# Reference Manual for *Combined Heat and Power Systems*



Professional Level Elective Module of Singapore Certified Energy Manager (SCEM) Programme

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For

National Environment Agency Singapore

By LJ Energy Pte Ltd Singapore

# **Table of Contents**

Acknowledgements 2
PREFACE 1
1.0 INTRODUCTION TO COMBINED HEAT AND POWER (CHP) SYSTEMS       2         1.1. Introduction       2
1.2. Benefits of CHP Systems
1.3. Main Candidates for CHP 4
1.4. Available Technologies for CHP 4
2.0 FUNDAMENTAL THERMODYNAMIC CONCEPTS       12         2.1 Basic Thermodynamics       12
2.2 SI Units and Conversions
2.3 First and Second Laws of Thermodynamics23
2.4 The First Law Applied to Control Volumes
2.6 Entropy
2.7 Illustrative Examples
3.0 STEAM PROPERTIES       47         3.1 Introduction       47
3.2 Saturation Temperature
3.3 Sensible and Latent Heat
3.4 Dryness Fraction
3.5 Superheated Steam
3.6 Steam Pressure vs Volume
3.7 Enthalpy
3.8 Steam Pressure vs Enthalpy of Evaporation
3.9 Condensate and Flash Steam
3.10 Use of Steam Tables
4.0 GAS POWER CYCLES       62         4.1 Gas Power Cycles       62
4.2 Gas Compressors
4.3 The Carnot Cycle
4.4 The Brayton Cycle
4.5 Illustrative Examples
5.0 VAPOUR POWER CYCLES

	5.2 The Carnot Vapour Cycle	93
	5.3 The Rankine Cycle	95
	5.4 Regeneration	108
	5.5 Illustrative Examples	113
(	6.1. Introduction	123 123
	6.2 Fuel-to-Power Equipment	129
	6.3 Heat-to-Power Equipment	147
	6.4 CHP Prime Mover Comparisons	151
	6.5. CHP Plant System Requirements	158
-	7.0 CO-GENERATION & TRI-GENERATION SYSTEMS 7.1. Ideal Co-generation Systems	161 161
	7.2 Practical Co-generation Systems	164
	7.3 Tri-generation Systems	166
	7.4 Combined Gas-vapour Cycle Co-generation Systems	167
	7.5 Performance Analysis of CHP Systems	169
	7.6 Illustrative Examples	176
8	3.0 THERMAL AND ELECTRICAL DESIGN OF CHP SYSTEMS 8.1 Introduction	187 187
	8.2 Thermal Load Profiles	188
	8.3. Electrical Load Profiles	190
	8.4 CHP System Configuration Options	193
ę	9.0 THE FEASIBILITY STUDY FOR CHP 9.1 Introduction	204 204
	9.2 CHP Simulation Software and Tools	206
	9.3 CHP Analysis of Existing Facility	208
	9.4 CHP Economic Analysis	216
	9.5 Calculating Estimated Energy Use and Cost of a CHP plant	227
	9.6 CHP System Budgetary Construction Costs	229
	9.7 Calculating Life Cycle Cost (LCC)	230
	10.0 THE CHP REGULATORY ISSUES IN SINGAPORE 10.1 Introduction	233 233
	10.2 Basic Terminologies and Key Players in the Singapore Electricity Market	233
	10.3 Key Players in the Singapore Electricity Industry	235
	10.4 Singapore Electricity Market and Electricity Generators	237

10.5 The Electricity Generators in Singapore	238
10.6 Factors Affecting Co-generation Plant Development	239
10.7 Treatment of Embedded Generators	257
10.8 CHP Registration Procedures in Singapore	259
11.0 COMBINED HEAT AND POWER (CHP) SYSTEMS CASE STUDIES	262 262
11.2 Case Study 1: A 2 MW Tri-generation (CHP) System in a Pharmaceutical Company A Singapore	A in 263
11.3. Case Study 2: A 10 MW Tri-generation (CHP) System in a Pharmaceutical Companin Singapore	y B 266
11.4. Case Study 3: A 6.3 MW Tri-generation (CHP) System in an Industrial Company C i Singapore	n 269
APPENDIX	276

# PREFACE

The Singapore Certified Energy Manager (SCEM) programme offers a formal training and certification system for energy managers in Singapore and is co-administered by the National Environment Agency (NEA) and The Institution of Engineers, Singapore (IES) since 2008. The programme equips facility managers, engineers, technicians and others who intend to build their careers as energy professionals with the technical skills and competencies needed to manage energy services within their organisations.

The Combined Heat and Power (CHP) Systems module is one of the elective modules in the SCEM programme. This reference manual aims to help SCEM candidates with their course work and serve as reference material for practising energy managers.

The reference manual contains eleven chapters on different aspects of how to design, implement, operate and evaluate typical CHP systems to be energy efficient subject to Singapore regulations. Each chapter includes a brief introduction to assist readers who may not be familiar with some of the basic concepts associated with each topic, and the expected learning objectives.

Chapters 1 to 5 cover introduction to CHP systems, fundamentals of thermodynamics including steam properties, and the building blocks of CHP systems - namely gas and vapour power cycles.

Chapters 6 to 8 provide a fairly good account of the prime movers used in CHP, co-generation and tri-generation systems as well as the various considerations to be followed in the thermal design of CHP systems. This is followed by Chapter 9 which gives a very good account of a technical and economic feasibility study that one should undertake prior to decision making with regard to the CHP system.

Chapter 10 provides an account of the Singapore authorities' regulatory requirements for the installation, testing, commissioning and operation of CHP systems in Singapore.

Chapter 11 contains a number of CHP success stories in the form of case studies.

We would like to take this opportunity to thank all those who have assisted in the preparation of this reference manual by providing technical information, images and case studies.

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# 1.0 INTRODUCTION TO COMBINED HEAT AND POWER (CHP) SYSTEMS

Combined Heat and Power (CHP) systems produce two or three useful outputs simultaneously. If the CHP system produces two simultaneous outputs, the system is known as a co-generation system. On the other hand, if it produces three useful outputs simultaneously, it is known as a tri-generation system. Generally, the simultaneous outputs are: electricity, heating and/or cooling required in an industrial set-up. The CHP system generally consists of a fuel handling system, combustion equipment as well as a prime mover and other conversion equipment like heat recovery steam generator and absorption chillers depending on the application.

This chapter provides an introduction to CHP systems used in industrial facilities.

# Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. The benefits of CHP systems
- 2. Potential industrial candidates for CHP systems
- 3. The different types of CHP systems

# 1.1. Introduction

In a conventional power plant, only a portion of the energy transferred from the fuel to the working fluid is converted to work. The remaining portion of the energy is rejected as waste heat to larger heat sinks like rivers, oceans or the atmosphere. The Sankey diagram for a conventional power plant is shown in Figure 1.1. As can be seen from the figure, the rejected heat is about 67% of the total energy input and is wasted unless it can be recovered and used for other heating applications.





Some manufacturing plants such as chemical, pharmaceutical, oil refining, steel manufacturing and food processing require significant amounts of heat energy. These industrial plants also consume a large amount of electrical energy. Therefore, in some cases, it can be economically viable to generate electricity and divert the waste heat generated for useful heating applications.

Such a system is commonly known as a CHP system. The schematic diagram and Sankey diagram for such a system are shown in Figures 1.2 and 1.3, respectively. The overall efficiency can be improved from 33% (Figure 1.1) to 75% or higher (Figure 1.3).



Figure 1.2 Schematic diagram of an ideal CHP plant



Figure 1.3 Sankey diagram for a CHP power plant

# 1.2. Benefits of CHP Systems

Some of the key benefits of a CHP system include:

- Increased total system thermodynamic efficiency
- Lower overall energy cost

- Improved power supply reliability (Singapore is an exception)
- Reduced overall CO<sub>2</sub> emissions
- Possible reduced investment in power transmission capacity (for non-critical applications)
- Lower transmission losses

#### 1.3. Main Candidates for CHP

CHP systems can benefit a wide variety of facilities and are widely used. Such facilities include:

- Oil refineries and Petrochemical Plants
- Waste water treatment plants
- Pharmaceutical plants

#### 1.4. Available Technologies for CHP

There are many technologies available in the market for combined heat and power generation. They can generally be classified as large, medium and small size systems based on their electrical power and heat output.

- Large size systems Steam turbine and combustion turbine (≥5 MW)
- Medium size systems Internal combustion engine (0.25 5 MW)
- Small size systems Microturbines (<250 kW)

A description of each type of system is provided below.

#### 1.4.1 Steam Turbine

The steam turbine power generation cycle has been the backbone of the power generation industry. There are three major components in a steam turbine power generation system: the heat source (the boiler which produces superheated steam), the turbine itself and a heat sink. The steam turbines are also categorised based on the exit steam conditions as either a back pressure or condensing steam turbine system.

#### 1.4.1.1 Back Pressure Turbine

In a back pressure steam turbine (Figure 1.4), the superheated steam produced by a boiler is supplied to a turbine where part of its energy is extracted to produce work. The low pressure steam available at the outlet of the turbine is then used for heating applications including the operation of absorption chillers for cooling. The condensate is usually recovered and returned back to the boiler.



Figure 1.4 Back pressure steam turbine system

The main advantage of such a system is the high overall efficiency because process heating applications are used as a heat sink to eliminate the need for a condenser which would otherwise reject heat to the environment.

One of the main disadvantages is that the steam flow rate and therefore the system output is governed by the thermal load, which is the heat sink. Hence, this system provides little flexibility for matching the electrical output to the actual electrical load. Another disadvantage of the back-pressure steam turbine is that it tends to be larger in size (for the same electrical output) compared to the turbines such as total condensing turbine as the reduction in enthalpy of steam is lower. It is to be noted that the power produced by the turbine is the product of the steam mass flow rate and difference in enthalpy. Hence, for the same power output, the steam mass flow rate becomes higher resulting in a larger turbine.

Back pressure steam turbines are generally used for applications that have a heat to power ratio of 4.0 to 14.3 kW<sub>t</sub>/kW<sub>e</sub> and an electrical output of more than a few MW.

# 1.4.1.2 Condensing Steam Turbine

The operation of a condensing turbine is similar to a back pressure turbine except that steam is exhausted from the turbine at a pressure lower than the atmospheric pressure to maximise the electrical output (Figure 1.5).

Since the steam available at the exit of the turbine would not be suitable for industrial applications, steam required for heating applications (including absorption chillers) is taken from an intermediate stage of the turbine. The low-pressure steam at the exit of the turbine is exhausted to a condenser, which acts as the main heat sink.



Figure 1.5 Condensing steam turbine system

Condensing steam turbines are generally used for applications where the heat to power ratio is in the range of 2.0 to  $10.0 \text{ kW}_t/\text{kW}_e$  and the electrical output is more than a few MW.

# 1.4.2 Combustion Turbine (Gas Turbine)

In the gas turbine power generation cycle, air enters the compressor at atmospheric pressure where it is compressed. It then enters a constant-pressure combustion chamber where fuel is injected. The exhaust gases then leave the combustor at a high temperature of around 1200°C and enter a gas turbine coupled to a generator. The hot gases after expansion leave the turbine at a temperature of about 450-600°C. The arrangement of a typical system is shown in Figure 1.6. The power generation efficiency of gas turbine systems is relatively low (about 35%) due to the need to discharge high temperature gases with a considerable amount of heat energy at the exit of the turbine.



#### Figure 1.6 Open cycle gas turbine cycle

The overall efficiency of the above system can be improved considerably if the waste heat available at the exit of the turbine is used for heating applications. This is usually achieved by directing the hot gases after the expansion process to a waste heat boiler. The steam produced by the waste heat boiler can be used directly for heating applications or in high capacity systems. The high-pressure steam can be expanded through a steam turbine coupled to a generator prior to using the low pressure steam for heating applications. A typical arrangement of a CHP system using a combustion gas turbine is shown in Figure 1.7.



Figure 1.7 CHP system with combustion turbine

Combustion turbine systems are generally used for applications where the heat to power ratio is in the range of 1.3 to  $2.0 \text{ kW}_t/\text{kW}_e$  and the electrical output is more than a few MW.

# 1.4.3 Internal Combustion Reciprocating Engines

The internal combustion engine works on the principle of the Otto cycle or the Diesel cycle. The Otto cycle is made up of four stages as shown in Figure 1.8: the intake stroke where a mixture of air and fuel is taken into a cylinder, the compression stroke where the mixture is compressed by a piston, the power stroke where the fuel-air

Air-fuel mixture Exhaust gases Δ ... Air-fuel mixture 4-1 2-3 Ш 1-2 3-4 (b) (c) (a) (d) Compression Power stroke Intake stroke Exhaust stroke stroke

mixture is ignited and expanded moving a piston which does mechanical work followed by the final stage where the exhaust is released to the atmosphere.

Figure 1.8 Four stages in an Otto cycle

The internal combustion engines operating on the Otto cycle can be fuelled by a wide variety of fuels. These include liquids such as gasoline and heavy oil as well as gases such as propane, biogas or natural gas.

In an internal combustion engine operating on the Diesel cycle, diesel based fuels are used instead of petroleum based fuels. The main difference in the cycle is that the fuel-air mixture is ignited by the heat generated by the compression.

Internal combustion engines can be coupled to generators for power generation. However, the overall operating efficiency is relatively low (20 to 32%) as a significant portion of the heat input from the fuel is wasted in the form of hot exhaust gases and jacket cooling.

Therefore, the overall operating efficiency of an internal combustion engine-driven power generation system can be improved significantly by operating as a combined heat and power generation system.

Figure 1.9 shows the arrangement of a typical CHP system using an internal combustion engine where hot water from jacket cooling is used for operating an absorption chiller and the exhaust gases are used to generate steam for process heating applications.



Figure 1.9 Arrangement of an internal combustion engine CHP system

The general sizes of internal combustion engines that can be used for such applications range from about 50 kW to around 5 MW. The heat to power ratio for such systems ranges from 1.1 to  $2.5 \text{ kW}_t/\text{kW}_e$ .

# 1.4.4 Micro Turbines

Micro turbines are essentially a smaller version of combustion turbine generators. They are available in output sizes ranging from 25 to 250 kW. Micro turbines are also designed to operate on a variety of gaseous fuels such as natural gas, biogas and propane. When used purely for electricity generation, their operating efficiency tends to be relatively low (20 to 30%).

Due to the high heat content of the exhaust gases, the overall efficiency of power generation using micro turbines can be improved by using the waste heat from the exhaust for heating and cooling applications.

A typical arrangement of a CHP system using micro turbines is shown in Figure 1.10.



Figure 1.10 Arrangement of a micro turbine CHP system

Single micro turbines can produce electrical power output up to about 250 kW. The typical heat to power ratio for such systems range from 1.4 to 2.5 kWt/kWe as summarised in Table 1.1 below.

Technology Steam Turbine Combustion Turbine		Combustion Turbine	Internal combustion Engine	Micro turbine
Size (MW)	0.5 - 150	1 - 200	0.05 - 10	0.025 - 0.25
Electric25 - 4025 - 40 (simpleEfficiency (%)40 - 60 (combine cycle)		25 - 40 (simple) 40 - 60 (combined cycle)	30 - 50	20 - 30
Overall system efficiency (%)	60 - 80	75 - 80	75 - 85	65 - 80
Typical uses of heat recovery	Hot water, LP steam, cooling	Hot water, LP-HP steam, cooling	Hot water, LP steam, cooling	Hot water, LP steam, cooling
Fuels	Solid, liquid and gaseous fuels	Natural gas, biogas, propane, distillate oil	Diesel and fuel oil	Natural gas, biogas, propane
Relative operational & maintenance	High	High	Low	Medium

expertise required				
Relative capital cost for system	High	High	Medium	Low
Heat to power ratio	2.0 - 10.0	1.3 – 2.0	1.1 - 2.5	1.4 - 2.5

Table1.1 Summary of the common technologies available for CHP

#### Summary

A CHP system is the simultaneous production of two or more useful outputs, namely electricity, heat and cooling. Different types of prime movers are associated with CHP systems. These prime movers include gas turbines, steam turbines, and reciprocating engines as well as micro turbines. Further details of CHP systems will be presented in the subsequent chapters of this reference manual.

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# 2.0 FUNDAMENTAL THERMODYNAMIC CONCEPTS

Combined Heat and Power (CHP) systems consist essentially of thermodynamic equipment in which the working fluid undergoes various thermodynamic processes. These processes are based on fundamental principles involving laws of conservation of energy and mass involving thermodynamic properties. A thermodynamic process can also be considered as a series of state points that the working fluid in the process undergoes. Therefore, it is of paramount interest for energy managers when handling CHP systems to understand the fundamental thermodynamic concepts.

This chapter provides the fundamental thermodynamic concepts behind CHP systems. This Chapter will follow SI units and conversion of the relevant units in this reference manual to other units is given.

#### Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. Basic thermodynamics
- 2. Relevant SI units and their conversions
- 3. First and Second Laws of thermodynamics as well as work and heat
- 4. Concept of entropy

# 2.1 Basic Thermodynamics

In thermodynamics, the storage, transformation and transfer of energy are studied. Energy can be stored as internal energy (by the temperature difference), kinetic energy (by virtue of change in velocity), potential energy (due to the change in elevation), and chemical energy (due to chemical composition). The energy is transformed from one of these forms to the another and at times transferred across a boundary as either heat or work. The fundamental understanding of thermodynamics will enable engineers to carry out the analysis or design of large scale systems like CHP systems, Heating, Ventilation and Air-conditioning (HVAC) systems and even nuclear power plants. A thermodynamic system may be regarded as a continuum in which the activity of the constituent molecules is averaged into measurable quantities such as pressure, temperature and velocity. The important fundamental thermodynamic terminologies are as follows:

# 2.1.1 System

A thermodynamic system is a definite quantity of matter contained within some closed surface. The surface is usually an obvious one like that enclosing the gas in the cylinder as shown in Figure 2.1. It may also be treated as an imaginary boundary like the deforming boundary of a certain amount of mass as it flows through a pump.

# 2.1.2 Surroundings

In thermodynamics, all matter and space external to a system is collectively called its surroundings. Thermodynamics is mainly concerned with the interactions of a system and its surroundings, or interaction between two or more systems. Normally, a system interacts with its surroundings by transferring energy across its boundary.



Figure 2.1 Arrangement of a typical pumping system

# 2.1.3 Boundary

The real or imaginary surface that separates the system from its surroundings. The boundary of a system can either be fixed or movable.

Thermodynamic systems are classified as **closed** or **open** systems.

# 2.1.4 Closed system

In a closed system, a fixed amount of mass exists and no mass can cross its boundary.

#### 2.1.5 Open system (control volume)

An open system is a properly selected region of interest in space. It usually encloses a device that involves a certain mass flow, e.g. a compressor, turbine, or nozzle. In many practical cases, an analysis is simplified if attention is just focused on this control volume in which there is a flow of working fluid, e.g. pumps and turbines. It is to be noted that both mass and energy can cross the boundary of a control volume.

#### 2.1.6 Control surface

The boundary of a control volume is called a control surface. It can be real or imaginary.

# 2.1.7 Property

Property is any characteristic of a system. A property is any quality that helps to describe a system. The properties determine the state of a system, which is a condition as described by the properties at a particular instant during a process or at idling. In thermodynamics, the common properties are pressure, temperature, volume, velocity, and position. In addition to the above properties, other properties must also be considered occasionally. Shape is important in thermodynamic analysis when surface effects are significant; whereas colour is important when radiation heat transfer is involved. Properties can be further classified as **intensive** or **extensive**.

#### 2.1.8 Intensive properties

Intensive properties are those independent of system size or the amount of material in a system, e.g. temperature, pressure, and density.

#### 2.1.9 Extensive properties

Extensive properties are those dependent on the size of the system or quantity of matter in a system, e.g. volume, number of particles and internal energy.

#### 2.10 Specific properties

Specific properties are the ratio of the properties to the mass of the substance.

# 2.11 Enthalpy

Enthalpy is the total heat content in a stream of fluid. The specific enthalpy is the enthalpy expressed per unit mass.

#### 2.12 Entropy

Entropy is a thermodynamic property which is helpful in assessing the energy quality during a thermodynamic process. Entropy is explained in detail in section 2.6.

#### 2.13 Thermodynamic equilibrium

Thermodynamic equilibrium exists when the properties are assumed to be constant from point to point and when there is no tendency to change with time. If the temperature, say, is suddenly increased at some part of the system boundary, spontaneous redistribution is assumed to occur until all parts of the system are at the same temperature and a new thermodynamic equilibrium is reached. In basic thermodynamic analysis, when the temperature or the pressure of a system is referred to, it is assumed that all points of the system have the same, or essentially the same, temperature or pressure.

#### 2.14 Process

A thermodynamic process is a series of successive state points - the path through which the system passes. If during the process the deviation from equilibrium is infinitesimal, a quasi-equilibrium process occurs and each state in the process may be idealised as an equilibrium state.

# 2.15 Cycle

When a system in a given initial state experiences a series of quasi-equilibrium processes and returns to the initial state, the system undergoes a cycle. At the end of the cycle, the properties of the system have the same values they had at the beginning.

#### 2.16 Isothermal process

An isothermal process is one in which the temperature is held constant. It is to be noted that the prefix iso- is attached to the name of any property that remains unchanged in a process.

#### 2.17 Adiabatic process

A process in which there is no heat transfer between a system and its surroundings.

# 2.18 Isobaric process

An isobaric process is one in which the pressure remains constant.

#### 2.19 Isochoric process

An isochoric process is one in which the volume remains constant.

Dimension	Unit
Length	metre (m)
Mass	kilogramme (kg)
Time	second (s)
Temperature	kelvin (K)
Electric current	ampere (A)
Amount of light	candela (cd)
Amount of matter	mole (mol)

Table 2.1 Fundamental dimensions and their units

#### 2.2 SI Units and Conversions

Any physical quantity can be characterised by dimensions. Units are the magnitudes assigned to the dimensions. The dimensions are classified as fundamental dimensions and derived dimensions.

#### 2.2.1 Fundamental dimensions

Fundamental dimensions describe the fundamental properties of a substance or system, e.g. mass, length, time and temperature. The units assigned to fundamental dimensions are called fundamental units, e.g. kilogramme (kg), metre (m), second (s) and degree Celsius (°C).

#### 2.2.2 Derived dimensions

Derived dimensions are the dimensions derived from the fundamental dimensions, e.g. momentum, speed, volume and work. The units assigned to derived dimensions are called derived units, e.g. kg.m/s, m/s, m<sup>3</sup>, kN.m.

#### 2.2.3 SI system of units

A simple and logical system based on a decimal relationship between the various units, e.g. kilogramme (kg), metre (m), second (s) and degree Celsius (°C).

Multiple	Prefix
10 <sup>12</sup>	tera, T
10 <sup>9</sup>	giga, G
10 <sup>6</sup>	mega, M
10 <sup>3</sup>	Kilo, k
10 <sup>2</sup>	hecto, h
10 <sup>1</sup>	deca, da
10 <sup>-1</sup>	deci, d
10-2	centi, c
10 <sup>-3</sup>	milli, m
10 <sup>-6</sup>	micro, µ
10 <sup>-9</sup>	nano, n
10 <sup>-12</sup>	pico, p

Table 2.2 Prefixes in SI units

Quantity	Symbol	SI Units	English Units	Multiplication factor
				for English to SI
				conversion
Length	L	m	Ft	0.3048
Mass	М	kg	lb	0.4536
Time	t	S	sec	
Area	А	m²	ft <sup>2</sup>	0.09290
Volume	V	m³	ft <sup>3</sup>	0.02832
Velocity	v	m/s	ft/sec	0.3048
Acceleration	а	m/s²	ft/sec <sup>2</sup>	
Angular velocity	ω	rad/s	S <sup>-1</sup>	
Weight, Force	W, F	Ν	lbf	4.448
Density	ρ	kg/m³	lb/ft <sup>3</sup>	16.02
Specific weight	w	N/m <sup>3</sup>	lbf/ft <sup>3</sup>	157.1
Pressure,	Р, т	kPa	lbf/ft <sup>2</sup>	0.04788
Stress	W, E	J	ft-lbf	1.356
Work, Energy	Q	J	Btu	1055.0
Heat transfer	Р	W	Ft-lbf/sec	1.356
Power	Ż	W or J/s	Btu/s	1055.0
Heat transfer	ṁ	kg/s	lb/sec	0.4536
rate	С	kJ/kg.K	Btu/lb.⁰R	4.187

Mass flow rate	<i>॑</i>	m³/s	ft <sup>3</sup> /sec	0.02832
Specific heat	h	kJ/kg	Btu/lb	2.326
Volume flow	S	kJ/kg.K	Btu/lb.ºR	4.187
rate	$\dot{v}$	m³/kg	ft <sup>3</sup> /lb	0.06242
Specific				
enthalpy				
Specific entropy				
Specific volume				

Table 2.3 Conversion multipliers from English to SI units

# 2.2.4 Density, Specific volume, Specific weight

The density of a substance is defined as the mass of the substance per unit volume. The reciprocal of density is called specific volume (that is volume per unit mass).

Mathematically, Density, $\rho = m/V$	(2.1)
---------------------------------------	-------

Therefore, the Specific volume, 
$$v = 1/\rho$$
 (2.2)

Associated with (mass) density is **weight density or specific weight w**, which is defined as the weight per unit volume.

Weight density, 
$$\rho_d = W/V$$
 (2.3)

Specific weight is related to density through W =mg as follows:

 $\rho_d = mg/m/\rho = \rho g$ (2.4)
where, g is the acceleration due to gravity, m/s<sup>2</sup>

For water, nominal values of density ( $\rho$ ) and specific weight ( $\rho_d$ ) are 1000 kg/m<sup>3</sup> and 9810 N/m<sup>3</sup>, respectively.

For air at sea level, the respective values are 1.21 kg/m<sup>3</sup> and 11.86 N/m<sup>3</sup>.

#### 2.2.5 Pressure

Pressure is defined as the force per unit area. The SI unit for pressure is Pa  $(N/m^2)$  or kPa  $(kN/m^2)$  and the working units of pressure are bar (SI) and PSI (British).

(2.5)



Pressure variation with elevation is expressed as:

Figure 2.2 Variation of pressure with elevation

In most thermodynamic relations, absolute pressure must be used. Absolute pressure is measured pressure, or gauge pressure, plus the local atmospheric pressure:

$$P_{abs} = P_{gauge} + P_{atm} \tag{2.6}$$

A negative gauge pressure is often called a vacuum, and gauges capable of reading negative pressures are called vacuum gauges. A gauge pressure of -50 kPa would be referred to as a vacuum of 50 kPa, with the sign omitted. The relationships between absolute and gauge pressures are shown in Figure 2.3. The word "gauge" is generally used in statements of gauge pressure; e.g. P =200 kPa gauge. If "gauge" is not present, the pressure will, in general, be an absolute pressure. Atmospheric pressure is an absolute pressure and will be taken as 100 kPa (at sea level), unless otherwise stated. It should be noted that atmospheric pressure is highly dependent on elevation. It is only about 53 kPa in a mountain city with an elevation of 4000 m.



Figure 2.3 Relationship between absolute and gauge pressure

#### 2.2.6 Temperature

Temperature is a measure of molecular activity. In thermodynamics, to understand the temperature, typically equality of temperature is discussed rather than defining the temperature at a macroscopic level.

Equality of Temperatures: Let two objects be isolated from the surroundings but placed in contact with each other. If one object is hotter than the other, the hotter object will become cooler and the cooler object will become hotter; both objects will undergo change until all properties (e.g. electrical resistance) of the bodies cease to change. Such a state of the two objects is known as thermal equilibrium. Hence, we state that two systems have equal temperatures if no change occurs in any of their properties when the systems are brought into contact with each other. In other words, if two systems are in thermal equilibrium, their temperatures are postulated to be equal. In thermodynamics, the above observation is referred to as the zeroth Law of Thermodynamics. The zeroth Law of Thermodynamics states that if two systems are equal in temperature to a third system, they are equal in temperature to each other.

# 2.2.6.1 Relative Temperature Scale

To establish a temperature scale, it is necessary to choose the number of subdivisions, called degrees, between two fixed, easily duplicable points - the ice point and the steam point. The ice point exists when ice and water are in equilibrium at a pressure of 101 kPa; the steam point exists when liquid water and its vapour are in a state of equilibrium at a pressure of 101 kPa. On the Fahrenheit scale, there are 180 degrees between these two points. On the Celsius (formerly called the Centigrade)

scale, there are 100 degrees. On the Fahrenheit scale, the ice point is assigned the value of 32 and on the Celsius scale it is assigned the value of 0.

Based on the above concepts, the conversion between degree Fahrenheit and degree Celsius is expressed as follows:

$$^{\circ}F = 1.8^{\circ}C + 32$$
 (2.7)

$$^{\circ}C = (^{\circ}F - 32)/1.8 \tag{2.8}$$

#### 2.2.6.2 Absolute Temperature Scale

The Second Law of Thermodynamics allows us to define an absolute temperature scale.

The relations between absolute and relative temperatures are:	
$T_R = t_F + 459.6$	(2.9)

$$T_k = t_c + 273.15 \tag{2.10}$$

where the subscript "F" refers to the Fahrenheit scale and the subscript "C" refers to the Celsius scale. (The values 460 and 273 are used where precise accuracy is not required.) The absolute temperature on the Fahrenheit scale is given in degrees Rankine (R), and on the Celsius scale, it is given in kelvins (K). Note: 300 K is read "300 kelvins," not "300 degrees Kelvin."

# 2.2.7 Energy

Energy is the capacity to do work. As a result, it takes the same unit as work. A system may possess several different forms of energy. Assuming uniform properties throughout the system, the kinetic energy, which is by virtue of motion, is given by:

$$KE = 1/2mV^2$$
 (2.11)

where V is the velocity of each lump of substance, assumed constant over the entire system. If the velocity is not constant for each lump, then the kinetic energy is found by integrating over the system.

The energy that a system possesses by virtue of its elevation h above some arbitrarily selected datum is its potential energy; it is determined from the equation:

PE = mgh

(2.12)

Where, m = Mass, kg g = Acceleration due to gravity, 9.81 m/s<sup>2</sup>

Other forms of energy include the energy stored in a battery, energy stored in an electrical condenser, electrostatic potential energy, and surface energy. In addition, there is the energy associated with the translation, rotation, and vibration of the molecules, electrons, protons, and neutrons, and the chemical energy due to bonding between atoms and between sub-atomic particles. These molecular and atomic forms of energy will be referred to as internal energy and designated by the letter U. In combustion, energy is released when the chemical bonds between atoms are rearranged.

In thermodynamics, one needs to focus first on the internal energy associated with the motion of molecules that is influenced by various macroscopic properties such as pressure, temperature, and specific volume. Internal energy, like pressure and temperature, is a property of fundamental thermodynamic importance. A substance always has internal energy, if there is molecular activity associated with it. In thermodynamics, one does not need to know the absolute value of internal energy, since the interest is only in its increase or decrease. In thermodynamics, an important law that is often used when considering isolated systems, is the Law of Conservation of Energy.

The Law of Conservation of Energy states that the energy of an isolated system remains constant. Energy can neither be created nor be destroyed in an isolated system. Energy can only be transformed from one form to another. Let us consider the system comprising two automobiles that hit head on and come to rest. As the energy of the system is the same before and after the collision, the initial KE must simply have been transformed into another kind of energy. In this case, internal energy is primarily stored in the deformed metal after the collision.

#### 2.3 First and Second Laws of Thermodynamics

The First Law of Thermodynamics, simply called the First Law, can be stated as "the net heat transfer is equal to the net work done for a system undergoing a cycle." This is expressed in equation form as:

$$\Sigma W = \Sigma Q \tag{2.13}$$

or, using the symbol to represent integration around a complete cycle as:

$$\delta W = \delta Q \tag{2.14}$$

The First Law can be illustrated by considering the following experiment. Let a weight be attached to a pulley-paddle-wheel setup, such as that shown in Figure 2.4. Let the weight fall a certain distance thereby doing work on the system contained in the tank shown. The work done is equal to the weight multiplied by the distance dropped. The temperature of the system (the substance in the tank) will immediately rise by an amount  $\Delta T$ . Now, the system is returned to its initial state (the completion of the cycle) by transferring heat to the surroundings, as implied by the Q in the Figure. This reduces the temperature of the system to its initial temperature. Applying the First Law, this heat transfer will be exactly equal to the work that was done by the falling weight.



Figure 2.4 The First Law applied to a cycle

The First Law of Thermodynamics is often applied to a process as the system changes from one state to another. Realising that a cycle results when a system undergoes two or more processes and returns to the initial state, we could consider a cycle composed of two processes.

Applying the First Law to this cycle involving these processes and after some mathematical manipulations, Eq. (2.14) can be expressed as:

$$\delta Q - \delta W = dE \tag{2.15}$$

For the two state points, the Eq. (2.15) can be written as:

$$Q_{1-2} - W_{1-2} = E_2 - E_1 \tag{2.16}$$

Where  $Q_{1-2}$  is the heat transferred into the system during the process from state 1 to 2,  $W_{1-2}$  is the work done by the system on the surroundings during the state change (process) and  $E_2$  and  $E_1$  are the quantity of energy at the state points 1 and 2, respectively.

The energy E consists of different forms of energy: kinetic energy, KE, potential energy, PE and internal energy, U. The internal energy includes chemical energy and energy associated with the atom. The internal energy for superheated steam depends only on pressure and temperature. The internal energy for saturated steam depends on temperature or pressure and quality of steam. The expression for internal energy connecting quality of steam is as follows:

$$u = u_f + x(u_g - u_f)$$
(2.17)

where, u is internal energy per unit mass with suffices f and g indicating liquid and vapour, respectively.

#### 2.3.1 Enthalpy

In the solution of problems involving systems, certain products or sums of properties occur with regularity. One such combination of properties is enthalpy H:

$$H = U + PV \tag{2.18}$$

where, U = Total internal energy, kJ P = Pressure, kPa

V = Volume, m<sup>3</sup>

The specific enthalpy, h, is found by dividing H by the mass: h = H/m. From Eq. (2.18):

h = u + Pv	(2.19)
------------	--------

where,

u = Internal energy, kJ/kg.K

- h = Specific enthalpy, kJ/kg
- P = Pressure, kPa
- v = Specific volume, m<sup>3</sup>/kg

Enthalpy is a property of a system and is also found in the steam tables. As in the case of internal energy, in thermodynamics it is only the change in enthalpy or internal energy that is important. Therefore, we can arbitrarily choose the datum from which to measure h and u. We choose saturated liquid at 0°C to be the datum point for water; there h = 0 and u = 0. From a practical point of view, the enthalpy can be treated as the total heat content in a stream of fluid or stream of air, water or steam, which are the common working fluids in most industries. This is illustrated in Figure 2.5 below.



Figure 2.5 The First Law applied to a cycle

Figure 2.5 shows a thermodynamic process in which a fluid stream enters with an enthalpy of  $h_1$  and emerges from the process with an enthalpy of  $h_2$ . If  $h_1$  is greater than  $h_2$ , based on the understanding of the enthalpy as total heat content, we can call the process cooling and the other way around, the process can be called heating.

#### 2.3.2 Latent Heat

The amount of energy that must be transferred in the form of heat to a substance held at constant pressure in order that a phase change occurs is called latent heat. It is the change in enthalpy of the substance at the saturated conditions of the two phases. The heat that is necessary to vapourise a unit mass of water at constant pressure is the latent heat of vapourisation and can be expressed as:

$$h_{fg} = h_g - h_f \tag{2.20}$$

where  $h_g$  is the enthalpy of saturated steam and  $h_f$  is the enthalpy of saturated liquid (obtained from the steam table for the corresponding pressure or temperature). Corresponding phase change and the associated heat rejected or absorbed for a solid is called latent heat of fusion or melting. When a solid changes phase directly to a gas, sublimation occurs. The latent heat of fusion and the latent heat of sublimation are relatively insensitive to pressure or temperature changes. For ice, the latent heat of fusion is approximately 330 kJ/kg and the latent heat of sublimation is about 2040 kJ/kg. The heat of vapourisation of water is very sensitive to pressure and temperature. Figures 2.6, 2.7 and 2.8 show the Pressure-volume (P-v) and Temperature-volume (T-v) diagrams for a pure liquid substance and the various regions while they undergo a phase change process by absorbing or rejecting the latent heat. The quality of the two-phase mixture in the two-phase region is defined by a parameter called quality or dryness fraction, x, defined as:

$$x = m_g/m \tag{2.21}$$

where,  $m = m_f + m_g$ ;  $m_g$  = the mass of vapour fraction and  $m_f$  = the mass of liquid fraction



Figure 2.6 P-v diagram for a pure liquid substance



Figure 2.7 T-v diagram for a pure liquid substance

Note that  $v_{cr}$  is the specific volume at the critical point of water

The various regions as shown in the P-v and T-v diagrams in Figures 2.6 and 2.7 are defined as follows:

Compressed liquid (subcooled liquid): A substance that is not about to vapourise Saturated liquid: A liquid that is about to vapourise

Saturated vapour: A vapour that is about to condense.

Saturated liquid–vapour mixture: The state at which the liquid and vapour phases coexist in equilibrium.

Superheated vapour: A vapour that is not about to condense (i.e., not a saturated vapour).

#### 2.3.3 Specific Heats

The specific heat capacity of a substance is the amount of heat required to raise the temperature of one kilogramme of that substance by 1°C. There are two types of specific heat; specific heat capacity at constant volume ( $C_v$ ) and constant pressure ( $C_p$ ). The expressions connecting specific heat capacity C, internal energy u, enthalpy h and gas constant R, are given below:

$$du = C_v \, dT \tag{2.22}$$

$$dh = C_p \, dT \tag{2.23}$$

$$C_p - C_v = R \tag{2.24}$$

$$C_p/C_v = \gamma \tag{2.25}$$

The gas constant R is related to the universal molar gas constant, R (kJ/K.mol) as follows:

R = R/M, where M is the molecular weight of the gas concerned. The universal gas constant R is 8.314 kJ/kg.K.

Also, 
$$C_p = R \gamma / (\gamma - 1), C_v = R / (\gamma - 1)$$
 (2.26)

Since R is a constant for an ideal gas, the specific heat ratio  $\gamma$  will depend only on the temperature.

For gases, the specific heats slowly increase with increasing temperature. Since they do not vary significantly over large temperature differences, it is often acceptable to treat  $C_v$  and  $C_p$  as constants.

For air, we will use  $C_v = 0.717$  kJ/kg.K and  $C_p = 1.00$  kJ/kg·K, unless otherwise stated. For more accurate calculations with air, or other gases, one should consult ideal-gas tables, which tabulate h(T) and u(T). For liquids and solids, the specific heat capacity  $C_p$  is available from temperature-specific capacity tables from any standard thermodynamic text book and can be used for the relevant computations. Since it is quite difficult to maintain constant volume while the temperature is changing,  $C_v$ values are usually not tabulated for liquids and solids; the difference  $C_p - C_v$  is quite small. For most liquids the specific heat is relatively insensitive to temperature change. For water we will use the nominal value of 4.19 kJ/kg·K.

#### 2.3.4 The constant-temperature process

By applying the First Law of Thermodynamics and the ideal gas equation, PV = mRT, the work done for an isothermal process is expressed as follows:

$$W = mRT \ln (P_2/P_1)$$
 (2.27)

#### 2.3.5 The constant-volume process

The work for a constant-volume quasi-equilibrium process is zero, since dV is zero. Hence, by applying the First Law and the definition of internal energy, one can quantify the amount of heat transferred during a constant volume process as:

$$Q = \dot{m} C_{\nu} \Delta T \tag{2.28}$$

#### 2.3.5 The constant-pressure process

Using the First Law and the definition of internal energy, one can quantify the amount of heat transferred during a constant pressure process as:

$$Q = \dot{m} C_p \Delta T \tag{2.29}$$

#### 2.3.6 The adiabatic process

There are numerous examples of processes where there is no or negligibly small heat transfer. In thermodynamics, such processes are called adiabatic processes. For an adiabatic process, the relevant expressions are:

$Tv^{(\gamma-1)} = \text{const.}$	(2.30)
$TP^{(1-\gamma)/\gamma} = \text{const.}$	(2.31)
$PV^{\gamma} = \text{const.}$	(2.32)

The work done during an adiabatic process is expressed as:

$$W = (P_2 V_2 - P_1 V_1) / (1 - \gamma)$$
(2.33)

For an adiabatic process, the ratio of specific heat,  $\gamma = 1.4$ .

#### 2.3.7 The polytropic process

In practical applications in industry, most of the processes follow the polytropic process in which there is some heat transfer, but do not maintain temperature, pressure or volume constant.

The work done during a polytropic process is expressed as:

$$W = (P_2 V_2 - P_1 V_1) / (1 - n)$$
(2.34)

where n is the polytropic index which is greater than 1 but less than 1.4.

#### 2.4 The First Law Applied to Control Volumes

The preceding sections focused on closed systems; those in which no mass crosses the boundary of a system. This is acceptable for many problems of interest and may, in fact, be imposed on the power plant schematic shown in Figure 2.8. However, if the First Law is applied to this system, only an incomplete analysis can be accomplished. For a more complete analysis one should relate W<sub>in</sub>, Q<sub>in</sub>, W<sub>out</sub>, and Q<sub>out</sub> to the pressure and temperature changes for the pump, boiler, turbine, and condenser respectively. For this, one should consider each device of the power plant as a control volume into which and from which a fluid flows. For example, water flows into the pump at a low pressure and leaves the pump at a high pressure; the work input into the pump is obviously related to this pressure rise. Hence, one must formulate equations that allow for making the necessary calculations. For most applications, it will be acceptable to assume a steady and uniform flow. In a steady flow, the flow variables do not change with time; whereas in a uniform flow, the velocity, pressure, and density are constant over the cross-sectional area.

#### 2.4.1 The conservation of mass

Many devices have an inlet (usually a pipe) and an outlet (also typically a pipe). Consider a device, a control volume, to be operating in a steady-flow mode with uniform profiles in the inlet and outlet pipes. During some time increment  $\Delta t$ , a small amount of mass  $\Delta m_1$  leaves the inlet pipe and enters the device, and the same amount of mass  $\Delta m_2$  leaves the device and enters the outlet pipe.

The amount of mass that enters the device can be expressed as:

$$m_1 = \rho_1 A_1 V_1 \Delta t \tag{2.35}$$

The amount of mass that leaves the device is expressed as:

$$m_2 = \rho_2 A_2 V_2 \Delta t \tag{2.36}$$

where,  $V_1$  and  $V_2$  are the velocities.

The quantity of mass entering or leaving a control volume per unit time is called mass flux or mass flow rate, which for steady state remains constant.

Therefore, for steady state processes, the mass flow rate, m, can be expressed as:

$$\dot{m} = m_1 / \Delta t = m_2 / \Delta t = \rho_1 A_1 V_1 = \rho_2 A_2 V_2 = \rho A V$$
 (2.37)

This expression in thermodynamics is known as conservation of mass. where,

V = velocity, m/s A = Area, m<sup>2</sup> P = Density, kg/m<sup>3</sup> m = mass, kg and  $\Delta t$  = time interval, s

#### 2.4.2 Applications of the energy equation

There are several points that must be considered in the analysis of most problems in which the energy equation is used. As a first step, it is very important to identify the control volume selected in the solution of the problems. If possible for simplicity, the control surface should be chosen so that the flow variables are uniform or known functions over the areas where the fluid enters and exits the control volume. The control surface should be chosen sufficiently far downstream from an abrupt area change (an entrance or a sudden contraction), that the velocity and pressure can be assumed to be uniform.

It is also necessary to specify the process by which the flow variables change, such as incompressible, isothermal, constant-pressure, adiabatic processes. If the working substance behaves as an ideal gas, then the appropriate equations may be used; if not, tabulated values must be used, e.g. the steam property table. For real gases that do not behave as ideal gases, properties can be found from the appropriate tables. Often heat transfer from a device or the internal energy change across a device, such as a pump, is not desired. For such situations, the heat transfer and internal energy change may be taken together as losses. In a pipeline, losses occur because of friction; in a pump, losses occur because of separated fluid flow around the rotating blades. For many devices the losses are included as an efficiency of the device. Some of the commonly used industrial equipment with steady state flows are described below:

#### **Throttling devices**

A throttling device involves a steady-flow adiabatic process that provides a pressure drop with no significant potential energy changes, kinetic energy changes, heat transfer, or work. As there is no heat transfer (being a rapid process) or work transfer during a throttling process, the energy equation for throttling can be simplified to  $h_1 = h_2$ .

Most of the valves used in industry are throttling devices, for which the above energy equation applies. Throttling valves used in refrigeration applications cause flashing resulting from the phase change due to a sudden pressure drop.

#### Compressors, pumps, and turbines

A pump is a device that transfers energy to a liquid by increasing its pressure. Compressors and blowers also fall into this category but have the primary purpose of increasing the pressure in a gas. A turbine, on the other hand, is a device in which work is done by the fluid on a set of rotating blades. As a result, there is a pressure drop from the inlet to the outlet of the turbine. In some situations, there may be heat transferred from the device to the surroundings, but often the heat transfer is negligible and can be ignored. In addition, the kinetic and potential energy changes can also be assumed to be negligible.
For such devices operating in a steady-state mode, the energy equation can be modified and written as:

$$-w_s = h_2 - h_1 \tag{2.38}$$

w<sub>s</sub> is negative for work consuming devices like compressors and positive for work producing devices like gas or steam turbines.

For liquids, such as water, by neglecting kinetic and potential energy changes, the energy equation can be written as follows:

$$(V_2^2 - V_1^2)/2 = -w_s = h_1 - h_2/\rho$$
(2.39)

#### **Nozzles and diffusers**

A nozzle is a device that is used to increase the velocity of a flowing fluid by reducing the pressure of the fluid. A diffuser is a device that increases the pressure in a flowing fluid by reducing the velocity. There is no work input into the devices and heat transfer is usually negligible. With the further assumptions of negligible internal energy and potential energy changes, the energy equation can be simplified to:

$$(V_2^2 - V_1^2)/2 = h_1 - h_2$$
(2.40)



Figure 2.8 Nozzle and diffuser.

As shown in Figure above, a nozzle has a decreasing area and a diffuser has an increasing area.

#### Heat exchangers

Heat exchangers are used to transfer energy from a hotter body to a colder body or to the surroundings by means of heat transfer. Practical industrial examples of such heat exchanges are: heat transfer from the combustion gas in a power plant to the water in the pipes of the boiler, and the heat transfer from the hot water that leaves an automobile engine to the atmosphere by use of a radiator. Many heat exchangers utilise a flow passage into which a fluid enters and from which the fluid exits at a different temperature. The velocity does not normally change in the heat exchanger and the pressure drop through the passage is usually neglected, and the potential energy change is assumed to be zero. The selection of control volume for thermodynamic system analysis purposes is depicted in Figure 2.9. The resulting simplified energy equation for a heat exchanger is:



Figure 2.9 A heat exchanger. (a) Combined unit. (b) Separated control volumes.

#### 2.5 Second Law of Thermodynamics

The Second Law of Thermodynamics can be stated by the Clausius statement and the Kelvin-Planck statement. An additional property, entropy, which can be used to determine whether the Second Law is being violated for any situation, is also introduced.

The Clausius statement of the Second Law of Thermodynamics states that it is impossible to construct a device that operates in a cycle and whose sole effect is the transfer of heat from a cooler body to a hotter body. This statement can be explained further by a refrigerator or a heat pump. It states that it is impossible to construct a refrigerator that transfers energy from a cooler body to a hotter body without the input of work (Kelvin-Planck statement). The violation of this statement is shown in Figure 2.10 (a). The violation of the Kelvin-Planck statement results in the violation of Clausius statement as shown in Figure 2.10 (b) because of the net effect of the refrigerator transferring heat from a colder source to a hotter sink without any work input.



Figure 2.10 Violation of Second Law of Thermodynamics

where,

 $T_H$  = Source temperature, K

 $T_L$  = Sink temperature, K

Q<sub>H</sub> = Heat absorbed, kJ

Q<sub>L</sub> = Heat rejected, kJ

W = Work produced, kJ

The Kelvin-Planck statement of the Second Law states that it is impossible to construct a device that operates in a cycle and produces no other effect than the production of work and the transfer of heat from a single body. In other words, it is impossible to construct a heat engine that extracts energy from a reservoir, does the work, and does not transfer heat to a low-temperature reservoir.

This rule out any heat engine that is 100 percent efficient. It may be worth noting that the two statements of the Second Law are negative statements. They are expressions of experimental observations. However, no experimental evidence has ever been obtained that violates either statement of the Second Law. The example below demonstrates that the two statements are equivalent.

The heat engine that operates most efficiently between a high-temperature reservoir and a low-temperature reservoir is the Carnot engine. It is an ideal engine that uses reversible processes to form its cycle of operation. Therefore, it is also called a reversible engine. The Carnot engine is very useful, since its efficiency establishes the maximum possible efficiency of any real engine working between two temperature limits. From an optimisation point of view, if the efficiency of a real engine is significantly lower than the efficiency of a Carnot engine operating between the same limits, then additional improvements may be possible.

#### 2.5.1 The Carnot Engine

The cycle associated with the Carnot engine is shown in Figure 2.11, using an ideal gas as the working substance. It is composed of the following four reversible processes:

 $1 \rightarrow 2$ : Isothermal expansion. Heat is transferred reversibly from the high-temperature reservoir at the constant temperature T<sub>H</sub>. The piston in the cylinder is withdrawn and the volume increases.



Figure 2.11 The Carnot engine cycle

 $2 \rightarrow 3$ : Adiabatic reversible expansion. The cylinder is completely insulated so that no heat transfer occurs during this reversible process. The piston continues to be withdrawn, with the volume increasing.

 $3 \rightarrow 4$ : Isothermal compression. Heat is transferred reversibly to the low temperature reservoir at the constant temperature T<sub>L</sub>. The piston compresses the working substance, with the volume decreasing.

 $4 \rightarrow 1$ : Adiabatic reversible compression. The completely insulated cylinder allows no heat transfer during this reversible process. The piston continues to compress the working substance until the original volume, temperature, and pressure are reached, whereupon the cycle repeats itself.

Applying the First Law to the Carnot cycle, we get:

$$Q_{H} - Q_{L} = W_{net} \tag{2.42}$$

where  $Q_L$  is assumed to be a positive value for the heat transfer to the lowtemperature reservoir. This allows us to write the thermal efficiency for the Carnot cycle as:

$$\eta_{Carnot} = 1 - \frac{Q_L}{Q_H} \tag{2.43}$$

In terms of temperature, equation (2.43) can be expressed as:

$$\eta_{Carnot} = 1 - \frac{T_L}{T_H}$$
(2.44)

where,

 $Q_H$  = Heat transferred from the source, kJ

 $Q_L$  = Heat rejected to the sink, kJ

W<sub>net</sub> = Net work produced, kJ

The three postulates pertaining to the Carnot engine are:

**Postulate 1** It is impossible to construct an engine, operating between two given temperature reservoirs, that is more efficient than the Carnot engine.

**Postulate 2** The efficiency of a Carnot engine is not dependent on the working substance used or any design feature of the engine.

**Postulate 3** All reversible engines operating between two given temperature reservoirs, have the same efficiency as a Carnot engine operating between the same two temperature reservoirs. The efficiency of a Carnot engine is dependent only on the two reservoir temperatures, and can be expressed as:

$$\eta = 1 - \frac{T_L}{T_H} \tag{2.45}$$

This expression is obtained by replacing the ratio of the associated heat transfers,  $Q_L/Q_H$  with the corresponding temperatures,  $T_L/T_H$ . We can make this replacement for all reversible engines or refrigerators. We see that the thermal efficiency of a Carnot engine is dependent only on the high and low absolute temperatures of the reservoirs. The fact that we used an ideal gas to perform the calculations is not important since

we have shown that Carnot efficiency is independent of the working substance. Consequently, the relationship (2.44) is applicable for all working substances, and for all reversible engines, regardless of the design characteristics.

The Carnot engine, when operated in reverse, becomes a Carnot heat pump or a refrigerator depending on the desired heat transfer. The coefficient of performance (COP) for a Carnot heat pump can be expressed as:

$$COP_{HP,Carnot} = \frac{Q_H}{W_{in}}$$
(2.46)

The performance indicator, Carnot COP, just sets a limit that real devices can only approach. The reversible cycles assumed are obviously unrealistic, but the fact that we have limits that we know cannot be exceeded is often very helpful.

#### 2.6 Entropy

To allow us to apply the Second Law of Thermodynamics to a process, we will identify a property called entropy. This will parallel our discussion on the First Law; first we stated the First Law for a cycle and then derived a relationship for a process.

Consider the reversible Carnot engine operating on a cycle consisting of the processes described in the preceding discussion. The quantity  $\int \delta Q/T$  is the cyclic integral of the heat transfer divided by the absolute temperature at which the heat transfer occurs. Since the temperature T<sub>H</sub> is constant during the heat transfer Q<sub>H</sub> and T<sub>L</sub> is constant during heat transfer Q<sub>L</sub>, the integral is given by:

$$\oint \frac{\delta Q}{T} = \frac{Q_H}{T_H} - \frac{Q_L}{T_L} \tag{2.47}$$

where the heat  $Q_L$  leaving the Carnot engine is considered to be positive. Using Eqs. (2.43) and (2.44), we see that for the Carnot cycle,

$$\frac{Q_L}{Q_H} = \frac{T_L}{T_H} \quad \text{or} \quad \frac{Q_H}{T_H} = \frac{Q_L}{T_L}$$
(2.48)

Substituting this into Eq. (2.47), we find the interesting result:

$$\oint \frac{\delta Q}{T} = 0 \tag{2.49}$$

Thus, the quantity  $\delta Q/T$  is a perfect differential, since its cyclic integral is zero. We let this differential be denoted by ds, where s depends only on the state of the system. Based on the preceding discussion, this is the definition of a property of a system. This extensive property is widely known as entropy; its differential is given by:

$$dS = \frac{\delta Q}{T} (rev) \tag{2.50}$$

where "rev" emphasises the reversibility of the process. This can be integrated for a process to give:

$$\Delta S = \int_{1}^{2} \frac{\delta Q}{T} (rev)$$
(2.51)

From the above equation, it is evident that the entropy change for a reversible process can be either positive or negative depending on whether energy is added to or extracted from the system during the heat transfer process. For a reversible adiabatic process (Q = 0) the entropy change is zero. If the process is adiabatic but irreversible, it is not generally true that  $\Delta s = 0$ .



Figure 2.12 Temperature – entropy (T-s) diagram for Carnot cycle

We often sketch a temperature-entropy diagram for cycles or processes of interest. The temperature-entropy diagram for a Carnot cycle is shown in Figure 2.12. The change in entropy for the first isothermal process from state 1 to state 2 is:

$$s_2 - s_1 = \int_1^2 \frac{\delta Q}{T} = \frac{Q_H}{T_H}$$
(2.52)

The entropy change for the reversible adiabatic process from state 2 to state 3 is zero. For the isothermal process from state 3 to state 4, the entropy change is the negative of the first process from state 1 to state 2; the process from state 4 to state 1 is also a reversible adiabatic process and is accompanied with a zero-entropy change. The heat transfer during a reversible process can be expressed in differential form as:

$$\delta Q = T ds \text{ or } Q = \int T ds \tag{2.53}$$

Hence, the area under the curve in the T-s diagram represents the heat transfer during any reversible process. The rectangular area in Figure 2.12 represents the net heat transfer during the Carnot cycle. Since the heat transfer is equal to the work done for a cycle, the area also represents the net work accomplished by the system during the cycle.

For this Carnot cycle: 
$$Q_{net} = W_{net} = \Delta T \Delta s$$
 (2.54)

For the First Law of Thermodynamics, a reversible infinitesimal change can be expressed as:

$$Tds - PdV = dU \tag{2.55}$$

Eq (2.55) is an important relationship in the study of simple systems assuming a reversible process. However, since it involves only properties of the system, it holds for any process including any irreversible process. If we have an irreversible process, in general:

$$\delta W \neq P dV and \, \delta Q \neq T ds$$
 (2.56)

Eq (2.55) still holds as a relationship between the properties since changes in properties do not depend on the process. Dividing by the mass, we get:

$$Tds - Pdv = du \tag{2.57}$$

where the specific entropy is s = S/m.

To relate the entropy change to the enthalpy change, we differentiate the definition of enthalpy and obtain:

$$dh = du + Pdv + vdP \tag{2.58}$$

Substituting into Eq. (2.57) for du, we get:

$$Tds = dh - vdP \tag{2.59}$$

#### 2.6.1 Entropy for an ideal gas with constant specific heats

Assuming an ideal gas (Pv = RT), and using Eqs. (2.58) & (2.59), we get:

$$ds = du/T + Pdv/T = C_v dT/T + Rdv/v$$
(2.60)

where,

$$du = C_{\nu}dT \tag{2.61}$$

$$Pv = RT \tag{2.62}$$

Integrating and rearranging Eq (2.59), and assuming constant specific heat, yields:

$$s_2 - s_1 = Cp \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$
 (2.63)

If the entropy change is zero, as in a reversible adiabatic process, Eqs. (2.60) and (2.61) can be used to obtain:

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{(\gamma-1)}$$
(2.64)

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma}$$
(2.65)

Combining these two expressions yields:

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{\gamma}$$
 (2.66)

#### 2.6.2 Entropy for an ideal gas with variable specific heats

If the specific heats for an ideal gas cannot be assumed constant over a particular temperature range, the pressure ratio is expressed as:

$$\frac{P_2}{P_1} = exp\left(\frac{\phi_2 - \phi_1}{R}\right) = \frac{exp\left(\frac{\phi_2}{R}\right)}{exp\left(\frac{\phi_1}{R}\right)} = \frac{f(T_2)}{f(T_1)}$$
(2.67)

Thus, we define a relative pressure P<sub>r</sub>, which depends only on the temperature, as:

$$P_r = exp\left(\frac{\emptyset}{R}\right) \tag{2.68}$$

$$\frac{P_2}{P_1} = \frac{P_{r2}}{P_{r1}} \tag{2.69}$$

#### 2.7 Illustrative Examples

#### Example 2.7.1

Express a pressure gauge reading of 20 mm Hg in absolute pressure at an elevation of 3,000 m. Specific gravity of mercury = 13.6 g/cm<sup>3</sup> Solution Converting the gauge pressure reading into pascals,  $P = \rho gh = 13.6 \times 1,000 (kg/m^3) \times 9.81(m/s^2) \times (20/1,000)(m) = 2,668 Pa = 2.67 kPa$ (1 kg/m.s<sup>2</sup>) = 1 Pa) The reduction in atmospheric pressure with an elevation of 3,000 m is:

dP<sub>atm@3000m</sub> = ρgh = 1.2 x 9.81 x 3,000/1000 = 35.32 kPa

Therefore,  $P_{atm@3000m} = 101.3 - 35.32 = 65.98$  kPa  $P_{abs} = P_{gauge} + P_{atm} = 2.668 + 65.98 = 68.65$  kPa

#### Example 2.7.2

A 3,000 kg automobile travelling at 75 km/h hits the rear of a stationary 900 kg automobile. After the collision the larger automobile slows to 60 km/h, and the smaller vehicle has a speed of 67 km/h. Determine the increase in internal energy, taking both vehicles as the system.

#### Solution

The kinetic energy before the collision can be determined as:  $K.E_{initial} = \frac{1}{2} m_a V_a^2$ =  $\frac{1}{2} \times 3,000 (kg) \times (75,000/3,600)^2 (m/s)^2 = 651,041 J$ 

After the collision the kinetic energy is:  $K.E_{final} = \frac{1}{2} m_a V_a^2 + \frac{1}{2} m_b V_b^2$ =  $\frac{1}{2} \times 3,000 \times (60,000/3,600)^2 + \frac{1}{2} \times 900 \times (67,000/3,600)^2 = 572,534 J$ (1 kg m<sup>2</sup>/s<sup>2</sup>) = 1 J) where the subscript a and b refers to the first and second automobiles, respectively.

The conservation of energy requires that:  $K.E_{initial} = K.E_{final} = \Delta U$  (Change in internal energy)

Therefore,  $\Delta U = 651,041 - 572,534 = 78,507 J = 78.5 kJ$ 

#### Example 2.7.3

A gas is contained in a vertical, frictionless piston–cylinder device. The piston has a mass of 5 kg and a cross-sectional area of 40 cm<sup>2</sup>. A compressed spring above the piston exerts a force of 70 N on the piston. If the atmospheric pressure is 90 kPa, what is the pressure inside the cylinder?

#### Solution:

Referring to the free body diagram of the piston and balancing the vertical forces will yield:



 $P x A = P_{atm} x A + W + F_{spring}$ 

Simplifying for P, we get:  $P = P_{atm} + (W + F_{spring})/A$ P = 90 + [(5 x 9.8 + 70)/1,000]/ (40 x 10<sup>-4</sup>) = 119.75 kPa

# Example 2.7.4

A 0.4m<sup>3</sup> rigid vessel initially contains saturated liquid–vapour mixture of water at 150°C. The water is now heated until it reaches the critical state. Determine the mass of the liquid water and the volume occupied by the liquid at the initial state.

#### Solution

Using the steam property table (at critical point)(221.2-bar),  $v_1 = v_2 = v_{cr} = 0.00317$  m<sup>3</sup>/kg

The total mass, m = volume/specific volume = V/v = 0.4/0.00317 = 126.18 kg



From the steam table (Reference 3), for 150°C,  $v_{\rm f}$  = 0.001091 m³/kg and  $v_g$  = 0.39248 m³/kg

Therefore, the quality of water at the initial state is:  $x_1 = (v_1 - v_f)/v_{fg} = (0.00317 - 0.001091)/(0.39248 - 0.001091) = 0.00531$ 

The mass of the liquid and its volume at the initial state are determined as:  $m_f = (1-x_1) x$  total mass (m) = (1 - 0.00531) x (126.18) = 125.5 kg

 $V_f = m_f x v_f = 125.5 x 0.001091 = 0.136 m^3$ 

# Example 2.7.5

Air enters an adiabatic nozzle steadily at 400 kPa, 250°C, and 25 m/s and leaves at 150 kPa and 170 m/s. The inlet area of the nozzle is 70 cm<sup>2</sup>. Determine (a) the mass flow rate through the nozzle, (b) the exit temperature of the air Assume the gas constant R = 0.287 kJ/kg.K

Solution



a) The mass is conserved, therefore,  $\dot{m}_1 = \dot{m}_2 = \dot{m}$ The specific volume and the mass flow rate are determined using the ideal gas expression:

 $v_1 = RT_1/P_1 = (0.287 \text{ x } 523)/400 = 0.375 \text{ m}^3/\text{kg}$ (kJ/kg.K x K x m<sup>2</sup> /N = (kN m/kg K) x K x m<sup>2</sup>/kN = m<sup>3</sup>/kg)  $\dot{m}_1 = A_1V_1/v_1 = (0.007 \text{ x } 25)/0.375 = 0.466 \text{ kg/s}$ 

b) Taking the nozzle as the system, which is also a control volume since the mass crosses the boundary, an energy balance is written as follows: Energy in = Energy out;  $E_{in} = E_{out}$ 

 $\dot{m}_1 = A_1 V_1 / v_1 = (0.007 \text{ x } 25) / 0.375 = 0.466 \text{ kg/s}$ 

 $\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2); h_2 - h_1 = V_1^2/2 - V_2^2/2$ 

But,  $h = C_p T$ ; therefore,  $C_{p,average}(T_2 - T_1) = V_1^2/2 - V_2^2/2$ 

 $T_2 = T_1 + [(V_1^2/2 - V_2^2/2/1,000)]/1.02 = 250 + [(25^2 - 170^2)/2/1,000]/1.02 = 222.2^{\circ}C$ (1 kJ/kg = 1,000 m<sup>2</sup>/s<sup>2</sup>)

#### Example 2.7.6

Steam flows steadily through an adiabatic turbine. The inlet conditions of the steam are 8 MPa, 500°C, and 70 m/s, and the exit conditions are 20 kPa, 94 percent quality, and 40 m/s. The mass flow rate of the steam is 15 kg/s. Determine (a) the power output, and (b) the turbine inlet area

Solution



The properties from the steam property tables are read as follows:

 $P_1 = 8 \text{ MPa} = 80\text{-bar and } T_1 = 500^{\circ}\text{C},$  $v_1 = 0.0417 \text{ m}^3/\text{kg}, h_1 = 3,398 \text{ kJ/kg}; \text{ and}$ 

 $P_2 = 20 \text{ kPa and } x_2 = 0.94,$  $h_2 = h_{f@20kPa} + X_2 x h_{f@20kPa}, h_2 = 251 + (0.94 x 2,358) = 2,467.5 \text{ kJ/kg}$ 

(a) Considering a steady state flow and negligible heat transfer (adiabatic flow), the energy balance is expressed as follows:  $\dot{m} (h_1 + V_1^2/2) = W_{out} + \dot{m} (h_2 + V_2^2/2)$ 

Therefore, 
$$W_{out} = \dot{m}(h_1 - h_2) + [(V_1^2/2 - V_2^2/2)/1,000]$$
  
= 15 x (3,398 - 2,467.5) + (70<sup>2</sup>/2 - 40<sup>2</sup>/2)/1,000 = 15.6 MW

b) The turbine inlet area is calculated as follows:  $\dot{m}_1 = A_1 V_1 / v_1$ , therefore,  $A_1 = (\dot{m}_1 \times v_1) / V_1 = (15 \times 0.0417) / 70 = 0.00893 \text{ m}^2 = 89.3 \text{ cm}^2$ 

#### Example 2.7.7

An inventor claims to have devised a cyclical engine for use in space vehicles that operates with a nuclear-fuel-generated energy source whose temperature is 550 K and a sink at 300 K that radiates waste heat to deep space. He also claims that this engine produces 5 kW while rejecting heat at a rate of 15,000 kJ/h. Is this claim valid?

Solution



The Carnot thermal efficiency is expressed as:  $\eta_{th,Carnot} = 1-T_L/T_H = 1-(300/550) = 0.455$  Applying First Law gives,

 $Q_H = W_{net} + Q_L = (5 \times 3,600) + 15,000 = 33,000 \text{ kJ/h}$ 

The actual thermal efficiency of the cycle is determined as:

 $\eta_{th,Carnot} = W_{net}/Q_H = (5 \times 3,600)/33,000 = 0.545$ 

Since the actual efficiency is more than the maximum possible efficiency, the inventor's claim is invalid.

# Summary

The fundamental thermodynamic concepts are summarised in this chapter beginning with thermodynamic systems and concepts. Following the basic concept discussion, a review of the important units and their conversions has been presented enabling practising engineers to perform with confidence. The other thermodynamic aspects discussed in this chapter included: The Laws of thermodynamics, various thermodynamic devices like pumps, compressors, nozzles and diffusers as well as heat exchangers. Finally, this chapter also discussed the concepts surrounding entropy with a number of illustrative examples.

# References

- 1. Cengel Yunus, Thermodynamics: an engineering approach, McGraw-Hill, New York, 2008.
- 2. Nag P. K, Engineering Thermodynamics: Tata-McGraw-Hill, New York, 1999.
- 3. Rogers, G.F.C. and Mayhew, Y.R., Thermodynamic and Transport Properties of Fluids, Oxford, 2014.

# 3.0 STEAM PROPERTIES

Water can exist as liquid, solid or gas. Steam is the gaseous phase of water. It can be produced by heating water in a boiler at constant pressure. The heat added to produce steam can later be extracted easily by condensing it back to water, hence steam is used for carrying large amounts of heat energy. In addition, steam is not toxic, it is easily transportable, can be generated relatively efficiently and is not very costly to generate. Therefore, steam is the most common heating medium in industrial facilities. Steam is also used in power generation and in various chemical reactions and processes.

This chapter provides an introduction to some of the basic properties of steam, which are useful in understanding how boilers and steam systems operate.

# Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. The relationship between saturation temperature and pressure
- 2. Various properties of steam
- 3. How to use steam tables

# 3.1 Introduction

Water ( $H_2O$ ) is abundant on earth. Like many substances, it can exist in three physical states, which are solid, liquid and gas. For  $H_2O$ , the three states are called ice, water and steam.

The molecular arrangement and the degree of excitation of the molecules determine the physical state of  $H_2O$ . When in the solid state, the molecules are locked together and can only vibrate about a mean bonded position. If heat is added, the vibration increases and when the heat added reaches a certain point, some of the molecules break away from their bonds. At this stage, the solid starts to melt to a liquid state. At atmospheric pressure, melting of ice occurs at 0°C.

In the liquid phase, the molecules are free to move, but are still close to each other due to mutual attraction. When heat is added, molecular agitation and collisions increase. The temperature of the liquid also rises until it reaches the boiling temperature when some molecules attain sufficient kinetic energy to allow them to escape from the liquid. However, this is momentary and they fall back into the liquid. Further addition of heat causes the excitation to increase so that some molecules will have sufficient energy to leave the liquid. When this happens, bubbles of steam will rise and break through the surface. When the number of molecules leaving the liquid surface is more than those re-entering, the water freely vapourises into steam.

#### 3.2 Saturation Temperature

When water is heated to its boiling point or its saturation temperature, it is saturated with heat energy. If more heat is added at this stage while maintaining the pressure constant, the temperature of the water will not rise but it will result in the water forming saturated (dry) steam. The temperature of both the boiling water and saturated (dry) steam will be the same, but the heat energy (per unit mass) will be much greater in the steam than the boiling water.

At atmospheric pressure, the boiling point or the saturation temperature of water is 100°C. However, if the pressure is increased, the saturation temperature will increase (water temperature will need to be increased beyond 100°C for it to boil). Similarly, if the pressure is reduced below atmospheric pressure, water will boil at a temperature less than 100°C. The relationship between the saturation temperature and the pressure is known as the steam saturation curve and is shown in Figure 3.1. Water and steam can coexist at any point on this curve, and both will be at the saturation temperature. In a steam generation process, the entire liquid water is converted to steam at a boiling point corresponding to the prevailing pressure at the boiler, the steam at that state is called saturated (dry) steam.



Figure 3.1 Steam saturation curve

The values of saturation temperature at different pressures are shown in Table 3.1.

Absolute pressure (bar)	Saturation Temperature (°C)		
1	99.6		
1.01 (atmospheric pressure)	100		
2	120.2		
3	133.5		
4	143.6		
5	151.8		
6	158.8		
7	165.0		
8	170.4		
9	175.4		
10	179.9		
11	184.1		
12	188.0		
13	191.6		
14	195.0		
15	198.3		

Table 3.1 Saturation temperature of steam at different pressures

# Example 3.1

What is the saturation temperature when the pressure is 2.5-bar?

#### Solution

From Figure 3.2, the saturation temperature at 2.5-bar pressure is approximately 125°C.



Figure 3.2 Steam temperature for Example 3.1

# Example 3.2

A heating process requires steam at 200°C. What is the minimum required steam pressure for this application?

# Solution

From Figure 3.3, a steam pressure of about 15-bar is required to achieve a saturation temperature of 200°C.



Figure 3.3 Steam temperature for Example 3.2

#### 3.3 Sensible and Latent Heat

When water is heated by adding heat energy, the temperature of the water rises. Such a process, where adding heat leads to a corresponding increase in temperature, is called sensible heating and the heat added is called sensible heat.

Specific heat capacity is the amount of heat energy required to raise the temperature per unit of mass of a substance by 1°C. In SI units, specific heat capacity is the amount of heat in joules required to raise 1 gram of a substance by 1 Kelvin. For water, the specific heat capacity at atmospheric pressure is 4.19 kJ/kg.K.

For water, the amount of sensible heat required to increase the temperature of 1 kg of water from 0°C to the boiling temperature is 419 kJ/kg (4.19 kJ/kg.K x 100 K). It is also called the "liquid enthalpy" or enthalpy of water (explained later in this chapter).

When water is heated to its boiling point, further adding of heat does not increase the temperature of water but only results in boiling of the water to form steam. Such heating which does not result in an increase in the temperature of the substance that is heated but only results in a change in phase is called latent heating. The heat added during such a process is called latent heat.

Latent heat is energy absorbed during evaporation of a liquid (or released during condensing of a vapour) that occurs without changing its temperature. The latent heat in SI units is expressed in joules per unit mass in grams of the substance undergoing a change of state. For water, the amount of heat required to evaporate 1 kg of water at its boiling point is termed the "enthalpy of evaporation". At atmospheric pressure, the enthalpy of evaporation of water is 2257 kJ/kg.

The liquid enthalpy and enthalpy of evaporation of water at different pressures is shown in Table 3.2.

Absolute pressure (bar)	Enthalpy of water (kJ/kg)	Enthalpy of evaporation (kJ/kg)	
1	417	2258	
1.01 (atmospheric pressure)	419	2257	
2	505	2202	
3 561		2164	
4	605	2134	

5	640	2109	
6	670	2087	
7	697	2067	
8	8 721 2048		
9	743	2031	
10	763	2015	
11	781	2000	
12	798	1986	
13	815	1972	
14	830	1960	
15	845	1947	

Table 3.2 Liquid enthalpy and enthalpy of evaporation of water

# Example 3.3

At atmospheric pressure, how much sensible heat is required to raise 1000 kg of water from 30°C to 70°C?

# Solution

Specific heat capacity of water atmospheric pressure is 4.19 kJ/kg.K

Therefore, the sensible heat required to raise the temperature of 1,000 kg is,

1,000 kg x 4.19 kJ/kg.K x (70 – 30) K = 167,600 kJ

# Example 3.4

At a pressure of 5-bar, how much latent heat is required to boil 1,000 kg of water at its saturation temperature?

# Solution

Latent heat capacity required at 5-bar pressure (from Table 3.2) is 2,109 kJ/kg

Therefore, the total heat required to boil 1,000 kg of water at 5-bar is,

1,000 kg x 2,109 kJ/kg = 2,109,000 kJ

#### 3.4 Dryness Fraction

In an industrial type boiler, where heat is supplied only to the water, it is not possible to produce dry steam. Due to turbulence and splashing in the boiler when bubbles of steam are released from the water surface, the steam contains some water droplets. Typically, steam produced by a shell-type boiler will contain about 5% water.

If the water content of the steam is say 4% by mass, then the steam is said to be 96% dry and has a dryness fraction of 0.96.

Dryness fraction of steam is defined as the ratio of the mass of actual dry steam to the total mass of wet steam and can be expressed as:

$$x = m_g / (m_g + m_f) \tag{3.1}$$

 $x = m_g/m$ 

where x = dryness fraction  $m_g$  = mass of dry steam  $m_f$  = mass of water in mixture m = mass of wet steam =  $m_g + m_f$ 

The concept of dryness fraction will be further explained in section 3.7 of this chapter.

# Example 3.5

1000 kg of steam contains 50 kg of water. What is the dryness fraction of the steam?

Solution Dryness fraction,  $x = m_g / m_g + m_f$ 

# 3.5 Superheated Steam

When saturated (dry) steam is further heated so that the temperature of the steam exceeds the saturation temperature at a particular pressure, the steam is superheated. Superheating produces steam that has a higher temperature and lower density than saturated (dry) steam at the same pressure.

For instance, steam at 6-bar is at a saturation temperature of 159°C. Now if this steam is heated to 179°C, the steam will have a superheat of 20°C at 6-bar.

Superheating steam ensures that the steam is completely dry. Superheating is used in steam driven equipment such as turbines to avoid drop in performance due to the presence of condensate and to prevent erosion and corrosion. However, superheated steam is normally not used for heating applications as a higher heat transfer area will be required when using superheated steam compared to saturated (dry) steam for the same application. This is because the superheated steam's density is lower in comparison to that of saturated (dry) steam.

#### Example 3.6

Steam at 10-bar is at a temperature of 210°C. What is the degree of superheat of the steam?

#### Solution

Saturation temperature of steam at 10-bar (from Table 3.1) is approximately 180°C.

Therefore, the degree of superheat is (210 - 180) °C = 30°C

#### 3.6 Steam Pressure vs Volume

The specific volume is the total volume of steam divided by the total mass of steam (volume per unit mass). It has units of cubic metre per kilogram (m<sup>3</sup>/kg). The density of steam is the reciprocal of its specific volume.

$\rho = m/V = 1/v$	(3.1)
where,	
$\rho$ = density (kg/m <sup>3</sup> )	
m = mass of steam (kg)	
V = volume of steam (m <sup>3</sup> )	
v = specific volume (m <sup>3</sup> /kg)	

As steam pressure increases, the density of steam also increases. Since the specific volume is inversely related to the density, the specific volume decreases with increased pressure.

Figure 3.4 shows the relationship between specific volume and pressure. As can be seen from the figure, the greatest change in specific volume occurs at lower pressures. At higher pressures, the change in specific volume is much less.



Figure 3.4 Steam pressure vs specific volume

As the specific volume of water is several orders of magnitude lower than that of steam, the droplets of water in wet steam will occupy negligible space. Therefore, the specific volume of wet steam is less than that of dry steam.

Specific volume of wet steam = 
$$v_g x$$
 (3.2)

where,

 $v_g$  = specific volume of dry steam x = dryness fraction of the steam

Table 3.3 shows values of specific volumes of dry steam at different pressures. Therefore, if the specific volume of wet steam is known, the dryness fraction can be computed as shown in Example 3.7.

Absolute pressure (bar)	Specific volume of saturated steam (m <sup>3</sup> /kg)		
1	1.7		
2	0.89		
3	0.61		
4	0.46		
5	0.37		
6	0.32		
7	0.27		
8	0.24		
9	0.21		
10	0.19		
11	0.18		
12	0.16		
13	0.15		
14	0.14		
15	0.13		

Table 3.3 Specific volume of dry steam

# Example 3.7

Steam is at 8-bar (absolute) pressure. The specific volume of the steam is 0.17 m<sup>3</sup>/kg. Compute the dryness fraction of the steam.

# Solution

From Table 3.3, the specific volume of dry steam at 8-bar is 0.24 m<sup>3</sup>/kg.

From equation (3.2), Specific volume of wet steam =  $v_g x$ 

Therefore, the dryness fraction,  $x = \text{specific volume of wet steam } / v_g$ 

= 0.17 / 0.24

= 0.71

# 3.7 Enthalpy

Specific enthalpy is a measure of the energy content of a unit of mass of a substance. The SI units of specific enthalpy are kJ/kg. For simplicity, specific enthalpy will be referred to as enthalpy in this reference manual. Figure 3.5 shows the relationship between temperature and enthalpy for water when it is heated at a constant pressure to form steam.



Figure 3.5 Relationship between temperature and enthalpy

As indicated in Figure 3.5, when water is heated, its temperature rises steadily until it reaches the saturation point where water cannot exist in the liquid form. At this point, addition of heat results in the boiling of water, producing steam. Further heating results in more steam being generated at the same temperature (saturation temperature). When all the water has been converted into steam, further addition of heat leads to an increase in steam temperature above the saturation temperature and the steam is then superheated.

Enthalpy value  $h_f$  refers to the enthalpy of water when it has reached the saturation temperature while  $h_g$  refers to the enthalpy of steam. The difference between  $h_g$  and  $h_f$  is the latent heat of vapourisation, which is denoted by the symbol  $h_{fg}$ . The value x refers to the dryness fraction.

Therefore,

Enthalpy of wet steam 
$$h_g = h_f + x h_{fg}$$
 (3.3)  
Enthalpy of dry steam  $h_g = h_f + h_{fg}$  (3.4)

Some values of h<sub>f</sub>, h<sub>g</sub> and h<sub>fg</sub> at different pressures are shown in Table 3.4.

For superheated steam,  $h = h_f + h_{fg} + C_p(t_{sup} - t_{sat} kJ/kg)$  (3.5)

where,

 $t_{sup}$  = superheated temperature of steam (K)  $t_{sat}$  = saturation temperature of steam (K) ( $t_{sup} - t_{sat}$ ) = degree of superheat (K) Cp = Specific heat capacity of steam (kJ/kg.K)

#### Example 3.8

A boiler is supplied with feedwater at a temperature of 70°C. The boiler produces steam at a pressure of 8-bar (abs.) and a temperature of 190°C.

Determine the quantity of heat supplied per kg of steam generation (excluding losses).

Take the specific heat capacity (Cp) of superheated steam to be 2.76 kJ/kg.K.

#### Solution

From Table 3.4, the following enthalpy values at 8-bar pressure can be obtained

 $\label{eq:hf} \begin{array}{l} h_{f} = 721 \ \text{kJ/kg} \\ h_{fg} = 2,048 \ \text{kJ/kg} \\ t_{sat} = 170.4^{\circ}\text{C} \ (\text{from Table 3.1}) \end{array}$ 

Since steam is produced at 190°C, the steam is superheated.

The enthalpy of superheated steam,

 $h_{sup} = h_f + h_{fg} + Cp (t_{sup} - t_{sat})$ 

= 721 + 2048 + 2.76 (190 - 170.4) = 2,823.1 kJ/kg

Enthalpy of feedwater at 70°C is 293 kJ/kg (70°C x 4.19 kJ/kg) Therefore, heat added = (2,823.1 - 293) kJ/kg = 2,530.1 kJ/kg

#### 3.8 Steam Pressure vs Enthalpy of Evaporation

Figure 3.5 showed how the enthalpy changes when water is heated to produce steam. However, the enthalpy of evaporation changes with pressure. As shown in Figure 3.6, when the pressure increases, the enthalpy of evaporation ( $h_{fg}$ ) reduces. Therefore, at higher operating pressures, more steam is required to produce the same amount of heating as the latent heat available becomes less.



Figure 3.6 Enthalpy of evaporation at different pressures

# 3.9 Condensate and Flash Steam

When high pressure steam is condensed, the condensate will be at the saturation temperature of the steam. Later when the condensate is returned to the condensate of the feedwater tank, the pressure is reduced to atmospheric pressure. When this happens, the enthalpy of the condensate has to reduce from the value at the higher pressure to the value at atmospheric pressure.

For example, condensate at 8-bar pressure (absolute) will have an enthalpy of 721 kJ/kg (from Table 3.4). However, when this condensate pressure is reduced to atmospheric pressure, the enthalpy can be 419 kJ/kg. The difference of 302 kJ/kg between the two enthalpy values (721 - 419) is released by evaporating part of the condensate. The steam formed during such a pressure reduction is called flash steam.

# 3.10 Use of Steam Tables

Since water and steam are commonly used for transferring heat energy, their properties are required for various calculations. Therefore, their properties are tabulated in so called "Steam Tables". In these tables, important properties such as pressure, temperature, enthalpy and specific volume are tabulated. A sample steam table is provided in Table 3.4. A more detailed version of the steam table is included in Appendix section of this reference manual.

Absolute pressure (bar)	Saturation temperature (°C)	Specific volume of dry steam (m <sup>3</sup> /kg)	Enthalpy of saturated water, h <sub>f</sub> (kJ/kg)	Enthalpy of evaporation, h <sub>fg</sub> (kJ/kg)	Enthalpy of saturated steam, h <sub>g</sub> (kJ/kg)
1	99.6	1.7	417	2258	2675
1.01 (atmospheric pressure)	100	1.67	419	2257	2676
2	120.2	0.89	505	2202	2707
3	133.5	0.61	561	2164	2725
4	143.6	0.46	605	2134	2739
5	151.8	0.37	640	2109	2749
6	158.8	0.32	670	2087	2757
7	165.0	0.27	697	2067	2764
8	170.4	0.24	721	2048	2769
9	175.4	0.21	743	2031	2774
10	179.9	0.19	763	2015	2778
11	184.1	0.18	781	2000	2781
12	188.0	0.16	798	1986	2784
13	191.6	0.15	815	1972	2787
14	195.0	0.14	830	1960	2790
15	198.3	0.13	845	1947	2792

Table 3.4 Properties of saturated (dry) steam and water

# Example 3.9

Determine the quantity of heat required to produce 1 kg of steam at pressure of 7-bar (absolute) using water at a temperature of 90°C, under the following conditions:

- a) When the steam is wet and has a dryness fraction of 0.8
- b) When the steam is dry saturated
- c) When the steam is superheated at a constant pressure to 240°C (assume the specific heat capacity of superheated steam to be 3.68 kJ/kg K)

# Solution

From Table 3.5, at 7-bar  $h_f = 697 \text{ kJ/kg}$   $h_{fg} = 2,067 \text{ kJ/kg}$  $t_{sat} = 165^{\circ}\text{C}$  a) When the steam is wet

 $h = h_f + x. h_{fg}$   $h = 697 + 0.8 \times 2,067$  h = 2,350 kJ/kgEnthalpy of water at 90°C = 377 kJ/kg

Therefore, actual heat required

h = 2,350 - 377h = 1,973 kJ/kg

b) When the steam is dry saturated

 $h = h_{f} + h_{fg}$  h = 697 + 2,067 h = 2,764 kJ/kgh = 2,764 - 377 = 2,387 kJ/kg

c) When the steam is superheated

$$\begin{split} h &= hf + h_{fg} + Cp \; (t_{sup} - t_{sat}) \\ h &= 697 + 2,067 + 3.68 \; (240 \; \text{--}\; 165) \\ h &= 3,040 \; \text{kJ/kg} \\ h &= 3,040 - 377 = 2,663 \; \text{kJ/kg} \end{split}$$

# Summary

This chapter provides an introduction to some of the basic properties of steam, which are useful in understanding how boilers and steam systems operate. The relationship between saturation temperature and pressure, various properties of steam and the use of steam tables were illustrated using worked examples.

# References

- 1. Jayamaha, Lal, Energy-Efficient Industrial Systems, Evaluation and Implementation, McGraw-Hill, New York, 2016.
- Rogers, G.F.C. and Mayhew, Y.R., Thermodynamic and Transport Properties of Fluids, Oxford, 2014.
- 3. The Steam and Condensate Loop, Spirax Sarco Limited, England 2008.

# 4.0 GAS POWER CYCLES

A CHP System including a Gas Turbine (GT) working on gas power cycles may be installed where clean premium fuels like natural gas are available. For thermodynamic analysis for evaluating the performance of such CHP systems, a fairly good understanding of the gas power cycle is very important. Among the gas power cycles, the engineer's main focus should be understanding the Brayton cycle and its analysis, to establish the thermal performance of the system.

This chapter presents the gas power cycles with the main focus on the gas turbine cycle or Brayton cycle.

# Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. The basics of gas power cycles
- 2. The Brayton gas power cycle used for CHP systems
- 3. Thermal efficiency of the Brayton cycle
- 4. Various thermal efficiency enhancement techniques for the Brayton cycle

# 4.1 Gas Power Cycles

CHP systems using steam or gas power plants are operated in a thermodynamic cycle as described in Chapter 2 of this reference manual. In a thermodynamic cycle, the working fluids such as water or gas undergo a series of processes and finally return to the initial thermodynamic state. In some other power plants like the internal combustion engine and the gas turbine, the working fluid does not go through a thermodynamic cycle despite the engine operating in a mechanical cycle. In this instance, the working fluid has a different composition or is in a different state at the end of the process than the starting state. Such equipment are sometimes said to operate on an open cycle, although the word 'cycle' is a misnomer. For simplicity, the performance of an idealised closed Brayton cycle similar to the actual cycle is analysed. Such a procedure will help in determining the influence of certain variables on the cycle performance. In the case of a cycle in which the working fluid is a gas, there is no change of phase. It is always a common practice to idealise the cycle first, which will simplify the analysis. Subsequently, a real-gas cycle is considered and how the actual apparatus deviates from the ideal is analysed. After the analysis of the actual cycle, various techniques are incorporated to modify the ideal cycle such that the actual cycle efficiency can be improved and tend towards the ideal cycle performance. The techniques include the use of devices such as regenerators, multistage compressors and expanders and intercoolers, and various combinations of these techniques.

The gas power cycle can work on an open or closed cycle. The four main components of a gas power cycle are compressor, combustor, turbine and heat exchanger (Figure 4.6). The gas undergoing these processes is assumed to have the characteristics of an ideal gas. To simplify the gas power cycle analysis, the following assumptions, known as air standard assumptions, are made:

- The air circulates continuously in a closed loop and always behaves as an ideal gas
- All the processes that constitute the cycle are internally reversible
- The combustion process is replaced by a heat addition process from an external source
- A heat rejection process that restores the state of the working fluid to initial state replaces the exhaust process

# 4.2 Gas Compressors

Gas compressors are used in gas power cycles to compress the air before it is subject to combustion process. As in the case of any other thermodynamic device, a control volume analysis can be done for the gas compressors.

In a gas compressor using the control volume energy equation, the power input to the compressor can be expressed as follows:

$$W_{comp} = \dot{m}(h_e - h_i) \tag{4.1}$$

where  $h_e$  and  $h_i$  are the exit and inlet enthalpies, respectively. Here, the compressor is considered as a fixed volume into which a gas flows and after undergoing the compression process, the gas emerges from the compressor. Negligible heat transfer is assumed to occur from the compressor along with negligible inlet and outlet kinetic and potential energy changes.

There are generally three types of compressors in industry namely reciprocating, centrifugal, and axial-flow. Reciprocating compressors are especially useful for producing high pressures but are limited to relatively low flow rates; upper limits of about 200 MPa with inlet flow rates of 160 m<sup>3</sup>/min are achievable with a two-stage

compressor unit. For high flow rates with relatively low pressure rise, a centrifugal or axial-flow compressor is more appropriate. In such compressors, a pressure rise of several MPa for an inlet flow rate of over 10,000 m<sup>3</sup>/min is achievable. A brief account of the three types of compressors is given below.

#### 4.2.1 The Reciprocating Compressor

The positions of the piston in a reciprocating compressor are shown in Figure 4.1. The intake and exhaust valves are closed as shown in (a) when state point 1 is reached in the P-v diagram of Figure 4.2. An isentropic compression follows as the piston travels inwards as shown in (b) until the maximum pressure at state point 2 is reached. The exhaust valve then opens, and the piston continues its inward motion while the air is exhausted as shown in (c) until state point 3 is reached at top dead centre.





The exhaust valve then closes as shown in (d), and the piston begins its outward motion with an isentropic expansion process until state point 4 is reached. At this

point, the intake valve opens and the piston moves outward during the intake process as shown in (e) until the cycle is completed.

Figure 4.2 (b) shows the actual operation of the P-v diagram. As the intake and exhaust valves do not open and close instantaneously, the airflow around the valves results in pressure gradients during the intake and exhaust strokes. In addition, losses occur due to the valves, and some heat transfer may also take place. However, the ideal cycle as shown in Figure 4.2a is simple to analyse and allows us to predict the influence of proposed design changes on input work, maximum pressure, flow rate, and other quantities of practical interest. The effectiveness of a compressor is partially measured by the volumetric efficiency. The volumetric efficiency of a compressor is defined as the ratio of the volume of gas drawn into the cylinder to the displacement volume.

That is, referring to Figure 4.2:

$$\eta_{vol} = (V_1 - V_4) / (V_1 - V_3) \tag{4.2}$$

The higher the volumetric efficiency the greater the volume of air drawn in as a percentage of the displacement volume. This can be increased if the clearance volume  $V_3$  is decreased. To improve the performance of the reciprocating compressor, one can remove heat from the compressor during the compression process 1-2.

Using the control volume inlet-outlet concept, the required work of compression for an adiabatic compressor can be expressed as:

$$w_{comp} = (h_2 - h_1) = C_p (T_2 - T_1)$$
(4.3)

Where,  $w_{comp} = compressor work$ , kJ/kg h = specific enthalpy, kJ/kg  $C_p = specific heat capacity, kJ/kg.K$ 

#### 4.2.2 Axial-Flow Compressors

A cut-away view of an axial-flow compressor is shown in Figure 4.3. It is similar in appearance to the steam turbine used in the Rankine power cycle (covered in detail in Chapter 5). Several stages of blades are needed to provide the desired pressure rise, with a relatively small rise occurring over each stage. Each stage has a stator, a series of blades that are attached to the stationary housing, and a rotor. All the rotors are attached to a common rotating shaft that utilises the power input to the compressor. The specially designed airfoil-type blades require extreme precision in manufacturing and installation to yield the maximum possible pressure rise while avoiding flow separation. The area through which the air passes decreases slightly as the pressure rises due to the increased density in the higher pressure air. Using fluid dynamic concepts, the velocity and pressure at each stage can be analysed, whereas from a thermodynamic point of view, we are concerned only with inlet and outlet conditions.



Figure 4.3 Cut-away of an axial-flow compressor

# 4.3 The Carnot Cycle

The most ideal of all the thermodynamic cycles is the Carnot cycle, which is used for performance comparison of actual cycles. The thermal efficiency of a system working on the Carnot cycle can be expressed as:

$$\eta_{carnot} = 1 - T_L / T_H \tag{4.4}$$

# 4.4 The Brayton Cycle

The Brayton cycle is a gas power cycle comprising two constant-pressure and two isentropic processes. In the Brayton cycle, a single-phase, gaseous working fluid undergoes the four processes. The air-standard Brayton cycle is the ideal cycle for the simple gas turbine. The schematic diagrams of a simple open cycle gas turbine

utilising an internal-combustion process and the simple closed cycle gas turbine, which utilises heat-transfer processes are shown in Figures 4.4 and 4.5, respectively. The P–v and T–s diagrams for the Brayton cycle are shown in Figures 4.6 and 4.7, respectively.



Figure 4.4 Simple open gas turbine generator



Figure 4.5 Simple closed cycle gas turbine generator


Figure 4.6 P-v diagram for Brayton cycle



Figure 4.7 T-s diagram for an ideal Brayton cycle

The analysis of the Brayton cycle can be done with the control volume concept discussed in Chapter 2 around each of the four devices shown in Figure 4.5. The energy and entropy equations for the components are expressed as follows:

## Compressor:

The energy equation can be expressed as:

$$h_1 + w_c - h_2 = 0 \tag{4.5}$$

The entropy equation can be expressed as:

$$s_1 - s_2 = 0$$
 (isentropic process) (4.6)

It is assumed that no heat is transferred from the compressor to the environment,

$$q = 0 \tag{4.7}$$

#### **Combustion chamber:**

The energy equation can be expressed as:

$$h_2 + h_3 - q_H = 0 (4.8)$$

The entropy equation can be expressed as:

$$s_2 - s_3 + \int \frac{dq}{T} = 0 \tag{4.9}$$

The process in the combustion chamber is a constant pressure process, i.e.

$$P_3 = P_2$$
 (4.10)

#### Turbine:

The energy equation can be expressed as:

$$h_3 - h_4 - w_T = 0 \tag{4.11}$$

The entropy equation can be expressed as:

$$s_3 - s_4 = 0 \text{ (isentropic)} \tag{4.12}$$

It is assumed that no heat is transferred from the turbine to the environment,

$$q = 0 \tag{4.13}$$

#### Heat exchanger:

The energy equation can be expressed as:

$$h_4 - h_1 - q_L = 0 \tag{4.14}$$

The entropy equation can be expressed as:

$$s_4 - s_1 - \int \frac{dq}{T} = 0 \tag{4.15}$$

The process in the heat exchanger is a constant pressure process, i.e.

$$P_4 = P_1$$
 (4.16)

The overall conversion efficiency for the cycle can be expressed as:

$$\eta_{th} = 1 - q_L/q_H = 1 - (h_4 - h_1)/(h_3 - h_2) \approx 1 - C_p(T_4 - T_1)/C_p(T_3 - T_2)$$
  
= 1 - T\_1(T\_4/T\_1 - 1)/T\_2(T\_3/T\_2 - 1) (4.17)

For an ideal cycle we know that the pressure increase in the compressor equals the pressure decrease in the turbine, i.e.

$$P_3/P_4 = P_2/P_1 \tag{4.18}$$

For the two isentropic processes we get the power relations as:

$$P_2/P_1 = (T_2/T_1)^{k/(k-1)} = P_3/P_4 = (T_3/T_4)^{k/(k-1)}$$
(4.19)

$$T_3/T_4 = T_2/T_1 \tag{4.20}$$

Therefore, 
$$T_3/T_2 = T_4/T_1$$
 and  $T_3/T_2 - 1 = T_4/T_1 - 1$  (4.21)

The cycle efficiency can be expressed as:

$$\eta_{th} = 1 - T_1/T_2 = 1 - 1/(P_2/P_1)^{k/(k-1)}$$
(4.22)

The above expression for cycle thermal efficiency was obtained assuming constant specific heats, i.e. the specific heat capacity is independent of temperature. For more accurate calculations, the gas tables should be used, which incorporate the effect of temperature on specific heat. Such a gas table is known as the ideal gas property table and is presented in the following section of this Chapter. In an actual gas turbine cycle, the compressor and the turbine are not isentropic because of the various losses occurring in them. These losses, which are about 15 to 20 percent, significantly reduce the efficiency of the gas turbine cycle. That means if the compressor requires about 60 percent of the turbine's output (a back-work ratio of 0.6), it leaves only 40 percent useful work output. This relatively low limit of turbine work output is experienced when the efficiencies of the compressor and turbine are too low.

The actual gas turbine engine differs from the ideal cycle primarily because of irreversibilities in the compressor and turbine, and because of pressure drop in the flow passages and combustion chamber (or in the heat exchanger of a closed-cycle turbine). Thus, the state points in a simple open-cycle gas turbine might be as shown in Figure 4.8.

The efficiencies of the compressor and turbine are defined in relation to isentropic processes.

Referring to the state points in Figure 4.8, the isentropic efficiencies of compressor and turbine are defined as follows:

$$\eta_{s,comp} = (h_{2s} - h_1)/(h_{2a} - h_1)$$
(4.23)

 $\eta_{s,turb} = (h_3 - h_{4a})/(h_3 - h_{4s}) \tag{4.24}$ 



Figure 4.8 T-s diagram for an actual Brayton cycle

Another important feature of the Brayton cycle is the large amount of compressor work (also called back-work ratio) compared to turbine work. The back-work ratio is the ratio of compressor work, w<sub>comp</sub> to turbine work, w<sub>turb</sub>. It is quite normal for the compressor to consume 40 to 80% of the output of the turbine. This is particularly important in the actual cycle as the effect of losses is considered. The losses result in a larger amount of compression work from the amount of turbine work produced. Thus, the overall efficiency drops very rapidly with a decrease in the efficiencies of the compressor and turbine. In fact, if these efficiencies drop below about 60%, all the work of the turbine will be required to drive the compressor, and the overall efficiency will be zero. This is in sharp contrast to the Rankine cycle (covered in Chapter 5), where only 1 or 2% of the turbine work is required to drive the pump. This demonstrates the inherent advantage of the cycle utilising a condensing working fluid, such that a much larger difference in specific volume between the expansion and compression processes is utilised effectively. The expansion of steam with a greater specific volume is taking place in a work producing device, namely, steam turbine (where a greater specific volume will result in production of more turbine work). Whereas, the compression process, which is by a work consuming device (pump) is for the boiler feedwater (water with a smaller specific volume will result in lower work of consumption of the pump). These two inherent advantages of the Rankine cycle results in greater net work produced and hence, achieving improved thermal efficiency for the cycle.

#### 4.4.1 The regenerative gas-turbine cycle

The efficiency of the gas turbine cycle could be improved by introducing a regenerator. The simple open-cycle gas turbine cycle with a regenerator is shown in Figure 4.9, and the corresponding ideal air-standard cycle with a regenerator is shown on the P– v and T–s diagrams. In cycle 1–2–x-3–4–y-1, the temperature of the exhaust gas leaving the turbine in state 4 is very much greater than the temperature of the gas leaving the compressor. Therefore, heat can be recovered from the exhaust gases to heat the high-pressure gases leaving the compressor. If this is done in a counterflow heat exchanger (a regenerator), the temperature of the high-pressure gas leaving the regenerator ideally will have a temperature equal to T<sub>4</sub>, the temperature of the gas leaving the turbine. Heat transfer from the external source is necessary only to increase the temperature from T<sub>x</sub> to T<sub>3</sub>.

The influence of pressure ratio on the simple gas turbine cycle with a regenerator is shown by considering cycle 1-2'-3'-4-1. In this cycle the temperature of the exhaust gas leaving the turbine is just equal to the temperature of the gas leaving the compressor; therefore, utilising a regenerator is not possible. This can be shown more exactly by determining the efficiency of the ideal gas turbine cycle with a regenerator.

Referring to the state points of Figure 4.10, the thermal efficiency of this cycle with regeneration can be derived as follows:

$$\eta_{th} = w_{net}/q_H = (w_t - w_c)/q_H$$
(4.25)

$$q_H = C_p (T_3 - T_x)$$
(4.26)

$$w_t = C_p (T_3 - T_4) \tag{4.27}$$



Figure 4.9 The regenerative Brayton cycle



Figure 4.10 T-s diagram for the regenerative Brayton cycle

But for an ideal regenerator,  $T_4 = T_x$ , and therefore  $q_H = w_t$ .the resulting thermal efficiency is:

$$\eta_{th} = 1 - w_c / w_t \approx 1 - C_p (T_2 - T_1) / C_p (T_3 - T_4)$$
  
=  $1 - T_1 (T_2 / T_1 - 1) / T_3 (1 - T_4 / T_3)$   
=  $1 - T_1 ((P_2 / P_1)^{(k-1)/k} - 1) / T_3 ((P_2 / P_1)^{(k-1)/k} - 1)$  (4.28)

$$\eta_{th} = 1 - T_1 / T_3 (P_2 / P_1)^{(k-1)/k} = 1 - T_2 / T_3$$
(4.29)

It is evident from the equation (4.28), for the ideal cycle with regeneration, the thermal efficiency depends not only on the pressure ratio but also on the ratio of the minimum to the maximum temperature. We note that, in contrast to the Brayton cycle, the efficiency decreases with an increase in pressure ratio. The effectiveness or efficiency of a regenerator is given by the regenerator efficiency, which is defined with reference to Figure 4.10. State x represents the high-pressure gas leaving the regenerator. In the ideal regenerator there would be only an infinitesimal temperature difference between the two streams, and the high-pressure gas would leave the regenerator at temperature  $T'_x$ , and  $T'_x = T_4$ . In an actual regenerator, which must operate with a finite temperature difference  $T_x$ , the actual temperature leaving the regenerator is therefore less than  $T'_x$ .

Referring to Figure 4.10, the regenerator efficiency is expressed as:

$$\eta_{reg} = (h_5 - h_{2a})/(h_5' - h_{2a}) \tag{4.30}$$

If the specific heat is assumed to be constant, the regenerator efficiency can also be written as:

$$\eta_{reg} = (T_5 - T_{2a}) / (T_5' - T_{2a}) \tag{4.31}$$

A higher efficiency can be achieved by using a regenerator with a greater heattransfer area. However, this also increases the pressure drop, resulting in additional loss. Therefore, both the pressure drop and the regenerator efficiency must be considered in determining which regenerator gives maximum thermal efficiency for the cycle. From an economic point of view, the cost of the regenerator must be weighed against the savings that can be achieved by its use.

Note that this expression for thermal efficiency given in eq.(4.28) is quite different from that for the Brayton cycle. For a given pressure ratio, the efficiency increases as the ratio of minimum to maximum temperature decreases. But, surprisingly, as the pressure ratio increases the efficiency decreases, an effect opposite to that of the Brayton cycle. Hence it is to be anticipated that for a given regenerative cycle temperature ratio, there is a particular pressure ratio for which the efficiency of the Brayton cycle will equal the efficiency of the regenerative cycle. This is shown for a temperature ratio of 0.25 in Figure 4.13.



Figure 4.11 The P-v diagram for a regenerative Brayton cycle



Figure 4.12 Effectiveness of Brayton and regenerative cycle

In actual practice, the temperature of the air leaving the regenerator at state 3 (Figure 4.11) must be less than the temperature of the air entering at state 5. Also,  $T_6 > T_2$ .

#### 4.4.2 The intercooling, reheat, regenerative gas-turbine cycle

In addition to the regenerator of the previous section there are two other common techniques for increasing the thermal efficiency of the gas turbine cycle. First, an intercooler can be incorporated into the compression process; air is compressed to an intermediate pressure, cooled in an intercooler, and then compressed to the final pressure. This reduces the work required for the compressor, as it tends towards the most efficient isothermal process and it reduces the maximum temperature reached in the cycle. The intermediate pressure is determined by equating the pressure ratio for each stage of compression; that is, referring to Figure 4.13,  $P_2/P_1 = P_4/P_3$ . Figure 4.15 shows the effect of multi-stage compression with intercooling. The shaded area is the savings in compressor power input.



Figure 4.13 Brayton cycle with multistage compression, intercooling, reheating and regenerative cycle



Figure 4.14 The effect of multi-stage compression and intercooling on the work input



Figure 4.15 Brayton cycle with multistage compression, intercooling, reheating and regenerative cycle

The second technique for increasing thermal efficiency is through the use of a reheater. The intermediate pressure is determined as in the compressor; we again require that the ratios be equal; that is,  $P_6/P_7 = P_8/P_9$ . Since  $P_9 = P_1$  and  $P_6 = P_4$ , we see that the intermediate turbine pressure is equal to the intermediate compressor pressure for our ideal-gas turbine. Finally, we should note that intercooling and reheating are never used without regeneration. In fact, if regeneration is not employed, intercooling and reheating reduce the efficiency of a gas turbine cycle.

In any actual gas turbine cycle, the temperature effect on specific heat capacity needs to be allowed for. Figure 4.16 shows the ideal gas property table taking into account the temperature effect of specific heat capacity.

Т	h	n	u		s°		
(K)	(kJ/kg)	<b>P</b> r	(kJ/kg)	vr	(kJ/kg*K)		
200	199.97	0.3363	142.56	1707.0	1.29559		
210	209.97	0.3987	149.69	1512.0	1.34444		
220	219.97	0.4690	156.82	1346.0	1.39105		
230	230.02	0.5477	164.00	1205.0	1.43557		
240	240.02	0.6355	171.13	1084.0	1.47824		
250	250.05	0.7329	178.28	979.0	1.51917		
260	260.09	0.8405	185.45	887.8	1.55848		
270	270.11	0.9590	192.60	808.0	1.59634		
280	280.13	1.0889	199.75	738.0	1.63279		
285	285.14	1.1584	203.33	706.1	1.65055		
290	290.16	1.2311	206.91	676.1	1.66802		
295	295.17	1.3068	210.49	647.9	1.68515		
300	300.19	1.3860	214.07	621.2	1.70203		
305	305.22	1.4686	217.67	596.0	1.71865		
310	310.24	1.5546	221.25	572.3	1.73498		
315	315.27	1.6442	224.85	549.8	1.75106		
320	320.29	1.7375	228.42	528.6	1.76690		
325	325.31	1.8345	232.02	508.4	1.78249		
330	330.34	1.9352	235.61	489.4	1.79783		
340	340.42	2.149	242.82	454.1	1.82790		
350	350.49	2.379	250.02	422.2	1.85708		
360	360.58	2.626	257.24	393.4	1.88543		
370	370.67	2.892	264.46	367.2	1.91313		
380	380.77	3.176	271.69	343.4	1.94001		
390	390.88	3.481	278.93	321.5	1.96633		
400	400.98	3.806	286.16	301.6	1.99194		
410	411.12	4.153	293.43	283.3	2.01699		
420	421.26	4.522	300.69	266.6	2.04142		
430	431.43	4.915	307.99	251.1	2.06533		
440	441.61	5.332	315.30	236.8	2.08870		

T	h		u		s°		
(K)	(kJ/kg)	<b>p</b> r	(kJ/kg)	vr	(kJ/kg*K)		
450	451.80	5.775	322.62	223.6	2.11161		
460	462.02	6.245	329.97	211.4	2.13407		
470	472.24	6.742	337.32	200.1	2.15604		
480	482.49	7.268	344.70	189.5	2.17760		
490	492.74	7.824	352.08	179.7	2.19876		
500	503.02	8.411	359.49	170.6	2.21952		
510	513.32	9.031	366.92	162.1	2.23993		
520	523.63	9.684	374.36	154.1	2.25997		
530	533.98	10.37	381.84	146.7	2.27967		
540	544.35	11.10	389.34	139.7	2.29906		
550	555.74	11.86	396.86	133.1	2.31809		
560	565.17	12.66	404.42	127.0	2.33685		
570	575.59	13.50	411.97	121.2	2.35531		
580	586.04	14.38	419.55	115.7	2.37348		
590	596.52	15.31	427.15	110.6	2.39140		
600	607.02	16.28	434.78	105.8	2.40902		
610	617.53	17.30	442.42	101.2	2.42644		
620	628.07	18.36	450.09	96.92	2.44356		
630	638.63	19.84	457.78	92.84	2.46048		
640	649.22	20.64	465.50	88.99	2.47716		
650	659.84	21.86	473.25	85.34	2.49364		
660	670.47	23.13	481.01	81.9	2.50985		
670	681.14	24.46	488.81	78.61	2.52589		
680	691.82	25.85	496.62	75.50	2.54175		
690	702.52	27.29	504.45	72.56	2.55731		
700	713.27	28.80	512.33	69.76	2.57277		
710	724.04	30.38	520.23	67.07	2.58810		
720	734.82	32.02	528.14	64.53	2.60319		
730	745.62	33.72	536.07	62.13	2.61803		
740	756.44	35.50	544.02	59.82	2.63280		

T	h	n	u	v	s°	Т	h	n	u	v	s°
(K)	(kJ/kg)	Pr	(kJ/kg)	vr	(kJ/kg*K)	(K)	(kJ/kg)	Pr	(kJ/kg)	vr	(kJ/kg*K)
750	767.29	37.35	551.99	57.63	2.64737	1340	1443.60	375.3	1058.94	10.247	3.30959
760	778.18	39.27	560.01	55.54	2.66176	1360	1467.49	399.1	1077.10	9.780	3.32724
780	800.03	43.35	576.12	51.64	2.69013	1380	1491.44	424.2	1095.26	9.337	3.34474
800	821.95	47.75	592.30	48.08	2.71787	1400	1515.42	450.5	1113.52	8.919	3.36200
820	843.98	52.59	608.59	44.84	2.74504	1420	1539.44	478.0	1131.77	8.526	3.37901
840	866.08	57.60	<mark>624.95</mark>	41.85	2.77170	1440	1563.51	506.9	1150.13	8.153	3.39586
860	888.27	63.09	<mark>6</mark> 41.40	39.12	2.79783	1460	1587.63	537.1	1168.49	7.801	3.41247
880	910.56	68.98	<mark>657.95</mark>	36.61	2.82344	1480	1611.79	568.8	1186.95	7.468	3.42892
900	932.93	75.29	674.58	34.31	2.84856	1500	1635.97	601.9	1205.41	7.152	3.44516
920	955.38	82.05	<mark>6</mark> 91.28	32.18	2.87324	1520	1660.23	636.5	1223.87	6.854	3.46120
940	977.92	89.28	708.08	30.22	2.89748	1540	1684.51	672.8	1242.43	6.569	3.47712
960	1000.55	97.00	725.02	28.40	2.92128	1560	1708.82	710.5	1260.99	6.301	3.49276
980	1023.25	105.2	741.98	26.73	2.94468	1580	1733.17	750.0	1279.65	6.046	3.50829
1000	1046.04	114.0	758.94	25.17	2.96770	1600	1757.57	791.2	1298.30	5.804	3.52364
1020	1068.89	123.4	776.10	23.72	2.99034	1620	1782.00	834.1	1316.96	5.574	3.53879
1040	1091.85	133.3	793.36	23.29	3.01260	1640	1806.46	878.9	1335.72	5.355	3.55381
1060	1114.86	143.9	810.62	21.14	3.03449	1660	1830.96	925.6	1354.48	5.147	3.56867
1080	1137.89	155.2	827.88	19.98	3.05608	1680	1855.50	974.2	1373.24	4.949	3.58335
1100	1161.07	167.1	845.33	18.896	3.07732	1700	1880.1	1025	1392.7	4.761	3.5979
1120	1184.28	179.7	862.79	17.886	3.09825	1750	1941.6	1161	1439.8	4.328	3.6336
1140	1207.57	193.1	880.35	16.946	3.11883	1800	2003.3	1310	1487.2	3.994	3.6684
1160	1230.92	207.2	897.91	16.064	3.1 <mark>391</mark> 6	1850	2065.3	1475	1534.9	3.601	3.7023
1180	1254.34	222.2	915.57	15.241	3.15916	1900	2127.4	1655	1582.6	3.295	3.7354
1200	1277.79	238.0	933.33	14.470	3.17888	1950	2189.7	1852	1630.6	3.022	3.7677
1220	1301.3 <mark>1</mark>	254.7	951.09	13.747	3.19834	2000	2252.1	2068	1678.7	2.776	3.7994
1240	1324.93	272.3	968.95	13.069	3.21751	2050	2314.6	2303	1726.8	2.555	3.8303
1260	1348.55	290.8	986.90	12.435	3.23638	2100	2377.7	2559	1775.3	2.356	3.8605
1280	1372.24	310.4	1004.76	11.835	3.255 <mark>1</mark> 0	2150	2440.3	2837	1823.8	2.175	3.8901
1300	1395.97	330.9	1022.82	11.275	3.27345	2200	2503.2	3138	1872.4	2.012	3.9191
1320	1419.76	352.5	1040.88	10.747	3.29160						

# Figure 4.16 Ideal gas property table with the temperature effect on specific heat

capacity

# 4.5 Illustrative Examples

# Example 4.5.1

Air enters the compressor of a gas turbine at 100 kPa and 25°C ( $T_1$ ). For a pressure ratio of 5 and a maximum temperature ( $T_3$ ) of 850°C, determine the back-work ratio (BWR) and the thermal efficiency for this Brayton cycle. Assume constant specific heat.



Solution

Back-work ratio, BWR =  $w_{comp}/w_{turb}$ Assuming constant specific heat, BWR =  $C_p(T_2 - T_1) / C_p(T_3 - T_4) = (T_2 - T_1) / (T_3 - T_4)$  $T_1 = 298$  K,  $T_3 = 1,123$  K,  $T_2 = T_1(P_2/P_1)^{(k-1)/k} = 298$  (5)<sup>0.2857</sup> = 472 K (refer to equation 4.19, k = 1.4 for an isentropic process) Similarly,  $T_4 = T_3(P_4/P_5)^{(k-1)/k} = 1,123 (1/5)^{0.2857} = 709.1$  K Therefore, the BWR = (472 - 298)/(1123 - 709) = 0.42 = 42%The thermal efficiency is calculated as follows:  $\eta_{thermal} = 1 - (P_2/P_1)^{(1-k)/k} = 1 - (5)^{-(0.2857)} = 0.369 = 36.9\%$ 

# Example 4.5.2

Assume the compressor and the gas turbine in the previous example each have an efficiency of 75 percent. Determine the back-work ratio (BWR) and the thermal efficiency for the Brayton cycle, assuming constant specific heats



# Solution

The isentropic efficiency of the compressor is given as:

#### $\prod_{s,comp} = w_{s,comp} / w_{a,comp}$

Therefore,  $w_{a,comp} = C_p(T_2 - T_1) / \prod_{s,comp} = (1/0.75) (472 - 298) = 232 \text{ kJ/kg}$ The isentropic efficiency of the turbine is given as:

$$\begin{split} & \prod_{s,turb} = w_{a,turb} / \ w_{s,turb} \\ & \text{Therefore, } w_{a,turb} = C_p(T_3 - T_4) \ x \ \prod_{s,turb} = (0.75) \ (1,123 - 709.1) = 310.42 \ kJ/kg \\ & \text{Therefore, actual BWR} = w_{a,comp} \ / \ w_{a,comp} = 232 \ / 310.42 = 0.747 = 74.7\% \\ & \prod_{thermal} = w_{net} / \ Q_{in} = \ (w_{a,turb} - w_{a,comp}) \ / \ Q_{in}, \ where, \ Q_{in} = C_p(T_3 - T_{2a}) \\ & \text{The actual compressor work also can be written as:} \\ & W_{a,comp} = C_p(T_{2a} - T_1), \ therefore, \ T_{2a} = W_{a,comp} + T_1 = 232 + 298 = 530 \ K \\ & \text{Therefore, the total heat input in the cycle is:} \\ & Q_{in} = C_p(T_3 - T_{2a}) = 1.0(1123 - 530) = 593 \ kJ/kg \\ & \Pi_{thermal} = w_{net} / \ Q_{in} = \ (w_{a,turb} - w_{a,comp}) \ / \ Q_{in} = (310.42 - 232) \ / \ 593 = 0.132 = 13.2\% \end{split}$$

#### Example 4.5.3

Add a regenerator of effectiveness 0.8 to the gas-turbine cycle in the previous example and calculate the thermal efficiency and the back-work ratio, assuming constant specific heats.

#### Solution:

Referring to the T-s diagram below:



 $\Pi_{\text{thermal}} = w_{\text{net}} / Q_{\text{in}} = (w_{a,\text{turb}} - w_{a,\text{comp}}) / Q_{\text{in}}$ 

Here, the presence of a regenerator results in a reduction in the addition of heat to the cycle.

Referring to the Figure above, the heat added in the cycle,  $Q_{in}$  can be expressed as:

$$Q_{in} = C_p(T_3 - T_5)$$

$$\Pi_{s,turb} = (T_3 - T_{4a})/(T_3 - T_4)$$

Therefore,  $T_{4a} = 1,123 - [0.75 (1,123 - 709.1)] = 812.5 \text{ K}$ 

The effectiveness of the regenerator is defined as:

 $\mathcal{E} = (T_3 - T_5)/(T_3 - T_{4a})$ 

Substituting the known temperatures in the effectiveness expression and cross multiplying yields:

$$\begin{split} T_5 &= 1123 - [0.8 \ (1,123 - 812.5)] = 874.6 \ K \\ \text{Therefore, } Q_{\text{in}} &= C_p(T_3 - T_5) = 1 \ (1,123 - 874.6) = 248.4 \ K \\ &\prod_{\text{thermal}} = w_{\text{net}} / \ Q_{\text{in}} = \ (w_{\text{a,turb}} - w_{\text{a,comp}}) \ / \ Q_{\text{in}} = \ (310.42 - 232) \ / \ 248.4 = 0.315 = 31.5\% \\ \text{The back-work ratio in this case does not change based on the information provided.} \end{split}$$

# Example 4.5.4

In an ideal Brayton cycle, the air enters the compressor at 1-bar and 30°C. The pressure of the air leaving the compressor is 12-bar and the maximum temperature in the power cycle is 1,100°C. Taking the temperature effect of specific heat capacity, determine:

- a. The pressure and temperature at each point in the cycle
- b. The compressor work
- c. The turbine work
- d. Back-work ratio
- e. The thermal efficiency of the cycle.

## Solution

Using the steam property table,  $v_1 = v_2 = v_{cr} = 0.003106 \text{ m}^3/\text{kg}$ The total mass, m = volume/specific volume = V/v = 0.4/0.003106 = 128.78 kg



(a) For the given information, T1 = 30 + 273 = 303 K, using the ideal gas property table of Figure 4.16,

 $h_1 = 303.2 \text{ kJ/kg}, P_{r1} = 1.435$ 

 $P_{r1}/P_{r2} = P_1/P_2$ , therefore,  $P_{r2} = P_{r1} \times P_2/P_1 = 1.435 \times 12 = 17.22$ 

For  $P_{r2}$  = 17.22, using the ideal gas property table,

$$\begin{split} h_2 &= 616.7 \text{ kJ/kg}, \text{ } \text{T}_2 = 609.2 \text{ K} \\ \text{For the given information, } \text{T}_3 = 1,100 + 273 = 1,373 \text{ K}, \text{ using the ideal gas property} \\ \text{table of Figure 4.17,} \\ h_3 &= 1,483.05 \text{ kJ/kg}, \text{P}_{r3} = 415.41 \\ \text{P}_{r3}/\text{ P}_{r4} &= \text{P}_3/\text{ P}_4, \text{ therefore, } \text{P}_{r4} = \text{P}_{r3} \times \text{P}_4/\text{ P}_3 = 415.41/12 = 34.61 \\ \text{For P}_{r4} &= 34.61, \text{ using the ideal gas property table,} \\ h_4 &= 751.03 \text{ kJ/kg}, \text{T}_4 = 735 \text{ K} \\ \text{(b) Referring to the ideal Brayton cycle T-s diagram,} \\ \text{The compressor work input, } w_{comp} = h_2 - h_1 = 616.7 - 303.2 = 313.5 \text{ kJ/kg} \\ \text{(c) Referring to the ideal Brayton cycle T-s diagram,} \\ \text{The turbine work produced, } w_{turb} = h_3 - h_4 = 1,483.05 - 751.03 = 732.02 \text{ kJ/kg} \end{split}$$

(d) Back work ratio, BWR =  $w_{comp}/w_{turb}$  = 313.5/732.02 = 0.428 = 42.8%

(e)  $\Pi_{\text{thermal}} = w_{\text{net}} / Q_{\text{in}} = (w_{\text{turb}} - w_{\text{comp}}) / Q_{\text{in}}$   $Q_{\text{in}} = h_3 - h_2 = 1483.05 - 616.7 = 866.35 \text{ kJ/kg}$ = (732.02 - 313.5) / 866.35 = 0.483 = 48.3%

## Example 4.5.5

In an actual Brayton cycle, the air enters the compressor at 1-bar and 30°C. The pressure of the air leaving the compressor is 12-bar and the maximum temperature in the power cycle is 1,100°C. Determine:

a. The temperature at each point in the cycle

- b. The compressor work
- c. The turbine work
- d. Back-work ratio
- e. The thermal efficiency of the cycle

Assume the isentropic efficiencies of compressor and turbine as 80% and 85%, respectively.

Solution



(a) For the given information, T1 = 30 + 273 = 303 K, using the ideal gas property table of Figure 4.16,

 $h_1 = 303.2 \text{ kJ/kg}, P_{r1} = 1.435$  $P_{r1}/P_{r2} = P_1/P_2$ , therefore,  $P_{r2} = P_{r1} \times P_2/P_1 = 1.435 \times 12 = 17.22$ For  $P_{r2} = 17.22$ , using the ideal gas property table,  $h_2 = 616.7 \text{ kJ/kg}, T_2 = 609.2 \text{ K}$  $\Pi_{s,comp} = W_{s,comp} / W_{a,comp} = (h_2 - h_1) / (h_{2a} - h_1)$ Therefore,  $h_{2a} = [(616.7 - 303.2)/0.8] + 303.2 = 695.08 \text{ kJ/kg}$ For  $h_{2a} = 695.08$  kJ/kg, using the ideal gas property table,  $T_{2a} = 683 \text{ K}$ For the given information,  $T_3 = 1100 + 273 = 1,373$  K, using the ideal gas property table of Figure 4.17,  $h_3 = 1,483.05 \text{ kJ/kg}, P_{r_3} = 415.41$  $P_{r3}/P_{r4} = P_3/P_4$ , therefore,  $P_{r4} = P_{r3} \times P_4/P_3 = 415.41/12 = 34.61$ For  $P_{r4} = 34.61$ , using the ideal gas property table, h<sub>4</sub> = 751.03 kJ/kg, T<sub>4</sub> = 735 K  $\Pi_{s,turb} = W_{a,turb} / W_{s,turb} = (h_3 - h_{4a}) / (h_3 - h_4)$ Therefore, h<sub>4a</sub> = 1,483.05 - [(1,483.05 - 751.03) x 0.85] = 860.83 kJ/kg For  $h_{4a} = 860.83$  kJ/kg, using the ideal gas property table,  $T_{4a} = 835.2 \text{ K}$ 

(b) Referring to the actual Brayton cycle T-s diagram, The compressor work input,  $w_{acomp} = h_{2a} - h_1 = 695.08 - 303.2 = 391.88 \text{ kJ/kg}$ 

(c) Referring to the ideal Brayton cycle T-s diagram, The turbine work produced,  $w_{aturb} = h_3 - h_{4a} = 1,483.05 - 860.83 = 622.22 \text{ kJ/kg}$ 

(d) Back work ratio, BWR =  $w_{acomp}/w_{aturb}$  = 391.88/622.22 = 0.63 = 63%

(e)  $\Pi_{\text{thermal}} = w_{\text{net}} / Q_{\text{in}} = (w_{\text{aturb}} - w_{\text{acomp}}) / Q_{\text{in}}$   $Q_{\text{in}} = h_3 - h_{2a} = 1,483.05 - 695.08 = 787.97 \text{ kJ/kg}$ = (622.22 - 391.88) / 787.97 = 0.292 = 29.2%

#### Example 4.5.6

Consider an ideal gas turbine cycle with two stages of compression and two stages of expansion. The pressure ratio across each stage for the compression and expansion is 8:1. Air enters the compressor at 1-bar and 30°C. The combustion gas enters each turbine with a temperature of 1,100°C. In addition, the cycle is incorporated with a regenerator with an effectiveness of 70%. Taking the effect of temperature on the specific heat capacity, determine:

- b. The compressor work
- c. The turbine work
- d. Back-work ratio
- e. The thermal efficiency of the cycle

Solution



(a) For the given information, T1 = 30 + 273 = 303 K, using the ideal gas property table of Figure 4.16,

 $\label{eq:h1} \begin{array}{l} h_1 = 303.2 \ \text{kJ/kg}, \ P_{r1} = 1.435 \\ P_{r1}/\ P_{r2} = P_1/\ P_2, \ \text{therefore}, \ P_{r2} = P_{r1} \ x \ P_2/\ P_1 = 1.435 \ x \ 8 = 11.48 \\ \text{For } P_{r2} = 11.48, \ \text{using the ideal gas property table}, \\ h_2 = 550.05 \ \text{kJ/kg}, \ T_2 = 560 \ \text{K} \end{array}$ 

For the given information,  $T_3 = 1,100 + 273 = 1,373$  K, using the ideal gas property table of Figure 4.16,

 $h_3 = 1,483.05 \text{ kJ/kg}, P_{r_3} = 415.41$ 

 $P_{r3}/P_{r4} = P_3/P_4$ , therefore,  $P_{r4} = P_{r3} \times P_4/P_3 = 415.41/12 = 34.61$ 

For  $P_{r4} = 34.61$ , using the ideal gas property table, h<sub>4</sub> = 751.03 kJ/kg, T<sub>4</sub> = 735 K

(b) Referring to the ideal Brayton cycle T-s diagram and assuming the intercooling after the first stage of the compression cools down the compressed gas to the initial temperature of 30°C,

The compressor work input,  $w_{comp} = 2 x (h_2 - h_1) = 2 x (550.05 - 303.2) = 493.7 kJ/kg$ 

(c) Referring to the ideal Brayton cycle T-s diagram and assuming the reheating after the first stage of the expansion reheats the compressed gas to the initial turbine inlet temperature of 1100°C,

The turbine work,  $w_{turb} = 2 x (h_3 - h_4) = 2 x (1,483.05 - 751.03) = 1,464 kJ/kg$ 

(d) Back work ratio, BWR =  $w_{comp}/w_{turb} = 493.7/1,464 = 0.337 = 33.7\%$ 

(e)  $\Pi_{thermal} = w_{net}/Q_{in} = (w_{turb} - w_{comp}) / Q_{in}$   $Q_{in} = (h_6 - h_4) + (h_8 - h_7)$ Referring to the two-stage T-s diagram above,  $T_6 = 1,373 \text{ kJ/kg}, P_{r6} = 415.41$   $P_{r6}/P_{r7} = P_6/P_7$ , therefore,  $P_{r7} = P_{r6} \times P_7/P_6 = 415.41/8 = 51.92$ For  $P_{r7} = 51.92$ , using the ideal gas property table,  $h_7 = 840.93 \text{ kJ/kg}, T_7 = 817.2 \text{ K}$ Therefore,  $Q_{in} = (1,483.05 - 751.03) + (1,483.05 - 840.93) = 1,374.14 \text{ kJ/kg}$  $\Pi_{thermal} = (1,464 - 493.7) / 1,374.14 = 0.706 = 70.6\%$ 

#### Example 4.5.7

A two-stage air compressor has an intercooler between the two stages. Air enters the compressor at 1-bar and 30°C and the air leaving the compressor has a pressure of 16-bar. The constant pressure intercooler cools the air to the inlet temperature. Determine the specific compressor work and the intercooler heat transfer rate.

#### Solution



(a) For the given information,  $T_1 = 30 + 273 = 303$  K, using the ideal gas property table of Figure 4.16,

 $h_1 = h_3 = 303.2 \text{ kJ/kg}, P_{r1} = 1.435$ 

For a two-stage compression process, the pressure ratio is 4.  $P_{r1}/P_{r2} = P_1/P_2$ , therefore,  $P_{r2} = P_{r1} \times P_2/P_1 = 1.435 \times 4 = 5.74$ 

For  $P_{r2} = 5.74$ , using the ideal gas property table,

h<sub>2</sub> = 451 kJ/kg, T<sub>2</sub> = 450 K

Referring to the two-stage ideal Brayton cycle T-s diagram and assuming the intercooling after the first stage of the compression cools down the compressed gas to the initial temperature of 30°C,

The compressor work input,  $w_{comp} = 2 \times (h_2 - h_1) = 2x(451 - 303.2) = 295.6 \text{ kJ/kg}$ The intercooler heat transfer rate,  $Q_{intercooler} = (h_2 - h_3) = (451 - 303.2) = 147.8 \text{ kJ/kg}$ 

## Example 4.5.8

The gas turbine cycle shown in the Figure is part of a CHP system. In the first turbine, the gas expands to a pressure of  $P_5$ . The gas is then expanded in the second turbine connected to the drive wheels. Considering the working fluid to be air throughout the entire cycle and assuming the cycle to be an ideal one, taking the temperature effect of specific heat, determine:

- a. The intermediate pressure P<sub>5</sub>
- b. The specific work output from the gas turbines
- c. The mass flow rate through the turbine
- d. The air temperature entering the combustor T<sub>3</sub>
- e. The thermal efficiency of the gas turbine cycle



#### Solution

(a) For the given pressure ratio of 8, the optimum intermediate pressure is:

 $P_{\text{intermediate}} = \sqrt{P_1 P_2} = \sqrt{8} = 2.83$ -bar

For the given information,  $T_4 = 1,500$  K, using the ideal gas property table of Figure 4.16,

 $h_4 = h_6 = 1,635.97 \text{ kJ/kg}, P_{r4} = 601.9 \\ P_{r4}/P_{r5} = P_4/P_5, \text{ therefore}, P_{r5} = P_{r1} \times P_5/P_4 = 601.9/2.83 = 212.6$ 

For  $P_{r5}$  = 212.6, using the ideal gas property table, h<sub>5</sub> = h<sub>7</sub> = 1,239.35 kJ/kg, T<sub>5</sub> = 1,167 K (b) Referring to the ideal Brayton cycle T-s diagram and assuming the reheating after the first stage of the expansion reheats the compressed gas to the initial turbine inlet temperature of 1500 K,

The turbine work,  $w_{turb} = 2 \times (h_4 - h_5) = 2 \times (1,635.97 - 1,239.35) = 793.24 \text{ kJ/kg}$ 

(c) For the given information,  $T_1 = 300$  K, using the ideal gas property table of Figure 4.16,

 $h_1 = 300.19 \text{ kJ/kg}, P_{r1} = 1.386$ 

 $P_{r2}/P_{r1} = P_2/P_1$ , therefore,  $P_{r2} = P_{r1} \times P_2/P_1 = 1.386 \times 8 = 11.08$ 

For  $P_{r2} = 11.08$ , using the ideal gas property table,

 $h_2 = 544.06 \text{ kJ/kg}, T_2 = 539.7 \text{ K}$ 

The compressor work,  $w_{comp} = h_2 - h_1 = 544.06 - 300.19 = 243.87 \text{ kJ/kg}$ 

Net power from the gas turbine cycle,  $w_{\text{net}}$  =  $w_{\text{turb}}$  -  $w_{\text{comp}}$ 

= 793.24 - 243.87 = 549.37 kJ/kg

 $P_{net} = \dot{m}_g x w_{net}$ ; therefore, the gas mass flowrate through the turbine is:

 $\dot{m}_g = P_{net} / w_{net} = 175/549.37 = 0.319 \text{ kg/s}$ 

(d) The effectiveness of the regenerator is assumed to be 0.75 Therefore,  $\mathcal{E} = (h_3 - h_2)/(h_7 - h_2)$ Solving the effectiveness expression for  $h_3$ :  $h_3 = [(1,239.35 - 544.06) \times 0.75] + 544.06 = 1,065.52 \text{ kJ/kg}$ 

For  $h_3 = 1,065.52$ , using the ideal gas property table,  $T_3 = 1,017 \text{ K}$   $\Pi_{\text{thermal}} = w_{\text{net}}/Q_{\text{in}} = (w_{\text{turb}} - w_{\text{comp}}) / Q_{\text{in}}$   $Q_{\text{in}} = (h_4 - h_3) + (h_6 - h_5) = (1,635.97 - 1,065.52) + (1,635.97 - 1,239.35)$  = 967.07 kJ/kg $\Pi_{\text{thermal}} = 549.37 / 967.07 = 0.568 = 56.8\%$ 

#### Summary

The gas power cycle or Brayton cycle, which is widely used in combined heat and power cycles, has been presented. The Brayton cycle analyses is done for both the ideal and actual cycles. Various thermal efficiency improvement techniques have also been discussed with their analysis. A number of illustrative examples are included at the end of the Chapter to give practising industrial professionals a good feel of the analysis of the ideal and actual Brayton cycles.

## References

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# **5.0 VAPOUR POWER CYCLES**

In CHP systems, it is very common to have a steam turbine deployed as a bottoming cycle to generate electricity and heat. This arrangement is very common in countries where there is no premium fuel like natural gas. The steam turbines work on vapour power cycles, in which the working fluid, water undergoes phase changes in the cycle components like boiler, turbine and condensers. Hence, practising engineers are recommended to have thorough understanding of the vapour power cycles with the phase change processes depicted in P-v and T-s diagrams.

This chapter presents the vapour power cycles with the main focus on the gas turbine cycle or Brayton cycle.

# Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. Basics of vapour power cycles
- 2. The Rankine vapour power cycle used for CHP systems
- 3. Thermal efficiency of the Rankine cycle
- 4. Various thermal efficiency enhancement techniques for the Rankine cycle

# 5.1 Vapour Power Cycles

CHP systems using steam power plants are operated in a cycle. In a cycle, the working fluids such as water or gas undergo a series of processes and finally return to the initial state. In some other power plants like the internal combustion engine and the gas turbine, the working fluid does not go through a thermodynamic cycle despite the engine operating in a mechanical cycle. In this instance, the working fluid is in a different composition or is in a different state at the end of the process than its starting state. Such equipment is sometimes said to operate on an open cycle (the word 'cycle' is a misnomer though). Steam power plants, which are widely used around the world, operate on a closed cycle.

In this Chapter, idealised vapour power cycles are discussed and analysed. Subsequently, an real-life cycle is considered and how the actual apparatus deviates from the ideal is analysed. In addition, various techniques are incorporated to modify the basic cycles that are intended to improve performance.

## 5.2 The Carnot Vapour Cycle

The Carnot cycle is the most efficient cycle operating between two specified temperature limits. Hence, the Carnot cycle is used as an ideal cycle for vapour power plants, as a comparison for the performance of the actual cycles. However, it is to be noted that the Carnot cycle is not a suitable model for vapour power cycles. In all our subsequent analysis of the Carnot cycle, we assume steam to be the working fluid since it is the working fluid used in vapour power cycles. Figure 5.1 shows a steady-flow Carnot cycle operated within the saturation dome of a pure substance. Heat is added to the fluid reversibly and isothermally in a boiler represented by the process line 1-2. The process 2-3 shows an isentropic expansion of the fluid in a turbine. The process 3-4 is a condensation process, which is reversible and isothermal, and takes place in a condenser. The fluid returns to its initial state after the isentropic compression process 4-1 in a compressor.



Figure 5.1 An ideal Carnot cycle



Figure 5.2 T-s diagram of an ideal Carnot cycle

However, there are a number of impracticalities associated with the cycle:

1. Isothermal heat transfer to or from a two-phase system is not difficult to achieve in practice since maintaining a constant pressure in the device automatically fixes the temperature at the saturation value. Therefore, processes 1-2 and 3-4 can be approached closely in actual boilers and condensers. The maximum temperature of the cycle is limited by limiting the heat transfer processes to two-phase systems as it has to remain under the critical-point value, which is 374°C for water. Based on the definition of Carnot cycle efficiency, the thermal efficiency of the cycle is limited by the maximum temperature of the cycle.

2. The expansion process 2-3 can be approximated as close to isentropic by a welldesigned turbine. However, as can be seen from Figure 5.2, the quality of the steam decreases during this process. The poor quality of the steam is not healthy for the turbine as it has to handle moisture laden steam, which may cause catastrophic damage. That is why it is a common practice that steam with quality (dryness fraction) of less than about 90 percent is not recommended in the operation of steam power plants.

3. The isentropic compression process 4-1 of the Carnot cycle as shown in Figure 5.2 is the compression of a two-phase (liquid–vapour) mixture to a saturated liquid. There are two difficulties associated with the process. Firstly, it is not easy to control the condensation process so precisely as to end up with the desired quality at state 4. Secondly, it is not practical to design a compressor that handles two phases. Some of these problems could be eliminated by modifying the Carnot cycle. However, such a modified Carnot cycle could introduce other problems such as isentropic compression to extremely high pressures and isothermal heat transfer at variable pressures. Therefore, it is evident that the Carnot cycle is not a practical cycle and is not a realistic model for vapour power cycles.

Referring to the T-s diagram for the Carnot cycle shown in Figure 5.2, the following equations for various processes can be written:

Heat input in boiler 
$$Q_{12} = h_2 - h_1$$
 (5.1)

Work output in turbine  $W_{23} = h_2 - h_3$  (5.2)

Heat rejected in condenser $Q_{34} = h_3 - h_4$ (5)	.3)
---	-----

Work input in compressor  $W_{41} = h_4 - h_1$  (5.4)

Net work 
$$W_{net} = W_{23} - W_{41} = Q_{12} - Q_{34}$$
 (5.5)

Work ratio 
$$r_w = (W_{23} - W_{41})/W_{23}$$
 (5.6)

For fixed upper and lower temperature limits, the thermal efficiency of the Carnot cycle is the highest and can be expressed as:

$$\eta_c = 1 - T_3 / T_1 \tag{5.7}$$

As can be seen from eq (5.7), the thermal efficiency increases with the increase of average temperature at which heat is added to the system, or with the decrease of average temperature at which heat is rejected from the system.

#### 5.3 The Rankine Cycle

Most of the impracticalities associated with the Carnot cycle can be eliminated by heating the steam to the saturated state in the boiler and condensing it completely in the condenser. Such a modification is depicted schematically on a T-s diagram in Figure 5.4. The resulting cycle is called the Rankine cycle, which is the ideal cycle for vapour power plants as shown in Figure 5.3. It is assumed that such an ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes. The processes, referring to both Figures 5.3 and 5.4, are:

Isentropic compression in a pump, process 5-6 Constant pressure heat addition in a boiler, process 6-2 Isentropic expansion in a turbine, process 2-3 Constant pressure heat rejection in a condenser, process 3-5



Figure 5.3 The modified Carnot cycle (ideal Rankine cycle)



Figure 5.4 T-s diagram of an ideal Rankine cycle

As seen from Figure 5.3, the water enters the pump at state 5 as saturated liquid and is compressed isentropically to the operating pressure and fed to the boiler as feedwater. The water temperature increases due to a slight decrease in the specific volume of water during this isentropic compression process. The vertical distance between states 5 and 6 on the T-s diagram is greatly exaggerated for clarity and simplification. It will be interesting to note that if water were truly incompressible, there would not be a temperature change at all during this process. The feedwater enters the boiler as a compressed liquid at state 6 and leaves as a saturated vapour at state

2. The boiler is essentially a large heat exchanger where the heat generated from combustion gases is transferred to the water at near constant pressure. The boiler, together with the section where the steam is superheated, if any, is often called the steam generator. The section of the boiler where the steam is superheated is called the superheater.

The saturated steam vapour at state 2 enters the turbine, where it is expanded isentropically and produces work by generating a torque on the shaft connected to an electric generator. Both the pressure and the temperature of the steam decrease during this process to values at state 3. At state point 3, the steam enters the condenser. At this point, the steam is usually a saturated liquid–vapour mixture with a relatively high quality. The steam undergoes a phase-change in the condenser and is condensed at near constant pressure. As in the case of the boiler, a condenser is basically a large heat exchanger exchanging heat (heat rejection) to a cooling medium such as a lake, a river, or the atmosphere. The steam leaves the condenser at a saturated liquid state and enters the pump, thus completing the cycle. In areas where water is scarce and precious, the power plants are cooled by air instead of water. This method of cooling is considered in those places where it is highly important to conserve water.

Referring to the T-s diagram of Figure 5.4, the area under process curve 6-2 represents the heat transferred to the water in the boiler and the area under the process curve 3-5 represents the heat rejected in the condenser. The area enclosed by the cycle curve is the net work produced during in the cycle. As seen from the T-s diagram, the area enclosed is the difference between the heat added in the boiler and heat rejected in the condenser. An energy analysis of the ideal Rankine cycle includes the four components in the cycle: the pump, boiler, turbine, and condenser, which are steady-flow devices as described in Chapter 2 of this manual. Therefore, the four processes that make up the Rankine cycle can be analysed as steady-flow processes. It is to be noted that the kinetic and potential energy changes of the steam are usually small relative to the work and heat transfer terms and are therefore usually neglected. Referring to the T-s diagram for the Rankine cycle and using the steady flow energy analysis described in Chapter 2, the thermal efficiency of the Rankine cycle can be established as follows:

Heat input in boiler  $Q_{62} = h_2 - h_6$ 

(5.8)

Work output in turbine 
$$W_{23} = h_2 - h_3$$
 (5.9)

Heat rejected in condenser 
$$Q_{34} = h_3 - h_4$$
 (5.10)

Work input in pump 
$$W_{56} = h_6 - h_5$$
 (5.11)

As the enthalpy at point 6 is difficult to establish from the steam property table, it is indirectly determined using the pump work equation:

$$W_{56} = v_{f@5}(P_6 - P_5) \tag{5.12}$$

Where,  $v_{f@5}$  = Specific volume at state point 5

Net work from the cycle, 
$$W_{net} = W_{23} - W_{56} = Q_{62} - Q_{34}$$
 (5.13)

Work ratio is expressed as: 
$$r_w = W_{net}/W_{23}$$
 (5.14)

Therefore, the thermal efficiency of the Rankine cycle:

$$\eta_{th,Rankine} = W_{net}/Q_{62} = (Q_{62} - Q_{34})/Q_{62}$$
(5.15)

The Carnot cycle efficiency is maximum because the heat added and rejected are at constant upper and lower temperature, respectively. In sharp contrast, for Rankine Cycle, the heat added and rejected in the Rankine cycle are not at constant upper and lower temperature, respectively. Therefore, the Rankine thermal efficiency is less than that of the Carnot cycle.

That is : 
$$\eta_{th,Rankine} < \eta_{th,Carnot}$$
 (5.16)

As previously stated, with reference to the efficiency of the Carnot cycle, the Rankine cycle thermal efficiency can be optimised through elevating the temperature at which the heat is added to the cycle or decreasing the temperature at which the heat is rejected. This can be achieved by carrying out one of the three techniques alone or through the combination of the techniques:

- (i) Lowering the condenser pressure
- (ii) Increasing boiler pressure

(iii) Superheating steam to high pressure

## 5.3.1 Deviation of actual vapour power cycles from idealised ones

The actual vapour power cycle is different from the ideal Rankine cycle depicted in Figure 5.3. This departure of the actual cycle is because of associated irreversibilities in various components of the cycle. The entropy generation resulting from fluid friction and heat loss (heat transfer between two finite temperature differences) to the surroundings is the source of the irreversibility. Steam leaves the boiler at a somewhat lower pressure because of the pressure drop resulting from fluid friction. This causes pressure drops in the boiler, the condenser, and the piping between various components. In addition, the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to the pressure drop in the connecting pipes. The pressure drop in the condenser is usually very small. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than that required by the ideal cycle. This necessitates a larger pump and larger work input to the pump contributing to the lower thermal efficiency of the actual Rankine cycle. The other major source of irreversibility is the transmission line heat loss from the steam to the surroundings as the steam flows through various components. The larger the steam transmission network, the higher the transmission heat loss, resulting in lower thermal efficiency of the cycle. To maintain the same level of cycle net work output, more heat needs to be transferred to the steam in the boiler to compensate for these undesirable cycle heat losses. This results in an undesirable drop in the thermal efficiency of the actual Rankine cycle. The T-s diagram of an actual Rankine cycle is shown in Figure 5.5.



Figure 5.5 T-s diagram of an actual Rankine cycle

Among the above stated irreversibilities, of particular importance are those associated with the pump and the turbine. Thermodynamically, pumps and compressors are classified as work-consuming devices whereas the turbines are work-producing devices. The deviation of actual pumps and turbines from the isentropic ones can be accounted for by defining isentropic efficiencies. With reference to Figure 5.5, the isentropic efficiencies of pump and turbine are defined as:

$$\eta_{s,pump} = W_{pump,ideal} / W_{pump,actual} = (h_{2s} - h_1) / (h_{2a} - h_1)$$
(5.17)

$$\eta_{s,turb} = W_{turb,actual} / W_{turb,ideal} = (h_3 - h_{4a}) / (h_3 - h_{4s})$$
(5.18)

As seen from the T-s diagram of Figure 5.5, the state points 2a and 4a are the actual exit state points of the pump and the turbine, respectively, and 2s and 4s are the corresponding states for the isentropic cases. The analysis of actual vapour power cycles should also take other factors into consideration. In actual condensers, for example, the liquid is usually sub-cooled to prevent cavitation. The rapid vapourisation and condensation of the fluid at the low-pressure side of the pump impeller may damage it. In addition, losses occur at the bearings between the moving parts as a result of friction. Also, steam and air leaks during the cycle operation are other two sources of loss, which affects the cycle thermal efficiency. Finally, the power consumed by auxiliary equipment such as fans that supply air to the furnace should also be considered in evaluating the overall performance of power plants. The power plant auxiliary equipment such as feedwater pump and fan, could if possible be optimised based on the load, which will help to improve the cycle efficiency.

**5.3.2 Improvement opportunities to increase the efficiency of the Rankine cycle** Considering the unprecedented technological advancement in recent years, there could be many improvement opportunities for steam power plants. Considering that steam power plants are responsible for the production of most electric power around the world, it is of paramount importance to attempt to improve their thermal efficiency by incorporating available state-of-the-art technologies as even small increase in thermal efficiency can result in huge fuel savings. As described in the preceding section, the basic idea behind all the modifications to increase the thermal efficiency of a power cycle remains the same: that is to increase the average temperature at which heat is added to the steam in the boiler or decrease the average temperature at which heat is rejected from the working fluid in the condenser.

The three techniques that can be used for realising this for the simple ideal Rankine cycle are described below:



## 5.3.2.1 Lowering the Condenser Pressure

Figure 5.6 T-s diagram showing lowering of condenser pressure in a Rankine cycle

As seen from the Rankine cycle T-s diagram in the preceding section, the heat rejection process line can be lowered by reducing the condenser pressure in the cycle. This in turn reduces the heat rejection temperature and improving the cycle thermal efficiency. Steam from the turbine enters as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. As a result, lowering the operating pressure of the condenser will translate into a lower temperature of the steam, which means the temperature at which heat is rejected is reduced as well. Figure 5.6 shows the effect of lowering the condenser pressure on the Rankine cycle efficiency. It is to be noted that for performance comparison, the turbine inlet state is maintained the same. The shaded area on this diagram represents the increase in net work output as a result of lowering the condenser pressure pressure from P4 to P4'.

As seen from the Figure, the heat input requirements also increase, represented by the area under curve 5'-1-2, but this increment is very small. Therefore, the net effect

of lowering the condenser pressure is an increase in the thermal efficiency of the Rankine cycle. Steam power plants usually operate well below the atmospheric pressure. This is to take maximum advantage of the increased efficiencies at low pressures. There is no major problem associated with this below-atmospheric operation of the condenser because of the closed loop operation of vapour power cycles. However, lowering of the condenser pressure is limited by the availability of cooling medium in terms of cooling temperature. Obviously, it cannot be lower than the saturation pressure corresponding to the temperature of the cooling medium. For effective heat transfer in the condenser, the difference between the saturation temperature of the steam and cooling medium temperature should at least be kept at 10°C. In a tropical country like Singapore, the cooling water temperature can range from 26 to 29.4°C. Based on this cooling water temperature and the temperature difference requirement, the lowest pressure corresponding to the saturation temperature of 36 to 39.4°C is in the range of 0.06 to 0.07-bar. This means there is a derating requirement when the power plant is procured from a country where the climate is different from that in tropical countries.

When the condenser pressure is lowered, it may cause leakage into the condenser of lower condenser pressure, which is less than atmospheric pressure. A very low condenser pressure also increases the moisture content of the steam at the final stages of the turbine, as can be seen from Figure 5.6. The presence of large quantities of moisture is not desirable as it decreases the turbine efficiency and subjects turbine blades to erosion and damage.

The effects of lowering the condenser pressure is summarised below:

- Condenser pressure is reduced from P<sub>3'</sub> to P<sub>3</sub>
- Steam rejects heat at saturation temperature corresponding to condenser
   pressure
- Lowering condenser pressure means lowering heat rejection temperature
- Net work output increases with decrease in condenser pressure
- Pump work W<sub>4'5'</sub>>W<sub>45</sub>, (however this increase is very small)
- Net effect is an increase of overall thermal efficiency
- Condenser pressure should be saturation pressure corresponding to about 10°C higher than the coolant temperature
- Dryness fraction of steam decreases

## 5.3.2.2 Superheating Steam to High Temperature

Lowering the condenser pressure to make the heat rejection at the condenser as low as possible results in the undesirable lower quality of the steam at the final stages of the expansion in the turbine. This undesirable effect could be mitigated through superheating the steam. Superheating the steam definitely elevates the average temperature at which heat is transferred to the steam in the boiler without increasing the boiler pressure. The effect of superheating on the performance of vapour power cycles is depicted on the T-s diagram shown in Figure 5.8. The net work increase through this superheating of the steam is indicated by the shaded area on the T-s diagram. However, there is an increase in the heat input to the cycle. Thus, both the net work and heat input increase as a result of superheating the steam to a higher temperature. The net effect is an increase in thermal efficiency of the cycle, resulting from the increase in the average temperature at which heat is added in the boiler. Metallurgical conditions limit the temperature to which steam can be superheated. Based on state-of-the-art technology, the highest steam temperature allowed at the turbine inlet is about 620°C. Any increase in this value depends on improving currently available materials or finding new ones that can withstand higher temperatures. In this regard ceramics are considered to be potential materials in the future.



Figure 5.7 Superheating of steam in a Rankine cycle


Figure 5.8 T-s diagram showing superheating of steam in a Rankine cycle

The effects of superheating the steam in the boiler are summarised below:

- Average temperature at which heat is added to steam can be increased without increasing boiler pressure by superheating the steam to high temperatures
- Both heat input and net work increase, while the steam flow rate remains unchanged
- The thermal efficiency of the cycle increases
- Dryness fraction of steam increases with superheating, which is desirable
- Specific steam consumption, SSC (mass flow of steam required per unit power output) decreases

# 5.3.2.3 Increasing Boiler Pressure





Figure 5.9 T-s diagram showing increasing of pressure in a Rankine cycle

Another way of increasing the average temperature during the heat-addition process is to increase the operating pressure of the boiler. This, in turn, raises the average temperature at which heat is transferred to the steam and thus raises the thermal efficiency of the cycle. From the first principle, it is clear that increasing boiler pressure will cause the temperature of heat addition to rise. This will result in a thermal efficiency increase of the Rankine cycle. The effect of increasing the boiler pressure is shown in Figure 5.9. As can be seen from the figure, the dryness fraction of steam decreases, which is not desirable. However, this low dryness fraction or steam quality issue can be corrected through reheating the steam.

Operating pressures of boilers have gradually increased over the years from about 28-bar in 1922 to over 300-bar today, generating enough steam to produce a net power output of 1 GW or more in a large power plant. There are many modern supercritical steam power plants.

The effects of increasing the boiler pressure is summarised below:

- Increasing boiler pressure will increase temperature of heat addition
- Thermal efficiency of cycle increases
- Dryness fraction of steam decreases, which is not desirable
- This undesirable effect can be corrected by reheating steam

# 5.3.2.4 The ideal reheat Rankine cycle

As noted in the previous section, increasing the boiler pressure increases the thermal efficiency of the Rankine cycle, but the quality of the steam towards the final stages of expansion in the turbine is compromised. This shortcoming can be addressed as follows:



Figure 5.10 Reheating of steam in a Rankine cycle



Figure 5.11 T-s diagram showing reheating of steam in a Rankine cycle

1. Superheat the steam to very high temperatures before it enters the turbine. This will bring in the desirable effect of increasing the average temperature thus increasing the cycle thermal efficiency. However, this is not a viable option because of the metallurgical limitation in withstanding very high temperatures.

2. Perform a two-stage expansion in the turbine with the deployment of reheat in between the stages. This is a modified version of a simple ideal Rankine cycle with the incorporation of a reheat process. Figures 5.10 and 5.11 show a Rankine cycle with two-stage expansion and the reheating, and the corresponding T-s diagram, respectively. Reheating improves the quality of the steam (lower moisture content) at the final stage of the expansion in the turbine. In the first stage, the high-pressure steam is expanded in the high pressure turbine isentropically to an intermediate pressure. The steam after the first stage of expansion is returned to the boiler where it is reheated at constant pressure, typically to the inlet temperature of the first turbine stage. The reheated steam is then expanded isentropically in the low-pressure turbine down to the condenser pressure.

Referring to the T-s diagram for the reheat cycle shown in Figure 5.11, the total heat input and the total turbine work output for a reheat cycle are expressed as follows:

$$q_{in,total} = (h_4 - h_1) + (h_6 - h_5)$$
(5.19)

$$W_{turb,out} = (h_4 - h_5) + (h_6 - h_7)$$
(5.20)

The incorporation of the single reheat generally improves the cycle thermal efficiency by about 4 to 5 percent resulting from the increase in the average temperature at which heat is added in the boiler. It is to be noted that the average temperature during the reheat process can be increased by increasing the number of expansion and reheat stages. As the number of stages is increased, the expansion and reheat processes approach to the most efficient process in thermodynamics, namely an isothermal process at the maximum temperature.

In general, the use of more than two reheat stages is not technically and economically feasible. The theoretical improvement in efficiency from the second reheat is about half of that which results from a single reheat. If the turbine inlet pressure is not high enough, double reheat would result in superheated exhaust. This is undesirable as it

would cause the average temperature for heat rejection to increase and thus the cycle efficiency to decrease. The reheat temperatures are very close or equal to the turbine inlet temperature. The optimum reheat pressure is about one-fourth of the maximum cycle pressure. For example, the optimum reheat pressure for a cycle with a boiler pressure of 160-bar is about 40-bar. The reheat cycle, which is essentially deployed to reduce the moisture content at the latter stage of the expansion, could be avoided if there are suitable materials available to withstand very high temperature resulting from superheating.

The effects of two-stage expansion of the steam with reheating is summarised below:

- Dryness fraction approaches to 1.0
- Thermal efficiency of cycle increases
- Represents a practical solution to low dryness fraction problem
- Used frequently in modern turbines

## 5.4 Regeneration

The average temperature at which the heat is added to the steam in the boiler is affected by the temperature at which the feedwater enters the boiler. If this temperature can also be increased by deploying certain practical techniques, the thermal efficiency of the Rankine cycle could be further improved. One such technique is called regeneration. The regeneration process is realised by extracting or "bleeding" steam from turbine at various points. Extracted steam is used to heat the feedwater using a heat exchanger called feedwater heater or regenerator. Feedwater heaters can be of two types - open feedwater heater and closed feedwater heater. In both types, heat exchange takes place between the bled steam and the feedwater in either a direct or indirect manner.

In an open feedwater heater, the heat is transferred from the steam to the feedwater by direct mixing of two fluid streams. Whereas in a closed feedwater heater, the heat is transferred from the steam to the feedwater without mixing the two fluid streams.

Regeneration also provides a convenient means of deaerating feedwater to prevent corrosion in the boiler. Deaerating is the removal of dissolved air from the feedwater resulting from air leaks into the condenser. Regeneration also helps to control the large volume flow rate of steam at the final stage of turbine expansion resulting in reduction of turbine size.

## **Open Feedwater Heater**

An open feedwater heater (OFWH) is a direct-contact feedwater heater, basically a mixing chamber where the steam extracted from the turbine mixes with the feedwater exiting the pump. The steam is extracted from the turbine such that the mixture leaves the heater as a saturated liquid at the heater pressure. Figures 5.12 and 5.13 show the schematic of a steam power plant with one open feedwater heater and the T-s diagram of the feedwater cycle, respectively.



Figure 5.12 Open feedwater in a Rankine cycle



Figure 5.13 T-s diagram for an open feedwater heater in a Rankine cycle

As shown in Figure 5.12, in an ideal regenerative Rankine cycle, steam enters the turbine at the boiler pressure at state point 2 and expands isentropically to an intermediate pressure of state point 3. After this intermediate expansion, some steam is extracted at this state point 3 and routed to the feedwater heater. The remaining steam continues to expand isentropically to the condenser pressure at state point 7. This steam leaves the condenser as a saturated liquid at the condenser pressure at state point 1. The condensed water (the feedwater) enters an isentropic pump, where it is pumped to the feedwater heater pressure of state point 2. The pressurised feedwater is routed to the feedwater heater, where it mixes with the steam extracted from the turbine. The fraction of the steam is extracted in such a manner that the mixture leaves the heater as a saturated liquid at the heater pressure of state point 3. A second pump increases the pressure of the feedwater to the boiler pressure of state point 4. The feedwater is then heated to the turbine inlet temperature at state point 2, hence, completing the cycle.

Referring to the T-s diagram of Figure 5.13, the open feedwater heater cycle is illustrated as follows:

For each 1 kg of steam leaving the boiler, y kg expands partially in the turbine and is extracted at state point 3. The remaining (1-y) kg expands completely to the condenser pressure. Therefore, the mass flow rates are different in different components. For instance, if the mass flow rate through the boiler is  $m \cdot$ , for example, it is (1-y)m· through the condenser. This aspect of the regenerative Rankine cycle should be considered in the analysis of the cycle as well as in the interpretation of the areas on the T-s diagram.

Referring to the T-s diagram of the open feedwater heater, the expression for the heat and work expressions in the cycle are expressed as follows:

$$q_{in} = h_2 - h_1 \tag{5.21}$$

$$q_{out} = (1 - y)(h_4 - h_5) \tag{5.22}$$

$$w_{turb,out} = (h_2 - h_3) + (1 - y)(h_3 - h_4)$$
(5.23)

 $w_{pump,in} = (1 - y)w_{pump5-6} + w_{pump7-1}$ (5.24) where,  $y = \dot{m}_3/\dot{m}_2$ , the fraction of steam extracted from the turbine

$$w_{pump5-6} = v_5(P_6 - P_5) \tag{5.25}$$

$$w_{pump7-1} = v_7(P_7 - P_1) \tag{5.26}$$

Referring to the Figure 5.13, for an isentropic process (entropy, S = constant), the entropies can be related as follows:

$$S_2 = S_3 = S_4 = S_{f@P5} + xS_{fg@P5}$$
(5.27)

As seen from the preceding discussion, the thermal efficiency of the Rankine cycle increases through the increase in the average temperature at which the heat is added through the use of OFWH. It is to be noted that the thermal efficiency can be further improved by increasing the number of OFWHs in the cycle. It is common in many large plants in operation today to use as many as eight open feedwater heaters. However, the optimum number of feedwater heaters needs to be determined by carrying out a cost benefit analysis in terms of economic considerations.

#### **Closed Feedwater Heater**

The closed feedwater heater is another type of feedwater heater frequently used in steam power plants. Heat is transferred from the extracted steam to the feedwater without any mixing taking place. Hence, the two streams now can be at different pressures as they do not mix. Figures 5.14 and 5.15 show a schematic of a closed feedwater heater incorporating a Rankine cycle and the T-s diagram for such a cycle, respectively. In an ideal CFWH, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure. In actual power plants, the feedwater leaves the heater below the exit temperature of the extracted steam because a temperature difference of at least a few degrees is required for any effective heat transfer to take place. The condensed steam is then either pumped to the feedwater line or routed to another heater or to the condenser through a device called a trap.



Figure 5.14 Closed feedwater heater in a Rankine cycle



Figure 5.15 T-s diagram showing closed feedwater heater in a Rankine cycle

The function of a trap is to trap the vapour while allowing liquid to be throttled to a lower pressure region. The process across the trap is an isenthalpic process, i.e. the inlet and outlet enthalpies remain the same.

The comparison between open and closed feedwater heaters can be summarised as follows:

- Open feedwater heaters (OFWH) are simple in construction and inexpensive and have good direct heat transfer characteristics
- OFWHs can bring the feedwater to the saturation state
- For each OFWH, a pump is required to handle the feedwater
- CFWHs are more complex because of the internal tubing network. As a result they are more expensive
- Heat transfer in a CFWH is less effective because it is indirect
- However, a CFWH does not require a separate pump for each heater since the extracted steam and the feedwater can be at different pressures
- Most steam power plants use a combination of open and closed feedwater heaters

# 5.5 Illustrative Examples

# Example 5.5.1

A steady-flow Carnot cycle uses water as the working fluid. Water changes from saturated liquid to saturated vapour as heat is transferred to it from a source at 300°C. Heat rejection takes place at a pressure of 30 kPa. Show the cycle on a T-s diagram relative to the saturation lines, and determine

(a) the thermal efficiency, (b) the amount of heat rejected, and (c) the net work output.

# Solution





(a)  $\eta_{thermal,C} = 1 - T_L / T_H = 1 - 342.1/573 = 0.403 = 40.3\%$ 

(b) The heat supplied during this cycle is the enthalpy of vapourisation

 $q_{in} = h_{fg@300C} = 2336 \text{ kJ/kg}$ 

Therefore, heat rejected,  $q_L = (T_L / T_H) x q_{in} = (342.1 / 573) x 2,336 = 1,394.7 kJ/kg$ 

(b) The net work output,  $w_{net} = \eta_{thermal,C} \times q_{in} = 0.403 \times 2,336 = 941.4 \text{ kJ/kg}$ 

#### Example 5.5.2

Consider a steady-flow Carnot cycle with water as the working fluid. The maximum and minimum temperatures in the cycle are 370.6°C and 70.6°C. The quality of water is 0.881 at the beginning of the heat-rejection process and 0.1 at the end. Show the cycle on a T-s diagram relative to the saturation lines, and determine (a) the thermal efficiency, (b) the pressure at the turbine inlet, and (c) the net work output.

Solution



Given:  $T_H = 370.6^{\circ}C = 643.6 \text{ K}$ ,  $T_L = 70.6^{\circ}C = 343.6 \text{ K}$ (a)  $\eta_{thermal,C} = 1 - T_L / T_H = 1 - 343.6/643.6 = 0.466 = 46.6\%$ (b) Also,  $s_2 = s_3 = s_f + x_3s_{fg}$ Therefore,  $s_2 = s_3 = 0.962 + 0.881 \times 6.783 = 6.937 \text{ kJ/kg K}$ Using  $s_2 = 6.937 \text{ kJ/kg K}$  and  $T = 370.6^{\circ}C$  from the steam table,  $P_2 = 24$ -bar. (c) The network,  $w_{net}$  is the enclosed area in the T-S diagram.  $s_4 = s_f + x_4s_{fg} = 0.962 + 0.1 \times 6.783 = 1.64 \text{ kJ/kg K}$  $w_{net} = (s_3 - s_4) (T_H - T_L) = (6.937 - 1.64) (643.6 - 343.6) = 1,589 \text{ kJ/kg}$ 

## Example 5.5.3

A steam power plant operates on a simple ideal Rankine cycle between the pressure limits of 40-bar and 0.06-bar. The temperature of the steam at the turbine inlet is 350°C, and the mass flow rate of steam through the cycle is 40 kg/s. Show the cycle on a T-s diagram with respect to saturation lines and determine (a) the thermal efficiency of the cycle and (b) the net power output of the power plant.

Solution:



Referring to the T-s diagram and using the given pressures and temperature, from the steam table:

$$\begin{split} h_4 &= 3094 \text{ kJ/kg}, \ s_4 &= 6.584 \text{ kJ/kg K} \\ Also, s_4 &= s_5 = s_{f@0.06bar} + x_5 s_{fg@0.06bar}, \ 6.584 = 0.521 + (x_5 \text{ x } 7.808) \\ Therefore, x_5 &= 0.777 \\ h_5 &= h_{f@0.06bar} + x_5 h_{fg@0.06bar}, \ h_5 &= 152 + (0.777 \text{ x } 2,415) = 2,028.5 \text{ kJ/k} \\ \Pi_{thermal, Rankine} &= w_{net}/Q_{in} = (w_{turb} - w_{pump}) / Q_{in} \end{split}$$

$$\begin{split} & w_{turb} = h_4 - h_5 = 3,094 - 2,028.5 = 1,065.5 \text{ kJ/kg} \\ & w_{pump} = v_{f@0.06bar} x \ (P_2 - P_5) \ x \ 100 = 0.001 \ x \ (40 - 0.06) \ x \ 100 = 3.99 \ \text{kJ/kg} \\ & \text{Also, } w_{pump} = h_1 - h_6 \ \text{, therefore, } h_1 = w_{pump} + h_6 = 3.99 + 152 = 156 \ \text{kJ/kg} \\ & Q_{in} = h_4 - h_1 = 3,094 - 156 = 2,938 \ \text{kJ/kg} \\ & (a) \ \Pi_{thermal, \ Rankine} = \ (w_{turb} - w_{pump}) \ / \ Q_{in} = (1,065.5 - 3.99)/2,938 = 0.361 = 36.1\% \\ & (b) \ \text{Net power from the turbine} = \ \dot{m}_{steam} \ x \ (w_{turb} - w_{pump}) \\ & = 40 \ x \ (1,065.5 - 3.99) = 42,460 \ \text{kW} = 42.46 \ \text{MW} \end{split}$$

#### Example 5.5.4

A simple ideal Rankine cycle which uses water as the working fluid operates its condenser at 45°C and its boiler at 350°C. Calculate the work produced by the turbine, the heat supplied in the boiler, and the thermal efficiency of this cycle when the steam enters the turbine without any superheating.

Solution



Referring to the T-s diagram and using the given pressures and temperature, from the steam table:

$$\begin{split} h_2 &= 2,565 \text{ kJ/kg}, \ s_2 &= 5.214 \text{ kJ/kg K} \\ \text{Also, } s_2 &= s_3 = s_{f@45C} + x_3 s_{fg@45C}, 5.214 = 0.521 + (x_5 \times 7.808) \\ \text{Therefore, } x_5 &= 0.777 = 77.7\% \\ h_5 &= h_{f@0.06bar} + x_5 h_{fg@0.06bar}, h_5 = 152 + (0.777 \times 2415) = 2,028.5 \text{ kJ/k} \\ \Pi_{\text{thermal, Rankine}} &= w_{\text{net}} / \text{Q}_{\text{in}} = (w_{\text{turb}} - w_{\text{pump}}) / \text{Q}_{\text{in}} \\ w_{\text{turb}} &= h_4 - h_5 = 3,094 - 2,028.5 = 1,065.5 \text{ kJ/kg} \\ w_{\text{pump}} &= v_{f@0.06bar} \times (P_2 - P_5) \times 100 = 0.001 \times (40 - 0.06) \times 100 = 3.99 \text{ kJ/kg} \\ \text{Also, } w_{\text{pump}} &= h_1 - h_6, \text{ therefore, } h_1 = w_{\text{pump}} + h_6 = 3.99 + 152 = 156 \text{ kJ/kg} \end{split}$$

 $\begin{aligned} Q_{in} &= h_4 - h_1 = 3,094 - 156 = 2,938 \text{ kJ/kg} \\ (a) &\prod_{\text{thermal, Rankine}} = (w_{\text{turb}} - w_{\text{pump}}) / Q_{in} = (1,065.5 - 3.99)/2,938 = 0.361 = 36.1\% \\ (b) &\text{Net power from the turbine} = \dot{m}_{\text{steam}} \times (w_{\text{turb}} - w_{\text{pump}}) \\ &= 40 \times (1,065.5 - 3.99) = 42,460 \text{ kW} = 42.46 \text{ MW} \end{aligned}$ 

#### Example 5.5.5

Consider a 250 MW steam power plant that operates on a simple ideal Rankine cycle. Steam enters the turbine at 10 MPa and 500°C and is cooled in the condenser at a pressure of 10 kPa. Show the cycle on a T-s diagram with respect to saturation lines, and determine (a) the quality of the steam at the turbine exit, (b) the thermal efficiency of the cycle, and (c) the mass flow rate of the steam.

Solution



Referring to the T-s diagram and using the given pressures and temperature, from the steam table:

(a) For 10 MPa = 100-bar and 500°C  $h_4 = 3,373 \text{ kJ/kg}, s_4 = 6.596 \text{ kJ/kg K}$ Also,  $s_4 = s_5 = s_{f@0.1bar} + x_5 s_{fg@0.1bar}, 6.596 = 0.649 + (x_5 x 7.5)$ Therefore,  $x_5 = 0.793 = 79.3\%$ 

(b)  $h_5 = h_{f@0.1bar} + x_5 h_{fg@0.1bar}$ ,  $h_5 = 192 + (0.793 \times 2392) = 2,088.9 \text{ kJ/k}$   $\Pi_{thermal, Rankine} = w_{net} / Q_{in} = (w_{turb} - w_{pump}) / Q_{in}$   $w_{turb} = h_4 - h_5 = 3,373 - 2,088.9 = 1,284.1 \text{ kJ/kg}$  $w_{pump} = v_{f@0.1bar} \times (P_1 - P_6) \times 100 = 0.001 \times (100 - 0.1) \times 100 = 9.99 \text{ kJ/kg}$  Also,  $w_{pump} = h_1 - h_6$ , therefore,  $h_1 = w_{pump} + h_6 = 9.99 + 192 = 202 \text{ kJ/kg}$  $Q_{in} = h_4 - h_1 = 3373 - 202 = 3171 \text{ kJ/kg}$  $\Pi_{thermal, Rankine} = (w_{turb} - w_{pump}) / Q_{in} = (1,284.1 - 9.99)/3,171 = 0.401 = 40.1\%$ 

(c) Given the net power from the turbine = 250 MW That is, 250 x 1,000 =  $\dot{m}_{steam}$  x (1284.1 – 9.99) Therefore,  $\dot{m}_{steam}$  = 196.2 kg/s

#### Example 5.5.6

A steam Rankine cycle operates between the pressure limits of 100-bar in the boiler and 20 kPa in the condenser. The turbine inlet temperature is 450°C. The turbine isentropic efficiency is 90 percent, the pump losses are negligible, and the cycle is sized to produce 3 MW of power. Calculate the mass flow rate through the boiler, the power produced by the turbine, the rate of heat supply in the boiler, and the thermal efficiency

Solution



Referring to the T-s diagram and using the given pressures and temperature, from the steam table:

(a) For 100-bar and 450°C  $h_4 = 3,241 \text{ kJ/kg}, s_4 = 6.484 \text{ kJ/kg K}$ Also,  $s_4 = s_{4s} = s_{f@0.2bar} + x_{4s} s_{fg@0.2bar}, 6.484 = 0.832 + (x_{4s} \times 7.5)$ Therefore,  $x_5 = 0.754 = 75.4\%$   $h_{4s} = h_{f@0.2bar} + x_{4s} h_{fg@0.2bar}, h_{4s} = 251 + (0.754 \times 2,358) = 2,028.9 \text{ kJ/k}$   $\Pi_{s,turb} = (w_{a,turb} / w_{s,turb}) = (h_4 - h_{4a}) / (h_4 - h_{4s})$ That is,  $0.9 = (3,241 - h_{4a}) / (3,241 - 2,028.9)$ Therefore,  $h_{4a} = 2,150.1 \text{ kJ/kg}$  
$$\begin{split} w_{pump} &= v_{f@0.2bar} x \ (P_5 - P_1) \ x \ 100 = 0.001 \ x \ (100 - 0.2) \ x \ 100 = 9.98 \ kJ/kg \\ w_{net} &= w_{turb} - w_{pump} = (3,241 - 2150.1) - 9.98 = 1,080.92 \ kJ/kg \\ P_{net} &= \dot{m}_{steam} \ x \ w_{net}; \ that \ is: \ 3 \ x \ 1,000 = \dot{m}_{steam} \ x \ 1,080.92 \\ Therefore, \ \dot{m}_{steam} = 2.78 \ kg/s \\ Power \ from \ the \ turbine, \ P_{turb} &= \dot{m}_{steam} \ x \ (h_4 - h_{4a}) = 2.78 \ x \ (3,241 - 2,150.1) = 3,032 \\ kW \\ Heat \ supplied \ in \ the \ boiler, \ Q_{in} = (h_4 - h_1) \\ w_{pump} &= h_1 - h_6 \ , \ therefore, \ h_1 = w_{pump} + h_6 = 9.98 + 251 = 261 \ kJ/kg \\ Therefore, \ P_{in} &= \dot{m}_{steam} \ x \ (h_4 - h_1) = 2.78 \ x \ (3241 - 261) = 8,284.4 \ kW \\ \eta_{thermal, \ Rankine} &= P_{net}/ \ P_{in} = 3,000/8,284.4 = 0.362 = 36.2\% \end{split}$$

#### Example 5.5.7

An ideal reheat Rankine cycle with water as the working fluid operates the boiler at 17,000 kPa, the reheater at 3,000 kPa, and the condenser at 90 kPa. The temperature is 500°C at the entrance of the high-pressure and low-pressure turbines. The mass flow rate through the cycle is 1.89 kg/s. Determine the power used by pumps, the power produced by the cycle, the rate of heat transfer in the reheater, and the thermal efficiency of this system

Solution



Referring to the T-s diagram and using the given pressures and temperature, from the steam table:

(a) For 170-bar and 500°C  $h_4 = 3,281 \text{ kJ/kg}, s_4 = 6.26 \text{ kJ/kg K}$   $s_{g@30bar} = 6.186 \text{ kJ/kg K}$ Also,  $s_4 = s_5 = 6.26 \text{ kJ/kg K}$  Since  $s_4 = 6.26 > s_{g@30bar}$ , the steam is still superheated after the first stage of expansion.

For 30-bar and 6.26 kJ/kg K, from the superheated steam property table,

$$\begin{split} h_5 &= 2,842.5 \text{ kJ/kg} \\ \text{For 30-bar and 500°C} \\ h_6 &= 3,456 \text{ kJ/kg}, \ s_6 &= 7.233 \text{ kJ/kg K} \\ \text{Also, } s_6 &= s_7 = s_{f@0.9bar} + x_7 s_{fg@0.9bar}, \ 7.233 = 1.27 + (x_{4s} \times 6.124) \\ \text{Therefore, } x_7 &= 0.973 = 97.3\% \\ h_7 &= h_{f@0.9bar} + x_7 h_{fg@0.9bar} = 405 + (0.973 \times 2,266) = 2,609.8 \text{ kJ/k} \\ \text{P}_{pump} &= \dot{m}_{fw} \times w_{pump} \text{ (Assumed } \dot{m}_{fw} = \dot{m}_{steam}) \\ w_{pump} &= v_{f@0.9bar} \times (P_1 - P_8) \times 100 = 0.001 \times (170 - 0.9) \times 100 = 16.91 \text{ kJ/kg} \\ \text{Therefore, } P_{pump} = 1.89 \times 16.91 = 31.96 \text{ kW} \end{split}$$

The net power produced by the cycle:

$$\begin{split} P_{turb} &= \dot{m}_{steam} \; x \; w_{net} \\ w_{net} &= w_{turb} - w_{pump} = (h_4 - h_5) + (h_6 - h_7) - w_{pump} \\ &= (3,281 - 2,842.5) + (3,456 - 2,609.8) - 16.91 = 1,267.8 \; kJ/kg \\ Therefore, \; P_{turb,net} &= \dot{m}_{steam} \; x \; w_{net} = 1.89 \; x \; 1267.8 = 2,396 \; kW \end{split}$$

The total heat input rate in the cycle,  $P_{in} = \dot{m}_{steam} x [(h_4 - h_1) + (h_6 - h_5)]$   $w_{pump} = h_1 - h_8$ , therefore,  $h_1 = w_{pump} + h_8 = 16.91 + 405 = 421.91 \text{ kJ/kg}$   $P_{in} = 1.89 x [(3,281 - 421.91) + (3,456 - 2,842.5)] = 6,563.2 \text{ kW}$  $\eta_{thermal, Rankine} = P_{net} / P_{in} = 2,396/6,563.2 = 0.365 = 36.5\%$ 

## Example 5.5.8

A steam power plant operates on an ideal regenerative Rankine cycle. Steam enters the turbine at 6 MPa and 450°C and is condensed in the condenser at 20 kPa. Steam is extracted from the turbine at 0.4 MPa to heat the feedwater in an open feedwater heater. Water leaves the feedwater heater as a saturated liquid. Show the cycle on a T-s diagram and determine (a) the net work output per kilogram of steam flowing through the boiler and (b) the thermal efficiency of the cycle.

#### Solution



Referring to the T-s diagram and using the given pressures and temperature, from the steam table:

(a) For 90-bar and 600°C

 $h_2 = 3,633 \text{ kJ/kg}, s_2 = s_3 = s_4 = 6.958 \text{ kJ/kg K}$ 

s<sub>g@4bar</sub> = 6.897 kJ/kg K

Since  $s_3 = 6.958 > s_{g@4bar}$ , the steam is still superheated after the first stage of expansion.

For 4-bar and 6.958 kJ/kg K, from the superheated steam property table,

$$\begin{split} h_3 &= 2,766 \text{ kJ/kg} \\ \text{Also, } s_3 &= s_4 = s_{f@0.2bar} + x_4 \, s_{fg@0.2bar}, \, 6.958 = 0.832 + (x_4 \times 7.075) \\ \text{Therefore, } x_4 &= 0.865 = 86.5\% \\ h_4 &= h_{f@0.2bar} + x_4 \, h_{fg@0.2bar} = 251 + (0.865 \times 2,358) = 2,290.7 \text{ kJ/k} \\ \text{Pump work input, } w_{5-6} &= v_{f@0.2bar} \times (P_6 - P_5) \times 100 = 0.001 \times (4 - 0.2) \times 100 \\ &= 0.38 \text{ kJ/kg} \\ \text{Pump work input, } w_{7-1} &= v_{f@4bar} \times (P_1 - P_7) \times 100 = 0.001 \times (90 - 4) \times 100 \\ &= 8.6 \text{ kJ/kg} \\ w_{net} &= w_{turb} - w_{pump} = (h_2 - h_3) + (h_3 - h_4) - (w_{5-6} + w_{7-1}) \\ &= (3,633 - 2,766) + (2,766 - 2,290.7) - (0.38 + 8.6) = 1,333.3 \text{ kJ/kg} \end{split}$$

The total heat input rate in the cycle,  $Q_{in} = h_2 - h_1$   $W_{7-1} = h_1 - h_7$ , therefore,  $h_1 = w_{7-1} + h_7$   $h_7 = h_{f@4bar} = 605 \text{ kJ/kg}$ Therefore,  $h_1 = 8.6 + 605 = 613.6 \text{ kJ/kg}$   $Q_{in} = 3,633 - 613.6 = 3,019.4 \text{ kJ/kg}$  $\eta_{thermal, Rankine} = w_{net}/Q_{in} = 1,333.3/3,019.4 = 0.442 = 44.2\%$ 

#### Summary

The vapour power cycle, also known as the Rankine cycle, which is widely used in combined heat and power cycles has been presented. The working fluid in the Rankine cycle is steam. Both the ideal and actual Rankine cycles have been presented and analysed thermodynamically. Various thermal efficiency improvement techniques have also been discussed with their analysis. A number of illustrative examples are included at the end of the chapter to give practising industrial professionals a good feel of the analysis of the ideal and actual Rankine cycles.

#### References

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# 6.0 CHP PRIME MOVERS

This chapter provides detailed information on various prime movers used in CHP Systems. The selection of prime movers in a CHP system is of paramount importance as far as the optimum operation of the system is concerned. One can select a prime mover with a better electrical conversion efficiency or low electrical conversion efficiency for the application. The decision on such a selection is arrived at by analysing the facility's thermal load vs electrical load requirements. This chapter covers the different types of prime movers available in the market along with their suitability for industrial applications for meeting thermal and electrical load requirements.

# Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. Different types of prime movers
- 2. Selection of prime movers
- 3. Advantages and disadvantages of different types of prime movers
- 4. Applications of CHP systems involving the prime movers

# 6.1. Introduction

As discussed in the preceding sections of this reference manual, co-generation or trigeneration CHP systems, produce both electrical power and useful heat from the same fuel source. In any CHP system, one group burns the fuel to produce heat in order to produce rotary power for electricity, where heat is a by-product. Some CHP systems derive electricity directly from the chemical process of oxidation of fuel, and again heat is a by-product. In the most common group of CHP systems, fuel is directly burned in a prime mover process. Examples of such a CHP group include internal combustion engines and combustion turbines. In each case, the combustion follows the expansion of the combustion gases, resulting in outputs of rotary power to drive an electrical generator. With internal combustion, the hot gases expand inside a cylinder with a piston and the expansion drives the piston. These prime movers are called internal combustion engines, e.g. automotive engines, and are closely related to vehicle engines.

A second group of fuel-fired CHP systems burns the fuel in a boiler to produce high pressure and often superheated steam. The steam produced is delivered to a steam turbine. Expansion of steam through the turbine blades produces rotary power which drives an electrical generator. The steam turbine process is often used in large commercial electricity generation power plants.

When the CHP system is used with steam driving the turbine, the heat energy rejected during condensation can be used for space heating or cooling of buildings or various industrial processes. For this to work, the discharged steam from the turbine must be piped to a low-pressure steam heat use, which is needed at the same time as the steam turbine operation. This necessitates a balance between a thermal load recovered from the turbine exhaust and the power output of the steam turbine. Boilers with supplementary firing and thermal storage systems are common in industry to help balance load requirements.

It is important to ensure that temperature and pressure of the exhaust steam matches the load requirements. It is worthwhile to note that the lower the temperature and pressure needed by the facility, the more energy is available to the steam turbine. Note that in comparison to gas turbines, boilers and steam-driven turbines have more flexibility in the fuel which drives the process. Steam necessary to drive a steam turbine can be produced by any fuel which can be burned in a boiler or waste heat derived from a process. In a combined cycle power plant, waste heat from a gas-fired combustion turbine produces steam that drives a steam turbine or is injected back into the combustion turbine for additional power. A common application today is to install a backpressure steam turbine in an existing steam boiler and steam distribution system. Boilers are sometimes operated at higher pressure than is required to deliver the steam, and with a backpressure steam turbine, steam expands through the turbine to the lower pressure needed to deliver the steam to the loads served, producing power as a by-product.

Another type of CHP set-up is using a fuel cell, which does not need mechanical energy to produce electricity. A chemical reaction occurs within the cells through the combining of hydrogen and oxygen. The products of that chemical reaction are electrical energy, water vapour, and heat. The source of the hydrogen could be natural gas or town gas depending upon the availability of such sources. The natural gas or town gas undergoes a fuel refining process producing the hydrogen gas. In CHP plants, the waste heat is used to meet the heating energy needs of buildings or processes. The temperature of the waste heat depends on the specific fuel cell process. While the temperature in some fuel cell processes is quite low (e.g. 65°C to 85°C for Polymer Electrolyte Membrane Fuel Cells, PEMFC), in other processes, the

waste heat temperature is high enough (e.g. 75°C to 1000°C for Solid Oxide Fuel Cells, SOFC) to produce steam for a combined process.

In this Chapter, the different types of prime movers used in CHP systems are discussed. The power produced by the prime mover is typically used to generate electricity. The power generated can also be used as mechanical power to drive pumps, chillers, and compressors in an industrial set-up. As noted, heat may be produced directly in the prime mover, and/or heat can be recovered from the prime mover exhaust stream using an appropriate heat recovery method. Heat recovery is a critical component of CHP systems which essentially reduces the fuel input energy wastage. Subsequently, the recovered heat is used to produce additional power, hot water, steam, chilled water, and/or desiccant dehumidification.

Generally, CHP prime movers can be classified into two types, namely: fuel-to-power equipment (FPE) and thermal-to-power equipment (TPE). The FPE prime mover equipment is fired with premium gaseous fuels such as natural gas, methane from wastewater plants or landfills, or liquid fuels such as light oils, biofuels. It is to be noted that fuel cells are a fuel-to-energy process. Hence fuel cells do not produce rotary power and are not considered as prime movers.

In Singapore and the region, natural gas (NG) is the preferred fuel as it is readily available via a piped distribution system from Indonesia and Malaysia. Furthermore, with the commissioning of the Liquified Natural Gas (LNG) terminals in Jurong Island, an NG based industrial system's operational reliability is unmatched. Natural gas is cleaner than fuel oil, coal, wood, or agricultural waste, etc.

NG has a smaller carbon footprint than most other fuels. NG, therefore, may not have the environmental problems associated with other such fuels.

Typical prime movers used in CHP systems include the following:

- Internal combustion (IC) reciprocating engine generators
- Spark ignition engines
- Diesel cycle engines
- Combustion turbine generators (CTGs)
- Microturbines

Thermal-to-power prime mover equipment includes processes where heat is developed by a source outside the prime mover. This includes both boiler-produced steam and waste heat derived from another process. It also includes processes where waste heat is generated by one of the primary prime movers discussed above.

The selection of prime mover type should be based on the following:

- Available heat recovery rate
- Electrical capacity
- Efficiency
- Size range

Typical thermal-to-power prime mover equipment include the following:

- Steam turbines
- Steam-driven reciprocating engines
- Stirling engines (external combustion engines)
- Organic Rankine cycles

A typical CHP system requires many other additional components and/or systems to constitute a complete CHP plant. However, the actual requirements may vary depending on the CHP system itself. Experience suggests that the following components/systems are common to many CHP applications in the industry:

- Fuel supply system(s)
- Gas compressors
- Combustion air
- Turbine inlet cooling
- Exhaust systems
- Exhaust heat recovery systems
- Lube oil systems
- Lube oil heat recovery or cooling system
- Engine jacket cooling water
- Water treatment systems
- Cooling towers for heat rejection
- Starting system (Battery or Compressed air)
- Black start generator/backup power system

- Integration technology/system for integrating voltage and phase with local grid supplies
- Control system for the plant and engine

A waste heat recovery system can be used to recover the waste heat available from prime movers and converting it to useful outputs such as power, process heat loads through the generation of hot water or steam or building cooling with the help of absorption chillers. It is to be noted that in Singapore and the region, the most common application of CHP systems is to provide steam, hot water, and/or chilled water for process heating and cooling. CHP systems producing two and three useful outputs simultaneously from the same fuel source are called co-generation and trigeneration systems, respectively and will be discussed in detail in Chapter 7.

A comparison of the prime movers commonly used in CHP systems is given in Table 6.1. This comparison will be helpful for facility managers, engineers and operators alike to understand equipment considerations and how prime movers integrate into their complete CHP system. This in turn will help them to be more effective and productive in their day-to-day management of their CHP systems and thereby increasing the CHP system's operational efficiency.

Prime Movers	Diesel	NG	Combustion	Steam	Micro	Fuel
	Engine	Engine	Turbine	Turbine	turbine	Cells
Size (MW)	0.05 - >10	0.05 - 7	1 - 200	Any size	0.025 -	0.02 -
					0.25	0.4
Electrical	30 - 50	25 - 40	25 – 40 (simple)	30 - 42	20 - 30	40 – <
Efficiency (%)			40 – 60 (combined)			50
Heat Rate	7,000 –	9,700 –	8,500 – 13,600	8,100 –	11,300 –	7,000 –
(Btu/kWh)	11,300	13,600		11,300	17,000	8,500
Waste Heat	1,000 –	1,000 -	3,400 – 12,000	NA	400 - 650	140 -
Recoverable	5,000	5,000				700
(Btu/kWh)						
Waste Heat	80 - 480	260 - 540	260 - 590	NA	200 - 340	60 -
Temperature						370
(°C)						
Typical Uses	Hot water,	Hot	Heat, hot water, LP-	LP	Heat, hot	Hot
of Waste Heat	LP steam,	water, LP	HP steam, district	steam,	water, LP	water,
Recovered	district	steam,	heating/cooling	district	steam	HP-LP
	heating/co	district		heating		steam
	oling	heating/		, i i i i i i i i i i i i i i i i i i i		
	5	coolina				
		5				
Fuels Used	Diesel and	NG,	NG, biogas,	All	NG,	H2,
	residual oil	biogas,	propane, distillate		biogas,	NG,
		propane	oil		propane,	propan
					distillate	е
					oil	
Fuel Pressure	<0.3	0.1 – 3.2	8 - 35	NA	2.5 – 7.0	0.03 –
(bar)						3.2
Typical	90 - 95	92 -97	90 - 98	Close to	90 - 98	>95
Availability (%)				100		
Time between	25,000 -	24,000 -	30,000 - 50,000	>50,000	5,000 to	10,000
Overhauls	30,000	60,000			40,000	_
(hours)						40,000
Start-up time	10 s	10 s	10 min – 1 h	1 h – 1	60 s	3 h – 1
				day		day
Noise	M - H	M - H	M	M - H	Μ	L

Legend: M – Moderate, H – High, L - Low

Table 6.1 Different types of prime movers and their characteristics

# 6.2 Fuel-to-Power Equipment

As discussed, most fuel-to-power equipment burn fuel in a combustion process that converts the chemical energy of the fuel into rotational kinetic energy, which can be transmitted through a shaft to produce electrical power in an electrical generator (fuel cells are an exception to this). Fuel-to-power prime mover equipment are usually connected to an electrical generator. While the great majority of CHP systems use mechanical energy to drive a generator to produce electricity, other alternatives involving chemical reactions are currently being studied and implemented. Several types of fuel-to-power equipment exist today that can effectively generate power. When a prime mover and generator is a factory assembled package, the package is generally known as a "gen-set."

It is worth noting that the basic criterion for a CHP system is that it produces both thermal and electrical energy from a single fuel source. In this respect, CHP differs from the typical electrical power generation plant today. The other factor in the economic feasibility of a CHP system is an attractive utilisation factor (U.F) i.e.the almost complete use of the input energy value with the help of an effective heat recovery system. The recovered heat is generally used for process heating to preheating applications depending on the quality of the recovered heat.

There are different choices and methods available when developing a CHP system. One basic configuration of a CHP system is the use of an internal combustion (IC) reciprocating engine or combustion turbine generator (CTG) with heat recovery equipment capturing exhaust heat and heat from the IC engine cooling jacket, etc. The IC engine type of system is most often used where the electrical loads are 2 to 3 MW or less, and where the need for thermal heat is low in comparison to the need for electrical power (for a relatively low heat to power ratio application). Since IC engine prime movers tend to be more fuel efficient than combustion turbines, there is proportionally more electrical power and less heat energy derived per unit fuel energy input.

IC engines are available in the market with sizes ranging from less than 50 kW to more than 15 MW. The CTGs' capacities range from approximately 1 MW to over 100 MW. The CTGs require high-pressure gas supply or gas compressors to provide the necessary gas pressure. Typically, the CTGs have higher exhaust temperatures and higher volume exhaust. Hence, CTGs are used for applications where high-temperature recovered heat is required, such as high-pressure steam, or low-

pressure high-temperature oil. CHP systems using CTG as the prime mover also have higher heat-to-power ratios than IC engine-based CHP systems. Hence, applications requiring high year-round, 24 hours-a-day heat loads are better candidates for CTG based CHP systems.

On the other hand, internal combustion engine-based systems are most suited for applications which require lower recovered waste heat temperatures, such as hot water or low-pressure steam of less than about 1 bar. It is worth noting that IC engines can be used with lower quality fuels, e.g. methane gas from a wastewater treatment plant or a garbage landfill. Generally, IC engine-based CHP systems require less specialised maintenance, training, and auxiliary equipment than CTG-based CHP systems.

# 6.2.1 IC Reciprocating Engines

## Engine Types

IC reciprocating engines are machines that translate the linear movement of pistons into the rotational movement of a crankshaft through a combustion process. Fuel subjected to combustion process, heats and expands the fuel-air mixture inside a cylinder which drives the piston. Most engines available in the market today are multi-cylinder for smoother power delivery. Figure 6.1 below shows a photograph of a reciprocating engine prime mover. The two basic types of IC reciprocating engines are spark ignition (SI) and compression ignition (CI) engines.



Figure 6.1 Photograph of a reciprocating engine prime mover (Courtesy of MWM Engines)

Both the engines (SI and CI) are available to operate in either a 4 or 3 stroke combustion cycle. The 4 strokes of the 4-stroke cycle are: intake, compression, power, and exhaust. Engines can be either naturally aspirated or turbocharged engines. In naturally aspirated engines, air and fuel are mixed in a carburetor and the intake stroke draws in the fuel-air mixture. Whereas a turbocharged engine has a compressor which compresses air and discharges that air into the combustion chamber during the intake stroke. To eliminate carburetors or external mixing of fuel and air, in most cases, only the air is injected by the turbocharger and fuel is directly injected into the combustion chamber. In either case, a turbocharged engine can deliver more power because there is a greater density of fuel and air in the process.

The 2-stroke cycle differs from the 4-stroke cycle in that it combines the power and intake strokes into one stroke while the exhaust and compression strokes are combined into a second common stroke. Start-up time for reciprocating engines can be fast, around 10 seconds for diesel-fueled engines. Generally, warm-up times take significantly longer and depend upon the mass of the system. In cold climates (not applicable in Singapore), the warm-up time can be reduced if the system includes a crankcase heater to keep the engine warm. Therefore, it is recommended to consider a crankcase heater as mandatory for a CHP system serving as an emergency power system in an industrial facility in cold climates.

As the name suggests, in spark ignition engines, a spark is added at the end of the compression stroke that ignites the fuel-air mixture to start the power stroke. Compression ignition engines differ from spark ignition engines, in that there is no spark added to the air-fuel mixture to start combustion. Instead, the intake air is compressed by the piston's motion reducing the volume of the cylinder. A CI engine uses a very high compression ratio which heats the air in the chamber to a point high enough to ignite the fuel. At the top of the compression stroke fuel is injected into the hot compressed air in the combustion chamber causing a spontaneous ignition. The heat of combustion develops very high pressure which drives the piston in an expansion stroke. When a compression engine is first started a glow plug is heated by an electric source. Once the engine is running and hot, the glow plug is no longer needed.

The most common fuel used in CI engine is diesel fuel (No. 2 oil). CI engines can also be fueled by a wide range of petroleum products (up to No. 6 oil). CI engines, also referred to as Diesel cycle engines, can also be fired with gaseous fuel in combination with liquid fuel, called pilot oil, used as the ignition agent in dual fuel engines. Note that the fuel in a compression cycle engine needs to have a fairly high flash point to prevent ignition until full compression is achieved at the top of the piston stroke.

## 6.2.2 Turbo- or Supercharger Power Boosters

Both SI and CI engines can be outfitted with turbo or superchargers to increase power output, leading to improved efficiency. As noted above, a turbocharger is a relatively small compressor that is mounted on a common shaft with a small turbine. As hot exhaust gas enters the turbine and expands, the turbine spins, spinning the compressor impeller at the same time. Engine intake air is routed through the compressor to pre-compress the combustion air, creating a denser air charge entering the cylinder. A supercharger is another type of compressor that also works to precompress the air. It is to be noted that a supercharger does not rely on exhaust gas to drive the compressor. Instead, superchargers are belt or gear-driven from the engine's crankshaft, using a small amount of engine output power to yield a greater overall power output. Nowadays, turbochargers are the most commonly utilised precompression tools in CHP systems. A turbo-charged engine arrangement is shown in Figure 6.2 below.



Figure 6.2 A turbo-charged engine arrangement

# 6.2.3 Size Ranges

IC engine generators are available in a wide range of sizes to meet many industrial applications. Both natural gas and diesel fuel fired engines can have electrical outputs ranging from 50 kW to 15 MW.

# 6.2.4 Useable Exhaust Temperatures/Useable Heat

As discussed, waste heat in the form of hot water or sometimes in the form of lowpressure steam can be recovered from reciprocating engine jacket manifolds, aftercoolers, lubrication systems, and engine exhaust. In terms of temperature, the highest potential is from exhaust gases followed by the engine jacket. The lowest temperature potential is heat recovery from lube oil cooling systems. Note that lube oil cooling systems are particularly temperature sensitive due to oil breakdown at high temperature.

The total amount of available waste heat from an engine is the total amount of fuel energy input less the energy value of the rotary power produced. It is to be noted that all the waste heat produced cannot be usefully recovered. As an example, engine heat loss by radiation and convection to the space is not possible to recover. Also, the heat carried away by the water vapour resulting from the combustion of fuel cannot be recovered. Condensation in the exhaust is avoided because of potential corrosion of the exhaust system due to the formation of carbonic acid ( $H_2CO_3$ ).

The latent heat in the water vapour is a significant part of the heat of combustion, around 10% in the case of natural gas, and depends on the type of fuel burned. For example, fuel oil derives more energy from carbon, which creates less water vapour than does natural gas. The Lower Calorific Value (LCV) is the heating value of the fuel less the latent heat of vapourisation for water in the fuel. The latent heat of vapourisation of the water vapour during combustion is a function of the fuel type and its chemistry, and most processes do not recover the water vapour energy. Hence, most engine manufacturers rate their engines based on LCV. On the other hand, the Higher Calorific Value (HCV) is the heating value of the fuel including latent heat of vapourisation of water. While engine performance may be rated based on the LCV of fuel, fuel purchases are typically based on the HCV. Therefore, owners, operators and engineers must take these differences into consideration in their engineering calculations.

The amount of waste heat which can be recovered from the IC engine depends on the type of engine, the temperature at which the heat recovery occurs, and on the type and capacity of the heat recovery equipment. In general, a turbocharged engine has more of its waste heat in the exhaust gases than a naturally aspirated engine. The higher the temperature at which beneficial heat recovery must occur, the less energy that can be recovered.

The typical distribution of input fuel energy for a reciprocating engine operating at rated load is tabulated as follows:



Figure 6.3 Sankey diagram for reciprocating engine energy distribution

The latent heat of vapourisation for the water vapour created by combustion of hydrogen is lost in the exhaust gases unless the gases are cooled to a point where the water vapour condenses. Condensing systems can be highly efficient and improve CHP sustainability, but, as noted, the exhaust system must be designed for the corrosive condensate (e.g., constructed of stainless steel). Most of the heat in jacket water and lube oil cooling can be recovered and used. Typical heat recovery practices can recover some 60 to 80% or higher of the heat in the exhaust gases depending on various factors including the thermal output temperature, with the highest efficiency achieved when the exhaust gases are cooled to near ambient temperatures. With respect to the fuel distribution percentages discussed above, it should be noted that the percentages vary with manufacturer and model as well as with engine load.

As noted, the quantity and quality of the heat that can be collected from reciprocating engines per kilowatt of power produced (heat to power ratio) is lower than what can be obtained from CTGs which have higher thermal-electric ratios. Firstly, less of the fuel energy is converted to shaft horsepower in a typical CTG versus a typical IC engine meaning more waste heat is available with a CTG. Secondly, with a CTG, nearly all the waste heat is in the exhaust air stream versus an IC engine. Thirdly, the exhaust gas temperatures are much higher with a CTG than most of the waste heat from a reciprocating engine. Most engine jacket cooling systems operate at around 90°C and offer a good opportunity to recover heat in the form of hot water. For many applications, the exhaust heat can be recovered into the coolant loop using an exhaust-to-liquid heat exchanger to provide a single form of heat recovery. As noted above, a few internal combustion engines permit coolant to reach around 120°C above atmospheric pressure and then allow the coolant to flash into low-pressure steam of about 1 bar after leaving the engine jacket in an ebullient cooling system.

## 6.2.5 Heat Rate and Electrical Efficiency

The heat rate is defined as the amount of input energy required by the prime mover to produce 1 unit of power. CHP heat rate for natural gas fueled SI engines can range between about 10,000 to 14,000 Btu/kWh, while the heat rate for CI engines can be as low as 7,000 Btu/kWh. It is to be noted that a lower heat rate means that less energy is required per kWh produced.

Typically, SI engines between 100 and 900 kW have observed electric efficiencies between about 25 and 30% based on the HCV. Larger SI engines, above 4,000 kW electrical output, have efficiencies of around 36%, based on the HCV. Note that given the electrical efficiency, the heat rate can be easily calculated (or vice versa) as heat rate is equal to the inverse of electric efficiency multiplied by 3,413 (the number of Btu per kWh).

## **Cooling Water Requirements**

The heat developed by reciprocating engines of either type (SI or CI engines) must be rejected to prevent overheating the engine parts which may lead to premature engine failure. A coolant loop is used to absorb this heat from the various engine components to ensure that all engine components remain functional. Cooling water with glycol is used as the coolant in many CHP system applications to absorb the generated heat and to transfer it to other useful applications through heat recovery principles.

A coolant loop required for engine components include the following:

- The engine jacket, or block
- Turbochargers
- Aftercoolers
- Lube oil coolers
- Exhaust heat recovery devices

The coolant loop can be the "jacket water system" which transfers heat to beneficial heat load applications and/or to heat rejection to keep the engine cool.

# Noise and Vibration

Due to the reciprocating motion of the internal components, IC engines tend to produce significant vibration and noise. The noise includes low-frequency rumble denoted by a continual, loud, thumping sound emanating from the engine block. This noise can be a problem and is typically addressed by sound attenuation inside an enclosure. The attenuation could be either incorporating the attenuation techniques in the CHP plant building housing or through a special dedicated enclosure, or both. Additional noise can occur from improperly positioned or designed air intake and discharge systems and/or inadequately muffled exhaust systems. Positioning exhaust discharges and air louvres and vents away from places where noise will cause problems can reduce some of the observed noise.

It is worth noting that not all the engine noise can be mitigated through these techniques. Installing reciprocating engines in sound-insulated engine rooms is one good way to help mitigate unwanted sound. Engines are usually mounted on vibration isolators to greatly reduce vibration transmission from the engine to surrounding areas. Where vibration is likely to cause problems, an isolated inertia base is often used in addition to vibration isolation on the engine. While a heat recovery heat exchanger will help reduce translated engine noise, an exhaust muffler may also be required.

# 6.2.6 Combustion Turbines

Combustion turbines and combustion turbine generators (CTGs) are gaining popularity in CHP installations nowadays, where power requirements are consistent throughout the day and there is consistent use for the relatively larger quantity of highgrade thermal energy throughout the year. Although the thermodynamics of combustion for turbines and reciprocating engines is similar, the mechanical process is vastly different. In a multistage combustion turbine, a multistage air compressor is mounted on a common shaft to a multistage turbine. Outside air is ducted to the compressor, where the pressure and temperature are increased before being delivered to the combustor. In the combustor, the hot, compressed air is mixed with fuel and ignited resulting in high-pressure, high temperature gas. The high temperature, high pressure gas is subsequently expanded in the turbine to provide shaft power. The shaft power is used to drive the generator and to drive the compressor with a certain percentage of back power ratio (ratio of compressor power to turbine power). Figure 6.4 below shows a photograph of a combustion turbine prime mover.



Figure 6.4 Photograph of a combustion turbine prime mover

Turbine capacity and efficiency is strongly dependent on the temperature of the air entering the compressor, and, therefore, many combustion turbine systems precool the air entering the CTG compressor. Precooling the combustion turbine inlet air provides more air flow and greater compressor efficiency. Greater turbine inlet air flow can produce more CTG power output resulting in greater efficiency. Some turbine inlet air cooling systems use evaporative cooling, since air density is related to dry bulb temperature. Other turbine inlet air cooling systems use chilled water from absorption chillers driven by recovered steam from the steam turbine generator exhaust. Still others cool the inlet air using chilled water from ice storage with the ice produced by electric power at night when other facility electrical loads are lower. The advantage of ice is that it can produce lower inlet air temperatures than can be produced with chilled water from absorption chillers.

# 6.2.6.1 Types and Sizes

Combustion turbines (CTs) are either single-shaft or two-shaft designs and are classified as either aero-derivative or industrial type. Aero-derivative combustion turbines are available from many manufacturers in electrical capacities ranging from about 1 MW up to about 15 MW. The CTs simple cycle efficiency can be up to 40% based upon LCV with recuperated turbine and no heat recovery. As previously discussed, the fuel-to-electrical efficiency can be increased by the use of waste heat steam to produce more power (combined cycle). Steam generated in a waste heat recovery steam generator (HRSG) located at the turbine discharge can produce more electrical power in a steam turbine generator (STG) or it can be injected into the CTG combustor after the compressor to increase the flow of gases through the turbine. This will also help to cool the gases to reduce NOx, and increase the CTG power produced.

Industrial combustion turbines are built for stationary electrical generation and are available in much higher capacities (up to around 500 MW). Industrial combustion turbines are heavier than their aero-derivative counterparts and are generally less efficient. Industrial combustion turbines have maximum simple cycle efficiencies of approximately 36% based on HCV. As discussed above, the heat rate for combustion turbines increases with increased inlet air temperature while, at the same time, power output capacity falls linearly. Experience suggests that a 5°C increase in air temperature approximately equates to about a 5% decrease in power output. Combustion turbine inlet cooling, as discussed, can be effective in maintaining consistent power output even at higher outside air temperatures.

The amount of inlet pressure loss and combustion turbine backpressure also affects the performance of the combustion turbine generator, and the CTG inlet and outlet pressure drops need to be kept within the turbine manufacturer's allowable limits. An approximate 0.5% decrease in power output can be expected for each inch of water column increase in air inlet pressure drop, therefore the design of the combustion turbine air inlet system is critical to successful, sustainable CHP operations.

# 6.2.6.2 Heat Rate and Electric Efficiency

Combustion turbines' average fuel-to-electrical shaft efficiencies generally range from about 25 to 40% based on the HCV. It is to be noted that larger CTGs are more efficient than smaller ones. Heat rates vary from manufacturer to manufacturer and model to model, and in general range from about 8,500 to almost 14,000 Btu/kWh. The balance of the fuel energy input is discharged in the exhaust and a minor amount

through convection and radiation or internal coolants in large turbines. A minimum stack exhaust temperature of approximately 150°C is typically required to prevent condensation (unless the exhaust system is specifically designed for exhaust gas condensation). Due to potential corrosion of exhaust systems, exhaust temperatures should not drop below 150°C in an exhaust system without design for condensation.

#### 6.2.6.3 Useable Exhaust Temperatures/Useable Heat

Combustion turbines typically run very hot with combustor exhaust gases sometimes exceeding 1,200°C. At the turbine exit, exhaust temperatures are reduced to temperatures between 450 and 600°C, due to the expansion of the hot gas through the turbine(s). The exhaust temperatures coupled with high exhaust flow rates lead to opportunities for heat recovery and duct firing, which are not feasible with reciprocating engines. In CHP plants, where combustion turbine electric generation efficiency is of utmost importance, regenerators or recuperators can be employed in the exhaust air stream to preheat the compressed air that enters the combustor thereby leading to higher electrical efficiency resulting in slightly decreased fuel consumption.

The combustion turbine exhaust contains a large percentage of excess air. Therefore, afterburners/duct burners may be installed in the exhaust to create a supplementary firing system providing additional steam. Installation of duct burners can be very efficient, reaching an estimated maximum efficiency that exceeds 90%.

## 6.2.6.4 Cooling Water Requirements

Combustion turbines do not have the same cooling requirements as IC reciprocating engines. Turbines do not have a crankcase or reciprocating parts that require cooling and the only typical requirement for internal cooling is for the oil that lubricates the compressor/ turbine shaft bearings and possibly the electric generator. As discussed above, cooling could often be utilised to pre-cool the intake air stream to improve the efficiency of the system. For example spraying water in the suction path is practised.

As a rule of thumb, the power output of a turbine decreases by approximately 1% for every 1°C rise in intake air temperature. As ambient air to the compressor intake is never the same temperature throughout the year. Combustion turbine inlet cooling (CTIC) systems lower or always maintain the desirable low intake temperature to ensure the stable power output. While indirect/direct evaporative cooling is the most
common CTIC system type, chilled water coils and direct expansion (DX) refrigerant coils can be used to provide even greater benefit in situations with high outside air temperatures especially those with high humidity.

Some of the advantages of using CTIC are:

- Increased power output capacity,
- Lower heat rate,
- Extended turbine life,
- Improved system efficiency



Figure 6.5 Effect of turbine inlet air cooling on output power

# Example:

If the turbine inlet cooling is improved from 42°C to 32 °C, estimate the improvement in the power output and heat rate for the turbine.

Solution:



From the above diagram :

The % improvement in relative output power ratio = 0.06% The % improvement in relative heat ratio = 0.025%

# 6.2.6.5 Noise/Vibration

Combustion turbines exhibit noise differently from reciprocating engines and generate high-frequency noise and vibrations. An operating combustion turbine can be loud and uncomfortable to the casual observer. Manufacturers often attenuate this noise by enclosing their turbine generators in sound-insulated enclosures. Locating the CTG in an enclosure reduces sound levels considerably but will not eliminate/attenuate the sound completely. Additional sound attenuation equipment is typically installed on turbine generators to further lower noise to acceptable levels.



Figure 6.6 Gas turbine sound attenuation enclosure

Sound attenuation equipment includes inlet air silencers and exhaust silencers. However, if the CTG exhaust flows through a HRSG and exhaust stack, the HRSG may attenuate the exhaust noise sufficiently without an exhaust silencer. Vibration levels in combustion turbines are generally low as the rotational nature of the assembly does not include reciprocating parts. It is to be noted that vibration is important as the machines operate at high speeds (around 10,000rpm) making the system easy to trip due to smallest imbalance in rotor due to compressor or turbine fouling.

### 6.2.7 Microturbines

Microturbines are very small combustion turbines, which feature an internal heat recovery heat exchanger called a recuperator. In a microturbine, the inlet air is compressed in a radial compressor and then preheated in the recuperator using heat from the turbine exhaust. Heated air from the recuperator is mixed with fuel and subject to ignition in the combustor chamber and the resulting hot combustion gas is then expanded in the expansion and power turbines. The expansion turbine drives both the compressor and generator connected to a common single-shaft. Whereas, two-shaft turbine designs use the turbine's exhaust to power a second turbine (called the power turbine) that drives the generator. The power turbine exhaust is then utilised in the recuperator to preheat the air from the compressor. Microturbines can be designed to operate on a number of fuels, including natural gas, propane, landfill gas, digester gas, sour gases, and liquid fuels such as biodiesel, gasoline, kerosene, and diesel fuel/heating oil, for example. It is to be noted that operating fuel pressures for microturbines may require built in fuel compressors that are offered as options by most manufacturers these days.

Microturbines are ideally suited for distributed generation applications due to their flexibility in connection methods, ability to provide stable and extremely reliable power, and low emissions.

Types of microturbine applications include the following:

- Peak shaving and base load power (grid parallel)
- CHP
- Stand-alone power
- Backup/standby power
- Primary power with grid as backup

Figure 6.7 below shows a photograph of a micro-turbine prime mover.



Figure 6.7 Photograph of a micro-turbine prime mover (Courtesy of Capstone Microturbine)

In CHP applications, the waste heat from the microturbine is used to produce hot water to heat building(s), to drive absorption cooling, desiccant dehumidification

equipment, and to supply other thermal energy needs in a building or industrial process.

#### 6.2.7.1 Commonly Available Microturbine Sizes

Microturbines are presently available with electrical outputs varying from about 30 to 300 kW. While this range of electrical outputs is relatively low compared to other prime mover technologies, the smaller footprint of the microturbine makes it ideal for installing them in parallel, creating large banks of microturbine generator sets to create larger power production arrays. This concept offers some benefits over a single, larger CTG. One of the advantages of a microturbine array is that if one machine is out of operation, the facility will not lose its entire electrical generation. A microturbine array can also maintain good efficiency throughout a variable electrical demand by controlling the number of microturbines in operation based on the load situation (better part-load control). This will make the operating machines in the array operate at full load efficiency. However, one disadvantage of microturbine is their construction cost which is more expensive than a single large prime mover of equal capacity.

### 6.2.7.2 Efficiencies and Heat Rate for Microturbines

Microturbines exhibit shaft efficiencies of between 20 and 30%, based upon the HCV of fuel, which corresponds to a heat rate of between 11,300 and 17,000 Btu/kWh. As microturbines reduce power output by reducing mass flow and combustion temperature, efficiency at part-load can be below that of full-load efficiency. Thermal output ranges from 200 to 350°C which is suitable for supplying heat for a variety of facilities' heat load applications.

### 6.2.8 Fuel Cells

Engine or gas turbine-based CHP systems rely on the combustion of fuel to produce high-pressure, high temperature gas that can expand to provide useful work as previously described. The expanded gases are utilised by the specific equipment and provide mechanical and thermal energy. In fuel cells, the oxidation process occurs across membranes which cause electron transfer. Note that fuel cells directly create electric power without a prime mover or generator. Traditionally, the power generation process in a fuel cell is considered as a pure chemical reaction rather than a combustion process eventhough most typical fuel cell-based systems have a fuel and an oxidiser and so the process is technically a combustion process. The working principle of a fuel cell is depicted in Figure 6.8 and that a commercial fuel cell based CHP system is shown in Figure 6.9.



Figure 6.8 Working principle of a hydrogen fuel cell



Figure 6.9 A commercial 1 kW fuel cell CHP system

Some of the advantages of fuel cells include the following:

- Fuel cells are practically emission free of undesirable exhaust gases
- Highly efficient
- Operate at very low noise levels
- Able to adjust to changes in electrical loads

The most common fuel cells (phosphoric acid process) reject heat (the chemical reaction byproduct) in the 60 to 90°C range and are about 40 to 55% efficient in electric generation. Other processes have different efficiencies, different temperature heat discharge and different cost per watt. For example, molten carbon is about 55% efficient and discharges heat at temperatures high enough (300 to 350°C) to produce high-pressure steam for the facilities' requirements.

## 6.2.8.1 Types of fuel cells

Fuel cells differ from simple batteries in that they use a continuous supply of fuel for the chemical reaction or can operate for extended periods of time provided there is a continuous uninterruptable supply of fuel. Although many variations exist, the most common type of fuel cell uses hydrogen as the fuel source and the oxygen in air to complete the chemical reaction. The source of the hydrogen is typically natural gas (however, pure hydrogen, propane, and diesel fuel can also be used). The by-product of the chemical reaction from the fuel cell is hot water, which can be used for domestic purposes. The use of smaller capacity fuel cells of about 1 to 2 kW is common in Japanese households.

As hydrogen (the fuel) enters the fuel cell and is mixed with air (containing oxygen), the fuel is oxidised as shown in Figure 6.5, broken down into protons and electrons. In the proton exchange membrane fuel cell (PEMFC) and phosphoric acid fuel cell (PAFC), positively charged ions move through the electrolyte across a voltage to produce electric power after which the protons and electrons are recombined with oxygen in the air to make hot water. As this water is removed from the fuel cell, more protons are pulled through the electrolyte, resulting in further power production.

### 6.2.8.2 Cost and Availability

Despite the fact that fuel cells are excellent candidates for CHP, the main disadvantage lies in the very high capital cost per installed kilowatt in comparison to other available CHP prime mover options. This cost disadvantage together with concerns about the exotic materials and developing technologies used in fuel cells, have limited their widespread application. For example, a fuel cell manufacturer in the United States of America produces a 200-kW unit that sells for approximately US\$ 1 million (US\$5,000/ kW). This is approximately 3 to 4 times the cost of an equivalent IC engine or combustion turbine generator system. Ongoing research and development suggest that larger fuel cells (1,000 kW) are also in development and are expected to sell for U.S. \$1,500 to \$2,000 per kilowatt sometime in the future. This

pricing is closer to what might be considered feasible for installation in a cost-effective CHP plant.

## 6.2.8.3 Efficiencies and Heat Rate

Fuel cell electric generation efficiencies range from 40 to more than 50% with hydrogen supplied fuel cells (versus hydrocarbon supplied), which corresponds to a heat rate of less than 7,000 to about 8,500 Btu/kWh.

## 6.3 Heat-to-Power Equipment

Heat-to-power generating equipment in CHP systems utilises heat produced by some other process to generate electricity or rotary power. The most common heat-to-power generating equipment is a steam turbine generator which is driven by either steam produced in a boiler or steam recovered from the waste heat of the fuel-fired prime mover(s) discussed above. When waste heat from a prime mover produces steam for use in a steam turbine, the waste heat produces additional power. The heat energy could also be used to generate hot water, steam, or chilled water that would have otherwise been produced using one of the conventional methods.

## 6.3.1 Steam Turbines

A steam turbine is a mechanical device that converts the heat content, enthalpy, of steam into rotational mechanical power. The rotational power can drive pumps, centrifugal chiller compressors, and other mechanical devices. Steam turbines are often used to drive an electrical power generator.

A steam turbine generator can make use of the thermal energy produced in a heat recovery steam generator (HRSG) to generate additional power. With a conventional boiler system, to qualify as combined heat and power, boiler produced steam must be used for both heating (and/or thermally driven cooling) and power. Sometimes, this means steam is produced at temperatures and pressures greater than needed for the facility's heating or cooling needs/applications, and the steam is expanded through a STG to the pressure needed for use by the facility. This type of CHP system produces power in direct relation to the thermal load. As a retrofit project where there is an existing steam boiler and steam distribution system, such a system is often very cost-effective to install. The backpressure turbine used is very efficient because all the steam exiting the turbine is beneficially used.

## 6.3.1.1 Types of Steam Turbines

Steam turbines are available in two types: axial-flow turbines and radial-flow turbines. Axial-flow steam turbines are those in which high-pressure steam is introduced into the turbine inlet at one end of the turbine and steam flows along the turbine's axis of rotation driving finned (bladed) wheels, or stages, that spin much like a windmill spins under the influence of the wind. Axial-flow steam turbines are sub-categorised as follows:

- Non-condensing / backpressure turbines
- Total condensing turbines
- Automatic extraction turbines
- Non-automatic extraction turbines
- Induction/mixed-pressure turbines
- Induction-extraction turbines



Figure 6.10 Different types of turbines (a) extraction turbine (b) total condensing turbine (c) extraction cum total condensing turbine

Axial-flow turbines are also defined by the type of stages and blades. The blades can either be impulse or reaction. Impulse blades are fixed to the turbine wheel and undergo rotation from the force of the steam hitting the turbine blades, while reaction blades also undergo rotation due to the nozzle effect as the steam leaves the blades. Radial-flow steam turbines are dramatically different from their axial-flow counterparts. In a radial-flow steam turbine, high-pressure steam enters the turbine in the center of the turbine impeller and decompresses radially, perpendicular to the turbine's axis of rotation. This drop in steam pressure (and energy) provides the motive force that causes the rotation of the turbine and, thus, the rotation of the shaft driving any mechanical device or generator. Multi-stage radial-inflow steam turbines are factory prepackaged equipment that includes two or more impellers connected through reduction gearing with steam piping installed between stages to transport steam from one stage to the next. Condensate, if any, is removed between stages, since turbines (of all types) operating at high rpm can be severely damaged if subjected to trace water droplets.

A non-condensing backpressure steam turbine's exhaust is under pressure, and is therefore called a backpressure turbine. The backpressure can be at any pressure required by the low-pressure secondary steam system, so long as that pressure is lower than the turbine inlet pressure. The greater the pressure difference the more potential for generation of power. Backpressure steam turbines provide an energy efficient method to reduce steam pressures compared to using pressure-reducing valves which lose much of the steam energy. Most of the energy difference between the steam entering and leaving a backpressure turbine is converted to shaft power so the process is quite efficient.

For example, if a steam boiler can produce about 14-bar absolute steam and only 5bar is needed for distribution, a backpressure steam turbine can be used to generate power operating on the energy difference between 14 and 5-bar absolute steam. The power produced is a function of the steam pressure difference across the backpressure turbine and the steam flow.

Typically, the steam flow rate is related to the heat loads served and usually varies depending on the applications they are serving. Another application may serve part of a steam plant needs where different pressures are needed. For example, a hospital may need about 10-bar steam for sterilizers and 1-bar steam for domestic water and space heating, absorption cooling, and other processes. When the steam is produced

by waste heat recovery from a prime mover like a combustion turbine the application is similar. Steam is generated at pressures higher than needed for the thermal loads served. The steam pressure is then reduced through a backpressure turbine to the pressure needed to serve the thermal loads.

A total condensing steam turbine is a steam turbine that exhausts into a condenser where the exhausted steam is condensed. The condenser will be in a vacuum allowing much more enthalpy to be obtained from each pound of steam, making the steam turbine thermodynamic process much more efficient. Most condensers are water cooled but some condensers are air cooled. Condensing turbines are the usual choice in commercial electrical power plants since the only need is for electrical power. A condensing turbine provides more rotational power for the steam available but most of the energy is lost in condensing. With a backpressure turbine, the condensing occurs in serving the thermal loads and is therefore beneficial. The overall efficiency in serving both power and thermal needs is therefore much greater for a backpressure turbine. In any case, a power plant that does not serve a heat load is not a CHP system.

Extraction condensing turbines allow steam to be removed from the turbine at any reduced pressure, including multiple reduced pressures. So, for example, steam could enter the steam turbine generator at 13 barg and a portion of the steam could be extracted at 7 barg to feed the medium pressure steam system; a second extraction port could also bleed off steam at 1 barg for use in the low-pressure steam system, and the remaining steam would drive the turbine to produce useful work as it expands through the rest of the turbine. Since the steam bleed-off serves beneficial thermal needs, such a plant is a CHP system.

The existence of multiple steam turbine types offers mechanical engineers several options to consider when analysing and designing the most efficient CHP plants. Steam turbine exhaust, when reduced in both pressure and temperature, can be used to supply heat exchangers, absorption chillers, pumps, or other equipment that are designed to operate with steam and that are installed in place of electrically driven equipment.

### 6.3.1.2 Steam Turbine Capacity Ranges

Steam turbines are commonly available in practically any size with some units installed in power generation plants exceeding 100 MW. Since there is no combustion

process, steam turbines have no environmental impacts, unlike combustion turbines. Steam turbines are likely to be available to make use of CHP produced steam of any quantity, and combustion turbine exhaust duct firing can very efficiently increase the steam production in a combined CTG CHP system.

### 6.3.1.3 Isentropic Efficiency Range

Steam turbine thermodynamic efficiencies are directly related to the efficiency of the Carnot cycle; therefore, the temperature of the heat source and the temperature of the heat sink set the maximum possible theoretical efficiency. The higher the steam temperature and the colder the condenser water, the higher the theoretical thermodynamic efficiency. Due to irreversibility (entropy), real systems will be less efficient than that predicted by the theoretical Carnot cycle. The efficiency of the turbine design at converting the energy of the steam into shaft energy is also an important factor. Steam turbines of high-quality construction can have isentropic efficiencies as high as 90%. Note, the isentropic efficiency is the efficiency of the steam turbine to convert steam energy into shaft power and is not the same as overall thermodynamic cycle efficiency, which is much lower. To achieve high thermodynamic cycle efficiency, commercial power plants have boilers able to produce very highpressure steam (often about 70 barg or more) and superheat the steam. On the other end of the process, the condenser produces as much vacuum as possible, which is a function of the condensing temperature. Some very large plants draw cold ocean or deep lake water. Still most of the steam energy is in the phase change from vapour to liquid and is thus only partially available to a condensing steam turbine.

### 6.3.1.4 Noise/Vibration

Steam turbines, like combustion turbines, experience high-frequency noise and rotational vibrations. Noise from a steam turbine is generally around 90 dBA or less, which requires hearing protection when spending an extended time near the equipment. Turbines are also provided with acoustic housing. Facility team should also note that noise from steam flow in pipes and the operation of pumps may be of greater concern.

### 6.4 CHP Prime Mover Comparisons

Reciprocating engines, gas combustion turbines, fuel cells, and steam turbines all have various advantages and disadvantages when compared to one another. This section compares the characteristics of the various prime movers deployed in CHP plants.

### 6.4.1 Electrical Output and Electric Efficiency

The electrical output of the various prime mover technologies ranges from a few kilowatts (microturbine) to hundreds of megawatts (steam turbine). The electric efficiency (total electrical output divided by the total energy input) of CHP systems involving different technologies ranges from 20% (microturbine) to over 50% (fuel cells). The CHP systems using natural gas reciprocating engine have electrical efficiencies ranging from 25 to 45% and power output ranging from 50 kW to 5 MW. Whereas, the CTG based CHP system has efficiencies ranging from 25 to 40% with a simple cycle and 40 to 60% with a combined cycle based on the HCV of the fuel. CTG power output typically ranges from 3 to 200 MW. However, nowadays, some manufacturers are producing CTG units with power output ranging from 1 MW to 1,000 MW (summarised in Table 6.2)

### 6.4.2 Heat Recovery Potentials

Based on the prime movers used, different CHP systems have different heat recovery potentials. Some CHP systems may produce low temperature hot water (LTHW), less than 120°C, low-pressure steam (1 barg or less), or medium-pressure steam. Some heat recovery systems are a part of the equipment served. A good example is an exhaust gas-fired absorption chiller-boiler that directly intakes the exhaust from the prime mover and uses the hot exhaust gases directly to drive the absorption process and to produce hot water for other applications. Another occasional thermal use is direct heating or drying, which can be highly efficient as exhaust gas transfers its energy as it cools to ambient temperature.

In some industrial applications, the exhaust from a gas turbine is directed to a process such as drying agricultural products or wood. This application beneficially uses the waste heat without an intermediate recovery process. Such applications are also extremely cost-effective. LTHW is typically recovered from IC reciprocating engines, although low-pressure steam (less than 2 barg) can be obtained from high temperature engine exhaust. Medium-pressure steam (up to about 17 barg) is typically recovered from a HRSG that uses the CTG exhaust as a heat source. Fuel cells typically generate LTHW (about 80°C depending upon the fuel cell technology), which can be used for pre-heating or domestic hot water production.

CTG generally exhibits higher heat to power ratios and generate substantially more heat in comparison to IC engines. The useable temperature of the recovered heat varies. For example, some applications can use cooling tower water from a steam turbine plant to heat agricultural processes, fish farms, or air-drying using heat at 25 to 32°C. On the other extreme, a duct-fired combustion turbine may recover heat from the 600°C exhaust.

#### 6.4.3 Fuels and Fuel Pressures

As discussed, CHP systems can be designed to operate on different types of fuels including, but not limited to, natural gas, diesel fuel, landfill gas, digester gas, propane, wood or agricultural waste, etc. However, as far as CHP plants in Singapore are concerned, the most commonly utilised fuel is natural gas. Natural gas is widely available through local utilities and is rarely subject to interruption that will affect the operations of the facility. When considering the use of natural gas, or any other fuel, the engineer must ensure that the selected prime mover equipment is able to operate with that fuel source. Typically, CTGs and microtubines can be designed to operate on fuels including natural gas, biogas, and propane. It is to be noted that natural gas and diesel-based IC engines, although similar in mechanical function, are designed to operate on different fuels. Natural gas IC engines can be designed to operate on natural gas, biogas, and propane, while diesel IC engines can be designed to operate on diesel fuel, biodiesel, or residual oil. Fuel cells can be designed to operate on natural gas, pure hydrogen gas ( $H_2$ ) or propane. If a CHP plant installation includes equipment involving the use of dirty fuels such as digester gas, additional fueltreatment equipment are required. Such fuels require cleaning and drying before it can be used in the combustor of the engines.

The fuel pressure required to operate fuel-to-power prime mover equipment in CHP systems varies between 0.03 and 3 barg for fuel cells to between 8 and 35 barg for CTGs. Natural gas-based IC engines are designed to operate on low-pressure gas. Typically, CHP Prime movers that require high-pressure fuel, including CTGs and microturbines, require gas compression equipment to increase the pressure of the utility delivered fuel to the required pressure at the prime mover (summarised in Table 6.2).

### 6.4.4 Power Density

The area needed for a CHP plant is a major consideration especially in space-scarce countries like Singapore. By knowing the power density in kW/m<sup>2</sup>, engineers can estimate the CHP plant square footage that will be required for a calculated facility level electrical load. Generally, the footprint for CHP plant fuel-to-power prime mover equipment installations does not vary widely between the systems considered. Experience suggests that CTG and microturbine based CHP Plant energy footprints

are small, varying from about 0.25 to 0.7 ft<sup>2</sup>/kW and 0.2 to 1.6 ft<sup>2</sup>/kW, respectively. Generally, IC reciprocating engines take more room than CTG for the same power output. The energy footprints for natural gas and diesel engines vary from about 0.25 to 0.35 ft<sup>2</sup>/kW. Note that fuel cell-based CHP plants can have some of the largest energy footprints among all the CHP plant fuel-to-power equipment at up to 5 ft<sup>2</sup>/kW.

STG-based CHP plant energy footprints are noted to be extremely low, typically less than 0.15 ft<sup>2</sup>/kW. However, it is to be noted that STGs represent the thermal-to-power classification of CHP prime mover equipment, therefore, additional space must be catered for the steam producing equipment such as HRSG and other similar heat exchangers.

### 6.4.5 Online Availability and Time between Overhauls

Electric power or heat generation capability of a CHP system is largely determined by the availability of the installed CHP system prime mover equipment's ability to consistently operate to meet the facility electrical and heat loads. All the fuel-to-power prime mover equipment described above have online availability between 90 and 95%, operating on various fuels such as natural gas, biogas, and propane.

Availability is a key issue while considering a CHP plant. The availability of the CHP system is associated with the nature of the load at the facility level. Therefore, if the load served by the CHP system is critical in nature, there must be a backup system. If the backup draws power from the local grid, there are associated charges which need to be addressed in the economic feasibility study of the CHP system. Obviously, CHP plants with multiple prime movers will be impacted less by down time because of the availability of other prime mover(s) during the down time of one prime mover. The appropriate, diligent selection of equipment in conjunction with proper preventive maintenance largely eliminates down time. However, it is to be noted that no CHP system will ever achieve 100% availability.

Overhaul of CHP systems involves the scrutiny of major components of the prime mover equipment for the purposes of restoring the operations through reconstruction, etc. Generally, equipment manufacturers/vendors have backup/replacement equipment available at their factories during the overhaul of the system thereby limiting the CHP system's down time. The recommended best practice is that prime mover equipment be overhauled at regular intervals to ensure that the CHP plant operates consistently and with optimum performance in terms of efficiency. CTGs are typically operated between 35,000 to 55,000 hours between major overhauls. Whereas, the microturbine based CHP system operates between 4000 and 45,000 hours between overhauls. Feedback from industry suggests that IC engines could be operated between 20,000 and 60,000 hours between overhauls. Fuel cells are recommended to operate between 8,000 and 42,000 hours. Finally, the operating hours for STG based CHP systems can range from 40,000 to 50,000 hours between overhauls. It is worth noting that the operating hours between overhauls are also dependent on the steam quality in terms of cleanliness and quality of the steam used to operate the machines.

### 6.4.6 Start-Up Time

Start-up times of CHP plant prime mover equipment should be assessed when evaluating electric and heat load profiles and selecting prime mover equipment to serve the estimated electric and heat loads. Start-up times vary greatly between all different types of CHP prime movers. For example, typical start-up time for diesel engines, microturbines and CTGs are about 10s, 60s and 10m to several hours, respectively. Note that some prime mover equipment may take even longer to start up. E.g. from 3 hours to up to 2 days for fuel cells and from 1 hour to 1 day for some STGs. Generally, steam turbine generators start quickly, significant time may be required to bring boiler and steam distribution pipes to a proper operation point. It is to be noted that although IC engines have very fast start-up times, the time required to properly warm up the plant will be significantly longer (e.g. no crankcase heaters in use).

# 6.4.7 Noise

As CHP systems consist of prime movers, which are essentially rotating machinery, such as reciprocating engines, combustion turbines, and compressors, they generate high decibel noise and vibration. Such noise and vibrations not only affect facility equipment and personnel but also neighbouring facilities and residents. Excessive vibration without proper control can propagate, causing damage to equipment as well as adjacent equipment and structures. In some areas, more recently developed zoning laws specify permissible noise levels in terms of octave band frequency levels. The noise produced by the CHP system must be added to existing noise levels before the system is installed to calculate the post-CHP noise level. CHP systems can generate significant levels of noise from multiple components, including engines, engine exhaust, compressors, and cooling fans.

Noise generated by the prime movers can be reduced or attenuated using acoustic insulation, acoustic barriers, air attenuation baffles, and exhaust silencers. Vibration reduction will not only contribute to noise reduction but is also necessary between rotating machinery and connected structures, piping, ductwork, and equipment. Vibration reduction is normally accomplished by mounting the equipment on spring vibration isolators and using flexible couplings between the rotating machinery and other equipment, piping, and ductwork. In almost all cases, CHP equipment requires noise attenuation and vibration isolation. This should be considered in the beginning phase of design and may require input from an acoustic specialist in sensitive areas. such as locations close to residential developments, libraries, theatres, and healthcare facilities. Consideration also should be given to other sources of vibration and their potential to interfere with CHP equipment operation. Locations should also be checked for existing sources of vibration that might impact the operation of the CHP plant components, which typically comprise high-speed rotating machinery that are also sensitive to high levels of vibration. In some cases, the location of the CHP plant may need to change to mitigate noise and vibration issues if other methods cannot provide sufficient attenuation.

Noise level is expressed in terms of Decibels. Noise produced by the various technology prime movers ranges from low to a very high decibel level (up to 95 dB(A)). If the noise levels from the prime movers are beyond the regulated limit set by the authorities (NEA), noise containing enclosures may have to deployed at the source. Generally, IC reciprocating engines exhibit more low-frequency higher amplitude linear vibrations than CTGs and STGs, which tend to exhibit higher frequency noise and vibration. Microturbines share the same type of noise characteristics as CTGs, only to a lesser degree due to their relatively smaller sizes. However, it is to be noted that noise is increased when multiple units are arranged in the plant room and are in operation simultaneously. Among the CHP technologies, fuel cells produce the least noise. It is to be noted that the high-pitched noise of a steam turbine is far easier to attenuate than the rumble of an IC engine.

In general, the prime movers are typically selected based on the heat to power ratio required for a facility's application. The chart in Figure 6.8 shows different options for the selection of prime movers based on heat to power ratio.

The above comparison is summarised in Table 6.2 below.

Prime	Electrical	Heat Recovery	Fuels and	Power	Availa-	Start-	Noise
Movers	Output	Potentials	Fuel	density,	bility, %	up	
	and		Pressures	kW/m²		time	
	Electric						
	Efficiency						
Reciproc-	50 kW to 5	HW & LP steam	Natural gas,	30 to 43	90 to 95	10 s	Low
ating	MW; 25 to	generation (2	biogas and				frequen
engine	40%	bar)	propane,				су
			diesel				noise
Micro-	20 to 50%	HW & MP steam	Natural gas,	6 to 53	90 to 95	60 s	High
turbine		generation	biogas and				frequen
			propane				су
							noise
CTG	3to 200	HW & MP steam	Natural gas,	15 to 43	90 to 95	10 m	High
	MW; 40 to	generation (up	biogas and				frequen
	60%	to 17 bar)	propane				су
	(combined						noise
	cycle)						

Table 6.2 Comparison of different prime movers used in CHP



Figure 6.11 Chart for selection of prime movers

## 6.5. CHP Plant System Requirements

At facility level, a CHP plant support system will be determined by the type of CHP prime mover used and the type of heat production and the subsequent uses for various applications.

Accordingly, typical support systems required for a CGT based CHP system include the following:

- Combustion-air system: louvers, ducting, air filters, turbine inlet cooling, inlet air silencer
- High-pressure gas system such as air compressors
- Low-pressure gas system such as pressure reducing valves
- Turbine exhaust system such as ducting, duct burners, superheater, Heat Recovery Steam Generators, (HRSG), emission system
- control system including continuous emissions monitoring system (CEMS) as required
- Electric power generator distribution system such as substations, switchgear, motor
- Control centres, electrical and mechanical protection devices
- Lube oil system such as lube oil cooler, lube oil pump, day tanks
- Fire protection systems such as smoke detectors, sprinkler systems
- Chemical storage and emergency showers

For a CTG CHP system using a HRSG, the following typical support systems will also be required:

- Main steam system such as non-return valve, pressure reducing valves, steam traps to remove condensate
- Condensate system: condensate receiver(s), condensate pumps, de-aerating feed tank
- Feedwater system: feedwater pumps, feedwater control station (feedwater control valves)
- Emission control systems typically have support subsystems, such as ammonia/ urea storage and delivery.

For an IC reciprocating engine CHP system, the following typical support systems will be required:

• Combustion-air system: louvres, ducting, air filters, inlet air silencer

- Low-pressure gas system: pressure reducing valves
- Engine exhaust system: piping, heat exchangers for hot water generation through heat recovery, emission control system, CEMS
- Electric power generator distribution system: substations, switchgear, motor control centres, electrical and mechanical protection devices
- Jacket water system: jacket water pumps, expansion tank, radiator, heat exchangers
- Hot water supply and return system: hot water pumps, heat exchangers, coils, control valves
- Lube oil system: lube oil cooler, lube oil pump (if needed or not supplied on the engine), day tanks

If a chiller plant is to be part of the CHP plant, the following additional support systems will likely be required as well:

Chilled water supply and return system: chilled water pumps, distribution system,

coils, control valves, water treatment

- Condenser water supply and return system: cooling towers, condenser water pumps, water treatment
- Auxiliary systems can include: Compressed air Backup fuel oil storage Makeup water Treated water: deionised/reverse osmosis (DI/RO) Fire protection systems

The above CHP system plant requirements are depicted with the help of a flow chart in Figure 6.9.



Figure 6.12 Flow chart showing CHP plant system requirements

It is the responsibility of the CHP design engineer to understand and evaluate the different fuel-to-power prime movers available in the market versus what will best serve the facility. Each prime mover option has various heat load options and the type of prime mover, generated heat quality, and heat uses incorporated into the CHP plant will determine the plant systems required. It is recommended that the design engineer should be familiar with each system listed in the preceding sections.

# Summary

The selection of suitable prime mover is important for the optimum operation of a CHP system at an industrial facility level. In this Chapter different types of prime movers are presented with their characteristics described, such that a practising facility level engineer could make the prime mover selection decision based on their heat to power ratio at their respective facilities.

# Reference

- 1. ASHRAE Handbook Fundamentals, 2009
- 2. U.S. Department of Energy, 1999. Office of Energy Efficiency and Renewable Energy: Review of Combined Heat and Power Technologies, October 2006
- 3. <u>https://www.energy.gov/eere/amo/combined-heat-and-power (Last accessed on 16 May 2019)</u>

# 7.0 CO-GENERATION & TRI-GENERATION SYSTEMS

Growing concern over climate change is prompting more and more industrial players to move towards energy efficient operations of their plants. The industry trend is to minimise the losses in systems and processes. One way to do that is the recovery and reuse of the energy content in industrial waste streams. By recovering and reusing energy in the waste streams, the utilisation factor (efficiency) will tend towards 90% to 95% at its best. One of the systems involving such an energy recovery is called Combined Heat and Power (CHP) System. CHP systems can be either cogeneration or tri-generation systems. In a co-generation system, two useful outputs are produced simultaneously; e.g. electricity, cooling or heating, whereas in a trigeneration system, there are three useful outputs, namely electricity, cooling and heating. This chapter provides a detailed discussion of co-generation and trigeneration systems.

### Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. Ideal co-generation system
- 2. Practical co-generation system
- 3. Combined gas-vapour cycle co-generation system
- 4. Performance analysis of CHP systems

### 7.1. Ideal Co-generation Systems

In a normal thermodynamic cycle, a portion of the heat transferred to the working fluid is converted to work, which is high-grade energy. The remaining portion of the heat is rejected to rivers, lakes, oceans, or the atmosphere as waste heat because its quality (or grade) is too low to be of any practical use. This wastage, which is a penalty paid for the production of the high-grade energy, causes the cycle to operate in an inefficient manner. The requirement of the high-grade energy emanates from the engineering fact that electrical or mechanical work is the only form of energy on which many engineering devices such as a fan, pump, compressor, etc. can operate. However, many industrial systems or devices require energy input in the form of heat for their various processes. Industries that rely heavily on process heat include chemical, pulp and paper, oil production and refining, steel making, food processing, and textile industries. The steam is generated by burning a readily available fuel like coal, oil, natural gas, or another fuel in a furnace and transferring the resulting heat to the feedwater in a closed vessel, called a boiler. Consider the operation of a process-

heating plant without any heat losses due to convection and radiation. All the heat transferred to the steam in the boiler is used in the process-heating units, as shown in Figure 7.1. In this case, the process heating seems like a perfect operation with practically no waste of energy. However, if one considers the Second Law of Thermodynamics, which essentially deals with the quality of the energy, no such perfect thermodynamic process is possible.

This can be attributed to the fact that there is a degradation of energy when the high furnace temperature in the order of around 1,400°C to 1,500°C is transferred to water to produce steam at about 200°C or below, which is an irreversible process. Whenever there is an irreversibility associated with any process, the resulting undesirable entropy generation leads to a loss in exergy or work potential. For the same reason, it is recommended not to use high-quality energy to accomplish a task that could be accomplished with low-quality energy.

Many industries around the world that use large amounts of process heat also consume a large amount of electric power. Therefore, it makes economical as well as engineering sense to use the already-existing work potential to produce power instead of letting it go to waste. This would result in a plant that co-generates both the electricity and heat, known as a co-generation plant. In general, co-generation is the production of more than one useful form of energy such as electricity and heat from the same energy source. Either a steam-turbine cycle (Rankine cycle) or a gas turbine cycle (Brayton cycle) or even a combined cycle can be used as the power cycle in a co-generation plant.

Figure 7.1 shows the schematic of an ideal steam-turbine co-generation plant. Assume that the plant is to supply process heat Q<sub>process</sub> at a pressure of 5-bar at a rate of 100 kJ/s (kW). To meet this demand, at first the steam is to be expanded in a turbine to a pressure of 5-bar producing power at a rate of 50 kW. It is possible to adjust the steam flow rate such that steam leaves the process heating section as a saturated liquid at 5-bar. Now, the steam is pumped to the boiler pressure with the help of a pump and is heated in the boiler to the required temperature. The feedwater pump work is usually very small and can be neglected. Disregarding any heat losses, the rate of heat input in the boiler is determined from an energy balance to be 150 kW. Looking at the ideal co-generation cycle, the most striking feature of the ideal steam-turbine co-generation plant shown in Figure 7.1 is the absence of a condenser. Thus, no heat is rejected from this plant as waste heat. In other words, all the energy

transferred to the steam in the boiler is utilised as either process heat or electric power. Thus, the efficiency (Utilisation Factor, U.F.) of an ideal co-generation plant is a perfect 100%. Such a co-generation plant is not practical though.



Figure 7.1 Schematic of an ideal co-generation system

The Utilisation Factor, U.F. of a co-generation plant is defined as:

 $U.F = (\dot{W}_{net} + Q_{process})/Q_{in}$ Where,  $\dot{W}_{net} =$  Net turbine work  $Q_{process} =$  Process heat, and  $Q_{in} =$  Boiler heat input

Alternatively, the utility factor is defined in terms of just heat rejected in the condenser and heat absorbed in the boiler as follows:

 $U.F = 1 - Q_{out}/Q_{in}$  (7.2) Where,  $Q_{out}$  = Heat rejected in the condenser, and

Qin = Heat added in the boiler

Strictly speaking,  $Q_{out}$  also includes all the undesirable heat losses from the piping and other components, but they are usually small and are thus neglected. It also includes combustion inefficiencies such as incomplete combustion and stack losses when the utilisation factor is defined based on the heating value of the fuel.

As mentioned in the preceding section, the utilisation factor of the ideal steam-turbine co-generation plant is 100%. However, actual co-generation plants can have

utilisation factors as high as around 80%. It is worthwhile to note that modern cogeneration plants can have an even higher utilisation factor.

In the co-generation plant of Figure 7.1, in the absence of the turbine, the heat needs to be transferred in the boiler at a rate of only 100 kW instead of at 150 kW. The additional 50 kW of heat supplied is converted to work.

## 7.2 Practical Co-generation Systems

In any industry, the power and process requirements constantly vary with respect to changes in production. The ideal steam-turbine co-generation plant described above is not practical because it cannot adjust to the variations in power and process-heat loads. The schematic of a more practical co-generation plant is shown in Figure 7.2. Figure 7.3 shows the T-s diagram for the practical co-generation system described in the Figure 7.2.

The practical co-generation cycle depicted in Figure 7.2 works as follows:

- 1. Under normal operation, some steam is extracted from the turbine at an intermediate pressure P<sub>6</sub> (which can be predetermined).
- 2. The remaining steam expands to the condenser pressure P<sub>7</sub> and is then condensed at constant pressure and temperature (saturation temperature).



Figure 7.2 Schematic of a practical co-generation system



Figure 7.3 T-s diagram of a practical co-generation system

- 3. When the process heat demand is high, all the steam is routed to the process heating by isolating the condenser path ( $m_7 = 0$ ). For this mode of the cogeneration operation, the waste heat available is zero.
- 4. If the process heat is still not sufficient, some steam leaving the boiler is throttled by an expansion or pressure-reducing valve (PRV) to the extraction pressure P<sub>6</sub> and is directed to the process-heating unit.
- 5. Maximum process heating is realised when all the steam leaving the boiler passes through the expansion valve ( $m_5 = m_4$ ). In this mode of operation, no power is produced.
- 6. When there is no demand for process heat, all the steam passes through the turbine and the condenser ( $m_5 = m_6 = 0$ ), and the co-generation plant operates as an ordinary steam power plant.

The rates of heat input, heat rejected, and process heat supply as well as the power produced for this co-generation plant can be expressed mathematically as follows:

$$Q_{in} = \dot{m}_3 (h_4 - h_3) \tag{7.3}$$

$$Q_{out} = \dot{m}_7 (h_7 - h_1) \tag{7.4}$$

Performing an energy balance at the process heater yields:

$$Q_p = \dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_8 h_8 \tag{7.5}$$

$$W_{turb} = (\dot{m}_4 - \dot{m}_5) (h_4 - h_6) + \dot{m}_6 (h_6 - h_7)$$
(7.6)

We call the above co-generation plant an ideal plant when the plant operates optimally. The conditions for such an optimum operation are as follows:

- 1. All the steam expands in the turbine to the extraction pressure and continues to the process heating unit.
- 2. No steam passes through the expansion valve or the condenser; thus, no waste heat is rejected ( $m_4 = m_6$  and  $m_5 = m_7 = 0$ ). This condition may be difficult to achieve in practice because of the constant variations in the process heat and power loads, and equipment limitation. But the plant should be designed so that the optimum operating conditions are approximated most of the time.

Over the years, the economic feasibility of the co-generation system was dependent on fuel tariffs. When the fuel tariffs were extremely high at the beginning of the last century, the use of co-generation systems was very common. Power plants were integrated to provide district heating/cooling (for thermal comfort), hot water, and process heating for residential and commercial buildings. When the fuel tariff became cheaper, the popularity of the district heating/cooling systems declined. However, the rapid rise in fuel prices in the 1970s brought about renewed interest in district heating. In recent years, co-generation plants have proved to be economically very attractive. As a result, more and more such plants have been installed all around the world including in Singapore.

# 7.3 Tri-generation Systems

A CHP system producing three simultaneous effects from a single fuel, a trigeneration system, is shown in Figure 7.4.



Figure 7.4 A Tri-generation CHP system

The three useful effects: electrical power, heating (steam) and cooling (chilled water) are circled in the figure. As seen from the figure, a gas turbine cycle is topping the steam turbine cycle. The steam is generated with the help of a waste heat boiler or heat recovery steam generator using the recovered waste heat from the gas turbine exhaust. The high pressure (HP) steam is expanded in the steam turbine generating additional power. The expanded low pressure steam (LP) is sent to the adsorption chiller, which in turn produces the chilled water.

#### 7.4 Combined Gas-vapour Cycle Co-generation Systems

The thermal efficiency of power plants working on gas turbine power (Brayton) cycle is about 35% in comparison to those working on vapour power (Rankine) cycles, which have a thermal efficiency of up to 45% at the best. However, the continued quest for higher thermal efficiencies has resulted in rather innovative modifications to conventional power plants. One such modification is involving a gas power cycle topping a vapour power cycle, which is called the combined gas-vapour power cycle, or simply the combined cycle. The combined cycle of greatest interest is the gas turbine (Brayton) cycle topping a steam turbine (Rankine) cycle, which has a higher thermal efficiency than either of the cycles executed individually. Typically, gas turbine cycles operate at considerably higher temperatures than steam cycles. Due to technological advancements in metallurgy and the development of new materials, the maximum temperature to which the turbines can be subjected to has increased over the years. For example, the maximum fluid temperature at the turbine inlet is about 650°C for modern steam power plants and the corresponding temperature for the gas turbine power plant is as high as about 1500°C. As heat is supplied at a higher average temperature, gas turbine cycles have a greater potential for higher thermal efficiencies (the basic concept derived from an ideal Carnot cycle). However, the main disadvantage in a gas turbine cycle is the relatively higher temperature exhaust gas leaving the gas turbine. However, this drawback can be mitigated through the use of a regenerator, whereby the compressed air coming out of the compressor of the cycle is preheated using the exhaust gas with the help of a heat exchanger (as shown in Figure 4.10 in Chapter 4). This will improve the thermal efficiency of the cycle by reducing the heat input at the combustion chamber of the cycle. It is also to be noted that the very high temperature exhaust gas (500 to 550°C) could well be used in a Waste Heat Recovery Steam Generator (HRSG) to produce superheated steam and run a steam power cycle. The resulting cycle is a combined vapour power cycle or combined cycle as shown in Figure 7.5.



Figure 7.5 Schematic of a combined vapour cycle co-generation system

In the combined cycle power plant, energy is recovered from the exhaust gases by transferring it to the feedwater in an HRSG. Typically, more than one gas turbine is needed to supply sufficient heat content to the steam. In addition, the steam cycle may involve regeneration as well as reheating. Energy for the reheating process can be supplied by burning some additional fuel in the oxygen-rich exhaust gases. Developments in gas turbine technology have made the combined gas-steam cycle economically very attractive. The combined cycle increases the efficiency without greatly increasing the initial cost. In fact, many new power plants operate on combined cycles. There is also a trend around the world to convert existing steam- or gas turbine plants to combined-cycle power plants. Such conversions result in thermal efficiencies of well over 50%.

All the three main power generators (Tuas Power, Senoko Power and Power Seraya) as well as some of the smaller ones in Singapore use the combined vapour power cycles for power generation, accounting for more than 95% of the 13 GW of installed power plant capacity. Hot combustion gases enter the gas turbines at about 1,150°C, and steam enters the steam turbines at about 500°C. Steam is cooled in the condenser by cooling water at an average temperature of about 25°C. A 1,350-MW

combined-cycle power plant was built in Ambarli, Turkey, in 1988 by Siemens of Germany. It was the first commercially operating thermal plant

in the world to attain an efficiency level of as high as 52.5 percent at design operating conditions. This plant has six 150MW gas turbines and three 173MW steam turbines. It is to be noted that some power plants using the combined vapour power cycle have even achieved a thermal efficiency of about 60%. In Singapore, an efficiency of 55% is reported from a power plant working on the combined vapour power cycle.

## 7.5 Performance Analysis of CHP Systems

Co-generation systems can be broadly classified as those using steam turbines, gas turbines and DG sets. Steam turbine co-generation systems involve different types of configurations with respect to mode of power generation such as extraction, backpressure or a combination of backpressure, extraction and condensing.

Gas turbines with heat recovery steam generators are another mode of co-generation.

The entire system is dynamic and depends on power and steam load variations in the plant. A performance assessment would yield valuable insights into co-generation system performance and any need for further optimisation.

The purpose of the co-generation plant performance test is to determine the power output and plant heat rate. In certain cases, the efficiency of individual components like the steam turbine is addressed specifically where performance deterioration is suspected. In general, the plant performance will be compared with the base line values arrived at for the plant operating condition rather than the design values. The other purpose of the performance test is to show the maintenance accomplishment after a major overhaul. In some cases the purpose of evaluation could even be for a total plant revamp.

# 7.5.1 Performance Terms and Definitions

The terminologies related to the performance evaluation of co-generation systems are defined as follows:

Overall plant heat rate is expressed as:

 $\dot{m}_{steam} (h_{steam} - h_{fw}) / P_{out}$ 

(7.7)

Overall plant fuel rate is expressed as:

$$\dot{m}_{fuel}/P_{out} \tag{7.8}$$

(total fuel consumption for turbine and steam)

Turbine isentropic efficiency is defined as:

$$\Delta h_{turb,actual} / \Delta h_{turb,s} \tag{7.9}$$

Air compressor efficiency is expressed as:

$$\Delta h_{comp,ideal} / \Delta h_{comp,s} \tag{7.10}$$

Overall gas turbine efficiency = 
$$P_{out} (kW) / \dot{m}_{fuel} GCV$$
 (7.11)

HRSG efficiency = 
$$\dot{m}_{steam} (h_{steam} - h_{fw}) / \dot{m}_{fg} C_{pfg} \Delta T_{fg} + \dot{m}_{f,aux} GCV$$
(7.12)

#### 7.5.2 Field Testing Procedure for Co-generation

The test procedure for each co-generation plant will be developed individually taking into consideration the plant configuration, instrumentation and plant operating conditions. A method is outlined in the following section for the measurement of heat rate and efficiency of a co-generation plant.

Test Duration:

The test duration is site-specific and in a continuous process industry, 8-hour test data should give reasonably reliable data. In the case of an industry with a fluctuating electrical/steam load profile a set 24-hour data sampling for a representative period is recommended.

#### 7.5.3 Measurements and Data Collection:

The suggested instrumentation (online/ field instruments) for the performance measurement is as follows:

Steam flow measurement : Orifice flow meters

Fuel flow measurements : Volumetric measurements / Mass flow meters

Air flow / Flue gas flow : Venturi / Orifice flow meter / Pitot tubes Flue gas Analysis : Zirconium Probe Oxygen analyser Unburnt Analysis : Gravimetric Analysis Temperature : Thermocouple Cooling water flow : Ultrasonic flowmeter Pressure : Pressure Gauges/transducers Power : Clamp on power meters Condensate : Ultrasonic flow meter

It is essential to ensure that the data is collected during steady state plant running conditions. Among others, the essential parameters to be collected for co-generation plant performance evaluation are tabulated below:

No	Quantity to be computed	Parameters
1	Steam inlet to turbine	Flow rate
		Pressure
		Temperature
2	Fuel input to boiler/gas turbine	Fuel flow rate
		Composition
3	Combustion air	Flow rate
		Pressure
		Temperature
4	Process extraction steam	Flow rate
		Pressure
		Temperature
5	Back pressure steam to process	Flow rate
		Pressure
		Temperature
6	Condensing steam	Pressure or
		Temperature
7	Condenante franciscutine	
· ·	Condensate from turbine	Flow rate
		Temperature
8	Turbine bypass steam	Flow rate
9	Flue das to HRSG	Pressure
5		
		Iemperature
10	Exit flue gas	Temperature
		Composition

		Excess O <sub>2</sub>	
11	Cooling water to condenser	Flow rate	
		Temperature in and out	

Table 7.1: List of parameters for the performance testing

The various electrical energy-related parameters are as follows:

- 1. Total power generation for the trial period from individual turbines
- 2. Hourly average power generation
- 3. Quantity of power import from utility (Grid)
- 4. Quantity of power generation from DG sets
- 5. Auxiliaries power consumption

## 7.5.4 Calculations for Steam Turbine Co-generation System

Figure 7.6 below shows the process flow diagram for a typical co-generation plant. The following calculation procedures have been provided:

- Turbine efficiency.
- Overall plant heat rate (kJ /kWh)

# Energy Performance Assessment of Co-generation and Turbine



Figure 7.6 Process flow diagram for co-generation plant

Establish the specific enthalpies from the steam table (Appendix) for the given steam conditions:

Using P<sub>1</sub> and T<sub>1</sub>, steam enthalpy at turbine inlet:  $h_1 kJ/kg$ Using P<sub>2</sub> and T<sub>2</sub>, S<sub>g2</sub> is read from the steam property table and compared with the entropy at the state point 1 (assuming isentropic first stage expansion). The type of steam after the expansion is determined and the enthalpy at the state point 2 is read from the appropriate steam property table (saturated or superheated steam table) i.e. enthalpy at 1st extraction:  $h_2$  kJ/kg

Using  $P_3$  and  $T_{3}$ ,  $S_{g3}$  is read from the steam property table and compared with the entropy at the state point 1 (assuming isentropic first stage expansion). The type of steam after the expansion is determined and the enthalpy at the state point 2 is read from the appropriate steam property table (saturated or superheated steam table) i.e. enthalpy at 1st extraction:  $h_3$  kJ/kg

Using  $P_3$  and  $T_{3}$ ,  $S_g$  is read from the steam property table and compared with the entropy at the state point 1 (assuming isentropic first stage expansion). The type of steam after the expansion is determined and the enthalpy at the state point 2 is read from the appropriate steam property table (saturated or superheated steam table) i.e. enthalpy at 1st extraction:  $h_2$  kJ/kg. Referring to Figure 7.6,  $h_{11}$  is the enthalpy of feedwater at boiler pressure, typically used to quantify heat input into the boiler.

Wherever there is wet steam after the expansion in the turbine, the dryness fraction and the enthalpy of the steam are determined as follows:

$$S_1 = S_2 = S_f + x S_{fg} (7.13)$$

Where, x is the dryness fraction of steam.

$$h = h_f + x h_{fg} \tag{7.14}$$

(Relevant pressure and temperature are read from the steam property table)

Dryness fraction of the steam after the last expansion is estimated by assuming the expansion processes in the turbine to be isentropic. The typical dryness fraction value after the final expansion is 0.88 to 0.92.

The specific work produced by the turbine at each extraction point is calculated as follows:

At the first extraction point = $h_1 - h_2$ kJ/kg		
At the second extraction point = $h_2 - h_3 \text{ kJ} / \text{kg}$	(7.16)	

From the second extraction to the condenser =  $h_3 - h_4$  kJ/kg (7.17)

The actual enthalpies, taking the isentropic efficiencies of the pumps and turbine into account, can be established based on the isentropic efficiency definition (refer to the T-s diagram shown in Figure of the illustrative example 7.6.4 below)

The process heat, the net power output and the Utility Factor of the co-generation plant are determined as follows:

$Q_{process} = \dot{m}_{steam} \Delta h_{process}$	(7.18)
$W_{net} = W_{turb} - W_{pump}$	(7.19)
$U.F = W_{net} + Q_{process}$	(7.20)

Where,

Q<sub>in</sub> = Heat input in the boiler.

## 7.5.5 Example of co-generation system performance analysis

A distillery plant having an average production of 40 kilolitres of ethanol is having a co-generation system with a backpressure turbine. The plant steam and electrical demand are 5.3 Tons/hr and 200 kW, respectively. The process flow diagram is shown in the figure below. The gross calorific value of natural gas is 38,000kJ/kg. Boiler feedwater temperature is assumed to be 85°C.



The co-generation system performance analysis is carried out as follows:

Referring to the diagram and using 16-bar and  $300^{\circ}$ C, from the superheated steam table, the enthalpy  $h_1 = 3,036.2 \text{ kJ/kg}$ 

Using 3-bar and 140°C, from the superheated steam table,  $h_2 = 2,748.8 \text{ kJ/kg}$ Heat energy input to turbine is calculated as:  $h_1 - h_2 = (3036.2 - 2748.8) = 287.4 \text{ kJ/kg}$ Total steam flow rate,  $Q_1 = 6000 \text{ kg/hr} = 1.67 \text{ kg/s}$ Power generation = 200 kW Energy input to the turbine =  $1.67 \times 287.4 = 480 \text{ kW}$ Power generation efficiency of the turbo alternator,  $\eta_{\text{turbo-alternator}} = 200/480 = 0.416 = 41.6 \%$  $\eta_{\text{turbo-alternator}} = 41.6\%$  $\eta_{\text{alternator}} = 92 \%$  (Typical efficiency of an alternator)  $\eta_{\text{gear}} = 98 \%$  (Typical efficiency of a very efficient gear)  $\eta_{\text{turbine}} = \eta_{\text{turbo-alternator}} / (\eta_{\text{alternator}} \times \eta_{\text{gear}}) = 0.416/(0.92 \times 0.98) = 0.461$ Quantity of steam bypassing the turbine = Nil Natural gas consumption of the boiler = 1,600 kg/hr.

Overall plant heat rate, kJ/kWh =  $\dot{m}_{steam} \times (h_{steam}, kJ/kg - h_{fw}, kJ/kg) / P_{output}, kW$ = 6,000 x (3,036.2 - 355.9)/200 = **80,409 kJ/kWh\***  $h_{fw} = h_{f@85C} = 355.9 kJ/kg$ 

\*Note: The plant heat rate is in the order of 80,409 kJ/kWh because of the use of a backpressure turbine. This value will be much reduced while operating on fully condensing mode. However, with a backpressure turbine, the energy in the steam is not wasted, as it is utilised in the process.

Overall plant fuel rate including boiler = 1,600/200 = 8 kg NG / kW

The following points are observed from the analysis of the co-generation cycle.

- The efficiency of the turbine generator set is as per manufacturer design specification. There is no steam bypass indicating that the power generation potential of the process steam is fully utilised.
- At present the power generation from the process steam completely meets the process electrical demand, or in other words the system is balanced.
- It is to be noted that similar steps can be followed for the evaluation of the performance of a gas turbine based co-generation system.
## 7.6 Illustrative Examples

## Example 7.6.1

Steam enters the turbine of a co-generation plant at 70-bar and 600°C. One-fourth of the steam is extracted from the turbine at 8-bar pressure for process heating. The remaining steam continues to expand to 15 kPa. The extracted steam is then condensed and mixed with feedwater at constant pressure and the mixture is pumped to the boiler pressure of 70-bar. The mass flow rate of steam through the boiler is 50 kg/s. Disregarding any pressure drops and heat losses in the piping, and assuming the turbine and the pump to be isentropic, determine the net power produced and the utilisation factor of the plant.



Referring to the T-s diagram and using steam property table

 $h_1 = h_{f@14kPa} = 220 \text{ kJ/kg}$ 

 $v_1 = v_{f@14kPa} = 0.001 \text{ m}^3/\text{kg}$ 

 $w_{pump-1} = v_1 (P_2 - P_1) = 0.001 (800 - 15) = 0.785 kJ/kg$   $h_2 = h_1 + w_{pump-1} = 220 + 0.785 = 220.785 kJ/kg$  $h_3 = h_{f@8bar} = 721 kJ/kg$ 

Performing an energy balance at the mixing chamber yields:  $\dot{m}_4h_4 = \dot{m}_2h_2 + \dot{m}_3h_3$ Therefore,  $h_4 = (\dot{m}_2h_2 + \dot{m}_3h_3)/\dot{m}_4 = [(12.5)(220.78) + (37.5)(721)]/50 = 595.6 kJ/kg$ 

 $w_{pump-2} = v_4 (P_5 - P_4) = 0.001 (7,000 - 800) = 6.2 \text{ kJ/kg}$   $h_5 = h_4 + w_{pump-2} = 595.6 + 6.2 = 601.8 \text{ kJ/kg}$ Using 70 bar and 600°C from the superheated steam property table,  $h_6 = 3,649 \text{ kJ/kg}$ ,  $s_6 = 7.088 \text{ kJ/kg.K}$ 

Using 8 bar and  $s_6 = s_7 = 7.088 \text{ kJ/kg.K}$ ,  $h_7 = 2977.4 \text{ kJ/kg}$ For 14 kPa,  $s_g = 8.031 \text{ kJ/kg.K}$ , therefore the steam at the state point 8 is wet. The dryness fraction  $x_8$  is determined as follows:  $s_6 = s_7 = s_8 = s_{f@14kPa} + x_8 s_{fg@14kPa} = 7.088 \text{ kJ/kg.K}$ That is,  $0.737 + x_8 7.294 = 7.088 \text{ kJ/kg.K}$ Therefore,  $x_8 = 0.87$  $h_8 = h_{f@14kPa} + x_8 h_{fg@14kPa} = 220 + 0.87 \times 2,376 = 2,287.12 \text{ kJ/kg}$ 

The net turbine work is calculated as follows:

$$\begin{split} & W_{\text{Turbine}} = \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8) \\ &= 50(3,649 - 2,977.4) + 37.5(2,977.4 - 2,287.12) = 59,465 \text{ kW} \\ & W_{\text{pump,in}} = W_{\text{pump-1}} + W_{\text{pump-2}} = 37.5(0.786) + 50(6.2) = 339.5 \text{ kW} \\ & W_{\text{net}} = W_{\text{Turbine}} - W_{\text{pump,in}} = 59,465 - 339.5 = \textbf{59,125 kW} \\ & Q_{\text{process}} = \dot{m}_7(h_7 - h_3) = 12.5(2,977.4 - 721) = 28,205 \text{ kW} \end{split}$$

 $Q_{in} = \dot{m}_5(h_6 - h_5) = 50(3,649 - 601.8) = 152,360 \text{ kW}$ 

The co-generation utility factor, U.F =  $(W_{net} + Q_{process})/Q_{in} = (59,125 + 28,205)/152,360 = 0.573 = 57.3\%$ 

#### Example 7.6.2

Steam is generated in the boiler of a co-generation plant at 10 MPa and 450°C at a steady rate of 5 kg/s. In normal operation, steam expands in a turbine to a pressure of 0.5 MPa and is then routed to the process heater, where it supplies the process

heat. Steam leaves the process heater as a saturated liquid and is pumped to the boiler pressure. In this mode, no steam passes through the condenser, which operates at 20 kPa. (a) Determine the power produced and the rate at which process heat is supplied in this mode. (b) Determine the power produced and the rate of process heat supplied if only 60 percent of the steam is routed to the process heater and the remainder is expanded to the condenser pressure. *Solution* 



Referring to the T-s diagram and using steam property table,

$$\begin{split} h_1 &= h_{f@20kPa} = 251 \text{ kJ/kg} \\ v_4 &= v_{f@5bar} = 0.001 \text{ m}^3\text{/kg} \\ w_{pump-2} &= v_4 (P_5 - P_4) = 0.001 (10,000 - 500) = 9.5 \text{ kJ/kg} (1 \text{ bar} = 100 \text{ kPa}) \\ \text{Here } h_3 &= h_4 \text{ as all the steam is passed through the process heater} \\ h_3 &= h_{f@5bar} = 640 \text{ kJ/kg} = h_4 \\ h_5 &= h_4 + w_{pump-2} = 640 + 9.5 = 649.5 \text{ kJ/kg} \end{split}$$

Using 100 bar and 450°C from the superheated steam property table (Reference 3),  $h_6 = 3241 \text{ kJ/kg}$ ,  $s_6 = 6.419 \text{ kJ/kg.K}$ For 5 bar,  $s_g = 6.822 \text{ kJ/kg.K}$ , therefore the steam at the state point 7 is wet. The dryness fraction  $x_7$  is determined as follows:  $s_6 = s_7 = s_{f@5bar} + x_7 s_{fg@5bar} = 6.419 \text{ kJ/kg.K}$ That is,  $1.86 + x_7 4.962 = 6.419 \text{ kJ/kg.K}$ Therefore,  $x_7 = 0.92$  $h_7 = h_{f@5bar} + x_7 h_{fg@5bar} = 640 + 0.92 \times 2,109 = 2,580.3 \text{ kJ/kg}$ 

The net turbine work is calculated as follows:

 $W_{\text{Turbine}} = \dot{m}_6(h_6 - h_7) + \dot{m}_6(h_6 - h_7)$ = 5(3,241 - 2,580.3) = 3,303 kW 
$$\begin{split} & w_{pump,in} = w_{pump-2} = 5(9.5) = 47.5 \text{ kW} \\ & W_{net} = W_{Turbine} - w_{pump,in} = 3,303 - 47.5 = \textbf{3,255 kW} \\ & Q_{process} = \dot{m}_7(h_7 - h_3) = 5(2,580.3 - 640) = \textbf{1,940 kW} \end{split}$$

For 20 kPa,  $s_g = 7.907$  kJ/kg.K, therefore the steam at the state point 8 is wet. The dryness fraction  $x_8$  is determined as follows:  $s_6 = s_8 = s_{f@20kPa} + x_8 s_{fg@20kPa} = 6.419$  kJ/kg.K That is,  $0.832 + x_8 7.075 = 6.419$  kJ/kg.K Therefore,  $x_8 = 0.789$   $h_8 = h_{f@20kPa} + x_8 h_{fg@20kPa} = 251 + 0.789 \times 2358 = 2,111.4$  kJ/kg  $w_{pump-1} = v_1 (P_2 - P_1) = 0.001 (500 - 20) = 0.48$  kJ/kg  $h_2 = h_1 + w_{pump-1} = 251 + 0.48 = 251.48$  kJ/kg Performing an energy balance at the mixing chamber yields:  $\dot{m}_4h_4 = \dot{m}_2h_2 + \dot{m}_3h_3$ Therefore,  $h_4 = (\dot{m}_2h_2 + \dot{m}_3h_3)/\dot{m}_4 = [(2)(251.48) + (3)(640)]/5 = 484.5$  kJ/kg  $h_5 = h_4 + w_{pump-2} = 484.5 + 9.5 = 494$  kJ/kg

The net turbine work is calculated as follows:  $W_{Turbine} = \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8)$  = 5(3,241 - 2,580.3) + 2(2,580.3 - 2,111.4) = 4,241.3 kW  $w_{pump,in} = w_{pump-2} + w_{pump-2} = 3(9.5) + 2(0.48) = 29.5 \text{ kW}$   $W_{net} = W_{Turbine} - w_{pump,in} = 4,241.3 - 29.5 = 4,211 \text{ kW}$  $Q_{process} = \dot{m}_7(h_7 - h_3) = 3(2,580.3 - 640) = 5,820 \text{ kW}$ 

#### Example 7.6.3

Consider a co-generation power plant modified with regeneration. Steam enters the turbine at 90-bar and 400°C and expands to a pressure of 16-bar. At this pressure, 35 percent of the steam is extracted from the turbine, and the remainder expands to 10 kPa. Part of the extracted steam is used to heat the feedwater in an open feedwater heater. The rest of the extracted steam is used for process heating and leaves the process heater as a saturated liquid at 16-bar. It is subsequently mixed with the feedwater leaving the feedwater heater, and the mixture is pumped to the boiler pressure. Assuming the turbines and the pumps to be isentropic, show the cycle on a T-s diagram with respect to saturation lines, and determine the mass flow rate of steam through the boiler for a net power output of 25 MW.



Solution:



16 bar

10 kPa

8

s

 $h_1 = h_{f@10kPa} = 192 \text{ kJ/kg}$  $v_1 = v_{f@10kPa} = 0.001 \text{ m}^3/\text{kg}$  $w_{pump-1} = v_1 (P_2 - P_1) = 0.001 (1600 - 10) = 15.9 \text{ kJ/kg}$  $h_2 = h_1 + w_{pump-1} = 192 + 15.9 = 207.9 \text{ kJ/kg}$  $h_9 = h_{f@16bar} = 721 \text{ kJ/kg}$ 

2

Performing an energy balance at the mixing chamber yields:

 $h_4 = 0.35h_9 + 0.65h_3$ 

Therefore,  $h_4 = (0.35h_9 + 0.65h_2) = (0.35)(721) + (0.65)(207.9) = 387.4 \text{ kJ/kg}$ 

 $W_{pump-2} = V_4 (P_5 - P_4) = 0.001 (9,000 - 1,600) = 7.4 \text{ kJ/kg}$  $h_5 = h_4 + w_{pump-2} = 387.4 + 7.4 = 394.8 \text{ kJ/kg}$ 

Using 90-bar and 400°C from the superheated steam property table (Reference 3),

 $h_6 = 3,118 \text{ kJ/kg}, s_6 = 6.286 \text{ kJ/kg.K}$ 

Using 16-bar and  $s_g = 6.422$  kJ/kg.K, which is greater than  $s_6 = s_7 = 6.286$  kJ/kg.K, therefore the steam at state point 7 is wet.

The dryness fraction x<sub>7</sub> is determined as follows:

 $S_6 = S_7 = S_8 = S_{f@16bar} + X_7 S_{fg@16bar} = 6.286 kJ/kg.K$ 

That is,  $2.344 + x_7 4.078 = 6.286 \text{ kJ/kg.K}$ 

Therefore,  $x_7 = 0.94$ 

 $h_7 = h_{f@16bar} + x_7 h_{fg@16bar} = 859 + 0.94 x 1,935 = 2,677.9 kJ/kg$ 

For 10 kPa,  $s_g = 8.149$  kJ/kg.K, therefore the steam at the state point 8 is wet. The dryness fraction  $x_8$  is determined as follows:

 $S_6 = S_7 = S_8 = S_{f@10kPa} + X_8 S_{fg@10kPa} = 6.286 kJ/kg.K$ 

That is,  $0.649 + x_8 7.5 = 6.286 \text{ kJ/kg.K}$ 

Therefore,  $x_8 = 0.75$ 

 $h_8 = h_{f@10kPa} + x_8 h_{fg@10kPa} = 192 + 0.75 \times 2,392 = 1,986 \text{ kJ/kg}$ 

The net turbine work is calculated as follows:

$$\begin{split} & w_{\text{pump,in}} = w_{\text{pump-1}} + w_{\text{pump-2}} = 0.65 \dot{m}_6 (7.4) + \dot{m}_6 (15.9) \\ & W_{\text{net,turbine}} = \dot{m}_6 (h_6 - h_7) + 0.65 \dot{m}_6 (h_7 - h_8) - w_{\text{pump,in}} = 25,000 \text{ kW} \\ & \dot{m}_6 (3,118 - 2,677.9) + 0.65 \dot{m}_6 (2,677.9 - 1,986) - [0.65 \dot{m}_6 (7.4) + \dot{m}_6 (15.9)] \\ & = 25,000 \text{ kW} \\ & \dot{m}_{\text{steam}} = \dot{m}_6 = \textbf{28.76 kg/s} \end{split}$$

#### Example 7.6.4

A textile plant requires 4 kg/s of saturated steam at 20bar, which is extracted from the turbine of a co-generation plant. Steam enters the turbine at 90bar and 500°C at a rate of 11 kg/s and leaves at 10 kPa. The extracted steam leaves the process heater as a saturated liquid and mixes with the feedwater at constant pressure. The mixture is pumped to the boiler pressure. Assuming an isentropic efficiency of 86 percent for both the turbine and the pumps, determine (a) the rate of process heat supplied, (b) the net power output from the system, and (c) the utilisation factor of the CHP plant.



Referring to the schematic and T-s diagram and using steam property table (Reference 3),

$$\begin{split} h_1 &= h_{f@10kPa} = 192 \text{ kJ/kg} \\ v_1 &= v_{f@10kPa} = 0.001 \text{ m}^3\text{/kg} \\ w_{pump-1} &= v_1 \left(P_2 - P_1\right) = 0.001 \left(2,000 - 10\right) = 19.9 \text{ kJ/kg} \\ h_2 &= h_1 + w_{pump-1} = 192 + 19.9 = 211.9 \text{ kJ/kg} \\ \text{Isentropic efficiency of pump and turbines are given as 0.86.} \\ \eta_{s,pump} &= (h_2 - h_1)/(h_2' - h_1) \\ \text{Therefore, } h_2' &= h_1 + (h_2 - h_1)/(\eta_{s,pump}) = 192 + (211.9 - 192)/0.86 = 215 \text{ kJ/kg} \end{split}$$

 $h_3 = h_{f@20bar} = 909 \text{ kJ/kg}$ 

Performing an energy balance at the mixing chamber yields:

 $\dot{m}_4h_4 = \dot{m}_3h_3 + \dot{m}_8 h_2'$ 

Therefore,  $h_4 = (\dot{m}_3h_3 + \dot{m}_8 h_2)/\dot{m}_4 = [(4.0)(909) + (11.0)(215)]/11.0 = 545.5 \text{ kJ/kg}$ 

$$\begin{split} w_{pump-2} &= v_4 \left( P_5 - P_4 \right) = 0.001 \ (9,000 - 2,000) = \ 7 \ kJ/kg \\ h_5 &= h_4 + w_{pump-2} = 542.4 \ + 7 = 394.8 \ kJ/kg \\ \eta_{s,pump} &= (h_5 - h_4)/ \ (h_5' - h_4) \\ \end{split}$$
 Therefore,  $h_5' = h_4 + (h_5 - h_4)/ \ \eta_{s,pump} = 545.5 + (549.4 - 545.5)/0.86 = 550 \ kJ/kg \end{split}$ 

Using 90-bar and 500°C from the superheated steam property table,

 $h_6 = 3633 \text{ kJ/kg}, s_6 = 6.958 \text{ kJ/kg.K}$ 

Using 20-bar,  $s_g = 6.340$  kJ/kg.K, which is less than  $s_6 = s_7 = 6.286$  kJ/kg.K, therefore the steam at state point 7 is superheated.

Therefore, using 20-bar and ,  $s_6 = s_7 = 6.958 \text{ kJ/kg.K}$ 

 $h_7 = 3,138 \text{ kJ/kg}$ 

 $\eta_{s,turb1} = (h_6 - h_7')/(h_6 - h_7)$ 

Therefore,  $h_7$ '=  $h_6 - (h_6 - h_7) \eta_{s,turb1} = 3633 - (3,633 - 3,138)0.86 = 3,207.3 kJ/kg$ For 10 kPa,  $s_g = 8.149 kJ/kg.K$ , therefore the steam at the state point 8 is wet.

The dryness fraction  $x_8$  is determined as follows:

 $s_6 = s_7 = s_8 = s_{f@10kPa} + x_8 s_{fg@10kPa} = 6.958 kJ/kg.K$ 

That is,  $0.649 + x_8 7.5 = 6.958 \text{ kJ/kg.K}$ 

Therefore,  $x_8 = 0.84$ 

 $h_8 = h_{f@10kPa} + x_8 h_{fg@10kPa} = 192 + 0.84 \times 2,392 = 2,201 \text{ kJ/kg}$ 

 $\eta_{s,turb2} = (h_7 - h_8')/(h_7 - h_8)$ 

Therefore,  $h_8' = h_7 - (h_7 - h_8) \eta_{s,turb2} = 3138 - (3,138 - 2,201)0.86 = 2,332 \text{ kJ/kg}$ 

(a) The rate of process heat supplied,

 $Q_{\text{process}} = \dot{m}_7(h_7' - h_3) = 4(3,207.3 - 909) = 9,193 \text{ kW}$ 

The net power output from the system is calculated as follows:  $P_{Turbine} = \dot{m}_6(h_6 - h_7') + \dot{m}_8(h_7' - h_8') = 4(3,633 - 3,207.3) + 7(3,207.3 - 2,332)$  = 7,830 kW  $P_{pump,in} = P_{pump-1} + P_{pump-2} = 7(19.9) + 11(7) = 216.3 \text{ kW}$  $P_{net,system} = P_{Turbine} - w_{pump,in} = 7,830 - 216.3 = 7,613 \text{ kW}$ 

 $Q_{in} = \dot{m}_5(h_6 - h_5) = 11(3,633 - 550) = 33,913 \text{ kW}$ 

The co-generation utility factor, U.F =  $(P_{net,system} + Q_{process})/Q_{in} = (7,613 + 9,193)/33,913 = 0.496 =$ **49.6%** 

#### Example 7.6.5

Consider a co-generation power plant that is modified with reheat and that produces 4 MW of power and supplies 8 MW of process heat. Steam enters the high-pressure turbine at 90-bar and 500°C and expands to a pressure of 12-bar. At this pressure, part of the steam is extracted from the turbine and routed to the process heater, while the remainder is reheated to 500°C and expanded in the low-pressure turbine to the condenser pressure of 10-kPa. The condensate from the condenser is pumped to 12-bar and is mixed with the extracted steam, which leaves the process heater as a compressed liquid at 110°C. The mixture is then pumped to the boiler pressure. Assuming the turbine to be isentropic, show the cycle on a T-s diagram with respect to saturation lines, and disregarding pump work, determine (a) the rate of heat input in the boiler and (b) the fraction of steam extracted for process heating.



Solution



Referring to the schematic and T-s diagram and using steam property table (Appendix),

$$\begin{split} h_1 &= h_{f@10kPa} = 192 \text{ kJ/kg} \\ v_1 &= v_{f@10kPa} = 0.001 \text{ m}^3\text{/kg} \\ w_{pump-1} &= v_1 \left(P_2 - P_1\right) = 0.001 \left(2,000 - 10\right) = 19.9 \text{ kJ/kg} \\ h_2 &= h_1 + w_{pump-1} = 192 + 19.9 = 211.9 \text{ kJ/kg} \\ h_3 &= h_{f@110}{}^o\text{C} = 461 \text{ kJ/kg} \end{split}$$

Using 90-bar and 500°C from the superheated steam property table (Appendix),

 $h_6 = 3385 \text{ kJ/kg}, s_6 = 6.657 \text{ kJ/kg.K}$ 

Using 12-bar,  $s_g = 6.523$  kJ/kg.K, which is less than  $s_6 = s_7 = 6.657$  kJ/kg.K, therefore the steam at state point 7 is superheated.

Therefore, using 12-bar and ,  $s_6 = s_7 = 6.657 \text{ kJ/kg.K}$ 

h7 = 2,810 kJ/kg

The amount of process heat supplied =  $\dot{m}_7 (h_7 - h_3) = 8,000 \text{ kW}$ 

That is, m<sub>7</sub> (2,810 - 461) = 8,000 kW

Therefore,  $\dot{m}_7 = 8,000 \text{ kW}/(2,810 - 461) = 3.41 \text{ kg/s}$ 

Using 12-bar and 500°C from the superheated steam property table,

 $h_8 = 3,220 \text{ kJ/kg}, s_8 = 7.539 \text{ kJ/kg.K}$ 

For 10 kPa,  $s_g = 8.149$  kJ/kg.K, therefore the steam at the state point 9 is wet.

The dryness fraction  $x_9$  is determined as follows:

 $s_8 = s_9 = s_{f@10kPa} + x_9 s_{fg@10kPa} = 7.539 \text{ kJ/kg.K}$ 

That is,  $0.649 + x_9 7.5 = 7.539 \text{ kJ/kg.K}$ 

Therefore,  $x_9 = 0.92$ 

 $h_9 = h_{f@10kPa} + x_9 h_{fg@10kPa} = 192 + 0.92 x 2,392 = 2,392 kJ/kg$ 

The power output from the system disregarding the pump power is expressed as follows:

$$\begin{split} \mathsf{P}_{\mathsf{Turbine}} &= \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_8 - h_9) = \dot{m}_6(h_6 - h_7) + (\dot{m}_6 - \dot{m}_7) \ (h_8 - h_9) = 4,000 \ \mathsf{kW} \\ \text{i.e.} \ \dot{m}_6(3,633 - 3,003) + (\dot{m}_6 - 3.14) \ (3,220 - 2,392) = 4,000 \ \mathsf{kW} \\ \text{Therefore,} \ \dot{m}_6 &= \textbf{4.52 kg/s} \end{split}$$

Performing an energy balance at the mixing chamber yields:  $\dot{m}_4h_4 = \dot{m}_3h_3 + \dot{m}_2 h_2$ Therefore,  $h_4 = [\dot{m}_3h_3 + (\dot{m}_6 - \dot{m}_7)h_2]/\dot{m}_6$ = [(3.14)(461) + (4.52 - 3.14)(192)]/4.52 = 378.8 kJ/kg Disregarding the pump work,  $h_5 = h_4 = 378.8 \text{ kJ/kg}$   $Q_{in,boiler} = \dot{m}_5(h_6 - h_5) + \dot{m}_9(h_8 - h_7)$ = 4.52(3,633 - 378.8) + 1.38(3,220 - 3,003) = 15,007 kW

## Summary

Industrial professionals dealing with CHP systems need to understand the optimum operation of a CHP system in terms of its heat and electricity requirements. As far as design is concerned, the CHP system can be for a co-generation or a tri-generation system with the gas turbine topping the cycle. This chapter provided a detailed introduction to CHP systems starting with the discussion of an ideal CHP cycle and finally the most widely used combined vapour power cycle, which is the most popular power generation cycle in Singapore as well. The chapter also includes a number of illustrative examples for better understanding of the optimum operation of co-generation and tri-generation systems.

# References

1. Cengel Yunus, Thermodynamics: an engineering approach, McGraw-Hill, New York, 2008.

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# 8.0 THERMAL AND ELECTRICAL DESIGN OF CHP SYSTEMS

This chapter provides a detailed account of the thermal and electrical design of a Combined Heat and Power (CHP) Systems. Prior to the thermal and electrical design of a CHP system for a facility, it is very important to understand the electrical and heat load requirements for the facility. The CHP system design can either be heat load matching or electricity load matching. This Chapter covers considerations such as understanding of electrical and thermal load requirements in order to ensure a CHP system is beneficial for a particular application. A Cost Benefit Analysis (CBA) of the proposed system subject to various local codes of practices and regulations has to be carried out to get a clear picture of the financial return that the investment on the proposed CHP system would bring in. The financial aspects that are to be considered prior to the planning of a CHP system will be covered in Chapter 9 of this reference manual.

#### Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. The thermal load profile and its estimation
- 2. The electrical load profile of a facility
- 3. Heat to power ratio and their significance in the CHP system design
- 4. The CHP system selection options
- 5. Factors other than electrical and thermal loads in the selection of a CHP system

#### 8.1 Introduction

There are many considerations to be made in designing an optimal CHP plant that meets the facility level requirements. The load profile and demand are two of the major considerations but these two alone may not provide the whole input required in the decision on CHP system design. However, the load profile and demand data will determine the heat to power ratio of a facility which in turn would provide the required information for the proper selection of a CHP system in an optimum manner. Therefore, it is very important to understand these data as a preliminary step. Load profiles are developed to properly understand the loads throughout the year to facilitate the optimum efficiency operation of the CHP system. These data will also

serve as a baseline against which the output of the CHP system can be examined from time to time.

#### 8.2 Thermal Load Profiles

Based on the historical data for at least 12 months, an annual load profile can be developed. Simultaneously, an hourly CHP output model can be run on design parameters including load factors, efficiencies, and standby or supplemental energy requirements. This type of detailed modeling does provide accurate analysis of CHP performance against historical data and necessary adjustments can be made for external parameters such as ambient conditions, production levels, and occupancy. For future predictions, some kind of statistical tools like regression analysis could be used, albeit with a certain degree of inaccuracy. However, this level of data is rarely available for loads other than the electrical loads, and estimates must be drawn from the available data on how the loads function. Generally, it should be possible to establish the thermal loads of an industrial facility from the monthly or annual gas usage, etc., for example, Million Metric British Thermal Unit (MMBTU) of natural gas billed to the facility by the gas vendor. In the case of oil and coal, data on delivery quantities can be used. In such cases, an assessment must be made on the monthly use of the fuel, which may best be determined through discussions with facility personnel. Typically, the total fuel throughput must also be adjusted using the existing thermal conversion devices to calculate the total monthly thermal loads. This must be further refined to apportion the correct volume of fuel to each addressable load. In the absence of individual data for each load, estimates should be made for the portion of energy use associated with each load. Figure 8.1 shows a two-year thermal load profile of a production facility considering a CHP system. In general, it will be useful to review three to five years of historical billings or monthly fuel use data to identify and ascertain the long-term trends and to assess whether the most recent 12-month data are typical and representative of the facility's operational thermal demand. It is to be noted that the latest 12-month data forms the basis of CHP system modelling and design. It is necessary to use any available data to develop profiles, which are often supplemented by short-term data collection or data collected using submetering. In the case of a boiler steam generation estimation in the absence of an onsite steam mass flow meter, measurement of boiler feedwater may provide a reasonably accurate and inexpensive measure of steam generation rate by the boiler. Figure 8.2 shows daily steam demand for the production facility considered for a CHP system installation.



Figure 8.1. Two-year thermal load profile of a production facility

It will be very useful to quantify steam system/boiler losses from an operational and maintenance point of view. It is common to use periodic readings of the gas meter in conjunction with an inventory of gas-fired equipment and more generic data such as water use patterns as a basis for creating a thermal load profile for the facility. If an energy audit was carried out in the recent past, the data acquired in the energy audits could be a reliable source of energy data too, assuming the data collected include salient data points. The energy audits may include short-term metering or other data collection efforts that can be the basis of information required for the CHP system. In addition, one should identify existing site operational problems related to the thermal application, anticipated future changes, and any other factors that should be considered in any CHP system analysis. Typically, a level-3 audit covering all the heating and cooling application in conjunction with the historical utility bills will be helpful for the development of a reliable load profile. In the absence of any detailed energy audit of the facility, the plant logs by the supervisors will also form a useful source of data provided one understands the basis of such logs. For this, the following questions



Figure 8.2 Daily steam load requirement for a Production Facility

could be prompted: When during the day or shift are data entered? Are data taken at the same time each day or during successive shifts? Are the terms used on the log consistent with generally accepted practice? Are the data available from the logs consistent with metered, utility-supplied data? Once these estimates have been developed and monthly use profiles developed, the data must be further refined to calculate the load usage during operating hours for a facility that does not support 8760-hour operation of a CHP plant. The resulting addressable loads during operating hours must then be plotted and further analysed to establish the base, average, and peak loads so that the true load factor for the CHP system could be established.

#### 8.3. Electrical Load Profiles

Electricity demand data can be trend logged at an appropriate time interval to provide a very precise baseline for facility electrical load profiling. These data may be less accurate in situations where the data provided are actually the aggregated values from two or more meters. In this situation, some effort is required to understand the actual load characteristics for each meter, because only a single meter may be served by a CHP system unless inverters and additional controls are added between the power output of the CHP system and the loads. The electrical load data are usually obtained by the facility owner from the local utility supplier, which continually monitors and records this information for its own billing purposes. In some situations, the facility owner should be able to download the data directly from the utility web portals wherever the access is granted. The data for a full year is a minimum requirement to properly understand the seasonal variations, and data for more than one year can be obtained and should be reviewed to properly understand ambient temperature impact.

The above data are normally provided as a CSV (comma separated value) file that can easily be read by programs like Excel spreadsheet for further analysis. The data are typically trend logged in kW. In some cases, the data may be provided as kWh readings, in which case the actual kW demand can be derived by dividing the kWh value by the ratio of the period to one hour. For example, if the data were provided as kWh readings in 15-minute intervals, then the actual demand is the reading value divided by 0.25. If the readings were in 30-minute intervals, then the actual demand is the reading divided by 0.5.

Figure 8.3 shows a weekly electrical load profile for a production facility based on utility-provided 5-minute interval demand data. As seen from the figure, the facility has a base load of approximately 800 kW. It is also obvious from the figure that there is a

trend of high demand during working days followed by low demand for weekends. There are also some anomalies where readings appear to go to zero for short periods three times during the year and a single event when the demand increased to approximately 75% more than the normal demand for a short period.



Figure 8.3 Weekly electrical load profile for a production facility

To better understand what exactly is happening, further analysis of the data is required. For this, a daily analysis of the acquired data could be performed. The daily load profile now reveals in greater detail how the facility operates throughout the day. Workdays would follow a regular pattern of startup, production during the day, and shutdown in the evening. In the daily load profile analysis, there could be some data anomalies as shown in Figure 8.4 that may need to be investigated further.



Figure 8.4 Hourly electric load profile for a production facility

For example, one of the anomalies could be a single zero reading that occurred on one of the week days. In this situation, the readings before and after this reading could be examined and if it found to be typical for the period, the reasons for the zero data anomaly, which is a short duration event could be one of the following

- a grid failure
- testing of standby generators
- a meter malfunction

It is to be noted that this does not affect the design of the CHP system other than possibly enhancing the value of CHP as a backup system if this and the other similar anomalies were in fact grid failures.

On the other hand, there are cases where a high demand anomaly is found between an interval. For example, in Figure 8.5, between 2:30 a.m. and 3:00 a.m, the electrical power demand surged to almost twice the demand in the previous and subsequent intervals. The demand recorded was also found to be significantly higher than the peak demand reading of about 850 kW. It is highly unlikely that the facility actually required this level of power to operate, especially at 2:30 a.m. on a Sunday, which normally is the lowest load time. The likely causes for such a surge in power demand could either be one of the following:

- a fault level or short circuit current
- a meter malfunction

In either case, this should be investigated with facility personnel, and, if no fault is recorded, this should be reported to the local utility provider and a corresponding adjustment needs to be made for the affected period.





#### 8.4 CHP System Configuration Options

The outputs from a CHP system are electrical or mechanical power and heat and/or cooling from the same input fuel energy source. The first of the two or three output energy streams produced by a topping-cycle CHP system is power. A topping CHP system cycle is a cycle whereby power or electricity is produced first followed by heat or cooling outputs. The waste heat exhaust in the electricity/power generation process is then recovered with the help of heat recovery devices and converted to useful energy that can be applied to facility process loads. In a topping cycle, the prime mover that generates the power can be of different types, e.g. reciprocating internal combustion engines, combustion turbines, micro-turbines or even fuel cells. A detailed discussion of the prime movers is provided in the Chapter 6 of this reference manual. Each prime mover has different characteristics, both in terms of the amount of input energy that is converted to power and heat and/or cooling as well as the form that the heat or thermal output takes.

Experience with the CHP systems in Singapore suggests that all topping-cycle CHP systems have a nominal overall efficiency (utilisation factor, U.F) of approximately 70%. The overall efficiency calculation takes into account the electrical or mechanical power as well as recovered thermal energy for the facility process loads or preheating. Bottoming-cycle systems have CHP efficiencies of approximately 80%. It is to be noted that the CHP system efficiency can be increased if the system is applied to a thermal load that requires low-quality heat, but for most practical applications the load temperature requirements keep the overall efficiency at these levels. Depending on the CHP configurations, the performance level in terms of power efficiency versus thermal efficiency varies. A good example is a topping-cycle CHP system which may have 32% electrical and 38% thermal efficiency or 38% electrical and 32% thermal efficiency.

In bottoming-cycle CHP systems, fuel is burned and utilised for thermal energy first and that is then used to drive a power generation device such as a steam turbine or generators working on organic Rankine cycles. The remaining thermal energy based on the quality of the thermal energy is further recovered to meet other small or medium scale thermal applications in the facility. Experience also suggests that CHP systems working on bottoming-cycle generally have lower electrical efficiencies, ranging from about 15% to 25%, with superior thermal efficiencies of about 55% to 65%. The key performance indicators helping to evaluate the performance of a CHP system are electrical and thermal efficiency. Along with many other characteristics to be considered, the first CHP configuration option to be reviewed is the heat to power (H/E) ratio. The heat to power ratio is a function of the power and thermal efficiencies of the system. The heat to power (H/E) ratio is a primary criterion in matching the facility loads with the appropriate CHP configuration.

The possible number of CHP configurations is largely determined by the facility level heat to power ratio. There is always a misconception regarding the CHP prime movers that are operated in a CHP system because the prime movers are often described in terms of their rated electrical efficiency, leading one to intuitively believe that the 45% efficient engine is "superior in performance" to the 40% efficient engine. However, it is to be noted that the "superior" engine is the one that best matches the load requirements. It is obvious that a 45% electrically efficient engine will convert more of the fuel to electrical base load and has a higher thermal load than the CHP system's output, then a better option could be to select an engine with a lower electrical efficiency. This will enable more of the fuel energy to be converted to thermal energy, matching most of the thermal load by the CHP plant.

Applications such as district heating systems that use bottoming cycle configurations require very high heat to power ratios, because the system addresses large thermal loads with relatively small electrical loads available in the plant. On the other hand, if the limiting factor is thermal load availability, the higher electrically efficient engine could be selected for the CHP system. This will provide more power while maintaining a high thermal load factor.

From an economic point of view, the better option should be reviewed against energy costs to determine how to optimise fuel conversion efficiencies with the help of a Cost Benefit Analysis (CBA), etc. A CHP system's heat to power ratio (H/E) depends on the cycle, prime mover type and design, heat recovery systems used, thermal conversion technology type and design, and the load quality requirements of the facility. A second CHP system characteristic equally important in determining the CHP configuration that best meets facility needs, is the temperature requirements of the thermal load applications. The CHP systems that provide high-temperature energy streams are more easily applied to various loads, whereas lower-temperature CHP systems have some restrictions on the type of loads they can serve. For example,

non-recuperated combustion turbines that have an exhaust temperature of about 450°C could generate high-pressure steam without losing much of their heat recovery potential. On the other hand, a recuperated combustion turbine with an exhaust temperature of about 320° C will lose a considerable amount of its heat recovery potential if high pressure steam is required. Combustion turbines with and without recuperator are shown in Figures 8.6 and 8.7, respectively. However, it is worth noting that both the systems will retain high heat recovery potential if they are designed to produce hot water at about 70°C.







Figure 8.7 A combustion turbine without recuperator

Besides the important heat to power (H/E) ratio criterion, there are numerous other issues when a CHP system is designed for a facility. A facility may meet all the general requirements for the application of a specific CHP plant size and configuration, but fuel availability or emissions limitations by the authorities could be among the pressing

issues that must be addressed for the successful implementation of the CHP system. Stringent and rigid requirements may affect the CHP system design in terms of heat to power ratio optimisation. An economic analysis performed with these stipulations may look less attractive to one without the considerations of the regulations and stipulations by the local authorities. Such issues related to CHP systems are discussed briefly in the following sections.

#### Example:

An industrial facility has decided to go for a CHP system (Co-generation plant). Steam is available at 35 T/h with a pressure of 70-bar and 500°C. The process steam requirements are given in the Table below.

Process	Steam flow, T/h	Pressure, bar	Temperature, °C
Process # 1	4	20	300
Process # 2	10	8	180
Process # 3	21	5	160

Design the CHP system with single turbine. The facility requires 5 MW electrical power. Find out whether the CHP system is self-sufficient, or any additional power needs to be purchased from the local grid. Assume efficiencies of turbine and generator are 72% and 92%, respectively.

#### Solution:

Enthalpy of steam at the turbine inlet: 3,410 kJ/kgSteam mass flow rate = 35 T/h = 35,000 kg/hHeat available at the turbine inlet =  $35,000 \times 3,410 = 119.35 \text{ GJ/h}$ 

Enthalpy of steam at the turbine extraction 1 (20-bar,  $300^{\circ}$ C) = 3,025 kJ/kg Heat extracted at the turbine extraction 1 = 4,000 x 3,025 = 12.10 GJ/h Enthalpy of steam at the turbine extraction 2 (8-bar,  $180^{\circ}$ C) = 2,792 kJ/kg Heat extracted at the turbine extraction 2 = 10,000 x 2,792 = 27.92 GJ/h Enthalpy of steam at the turbine extraction 3 (5-bar,  $160^{\circ}$ C) = 2,767 kJ/kg Heat extracted at the turbine extraction 3 = 21,000 x 2,767 = 58.11 GJ/h

Overall turbo-alternator efficiency = Output power/Input power =  $\eta_t x \eta_g$ Input power = total heat at the turbine inlet – total extracted heat from the turbine = 119.35 – (12.10 + 27.92 + 58.11) = 21.22 GJ/h Power generation (Output power) =  $21.22 \text{ x} \eta_t \text{ x} \eta_g = 21.22 \text{ x} 0.72 \text{ x} 0.92 = 14.06 \text{ GJ/h}$ = **3.91 MW** 

Therefore, the electrical power to be imported from the grid = 5.0 - 3.91 = 1.09 MW

## 8.4.1 CHP system power quality

When a CHP system is planned and designed, due diligence and consideration should be given for the quality of the power, simply called power quality. The main power quality issues are:

- unbalanced voltage the ratio of max. voltage deviation from mean of 3 phases to the average voltage of the 3 phases)
- high reactive currents leading to poor power factor ( Power factor is defined as the ratio of actual power (kW) to the apparent power (kVA).
- high level of harmonics (Harmonics are electrical signal distortions caused by inadvertent interference of nearby signals)

These power quality issues will impair the operation of some sensitive equipment or cause them to underperform. In addition, poor power quality can negatively impact the performance and life of a CHP system. The power quality issues will also prevent the CHP system getting grid synchronised, affecting the import of electricity from the local grid. Hence, power quality issues will be thorny ones regardless of whether the CHP system is in island or grid-syncronised modes of operations.

# 8.4.2 Fuel characteristics of a CHP system

The reliability of the supply of fuel to a CHP unit will affect the project in many ways depending upon the critical nature of the application. Hence, the first thing that needs to be checked as far as the CHP system fuel supply is concerned is its availability. As far as CHP systems in Singapore are concerned, premium fuel such as Natural Gas (NG) is widely used. With the commissioning of the Liquefied Natural Gas (LNG) terminals in Singapore, NG is starting to flow through most of the country. Hence, NG is more reliable in Singapore for use as a premium fuel in gas turbines, etc. In many instances, the application of CHP will increase the peak fuel throughput in a facility that has an existing supply, and so a sufficient amount of fuel should be contracted for CHP systems involving critical applications.

In addition, careful consideration should also be given to fuel intake pressure because many prime movers require high-pressure fuel to operate. For example, combustion turbines require from 5 to 21-bar absolute pressure for the smallest to largest unit, respectively. Reciprocating engines and fuel cells generally require from 1.2-bar for smaller units to over 2-bar for larger units. Fuel pressure can be augmented with gas pressure boosters/compressors, but these systems will consume additional power ranging from under 3% for smaller combustion turbines to over 5% for larger units with low pressure supply.

When biogas is used as CHP system fuel, it may consume very high booster system power because the pressure at the supply point can be close to zero. In addition to the fuel availability and off-take pressure, fuel quality must also be checked. CHP systems using fuel cells may have fuel quality requirements (Contaminant free fuel, the contaminants are CO,  $H_2S$  and  $NH_3$ ) that exceed that provided by the local utility. It is important to recognise that not all gas supplies are of equal quality, with many gas grids accepting synthetic gas or other forms of gas that may have a deleterious effect on the gas quality, particularly just downstream from the point of injection. It is very common practice to have a fuel quality statement from the gas utility provided to the engine supplier to confirm the fuel compatibility. When considering CHP systems that are fueled by biogas or synthetic gas, a higher level of investigation regarding fuel energy content and impurities is required. Gas pretreatment equipment will most likely be required to remove moisture, sulfur, particulate matter, siloxanes, and other impurities. In addition, the operation and performance ratings of the engine must be characterised for the specific energy content of the fuel.

Low energy gases, such as municipal wastewater treatment plant anaerobic digester gas, are compatible with many engines, but these engines must be configured for the application and will not perform at the same capacity or efficiency as if the engine were configured for typical pipeline quality natural gas. Gas cleanup requirements can be relaxed if higher maintenance costs and more frequent maintenance intervals are allowed, but there may be little alternative to implementing high level gas pretreatment equipment if engine exhaust aftertreatment is required, because fouling of catalysts can occur very quickly if fuel quality is low.

Solid fuel biomass, such as woody biomass, can be directly combusted in a boiler to produce steam that turns a backpressure steam turbine to generate power, with the exhaust steam going to process or heating. Alternatively, the biomass can be pyrolysed to produce syngas, which can be combusted in an engine to produce power and heat. In these CHP systems, the fuel quality and energy content are a function of the type and moisture content of the biomass as well as the combustion or pyrolysis

process used. Such systems are generally designed around the specific feedstock and must be appropriately sized to the throughput of fuel necessary to provide the energy output required by the facility.

Although natural gas is the predominant fuel for CHP systems in the continental United States, CHP systems can also be operated with a variety of fuels, including diesel or propane. In all cases, the fuel specifications should be reviewed with the prime mover supplier to ensure that the available fuel quality is in line with the manufacturer's requirements as well as its impact on warranty, maintenance, and emissions.

#### 8.4.3 Electric Interconnection

Interconnection with the electrical utility grid provides a number of benefits to CHP systems, including the availability of supplemental and standby power, increased reliability, and operating flexibility. Most CHP systems are interconnected to the electrical utility grid, and interconnection issues are a critical concern in the design and operation of a CHP system. Though utility interconnection issues typically deal with protection of the utility from the effects of the CHP system, it is also necessary to consider protection of the on-site generator from problems caused by the utility grid. Utility interconnection concerns include the following:

**Power Quality:** One of the concerns by grid operators in Singapore, the Power System Operator (PSO) is that an interconnected on-site generator should not degrade the quality of power supplied by the utility in any circumstances. This is measured by voltage and frequency stability, power factor, and harmonic content. In general, with the exception of induction generators and inverters, the quality of power available from an on-site generator will exceed the quality of power that is available from the grid. From the owner's perspective, poor grid power quality, whether caused by facility loads or the utility, are of equal concern. Short-term metering to document power variations of a few seconds or less may be required where the end-user processes are characterised by short, significant changes in load. In this case, it is necessary to establish that the selected prime mover can respond to short-term variations in load.

**Power Safety:** Grid operators in Singapore (PSO) are concerned that an interconnected generator has the potential to energise a utility circuit that is not being powered by the utility. This condition can result in a safety hazard to utility personnel

working on that circuit. Most utilities will require the CHP system to install a reverse power relay and an external disconnect switch that is accessible by utility personnel and that can be used to disconnect and lock out the CHP system. For CHP systems connected to local grids, additional protective devices will be required to ensure the utility that the generator can be disconnected when required. These devices generally must be reviewed by and meet the approval of the authorities before an interconnection will be allowed. Generally, expertise of an electrical professional engineer is needed.

**Grid Fault Protection:** Utility operation of the grid can be quite complex and include the coordination of relays, switches, and fault control. The interconnection of a CHP system or any other active source of power within the grid generally must be reviewed by the utility to avoid jeopardising the ability of the utility to manage grid operations. This includes the calculation and declaration of the fault level or short circuit current during the design stage of the CHP system.

#### 8.4.4 Ancillary and other equipment

Power-house ancillary subsystems such as water treatment, power supply, condensate return, deaeration, cooling towers, and controls are used to support the CHP system. If existing subsystems are to be used, then they must be adequately maintained, have capacity, and be able to provide the quality required for the CHP system. If the existing subsystems are not adequate, then the cost of new subsystems and equipment should be included in the CHP project budget. Major existing energy components such as boilers, hot water generators, emergency power generators, and chillers will often still provide for the needs of the facility beyond the capacity of the CHP system. When CHP plants are designed to meet baseload requirements, the additional load is typically met using the existing equipment in parallel with the CHP plant. In some situations, the installation of a CHP system may defer or eliminate the need to replace a major component, thus providing a capital cost credit toward the project. For new construction or major renovations, the installation of a CHP system can be integrated with the installation of other major components, including boilers and chillers, thus minimising total project cost.

In all scenarios and particularly where existing equipment is used to support a new CHP plant, it is important to review the equipment loads when the CHP system is fully operational in order to ensure that the existing ancillary equipment has the turndown necessary to accommodate the CHP plant. For example, where a CHP plant that

generates 9 million Btu (9.5 GJ) of steam is added to an existing facility with a 300 hp steam boiler (10 million Btu/h) (2943 kW) that is at times fully loaded, it will be necessary to either adjust the existing boiler to provide for stable operation at 1 million Btu/h (294 kW) or add a new small boiler to provide the supplementary steam. It is not unusual that existing facilities have boilers that cannot operate efficiently below 30% of their nominal capacity such that, if the CHP thermal output is slightly less than the load, a new boiler may need to be incorporated as part of the CHP plant.

It is also important to recognise that, while CHP may provide a less costly source of power and thermal energy, it will not solve pre-existing problems with the site steam and hot or chilled-water systems or inadequacies in the power distribution system or automated control systems. Any CHP review or audit should attempt to identify any such inadequacies in the facility thermal and electrical distribution systems which should be fixed before installation of a CHP plant. Equally, any potential energy efficiency measures that would impact the loads intended to be addressed by the CHP plant should be implemented before installation of CHP equipment. Addressing such issues after installation of CHP may not only result in wasted resources, but also lead to poor performance of the CHP plant, resulting in lower efficiencies and higher-than anticipated maintenance costs.

#### 8.4.5 Emissions

Local air quality emissions requirements for stationary engines, fuel cells, or boilers, in the case of biomass CHP systems, are an important consideration for CHP system design. Most projects, as a minimum, are required to ensure that project emissions comply with local air quality requirements as defined by the local jurisdictional authority. (See Chapter 10 for further information on emissions considerations.) This may require the addition of exhaust aftertreatment devices, which must be included in the project capital cost as well as operation and maintenance costs, because most of these devices do consume chemicals and require replacement of components. Lower restrictions may be placed on renewable fuels, such as landfill gas or digester gas, but if these systems require exhaust aftertreatment, then a high level of gas pretreatment will most likely be required to prevent fouling of any catalysts used for exhaust aftertreatment. Consideration must also be given to the location of the point of emissions of the stack. The location must comply with local regulations that normally prescribe distances from air intakes, open windows, etc., as well as sensitive receptors (people).

#### 8.4.6 Maintenance requirements of CHP Systems

CHP plants' availability is about 95% and they operate closer to the availability of the system. Since a CHP system operates continuously, regular maintenance of the system involving the prime mover, thermal recovery and conversion equipment, and accessories such as gas pressure boosters, controls, pumps, and exhaust aftertreatment is of paramount importance. Continual preventive and predictive maintenance are required to sustain the performance and reliability of the plant as well as maximise the life of the CHP system equipment. It is also important to comply with emissions standards by the regulatory agencies like the National Environment Agency (NEA). Generally, maintenance costs of the CHP system can be a significant part of the total operating costs, typically representing up to 25% to 30% of operating costs when including engine rebuild or major overhaul. Maintenance requirements for various types of prime movers depends on their types. Experience suggests that to maintain a plant in good working condition through its full life cycle (which can go up to 20 to 30 years), the prime movers need to be subject to full overhauls or changeout of major components. The main variable in terms of maintenance requirements is the frequency of maintenance intervals, which can be a design consideration, particularly for critical or remote applications.

#### 8.4.7 Operating requirements

CHP systems are designed to operate on auto mode with some kind of load control based on power or thermal load requirements. Local electric utility grid interconnection normally requires automation of the prime mover to ensure compliance with certain grid connection protocols. Like many other systems, sophisticated state-of-the-art remote monitoring has minimised the man and machine interaction - in a CHP system, the interaction between the man and the prime mover. From an operational point of view, the prime mover requires people with specialised operational skills, typically provided by the prime mover manufacturers themselves or by the facility personnel trained by the prime mover specialists. In contrast, the operation requirements for heat recovery and thermal conversion equipment are typically the same as those for standard boilers, heat exchangers, absorbers, or other such equipment. The CHP system plant thermal equipment is also subject to existing equipment regulations by the government agencies like National Environment Agency, etc. For example, if the CHP plant is generating high-pressure steam, and regulations require specific boilerlicensed personnel to operate such equipment, then the CHP plant thermal equipment will also be subject to the same set of requirements. For cost effectiveness, it is recommended to have the existing boiler/chiller plant personnel operate and maintain the CHP system plant thermal equipment.

#### Summary

Understanding the basic concepts underlying the design and selection of a CHP system is of paramount importance. This chapter provided an account of the thermal system design of a CHP system for a facility based on the heat to power ratio of the facility as well as the other relevant local regulations and codes of practices. The importance of considering beyond the heat and load requirements are also emphasised in this chapter.

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# 9.0 THE FEASIBILITY STUDY FOR CHP

This chapter provides a detailed account of how a feasibility study is to be carried out prior to considering a CHP system for an industrial facility. Understanding the current energy sources and conversion technology along with their consumption and cost can form a baseline. This allows the facility owner to explore reducing annual energy consumption and cost. One way to make use of most of the fuel energy input is by using a CHP system described in the preceding chapters. However, before deciding on such a CHP system, it is very important to carry out a technical and economic feasibility study. In this chapter, various levels of such a feasibility study along with the economic analysis tools that can be used for the cost benefit analysis of a CHP system are presented.

#### Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. The basics of feasibility analysis
- 2. CHP simulation software and tools
- 3. Analysis of existing CHP systems
- 4. Different levels of CHP feasibility studies
- 5. CHP economic analysis

#### 9.1 Introduction

Prior to the consideration of a successful Combined Heat and Power (CHP) system, a careful economic and technical feasibility evaluation of the system needs to be carried out to convince the top management of the firm. The process of performing the feasibility study for a CHP system includes a preliminary screening study as well as a detailed and comprehensive study. During the feasibility study, information on existing or proposed facilities including cooling, electrical, heating load data, etc. are collected and analysed. In addition, technically feasible solutions to effectively and efficiently meet the facility's load requirements are developed. Finally, an economic analysis using a suitable economic tool is conducted. The economic analysis shall include estimating energy usage and the resulting cost, preparing budget cost estimates, and calculating life-cycle costs to determine the recommended plant size and configuration. The three phases of a CHP feasibility study for an existing installation are shown in Table 9.1. Table 9.2 provides an indication of the time, effort and information required, and typical cost.

Stages	Purpose
Site screening and defining scopes	To evaluate and ascertain if the site is a good
	candidate for a CHP system
Preliminary assessment (Level 1)	To evaluate and ascertain if a CHP system is
	technically appropriate and has economic
	potential
Detailed assessment (Level 2)	Data collection, analysis and optimisation of
	the CHP system including conceptual design

Table 9.1. Three phases of a CHP feasibility study					
Stages	Time required	Information required	Co		

Stages	I ime required	Information required	Cost of study
Site screening and	2 days	Site information to a	Negligible
defining scopes		minimal level and average	
		utility cost. e.g. Site layout,	
		nature of business, types of	
		energy used, energy tariff	
		and annual energy cost etc.	
Preliminary	3 – 5 weeks	3 years of utility bills,	\$10,000
assessment		building operation	
(Level 1)		information, building loads	
		including electrical, ACMV	
		and process heating loads,	
		future forecast on	
		production expansion,	
		equipment addition,	
		projected energy cost	
Detailed	1 – 4 months	Data collection for two to	\$10,000 to \$100,000
assessment		three weeks period, data	
(Level 2)		analysis and modelling,	
		conceptual design and cost	
		estimates	

Table 9.2. Information required for the feasibility study

For new installations, the integration of a CHP system is like any other mechanical and electrical system (such as hybrid chiller plants and air side systems). The feasibility of CHP systems in new installations should be part of the project design process, which is typically performed during the programming or planning stage. For the above-mentioned CHP feasibility studies, currently, quite a number of popular and proven building energy simulation programs are available. Some of the commonly used software are: HAP, Visual DOE, eQuest, Energy Plus and IES, etc.

The cost-effectiveness of CHP systems can be evaluated in the early stages of the system design. A section of this chapter describes the procedure of analysing a CHP system for new construction.

#### 9.2 CHP Simulation Software and Tools

Due to the variable nature of loads in an industrial environment, detailed sizing of CHP systems is not straightforward for most practical situations. The occurrence of concurrent heating and cooling loads as well as the varying nature of electrical demand, along with plant-level equipment performance factors and the time-variant nature of electrical and gas tariffs make the sizing an even more complicated process.

In addition, due to unprecedented technological advancement, several state-of-theart feasible equipment and system configurations are available. Selection of the best among them while considering issues of uncertainty and variability is a challenging task indeed. However, the availability of numerous proven energy simulation and analysis software provides some degree of respite in the CHP system feasibility study. If one does not have access to these software, a customised Excel spreadsheet could also be developed for the feasibility study of the system. Figure 9.1 shows a computer screen shot of such a customised Excel spreadsheet developed for CHP systems in Singapore.

PLANT INFORMATION	Parameters	
	5MW Gas	
System:	Turbine	
No of machines	1	
Max electricity Load in MW		* assume 30% electrica
(gross)	4.7	efficiency
Max auxiliary electricity Load		
in MW	0.3	
		1
Actual CHP Electrical Load in	. ==	
MW (gross):	4.70	4
Actual CHP Electrical Load in		
MW (net):	4.40	
Actual Steam from CHP (t/h)	12.00	
CHP Load factor	100%	1
CHP Availability	100%	-
Load profile	Steady	
	0.000.07	
Annual Operating hours	8300	
HSFO (US\$/MT)	400	
FX (US\$ to S\$)	1.5	
Hurdle Rate	10%	
	\$	
Cost per CHP (S\$)	8,500,000.00	4
No of years amortised	15	J

	Units	Current	With CHP	Assumptions
Electricity				
Electrical Base Load demand	MW	6	6	
Annual Electricity imported from Grid	MWh	49800	13280	For CHP - electricity import during downtime to be considered
Annual Electricity generated from CHP	MWh	0	36520	CHP system availability to be considered
Steam demand (8 barg, saturated)	t/h	12	12	1 t/h approx 0.644MW
Annual Steam generated by Boilers	Tons/A	99600	0	
Annual Steam generated by CHP	Tons/A	0	99600	CHP system availability to be considered
Annual Fuel consumption (LHV basis)				·
Boiler	mmbtu	243171	0	To cater for CHP downtime
CHP (no duct firing capability)	mmbtu (HHV)	0	488041	CHP system availability to be considered

Operational						
Fuel (Natural Gas) Total Cost	SG\$ Mil/yr	\$	4,486,504.31	\$	8,872,587.32	
Electricity Purchased Cost (Energy only)	SG\$ Mil/yr	\$	9,063,600.00	\$	2,416,960.00	
Total 3rd Parties Charges						*AFP charges for power from CHP systems
(Network UOS, MSSL, EMC & PSO, AFP, MEUC						are incurred twice (considered as import
Charges)	SG\$ Mil/yr	\$	653,874.00	\$	332,132.80	and export)
						*No reduction considered (as a
Contracted Capacity Charge	SG\$ Mil/yr	\$	506,880.00	\$	506,880.00	conservative assumption)
O & M Cost	SG\$ Mil/yr	\$		\$	500,000.00	
Total Operational Cost	SG\$ Mil/yr	\$	14,710,858.31	\$	12,628,560.12	
Capital Cost						
Total Capex and development cost	SG\$ Mil	\$	-	\$	8,500,000.00	
						Hurdle Rate (see cell B18)
Annualised capital cost	SG\$ Mil/yr	\$	-	\$	1,117,527.10	No. of Years amortised (see cell B20)
Total annual Cost						
Annual Total Cost	SG\$ Mil/yr	\$	14,710,858.31	\$	13,746,087.22	
Appual Savings	SG¢ Mil/yr	4			064 771 00	
Annual Savings	100¢ Pill/yi	4			304/11103	
% Operational Cost Avoidance	14%	1				
	14-70					
% Total Cost Avoidance	/%					
Payback (years)	8.8					
Electricity (3rd Party charges)						
Network	\$ 8.37		\$/MWh			
MSSL	\$ 1.61		\$/MWh			
EMC / PSO	\$ 0.64		\$/MWh			
AFP	\$ 2.16		\$/MWh			
MEUC	\$ 0.35		\$/MWh			
Total	\$ 13.13		\$/MWh			
Energy (USEP)	\$ 182.00		\$/MWh			
Contracted Capacity (CC)	\$ 7.04		\$/kW			
Fuel cost						
Gas cost (boiler only)	\$ 18.45		\$/mmbtu	*D	ifference in price	
			-	att	ributed to cost of	
	1	1		Itra	nemission at	
				1 ci ci	nonnoon ac	
Gas cost (with chp)	\$ 18.18		\$/mmbtu	dif	ferent pressure	
Gas cost (with chp)	\$ 18.18		\$/mmbtu	dif	ferent pressure	

The above scenario is for a brownfield development, hence existing sunk costs on assets (such as boilers) are not factored in.
Maintenance cost for existing systems are not factored in the CBA, since such costs are still being expensed
The annualised cost varies depending on the hurdle rate of the developer/company, as well as the number of annualised years.

Figure 9.1 A sample CHP feasibility analysis Excel spreadsheet

#### 9.3 CHP Analysis of Existing Facility

CHP analysis of an existing industrial facility is initiated with a site screening taking a maximum of two days. This site screening will help to determine if a CHP system makes technical and economic sense for the facility's year-round operations. This screening is typically conducted in liaison with the facility managing team, answering specific questions before undertaking the engineering and economic analyses. As indicated in Table 9.2, the required information for this screening is minimal. Formulation of a simple spreadsheet will also be helpful for this case with the spreadsheet addressing the question "Is My Facility a Good Candidate for CHP?" A set of 12 questions is shown in Table 9.3; if the answer is yes for three or more of these questions, the facility may be a good candidate for CHP. If the site is indeed found to be a good candidate, a Level 1 feasibility study could be undertaken.

No.	Questions
1	What is your current electricity tariff? More than \$0.13/kWh?
2	Are you concerned about the impact of current or future energy tariffs on your
	business operations?
3	Are you concerned about power reliability? Is there a substantial financial or
	other impact to your business if the business operation is disrupted for 1 hr.?/
	for 10 minutes?
4	Does your facility operate more than 5,000 hours per year?
5	Do you have heating (steam, hot water, chilled water using absorption chiller,
	hot air, etc.) requirement throughout the year?
6	Currently, do you have a central plant?
7	Do you anticipate a replacement, upgrading or retrofitting of central plant
	equipment within the next 3-5 years?
8	Do you anticipate a facility/production expansion within the next 3-5 years?
9	Have you already retrofitted your facility with any energy efficient measures
	but still have high energy cost?
10	Do you intend to reduce your inefficient energy consumption impact on the
	environment?
11	Do you have access to any nearby biomass resources? e.g. food processing
	waste etc.

Table 9.3. Questionnaire for the CHP suitability check

# 9.3.1 CHP Level 1 feasibility study

The purpose of the CHP Level 1 feasibility study is to determine the technical applicability and the economic benefits of the CHP system for the facility under consideration. Essentially, the level 1 feasibility study requires an experienced facility level engineer familiar with CHP technology and with a good understanding of the electrical, thermal, and cooling loads as well as the equipment operation. The main task of this facility engineer will be to collect and analyse data, and apply his/her engineering judgement to whether the CHP system will deliver economic benefits to the facility.

The CHP level 1 feasibility study starts with data collection with highly accurate instrumentation. For example, temperature, flow rate and power are trend logged for a sufficient period of time with thermistors, non-intrusive type flow meters (ultrasonic flow meter) and clamp-on power meters, respectively.

Prior to the data gathering, a level 1 feasibility analysis checklist comprising the following main elements is prepared:

- Contact information
- Site information and data
- As-built drawings of building, chiller and boiler plants, etc.
- Electricity and fuel use data
- Thermal loads like heating, cooling, domestic hot water, etc.
- Existing equipment design specifications
- Other relevant data

These data can be acquired by liaising with the facility team that is in charge and responsible for the daily operation of the various systems and equipment.

Subsequent to the data collection, the engineer will proceed with the following tasks:

# **Hurdle Identification**

This first and foremost step involves the identification of any major uncontrollable hurdles that will prevent the project from being implemented. Typical examples of such hurdles are:

- long-term corporate power purchase contracts that will not allow installation of on-site power generation
- local utility and regulatory policies that add CHP constraints and costs
- stringent local building and environmental codes
- special requirements for the stack to exhaust the products of combustion
- space for prime mover and auxiliary equipment
- noise level constraints, etc.

If any one of these hurdles is revealed during this stage of the economic analysis, it should be eliminated by adopting the optimum and best available solution.

# **Conceptual Engineering Design**

This stage includes the sizing and identifying of prime mover technology along with thermally operated equipment such as absorption chillers for waste heat utilisation. Basically, the conceptual engineering design of the CHP system will be based on the site load requirements such as electrical, cooling and heating load profiles. The electrical energy consumption profile could be obtained from utility bills, submetering (electrical, steam/hot water, chilled water) or in some cases, from trend data. The cooling and heating load profiles could be obtained from the trend logged data of a sufficient duration (e.g. two weeks).

Another approach for obtaining the building load profiles is by calibrated simulation. This is the use of hour-by-hour building energy simulation by using energy simulation software such as Carrier Hourly Analysis Program (HAP) or Equest to calibrate various physical inputs to the developed building model so that the actual energy use from utility bills or other sources matches closely with that predicted by the building energy simulation. The accuracy of the calibrated simulation depends heavily on the data available from the site personnel. The results of the calibrated simulation are a set of 8760 hourly values for electrical demand, thermal energy for space heating, domestic hot water, and cooling energy. This information along with proper tools can be used for optimal sizing of the prime mover and thermally operated chiller with available analytical tools. It can be as simple as a customised Excel spreadsheet.

In cases where the site has already implemented (or plans to implement) energy conservation measures (ECM), it is important to take these measures into account in the optimal sizing of the prime mover(s) and, if applicable, the absorption chiller. In addition to equipment sizing, the engineer or the CHP project developer will investigate the proper prime mover technology (reciprocating engines, gas turbine, microturbine, etc.). Despite the capability of the available analytical tools to optimise the prime movers, it is suggested that several prime mover alternatives be considered. This could be in terms of the prime mover sizes, technology, the best available absorption chillers, etc.

In the event of the unavailability of an optimum sizing tool, it is advisable for the design engineer to start off with a prime mover or movers with the capability of providing a portion of the site electrical power demand and the majority of the site thermal load. The shortfall site electrical power resulting from this design approach could be imported from the Singapore electricity grid. This approach is known as thermal load matching resulting in higher system efficiency since it maximises the use of waste heat.

The following is a summary of different types of CHP design options:
**Island Mode Operation:** where the CHP site is stand-alone, i.e. the CHP plant is not connected to the local grid, and hence all the thermal and electrical needs have to be met by the CHP system. In this mode of operation, excess standby capacity for scheduled and unscheduled maintenance as well as momentary demand spikes and energy creep issues must also be considered.

**Electric Base Load Design:** Sized by electric base-load where the CHP system is sized such that it meets the minimum electricity billing demand which can be established from the past electricity bills. Any shortfall in electrical power is purchased from the local electricity grid. Similarly, any thermal energy shortfalls have to be met by a separate economical heating source.

**Thermal Base Load Design:** where the CHP system is sized so that most of the facility's thermal load is met with heat recovered from the CHP prime mover, with any excess electrical power sold to the electricity grid and any shortfalls in electrical power met by importing from the local grid.

Intermediate Loads Design: where some amount of thermal load and some amount of electrical load are met by the CHP system. In reality, this is probably the most common design option because of the fact that the final CHP design and equipment sizing will depend on location-specific economics and issues such as energy security and reliability. Economic issues not only involve the consideration of cost of thermal and electrical energies, but also the operation and maintenance costs of the equipment as well as environmental costs.

**Peak Load Design:** where the CHP system is specifically designed to shed electrical demand during the peak period, and thereby save on demand charges.

# **Economic Analysis**

The CHP level 1 feasibility study is intended to determine the economic feasibility of a CHP system. An economic analysis is very important as well at this stage of the CHP system feasibility study. Typically, one should carry out an economic analysis using just a simple payback period tool to quantify the economic benefits without taking into account the time value of money and discounting factors. etc. As the name suggests, this simple payback way of evaluating the economic return is the simplest compared to more accurate and sophisticated methods such as net present value, internal rate of return (IRR), and life-cycle costs (LCC). The simple payback period method is the least accurate of all the methods because it does not take into account the time value of money, unlike all the other more accurate economic tools.

The simple payback period is the ratio of the capital or investment cost to the annual net cost savings. The cost of borrowing money, inflation, and other factors associated with the operation of the system during its lifetime are ignored in this method.

However, the simple payback analysis does take into account the following:

- Heat and power produced by the CHP system, and the estimated amount of each to be used on the site
- Avoided costs of utility-purchased heat and power
- Cost of fuel associated with running the CHP system
- Cost estimates to install and maintain the system
- Available incentives for CHP installations

These variables are applied to each of the proposed alternatives. It should be noted that, often, estimated equipment pricing is quite accurate at this initial stage, but other project development costs (such as the cost of CHP system tie-in and site construction expenditures, additional structural work, noise and pollution) are preliminary. Given these uncertainties it is important that reasonable estimates for all other turnkey costs associated with CHP system implementation, operation, and maintenance be included in this preliminary budget. Sometimes additional analysis will be required to account for benefits such as backup power in events of utility outage or potential increase in the utility rates.

The determination to proceed to a level 2 feasibility study will depend on the simple payback estimates, since owners have an upper threshold value based on their own economic criteria., If all of the previously mentioned costs and benefits are included in the preliminary economic analysis, it should provide a fairly accurate representation of the opportunity or benefit of the CHP project. It should be clear, however, that the results of this economic analysis are simply a necessary phase before proceeding to the more accurate economic study that is part of the level 2 feasibility study.

#### Typical Report Outline for CHP Level 1 Feasibility Study

The recommended structure of a level 1 CHP feasibility study shall be as follows:

1. Executive summary

- 2. Preliminary analysis and assessment
  - (a) Facility description
  - (b) Baseline utility cost
  - (c) Facility electrical, thermal, and cooling load profiles
  - (d) CHP systems design options and alternatives
  - (e) Engineering and energy analysis of CHP system design alternatives
  - (f) Emissions
  - (g) Utility interconnection
  - (h) Power reliability
  - (i) Budgetary installation and maintenance costs
- 3. Economic analysis
- 4. Conclusions and recommendations for level 2 feasibility study
- 5. Appendix

#### 9.3.2 CHP level 2 feasibility study

Once a level 1 feasibility study has found that CHP is economically and technically feasible, the level 2 feasibility study can be started. Many of the preliminary assumptions used in the level 1 feasibility study will be replaced with more accurate data. Additional data such as operational goals, controls, monitoring, and off-grid capabilities will also be considered in this study. This will help to revise and optimise the preliminary sizing presented in the level 1 feasibility study. The results of level 2 feasibility study should include all the information needed to make a decision on whether to proceed with the project, and typically include the following:

- More accurate estimated construction, operation, and maintenance pricing
- Estimates of the final project economics with a simple payback schedule and a life-cycle cost analysis of the total investment

The economic analysis will be based on final system sizing and proposed operation and will be based on more accurate thermal and electrical load profiles. Accurate data in this regard is measured data obtained from trending (utilising the existing control system or installing new instrumentation) or from electrical utility interval data. Planned site expansion or new construction has to be considered and coordinated with various entities in this facility; for example, engineering, planning, and construction. In cases where the CHP plant is part of a new construction substantial cost savings can be achieved, and these avoided costs have to be incorporated in the total implementation cost resulting in improved return on investment of the system. Several site visits and a comprehensive review of the existing conditions will be required as part of this study, thereby allowing the decision maker to make a well supported decision.

Typically, a CHP level 2 feasibility analysis report should include the following:

- Site load profiles
- System operational schedule
- · Mechanical and electrical system components
- Heat recovery
- Systems efficiency
- Sound levels
- System vibration
- Space considerations
- System availability during utility outage
- Utility interconnection
- Emissions and permitting
- Capital cost
- Fuel costs
- Maintenance costs
- Availability of incentives

Economic analysis including life-cycle analysis

- Financing options
- Preliminary project schedule

• Supporting documents for project execution (proposals, costs, design documents, etc.)

# Typical Report Outline for Level 2 Feasibility Study

As in the case of CHP level 1 feasibility study report, the recommended structure of a level 2 CHP feasibility study shall be as follows:

- Executive summary
- Description of existing site plan and equipment
- Site energy requirement
- CHP equipment selection
- Description of preferred CHP system
- System operation
- · Regulatory and permitting requirements overview

- Total CHP systems costs
- Assumptions for cash flow analysis
- Discounted cash flow analysis for preferred system
- Appendices

#### 9.3.3 CHP feasibility study for new facilities

CHP systems for new installations can be considered during the early stages of the design or called conceptual design phase. The same qualification test of existing facility can be applied for the new facilities as well. If a CHP system is found to be favourable, the designer can propose a CHP system as part of the development of the design alternatives. With the increased utilisation of building energy simulation programmes like IES, Equest and Energy Plus, etc., a preliminary model of the facility can be developed to assist the designer in analysing various design alternatives. A proposed CHP system can be one of the alternatives or one of several design alternatives involving different CHP system sizes. Since optimal CHP prime mover sizing is more complex than other mechanical and electrical equipment in buildings, it can be very beneficial to combine the strength of the building energy simulation program and other tools for optimal selection of the CHP prime mover and absorption chiller.

#### 9.4 CHP Economic Analysis

CHP economic analysis is the process by which the economic factors surrounding a proposed CHP plant are analysed to determine if the project makes economic sense and that the project is a good investment of the stakeholders' funds. The criteria for defining a project as economically viable will vary from project to project, but at a minimum the project typically needs to save money over the "business-as-usual" (BAU) case. More stringent criteria may require the project to perform at least as well as a competing investment. Methods of economic analysis vary from a simple payback analysis to the more complicated and detailed life-cycle-cost (LCC) analysis discussed in detail in this chapter.

#### **Economic Feasibility**

It is recommended to perform an energy audit, to identify areas where improvements to existing utility systems can be made. Design of a CHP system should be based on an optimised energy base-load. The cost benefit of installing a CHP plant shall be made by comparing with the following:

- Current methods/ costs of power procurement
- Current methods / costs of heat (steam) procurement (fuel / technology)
- Age of current systems (do they need to be replaced?)
- Other factors to consider include:
- Company's hurdle rate
- Any plant expansion
- Availability of fuel
- Engineering Capability (O&M support, Facility Management, etc.)
- Modifications to Infrastructure (If any)
- Load profile

A Cost Benefit Analysis (CBA) provides the CHP system developer with a snapshot of the current market condition. The CBA should not be used to make a decision to invest in a CHP system because other factors also need to be considered. The other factors which are to be considered include oil prices, availability of fuel, fuel switching as well as foreign exchange. One or more of these factors may change with time which would diminish the attractiveness of implementing a CHP system. Therefore, a sensitivity study should also be made based on changes in these factors. It is also a good idea to determine the breakeven point at which it does not make sense to invest.



Figure 9.2 Sensitivity analysis for a CHP system

The various economic tools used in the economic analysis of a CHP system are:

- Simple payback period
- Return on investment (ROI)
- Cash flow analysis
- Net present value (NPV)
- Discounted payback period

• Internal rate of return (IRR)

Each of these economic tools is briefly described in the following sections.

#### Simple Payback Period

Simple payback period is an estimation of how long it will take to recover the initial investment.

Mathematically, Simple payback period = Capital investment / annual savings

#### Example:

The investment for a new energy efficient chiller is \$1 million. The expected annual savings is \$100,000 *Solution* 

Simple payback period = \$ 1million / \$100,000 = 10 years

The advantages and disadvantages of the Simple Payback Period are as follows:

#### Advantages:

- Simple to use
- Gives an immediate indication of how long it will take to recover the investments
- Useful when time scale is small
- Provides an indication of when cash flow will become positive

#### **Disadvantages:**

- Does not consider time value of money
- Does not account for cash flows after payback period

#### Return on Investment (ROI)

The Return on Investment expresses the annual return from the project as a percentage of the investment cost

Mathematically, ROI = (Annual net cash flow / capital cost) x 100

In order to make an investment decision, the rate of return should be higher than cost of capital (interest rate). The greater the difference between ROI and interest rate, the better the investment for the project

The advantages and disadvantages of the ROI are as follows:

#### Advantages:

- Simple to use
- Provides indication of Return from investment (to compare with cost of capital)

#### Disadvantages:

- Does not consider time value of money
- Does not consider variable nature of cash flow

#### Example

A new chiller A will cost \$500,000 and will provide savings of \$80,000 a year. Another new chiller B will cost \$550,000 and will provide savings of \$85,000 a year. Compare the ROI for the above 2 options.

Solution

Return on Investment (ROI) for the chiller A = 80,000/500,000 = 0.16 = 16%Return on Investment (ROI) for the chiller B = 85,000/550,000 = 0.155 = 15.5%

# **Cashflow Analysis**

Cash flow resulting from a project may not be uniform due to reasons such as varying nature of electricity tariff, maintenance cost, utilisation hours of the new system, tax rates, etc. Hence, a cashflow analysis covering the entire life span of the new system or equipment needs to be carried out. An example of such a cash flow analysis is given below:

# Example 9.1

The investment cost for new equipment is \$250,000. Expected energy savings are \$50,000 based on the present electricity tariff. There will be no maintenance cost for the first year of operation. Thereafter, the annual preventive maintenance cost will be \$5,000 a year. An additional maintenance cost of \$5,000 will be charged every 5 years starting from the 5th year of operation. Assume that the electricity tariff will increase by 5% a year from the 2nd year, prepare the cash flow for the first 10 years of operation.

#### Solution

	Year 0	Year 1	Year 2	Year 3	Year 4	Year 5	Year 6	Year 7	Year 8	Year 9	Year 10
Capital cost	(250,000)										
Energy savings		50,000	52,500	55,125	57,881	60,775	63,814	67,005	70,355	73,873	77,566
Maintenance cost			(5,000)	(5,000)	(5,000)	(10,000)	(5,000)	(5,000)	(5,000)	(5,000)	(10,000)
Net cash flow	(250,000)	50,000	47,500	50,125	52,881	50,775	58,814	62,005	65,355	68,873	67,566
Cumulative cash flow	(250,000)	(200,000)	(152,500)	(102,375)	(49,494)	1,282	60,096	122,100	187,455	256,328	323,895

#### Time Value of Money

Project cash flows (expenditure and savings) will result during the expected life of the project. The typical project life of a CHP system would be close to 20 years or more. Therefore, the value of cash flows resulting during different years will not be the same. To assess the feasibility of a project taking this into account, the present and future cash flows need to be converted to a common basis. A discounting factor is used to convert all the future cash flows to the present value (PV) of money. For example, if \$1.00 is deposited in a bank account which pays 5% interest, the money deposited will be worth \$1.05 in one year. Therefore, money value of \$1.05 in one year has a present value of \$1.00. Similarly, the value of \$1.00 in one year has a present value of \$0.95 (i.e. 1/1.05 = 0.95).

Present value can be expressed mathematically as:

 $PV = FV / (1 + i)^n$ 

where,

FV = future value of cash flow at end of year "n"

i = interest rate or discounting rate

n = number of years in the future

#### Example 9.2

Calculate the PV for a cash flow of \$100,000 which will result in the 2nd year if the discounting rate is 5%.

Solution

 $PV = FV / (1 + i)^n = 100,000/(1+0.05)^2 = $90,702$ 

#### Net Present Value (NPV)

Net present value is the sum of the present values of cash flows resulting from a project.

Net present value can be expressed mathematically as:

$$\mathsf{NPV} = \sum_{1}^{n} \frac{Fv_n}{(1+i)^n} - I_0$$

where,

FVn = future value of cash flow at end of year "n"

i = interest rate or discounting rate

n = number of years in the future

lo = initial investment

For a project economic evaluation using NPV,

If NPV > 0, the investment would add value to the company and the project may be accepted

If NPV < 0, the investment would diminish value from the company and the project should be rejected

If NPV = 0, the investment would not gain or lose value for the company and the decision whether to proceed or reject the project would depend on other factors

# Example 9.3

Calculate the NPV for the following two projects taking the discounting rate to be 5%

Year	Project A	Project B
Year 0	(500,000)	(700,000)
Year 1	200,000	260,000
Year 2	200,000	260,000
Year 3	200,000	260,000
Year 4	200,000	260,000

Solution

Project A:

 $\mathsf{NPV} = 200,000/(1+0.05)^1 + 200,000/(1+0.05)^2 + 200,000/(1+0.05)^3 + 200,000/(1+0.05)^4$ 

- 500,000

= 190,476 + 181,406 + 172,768 + 164,540 - 500,000

= \$209,190

Project B:

 $NPV = 260,000/(1+0.05)^{1}+260,000/(1+0.05)^{2}+260,000/(1+0.05)^{3}+260,000/(1+0.05)^{4}$ 

- 700,000
- = 247,619 + 235,828 + 224,598 + 213,903 700,000
- = \$221,948

Year	1%	3%	5%	6%	8%	10%	12%	15%	20%
1	.990	.971	.952	.943	.926	.909	.893	.870	.833
2	.980	.943	.907	.890	.857	.826	.797	.756	.694
3	.971	.915	.864	.840	.794	.751	.712	.658	.579
4	.916	.888.	.823	.763	.735	.683	.636	.572	.482
5	.951	.863	.784	.747	.681	.621	.567	.497	.402
6	.942	.837	.746	.705	.630	.564	.507	.432	.335
7	.933	.813	.711	.665	.583	.513	.452	.376	.279
8	.923	.789	.677	.627	.540	.467	.404	.327	.233
9	.914	.766	.645	.592	.500	.424	.361	.284	.194
10	.905	.744	.614	.558	.463	.386	.322	.247	.162
11	.896	.722	.585	.527	.429	.350	.287	.215	.135
12	.887	.701	.557	.497	.397	.319	.257	.187	.112
13	.879	.681	.530	.469	.368	.290	.229	.163	.093
14	.870	.661	.505	.442	.340	.263	.205	.141	.078
15	.861	.642	.481	.417	.315	.239	.183	.123	.065

Table 9.4 Discounting factor

If the project NPV (based on the discounting rate used) is positive, the project is considered to be acceptable.

The advantages and disadvantages of NPV are as follows:

#### Advantages:

Considers time value of money Takes into account variable nature of cash flow Can tell whether the investment will increase the company's value

#### **Disadvantages:**

Requires an estimate of the cost of capital Provides an absolute \$ value (not a %)

The discounting rate used in the economic analysis of a CHP system project should take into consideration the following:

- borrowing cost
- return on investment (opportunity cost)

• project risk and other factors

# Examples (without considering risk) :

If a CHP system project requires funds to be borrowed at 7% interest rate - the discounting rate should be 7%. On the other hand, using the firm's own funds, if an alternative investment can bring returns of 10%, then the discounting rate should be taken as 10%.

If one of the CHP system projects requires \$4 million for which \$1 million is to be borrowed at 7% while \$3 million will be from the company's own funds which can yield 10% in an alternative investment, then the discounting rate is calculated by giving the respective weightage on the borrowed and own money as follows:

Discounting rate can be taken as 10%, or weighted average =  $(0.25 \times 7\%) + (0.75 \times 10\%) = 9.25\%$ 

# **Discounted Payback Period**

This method uses the discounted cash flows to compute the payback period and therefore accounts for time value of money.

Mathematically, discounted payback period is expressed as:

Discounted payback period = year before recovery + (unrecovered cost at the beginning of the year / cash flow during the year)

# Example 9.4

Calculate the discounted payback period for the following project taking the discounting rate to be 7%

Year	Cash flows
Year 0	(1,500,000)
Year 1	400,000
Year 2	450,000
Year 3	500,000
Year 4	600,000

Year	Cash flows	Discounted cash flows	Cumulative DCF
Year 0	(1,500,000)		
Year 1	400,000	373,832	373,832
Year 2	450,000	393,047	766,879
Year 3	500,000	408,149	1,175,028
Year 4	600,000	457,737	1,632,765

#### Solution

Discounted payback period = year before recovery +

(unrecovered cost at the beginning of the year / cash flow during the year)

= 3 + (1,500,000 - 1,175,028) / 457,737

= 3 + 0.71

= 3.71 years

#### Internal Rate of Return (IRR)

The economic evaluation tool, Internal Rate of Return, calculates the rate of return a CHP system investment is expected to make. The rate of return is the interest (discounting) rate at which the NPV = 0 (i.e. discounted benefits = discounted costs) NPV =  $0 = FV_0/(1+i)_0 + FV_1/(1+i)^1 + FV_2/(1+i)_2 + \dots + FV_n/(1+i)^n$  where,

 $FV_n$  = future value of cash flow at end of year "n" (positive for savings and negative for expenditure)

i = interest rate or discounting rate

n = number of years in the future

In the IRR method of economic evaluation, the value of "i" (discounting rate) needs to be calculated by trial & error. IRR should be higher than the "hurdle rate" for the company investing in the CHP system project. For different CHP system investment options, the one with the highest rate of return should normally be chosen.

# Example 9.5

For the following project cash flow, compute the IRR

Year	Cash flow
Year 0	(500,000)
Year 1	200,000
Year 2	200,000
Year 3	200,000
Year 4	200,000

Solution

$$NPV = 0 = 200,000/(1+i)^{1} + 200,000/(1+i)^{2} + 200,000/(1+i)^{3} + 200,000/(1+i)^{4} - 500,000$$

By trial & error, use i = 5%

NPV = 
$$200,000/(1+0.05)^{1} + 200,000/(1+0.05)^{2} + 200,000/(1+0.05)^{3} +$$

 $200,000/(1+0.05)^{4}-500,000 = 190,476 + 181,406 + 172,768 + 164,540 - 500,000$ 

= \$209,190

Next try i = 15%

NPV = 
$$200,000/(1+0.15)^{1} + 200,000/(1+0.15)^{2} + 200,000/(1+0.15)^{3} + 200,000/(1+0.15)^{4} - 500,000$$

= \$70,996 (still positive)

Therefore, this project investment, the IRR is between 15% and 25%.

It is to be noted that the NPV, IRR types of economic evaluation can easily be carried out using the built-in functions of the Excel spreadsheet.

The advantages and disadvantages of the IRR method are as follows:

#### Advantages:

Uses time value of money Considers variable cash flows Able to compare with hurdle rate

#### **Disadvantages:**

Requires an estimate of the cost of capital Does not show improvement in \$ value Cannot allow more than one change in sign of cash flows

Some of the terminologies related to the economic analysis are described below:

#### Inflation Rates

Inflation rates are the rates at which the costs of goods or services increase. The inflation rates vary from country to country.

The inflation rates that are typically considered in a Life Cycle Cost analysis are as follows:

- Cost of energy (purchased electricity and fuel),
- Cost of labour (operations and maintenance labor, as well as administrative labour costs),
- Cost of permits,
- Cost of spare parts and general goods

#### Length of Analysis

The length of the LCC analysis of a CHP system is an important consideration. A typical Life Cycle Cost Analysis is approximately 20 to 25 years, though it will vary from project to project. As a CHP system consists of different components with varying useful service life, it would be more difficult to estimate the useful life span of the entire CHP plant. For example, the plant building may have a 50-year life, the prime mover

a 20-year life, and the piping a 30-year life. The confidence level of the economic analysis weakens as one gets into more micro levels of the CHP project. However, the project investors may have a standard analysis length that is used for all analyses, or a specified time period in which they need to achieve the projected savings.

### Salvage Value

Salvage value, or scrap value as it is interchangeably called, is the value of an asset, usually with a fairly high capital cost, at the end of its useful service life, e.g. car, chiller, boiler, building, etc. Whether or not salvage value is considered is primarily dependent on the length of the analysis, as discussed in the above paragraph. If the salvage value of any equipment that reaches the end of its useful service life is considered in the Cost Benefit Analysis, one of the stumbling blocks or hurdles in the economic analysis could be overcome. It is also to be noted that the actual effective amount of the salvage value and its significance to the overall analysis depends on many factors, such as whether the equipment was fully depreciated at the point of salvaging, market value of the equipment, which in turn can be affected strongly by the advancement of related technologies, and the cost of demolition/removal of the item, etc.

#### 9.5 Calculating Estimated Energy Use and Cost of a CHP plant

The energy cost constitutes the lion's share of the annual costs in the Life Cycle Cost analysis of a CHP plant. The energy usage could be estimated from the historical data or simulated from computer modeling. In Singapore, typically, the energy use in the industrial sector is in the form of electricity or natural gas consumption. In Singapore, the electricity tariffs are expressed in \$/kWh, while natural gas tariff is usually expressed in \$/MMBtu. Electricity rate schedules also include demand charge, which is an additional charge separate from the rate charge. Demand charge depends on the maximum power usage during the on-peak period, also referred to as the demand period. The purpose of the demand charge is to reflect the cost of providing utility and distribution capacity to meet a facility's peak electrical requirement. With the recent liberalisation of the Singapore electricity market, the utility suppliers in the contestable sectors tailor their rates specifically to the needs of their consumers. In the Singapore electricity market, there are peak and off-peak tariffs.

The various energy costs that are to be considered in the CHP system cost estimates are as follows:

• Electrical energy costs.

- Electricity power costs (demand charges).
- Standby charges. (Some electrical utilities will charge an interconnected facility a standby charge to guarantee electrical power capacity equal to the installed capacity of the CHP plant.
- Natural gas or fuel oil charges.

In order to estimate the energy costs, one must first estimate the energy usage in the facility where the CHP system is going to be installed. The first step in establishing the energy usage is to analyse the facility energy usage profiles through measurement or from the historical utility bills for at least 3 or 4 years.

Energy usage profiles that are considered in analysing a CHP plant are electricity usage, thermal usage (e.g., heating, domestic hot water production), and cooling usage. The interaction of these energy usages determines how the facility's energy needs will be met using the conventional method, as well as under each of the CHP options being considered. Once the energy usage is determined, costs can be assigned to each component of energy used (or saved).

These days, several commercially available proven and user-friendly software programmes are available for estimating the energy usage of many types of CHP system applications. Some of the advantages in using the software models are ease of use, repeatability, and presumed quality assurance of the model. Disadvantages of using the software include limitations in modelling unique CHP plant applications, and the model may have unnoticed flaws producing inaccurate but sensible outputs. Customised spreadsheet models can be created that allow for very detailed analysis of unique or "out of the ordinary" CHP applications, or allow the modelling to report a unique aspect of the results.

Advantages of spreadsheet modelling with for example, Microsoft Excel, include:

- Ability to build a detailed and customised model
- Increased ability to follow the logic behind each calculation step

Some of the disadvantages of spreadsheets are:

- Models require more time for initial creation and checking
- A higher chance of errors if close attention is not paid to detail
- A lack of annual hour-by-hour calculation

 Making even small changes to the model once it is substantially complete can prove challenging

Estimating Annual Operation and Maintenance Costs involves calculating the following costs:

- Preventive/periodic maintenance of CHP equipment
- Costs of consumables (lubricants, test gas for emissions monitoring, etc.)
- Repair of CHP equipment
- Rebuilding equipment during the life of the analysis
- Cost of permitting and annual testing (i.e., permit to operate, emissions control)
- testing
- Cost of operators and maintenance personnel
- Cost of administrative staff

#### 9.6 CHP System Budgetary Construction Costs

Depending on the depth of the Life Cycle Cost (LCC) analysis being prepared, budget construction cost estimates can range anywhere from a cost (\$) per kilowatt basis to a detailed item-by-item cost estimate basis. One can obtain equipment manufacturer/vendor quotes, cost estimating publications, contractor estimates, and professional costs estimators to come up with the total budgetary construction cost. It is advisable to always obtain multiple vendor quotes for all major equipment because of the expensive nature of such equipment. The key to preparing an accurate budget construction cost estimate is having a keen understanding of how the plant will be constructed. For example, knowing the installation cost of a main item alone may not be adequate and will not provide an accurate cost indication unless a clear picture is also available of the cost of all components., For example, in the case of a piping installation, the piping cost has to be furnished along with that of hangers, supports, and bracing in order to estimate the correct cost. Thinking further about possible constraints of the installation location, if the installer will be working in a tight space or at high platform levels, then the earlier assumed project cost may increase as a result of higher than anticipated installation costs.

CHP system construction cost estimation based on publications provide the raw material and labour costs. However, the actual cost may vary by 10% to 20 % for

materials and up to 50% for labour. Location factors also need to be considered while estimating the budgetary construction cost of CHP systems. Other factors include the following:

*Local Taxes:* Sales tax may be applicable to all purchased material depending on the locale and is applied to the subtotal including the above markups and location factors. *General requirements:* General requirements such as contractors' cost of reproduction, office equipment, construction trailers, mobilisation and demobilisation, project management, etc. Typical values are often around 5 percent and are either estimated individually or the 5 percent factor is applied to the subtotal including the sales tax.

*Contingency:* It is common to have a contingency amount to be added to cover unexpected costs. Contingencies may range from 5% to 25% based on the nature of the cost estimates.

*Insurance and bonds:* This is for the contractors and their labour and even for an expensive piece of equipment, this could be roughly around 3%.

Contractor's overhead and profit. This typically ranges from 10 to 15 percent and is applied to the subtotal including the insurance and bonds.

*Owner's project costs.* Project costs additional to the budget construction costs should be included in the LCC analysis. These include: cost of engineering design, testing, and inspection fees, and the owner's construction administration. Project costs are typically about 20% of the construction costs.

# 9.7 Calculating Life Cycle Cost (LCC)

Life Cycle Cost is the summation of all the component costs such as the capital cost, the annual operating and maintenance costs including energy costs, the borrowing cost and any taxes. The time value of money is used for converting the future value of money to present value (the net present value) so that alternatives can be compared on an 'apples to apples' basis.

Discount rate	5%
Maintenance escalation rate	3%
Fuel tariff escalation rate	2%
Electricity tariff escalation rate	2%
Adm. & Permit cost escalation rate	3%

Table 9.5 Example of economic factors assumed for the LCC

A sample of Life Cycle Cost calculation is shown in Tables 9.6 to 9.7. Table 9.5 shows assumed economic factors like discounting rate, fuel and electricity escalation rates, etc. Table 9.5 shows the annual costs, escalated year by year over the project life and equated to the present worth, and Table 9.6 provides a summary of the LCC calculation. In a sample calculation as shown in Table 9.6, the total present cost is \$49,378,164 which is the sum of the \$12,000,000 project cost and the \$37,378,164 NPV of the annual costs.

	Annual Cost					
Year	Maintenace Cost (\$)	Admin. & Permit Cost (\$)	Fuel Cost (\$)	Electricity Cost (\$)	Total Annual Costs (\$)	Present Worth (\$)
1	500,000	50000	1,500,000	450,000	2,500,000	2,380,952
2	515,000	51500	1,530,000	459,000	2,555,500	2,317,914
3	530,450	53045	1,560,600	468,180	2,612,275	2,256,581
4	546,364	54636	1,591,812	477,544	2,670,355	2,196,908
5	562,754	56275	1,623,648	487,094	2,729,773	2,138,848
6	579,637	57964	1,656,121	496,836	2,790,558	2,082,358
7	597,026	59703	1,689,244	506,773	2,852,745	2,027,393
8	614,937	61494	1,723,029	516,909	2,916,368	1,973,912
9	633,385	63339	1,757,489	527,247	2,981,459	1,921,875
10	652,387	65239	1,792,639	537,792	3,048,056	1,871,242
11	671,958	67196	1,828,492	548,547	3,116,193	1,821,974
12	692,117	69212	1,865,061	559,518	3,185,909	1,774,033
13	712,880	71288	1,902,363	570,709	3,257,240	1,727,384
14	734,267	73427	1,940,410	582,123	3,330,226	1,681,991
15	756,295	75629	1,979,218	593,765	3,404,908	1,637,819
16	778,984	77898	2,018,803	605,641	3,481,325	1,594,835
17	802,353	80235	2,059,179	617,754	3,559,521	1,553,007
18	826,424	82642	2,100,362	630,109	3,639,537	1,512,303
19	851,217	85122	2,142,369	642,711	3,721,418	1,472,692
20	876,753	87675	2,185,217	655,565	3,805,210	1,434,144
Total	13,435,187	1,343,519	36,446,055	10,933,816	62,158,577	37,378,164

Table 9.6 Sample Life Cycle Cost Analysis (LCC)

Note that the sum of the annual costs is nearly \$62 million, but the future value is discounted to the amount shown in Table 9.6. The total present costs of an alternative would then be compared with that of other alternatives to determine the alternative with the lowest LCC.

Project Cost (\$)	NPV of Annual Cost (\$)	Total Present Cost (\$)
12,000,000	37,378,164	49,378,164

Table 9.7 Sample Life Cycle Cost Analysis (LCC)

Note that the sum of the annual costs is nearly \$49 million, but the future value is discounted to the amount shown in Table 9.7. The total NPV of an alternative would

then be compared with that of other alternatives to determine the alternative with the lowest LCC.

#### Summary

As with any major engineering project, the CHP system planning and design stage needs a detailed technical and economic feasibility study to evaluate the feasibility of going ahead with the project. In this chapter, such a feasibility study along with various economic analysis tools are presented to aid practising industry professionals currently dealing with or who may deal with CHP systems in future.

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# **10.0 THE CHP REGULATORY ISSUES IN SINGAPORE**

This chapter provides detailed information on the regulatory framework one should follow in the planning and implementation stages of Combined Heat and Power (CHP) Systems in Singapore.

#### Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. The registration of different types of CHP systems
- 2. The basic terminologies associated with the regulatory requirements
- 3. The Singapore electricity market
- 4. The various local authorities involved in the CHP system registration

#### **10.1 Introduction**

Prior to the planning, design and implementation of a CHP system, it is very important to have a good understanding of various local regulatory requirements in Singapore as far as CHP systems are concerned. Singapore's national electricity grid is highly interconnected and complex, and regulatory requirements are also stringent for gridconnected CHP systems. The important CHP related regulatory requirements are controlled by EMA, EMC and NEA, etc. In this Chapter, such regulatory requirements are described for practising engineers to have a better understanding of the relevant regulatory requirements pertaining to CHP systems.

#### 10.2 Basic Terminologies and Key Players in the Singapore Electricity Market

In this section, a brief account of the basic terminologies and the key players in the Singapore electricity market are provided.

#### **Basic terminologies**

- 1. Co-generation: Simultaneous generation of heat and electrical power
- 2. CCGT: Combined Cycle Gas Turbine
- 3. Dispatch: The process by which generation is coordinated in real time to meet demand
- 4. PSO: Power system operator. In Singapore, the power system operator is the Energy Market Authority of Singapore (EMA)

- 5. Dispatchable generator: A generator that is capable of following dispatch instructions from the PSO
- 6. Dispatch Period: A thirty-minute time interval beginning on the hour or the halfhour during which dispatch is being effected
- 7. Node: Any of the injection or exit points on the transmission system in the market model
- 8. Nodal price: An electricity price at a specific location
- Real-time dispatch: A schedule determined by the Energy Market Authority that contains the quantities of energy, reserve and regulation scheduled in respect of a registered facility
- 10. EMA: Energy Market Authority, the PSO in Singapore
- 11. EMC: Energy Market Company
- 12. Real-time market: The wholesale electricity markets operated by the EMC for energy, reserve or regulation
- 13. Regulation: In relation to a generating unit, the frequent adjustment to its output so that any power system frequency variations or imbalances between load and the output from generation facilities can be corrected
- 14. Regulator: The entity that has regulatory oversight over the Singapore electricity market (the EMA)
- 15. Reserve: Spare capacity that can be used quickly to maintain supply when there is mismatch between supply and demand
- 16. WEQ: Withdrawal Energy Quantity from the grid
- 17. IEQ: Injection Energy Quantity into the grid
- 18. USEP: Uniform Singapore Energy Price

#### 19. HEUC: Hourly Energy Uplift Charges

#### 20. MEP: Market Energy Price

The size of the CHP systems and their corresponding categorisation is tabulated as follows:

CHP Capacity	Category
< 1MW <sub>e</sub>	Small
1 to 10 MW <sub>e</sub>	Medium
> 10 MW <sub>e</sub>	Large

Table 10.1: CHP system categorisation based on the size

#### 10.3 Key Players in the Singapore Electricity Industry

#### 10.3.1 The Market Operator

The Energy Market Company (EMC) is the electricity market operator in Singapore. The EMC is a joint venture between the EMA and M-Co (The Marketplace Company) Pte Ltd, a wholly-owned subsidiary of M-Co International Ltd of New Zealand.

#### 10.3.2 The Power System Operator

The Energy Market Authority is the Power System Operator (PSO) in Singapore. As the PSO, the EMA ensures the reliable supply of electricity to consumers and the secure operation of the power system. The EMA as the PSO also controls the dispatch of generation facilities in the National Electricity Market of Singapore (NEMS). In addition, the EMA coordinates outage and emergency planning, and directs the operation of Singapore's high-voltage transmission system.

#### 10.3.3 The Transmission Licensee

SP Power Assets is the Transmission Licensee in Singapore. It owns the national power grid in Singapore. SP Power Assets has appointed SP Power Grid as the transmission agent to operate and maintain the Singapore power grid.

#### 10.3.4 The Generation Licensees

Singapore's three largest electricity generating companies are Senoko Power, Power Seraya and Tuas Power. There are other electricity generation companies as well, who contribute to the Singapore electricity market with their smaller capacity generators. More information on the Singapore power generators and their respective generation capacities will be described in the next section of this chapter. As per the Singapore regulations, any company that generates electricity needs a generation licence if one or more generating units have an individual nameplate rating of 10 MW or above. If the generation system is connected to the national power grid, the generating unit(s) must be registered with EMC and will have to compete to secure dispatch in NEMS.

#### 10.3.5 The Wholesaler (Generation) Licensees

In Singapore, if a power generation company has a generating unit with an individual nameplate rating of less than 10 MW but more than 1 MW, and if that generator is connected to the grid, it will need a wholesaler (Generation) licence.

A licence is also needed if a company has a generating unit whose nameplate rating is less than 1 MW but wishes to sell this electricity to the grid.

#### 10.3.6 The Wholesaler (Interruptible Load) Licensees

The wholesaler interruptible load licensees are the electricity consumers or companies that provide services to other consumers who are willing to offer their loads to be interrupted. In Singapore, all the wholesaler interruptible load licensees need to have a wholesaler interruptible load licence.

#### 10.3.7 The Market Support Services Licensee (MSSL)

Singapore Power (SP) Services is the Market Support Services Licensee (MSSL) in Singapore. As an MSSL, SP Services provides market support services such as retail settlement, meter reading and meter data management, consumer registration, and transfer processing for contestable consumers who switch from one retailer to another. The contestable consumers, classified based on their annual electricity consumption, are those who are entitled to choose their electricity retailer. In addition, SP Services also provides the following:

- 1. Indirect access to the NEMS by contestable consumers who have not appointed a retailer
- 2. Supply of electricity to non-contestable consumers at regulated tariffs
- 3. Providing of billing and payment collection of charges for use of the power grid on behalf of the Transmission Licensee, SP Power Assets

#### 10.3.8 The Electricity Retail Licensees

In Singapore there are two types of licensed electricity retailers, namely market participant retailers (MPRs) and non-market participant retailers (NMPRs).

MPRs have to be registered with EMC to purchase electricity from the NEMS to sell that electricity on to contestable consumers.

NMPRs do not have to register with EMC to participate in the NEMS since they purchase electricity indirectly through SP Services, the MSSL.

#### **10.4 Singapore Electricity Market and Electricity Generators**

The structure of Singapore's electricity market is illustrated in Figure 10.1 below. As can be seen from the Figure, the main players include generation companies, the power grid operator, the power system operators, the electricity market operator, the market support services licensee and the electricity retailers.



Figure 10.1 The Singapore electricity market structure

The Singapore electricity market settlement is illustrated in Figure 10.2 below. As can be seen from the Figure, the Energy Market Company (EMC) plays a central role in the Singapore electricity market settlement.



#### 10.5 The Electricity Generators in Singapore

The electricity generating companies in Singapore and their generation capacities are tabulated below (as of January 2019):

Generating Companies	Generation Capacity (MW)
Senoko Energy	3,300
YTL Power Seraya	3,100
Tuas Power Generation	2,670
Sembcorp Cogen	1,215
Keppel Merlimau Cogen	1,340
National Environment Agency	178
Keppel Seghers waste-to-energy plant	22
Senoko waste-to-energy plant	55
Shell Eastern Petroleum	60
Exxon Mobil Asia Pacific Pte Ltd	633
PacificLight Power (formerly GMR /	800
Island Power)	

Table 10.2 Generation companies and their installed capacities in Singapore

The total installed capacity in Singapore is about 13 GW. Considering Singapore's peak demand during weekdays and weekends as shown in Figures 10.3 and 10.4, respectively, Singapore has surplus electricity generation capacity.







Figure 10.4 The typical weekend energy demand profile in Singapore

# **10.6 Factors Affecting Co-generation Plant Development**

The various factors affecting co-generation plant development in Singapore are listed below, followed by a brief account of each factor.

The factors considered:

- Plant or Factory Requirements
- Energy Balance
- Availability of Fuel
- Location of Co-generation plant (Availability of Land)
- Operating Mode Grid-Synchronised or Island mode
- Economic Feasibility
- Regulatory requirements
- Fault Level Considerations

# 10.6.1 Plant or Factory requirements

Understanding the type of business helps provide insight into the economic feasibility of operating a CHP plant. Generally, businesses with the following characteristics are good candidates for CHP applications in Singapore.

- Continuous and steady operation (in service for 24 hours)
- Have a high heat load requirement relative to electrical load
- Typical annual operational hours exceed 85% (i.e. > 7446 hours)

A typical power consumption profile of an industrial set-up is shown in Figure 10.5 below.



Figure 10.5 Typical power consumption profile

When a CHP plant is going to be installed in a greenfield kind of facility, the facility may not have adequate infrastructure and availability of backup utility supply at the beginning. The advantages and disadvantages when a CHP system is planned in such a greenfield set-up are as follows:

Advantages:

- Infrastructure for interfacing is planned or made ready, e.g. connection to the grid takes fault level contribution into account
- Disruptions to operations are minimal
- Operational savings achieved from start of operations (in an optimised system)

Disadvantages:

- Lack of historical utility demand data over-design of systems could occur
- Construction team have different objectives (capacity over optimal operation)
  e.g. electrical demand projected based on Maximum Power Rating
- Sunk cost of new equipment has to be factored into cost benefit study

On the other hand, when a CHP plant is going to be installed in a brownfield kind of facility, the facility may have adequate infrastructure and availability of backup utility supply in the beginning. The advantages and disadvantages when a CHP system is planned in such a brownfield set-up are as follows:

# Advantages:

- Historical data is available for analysis design with confidence
- Utility assets are close to full depreciation, or need to be replaced
- Plant improvements can be incorporated into the design

Disadvantages:

- Plant operations may be momentarily disrupted, e.g. during the tie-in between the CHP system and existing plant systems
- Lack of available or suitable space
- Substantial modifications (including hot taps) may have to be made
- Brownfield construction cost is high

The typical requirements for electrical power and heat for the various sectors in Singapore are tabulated as follows:

	Types of Industry (in Singapore)					
Typical Requirements	Retail	Hospitality / Building Svcs	Chemical / Pharmaceutical	Semi- conductor		
Power	< 1MW	< 3 MW	> 5 MW	> 10MW		
Heat (Steam)	0 MW	< 3MW	> 8 MW	< 3 MW		
Good CHP Candidate?	No	Marginal - with absorption chilling	Yes	Marginal - with absorption chilling		

Table 10.3 Typical CHP heat to power ratio for industries in Singapore

As can be seen from the Table, good CHP candidates are those with a fairly good heat to power ratio. This suggests that when the waste heat produced could be well utilised for their heating applications, the CHP system becomes optimum.

It is to be noted that even within similar industries, utility requirements may differ widely. One good example is the common trend in the pharmaceutical industries where the Active Pharmaceutical Ingredient (API plants) are better CHP candidates than the Biologics plants.

Consideration of the current method of energy procurement is also important in the planning stage of the CHP system. For instance, currently the electricity is from the electrical grid, and the heat requirement from the boiler, heat pump and cooling requirement is from the electrical chillers or absorption chillers. Some industries (such as in food processing) burn their spent or waste feed stocks (such as cocoa shells) in

biomass boilers to make steam or hot water. Compared to gas or diesel-based boilers, their fuel costs \$0. In such a scenario, their fuel cost will be zero. Factors such as this also need to be considered in the planning stage of the CHP system.

Some industries obtain their cooling/heating requirement by connecting to a District Cooling System (DCS), but these are usually seen in commercial buildings and large plant complexes. The bottom-line is: various methods of energy procurement will affect the economic considerations of implementing a CHP system.

The motivation of industrial players for the development of a co-generation system in their industrial premises may differ as follows:

- Competitors have implemented such systems
- Overseas plants have implemented similar systems, why not here?
- News reports indicate that CHP systems make sense
- Our electricity demand is very large
- We need to reduce our carbon output

However, the economic feasibility of a co-generation plant can only be established by performing a Cost Benefit Analysis (CBA).

A CBA is a comparison of current energy procurement costs versus the costs incurred by a potential CHP system. It provides a snap shot of the various cost issues at any point in time and gives insight into the feasibility of the potential CHP development.

Load consumption pattern also plays an important role in deriving the economic benefit from a CHP system. For example, a steady continuous consumption pattern is desirable when considering CHP system implementation for the following reasons:

- It is preferable for certain systems, such as gas turbines, to operate at an economically optimal load (generally > 80% running load)
- Below the optimal load condition, the economic benefits of running a CHP system diminishes
- The payback period for the investment will be shorter
- From the system viewpoint, discontinuous sub-optimal loading creates stress, which may shorten the life of the machine

• The efficiency of the CHP system also decreases with sub-optimal heat to power ratios (due to the discharge of unused heat into the atmosphere)

#### 10.6.2 Operating hours

The number of operating hours is a contributing factor to the economic evaluation when deciding whether to implement a CHP system.

The operating hours of the industrial plant affects how the CHP system will be scheduled for maintenance. Plants that produce different products will see differing utility loads, This will also affect the design at which the operating load of the CHP system is optimal, as well as the operation strategy. As far as the load consideration for the CHP system design is concerned, a reliable monthly electrical load profile needs to be established as shown in Figure 10.6. From the monthly load profile, the continuous load and the CHP base load can be established. The CHP design is based on the baseload in order to avoid the part-load operations of the CHP system.



Figure 10.6 Example of monthly electrical load profile

# 10.6.3 Critical load consideration

The distinction between critical load and non-critical load affects the way the grid connection is made, as well as the type of contingencies and level of contracted capacity required.



Figure 10.7 Connection of critical and non-critical loads to the grid

It would make a clear distinction if the CHP system can be connected to non-critical loads. This could reduce the contracted capacity significantly. It is to be noted that in some cases, island mode operation (a CHP system operation without connecting to local grid) may be considered by the facility owner.

On the other hand, it will not make any clear distinction if the grid connection is necessary for backup purposes. In such situations, the contracted capacity for the industrial plant may not be reduced or just partially reduced.

# 10.6.4 Energy Balance

Energy balance or heat to power ratio, as the name suggests, is the ratio of a plant's heat requirement to the electrical power requirement. Generally, heat to power ratio varies based on the application. Typically, a CHP system is running optimally as far as its operations are concerned if the heat to power ratio is 1.7 or more.

It is to be noted that most plants with CHP systems installed are net importers of electricity. The heat requirements of any plant or facility can be boosted through supplementary firing.

Example for Heat to Power Ratio Heat required: 7.5 MW (approximately equivalent to 10 tons of saturated steam) Power required: 4.5 MW Heat to Power ratio: 1.67

Experience suggests that typical industries in Singapore with high heat requirements are:

- Pharmaceutical plants
- Chemical industry
- Petro-Chemical industry
- Food processing industry

Generally, the industries with CHP systems export excess electricity to the grid only under special circumstances. Some of the circumstances are as follows:

- Excess process fuel that needs to be consumed, for example:
  - o a chemical plant's process waste gas
  - special purpose bio-digestion plants
- Where the co-generation plant is the primary heat generator, for example:
  - $\circ$  a refinery
- Where the factory/plant has to be split into different sites due to space constraints (subject to EMA's approval), for example:
  - The company operates two sites that are on separate plots, but both sites are majority owned by the company. Also, the company must demonstrate that the reason for the two different plots was due to a lack of space

#### 10.6.5 Availability of fuel

The availability of a reliable premium fuel such as natural gas is one of the important factors in the efficient and seamless operation of a CHP system.

Some of the fuel types most commonly available and used - or planned for use - in co-generation plants in Singapore are:

- Natural Gas
- Process / Waste Gas / Waste Oil
- Bio-gas
- Biomass
- Diesel / Renewable diesel
- Town gas

In Singapore, natural gas is available in the Western and Northern areas. Currently, most of the power generation, about 95%, uses natural gas. In addition, all the cogeneration plants in Singapore run on natural gas. With the commissioning of the Liquefied Natural Gas (LNG) terminal in Jurong Island in 2013, the natural gas availability in Singapore is highly stable and reliable, with the power generation companies largely benefitting from it. In Singapore, the Natural Gas price is tied to HSFO price in Asia and converted based on calorific value in mmBTU.



Figure 10.8 Natural Gas transmission network in Singapore (Reference 5)

The factors that need to be considered when using Natural Gas include gas off take pressure (depending on the location, technology and availability). The current (as of May 2019) average natural gas tariff in Singapore for large consumers is about S\$16/mmBTU.



Figure 10.9 High pressure and low pressure Natural Gas lines in Singapore (Reference 5)

#### 10.6.6 Location of Co-generation plant (Availability of Land)

Another important factor that needs to be considered while implementing a CHP system in an industrial set up is the space requirement. While allocating the space for the system, maintenance of the CHP in terms of easy and safe access to the CHP equipment should also be considered. The recommended distance allowance from nearby obstructions and buildings is shown in Figure 10.10. Any shortcomings in such stipulated distances need to be readjusted by consulting with the CHP supplier and the relevant authorities.



Figure 10.10 Typical CHP system installation distance guideline

The height of the CHP stack needs to be in compliance with the National Environment Agency's (NEA) guidelines (refer to the NEA web portal <u>www.nea.gov.sg</u> for more
information). The stack height requirement for the CHP system installation is shown in Figure 10.11.

The NEA Code of Practice on Pollution Control for fuel-burning equipment also needs to be complied with during the CHP implementation. The NEA Code of Practice on Pollution Control for emission control of fuel burning states that: A chimney of an approved height should be provided for safe dispersion of flue gases from fuel burning equipment. The design chimney height computed from the SO<sub>2</sub> emission calculation shall not be lower than 3m from roof of factory or 15m from ground level, whichever is higher.



Figure 10.11 Typical CHP system stack height guideline

A CHP plant should be situated near to the utility building. It is generally better to site CHP plants close to the point where the steam interface occurs, to reduce the complexity and distance of the steam and condensate pipes, as well as other services (such as instrument air).



Figure 10.12 Recommended CHP system installation near the utilities

While planning the CHP system installation, the noise emitted from the CHP system needs to be addressed. The noise emission data in decibels can be obtained from the CHP engine supplier and that should be compared with the noise level permitted by the authorities. If the noise level exceeds the level allowed by the local authorities, some kind of remedial actions to mitigate the noise level must be considered to reduce it to the acceptable level. In addressing the noise emission through various strategies, be mindful about the following:

- Sound propagation to surrounding buildings and public roads
- Proximity to buildings nearby has the effect of amplifying high frequency noise

The noise emission of a CHP plant and amplification of the noise is illustrated in Figure 10.13 below.



Figure 10.13 CHP system noise emission and propagation

The other factors that need to be considered are:

- Preferable to site away from tall buildings / Hazardous work areas
- If the CHP Plant is sited in a Class 1 area, equipment, pipes and cables may have to be explosion-proof
- The CHP Plant stack height will have to be taller than nearby buildings, and additional height adds to costs

A CHP plant usually requires space for the following:

- Selected prime mover
- Heat recovery system / blowdown sump

- Control room
- MCC room
- Transformers
- Water Treatment / Chemical dozing skid (for greenfield facility)

Industrial experience suggests that a 5MW Gas Turbine Driven CHP plant will require approximately 30m x 20m of space.

#### 10.6.7 Operating Mode – Grid-Synchronised or Island mode

Co-generation plants are usually synchronised to Grid (50Hz). Island mode is considered only when the loads are not critical. In industry, co-generation plants usually provide for base load, with the grid providing residual and backup supply. To ensure reliable seamless power supply for industrial applications with critical load, the industry contracts a particular capacity (kW) from the utility, called contracted capacity. In Singapore, there are various schemes available for Contracted Capacity.

Three models for the declared contracted capacity are shown in Figures 10.14 and 10.15. In the model shown in Figure 10.14, there is one tri-generation system producing B MW of power and A MW is imported from the grid. The total requirement for the load is (A+B) MW and the declared capacity is A+B MW. This model is operating without a load limiter switch. In the right-hand side model shown in Figure 10.14, the difference is on the declared capacity. This model declares only A MW assuming the load limiter will do the load shedding within the allotted time frame. The allowable capacity in this model is 20% more than the declared capacity of A MW.

Figure 10.15 shows a model with two tri-generation systems with each producing 1/2B MW. The declared capacity is only (A+1/2B) MW despite the load requirement of (A+B) MW. In this model, the assumption is that at any time only one generator will fail. As in the case of the previous model, the allowable capacity that can be drawn from the grid is 20% more than the declared (A+1/2B) MW, based on the condition that the load limiter will kick in and perform the load shedding within the allotted time frame.







single generator

Allowable Max Demand = (A+1/2B)\*1.2 MW

Figure 10.15 Contracted capacity scheme with load limiter and two generators

#### **10.6.8 Contracted Capacity Schemes**

The three contracted capacity schemes are summarised below:

- 1. Summation Scheme (SS)
- (i) Declare Contracted Capacity
- (ii) Maximum Demand = Import + Generation

#### 2. Capped Capacity Scheme (CCS)

- (i) Grid committed to provide capacity requested by customer
- (ii) Tripping requirement at customer's switchboard
- (iii) Customer installs Load Limiting Device (LLD)

120% of Contracted Capacity

Given 10 sec to manage load shedding



#### **Capped Capacity Scheme**



3. Extended Capped Capacity Scheme (ECCS)

Tripping requirements (for activation of tripping device) as follows:

 120% of the Contracted Capacity (CC) for more than 100 seconds; or 200% of CC for more than 10 seconds

The penalty for exceeding the allotted time frame for the load shedding is as follows:

- 5 times CC charge for demand exceeding 120% and up to 200% of CC
- 12 times CC charge for demand exceeding 200% of CC



Figure 10.17 Extended Capped Capacity Scheme (courtesy of EMA)

Energy	Conventional	CHP (with grid support)	CHP (Island mode)
Peak Off Peak	イイ	x	x x
Market related charges PSO EMC AFP MEUC HEUC	イント	√ Net √ Net √ Gross √ Net √ Net	X X X X X
Use-of-System Peak Off peak Contracted Capacity Uncontracted Capacity Reactive Power	イント	X X V V	X X X X X X
MSSL Charges Meter reading MSS basic services Billing & collection Retail market system related charge	イン	√ Net √ Net √ Net √ Net	X X X X

#### Legend:

√ Applicable

x Not Applicable

Energy costs based solely on fuel consumption

Substantial savings from eduction in UOS

Figure 10.18 Various charges for CHP with and without grid synchronization

#### 10.6.9 Fault Level Considerations

The fault level is measured in current or MVA. Current = Voltage divided by Resistance. When a short circuit occurs (low resistance), a large current flows. This causes the operation of protective devices (fuses and circuit breakers open) to isolate the fault.

The amount of Fault Current depends on the following parameters:

- Voltage
- Capacity of Transformer (higher capacity, higher fault current)
- Impedance (lower impedance, higher fault current)

Types of fault can be one of the following:

- 3 Phase
- Phase to Phase
- Single Phase to Ground

Protection systems are designed to isolate only the faulty section of network. If the fault current rating is too high, it may not be interrupted. If the fault current rating is too low, the protective devices may not operate.

Electrical Switchgear is designed to withstand normal and fault current E.g. 25kA, 3 sec for 22kV, 65kA, 3 sec for 400V, etc.

Circuit Breakers are designed to interrupt normal and maximum fault current. E.g. 25kA for 22kV, 65kA for 400V, etc.

If the actual fault current is higher than the equipment designed rating, the equipment will fail. Singapore's electrical network is interconnected to improve reliability and availability. This will result in an increase in fault current contribution.

Every transformer has "%" impedance value stamped on the nameplate. It is stamped because it is a tested value after the transformer has been manufactured. The test is as follows: A voltmeter is connected to the primary of the transformer and the secondary 3-Phase windings are bolted together with an ampere meter to read the value of current flowing in the 3-Phase bolted fault on the secondary. The voltage is brought up in steps until the secondary full load current is reached on the ampere meter connected on the transformer secondary.

**Example 1:** Determine the % impedance stamping for a 1000KVA 13.8KV – 480Y/277V transformer *Solution* 

Determine the transformer Full Load Amps (FLA)  $FLA = KVA / 1.73 \times L-L KV$   $FLA = 1000 / 1.732 \times 0.48$  FLA = 1,202.85The 1000KVA 480V secondary full load ampere is 1,202A. When the secondary ampere meter reads 1,202A and the primary Voltage Meter reads 793.5V. The percent of impedance value is 793.5 / 13800 = 0.0575. Therefore, % Z = 0.0575 \times 100 = 5.75% This implies that if there was a 3-Phase Bolted fault on the secondary of the transformer then the maximum fault current that could flow through the transformer would be the ratio of 100 / 5.75 times the FLA of the transformer, i.e.  $17.39 \times 10^{-2} \times 10^{-2} \times 10^{-2} \times 10^{-2}$ 

based on the infinite source method at the primary of the transformer. A quick calculation for the Maximum Fault Current at the transformer secondary terminals is:

FC = FLA / %PU Z FC = 1202 / 0.0575 = 20,904A

This quick calculation can help to determine the fault current on the secondary of a transformer for the purpose of selecting the correct overcurrent protective devices that can interrupt the available fault current. The main breaker to be installed in the circuit on the secondary of the transformer has to have a KA Interrupting Rating greater than 21,000A. Be aware that feeder breakers should include the estimated motor contribution too. If the actual connected motors are not known, then assume the contribution to be  $4 \times FLA$  of the transformer. Therefore, in this case the feeders would be sized at 20.904 + ( $4 \times 1202$ ) = 25,712 Amps.

**Example 2:** Determine the Fault Level Current (FLC) for the transformers shown in the Figure below: *Solution* 

FLA = KVA / 1.73 x L-L KV FLA = 1000 / 1.732 x 0.4 FLA = 1,443.4 A FLC = FLA x 100/6 = 1,443.4 x 100/6 = 24,056 A = **24 kA** For the circuit on the right-hand side, the FLC = 24 kA x 2 = **48 kA** 



Figure 10.19 Fault level current

#### 10.6.10 Regulatory requirements

The genset, the land on which the on-site load and genset are situated, and the facility constituting the load, must be majority owned by the same company. The genset and load must be located on the same contiguous plot. Generally, cable between genset and load cannot be connected between two separate plots of land. The company must hold an electricity license. The genset shall be registered at the wholesale market according to the Market Rules.

CHP capacity	Registration	Characteristics
(MW)		
(1117)		
≥ 10 MW	Registered as generation facility	- Centrally scheduled - dispatched
		based on "offers" into market
		- Metered on a half hour basis
		- Settled by wholesale market - based
		on nodal energy price
1- 10 MW	Registered as generation settlement	- Self-Scheduled (but may choose to be
	facility	centrally scheduled)
		- Metered on a half hour basis
		- Paid based on nodal energy price
≤ 1 MW	Not Registered or Settled in	- Self-scheduled
	Wholesale Market	- Settled in retail market - based on
		monthly meter readings
		- Effectively receive Uniform Singapore
		Energy Price

In Singapore, the registration of the co-generation facility is done as tabulated below:

Table	10.4:	CHP	system	registration	categories

#### **10.7 Treatment of Embedded Generators**

'Embedded generation' means generators producing power principally for internal purposes. While this includes on-site generation on the same premises owned by the same party as a large load, various other ownership and locational structures will also be considered.

Such a CHP system would typically use an external fuel source (e.g. gas or distillate). 'Embedded co-generation' refers to embedded generation that provides electricity as well as other products such as process steam, hot water and/or chilled water. Such generation may use an external fuel source (e.g. gas or distillate) or within plant fuel (e.g. non-saleable refinery product).

#### 10.7.1 Benefits of Embedded Generation

Many of the benefits of embedded generation are recognised in the regulatory arrangements.

However, there are some areas where the benefits of embedded cogenerators in particular are not reflected in the charges they pay or receive. The key areas where the NEMS does not reward embedded cogenerators for the benefits they could potentially provide are: Greenhouse gas emission reductions and fuel diversity during a gas crisis.

The potential options to recognise these benefits are as follows:

Implement a more comprehensive GHG reduction scheme such as:

- Tradable permits or carbon tax;
- Provide an explicit subsidy for co-generation
- Offer reduced license fee for co-generation
   Offer limited 'net' treatment of EMC, PSO, MEUC and MSS charges (depending on value of GHG & fuel diversity benefits)
   The current EMA policy is a 'gross' treatment on the embedded generators as far as the above charges are concerned. For example: a 30 MW embedded generating company having an onsite demand of 50 MW draws the remaining 20 MW from the grid. But, the EMC, PSO, MEUC and MSS charges are based on 50 MW instead of a net treatment of 20 MW.

EMA policy on the treatment of embedded generators is as follows:

- Reserve charges: Embedded generators should continue to pay charges for reserve and regulation as per current treatment ('gross' treatment)
- Non-reserve charges: Embedded generators will be granted net treatment on non-reserve charges, provided they will not export to the grid

#### 10.7.2 General Comments on Embedded Generators

- Embedded cogenerators provide several benefits to the system including energy efficiency, lower emissions and supply security
- Therefore, efficient embedded co-generation should be encouraged but the current 'gross load' treatment penalises embedded co-generation – the NEMS is one of the few markets that applies a gross treatment

#### 10.7.3 EMC directed by EMA

- Embedded generation should also not be required to be centrally dispatched
- EMC to modify the Market Rules for net treatment of embedded generators
- EMC to implement the net treatment of EMC fees, PSO fees and MEUC for embedded generators
- EMC to enable price neutralisation to be performed if the market participant with embedded generation facilities (who has granted authorisation for price neutralisation by EMA) purchases electricity via a retailer

#### 10.7.4 Nodal Price Neutralization

The Singapore wholesale electricity market pays generators at the generator's nodal price (MEP) and charges loads USEP plus HEUC. USEP is a volume weighted average nodal price at all consumption nodes. When constraints occur, any one generator's nodal price could vary significantly from USEP plus HEUC. Thus, an embedded generator could be paid a price for its generation which is substantially different from the price charged for its consumption. This exposes it to nodal price risk.

#### Example:

Under the Market Rules, one settlement account is assigned to a group of embedded generators.

To calculate the amount of neutralisation adjustment for each such settlement account, there are separate 2 situations: (For simplicity, we use an example where there is only one embedded generation facility in an embedded generator group.)

**Case 1:** Associated Load (WEQ) is greater than or equal to Generation (IEQ) Here, the embedded generator has withdrawn more than or equal to its injection. Before neutralization, it would have been charged (USEP+HEUC)\*WEQ for withdrawal and paid MEP\*IEQ.

With neutralization, it should receive a credit of IEQ \* (USEP+HEUC - MEP).

#### Case 2: Associated Load (WEQ) is less than Generation (IEQ)

Here, the embedded generator has injected more than its withdrawal. With neutralization, it should receive a credit of WEQ \* (USEP+HEUC – MEP).

#### **10.8 CHP Registration Procedures in Singapore**

Preliminary: Submit Official Letter to EMA informing of the intention to install embedded generator

After Approval from EMA, the various steps that need to be followed are tabulated below:

St	teps	Activity Ar							
1									
	1.1	Submit Consultation form to connect to transmission system	PGrid						
	1.2	Consultation with PG							
2									
	2.1	Submit Application form to connect to transmission system	PGrid						
	2.2	Submit gas connection application	PGas						
	2.3	Submit application for wholesaler license (generation)	EMA						
	2.4	Submit PSO/MP Agreement	PSO						
3									
	3.1	Check - Documents 2.1, 2.3 and 2.4 approved or signed							
4									
	4.1	2nd Consultation with PG (power quality, protection, etc)	PGrid						
	4.2	Submit application for Market participant registartion	EMC						
	4.3	Consultation with PSO	PSO						
5									
	5.1	Sign Supplementary agreement to connection agreement on metering charges	PGrid						
_	tone	Activity	A						
St	lehs	Activity	Agency concerned						
<b>S</b> 1 6	leps	Activity	Agency concerned						
6	6.1	PGrid meter section will issue Meter ID	Agency concerned PGrid						
6	6.1 6.2	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement)	PGrid EMC						
6 7	6.1 6.2	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement)	PGrid EMC						
51 6 7	6.1 6.2 7.1	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement	PGrid EMC MSSL						
51 6 7	6.1 6.2 7.1 7.2	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements	Agency concerned PGrid EMC MSSL PGrid						
51 6 7	6.1 6.2 7.1 7.2 7.3	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement	Agency concerned PGrid EMC MSSL PGrid EMC						
51 6 7	6.1 6.2 7.1 7.2 7.3 7.4	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements	Agency concerned PGrid EMC MSSL PGrid EMC PSO						
51 6 7 8	6.1 6.2 7.1 7.2 7.3 7.4	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements	Agency concerned PGrid EMC MSSL PGrid EMC PSO						
Si           6           7           8	6.1 6.2 7.1 7.2 7.3 7.4 8.1	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements Check - Documents 4.2 approved (Settlement ID issued officially)	Agency concerned PGrid EMC MSSL PGrid EMC PSO EMC						
St           6           7           8           9	6.1 6.2 7.1 7.2 7.3 7.4 8.1	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements Check - Documents 4.2 approved (Settlement ID issued officially)	Agency concerned PGrid EMC MSSL PGrid EMC PSO EMC						
St           6           7           8           9	6.1 6.2 7.1 7.2 7.3 7.4 8.1 9.1	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements Check - Documents 4.2 approved (Settlement ID issued officially) Arrange for PGrid to install Generation meters	Agency concerned PGrid EMC MSSL PGrid EMC PSO EMC PGrid						
St           6           7           8           9	6.1 6.2 7.1 7.2 7.3 7.4 8.1 9.1 9.2	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements Check - Documents 4.2 approved (Settlement ID issued officially) Arrange for PGrid to install Generation meters Submit price neutralization for embedded generators	Agency concerned PGrid EMC MSSL PGrid EMC PSO EMC PGrid EMC						
St           6           7           8           9           10	6.1 6.2 7.1 7.2 7.3 7.4 8.1 9.1 9.2	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements Check - Documents 4.2 approved (Settlement ID issued officially) Arrange for PGrid to install Generation meters Submit price neutralization for embedded generators	Agency concerned PGrid EMC MSSL PGrid EMC PSO EMC PGrid EMC						
St           6           7           9           10	6.1 6.2 7.1 7.2 7.3 7.4 8.1 9.1 9.2 10.1	PGrid meter section will issue Meter ID EMC will issue temporary settlement ID to MSSL (for use in MSSL/MP agreement) Sign MSSL/MP Agreement Ensure compliance with PGrid's requirements Submit signed MSSL/MP agreement Ensure compliance with real time monitoring and load forecast requirements Check - Documents 4.2 approved (Settlement ID issued officially) Arrange for PGrid to install Generation meters Submit price neutralization for embedded generators Check - Documents 9.2 approved	Agency concerned PGrid EMC MSSL PGrid EMC PSO EMC PGrid EMC PGrid EMA						

S	teps	Activity	Agency concerned
11			
	11.1	Submission of letter of undertaking by <b>LEW</b> to Power Quality section and distribution planning	PGrid
	11.2	Check - 9.1 (Meters installed)	PGrid
	11.3	Submit information for Generation account set up	MSSL
	11.4	Submit application for registration of comissioning generation facility	EMC
12			
	12.1	Ensure Generation MSSL account set up	MSSL
	12.2	Check - Document 2.2 approved	PGas
	12.3	Check - 7.4 conditions fulfilled	PSO
	12.4	Check - Document 11.4 approved	EMC
	12.5	Check Generation MSSL account activated	MSSL
13			
	13.1	Comissioning of plant	Owner
14			
	14.1	Arrange for PGrid to oversee installation of LMD (and LLD - if opting for ECCS)) equipment	PGrid
	14.2	Submit application to register Generation Settlement Facility	EMC

Table 10.5 Typical CHP registration flow diagram in Singapore

#### Summary

Understanding the Singapore regulatory framework to register a CHP system in the Singapore electricity market is important for industry professionals taking care of the CHP systems. In this chapter, the Singapore electricity market and the various players in the electricity market are presented. The various regulations for the registration of a grid synchronised CHP system is also presented.

#### References

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- Energy Market Authority (EMA) web page, URL: <u>https://www.ema.gov.sg/index.aspx</u> (Last accessed on 16 May 2019)
- 4. Singapore Power web page
- 5. Singapore Power Gas web page, http://www.powergas.com.sg
- Singapore Certified Energy Manager (SCEM) Lecture Notes by LJ Energy Pte Ltd

### 11.0 COMBINED HEAT AND POWER (CHP) SYSTEMS CASE STUDIES

This chapter provides three CHP system installations in Singapore as case studies. The CHP systems described in this chapter are either tri-generation or co-generation systems. All three CHP systems are installed by industrial companies for their plant utility requirements. In order to respect the sensitivity of the information provided by the three companies, the case studies are kept anonymous.

#### Learning Outcomes:

The main learning outcomes from this chapter are to understand:

- 1. The actual CHP system implementation
- 2. Actual performance of the CHP
- 3. The benefits attained through the CHP implementation

#### 11.1. Introduction

As discussed in the preceding Chapters, in a conventional power plant, only a portion of the energy transferred from the fuel to the working fluid is converted to work. The remaining portion of the energy is rejected as waste heat to larger heat sinks such as rivers, oceans or atmosphere. This was evident from the Sankey diagram for a conventional power plant shown in Chapter 1. As seen from the figure, the rejected heat is about 67% of the total energy input and is wasted unless it can be recovered and used for other heating applications.

Some manufacturing plants like chemical, pharmaceutical, oil refining, steel manufacturing and food processing require significant amount of heat energy. These industrial plants also consume a large amount of electrical energy. Therefore, in some cases, it can be economically viable to generate electricity and divert the waste heat generated for useful heating applications.



Figure 11.1 Schematic diagram of an ideal CHP plant

The following benefits are derived by the companies implementing a CHP system:

- Increased total system thermodynamic efficiency
- Lower overall energy cost
- Improved power supply reliability
- Reduced overall CO<sub>2</sub> emissions
- Reduced investment in power transmission capacity
- Lower transmission losses

# 11.2 Case Study 1: A 2 MW Tri-generation (CHP) System in a Pharmaceutical Company A in Singapore

#### Background

The principal activity of this pharmaceutical company is the manufacture of vaccine products. The company started its operation in 2009 and currently operates at 40% of the full production capacity. The production is expected to continue ramping up during the planning stage of this tri-generation project.

The company uses electricity, steam, chilled water and hot water in its operations. Currently, the company buys electricity from the grid, while steam is produced separately by its natural gas fired fire-tube boilers. Chilled water is produced by the electrical chillers and hot water is produced from steam with the help of a heat exchanger. The utility demand of the company prior to the tri-generation system implementation is schematically represented in Figure 11.2.



Figure 11.2 The utility demand of the company A before the CHP system implementation

Before the tri-generation system implementation, the company had 2 nos. of 22 tons/hour boilers (1 operating and 1 serving as a standby unit). The current average steam demand is 3 tons per hour while the peak steam demand is 5 tons per hour. Since the boilers are considerably oversized, the boiler is running at sub-optimal efficiency of 80% due to the poor turn-down ratio.

In order to improve the energy efficiency of its plant, the industrial company decided to install a tri-generation plant to partially meet the demand for electricity, steam, and chilled water, and fully meet the demand for hot water. The rationale for the sizing of the tri-gen facility is to achieve the highest fuel utilisation efficiency of about 83%. The company also decided to replace the existing boilers with high efficiency boilers. The new boilers' efficiency was estimated to be 90% at a steam load of 1.3 tons per hour. It was anticipated that the implementation of these measures would help to reduce the energy consumption of the company by 10% at the facility level. The proposed system is shown schematically in Figure 11.3.



Figure 11.3 The proposed energy efficient system including a CHP system for Company A

Figure 11.3 is a schematic diagram of the proposed 1.8 MW tri-generation system comprising a natural gas fired engine, heat recovery steam generator (HRSG) and a 200 RT absorption chiller.

#### **Project Description**

The tri-generation project involves the installation of the following key equipment:

- A 1.8 MW gas engine to generate electricity
- A heat recovery steam generator to generate 1.2 tons per hour of steam at 8 barg pressure
- A hot water economiser to generate 350 kW of hot water
- An absorption chiller to produce 200 RT of chilled water using hot water from gas engine jacket

Considering the steam usage profile (peak demand of 5 tons/hour and average demand of 3 tons at 40% production load) and the projected average steam demand of 4.5 tons/hour at production full load, the company proposed to replace the existing 2 nos. of 22 tons/hour boilers with 2 nos. of 5 tons/hour fire-tube boilers (1 operating and 1 serving as standby) with economisers to supplement the steam production from the tri-generation plant.

The company estimated that the annual energy savings of the project is 33,647,050 MJ or \$1,364,126 (using a natural gas tariff of \$20.5821/mmBtu and an electricity tariff of \$0.2128/kWh, the electricity and gas tariffs in Singapore in 2013). The project was completed in a year.

A summary of the baseline and post-implementation energy consumption and the energy savings calculations are tabulated below.

Item	Baseline	Post-	Savings
		implementation	
Energy (MJ/year)	319,792,438	288,681,487	31,110,950
Cost (S\$)	8,909,603	7,618,400	1,291,203
CO <sub>2</sub> emission (Ton)	21,417	17,880	3,537

#### Outcome of the CHP system implementation

A CHP (Tri-generation system) comprising an engine, waste heat recovery boiler and an absorption chiller have been installed by the company A. An energy savings calculation has been performed for the baseline energy consumption and postimplementation energy consumption. The computation results suggest that an annual energy savings of 31,110,950 MJ/year has been achieved resulting from the implementation of the tri-generation system, which translates into an annual cost savings of about \$1,291,203 /year based on the 2013 electricity and NG tariffs in Singapore. It has also been established from the measurements that a postimplementation CO<sub>2</sub> emission reduction of about 3,500 tons/year is achievable for the company.

## 11.3. Case Study 2: A 10 MW Tri-generation (CHP) System in a Pharmaceutical Company B in Singapore

#### Background

Pharmaceutical company B used to depend on the electricity from the national grid for plant use and to produce chilled water for air conditioning and plant cooling systems. The electricity drawn from the grid prior to the implementation of the CHP system (tri-generation) was about 12 to 13 MW. Out of this electricity drawn from the grid, about 9 to 10 MW of electricity was for plant use, and 3 MW for operating the electrical chiller producing chilled water for plant use. The company plant also used to consume significant amount of fuel energy (natural gas) for the production of steam for plant use. The electricity and fuel use details for the plant are depicted in Figure 11.4. The tri-generation system using a gas turbine as the prime mover is shown in Figure 11.5.





In order to improve the energy efficiency of its plant, the industrial company B decided to install a 10 MW tri-generation plant to meet the majority of the company's demand for electricity and partial demand of steam, chilled water, and hot water. As with company A, the rationale for the sizing of the tri-gen facility is to achieve the highest fuel utilisation efficiency of about 83%. It was anticipated that the implementation of these measures would help to reduce the energy consumption of the company by about 12% at the facility level. The proposed tri-generation system is shown schematically in Figure 11.7.





The process flow diagram after the installation of the CHP system is shown in Figure 11.6 below. As can be seen from the Figure, the CHP plant produces about 10 MW of electricity reducing the grid drawn electricity amount to 3 MW. In addition, the waste exhaust from the two gas turbines are producing 4,000 RT of cooling with the help of the steam driven absorption chiller. The steam quantity used in the absorption chiller is about 15.5 tons/hr which is part of about 24 tons/hr steam generated by the waste heat recovery boiler.









The post-implementation utility situation of the company in a nutshell is depicted in Figure 11.7. As seen from the Figure, the bulk of the plant's electricity and heat comes from the CHP system. The plant keeps standby boilers for their critical loads in the event of the CHP undergoing maintenance, etc. In addition, the company has reduced contracted capacity quite substantially.

#### Outcome of the CHP system implementation

A 10 MW CHP (Tri-generation system) comprising two gas turbines, waste heat recovery boiler and 4,000 RT absorption chiller have been installed by company B. Due to the implementation of the CHP system, the electricity drawn from the plant is reduced quite significantly from 13 MW to 3 MW resulting in notable cost savings. An energy savings calculation has been performed for the baseline energy consumption and post-implementation energy consumption. In addition, company B has also achieved reliable and seamless energy supply for their business continuity. The company also managed to build a positive image from the environmentally friendly nature of their business through significant reduction of CO<sub>2</sub> emission resulting from the energy efficiency achieved from the CHP system project. Company B became a role model for other pharmaceutical companies in Singapore to invest in green and sustainable technologies. From the point of view of business competitiveness, this kind of project helps to attract more investment due to low energy costs.

## 11.4. Case Study 3: A 6.3 MW Tri-generation (CHP) System in an Industrial Company C in Singapore

#### Background

Company C is a Japanese company, which is a leader in innovative technologies and materials for food products. The company is a global manufacturer and supplier of high-quality cocoa butter equivalents (CBE), non-lauric cocoa butter replacers (CBR), lauric cocoa butter substitute (CBS) as well as other specialty oils and fats. Prior to the CHP implementation, the company's electricity and steam demands were met by the local electricity grid and steam boilers, respectively. Before the implementation of the CHP system, the annual electricity and steam demands for the company C used to be about 3.4 MW & 166,000 Tons, respectively. This constitutes a total annual energy consumption (fuel and electrical energy) of about 660 TJ. The high demand for heat and electricity which are sourced in a conventional manner makes the plant

operation inefficient. Company C potentially risks losing its competitive edge in this highly competitive and volatile market. Therefore, the management decided to introduce a CHP system producing electricity and steam simultaneously. The utility source details prior to the CHP implementation at company C are shown in Figure 11.8. As can be seen from the Figure, company C procured the electricity from the national grid and used LFO and diesel to generate the steam using medium pressure boilers.



Figure 11.8 The utility source details of company C



Figure 11.9 The post-CHP system utility source details of company C

The CHP system (co-generation) generates about 6.32 MW of electricity and 30 tons/hr of saturated medium pressure steam at 13 barg. Out of the generated electrical power, it supplies 4.5 MWe and 30 tons/hr to the facilities of company C and exports the rest of the electricity to the national grid. The co-generation plant has replaced the existing oil-fired boilers and now parallel feed on grid electricity. The co-generation plant supplies electricity to company C and exports excess electricity to the power grid so as to maintain the highest efficiency. All the exhaust heat from the

gas turbine is recovered by a Heat Recovery Steam Generator (HRSG). The details of the gas fired co-generation system is shown in Figure 11.10 and 11.11 for the unfired and fired mode, respectively.



Figure 11.10 The gas turbine-based co-generation system (unfired mode) installed at company C

The specifications of the Kawasaki gas turbine generator are tabulated in Table 11.1 and a photograph of the gas turbine is shown in Figure 11.14.

Turbine type	GPB80D
Model	Kawasaki M7A-03D
Electrical power output	6.32 MW
Generator capacity	8,400 kVA
Voltage	3.3 kV
Frequency	50 Hz
Fuel type	Natural gas



#### Table 11.1 The gas turbine specifications

Figure 11.11 The gas turbine-based co-generation system (fired mode) installed at company C



Figure 11.12 Photograph of the gas turbine installed at company C

#### Outcome of the CHP system implementation

The benefits derived by company C through the implementation of the CHP (cogeneration) system, as with the two previous case studies presented, are the significant amount of electrical energy and fuel energy savings along with the substantial amount of  $CO_2$  emission reductions. The annual electricity and steam demand of company C are about 3,400 kW (28,700 MWh) and 20 Tons/hr (166,200 Tons), respectively.

The annual energy savings for Company C resulting from the installation of the CHP system is depicted in Figure 11.13 and the corresponding cost savings in Figure 11.14.



Figure 11.13 The annual energy savings for Company C through the installation of the CHP system

The annual energy savings in absolute and percentage terms are as follows:Annual energy savings: 147.8 TJ (140,095 MMBtu)% energy savings: 22 %



Figure 11.14 The annual cost savings for Company C through the installation of the CHP system

Annual energy cost savings : \$ 3,564,392 % energy cost savings : 31 %

The annual reduction in  $CO_2$  emissions for company C resulting from the installation of the CHP system is depicted in Figure 11.15



Figure 11.15 The annual CO<sub>2</sub> emissions reduction for company C through the installation of the CHP system

The annual CO<sub>2</sub> emissions reduction in absolute and percentage terms are as follows:

Annual CO2 emissions reduction: 18,300 tons of CO2

% CO\_2 emissions reduction: 39 %

Some of the useful references for further reading on various programmes on CHP in Europe and USA are included in the references.

#### Summary

A CHP system is the simultaneous production of two (co-generation) or more useful outputs, namely electricity, heat and cooling (tri-generation). Three industrial implementations of CHP systems as case studies have been presented in this chapter. At the end of each case study, the benefits achieved from the installation of the CHP systems are highlighted in terms of the energy and cost savings as well as the reduction in  $CO_2$  emissions.

#### References

1. Younus Cengel, Thermodynamics: An Engineering Approach ed.6, McGraw-Hill, 2006.

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3. Neil Petchers, Combined heating, cooling and power handbook: Technologies and Applications", Fairmont Press, second edition, 2012

4. Horlock, J.H, "Combined power plants: including combined cycle gas turbine (CCGT) plants", Oxford Pergamon Press, 1992

5. Web page, Department of Energy, USA. URL: <u>https://www.energy.gov/</u> (Last accessed on 16 May 2019)

Recommended further reading on CHP:

- European Commission Energy Efficiency Programmes; https://ec.europa.eu/energy/en/topics/energy-efficiency/cogeneration-heatand-power
- 2. European Environment Agency web page; https://www.eea.europa.eu/dataand-maps/indicators/combined-heat-and-power-chp-1/combined-heat-andpower-chp-2
- European Union Energy Policy: Co-generation directive; http://www.inforse.org/europe/eu\_cogen-di.htm
- Combined Heat and Power (CHP) Partnership/US EPA; https://www.epa.gov/chp
- CHP Project Development Handbook; https://www.energy.gov/eere/amo/downloads/chp-project-developmenthandbook-us-environmental-protection-agency-chp

#### APPENDIX

#### STEAM PROPERTY TABLE (Courtesy of Wiley Blackwell Publisher)

#### Saturated Water and Steam

p [bar]	$\frac{T_s}{\Gamma^{c} \Gamma^{c}}$	$\frac{v_g}{[m^3/kg]}$	$\frac{u_f u_g}{[k ]/k \sigma]}$	h <sub>r</sub>	h <sub>fg</sub> [k]/kg]	h <sub>g</sub>	<u>s</u> f	s <sub>fg</sub> [kJ/kgK]	Sg
10		1 604	417 2506	417	2258	2675	1 303	6.056	7 359
1.1	102.3	1.549	429 2510	429	2251	2680	1.333	5.994	7.327
1.2	104.8	1.428	439 2512	439	2244	2683	1.361	5.937	7.298
1.3	107.1	1.325	449 2515	449	2238	2687	1.387	5.884	7.271
1.4	109.3	1.236	458 2517	458	2232	2690	1.411	5.835	7.246
1.5	111.4	1.159	467 2519	467	2226	2693	1.434	5.789	7.223
1.6	113.3	1.091	475 2521	475	2221	2696	1.455	5.747	7.202
1.7	115.2	1.031	483 2524	483	2216	2699	1.475	5.707	7.182
1.8	116.9	0.9774	491 2526	491	2211	2702	1.494	5.669	7.163
1.9	118.6	0.9292	498 2528	498	2206	2704	1.513	5.632	7.145
2.0	120.2	0.8856	505 2530	505	2202	2707	1.530	5.597	7.127
2.1 2.2 2.3 2.4 2.5	121.8 123.3 124.7 126.1 127.4	0.8461 0.8100 0.7770 0.7466 0.7186	511         2531           518         2533           524         2534           530         2536           535         2537	511 518 524 530 535	2198 2193 2189 2185 2185 2182	2709 2711 2713 2715 2717	1.547 1.563 1.578 1.593 1.607	5.564 5.533 5.503 5.474 5.446	7.111 7.096 7.081 7.067 7.053
2.6	128.7	0.6927	541 2539	541	2178	2719	1.621	5.419	7.040
2.7	130.0	0.6686	546 2540	546	2174	2720	1.634	5.393	7.027
2.8	131.2	0.6462	551 2541	551	2171	2722	1.647	5.368	7.015
2.9	132.4	0.6253	556 2543	556	2168	2724	1.660	5.344	7.004
3.0	133.5	0.6057	561 2544	561	2164	2725	1.672	5.321	6.993
3.5	138.9	0.5241	58425496052554623255863925626552565	584	2148	2732	1.727	5.214	6.941
4.0	143.6	0.4623		605	2134	2739	1.776	5.121	6.897
4.5	147.9	0.4139		623	2121	2744	1.820	5.037	6.857
5.0	151.8	0.3748		640	2109	2749	1.860	4.962	6.822
5.5	155.5	0.3427		656	2097	2753	1.897	4.893	6.790
6	158.8	0.3156	669         2568           696         2573           720         2577           742         2581           762         2584	670	2087	2757	1.931	4.830	6.761
7	165.0	0.2728		697	2067	2764	1.992	4.717	6.709
8	170.4	0.2403		721	2048	2769	2.046	4.617	6.663
9	175.4	0.2149		743	2031	2774	2.094	4.529	6.623
10	179.9	0.1944		763	2015	2778	2.138	4.448	6.586
11	184.1	0.1774	780 2586	781	2000	2781	2.179	4.375	6.554
12	188.0	0.1632	797 2588	798	1986	2784	2.216	4.307	6.523
13	191.6	0.1512	813 2590	815	1972	2787	2.251	4.244	6.495
14	195.0	0.1408	828 2593	830	1960	2790	2.284	4.185	6.469
15	198.3	0.1317	843 2595	845	1947	2792	2.315	4.130	6.445
16	201.4	0.1237	857         2596           870         2597           883         2598           895         2599           907         2600	859	1935	2794	2.344	4.078	6.422
17	204.3	0.1167		872	1923	2795	2.372	4.028	6.400
18	207.1	0.1104		885	1912	2797	2.398	3.981	6.379
19	209.8	0.1047		897	1901	2798	2.423	3.936	6.359
20	212.4	0.09957		909	1890	2799	2.447	3.893	6.340
22	217.2	0.09069	928 2601	931	1870	2801	2.492	3.813	6.305
24	221.8	0.08323	949 2602	952	1850	2802	2.534	3.738	6.272
26	226.0	0.07689	969 2603	972	1831	2803	2.574	3.668	6.242
28	230.0	0.07142	988 2603	991	1812	2803	2.611	3.602	6.213
30	233.8	0.06665	1004 2603	1008	1795	2803	2.645	3.541	6.186
32 34 36 38 40	237.4 240.9 244.2 247.3 250.3	0.06246 0.05875 0.05544 0.05246 0.04977	1021         2603           1038         2603           1054         2602           1068         2602           1082         2602	1025 1042 1058 1073 1087	1778 1761 1744 1729 1714	2803 2803 2802 2802 2802 2801	2.679 2.710 2.740 2.769 2.797	3.482 3.426 3.373 3.322 3.273	6.161 6.136 6.113 6.091 6.070

p [bar]	$\frac{T_{s}}{[^{\circ}C]}$	$\frac{v_g}{[m^3/kg]}$	u <sub>f</sub> [kJ/kg	u <sub>g</sub> ]	h <sub>f</sub>	h <sub>fg</sub> [kJ/kg]	h <sub>g</sub>	s <sub>f</sub>	<sup>S</sup> fg kJ/kgK]	Sg
40	250.3	0.04977	1082 2	602	1087	1714	2801	2.797	3.273	6.070
42	253.2	0.04732	1097 2	601	1102	1698	2800	2.823	3.226	6.049
44	256.0	0.04509	1109 2	600	1115	1683	2798	2.849	3.180	6.029
46	258.8	0.04305	1123 2	599	1129	1668	2797	2.874	3.136	6.010
48	261.4	0.04117	1136 2	598	1142	1654	2796	2.897	3.094	5.991
50	263.9	0.03944	1149 2	597	1155	1639	2794	2.921	3.052	5.973
55	269.9	0.03563	1178 2	594	1185	1605	2790	2.976	2.955	5.931
60	275.6	0.03244	1206 2	590	1214	1570	2784	3.027	2.863	5.890
65	280.8	0.02972	1232 2	586	1241	1538	2779	3.076	2.775	5.851
70	285.8	0.02737	1258 2	581	1267	1505	2772	3.122	2.692	5.814
75	290.5	0.02532	1283 2	576	1293	1473	2766	3.166	2.613	5.779
80	295.0	0.02352	1306 2	570	1317	1441	2758	3.207	2.537	5.744
85	299.2	0.02192	1329 2	565	1341	1410	2751	3.248	2.463	5.711
90	303.3	0.02048	1351 2	559	1364	1379	2743	3.286	2.393	5.679
95	307.2	0.01919	1372 2	552	1386	1348	2734	3.324	2.323	5.647
100	311.0	0.01802	1393 2	545	1408	1317	2725	3.360	2.255	5.615
105	314.6	0.01696	1414 2	537	1429	1286	2715	3.395	2.189	5.584
110	318.0	0.01598	1434 2	529	1450	1255	2705	3.430	2.123	5.553
115	321.4	0.01508	1454 2	522	1471	1224	2695	3.463	2.060	5.523
120	324.6	0.01426	1473 2	514	1491	1194	2685	3.496	1.997	5.493
125	327.8	0.01349	1492 2	505	1511	1163	2674	3.529	1.934	5.463
130	330.8	0.01278	1511 2	496	1531	1131	2662	3.561	1.872	5.433
135	333.8	0.01211	1530 2	487	1551	1099	2650	3.592	1.811	5.403
140	336.6	0.01149	1548 2	477	1571	1067	2638	3.623	1.750	5.373
145	339.4	0.01090	1567 2	467	1591	1034	2625	3.654	1.689	5.343
150	342.1	0.01035	1585 2	456	1610	1001	2611	3.685	1.627	5.312
155	344.8	0.00982	1604 2	445	1630	967	2597	3.715	1.565	5.280
160	347.3	0.00932	1623 2	433	1650	932	2582	3.746	1.502	5.248
165	349.8	0.00884	1641 2	420	1670	895	2565	3.777	1.437	5.214
170	352.3	0.00838	1660 2	406	1690	858	2548	3.808	1.373	5.181
175	354.6	0.00794	1679 2	391	1711	819	2530	3.839	1.305	5.144
180	357.0	0.00751	1699 2	375	1732	778	2510	3.872	1.236	5.108
185	359.2	0.00709	1719 2	358	1754	735	2489	3.905	1.163	5.068
190	361.4	0.00668	1740 2	339	1777	689	2466	3.941	1.086	5.027
195	363.6	0.00627	1762 2	318	1801	639	2440	3.977	1.004	4.981
200	365.7	0.00585	1786 2	294	1827	584	2411	4.014	0.914	4.928
202	366.5	0.00569	1796 2	283	1838	560	2398	4.031	0.875	4.906
204	367.4	0.00552	1806 2	271	1849	535	2384	4.049	0.835	4.884
206	368.2	0.00534	1817 2	259	1861	508	2369	4.067	0.792	4.859
208	369.0	0.00517	1829 2	245	1874	479	2353	4.087	0.745	4.832
210	369.8	0.00498	1842 2	231	1889	447	2336	4.108	0.695	4.803
212	370.6	0.00479	1856 2	214	1904	412	2316	4.131	0.640	4.771
214	371.4	0.00458	1871 2	196	1921	373	2294	4.157	0.579	4.736
216	372.1	0.00436	1888 2	174	1940	328	2268	4.186	0.508	4.694
218	372.9	0.00409	1911 2	146	1965	270	2235	4.224	0.417	4.641
220	373.7	0.00368	1949 2	097	2008	170	2178	4.289	0.263	4.552
221.2	374.15	0.00317	2014 2	014	2084	0	2084	4.406	0.000	4.406

Saturated Water and Steam

Superheated Steam\*

p/[bar] (T <sub>s</sub> /[°C])		<u>Т</u> [°С]	200	250	300	350	400	450	500	600
5 (151.8)	$\begin{array}{c} v_{g} & 0.3748 \\ u_{g} & 2562 \\ h_{g} & 2749 \\ s_{g} & 6.822 \end{array}$	v u h s	0.4252 2644 2857 7.060	0.4745 2725 2962 7.271	0.5226 2804 3065 7.460	0.5701 2883 3168 7.633	0.6172 2963 3272 7.793	0.6641 3045 3377 7.944	0.7108 3129 3484 8.087	0.8040 3300 3702 8.351
6 (158.8)	$v_{g} = 0.3156$ $u_{g} = 2568$ $h_{g} = 2757$ $s_{g} = 6.761$	v u h s	0.3522 2640 2851 6.968	0.3940 2722 2958 7.182	0.4344 2801 3062 7.373	0.4743 2881 3166 7.546	0.5136 2962 3270 7.707	0.5528 3044 3376 7.858	0.5919 3128 3483 8.001	0.6697 3299 3701 8.267
7 (165.0)	$v_{g} = 0.2728$ $u_{g} = 2573$ $h_{g} = 2764$ $s_{g} = 6.709$	v u h s	0.3001 2636 2846 6.888	0.3364 2720 2955 7.106	0.3714 2800 3060 7.298	0.4058 2880 3164 7.473	0.4397 2961 3269 7.634	0.4734 3043 3374 7.786	0.5069 3127 3482 7.929	0.5737 3298 3700 8.195
8 (170.4)	$v_{g} = 0.2403$ $u_{g} = 2577$ $h_{g} = 2769$ $s_{g} = 6.663$	v u h s	0.2610 2631 2840 6.817	0.2933 2716 2951 7.040	0.3242 2798 3057 7.233	0.3544 2878 3162 7.409	0.3842 2960 3267 7.571	0.4138 3042 3373 7.723	0.4432 3126 3481 7.866	0.5018 3298 3699 8.132
9 (175.4)	$v_{g} = 0.2149$ $u_{g} = 2581$ $h_{g} = 2774$ $s_{g} = 6.623$	v u h s	0.2305 2628 2835 6.753	0.2597 2714 2948 6.980	0.2874 2796 3055 7.176	0.3144 2877 3160 7.352	0.3410 2959 3266 7.515	0.3674 3041 3372 7.667	0.3937 3126 3480 7.811	0.4458 3298 3699 8.077
10 (179.9)	$v_{g} = 0.1944$ $u_{g} = 2584$ $h_{g} = 2778$ $s_{g} = 6.586$	v u h s	0.2061 2623 2829 6.695	0.2328 2711 2944 6.926	0.2580 2794 3052 7.124	0.2825 2875 3158 7.301	0.3065 2957 3264 7.464	0.3303 3040 3370 7.617	0.3540 3124 3478 7.761	0.4010 3297 3698 8.028
15 (198.3)	$v_{g} = 0.1317$ $u_{g} = 2595$ $h_{g} = 2792$ $s_{g} = 6.445$	v u h s	0.1324 2597 2796 6.452	0.1520 2697 2925 6.711	0.1697 2784 3039 6.919	0.1865 2868 3148 7.102	0.2029 2952 3256 7.268	0.2191 3035 3364 7.423	0.2351 3120 3473 7.569	0.2667 3294 3694 7.838
20 (212.4)	$v_{g} = 0.0996$ $u_{g} = 2600$ $h_{g} = 2799$ $s_{g} = 6.340$	v u h s		0.1115 2681 2904 6.547	0.1255 2774 3025 6.768	0.1386 2861 3138 6.957	0.1511 2946 3248 7.126	0.1634 3030 3357 7.283	0.1756 3116 3467 7.431	0.1995 3291 3690 7.701
30 (233.8)	$v_{g} = 0.0666$ $u_{g} = 2603$ $h_{g} = 2803$ $s_{g} = 6.186$	v u h s		0.0706 2646 2858 6.289	0.0812 2751 2995 6.541	0.0905 2845 3117 6.744	0.0993 2933 3231 6.921	0.1078 3020 3343 7.082	0.1161 3108 3456 7.233	0.1324 3285 3682 7.507
40 (250.3)	$\begin{array}{ccc} v_{\rm g} & 0.0498 \\ u_{\rm g} & 2602 \\ h_{\rm g} & 2801 \\ s_{\rm g} & 6.070 \end{array}$	v u h s			0.0588 2728 2963 6.364	0.0664 2828 3094 6.584	0.0733 2921 3214 6.769	0.0800 3010 3330 6.935	0.0864 3099 3445 7.089	0.0988 3279 3674 7.368
50 (263.9)	$v_{g} = 0.0394$ $u_{g} = 2597$ $h_{g} = 2794$ $s_{g} = 5.973$	v u h s			0.0453 2700 2927 6.212	0.0519 2810 3070 6.451	0.0578 2907 3196 6.646	0.0632 3000 3316 6.818	0.0685 3090 3433 6.975	0.0786 3273 3666 7.258
60 (275.6)	$\begin{array}{ccc} v_{g} & 0.0324 \\ u_{g} & 2590 \\ h_{g} & 2784 \\ s_{g} & 5.890 \end{array}$	v u h s			0.0362 2670 2887 6.071	0.0422 2792 3045 6.336	0.0473 2893 3177 6.541	0.0521 2988 3301 6.719	0.0566 3081 3421 6.879	0.0652 3266 3657 7.166
70 (285.8)	$v_{g} = 0.0274$ $u_{g} = 2581$ $h_{g} = 2772$ $s_{g} = 5.814$	v/[m <sup>3</sup> u/[kJ h/[kJ s/[kJ	<sup>3</sup> /kg] /kg] /kg] /kgK]		0.0295 2634 2841 5.934	0.0352 2772 3018 6.231	0.0399 2879 3158 6.448	0.0441 2978 3287 6.632	0.0481 3073 3410 6.796	0.0556 3260 3649 7.088

<i>p</i> /[bar] ( <i>T</i> <sub>s</sub> /[°C])		$\frac{T}{[°C]}$	350	375	400	425	450	500	600	700
80 (295.0)	v, 0.02352 h, 2758 s, 5.744	$\frac{v}{10^{-2}}$ h s	2.994 2990 6.133	3.220 3067 6.255	3.428 3139 6.364	3.625 3207 6.463	3.812 3272 6.555	4.170 3398 6.723	4.839 3641 7.019	5.476 3881 7.279
90 (303.3)	v, 0.02048 h, 2743 s <sub>g</sub> 5.679	$\frac{v}{10^{-2}}$ h s	2.578 2959 6.039	2.794 3042 6.171	2.991 3118 6.286	3.173 3189 6.390	3.346 3256 6.484	3.673 3385 6.657	4.279 3633 6.958	4.852 3874 7.220
100 (311.0)	v, 0.01802 h, 2725 s, 5.615	v/10 <sup>-2</sup> h s	2.241 2926 5.947	2.453 3017 6.091	2.639 3097 6.213	2.812 3172 6.321	2.972 3241 6.419	3.275 3373 6.596	3.831 3624 6.902	4.353 3868 7.166
110 (318.0)	v, 0.01598 h, 2705 s <sub>g</sub> 5.553	$\frac{v}{10^{-2}}$ h s	1.960 2889 5.856	2.169 2989 6.014	2.350 3075 6.143	2.514 3153 6.257	2.666 3225 6.358	2.949 3360 6.539	3.465 3616 6.850	3.945 3862 7.117
120 (324.6)	v, 0.01426 h, 2685 s <sub>g</sub> 5.493	$\frac{v}{10^{-2}}$ h s	1.719 2849 5.762	1.931 2960 5.937	2.107 3052 6.076	2.265 3134 6.195	2.410 3209 6.301	2.677 3348 6.487	3.159 3607 6.802	3.605 3856 7.072
130 (330.8)	v <sub>g</sub> 0.01278 <b>h</b> , 2662 s, 5.433	$v/10^{-2}$ h s	1.509 2804 5.664	1.726 2929 5.862	1.901 3028 6.011	2.053 3114 6.136	2.193 3192 6.246	2.447 3335 6.437	2.901 3599 6.758	3.318 3850 7.030
140 (336.6)	$v_{,} 0.01149$ $h_{g} 2638$ $s_{g} 5.373$	$\frac{v}{10^{-2}}$ h s	1.321 2753 5.559	1.548 2896 5.784	1.722 3003 5.946	1.872 3093 6.079	2.006 3175 6.193	2.250 3322 6.390	2.679 3590 6.716	3.071 3843 6.991
150 (342.1)	v, 0.01035 h, 2611 s <sub>g</sub> 5.312	$\frac{v}{10^{-2}}$ h s	1.146 2693 5.443	1.391 2861 5.707	1.566 2977 5.883	1.714 3073 6.023	1.844 3157 6.142	2.078 3309 6.345	2.487 3581 6.677	2.857 3837 6.954
160 (347.3)	v 0.00932 h, 2582 s <sub>g</sub> 5.248	$\frac{v}{10^{-2}}$ h s	0.976 2617 5.304	1.248 2821 5.626	1.427 2949 5.820	1.573 3051 5.968	1.702 3139 6.093	1.928 3295 6.301	2.319 3573 6.639	2.670 3831 6.919
170 (352.3)	v, 0.00838 h, 2548 s <sub>g</sub> 5.181	$\frac{v}{10^{-2}}$ h s		1.117 2778 5.541	1.303 2920 5.756	1.449 3028 5.914	1.576 3121 6.044	1.796 3281 6.260	2.171 3564 6.603	2.506 3825 6.886
180 (357.0)	v <sub>g</sub> 0.00751 h, 2510 s <sub>g</sub> 5.108	$\frac{v}{10^{-2}}$ h s		0.997 2729 5.449	1.191 2888 5.691	1.338 3004 5.861	1.463 3102 5.997	1.678 3268 6.219	2.039 3555 6.569	2.359 3818 6.855
190 (361.4)	v, 0.00668 h, 2466 s <sub>g</sub> 5.027	$\frac{v}{10^{-2}}$ h s		0.882 2674 5.348	1.089 2855 5.625	1.238 2980 5.807	1.362 3082 5.950	1.572 3254 6.180	1.921 3546 6.536	2.228 3812 6.825
200 (365.7)	v, 0.00585 h, 2411 s <sub>g</sub> 4.928	v/10 <sup>-2</sup> [1 h/[kJ/kg s/[kJ/kg	m <sup>3</sup> /kg] ;] K]	0.768 2605 5.228	0.995 2819 5.556	1.147 2955 5.753	1.270 3062 5.904	1.477 3239 6.142	1.815 3537 6.505	2.110 3806 6.796
210 (369.8)	v, 0.00498 <b>h</b> , 2336 <b>s</b> , 4.803	$v/10^{-2}$ h s		0.650 2500 5.050	0.908 2781 5.484	1.064 2928 5.699	1.187 3041 5.859	1.390 3225 6.105	1.719 3528 6.474	2.003 3799 6.768
220 (373.7)	v, 0.00368 h, 2178 s <sub>g</sub> 4.552	$\frac{v}{10^{-2}}$ h s		0.450 2300 4.725	0.825 2738 5.409	0.987 2900 5.645	1.111 3020 5.813	1.312 3210 6.068	1.632 3519 6.444	1.906 3793 6.742
221.2 (374.15)	$v_{c} = 0.00317$ $h_{c} = 2084$ $s_{c} = 4.406$	$\frac{v}{10^{-2}}$ h s	0.163 1637 3.708	0.351 2139 4.490	0.816 2733 5.398	0.978 2896 5.638	1.103 3017 5.807	1.303 3208 6.064	1.622 3518 6.441	1.895 3792 6.739