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وزارة التعليم العالي والبحث العلمي

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كلية التكنولوجيا
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تحت عنوان : - ENERGY CONVERSION

وفق البرنامج المقترح لطلبة السنة الثالثة ليسانس فرع الطاقويات
.

رئيس اللجنة العلمية للقسم
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ABSTRACT

In the context of increasing global energy demand and growing environmental concerns, a thorough understanding of energy conversion processes is essential for the development of efficient and sustainable energy systems. This book is designed to provide undergraduate students, particularly those specializing in Energy Engineering, with a solid foundation in the principles of applied thermodynamics relevant to modern energy technologies.

The content is organized into four interconnected chapters that progressively address the fundamental aspects of energy systems. The first chapter examines single-phase thermodynamic cycles that underpin the operation of heat engines and gas-based power systems, including classical cycles such as Carnot, Otto, Diesel, Brayton, and Stirling, along with techniques for improving thermal efficiency. The second chapter focuses on two-phase cycles used in steam power generation, covering the Rankine cycle and its extensions, as well as hybrid energy systems, cogeneration, and an introduction to nuclear power systems. The third chapter introduces exergy analysis as an advanced thermodynamic tool for identifying energy losses and system inefficiencies, with practical applications to gas and steam power plants.

Through a structured and pedagogical approach, this book aims to equip students with the theoretical knowledge and analytical tools necessary to understand, evaluate, and optimize energy conversion systems in a sustainable engineering context.

The fourth and final chapter addresses the thermodynamics of combustion, a topic that underpins many energy systems based on chemical energy conversion. Students are introduced to the behavior of reactive gas mixtures, stoichiometric calculations, flame temperatures, and the principles of ignition, both spontaneous and controlled. The chapter also delves into chemical kinetics, examining reaction mechanisms and factors that influence combustion processes, such as pressure, temperature, and reaction rates.

Overall, this coursebook is intended to reinforce the knowledge gained through lectures, tutorials, and practical laboratory sessions. It also serves as a reference guide for academic projects, research endeavors, and more advanced coursework in the field of energy engineering.

Keywords

Energy engineering, single-phase, thermodynamic cycles, two-phase cycles, concept of exergy, thermodynamics of combustion

ENERGY CONVERSION

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PREFACE

In the contemporary world, where technological advancement and environmental sustainability are deeply intertwined, the subject of energy occupies a pivotal role. The growing global demand for energy, coupled with the urgent need to transition toward cleaner and more efficient energy systems, makes it essential for future engineers to develop a solid and comprehensive understanding of energy conversion processes. The transformation of energy from one form to another—whether mechanical, thermal, electrical, or chemical—is central to nearly all modern technologies, and understanding these conversions is critical for innovation and responsible engineering practice.

This coursebook has been carefully developed as a structured and pedagogically coherent resource for students in the third year of their bachelor's degree program, particularly those pursuing a specialization in Energy Engineering. Its primary aim is to equip students with the theoretical knowledge to grasp the complex workings of thermodynamic systems used in energy production. Through clear explanations, illustrative examples, and methodical progression, the text supports learners in mastering key concepts in energy conversion and analysis.

The content of this book is divided into four major chapters, each building upon the last to form a logical and comprehensive journey through the fundamental aspects of energy systems.

The first chapter introduces students to single-phase thermodynamic cycles, which are fundamental to the operation of many heat engines and gas-powered systems. It covers classic cycles such as the Carnot, Otto, Diesel, Brayton, and Stirling cycles. These foundational cycles are presented along with their practical adaptations and enhancements, including configurations with regenerative heating, intercooling, and reheating, all of which aim to improve thermal efficiency and operational performance. This chapter lays the groundwork for understanding how various engine types operate and how engineers can optimize their functionality.

In the second chapter, the focus shifts to two-phase cycles, particularly those used in steam-based power generation. Here, students explore the Rankine cycle and its extensions, such as the Hirn cycle, along with methods involving reheating and steam extraction. This chapter also introduces hybrid energy systems, which integrate technologies like solar thermal input with gas or steam cycles, as well as cogeneration systems that produce both electricity and useful heat. Additionally, it provides an overview of nuclear power systems, discussing their basic operating principles and their role in the broader energy mix.

The third chapter introduces the concept of exergy, a vital tool in modern thermodynamic analysis. Unlike traditional energy balances, exergy analysis allows engineers to pinpoint where and how energy losses occur in a system, offering deeper insights into inefficiencies and potential areas for improvement. This section applies the exergy framework specifically to gas and steam power plants, demonstrating its practical value in real-world system evaluations.

The fourth and final chapter addresses the thermodynamics of combustion, a topic that underpins many energy systems based on chemical energy conversion. Students are introduced to the behavior of reactive gas mixtures, stoichiometric calculations, flame temperatures, and the principles of ignition, both spontaneous and controlled. The chapter also delves into chemical kinetics, examining reaction mechanisms and factors that influence combustion processes, such as pressure, temperature, and reaction rates.

Overall, this coursebook is intended to reinforce the knowledge gained through lectures, tutorials, and practical laboratory sessions. It also serves as a reference guide for academic projects, research endeavors, and more advanced coursework in the field of energy engineering.

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CHAPTER ONE SINGLE-PHASE POWER CYCLES

I.1 Definition

A single-phase power cycle is a successive series of transformations, after which the system returns to its initial state. In this type of cycle, the motor fluid used keeps the same phase during all the transformations undergone during this cycle.

I.2 Carnot Cycle

I.2.1 Description of the cycle

In thermodynamics, the Carnot cycle is the reversible cyclic process of the Carnot machine. This machine produces work (it is an engine) from two heat sources of different temperatures. A gas, considered to be perfect, undergoes characteristic transformations to provide mechanical work. Carnot sought to make a cycle with the best possible efficiency. Thus, each efficiency of a thermodynamic machine can be compared with the efficiency of the Carnot cycle. It serves as a reference cycle.

The cycle is composed of 4 processes (2 isotherms and 2 isentropic):

- Isentropic compression (reversible adiabatic)
- Isothermal expansion
- Isentropic expansion (reversible adiabatic)
- Isothermal compression

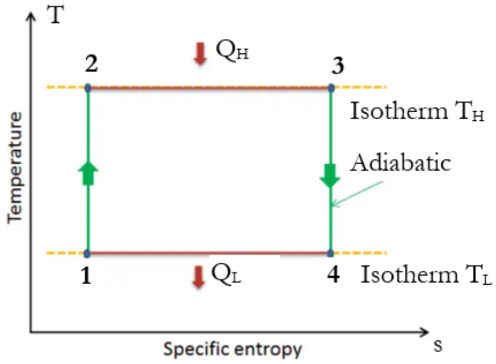


Fig. I 1 Carnot Ts Diagram

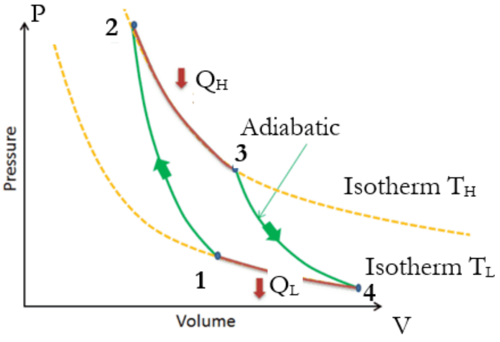


Fig. I 2 Carnot Pv Diagram

T_H is the temperature of the hot source and T_L is the temperature of the cold source The transformation 1-2 is isentropic ($S_1=S_2$) so we can write the following equality:

$$\Delta S_{12}=S_2\text{-}S_1=0 \qquad \text{0-1)}$$

where S is the entropy of the gas

The 2-3 transformation is isothermal so according to the second law of thermodynamics, we can write:

$$\Delta S_{23}=\frac{Q_H}{T_H} \qquad \text{0-2)}$$

This gives

$$Q_H = T_H(S_3 - S_2) \qquad \text{0-3)}$$

The 3-4 transformation is isentropic (S3=S4) so we can write the following equality:

$$\Delta S_{34} = S_4 - S_3 = 0 \qquad \text{0-4)}$$

The 4-1 transformation is isothermal so according to the second law of thermodynamics, we can write:

$$\Delta S_{41} = \frac{Q_L}{T_L} \tag{0-5}$$

This gives

$$Q_L = T_L(S_1 - S_4) \tag{0-6}$$

On the other hand, we have

$$\Delta S\,(2-3) = -\Delta S\,(4-1) \tag{0-7}$$

Based on the definition of yield and equations (2) and (6), the cycle yield can be written as follows:

$$\eta_{Carnot} = \frac{\text{Net work produced by the cycle}}{\text{Heat supplied to the cycle}} = \frac{W_{net}}{Q_H} \tag{0-8}$$

Therefore

$$\eta_{Carnot} = \frac{W_{net}}{Q_H} = \frac{Q_H - Q_L}{Q_H} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{T_L}{T_H} \tag{0-9}$$

Carnot yield for ideal cycle

$$\eta_{Carnot} = 1 - \frac{T_L}{T_H} \tag{0-10}$$

As we consider an ideal gas, the working fluid for a Carnot-cycle heat engine can be calculate by analyzing the four processes that make up the cycle. The four state points, 1, 2, 3, and 4, and the four processes are as shown in Fig. 5.24. For convenience, let us consider a unit mass of gas inside the cylinder. Now for each of the four processes, the reversible work done at the moving boundary is given by

$$\delta w = P\,dV \tag{0-11}$$

Similarly, for each process the gas behavior is, from the ideal-gas relation,

$$PV = RT \tag{0-12}$$

and the internal energy is

$$du = C_v\,dT \tag{0-13}$$

Assuming no changes in kinetic or potential energies, the energy equation is, at unit mass,

$$\delta q = du + \delta w \tag{0-14}$$

Substituting the three previous expressions into this equation, we have for each of the four processes

$$\delta q = C_v\,dT + \frac{RT}{v}dv \tag{0-15}$$

The isothermal heat addition process 1–2 is an expansion at T_H, such that v_2 is larger than v_1. Similarly, the isothermal heat rejection process 3–4 is a compression at a lower temperature, T_L, and v_4 is smaller than v_3. The adiabatic expansion process 2–3 is an expansion from T_H to T_L, with an increase in specific volume, while the adiabatic compression process 4–1 is a compression from T_L to T_H, with a decrease in specific volume. The area below each process line represents the work for that process, as given by Eq. (I.11).

We now proceed to integrate Eq. (I.15) for each of the four processes that make up the Carnot cycle. For the isothermal heat addition process 2–3, we have

$$q_H = {}_2q_3 = 0 + RT_H \ln\frac{v_3}{v2} \tag{0-16}$$

For the adiabatic expansion process 3–4 we divide by *T* to get,

$$0 = \int_{T_H}^{T_L} \frac{C_{v0}}{T} dT + R \ln\frac{v_4}{v_3} \qquad 0\text{-}17)$$

For the isothermal heat rejection process 4–1,

$$q_L \; = \; -_4q_1 \; = \; -0 \; - \; RT_L \ln\frac{v_1}{v_4} = RT_L \ln\frac{v_4}{v_1} \qquad 0\text{-}18)$$

and for the adiabatic compression process 1–2 we divide by *T* to get,

$$0 = \int_{T_L}^{T_H} \frac{C_{v0}}{T} dT + R \ln\frac{v_2}{v_1} \qquad 0\text{-}19)$$

From Equations. (I.17) and (I.19), we get

$$\int_{T_H}^{T_L} \frac{C_{v0}}{T} dT = \; -R\ln\frac{v_4}{v_3} = R \ln\frac{v_2}{v_1} \qquad 0\text{-}20)$$

Therefore,

$$\frac{v_4}{v_3} = \frac{v_1}{v_2} \qquad 0\text{-}21)$$

Thus, from Equations. (I.21) , (I.18) and Eq. (I.17), we find that

$$\frac{Q_H}{Q_L} = \frac{RT_H \ln\frac{v_2}{v1}}{RT_L \ln\frac{v_3}{v_4}} = \frac{T_H}{T_L} \qquad 0\text{-}22)$$

I.2.2 Carnot exercises

I.2.2.1 Exercise 1

Let us consider the heat engine, shown schematically in Fig. I.2, that receives a heat transfer rate of 1 MW at a high temperature of 550°C and rejects energy to the ambient surroundings at 300 K. Work is produced at a rate of 450 kW. Calculate:

1. The energy discarded to the ambient surroundings
2. The engine efficiency
3. compare this efficiency to an efficiency cycle Carnot between the same T_H and T_L

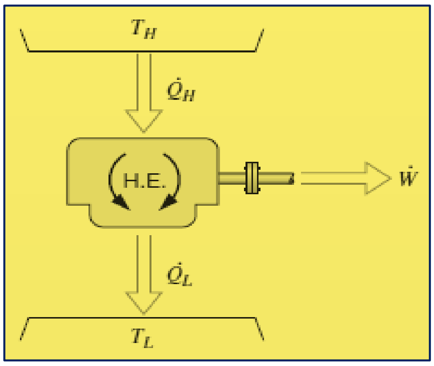


Fig. I 3. A heat engine operating between two constant-temperature energy reservoirs

Solution

1. If we take the heat engine as a control volume, the energy equation gives
$$\dot{Q}_L = \dot{Q}_H - \dot{W} = 1000 - 450 = 550\,kW$$
2. from the definition of efficiency
$$\eta_{thermal} = \frac{\dot{W}}{\dot{Q}_H} = \frac{450}{1000} = 0.45 = 45\%$$
3. For the Carnot heat engine, the efficiency is given by the temperature of the reservoirs:
$$\eta_{Carnot} = 1 - \frac{T_L}{T_H} = 1 - \frac{300}{550 + 273} = 0.635 = 63.5\%$$

The rates of work and heat rejection become

$$\dot{W} = \eta_{Carnot}\ \dot{Q}_H = 0.635 \times 1000 = 635\,kW$$

$$\dot{Q}_L = \dot{Q}_H - \dot{W} = 1000 - 635 = 365\,kW$$

The actual heat engine thus has a lower efficiency than the Carnot (ideal) heat engine, with a value of 45% typical for a modern steam power plant. This also implies that the actual engine rejects a larger amount of energy to the ambient surroundings (55%) compared with the Carnot heat engine (36%).

I.2.2.2 Exercise 2

Calculate the thermal efficiency of a Carnot cycle heat engine operating between reservoirs at T_H =300°C and T_L=45°C.

Response

$$\eta_{Carnot} = 1 - \frac{T_L}{T_H} = 1 - \frac{45 + 273}{300 + 273} = 0.445 = 44.5\%$$

I.2.2.3 Exercise 3

An ideal (Carnot) heat engine has an efficiency of 40%. If the high temperature is raised 15%, what is the new efficiency keeping the same low temperature?

Solution

$$OLD\eta_{Carnot} = 1 - \frac{T_L}{T_{H.OLD}} \quad 1 - \frac{T_L}{T_H}$$

$$\frac{T_L}{T_{H.OLD}} = 1 - OLD\eta_{Carnot} = 0.6$$

$$T_L = 0.6T_{H.OLD}$$

$$NEW\eta_{Carnot} = 1 - \frac{T_L}{T_{H.NEW}} \quad 1 - \frac{T_L}{T_H}\ T_L$$

$$T_{H.NEW} = T_{H.OLD} + 15\%T_{H.OLD} = T_{H-OLD}\left(\frac{15}{100} + 1\right) = 1.15T_{H.OLD}$$

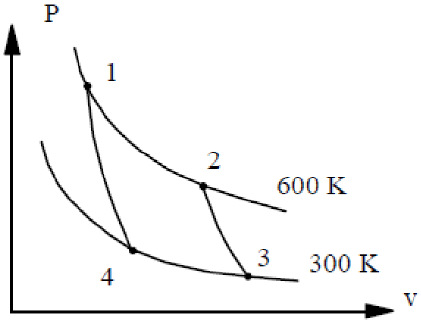
$$NEW\eta_{Carnot} = 1 - \frac{0.6\ T_{H.OLD}}{1.15\ T_{H.OLD}}$$

$$NEW\eta_{Carnot} = 47.8\%$$

I.2.2.4 Exercise 4

An ideal gas Carnot cycle with air in a piston/cylinder has a high temperature of 1000 K and heat rejection at 400 K. During heat addition the volume triples. Find :

1. The two specific heat transfers in the cycle
2. The overall cycle efficiency.



Solution

TH= 1000K; TL = 400K; $\frac{v_3}{v_2} = 3$

$$q_H = 0 + RT_H \ln\frac{v_3}{v_2} = 8.31 * 1000ln3$$

$$= 9\,129{,}47\frac{kJ}{kmol} q_H = 9\,129{,}47\ kJ/kmol$$

$$\frac{q_H}{q_L} = \frac{T_H}{T_L}\ then\ q_L = q_H * \frac{T_L}{T_H} = 9\,129{,}47 * \frac{400}{1000} = \frac{3\,651{,}79kJ}{kmol}$$

$$q_L = 3\,651,\frac{79kJ}{kmol}$$

$$\eta_{th} = 1 - \frac{3\,651{,}79}{9\,129{,}47} = 0.61 - \frac{Q_L}{Q_H} = 1 -= 0{,}6$$

I.2.2.5 Exercise 5

Find the power output and the low T heat rejection rate for a Carnot cycle heat engine that receives 6 kW at 250°C and rejects heat at 30°C

Solution:

From the definition of the Carnot cycle:

$$\eta_{carnot} = 1 - \frac{T_L}{T_H} = 1 - \frac{303}{523} = 0.42$$

Definition of the heat engine efficiency gives the work as

$$\dot{W} = \eta\,\dot{Q}_H = 0.42 \times 6 = 2.52\ kW$$

Apply the energy equation

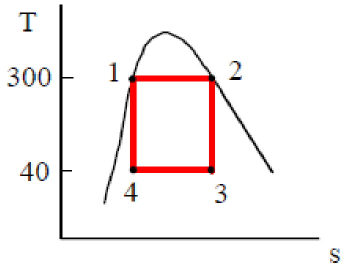
$$\dot{Q}_L = \dot{Q}_H - \dot{W} = 6 - 2.52 = 3.48\ kW$$

I.2.2.6 Exercise 6

Air in a piston/cylinder goes through a Carnot cycle with the P-v diagram shown in figure bellow. The high and low temperatures are 600 K and 300 K respectively. The heat added at the high temperature is 250 kJ/kg and the lowest pressure in the cycle is 75 kPa.

Find

1. The network (W_{net}) per unit mass
2. The specific volume (V_4) and pressure (P_4) after heat rejection.



Solution:

Given: Q_H = 250 kJ/kg, T_H = 600 K, T_L = 300 K, P_3 = 75 kPa

1. Since this is a Carnot cycle and we know the temperatures, the efficiency is

$$\eta_{Carnot} = 1 - \frac{T_L}{T_H} = 1 - \frac{300}{600} = 0.5$$

and the network becomes

$$W_{net} = \eta Q_H = 0.5 \times 250 = \frac{125kJ}{kg}$$

The heat rejected is $Q_L = Q_H - W_{net} = \frac{125kJ}{kg}$

After heat rejection is state 4.

$$Q_L = RT_L \ln\left(\frac{V_3}{V_4}\right)$$

$$V_3 = R\frac{T_3}{P_3} = 0.287 \times \frac{300}{75} = \frac{1.148\, m^3}{kg}$$

$$V_4 = V_3 \exp\left(-\frac{Q_L}{RT_L}\right) = 1.148 \exp\left(-\frac{125}{0.287} \times 300\right) = \frac{0.2688\, m^3}{kg}$$

$$P_4 = R\frac{T_4}{V_4} = 0.287 \times \frac{300}{0.2688} = 320\, kPa$$

I.2.2.7 Exercise 7

Consider a Carnot-cycle heat engine with water as the working fluid. The heat transfer to the water occurs at 300°C, during which process the water changes from saturated liquid to saturated vapor. The heat is rejected from the water at 40°C. Show the cycle on a T–s diagram and find the quality of the water at the beginning and end of the heat rejection process. Determine

1. The network output per kilogram of water
2. The cycle thermal efficiency.

Solution:

From the definition of the Carnot cycle, two constant s and two constant T processes.
From table B.1.1 State 2 is saturated vapor so

$$s_3 = s_2 = 5.7044\frac{kJ}{kg}.K = 0.5724 + x_3(7.6845) \text{then } x_3 = 0.6678$$

State 1 is saturated liquid so

$$s_4 = s_1 = \frac{3.2533kJ}{kg}.K = 0.5724 + x_4(7.6845) \text{then } x_4 = 0.3489$$

$$\eta_{Carnot} = Q_H * W_{net} = \frac{T_H - T_L}{T_H} = 1 - \frac{300}{573.2} \ 0.4536$$

$$Q_H = T_H(s_2 - s_1) = 573.2\,(5.7044 - 3.2533) = 1405.0\frac{kJ}{kg}$$

$$W_{net} = \eta T_H \times Q_H = 637.\frac{3kJ}{kg}$$

I.2.2.8 Exercice8

Let us consider a Carnot cycle that uses water vapor as a coolant. A source at a temperature of 250°C transmits its heat to the coolant as it transitions from a saturated liquid state to a saturated vapor. Heat is removed at a pressure of 10 kPa

Represent the cycle in the T.S diagram

Determine:

1. The thermal efficiency of the cycle
2. The amount of heat removed
3. Network produced

Solution:

According to the previous thermodynamic tables and in the appendix to P = 0.001 MPa = 10 kPA we have:

The corresponding saturation temperature = 45.81 °C = 318.81 K

The entropy of the fluid at point 1 (liquid) = 0.6493 kJ/kg

The entropy of the fluid at point 4 (steam), s_{4vap}=8.1502 kJ/kg

and at T=250°c

The entropy of the fluid at point 1 (liquid) = 2.7927 kJ/kg

The entropy of the fluid at point 4 (steam) = 6.0730 kJ/kg

$$\eta = 1 - \frac{T_f}{T_c} = 0.39 = 39\%$$

$$\mathrm{Q_L} = \mathrm{T_L}(\mathrm{s_{1f}} - \mathrm{s_{4g}}) = 318.81(6.0730 \ - 2.7927) = -1\ 045{,}79\frac{\mathrm{kJ}}{\mathrm{kg}}$$

$$\mathrm{Q_H} = \mathrm{T_H}(\ \mathrm{s_{4g}} - \mathrm{s_{1f}}) = 523(6.0730 \ - 2.7927) = 1762.5\frac{\mathrm{kJ}}{\mathrm{kg}}$$

$$\Delta\mathrm{U} = \mathrm{W_{net}} + \mathrm{Q_H} + \mathrm{Q_L} = 0 \ \text{ then}$$

$$\mathrm{W_{net}} = -\mathrm{Q_H} - \mathrm{Q_L} = 1\ 762{,}5 + 1\ 045{,}79 = -7160\frac{\mathrm{kJ}}{\mathrm{kg}}$$

I.3 Otto Cycle

The air-standard Otto cycle is an ideal cycle that approximates a spark-ignition internal combustion engine. This cycle is shown on the *P–v* and *T–s* diagrams in Fig I.6 and the processes are listed in *Table I. 1*. Process 1–2 is an isentropic compression of the air as the piston moves from bottom dead center (BDC) to top dead center (TDC). Heat is then added at constant volume while the piston is momentarily at rest at TDC. This process corresponds to the ignition of the fuel–air mixture by the spark and the subsequent burning in the actual engine. Process 3–4 is an isentropic expansion, and process 4–1 is the rejection of heat from the air while the piston is at BDC.

Table 0-1 The Otto Cycle Processes

Process	Energy Equation	Entropy Equation	Process Equation
Compression	u_2-u_1=-$_1w_2$	s_2-s_1=(0/T)+0	q=0, $s_1 - s_2$
Combustion	u_3-u_2=q_H	$s_2 - s_1 = \int \frac{dq_H}{T} + 0$	v_2=v_3=C
Expansion	u_4-u_3=-$_3w_4$	s_4-s_3=(0/T)+0	q=0, $s_3 - s_4$
Heat rejection	u_1-u_4=-q_L	$s_1 - s_4 = -\int \frac{dq_L}{T} + 0$	v_4=v_1=C

Since the Carnot Principle states that no engine can be more efficient than a reversible engine (a Carnot combustion engine) operating between the same high-temperature and low-temperature tanks, the Otto engine must have a lower efficiency than the Carnot efficiency. A typical gasoline automotive engine operates at about 25% to 30% of thermal efficiency. About 70 to 75% is rejected as waste heat without being converted into useful work, i.e. work delivered to the wheels. The four-stroke internal combustion engine, like that in a car, operates under this cycle

Volume compression ratio :
τ = (V + v)/v
V = volume during piston stroke
v = volume of the combustion chamber
T_H is the temperature of the hot source
T_L is the temperature of the cold source
The 0 point is called the top dead center: TDC
Point 1 is called the low dead center: BDC

Fig. I 4. Air-standard Otto cycle

Stroke 0-1: Volume increases from v to V isobarically
Stroke 1-2: The gas is compressed adiabatic and reversible (isentropic), the volume of the cylinder decreases from (V + v) to v, its pressure increases from P_1 to P_2, and its temperature increases from T_1 to T_2 ,so we will have

$$Q = 0 \rightarrow W = -\Delta H = -C_P\,\Delta T = -C_P\,(T_2 - T_1) \tag*{0-23)}$$

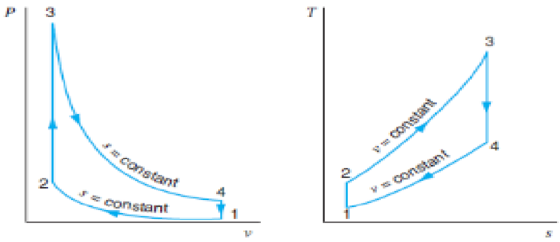
$$P_1V_1^{\gamma} = P_2V_2^{\gamma} \Rightarrow P_2 = P_1\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1\tau^{\gamma} \tag*{0-24)}$$

$$P_1V_1^{\gamma} = P_2V_2^{\gamma} \Rightarrow P_2 = P_1\left(\frac{V_1}{V_2}\right)^{\gamma} = P_1\tau^{\gamma} \tag*{0-25)}$$

$$T_1V_1^{\gamma-1} = T_2V_2^{\gamma-1} \Rightarrow T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1\tau^{\gamma-1} \tag*{0-26)}$$

$$T_2 = T_1\tau^{\gamma-1} \tag*{0-27)}$$

τ: Compression ratio
Stroke 2-3: The combustion being isochor, so the heat supplied to the gases is



$$Q_H = \Delta U = C_v\,\Delta T = C_v\,(T_3 - T_2) \tag*{0-28)}$$

Where from

$$T_3 = T_2 + \frac{Q_H}{C_v} \tag*{0-29)}$$

Since

$$\frac{P_3}{P_2} = \frac{T_3}{T_2} \tag*{0-30)}$$

therefore:

$$P_3 = P_2\frac{T_3}{T_2} = P_1\,\tau^{\gamma}\frac{T_3}{T_2} \tag*{0-31)}$$

$$P_3 = P_1\,\tau^{\gamma}\left(\frac{Q_H}{T_2}\,C_v + 1\right) \tag*{0-32)}$$

$$P_3 = P_1\,\tau^{\gamma}\left(\frac{Q_H}{T_1\tau^{\gamma-1}}\,C_v + 1\right) \tag*{0-33)}$$

Stroke 3-4: the transformation is isentropic, then

$$T_3V_3^{\gamma-1} = T_4V_4^{\gamma-1} \Rightarrow T_4 = T_3\left(\frac{V_3}{V_4}\right)^{\gamma-1} = \frac{T_3}{\tau^{\gamma-1}} \quad 0\text{-}34)$$

$$T_4 = \frac{T_3}{\tau^{\gamma-1}} \quad 0\text{-}35)$$

We have

$$P_3V_3^{\gamma} = P_4V_4^{\gamma} \Rightarrow P_4 = P_3\left(\frac{V_3}{V_4}\right)^{\gamma} = \frac{P_3}{\tau^{\gamma}} \quad 0\text{-}36)$$

Then:

$$P_4 = \frac{P_3}{\tau^{\gamma}} \quad 0\text{-}37)$$

The stroke 4-1: system gives up heat to the outside.

$$Q_L = \Delta U = C_v\,\Delta T = C_v\,(T_1 - T_4) = C_v\frac{T_1 - T_3}{\tau^{\gamma-1}} \quad 0\text{-}38)$$

Stroke 1-0: The gases are released into the atmosphere in an isobaric manner and the cycle begins again.

Performance:

$$\eta_{Otto} = \frac{W_{net}}{Q_H} = \frac{Q_H - Q_L}{Q_H} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{C_v\,(T_1 - T_4)}{C_v\,(T_3 - T_2)} = 1 - \frac{T_1 - \frac{T_3}{\tau^{\gamma-1}}}{T_3 - T_1\tau^{\gamma-1}} = 1 - \frac{1}{\tau^{\gamma-1}} \quad 0\text{-}39)$$

$$\boxed{\eta_{Otto} = 1 - \frac{1}{\tau^{\gamma-1}}} \quad 0\text{-}40)$$

I.3.1 Otto cycle exercises

I.3.1.1 **Exercise 1**

The compression ratio of a theoretical Otto cycle is 8. At the beginning of the compression stroke, the pressure is 0.1 Mpa and the temperature is 15°C. The heat supplied to the air per cycle is 1800 kJ/kg of air.

Determine:

1. The pressure and temperature at the end of each cycle evolution
2. Thermal efficiency

Data: r (air) =287 J/kg. K; Cv(air) = 710 J/kg. K; γ=1.4

I.3.1.2 Exercise 2

An Otto cycle with a compression ratio τ = 9. The intake air is at 100 kPa = (1 bar) and 20°C, and the chamber volume is 500 cm³ before the compression stroke. The temperature at the end of adiabatic expansion (expansion) is T_4 = 800 K. Cv = 0.718 kJ/kg. K g = Cp/C $_v$ = 1.4

Calculate:

1. The intake air mass
2. The temperature T_3 and the pressure P_3
3. The amount of heat added by the combustion of the air-fuel mixture
4. The thermal efficiency (yield) of this cycle
5. The average effective pressure

Solution:

1. The intake air mass

At the beginning of the calculations, we need to determine the amount of gas in the cylinder before the compression stroke. Using the ideal gas law, we can find the mass:
PV = m.r.T
where:
P is the absolute pressure of the gas, *m* is the mass of substance, *T* is the absolute temperature
V is the volume, *r* is the specific gas constant, equal to the universal gas constant divided by the molar mass (M) of the gas or mixture. For dry air r (air) = 287.1 $J.kg^{-1}. K^{-1}$.
Consequently
$m = P_1V_1/r\,T_1 = (100000 \times 500 \times 10^{-6})/\ (287,1 \times 293) = 5,95 \times 10^{-4}$ kg
In this exercise, all volumes are known:
$V_1 = V_4 = V_{max} = 500 \times 10\text{-}6\ m3 = 0,5$ litres
$V_2 = V_3 = V_{min} = V_{max}/\tau = 55,56 \times 10^{-6}\ m^3$
$(V_{max} – V_{min})$ x: Number of cylinders = total engine displacement.

2. Since the process is isentropic, we can use Laplace relations for isentropic transformations
Therefore, $T_3 = T_4.\tau$ g – 1 = 800. 9 0,4 = 1926 K
Again, we can use the ideal gas law to find the pressure at the beginning of the expansion:
P3 = m r T3/V3 = 5,95 × 10-4 x 287,1 x 1926 / 55,56 × 10-6 = 5920000 Pa = 59,2 bars

3. The amount of heat added
To calculate the amount of heat added by the combustion of the air-fuel mixture, Q_{add}, we need to use the first law of thermodynamics for the isochoric process, which states that $Q_{add} = \Delta U$, so:
The temperature at the end of the compression stroke can be determined using the relationship P, V, T for adiabatic processes between points 1 → 2.

$$T_1V_1^{\gamma-1} = T_2V_2^{\gamma-1} \Rightarrow T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1\tau^{\gamma-1} = 706\ \text{K}$$

$Q_{add} = m.c_v\ (T_3 – T_2) = 5,95 \times 10\ \text{-}4 \text{ x } 718 \text{ x } 1220 = 521,2$ J

4. Thermal efficiency

$$\eta_{Otto} = 1 - \frac{1}{\tau^{\gamma-1}} = 1 - \frac{1}{\tau^{\gamma-1}} = 1 - \frac{1}{9^{0,4}} = 0,58 = 58\%$$

5. Average effective pressure
The average effective pressure is given by the following formula**:**
W_{net} =AEP $(V_{max}\text{-}V_{min})$, which gives AEP = W_{net} /(Vmax-Vmin) = $Q_{add}.\ \eta_{Otto}\ /(V_{max}\text{-}V_{min})$
AEP = 304,7/ *(*500 × 10 -6 – 55,56 × 10 -6*)* = 685,6 kPa = 6,856 bars

I.3.1.3 Exercise 3

Air flows into a gasoline engine at 95 kPa, 300 K. The air is then compressed with a volumetric compression ratio of 8:1. In the combustion process 1300 kJ/kg of energy is released as the fuel burns. Find the temperature and pressure after combustion.

Solution:

Solve the problem with constant heat capacity.
Compression 1 to 2: $s_2 = s_1$

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{\gamma-1} = 300 \times 8^{\,0.4} = 689.2\ \text{K}$$

$$P_2 = P_1 \times \left(\frac{v_1}{v_2}\right)^{\gamma} = 95 \times 8^{1.4} = 1746\ \text{kPa}$$

Combustion 2 to 3 at constant volume: $u_3 = u_2 + q_H$

$$T_3 = T_2 + \frac{q_H}{Cv} = 689.2 + \frac{1300}{0.717} = 2502\ K$$

$$P_3 = P_2 \times \left(\frac{T_3}{T_2}\right) = 1746\left(\frac{2502}{689.2}\right) = 6338\ kPa$$

I.3.1.4 Exercise 4

To approximate an actual spark-ignition engine consider an air-standard Otto cycle that has a heat addition of 1800 kJ/kg of air, a compression ratio of 7, and a pressure and temperature at the beginning of the compression process of 90 kPa, 10°C. Assuming constant specific heat, with the value from Table A.5, determine:

1. The maximum pressure and temperature of the cycle,
2. The thermal efficiency of the cycle
3. The mean effective pressure (MEP).

Solution

1. *Compression: Reversible and adiabatic, so from Laplace laws we* get

$$P_2 = P_1 \times \left(\frac{v_1}{v_2}\right)^{\gamma} = 1372\ kPa\,;\ T_2 = T_1\left(\frac{v_1}{v_2}\right)^{\gamma-1} = 616.6\ K$$

Combustion: constant volume

$$T_3 = T_2 + \frac{q_H}{Cv} = 616.6 + \frac{1800}{0.717} = 3127\ K$$

$$P_3 = P_2 \times \left(\frac{T_3}{T_2}\right) = 1372\left(\frac{3127}{616.6}\right) = 6958\ kPa$$

Efficiency and network

$$\eta_{Otto} = 1 - \frac{1}{\tau^{\gamma-1}} = 1 - \frac{1}{\tau^{\gamma-1}} = 1 - \frac{1}{7^{0,4}} = 0,54 = 54\%$$

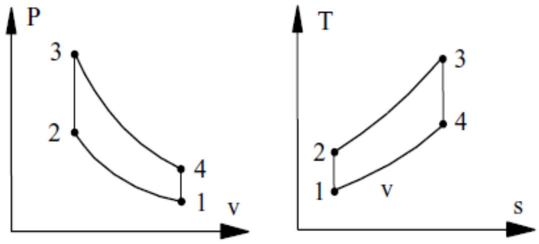
$$W_{net} = \eta_{otto} \times Q_H = 0.541 \times 1800 = 973.8\frac{kJ}{kg}$$

Displacement v_1 and MEP_{eff}

$$v_1 = \frac{RT_1}{P_1} = \frac{0.287 \times 283.2}{90} = 0.9029\frac{m^3}{kg}$$

$$v_2 = \left(\frac{1}{7}\right)v1 = 0.\frac{1290\ m^3}{kg}$$

$$MEP_{eff} = W_{net}(v_1 - v_2) = 973.8(0.9029 - 0.129)$$
$$= 1258\ kPa$$



I.3.1.5 Exercise 5

A gasoline engine takes air in at 290 K, 90 kPa and then compresses it. The combustion adds 1000 kJ/kg to the air after which the temperature is 2050 K. Use the cold air properties (i.e. constant heat capacities at 300 K) and find

1. The compression ratio,
2. The compression specific work
3. The highest pressure in the cycle.

Solution

Standard Otto Cycle

Combustion process: $T_3 = 2050$ K; $u_2 = u_3 - q_H$

$$T_2 = T_3 - \frac{q_H}{C_v} = 2050 - \frac{1000}{0.717} = 655.3\ K$$

Compression process

$$P_2 = P_1 \times \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = 90\left(\frac{655.3}{290}\right)^{3.5} = 1561\,\mathrm{kPa}$$

$$CR = \tau = \frac{v_1}{v_2} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{\gamma}-1} = \left(\frac{655.3}{290}\right)^{2.5} = 7.67$$

$$-{}_1w_2 = u_2 - u_2 = C_v(T_2 - T_1)$$

$$= 0.717(655.3 - 290) = 262\frac{kJ}{kg}$$

Highest pressure is after the combustion

$$P_3 = P_2\left(\frac{T_3}{T_2}\right) = 1561 \times \frac{2050}{655.3} = 4883\,kPa$$

I.4 Cycle Diesel [3]

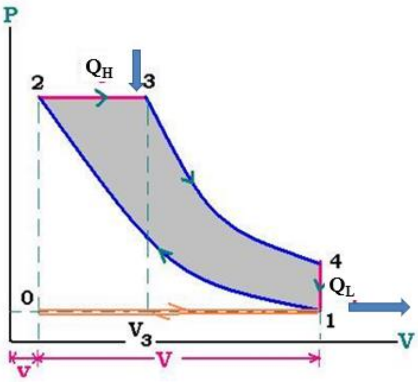


Fig. I 5. Diesel Cycle

The diesel cycle is one of the most common thermodynamic cycles that can be found in automobile engines and describes how a compression-ignition piston engine works. The diesel engine works in a similar way to the petrol engine. The most important difference is that there is no gasoline in the cylinder at the beginning of the compression stroke; Therefore, auto-ignition does not occur in diesel engines. The diesel engine uses compression ignition instead of spark ignition. Due to the high temperature reached during adiabatic compression, the fuel ignites spontaneously during injection. Therefore, no candle is needed. Before the piston stroke begins, the injectors begin injecting fuel directly into the combustion chamber. Therefore, the first part of the stroke occurs approximately at constant pressure. Higher compression ratios can be achieved in diesel engines, compared to Otto engines

Unlike the Otto cycle, the Diesel cycle does not perform isochoric heat addition. In an ideal Diesel cycle, the system running the cycle undergoes a series of four processes: two isentropic (reversible adiabatic) processes alternating with an isochoric process and an isobaric process. Since the Carnot Principle states that no engine can be more efficient than a reversible engine (a Carnot combustion engine) operating between the same high-temperature and low-temperature tanks, the diesel engine must have a lower efficiency (efficiency) than the efficiency of the Carnot cycle. A typical diesel engine has a thermal efficiency of about 30% to 35%. About 65 to 70% is rejected as waste heat without being converted into useful work, i.e., work delivered to the wheels. In general, engines using the Diesel cycle are more efficient than engines using the Otto cycle. The diesel engine has the highest thermal efficiency of all practical combustion engines. Low-speed diesel engines (used on ships) can have a thermal efficiency of more than 50%. The world's largest diesel engine peaks at 51.7%. [4]

2-3 where the heat received Qc is isobaric = constant pressure

We have:

$$P_2 = P_3 \qquad 0\text{-}41)$$

Volume changes from V_2 to V_3

$$Q_H = \Delta H = C_p\,\Delta T = C_p\,(T_3 - T_2) \qquad 0\text{-}42)$$

Where from

$$T_3 = Q_H/C_P + T_2 \qquad 0\text{-}43)$$

The volume expansion rate τ (3-4) is the ratio of V_3 and V_4

$$V_4 = V_1 \text{ and } P_2 = P_3 \qquad 0\text{-}43)$$

In the case of an ideal gas, we have:

$$V_3 = RT_3/P_3 \; and \; V_4 = V_1 = RT_1/P1 \rightarrow V_3/V_4 = (T_3/T_1) \; x \; (P_1/P_3) \quad \text{(I -44)}$$

τ (1-2) is the compression ratio during phase 1-2.

$$\tau\,(1-2) \; = \; (V_1/V_2) \; = (P_2/P_1)^{1/\gamma} \quad \text{0-45)}$$

Therefore:

$$\tau\ (3\text{-}4) = (T_3/T_1) \;/\; \tau\ (1\text{-}2)^{\gamma} \quad \text{0-46)}$$

Yield : $\boxed{\eta = 1 - (|Q_L|/Q_H)}$ 0-47)

Thermal efficiency of the diesel cycle

In general, thermal efficiency or efficiency η_{th} of a heat engine is defined as the ratio between the network it provides, W_{net} , on the heat it receives, Q_H .

$$\eta_{th} = \frac{W_{net}}{Q_H} \quad \text{0-48)}$$

It represents the fraction of heat, Q_H, which is converted into work. Since energy is conserved according to the first law of thermodynamics and the energy cannot be converted to function completely, the heat input, Qc, must be equal to the work performed, W, plus the heat that must be dissipated as waste heat Q_L into the environment. Therefore, we can rewrite the formula for thermal efficiency as follows:

$$\eta_{th} = \frac{W_{net}}{Q_H} = \frac{Q_H - Q_L}{Q_H} = 1 - \frac{Q_L}{Q_H} \quad \text{0-49)}$$

The absorbed heat occurs during the combustion of the fuel-air mixture, when the spark occurs, roughly at constant volume. Since during an isochoric process, no work is done by or on the system, the first law of thermodynamics stimulates $\Delta U = \Delta Q$. Therefore, the added heat and the rejected heat are given by:

$$Q_H = m.c_p\ (T_3 - T_2);\ Q_L = m.c_v\ (T_4 - T_1) \quad \text{0-50)}$$

By substituting these expressions for the heat added and rejected in the expression of thermal efficiency, we obtain:

$$\eta_{th} = \frac{W_{net}}{Q_H} = \frac{Q_H - Q_L}{Q_H} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{C_v\ (T_1 - T_4)}{C_P\ (T_3 - T_2)} = 1 - \frac{1}{\gamma}\frac{(T_1 - T_4)}{(T_3 - T_2)} \quad \text{0-51)}$$

This equation can be rearranged in this form

$$\boxed{\eta_{Diesel} = 1 - \frac{1}{\tau^{\gamma-1}}\left(\frac{(\alpha^{\gamma} - 1)}{\gamma(\alpha - 1)}\right)} \quad \text{0-52)}$$

With:

η_{Diesel} is the maximum thermal efficiency of a diesel cycle

α is the rate of relaxation (V_3/V_{2}) (i.e. the ratio of volumes at the end and beginning of the combustion phase)

τ is the compression ratio

$\gamma = c_p/c_v = 1{,}4$

This is a very useful conclusion, as it is desirable to achieve a high compression ratio to extract more mechanical energy from a given mass of fuel. As we concluded in the previous section, the thermal efficiency of the Otto cycle is also a function of the compression ratio τ and g.

$$\boxed{\eta_{Otto} = 1 - \frac{1}{\tau^{\gamma-1}}} \quad \text{0.54)}$$

When we compare the two outputs Otto and Diesel, we see that for a given compression ratio (τ) the Otto cycle is more efficient than the Diesel cycle. But diesel engines are generally more efficient because they can operate at higher compression ratios.
In regular Otto engines, the compression ratio has its limits. The compression ratio in a gasoline engine will usually not be much higher than 10:1. Higher compression ratios will subject gasoline engines to engine knocking, which is caused by auto-ignition. In diesel engines, the risk of fuel auto-ignition is minimal because diesel engines are compression-ignition engines and there is no fuel in the cylinder at the beginning of the compression stroke.

Table 0-2 The Diesel Cycle Processes

Process	Energy Equation	Entropy Equation	Process Equation
Compression	$u_2-u_1=-{}_1w_2$	$s_2-s_1=(0/T)+0$	$q=0, s_1 - s_2$
Combustion	$u_3-u_2=q_H-{}_2w_3$	$s_2 - s_1 = \int \frac{dq_H}{T} + 0$	$P_2=P_3=C$
Expansion	$u_4-u_3=-{}_3w_4$	$s_4-s_3=(0/T)+0$	$q=0, s_3 - s_4$
Heat rejection	$u_1-u_4=-q_L$	$s_1 - s_4 = -\int \frac{dq_L}{T} + 0$	$v_4=v_1=C$

I.4.1 Diesel Cycle Exercises

I.4.1.1 Exercise1

Let's assume the Diesel cycle, which is one of the most common thermodynamic cycles that can be found in automobile engines. One of the key parameters of these engines is the change in volume between the top dead center (TDC) and the bottom dead center (BDC). The ratio of these volumes (V_1/V_2) is called the compression ratio. Also, the expansion ratio (V_3/V_2), which is the ratio of volumes at the end and beginning of the combustion phase, is called the cut-off rate.

In this example, we consider a diesel cycle with a compression ratio of $\tau=20$ and a cut-off rate $\alpha = (V_3/V_2) = 2$. The air is at 100 kPa=1 bar, 20°C and the chamber volume is 500 ccm³before the compression stroke. $C_v = 0{,}718$ kJ/kg. K; $\gamma = C_p/C_v = 1{,}4$
Calculate:

1. The intake air mass
2. The temperature T_2 and the pressure P_2
3. The temperature T_3
4. The amount of heat added by the combustion of the air-fuel mixture
5. The efficiency or thermal yield of this cycle
6. The average effective pressure

Solution:

1. At the beginning of the calculations, we need to determine the amount of gas in the cylinder before the compression stroke. Using the law of the ideal gas, we can find the mass:

$PV = mrT$
Where:
P is the absolute pressure of the gas
m is the mass of substance
T is the absolute temperature
V is the volume
r is the specific gas constant, equal to the universal gas constant divided by the molar mass (M) of the gas or mixture. For dry air $r = 287{,}1$ J.kg^{-1}.K^{-1}.
Consequently

$$m = \frac{P_1 V_1}{r\,T_1} = \frac{100000 \times 500\,.10^{-6}}{287{,}1 \times 293} = 5{,}95\,.10^{-4}\,kg$$

In this issue, all volumes are known:

$$V_1 = V_4 = V_{max} = 500\,.10^{-6}\,m^3\ (0{,}5\ l)$$

$$V_2 = V_{min} = \frac{V_{max}}{\tau} = 25.10^{-6}\, m^3$$

$(V_{max} - V_{min}) x\ Number\ of\ cylinders\ =\ total\ engine\ displacement.$

2. The temperature T_2 and pressure P_2

The temperature at the end of the compression stroke can be determined using the relationship P, V, T for adiabatic processes between points1 → 2.

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1} \Rightarrow T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{\gamma-1} = T_1 \tau^{\gamma-1}$$

T_2 = 971 K

$$P_2 = \frac{mrT_2}{V_2} = 5{,}95\,.10^{-4}\,.287{,}1.\frac{971}{25.10^{-6}} = 6635000\, Pa = 66{,}35\, bar$$

3. Temperature T_3

Like the process 2 → 3 occurs at constant pressure, the equation of state of ideal gases gives

$$T_3 = T_2 \left(\frac{V_3}{V_2}\right) = 971 * 2 = 1942\, K$$

4. The amount of heat added

To calculate the amount of heat added by the combustion of the air-fuel mixture, Q_{add}, we need to use the first law of thermodynamics for the isochorus process, which states that $Q_{add} = \Delta U$, therefore:

$$Q_{add} = m.C_P\,(T_3 - T_2) = 5{,}95\,.10^{-4}.1010\,.971 = 583{,}5\, J$$

5. Thermal efficiency

$$\eta_{Diesel} = 1 - \frac{1}{\tau^{\gamma-1}}\left(\frac{(\alpha^{\gamma} - 1)}{\gamma(\alpha - 1)}\right) = 0{,}6467 = 64{,}7\%$$

6. Mean effective pressure (MEP)

The average effective pressure is given by the following formula:

$$W_{net} = \text{MEP}\,(V_{max} - V_{min})$$

$$\text{MEP} = \frac{Wnet}{V_{max} - V_{min}} = Q_{add}.\frac{\eta_{Diesel}}{V_{max} - V_{min}}$$

$$\text{MEP} = \frac{583{,}5x0{,}6467}{(500.10^{-6} - 25\,.10^{-6})} = 794{,}3\, kPa = 7{,}943\, bar$$

I.4.1.2 Exercice2

The compression ratio of a theoretical diesel cycle is 16. At the beginning of the compression stroke, the pressure is 0.1 MPa and the temperature is 15°C. The heat supplied to the air per set cycle of 1800 kJ/kg of air.

Determine:

1. The pressure and temperature at the end of each cycle evolution
2. Thermal efficiency

I.4.1.3 Exercice3

A diesel engine has a state before compression of 95 kPa, 290 K, and a peak pressure of 6000 kPa, a maximum temperature of 2400 K. Find:

1. The volumetric compression ratio
2. The thermal efficiency.

Solution:

1. Compression process (isentropic) then from Laplace laws:

$$\left(\frac{P_2}{P_1}\right) = \left(\frac{v_1}{v_2}\right)^{\gamma} = Compressin\ Ratio^{\gamma} = CR^{\gamma} = \tau^{\gamma}$$

$$\tau = CR = \frac{v_1}{v_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} = \left(\frac{6000}{95}\right)^{\frac{1}{1.4}} = 19.32$$

$$T_2 = T_1\left(\frac{P2}{P1}\right)\gamma - \frac{1}{\gamma} = 290 \times \left(\frac{6000}{95}\right)^{0.2857} = 947.9\,K$$

Combustion and expansion volumes

$$v_3 = v_2 \times \frac{T_3}{T_2} = v_1 . \frac{T_3}{T_2 \times CR}\, ; v_4 = v_1$$

Expansion process, isentropic

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{\gamma-1} = T_3\left[\frac{T_3}{\mathrm{CR}\,.\,T_2}\right]^{\gamma-1} = 2400 \times \left[\frac{2400}{19.32 \times 947.9}\right]0.4 = 1064.6\ \mathrm{K}$$

Efficiency

$$\eta = 1 - \frac{1}{\gamma}.\frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{1}{1.4}.\frac{1064.6 - 290}{2400 - 947.9} = 0.619$$

I.4.1.4 Exercise4

At the beginning of compression in a diesel cycle, $T = 300$ K, $P = 200$ kPa, and after combustion (heat addition) is complete, $T = 1500$ K and $P = 7.0$ MPa. Find:

1. The compression ratio.
2. The thermal efficiency.
3. The mean effective pressure.

Solution:

1. Compression process (isentropic)

$$P_2 = P_3 = 7000\,kPa \Rightarrow \frac{v_1}{v_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} = \left(\frac{7000}{200}\right)^{0.7143} = 12.67$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 300\left(\frac{7000}{200}\right)^{0.2857} = 828.4\,K$$

2. Expansion process (isentropic), first get the volume ratios

$$\frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{1500}{828.4} = 1.81$$

$$\frac{v_4}{v_3} = \frac{v_1}{v_3} = \left(\frac{v_1}{v_2}\right)\left(\frac{v_2}{v_3}\right) = \frac{12.67}{1.81} = 7$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{\gamma-1} = \left(\frac{1500}{7}\right)^{0.4} = 688.7\,K$$

$$q_L = C_v(T_4 - T_1) = 0.717(688.7 - 300) = 278.5\frac{kJ}{kg}$$

$$q_H = h_3 - h_2 \approx C_P(T_3 - T_2) = 1.004(1500 - 828.4) = \frac{674kJ}{kg}$$

Overall performance

$$\eta = 1 - \frac{q_L}{q_H} = 1 - \frac{278.5}{674} = 0.587$$

3. $w_{net} = q_{net} = q_H - q_L = 674 - 278.5 = \frac{395.5kJ}{kg}$

$$v_{max} = v_1 = \frac{R\,T_1}{P_1} = 0.287 \times \frac{300}{200} = 0.4305\frac{m^3}{kg}$$

$$v_{min} = v_{max}\frac{v1}{v2} = \frac{0.4305}{12.67} = \frac{0.034\,m^3}{kg}$$

$$MEP = \frac{w_{net}}{v_{max} - v_{min}} = \frac{395.5}{0.4305 - 0.034} = 997\,kPa$$

I.4.1.5 Exercise5

An air-standard diesel cycle has a compression ratio of 20, and the heat transferred to the working fluid per cycle is 1800 kJ/kg. At the beginning of the compression process, the pressure is 0.1 MPa and the temperature is 15ºC. Determine

1. The pressure and temperature at each point in the cycle.
2. The thermal efficiency.
3. The mean effective pressure.

Response

1. T_2 = 955.2K ; P_2 = 6.629 MPa ; T_3 = 2748 K ; v_3 = 0.118 96 m3/kg ; T_4 = 1265 K
2. η_{th} = 61.1%
3. MEP= 1400 kPa

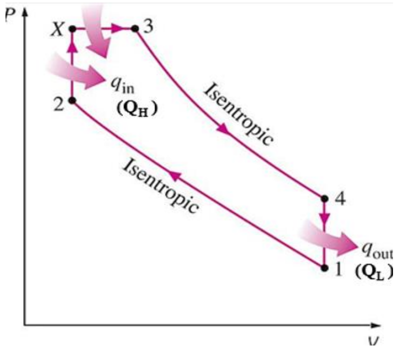


Fig. I 6. Sabathé cycle

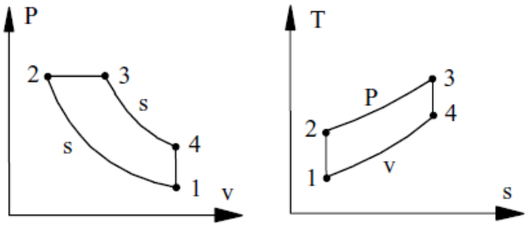
I.5 Sabathé cycle; dual cycle.

The Sabathé cycle, also known as the dual cycle, is a thermodynamic cycle that combines aspects of both the Otto and Diesel cycles. It is often used to model internal combustion engines, where part of the heat addition occurs at constant volume (like in the Otto cycle) and another part at constant pressure (similar to the Diesel cycle).

Key Steps in the Sabathé Cycle:

1. Isentropic Compression
 Air is compressed, increasing its temperature and pressure.
2. Constant Volume Heat Addition
 Heat is added at constant volume, causing a rapid increase in temperature and pressure.
3. Constant Pressure Heat Addition
 Further heat is added at constant pressure, which increases the volume of the gas.
4. Isentropic Expansion
 The gas expands, producing work as it drives the piston or turbine.
5. Constant Volume Heat Rejection
 Heat is rejected at constant volume, completing the cycle.

Characteristics and Applications:



The Sabathé cycle balances the benefits of both constant-volume and constant-pressure heat addition. This makes it ideal for engines requiring high efficiency and flexibility, especially in conditions where varying load demands are present. It's commonly used in diesel and some hybrid-type engines in transportation and power generation due to its efficiency advantages over using the Otto or Diesel cycles alone.

I.5.1 Dual cycle exercises

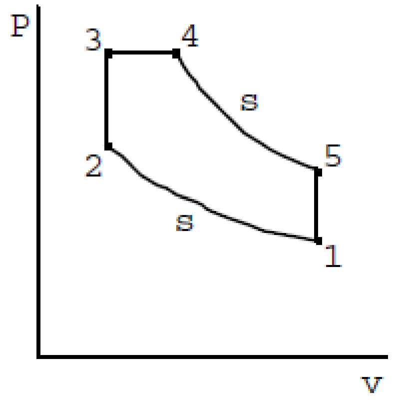
I.5.1.1 Exercise1

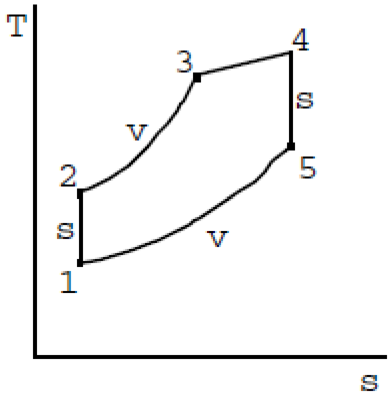
In the Otto cycle, all the heat transfer Q_H occurs at constant volume. It is more realistic to assume that part of Q_H occurs after the piston has started its downward motion in the expansion stroke. Therefore, consider a cycle identical to the Otto cycle, except that the first two-thirds of the total Q_H occurs at constant volume and the last one-third occurs at constant pressure. Assume that the total qH is 2100 kJ/kg, that the state at the beginning of the compression process is 90 kPa, 20°C, and that the compression ratio is 9.

1. Calculate the maximum pressure, the temperature, and the thermal efficiency of this cycle.
2. Compare the results with those of a conventional Otto cycle having the same given variables.

$P_1 = 90$ kPa, $T_1 = 20$°C
$\tau = V_1/V_2 = 7$

Solution





$$a)\ Q_{23} = (2/3) \times 2100 = 1400\ kJ/kg;$$

$$Q_{34} = \frac{2100}{3} = 700\frac{kJ}{kg}$$

$$b)\ P_2 = P_1\left(\frac{V_1}{V_2}\right)^{\gamma} = 90(9)1.4 = 1951\ kPa$$

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1} = 293.15(9)0.4 = 706\ K$$

$$T_3 = T_2 + \frac{Q_{23}}{C_V} = 706 + \frac{1400}{0.717} = 2660\ K$$

$$P_3 = \frac{P_2 T_3}{T_2} = 1951\left(\frac{2660}{706}\right) = 7350.8\ kPa = P4$$

$$T_4 = T_3 + \frac{Q_{34}}{C_P} = 2660 + \frac{700}{1.004} = 3357\ K$$

$$\frac{V_5}{V_4} = \frac{V_1}{V_4} = \frac{P_4}{P_1} \times \frac{T_1}{T_4} = \frac{7350.8}{90} \times \frac{293.15}{3357} = 7.131$$

$$T_5 = T_4 \left(\frac{V_4}{V_5}\right)^{\gamma-1} = 3357 \left(\frac{1}{7.131}\right) 0.4 = 1530\, K$$

$$Q_L = C_V (T_5 - T_1) = 0.717(1530 - 293.15)$$
$$= 886.2 \frac{kJ}{kg}$$

$$\eta_{th} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{886.2}{2100} = 0.578$$

$$\eta_{th} = 1 - (9) - 0.4 = 0.585, small\ difference$$

I.6 Brayton cycle: the ideal cycle for gas-turbine engines

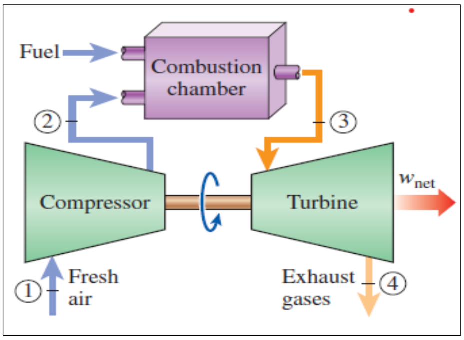


Fig. I 7. An open-cycle gas-turbine engine

The Brayton cycle is used for gas turbines only where both the compression and expansion processes take place in rotating machinery. Gas turbines usually operate on an *open cycle,* as shown in Figure I.7. Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. The high-pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure. The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power. The exhaust gases leaving the turbine are thrown out (not recirculated), causing the cycle to be classified as an open cycle. The open gas-turbine cycle described above can be modeled as a *closed cycle,* as shown in Figure I.8, by utilizing the air-standard assumptions. Here the compression and expansion processes remain the same, but the combustion process is replaced by a constant-pressure heat-addition process from an external source, and the exhaust process is replaced by a constant-pressure heat-rejection process to the ambient air.

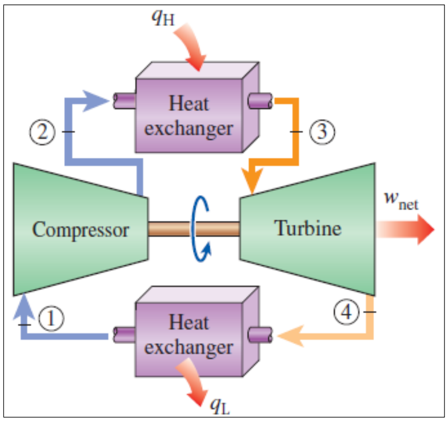


Fig. I 8. A closed-cycle gas-turbine engine

The ideal cycle that the working
fluid undergoes in this closed loop is the Brayton cycle, which is made up
of four internally reversible processes:
1-2 Isentropic compression (in a compressor)
2-3 Constant-pressure heat addition
3-4 Isentropic expansion (in a turbine)
4-1 Constant-pressure heat rejection
The *T-s* and *P-v* diagrams of an ideal Brayton cycle are shown in Figure. I.9 and Figure I.10

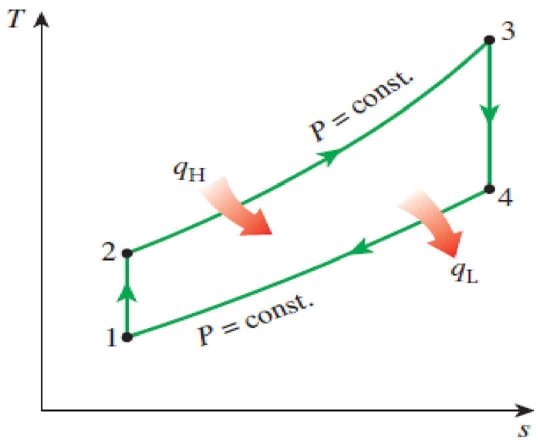


Fig. I 9. T-s diagrams ideal Brayton cycle

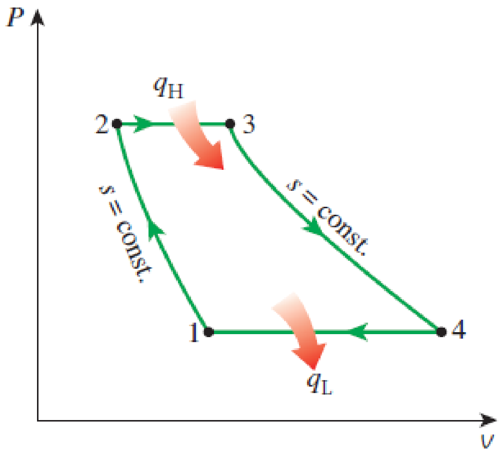


Fig. I 10 P-v diagrams ideal Brayton cycle

I.6.1 Brayton Efficiency

The process 1-2 is adiabatic

$$T_2 = T_1[P_2/P_1]^{(\gamma - 1)/\gamma} \tag{I.55}$$

$\Delta T = T_2 - T_1$,

$$\text{There fore } T_2 - T_1 = T_1[[P_2/P_1]^{(\gamma - 1)/\gamma} - 1] \tag{I.56}$$

With $\tau = P_2/P_1$

$$T_2 - T_1 = T_1[[\tau]^{(\gamma - 1)/\gamma} - 1] \tag{I.57}$$

$$\text{and } W(\text{receipt}) = Cp\Delta T = Cp\ (T_2 - T_1) \tag{I.58}$$

2-3: At the combustion chamber:

The process is isobaric $P_2 = P_3$.

the temperature rises from T2 to T3

We have: $Q_H = C_p\Delta T = C_p\ (T_3 - T_2)$

(I.59)

Process 3-4: At the turbine:

The temperature drops from T_3 to T_4.

The process is adiabatic,

so, $$Q = 0 \rightarrow W = -\Delta H = -Cp\Delta T \tag{I.60}$$

W(supplied)= - $Cp\Delta T$ with $\Delta T = T_4 - T_3$

$$W(\text{supplied}) = -Cp\ (T_4 - T_3) \tag{I.61}$$

$T4 = T_3[P_4/P_3]^{(\gamma - 1)/\gamma}$

Therefore $$T_4 - T_3 = T_3[[P_4/P_3]^{(\gamma - 1)/\gamma} - 1] \tag{I.62}$$

Where $$T_4 = T_3[1/\tau]^{(\gamma - 1)/\gamma} \text{ and } T_4 - T_3 = T_3[[1/\tau]^{(\gamma - 1)/\gamma} - 1] \tag{I.63}$$

4-1 process at the cooling level:

During the process 4-1, which is isobaric, the gas is cooled by giving up a quantity of heat Q_L

We have $Q_L = -C_p\Delta T = -Cp\ (T_1 - T_4)$

$$T_1 - T_4 = T_1 - T_3[1/\tau]^{(\gamma - 1)/\gamma} \tag{I.64}$$

The efficiency of this machine is:

$\eta_{\text{Brayton}} = W(\text{net})/W(\text{receipt})$ (I.65)

$W(\text{net}) = W(\text{receipt}) - W(\text{supplied}) = C_p (T_2 - T_1) - C_p(T_4 - T_3)$

$\eta_{\text{Brayton}} = [C_p(T_2 - T_1) - C_p(T_4 - T_3)]/ [C_p(T_2 - T_1)] = 1 - [|(T_4 - T_3)|/(T_2 - T_1)]$ (I.66)

$\eta_{\text{Brayton}} = (Q_H - Q_L)/Q_H = 1-(Q_L/Q_H)=1-[C_p(T_1 - T_4)/C_p(T_3 - T_2)]=1-[|(T_1 - T_4)|/(T_3 - T_2)]$ (I.65)

Let be the compression ratio: $\tau_p = P_2/P_1 = P_3/P_4$

The Brayton efficiency Cycle is then (I.68)

$$\boxed{\eta_{\text{brayton}} = 1 - [[1/\tau_p]^{(\gamma - 1)/\gamma}]}$$

Table 0-3 The Brayton Cycle Processes

Process	Energy Equation	Entropy Equation	Process Equation
Compressor	$W_C = h_2 - h_1$	$s_2 - s_1 = (0/T) + 0$	$Q=0,\ s_1 = s_2$
Combustion	\$Q_H = h_3 - h_2$	$s_2 - s_1 = \int \frac{dq_H}{T} + 0$	$P_2 = P_3 = C$
Turbine	$W_T = -(h_4 - h_3)$	$s_4 - s_3 = (0/T) + 0$	$Q=0, s_3 - s_4$
Heat exchanger	$Q_L = -(h_1 - h_4)$	$s_1 - s_4 = -\int \frac{dq_L}{T} + 0$	$P_4 = P_1 = C$

I.6.2 Brayton Cycle Exercises

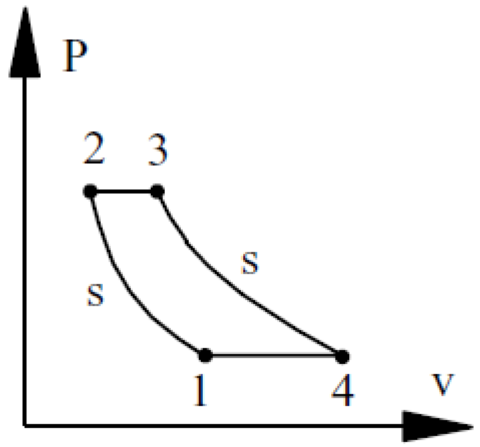
I.6.2.1 Exercise 1

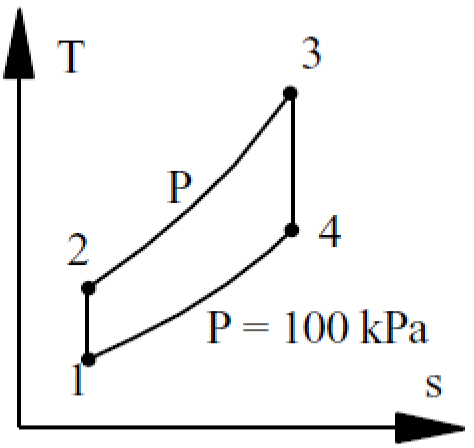
Consider an ideal air-standard Brayton cycle in which the air into the compressor is at 100 kPa, 20°C, and the pressure ratio across the compressor is 12:1. The maximum temperature in the cycle is 1100°C, and the airflow rate is 10 kg/s. Assume constant specific heat for the air, value from Table A.5. Determine:

1. The compressor work,
2. The turbine work,
3. The thermal efficiency of the cycle.

Data: Cp = 1.0035kJ/kg.K

Solution:





1. Compression ratio $P_2/P_1 = 12$

Max temperature $T_3 = 1100°C$
m. = 10 kg/s
The compression is reversible and adiabatic so constant s. From Laplace laws

$$T_2 = T_2\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 293.2(12)^{0.286} = 596.8K$$

Energy equation with compressor work

$W_C = -{}_1W_2 = C_P(T_2 - T_1) = 1.004(596.8 - 293.2) = 304.8$ kJ/kg

2. The expansion is reversible and adiabatic so constant s.

$$T_4 = T_2\left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = 13733.2\left(\frac{1}{12}\right)^{0.286} = 674.7K$$

3. Energy equation with turbine work out

$W_T = C_P(T_3 - T_4) = 1.004(1373.2 - 674.7) = 701.3$ kJ/kg

Scale the work with the mass flow rate

$W_C = m.W_C = 3048$ kW, $W_T = m.W_T = 7013$ kW

Energy added by the combustion process

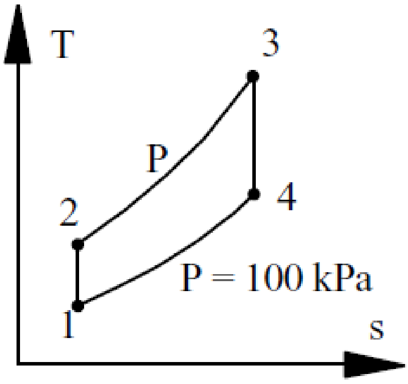
$Q_H = C_P(T_3 - T_2) = 1.004(1373.2 - 596.8) = 779.5$ kJ/kg

$\eta_{th} = W_{net}/Q_H = (701.3 - 304.8)/779.5 = 0.509$

I.6.2.2 Exercise 2

A large stationary Brayton cycle gas-turbine power plant delivers a power output of 100 MW to an electric generator. The minimum temperature in the cycle is 300 K, and the maximum temperature is 1600 K. The minimum pressure in the cycle is 100 kPa, and the compressor pressure ratio is 14 to 1.

1. Calculate the power output of the turbine.
2. What fraction of the turbine output is required to drive the compressor?
3. What is the thermal efficiency of the cycle?



Solution

Minimum T : $T_1 = 300$ K

Maximum T : $T_3 = 1600$ K

Pressure ratio: $P_2/P_1 = 14$

1. Solve using constant C_{P0}

Compression in compressor: $s2 = s1 \Rightarrow$

$$T2 = T1\left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}} = 300(14)^{0.286} = 638.1\,K$$

$W_C = h_2 - h_1 = C_P(T_2 - T_1) = 1.004\ (638.1 - 300) = 339.5$ kJ/kg

Expansion in turbine: $s4 = s3 \Rightarrow$

$$T_4 = T_2\left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = 1600\left(\frac{1}{14}\right)^{0.286} = 752.2K$$

$W_T = h_3 - h_4 = C_P(T_3 - T_4) = 1.004\ (1600 - 752.2) = 851.2$ kJ/kg

$W_{net} = 851.2 - 339.5 = 511.7$ kJ/kg

2. $\dot{m} = \frac{\dot{W}net}{Wnet} = \frac{100000}{511.7} = 195.4\frac{kg}{s}$

$Q_H = C_P\ (T_3 - T_2) = 1.004\ (1600 - 638.1) = 965.7$ kJ/kg

3. $\eta_{th} = W_{net}/Q_H = 511.7/965.7 = 0.530$

I.6.2.3 Exercise 3

A Brayton cycle produces 14 MW with an inlet state of 17°C, 100 kPa, and a compression ratio of 16:1. The heat added in the combustion is 960 kJ/kg. What are

1. The mass flow rate of air,
2. The highest temperature

Response:

$T_3 = 1596.5$K

$\dot{m} = 26.66\frac{kg}{s}$

I.6.2.4 Exercise 4

Repeat Exercise 5 but assume that the compressor has an isentropic efficiency of 85% and the turbine an isentropic efficiency of 88%.

Response:

1. $\dot{W}_T = m = 214.2\, MW$
2. $\frac{W_C}{W_T} = \frac{399.4}{749.1} = 0.533$
3. $\eta_{th} = 0.386$

I.7 Stirling and Ericsson Cycles

There are two other cycles that involve an isothermal heat addition process at Tc and an isothermal heat rejection process at T_f: the Stirling cycle and the Ericsson cycle. They differ from the Carnot cycle in that the two isentropic processes are replaced by two constant-volume regeneration processes in the Stirling cycle, and by two constant-pressure regeneration processes in the Ericsson cycle. The Stirling and Ericsson cycles are both thermodynamic cycles used as models for high-efficiency heat engines. Both aim to approximate the Carnot cycle's efficiency, often incorporating regenerative processes to enhance heat exchange. They are distinguished primarily by their types of heat addition and rejection processes.

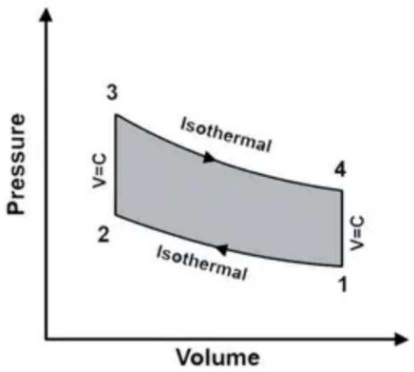


Fig. I 11.Stirling cycle Pv diagram

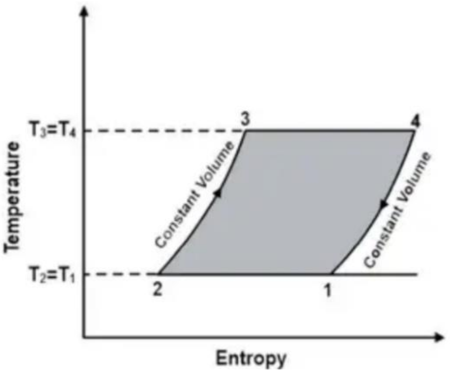


Fig. I 12. Stirling cycle Ts diagram

I.7.1 Stirling Cycle

The Stirling cycle is a closed-cycle engine that uses two isothermal and two isochoric (constant-volume) processes:

- *1-2 Isothermal Compression:* The working fluid is compressed at a constant temperature, and heat is released to an external heat sink.
- *2-3 Isochoric Compression (Constant Volume) Heating:* The fluid's temperature is increased at constant volume, typically through a regenerative heat exchange.
- *3-4 Isothermal Expansion:* The fluid expands at constant temperature, absorbing heat from an external heat source.
- *4-1 Isochoric Expansion Cooling:* The fluid cools at a constant volume, transferring heat to a regenerative device to be reused in the cycle.

Characteristics of Stirling cycle:

The Stirling cycle is used in Stirling engines, often for applications requiring high efficiency with relatively low power outputs, such as in solar energy and small-scale power generation.
Regeneration (the process of storing heat in a regenerator and reusing it) is essential in the Stirling cycle to improve efficiency by reducing the heat lost.

I.7.2 Ericsson Cycle

The Ericsson cycle is similar to the Stirling cycle but uses two isothermal and two isentropic (adiabatic) processes instead of isochoric processes:

- *1-2 Isothermal Compression:* The gas is compressed at constant temperature, with heat removed to maintain the isothermal condition.
- *2-3 Isobaric Expansion:* The gas expands without any heat exchange, reducing its temperature and doing work.
- *3-4 Isothermal Expansion:* The gas expands at constant temperature while absorbing heat from an external source.
- *4-1 Isentropic Expansion:* The gas is compressed without heat exchange, returning to the initial state.

Characteristics of Ericsson cycle

The Ericsson cycle theoretically has high efficiency and is idealized as close to the Carnot cycle due to its isothermal heat addition and rejection.

Rarely implemented practically due to the challenges of maintaining isothermal conditions in real engines, though it serves as a model for designing efficient heat engines.

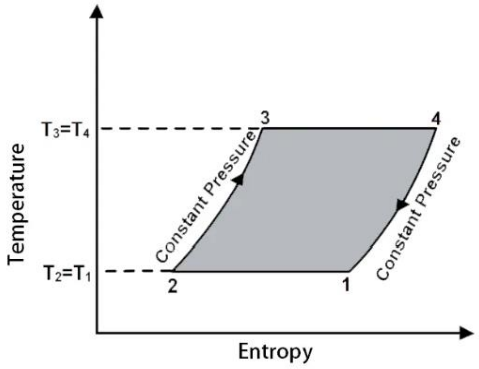


Fig. I 13. Ericson cycle Ts diagram

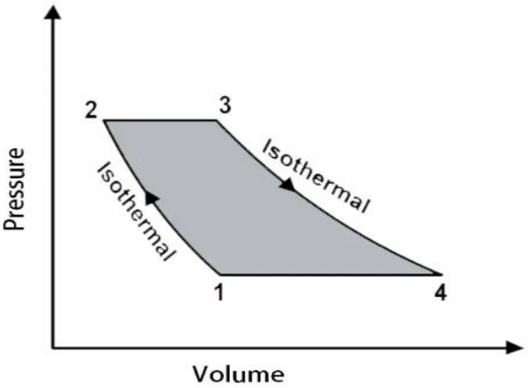


Fig. I 14. Ericson cycle Pv diagram

Comparison: Stirling vs. Ericsson Cycle

Processes: The Stirling cycle includes isochoric (constant-volume) processes, while the Ericsson cycle incorporates isentropic (adiabatic)

Regeneration: Both cycles use regeneration to improve efficiency, but the Stirling cycle relies on it more extensively.

Practicality: Stirling engines are more common than Ericsson engines in practical applications due to simpler operation requirements and ease of achieving constant volume processes over isentropic compression and expansion.

I.7.3 Stirling Cycle thermal efficiency

For the calculation of the thermal efficiency of the Stirling cycle (the same method may be used for the Ericsson cycle), the basic equation is again

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{\sum Q}{Q_{in}} \quad 0\text{-}66)$$

The first law for a closed system is

$$Q = \Delta U + W \quad 0\text{-}67)$$

$$\Delta U = 0 \; for \; T = Constante$$

$$Q_{in} = W = mRT_1 ln\left(\frac{V_2}{V_1}\right) \quad 0\text{-}68)$$

Similarly,

$$Q_{out} = -mRT_3 ln\left(\frac{V_3}{V_4}\right) \qquad \text{0-69)}$$

However,

$$V_2 = V_3 \; and \; V_4 = V_1 \qquad \text{0-70)}$$

and so

$$\eta_{th} = \frac{T_1 - T_3}{T_1} = 1 - \frac{T_L}{T_H} \qquad \text{0-71)}$$

where T_L is the low temperature and T_H is the high temperature. The thermal efficiency is the same as in the Carnot cycle.

I.7.4 Stirling and Ericson exercises

I.7.4.1 Exercise 1

An engine operating on the air-standard Stirling cycle is examined and found to have the following conditions: $P_1 = 725$ kPa, $T_1 = 590$ K, and $V_1 = 0.056\,7$ m^3 at the beginning of isothermal expansion; $V_2/V_1 = 1.5$; $T_3 = 300$ K.
For one cycle, determine

1. The network
2. The thermal efficiency
3. The mean effective pressure
4. The heat rejected
5. The heat added.

Given: air-standard Stirling cycle
P_1 725 kPa
$T_1 = 590$ K $= T_2$
$V_1 = 0.056\,7$ m^3
$V_2/V_1 = 1.5$
$T_3 = 300$ K

Solution :

1. For the network

$$Q_{in} = mRT_1 ln\frac{V_2}{V_1} = p_1 V_1 ln\frac{V_2}{V_1}$$

$$Q_{in} = \left(725\,\frac{kN}{m^2}\right)(0.056\,7\,m^3)ln(1.5)$$

$$Q_{in} = 16.7\,kJ$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$

$$W_{net} = \eta_{th} Q_{in} = \left[1 - \frac{T_L}{T_H}\right] Q_{in}$$

$$W_{net} = \left[1 - \frac{300}{590}\right](16.7\,kJ)$$

$$W_{net} = 8.2\,kJ$$

2. The thermal efficiency is given by

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{300}{590} == 0.491\ 5$$

3. The mean effective pressure,

$$p_m = \frac{W_{net}}{V_D} = \frac{W_{net}}{V_2 - V_1} = \frac{8.2}{(0.056\ 7)(1.5 - 1)} = 289\ \frac{kN}{m^2}$$

4. The heat rejected,

$$Q_{out} = -mRT_3 \ln\frac{V_3}{V_4}$$

Since $V_1 = V_4$ and $V_2 = V_3$, then

$$\frac{V_2}{V_1} = \frac{V_3}{V_4} = 1.5$$

$$m = \frac{p_1 V_1}{RT_1} = \frac{(725\ )(0.056\ 7\ )}{(0.287\ 08\ )(590\ )} = 0.243\ kg$$

$$Q_{out} = -(0.243\ kg)(0.287\ 08\ )(300\ K) ln(1.5) = -8.5\ kJ$$

5. For the heat added,

$$Q_{in} = 16.7\ kJ$$

I.7.4.2 Exercise 2

In an air-standard Ericsson cycle, the maximum pressure is 4.1 MPa and the minimum pressure is 210 kPa. The heat supplied is 581 kJ/kg, and the minimum temperature is 21°C. Determine:

1. The cycle work
2. the heat rejected
3. The heat stored in the regenerator
4. The entropy changed during the heat addition.

Given: air-standard Ericsson cycle

$P_1 = 4.1$ MPa $= P_4$
$P_2 = 210$ kPa $= P_3$
$Q_{in} = 581$ kJ/kg
$T_3 = 21$°C + 273 = 294 K $= T_L$

Solution :

1. For the network,

$$\eta_{th} = \frac{W_{net}}{Q_{in}}$$

$$W_{net} = \eta_{th} Q_{in} = \left[1 - \frac{T_L}{T_H}\right] Q_{in}$$

Solve for $T_H = T_1$, thus,

$$Q_{in} = RT_1 ln\frac{V_2}{V_1} = RT_1 ln\frac{p_1}{p_2}$$

$$T_1 = \frac{Q_{in}}{Rln\frac{p_1}{p_2}} = \frac{581}{(0.287\ 08\ )ln\left(\frac{4\ 100}{210}\right)} = 681K$$

$$W_{net} = \left[1 - \frac{294}{681}\right]\left(581\ \frac{kJ}{kg}\right) = 330\ \frac{kJ}{kg}$$

2. For the heat rejected,

$$Q_{out} = -RT_3 ln\frac{V_3}{V_4} = -RT_3 ln\frac{p_4}{p_3}$$

$$Q_{out} = -\left(0.287\ 08\ \frac{kJ}{kg-K}\right)(294\ K) ln\left(\frac{4\ 100}{210}\right) = -251\ \frac{kJ}{kg}$$

3. The heat stored in the regenerator,

$$Q_{reg} = c_p(T_3 - T_2) = \left(1.004\,7\ \frac{kJ}{kg-K}\right)(294 - 681)K = -288.8\ kJ$$

4. Since the air is assumed to be an ideal gas, the change in entropy is 0

$$\Delta s_{1-2} = c_p ln\frac{T_2}{T_1} - Rln\frac{p_2}{p_1}$$

$$\Delta s_{1-2} = -\left(0.287\,08\ \frac{kJ}{kg-K}\right) ln\left(\frac{210}{4\,100}\right)$$

$$\Delta s_{1-2} = 0.853\,1\ \frac{kJ}{kg-K}$$

I.7.4.3 Exercise 3

Consider an ideal Stirling-cycle engine in which the state at the beginning of the isothermal compression process is 100 kPa, 25°C, the compression ratio is 6, and the maximum temperature in the cycle is 1100°C. Calculate

1. the maximum cycle pressure
2. the thermal efficiency of the cycle with and without regenerators.

Solution

Ideal Stirling cycle
$T_1 = T_2 = 25°C$
$P_1 = 100$ kPa
$CR = v_1/v_2 = 6$
$T_3 = T_4 = 1100°C$
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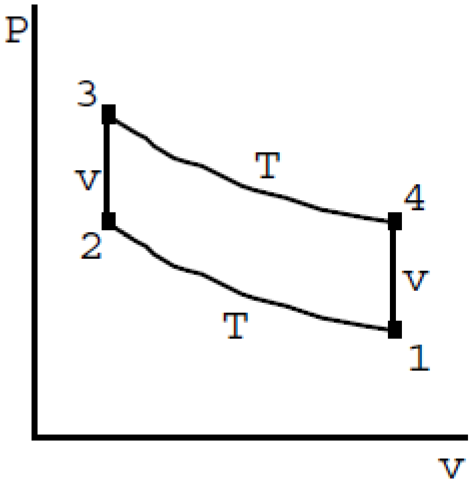
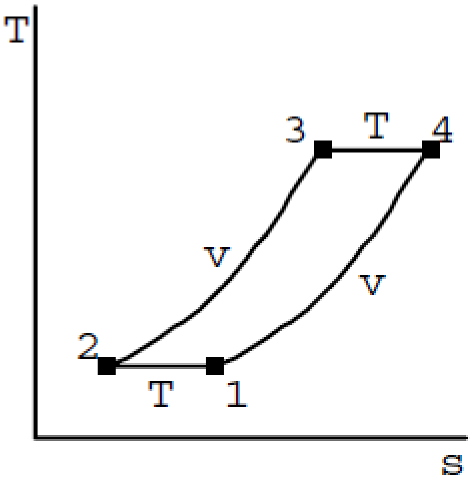




Isothermal compression (heat goes out)

$$T_1 = T_2 \Rightarrow P_2 = P_1\left(\frac{V_1}{V_2}\right) = 100 \times 6 = 600\ kPa$$

$${}_1w_2 = {}_1q_2 = -RT_1\ln\left(\frac{V_1}{V_2}\right) = -0.287\times 298.2\ln(6) = -153.3\frac{kJ}{kg}$$

Constant volume heat addition

$$V_2 = V_3 \Rightarrow V_3 = \frac{T_3P_2}{T_2} = 600\times\frac{1373.2}{298.2} = 2763\ kPa$$

$$q_{23} = u_3 - u_2 = C_{vo}(T_3 - T_2) = 0.717\ (1100 - 25) = 770.8\frac{kJ}{kg}$$

Isothermal expansion (heat comes in)

$$w_{34} = q_{34} = RT_3\ln\left(\frac{V_4}{V_3}\right) = 0.287 \times 1373.2 \times ln6 = 706.1\frac{kJ}{kg}$$

$$wnet = 706.1 - 153.3 = 552.8\frac{kJ}{kg}$$

Efficiency without regenerator, ($q_{23}\ and\ q_{34}$ are coming in from source)

$$\eta_{NO\ REGEN} = \frac{wnet}{q_{23} + q_{34}} = \frac{552.8}{770.8 + 706.1} = 0.374$$

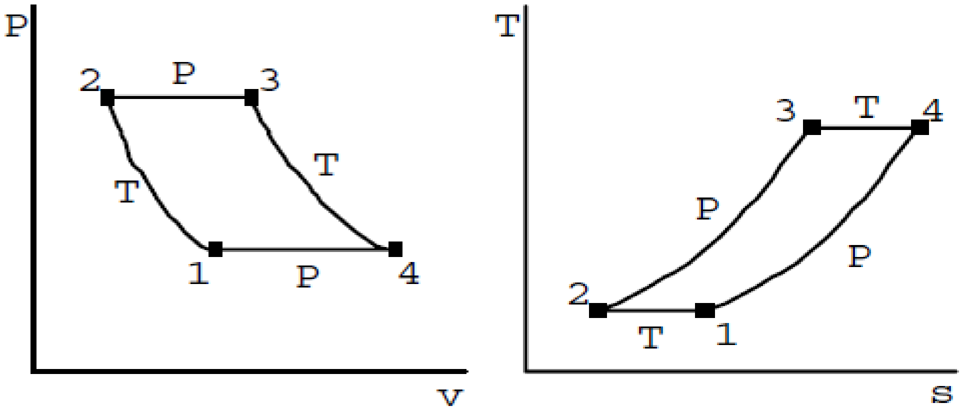
Efficiency with regenerator, (Now only q_{34} is coming in from source)

$$\eta_{WITH\ REGEN} = \frac{wnet}{q34} = \frac{552.87}{06.1} = 0.783$$

I.7.4.4 Exercise 4

Consider an ideal air-standard Ericsson cycle that has an ideal regenerator as shown in Figure bellow. The high pressure is 1 MPa and the cycle efficiency is 70%. Heat is rejected in the cycle at a temperature of 300 K, and the cycle pressure at the beginning of the isothermal compression process is 100 kPa. Determine

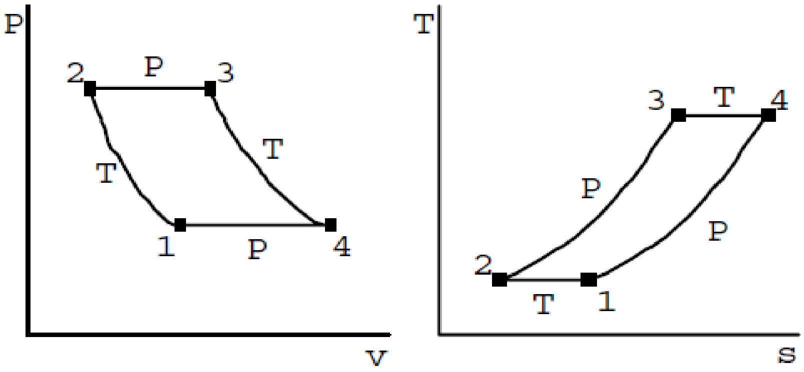
1. The high temperature,
2. The compressor work,
3. The turbine work per kilogram of air.



$P_2 = P_3 = 1$ MPa ; $T_1 = T_2 = 300$ K ; $P_1 = 100$ kPa

${}_2q_3 = -\,{}_4q_1$ (ideal reg.) ⇒ $q_H = {}_3q_4$ and $w_T = q_H$

$r_p = P_2/P_1 = 10$



$$\eta_{TH} = \eta_{CARNOT\ TH} = 1 - \frac{T_L}{T_H} = 0.7 \Rightarrow T_3 = T_4 = T_H = 1000\,K$$

$$q_L = -w_C = \int v\,dP = RT1ln\left(\frac{P_2}{P_1}\right) = 0.287 \times 300 \times ln\left(\frac{1000}{100}\right) = 198.25$$

$$w_T = q_H = -\int v\,dP = -RT3\ln\left(\frac{P_4}{P_3}\right) = 660.8\frac{kJ}{kg}$$

I.7.4.5 Exercise 5

An air-standard Ericsson cycle has an ideal regenerator. Heat is supplied at 1000°C, and heat is rejected at 20°C. Pressure at the beginning of the isothermal compression process is 70 kPa. The heat added is 600 kJ/kg. Find

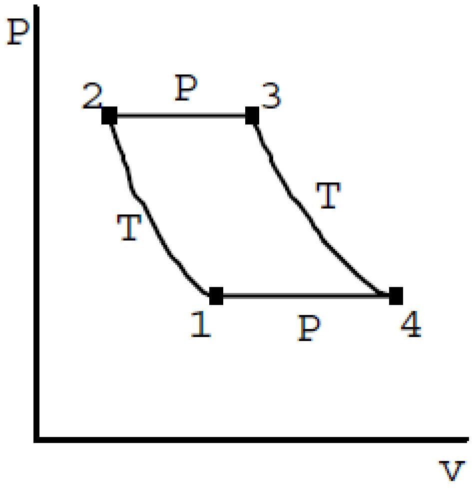
1. The compressor work
2. The turbine work
3. The cycle efficiency.

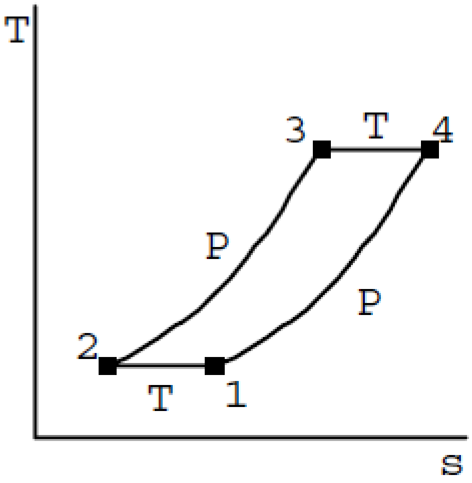
Solution:
Identify the states
Heat supplied at high temperature $T_3 = T_4 = 1000°C = 1273.15$ K
Heat rejected at low temperature $T_1 = T_2 = 20°C = 293.15$ K
Beginning of the compression: $P_1 = 70$ kPa
Ideal regenerator: ${}_2q_3 = -{}_4q_1 \Rightarrow q_H = {}_3q_4 = 600\frac{kJ}{kg} \Rightarrow w_T = q_H = 600\frac{\mathrm{kJ}}{\mathrm{kg}}$

$$\eta_{TH} = \eta_{CARNOT\ TH} = 1 - 293.151273.15 = 0.7697$$

$$w_{net} = \eta_{TH} q_H = 0.7697 \times 600 = 461.82\frac{kJ}{kg}$$

$$q_L = -w_C = 600 - 461.82 = 138.2\frac{kJ}{kg}$$





I.8 Preheat or regenerative cycle

I.8.1 The simple gas-turbine cycle with a regenerator

The efficiency of the gas turbine cycle may be improved by introducing a regenerator. The simple open-cycle gas turbine cycle with a regenerator is shown in Figure. I.15, and the corresponding ideal air-standard cycle with a regenerator is shown in the *P–v* and *T–s* diagrams in Figure. I.15. In cycle 1–2–*x*–3–4–*y*–1, the temperature of the exhaust gas leaving the turbine in state 4 is higher than the temperature of the gas leaving the compressor. Therefore, heat can be transferred from the exhaust gases to 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, *Tx*, may, in the ideal case, have a temperature equal to *T*4, the temperature of the gas leaving the turbine. Heat transfer from the external source is necessary only to increase the temperature from *Tx* to *T*3. Area *x*–3–*d*–*b*–*x* represents the heat transferred, and area *y*–1–*a*–*c*–*y* represents the heat rejected. 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, utilizing a regenerator is not possible. This can be shown more exactly by determining the efficiency of the ideal gas-turbine cycle with a regenerator.

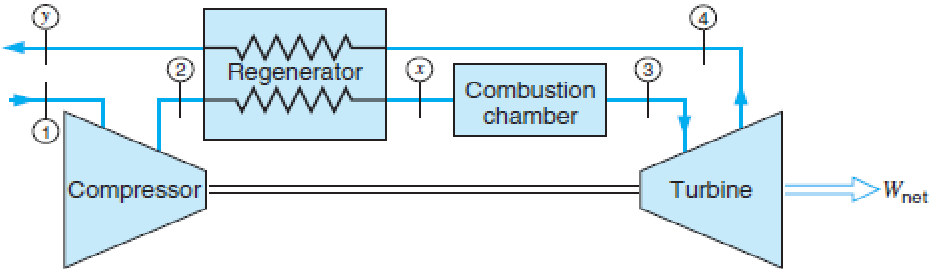


Fig. I 15 Simple gas-turbine cycle with a regenerator

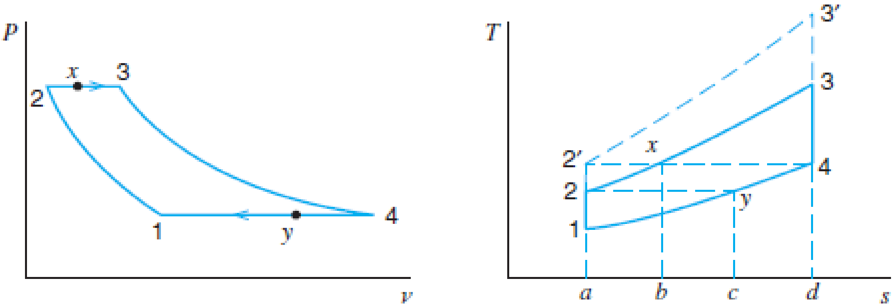


Fig. I 16 The ideal regenerative cycle

The efficiency of this cycle with regeneration is found as follows:

$$\eta_{TH} = \frac{w_{net}}{q_H} = \frac{w_t - w_c}{q_H} \qquad \text{(I-72)}$$

$$q_H \approx C_p(T_3 - T_x)$$

$$w_t \approx C_p(T_3 - T_4)$$

But for an ideal regenerator $T_4 = T_x$ And $q_H = w_t$. Consequently

$$\eta_{TH} = 1 - \frac{w_c}{w_t} \approx 1 - \frac{C_p(T_2 - T_1)}{C_p(T_3 - T_4)} \approx 1 - \frac{T_1\left(\frac{T_2}{T_1} - 1\right)}{T_3\left(1 - \frac{T_4}{T_3}\right)}$$

$$\eta_{TH} = 1 - \frac{T_1\left(\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right)}{T_3\left(1 - \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}\right)} = 1 - \frac{T_1}{T_3}\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 1 - \frac{T_2}{T_3} \qquad \text{(I-73)}$$

Thus, 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 can best be defined by reference to Figure I.17. 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 a temperature $T_{x'}$ and $T_{x'} = T_4$ In an actual regenerator, which must operate with a finite temperature difference *Tx*, the actual temperature leaving the regenerator is less than *Tx'*. The regenerator efficiency is defined by

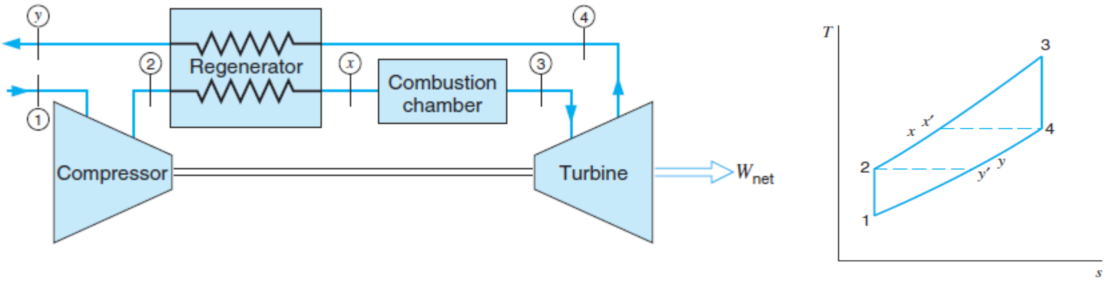


Fig. I 17 T–s diagram illustrating the definition of regenerator

$$\eta_{reg} = \frac{h_x - h_2}{h'_x - h_2} \qquad \text{(I-74)}$$

If the specific heat is assumed to be constant, the regenerator efficiency is also given by the relation

$$\eta_{reg} = \frac{T_x - T_2}{T_{x'} - T_2} \qquad \text{(I-75)}$$

A higher efficiency can be achieved by using a regenerator with a greater heat-transfer area

I.8.2 Multi-stage cycle with a regenerator, cooling, and intermediate heating

There is a modification of the Brayton/gas turbine cycle that tends to change its performance in the direction of the Ericsson cycle. This modification is to use multiple stages of compression with intercooling and multiple stages of expansion with reheat.

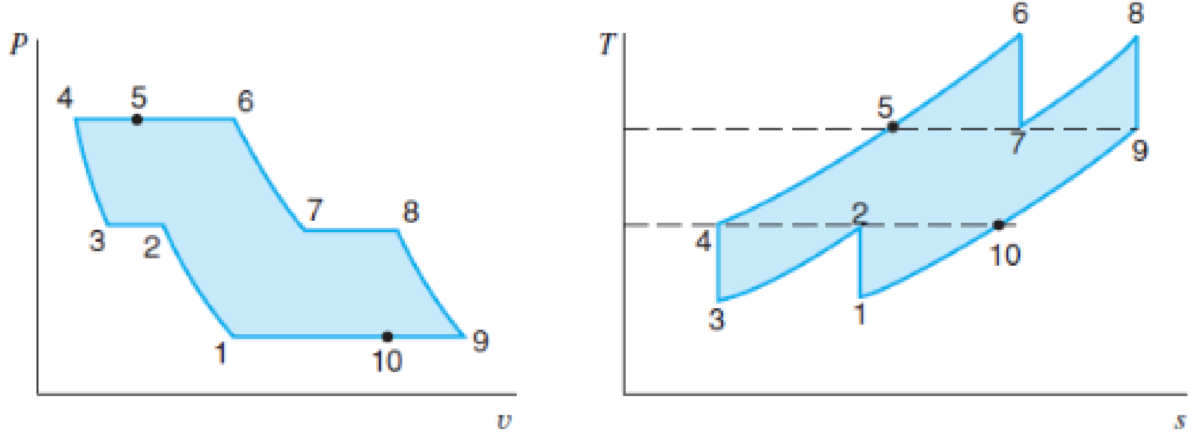
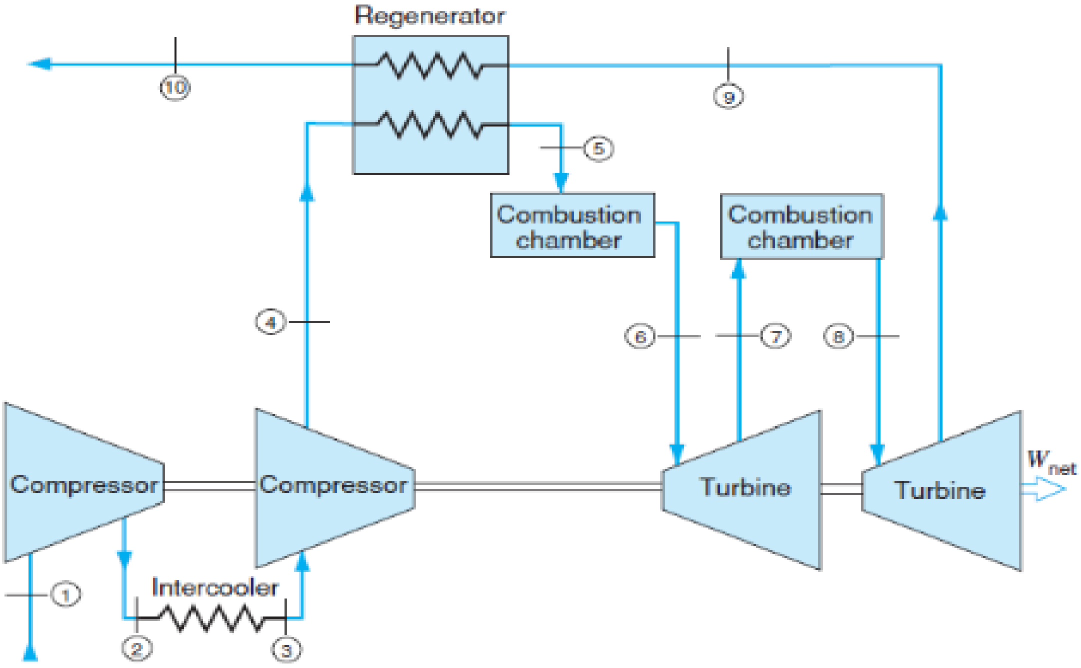


Fig. I 18 The ideal gas-turbine cycle utilizing intercooling, reheat, and a regenerator.

Such a cycle with two stages of compression and also expansion, and incorporating a regenerator, is shown in Fig. I 18. The air-standard cycle is given on the corresponding *T–s* diagram. It may be shown that for this cycle, the maximum efficiency is obtained if equal pressure ratios are maintained across the two compressors and the two turbines. In this ideal cycle, it is assumed that the temperature of the air leaving the intercooler, T_3, is equal to the temperature of the air entering the first stage of compression, T_1, and that the temperature after reheating, T_8, is equal to the temperature entering the first turbine, T_6. Furthermore, in the ideal cycle, it is assumed that the temperature of the high-pressure air leaving the regenerator, T_5, is equal to the temperature of the low-pressure air leaving the turbine, T_9.



From the discussion of the expression for a steady-state shaft, work is given by this equation:

$$w = -\int v\, dP + \Delta ke + \Delta pe - loss \qquad \text{(I-76)}$$

We recognized that the work will be less if the specific volume is smaller for a given change in pressure. This fact is used in the application of *intercoolers* used in many compression processes.

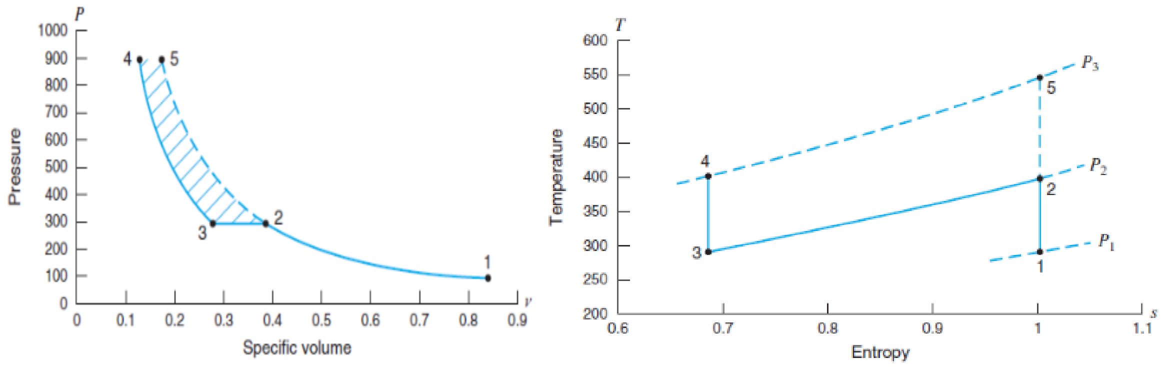


Fig. I 19. A compressor with an intercooler.

By cooling a gas at constant pressure, as in a heat exchanger, the specific volume is reduced, and any subsequent compression can then be done with a lower work input. Consider the reversible compression process between an initial state 1 and a final state 4 shown in Fig. I.18, which requires an amount of work equal to the area under the curve, as seen from the *P* axis in the *P–v* diagram. The flow is taken out at an intermediate pressure at state 2 and is cooled to the original inlet temperature before the compression to the final pressure. The whole process is illustrated in Fig. I.19 in both the *P–v* and *T–s* diagrams. If the process is done without the intercooler it follows the path 1–2–5, which requires a larger amount of work since the specific volume is larger for the last part of the process. The work input difference corresponds to the area enclosed by the curves 2–3–4–5–2 shown as shaded in Fig. I.19. If many compression and expansion stages are used, it is evident that the Ericsson cycle is approached. This is shown in Fig. I.20.

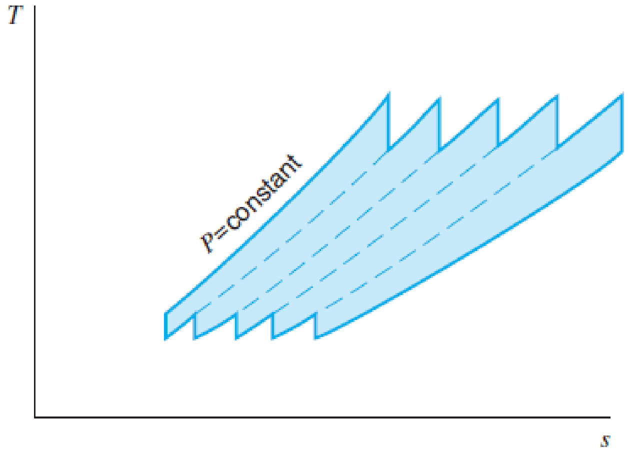


Fig. I 20 T–s diagram that shows how the gas-turbine cycle with many stages approaches the Ericsson cycle.

I.8.2.1 Exercise 1

Consider a gas turbine cycle with two stages of compression and two stages of expansion. The pressure ratio across each compressor stage and each turbine stage is 8 to 1. The pressure at the entrance of the first compressor is 100 kPa, the temperature entering each compressor is 20oC, and the temperature entering each turbine is 1100oC. A regenerator is also incorporated into the cycle, and it has an efficiency of 70%. Determine:

1. The compressor work,
2. The turbine work,
3. The thermal efficiency of the cycle.

Solution

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} = \frac{P_6}{P_7} = \frac{P_8}{P_9} = 8.0$$

$$P_1 = 100\,kPa$$

$$T_1 = T_3 = 20^0C\,, T_6 = T_8 = 1100^0C$$

Assume constant specific heat

$$s_2 = s_1 \; and \; s_4 = s_3$$

$$T_4 = T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 293.15(8)0.286 = 531\,K$$

1. $Total\ -w_C = 2\times(-w_{12}) = 2C_{P0}(T_2 - T_1)2\times1.004(531 - 293.15) = 477.6\frac{kJ}{kg}$

$$Also\ s_6 = s_7 \; and \; s_8 = s_9 : \Rightarrow T_7 = T_9 = T_6\left(\frac{P_7}{P_6}\right)^{\frac{\gamma-1}{\gamma}} = 1373.15180.286 = 758\,K$$

2. $Total\ w_T = 2\times w_{67} = 2C_{P0}(T_6 - T_7)2\times1.004(1373.15 - 758) = 1235.2\frac{\mathrm{kJ}}{\mathrm{kg}}$

$$w_{NET} = 1235.2 - 477.6 = 757.6\frac{kJ}{kg}$$

Ideal regenerator:

$$T5 = T9, T10 = T4$$

so the actual one has

3. $\eta_{REG} = \frac{h_5 - h_4}{h_9 - h_4} = \frac{T_5 - T_4}{T_9 - T_4} = \frac{T_5 - 531}{758 - 531} = 0.7 \Rightarrow T_5 = 689.9\,K$

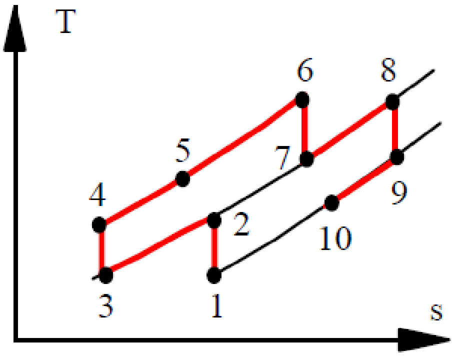
$$\Rightarrow q_H = (h_6 - h_5) + (h_8 - h_7)$$

$$= C_{P0}(T_6 - T_5) + C_{P0}(T_8 - T_7)$$

$$= 1.004(1373.15 - 689.9) + 1.004\,(1373.15 - 758)$$

$$= 1303.6\frac{kJ}{kg}$$

$$\eta_{TH} = \frac{w_{NET}}{q_H} = \frac{757.6}{1303.6} = 0.581$$



I.9 Thermal power plant [5]

I.9.1 Definition

A thermal power plant is a type of power station in which the heat energy generated from various fuel sources (e.g., coal, natural gas, nuclear fuel, etc.) is converted to electrical energy. The heat from the source is converted into mechanical energy using a thermodynamic power cycle (such as a Diesel cycle, Rankine cycle, Brayton cycle, etc.). The most common cycle involves a working fluid (often water) heated and boiled under high pressure in a pressure vessel to produce high-pressure steam. This high pressure-steam is then directed to a turbine, where it rotates the turbine's blades. The rotating turbine is mechanically connected to an electric generator which converts rotary motion into electricity. Fuels such as natural gas or oil can also be burnt directly in gas turbines (internal combustion), skipping the steam generation step. These plants can be of the open cycle or the more efficient combined cycle type.

I.9.2 Components of Thermal Power Plant.

The basic layout and diagram of a thermal power plant are shown in the following figure.

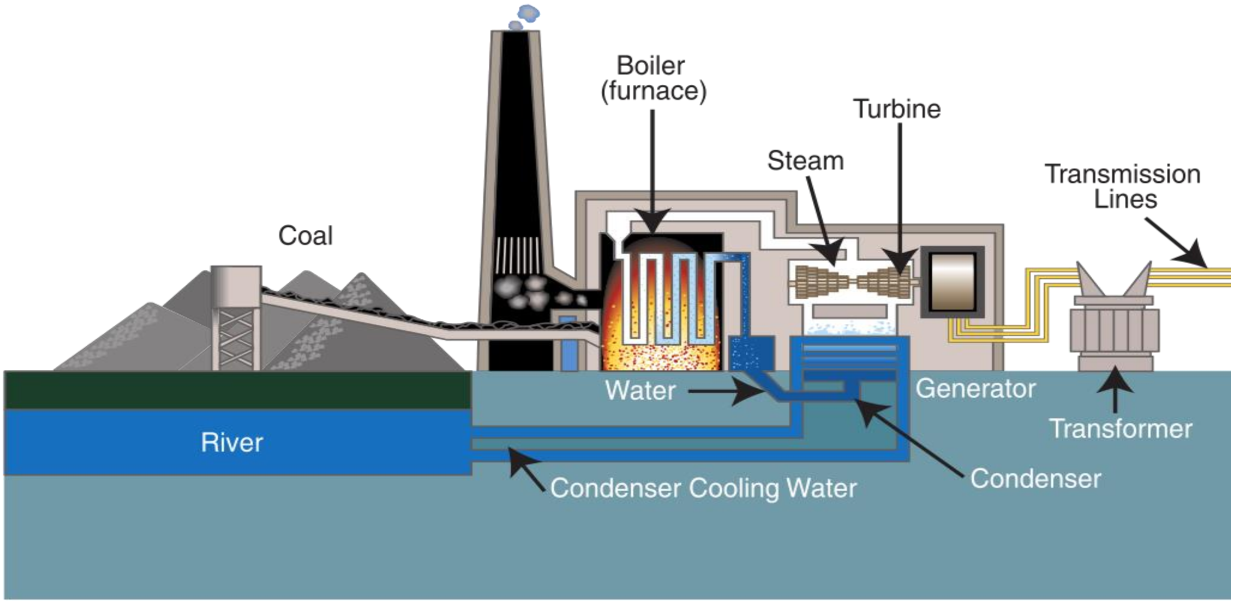


Fig. I 181 Components of Thermal Power Plant [6]

The diagram of a thermal power plant consists of the following major components:

1. Coal storage and pulverizer
2. Water Boiler
3. Steam Turbine
4. Alternator
5. Condenser
6. Feed water pump
7. Chimney

I.9.2.1 Coal Storage and Pulverizer:

In a thermal power plant, there is a dedicated space where the coal is brought from coal mines. The coal may contain iron particles that may damage the equipment, and therefore, the coal is first cleaned in a magnetic cleaner to remove the iron particles. The pulverizer is a system used to crush the raw coal into small particles to convert it into coal powder. This is an essential process that ensures the complete combustion of the coal. The ash produced after coal combustion is removed from the boiler. The ash is known as fly ash, and cement industries use it to produce cement.

I.9.2.2 Water Boiler:

A boiler is a device used to produce steam from water. In the boiler, there is a combustion chamber where the pulverized coal is burnt. The coal and preheated air are burned in the combustion zone. When coal is ignited in the boiler, a large fireball forms at the center, releasing a huge amount of heat energy and high-temperature flue gases. The water flows through the steel tubes that run along the boiler walls. The flue gases heat the water and pass through the superheater, economizer, and air preheater, and, finally, exhaust from the chimney in the atmosphere.
Let us have a brief overview of the superheater, economizer, and air preheater.

Superheater:

The superheater tubes are a group of tubes located in the hottest part of the boiler. The boiler tubes produce the saturated steam, and to increase its enthalpy, the steam is superheated in a superheater at around 540 °C. The superheated high-pressure steam is then fed to the steam turbine.

Air pre-heater:

The air is used for combustion of the coal in the boiler. The primary air fan takes the atmospheric air, and then the air is heated up in the air pre-heater. The coal with preheated air is injected into the boiler. The preheated air improves the coal combustion in the boiler.

Economizer:

An economizer recovers waste heat from the exhausted flue gases and preheats the feed water to increase the boiler's efficiency. The preheating of water reduces the amount of coal and saves energy.

The heat produced is then transferred to boil the water and produce steam. Overall, the boiler is a piece of equipment that converts water into steam.

I.9.2.3 Steam Turbine:

A steam turbine is a prime mover that converts the heat and pressure energy of steam into kinetic energy. The superheated and high-pressure steam is passed over the blades of the steam turbine, which causes the turbine to rotate. The steam turbine is further connected to the electric generator and drives the generator to produce electricity.

The steam condenses in a condenser when it passes through the steam turbine, and the pressure and temperature of the steam decrease, causing an increase in volume. The expanded low-pressure steam is exhausted in the condenser.

I.9.2.4 Condenser:

A condenser condenses the exhaust steam from the turbine by cold water circulation. The steam loses its pressure and temperature, and it is again converted into water. The condenser improves the overall efficiency of the power plant.

I.9.2.5 Alternator:

An alternator is an electric generator that converts the rotational mechanical energy of the steam turbine into alternating current electricity. In practice, the alternator produces electricity at 11 kV, which is increased to higher voltages for transmission.

I.9.2.6 Feed Water Pump:

The feed water pump feeds the condensed water from the condenser to the boiler to produce steam again.

I.9.2.7 Chimney:

A chimney is the exhaust point of the gases produced due to coal combustion. It is typically made up of bricks and concrete. The size of the chimney depends on the volume of gases to be exhausted. Chimneys also have a mechanism to filter the exhaust gases before releasing them into the environment, so they cause minimal pollution. It is also important to note that a practical thermal power plant has several other modern components and auxiliary

I.9.3 Working of Thermal Power Plant

Equipment that helps maximize the overall plant efficiency.

The complement working of a typical thermal power plant is explained here.

In a thermal power plant, the pulverized coal is burnt in a combustion chamber to produce heat. The boiler then uses this heat to convert water into steam with high temperature and pressure.

The high-pressure superheated steam is passed over the blades of the steam turbine coupled to an electric generator.

The generator coupled to a steam turbine rotates and produces electricity. The generator voltage is stepped up to transmit electrical energy to load points for utilization.

This is the basic process in a thermal power plant to generate electricity.

I.9.4 Advantages of Thermal Power Plant

A thermal power plant has several advantages over other types of power plants. Some key benefits of thermal power plants are given below:

- Thermal power plants utilize coal as fuel to produce heat. Hence, the fuel cost is low compared to nuclear power plants.
- They can generate massive amounts of electric power to supply significant loads.
- Their operating cost is lower than that of the diesel generating station.
- Thermal power plants utilize less water than hydroelectric power plants.

I.9.5 Disadvantages of Thermal Power Plant

Apart from benefits, thermal power plants also have some downsides. Some key disadvantages of thermal power plants are given below:

- The maintenance cost of thermal power plants is high.
- The exhaust gases pollute the environment.
  Coal and ash handling is complicated. The operational cost of thermal power plants is higher than that of hydroelectric power plants.

CHAPTER TWO TWO PHASES POWER CYCLES

II.1 Phase change reminders

The phase changes of water involve transitions between its three primary states: solid, liquid, and gas. Here's a breakdown:

II.1.1 Melting:

- Solid (ice) → Liquid (water)
- Occurs at 0°C under normal atmospheric pressure.
- Heat energy is absorbed, breaking the hydrogen bonds holding water molecules in a rigid structure.

II.1.2 Freezing:

- Liquid (water) → Solid (ice)
- Happens at 0°C when water loses heat.
- Molecules slow down and form a crystalline structure.

II.1.3 Evaporation:

- Liquid (water) → Gas (water vapor)
- Occurs at any temperature but more rapidly at higher temperatures.
- Molecules at the surface gain enough energy to escape into the gas phase.

II.1.4 Boiling:

- Liquid (water) → Gas (water vapor)
- Occurs at 100°C under standard atmospheric pressure.
- Heat energy causes the entire liquid to transition into vapor.

II.1.5 Condensation:

- Gas (water vapor) → Liquid (water)
- Happens when water vapor cools, losing energy, and molecules come closer together to form liquid droplets.

II.1.6 Sublimation:

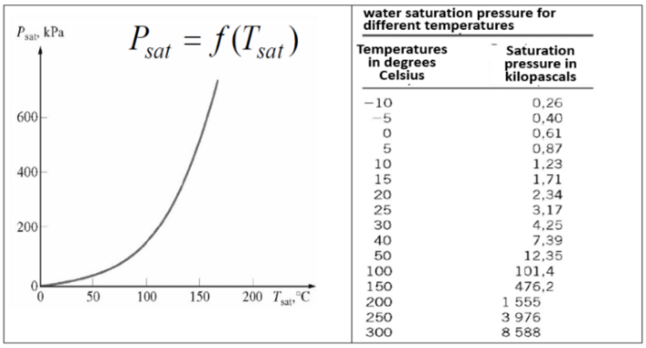
- Solid (ice) → Gas (water vapor)
- Occurs under low pressure or when ice absorbs enough energy to bypass the liquid phase

II.1.7 Deposition:

- Gas (water vapor) → Solid (ice)
- Happens when water vapor freezes directly into ice without becoming liquid

II.1.8 vaporization curve of water

The vaporization curve of water, also called the boiling curve or the liquid-vapor equilibrium curve, is the graph that represents the relationship between temperature and pressure at which water can coexist as both a liquid and a vapor (gas).
This curve is part of the phase diagram of water and indicates the conditions under which water transitions between the liquid and gaseous phases through vaporization (boiling or evaporation) or condensation.



water saturation pressure for different temperatures

Temperatures in degrees Celsius	Saturation pressure in kilopascals
−10	0,26
−5	0,40
0	0,61
5	0,87
10	1,23
15	1,71
20	2,34
25	3,17
30	4,25
40	7,39
50	12,35
100	101,4
150	476,2
200	1 555
250	3 976
300	8 588

Fig.II 1 vaporization curve of water

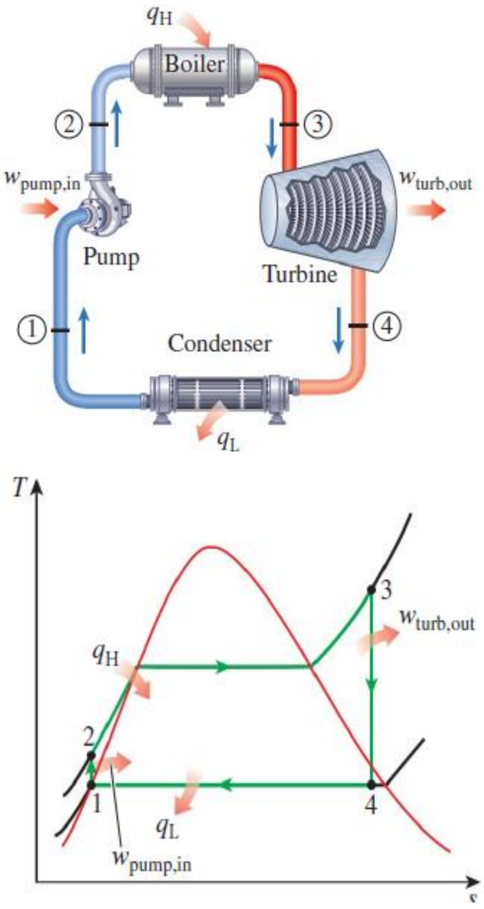
II.1.8.1 Key characteristics of the vaporization curve:

- Critical Point: The curve ends at the critical point, beyond which water exists as a supercritical fluid and there is no distinct liquid or gas phase.
- Boiling Point: At standard atmospheric pressure (1 atm or 101.325 kPa), the boiling point of water is 100°C. The boiling point increases with higher pressures and decreases at lower pressures.

The vaporization curve tells us the temperature at which water will boil at different pressures.
The temperature at which vaporization occurs for a given pressure is called the saturation temperature. Similarly, the pressure at the given temperature is called the saturation pressure. Saturation pressure and saturation temperature are thus related by a functional relationship known as the vaporization curve.

II.2 Rankine cycle.

The Rankine cycle, which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:
1-2 Isentropic compression in a pump
2-3 Constant pressure heat addition in a boiler
3-4 Isentropic expansion in a turbine
4-1 Constant pressure heat rejection in a condenser
Water enters the pump at state 1 as saturated liquid and is compressed isentropically to the operating pressure of the boiler. The water temperature increases somewhat during this isentropic compression process due to a slight decrease in the specific volume of water. Water enters the boiler as a compressed liquid at state 2 and leaves as a superheated vapor at state 3. The superheated vapor at state 3 enters the turbine, where it expands isentropically and produces work by rotating the shaft connected to an electric generator. The pressure and the temperature of steam drop during this process to the values at state 4, here steam enters the condenser. At this state, steam is usually a saturated liquid–vapor mixture with a high quality. Steam is condensed at constant pressure in the condenser. Steam leaves the condenser as saturated liquid and enters the pump, completing the cycle. the heat transfer for internally reversible processes, we see that the area under process curve 2-3 represents the heat transferred to the water in the boiler and the area under the process curve 4-1 represents the heat rejected in the condenser. The difference between these two (the area enclosed by the cycle curve) is the network produced during the cycle.



0 2-2 Ideal Rankine cycle

1-2 Isentropic compression in the pump:

$$W_{Pump} = h_2 - h_1 = v_1(P_2 - P_1) \qquad 0(II-1)$$

2-3 Heat input at constant pressure in the boiler:

$$Q_H = h_3 - h_2 \qquad 0(II-2)$$

3-4 Isentropic expansion in the turbine:

$$W_{Turbine} = -(h_4 - h_3) = h_3 - h_4 \qquad 0(II-3)$$

4-1 Constant pressure heat removal in the condenser:

$$Q_L = -(h_1 - h_4) = h_4 - h_1 \qquad 0(II\text{-}4)$$

Efficiency or Yield

$$\eta_{th} = \frac{W_{net}}{Q_H} = \frac{W_{Turbine,out} - W_{Pump,in}}{Q_H} = \frac{Q_H - Q_L}{Q_H} = 1 - \frac{|Q_L|}{Q_H} \qquad 0\text{-}1)$$

II.3 The ideal reheat Rankine cycle (Hirn cycle)

The Hirn cycle is used to increase the simple Rankine cycle. To do this we can use two possibilities

1. Superheat the steam to very high temperatures before it enters the turbine. This would be the desirable solution since the average temperature at which heat is added would also increase, thus increasing the cycle efficiency. This is not a viable solution, however, since it requires raising the steam temperature to metallurgically unsafe levels.
2. Expand the steam in the turbine in two stages and reheat it in between. In other words, modify the simple ideal Rankine cycle with a *reheat* process. Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants. The *T-s* diagram of the ideal reheat Rankine cycle and the schematic of the power plant operating on this cycle are shown in Fig. II. 4

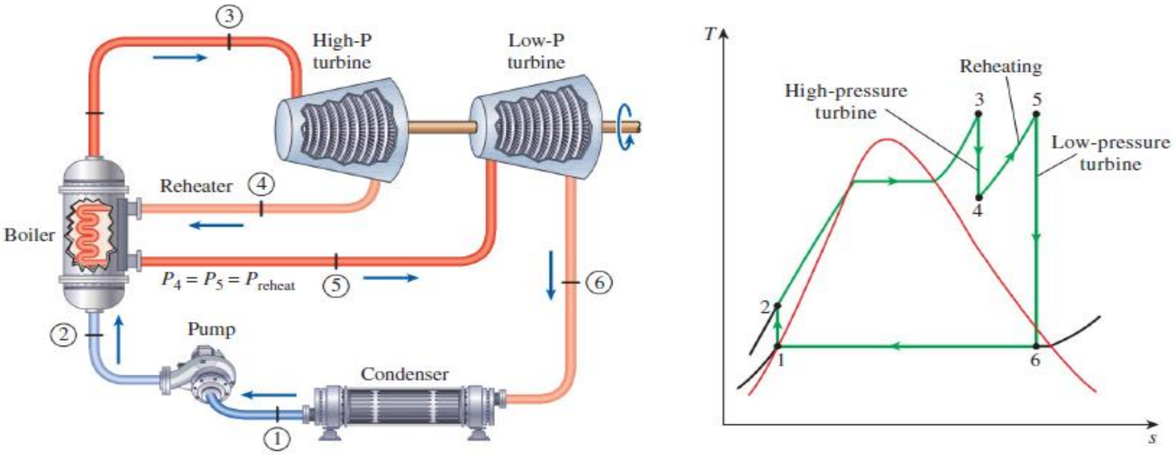


Fig. II 1 The ideal reheats Rankine cycle.

The ideal reheat Rankine cycle differs from the simple ideal Rankine cycle in that the expansion process takes place in two stages. In the first stage (the high-pressure turbine), steam is expanded isentropically to an intermediate pressure and sent back to the boiler where it is reheated at constant pressure, usually to the inlet temperature of the first turbine stage. Steam then expands isentropically in the second stage (low-pressure turbine) to the condenser pressure. Thus, the total heat input and the total turbine work output for a reheat cycle become:

$$Q_{\mathrm{H}} = Q_{Hprimary} + Q_{Hreheat} = (h_3 - h_2) + (h_5 - h_4) \qquad 0(II\text{-}5)$$

and

$$W_{\mathrm{turb,out}} = W_{\mathrm{turb,1}} + W_{\mathrm{turb,2}} = (h_3 - h_4) + (h_5 - h_6) \qquad 0(II\text{-}6)$$

II.3.1 Rankine Cycle Exercises

II.3.1.1 Exercise 1

A smaller power plant produces steam at 3 MPa, 600°C in the boiler. It keeps the condenser at 45°C by transfer of 10 MW out as heat transfer. The first turbine section expands to 500 kPa and then flow is reheated followed by the expansion in the low-pressure turbine. Find:

1. The reheat temperature so the turbine output is saturated vapor.
2. For this reheat find the total turbine power output and the boiler heat transfer.

Solution

The states properties from Tables B.1.1 and B.1.3

Point1:45°C, x=0: h_1=188.42 kJ/kg, v_1=0.00101 m^3/kg, P_{sat}=9.59 kPa

Point3:3.0 MPa, 600°C: h_3 = 3682.34 kJ/kg, s_3 = 7.5084 kJ/kg K

Point6:45°C, x = 1: h_6 = 2583.19 kJ/kg, s_6 = 8.1647 kJ/kg K

Energy: $W_p = h_2 - h_1$; Entropy: $s_2=s_1$

$W_p = \int v\, dP = v_1(P_2 - P_1) = 0.00101\ (3000 - 9.59) = 3.02$ kJ/kg

$h_2 = h_1 + W_p = 188.42 + 3.02 = 191.44$ kJ/kg

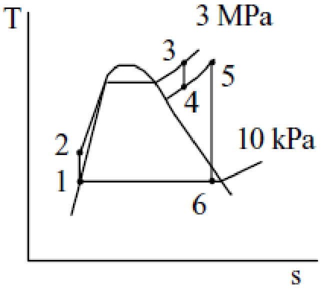
$s_4 = s_3 \Rightarrow h4 = 3093.26$ kJ/kg; $T_4 = 314$°C

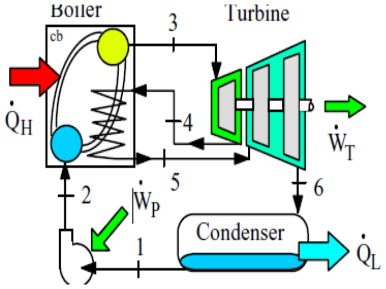
$s_6 = s_5 = 8.1647$ kJ/kg K

State 5: 500 kPa, $s_5 \Rightarrow h_5 = 3547.55$ kJ/kg, $T_5 = 529$°C

$Q_L = h_6 - h_1 = h_{fg} = 2394.77$ kJ/kg

$$\dot{m} = \frac{\dot{Q}_L}{Q_L} = 10\frac{000}{2394.77} = 4.176\frac{kg}{s}$$





Both turbine sections

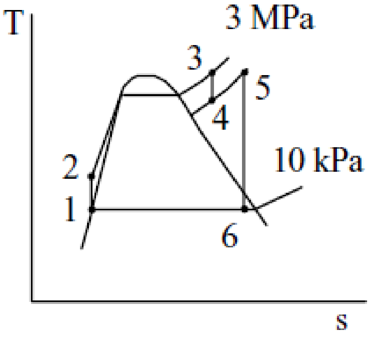
$$\dot{W}_{T,tot} = \dot{m}W_{T,tot} = \dot{m}(h_3 - h_4 + h_5 - h_6)$$
$$= 4.176\,(3682.34 - 3093.26 + 3547.55 - 2583.19) = 6487\ kW$$

Both boiler sections

$$Q_H = \dot{m}(h_3 - h_2 + h_5 - h_4)$$
$$= 4.176\,(3682.34 - 191.44 + 3547.55 - 3093.26) = 16\,475\ kW$$

II.3.1.2 Exercise 2

Consider an ideal steam reheat cycle where steam enters the high-pressure turbine at 3.0 MPa, 400°C, and then expands to 0.8 MPa. It is then reheated to 400°C and expands to 10 kPa in the low-pressure turbine.
Calculate: The cycle thermal efficiency



Solution

$W_P = v_1(P_2 - P_1) = 0.00101(3000 - 10) = 3.02$ kJ/kg,

$h_2 = 191.81 + 3.02 = 194.83$ kJ/kg

$P_3 = 3$ MPa, $T_3 = 400$ºC $\Rightarrow h_3 = 3230.82$ kJ/kg, $s_3 = 6.9211$ kJ/kg K

$s_4 = s_3 \Rightarrow h4 = 2891.6$ kJ/kg;

State 5:

Point5:400°C, 0.8 MPa $\Rightarrow h_5 = 3267.1$ kJ/kg, $s_5 = 7.5715$ kJ/kg K

$s_6 = s_5 = 7.5715$ kJ/kg K => two-phase state

$x_6 = (s_6 - s_f)/s_{fg} = 7.5715 - 0.64927.501 = 0.92285 = 0.923$

$h_6 = 191.81 + 0.92285 \times 2392.82 = 2400$ kJ/kg

$W_{T,tot} = h_3 - h_4 + h_5 - h_6 = 3230.82 - 2891.6 + 3267.1 - 2400 = 1237.8$ kJ/kg

$Q_{H1} = h_3 - h_2 = 3230.82 - 194.83 = 3036$ kJ/kg
$Q_H = Q_{H1} + h_5 - h_4 = 3036 + 3267.1 - 2891.6 = 3411.5$ kJ/kg
$\eta_{CYCLE} = (1237.8 - 3.02)/3411.5 = 0.362$

II.3.1.3 Exercise 3

A smaller power plant produces 25 kg/s of steam at 3 MPa, 600°C in the boiler. It cools the condenser with ocean water so the condenser exit is at 45°C. There is a reheat done at 500 kPa up to 400°C and then expansion in the low-pressure turbine. Find:

1. The net power output
2. The total heat transfer in the boiler.

Solution:

The states properties from Tables B.1.1 and B.1.3
Point1:45°C, x = 0: h1 = 188.42 kJ/kg, v1 = 0.00101 m3/kg, Psat = 9.59 kPa
Point3:3.0 MPa, 600oC: h3 = 3682.34 kJ/kg, s3 = 7.5084 kJ/kg K
Point5:500 kPa, 400oC: h5 = 3271.83 kJ/kg, s5 = 7.7937 kJ/kg K
$s_6 = s_5 = 7.7937$ kJ/kg K => two-phase state
$x_6 = (s_6 - s_f)/s_{fg} = 7.7937 - 0.63867.5261 = 0.9507$
$h_6 = 188.42 + 0.9507 \times 2394.77 = 2465.1$ kJ/kg
Both turbine
$w_{T,tot} = h_3 - h_4 + h_5 - h_6 = 3682.34 - 3093.26 + 3271.83 – 2465.1 = 1395.81$ kJ/kg

$$\dot{W}_{net} = \dot{W}_{T,tot} - \dot{W}_P = \dot{m}(\dot{W}_{T,tot} - \dot{W}_P) = 25\,(1395.81 - 3.02) = 34\,820\ kW$$

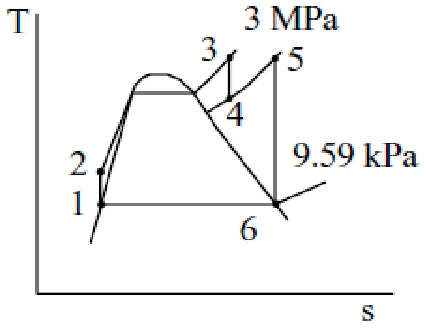
Both boiler sections
$\dot{Q}_H = \dot{m}\,(h_3 - h_2 + h_5 - h_4) = 25\,(3682.34 – 191.44 + 3271.83 - 3093.26) = 91\,737$ kW

II.3.1.4 Exercise 4

The reheat pressure affects the operating variables and thus turbine performance. Repeat Exercise II.3.1.2 twice, using 0.6 and 1.0 MPa for the reheat pressure.

Solution

$W_P = v_1(P_2 - P_1) = 0.00101(3000 - 10) = 3.02$ kJ/kg,
$h_2 = h_1 + w_P = 191.81 + 3.02 = 194.83$ kJ/kg
State 3: 3 MPa, 400°C => h3 = 3230.82 kJ/kg, s3 = 6.9211 kJ/kg K
Low T boiler section: $Q_{H1} = h_3 - h_2 = 3230.82 - 194.83 = 3035.99$ kJ/kg
State 4: P_4, $s_4 = s_3$
For $P_4 = 1$ MPa: $h_4 = 2940.85$ kJ/kg, state 4 is sup. vapor
State 5: 400°C, $P_5 = P_4$ => $h_5 = 3263.9$ kJ/kg, $s_5 = 7.465$ kJ/kg K,
For $P_4 = 0.6$ MPa: $h_4 = 2793.2$ kJ/kg, state 4 is sup. vapor
State 5: 400°C, $P_5 = P_4$ => $h_5 = 3270.3$ kJ/kg, $s_5 = 7.7078$ kJ/kg K,
State 6: 10 kPa, $s_6 = s_5$ => $x_6 = (s_6 - s_f)/s_{fg}$
Total turbine work: $w_{T,tot} = h_3 - h_4 + h_5 - h_6$
Total boiler H.Tr.: $Q_H = Q_{H1} + h_5 - h_4$
Cycle efficiency:
$\eta_{CYCLE} = (W_{T,tot} – w_P)/Q_H$



P_4=P_5	x_6	h_6	W_T	Q_H	η_{CYCLE}
1	0.9087	2366	1187.9	3359.0	0.3527
0.6	0.9410	2443.5	1228.0	3437.7	0.3563

Notice the very small changes in efficiency.

II.3.1.5 Exercise 5

The effect of a number of reheat stages on the ideal steam reheat cycle is to be studied. Repeat exercise2 using two reheat stages, one stage at 1.2 MPa and the second at 0.2 MPa, instead of the single reheat stage at 0.8 MPa.

Solution

$W_P = v_1(P_2 - P_1) = 0.00101(3000 - 10) = 3.02$ kJ/kg,

$h_2 = h_1 + W_P = 191.81 + 3.02 = 194.83$ kJ/kg

$P_4 = P_5 = 1.2$ MPa, $P6 = P7 = 0.2$ MPa

Point3: $h_3 = 3230.82$ kJ/kg, $s3 = 6.9211$ kJ/kg K

Point 4: P_4, $s_4 = s_3 \Rightarrow$ sup. vap. $h4 = 2985.3$

Point 5: $h_5 = 3260.7$ kJ/kg, $s5 = 7.3773$ kJ/kg K

Point 6: P_6, $s_6 = s5 \Rightarrow$ sup. vapor

$h_6 = 2811.2$ kJ/kg

Point 7: $h_7 = 3276.5$ kJ/kg, $s7 = 8.2217$ kJ/kg K

Point 8: P_8, $s_8 = s_7 \Rightarrow$ sup. vapor $h8 = 2607.9$ kJ/kg

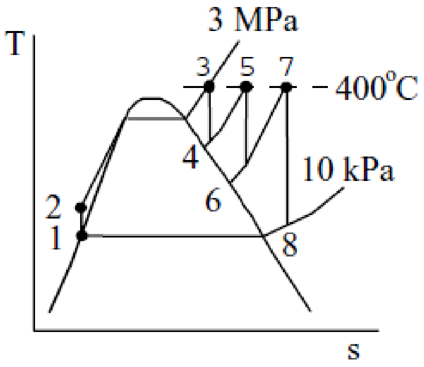
Total turbine work, same flow rate through all sections

$W_T = (h_3 - h_4) + (h_5 - h_6) + (h_7 - h_8) = 245.5 + 449.5 + 668.6 = 1363.6$ kJ/kg

Total heat transfer in boiler, same flow rate through all sections

$Q_H = (h_3 - h_2) + (h_5 - h_4) + (h_7 - h_6) = 3036 + 319.8 + 465.3 = 3821.1$ kJ/kg

Cycle efficiency: $\eta_{TH} = (W_T - W_P)/Q_H = 1363.6 - 3.023821.1 = 0.356$



II.3.1.6 Exercise 6

A steam power plant as shown in Fig. 11.3 operating in a Rankine cycle has saturated vapor at 3.0 MPa leaving the boiler. The turbine exhausts to the condenser operating at 10 kPa. Find the specific work and heat transfer in each of the ideal components and the cycle efficiency.

Solution:

C.V. Pump Reversible and adiabatic.

Energy: $wp = h_2 - h_1$; Entropy: $s_2 = s_1$

since incompressible it is easier to find work (positive in) as

$wp = \int v\, dP = v1\ (P_2 - P_1) = 0.00101\ (3000 - 10) = 3.02$ kJ/kg

$\Rightarrow h2 = h1 + wp = 191.81 + 3.02 = 194.83$ kJ/kg

C.V. Boiler : $q_H = h_3 - h_2 = 2804.14 - 194.83 = 2609.3$ kJ/kg

C.V. Turbine : $w_T = h_3 - h_4$; $s_4 = s_3$

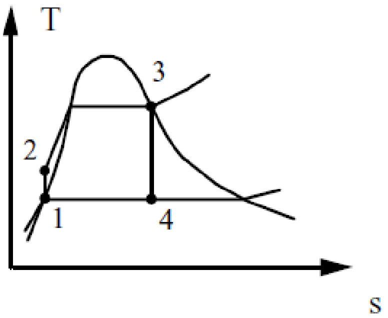
$s4 = s3 = 6.1869 = 0.6492 + x4\ (7.501) \Rightarrow x4 = 0.7383$

$\Rightarrow h4 = 191.81 + 0.7383\ (2392.82) = 1958.34$ kJ/kg

$w_T = 2804.14 - 1958.34 = 845.8$ kJ/kg

C.V. Condenser : $q_L = h_4 - h_1 = 1958.34 - 191.81 = 1766.5$ kJ/kg

$\eta_{cycle} = wnet / q_H = (w_T + wp) / q_H = (845.8 - 3.0) / 2609.3 = 0.323$



II.3.1.7 Exercise 7

Consider a solar-energy-powered ideal Rankine cycle that uses water as the working fluid. Saturated vapor leaves the solar collector at 175°C, and the condenser pressure is 10 kPa. Determine the thermal efficiency of this cycle.

Solution:

C.V. H_2O ideal Rankine cycle

State 3: T3 = 175°C ⇒ P3 = PG 175°C = 892 kPa, s3 = 6.6256

CV Turbine adiabatic and reversible so second law gives

s4 = s3 = 6.6256 = 0.6493 + x4 × 7.5009 => x4 = 0.797

h4 = 191.83 + 0.797 × 2392.8 = 2098.3 kJ/kg

The energy equation gives

wT = h3 - h4 = 2773.6 - 2098.3 = 675.3 kJ/kg

C.V. pump and incompressible liquid gives work into pump

wP = v1(P2 - P1) = 0.00101(892 - 10) = 0.89 kJ/kg

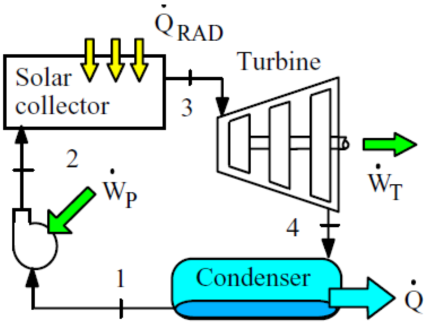
h2 = h1 + wP = 191.83 + 0.89 = 192.72 kJ/kg

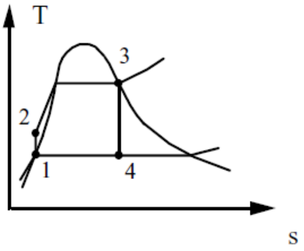
C.V. boiler gives the heat transfer from the energy equation as

qH = h3 - h2 = 2773.6 - 192.72 = 2580.9 kJ/kg

The cycle network and efficiency are found as

wNET = wT - wP = 675.3 - 0.89 = 674.4 kJ/kg





II.3.1.8 Exercise 8

A utility runs a Rankine cycle with a water boiler at 3.0 MPa, and the cycle has the highest and lowest temperatures of 450°C and 45°C, respectively. Find the plant efficiency and the efficiency of a Carnot cycle with the same temperatures.

Solution:

The states properties from Tables B.1.1 and B.1.3

1: 45°C , x = 0 => h1 = 188.42 , v1 = 0.00101 , Psat = 9.6 kPa

3: 3.0 Mpa, 450°C => h3 = 3344, s3 = 7.0833

C.V. Pump Reversible and adiabatic.

Energy: $w_p = h_2 – h_1$; Entropy: $s_2 = s_1$

since incompressible, it is easier to find work (positive in) as

$w_p = \int v\, dP = v_1 (P_2 – P_1) = 0.00101 (3000 – 9.6) = 3.02$ kJ/kg

=> $h_2 = h_1 + wp = 188.42 + 3.02 = 191.44$ kJ/kg

C.V. Boiler: $q_H = h_3 – h_2 = 3344 – 191 = 3152.56$ kJ/kg

C.V. Turbine: $w_T = h_3 – h_4$; $s_4 = s_3$

$s_4 = s_3 = 7.0833 = 0.6386 + x_4 (7.5261)$ => $x_4 = 0.8563$

=> $h_4 = 188.42 + 0.8563 (2394.77) = 2239.06$ kJ/kg

$w_T = 3344 – 2239.06 = 1105$ kJ/kg

C.V. Condenser: $q_L = h_4 – h_1 = 2239.06 – 188.42 = 2050.64$ kJ/kg

$\eta_{cycle} = (w_T+w_p)/qH=(1105 – 3.02)/3152.56 = 0.349$

II.4 Cycle with one or more steam draws. (the ideal regenerative Rankine cycle)

A practical regeneration process in steam power plants is accomplished by extracting steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feedwater instead. The device where the feedwater is heated by regeneration is called a regenerator, or a feedwater heater (FWH). A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (open feedwater heaters) or without mixing them (closed feedwater heaters).

II.4.1 Open Feedwater Heaters with one steam draws

An open (or direct contact) feedwater heater is basically a mixing chamber, where the steam extracted from the turbine mixes with the feedwater exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure. The schematic of a steam power plant with one open feedwater heater (also called single-stage regenerative cycle) and the T-s diagram of the cycle are shown in Fig. II.5. In an ideal regenerative Rankine cycle, steam enters the turbine at the boiler pressure (state 5) and expands isentropically to an intermediate pressure (state 6). Some steam is extracted at this state and routed to the feedwater heater, while the remaining steam continues to expand isentropically to the condenser pressure (state 7). This steam leaves the condenser as a saturated liquid at the condenser pressure (state 1). The condensed water, which is also called the feedwater, then enters an isentropic pump, where it is compressed to the feedwater heater pressure (state 2) and is routed to the feedwater heater, where it mixes with the steam extracted from the turbine. The fraction of the steam extracted is such that the mixture leaves the heater as a saturated liquid at the heater pressure (state 3). A second pump raises the pressure of the water to the boiler pressure (state 4). The cycle is completed by heating the water in the boiler to the turbine inlet state (state 5).

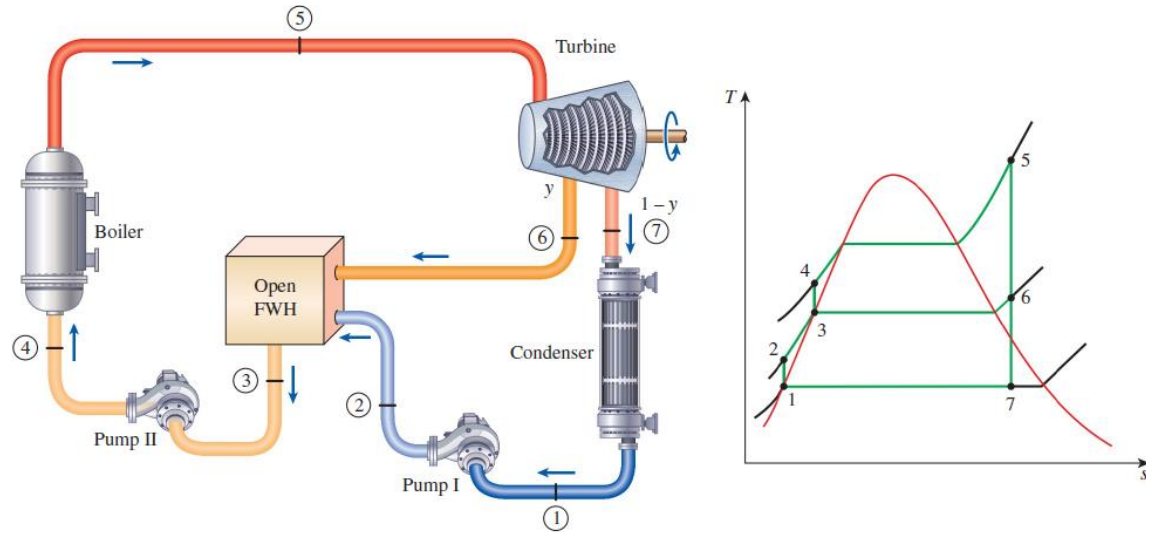


Fig.II 2 The ideal regenerative Rankine cycle with an open feedwater heater.

In the analysis of steam power plants, it is more convenient to work with quantities expressed per unit mass of the steam flowing through the boiler. For each 1 kg of steam leaving the boiler, y kg expands partially in the turbine and is extracted at state 6. The remaining (1-y) kg expands completely to the condenser pressure. Therefore, the mass flow rates are different in different components. If the mass flow rate through the boiler is $\dot{m}$, for example, it is (1-y)$\dot{m}$ 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. Considering Fig. II.5, the heat and work interactions of a regenerative Rankine cycle with one feedwater heater can be expressed per unit mass of steam flowing through the boiler as follows:

$$Q_H = h_5 - h_4 \qquad 0(II\text{-}1)$$

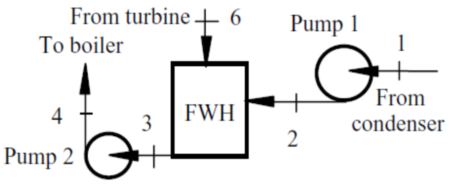
$$Q_L = (1-y)(h_7 - h_1) \qquad 0(II\text{-}2)$$

$$Q_L = (1-y)(h_7 - h_1) \qquad 0(II\text{-}3)$$

$$Q_L = (1-y)(h_7 - h_1) \qquad 0(II\text{-}4)$$

$$Q_L = (1-y)(h_7 - h_1) \qquad 0(II\text{-}5)$$

$$W_{turb,out} = (h_5 - h_6) + (1-y)(h_6 - h_7) \qquad 0(II\text{-}6)$$



$$W_{pump,in} = (1 - y)W_{pump\ I,in} + Wp_{ump\ II,in} \qquad 0(II\text{-}7)$$

Where

$$y = \frac{\dot{m}_6}{\dot{m}_5} \qquad 0(II\text{-}8)$$

$$W_{pump\ I,in} = V_1(P_2 - P_1) \qquad 0(II\text{-}9)$$

$$Wp_{ump\ II,in} = V_3(P_4 - P_3) \qquad 0(II\text{-}10)$$

II.4.2 Open Feedwater Heaters with several steam draws

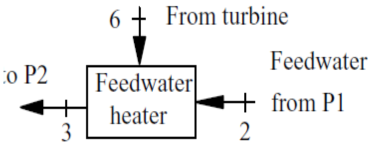
The thermal efficiency of the Rankine cycle increases because of regeneration. This is because regeneration raises the average temperature at which heat is transferred to the steam in the boiler by raising the temperature of the water before it enters the boiler. The cycle efficiency increases further as the number of feedwater heaters is increased. Many large plants in operation today use as many as eight feedwater heaters. The optimum number of feedwater heaters is determined from economical considerations. The use of an additional feedwater heater cannot be justified unless it saves more from the fuel costs than its own cost.

II.4.2.1 Exercise 1

An open feedwater heater in a regenerative steam power cycle receives 20 kg/s of water at 100°C, 2 MPa. The extraction steam from the turbine enters the heater at 2 MPa, 275°C, and all the feedwater leaves as saturated liquid.

What is the required mass flow rate of the extraction steam?

Solution



C.V Feedwater heater

Continuity Eq.: $\dot{m}_2 + \dot{m}_6 = \dot{m}_3$

Energy Eq.: $\dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 = (\dot{m}_2 + \dot{m}_6)\, h_3$

Table B.1.4: h_2 = 420.45 kJ/kg, Table B.1.2: h_3 = 908.77 kJ/kg

Table B.1.3: h_6 = 2963 kJ/kg, this is interpolated

With the values substituted into the energy equation, we get

$$\dot{m}_6 = \dot{m}_2 \frac{h_3 - h_2}{h_6 - h_3} = 20\frac{908.77 - 420.45}{2963 - 908.77} = 4.754\ kg/s$$

II.4.2.2 Exercise 2

A power plant with one open feedwater heater has a condenser temperature of 45°C, a maximum pressure of 5 MPa, and a boiler exit temperature of 900°C. Extraction steam at 1 MPa to the feedwater heater is mixed with the feedwater line so the exit is saturated liquid into the second pump.

Find the fraction of the extraction steam flow and the two specific pump work inputs.

Solution

State out of boiler 5: h_5 = 4378.82 kJ/kg, s_5 = 7.9593 kJ/kg K

C.V. Turbine reversible, adiabatic: $s_6 = s_5$

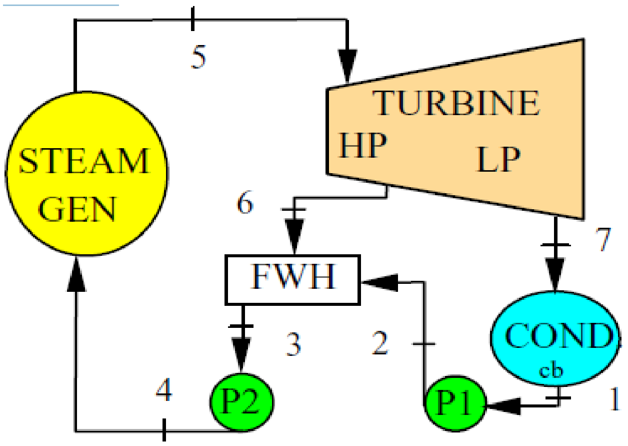
State 6: P_6 , s_6 => h_6 = 3640.6 kJ/kg, T_6 = 574°C

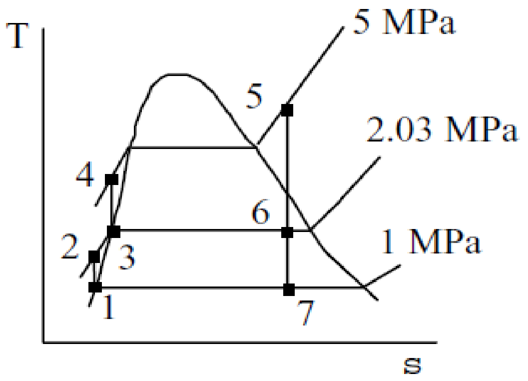
C.V Pump P1
wP1 = h2 - h1 = v1(P2 - P1) = 0.00101(1000 - 9.6) = 1.0 kJ/kg
=> h2 = h1 + wP1 = 188.42 + 1.0 = 189.42 kJ/kg
C.V. Feedwater heater: Call m.6 / m.tot = x (the extraction fraction)
Energy Eq.: (1 - x) h2 + x h6 = 1 h3
x = (h3 - h2h)/(6 - h2)= (762.79 - 189.42)/(3640.6 - 189.42) = 0.1661
C.V Pump P2
wP2 = h4 - h3 = v3(P4 - P3) = 0.001127(5000 - 1000) = 4.5 kJ/kg

II.4.2.3 Exercise 4

A Rankine cycle operating with ammonia is heated by some low temperature source so the highest T is 120oC at a pressure of 5000 kPa. Its low pressure is 1003 kPa and it operates with one open feedwater heater at 2033 kPa. The total flow rate is 5 kg/s. Find

1. The extraction flow rate to the feedwater heater assuming its outlet state is saturated liquid at 2033 kPa.
2. The total power to the two pumps.



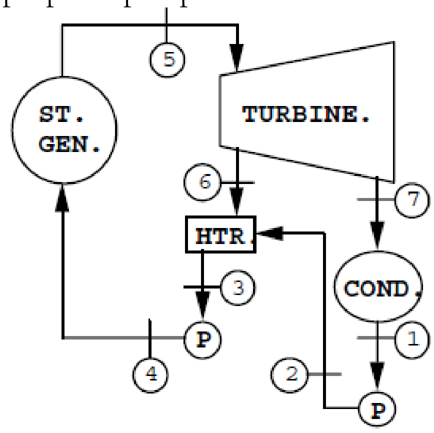


State 1: $x_1 = 0$, $h_1 = 298.25$ kJ/kg, $v_1 = 0.001658$ m3/kg
State 3: $x_3 = 0$, $h_3 = 421.48$ kJ/kg, v3 = 0.001777 m3/kg
State 5: $h_5 = 421.48$ kJ/kg, $s_5 = 4.7306$ kJ/kg K
State 6: $s_6 = s_5 \Rightarrow x_6 = (s_6 – s_f)/s_{fg} = 0.99052$, h6 = 1461.53 kJ/kg
C.V Pump P1
$w_{P1} = h_2 - h_1 = v_1(P_2 - P_1) = 0.001658(2033 - 1003) = 1.708$ kJ/kg
=> $h_2 = h_1 + w_P1 = 298.25 + 1.708 = 299.96$ kJ/kg
$\dot{m}_6 / \dot{m}_{tot} = x$ (the extraction fraction)
Energy Eq.: $(1 - x) h_2 + x h_6 = 1 h_3$
$x = (h_3 - h_2)/(h_6 - h_2) = (762.79 - 189.42)/(3640.6 - 189.42)= 0.1046$
$\dot{m}_{ext} = x\dot{m}_{tot} =$ 0.1046 × 5 = 0.523 kg/s
$\dot{m}_1 = (1 - x)\dot{m}_{tot}$ = (1 – 0.1046) 5 = 4.477 kg/s
C.V Pump P2
$wP_2 = h_4 - h_3 = v3(P_4 - P_3) = 0.001777(5000 - 2033) = 5.272$ kJ/kg
Total pump work
$\dot{W}p = \dot{m}_1\dot{W}p1 + \dot{m}_{tot}\,\dot{W}p2$ = 4.477 × 1.708 + 5 × 5.272 = 34 kW

II.4.2.4 Exercise 4

Consider an ideal steam regenerative cycle in which steam enters the turbine at 3.0 MPa, 400°C, and exhausts to the condenser at 10 kPa. Steam is extracted from the turbine at 0.8 MPa for an open feedwater heater. The feedwater leaves the heater as saturated liquid. The appropriate pumps are used for the water leaving the condenser and the feedwater heater.

Calculate the thermal efficiency of the cycle and the network per kilogram of steam.



Solution:

This is a standard Rankine cycle with an open FWH as shown in figure bellow

Control Volume (C.V) Pump P1

$w_{P1} = h_2 - h_1 = v_1(P_2 - P_1) = 0.00101(800 - 10) = 0.798$ kJ/kg

=> $h_2 = h_1 + w_{P1} = 191.81 + 0.798 = 192.61$ kJ/kg

C.V. FWH

$\dot{W}p = \dot{m}_1\dot{W}p1 = x$ (the extraction fraction)

$(1 - x)\ h_2 + x\ h_6 = 1\ h3$

$x = (h_3 - h_2)/(h_6 - h_2) = (721.1 - 192.61)/(2891.6 - 192.61) = 0.1958$

C.V Pump P2

$w_{P2} = h_4 - h_3 = v_3(P_4 - P_3) = 0.001115(3000 - 800) = 2.45$ kJ/kg

$h_4 = h_3 + w_{P2} = 721.1 + 2.45 = 723.55$ kJ/kg

CV Boiler: $q_H = h_5 - h_4 = 3230.82 - 723.55 = 2507.3$ kJ/kg

CV Turbine

2nd Law $s_7 = s_6 = s_5 = 6.9211$ kJ/kg K

P_6 , s_6 => $h_6 = 2891.6$ kJ/kg (superheated vapor)

$s_7 = s_6 = s_5 = 6.9211$ => $x_7 = 6.9211 - 0.64927.501 = 0.83614$

=> $h_7 = 191.81 + x7\ 2392.82 = 2192.55$ kJ/kg

Turbine has full flow in the HP section and a fraction 1-x in the LP section

$\dot{m}_T / \dot{m}_5 = h_5 - h_6 + (1 - x)\ (h_6 - h_7)$

$w_T = 3230.82 – 2891.6 + (1 - 0.1988)\ (2891.6 – 2192.55) = 899.3$

P_2 has the full flow and P1 has the fraction 1-x of the flow

$w_{net} = w_T - (1 - x)\ w_{P1} - w_{P2}$

$= 899.3 - (1 - 0.1988)0.798 – 2.45 = 896.2$ kJ/kg

$\eta_{cycle} = w_{net}/q_H = 896.2 / 2507.3 = 0.357$

II.4.2.5 Exercise 5

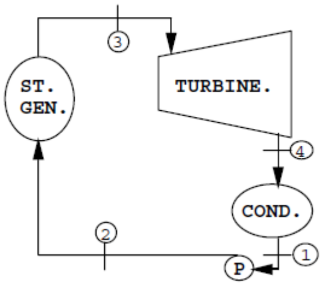
The effect of a number of open feedwater heaters on the thermal efficiency of an ideal cycle is to be studied. Steam leaves the steam generator at 20 MPa, 600°C, and the cycle has a condenser pressure of 10 kPa.

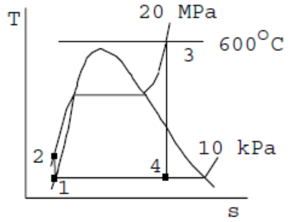
Determine the thermal efficiency for each of the following cases.

A: No feedwater heater.

B: One feedwater heater operating at 1 MPa.

C: Two feedwater heaters, one operating at 3 MPa and the other at 0.2 MPa.





A) no feed water heater

$w_P = \int_1^2 vdP \approx 0.00101(20000 - 10) = 20.2$ kJ/kg

$h_2 = h_1 + w_P = 191.8 + 20.2 = 212.0$

$s_4 = s_3 = 6.5048 = 0.6493 + x_4 \times 7.5009$

$x_4 = 0.78064$

ST. GEN.P 1 2 TURBINE.COND. 4 3

$h_4 = 191.83 + 0.780\ 64 \times 2392.8 = 2059.7$

$w_T = h_3 - h_4 = 3537.6 - 2059.7 = 1477.9$ kJ/kg

$w_{Net} = w_T - w_P = 1477.9 - 20.2 = 1457.7$ kJ/kg

$q_H = h_3 - h_2 = 3537.6 - 212.0 = 3325.6$ kJ/kg

$\eta_{TH} = w_{Net}/q_H = 1457.7/3325.6 = 0.438$

B) one feedwater heater

$w_{P12} = 0.00101(1000 - 10) = 1.0$ kJ/kg

$h_2 = h_1 + w_{P12} = 191.8 + 1.0 = 192.8$

$w_{P34} = 0.001127\ (20000- 1000) = 21.4$ kJ/kg

$h_4 = h_3 + w_{P34} = 762.8 + 21.4 = 784.2$

$s_6 = s_5 = 6.5048 = 2.1387 + x_6 \times 4.4478$

$x_6 = 0.9816$

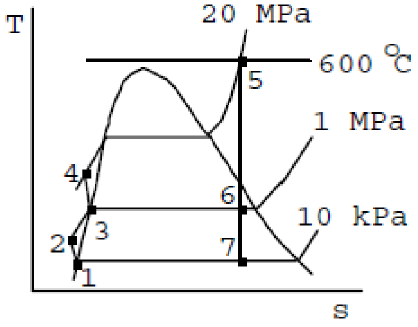
$h_6 = 762.8 + 0.9816 \times 2015.3 = 2741.1$

CV: heater

continuity: $m_3 = m_6 + m_2 = 1.0$ kg

1st law: $m_6h_6 + m_2h_2 = m_3h_3$

$m_6 = 762.8 - 192.82741.1 - 192.8 = 0.2237$ kg



C) two feedwater heaters

$w_{P12} = 0.00101 \times(200 - 10) = 0.2$ kJ/kg

$h_2 = w_{P12} + h_1 = 191.8 + 0.2 = 192.0$

$w_{P34} = 0.001061 \times(3000 - 200)= 3.0$ kJ/kg

$h_4 = h_3 + w_{P34} = 504 .7 + 3.0 = 507.7$

$w_{P56} = 0.001217(20000 - 3000)= 20.7$ kJ/kg

$h_6 = h_5 + w_{P56} = 1008.4 + 20.7 = 1029.1$

$s_8 = s_7 = 6.5048$ $T= 293.2$ °C

at P8 = 3 MPa $h_8 = 2974.8$

$s_9 = s_8 = 6.5048 = 1.5301 + x_9$ x 5.5970

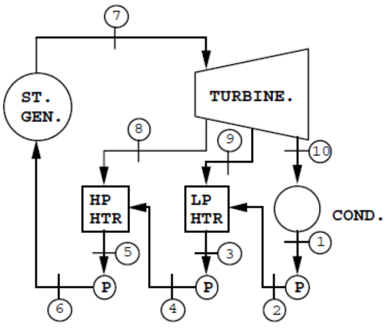
$x_9 = 0.8888$ => $h_9 = 504.7 + 0.888$ x $2201.9 = 2461.8$ kJ/kg

CV: high-pressure heater

continuity: $m_5=m_4+ m_8 =1.0$ kg.

1st law: $m_5h_5 = m_4h_4 + m_8h_8$

$m_8=(1008.4 - 507.7)/(2974.8 - 507.7)= 0.2030$; $m_4= 0.7970$ kg



CV: low pressure heater

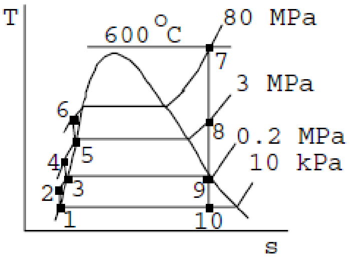
cont: $m_9 + m_2 = m_3 = m_4$; 1st law: $m_9h_9 + m_2h_2 = m_3h_3$

$m_9=(0.7970(504.7 - 192.0)/(2461.8 - 192.0)= 0.2030$; $m_2= 0.7970$CV:

$m_2 = 0.7970 - 0.1098 = 0.6872$

turbine

$w_T = (h_7 - h_8) + (1 - m_8)(h_8 - h_9) + (1 - m_8 - m_9)(h_9 - h_{10})= (3537.6 - 2974.8) + 0.797(2974.8 - 2461.8)$
$+ 0.6872(2461.8 - 2059.7) = 1248.0$ kJ/kg



CV: pumps

$w_P = m_1w_{P12} + m_3w_{P34} + m_5w_{P56} = 0.6872(0.2) + 0.797(3.0) + 1(20.7) = 23.2$ kJ/kg

$w_N = 1248.0 - 23.2 = 1224.8$ kJ/kg

CV: steam generator

$q_H = h_7 - h_6 = 3537.6 - 1029.1 = 2508.5$ kJ/kg

$\eta_{TH} = w_N/q_H = 1224.8/2508.5 = 0.488$

II.5 Combined cycle (gas-steam)

A combined gas-steam plant is a type of power generation facility that combines a gas turbine and a steam turbine to achieve higher efficiency in electricity production. This integration takes advantage of the waste heat from the gas turbine to produce steam, which is then used to drive a steam turbine, resulting in a more efficient energy conversion process.

II.5.1 Role of each component

- Gas Turbine:

Burns natural gas (or other fuels) to produce high-temperature, high-pressure gases.
Drives a generator to produce electricity.
Exhaust gases are typically very hot (500–600°C), which would otherwise be wasted.

- Heat Recovery Steam Generator (HRSG):

Captures the waste heat from the gas turbine's exhaust.
Uses this heat to produce steam, often at multiple pressure levels (high, medium, and low).

- Steam Turbine:

Utilizes the steam produced in the HRSG to drive a generator for additional electricity production.

- Generator:

Converts mechanical energy from both turbines into electrical energy.

- Cooling System:

Condenses the steam exiting the steam turbine back into water, which is recirculated to the HRSG.

II.5.2 How It Works:

- Step 1: Gas Turbine Cycle (Brayton Cycle)

Air is compressed, mixed with fuel, and combusted in the gas turbine.
The high-temperature gases expand, spinning the turbine and generating electricity.

- Step 2: Waste Heat Utilization

Exhaust gases from the gas turbine pass through the HRSG.
The HRSG captures this heat to produce steam.

- Step 3: Steam Turbine Cycle (Rankine Cycle)

The generated steam is used in a steam turbine to produce additional electricity.
Steam exiting the turbine is cooled and recycled.

II.5.3 Advantages:

- High Efficiency:

Efficiency can reach 50–60%, compared to 30–40% for standalone gas or steam plants.

- Lower Emissions:

More energy is extracted from the same fuel input, reducing CO_2 emissions per unit of electricity.

- Fuel Flexibility:

Can operate on natural gas, synthetic gas, or other fuels.

- Flexibility in Operation:

Gas turbines can ramp up or down quickly, complementing renewable energy sources.

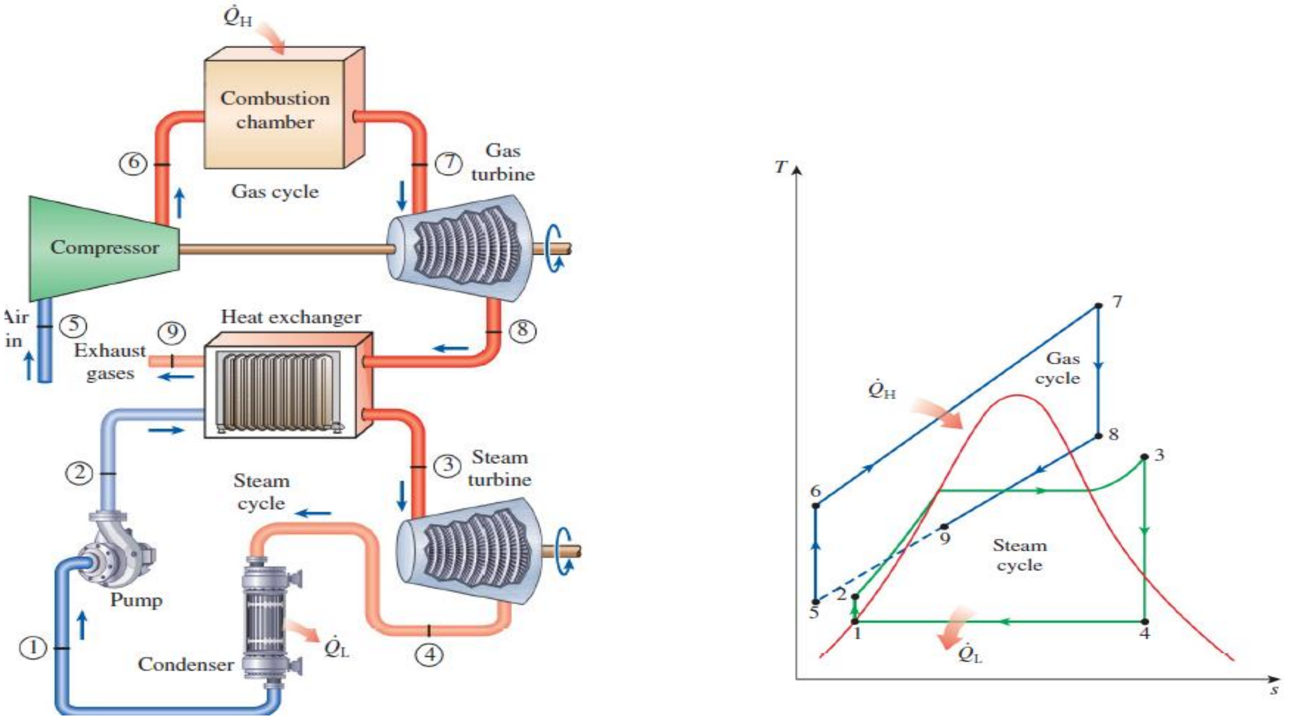


Fig.II 3 Combined gas–steam power plant

Gas-turbine plants are being converted to combined-cycle power plants. Thermal efficiencies well over 40 percent are reported as a result of conversion.

II.5.4 exercises

II.5.4.1 Exercise 1

Consider the combined gas–steam power cycle shown in Fig.II 4. The topping cycle is a gas-turbine cycle that has a pressure ratio of 8. Air enters the compressor at 300 K and the turbine at 1300 K. The isentropic efficiency of the compressor is 80 percent, and that of the gas turbine is 85 percent. The bottoming cycle is a simple ideal Rankine cycle operating between the pressure limits of 7 MPa and 5 kPa. Steam is heated in a heat exchanger by the exhaust gases to a temperature of 500°C. The exhaust gases leave the heat exchanger at 450 K. Determine:

1. The ratio of the mass flow rates of the steam and the combustion gases
2. The thermal efficiency of the combined cycle.

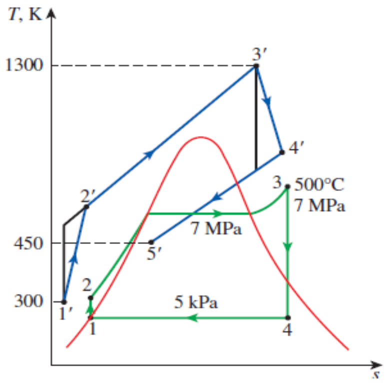


Fig.II 5 T-s diagram of the gas–steam combined cycle described in Example II.5.4.1.

Responses

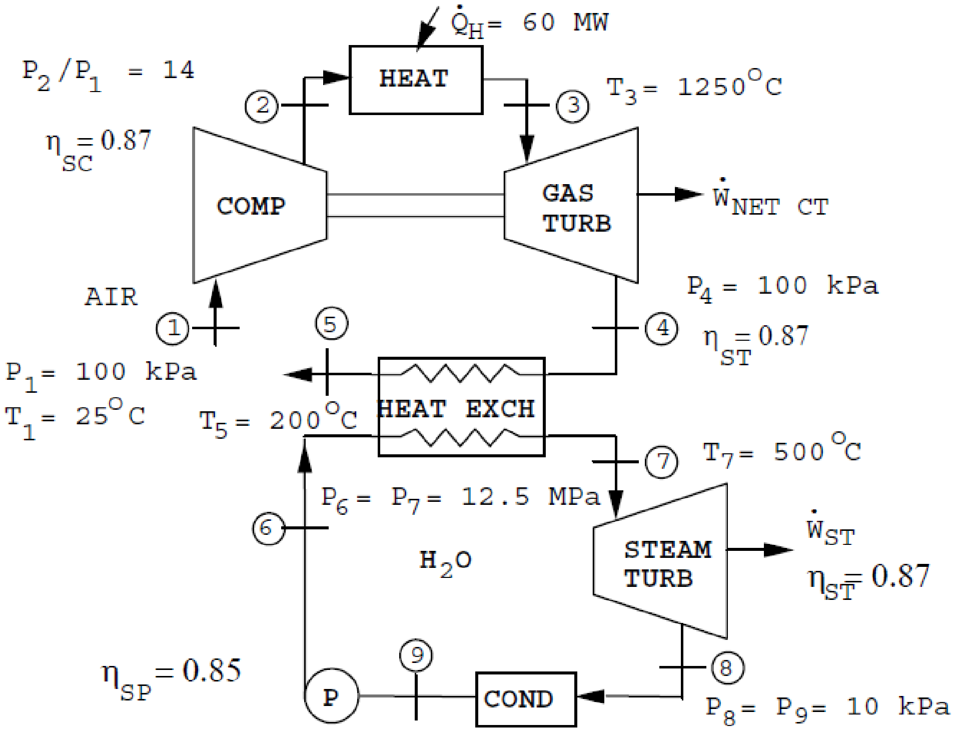
1. y=0.131
2. η=0.487

II.5.4.2 exercise 2

The power plant shown in Fig. bellow combines a gas-turbine cycle and a steam-turbine cycle. The following data are known for the gas-turbine cycle. Air enters the compressor at 100 kPa, 25°C, the compressor pressure ratio is 14, and the isentropic compressor efficiency is 87%; the heater input rate is 60 MW; the turbine inlet temperature is 1250°C, the exhaust pressure is 100 kPa, and the isentropic turbine efficiency is 87%; the cycle exhaust temperature from the heat exchanger is 200°C. The following data are known for the steam-turbine cycle. The pump inlet state is saturated liquid at 10 kPa, the pump exit pressure is 12.5 MPa, and the isentropic pump efficiency is 85%; turbine inlet temperature is 500°C and the isentropic turbine efficiency is 87%. Determine

a. The mass flow rate of air in the gas-turbine cycle.
b. The mass flow rate of water in the steam cycle.
c. The overall thermal efficiency of the combined cycle.

Temperature T (K)	Enthalpy h (kJ/kg)	Internal energy u (kJ/kg)	Entropy standard s0 (kJ/kg·K)	Fonction isentropic Pr
280	280.13	199.75	1.63279	1.0889
290	289.34	207.74	1.66402	1.0901
298	298.66	213.74	1.7010	1.0913
300	300.19	215.01	1.70713	1.0916
629	634.48			15.2782
791				36.8209
1523.2	1663.91	812.68		515.493



a) From Air Tables given, Pr_1 = 1.0913, h_1 = 298.66, h_5 = 475.84 kJ/kg

Isentropic process An isentropic process is a process during which the entropy remains constant (i.e. ,s_2.=,s_1.). In the Air Tables, this criterion is used to determine the other properties in the final state. Calculating Pr_{2s} For such a process, the relationship used is

$$Pr_{2s} = Pr_1 \times \frac{P_2}{P_1}$$

Where :

Pr_1 is the initial tabulated value.

$\frac{P_2}{P_1}$ is the pressure ratio between the final state and the initial state.

$$Pr_{2s} = 1.0913 \times 14 \approx 15.2782$$

This tabulated value Pr_{2s} is then used to retrieve, from the tables, other properties of the final state (such as temperature, enthalpy, etc.) assuming that the process is isentropic.

◈ Gas-Turbine (Brayton) Cycle

1. Compressor (State 1 → State 2_s → State 2)

State 1: $T_1 = 298.15\,K$, $h_1 = 298.66$ kJ/kg, $Pr_1 = 1.0913$

Ideal-isentropic compression: $Pr_{2s} = Pr_1 \times (P_2/P_1) = 1.0913 \times 14 = 15.2782$ → From table: $T_{2s} = 629\,K$, $h_{2s} = 634.48$ kJ/kg

Isentropic work: $w_{SC} = h_1 - h_{2s} = 298.66 - 634.48 = -335.82$ kJ/kg

Real work with efficiency $\eta_C = 0.87$: $w_C = \frac{w_{SC}}{\eta_C} = \frac{-335.82}{0.87} = -386.0$ kJ/kg → $h_2 = h_1 - w_C = 684.66$ kJ/kg

2. Combustion heating (State 2 → State 3)

$T_3 = 1523.2\,K$ → $h_3 = 1663.91$ kJ/kg, $Pr_3 = 515.493$

3. Air mass flow

$$\dot{m}_{air} = \frac{\dot{Q}_H}{h_3 - h_2} = \frac{60\,000}{1663.91 - 684.66} = \boxed{61.27\ \mathrm{kg/s}}$$

4. Turbine (State 3 → State 4_s → State 4)

$Pr_{4s} = Pr_3 \div 14 = 36.8209$ → From table: $T_{4s} = 791\,K$, $h_{4s} = 812.68$ kJ/kg

Isentropic turbine work: $w_{ST} = h_3 - h_{4s} = 851.23$ kJ/kg

Real turbine work with $\eta_T = 0.87$: $w_T = 0.87 \times 851.23 = 740.57$ kJ/kg → $h_4 = h_3 - w_T = 923.34$ kJ/kg

5. Heat recovery

Exhaust air leaves at $T_5 = 473.15\,K$, $h_5 = 475.84$ kJ/kg

◈ Steam (Rankine) Cycle

6. Pump (State 9 → State 6)

Inlet: saturated liquid at 10 kPa → $h_9 = 191.83$ kJ/kg, $v_f = 0.00101\ \mathrm{m^3/kg}$

Pump work: $w_{sp} = v_f(P_6 - P_9) = 12.615$ kJ/kg $w_p = \frac{12.615}{0.85} = 14.84$ kJ/kg → Outlet enthalpy: $h_6 = 206.67$ kJ/kg

7. Boiler and Turbine (State 6 → State 7 → State 8 → State 9)

Steam conditions: 12.5 MPa, 500 °C → $h_7 = 3341.7$ kJ/kg, $s_7 = 6.4617$ kJ/kg · K

Dry expansion to condenser at 10 kPa:

$$x_{8s} = \frac{s_7 - s_f}{s_{fg}} = 0.7749$$

→ $h_{8s} = 2046.0$ kJ/kg

Isentropic turbine work: $w_{ST} = 1295.7$ kJ/kg

Real turbine work ($\eta = 0.87$): $w_{T,steam} = 1127.3$ kJ/kg

8. Steam mass flow

$$\dot{m}_{H_2O} = \dot{m}_{air} \cdot \frac{h_4 - h_5}{h_7 - h_6} = 61.27 \cdot \frac{923.34 - 475.84}{3341.7 - 206.67} = \boxed{8.746\ \mathrm{kg/s}}$$

◈ Combined Output and Efficiency

9. Net power outputs

Gas cycle net: $\dot{W}_{gas} = 61.27 \times (740.57 - 386) = 21\,725$ kW

Steam cycle net: $\dot{W}_{steam} = 8.746 \times (1127.3 - 14.84) = 9730$ kW

Total net: 31.455 MW

10. Thermal efficiency

$$\eta_{th} = \frac{31.455}{60} = \boxed{52.4\%}$$

Summary Table

Parameter	Value
Mass flow of air ($\dot{m}_{air}$)	61.27 kg/s
Mass flow of steam ($\dot{m}_{H_2O}$)	8.746 kg/s
Net power output	31.455 MW
Thermal efficiency (η_{th})	52.4 %

II.6 Hybrid installations (solar gas).

Hybrid Solar-Gas Power Installations are innovative energy systems that combine solar power with gas-based power generation, providing a reliable, efficient, and sustainable solution for energy production. By integrating renewable solar energy with the flexibility of gas turbines, these hybrid systems optimize energy production, reduce greenhouse gas emissions, and enhance grid stability.

II.6.1 Components of Solar-Gas Hybrid Systems

II.6.1.1 Solar Energy System:

- Solar Photovoltaic (PV): Converts sunlight directly into electricity using solar panels.
- Concentrated Solar Power (CSP): Uses mirrors to focus sunlight, generating high-temperature heat to produce steam or electricity.

II.6.1.2 Gas Turbine System:

Utilizes natural gas or other gaseous fuels to generate electricity.
Offers a reliable backup source during periods of low solar output (e.g., at night or during cloudy weather).

II.6.1.3 Energy Storage (Optional):

- Batteries: Store excess solar energy for later use.
- Thermal Storage: Stores heat from CSP systems to produce power even when the sun is not shining.

II.6.2 How Solar-Gas Hybrid Systems Work

- Primary Solar Operation:

During the day, solar energy is the primary source of power. Electricity generated by solar PV(Photovoltaic) panels
Gas turbines are activated when solar energy production is insufficient to meet demand, such as during peak hours or at night.

- Continuous Operation:

The hybrid system dynamically adjusts the contribution of solar and gas energy based on demand and solar availability.
Excess solar energy can be stored in batteries or thermal systems for future use.

II.6.3 Types of Solar-Gas Hybrid Installations

II.6.3.1 PV + Gas Turbine:

Solar PV systems generate electricity during the day.
Gas turbines supply power when solar production is insufficient.

II.6.3.2 CSP (Concentrated Solar Power) + Gas Turbine:

CSP systems use concentrated sunlight to produce steam, which can augment the operation of gas turbines. Thermal energy storage (e.g., molten salt) allows power generation even after sunset.

II.6.3.3 Solar-Assisted Combined-Cycle (SACC) Plants:

A combined-cycle gas plant is integrated with solar energy, where waste heat from the gas turbine is used alongside solar-generated steam to drive a steam turbine.
Significantly boosts the thermal efficiency of the plant.

II.6.4 Benefits of Solar-Gas Hybrid Systems

- Increased Efficiency:

Solar energy offsets a portion of the fuel used by gas turbines, enhancing overall efficiency.

- Reduced Emissions:

Combines the clean energy from solar with the lower carbon footprint of natural gas, leading to significantly reduced greenhouse gas emissions compared to conventional fossil fuel plants.

- Reliable Power Supply:

Gas turbines provide a dependable power source, ensuring uninterrupted electricity even during low solar production periods.

- Flexibility:

The system can rapidly adjust to changes in energy demand, making it suitable for peak-load management.

- Cost Savings:

Reduced fuel consumption and lower operational costs due to the incorporation of free solar energy.

- Grid Stability:

Hybrid systems help stabilize grids by balancing intermittent solar energy with the consistent output of gas turbines. Combines solar PV, battery storage, and gas turbines for enhanced grid stability.

II.6.5 Exercise: Designing a Hybrid Solar-Gas Power Plant

Objective:

Develop a conceptual design for a hybrid solar-gas power installation, calculate its energy output, and analyze its efficiency and environmental benefits.

Problem Statement:

You are tasked with designing a 50 MW hybrid solar-gas power plant for a remote community. The plant will consist of a solar photovoltaic (PV) array and a natural gas turbine. The community has the following requirements:

1. Solar energy should meet at least 50% of the energy demand during daylight hours.
2. The natural gas turbine will provide backup energy during periods of low solar output.
3. The plant should operate at an average efficiency of 55%.
4. Minimize the CO_2 emissions compared to a standalone gas turbine power plant.

Part 1: Solar System Design: Determine Solar Array Capacity:
Assume the location receives 6 peak sunlight hours per day on average.
What size (in MW) should the solar PV array be to meet 50% of the plant's total daily energy output?
Calculate Solar Energy Production:
If the solar panels have an efficiency of 20% and the total area required for the solar array is 5 hectares per MW, how much land area is needed?
Part 2: Gas Turbine Design
Calculate Gas Turbine Capacity:
If the plant must provide a constant 50 MW output, what is the required capacity of the gas turbine to supplement the solar array during nightstroke and cloudy days?
Fuel Consumption:
Assume the gas turbine has a thermal efficiency of 40%, and natural gas has an energy content of 50 MJ/kg.

How much natural gas (in kilograms) will the turbine consume daily if it operates for 12 hours?

Part 3: Efficiency and Emissions

Overall Efficiency:

Combine the contributions of the solar PV array and the gas turbine to calculate the plant's average efficiency.

CO_2 Emissions Reduction:

A standalone gas turbine plant of the same capacity emits 0.5 kg of CO_2 per kWh.

How much CO_2 is avoided daily by integrating solar energy into the system?

Solution

Part 1: Solar System Design

1. Determine Solar Array Capacity

The plant's total daily energy output must be:

50 MW×24 hours=1200 MWh.

If solar energy contributes 50% of the total daily energy, the solar contribution is:

600 MWh.

Assume the location receives 6 peak sunlight hours per day:

Solar Capacity=Solar Energy ContributionSunlight Hours=600 MWh/6 hours=100 MW

2. Calculate Solar Energy Production and Land Area

Each MW of solar capacity requires 5 hectares of land:

Land Area=100 MW×5 hectares/MW=500 hectares

Part 2: Gas Turbine Design

1. Calculate Gas Turbine Capacity

The plant must generate 50 MW continuously. During 12 hours/day when solar energy is not available, the gas turbine must supply the full load:

Gas Turbine Capacity=50 MW.

2. Fuel Consumption of the Gas Turbine

The gas turbine has a thermal efficiency of 40%, and natural gas has an energy content of 50 MJ/kg.

First, calculate the total electrical energy output of the gas turbine during 12 hours of operation:

Electrical Energy Output=50 MW×12 hours=600 MWh

Convert to megajoules (MJ):

600 MWh=600×3600 MJ=2160000 MJ.

The thermal energy input required (using efficiency η=40%=0.4 is:

Fuel Energy Required=Electrical Energy Output *η=2160000 MJ*0.4=5400000 MJ

Using the energy content of natural gas 50 MJ/kg

Fuel Consumption=Fuel Energy Required/Energy Content per kg=5400000 MJ/50 MJ/kg=108000 kg

Daily fuel consumption is 108000 kg of natural gas.

Part 3: Efficiency and Emissions

1. Overall Efficiency of the Hybrid System

The plant operates as a combination of solar and gas systems. Calculate the total energy produced by both sources:

- Solar contribution: 600 MWh
- Gas turbine contribution: 600 MWh

The total energy output is:

Total Energy Output=600+600=1200 MWh.

The solar system is 100% efficient, and the gas turbine has a thermal efficiency of 40%.

The weighted average efficiency η_{hybrid} is:

η_{hybrid}=Solar Energy Output×η_{solar}+Gas Energy Output×η_{gas}
Substitute values:
η_{hybrid}=600×1+600×0.41200=600+2401200=0.7
The overall efficiency of the hybrid plant is 70%.

2. CO₂ Emissions Reduction
A standalone gas turbine emits 0.5 kg of CO_2 per kWh. If the gas turbine were to generate the entire daily output of 1200 MWh
Standalone CO_2 Emissions=1200×1000×0.5=600000 kg CO_2/day
In the hybrid system, only 600 MWh is generated by the gas turbine:
Hybrid Gas CO_2 Emissions=600×1000×0.5=300000 kg CO_2/day.
The reduction in CO_2 emissions is:
CO_2 Reduction=600000−300000=300000 kg CO_2/day

II.7 Cogeneration plants.

II.7.1 Introduction

Cogeneration is *the production of more than one useful form of energy (such as process heat and electric power) from the same energy source*. Either a steam-turbine (Rankine) cycle or a gas-turbine (Brayton) cycle or even a combined cycle can be used as the power cycle in a cogeneration plant. The schematic of an ideal steam-turbine cogeneration plant is shown in Fig.II 6. This plant is to supply process heat $\dot{Q}_p$ at 500 kPa at a rate of 100 kW. To meet this demand, steam is expanded in the turbine to a pressure of 500 kPa, producing power at a rate of 20 kW. The flow rate of the steam can be adjusted such that steam leaves the process heating section as a saturated liquid at 500 kPa. Steam is then pumped to the boiler pressure and is heated in the boiler to state 3. The 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 120 kW. Probably the most striking feature of the ideal steam-turbine cogeneration plant shown in Fig.II 8 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 utilized as either process heat or electric power. Thus it is appropriate to define a utilization factor ε_u for a cogeneration plant as:

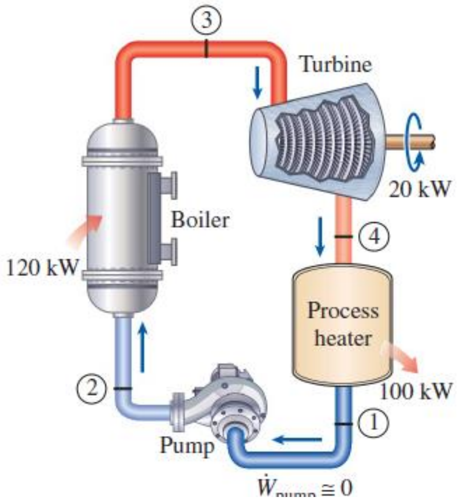
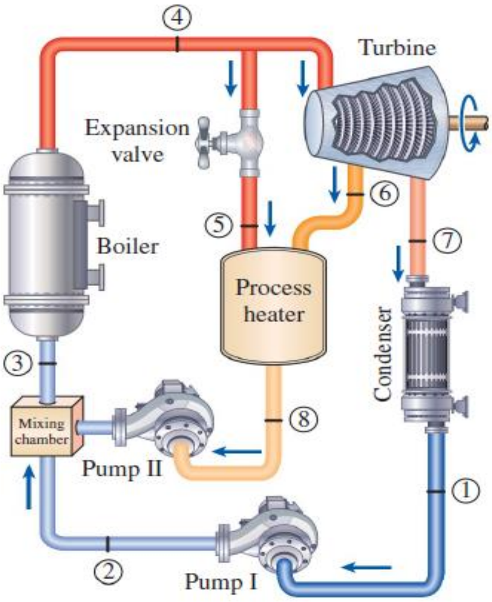


Fig.II 6 An ideal cogeneration plant.

$$\varepsilon_u = \frac{Net\ power\ output + \ Process\ heat\ delivered}{Total\ heat\ input} = \frac{\dot{W}_{net} + \dot{Q}_p}{\dot{Q}_H} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}} \quad 0(II-11)$$

Where $\dot{Q}_{out}$ represents the heat rejected in the condenser. Strictly speaking, $\dot{Q}_{out}$ also includes all the undesirable heat losses from the piping and other components, but they are usually small and thus neglected. It also includes combustion inefficiencies such as incomplete combustion and stack losses when the utilization factor is defined on the basis of the heating value of the fuel. The utilization factor of the ideal steam-turbine cogeneration plant is obviously 100 percent. Actual cogeneration plants have utilization factors as high as 80 percent. Some recent cogeneration plants have even higher utilization factors. Notice that without the turbine, we would need to supply heat to the steam in the boiler at a rate of only 100 kW instead of at 120 kW. The additional 20 kW of heat supplied is converted to work. Therefore, a cogeneration power plant is equivalent to a process-heating plant combined with a power plant that has a thermal efficiency of 100 percent. The ideal steam-turbine cogeneration 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 (but more complex) cogeneration plant is shown in Fig.II 7. Under normal operation, some steam is extracted from the turbine at some predetermined intermediate pressure *P*6. The rest of the steam expands to the condenser pressure *P*7 and is then cooled at constant pressure. The heat rejected from the condenser represents the waste heat for the cycle. At strokes of high demand for process heat, all the steam is routed to the process-heating units and none to the condenser ($\dot{m}_7 = 0$). The waste heat is zero in this mode. If this is not sufficient, some steam leaving the boiler is throttled by an expansion or pressure-reducing valve (PRV) to the extraction pressure *P*6 and is directed to the process-heating unit. Maximum process heating is realized when all the steam leaving the boiler passes through the expansion valve ($\dot{m}_5 = \dot{m}_4$). No power is produced in this mode. When there is no demand for process heat, all the steam passes through the turbine and the condenser ($\dot{m}_5 = \dot{m}_6 = 0$), and the cogeneration 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 cogeneration plant can be expressed as follows:



0(II-12)

$$\dot{Q}_{in} = \dot{m}_3 (h_4 - h_3)$$

0(II-13)

$$\dot{Q}_{out} = \dot{m}_7 (h_7 - h_1)$$

0(II-14)

$$\dot{Q}_P = \dot{m}_5 h_5 + \dot{m}_6 h_6 - \dot{m}_8 h_8$$

0(II-15)

$$\dot{W}_{turb} = (\dot{m}_4 - \dot{m}_5)(h_4 - h_6) + \dot{m}_7 (h_6 - h_7)$$

II.7.2 EXERCISE *:* An Ideal Cogeneration Plant

Consider the cogeneration plant shown in Fig.II 9. Steam enters the turbine at 7 MPa and 500°C. Some steam is extracted from the turbine at 500 kPa for process heating. The remaining steam continues to expand to 5 kPa. Steam is then condensed at constant pressure and pumped to the boiler pressure of 7 MPa. At strokes of high demand for process heat, some steam leaving the boiler is throttled to 500 kPa and is routed to the process heater. The extraction fractions are adjusted so that steam leaves the process heater as a saturated liquid at 500 kPa. It is subsequently pumped to 7 MPa. The mass flow rate of steam through the boiler is 15 kg/s. Disregarding any pressure drops and heat losses in the piping and assuming the turbine and the pump to be isentropic, determine:

1. The maximum rate at which process heat can be supplied,
2. The power produced and the utilization factor when no process heat is supplied,

3. The rate of process heat supply when 10 percent of the steam is extracted before it enters the turbine and 70 percent of the steam is extracted from the turbine at 500 kPa for process heating.

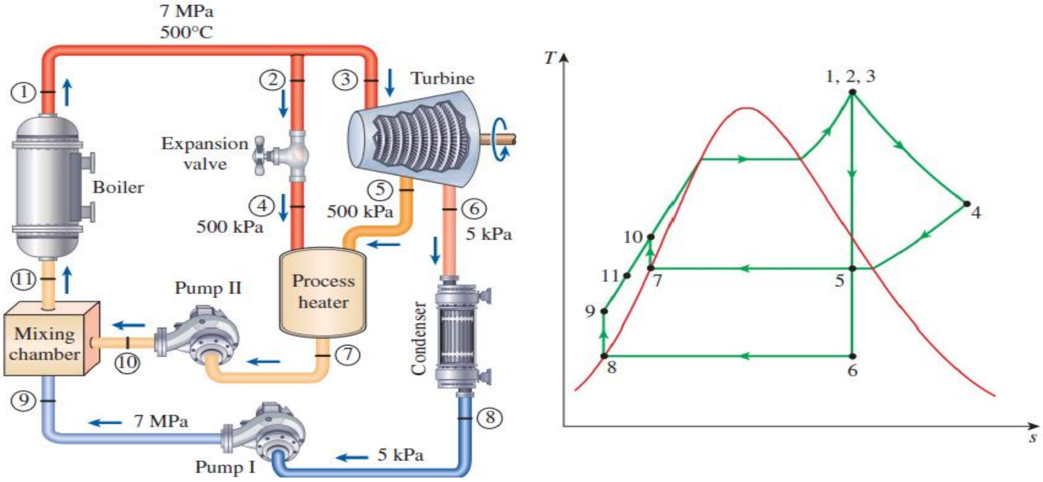


Fig.II 8 Schematic and T-s diagram for Example II.7.2

Solution

This is a classic thermodynamics problem involving a cogeneration (combined heat and power) system, and we'll analyze the Rankine cycle with process heat extraction. The system includes a turbine with extraction, a condenser, pumps, and boiler, with isentropic turbine and pump behavior.
We are given:
Steam inlet to turbine: 7 MPa, 500°C
Extraction pressure for process heat: 500 kPa
Final expansion to: 5 kPa
Condenser at 5 kPa
Steam leaves process heater as saturated liquid at 500 kPa
Mass flow rate through boiler: 15 kg/s
All pumps/turbines are isentropic
Ignore pressure losses

Step-by-step plan:
We'll approach this in three parts, per the questions:

1. Maximum process heat rate
This occurs when all 15 kg/s bypasses the turbine and is throttled directly from 7 MPa to 500 kPa for process heat.
States:
State 1: Boiler outlet = 7 MPa, 500°C → Superheated steam
State 2: Throttled to 500 kPa (isenthalpic process) → Used in process heater
State 3: Leaves heater as saturated liquid at 500 kPa
Use steam tables:
At 7 MPa and 500°C: $h_1 \approx 3410$ kJ/kg
After throttling to 500 kPa (isenthalpic):
$h_2 = h_1 = 3410$ kJ/kg
At 500 kPa, saturated liquid: $h_3 = h_f$ at 500 kPa ≈ 640 kJ/kg
Heat extracted for process:

$$\dot{Q}_{\text{process, max}} = \dot{m} \cdot (h_2 - h_3) = 15 \cdot (3410 - 640) = 15 \cdot 2770 = \boxed{41{,}550 \text{ kW}}$$

2. Power and utilization factor (no process heat)

All 15 kg/s goes through turbine, no extraction. Full expansion from 7 MPa, 500°C to 5 kPa.

States:

State 1: 7 MPa, 500°C → $h_1 = 3410, s_1 \approx 6.6$ kJ/kg·K

State 2: Isentropic expansion to 5 kPa → find h_2 from entropy

From steam tables:

At 5 kPa: $s_f = 0.476, s_g = 8.396$ → Use interpolation

Let:

$$x_2 = \frac{s_1 - s_f}{s_g - s_f} = \frac{6.6 - 0.476}{8.396 - 0.476} \approx 0.798$$

$$h_2 = h_f + x_2(h_{fg}) = 137.8 + 0.798 \cdot (2562.5) \approx 137.8 + 2045 = \boxed{2183 \text{ kJ/kg}}$$

Turbine work:

$$w_t = h_1 - h_2 = 3410 - 2183 = 1227 \text{ kJ/kg}$$

$$\dot{W}_{\text{turbine}} = \dot{m} \cdot w_t = 15 \cdot 1227 = \boxed{18{,}405 \text{ kW}}$$

Pump work:

Pump raises from 5 kPa to 7 MPa.

At 5 kPa: saturated liquid, $v \approx 0.001\, m^3/kg$

$$w_p = v(P_2 - P_1) = 0.001(7000 - 5) = 6.995 \text{ kJ/kg}$$

$$\dot{W}_{\text{pump}} = 15 \cdot 6.995 \approx \boxed{105 \text{ kW}}$$

Net power:

$$\dot{W}_{\text{net}} = \dot{W}_{\text{turbine}} - \dot{W}_{\text{pump}} = 18{,}405 - 105 = \boxed{18{,}300 \text{ kW}}$$

Utilization factor:

$$\text{Utilization factor} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{boiler}}}$$

Boiler input:

Inlet to boiler = after condenser + pump

Enthalpy before boiler $h_{pump\_in} = h_f @ 5kPa = 137.8$

Pumped to 7 MPa adds 6.995 → $h_{boiler\_in} \approx 144.8$

$$q_{in} = h_1 - h_{\text{boiler in}} = 3410 - 144.8 = 3265.2 \text{ kJ/kg}$$

$$\dot{Q}_{\text{in}} = 15 \cdot 3265.2 \approx 48{,}978 \text{ kW}$$

$$\boxed{\text{Utilization factor} = \frac{18{,}300}{48{,}978} \approx 0.3737 \text{ or } 37.4\%}$$

3. Process heat with 10% pre-throttled, 70% extracted

Breakdown:

10% of 15 kg/s → 1.5 kg/s throttled to 500 kPa

70% of 15 kg/s → 10.5 kg/s extracted at 500 kPa

Remaining 20% → 3 kg/s expands fully to 5 kPa

First: Process heat from throttled steam

Same as before:

$h_1 = 3410, h_3 = 640$

$$\dot{Q}_{\text{throttle}} = 1.5 \cdot (3410 - 640) = \boxed{4{,}155 \text{ kW}}$$

Second: Process heat from extracted steam

Steam expands from 7 MPa, 500°C to 500 kPa

Isentropic, so find $h_{extracted}$

At 500 kPa: $s_f = 1.303, s_g = 7.707$

$$x = \frac{6.6 - 1.303}{7.707 - 1.303} \approx 0.88$$

$$h_{extracted} = h_f + xh_{fg} = 640 + 0.88 \cdot 2108 = 640 + 1855 = \boxed{2495\ \text{kJ/kg}}$$

$$\dot{Q}_{\text{extraction}} = 10.5 \cdot (2495 - 640) = 10.5 \cdot 1855 = \boxed{19{,}478\ \text{kW}}$$

Total process heat:

$$\boxed{\dot{Q}_{\text{process, total}} = 4{,}155 + 19{,}478 = 23{,}633\ \text{kW}}$$

II.8 Concept of Nuclear Power Plants

II.8.1 Introduction

A nuclear power plant is a facility that generates electricity by harnessing the energy released during nuclear reactions. The process relies on nuclear fission, where heavy atomic nuclei (usually uranium-235 or plutonium-239) split into smaller fragments when struck by neutrons, releasing a tremendous amount of energy in the form of heat. This heat is used to produce steam, which drives turbines connected to generators, ultimately converting the thermal energy into electricity. Nuclear power plants are a significant source of low-carbon electricity and play a critical role in meeting global energy demands while reducing greenhouse gas emissions.

II.8.2 Main Components of a Nuclear Reactor:

The basic components of nuclear power plant are presented in figure Fig.II 10

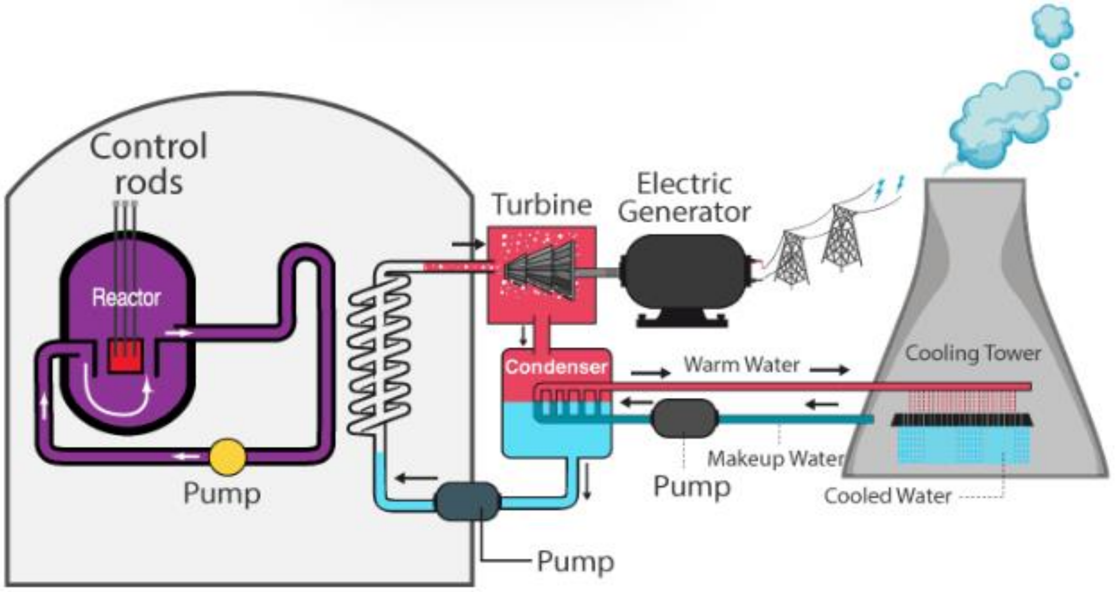


Fig.II 9 Main Components of a Nuclear Reactor

1. **The core of the plant where nuclear fission occurs. It contains:**
    - Fuel Rods: Cylindrical tubes containing enriched uranium or plutonium pellets.
    - Control Rods: Used to regulate the fission reaction by absorbing neutrons.
    - Moderator: A material (e.g., water, graphite) that slows down neutrons to sustain the chain reaction.
2. **Coolant System:**
    - Transfers heat generated in the reactor to the steam generator. Common coolants include water, liquid sodium, or gas.
3. **Steam Generator:**
    - Converts water into steam using the heat from the reactor's coolant system.

4. **Turbine and Generator:**
    - The steam drives a turbine connected to a generator, producing electricity.
5. **Condenser:**
    - Converts the used steam back into water to be reused in the cycle.
6. **Containment Structure:**
    - A robust, sealed building made of steel and concrete that encloses the reactor to prevent the escape of radioactive materials.
7. **Cooling System:**
    - A cooling tower or system that dissipates excess heat into the environment.

II.8.3 Principle of Operation

1. **Nuclear Fission:**
    - The plant begins with the controlled splitting of uranium or plutonium atoms inside the reactor core.
    - Each fission event releases heat and additional neutrons, which cause a self-sustaining chain reaction.
2. **Heat Transfer:**
    - The heat produced by fission is absorbed by the coolant, which is then used to produce steam in the steam generator.
3. **Electricity Generation:**
    - High-pressure steam drives the turbine, which rotates the generator to produce electricity.
4. **Recycling:**
    - The steam is cooled, condensed, and returned to the cycle.

II.8.4 Types of Nuclear Power Plants

1. **Pressurized Water Reactor (PWR):**
    - Uses pressurized water as both a coolant and a moderator. The steam is produced in a separate steam generator.
2. **Boiling Water Reactor (BWR):**
    - Boils water directly in the reactor core to produce steam that drives the turbine.
3. **Fast Breeder Reactor (FBR):**
    - Utilizes fast neutrons to sustain fission and produces more fissile material than it consumes.
4. **Small Modular Reactors (SMRs):**
    - Compact and scalable reactors designed for smaller power requirements and quicker deployment.

II.8.5 Advantages of Nuclear Power Plants

1. **Low Carbon Emissions:**
    - Nuclear energy produces minimal greenhouse gases, making it a key technology for combating climate change.
2. **High Energy Density:**
    - A small amount of nuclear fuel produces a vast amount of energy compared to fossil fuels.

3. **Reliable Power Supply:**
    - Unlike renewable sources like solar or wind, nuclear power provides continuous, stable energy output.
4. **Long-Term Fuel Availability:**
    - With advancements like breeder reactors and thorium fuel cycles, nuclear energy has the potential for long-term sustainability.

II.8.5.1 Exercise: Nuclear Power Plant Efficiency and Output Calculation

A nuclear power plant operates with a thermal power output of 3000 MW from its reactor core. The plant uses a steam turbine generator with an overall thermal-to-electrical efficiency of 35%.

I. Questions:

1. Electrical Power Output:
   Calculate the electrical power output of the plant.
2. Waste Heat:
   Determine the amount of waste heat released into the environment.
3. Fuel Consumption:
   Assume the plant uses enriched uranium-235 as fuel. One gram of uranium-235 releases 8.2×10^{13} J during fission. Calculate the amount of uranium-235 consumed per day.
4. Efficiency Improvement:
   If the efficiency of the plant is improved to 40%, how much additional electrical power would be generated?

Solution to the Nuclear Power Plant Exercise

II. Given Data:

- Thermal power output (P_{th}) = 3000 MW
- Efficiency η = 35%
- Energy released per gram of uranium-235 = 8.2×10^{13} J
- 1 MW = 1×10^{6} W
- Stroke per day = 24 hours = 86400 seconds

1. Electrical Power Output

The formula to calculate electrical power output is:

$P_{el}=P_{th}\times\eta_P$

Substitute the values:

$P_{el}=3000\times0.35=1050$ MW

Electrical Power Output = 1050 MW

2. Waste Heat Released

The waste heat (Q_{waste}) is the remaining thermal energy not converted to electricity:

$Q_{waste}=P_{th}-P_{el}$

Substitute the values:

$Q_{waste}=3000-1050=1950$ MW

Waste Heat Released = 1950 MW

3. Uranium-235 Consumption

First, convert the thermal power to energy per second:

$P_{th}=3000\times10^{6}\text{ W}=3\times10^{9}\text{ J/s}$

The total thermal energy per day is:

$E_{th}=P_{th}\times$Stroke per day

$E_{th}=3\times10^{9}\times86400=2.592\times10^{14}$ J/day

The energy released per gram of uranium-235 is:

8.2×10^{13} J/gram

To find the mass of uranium-235 consumed per day:

Mass=E_{th}/Energy per Gram

Mass=$2.592\times10^{14}/8.2\times10^{13}\approx3.16$ grams/day

Uranium-235 Consumption = 3.16 grams per day

4. Efficiency Improvement to 40%

The new electrical power output ($P_{el'}$) is:

$P'_{el}=P_{th}\times\eta'_{P}$

Where $\eta'=0.40$:

$P'_{el}=3000\times0.40=1200$ MWP

The additional power generated is:

$\Delta P=1200-1050=150$ MW

Additional Electrical Power = 150 MW

CHAPTER THREE. EXERGY AND EXERGETIC

III.1 Definition

The exergy is s the possible work we can extract from a given physical setup when it is allowed to interact with the ambient and the process end state is at P_0, T_0.

III.2 Dead state

A system is said to be in the dead state when it is in thermodynamic equilibrium with the environment it is in (Fig. 3–1). At the dead state, a system is at the temperature and pressure of its environment (in thermal and mechanical equilibrium); it has no kinetic or potential energy relative to the environment (zero velocity and zero elevation above a reference level); and it does not react with the environment

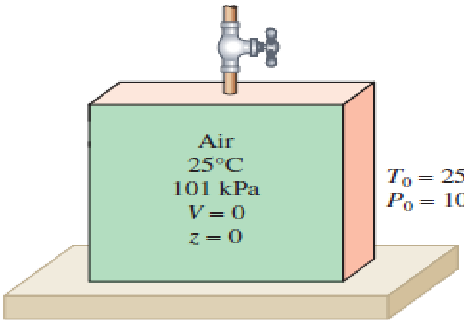


Fig.III 1 A system that is in equilibrium with its environment is said to be at the dead state.

III.3 Surroundings, immediate surroundings, environment

III.3.1 Surroundings

Surroundings are everything outside the system boundaries.

III.3.2 Immediate surroundings

Immediate surroundings refers to the portion of the surroundings that is affected by the process,

III.3.3 Environment

Environment refers to the region beyond the immediate surroundings whose properties are not affected by the process at any point.

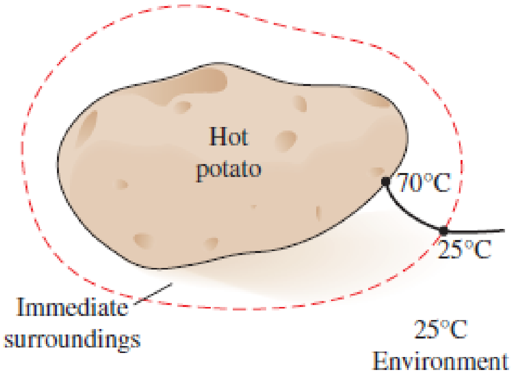


Fig.III 2 The immediate surroundings of a hot potato are simply the temperature gradient zone of the air next to the potato.

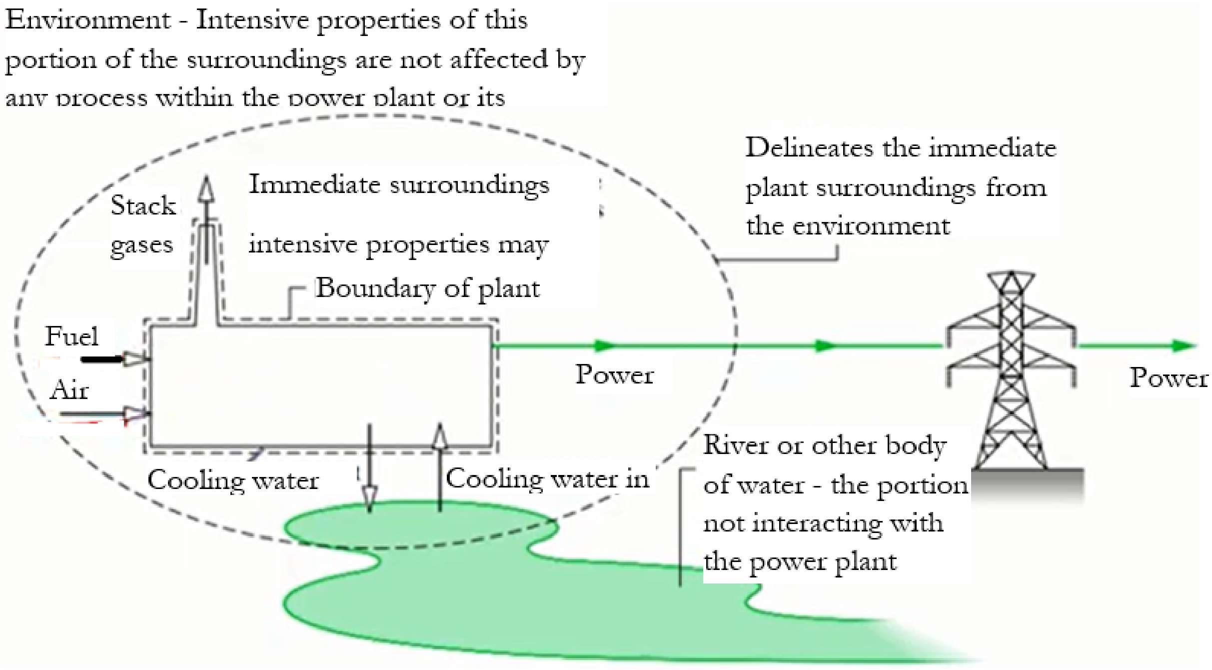


Fig.III 3 The immediate surroundings of electrical power plant

A system delivers the maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state. This represents the useful work potential of the system at the specified state and is called exergy.

III.4 Exergy Associated with Kinetic and Potential Energy

Kinetic energy is a form of mechanical energy, and thus it can be converted to work entirely. Therefore, the exergy of the kinetic energy of a system is equal to the kinetic energy itself regardless of the temperature and pressure of the environment. That is, the **Exergy of kinetic energy**

$$x_{ke} = ke = \frac{V^2}{2}\left(\frac{kJ}{kg}\right) \qquad 0(\text{III-1})$$

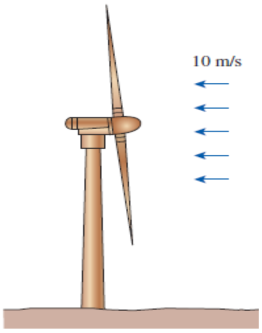
where V is the velocity of the system relative to the environment.

Potential energy is also a form of mechanical energy, and thus it can be converted to work entirely. Therefore, the exergy of the potential energy is equal to the potential energy itself, regardless of the temperature and pressure of the environment. That is, **Exergy of potential energy**:

$$x_e = pe = gz\left(\frac{kJ}{kg}\right) \qquad 0(\text{III-2})$$

where g is the gravitational acceleration and z is the elevation of the system relative to a reference level in the environment. Therefore, the exergies of kinetic and potential energies are equal to themselves, and they are entirely available for work. However, the internal energy u and enthalpy h of a system are not entirely available for work

III.4.1.1 exercise: Maximum Power Generation by a Wind Turbine



A wind turbine with a 12 m diameter rotor, as shown in Figure below, is to be installed at a location where the wind is blowing steadily at an average velocity of 10 m/s.

Determine the maximum power that can be generated by the wind turbine. Air is at standard conditions of 1 atm and 25°C, and thus its density is 1.18 kg/m^3

III.5 EXERGY CHANGE OF A SYSTEM

The property exergy is the work potential of a system in a specified environment and represents the maximum amount of useful work that can be obtained as the system is brought to equilibrium with the environment. The exergy of a system that is in equilibrium with its environment is zero. The state of the environment is referred to as the "dead state" since the system is practically "dead" (cannot do any work) from a thermodynamic point of view when it reaches that state

III.5.1 Exergy of a Fixed Mass: Nonflow (or Closed System) Exergy

Consider a piston–cylinder device that contains a fluid of mass m at temperature T and pressure P. The system (the mass inside the cylinder) has a volume V, internal energy U, and entropy S. The system is now allowed to undergo a differential change of state during which the volume changes by a differential amount dV and heat is transferred in the differential amount of dQ. Taking the direction of heat and work transfers to be from the system (heat and work outputs), the energy balance for the system during this differential process can be expressed as

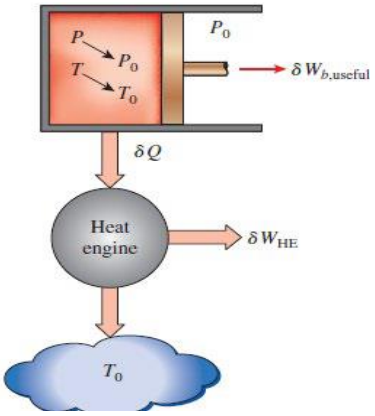


Fig.III 4 The exergy of a specified mass at a specified state is the useful work that can be produced as the mass undergoes a reversible process to the state of the environment

$$\underbrace{\delta_{Ein} - \delta_{Eout}}_{\text{Net energy transfer}} = \underbrace{dE_{system}}_{\text{Change in internal, kinetic,}} \qquad 0(\text{III-3})$$

Since the only form of energy the system contains is internal energy, and the only forms of energy transfer a fixed mass can involve are heat and work.

$$dU = -\delta Q - \delta W \qquad 0(\text{III-4})$$

Also, the only form of work a simple compressible system can involve during a reversible process is the boundary work, which is given to be

$$\delta W = P\,dV \qquad 0(\text{III-5})$$

when the direction of work is taken to be from the system, otherwise it would be
$\delta W = -P\,dV$ (III.2)
The pressure P in the PdV expression is the absolute pressure, which is measured from absolute zero. Any useful work delivered by a piston–cylinder device is due to the pressure above the

atmospheric level. The pressure P in the PdV expression is the absolute pressure, which is measured from absolute zero. Any useful work delivered by a piston–cylinder device is due to the pressure above the atmospheric level. Therefore,

$$\delta W = P\,dV = (P - P_0)dV + P_0\,dV = \delta W_{b,useful} + P_0 dV \tag{0(III-6)}$$

Noting that $dS = \frac{dQ}{T}$ for a reversible process, and the thermal efficiency of a reversible heat engine operating between the temperatures of T and T_0 is $\eta = 1 - \left(\frac{T_0}{T}\right)$, the differential work produced by the engine as a result of this heat transfer is

$$\delta W_{HE} = \left(1 - \frac{T_0}{T}\right)\delta Q = \delta Q - \frac{T_0}{T}\delta Q = \delta Q - (-T_0 dS) \rightarrow \delta Q = \delta W_{HE} - T_0 dS \tag{0(III-7)}$$

We have

$$\delta W = \delta W_{b,useful} + P_0 dV \tag{0(III-8)}$$

And we have to

$$\delta Q = \delta W_{HE} - T_0 dS \tag{0(III-9)}$$

By substituting (III.8) and (III.9) in (III.6), we obtain:

$$dU = -\delta Q - \delta W = -\delta W_{HE} + T_0 dS - \delta W_{b,useful} - P_0 dV \tag{0(III-10)}$$

From this equation (III.10), the total useful work is derived

$$\delta W_{total\ usuful} = \delta W_{HE} + \delta W_{b,useful} = T_0 dS - dU - P_0 dV \tag{0(III-11)}$$

Integrating from the given state (no subscript) to the dead state (0 subscript) we obtain

$$W_{total\ usuful} = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) \tag{0(III-12)}$$

Where $W_{total\ usuful}$ is the total useful work delivered as the system undergoes a reversible process from the given state to the dead state, which is *exergy* by definition.

A closed system, in general, may possess kinetic and potential energies, and the total energy of a closed system is equal to the sum of its internal, kinetic, and potential energies. Noting that kinetic and potential energies themselves are forms of exergy, the exergy of a closed system of mass m is

$$X = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) + m\frac{V^2}{2} + mgz \tag{0(III-13)}$$

On a unit mass basis, the closed system (or nonflow) exergy Φ is expressed as

$$\Phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \tag{0(III-14)}$$

Since the total energy

$$(e - e_0) = (u - u_0) + \frac{V^2}{2} + gz \tag{0(III-15)}$$

Then we obtain

$$\Phi = (e - e_0) + P_0(v - v_0) - T_0(s - s_0) \tag{0(III-16)}$$

Theo exergy change of a closed system during a process is simply the difference between the final and initial exergies of the system,

$$\Phi = (e - e_0) + P_0(v - v_0) - T_0(s - s_0) \qquad 0(III\text{-}17)$$

or, on a unit mass basis,

$$\begin{aligned}\Delta X &= X_2 - X_1 = m(\Phi_2 - \Phi_1) \\ &= (E_2 - E_1) + P_0(V_2 - V_1) \\ &- T_0(S_2 - S_1)\end{aligned} \qquad 0(III\text{-}18)$$

$$\begin{aligned}\Delta X = &(U_2 - U_1) + P_0(V_2 - V_1) \\ &- T_0(S_2 - S_1) + m\frac{V_2^2 - V_1^2}{2} \\ &+ mg(z_2 - z_1)\end{aligned} \qquad 0(III\text{-}19)$$

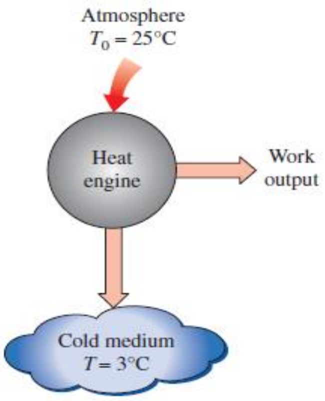
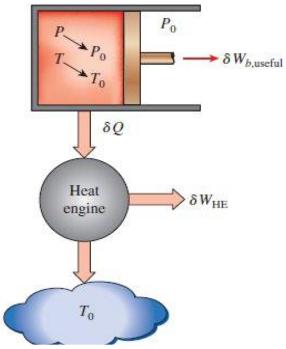


Fig.III 5 The exergy of a cold medium is also a positive quantity since work can be produced by transferring heat to it.

$$\Delta\Phi = \Phi_2 - \Phi_1 = (u_2 - u_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1 \qquad 0(III\text{-}20)$$

$$\Delta\Phi = (e_2 - e_1) + P_0(v_2 - v_1) - T_0(s_2 - s) \qquad 0(III\text{-}21)$$

The exergy of a closed system is either positive or zero. It is never negative. Even a medium at low temperature ($T<T_0$) and/or low pressure ($P<P_0$) contains exergy since a cold medium can serve as the heat sink to a heat engine that absorbs heat from the environment at T_0, and an evacuated space makes it possible for the atmospheric pressure to move a piston and do useful work



III.5.2 Exergy of a Flow Stream: Flow (or Stream) Exergy

A flowing fluid has an additional form of energy, called the flow energy, which is the energy needed to maintain flow in a pipe or duct, and it is expressed as $w_{flow} = Pv$ where v is the specific volume of the fluid, which is equivalent to the volume change of a unit mass of the fluid as it is displaced during flow. The flow work is essentially the boundary work done by a fluid on the fluid downstream, and thus the exergy associated with flow work is equivalent to the exergy associated with the boundary work, which is the boundary work in excess of the work done against the atmospheric air at P_0 to displace it by a volume v (Fig). Noting that the flow work is Pv and the work done against the atmosphere is P_0v, the exergy associated with flow energy can be expressed as

$$\Delta\Phi = (e_2 - e_1) + P_0(v_2 - v_1) - T_0(s_2 - s) \qquad 0(III\text{-}22)$$

$$x_{flow} = Pv - P_0v = (P - P_0)v \qquad 0(III\text{-}23)$$

Therefore, the exergy associated with flow energy is obtained by replacing the pressure P in the flow work relation by the pressure in excess of the atmospheric pressure, P 2 P0. Then the exergy of a flow stream is determined by simply adding the flow exergy relation above to the exergy relation in Eq. III–10 for a nonflowing fluid,

$$x_{flowingfluid} = x_{nonflowingfluid} + x_{flow} \quad 0(III\text{-}24)$$

$$\psi = (u-u_0)+P_0(v-v_0)-T_0(s-s_0)+\frac{V^2}{2}+gz+(P-P_0)v \quad 0(III\text{-}25)$$

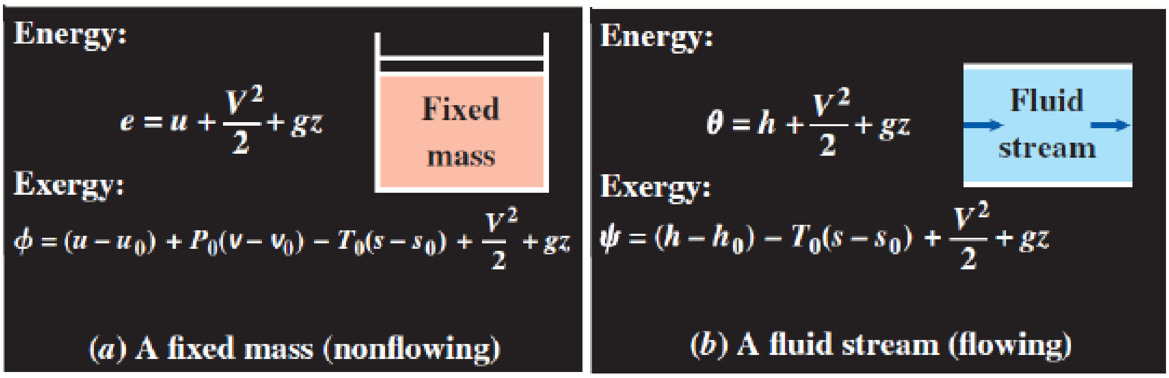
$$\psi = (u-u_0)+P_0(v-v_0)-T_0(s-s_0)+\frac{V^2}{2}+gz+(P-P_0)v \quad 0(III\text{-}26)$$

$$\psi = (u+Pv)-(u_0+P_0v_0)-T_0(s-s_0)+\frac{V^2}{2}+gz \quad 0(III\text{-}27)$$

$$\psi = (h-h_0)-T_0(s-s_0)+\frac{V^2}{2}+gz \quad 0(III\text{-}28)$$

the exergy change of a fluid stream as it undergoes a process from state 1 to state 2 becomes

$$\Delta\psi = \psi_2-\psi_1=(h_2-h_1)-T_0(s_2-s_1)+\frac{V_2^2-V_1^2}{2}+g(z_2-z_1) \quad 0(III\text{-}29)$$



The *energy* and *exergy* contents of (*a*) a fixed mass and (*b*) a fluid stream.

For fluid streams with negligible kinetic and potential energies, the kinetic and potential energy terms drop out. Note that the exergy change of a closed system or a fluid stream represents the maximum amount of useful work that can be done (or the minimum amount of useful work that needs to be supplied if it is negative) as the system changes from state 1 to state 2 in a specified environment, and represents the reversible work $R_{eversible}$. It is independent of the type of process executed, the kind of system used, and the nature of energy interactions with the surroundings. Also note that the exergy of a closed system cannot be negative, but the exergy of a flow stream can at pressures below the environment pressure P_0.

III.6 EXERGY TRANSFER BY HEAT, WORK, AND MASS

Exergy, like energy, can be transferred to or from a system in three forms: *heat, work,* and *mass flow*. Exergy transfer is recognized at the system boundary as exergy crosses it, and it represents the exergy gained or lost by a system during a process. The only two forms of exergy interactions associated with a fixed mass or closed system are heat transfer and work.

III.6.1 Exergy by Heat Transfer, *Q*

The work potential of the energy transferred from a heat source at temperature *T* is the maximum work that can be obtained from that energy in an environment at temperature T_0 and is equivalent to the work produced by a Carnot heat engine operating between the source and the environment.

Therefore, the Carnot efficiency $\eta_C = 1-(T_0/T)$ represents the fraction of energy of a heat source at temperature T that can be converted to work

- Exergy transfer by heat:

$$X_{heat} = \left(1-\frac{T_0}{T}\right)Q \quad 0(\text{III-30})$$

When the temperature T at the location where heat transfer is taking place is not constant, the exergy transfer accompanying heat transfer is determined by integration to be:

$$X_{heat} = \int\left(1-\frac{T_0}{T}\right)\delta Q \quad 0(\text{III-31})$$

- Exergy Transfer by Work, W

Exergy is the useful work potential, and the exergy transfer by work can simply be expressed as

$$X_{work} = \begin{cases} W - W_{surr}\ (for\ bondary\ work) \\ W\ (for\ other\ forms\ of\ work) \end{cases} \quad 0(\text{III-32})$$

where $W_{surr} = P_0(V_2 - V_1)$, P_0 is atmospheric pressure, and V_1 and V_2 are the initial and final volumes of the system.

- Exergy Transfer by Mass, m

Mass contains exergy as well as energy and entropy, and the exergy, energy, and entropy contents of a system are proportional to mass. Also, the rates of exergy, entropy, and energy transport into or out of a system are proportional to the mass flow rate. Mass flow is a mechanism to transport exergy, entropy, and energy into or out of a system. When mass in the amount of m enters or leaves a system, exergy in the amount of $m\psi$, where $\psi = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz$, accompanies it. That is,

$$X_{mass} = m\psi \quad 0(\text{III-33})$$

Therefore, the exergy of a system increases by $m\psi$ when mass in the amount of m enters and decreases by the same amount when the same amount of mass at the same state leaves the system. Exergy flow associated with a fluid stream when the fluid properties are variable can be determined by integration from

$$\dot{X}_{mass} = \int_{A_C}\psi\rho V_n dA_C \quad and\ X_{mass} = \int\psi\delta m = \int_{\Delta t}\dot{X}_{mass}dt \quad 0(\text{III-34})$$

where A_c is the cross-sectional area of the flow and V_n is the local velocity normal to dA_c.

III.6.2 Exergy Destruction

Irreversibilities such as friction, mixing, chemical reactions, heat transfer through a finite temperature difference, unrestrained expansion, non quasi equilibrium compression or expansion always generate entropy, and anything that generates entropy always destroys exergy. The exergy destroyed is proportional to the entropy generated, , and is expressed as

$$X_{destroyed} = T_0S_{gen} \geq 0 \quad 0(\text{III-35})$$

Note that exergy destroyed is a positive quantity for any actual process and becomes zero for a reversible process. Exergy destroyed represents the lost work potential and is also called the irreversibility or lost work. No exergy is destroyed during a reversible process ($X_{destroyed,rev} = 0$). The decrease of exergy principle does not imply that the exergy of a system cannot increase. The exergy change of a system can be positive or negative during a process, but exergy destroyed cannot be negative. The decrease of exergy principle can be summarized as follows:

$$X_{destroyed} = \begin{cases} > 0 \ (Irreversible\ process) \\ = 0 \ (Reversible\ process) \\ < 0 \ (Impossible\ process) \end{cases}$$

0(III-36)

III.7 EXERGY BALANCE: CLOSED SYSTEMS

The nature of exergy is opposite to that of entropy in that exergy can be destroyed, but it cannot be created. Therefore, the exergy change of a system during a process is less than the exergy transfer by an amount equal to the exergy destroyed during the process within the system boundaries. Then the decrease of exergy principle can be expressed as

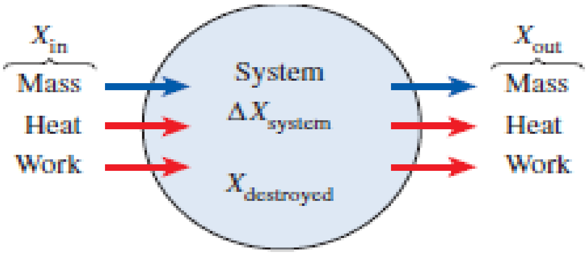


Fig.III 6 Mechanisms of exergy transfer

$$\begin{pmatrix} Total \\ exergy \\ entering \end{pmatrix} - \begin{pmatrix} Total \\ exergy \\ leaving \end{pmatrix} - \begin{pmatrix} Total \\ exergy \\ destroyed \end{pmatrix} = \begin{pmatrix} Change\ in\ the \\ totale\ exergy \\ of\ the\ system \end{pmatrix}$$

0(III-37)

or

$$X_{in} - X_{out} - X_{destroyed} = \Delta X_{system}$$

0(III-38)

Then the exergy balance for any system undergoing any process can be expressed more explicitly as

General:

$$\underbrace{X_{in} \quad - \quad X_{out}}_{\text{Net exergy transfer}} \quad - \quad \underbrace{X_{destroyed}}_{\text{Exergy destruction}} \quad = \quad \underbrace{\Delta X_{system} \qquad (kJ)}_{\text{Change in exergy}}$$

0(III-39)

Or in rate form, as

$$\underset{\text{Rate of net exergy transfer}}{\dot{X}_{in} \quad - \quad \dot{X}_{out}} \quad - \quad \underset{\text{Rate of Exergy destruction}}{\dot{X}_{destroyed}} \quad = \quad \underset{\text{Rate of change in exergy}}{\Delta \dot{X}_{system} \ (\text{kW})}$$

0(III-40)

where the rates of exergy transfer by heat, work, and mass are expressed as

$$\dot{X}_{heat} = \left(1 - \frac{T_0}{T}\right)\dot{Q},\ \dot{W}_{work} = \dot{W}_{usuful}, and\ \dot{X}_{mass} = \dot{m}\psi$$ 0(III-41)

The exergy balance can also be expressed per unit mass as

General, unit-mass basis:

$$X_{in} - X_{out} - X_{destroyed} = \Delta X_{system}\ (\frac{kJ}{kg}$$ 0(III-42)

Also, it is usually more convenient to find the entropy generation S_{gen} first, and then to evaluate the exergy destroyed directly.

$$X_{destroyed} = T_0 S_{gen}\ or\ \dot{X}_{destroyed} = T_0 \dot{S}_{gen}$$ 0(III-43)

A closed system does not involve any mass flow and thus any exergy transfer associated with mass flow. Taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the exergy balance for a closed system can be expressed more explicitly as (See precedent Figure)

$$X_{heat} - X_{work} - X_{destroyed} = \Delta X_{system}$$ 0(III-44)

Or *for a closed system*

$$\sum\left(1 - \frac{T_0}{T_k}\right)Q_k - [W - P_0(V - V_0)] - T_0 S_{gen} = X_2 - X_1$$ 0(III-45)

where Q_k is he the heat transfer through the boundary at the temperature T_k at location *k*. Dividing the previous equation by the time interval Δt and taking the limit as $\Delta t \rightarrow 0$ gives the rate form of the exergy balance for a closed system,

for rate form

$$\sum\left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k - \left(\dot{W} - P_0\frac{dV_{system}}{dt}\right) - T_0\dot{S}_{gen} = \frac{dX_{system}}{dt}$$ 0(III-46)

III.7.1 EXERCISES

III.7.1.1 EXERCISE1

The electric power needs of a community are to be met by windmills with 40 m diameter rotors. The windmills are to be located where the wind is blowing steadily at an average velocity of 6 m/s. Determine the minimum number of windmills that need to be installed if the required power output is 1500 kW

$$x_{ke} = ke = \frac{v^2}{2} = \frac{6^2}{2} = 18J/kg$$

$$\dot{w} = \dot{m}x$$

$$\dot{m} = \rho v A = \rho v\frac{3.14\ D^2}{4} = 1.18 * 6 * \frac{3.14\ 40^2}{4} = 89.04\ kg/s$$

$$P = \rho RT\ \rightarrow \rho = \frac{P}{RT} = \frac{101.325}{287.10^{-3} * 298} = 1.18m^3/kg$$

$$\dot{w} = \dot{m}x = \frac{8904 * 18}{1000} = 160.3\ kW$$

$N = \frac{W_{total}}{W} = \frac{1500}{160.3} = 10$ Windmill

III.7.1.2 EXERCISE 2

Determine the specific exergy of saturated water vapor at 120°C, having a velocity of 30 m/s and an elevation of 6 m, each relative to an exergy reference environment where
To = 298 K (25°C), Po = 1 atm, and g = 9.8 m/s^2.

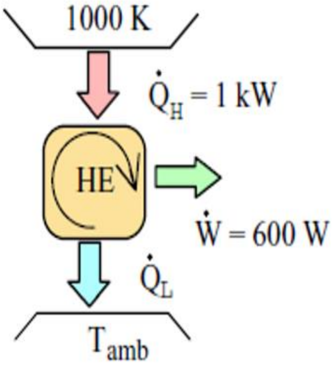
Solution

For water as saturated vapor at 120°C,
Table A-2 gives
v = 0.8919 m^3/kg, u = 2529.3 kJ/kg, s = 7.1296 kJ/kg. K.
At the dead state, where T_o = 298 K (25°C) and p_1 = 1 atm, water is a liquid.

From Table A-2, we get v_0= 1.0029 × 10^{-3} m^3/kg,
u_0 = 104.88 kJ/kg, S_0 = 0.3674 kJ/kg. K. Substituting values
e = (u - u_o) + P_o(v - v_o) - To(s - s_o) +V_2/2+gZ
= [(2529.3-104.88)] + [1.01325 ×105(0.8919 – 1.0029 × 10-3]
-298 (7.1296 – 0.3674) + [(30)^2/2 + 9.8 *6
= (2424.42 + 90.27 - 2015.14 + 0.45 + 0.06) = 500 kJ/ kg

III.7.1.3 EXERCISE 3

A heat engine receives 1 kW heat transfer at 1000 K and gives out 600 W as work with the rest as heat transfer to the ambient.
What are the fluxes of exergy in and out?

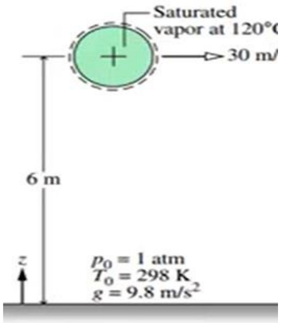


Solution

Exergy flux in: $\dot{\Phi}_H = \left(1 - \frac{T_0}{T_H}\right)\dot{\Phi}_H = \left(1 - \frac{298.15}{1000}\right)1kW = 0.702\ kW$

Exergy flux out: $\dot{\Phi}_L = \left(1 - \frac{T_0}{T_L}\right)\dot{\Phi}_L = 0\ \ because\ T_0 = T_L$

The other exergy flux out is the power $\dot{\Phi}_{out} = 0.6\ kW$



CHAPTER FOUR. THERMODYNAMICS OF COMBUSTION

IV.1 Properties of mixtures

In thermodynamics and combustion, mixtures consist of multiple substances, such as fuel and air before combustion or exhaust gases after combustion. Understanding the properties of mixtures is essential for analyzing energy transfer, chemical reactions, and system performance.

IV.1.1 Composition of Mixtures

IV.1.1.1 Mass and Mole Fractions

For a mixture containing different components, the mass fraction (Yi) and mole fraction (Xi) of each component are defined as:

$$Y_i = \frac{m_i}{m_{total}} \quad 0.1)$$

$$X_i = \frac{n_i}{n_{total}} \quad 0.2)$$

where:
m_i and n_i are the mass and mole amount of component *i.*
m_{total} and n_{total} are the total mass and moles of the mixture.
The sum of all mass fractions and mole fractions must be equal to 1:

$$\sum Y_i = 1 \; and \; \sum X_i = 1 \quad 0.3)$$

IV.1.1.2 Molecular Weight of a Mixture

The average molar mass $(M_{mix})$of a mixture is determined from mole fractions and individual molar masses (M_i):
Alternatively, it can be expressed in terms of mass fractions:

$$\sum Y_i = 1 \; and \; \sum X_i = 1 \quad 0.4)$$

$$M_{mix} = \sum X_i \, M_i \quad 0.5)$$

IV.1.1.3 Specific Gas Constant

The specific gas constant of a mixture R_{mix} is given by:

$$R_{mix} = \frac{R_{mix}}{M_{mix}} \quad 0.6)$$

Where is the universal gas constant (8.314J/mol.K)
For an ideal gas mixture, the equation of state remains:

$$PV = mR_{mix}T \quad 0.7)$$

IV.1.1.4 Specific Heat Capacity of Mixtures

For a gas mixture, the specific heat capacities $(Cp \ and \ Cv)$ are found using mass or mole fractions:

$$C_{P,mix} = Y_i C_{P,i} \ and \ C_{V,mix} = Y_i C_{V,i} \quad 0.8)$$

For mole fractions:

$$C_{P,mix} = \sum Y_i \, C_{P,i} \text{ and } C_{V,mix} = \sum Y_i \, C_{V,i} \quad 0.9)$$

The ratio of specific heats (γ_{mix}) is:

$$\gamma_{mix} = \frac{C_{P,mix}}{C_{V,mix}} \quad 0.10)$$

IV.1.1.5 Enthalpy and Internal Energy of Mixtures

For a mixture of ideal gases, the enthalpy and internal energy are additive:

$$h_{mix} = \sum Y_i \, h_i \ \ and \ u_{mix} = \sum Y_i \, u_i \quad 0.11)$$

Similarly, using mole fractions:

$$h_{mix} = \sum X_i \, h_i \ and \ u_{mix} = \sum X_i \, h_i \quad 0.12)$$

IV.1.2 Dalton’s Law for Gas Mixtures

Dalton’s Law states that in an ideal gas mixture, the total pressure is the sum of the partial pressures of each gas:

$$P_{total} = \sum P_i \quad 0.13)$$

Where $P_i = X_i P_{total}$ is the partial pressure of each gas.

IV.2 stoichiometric combustion,

IV.2.1 Definition

Stoichiometric combustion is the complete combustion of a fuel with the exact amount of oxygen (or air) required to convert the fuel into carbon dioxide fully (CO_2) and water (H_2O), without leaving any excess oxygen or unburned fuel.

$Fuel + Oxygen \ \rightarrow CO_2 + H_2O$

This process ensures maximum energy release with minimal unburned products, making it ideal for efficiency in combustion engines, power plants, and heating systems.

IV.2.2 Stoichiometric Air-Fuel Ratio (AFR)

The stoichiometric air-fuel ratio (AFR_{stoich}) is the mass ratio of air to fuel required for complete combustion. It is determined by balancing the chemical reaction equation.

$$AFR_{stoich} = \frac{Mass \ of \ fuel \ required}{Mass \ of \ fuel} \quad 0.14)$$

IV.2.3 Example: Stoichiometric Combustion of Methane (CH_4)

Balanced chemical equation:

$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$

Since air contains 21% oxygen by volume, the required mass of air can be calculated using molar masses:

$$M_{CH4} = 16\frac{g}{mol}$$

$$M_{O2} = 32\frac{g}{mol}$$

$$M_{air} = 28.97\frac{g}{mol}$$

(since air is mostly nitrogen and oxygen)

Using mass ratios:

$$\frac{2 \times 32}{16} = 4$$

Since oxygen is only 21% of air, the actual air needed is:

$$\frac{4}{0.21} = 17.2$$

Thus, AFR for methane = 17.2:1.

IV.2.4 Equivalence Ratio (ϕ\phi)

The equivalence ratio determines whether combustion is stoichiometric, lean, or rich:

$$\phi = \frac{Actual\frac{fuel}{Air}\ Ratio}{Stoichiometric\frac{fuel}{Air}Ratio} \qquad 0.15)$$

$$\phi = 1\ =\ 1\ \rightarrow\ Stoichiometric\ combustion\ (ideal)$$

$$\phi < 1\ <\ 1\ \rightarrow\ Lean\ combustion\ (excess\ air, lower\ temperature)$$

$$\phi > 11\ \rightarrow\ Rich\ combustion\ (excess\ fuel, more\ CO\ and\ unburned\ hydrocarbons)$$

IV.2.5 heat of formation and calorific values,

IV.2.5.1 Heat of Formation (Enthalpy of Formation)

The heat of formation (also called enthalpy of formation, ΔH_0^f is the heat change when one mole of a compound is formed from its constituent elements in their standard states at 25°C (298 K) and 1 atm pressure.

◆ Standard Heat of Formation (ΔH_0^f)

- For elements in their natural state (e.g., O_2,H_2,N_2O, H_2, N_2O_2,H_2,N_2), the heat of formation is always zero.
- For compounds, it is obtained from experimental data.

Example: Heat of Formation of Some Common Substances

Substance	Chemical Formula	$\Delta H_0^f\ \left(\frac{\text{kJ}}{\text{mol}}\right)$
Oxygen (gas)	O_2	0.00
Nitrogen (gas)	N_2	0.00
Water (liquid)	H_2O	-285.8

Substance	Chemical Formula	$\Delta H_0^f \left(\frac{\mathrm{kJ}}{\mathrm{mol}}\right)$
Carbon Dioxide	CO_2	-393.5
Methane	CH_4	-74.8
Octane	C_8H_{18}	-250.1

IV.2.5.2 Heat of Reaction Using Heat of Formation

The heat released or absorbed in a combustion reaction can be calculated using the enthalpy of formation values of reactants and products.

$$\Delta Hreaction = \sum \Delta H_0^f (products) - \sum \Delta H_0^f (reactants) \qquad 0.16)$$

Example: Combustion of Methane

$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$

Using standard heat of formation values:

$$\Delta H_{reaction} = [(-393.5) + 2(-285.8)] - [(-74.8) + 2(0)]$$
$$= (-393.5 - 571.6) - (-74.8) = -964.3 + 74.8 = -\frac{889.5kJ}{mol}$$

IV.2.5.3 Calorific Value (Heating Value)

The calorific value (also called heat of combustion) is the total amount of heat released when 1 kg of fuel is completely burned.

There are two types:

1. Higher Heating Value (HHV) or Gross Calorific Value (GCV)
   - Includes the latent heat of vaporization of water.
   - Used in power plants where water condensation occurs.
2. Lower Heating Value (LHV) or Net Calorific Value (NCV)
   - Excludes the latent heat of water vapor.
   - Used in internal combustion engines.

Calorific Values of Common Fuels

Fuel	Formula	HHV (MJ/kg)	LHV (MJ/kg)
Methane	CH_4	55.5	50.0
Propane	C_3H_8	50.3	46.4
Octane (Gasoline)	C_8H_{18}	47.9	44.4
Diesel	-	45.5	42.5
Coal	-	24 - 35	22 - 33
Hydrogen	H_2	141.8	120.0

- Hydrogen has the highest HHV (141.8 MJ/kg), making it a strong candidate for clean energy.
- Methane (Natural Gas) has a high LHV and is used widely for domestic and industrial heating.
- Coal has a wide range of values depending on its composition.

IV.2.5.4 Relationship Between Heat of Formation and Calorific Value

- The calorific value of a fuel depends on the heat of formation of its combustion products.

- The more negative the heat of formation of the products, the higher the energy released.
- Example: Methane combustion releases a high amount of heat because CO_2 and H_2O have highly negative enthalpies of formation.

IV.2.5.5 Example: Calculating Heat of Reaction for Propane Combustion

We'll determine how much energy is released when 1 mole of propane (C_3H_8) undergoes complete combustion.
Step 1: Write the Balanced Combustion Equation
$C_3H_8+5O_2\rightarrow 3CO_2+4H_2O$
This equation tells us that:

- 1 mole of propane reacts with 5 moles of oxygen.
- The products are 3 moles of carbon dioxide and 4 moles of water.

Step 2: Use Standard Heat of Formation Values
We use the standard enthalpies of formation (ΔH_0^f) from tables:

Substance	Formula	$\Delta H_0^f\ \left(\frac{\mathrm{kJ}}{\mathrm{mol}}\right)$
Propane (gas)	C_3H_8	-103.8
Oxygen (gas)	O_2	0 (by definition)
Carbon Dioxide	CO_2	-393.5
Water (liquid)	H_2O	-285.8

Step 3: Apply the Heat of Reaction Formula

$$\Delta H_{reaction} = \sum \Delta H_0^f(products) - \sum \Delta H_0^f(reactants) \quad 0.17)$$

Calculate the Total Heat of Formation for Products:

$$\left(3\times \Delta H_0^f CO_2\right) + \left(4\times \Delta H_0^f H_2O\right) = (3\times -393.5) + (4\times -285.8)$$
$$= -1180.5 + (-1143.2) = -2323.7kJ$$

Calculate the Total Heat of Formation for Reactants:

$$(1\times \Delta H_0^f\ C_3H_8) + (5\times \Delta H_0^f O_2) = (-103.8) + (5\times 0) = -103.8\,kJ$$

Final Calculation:

$$\Delta H_{reaction} = (-2323.7) - (-103.8) = -2219.9\,kJ$$

Example: Converting to Heating Value (Calorific Value)
Since fuels are often measured by mass, we convert kJ/mol to MJ/kg.
1 Molar Mass of Propane:

$$C_3H_8 = (3\times 12.01) + (8\times 1.008) = 44.1\,g/mol = 0.0441kg/mol$$

2 Convert to MJ/kg:

$$\frac{2219.9\,kJ/mol}{0.0441\,kg/mol} = 50.3MJ/kg$$

IV.2.6 adiabatic flame temperature.

IV.2.6.1 What is Adiabatic Flame Temperature?

The adiabatic flame temperature is the maximum temperature that can be achieved during a combustion reaction without any heat loss to the surroundings. It assumes that all the chemical energy released is converted into thermal energy of the products.

- Higher Adiabatic Flame Temperature → More efficient combustion
- Lower Adiabatic Flame Temperature → Indicates heat losses or incomplete combustion

There are two types:

1. Constant Pressure Adiabatic Flame Temperature ($T_{ad,p}$) → Occurs in open systems (e.g., jet engines, gas turbines).
2. Constant Volume Adiabatic Flame Temperature ($T_{ad,v}$) → Occurs in closed systems (e.g., internal combustion engines).

IV.2.6.2 How to Calculate Adiabatic Flame Temperature?

It is found using the First Law of Thermodynamics (Energy Conservation):

$\sum H_{reactants}=\sum H_{products}$

where:

$H_{reactants}$ = total enthalpy of reactants

$H_{products}$ = total enthalpy of products

Since no heat loss occurs:

$$\sum \Delta H_0^f + \sum Cp(Tad - Tref) = 0 \qquad 0.18)$$

where:

ΔH_0^f = Standard heat of formation (kJ/mol)

Cp = Specific heat at constant pressure (kJ/kmol·K)

Tref = Reference temperature (usually 298 K)

Tad= Adiabatic flame temperature

3. ***Example Calculation:***

Adiabatic Flame Temperature of Methane (CH_4) Combustion

Step 1: Write the Balanced Reaction

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

Step 2: Use Standard Enthalpy of Formation

From tables:

Substance	Formula	ΔH_0^f (kJ/mol)
Methane (gas)	CH_4	-74.8
Oxygen (gas)	O_2	0
Carbon Dioxide	CO_2	-393.5
Water (gas)	H_2O	-241.8

Total enthalpy of reactants:

$H_{reactants}$= (−74.8)+2(0)=−74.8 kJ/mol = (-74.8) + 2(0) = -74.8 kJ/mol

Total enthalpy of products:

$H_{products}$= (−393.5)+2(−241.8)=−877.1 kJ/mol = (-393.5) + 2(-241.8) = -877.1 kJ/mol

Step 3: Solve for Tad

Using heat capacity values (Cp):

Gas	Cp (kJ/kmol·K)
CO_2	37.1
H_2O (gas)	34.0

Using:

$$\sum \Delta H_0^f + \sum Cp(T_{ad} - 298) = 0$$
$$-74.8 = (37.1 + 2(34.0))(Tad - 298)$$
$$-74.8 = (105.1)(Tad - 298)$$

$$T_{ad} = -\frac{74.8}{105.1} + 298 = 2220\ K$$

IV.2.6.3 Factors Affecting Adiabatic Flame Temperature

Fuel Type: Hydrocarbons have different flame temperatures (e.g., hydrogen burns hotter than methane).
Air-Fuel Ratio: More oxygen generally increases Tad, but excess air reduces it.
Pressure: Higher pressure increases Tad due to enhanced reaction rates.
Phase of Water: If water remains as vapor, Tad is higher. If it condenses, latent heat loss reduces Tad.

IV.2.6.4 Adiabatic Flame Temperature of Common Fuels

Fuel	Chemical Formula	Tad (K)
Hydrogen	H_2	2800 K
Methane	CH_4	2220 K
Propane	C_3H_8	2260 K
Gasoline	C_8H_{18}	2480 K
Carbon Monoxide	CO	2110 K

IV.3 Chemical kinetics

IV.3.1 Definition of Chemical Kinetics

Chemical kinetics is the branch of chemistry that studies the **rates of chemical reactions** and the **factors that influence these rates**. It examines how fast reactants are converted into products and provides insight into the **mechanisms** of reactions at a molecular level.
Key aspects of chemical kinetics include:

- **Reaction Rate:** The speed at which reactants are consumed and products are formed.
- **Reaction Mechanisms:** The sequence of steps that describe how a reaction occurs at a molecular level.
- **Factors Affecting Reaction Rate:**
    - **Temperature** (higher temperature increases reaction rate).
    - **Concentration** (higher reactant concentration usually increases rate).
    - **Catalysts** (substances that speed up a reaction without being consumed).
    - **Pressure** (affects reactions involving gases).
    - **Surface Area** (more surface area leads to faster reactions).
- **Rate Laws and Reaction Order:** Mathematical expressions that describe how the rate depends on reactant concentrations.
- **Arrhenius Equation:** A formula that relates the reaction rate to temperature and activation energy.

IV.3.2 Importance of Chemical Kinetics in Combustion:

In **internal combustion engines**, chemical kinetics helps predict how fuel burns, optimize engine efficiency, reduce emissions, and design better fuels and catalytic converters.

IV.3.3 Exercise: Chemical Kinetics in an Internal Combustion Engine

Problem Statement:

A **four-stroke gasoline engine** with a **cylinder volume of 2.5 L** operates at **3000 rpm** and uses **octane (C_8H_{18})** as fuel. The combustion process follows the simplified reaction:

$$C_8H_{18} + 12.5O_2 \rightarrow 8CO_2 + 9H_2O$$

The reaction rate follows the **Arrhenius equation**:

$$k = Ae^{-Ea/RT} \qquad 0.19)$$

where:
A=1.2×10^{8} s^{-1}(pre-exponential factor)
Ea=135,000 J/mol (activation energy)
R=8.314 J/mol·K (universal gas constant)
T = 2000 K (combustion temperature)
Given Data:
Engine speed: 3000 rpm
Cylinder volume: 2.5 L
Fuel type: Octane (C_8H_{18})
Stoichiometric air-fuel ratio (AFR): 14.7:1 (mass basis)
Lower heating value (LHV) of octane: 44.4 MJ/kg
Density of octane: 0.72 kg/L
Molar mass of octane: 114 g/mol

Questions:
1. Air-Fuel Ratio and Combustion Reaction
(a) Write the **balanced chemical equation** for the complete combustion of octane.
(b) Calculate the **mass of oxygen required** to completely burn **1 kg of octane**.
(c) Determine the **mass of air needed** per **kg of fuel** using the stoichiometric AFR.
2. Reaction Kinetics and Rate Constant
(a) Calculate the **reaction rate constant kk at 2000 K** using the Arrhenius equation.
(b) If the temperature increases to **2200 K**, determine the new reaction rate constant k'k'.
(c) By what factor does the reaction rate increase when the temperature rises from **2000 K to 2200 K**?
3. Fuel Consumption and Power Output
(a) Calculate the **mass flow rate of fuel (kg/s)** if the engine operates at **full load** with an **efficiency of 30%**.
(b) Determine the **volume of fuel consumed per hour (L/h)**.
(c) Calculate the **total heat energy released per second (kW)** due to combustion.
(d) Determine the **brake power output (kW)** of the engine, assuming **30% efficiency**.

Solution

Step 1: Air-Fuel Ratio and Combustion Reaction
(a) Balanced Chemical Equation

The complete combustion of **octane (C_8H_{18})** with **oxygen (O_2)** follows:
$C_8H_{18} + 12.5O_2 \rightarrow 8CO_2 + 9H_2O$ This equation shows that **one mole** of octane reacts with **12.5 moles** of oxygen to produce **8 moles of CO_2** and **9 moles of H_2O**.

(b) Mass of Oxygen Required per kg of Octane

Molar mass of octane: **114 g/mol**
Molar mass of oxygen: **32 g/mol**
Mass of oxygen required per mole of octane:
12.5×32=400 g of O_2
For **1 kg (1000 g) of octane**:

$$\frac{1000}{114} \times 400 = 3508.77\ g = 3.51\ kg\ of\ O_2$$

(c) Mass of Air Required per kg of Octane

Air is **21% oxygen by mass**, so for every kg of oxygen, the mass of air required is:
3.510.21=16.7 kg of air
Thus, **16.7 kg of air** is required per **1 kg of octane** under stoichiometric conditions.

Step 2: Reaction Kinetics and Rate Constant

The **Arrhenius equation** for reaction rate constant kk is:

$$k = Ae^{-Ea/RT} \qquad 0.20)$$

(a) Calculate k at 2000 K

Given:
A=$1.2\times10^{8}\ s^{-1}$(pre-exponential factor)
Ea=135,000 J/mol (activation energy)
R=8.314 J/mol·K (universal gas constant)
T = 2000 K (combustion temperature)

$$k = 1.2 \times 108\ e^{-135/8.314.2000}$$

$$k = 3.58\ 10^{4} s^{-1}$$

(b) Calculate k' at 2200 K

(c) Factor of Increase in Reaction Rate

$$k' = 1.2 \times 108\ e^{-135/8.314.2200}$$

$$k' = 7.35\ 10^{4} s^{-1}$$

So, increasing the temperature from **2000 K to 2200 K doubles the reaction rate**.

$$k'/k = 7.53\times10^{4}/3.58\times10^{4} = 2.1$$

Step 3: Fuel Consumption and Power Output

(a) Mass Flow Rate of Fuel

The **power output equation** is:
Power=Fuel Energy Input×Efficiency\text{Power}
Given:
Engine efficiency = **30%** = 0.30
LHV of octane = **44.4 MJ/kg**
Target power output = **100 kW** (assumed)
Fuel Energy Input=Power Output/Efficiency=100/0.30=333.3 kW
Fuel Mass Flow Rate=333.3/44.4=7.51 g/s

(b) Volume of Fuel Consumed per Hour

Using **fuel density** ρ=0.72 kg/L:
Fuel Volume Flow Rate=(7.51×3600)/(0.72×103=37.6 L/h
So, the engine **consumes 37.6 L of fuel per hour**.

(c) Total Heat Energy Released per Second

Total Heat Energy=Fuel Flow Rate×LHV =7.51×44.4=333.3 kW

(d) Brake Power Output

Brake Power=Total Heat Energy×Efficiency=333.3×0.30=100 kW
So, the **brake power output is 100 kW**.

Final Summary of Results:

Calculation	Value
Mass of Oxygen per kg of Octane	3.51 kg O_2

Calculation	Value
Mass of Air per kg of Octane	16.7 kg Air
Reaction Rate at 2000 K	$3.58\times10^{4}s^{-1}$
Reaction Rate at 2200 K	$7.53\times10^{4}s^{-1}$
Factor Increase in Rate	2.1×
Fuel Consumption	7.51 g/s
Fuel Volume per Hour	37.6 L/h
Total Heat Energy Released	333.3 kW
Brake Power Output	100 kW

IV.4 Elemental reactions

Elemental reactions play a crucial role in the combustion process inside an internal combustion engine (ICE). They govern fuel oxidation, energy release, flame propagation, and pollutant formation. Understanding these reactions helps in optimizing engine performance, fuel efficiency, and emission control.

IV.4.1 Fuel Combustion and Energy Release

Combustion in an engine occurs through a series of fast elemental reactions that break down hydrocarbon fuels (e.g., gasoline, diesel).
The primary reaction for octane (C8H18) combustion is: C8H18+12.5O2→8CO2+9H2O
However, this is a simplified global reaction; in reality, combustion occurs in multiple elementary steps involving radical species.

IV.4.2 Ignition and Flame Propagation

The autoignition of fuel in an ICE depends on elemental chain reactions.
Key free radicals (O,H,OH) control ignition delay and flame speed.
Example of an important reaction: H+O2→ OH + O
Faster radical formation → quicker ignition and complete combustion → better efficiency.

IV.4.3 Knock Formation and Octane Rating

Knocking occurs due to the uncontrolled autoignition of the air-fuel mixture before the spark plug fires.
Higher octane fuels resist knock because they have slower elementary reaction rates in pre-ignition conditions.
Knock-related reactions involve low-temperature oxidation of fuel molecules leading to explosive autoignition: RH+O2→RO2 (where RH is a hydrocarbon fuel molecule).
Controlling these reactions through additives (like ethanol) improves fuel anti-knock performance.

IV.4.4 Pollutant Formation and Emission Control

Incomplete combustion or high-temperature reactions lead to pollutants such as NOx, CO, and unburned hydrocarbons.
NOx formation follows the Zeldovich mechanism: N2+O→NO+ N
N+O2→ NO + O
CO formation occurs in fuel-rich conditions: CxHy+O2→CO+H2O

Modern catalytic converters use elemental reactions to reduce emissions by converting:
$CO+O2 \rightarrow CO2$
$NO+CO \rightarrow N2+CO2$

IV.4.5 Engine Optimization and Fuel Efficiency

Understanding reaction kinetics helps design:
Turbochargers → Improve air intake for better combustion.
Variable valve timing (VVT) → Optimizes air-fuel mixing for efficient burning.
EGR (Exhaust Gas Recirculation) → Reduces NOx by lowering peak temperatures.
Alternative fuels (Hydrogen, Biofuels) → Modify reaction pathways for cleaner burning.

IV.5 Chain reactions and free radical production

n an internal combustion engine (ICE), the combustion process involves chain reactions driven by free radicals. These reactions are essential for flame propagation, ignition, and pollutant formation.
1. What is a Chain Reaction?
A chain reaction is a self-sustaining sequence of chemical reactions where reactive species (radicals) continue to generate new reactive species, propagating the reaction.
Stages of a Chain Reaction in Combustion:
Initiation → Formation of free radicals
Propagation → Radicals react with fuel and oxygen to form more radicals
Branching → A single radical generates multiple new radicals (increasing reaction rate)
Termination → Radicals recombine, stopping the chain reaction
2. Free Radical Production in Combustion
What are Free Radicals?
Free radicals are highly reactive unstable molecules with unpaired electrons. They are crucial for combustion as they speed up reactions.
Key Free Radicals in Hydrocarbon Combustion:
$H\bullet$ (Hydrogen radical)
$O\bullet$ (Oxygen radical)
$OH\bullet$ (Hydroxyl radical)
$HO_2\bullet$ (Hydroperoxy radical)
These radicals form through high-temperature decomposition of fuel and oxygen molecules.
3. Chain Reaction Mechanism in Hydrocarbon Combustion
(1) Initiation – Radical Formation
The first step breaks fuel molecules to create radicals.
Example (Methane combustion):
$CH4+O2 \rightarrow CH3\bullet+HO2\bullet$
(Heat or spark breaks the C-H bond, producing radicals.)
For hydrogen combustion:
$H2+O2 \rightarrow 2OH\bullet$
(High temperature causes oxygen to split and attack hydrogen.)

(2) Propagation – Radical Chain Reaction
Radicals react with fuel and oxygen, forming new radicals and sustaining combustion.
Examples:
$H\bullet+O2 \rightarrow OH\bullet+O\bullet$
$O\bullet+H2 \rightarrow OH\bullet+H\bullet$
$OH\bullet+CH4 \rightarrow CH3\bullet+H2O$
These reactions release energy and sustain the flame.
(3) Branching – Rapid Increase in Radicals

Branching increases reaction speed, leading to explosive combustion.
Key reaction:
$O\bullet + H_2 \rightarrow OH\bullet + H\bullet$
Each step produces multiple radicals, causing a chain explosion.
(4) Termination – Radical Removal
When radicals recombine, the reaction slows down and stops.
Examples:
$H\bullet + OH\bullet \rightarrow H_2O$
$O\bullet + O\bullet \rightarrow O_2$
$OH\bullet + OH\bullet \rightarrow H_2O_2$
These reactions stabilize the system and prevent uncontrolled explosions.
4. Role in Internal Combustion Engines
Why Are Chain Reactions Important in ICEs?
✓ Fast ignition – Radicals speed up combustion
✓ Complete burning – Ensures fuel is fully oxidized
✓ Knock prevention – Controlled radical formation prevents engine knocking
✓ Emission control – Managing radicals reduces NOx and CO formation
Knocking and Radical Chain Reactions
In knocking, unwanted radicals ($HO_2\bullet$, $CH_3\bullet$) form before the spark plug fires, causing pre-ignition and pressure spikes.
Octane rating measures fuel's ability to resist knock by slowing radical formation.
5. How to Control Radical Reactions in Engines?
-Using High-Octane Fuel – Reduces radical chain branching to prevent knock
-Turbocharging – Ensures complete combustion by increasing oxygen supply
-EGR (Exhaust Gas Recirculation) – Reduces flame temperature, limiting radical formation
-Catalytic Converters – Remove radicals and pollutants (e.g., NO•, CO•)

IV.6 Recombinations

1. What is Recombination?
Recombination is the process by which two radicals combine or react with a third body to form a stable molecule, stopping the propagation of chain reactions.
Types of Recombination:
Radical-Radical Combination → Direct merging of radicals.
Three-Body Recombination → A third molecule absorbs excess energy to stabilize the product.

2. Key Recombination Reactions in Combustion
(1) H and OH Termination
$H\bullet + OH\bullet \rightarrow H_2O$
Forms water vapor, an important combustion product.
Reduces reactive radical concentration, slowing down the flame.
(2) Oxygen Radical Termination
$O\bullet + O\bullet \rightarrow O_2$
Occurs in cooling zones of combustion.
Prevents excessive oxidation reactions.
(3) Hydrogen Radical Recombination
$H\bullet + H\bullet + M \rightarrow H_2 + M$
A third body (M) absorbs excess energy, stabilizing the molecule.
Prevents excessive flame acceleration.
(4) Hydroxyl Radical Recombination

OH•+OH•→H2O2
Forms hydrogen peroxide, which can decompose into oxygen and radicals.
Important in low-temperature ignition and pollutant formation.

3. Importance of Internal Combustion Engines
Flame Quenching and Knock Prevention
Recombination limits the number of active radicals, preventing pre-ignition and knocking.
Flame quenching near engine walls occurs due to rapid recombination, stopping combustion before the fuel fully burns.
Emission Control
Catalytic converters promote recombination to remove harmful gases: NO•+CO→N2+CO2
Converts nitrogen oxides (NOx) into harmless nitrogen.
Reduces toxic emissions from vehicle exhaust.
4. Engineering Applications: How to Control Recombination?
✓ Fuel additives adjust radical recombination to optimize engine performance.
✓ Exhaust Gas Recirculation (EGR) increases recombination to lower NOx emissions.
✓ Turbocharging increases oxygen, preventing excess recombination that could slow combustion.
Conclusion:
Recombination reactions stop combustion chain reactions, controlling flame speed, reducing knocking, and lowering emissions. Properly managing recombination improves fuel efficiency and environmental impact in internal combustion engines.

IV.7 Equilibrium constants

Equilibrium Constants in Combustion Reactions
In combustion chemistry, the equilibrium constant determines the balance between reactants and products at a given temperature. It is crucial for understanding fuel oxidation, flame stability, pollutant formation, and engine efficiency.

1. Definition of the Equilibrium Constant
For a general reaction:

$$aA + bB \rightleftharpoons cC + dD \qquad 0.21)$$

The equilibrium constant (K_{eq}) is given by:

$$K_{eq} = \frac{[C]^c[D]^d}{[A]^a[B]^b} \qquad 0.22)$$

where:
$[C]$, $[D]$, $[A]$, $[B]$ are the molar concentrations (for gases, partial pressures are used).
K_{eq} depends only on temperature (not on pressure or concentration changes).
2. Types of Equilibrium Constants in Combustion
(1) K_c – Concentration-Based Equilibrium Constant

$$K_c = \frac{[products]}{[reactants]}\text{Type equation here.} \qquad 0.23)$$

Used when dealing with concentration (mol/L) in liquid or low-pressure gas reactions.
(2) K_p – Pressure-Based Equilibrium Constant

$$K_p = \frac{P_C^c P_D^d}{P_A^a P_B^b} \qquad 0.24)$$

Used when dealing with gaseous reactions, where P represents partial pressure.
$\Delta n = (c + d) - (a + b)$, so:

$$K_p = K_c (RT)^{\Delta n} \qquad 0.25)$$

where:
$R = 0.0821$ L·atm/(mol·K) (gas constant)
T = temperature in Kelvin
(3) K_f/K_r – Forward and Reverse Rate Constants

$$K_{eq} = \frac{k_f}{k_r} \qquad 0.26)$$

where:
k_f = rate constant for forward reaction
k_r = rate constant for reverse reaction
3. Example: Equilibrium in Combustion Reactions
Example Reaction – Carbon Monoxide Oxidation

$$CO + \frac{1}{2}O_2 \rightleftharpoons CO_2$$

The equilibrium constant:

$$K_p = \frac{P_{CO_2}}{P_{CO} P_{O_2}^{1/2}} \qquad 0.27)$$

High K_p → Complete combustion (efficient engine)
Low K_p → Incomplete combustion (more CO emissions)
Example Reaction – NO Formation in Engines (Zeldovich Mechanism)

$$N_2 + O_2 \rightleftharpoons 2NO$$

$$K_p = \frac{P_{NO}^2}{P_{N_2} P_{O_2}} \qquad 0.28)$$

At high temperatures ($> 2000K$), K_p increases, leading to higher NOx emissions.
Engine tuning aims to reduce K_p to minimize NO formation.
4. Temperature Dependence – Van't Hoff Equation
The equilibrium constant changes with temperature:

$$\frac{d\ln K}{dT} = \frac{\Delta H^\circ}{RT^2} \qquad 0.29)$$

Endothermic reactions ($\Delta H^\circ > 0$) → K increases with temperature.
Exothermic reactions ($\Delta H^\circ < 0$) → K decreases with temperature.
* In combustion, high temperatures shift equilibrium toward complete oxidation, improving efficiency.
5. Engineering Applications

✓ Optimizing Air-Fuel Ratio – Ensures complete combustion by maintaining a high K_p.

✓ Controlling NOx Emissions – Lowering combustion temperature reduces K_p for NO formation.

✓ Catalytic Converters – Promote equilibrium shift to convert CO and NO into harmless gases.

Conclusion

Equilibrium constants help predict combustion efficiency, pollutant formation, and engine performance. By adjusting temperature, pressure, and reactant ratios, engineers optimize fuel efficiency and emissions.

Equilibrium Constants in Combustion Reactions

In combustion chemistry, the equilibrium constant determines the balance between reactants and products at a given temperature. It is crucial for understanding fuel oxidation, flame stability, pollutant formation, and engine efficiency.

1. Definition of the Equilibrium Constant

For a general reaction:

$$aA + bB \rightleftharpoons cC + dD \tag{IV.30}$$

The equilibium constant (K_{eq}) is given by:

$$K_{eq} = \frac{[C]^c[D]^d}{[A]^a[B]^b} \qquad 0.31)$$

where:

$[C]$, $[D]$, $[A]$, $[B]$ are the molar concentrations (for gases, partial pressures are used).

K_{eq} depends only on temperature (not on pressure or concentration changes).

2. Types of Equilibrium Constants in Combustion

(1) K_c – Concentration-Based Equilibrium Constant

$$K_c = \frac{[products]}{[reactants]} \qquad 0.32)$$

Used when dealing with concentration (mol/L) in liquid or low-pressure gas reactions.

(2) K_p – Pressure-Based Equilibrium Constant

$$K_p = \frac{P_C^c P_D^d}{P_A^a P_B^b} \qquad 0.33)$$

Used when dealing with gaseous reactions, where P represents partial pressure.

$\Delta n = (c + d) - (a + b)$, so:

$$K_p = \frac{P_C^c P_D^d}{P_A^a P_B^b} \qquad 0.34)$$

$$K_p = K_c(RT)^{\Delta n}$$

where:

$R = 0.0821$ L·atm/(mol·K) (gas constant)

T = temperature in Kelvin

(3) K_f/K_r – Forward and Reverse Rate Constants

$$K_{eq} = \frac{k_f}{k_r} \qquad 0.35)$$

where:

k_f = rate constant for forward reaction
k_r = rate constant for reverse reaction

3. Example: Equilibrium in Combustion Reactions
Example Reaction – Carbon Monoxide Oxidation

$$CO + \frac{1}{2} O_2 \rightleftharpoons CO_2$$

The equilibrium constant:

$$K_p = \frac{P_{CO_2}}{P_{CO} P_{O_2}^{1/2}} \quad 0.36)$$

High K_p → Complete combustion (efficient engine)
Low K_p → Incomplete combustion (more CO emissions)
Example Reaction – NO Formation in Engines (Zeldovich Mechanism)

$$N_2 + O_2 \rightleftharpoons 2NO$$

$$K_p = \frac{P_{NO}^2}{P_{N_2} P_{O_2}} \quad 0.37)$$

At high temperatures ($> 2000K$), K_p increases, leading to higher NOx emissions.
Engine tuning aims to reduce K_p to minimize NO formation.

4. Temperature Dependence – Van't Hoff Equation
The equilibrium constant changes with temperature:

$$\frac{d\ln K}{dT} = \frac{\Delta H^\circ}{RT^2} \quad 0.38)$$

Endothermic reactions ($\Delta H^\circ > 0$) → K increases with temperature.
Exothermic reactions ($\Delta H^\circ < 0$) → K decreases with temperature.
In combustion, high temperatures shift equilibrium toward complete oxidation, improving efficiency.

5. Engineering Applications
✓ Optimizing Air-Fuel Ratio – Ensures complete combustion by maintaining a high K_p.
✓ Controlling NOx Emissions – Lowering combustion temperature reduces K_p for NO formation.
✓ Catalytic Converters – Promote equilibrium shift to convert CO and NO into harmless gases.

Conclusion
Equilibrium constants help predict combustion efficiency, pollutant formation, and engine performance. By adjusting temperature, pressure, and reactant ratios, engineers optimize fuel *efficiency and emissions.*
Numerical Example: Equilibrium Constant for NO Formation
For the reaction (NOx formation in engines):

$$N_2 + O_2 \rightleftharpoons 2NO$$

The equilibrium constant is calculated as:

$$K_p = e^{-\frac{\Delta G^\circ}{RT}} \quad 0.39)$$

Given:
$\Delta G^\circ = 86500$ J/mol (Gibbs free energy change at 2500 K)

$R = 8.314$ J/mol·K (gas constant)
$T = 2500$ K

$$K_p = e^{\frac{-86500}{(8.314\times 2500)}}$$

IV.8 Reaction rates in combustion

In combustion, reaction rate determines how quickly fuel and oxygen react to form products like CO₂ and H₂O. It plays a crucial role in flame speed, engine efficiency, and pollutant formation.

1. Definition

For a general reaction:

$$aA + bB \rightarrow cC + dD \qquad 0.40)$$

The reaction rate is given by:

$$r = k[A]^m[B]^n \qquad 0.41)$$

where:

- r = **reaction rate** (mol/L·s)
- k = **rate constant** (depends on temperature)
- $[A]$, $[B]$ = **reactant concentrations** (mol/L)
- m, n = **reaction orders** (determined experimentally)

2. Arrhenius Equation for Rate Constant k

The rate constant changes with **temperature** following the **Arrhenius equation**:

$$k = Ae^{-\frac{E_a}{RT}} \qquad 0.42)$$

where:

- A = **pre-exponential factor** (depends on reaction type)
- E_a = **activation energy** (J/mol)
- $R = 8.314$ J/mol·K (gas constant)
- T = temperature (K)

Higher temperatures → Faster reactions (because more molecules have enough energy to react).

3. Example: Reaction Rate in Internal Combustion Engine

Example Reaction: Hydrogen Combustion

$$H_2 + \frac{1}{2}O_2 \rightarrow H_2O$$

Rate equation:

$$r = k[H_2]^1[O_2]^{0.5}$$

If:

- $k = 2.5 \times 10^8$ L/mol·s
- $[H_2] = 0.02$ mol/L
- $[O_2] = 0.01$ mol/L

$$r = (2.5 \times 10^8)(0.02)^1(0.01)^{0.5}$$

$r = 1.58 \times 10^6$ mol/L\cdotps

Interpretation:

- **Higher oxygen concentration → Faster combustion.**
- **Increasing temperature → Higher k, faster reaction.**

4. Engineering Applications in Combustion Engines

✓ **Fuel-Air Ratio Optimization** – Ensures maximum energy release.

✓ **Turbocharging** – Increases oxygen concentration, boosting reaction rate.

✓ **Knock Prevention** – Controlling temperature prevents uncontrolled reaction rates.

IV.9 Simplified combustion models

In combustion analysis, **simplified models** help predict flame behavior, fuel consumption, and emissions with reduced computational complexity. These models are widely used in **internal combustion engines, gas turbines, and industrial burners**.

IV.9.1 Types of Simplified Combustion Models

IV.9.1.1 Global Reaction Model (Single-Step Model)

This model represents combustion using a **single-step reaction**:

$$C_xH_y + O_2 \rightarrow CO_2 + H_2O \qquad 0.43)$$

***Advantages:**

Simple and computationally fast.

Useful for estimating heat release and fuel consumption.

***Limitations:**

Cannot predict **intermediate species** (CO, radicals).

Does not account for **flame propagation and pollutant formation**.

IV.9.1.2 (2) Eddy Dissipation Model (EDM)

This model assumes that **turbulence controls the reaction rate**, rather than chemical kinetics.

***Advantages:**

Works well for **premixed and non-premixed flames**.

Simple and widely used in CFD simulations (e.g., in ANSYS Fluent).

*** Limitations:**

Fails for **low turbulence flames**.

Does not model **detailed chemical kinetics**.

IV.9.1.3 (3) Finite-Rate Chemistry Model

This model considers **chemical kinetics** by solving rate equations:

$$r = k[A]^m[B]^n \qquad 0.44)$$

***Advantages:**

More accurate than EDM.

Predicts **pollutant formation (CO, NOx, unburned HC)**.

*** Limitations:**

Requires **detailed reaction mechanisms**.

Higher computational cost.

IV.9.1.4 Flamelet Model

This model assumes that **flames consist of thin layers (flamelets)** where chemistry and turbulence interact.

* **Advantages:**
Captures **detailed chemistry** while reducing computation.
Used in **diesel and gas turbine** simulations.

* **Limitations:**
Cannot handle **extinction and ignition delay** well.

IV.9.1.5 5. Choosing the Right Model for Internal Combustion Engines

Model	Best for	Limitations
Single-Step Model	Fast energy balance estimates	No intermediate species
Eddy Dissipation Model (EDM)	High-turbulence flames	No detailed chemistry
Finite-Rate Model	Detailed pollutant formation	Computationally expensive
Flamelet Model	Diesel and gas turbines	Limited for ignition/extinction

IV.10 Pressure dependence

Simplified Combustion Models
In combustion analysis, simplified models help predict flame behavior, fuel consumption, and emissions with reduced computational complexity. These models are widely used in internal combustion engines, gas turbines, and industrial burners.

1. Types of Simplified Combustion Models

IV.10.1.1 (1) Global Reaction Model (Single-Step Model**)**

This model represents combustion using a single-step reaction:

$$C_xH_y + O_2 \rightarrow CO_2 + H_2O \qquad 0.45)$$

* Advantages:
Simple and computationally fast.
Useful for estimating heat release and fuel consumption.

*Limitations:
Cannot predict intermediate species (CO, radicals).
Does not account for flame propagation and pollutant formation.

IV.10.1.2 Eddy Dissipation Model (EDM)

This model assumes that turbulence controls the reaction rate, rather than chemical kinetics.

*Advantages:
Works well for premixed and non-premixed flames.
Simple and widely used in CFD simulations (e.g., in ANSYS Fluent).

** Limitations:
Fails for low turbulence flames.
Does not model detailed chemical kinetics.

IV.10.1.3 Finite-Rate Chemistry Model

This model considers chemical kinetics by solving rate equations:

$$r = k[A]^m[B]^n \quad 0.46)$$

*Advantages:
More accurate than EDM.
Predicts pollutant formation (CO, NOx, unburned HC).

* Limitations:
Requires detailed reaction mechanisms.
Higher computational cost.

IV.10.1.4 Flamelet Model

This model assumes that flames consist of thin layers (flamelets) where chemistry and turbulence interact.

* Advantages:
Captures detailed chemistry while reducing computation.
Used in diesel and gas turbine simulations.

IV.10.1.55. Choosing the Right Model for Internal Combustion Engines

Model	Best for	Limitations
Single-Step Model	Fast energy balance estimates	No intermediate species
Eddy Dissipation Model (EDM)	High-turbulence flames	No detailed chemistry
Finite-Rate Model	Detailed pollutant formation	Computationally expensive
Flamelet Model	Diesel and gas turbines	Limited for ignition/extinction

Conclusion

Simplified combustion models balance accuracy and computational cost. Engineers use global models for quick analysis and finite-rate chemistry for detailed emissions studies.

Pressure Dependence in Combustion Reactions

The reaction rate and equilibrium of combustion reactions often depend on pressure. This is crucial in internal combustion engines, gas turbines, and high-altitude combustion systems.

1. How Pressure Affects Reaction Rate

For a general reaction:

$$aA + bB \rightarrow cC + dD \quad 0.47)$$

The reaction rate is given by:

$$r = k[A]^m[B]^n \quad 0.48)$$

where k (rate constant) is affected by pressure, especially in gas-phase reactions.

✓ For simple reactions: Rate depends on reactant concentrations, which increase with pressure.

✓ For complex reactions: Pressure affects molecular collisions and the transition state.

Example: Hydrogen-Oxygen Combustion

$$2H_2 + O_2 \rightarrow 2H_2O$$

At higher pressure, more H_2 and O_2 molecules collide → faster reaction.
In knocking conditions (SI engines), excessive pressure can lead to uncontrolled combustion.

2. Pressure Dependence of the Rate Constant k
The Arrhenius equation:

$$k = Ae^{-\frac{E_a}{RT}} \quad 0.49)$$

does not include pressure, but pressure affects activation energy (E_a) in chain reactions and radical chemistry.
For three-body reactions (where an extra molecule helps stabilize the products):

$$A + B + M \rightarrow AB + M$$

At low pressure, there are fewer stabilizing collisions → reaction slows down.
At high pressure, stabilization is more effective → reaction speeds up.

3. Pressure Dependence in Equilibrium (Le Chatelier's Principle)
For a reaction:

$$N_2 + 3H_2 \rightleftharpoons 2NH_3$$

Increasing pressure shifts equilibrium toward fewer gas molecules → higher NH_3 yield.
In SI engines, high pressure shifts equilibrium to more NOx production → more pollutants.

4. Application in Internal Combustion Engines
*Spark-Ignition (SI) Engines:
✓ High pressure promotes complete combustion but increases knock tendency.
✓ Turbocharging increases pressure → higher power output but needs knock control.
*Compression-Ignition (CI) Engines:
✓ Diesel engines rely on high-pressure autoignition for combustion.
✓ High pressure promotes NOx formation, requiring EGR (Exhaust Gas Recirculation) for control.

IV.11 Partial equilibrium and quasi-stationary states

Partial Equilibrium and Quasi-Stationary States in Combustion
In combustion and chemical kinetics, partial equilibrium and quasi-stationary state (QSS) approximations simplify complex reaction mechanisms by assuming that certain steps reach equilibrium or maintain a steady concentration over time. These methods help model radical formation, pollutant emissions (NOx, CO), and ignition delays in combustion systems.

1. Partial Equilibrium Approximation
This assumes that some reactions in a complex mechanism reach equilibrium much faster than the overall process.
Example: CO Oxidation in Combustion

$$CO + \frac{1}{2}O_2 \rightleftharpoons CO_2$$

This reaction reaches equilibrium quickly, meaning its forward and backward rates balance:

$$k_f[CO][O_2]^{1/2} = k_b[CO_2] \quad 0.50)$$

Using the equilibrium constant $K = \frac{k_f}{k_b}$, we can express one species in terms of others, reducing complexity.

*Application:
Used in turbulent combustion models (flamelet, PDF methods).
Helps predict pollutant emissions (CO, NOx) efficiently.

2. Quasi-Stationary State (QSS) Approximation
This assumes that the concentration of intermediate species (e.g., free radicals like OH, O, HO_2) remains constant over time because their production and consumption rates are nearly equal.
Example: NO Formation in Engines
During high-temperature combustion, NO forms via the Zeldovich mechanism:

$$O + N_2 \rightarrow NO + N$$
$$N + O_2 \rightarrow NO + O$$
$$N + OH \rightarrow NO + H$$

Since $[O]$ and $[OH]$ are radicals, they exist in very low, nearly steady concentrations. The QSS assumption simplifies the system by setting:

$$\frac{d[O]}{dt} \approx 0, \quad \frac{d[N]}{dt} \approx 0$$

which allows us to derive a simplified rate expression for NO formation.
*Application:
Used in pollutant modeling (NOx, soot formation).
Helps predict ignition delay in internal combustion engines.

3. Comparison and Applications in Internal Combustion Engines

Method	Assumption	Application in Combustion
Partial Equilibrium	Fast reactions reach equilibrium	CO oxidation, fuel reforming
Quasi-Steady State (QSS)	Intermediate species remain constant	NOx formation, radical chemistry

* In IC engines:
✓ Partial equilibrium helps model CO and hydrocarbon oxidation.
✓ QSS approximation is used for NOx formation, ignition delay, and flame propagation.

Conclusion
Partial equilibrium simplifies fast equilibrating reactions.
QSS approximation helps track low-concentration intermediates.
Both methods reduce complexity while maintaining accuracy in combustion modeling.

IV.12 Auto-ignition and spontaneous ignition

Auto-Ignition and Spontaneous Ignition in Combustion
Auto-ignition and spontaneous ignition are critical phenomena in internal combustion engines, gas turbines, and detonation processes. They determine ignition delay, knocking tendency, and efficiency in engines.

1. Definition of Auto-Ignition and Spontaneous Ignition

*Auto-Ignition: A fuel-air mixture self-ignites due to high temperature and pressure, without an external spark or flame.

* Spontaneous Ignition: The moment at which a mixture reaches its auto-ignition temperature (AIT) and ignites.

* Example:

In spark-ignition (SI) engines, fuel is intended to ignite by a spark, but auto-ignition can lead to knocking.

In diesel engines, fuel is injected into hot, compressed air, relying on auto-ignition for combustion.

2. Auto-Ignition Temperature (AIT)

Each fuel has a minimum temperature at which it ignites without a spark.

Fuel	Auto-Ignition Temperature (°C)
Gasoline	280–470
Diesel	210–350
Methane (CH_4)	580
Hydrogen (H_2)	500–570

✓ High AIT → More stable fuel (e.g., Methane, Hydrogen).

✓ Low AIT → Easier auto-ignition (e.g., Diesel, Gasoline).

3. Auto-Ignition Delay (Ignition Delay Time, τ)

The time between fuel injection and ignition is called the ignition delay time (τ). It depends on:

✓ Temperature (T) → Higher T, shorter delay.

✓ Pressure (P) → Higher P, shorter delay.

✓ Fuel type → Diesel ignites faster than gasoline.

The Arrhenius equation models ignition delay:

$$\tau = Ae^{\frac{E_a}{RT}} \qquad \text{(IV 51)}$$

where:

A = Pre-exponential factor (fuel-dependent)

E_a = Activation energy (J/mol)

R = Gas constant (8.314 J/mol·K)

T = Temperature (K)

*In diesel engines: Short ignition delay → Better combustion efficiency.

** In gasoline engines: Short ignition delay → More knocking risk.

4. Applications in Internal Combustion Engines

Engine Type	Auto-Ignition Role	Effects
SI Engines (Gasoline)	Unwanted auto-ignition leads to knock	Reduces efficiency, damages engine
CI Engines (Diesel)	Desired auto-ignition for combustion	Controls fuel injection timing
HCCI Engines	Uses controlled auto-ignition	Increases efficiency, reduces NOx

Knock Prevention in Gasoline Engines:

✓ Use high-octane fuel (higher auto-ignition temperature).

✓ Reduce compression ratio (prevents premature ignition).

✓ Optimize ignition timing with sensors.

* Optimizing Auto-Ignition in Diesel Engines:
✓ Control injection timing (shorten ignition delay).
✓ Increase compression ratio (higher temperature).
✓ Use cetane enhancers (reduce ignition delay).

IV.13 Effect of pressure on auto-ignition temperature

Auto-ignition temperature (AIT) is the minimum temperature at which a fuel-air mixture spontaneously ignites without an external spark. The AIT of a fuel decreases as pressure increases, making combustion more likely at lower temperatures under high-pressure conditions.

1. Relationship Between Pressure and Auto-Ignition Temperature
The relationship between AIT and pressure follows an inverse trend:

$$T_{\mathrm{auto}} \propto \frac{1}{P^n} \qquad 0.52)$$

where:
T_{auto} = Auto-ignition temperature
P = Pressure
n = Empirical constant (fuel-dependent, typically between 0.1 and 0.3)
This means that as pressure increases, AIT decreases, making ignition easier.

2. Explanation: Why Does AIT Decrease with Pressure?
* Increased Molecular Collisions:
At higher pressure, molecules are packed closer together, increasing the collision frequency. More collisions mean more energy transfer, which accelerates chemical reactions.
*Higher Radical Formation:
At high pressure, reactive radicals (OH, O, HO_2) form more rapidly, speeding up ignition.
*Reduced Activation Energy:
At higher pressures, the effective activation energy for ignition is lower, requiring less thermal energy for reaction onset.

3. Effect of Pressure on Auto-Ignition in Internal Combustion Engines

Engine Type	Effect of Pressure on Auto-Ignition	Impact on Performance
Spark-Ignition (SI) Engines (Gasoline)	Higher pressure lowers AIT, increasing knock risk	Engine knock reduces efficiency and causes damage
Compression-Ignition (CI) Engines (Diesel)	High compression increases pressure, lowering AIT	Ensures reliable auto-ignition for diesel combustion
HCCI (Homogeneous Charge Compression Ignition)	Relies on controlled AIT reduction at high pressures	Improves efficiency while reducing NOx emissions

Knock Prevention in SI Engines:
✓ Use high-octane fuel (higher AIT).
✓ Reduce compression ratio to limit pressure rise.
✓ Adjust spark timing to avoid premature ignition.
Optimizing Diesel Engine Auto-Ignition:
✓ Increase compression ratio to ensure timely auto-ignition.
✓ Use cetane improvers to reduce ignition delay.

4. Experimental Observations of AIT vs. Pressure

Fuel	AIT at 1 bar (°C)	AIT at 10 bar (°C)	AIT at 50 bar (°C)
Gasoline	280–470	~250	~200
Diesel	210–350	~180	~150
Methane (CH_4)	580	~500	~450

✓ Trend: Higher pressure lowers AIT across all fuels.

IV.14 Controlled ignition

Controlled Ignition in Combustion Systems

Controlled ignition refers to the precise timing and regulation of fuel combustion to optimize engine performance, efficiency, and emissions. It is critical in spark-ignition (SI) engines, compression-ignition (CI) engines, and advanced combustion systems like HCCI (Homogeneous Charge Compression Ignition).

1. Why is Controlled Ignition Important?

✓ Prevents knocking in gasoline engines.

✓ Improves fuel efficiency by ensuring complete combustion.

✓ Reduces emissions (NOx, CO, unburned hydrocarbons).

✓ Enhances engine power output and longevity.

2. Controlled Ignition in Different Engine Types

Engine Type	How Ignition is Controlled	Key Challenges
Spark-Ignition (SI) Engine	Spark plug ignites the air-fuel mixture at the right moment	Avoiding knock, optimizing timing
Compression-Ignition (CI) Engine (Diesel)	Auto-ignition occurs due to high compression temperature	Controlling ignition delay, emissions
HCCI (Homogeneous Charge Compression Ignition)	Compression + precise air-fuel mixing for spontaneous ignition	Difficult to control ignition timing

In SI Engines (Gasoline, LPG, CNG)

◈ Ignition Timing Optimization:

Ignition advance or retardation affects efficiency and power.

Modern engines use Electronic Control Units (ECUs) for real-time ignition control.

◈ Knock Prevention:

Knocking occurs if the fuel-air mixture auto-ignites before the spark plug fires.

High-octane fuels resist knock.

Knock sensors adjust ignition timing dynamically.

In CI Engines (Diesel, Jet Engines)

◈ Fuel Injection Control:

Precise timing of fuel injection determines ignition timing.

Shorter ignition delay = faster combustion, lower soot formation.

◈ EGR (Exhaust Gas Recirculation):

Reduces peak temperatures to limit NOx formation.

3. Advanced Controlled Ignition Strategies

◈ Variable Valve Timing (VVT) & Variable Ignition Timing

Adjusts intake/exhaust valve timing to improve power and efficiency.

Found in modern gasoline and diesel engines.

◈ Homogeneous Charge Compression Ignition (HCCI)

Uses compression + controlled temperature to ignite a premixed charge.

No spark plug needed (like diesel), but burns like gasoline.
Lower NOx emissions, higher efficiency.

◈ Reactivity Controlled Compression Ignition (RCCI)
Uses two fuels with different auto-ignition properties (e.g., diesel + gasoline).
Provides better control over ignition and emissions.

4. Factors Affecting Controlled Ignition

- ✓ Fuel Properties: Octane (gasoline), Cetane (diesel), AIT (auto-ignition temperature).
- ✓ Compression Ratio: Higher ratio = more heat = easier ignition.
- ✓ Engine Load & Speed: Timing must adjust dynamically.
- ✓ Temperature & Pressure: Sensors help regulate combustion timing.

IV.15 Heat flow critical for ignition

Heat flow plays a crucial role in ignition by influencing the temperature rise of a fuel-air mixture until it reaches the auto-ignition temperature (AIT). Proper heat transfer mechanisms determine whether ignition will occur and how efficiently combustion will proceed.

1. Importance of Heat Flow in Ignition
For ignition to occur, the fuel-air mixture must absorb sufficient heat to initiate chemical reactions leading to combustion. Heat flow affects:

- ✓ Ignition delay: Faster heat accumulation → shorter ignition delay.
- ✓ Flame stability: Proper heat balance prevents misfire or flame extinction.
- ✓ Engine knock: Uncontrolled heat transfer can lead to knock in SI engines.

Equation for Energy Balance in Ignition

$$\dot{Q} = mc_p \frac{dT}{dt} \qquad 0.53)$$

where:

$\dot{Q}$ = Heat flow (W)
m = Mass of the mixture (kg)
c_p = Specific heat capacity (J/kg·K)
dT/dt = Rate of temperature rise (K/s)
Ignition occurs when the accumulated heat raises the mixture temperature to AIT.

3. Mechanisms of Heat Transfer Affecting Ignition

Heat Transfer Mode	Role in Ignition
Conduction	Transfers heat from hot surfaces to the fuel-air mixture.
Convection	Heated air transfers energy to the mixture, accelerating ignition.
Radiation	Hot gases and flames emit thermal radiation, further heating the mixture.

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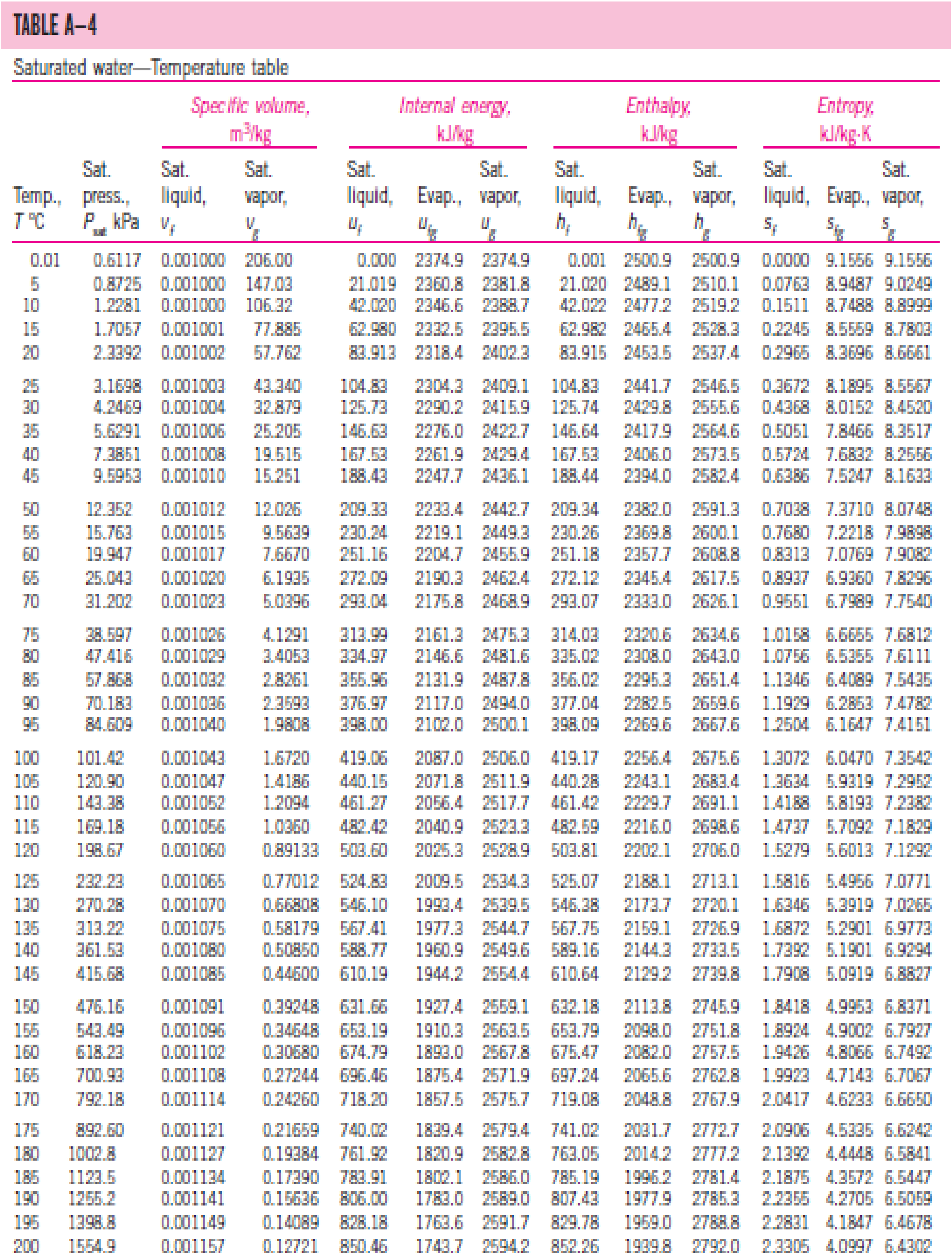

TABLE A–4

Saturated water—Temperature table

		Specific volume, m^3/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg			Entropy, kJ/kg·K		
Temp., T °C	Sat. press., P_{sat} kPa	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
0.01	0.6117	0.001000	206.00	0.000	2374.9	2374.9	0.001	2500.9	2500.9	0.0000	9.1556	9.1556
5	0.8725	0.001000	147.03	21.019	2360.8	2381.8	21.020	2489.1	2510.1	0.0763	8.9487	9.0249
10	1.2281	0.001000	106.32	42.020	2346.6	2388.7	42.022	2477.2	2519.2	0.1511	8.7488	8.8999
15	1.7057	0.001001	77.885	62.980	2332.5	2395.5	62.982	2465.4	2528.3	0.2245	8.5559	8.7803
20	2.3392	0.001002	57.762	83.913	2318.4	2402.3	83.915	2453.5	2537.4	0.2965	8.3696	8.6661
25	3.1698	0.001003	43.340	104.83	2304.3	2409.1	104.83	2441.7	2546.5	0.3672	8.1895	8.5567
30	4.2469	0.001004	32.879	125.73	2290.2	2415.9	125.74	2429.8	2555.6	0.4368	8.0152	8.4520
35	5.6291	0.001006	25.205	146.63	2276.0	2422.7	146.64	2417.9	2564.6	0.5051	7.8466	8.3517
40	7.3851	0.001008	19.515	167.53	2261.9	2429.4	167.53	2406.0	2573.5	0.5724	7.6832	8.2556
45	9.5953	0.001010	15.251	188.43	2247.7	2436.1	188.44	2394.0	2582.4	0.6386	7.5247	8.1633
50	12.352	0.001012	12.026	209.33	2233.4	2442.7	209.34	2382.0	2591.3	0.7038	7.3710	8.0748
55	15.763	0.001015	9.5639	230.24	2219.1	2449.3	230.26	2369.8	2600.1	0.7680	7.2218	7.9898
60	19.947	0.001017	7.6670	251.16	2204.7	2455.9	251.18	2357.7	2608.8	0.8313	7.0769	7.9082
65	25.043	0.001020	6.1935	272.09	2190.3	2462.4	272.12	2345.4	2617.5	0.8937	6.9360	7.8296
70	31.202	0.001023	5.0396	293.04	2175.8	2468.9	293.07	2333.0	2626.1	0.9551	6.7989	7.7540
75	38.597	0.001026	4.1291	313.99	2161.3	2475.3	314.03	2320.6	2634.6	1.0158	6.6655	7.6812
80	47.416	0.001029	3.4053	334.97	2146.6	2481.6	335.02	2308.0	2643.0	1.0756	6.5355	7.6111
85	57.868	0.001032	2.8261	355.96	2131.9	2487.8	356.02	2295.3	2651.4	1.1346	6.4089	7.5435
90	70.183	0.001036	2.3593	376.97	2117.0	2494.0	377.04	2282.5	2659.6	1.1929	6.2853	7.4782
95	84.609	0.001040	1.9808	398.00	2102.0	2500.1	398.09	2269.6	2667.6	1.2504	6.1647	7.4151
100	101.42	0.001043	1.6720	419.06	2087.0	2506.0	419.17	2256.4	2675.6	1.3072	6.0470	7.3542
105	120.90	0.001047	1.4186	440.15	2071.8	2511.9	440.28	2243.1	2683.4	1.3634	5.9319	7.2952
110	143.38	0.001052	1.2094	461.27	2056.4	2517.7	461.42	2229.7	2691.1	1.4188	5.8193	7.2382
115	169.18	0.001056	1.0360	482.42	2040.9	2523.3	482.59	2216.0	2698.6	1.4737	5.7092	7.1829
120	198.67	0.001060	0.89133	503.60	2025.3	2528.9	503.81	2202.1	2706.0	1.5279	5.6013	7.1292
125	232.23	0.001065	0.77012	524.83	2009.5	2534.3	525.07	2188.1	2713.1	1.5816	5.4956	7.0771
130	270.28	0.001070	0.66808	546.10	1993.4	2539.5	546.38	2173.7	2720.1	1.6346	5.3919	7.0265
135	313.22	0.001075	0.58179	567.41	1977.3	2544.7	567.75	2159.1	2726.9	1.6872	5.2901	6.9773
140	361.53	0.001080	0.50850	588.77	1960.9	2549.6	589.16	2144.3	2733.5	1.7392	5.1901	6.9294
145	415.68	0.001085	0.44600	610.19	1944.2	2554.4	610.64	2129.2	2739.8	1.7908	5.0919	6.8827
150	476.16	0.001091	0.39248	631.66	1927.4	2559.1	632.18	2113.8	2745.9	1.8418	4.9953	6.8371
155	543.49	0.001096	0.34648	653.19	1910.3	2563.5	653.79	2098.0	2751.8	1.8924	4.9002	6.7927
160	618.23	0.001102	0.30680	674.79	1893.0	2567.8	675.47	2082.0	2757.5	1.9426	4.8066	6.7492
165	700.93	0.001108	0.27244	696.46	1875.4	2571.9	697.24	2065.6	2762.8	1.9923	4.7143	6.7067
170	792.18	0.001114	0.24260	718.20	1857.5	2575.7	719.08	2048.8	2767.9	2.0417	4.6233	6.6650
175	892.60	0.001121	0.21659	740.02	1839.4	2579.4	741.02	2031.7	2772.7	2.0906	4.5335	6.6242
180	1002.8	0.001127	0.19384	761.92	1820.9	2582.8	763.05	2014.2	2777.2	2.1392	4.4448	6.5841
185	1123.5	0.001134	0.17390	783.91	1802.1	2586.0	785.19	1996.2	2781.4	2.1875	4.3572	6.5447
190	1255.2	0.001141	0.15636	806.00	1783.0	2589.0	807.43	1977.9	2785.3	2.2355	4.2705	6.5059
195	1398.8	0.001149	0.14089	828.18	1763.6	2591.7	829.78	1959.0	2788.8	2.2831	4.1847	6.4678
200	1554.9	0.001157	0.12721	850.46	1743.7	2594.2	852.26	1939.8	2792.0	2.3305	4.0997	6.4302

TABLE A–4

Saturated water—Temperature table (*Concluded*)

		Specific volume, m^3/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg			Entropy, kJ/kg·K		
Temp., T °C	Sat. press., P_{sat} kPa	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
205	1724.3	0.001164	0.11508	872.86	1723.5	2596.4	874.87	1920.0	2794.8	2.3776	4.0154	6.3930
210	1907.7	0.001173	0.10429	895.38	1702.9	2598.3	897.61	1899.7	2797.3	2.4245	3.9318	6.3563
215	2105.9	0.001181	0.094680	918.02	1681.9	2599.9	920.50	1878.8	2799.3	2.4712	3.8489	6.3200
220	2319.6	0.001190	0.086094	940.79	1660.5	2601.3	943.55	1857.4	2801.0	2.5176	3.7664	6.2840
225	2549.7	0.001199	0.078405	963.70	1638.6	2602.3	966.76	1835.4	2802.2	2.5639	3.6844	6.2483
230	2797.1	0.001209	0.071505	986.76	1616.1	2602.9	990.14	1812.8	2802.9	2.6100	3.6028	6.2128
235	3062.6	0.001219	0.065300	1010.0	1593.2	2603.2	1013.7	1789.5	2803.2	2.6560	3.5216	6.1775
240	3347.0	0.001229	0.059707	1033.4	1569.8	2603.1	1037.5	1765.5	2803.0	2.7018	3.4405	6.1424
245	3651.2	0.001240	0.054656	1056.9	1545.7	2602.7	1061.5	1740.8	2802.2	2.7476	3.3596	6.1072
250	3976.2	0.001252	0.050085	1080.7	1521.1	2601.8	1085.7	1715.3	2801.0	2.7933	3.2788	6.0721
255	4322.9	0.001263	0.045941	1104.7	1495.8	2600.5	1110.1	1689.0	2799.1	2.8390	3.1979	6.0369
260	4692.3	0.001276	0.042175	1128.8	1469.9	2598.7	1134.8	1661.8	2796.6	2.8847	3.1169	6.0017
265	5085.3	0.001289	0.038748	1153.3	1443.2	2596.5	1159.8	1633.7	2793.5	2.9304	3.0358	5.9662
270	5503.0	0.001303	0.035622	1177.9	1415.7	2593.7	1185.1	1604.6	2789.7	2.9762	2.9542	5.9305
275	5946.4	0.001317	0.032767	1202.9	1387.4	2590.3	1210.7	1574.5	2785.2	3.0221	2.8723	5.8944
280	6416.6	0.001333	0.030153	1228.2	1358.2	2586.4	1236.7	1543.2	2779.9	3.0681	2.7898	5.8579
285	6914.6	0.001349	0.027756	1253.7	1328.1	2581.8	1263.1	1510.7	2773.7	3.1144	2.7066	5.8210
290	7441.8	0.001366	0.025554	1279.7	1296.9	2576.5	1289.8	1476.9	2766.7	3.1608	2.6225	5.7834
295	7999.0	0.001384	0.023528	1306.0	1264.5	2570.5	1317.1	1441.6	2758.7	3.2076	2.5374	5.7450
300	8587.9	0.001404	0.021659	1332.7	1230.9	2563.6	1344.8	1404.8	2749.6	3.2548	2.4511	5.7059
305	9209.4	0.001425	0.019932	1360.0	1195.9	2555.8	1373.1	1366.3	2739.4	3.3024	2.3633	5.6657
310	9865.0	0.001447	0.018333	1387.7	1159.3	2547.1	1402.0	1325.9	2727.9	3.3506	2.2737	5.6243
315	10,556	0.001472	0.016849	1416.1	1121.1	2537.2	1431.6	1283.4	2715.0	3.3994	2.1821	5.5816
320	11,284	0.001499	0.015470	1445.1	1080.9	2526.0	1462.0	1238.5	2700.6	3.4491	2.0881	5.5372
325	12,051	0.001528	0.014183	1475.0	1038.5	2513.4	1493.4	1191.0	2684.3	3.4998	1.9911	5.4908
330	12,858	0.001560	0.012979	1505.7	993.5	2499.2	1525.8	1140.3	2666.0	3.5516	1.8906	5.4422
335	13,707	0.001597	0.011848	1537.5	945.5	2483.0	1559.4	1086.0	2645.4	3.6050	1.7857	5.3907
340	14,601	0.001638	0.010783	1570.7	893.8	2464.5	1594.6	1027.4	2622.0	3.6602	1.6756	5.3358
345	15,541	0.001685	0.009772	1605.5	837.7	2443.2	1631.7	963.4	2595.1	3.7179	1.5585	5.2765
350	16,529	0.001741	0.008806	1642.4	775.9	2418.3	1671.2	892.7	2563.9	3.7788	1.4326	5.2114
355	17,570	0.001808	0.007872	1682.2	706.4	2388.6	1714.0	812.9	2526.9	3.8442	1.2942	5.1384
360	18,666	0.001895	0.006950	1726.2	625.7	2351.9	1761.5	720.1	2481.6	3.9165	1.1373	5.0537
365	19,822	0.002015	0.006009	1777.2	526.4	2303.6	1817.2	605.5	2422.7	4.0004	0.9489	4.9493
370	21,044	0.002217	0.004953	1844.5	385.6	2230.1	1891.2	443.1	2334.3	4.1119	0.6890	4.8009
373.95	22,064	0.003106	0.003106	2015.7	0	2015.7	2084.3	0	2084.3	4.4070	0	4.4070

TABLE A–5

Saturated water—Pressure table

		Specific volume, m³/kg		*Internal energy, kJ/kg*			*Enthalpy, kJ/kg*			*Entropy, kJ/kg·K*		
Press., P kPa	Sat. temp., T_{sat} °C	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
1.0	6.97	0.001000	129.19	29.302	2355.2	2384.5	29.303	2484.4	2513.7	0.1059	8.8690	8.9749
1.5	13.02	0.001001	87.964	54.686	2338.1	2392.8	54.688	2470.1	2524.7	0.1956	8.6314	8.8270
2.0	17.50	0.001001	66.990	73.431	2325.5	2398.9	73.433	2459.5	2532.9	0.2606	8.4621	8.7227
2.5	21.08	0.001002	54.242	88.422	2315.4	2403.8	88.424	2451.0	2539.4	0.3118	8.3302	8.6421
3.0	24.08	0.001003	45.654	100.98	2306.9	2407.9	100.98	2443.9	2544.8	0.3543	8.2222	8.5765
4.0	28.96	0.001004	34.791	121.39	2293.1	2414.5	121.39	2432.3	2553.7	0.4224	8.0510	8.4734
5.0	32.87	0.001005	28.185	137.75	2282.1	2419.8	137.75	2423.0	2560.7	0.4762	7.9176	8.3938
7.5	40.29	0.001008	19.233	168.74	2261.1	2429.8	168.75	2405.3	2574.0	0.5763	7.6738	8.2501
10	45.81	0.001010	14.670	191.79	2245.4	2437.2	191.81	2392.1	2583.9	0.6492	7.4996	8.1488
15	53.97	0.001014	10.020	225.93	2222.1	2448.0	225.94	2372.3	2598.3	0.7549	7.2522	8.0071
20	60.06	0.001017	7.6481	251.40	2204.6	2456.0	251.42	2357.5	2608.9	0.8320	7.0752	7.9073
25	64.96	0.001020	6.2034	271.93	2190.4	2462.4	271.96	2345.5	2617.5	0.8932	6.9370	7.8302
30	69.09	0.001022	5.2287	289.24	2178.5	2467.7	289.27	2335.3	2624.6	0.9441	6.8234	7.7675
40	75.86	0.001026	3.9933	317.58	2158.8	2476.3	317.62	2318.4	2636.1	1.0261	6.6430	7.6691
50	81.32	0.001030	3.2403	340.49	2142.7	2483.2	340.54	2304.7	2645.2	1.0912	6.5019	7.5931
75	91.76	0.001037	2.2172	384.36	2111.8	2496.1	384.44	2278.0	2662.4	1.2132	6.2426	7.4558
100	99.61	0.001043	1.6941	417.40	2088.2	2505.6	417.51	2257.5	2675.0	1.3028	6.0562	7.3589
101.325	99.97	0.001043	1.6734	418.95	2087.0	2506.0	419.06	2256.5	2675.6	1.3069	6.0476	7.3545
125	105.97	0.001048	1.3750	444.23	2068.8	2513.0	444.36	2240.6	2684.9	1.3741	5.9100	7.2841
150	111.35	0.001053	1.1594	466.97	2052.3	2519.2	467.13	2226.0	2693.1	1.4337	5.7894	7.2231
175	116.04	0.001057	1.0037	486.82	2037.7	2524.5	487.01	2213.1	2700.2	1.4850	5.6865	7.1716
200	120.21	0.001061	0.88578	504.50	2024.6	2529.1	504.71	2201.6	2706.3	1.5302	5.5968	7.1270
225	123.97	0.001064	0.79329	520.47	2012.7	2533.2	520.71	2191.0	2711.7	1.5706	5.5171	7.0877
250	127.41	0.001067	0.71873	535.08	2001.8	2536.8	535.35	2181.2	2716.5	1.6072	5.4453	7.0525
275	130.58	0.001070	0.65732	548.57	1991.6	2540.1	548.86	2172.0	2720.9	1.6408	5.3800	7.0207
300	133.52	0.001073	0.60582	561.11	1982.1	2543.2	561.43	2163.5	2724.9	1.6717	5.3200	6.9917
325	136.27	0.001076	0.56199	572.84	1973.1	2545.9	573.19	2155.4	2728.6	1.7005	5.2645	6.9650
350	138.86	0.001079	0.52422	583.89	1964.6	2548.5	584.26	2147.7	2732.0	1.7274	5.2128	6.9402
375	141.30	0.001081	0.49133	594.32	1956.6	2550.9	594.73	2140.4	2735.1	1.7526	5.1645	6.9171
400	143.61	0.001084	0.46242	604.22	1948.9	2553.1	604.66	2133.4	2738.1	1.7765	5.1191	6.8955
450	147.90	0.001088	0.41392	622.65	1934.5	2557.1	623.14	2120.3	2743.4	1.8205	5.0356	6.8561
500	151.83	0.001093	0.37483	639.54	1921.2	2560.7	640.09	2108.0	2748.1	1.8604	4.9603	6.8207
550	155.46	0.001097	0.34261	655.16	1908.8	2563.9	655.77	2096.6	2752.4	1.8970	4.8916	6.7886
600	158.83	0.001101	0.31560	669.72	1897.1	2566.8	670.38	2085.8	2756.2	1.9308	4.8285	6.7593
650	161.98	0.001104	0.29260	683.37	1886.1	2569.4	684.08	2075.5	2759.6	1.9623	4.7699	6.7322
700	164.95	0.001108	0.27278	696.23	1875.6	2571.8	697.00	2065.8	2762.8	1.9918	4.7153	6.7071
750	167.75	0.001111	0.25552	708.40	1865.6	2574.0	709.24	2056.4	2765.7	2.0195	4.6642	6.6837

TABLE A–5

Saturated water—Pressure table (*Concluded*)

		Specific volume, m^3/kg		*Internal energy,* kJ/kg			*Enthalpy,* kJ/kg			*Entropy,* kJ/kg·K		
Press., P kPa	Sat. temp., T_{sat} °C	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
800	170.41	0.001115	0.24035	719.97	1856.1	2576.0	720.87	2047.5	2768.3	2.0457	4.6160	6.6616
850	172.94	0.001118	0.22690	731.00	1846.9	2577.9	731.95	2038.8	2770.8	2.0705	4.5705	6.6409
900	175.35	0.001121	0.21489	741.55	1838.1	2579.6	742.56	2030.5	2773.0	2.0941	4.5273	6.6213
950	177.66	0.001124	0.20411	751.67	1829.6	2581.3	752.74	2022.4	2775.2	2.1166	4.4862	6.6027
1000	179.88	0.001127	0.19436	761.39	1821.4	2582.8	762.51	2014.6	2777.1	2.1381	4.4470	6.5850
1100	184.06	0.001133	0.17745	779.78	1805.7	2585.5	781.03	1999.6	2780.7	2.1785	4.3735	6.5520
1200	187.96	0.001138	0.16326	796.96	1790.9	2587.8	798.33	1985.4	2783.8	2.2159	4.3058	6.5217
1300	191.60	0.001144	0.15119	813.10	1776.8	2589.9	814.59	1971.9	2786.5	2.2508	4.2428	6.4936
1400	195.04	0.001149	0.14078	828.35	1763.4	2591.8	829.96	1958.9	2788.9	2.2835	4.1840	6.4675
1500	198.29	0.001154	0.13171	842.82	1750.6	2593.4	844.55	1946.4	2791.0	2.3143	4.1287	6.4430
1750	205.72	0.001166	0.11344	876.12	1720.6	2596.7	878.16	1917.1	2795.2	2.3844	4.0033	6.3877
2000	212.38	0.001177	0.099587	906.12	1693.0	2599.1	908.47	1889.8	2798.3	2.4467	3.8923	6.3390
2250	218.41	0.001187	0.088717	933.54	1667.3	2600.9	936.21	1864.3	2800.5	2.5029	3.7926	6.2954
2500	223.95	0.001197	0.079952	958.87	1643.2	2602.1	961.87	1840.1	2801.9	2.5542	3.7016	6.2558
3000	233.85	0.001217	0.066667	1004.6	1598.5	2603.2	1008.3	1794.9	2803.2	2.6454	3.5402	6.1856
3500	242.56	0.001235	0.057061	1045.4	1557.6	2603.0	1049.7	1753.0	2802.7	2.7253	3.3991	6.1244
4000	250.35	0.001252	0.049779	1082.4	1519.3	2601.7	1087.4	1713.5	2800.8	2.7966	3.2731	6.0696
5000	263.94	0.001286	0.039448	1148.1	1448.9	2597.0	1154.5	1639.7	2794.2	2.9207	3.0530	5.9737
6000	275.59	0.001319	0.032449	1205.8	1384.1	2589.9	1213.8	1570.9	2784.6	3.0275	2.8627	5.8902
7000	285.83	0.001352	0.027378	1258.0	1323.0	2581.0	1267.5	1505.2	2772.6	3.1220	2.6927	5.8148
8000	295.01	0.001384	0.023525	1306.0	1264.5	2570.5	1317.1	1441.6	2758.7	3.2077	2.5373	5.7450
9000	303.35	0.001418	0.020489	1350.9	1207.6	2558.5	1363.7	1379.3	2742.9	3.2866	2.3925	5.6791
10,000	311.00	0.001452	0.018028	1393.3	1151.8	2545.2	1407.8	1317.6	2725.5	3.3603	2.2556	5.6159
11,000	318.08	0.001488	0.015988	1433.9	1096.6	2530.4	1450.2	1256.1	2706.3	3.4299	2.1245	5.5544
12,000	324.68	0.001526	0.014264	1473.0	1041.3	2514.3	1491.3	1194.1	2685.4	3.4964	1.9975	5.4939
13,000	330.85	0.001566	0.012781	1511.0	985.5	2496.6	1531.4	1131.3	2662.7	3.5606	1.8730	5.4336
14,000	336.67	0.001610	0.011487	1548.4	928.7	2477.1	1571.0	1067.0	2637.9	3.6232	1.7497	5.3728
15,000	342.16	0.001657	0.010341	1585.5	870.3	2455.7	1610.3	1000.5	2610.8	3.6848	1.6261	5.3108
16,000	347.36	0.001710	0.009312	1622.6	809.4	2432.0	1649.9	931.1	2581.0	3.7461	1.5005	5.2466
17,000	352.29	0.001770	0.008374	1660.2	745.1	2405.4	1690.3	857.4	2547.7	3.8082	1.3709	5.1791
18,000	356.99	0.001840	0.007504	1699.1	675.9	2375.0	1732.2	777.8	2510.0	3.8720	1.2343	5.1064
19,000	361.47	0.001926	0.006677	1740.3	598.9	2339.2	1776.8	689.2	2466.0	3.9396	1.0860	5.0256
20,000	365.75	0.002038	0.005862	1785.8	509.0	2294.8	1826.6	585.5	2412.1	4.0146	0.9164	4.9310
21,000	369.83	0.002207	0.004994	1841.6	391.9	2233.5	1888.0	450.4	2338.4	4.1071	0.7005	4.8076
22,000	373.71	0.002703	0.003644	1951.7	140.8	2092.4	2011.1	161.5	2172.6	4.2942	0.2496	4.5439
22,064	373.95	0.003106	0.003106	2015.7	0	2015.7	2084.3	0	2084.3	4.4070	0	4.4070

TABLE A–6

Superheated water

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg·K
	P = 0.01 MPa (45.81°C)*				P = 0.05 MPa (81.32°C)				P = 0.10 MPa (99.61°C)			
Sat.†	14.670	2437.2	2583.9	8.1488	3.2403	2483.2	2645.2	7.5931	1.6941	2505.6	2675.0	7.3589
50	14.867	2443.3	2592.0	8.1741								
100	17.196	2515.5	2687.5	8.4489	3.4187	2511.5	2682.4	7.6953	1.6959	2506.2	2675.8	7.3611
150	19.513	2587.9	2783.0	8.6893	3.8897	2585.7	2780.2	7.9413	1.9367	2582.9	2776.6	7.6148
200	21.826	2661.4	2879.6	8.9049	4.3562	2660.0	2877.8	8.1592	2.1724	2658.2	2875.5	7.8356
250	24.136	2736.1	2977.5	9.1015	4.8206	2735.1	2976.2	8.3568	2.4062	2733.9	2974.5	8.0346
300	26.446	2812.3	3076.7	9.2827	5.2841	2811.6	3075.8	8.5387	2.6389	2810.7	3074.5	8.2172
400	31.063	2969.3	3280.0	9.6094	6.2094	2968.9	3279.3	8.8659	3.1027	2968.3	3278.6	8.5452
500	35.680	3132.9	3489.7	9.8998	7.1338	3132.6	3489.3	9.1566	3.5655	3132.2	3488.7	8.8362
600	40.296	3303.3	3706.3	10.1631	8.0577	3303.1	3706.0	9.4201	4.0279	3302.8	3705.6	9.0999
700	44.911	3480.8	3929.9	10.4056	8.9813	3480.6	3929.7	9.6626	4.4900	3480.4	3929.4	9.3424
800	49.527	3665.4	4160.6	10.6312	9.9047	3665.2	4160.4	9.8883	4.9519	3665.0	4160.2	9.5682
900	54.143	3856.9	4398.3	10.8429	10.8280	3856.8	4398.2	10.1000	5.4137	3856.7	4398.0	9.7800
1000	58.758	4055.3	4642.8	11.0429	11.7513	4055.2	4642.7	10.3000	5.8755	4055.0	4642.6	9.9800
1100	63.373	4260.0	4893.8	11.2326	12.6745	4259.9	4893.7	10.4897	6.3372	4259.8	4893.6	10.1698
1200	67.989	4470.9	5150.8	11.4132	13.5977	4470.8	5150.7	10.6704	6.7988	4470.7	5150.6	10.3504
1300	72.604	4687.4	5413.4	11.5857	14.5209	4687.3	5413.3	10.8429	7.2605	4687.2	5413.3	10.5229
	P = 0.20 MPa (120.21°C)				P = 0.30 MPa (133.52°C)				P = 0.40 MPa (143.61°C)			
Sat.	0.88578	2529.1	2706.3	7.1270	0.60582	2543.2	2724.9	6.9917	0.46242	2553.1	2738.1	6.8955
150	0.95986	2577.1	2769.1	7.2810	0.63402	2571.0	2761.2	7.0792	0.47088	2564.4	2752.8	6.9306
200	1.08049	2654.6	2870.7	7.5081	0.71643	2651.0	2865.9	7.3132	0.53434	2647.2	2860.9	7.1723
250	1.19890	2731.4	2971.2	7.7100	0.79645	2728.9	2967.9	7.5180	0.59520	2726.4	2964.5	7.3804
300	1.31623	2808.8	3072.1	7.8941	0.87535	2807.0	3069.6	7.7037	0.65489	2805.1	3067.1	7.5677
400	1.54934	2967.2	3277.0	8.2236	1.03155	2966.0	3275.5	8.0347	0.77265	2964.9	3273.9	7.9003
500	1.78142	3131.4	3487.7	8.5153	1.18672	3130.6	3486.6	8.3271	0.88936	3129.8	3485.5	8.1933
600	2.01302	3302.2	3704.8	8.7793	1.34139	3301.6	3704.0	8.5915	1.00558	3301.0	3703.3	8.4580
700	2.24434	3479.9	3928.8	9.0221	1.49580	3479.5	3928.2	8.8345	1.12152	3479.0	3927.6	8.7012
800	2.47550	3664.7	4159.8	9.2479	1.65004	3664.3	4159.3	9.0605	1.23730	3663.9	4158.9	8.9274
900	2.70656	3856.3	4397.7	9.4598	1.80417	3856.0	4397.3	9.2725	1.35298	3855.7	4396.9	9.1394
1000	2.93755	4054.8	4642.3	9.6599	1.95824	4054.5	4642.0	9.4726	1.46859	4054.3	4641.7	9.3396
1100	3.16848	4259.6	4893.3	9.8497	2.11226	4259.4	4893.1	9.6624	1.58414	4259.2	4892.9	9.5295
1200	3.39938	4470.5	5150.4	10.0304	2.26624	4470.3	5150.2	9.8431	1.69966	4470.2	5150.0	9.7102
1300	3.63026	4687.1	5413.1	10.2029	2.42019	4686.9	5413.0	10.0157	1.81516	4686.7	5412.8	9.8828
	P = 0.50 MPa (151.83°C)				P = 0.60 MPa (158.83°C)				P = 0.80 MPa (170.41°C)			
Sat.	0.37483	2560.7	2748.1	6.8207	0.31560	2566.8	2756.2	6.7593	0.24035	2576.0	2768.3	6.6616
200	0.42503	2643.3	2855.8	7.0610	0.35212	2639.4	2850.6	6.9683	0.26088	2631.1	2839.8	6.8177
250	0.47443	2723.8	2961.0	7.2725	0.39390	2721.2	2957.6	7.1833	0.29321	2715.9	2950.4	7.0402
300	0.52261	2803.3	3064.6	7.4614	0.43442	2801.4	3062.0	7.3740	0.32416	2797.5	3056.9	7.2345
350	0.57015	2883.0	3168.1	7.6346	0.47428	2881.6	3166.1	7.5481	0.35442	2878.6	3162.2	7.4107
400	0.61731	2963.7	3272.4	7.7956	0.51374	2962.5	3270.8	7.7097	0.38429	2960.2	3267.7	7.5735
500	0.71095	3129.0	3484.5	8.0893	0.59200	3128.2	3483.4	8.0041	0.44332	3126.6	3481.3	7.8692
600	0.80409	3300.4	3702.5	8.3544	0.66976	3299.8	3701.7	8.2695	0.50186	3298.7	3700.1	8.1354
700	0.89696	3478.6	3927.0	8.5978	0.74725	3478.1	3926.4	8.5132	0.56011	3477.2	3925.3	8.3794
800	0.98966	3663.6	4158.4	8.8240	0.82457	3663.2	4157.9	8.7395	0.61820	3662.5	4157.0	8.6061
900	1.08227	3855.4	4396.6	9.0362	0.90179	3855.1	4396.2	8.9518	0.67619	3854.5	4395.5	8.8185
1000	1.17480	4054.0	4641.4	9.2364	0.97893	4053.8	4641.1	9.1521	0.73411	4053.3	4640.5	9.0189
1100	1.26728	4259.0	4892.6	9.4263	1.05603	4258.8	4892.4	9.3420	0.79197	4258.3	4891.9	9.2090
1200	1.35972	4470.0	5149.8	9.6071	1.13309	4469.8	5149.6	9.5229	0.84980	4469.4	5149.3	9.3898
1300	1.45214	4686.6	5412.6	9.7797	1.21012	4686.4	5412.5	9.6955	0.90761	4686.1	5412.2	9.5625

TABLE A–6

Superheated water (*Concluded*)

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K
	P = 1.00 MPa (179.88°C)				*P* = 1.20 MPa (187.96°C)				*P* = 1.40 MPa (195.04°C)			
Sat.	0.19437	2582.8	2777.1	6.5850	0.16326	2587.8	2783.8	6.5217	0.14078	2591.8	2788.9	6.4675
200	0.20602	2622.3	2828.3	6.6956	0.16934	2612.9	2816.1	6.5909	0.14303	2602.7	2803.0	6.4975
250	0.23275	2710.4	2943.1	6.9265	0.19241	2704.7	2935.6	6.8313	0.16356	2698.9	2927.9	6.7488
300	0.25799	2793.7	3051.6	7.1246	0.21386	2789.7	3046.3	7.0335	0.18233	2785.7	3040.9	6.9553
350	0.28250	2875.7	3158.2	7.3029	0.23455	2872.7	3154.2	7.2139	0.20029	2869.7	3150.1	7.1379
400	0.30661	2957.9	3264.5	7.4670	0.25482	2955.5	3261.3	7.3793	0.21782	2953.1	3258.1	7.3046
500	0.35411	3125.0	3479.1	7.7642	0.29464	3123.4	3477.0	7.6779	0.25216	3121.8	3474.8	7.6047
600	0.40111	3297.5	3698.6	8.0311	0.33395	3296.3	3697.0	7.9456	0.28597	3295.1	3695.5	7.8730
700	0.44783	3476.3	3924.1	8.2755	0.37297	3475.3	3922.9	8.1904	0.31951	3474.4	3921.7	8.1183
800	0.49438	3661.7	4156.1	8.5024	0.41184	3661.0	4155.2	8.4176	0.35288	3660.3	4154.3	8.3458
900	0.54083	3853.9	4394.8	8.7150	0.45059	3853.3	4394.0	8.6303	0.38614	3852.7	4393.3	8.5587
1000	0.58721	4052.7	4640.0	8.9155	0.48928	4052.2	4639.4	8.8310	0.41933	4051.7	4638.8	8.7595
1100	0.63354	4257.9	4891.4	9.1057	0.52792	4257.5	4891.0	9.0212	0.45247	4257.0	4890.5	8.9497
1200	0.67983	4469.0	5148.9	9.2866	0.56652	4468.7	5148.5	9.2022	0.48558	4468.3	5148.1	9.1308
1300	0.72610	4685.8	5411.9	9.4593	0.60509	4685.5	5411.6	9.3750	0.51866	4685.1	5411.3	9.3036
	P = 1.60 MPa (201.37°C)				*P* = 1.80 MPa (207.11°C)				*P* = 2.00 MPa (212.38°C)			
Sat.	0.12374	2594.8	2792.8	6.4200	0.11037	2597.3	2795.9	6.3775	0.09959	2599.1	2798.3	6.3390
225	0.13293	2645.1	2857.8	6.5537	0.11678	2637.0	2847.2	6.4825	0.10381	2628.5	2836.1	6.4160
250	0.14190	2692.9	2919.9	6.6753	0.12502	2686.7	2911.7	6.6088	0.11150	2680.3	2903.3	6.5475
300	0.15866	2781.6	3035.4	6.8864	0.14025	2777.4	3029.9	6.8246	0.12551	2773.2	3024.2	6.7684
350	0.17459	2866.6	3146.0	7.0713	0.15460	2863.6	3141.9	7.0120	0.13860	2860.5	3137.7	6.9583
400	0.19007	2950.8	3254.9	7.2394	0.16849	2948.3	3251.6	7.1814	0.15122	2945.9	3248.4	7.1292
500	0.22029	3120.1	3472.6	7.5410	0.19551	3118.5	3470.4	7.4845	0.17568	3116.9	3468.3	7.4337
600	0.24999	3293.9	3693.9	7.8101	0.22200	3292.7	3692.3	7.7543	0.19962	3291.5	3690.7	7.7043
700	0.27941	3473.5	3920.5	8.0558	0.24822	3472.6	3919.4	8.0005	0.22326	3471.7	3918.2	7.9509
800	0.30865	3659.5	4153.4	8.2834	0.27426	3658.8	4152.4	8.2284	0.24674	3658.0	4151.5	8.1791
900	0.33780	3852.1	4392.6	8.4965	0.30020	3851.5	4391.9	8.4417	0.27012	3850.9	4391.1	8.3925
1000	0.36687	4051.2	4638.2	8.6974	0.32606	4050.7	4637.6	8.6427	0.29342	4050.2	4637.1	8.5936
1100	0.39589	4256.6	4890.0	8.8878	0.35188	4256.2	4889.6	8.8331	0.31667	4255.7	4889.1	8.7842
1200	0.42488	4467.9	5147.7	9.0689	0.37766	4467.6	5147.3	9.0143	0.33989	4467.2	5147.0	8.9654
1300	0.45383	4684.8	5410.9	9.2418	0.40341	4684.5	5410.6	9.1872	0.36308	4684.2	5410.3	9.1384
	P = 2.50 MPa (223.95°C)				*P* = 3.00 MPa (233.85°C)				*P* = 3.50 MPa (242.56°C)			
Sat.	0.07995	2602.1	2801.9	6.2558	0.06667	2603.2	2803.2	6.1856	0.05706	2603.0	2802.7	6.1244
225	0.08026	2604.8	2805.5	6.2629								
250	0.08705	2663.3	2880.9	6.4107	0.07063	2644.7	2856.5	6.2893	0.05876	2624.0	2829.7	6.1764
300	0.09894	2762.2	3009.6	6.6459	0.08118	2750.8	2994.3	6.5412	0.06845	2738.8	2978.4	6.4484
350	0.10979	2852.5	3127.0	6.8424	0.09056	2844.4	3116.1	6.7450	0.07680	2836.0	3104.9	6.6601
400	0.12012	2939.8	3240.1	7.0170	0.09938	2933.6	3231.7	6.9235	0.08456	2927.2	3223.2	6.8428
450	0.13015	3026.2	3351.6	7.1768	0.10789	3021.2	3344.9	7.0856	0.09198	3016.1	3338.1	7.0074
500	0.13999	3112.8	3462.8	7.3254	0.11620	3108.6	3457.2	7.2359	0.09919	3104.5	3451.7	7.1593
600	0.15931	3288.5	3686.8	7.5979	0.13245	3285.5	3682.8	7.5103	0.11325	3282.5	3678.9	7.4357
700	0.17835	3469.3	3915.2	7.8455	0.14841	3467.0	3912.2	7.7590	0.12702	3464.7	3909.3	7.6855
800	0.19722	3656.2	4149.2	8.0744	0.16420	3654.3	4146.9	7.9885	0.14061	3652.5	4144.6	7.9156
900	0.21597	3849.4	4389.3	8.2882	0.17988	3847.9	4387.5	8.2028	0.15410	3846.4	4385.7	8.1304
1000	0.23466	4049.0	4635.6	8.4897	0.19549	4047.7	4634.2	8.4045	0.16751	4046.4	4632.7	8.3324
1100	0.25330	4254.7	4887.9	8.6804	0.21105	4253.6	4886.7	8.5955	0.18087	4252.5	4885.6	8.5236
1200	0.27190	4466.3	5146.0	8.8618	0.22658	4465.3	5145.1	8.7771	0.19420	4464.4	5144.1	8.7053
1300	0.29048	4683.4	5409.5	9.0349	0.24207	4682.6	5408.8	8.9502	0.20750	4681.8	5408.0	8.8786

TABLE A–6

Superheated water (*Continued*)

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K
	P = 4.0 MPa (250.35°C)				*P* = 4.5 MPa (257.44°C)				*P* = 5.0 MPa (263.94°C)			
Sat.	0.04978	2601.7	2800.8	6.0696	0.04406	2599.7	2798.0	6.0198	0.03945	2597.0	2794.2	5.9737
275	0.05461	2668.9	2887.3	6.2312	0.04733	2651.4	2864.4	6.1429	0.04144	2632.3	2839.5	6.0571
300	0.05887	2726.2	2961.7	6.3639	0.05138	2713.0	2944.2	6.2854	0.04535	2699.0	2925.7	6.2111
350	0.06647	2827.4	3093.3	6.5843	0.05842	2818.6	3081.5	6.5153	0.05197	2809.5	3069.3	6.4516
400	0.07343	2920.8	3214.5	6.7714	0.06477	2914.2	3205.7	6.7071	0.05784	2907.5	3196.7	6.6483
450	0.08004	3011.0	3331.2	6.9386	0.07076	3005.8	3324.2	6.8770	0.06332	3000.6	3317.2	6.8210
500	0.08644	3100.3	3446.0	7.0922	0.07652	3096.0	3440.4	7.0323	0.06858	3091.8	3434.7	6.9781
600	0.09886	3279.4	3674.9	7.3706	0.08766	3276.4	3670.9	7.3127	0.07870	3273.3	3666.9	7.2605
700	0.11098	3462.4	3906.3	7.6214	0.09850	3460.0	3903.3	7.5647	0.08852	3457.7	3900.3	7.5136
800	0.12292	3650.6	4142.3	7.8523	0.10916	3648.8	4140.0	7.7962	0.09816	3646.9	4137.7	7.7458
900	0.13476	3844.8	4383.9	8.0675	0.11972	3843.3	4382.1	8.0118	0.10769	3841.8	4380.2	7.9619
1000	0.14653	4045.1	4631.2	8.2698	0.13020	4043.9	4629.8	8.2144	0.11715	4042.6	4628.3	8.1648
1100	0.15824	4251.4	4884.4	8.4612	0.14064	4250.4	4883.2	8.4060	0.12655	4249.3	4882.1	8.3566
1200	0.16992	4463.5	5143.2	8.6430	0.15103	4462.6	5142.2	8.5880	0.13592	4461.6	5141.3	8.5388
1300	0.18157	4680.9	5407.2	8.8164	0.16140	4680.1	5406.5	8.7616	0.14527	4679.3	5405.7	8.7124
	P = 6.0 MPa (275.59°C)				*P* = 7.0 MPa (285.83°C)				*P* = 8.0 MPa (295.01°C)			
Sat.	0.03245	2589.9	2784.6	5.8902	0.027378	2581.0	2772.6	5.8148	0.023525	2570.5	2758.7	5.7450
300	0.03619	2668.4	2885.6	6.0703	0.029492	2633.5	2839.9	5.9337	0.024279	2592.3	2786.5	5.7937
350	0.04225	2790.4	3043.9	6.3357	0.035262	2770.1	3016.9	6.2305	0.029975	2748.3	2988.1	6.1321
400	0.04742	2893.7	3178.3	6.5432	0.039958	2879.5	3159.2	6.4502	0.034344	2864.6	3139.4	6.3658
450	0.05217	2989.9	3302.9	6.7219	0.044187	2979.0	3288.3	6.6353	0.038194	2967.8	3273.3	6.5579
500	0.05667	3083.1	3423.1	6.8826	0.048157	3074.3	3411.4	6.8000	0.041767	3065.4	3399.5	6.7266
550	0.06102	3175.2	3541.3	7.0308	0.051966	3167.9	3531.6	6.9507	0.045172	3160.5	3521.8	6.8800
600	0.06527	3267.2	3658.8	7.1693	0.055665	3261.0	3650.6	7.0910	0.048463	3254.7	3642.4	7.0221
700	0.07355	3453.0	3894.3	7.4247	0.062850	3448.3	3888.3	7.3487	0.054829	3443.6	3882.2	7.2822
800	0.08165	3643.2	4133.1	7.6582	0.069856	3639.5	4128.5	7.5836	0.061011	3635.7	4123.8	7.5185
900	0.08964	3838.8	4376.6	7.8751	0.076750	3835.7	4373.0	7.8014	0.067082	3832.7	4369.3	7.7372
1000	0.09756	4040.1	4625.4	8.0786	0.083571	4037.5	4622.5	8.0055	0.073079	4035.0	4619.6	7.9419
1100	0.10543	4247.1	4879.7	8.2709	0.090341	4245.0	4877.4	8.1982	0.079025	4242.8	4875.0	8.1350
1200	0.11326	4459.8	5139.4	8.4534	0.097075	4457.9	5137.4	8.3810	0.084934	4456.1	5135.5	8.3181
1300	0.12107	4677.7	5404.1	8.6273	0.103781	4676.1	5402.6	8.5551	0.090817	4674.5	5401.0	8.4925
	P = 9.0 MPa (303.35°C)				*P* = 10.0 MPa (311.00°C)				*P* = 12.5 MPa (327.81°C)			
Sat.	0.020489	2558.5	2742.9	5.6791	0.018028	2545.2	2725.5	5.6159	0.013496	2505.6	2674.3	5.4638
325	0.023284	2647.6	2857.1	5.8738	0.019877	2611.6	2810.3	5.7596				
350	0.025816	2725.0	2957.3	6.0380	0.022440	2699.6	2924.0	5.9460	0.016138	2624.9	2826.6	5.7130
400	0.029960	2849.2	3118.8	6.2876	0.026436	2833.1	3097.5	6.2141	0.020030	2789.6	3040.0	6.0433
450	0.033524	2956.3	3258.0	6.4872	0.029782	2944.5	3242.4	6.4219	0.023019	2913.7	3201.5	6.2749
500	0.036793	3056.3	3387.4	6.6603	0.032811	3047.0	3375.1	6.5995	0.025630	3023.2	3343.6	6.4651
550	0.039885	3153.0	3512.0	6.8164	0.035655	3145.4	3502.0	6.7585	0.028033	3126.1	3476.5	6.6317
600	0.042861	3248.4	3634.1	6.9605	0.038378	3242.0	3625.8	6.9045	0.030306	3225.8	3604.6	6.7828
650	0.045755	3343.4	3755.2	7.0954	0.041018	3338.0	3748.1	7.0408	0.032491	3324.1	3730.2	6.9227
700	0.048589	3438.8	3876.1	7.2229	0.043597	3434.0	3870.0	7.1693	0.034612	3422.0	3854.6	7.0540
800	0.054132	3632.0	4119.2	7.4606	0.048629	3628.2	4114.5	7.4085	0.038724	3618.8	4102.8	7.2967
900	0.059562	3829.6	4365.7	7.6802	0.053547	3826.5	4362.0	7.6290	0.042720	3818.9	4352.9	7.5195
1000	0.064919	4032.4	4616.7	7.8855	0.058391	4029.9	4613.8	7.8349	0.046641	4023.5	4606.5	7.7269
1100	0.070224	4240.7	4872.7	8.0791	0.063183	4238.5	4870.3	8.0289	0.050510	4233.1	4864.5	7.9220
1200	0.075492	4454.2	5133.6	8.2625	0.067938	4452.4	5131.7	8.2126	0.054342	4447.7	5127.0	8.1065
1300	0.080733	4672.9	5399.5	8.4371	0.072667	4671.3	5398.0	8.3874	0.058147	4667.3	5394.1	8.2819

TABLE A–6

Superheated water (*Concluded*)

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K
	P = 15.0 MPa (342.16°C)				*P* = 17.5 MPa (354.67°C)				*P* = 20.0 MPa (365.75°C)			
Sat.	0.010341	2455.7	2610.8	5.3108	0.007932	2390.7	2529.5	5.1435	0.005862	2294.8	2412.1	4.9310
350	0.011481	2520.9	2693.1	5.4438								
400	0.015671	2740.6	2975.7	5.8819	0.012463	2684.3	2902.4	5.7211	0.009950	2617.9	2816.9	5.5526
450	0.018477	2880.8	3157.9	6.1434	0.015204	2845.4	3111.4	6.0212	0.012721	2807.3	3061.7	5.9043
500	0.020828	2998.4	3310.8	6.3480	0.017385	2972.4	3276.7	6.2424	0.014793	2945.3	3241.2	6.1446
550	0.022945	3106.2	3450.4	6.5230	0.019305	3085.8	3423.6	6.4266	0.016571	3064.7	3396.2	6.3390
600	0.024921	3209.3	3583.1	6.6796	0.021073	3192.5	3561.3	6.5890	0.018185	3175.3	3539.0	6.5075
650	0.026804	3310.1	3712.1	6.8233	0.022742	3295.8	3693.8	6.7366	0.019695	3281.4	3675.3	6.6593
700	0.028621	3409.8	3839.1	6.9573	0.024342	3397.5	3823.5	6.8735	0.021134	3385.1	3807.8	6.7991
800	0.032121	3609.3	4091.1	7.2037	0.027405	3599.7	4079.3	7.1237	0.023870	3590.1	4067.5	7.0531
900	0.035503	3811.2	4343.7	7.4288	0.030348	3803.5	4334.6	7.3511	0.026484	3795.7	4325.4	7.2829
1000	0.038808	4017.1	4599.2	7.6378	0.033215	4010.7	4592.0	7.5616	0.029020	4004.3	4584.7	7.4950
1100	0.042062	4227.7	4858.6	7.8339	0.036029	4222.3	4852.8	7.7588	0.031504	4216.9	4847.0	7.6933
1200	0.045279	4443.1	5122.3	8.0192	0.038806	4438.5	5117.6	7.9449	0.033952	4433.8	5112.9	7.8802
1300	0.048469	4663.3	5390.3	8.1952	0.041556	4659.2	5386.5	8.1215	0.036371	4655.2	5382.7	8.0574
	P = 25.0 MPa				*P* = 30.0 MPa				*P* = 35.0 MPa			
375	0.001978	1799.9	1849.4	4.0345	0.001792	1738.1	1791.9	3.9313	0.001701	1702.8	1762.4	3.8724
400	0.006005	2428.5	2578.7	5.1400	0.002798	2068.9	2152.8	4.4758	0.002105	1914.9	1988.6	4.2144
425	0.007886	2607.8	2805.0	5.4708	0.005299	2452.9	2611.8	5.1473	0.003434	2253.3	2373.5	4.7751
450	0.009176	2721.2	2950.6	5.6759	0.006737	2618.9	2821.0	5.4422	0.004957	2497.5	2671.0	5.1946
500	0.011143	2887.3	3165.9	5.9643	0.008691	2824.0	3084.8	5.7956	0.006933	2755.3	2997.9	5.6331
550	0.012736	3020.8	3339.2	6.1816	0.010175	2974.5	3279.7	6.0403	0.008348	2925.8	3218.0	5.9093
600	0.014140	3140.0	3493.5	6.3637	0.011445	3103.4	3446.8	6.2373	0.009523	3065.6	3399.0	6.1229
650	0.015430	3251.9	3637.7	6.5243	0.012590	3221.7	3599.4	6.4074	0.010565	3190.9	3560.7	6.3030
700	0.016643	3359.9	3776.0	6.6702	0.013654	3334.3	3743.9	6.5599	0.011523	3308.3	3711.6	6.4623
800	0.018922	3570.7	4043.8	6.9322	0.015628	3551.2	4020.0	6.8301	0.013278	3531.6	3996.3	6.7409
900	0.021075	3780.2	4307.1	7.1668	0.017473	3764.6	4288.8	7.0695	0.014904	3749.0	4270.6	6.9853
1000	0.023150	3991.5	4570.2	7.3821	0.019240	3978.6	4555.8	7.2880	0.016450	3965.8	4541.5	7.2069
1100	0.025172	4206.1	4835.4	7.5825	0.020954	4195.2	4823.9	7.4906	0.017942	4184.4	4812.4	7.4118
1200	0.027157	4424.6	5103.5	7.7710	0.022630	4415.3	5094.2	7.6807	0.019398	4406.1	5085.0	7.6034
1300	0.029115	4647.2	5375.1	7.9494	0.024279	4639.2	5367.6	7.8602	0.020827	4631.2	5360.2	7.7841
	P = 40.0 MPa				*P* = 50.0 MPa				*P* = 60.0 MPa			
375	0.001641	1677.0	1742.6	3.8290	0.001560	1638.6	1716.6	3.7642	0.001503	1609.7	1699.9	3.7149
400	0.001911	1855.0	1931.4	4.1145	0.001731	1787.8	1874.4	4.0029	0.001633	1745.2	1843.2	3.9317
425	0.002538	2097.5	2199.0	4.5044	0.002009	1960.3	2060.7	4.2746	0.001816	1892.9	2001.8	4.1630
450	0.003692	2364.2	2511.8	4.9449	0.002487	2160.3	2284.7	4.5896	0.002086	2055.1	2180.2	4.4140
500	0.005623	2681.6	2906.5	5.4744	0.003890	2528.1	2722.6	5.1762	0.002952	2393.2	2570.3	4.9356
550	0.006985	2875.1	3154.4	5.7857	0.005118	2769.5	3025.4	5.5563	0.003955	2664.6	2901.9	5.3517
600	0.008089	3026.8	3350.4	6.0170	0.006108	2947.1	3252.6	5.8245	0.004833	2866.8	3156.8	5.6527
650	0.009053	3159.5	3521.6	6.2078	0.006957	3095.6	3443.5	6.0373	0.005591	3031.3	3366.8	5.8867
700	0.009930	3282.0	3679.2	6.3740	0.007717	3228.7	3614.6	6.2179	0.006265	3175.4	3551.3	6.0814
800	0.011521	3511.8	3972.6	6.6613	0.009073	3472.2	3925.8	6.5225	0.007456	3432.6	3880.0	6.4033
900	0.012980	3733.3	4252.5	6.9107	0.010296	3702.0	4216.8	6.7819	0.008519	3670.9	4182.1	6.6725
1000	0.014360	3952.9	4527.3	7.1355	0.011441	3927.4	4499.4	7.0131	0.009504	3902.0	4472.2	6.9099
1100	0.015686	4173.7	4801.1	7.3425	0.012534	4152.2	4778.9	7.2244	0.010439	4130.9	4757.3	7.1255
1200	0.016976	4396.9	5075.9	7.5357	0.013590	4378.6	5058.1	7.4207	0.011339	4360.5	5040.8	7.3248
1300	0.018239	4623.3	5352.8	7.7175	0.014620	4607.5	5338.5	7.6048	0.012213	4591.8	5324.5	7.5111

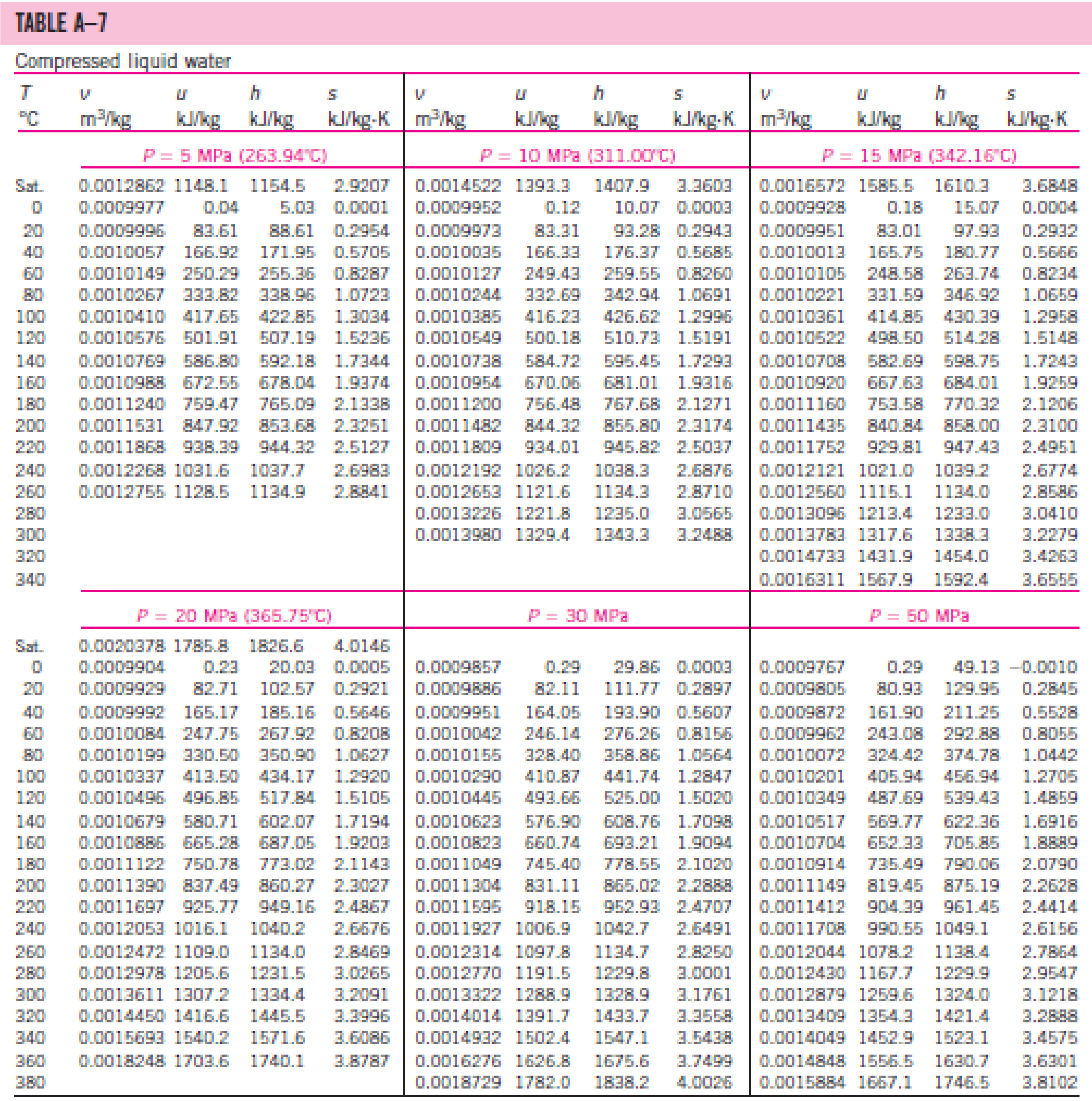

TABLE A–7

Compressed liquid water

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kg·K
	P = 5 MPa (263.94°C)				P = 10 MPa (311.00°C)				P = 15 MPa (342.16°C)			
Sat.	0.0012862	1148.1	1154.5	2.9207	0.0014522	1393.3	1407.9	3.3603	0.0016572	1585.5	1610.3	3.6848
0	0.0009977	0.04	5.03	0.0001	0.0009952	0.12	10.07	0.0003	0.0009928	0.18	15.07	0.0004
20	0.0009996	83.61	88.61	0.2954	0.0009973	83.31	93.28	0.2943	0.0009951	83.01	97.93	0.2932
40	0.0010057	166.92	171.95	0.5705	0.0010035	166.33	176.37	0.5685	0.0010013	165.75	180.77	0.5666
60	0.0010149	250.29	255.36	0.8287	0.0010127	249.43	259.55	0.8260	0.0010105	248.58	263.74	0.8234
80	0.0010267	333.82	338.96	1.0723	0.0010244	332.69	342.94	1.0691	0.0010221	331.59	346.92	1.0659
100	0.0010410	417.65	422.85	1.3034	0.0010385	416.23	426.62	1.2996	0.0010361	414.85	430.39	1.2958
120	0.0010576	501.91	507.19	1.5236	0.0010549	500.18	510.73	1.5191	0.0010522	498.50	514.28	1.5148
140	0.0010769	586.80	592.18	1.7344	0.0010738	584.72	595.45	1.7293	0.0010708	582.69	598.75	1.7243
160	0.0010988	672.55	678.04	1.9374	0.0010954	670.06	681.01	1.9316	0.0010920	667.63	684.01	1.9259
180	0.0011240	759.47	765.09	2.1338	0.0011200	756.48	767.68	2.1271	0.0011160	753.58	770.32	2.1206
200	0.0011531	847.92	853.68	2.3251	0.0011482	844.32	855.80	2.3174	0.0011435	840.84	858.00	2.3100
220	0.0011868	938.39	944.32	2.5127	0.0011809	934.01	945.82	2.5037	0.0011752	929.81	947.43	2.4951
240	0.0012268	1031.6	1037.7	2.6983	0.0012192	1026.2	1038.3	2.6876	0.0012121	1021.0	1039.2	2.6774
260	0.0012755	1128.5	1134.9	2.8841	0.0012653	1121.6	1134.3	2.8710	0.0012560	1115.1	1134.0	2.8586
280					0.0013226	1221.8	1235.0	3.0565	0.0013096	1213.4	1233.0	3.0410
300					0.0013980	1329.4	1343.3	3.2488	0.0013783	1317.6	1338.3	3.2279
320									0.0014733	1431.9	1454.0	3.4263
340									0.0016311	1567.9	1592.4	3.6555
	P = 20 MPa (365.75°C)				P = 30 MPa				P = 50 MPa			
Sat.	0.0020378	1785.8	1826.6	4.0146								
0	0.0009904	0.23	20.03	0.0005	0.0009857	0.29	29.86	0.0003	0.0009767	0.29	49.13	−0.0010
20	0.0009929	82.71	102.57	0.2921	0.0009886	82.11	111.77	0.2897	0.0009805	80.93	129.95	0.2845
40	0.0009992	165.17	185.16	0.5646	0.0009951	164.05	193.90	0.5607	0.0009872	161.90	211.25	0.5528
60	0.0010084	247.75	267.92	0.8208	0.0010042	246.14	276.26	0.8156	0.0009962	243.08	292.88	0.8055
80	0.0010199	330.50	350.90	1.0627	0.0010155	328.40	358.86	1.0564	0.0010072	324.42	374.78	1.0442
100	0.0010337	413.50	434.17	1.2920	0.0010290	410.87	441.74	1.2847	0.0010201	405.94	456.94	1.2705
120	0.0010496	496.85	517.84	1.5105	0.0010445	493.66	525.00	1.5020	0.0010349	487.69	539.43	1.4859
140	0.0010679	580.71	602.07	1.7194	0.0010623	576.90	608.76	1.7098	0.0010517	569.77	622.36	1.6916
160	0.0010886	665.28	687.05	1.9203	0.0010823	660.74	693.21	1.9094	0.0010704	652.33	705.85	1.8889
180	0.0011122	750.78	773.02	2.1143	0.0011049	745.40	778.55	2.1020	0.0010914	735.49	790.06	2.0790
200	0.0011390	837.49	860.27	2.3027	0.0011304	831.11	865.02	2.2888	0.0011149	819.45	875.19	2.2628
220	0.0011697	925.77	949.16	2.4867	0.0011595	918.15	952.93	2.4707	0.0011412	904.39	961.45	2.4414
240	0.0012053	1016.1	1040.2	2.6676	0.0011927	1006.9	1042.7	2.6491	0.0011708	990.55	1049.1	2.6156
260	0.0012472	1109.0	1134.0	2.8469	0.0012314	1097.8	1134.7	2.8250	0.0012044	1078.2	1138.4	2.7864
280	0.0012978	1205.6	1231.5	3.0265	0.0012770	1191.5	1229.8	3.0001	0.0012430	1167.7	1229.9	2.9547
300	0.0013611	1307.2	1334.4	3.2091	0.0013322	1288.9	1328.9	3.1761	0.0012879	1259.6	1324.0	3.1218
320	0.0014450	1416.6	1445.5	3.3996	0.0014014	1391.7	1433.7	3.3558	0.0013409	1354.3	1421.4	3.2888
340	0.0015693	1540.2	1571.6	3.6086	0.0014932	1502.4	1547.1	3.5438	0.0014049	1452.9	1523.1	3.4575
360	0.0018248	1703.6	1740.1	3.8787	0.0016276	1626.8	1675.6	3.7499	0.0014848	1556.5	1630.7	3.6301
380					0.0018729	1782.0	1838.2	4.0026	0.0015884	1667.1	1746.5	3.8102