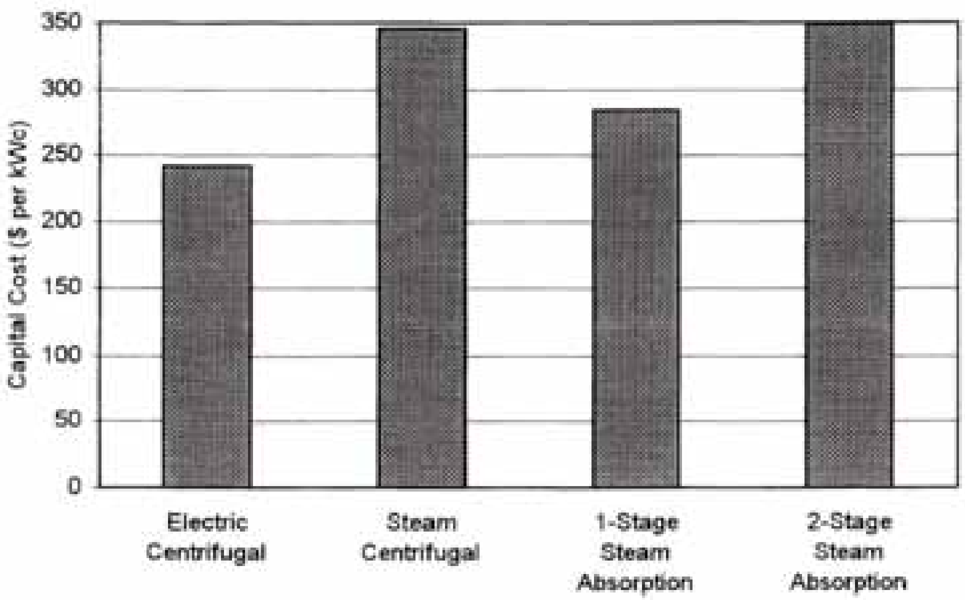

Figure 4.15 Comparative capital costs of chiller technologies, including installation, auxiliaries and building

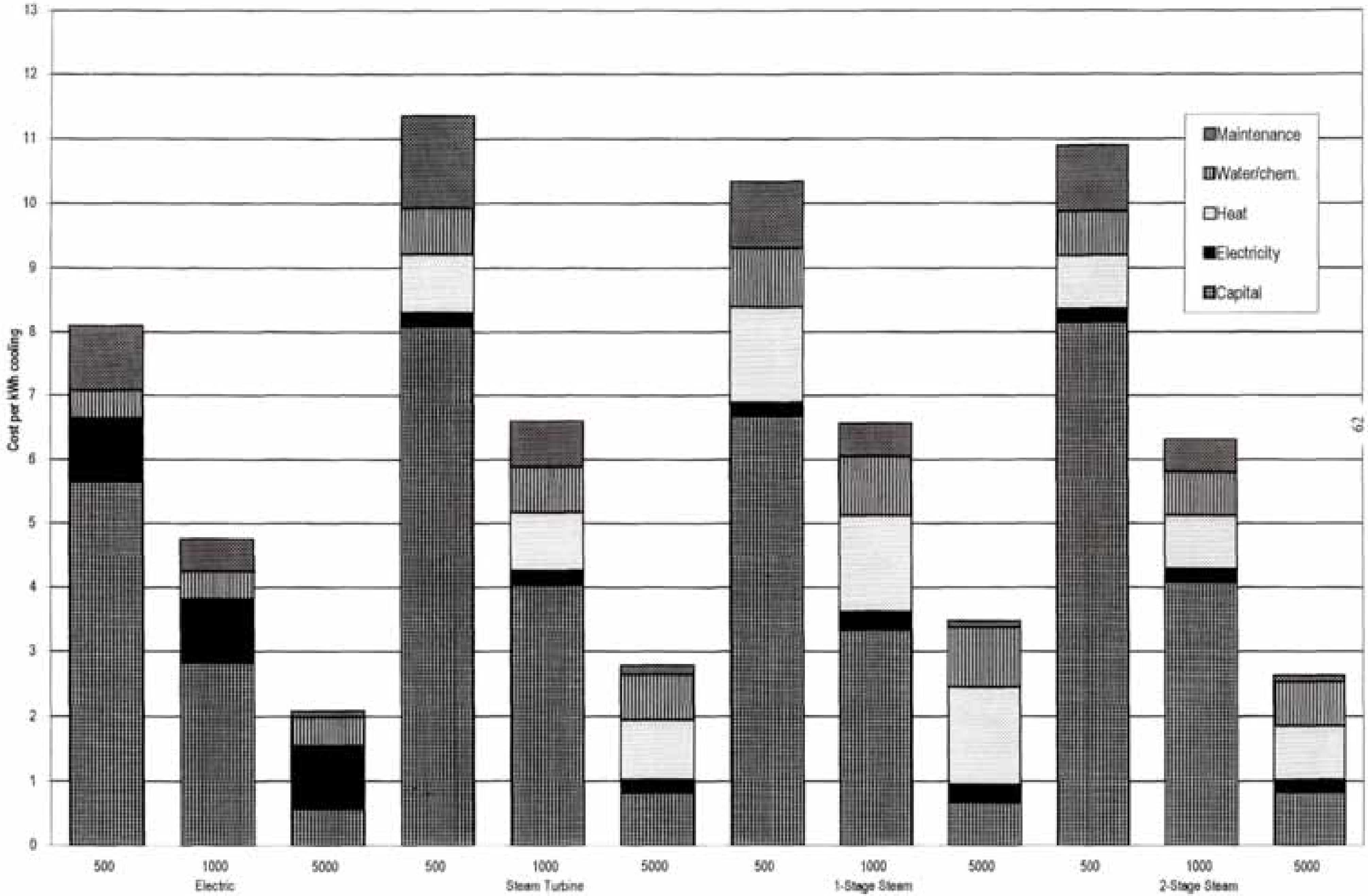

Figure 4.16 Component costs of chiller technologies assuming 1 cent/kWh heat and 5 cents/kWh electricity

Whereas **Figure 4.16** compares costs at static assumptions regarding the cost of heat and electricity, **Figures 4.17 - 4.20** compare the total costs, for 500, 1000, 2000 and 5000 EFLH, respectively, of thermal driven chiller technologies (expressed as cost per kWh of cooling) at a range of heat costs. Generally, two-stage absorption is the lowest cost option, followed by steam turbine drive. At low heat costs and low levels of utilization (under 1.5 cents/kW$_c$ at 500 EFLH and under 0.5 cents/kW$_c$ at 1000 EFLH), the relatively inefficient one-stage steam absorption can show a cost advantage over the other thermal driven chillers. As discussed in **Chapter 6**, the lower temperature requirements of one-stage absorption reduce the effective cost of the heat and make one-stage absorption more attractive in the context of integration with CHP.

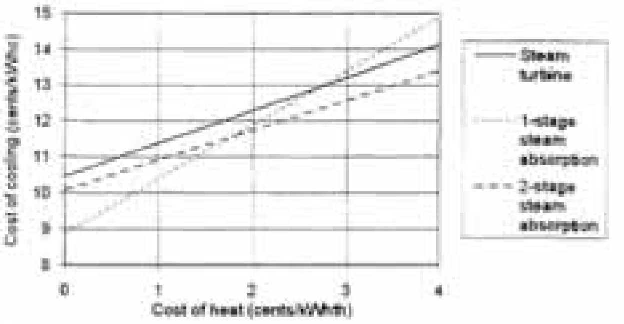

Figure 4.17 Comparative costs of heat-based chillers at 500 EFLH

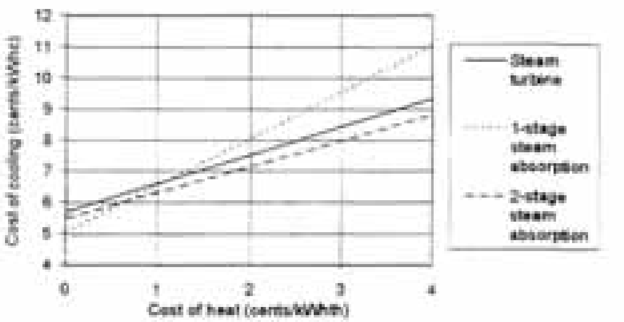

Figure 4.18 Comparative costs of heat-based chillers at 1000 EFLH

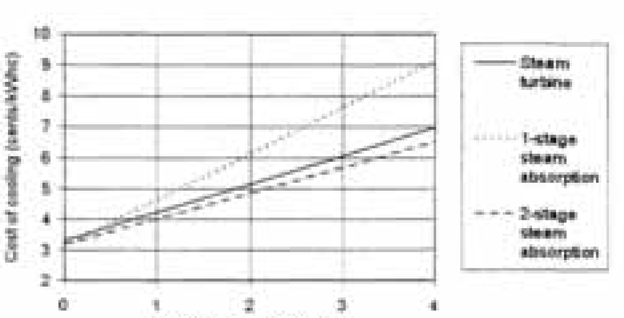

Figure 4.19 Comparative costs of heat-based chillers at 2000 EFLH

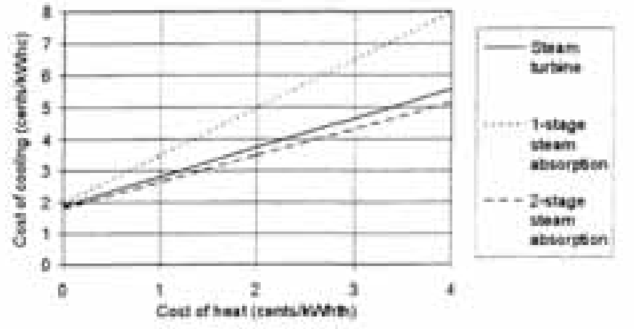

Figure 4.20 Comparative costs of heat-based chillers at 5000 EFLH

If electricity is sufficiently costly relative to the cost of heat for thermal drive chillers, the latter can compete effectively against electric chillers. **Figures 4.21-4.23** show the minimum electric price/heat price ratio required for the cost of thermal drive chiller output to equal the cost of electric centrifugal chiller output, for the heat cost range of 0-4 cents/kWh$_{th}$ and at various EFLH, based on the parameters previously presented. These graphs represent a given set of generalized assumptions, and many site-specific variables can affect the cost comparison in a particular situation.

A number of generalizations can be made based on the calculations illustrated in **Figures 4.21-4.23**:

- At 500 EFLH, thermal drive equipment can only compete with electric centrifugal chillers if the electric/heat price ratio ranges from 21 to 26 with heat costing 1 cent/kWh$_{th}$.
- At 1000 EFLH, steam absorption can compete with very low heat costs and high electric prices. The required electric/heat price ratio ranges from 15 to 18 with heat costing 1 cent/kWh$_{th}$.
- At 2000 EFLH, thermal drive chillers become more competitive, with the required electric/heat price ratio ranging from 11 to 16 with heat costing 1 cent/kWh$_{th}$.
- At 5000 EFLH, the required electric/heat price ratio drops to a range of 8 to 15 with heat costing 1 cent/kWh$_{th}$.

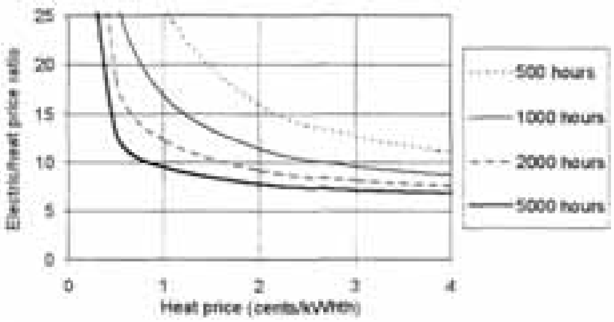

Figure 4.21 Electric/heat price ratio required for cost of *steam turbine chiller* output to equal the cost of electric centrifugal chiller output

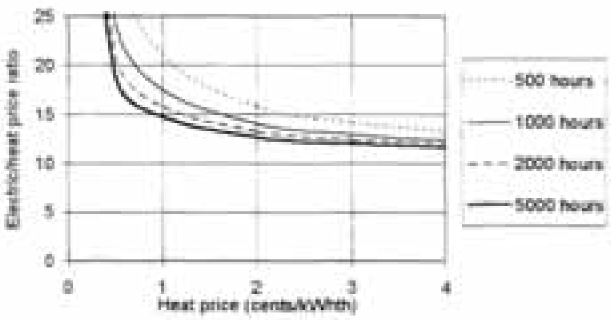

Figure 4.22 Electric/heat price ratio required for cost of *1-stage steam absorption chiller* output to equal the cost of electric centrifugal chiller output

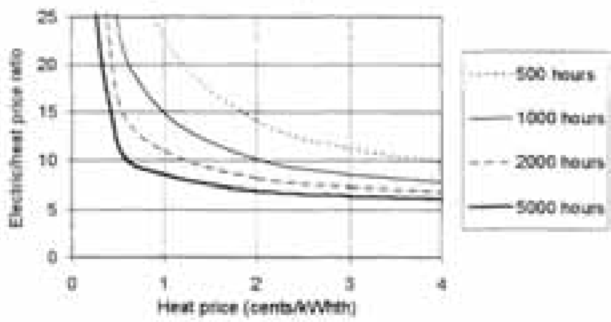

Figure 4.23 Electric/heat price ratio required for cost of *2-stage steam absorption chiller* output to equal the cost of electric centrifugal chiller output

4.6 ABSORPTION CHILLERS DRIVEN WITH DISTRICT HOT WATER

This section describes and quantifies the performance and economics of absorption chiller technologies driven with district hot water. For the reasons described in **Section 4.1**, this discussion is presented separately from the prior discussion of chillers used in central chilled water plants. For a review of relevant refrigerants, see the discussion in **Section 4.4.3**.

Interest is growing in the use of district heating systems to provide district cooling, either through *dispersed absorption* (absorption chillers in each building) or through *decentralized chilled water plants* (small district heat-driven absorption plants distributing chilled water). The generation of cooling through delivery of district heat increases utilization of existing plant and distribution systems and provides opportunities for increased service and new business. Although there are examples of steam district heat being used to provide district cooling, most of the current interest in using district heat for cooling is in district hot water systems. In addition, the low temperatures of district hot water present additional design considerations compared to steam absorption. Therefore, this section focuses on hot water absorption.

4.6.1 Description of Technology

Steam and hot water absorption chillers driven with district heat use a one-stage absorption cycle as illustrated in **Figure 4.6**. This is the district heat-driven technology most commonly implemented or considered for district cooling. Adsorption and dessicant (sorption) units are other thermal drive technologies which are capable of generating cooling.

In adsorption chillers the refrigerant (water) is bound in fluids or on the surface of a hygrospopic solid substance (e.g., silica gel) in the desorber. The heat of the driving energy vaporizes the refrigerant in the desorber, then the refrigerant vapor is condensed in the condenser. The condensed refrigerant is atomized in the evaporator, thereby cooling the cooling loop. The evaporated refrigerant flows to the collector and is bound on the surface of the hygrospopic substance. After saturation of the collector with the refrigerant the collector has to be switched to desorption. During desorption the hygrospopic substance is regenerated (dried) for operation as a collector. The operation is switched between collection and desorption in five-minute intervals.

Dessicant or sorption machines use a dessicating , or drying, material to remove moisture from air being conditioned. Dessicants can be solid (e.g., silica gel) or liquid (e.g., lithium chloride). The dehumidification process is essentially adiabatic, converting the latent energy of the water vaport to sensible energy of the air, resulting in an increase in the air temperature. However, dehumidification combined with heat rejection results in an overall cooling of the air.

4.6.2 Performance

Absorption cooling is a well-established technology. However, until recently absorption chillers were generally designed for steam and were poorly suited for use with low-temperature district heat. Most district hot water systems are designed for a variable temperature, with send-out temperatures during summer dropping to 70-90°C. In contrast, many one-stage absorption chillers are designed for 2 bar (120°C) steam.

Generally, the heat transfer surface area in the generator in a steam absorption chiller used for hot water will be undersized. Providing the required heat transfer surface area for chillers designed for steam results in higher investment costs per unit of cooling capacity when these chillers are used for hot water applications. This creates the potential for decreasing investment costs by designing and marketing chillers specifically for hot water applications, and in fact a number of manufacturers are now doing so. Further optimization of chiller designs for hot water applications may be possible relative to reducing capital cost and/or increasing the hot water temperature difference.

On the other hand, many single-effect steam absorption chillers are designed for low pressure steam (e.g., 2 bar), whereas pressures in hot water district heating distribution systems may be operating at peak pressures of 12 bar or more. Designing a chiller to operate under this higher pressure will tend to increase investment costs.

The performance of absorption chillers driven with hot water will vary depending on machine design, the hot water supply temperature and other factors. Key performance parameters include chiller capacity (which affects investment costs), COP (which affects the operating costs and environmental impacts) and the hot water temperature difference (which affects the district heating distribution pipe capacity).

Chiller capacity as a function of hot water supply temperature

The capacity of an absorption chiller will drop as the temperature of the driving energy is decreased, resulting in a higher investment cost per unit of cooling capacity because more heat transfer surface is required to produce a given amount of cooling. This can also result in higher investment costs for additional building space and structural reinforcement due to the greater size and weight of the chiller per unit of cooling capacity.

The extent of this drop in capacity depends on the specific machine design. Capacity derate curves for one-stage steam absorption chillers were shown in **Figure 4.9** for steam pressures down to 0.3 bar. **Figure 4.24** shows the capacity derate of absorption chillers as a function of the hot water supply temperature. The capacity for the different chillers in the figure has been equalized to 100% at 100°C hot water supply temperature and is based on a 10°C temperature difference.

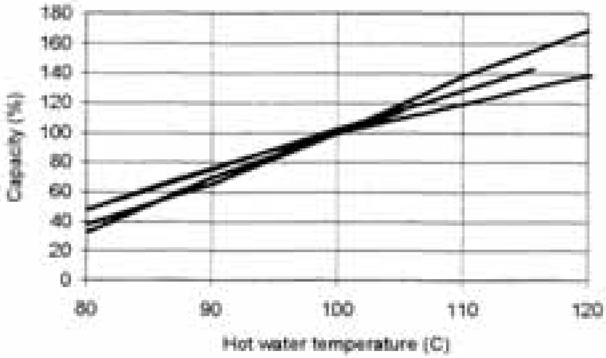

Figure 4.24 Capacity derate for different chillers at a range of hot water supply temperatures, assuming 10°C hot water temperature difference [4-3, 4-14]

In planning a system of decentralized chilled water plants using absorption chillers driven by the hot water district heating system, Gothenburg Energy examined the impacts on investment costs of installing chiller capacity to utilize the normal 75°C summertime hot water, and concluded that it was more economical to increase the summer operating temperature of the district heating system to 100 °C. It is significant that in this system CHP is not a source of heat during the summer, although it is an important source of heat during the rest of the year. During summer, waste incineration and industrial waste heat are key sources of thermal energy. [4-15]

Chiller capacity as a function of hot water temperature difference

For an absorption chiller driven with hot water, both the supply temperature and the temperature difference will affect the capacity. **Figure 4.25** shows the capacity derate for a chiller at different hot water supply temperatures and temperature differences. Hot water at temperatures of 90° C and 100°C would result in capacity derates to 45% and 65%, respectively, of the capacity with 120°C driving energy, assuming a 5°C temperature difference. If the temperature difference is 10°C, the capacities drop further, to 40% and 57%, for hot water supply temperatures of 90° C and 100°C.

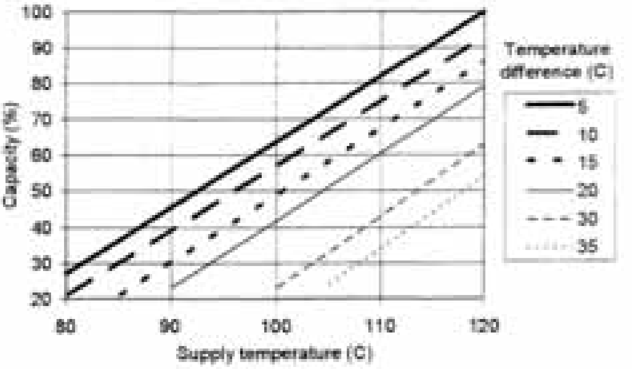

Figure 4.25 Capacity derate for different supply temperatures and temperature differences for steam absorption chiller driven with hot water [4-3]

If steam driven absorption chillers are converted to hot water, the capacity derate using hot water can be estimated by calculating the average of the supply and return temperatures for the hot water absorption chiller. This is a good approximation of the saturation temperature of equivalent steam pressure for a steam driven absorption chiller. Thus, for example, for hot water supply/return temperatures of 90/80°C, the average is 85°C and the equivalent steam pressure is 0.58 bar, as shown in **Figure 4.26**. This equivalent steam pressure can then be applied to the capacity derate chart for the particular steam chiller, such as **Figure 4.9**.

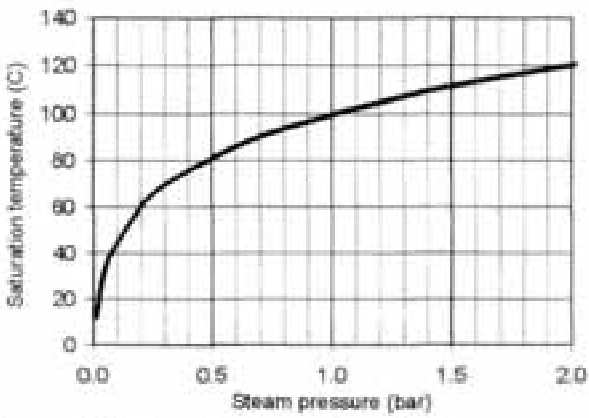

Figure 4.26 Saturation temperature of steam at pressures up to 2 bar

COP as a function of hot water supply temperature

Apart from affecting the chiller capacity, the hot water supply temperature will also affect the chiller COP. COP data (chiller only) from manufacturers of absorption chillers marketed for hot water applications are summarized in **Table 4.12**. The COP values are based on a leaving chilled water temperature of 7°C and a condenser entering water temperature of 30°C. Given the significant differences in values for the different manufacturers, the choice of chiller manufacturer would appear to be at least as important as the choice of supply temperature.

	Hot water supply/return temperatures (C)	
	95/80	105/95
Manufacturer		
1	0.75	not available
2	0.70	0.75
3	0.69	0.71

Table 4.12 COP data from manufacturers of hot water absorption chillers and other sources [4-14]

The impact on COP due to different hot water supply temperatures is shown in **Figure 4.27** for two chillers being used with hot water, under the base case conditions of 6°C chilled water temperature, 30°C condenser entering water temperature and 10°C hot water temperature difference. The difference in the effect of the supply temperature on the COP could be explained by differences in the design of the chillers, but may also be affected by how the two manufacturers present performance data.

Electric usage for auxilary equipment will vary slightly depending on the COP of the particular chiller, the hot water temperature and other design conditions. As shown in **Figure 4.27**, the steam chiller driven with 95°C hot water will have a COP of about 0.67 -- the same as for 2 bar (120°C) steam. Auxiliaries would be expected to have the same electric requirements as shown for one-stage steam absorption in **Table 4.7** (0.06 kW_e/kW_c), for a total chiller system COP of 0.64.

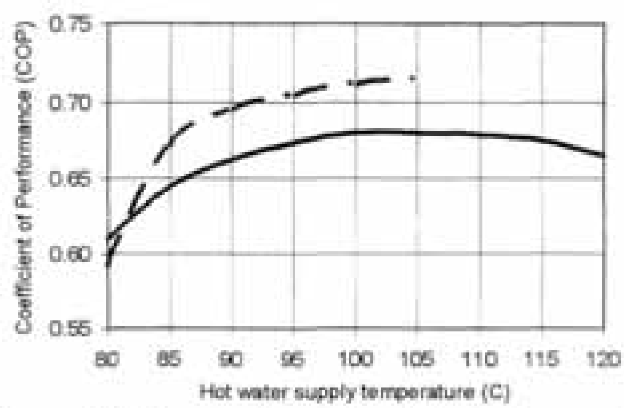

Figure 4.27 Impact of variations in hot water supply temperature on COP for two chillers being used with hot water [4-16, 4-17]

Distribution capacity as a function of hot water temperature difference

District hot water pipes are typically designed for a hot water temperature difference of 30-50°C under peak heating conditions, whereas hot water absorption generally produces hot water temperature differences of 15°C or less. Depending on the relative size of heating and cooling loads, this can result in pipe capacity constraints.

For example, in Seoul, Korea, where the average cooling demand per square meter of building space is 15% higher than the average heating demand, district hot water pipes used for dispersed absorption are constraining expansion of cooling service. In contrast, in the Netherlands, where the average cooling demand is 40-45% lower than the average heating demand, use of district heating systems for absorption cooling is less of a problem. However, while the average demands for heating and cooling can be used as a capacity criterion for the main distribution pipes, problems can still arise in certain parts of the main distribution system and the service pipes. In downtown areas with mainly office buildings, the ratio of cooling to heating is higher than average, and distribution pipes sized for heating can still be a restriction to absorption cooling.

The pipeline capacity constraint becomes more severe with lower temperature district heating systems, because the hot water temperature difference and/or the chiller capacity is reduced with lower hot water supply temperatures, as shown in **Figure 4.25**.

Condenser water temperature

Regardless of the type of chiller, capacity and efficiency will decrease if the condenser water temperature is increased. For a central chiller plant, regardless of chiller type, the design condenser water temperature usually is proportional to the design ambient wet bulb temperature or, if available, river or lake water temperature. Normally, the approach temperature between the design ambient wet bulb temperature and the condenser water temperature is about 5°C. To limit the number of variables, the earlier performance comparisons for chillers to be used in central plants were based on one condenser water temperature.

With dispersed chillers, the design condenser water temperature can vary due to space limitations and other constraints for local building cooling towers (e.g., esthetic or visibility concerns due to cooling tower drift). These constraints may result in higher condenser temperatures due to undersized cooling towers. Therefore, some discussion is warranted regarding the impact of condenser temperatures on capacity and COP.

Higher condenser water temperatures reduce the chiller capacity, as shown in **Figure 4.28** for one chiller at different hot water and condenser water temperatures.

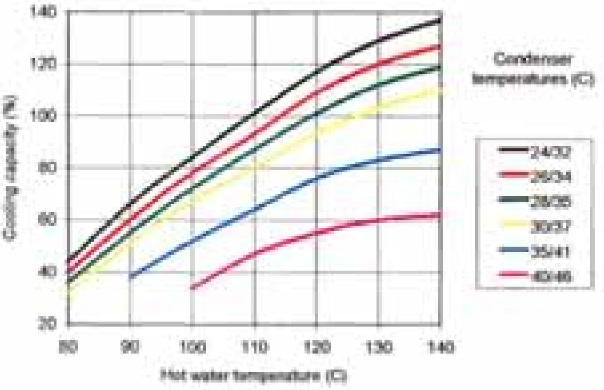

Figure 4.28 Impact of hot water supply temperature on absorption chiller capacity at different condenser temperatures (chilled water 6°C, hot water temperature difference 10°C) [4-17]

Higher condenser temperatures also affect COP, particularly at lower hot water driving temperatures. If the temperature of driving energy is relatively high, the COP is relatively insensitive to the condenser water temperature. However, with lower driving temperatures the COP becomes sensitive to changes in condenser water temperature, and this sensitivity increases with lower driving temperatures. **Figure 4.29** shows the COP of an absorption chiller used with a range of hot water supply temperatures and for different condenser temperatures.

Other chillers will exhibit a different pattern of performance under varying conditions, based on the particular machine design. Note that for this chiller at lower condenser inlet temperatures (under 30°C), the COP actually increases slightly as the hot water inlet temperature drops from a design condition of 120°C to 100-105°C, where it peaks at 0.68 and 0.71 for condenser inlet temperatures of 30°C and 25°C, respectively.

Part-load performance

Part-load performance of absorption chillers driven with hot water is similar to performance with steam, as illustrated in **Figure 4.8**.

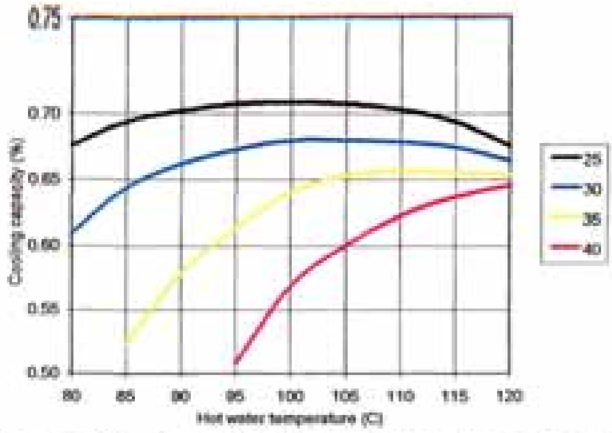

Figure 4.29 Impact of hot water supply temperature on absorption chiller COP at different condenser temperatures (chilled water 6°C) [4-17]

4.6.3 Economics

Capital Costs

Capital costs for absorption chillers driven with hot water can be estimated based on costs for steam absorption chillers, adjusting for: 1) the capacity derate when hot water is used instead of the normal design condition of 2 bar steam; and 2) cost differences due to economies of scale in the chiller and plant components. Estimates are made for *decentralized chilled water plants* (assuming generation of chilled water in 5 MW_c absorption chiller plants) and for *dispersed absorption* (assuming 1 MW_c absorption chiller plants in buildings).

As indicated in **Figure 4.25** for a representative steam chiller, 100% capacity is reached with 120°C driving temperature and 5°C temperature difference. With a 10°C temperature difference the capacity would be reduced to 93% of full capacity. Driving the chiller with 95°C hot water with a 10°C temperature difference would reduce the capacity to about 50% of full capacity.

Cost adjustments were also made to account for the diseconomies of scale with smaller chiller and plant sizes likely in decentralized chilled water plants and dispersed absorption plants compared to centralized steam absorption chiller plants. Assumptions for chiller and plant sizes for each type of facility are summarized in **Table 4.13**.

Facility Type	Chiller size (kWc)	Plant size (kWc)
Centralized steam absorption	5,280	35,200
Decentralized chilled water plant	1,250	5,000
Dispersed absorption plant	500	1,000

Table 4.13 Chiller and plant size assumptions for economic calculations

Cost adjustments were made for the bare chiller and various plant components, generally using a 0.8 exponent factor as illustrated in the following example formula:

S^A = Size of chiller A in kW_c
S^B = Size of chiller B in kW_c
C^A = Cost of chiller A in $\$/kW_c$

Cost of chiller B in $\$/kW_c$ =

$$C^A * (S^A/S^B) * (S^B/S^A)^{0.8}$$

Based on the derate for 95°C hot water, and adjusting for size differences compared to the bare chiller capital cost presented in **Section 4.4** for a 5 MW_c one-stage steam absorption chiller, bare chiller capacity for 95/85°C hot water would cost \$166 per kW_c for a 1,250 kWc chiller in a decentralized chilled water plant and \$200 per kW_c for a 500 kWc dispersed absorption chiller.

These values are reasonably consistent with bare chiller capital cost values provided by manufacturers of chillers designed for hot water, as summarized in **Table 4.14**.

Manufacturer	Size (kWc)	\$ / kWc
1	140-560	303
2	400-5000	200
3	970	113

Table 4.14 Capital costs for bare chillers designed for 95°C hot water [4-14]

Given the wide range of the data in **Table 4.14**, capital cost assumptions will be derived based on adjusted costs of steam absorption chillers. **Table 4.15** summarizes total installed capital costs per kW_c for representative hot water absorption chiller capacity driven with 95°C hot water, including auxiliaries and building space. No costs are included for chilled water distribution for the decentralized chilled water plant.

Although these capital costs appear high relative to the other chiller technologies discussed earlier, by using the district heating distribution system to deliver cooling energy, significant savings in chilled water distribution costs are possible: either their complete elimination (in dispersed absorption chillers in individual buildings) or large reductions (in decentralized chilled water plants).

Operation and Maintenance

Operation and maintenance costs are discussed generally in **Section 4.3.4**. Key assumptions regarding make-up water and costs for hot water absorption chillers are shown in **Table 4.16**, assuming 95°C hot water supply.

Make-up water consumption (liters/kWhth)	3.50
Make-up water consumption (liters/kWhc)	8.76
Water cost (\$/1000 liters)	0.79
Chemical cost (\$/1000 liters)	0.26
Total water and chemical cost (\$/1000 liters)	1.06
Total water and chemical cost (cent/kWhc)	0.93

Table 4.16 Makeup water for hot water absorption chiller cooling towers

Water and chemical costs will change slightly with higher or lower driving temperature, depending on the impact on COP. For example, for the steam chiller depicted in **Figure 4.27** above, if the driving temperature is increased from 90°C to 100°C, the COP would increase from 0.665 to 0.68. The amount of heat rejected per unit of cooling output is equal to the sum of the cooling output (i.e., the thermal energy removed from the building) plus the thermal energy driving the absorption process. Therefore, in the example just cited, the percentage decrease in the heat rejected is:

$$\frac{(1.665/0.665) - (1.68/0.68)}{(1.665/0.665)} = 1.3\%$$

	Dispersed Absorption	Decentralized chilled water
Bare chiller	200	166
Chiller pumps (primary)	7	6
Cooling tower, condenser pumps	77	56
Mechanical installation and piping	90	58
Instrumentation and control	14	9
Electrical	18	12
Building and foundation	23	40
Subtotal	429	347
Project management (5%)	21	17
Engineering (10%)	43	35
Contingency (10%)	43	35
TOTAL	537	433

Table 4.15 Representative capital costs for absorption chiller capacity driven with 95/85°C district hot water (\$/kWc)

Overall Economics

Figures 4.30 and 4.31 illustrate representative economics of hot water absorption (95/85°C) for dispersed absorption and decentralized chilled water plants, respectively, at various heat prices and Equivalent Full Load Hours (EFLH). Key assumptions are detailed in **Table 4.17**. Auxiliaries were assumed to be powered by electricity costing 5.0 cents/kWh$_e$. Total costs per unit of cooling were not significantly affected by variations in the cost of electricity. Compared to maintenance costs for centralized chilled water plants, maintenance costs per ton were assumed to be 20% higher for 1,250 kW$_c$ chillers in decentralized chilled water plants and 100% higher for 500 kW$_c$ chillers in dispersed absorption plants.[4-4]

The cost of hot water absorption cooling will depend on many site-specific variables. As indicated earlier, this approach can provide substantial savings in distribution costs. Considering only plant-related costs, generating cooling with district hot water will be quite sensitive to utilization due to the relatively high capital costs. Absorption chillers driven with district heating will tend to have lower utilization than other chillers potentially integrated with CHP because this approach will generally be used in regions with well-developed district heating systems, which tend to areas with cooler climates and relatively low cooling requirements. In addition, in the dispersed absorption approach the utilization of the chiller will be limited to the individual building's requirements, whereas in decentralized chilled water plants it is possible to use a given chiller to provide baseload cooling, thereby increasing the utilization hours and decreasing the impact of high capital costs.

Based on this generalized analysis, decentralized chilled water plants with a relatively high utilization factor (2000 EFLH) would cost 4-10 cents/kWh$_c$, based on heat costs ranging from 0-4 cents/kWh$_{th}$. At 1000 EFLH this range increases to 7-13 cents/kWh$_c$. Corresponding ranges for dispersed absorption are 5-11 cents/kWh$_c$ and 8.5-14.5 cents/kWh$_c$ for 2000 and 1000 EFLH, respectively.

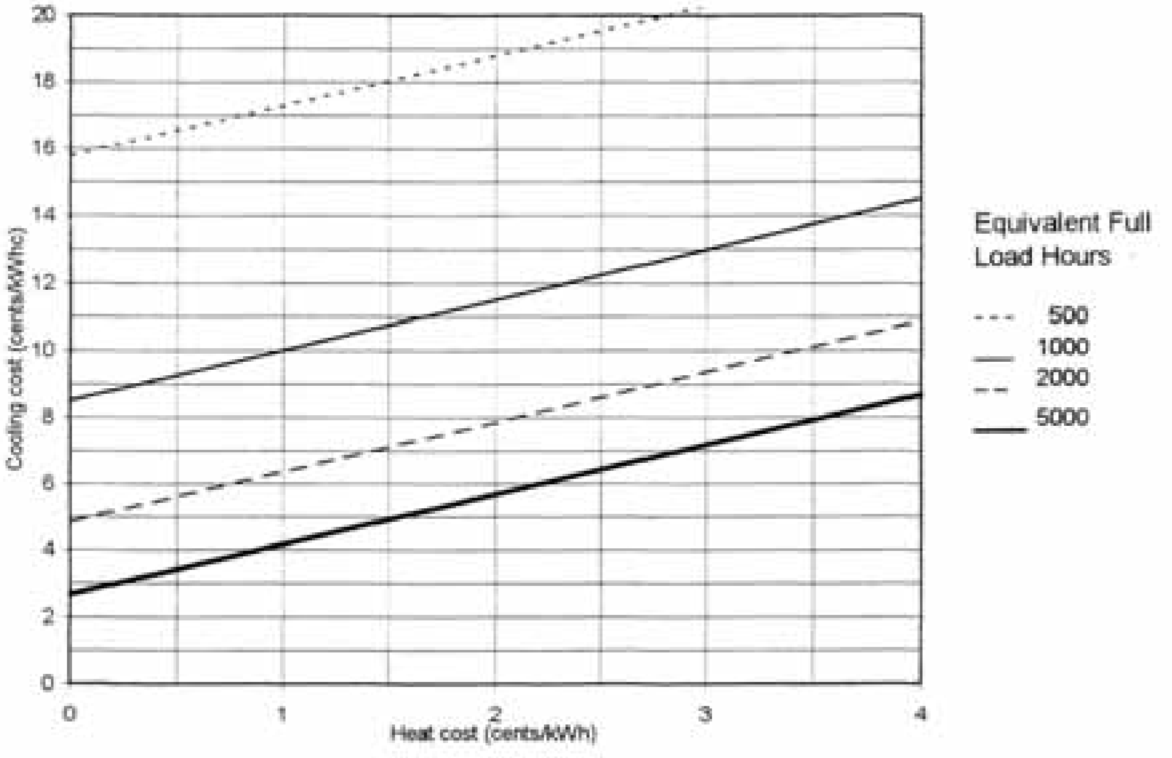

Figure 4.30 Representative economics for dispersed hot water absorption (500 kW$_c$ chillers in 1,000 kW$_c$ plant)

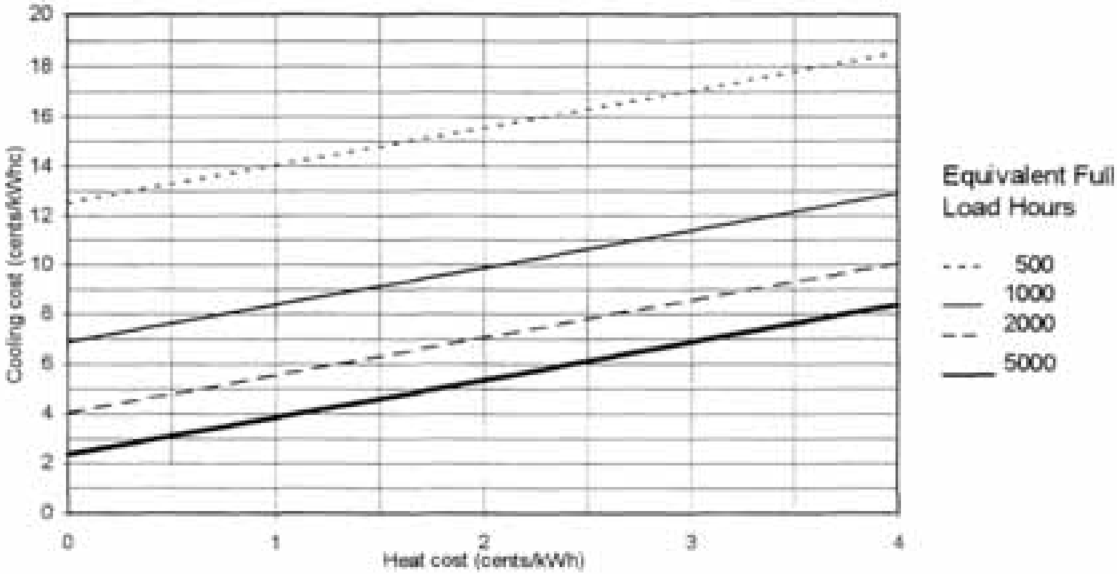

Figure 4.31 Representative economics for decenntralized chilled water plant using hot water absorption (1,250 kW_c chillers in 5,000 kW_c plant)

		Dispersed Absorption	Decentralized Chilled Water
	Chiller unit size / plant size (MWc)	0.5/1.0	1.25/5.0
A	Capital cost ($/kWc)	537	433
B	Real interest rate (%)	8.00%	8.00%
C	Capitalization period (years)	15	15
D	Capital recovery factor (%)	11.68%	11.68%
E	Water & chemical costs (cents/kWhc)	0.93	0.93
F	Maintenance cost ($/kWc/year)	10.24	6.14
G	Electricity to chiller (kWhe/kWhc)		
H	Electricity to auxilliaries (kWhe/kWhc)	0.056	0.056
I	Cost of electricity (cents/kWhe)	5	5
J	Heat to chiller (kWhth/kWhc)	1.50	1.50
K	Cost of heat (cents/kWhth)	0-4	0-4
L	Equivalent Full Load Hours	500-5000	500-5000

Formulas: Cooling price (cents/kWh_c) = $(A \times D \times 100/L) + E + (F \times 100/L) + [(G+H) * I] + (J * K)$

Capital recovery factor (D) = $[B \times (1 + B)^C] / [(1 + B)^C - 1]$

Table 4.17 Assumptions for hot water absorption chiller economics

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Chapter 5
Fundamental of District Heating and Cooling

5.1 INTRODUCTION

This chapter briefly describes some aspects of overall district heating and cooling design which are relevant to integration of district cooling with CHP and district heating. **Section 5.2** discusses heating and cooling loads using the example of St. Paul, Minnesota, USA. **Section 5.3** briefly addresses distribution considerations. **Section 5.4** discusses the role of thermal storage.

5.2 HEATING AND COOLING DEMAND

5.2.1 Heating

Depending on the country and location, different design ambient temperatures and parameters are used for designing the heating system of a building. The discussion in this section uses the design values for St. Paul, Minnesota, USA as examples.

Building heating systems in North America are usually designed to meet the 97.5% dry bulb temperature (the temperature which is higher than only 2.5% of the hours in December through February -- a total of 2,160 hours). [5-1] For St. Paul this design outdoor temperature is -24°C. An indoor temperature of about 20-22°C is usually desired, but the building heating system usually only has to provide heating up to 18-19°C due to internal heat gain from people, electric equipment, etc.

Depending on the building construction (insulation, heat capacity and glass area), outdoor temperatures below the 97.5% point will affect the indoor temperature differently. The indoor temperature in "heavy" buildings with a high heat capacity will not be affected as quickly as the indoor temperature in "light" buildings with poor insulation. For light buildings the 99% dry bulb temperature (-27°C in St. Paul) could therefore be a better design parameter.

Heating degree-days are used to calculate the heating energy required. The number of heating degree-days for a particular day is defined as the difference between the desired indoor temperature and the average daily ambient temperature. Normally, degree-days below an outdoor temperature of 18°C are used in North America. The indoor temperature will actually be greater than this due to the internal heat gains. The annual heating degree-days for St Paul is 4,434. [5-2]

The district heating system will include different types of buildings with different usage patterns for space heating and domestic hot water. Due to these different patterns, the peak load for the district heating system will be lower than the sum of the peak loads for each building. The ratio between the system peak load and the individual peak loads, usually called the "diversification factor," is normally in the range of 0.8.

A study of an area of 72 identical single family houses in Sweden showed a diversification factor of about 0.7. [5-3] Because of the less uniform usage of space heating and domestic hot water and the higher usage of domestic hot water in residential buildings compared to offices and stores, the residential load should be more diversified. The study also showed that the maximum diversification (lowest diversification factor) was rapidly reached when 20 or more buildings were connected to the same system.

The equivalent full load hours (EFLH), defined as the annual energy consumption divided by the peak demand, are estimated to be 1,700 for offices and stores and 1,900 for apartments and hospitals in St. Paul. With an average of about 1,760 hours for all individual buildings, and a diversification factor of 0.8, EFLH for the overall district heating system is 2,200.

An estimated load duration curve for the system, based on empirical data from operating systems and meteorological data for St Paul (monthly degree-days, a design outdoor temperature of -29°C and about 20% degree-day-independent load) is shown in **Figure 5.1**. The lower design outdoor temperature used (1% dry bulb minus 2°C) compared to the design of individual buildings is a safety factor to ensure that heat can always be provided to the buildings. The EFLH using this method are 2,200, which is comparable to the actual value. An exact value for a new system cannot be accurately anticipated due to the multiplicity of buildings with different standards and different human behaviors involved.

Figure 5.1 shows the time duration for different heat loads in the system. **Figure 5.2** shows the percentage of the annual energy production that can be supplied by heat sources of different sizes. Heat sources sized at 20% or 50% of the total heat demand can, for example, provide about 55% or over 90%, respectively, of the annual heat production.

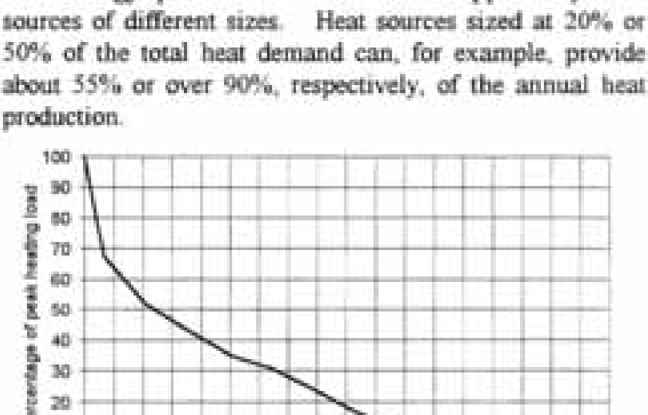

Figure 5.1 District heating load duration curve for St Paul, USA

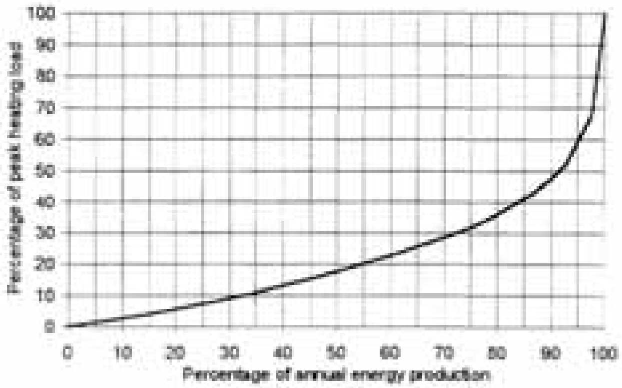

Figure 5.2 Energy distribution based on load duration in Figure 5.1

5.2.2 Cooling

Building cooling systems are usually designed to meet the 1% dry bulb temperature (the temperature which is below only 1% of the hours in June through September -- a total of 2,928 hours). [5-1] For St. Paul this outdoor design temperature is 33°C. The design of the cooling system must also account for humidity (wet-bulb temperature), solar heat gain and internal heat gain from people and equipment.

Cooling degree-days can be used to calculate the cooling energy required. The cooling degree-day is normally defined as the temperature above 18°C. Due to internal heat gains, especially in offices, degree-days above an outdoor temperature of 10°C may in some cases be a more appropriate basis for calculating the energy required. Depending on the base temperature used, the cooling degree-days for St. Paul will be 369 with 18°C base temperature and 1,497 with 10°C base temperature.

As with the district heating system, the district cooling system will include different types of buildings with different usage patterns. The diversification effect will be smaller, though, because the main load will be from offices and stores with similar usage patterns. Due to the more uniform usage patterns and the lack of a cooling consumption similar to the domestic hot water, the diversification factor will be higher compared to district heating. Very little research has been done on diversification in district cooling systems but a recent survey of 13 central chilled water systems showed a range in diversification factors from 46% to 100%, with an average of 82% and a median of 86%. [5-4] Until more data are available, a conservative diversification factor of 0.9-0.95 is recommended for initial calculations.

The cooling equivalent full load hours in St. Paul are estimated to average 1,100 for all individual buildings in St. Paul. This is a cautious estimate and the EFLH for some individual buildings can be as high as 2,000 hours. With a diversification factor of 0.9-0.95 the EFLH for the district cooling system is around 1,200 hours. The EFLH will also be affected by the rate structure. A relatively high demand charge will encourage customers to lower their individual peak demands, thereby increasing the EFLH.

An estimated load duration curve for the system in St. Paul, based on empirical data from operating cooling systems in St. Paul and Minneapolis is shown in **Figure 5.3**, and represents 1,200 EFLH. As with the heating load duration curve, the cooling load duration curve for other locations can be estimated based on meteorological data such as monthly degree-days, design outdoor temperature and degree-day-independent load such as computer cooling. **Figure 5.4** shows the percentage of the annual energy production that can be supplied by cooling sources of different sizes.

For district cooling, hourly load profiles for the peak "design day" and for other load conditions is more important than the annual load curve for optimizing the system and deciding how to displace production equipment and utilizing chilled water storage, as discussed further in **Section 5.4.1**. Detailed analysis of the projected coincidence of heating and cooling loads, if usable data are available, can aid significantly in system optimization.

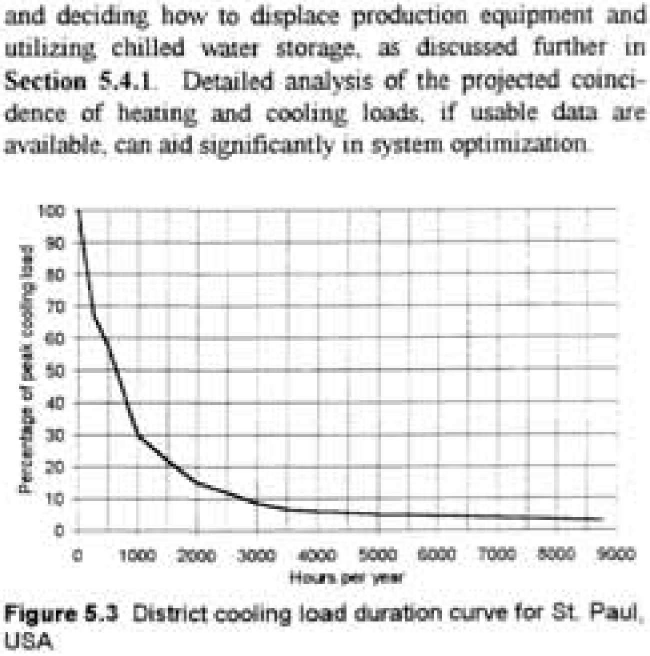

Figure 5.3 District cooling load duration curve for St. Paul, USA

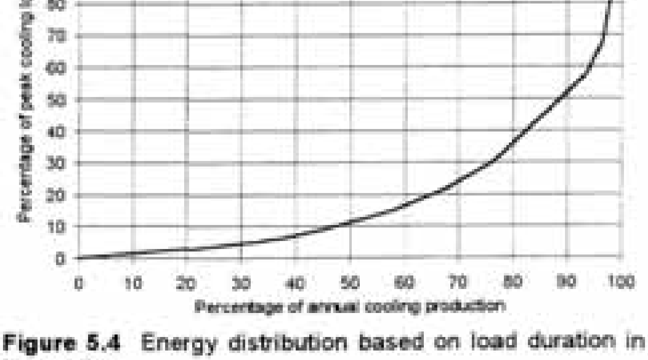

Figure 5.4 Energy distribution based on load duration in Figure 5.3

The heating and cooling load duration curves can then be illustrated in the same graph, with the cooling peak at the right side and the heating peak at the left side, as shown in **Figure 5.5**. (This is based on 120 MW peak heating demand and 63 MW peak cooling demand, assuming a mature heating market penetration and relatively low cooling market penetration. A higher ratio of cooling to heating would result from comparable assumptions regarding market penetration.)

The implications for CHP thermal load can then be examined. For example, **Figure 5.6** shows the combined thermal load curve assuming all cooling is provided with one-stage absorption chillers.

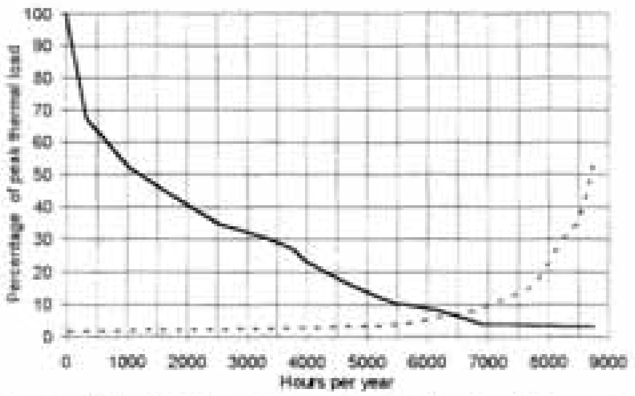

Figure 5.5 Heating and cooling load duration curves (heating peak 120 MW and cooling peak 63 MW)

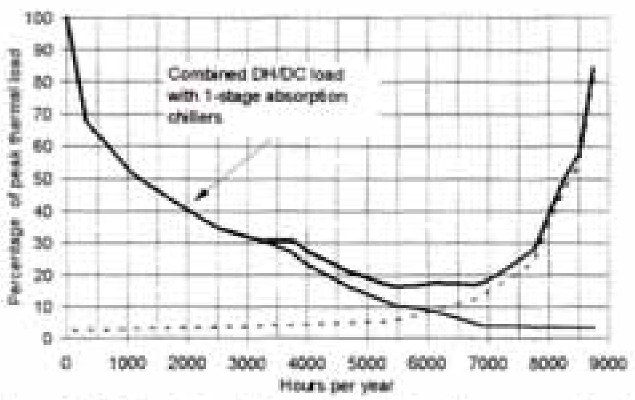

Figure 5.6 Combined thermal load duration curve based on Figure 5.5 assuming one-stage steam absorption chillers

5.3 DISTRIBUTION

The distribution medium in district cooling systems is generally chilled water with a supply temperature of between 5 and 9°C and a return temperature between 12 and 15°C. Distribution pipe diameters and pumping energy requirements can be reduced through distribution of lower-temperature water. Further reductions are possible through additives which depress the freezing point of water, as described in **Section 7.3**.

Research on the generation and distribution of ice slurries holds promise for even more significant reductions in pipe diameters and pumping energy requirements. Ice slurries are suspensions of small ice crystals of approximately 20% by weight. An experimental ice slurry district cooling system is now operating in Ottawa, Canada serving government buildings.[5-5] Still unresolved are a variety of technical challenges, including avoiding plugging in branch lines in district cooling distribution systems.

5.4 THERMAL STORAGE

Thermal storage can be an important strategy for optimizing a CHP/district heating/district cooling system by increasing equipment utilization and maintaining a more even CHP thermal load. The following discussion is a brief introduction to the basics of cool storage and heat storage.

5.4.1 Cool Storage

The variation between maximum and minimum loads for cooling is much greater than for heating. Building cooling systems are usually operated more on/off than heating systems. During nighttime when the ventilation air to an office can be shut off, the outdoor temperature is lower and there is less internal heat gain, so the cooling system can be shut off. In contrast, a heating system still must be operated at night. With the on/off operation of building cooling systems, a morning peak can occur when the buildings are cooled down before office hours. However, the cooling load profile for a specific system depends on weather conditions, types of buildings served, operation of the building cooling systems and the district cooling system rate structure.

Little actual data is normally available to determine a cooling load profile for a district cooling system. A cooling design day load profile is shown in **Figure 5.7** with a daily average load of about 50% of the peak load. This load profile, with a relatively low load during nighttime, is an attractive candidate for storage because of the significant impact of storage on reducing requirements for expensive chiller capacity.

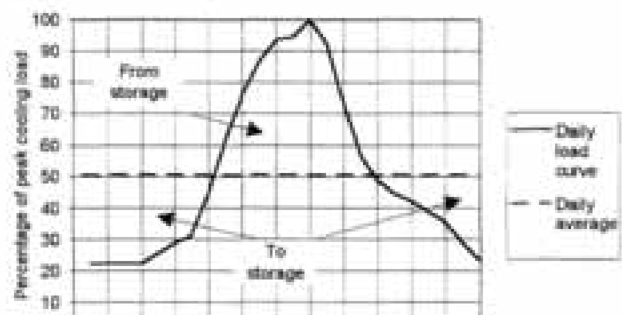

Figure 5.7 Generalized cooling design day load profile with possible storage [5-6]

Metered data from St Paul and Vasteras, Sweden (see **Figure 5.8**) show a flatter load curve with a higher average load -- up at 70-75% of the peak load. The high night load in Vasteras can be partly attributed to customers with high base loads such as hotels and stores with condenser cooling of refrigerators connected to the district cooling system. For St. Paul the high night loads are coincidental with days in which extreme humidity continues during the night.

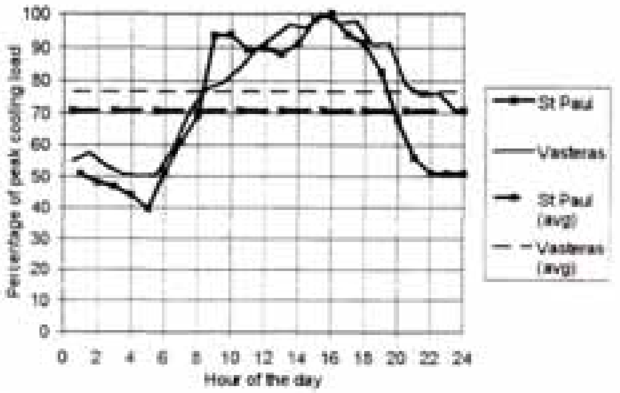

Figure 5.8 Design day load profile based on metered data

With a daily storage system, the installed chiller capacity can be reduced by 25-50% depending on local conditions. Because chiller capacity is usually the single most expensive item in a district cooling system, the reduction in installed chiller capacity is the main benefit. Other benefits are:

- Reduced electric demand cost;
- Possibility of operating the individual chillers closer to maximum efficiency;
- In the spring and autumn more "free cooling" can be produced during the night when the outdoor temperature is low enough, thereby increasing the period for free cooling; and
- In integrated CHP/district cooling plants using gas turbines, cool storage can also be used to boost electric output through inlet cooling.

Cool storage can be provided through storage of chilled water, ice or ice slurry. Chilled water is the most common form of cool storage,using concrete or steel tanks to store chilled water generated with any type of conventional chiller. Where space is available for chilled water storage, the economies of scale for this technology can provide significant economic advantages over ice storage.

Under normal conditions a chilled water storage tank is always filled with water. During discharge, cold water is pumped from the bottom of the tank and warm return water is supplied in the top. Because of the lower differential temperature, the inlet and outlet water velocities must be lower compared to hot water storage to ensure that warm and cold water is not mixed. Due to the different densities for warm and cold water a stable stratification can be obtained. At design conditions a volume of around 110 m^3 is needed to store 1 MWh with a differential temperature of 8°C.

The chilled water storage tank also can provide a fast supply of feedwater to the system. The sizing of the feedwater treatment equipment can thereby be reduced without having to use untreated water when bringing new pipe sections into operation or after a large leak. In addition, it is possible to reduce the size of the main pipes from a chiller plant if some remote storage can be sited.

Ice generation and storage is a well-developed technology, and allows storage in a more compact space -- often a key issue in urban environments. The volume required for ice storage is 4 to 6 times smaller compared to chilled water storage for the same energy storage capacity. [5-7] Ice storage also provides an oppportunity to reduce the temperature of cooling distribution and therefore reduce distribution costs. These advantages must be weighed against higher capital and operating costs for ice-making equipment compared to water chillers. According to a recent survey, the average capital costs of ice storage are about twice those of chilled water storage, and the energy requirements are higher by about one third. [5-7]

Ice slurry generation and distribution offers many of the same advantages of ice storage relative to compactness and lower distribution costs. However, this technology is still in the development stage.

5.4.2 Hot Water Storage

Hot water storage is used in district heating systems for four main purposes:

- To gain the ability to follow the electric demand instead of the heating demand with a CHP plant by storing the heat produced during peak electric demands until needed in the heating system;
- To be able to extend the use of cheaper heat sources where the supply can not be adjusted to the heat demand and/or where low fuel cost/high capital cost production (such as sewage water heat pumps or biomass boilers) is available;
- To get a more even load, and thereby better technical and environmental performance, for production equipment such as biomass boilers and incinerators; and
- To provide peaking and back-up, thereby reducing the investment in peaking and back-up boilers.

Apart from above benefits, the water volume in the tank provides a fast supply of feedwater to the system. The sizing of the feedwater treatment equipment can thereby be reduced without having to use untreated water when taking new pipe sections into operation or after an large leak.

Under normal conditions the storage tank is always filled with water. During discharge warm water is pumped from the upper level of the tank and cold return water is supplied in the bottom. The inlet and outlet water velocities must be low enough to ensure that warm and cold water are not mixed. Due to the different densities for warm and cold water a stable stratification can be obtained.
Pressurized tanks are expensive in sizes that are appropriate for hot water storage systems for larger district heating

systems. The storage tanks are instead usually built as atmospheric tanks, which in practice limits the maximum temperature in the tank to below 90°C. With a return temperature of 75°C at design conditions, a volume of around 45 m^3 is needed to store 1 MWh_{th}. As low a return temperature as possible is critical for the economy of a storage system.[5-8]

Due to the pressure difference between the atmospheric tank and the district heating network it is quite common to install a combined pump and turbine to be able to regain some of the energy that has to be used to pump the water from the tank to the district heating system. A steam blanket on top of the tank is also advisable to minimize the availability of oxygen to the water.

Because of the limited energy that can be stored per cubic meter, hot water tanks are usually designed as daily storage tanks. In certain cases, where lower return temperatures can be obtained, rock caverns have been used for weekly and annual energy storage.[5-9]

Daily heating load profiles will vary depending on the climate and mix of buildings on the system. In **Figure 5.9** a daily district heating profile is shown based on the district heating system in St. Paul, a downtown district heating system serving primarily offices, stores and hotels. Some apartment buildings with higher domestic hot water usage are also connected, but the morning and afternoon peaks can more probably be explained partly by return from night setback of the space heating and partly by domestic hot water usage in hotels. However, even with more apartments connected to a system the profile does not change very much. The domestic hot water usage will increase slightly and the night setback will probably decrease.

Based on the daily load profile, the peak demand could be reduced by around 10% with hot water storage. However, use of hot water storage for a district heating and cooling system will not be decided based on the possibility of reducing the heating peak demand so much as on the ability to transfer heat from base-loaded CHP plants, produced at peak electric demand periods, to other periods of the day or week with higher heating demand.

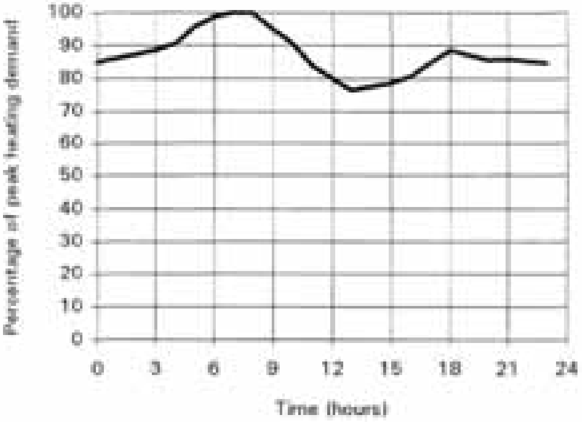

Figure 5.9 Daily load curve for heating in St. Paul, USA

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5-7 "ASHRAE RP-766: Study of Operational Experience with Thermal Energy Storage Systems," ASHRAE Transactions 1995, V. 101, Pt. 2," as noted in "Energy and Economic Implications of Combining District Cooling and Thermal Storage," Andrepont, Kooy and Winters, 10th Annual Cooling Conference, International District Energy Association, October 1995.

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Chapter 6
Integrating District Cooling and CHP

6.1 INTRODUCTION

This chapter addresses the energy efficiency and economic implications of alternatives for integrating district cooling and Combined Heat and Power (CHP). Throughout this chapter, references will be made to the major approaches to distributing cooling energy as described and illustrated in **Section 1.2**.

Section 6.2 compares the efficiency of cooling/CHP technology alternatives, based on maximizing cooling production under the technology assumptions presented in **Chapters 3 and 4**. Efficiency comparisons are not made on the basis of total annual energy outputs (electricity, heating and cooling). As discussed in **Chapter 2**, a consistent "figure of merit" for comparing the energy efficiencies of different options for combining CHP and cooling is problematic because each option, employed in a given circumstance, will produce different annual quantities of electricity, heating and cooling. Efficiency comparisons based on summing these three types of energy outputs will be misleading because they ignore the differing qualities of electricity, heating and cooling.

Similarly, environmental performance of CHP/cooling combinations cannot be expressed as emissions per unit of total energy output, for the reasons just summarized. In addition, the air emissions performance is highly case-specific and is driven by the CHP technology. Therefore, this chapter does not further address environmental impacts. The generalized information presented in **Chapters 3 and 4** on CHP air emissions and chiller refrigerant impacts can be used in initial evaluations of the environmental implications of cooling/CHP alternatives.

Section 6.3 describes how the economic analysis of cooling/CHP alternatives can be structured, discusses the influence of key variables and presents formulas for calculating the costs of cooling integrated with CHP.

Section 6.4 presents illustrative hypothetical scenarios for integrating district cooling with CHP, using the economic formulas.

Section 6.5 summarizes key findings regarding the energy efficiency and economics of the illustrative scenarios.

6.2 ENERGY EFFICIENCY

6.2.1 Assumptions

This section compares the efficiency of cooling/CHP technology alternatives based on maximizing cooling production under the technology assumptions previously presented. Efficiency comparisons are not made on the basis of annual outputs of electricity, heating and cooling for the reasons discussed in **Section 6.1**. In the following calculations, all electric and thermal output from CHP is converted to cooling. The net heating is hot water at available at 100/75°C, with a hot water economizer assumed in all cases except steam turbine CHP. (Steam turbine CHP was assumed to be fueled with coal, so reduction of the stack gas temperature with an economizer would not be desirable due to acid dew point concerns.) Thermal extraction conditions are as summarized in **Table 6.1**, consistent with **Chapters 3 and 4**.

District heating	100/75°C
Steam turbine drive chiller	11 bar steam
Hot water absorption chiller	95/85°C
1-stage steam absorption chiller	2 bar steam
2-stage steam absorption chiller	8 bar steam

Table 6.1 Summary of thermal extraction temperatures

6.2.2 Simple Cycle Gas Turbine CHP

Figure 6.1 shows that maximum cooling output from simple cycle gas turbine CHP is provided with a combination of two-stage absorption plus electric drive chillers. With a simple cycle, the higher-temperature heat-driven chillers provide a higher output than the lower-temperature options, with the least cooling output provided with all electric chillers. However, as discussed in **Chapter 2**, a simple cycle can be considered a thermodynamically suboptimal design for a new power plant.

Key data used to generate the figure are shown in **Table 6.2.** In **Tables 6.2-6.5**, the following symbols are used for chiller types:

- Elec. = Electric chillers only
- ST+E = Steam turbine drive plus electric
- HWA+E = Hot water absorption plus electric
- 1SA+E = 1-stage steam absorption plus electric
- 2SA+E = 2-stage steam absorption plus electric

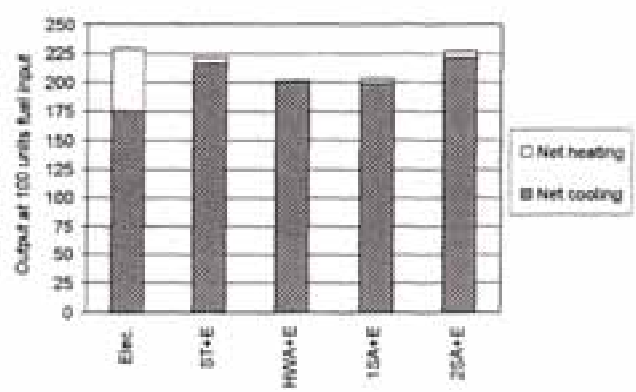

Figure 6.1 Simple cycle gas turbine CHP with maximum cooling production

Chiller type(s) ==>	Elec.	ST+E	HWA+E	1SA+E	2SA+E
Fuel input	100.0	100.0	100.0	100.0	100.0
Electric production	34.6	34.6	34.6	34.6	34.6
Thermal to chillers	0.0	47.8	53.1	50.1	48.5
Electricity to chillers	29.9	27.9	28.2	28.3	27.8
Electricity to aux.	4.8	6.7	6.5	6.4	6.9
Net electricity	0.0	0.0	0.0	0.0	0.0
Net cooling	175.1	215.9	200.5	199.1	221.1
Net heating	53.7	5.9	0.6	3.6	5.2

Table 6.2 Simple cycle gas turbine CHP with maximum cooling production

6.2.3 Diesel Engine CHP

Figure 6.2 shows that net cooling with gas diesel engine CHP is the highest with a combination of hot water absorption and electric drive chillers, although the output is barely above that of two-stage absorption plus electric drive chillers. Lower-temperature heat-driven options compare favorably with diesel CHP than with gas turbine CHP because the temperature of the thermal output of diesel engines is more limited compared to the gas turbine. Overall, the differences between the chiller scenarios are slight. As with the simple cycle gas turbine, the least cooling output is provided with electric chillers, but this option provides more 100°C district hot water. Key data used to generate the figure are shown in **Table 6.3**.

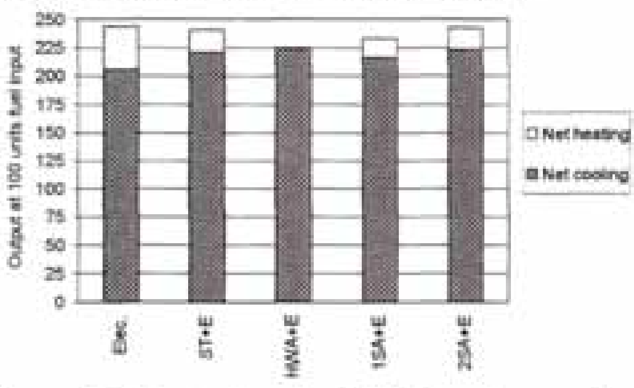

Figure 6.2 Gas diesel engine CHP with maximum cooling production

Chiller type(s) ==>	Elec.	ST+E	HWA+E	1SA+E	2SA+E
Fuel input	100.0	100.0	100.0	100.0	100.0
Electric production	40.6	40.6	40.6	40.6	40.6
Thermal to chillers	0.0	18.0	37.6	21.1	18.8
Electricity to chillers	35.0	34.3	33.8	34.3	34.2
Electricity to aux.	5.6	6.3	6.8	6.3	6.4
Net electricity	0.0	0.0	0.0	0.0	0.0
Net cooling	205.4	220.8	223.4	215.5	223.3
Net heating	38.0	20.0	0.4	16.9	19.2

Table 6.3 Gas diesel engine CHP with maximum cooling

6.2.4 Steam Cycle CHP

Figure 6.3 shows that net cooling with the reference steam cycle CHP is the highest with electric drive chillers. This is due to the electric derate which occurs when thermal energy is extracted for heat-driven chilling. In general, the cooling output is inversely related to the temperature of thermal extraction, which is the opposite of the simple cycle gas turbine and diesel engine CHP. Key data used to generate the figure are shown in **Table 6.4**.

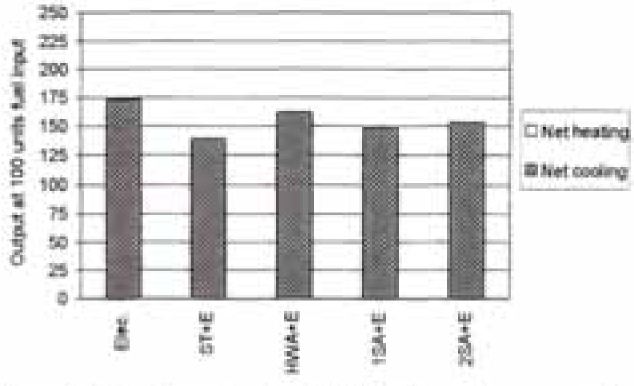

Figure 6.3 Steam turbine CHP with maximum cooling production

Chiller type(s) ==>	Elec.	ST+E	HWA+E	1SA+E	2SA+E
Fuel input	100.0	100.0	100.0	100.0	100.0
Electric production	34.2	15.4	26.3	23.2	17.1
Thermal to chillers	0.0	71.3	60.4	63.5	69.6
Electricity to chillers	29.5	10.3	20.8	18.0	11.6
Electricity to aux.	4.7	5.0	5.6	5.2	5.4
Net electricity	0.0	0.0	0.0	0.0	0.0
Net cooling	172.9	138.4	162.1	147.7	152.6
Net heating	0.0	0.0	0.0	0.0	0.0

Table 6.4 Steam turbine CHP with maximum cooling production

6.2.5 Combined Cycle Gas Turbine CHP

Figure 6.4 shows that gas turbine combined cycle CHP results in the same ranking of chiller technologies as shown for steam turbine CHP, although the differences between chiller technologies are much smaller than for steam turbine CHP. With a thermodynamically optimized new CHP plant (making use of gas turbine exhaust temperatures through a combined cycle), the lower-temperature chiller technologies generally show a higher output than higher-temperature options. Key data used to generate the figure are shown in **Table 6.5**.

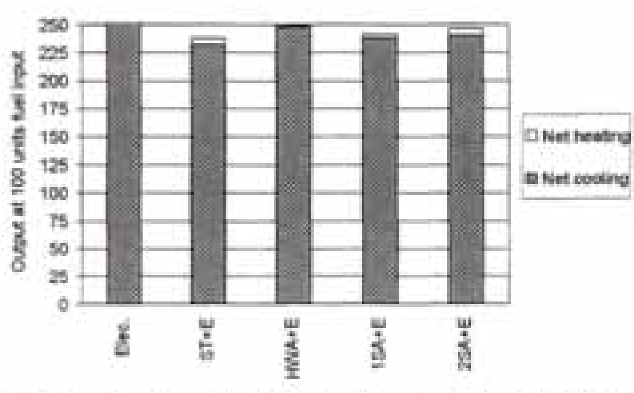

Figure 6.4 Gas turbine combined cycle CHP with maximum cooling production

Chiller type(s) ==>	Elec.	ST+E	HWA+E	1SA+E	2SA+E
Fuel input	100.0	100.0	100.0	100.0	100.0
Electric production	50.0	38.6	45.1	43.3	39.7
Thermal to chillers	0.0	43.4	41.6	41.1	42.3
Electricity to chillers	43.1	31.5	37.6	36.0	32.4
Electricity to aux.	6.9	7.1	7.6	7.3	7.3
Net electricity	0.0	0.0	0.0	0.0	0.0
Net cooling	252.9	232.1	248.1	238.5	240.7
Net heating	5.3	5.3	0.6	3.0	5.4

Table 6.5 Gas turbine combined cycle CHP with maximum cooling production

6.2.6 Comparison of CHP Technologies For Cooling Production

Figures 6.5-6.9 illustrate the comparative efficiencies, when maximizing chilled water production, of the CHP technologies when combined with electric drive, steam turbine drive, hot water absorption, one-stage steam absorption and two-stage steam absorption, respectively. These data are the same as presented in **Figures 6.1 - 6.4**, using the assumptions summarized in **Section 6.2.1**, but are sorted by chiller technology rather than CHP technology and show the split between electric-drive cooling and heat-driven cooling for each technology combination.

Following is the legend for **Figures 6.5-6.9**:

□ Net heating

▦ Electric cooling

▥ Thermal cooling

The following generalization holds true for all chiller technologies: gas turbine combined cycle CHP provides the highest net cooling for any of the chiller technologies, followed by diesel engine CHP and gas turbine simple cycle, with steam turbine CHP providing the lowest cooling output. The advantage of the gas turbine combined cycle is generally highest with electric drive chillers. For the heat-driven chiller options, the advantage of the gas turbine combined cycle is generally greater for lower-temperature chiller driving energy.

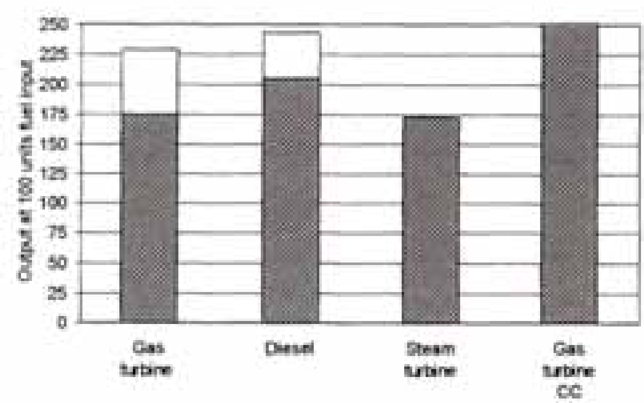

Figure 6.5 Comparative efficiencies of CHP combined with electric drive chillers (maximum chilled water scenarios)

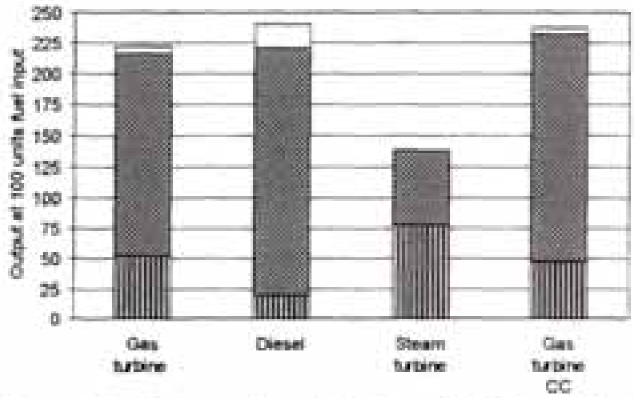

Figure 6.6 Comparative efficiencies of CHP combined with steam turbine drive chillers (maximum chilled water scenarios)

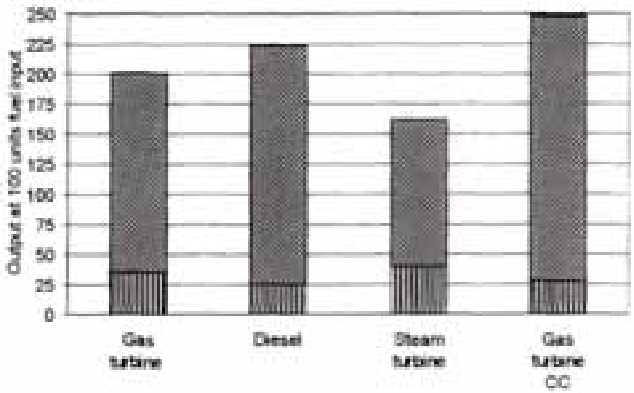

Figure 6.7 Comparative efficiencies of CHP combined with hot water absorption chillers (maximum chilled water scenarios)

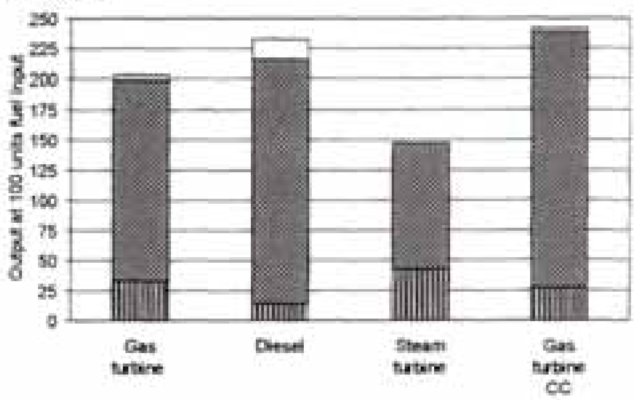

Figure 6.8 Comparative efficiencies of CHP combined with one-stage steam absorption chillers (maximum chilled water scenarios)

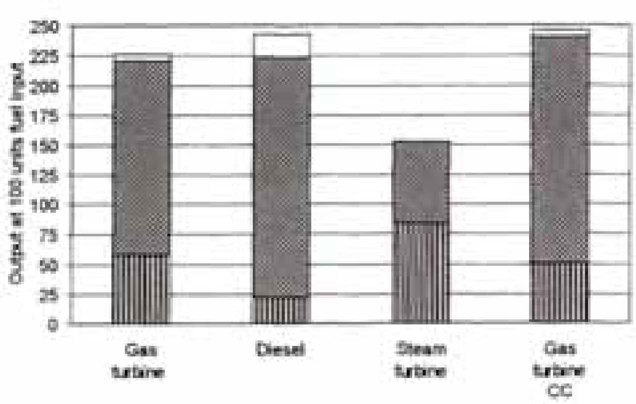

Figure 6.9 Comparative efficiencies of CHP combined with two-stage steam absorption chillers (maximum chilled water scenarios)

6.3 ANALYSIS OF COOLING/CHP OPTIONS

6.3.1 Distribution approaches

Choice of the optimal system in a specific circumstance will require more than a comparative analysis of CHP and chiller technologies for generating cooling. A related fundamental choice is determining the approach to energy distribution for cooling. Although distribution costs are not within the scope of this report, they are critical to a complete analysis of cooling options. Major types of distribution approaches were introduced and illustrated in schematic form in **Section 1.2**.

6.3.2 Calculating the cooling cost

This section presents a formula for calculating the net cost of cooling in new CHP/cooling plants being integrated with district heating systems. In the formula, the total annual costs of owning and operating the CHP and chiller facilities are calculated under a given assumption for the price of fuel. Revenues from sale of electricity and district heat based on given assumptions offset a portion of the costs. The remaining costs are then allocated to the cooling production to determine the net cost per unit of cooling output. Other approaches are possible, such assigning the net costs of CHP (after electric revenues) to heating and cooling according to the amount of energy used. However, in addition to the problem of differing exergy values for heating and cooling (see **Chapter 2**), the value of the heat is usually relatively inflexible.

In contrast to the "greenfield" situations addressed in this report, evaluation of CHP and/or cooling alternatives within existing CHP and/or cooling facilities can be strongly affected by the sunk costs and performance characteristics of the existing equipment.

Formula

The cost of cooling can be calculated as follows:

Cost of cooling production (cents/kWh$_c$) =

(CHP amortization cost + fuel costs + CHP non-fuel operating costs
- electricity revenue - heat revenue + chiller amortization cost
+ chiller non-energy operating cost) / cooling output in kWh$_c$

Symbols for each formula element and subsidiary formulas follow.

Symbols for each formula element

Basic values	A	Fuel price (cents/kWh fuel)
	B	Value of electricity (cents/kWh$_e$)
	C	Value of heat (cents/kWh$_{th}$)
Plant sizing	D	Size of CHP plant (MW$_e$ in district heating mode)
	E	Size of chiller plant (MW$_c$)
Equipment utilization	F	CHP total Equivalent Full Load Hours
	G	Chiller Equivalent Full Load Hours
Capital costs	H	Capital cost of CHP plant (\$/kW$_e$ in district heating mode)
	J	Capital cost of chiller plant (\$/kW$_c$)
	K	Real interest rate (%)
	L	Capitalization period (years)
	M	Capital recovery factor
Efficiencies and outputs	N	Cooling output in kWh$_c$
	P	Thermal output used for heating (kWh$_{th}$)
	R	Electric efficiency in CHP in district heating mode
	S	Electric efficiency in CHP in district cooling mode
Operating costs	T	Chiller electricity use (kWh$_e$/kWh$_c$)
	U	Number of Full-Time-Equivalent plant staff
	V	Labor cost per Full-Time-Equivalent (\$/year)
	W	CHP O & M costs (cents/kWh$_e$ in district heating mode)
	Y	Chiller water/chemical costs (cents/kWh$_c$)
	Z	Chiller maintenance costs (\$/kW$_c$ per year)

Subsidiary formulas

Formulas for key cost components are:

Capital recovery factor (M) = $K \times (1 + K)^{L}] / [(1 + K)^{L} - 1]$

CHP amortization cost = $(E \times H \times 1000 \times R / S)$

Fuel costs = $A \times D \times F \times 10 / S$

CHP non-fuel operating costs = $(U \times V \times R / S) + (W \times D \times G \times 10 \times R / S)$

Electricity revenue = $[(D \times F \times S / R) - (N \times T)] \times B \times 10$

Heat revenue = $P \times C \times 10$

Chiller amortization cost = $E \times 1000 \times J \times M$

Chiller non-energy operating cost = $(N \times Y \times 10) + (Z \times E \times 1000)$

6.3.3 Key variables

The efficiency, environmental and economic implications of alternative configurations for integrating cooling with CHP will vary depending on many site-specific factors, including:

1. cooling and heating peak demand and utilization hours;
2. economic variables, including types and costs of fuels and the values of electricity and heat;
3. characteristics of the CHP and chiller technologies;
4. temperature and other characteristics of the district heating system;
5. site-related design factors;
6. type and size of thermal storage; and
7. environmental restrictions.

Cooling and heating demand and utilization

Cooling and heating peak demand and utilization hours must be established from site-specific data. It is important to examine the specific characteristics of the buildings expected to be connected to the district cooling system. Buildings which opt for district cooling may tend to have a higher cooling load than average cooling data might imply (e.g., office buildings compared to a mix of buildings including residential buildings).

The type, age and specific cooling system characteristics of the buildings also affects the chilled water temperature difference. If a substantial portion of the long-term cooling load will come from newer buildings, the potential for a relatively high system-wide chilled water temperature difference increases the feasibility of chilled water distribution (as opposed to district-heat-driven options) by reducing the size of chilled water distribution pipes.

The density and pattern of likely cooling load are also relevant. For example, a high cooling load within a small area increases the feasibility a central chilled water plant. In contrast, if the likely cooling loads are dispersed in a large area served by a district heating system, dispersed absorption may be an attractive choice. If near-term cooling loads are somewhat divided geographically but substantial market penetration is expected in the long term, the decentralized district-heat-driven chilled water approach may be appropriate.

The summertime heating demand and the temperature of summer heating send-out (discussed further under "District heating system" below) are also relevant because they affect the degree of synergy of CHP district heating and cooling. If large amounts of low-temperature district heat can be used, CHP utilization can be increased and efficiency can be improved by making productive use of low-temperature heat which is not recoverable for cooling purposes. To the extent that operation of the CHP plant in district cooling mode produces byproduct thermal energy, the allocation of costs can be adjusted to credit the cooling side for this production.

Cooling and heating loads must be characterized, through load duration curves as discussed in **Chapter 5** and/or analysis of the seasonal range of the hourly load profiles, in sufficient detail to enable estimation of the size and coincidence of cooling and heating loads. The relative size of heating and cooling demand affects the feasibility of using district-heat-driven absorption, as discussed in **Section 4.6**. The size and shape of the cooling and heating load profiles will affect the sizing and utilization hours of the CHP and chiller facilities and the amount of CHP thermal output used for heating.

Economic variables

Fundamental economic factors include:

1. the availability and cost of fuel alternatives;
2. the value of electricity;
3. the value of heat supplied from a CHP facility to the district heating system; and
4. the cost of capital.

In a specific situation, the values of electricity and heat can be separated into two components: capacity (the ability to reliably meet a peak demand) and energy. Although in this report the economic values of electricity and heat have been treated as a single value, for a detailed specific analysis the values must be based on the actual case-specific economic values, which at least for electricity generally requires separation into capacity and energy components.

To the extent that the new CHP facility under consideration has higher running costs during certain operating hours compared to other available electric generation plants, this should be reflected in the value paid for the electricity generated in the CHP plant.

For all economic analyses in this report, the cost of capital is based on an interest rate of 8% over 15 years.

CHP and chiller technologies

The particular CHP and chiller technologies under study will obviously have significant implications for performance and costs, as presented in **Chapters 3 and 4**. CHP costs related to CHP performance in district heating mode must be adjusted to reflect the impact on electric efficiency when operating in district cooling mode for the given chiller technology. There are no such impacts with gas turbine and reciprocating engine CHP, but there are significant impacts with steam turbine CHP and, to a lesser extent, with gas turbine combined cycle CHP.

District heating system

Key characteristics of the existing district heating system which are relevant to the analysis of CHP/cooling alternatives include:

1. the current summer temperature of the district heat, and the potential to increase the temperature;
2. the cost, efficiency and environmental impacts of excess summertime heating capacity;
3. the geographic extent of distribution piping; and
4. the size and type of existing plant facilities.

There are significant variations in recovery temperatures required for district heating systems in the member countries of the IEA District Heating and Cooling Implementing Agreement, from the low supply temperatures used in Denmark and Holland (typically 90°C, but can be as low as 80°C) to high supply temperatures used in Germany (up to 130°C) and North America (as high as 180°C). The sizing of the CHP plant will also affect the level of recovery temperature. If the CHP plant is supplying less than the peak demand, it is possible to reduce the CHP recovery temperature below the level required at peak conditions, as has been assumed in this report. (See **Section 3.2**.)

Under the right circumstances, an increase in the summer district heating temperature can be the best approach to providing district-heat-driven cooling. For example, in the case of Gothenburg, Sweden (**Section 7.1**), the summer operating temperature of the district heating system has been increased in order to reduce absorption chiller investment costs. This was acceptable from an efficiency, environmental and economic standpoint due to the system's access to a variety of waste heat sources.

The recoverable heat from CHP equipment varies depending on the heat sink temperature from the district energy system, with the amount of recoverable heat usually increasing as the recovery temperature decreases. With reciprocating engines and gas turbines, the electric efficiency is unchanged for different heat supply temperatures while the total efficiency decreases with increased heat supply temperature. In contrast, the electric efficiency for a steam turbine decreases with increasing heat supply temperature while the total efficiency is unchanged.

Site-related design factors

Of the many possible site-specific design factors, several deserve particular mention:

1. available heat sinks for condenser cooling and "free cooling;"
2. ambient temperatures under which CHP/cooling will be operated;
3. space available in streets for piping;
4. space available for plants and/or thermal storage; and
5. local codes or other constraints.

The availability and temperature of heat sinks for condenser cooling and wintertime "free cooling" (cooling of the district chilled water loop solely through heat exchange with the condenser water) can have a significant effect on costs and efficiencies. Access to low-temperature heat sinks such as river, lake or ocean water can improve efficiencies. With dispersed absorption chillers, the design condenser water temperature can vary due to space limitations and other constraints for local building cooling towers (e.g., noise or esthetic/visibility concerns due to cooling tower drift). These constraints may result in higher condenser temperatures due to inappropriately sized cooling towers, which would reduce the capacity of the chillers.

In a site-specific evaluation it is important to consider performance at actual ambient temperatures during which most of the cooling/CHP operations will take place. Technology choices can be affected because of the impact of inlet air temperatures on gas turbine performance.

Design modifications may be appropriate, such as boosting power output by chilling gas turbine inlet air, either cooling directly on a baseload basis or indirectly through a thermal storage system.

If streets are crowded with below-ground utilities, subways or other obstructions, installation of chilled water distribution pipes may be expensive or practically impossible. In such circumstances, a district-heat-driven cooling system may be the best approach. Space availability for plant facilities, in addition to the geographic pattern of cooling load, affect the degree of centralization of the approach used for distributing cooling energy, and this in turn affects technology choices. For example, for widely dispersed small cooling loads which cannot be feasibly served via district-heat-driven absorption, the most economical CHP option may be small reciprocating engines.

Finally, site-specific local codes or other local considerations, such as those discussed above regarding cooling towers, may affect a variety of design factors.

Thermal storage

As discussed in **Chapter 5**, thermal storage can improve the economics of CHP and cooling, and hot water storage can affect overall CHP economics. The type of cool storage medium (chilled water, ice, ice slurry) will affect the size, performance and costs of chillers.

Environmental

Key environmental factors include:

1. emission standards for various power plant technologies and fuels;
2. restrictions affecting the availability and use of ozone-depleting refrigerants; and
3. local codes (e.g., restrictions on ammonia chillers).

6.4 ILLUSTRATIVE SCENARIOS

6.4.1 Introduction

There is an enormous range of economic and technical conditions under which integrated cooling/CHP could be implemented, and a case-specific analysis is essential. However, in order to illuminate some of the strengths and weaknesses of particular configurations, the following scenarios are presented, illustrating the possible results under a variety of assumed conditions for the facility size range emphasized in **Chapter 3** -- CHP plants with an electric generation capacity of 20-25 MW_e.

It is important to note that this chapter provides a consistent set of analyses, using the CHP and chiller data presented earlier in this report, of conceptualized new CHP/cooling facilities integrated with an existing district heating system. Integration of district cooling with an existing CHP system is highly dependent on many additional site-specific conditions, including "sunk" investments and performance characteristics of the existing equipment.

6.4.2 Heating and cooling load assumptions

Heating and cooling load assumptions were developed for three illustrative climate conditions as summarized in **Table 6.6**. The CHP utilization assumptions are illustrated in **Figure 6.10**. The relationships between heating and cooling peak demand were fixed as shown, and amounts of heating and cooling energy were derived from calculations of CHP Equivalent Full Load Hours (EFLH) based on load duration curves shown in **Figures 6.11-6.13**. CHP was assumed to provide 50% of the total heating demand for all climate scenarios except the "warm" climate. CHP was assumed to provide 50% of the total cooling demand for all scenarios, even in the "cold" climate, in order to maintain a reasonably economical utilization of the heat-driven chiller capacity.

Heat with a temperature of 100/75°C which is not usable for cooling production in CHP cooling mode was assumed to be used to offset heat that otherwise has to be produced in CHP heating mode. In cases where thermal drive chillers cannot provide enough cooling capacity to satisfy the assumed ratio between heating and cooling, electric drive chillers are used to make up the difference. In cases where the CHP is not fully utilized during the cooling peak for cooling production, the available extra CHP capacity is assumed to be utilized for condensing power production. Based on the capacity value of electricity in a specific case, a plant might be operated differently and therefore provide different economic results.

Use of chilled water storage has not been applied in the calculations. Chilled water storage will raise the utilization of chiller capacity. However, the annual effect of storage is difficult to calculate on a generalized basis.

	Cold	Medium	Warm
Total heating and cooling loads (EFLH)			
Heating	2500	2200	1000
Cooling	900	1200	2000
Heating peak relative to cooling peak	200%	100%	50%
CHP production of heating peak	50%	50%	100%
CHP production of cooling peak	50%	50%	50%
Calculated CHP EFLH			
Heating	4740	4000	1000
Cooling	1650	2220	3330
Total	6390	6220	4330
Chiller EFLH	1650	2220	3330

Table 6.6 Load and utilization assumptions for climate scenarios

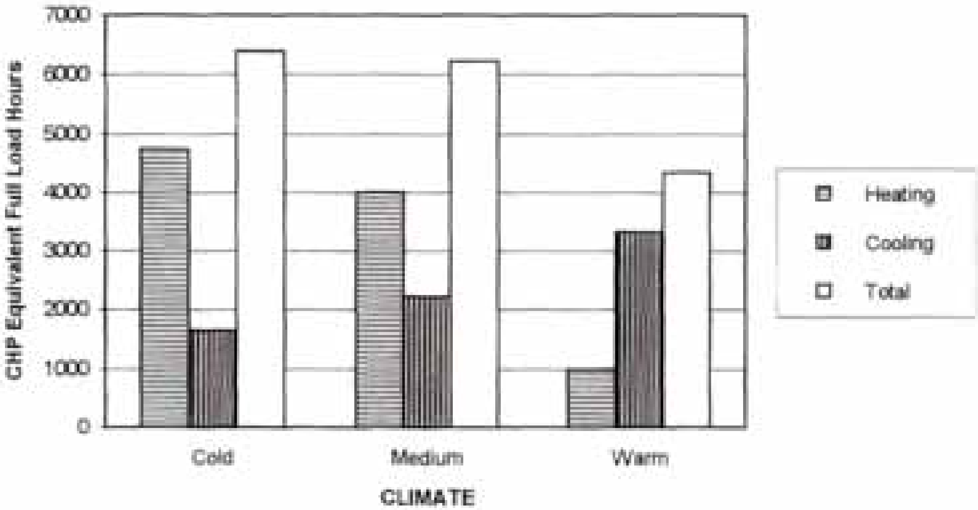

Figure 6.10 CHP utilization assumptions for climate scenarios

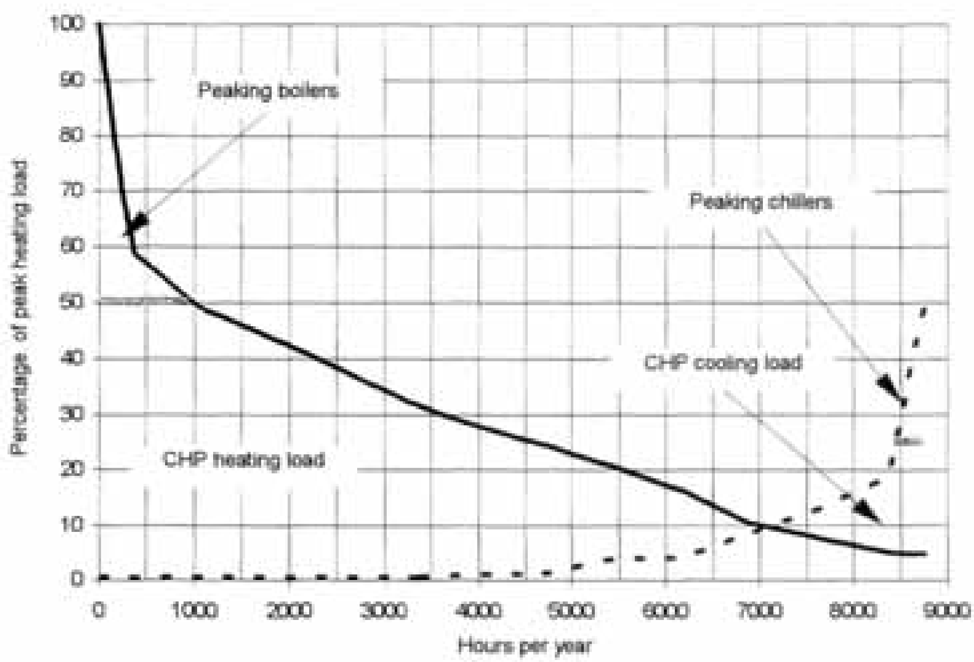

Figure 6.11 Cold climate heating and cooling load duration curves

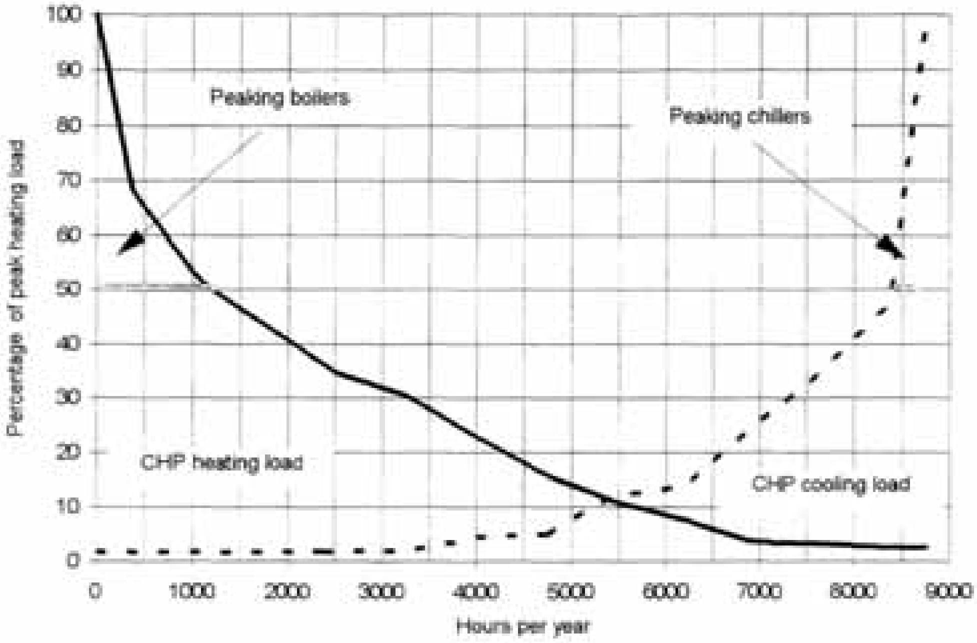

Figure 6.12 Medium climate heating and cooling load duration curves

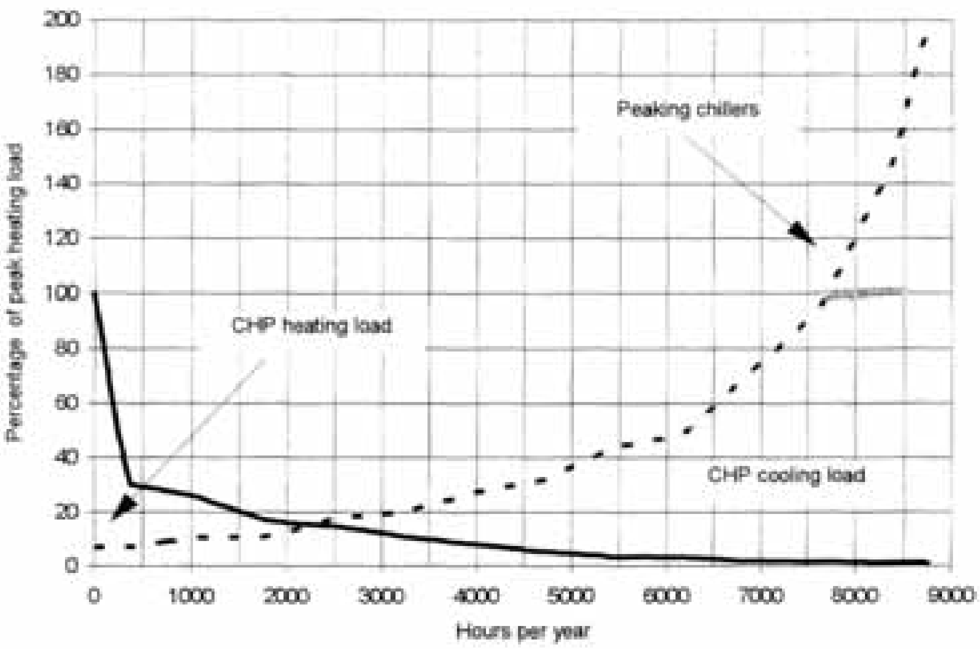

Figure 6.13 Warm climate heating and cooling load duration curves

For a given climate condition and CHP technology, the net cooling production and the CHP heating production are the same for each chiller technology combined with that CHP technology. In many of the scenarios for heat-driven cooling, some electric drive cooling is required in order to produce the same net cooling output for comparison with the other scenarios. Supplemental electric drive cooling was generally not required or was minimal for the cold climate scenarios. Some supplemental electric drive cooling was required for the medium and warm climate scenarios, particularly for the gas turbine and diesel engine scenarios and for combinations including one-stage steam absorption or hot water absorption. The net electricity varied slightly depending on how much electricity was used in order to produce additional cooling as required in order to ensure that the cooling production is equal for a given technology comparison.

It is important to note that, as discussed in **Section 6.3.3**, the values of electricity and heat can be separated into two components: capacity (the ability to reliably meet a peak demand) and energy. The different technology combinations will yield different levels of electric export capacity at peak conditions, and this will affect the capacity value of the exported electricity in specific cases. However, in these generalized comparative analyses, the same single value for exported electricity has been assumed. It is important to evaluate the capacity value of electricity exports, which can only be done on a case-specific basis.

The following scenarios are based on the assumptions summarized in **Table 6.7**.

Scenario	Climate	Heat value (c/kWh$_{th}$)	Electric value (c/kWh$_e$)	Potential for district-heat-driven absorption
1	Medium	1.25	3.00	No
2	Medium	1.25	4.00	No
3	Warm	1.25	3.00	No
4	Warm	1.25	6.00	No
5	Cold	2.50	3.00	Yes
6	Cold	2.50	5.00	Yes
7	Medium	2.50	4.00	Yes
8	Medium	2.50	5.00	Yes

Table 6.7 Scenario assumptions

6.4.3 Scenario 1: Medium climate/central chilled water/low heat and electricity values

Heat and electricity values were assumed to be 1.25 cent/kWh$_{th}$ and 3.0 cents/kWh$_e$, respectively. The comparative analysis shows that gas turbine CHP with electric drive chillers provided the lowest cooling costs. **Figure 6.14** compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. Steam turbine CHP can compete with the gas turbine using 1.0 cent/kWh fuel if solid fuel with a cost below 0.25 cent/kWh can be obtained for firing the steam turbine boiler.

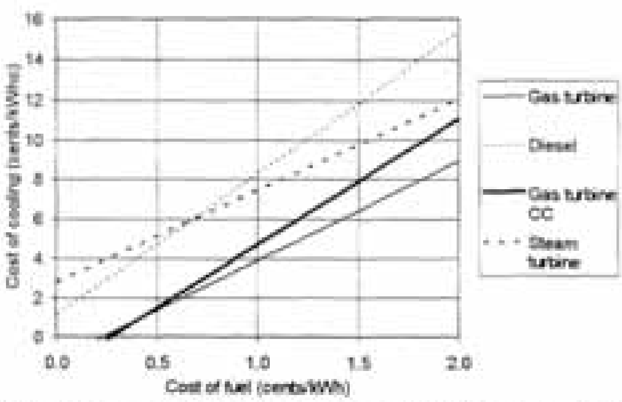

Figure 6.14 Scenario 1 cooling costs with electric drive chillers combined with various CHP technologies

The cost differences between the chiller types when combined with gas turbine CHP are very small, as summarized in **Table 6.8** under the assumption that fuel costs 1.0 cent/kWh.

Electric	3.9
Steam turbine	4.3
1-stage absorption	4.1
2-stage absorption	4.2

Table 6.8 Scenario 1 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine CHP with 1.0 cent/kWh fuel

However, the cost differences between the chiller types when combined with steam turbine CHP are more significant, as shown in **Figure 6.15** for a range of fuel costs.

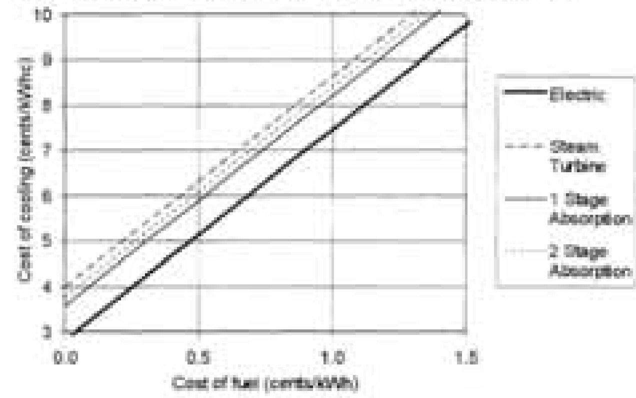

Figure 6.15 Scenario 1 cooling costs (cents/kWhc) for chiller technologies combined with steam turbine CHP

6.4.4 Scenario 2: Medium climate/central chilled water/low heat value/high electricity value

This scenario is the same as Scenario 1 except that the electricity value has been increased to 4.0 cents/kWh$_e$. The comparative analysis shows that gas turbine combined cycle CHP with electric drive chillers provided the lowest cooling costs at a fuel cost of 1.0 cent/kWh. **Figure 6.16** compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. At fuel costs above 1.3 cents/kWh, the simple cycle gas turbine becomes the lowest-cost option. In this scenario, solid fuel for firing steam turbine CHP must be available at zero cost in order to compete with gas turbine combined cycle using fuel costing 1.0 cent/kWh.

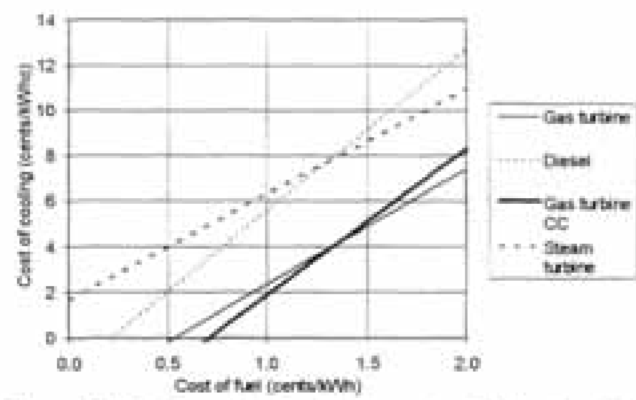

Figure 6.16 Scenario 2 cooling costs with electric drive chillers combined with various CHP technologies

In Scenario 2, the cost differences between chiller technologies combined with the lowest-cost CHP option are larger than is the case in Scenario 1, as summarized in **Table 6.9** under the assumption that fuel costs 1.0 cent/kWh. These cost differences are shown in **Figure 6.17** for a range of fuel costs.

Electric	1.9
Steam turbine	3.1
1-stage absorption	2.6
2-stage absorption	2.9

Table 6.9 Scenario 2 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine combined cycle CHP with 1.0 cent/kWh fuel

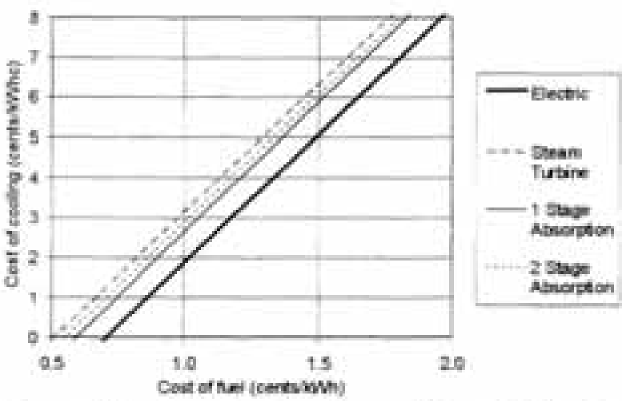

Figure 6.17 Scenario 2 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine combined cycle CHP

Figure 6.18 compares the cost per kWh of cooling using the most competitive heat-driven chiller option (one-stage steam absorption) coupled with the four major CHP technologies. The "cross-over" point, at which gas turbine CHP becomes lower cost than gas turbine combined cycle, is at a fuel cost of about 0.8 cent/kWh.

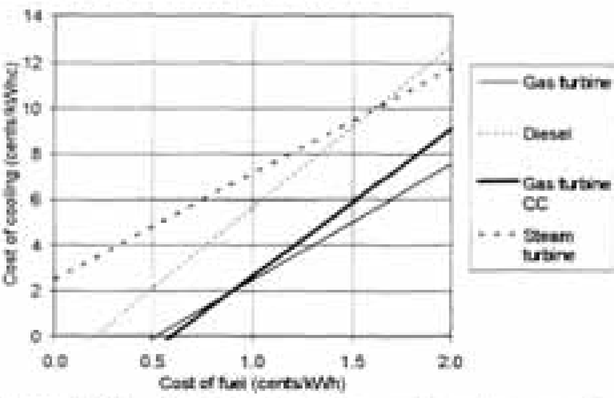

Figure 6.18 Scenario 2 cooling costs with one-stage steam absorption plus electric drive chillers combined with various CHP technologies

6.4.5 Scenario 3: Warm climate/central chilled water/low heat and electricity values

Heat and electricity values were assumed to be 1.25 cent/kWh$_{th}$ and 3.0 cents/kWh$_e$, respectively. The comparative analysis shows that gas turbine CHP with electric drive chillers provided the lowest cooling costs. **Figure 6.19** compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. Sensitivity of cooling costs to changes in fuel cost is lower in the warm climate compared to the medium climate because net CHP costs are spread over a larger number of cooling utilization hours. Steam turbine CHP can compete with the gas turbine using 1.0 cent/kWh fuel if solid fuel with a cost below approximately 0.25 cent/kWh can be obtained for firing the steam turbine boiler.

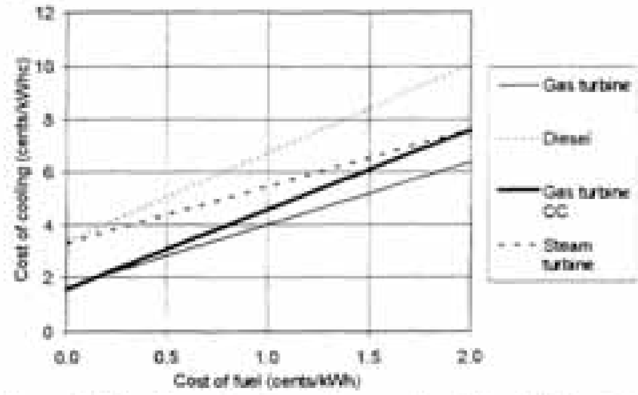

Figure 6.19 Scenario 3 cooling costs with electric drive chillers combined with various CHP technologies

The cost differences between the chiller types combined with gas turbine CHP are very small, as summarized in **Table 6.10** under the assumption that fuel costs 1.0 cent/kWh.

Electric	4.0
Steam turbine	4.2
1-stage absorption	4.1
2-stage absorption	4.2

Table 6.10 Scenario 3 cooling costs (cents/kWhc) or chiller technologies combined with gas turbine CHP with 1.0 cent/kWh fuel

However, the cost differences between the chiller types combined with steam turbine CHP are more significant, as shown in **Figure 6.20** for a range of fuel costs.

Figure 6.20 Scenario 3 cooling costs (cents/kWhc) for chiller technologies combined with steam turbine CHP

6.4.6 Scenario 4: Warm climate/central chilled water/low heat value/high electricity value

This scenario is the same as Scenario 3 except that the electricity value has been increased to 6.0 cents/kWh$_e$. The comparative analysis shows that gas turbine combined cycle CHP with electric drive chillers provided the lowest cooling costs. **Figure 6.21** compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. Note that in this scenario steam turbine CHP cannot compete with gas turbine combined cycle CHP unless gas fuel costs over 1.4 cents/kWh and solid fuel for firing the steam turbine boilers has zero cost.

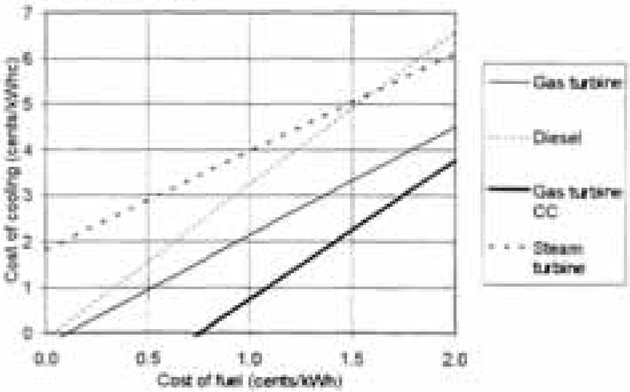

Figure 6.21 Scenario 4 cooling costs with electric drive chillers combined with various CHP technologies

Unlike with simple cycle gas turbine CHP, with gas turbine combined cycle CHP the electric drive option has a larger cost advantage over the other chiller options, as summarized in **Table 6.11** under the assumption that fuel costs 1.0 cent/kWh. These cost differences are shown in **Figure 6.22** for a range of fuel costs.

Electric	0.7
Steam turbine	2.0
1-stage absorption	1.6
2-stage absorption	1.6

Table 6.11 Scenario 4 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine combined cycle CHP with 1.0 cent/kWh fuel

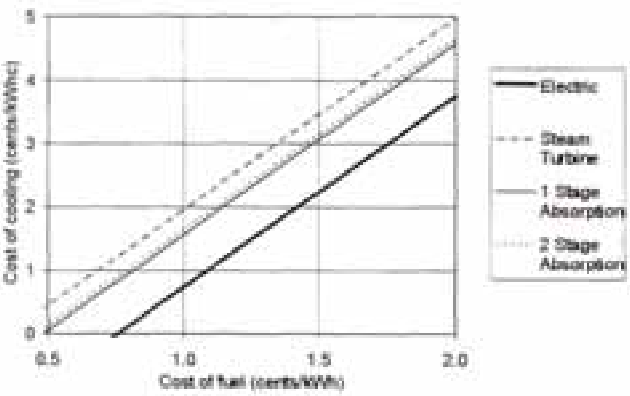

Figure 6.22 Scenario 4 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine combined cycle CHP

Figure 6.23 compares the cost per kWh of cooling using the most competitive heat-driven chiller option (two-stage steam absorption) coupled with the four major CHP technologies. The "cross-over" point, at which gas turbine CHP becomes lower cost than gas turbine combined cycle, is at a fuel cost of about 1.3 cents/kWh.

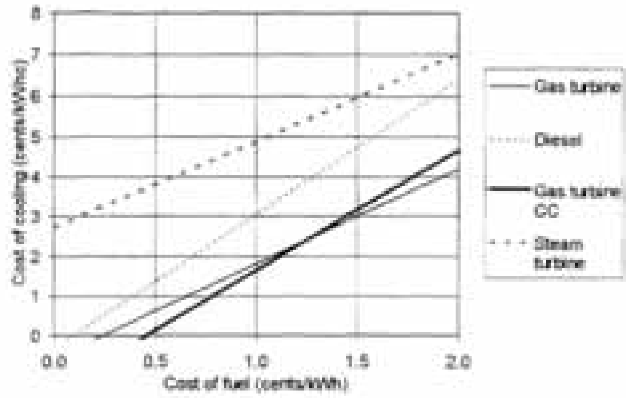

Figure 6.23 Scenario 4 cooling costs with two-stage steam absorption plus electric drive chillers combined with various CHP technologies

6.4.7 Scenario 5: Cold climate/high heat value/low electricity value

Heat and electricity values were assumed to be 2.5 cents/kWh$_{th}$ and 3.0 cents/kWh$_{e}$, respectively. In this case it is assumed that a well-developed district heating system offers the possibility of dispersed absorption or decentralized district-heat-driven chilled water approaches as alternatives to centralized chilled water.

The comparative analysis shows that, for centralized chilled water district cooling, gas turbine CHP with electric drive chillers provided the lowest cooling costs. **Figure 6.24** compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. In the cold climate, costs increase steeply with increasing fuel costs because the increasing fuel cost burden is carried by relatively few cooling utilization hours. Steam turbine CHP can compete with the gas turbine using 2.0 cents/kWh fuel if solid fuel with a cost below approximately 1.2 cents/kWh can be obtained for firing the steam turbine boiler.

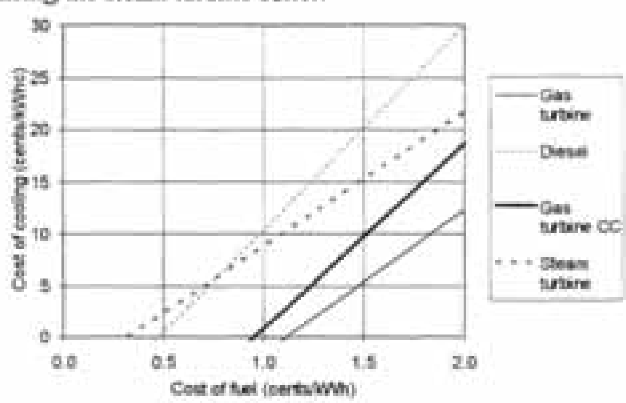

Figure 6.24 Scenario 5 cooling costs with centralized electric drive chillers combined with various CHP technologies

The cost differences between the chiller types for centralized district cooling are relatively small, as summarized in **Table 6.12** under the assumption that fuel costs 2.0 cents/kWh.

Electric	12.3
Steam turbine	13.0
1-stage absorption	12.7
2-stage absorption	12.9

Table 6.12 Scenario 5 cooling costs (cents/kWhc) for centralized chiller technologies combined with gas turbine CHP with 2.0 cents/kWh fuel

However, the cost differences between the chiller types, when combined with steam turbine CHP, are more significant, as shown in **Figure 6.25** for a range of fuel costs.

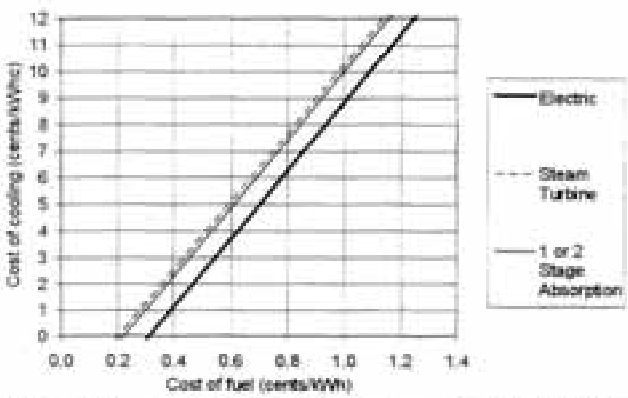

Figure 6.25 Scenario 5 cooling costs (cents/kWhc) for chiller technologies combined with steam turbine CHP

Figure 6.26 compares the cost per kWh of cooling using dispersed chillers (hot water absorption plus electric drive chillers) coupled with the four major CHP technologies. The cost of cooling with the dispersed approach, assuming 2.0 cents/kWh fuel, is 14.8 cents/kWh$_c$ compared to 12.3 cents for the centralized chilled water approach. This cost differential of 2.5 cents/kWh$_c$ must be weighed against the costs of constructing a chilled water distribution system. The decentralized district-heat-driven chilled water approach would significantly reduce, although not eliminate, costs for chilled water distribution. However, the chiller plant-related capital costs would be lower compared to the dispersed absorption approach, as discussed in **Chapter 4**.

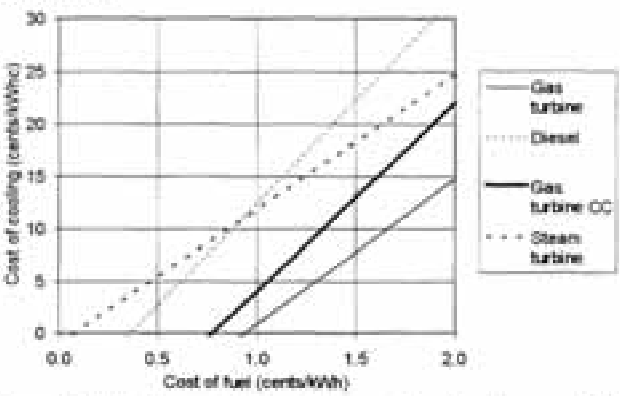

Figure 6.26 Scenario 5 cooling costs with dispersed hot water and electric drive chillers combined with various CHP technologies

6.4.8 Scenario 6: Cold climate/high heat value/high electricity value

This scenario is the same as Scenario 5 except that the electricity value has been increased to 5.0 cents/kWh$_e$. The comparative analysis shows that with this increase in the value of electricity the gas turbine combined cycle barely overtakes the simple cycle gas turbine as the CHP technology which, combined with electric drive chillers, provides the lowest cooling costs.

Figure 6.27 compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. Steam turbine CHP can compete with the gas turbine combined cycle using 2.0 cent/kWh fuel if solid fuel with a cost below approximately 1.0 cents/kWh can be obtained for firing the steam turbine boiler.

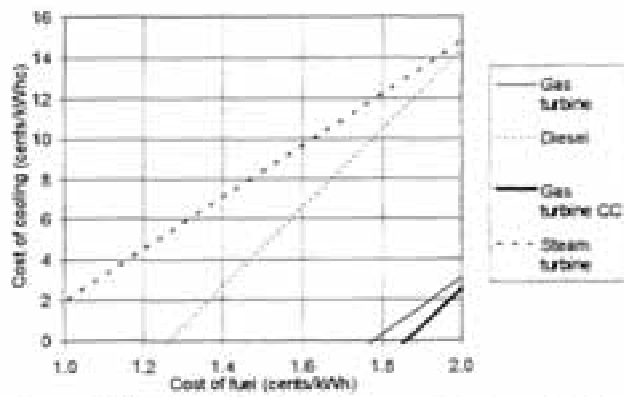

Figure 6.27 Scenario 6 cooling costs with centralized electric drive chillers combined with various CHP technologies

The cost differences between the chiller types for centralized district cooling are relatively small, as summarized in **Table 6.13** under the assumption that fuel costs 2.0 cent/kWh. These cost differences are shown in **Figure 6.28** for a range of fuel costs.

Electric	3.1
Steam turbine	3.5
1-stage absorption	3.2
2-stage absorption	3.3

Table 6.13 Scenario 6 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine combined cycle CHP with 2.0 cents/kWh fuel

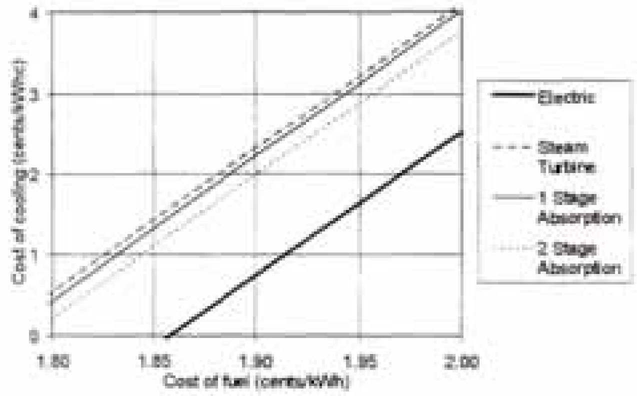

Figure 6.28 Scenario 6 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine combined cycle CHP

The cost differential between centralized electric drive chillers and dispersed hot water absorption (supplemented by electric drive chillers) for Scenario 6 is about the same as calculated for Scenario 5.

6.4.9 Scenario 7: Medium climate/high heat value/low electricity value

Heat and electricity values were assumed to be 2.5 cents/kWh$_{th}$ and 4.0 cents/kWh$_e$, respectively. As in Scenarios 5 and 6, it was assumed that a well-developed district heating system offers the possibility of district-heat-driven cooling approaches as alternatives to centralized chilled water.

The comparative analysis showed that, for centralized chilled water district cooling, gas turbine CHP with electric drive chillers provided the lowest cooling costs at fuel costs over 1.3 cents/kWh. **Figure 6.29** compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. Steam turbine CHP can compete with the gas turbine using 2.0 cent/kWh fuel if solid fuel with a cost below approximately 1.2 cents/kWh can be obtained for firing the steam turbine boiler.

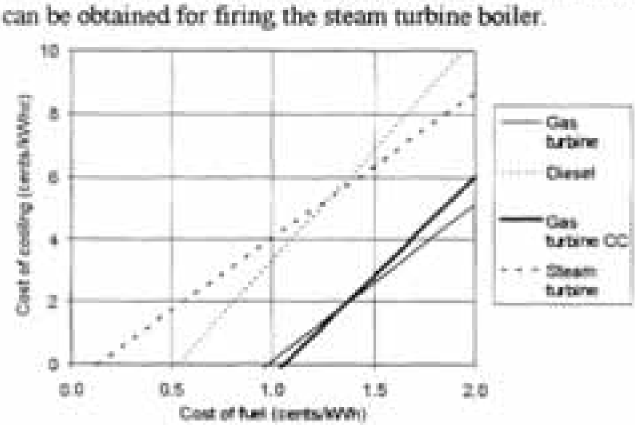

Figure 6.29 Scenario 7 cooling costs with centralized electric drive chillers combined with various CHP technologies

The cost differences between the chiller types for centralized district cooling are relatively small, as summarized in **Table 6.14** under the assumption that fuel costs 2.0 cent/kWh.

Electric	5.1
Steam turbine	5.4
1-stage absorption	5.3
2-stage absorption	5.3

Table 6.14 Scenario 7 cooling costs (cents/kWhc) for centralized chiller technologies combined with gas turbine CHP with 2.0 cent/kWh fuel

However, the cost differences between the chiller types when combined with steam turbine CHP are more significant, as shown in **Figure 6.30** for a range of fuel costs.

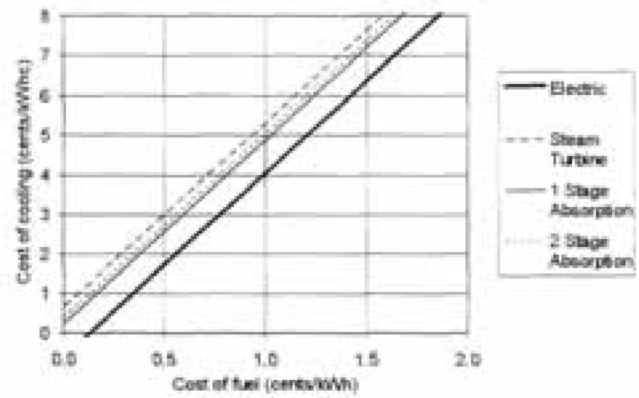

Figure 6.30 Scenario 7 cooling costs (cents/kWhc) for chiller technologies combined with steam turbine CHP

Figure 6.31 compares the cost per kWh of cooling using dispersed chillers (hot water absorption plus electric drive chillers) coupled with the four major CHP technologies. The cost of cooling with the dispersed approach, assuming 2.0 cents/kWh fuel, is 6.5 cents/kWh$_c$ compared to 5.1 cents for the centralized chilled water approach. This cost differential of 1.4 cents/kWh$_c$ must be weighed against the costs of constructing a chilled water distribution system.

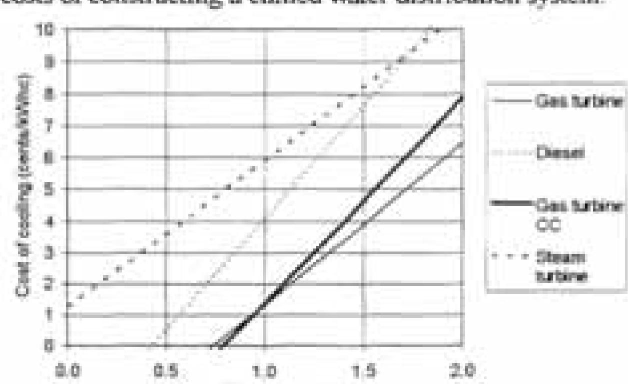

Figure 6.31 Scenario 7 cooling costs with dispersed hot water and electric drive chillers combined with various CHP technologies

6.4.10 Scenario 8: Medium climate/high heat value/high electricity value

This scenario is the same as Scenario 7 except that the electricity value has been increased to 5.0 cents/kWh$_e$. With this increase in the value of electricity, at a fuel cost of 2.0 cents/kWh the gas turbine combined cycle barely overtakes the simple cycle gas turbine as the CHP technology which, combined with electric drive chillers, provides the lowest cooling costs.

Figure 6.32 compares the cost per kWh of cooling using the four major CHP technologies combined with electric drive chillers. Steam turbine CHP can compete with the gas turbine combined cycle using 2.0 cents/kWh fuel if solid fuel with a cost below approximately 1.0 cents/kWh can be obtained for firing the steam turbine boiler.

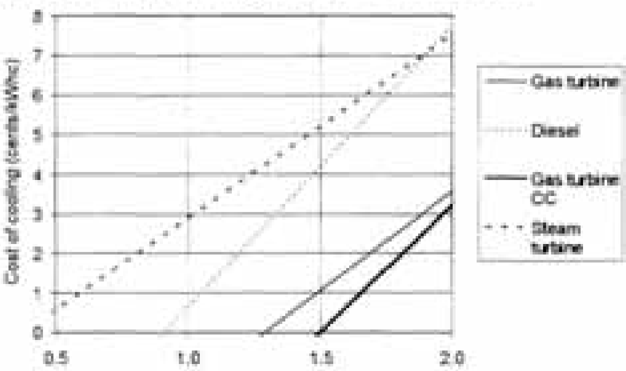

Figure 6.32 Scenario 8 cooling costs with centralized electric drive chillers combined with various CHP technologies

The cost differences between the chiller types for centralized district cooling are relatively small, as summarized in **Table 6.15** under the assumption that fuel costs 2.0 cent/kWh. These cost differences are shown in **Figure 6.33** for a range of fuel costs.

Electric	3.6
Steam turbine	3.7
1-stage absorption	3.7
2-stage absorption	3.6

Table 6.15 Scenario 8 cooling for chiller technologies combined with gas turbine combined cycle CHP with 2.0 cents/kWh fuel costs (cents/kWhc)

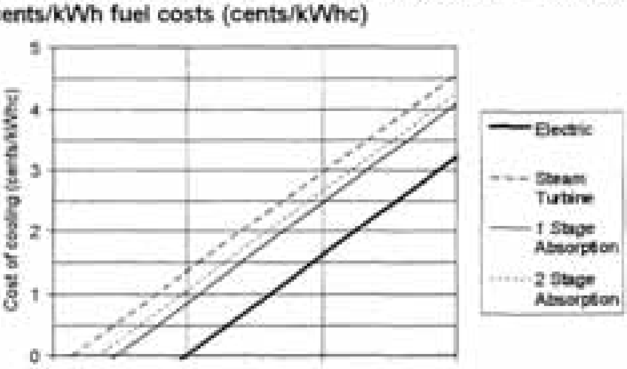

Figure 6.33 Scenario 8 cooling costs (cents/kWhc) for chiller technologies combined with gas turbine combined cycle CHP

The cost differential between centralized electric drive chillers and dispersed hot water absorption (supplemented by electric drive chillers) for Scenario 8 is about the same as calculated for Scenario 7.

6.5 FINDINGS

6.5.1 Energy efficiency

- If the goal is maximum cooling output per unit of fuel used, the CHP technologies rank as follows, from highest to lowest output:

  1. Gas turbine combined cycle
  2. Diesel engine
  3. Gas turbine
  4. Steam turbine

  This ranking holds true regardless of the chiller technologies employed, although the extent of differences between the CHP types varied depending on the chiller technologies.

- With a simple cycle gas turbine, the higher-temperature heat-driven chillers (supplemented by electric drive chillers) provide more cooling output than the lower-temperature options, with the electric-chiller-only option providing the lowest cooling output. This is also roughly true with a diesel engine, although the lower-temperature heat-driven options compare more favorably because the temperature of useful thermal output of diesel engines is more limited compared to the gas turbine.

- With steam turbine and gas turbine combined cycle CHP, the electric drive chiller provides the highest cooling output, followed by hot water absorption and other heat-driven options, roughly in order of increasing driving temperature. The differences between chiller types with gas turbine combined cycle are less than those for steam turbine CHP.

- In the analyses presented in **Section 6.3**, there is virtually no difference in cooling output from gas turbine combined cycle between:

  1. operation in condensing mode with all electric chillers; and
  2. operation in CHP mode using a combination of one-stage steam absorption and electric drive chillers to maximize cooling output.

- Simple cycle gas turbine CHP can appear attractive from an efficiency standpoint when the thermal output is viewed as "waste heat." However, it can be argued that this is because, from the standpoint of new plant design, total efficiency has not really been optimized with a simple cycle, i.e., generally there is the capability to generate additional electricity in a combined cycle.

- For a new CHP facility, there is not a compelling argument for using heat generated through CHP to drive chillers as opposed to installing a condensing tail to drive electric chillers. However, this argument does not hold for the smaller end of the scale of CHP facilities (e.g., 5 MW_e), where due to economies of scale it is generally not cost-effective to install a steam turbine to drive a generator in a combined cycle. In these circumstances, the thermal energy can be appropriately regarded as "waste," and the economics and perceived efficiency of absorption is favorable.

6.5.2 Economics

The following discussion summarizes the results of the illustrative scenarios presented in **Section 6.5** for new CHP systems in the 20-25 MW_e size range under stated load and economic assumptions. The economics in a specific case are highly dependent on case-specific factors.

CHP options

- In the illustrative scenarios, simple cycle gas turbine CHP provides the lowest cooling cost at low values of electricity (3 cents/kWh_e), due in large part to its low investment cost.

- Combined cycle gas turbine CHP provides the lowest cooling cost at higher electricity values (above 5 cents/kWH_e) as a result of its high electric efficiency. As electricity value rises, the competitiveness of the gas turbine combined cycle increases faster than the other CHP options.

- With the potential for steam turbine CHP to be fired with lower-cost fuel, this CHP option has the potential to be the most cost-effective option depending on specific fuel costs.

- In CHP plants under 20 MW_e, reciprocating engine CHP can become more competitive than indicated in the illustrative scenarios, and in CHP plants above 50 MW_e, steam turbine CHP has the potential to be more competitive than indicated.

- Sensitivity of cooling costs to changes in fuel cost, heat value and electricity value is lowest in the warm climate because net CHP costs are spread over a relatively large number of cooling utilization hours. Conversely, sensitivity of cooling costs to these factors is highest in the cold climate because net CHP costs are spread over a relatively small number of cooling utilization hours.

Chiller options

- Based on the illustrative scenarios, electric drive chillers combined with gas turbine CHP (at low electric values) and gas turbine combined cycle CHP (at high electric values) provided the lowest cooling costs for centralized chilled water district cooling. However, in many scenarios the cost differences between electric drive cooling and heat-driven options (supplemented with electric drive) were quite small and can be considered insignificant in view of the many case-specific variables which can affect the calculations. In general, the costs of the CHP are more significant than the costs of the chiller equipment.

- Generally, cost differences between the cooling technologies combined with simple cycle gas turbine and diesel engine CHP are very small because the electric output of these CHP technologies is not affected by thermal extraction. In contrast, with steam turbine CHP and to a lesser extent gas turbine combined cycle CHP, cost differences between chiller technologies are more significant because with the steam cycle the electric output decreases when thermal energy is extracted, and this derate increases with increasing thermal extraction temperature.

- Aside from direct economic considerations, the value of flexibility and reliability may lead the system designer to install heat-driven chillers. For example, heat-driven cooling can help protect against penalties associated with a loss of power generation capacity at peak, since with heat-driven chillers the system operator can fire up relatively inexpensive standby boiler capacity.

- For all CHP types, the economic differences between the heat-driven chiller options were relatively small, with costs slightly higher for chillers requiring higher-temperature driving energy. In essence, the higher investment costs for higher-temperature heat-driven options was to a large extent offset by their higher efficiencies.

Chapter 7
Case Studies

7.1 GOTHENBURG [7-1, 7-2, 7-3]

The city of Gothenburg, Sweden is served by a hot water district heating system which supplies commercial buildings, hospitals, a university and about 150,000 households. The hot water temperature varies from 75°C in summer to a maximum of 120°C during the winter. The system derives about 70% of its energy from waste heat, including industrial waste heat from an oil refinery, heat recovered from wastewater via heat pumps, and heat from refuse incineration. In addition, the system also uses waste heat produced from engine testing at a truck engine testing facility.

A refuse incineration plant is owned by the city of Gothenburg and burns about 300,000 tons/year. The heat generated through incineration is used in the district heating system. Heat from 5-18°C sewage treatment plant effluent is used in a heat pump plant to heat district heating return water (48-70°C) to a temperature of 80-85°C. This water is then pumped to the district heating plant where the temperature is further increased to provide the temperatures needed during the winter.

Steam turbine CHP, fueled with coal, oil and natural gas, is used to provide additional wintertime heat. A new gas-fired CHP plant is planned, and use of biomass fuel is being evaluated.

Provision of district cooling services was initiated in 1995. Cooling is provided with absorption chillers driven by the hot water district heating system. With the addition of district-heat-driven cooling, the system will be able to optimize the use of waste heat, providing a heat load throughout the summer when heating demand would otherwise be low. In order to reduce investment cost for the absorption chillers, the summer district heat temperature is being maintained at 90°C.

The initial plans were to build decentralized chilled water plants to serve adjacent buildings. However, as of March 1996 only one of the six chiller plants constructed to date has been built to serve more than one building. Although decentralized chilled water remains the goal, it has proven difficult to obtain multiple customer contracts at the same time in the same immediate area.

A survey of central Gothenburg has shown that about 30 prospective customers have a cooling demand of approximately 25 MW_c, with a utilization time of 1000 hours. The estimated capital cost of the entire cooling system is $12 million (US), of which about $4.3 million (US) is for absorption chillers. The installed chiller capacity is expected to reach 9.5 MW_c by the end of 1996.

7.2 SEOUL [7-4, 7-5, 7-6]

Korea District Heating Corp. (KDHC) was established in 1985 and restructured as a public corporation with the goal of saving energy and improving the environment through district heating. The government designates a district heating area for any newly developed area that is more than 3.3 square kilometers. District heating was mandated by the government for five new satellite cities near Seoul. As of 1995, district hot water was distributed in 8 systems with a total peak demand of 3,500 MW_{th}. The total demand is expected to grow to 6,300 MW_{th} in 11 systems by the year 2001. District hot water has a peak winter supply temperature of 115°C in and a summer temperature of 95°C.

Approximately 90% of the heat is produced in gas turbine combined cycle and steam turbine CHP facilities owned by the Korea Electric Power Corporation (KEPCO).

Since 1992 KDHC has been providing district cooling through a dispersed absorption approach, using the district heating network to deliver hot water to absorption chillers located in customer buildings. Hot water supply/return temperatures are 95/80°C. As of Dec. 1995, a total cooling load of 49 MW_c was being served in 60 buildings with a total floor space of 508,000 square meters. The typical peak cooling demand in a commercial building is approximately 17% higher than the typical peak heating demand.

7.3 CHICAGO [7-7, 7-8]

In Chicago, Illinois, USA, Trigen-People's District Energy Corp. is developing a cogeneration district heating and cooling system to serve a large convention center and surrounding areas. As of 1995, the company had overhauled existing chiller capacity, taken over operation of the existing chillers and boilers and constructed a 32,000 cubic meter chilled water storage tank. The storage system allows all 35 MW_c of peak cooling demand to be generated with electric centrifugal chillers using off-peak power purchased from the local utility, supplemented by steam absorption chillers. Peak heating demand is 29 MW_{th} of steam, currently supplied using existing heat-only boilers.

Beginning in January 1997, expansion of the convention center will double heating and cooling loads to about 58 MW_{th} heating and 70 MW_c cooling. Three gas turbines will be installed to generate electricity, cooling and heating using a unique design. The gas turbine, motor/generator and ammonia screw compressor are all connected by a common shaft. The gas turbine provides the driving torque to the screw compressor and the motor/generator absorbs or provides the balance. Existing loads are served with 5.5°C chilled water, but the new cooling loads will be served with water containing a mix of nitrates and nitrites which depress the freezing point to -1°C.

Each of the three gas turbine units can produce 1.2 MW of electricity or 7.7 MW_c of 5.5°C chilled water, or a mix of electricity and chilled water. There is a common Heat Recovery Steam Generator (HRSG) which can generate steam from the exhaust gases and/or supplemental firing. The peak output of the HRSG including supplemental firing is about 23 MW_{th}.

7.4 TRENTON [7-9]

The Trenton, New Jersey, USA, district energy system was established in 1981 to provide heating, with the State of New Jersey's Capitol Complex and a large prison providing 85% of the thermal load. Electricity and district hot water are cogenerated in two 6 MW_e 20-cylinder diesel engines burning 95% natural gas and 5% fuel oil as pilot fuel. Most of the district heat is supplied as 175°C hot water, but 105°C and 205°C heating are also provided through separate supply and return pipes.

As district heating expansion marketing proceeded, it became clear that success in marketing heating would be increased if cooling service could also be provided, particularly for new government and commercial buildings. The prospect of CFC refrigerant phaseout increased interest in the possibility of district cooling. A small chilled water distribution loop was implemented in 1989, linking available excess chiller capacity in existing buildings with a chilled water storage system. The cooling facilities now consist of 3 two-stage absorption chillers supplying a total of 7.5 MW_c (using cogenerated heat), five electric centrifugal chillers supplying a total of 21 MW_c, and a 10,500 cubic meter chilled water storage tank. Peak heating demand is 50 MW_{th} and peak cooling demand is about 35 MW_c.

7.5 ST. PAUL [7-10, 7-11]

A new hot water district heating system serving downtown St. Paul, Minnesota, USA began operation in 1983 under the management of District Energy St. Paul, Inc., a private, non-profit corporation. This system replaced a steam district heating system dating to the early 1900s. Substantial improvements in plant and distribution efficiencies resulted, with the district heating system now serving 2 million square meters, or twice the previously served building space, for the same consumption of fuel.

The district heating system currently has a market share of 75% of the building floorspace in downtown St. Paul, and serves downtown offices, hotels, government buildings and stores, as well as hospitals, a housing complex and an industrial park adjacent to downtown. The supply temperature varies from 120°C at winter peak to 90°C during the summer. (The summer temperature is kept relatively high to meet industrial process requirements.)

The heating plant includes 3 coal/gas-fired steam boilers, 1 gas/oil-fired steam boiler and 2 gas/oil-fired hot water boilers. The central plant is backed up by a boiler plant at a customer site. The peak coincident heating demand is about 130 MW_{th}. An 860 kWe backpressure steam turbine cogeneration facility was installed in 1990 to provide in-house electricity requirements.

Construction of a new centralized district cooling system began in 1992, and distribution of chilled water began in 1993. The system started with two 7.9 MW_c electric centrifugal chillers, then added a 9,500 cubic meter chilled water storage system and two 1.8 MW_c steam absorption chillers using 1.4 bar exhaust from the cogeneration backpressure turbine. As of 1995, plant capacity was about 33 MW_c, serving an aggregate contract load of 31 MW_c in 500,000 square meters of building space. Plant capacity is expected to increase to 40 MW_c during 1996.

7.6 GERMANY [7-12]

Concern about the global environmental impacts of CFC refrigerants has contributed to a loss of dominance of electric compression chillers in the German cooling market. Today, district heating utilities and individual consumers are making increasing use of absorption chiller systems for meeting cooling requirements. Absorption chillers represent almost 50% of recent chiller installations. Most of the installations are one-stage, but two-stage absorption and direct-fired lithium bromide absorption chillers are also being installed. A variety of types of driving energy are used, including district hot water ((85-120°C), steam (2-9 bar) and exhaust gases from gas turbines and reciprocating engines. One plant is now using adsorption chillers with silica gel absorbent. Characteristic projects include:

Chemnitz -- In the early 1970s a district cooling system (6 °C) was constructed with 8.4 MW_c cooling capacity for eight customers. After 1990 the four centrifugal and two reciprocating compressors with CFC-12 were replaced by two one-stage lithium bromide absorption chillers driven with district hot water (120/100°C).

Mannheim -- Mannheimer Versorgungs- und Verkehrsgesellschaft (MVV) is a public utility supplying district heating, electricity, gas and water in the Rhine-Neckar region. It has installed two one-stage lithium bromide absorption chillers to produce 6°C chilled water at customer sites. The chillers are driven with district hot water (86/75°C) produced by a 800 MW_e CHP plant. The chillers were financed by MVV and will remain MVV's property through the duration of the supply contract.

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Appendix A
Convesion Factors

	To convert	to	Multiply by
ENERGY	kiloWatt-hour	Btu	3413
	MegaWatt-hour	million Btu	3.413
	kiloWatt-hour	ton-hour	0.284
	ton-hour	Btu	12000
	MegaWatt-hour	GigaJoule	3.6
	kiloWatt-hour	MegaJoule	3.6
	kiloWatt-hour	kcal	860
POWER	kiloWatt	Btu/hour	3413
	MegaWatt	million Btu/hour	3.413
	kiloWatt	tons	0.284
	tons	Btu/hour	12000
	MegaWatt	MegaJoule/second	1
	kiloWatt	kiloJoule/second	1
	kiloWatt	kcal/hour	860
PRESSURE	bar	pounds per square inch	14.5
	bar	MegaPascals	0.1
TEMPERATURE	degree C	degree F	F = (C * 1.8) +32
VOLUME	cubic meter	cubic feet	35.32
	liter	gallon (U.S.)	0.264
LENGTH	meters	feet	3.28
	millimeters	inches	0.03937
AREA	square meters	square feet	10.764
COST	cost/kWhc	cost/ton-hour	3.52
	cost/kWh	cost/million Btu	293

Appendix B
Currency Exchange Rates

The following exchange rates were in effect at the time of publication (March 1996):

To convert U.S. dollars to	Multiply by
Danish Krone	5.7
Finnish Markka	4.6
German Mark	1.48
1000 Korean Won	0.78
Dutch Guilder	1.65
Norwegian Krone	6.42
Swedish Krona	6.63
British Pound	0.65
Canadian dollar	1.36
European currency unit	0.80

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