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كلية العلوم و التكنولوجيا
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Educational Handout

Title

Energy Conversion

Handout intended for third-year Energy Engineering bachelor's students.

مطبوعة مخصصة للسنة الثالثة ليسانس طاقوية

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Foreword

This handout is intended for third-year students in the Energy Engineering bachelor's degree program. It covers the curriculum of the Energy Conversion module, aimed at applying the thermodynamics concepts acquired in previous years to various energy-producing or energy-consuming machines. The educational content is in line with the training framework approved by the CRUO/MESRS. It is structured according to the course program presented in the LMD student curriculum, covering the following chapters:

The first chapter deals with single-phase power cycles, including definitions and different types of single-phase power cycles, where the working fluid returns to its initial state after undergoing a set of transformations. Knowledge of these types of cycles is essential with a sufficient theoretical degree in order to understand and master all the physical phenomena governing the study of energy production or consumption machines.

The second chapter provides a thermal study of two-phase power cycles, where the working fluid exits the device in a different state, meaning that the composition and pressure or temperature conditions differ from those it had when entering. This chapter includes a review of phase change, different two-phase cycles such as the Rankine cycle, Hirn cycle, reheat cycle, cycles with one or multiple steam extractions, and mixed cycles (gas-vapor). It also introduces some concepts related to steam power plants and nuclear power plants.

The third chapter discusses exergy and exergetic analysis of thermodynamic systems, providing examples of their application domains in gas and steam power plants. It relies on demonstrations to calculate the thermal efficiency of each machine.

The fourth chapter defines the thermodynamics of combustion, mixture properties, stoichiometric combustion, heat of formation and calorific values, adiabatic flame temperature, chemical kinetics, simplified combustion models, concepts of autoignition, spontaneous ignition, controlled ignition, as well as the critical heat flux for ignition.

Contents

Chapter 1:
Single-phase power cycles

1.1 Definitions of Thermal Engines

A thermal engine is a mechanical device that converts thermal energy into mechanical energy during a specific cycle. In this process, it absorbs a quantity of heat Q_{in} from a hot source at temperature T_H, typically through a working fluid, while simultaneously releasing a quantity of heat Q_L to a cold source at temperature T_C, as illustrated in Figure 1.1. A fraction of the absorbed heat Q_{in} is converted into mechanical work W, which is then supplied by the engine to the external environment.

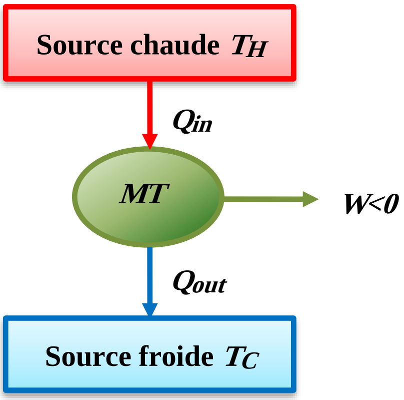


Figure 1.1 Operating diagram of a heat engine

1.2 Thermal Efficiency of an Engine Cycle

The thermal efficiency of an engine cycle, is defined as the ratio of the energy produced in a useful or exploitable form to the energy invested as input. In other words, it is the ratio between the desired output (W) and the input required to obtain it (Q_{in}).

$$\eta_{th}=\frac{\text{gain}}{\text{expenditure}}=\frac{\text{useful energy}}{\text{costly energy}}=\left|\frac{W_{Cycle}}{Q_{in}}\right|$$

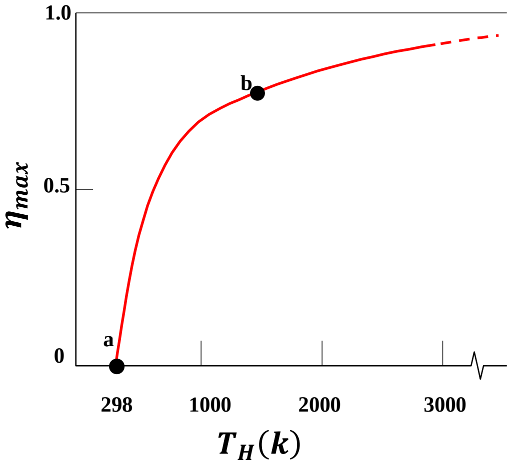


Figure 1.2 Carnot efficiency versus T_H, for T_C =298 K.

Figure 1.2 illustrates that the thermal efficiency of a system increases as T_H (the higher temperature reservoir) increases. Examining the segment a-b of the curve, characterized by relatively low T_H and T, it can be observed that efficiency increases rapidly with higher T_H values. This indicates that even a small increase in T_H within this range can significantly impact efficiency. It is important to note that these conclusions, derived from Figure 1.2, strictly apply to systems undergoing reversible cycles. However, they provide a qualitatively accurate representation for actual power cycles.

Empirical observations demonstrate that the thermal efficiencies of actual cycles tend to improve when the average temperature at which energy is added through heat transfer increases, and/or the average temperature at which energy is discharged through heat transfer decreases.

Comment. Conventional power-producing cycles have thermal efficiencies ranging up to about 40%. This value may seem low, but the comparison should be made with an appropri- ate limiting value and not 100%.

1.3 The carnot cycle.

The Carnot cycle, named after the French engineer Carnot, refers to the ideal cycle comprised of four reversible processes, which include two isothermal processes (at temperatures equal to those of the sources) and two adiabatic processes (isentropic). The depiction of this cycle in the Clapeyron diagram (P, v) or the entropy diagram (T, s) is illustrated in Figure 1.3.

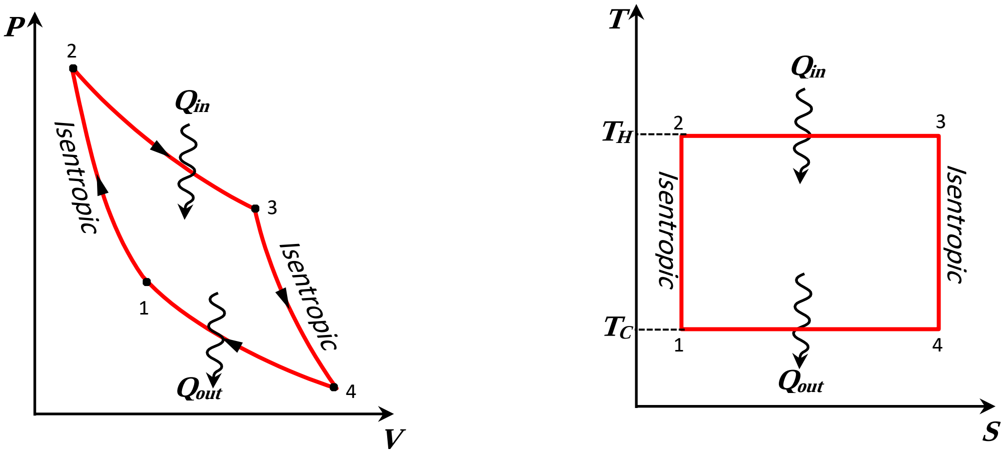


Figure 1.3 P-v and T-s diagrams of a Carnot cycle.

The four processes of the cycle are :

Process 1–2: The gas is compressed isentropically to state 2, where the temperature is $T_{H,}$ $(Q_{12} = 0)$.

$W_{is}^{C} = W_{12} = \Delta u_{12} = C_v(T_H - T_C)$

Process 2–3: The assembly is placed in contact with the reservoir at T_H. The gas expands isothermally while receiving energy Q_{in} from the hot reservoir by heat transfer.

$W_{23} = -Q_{in} = -T_H(S_3 - S_2) \quad avec \quad S_3 = S_4 \; et \; S_1 = S_2$

Process 3–4: The assembly is again placed on the insulating stand and the gas is allowed to continue to expand isentropically until the temperature drops to $T_{C,}$

$(Q_{34} = 0)$.

$$W_{is}^{E} = W_{34} = \Delta u_{34} = C_v(T_C - T_H)$$

Process 4–1: The assembly is placed in contact with the reservoir at T_C. The gas is compressed isothermally to its initial state while it discharges energy Q_{out} to the cold reservoir by heat transfer.

$$W_{41} = -Q_{out} = -T_C(S_1 - S_4).$$

Cycle assessment.

Useful energy.

$$W_{cycle} = W_{12} + W_{23} + W_{34} + W_{41} = -(Q_{in} + Q_{out})$$

$$= (S_1 - S_4)(T_H - T_C)$$

Costly energy.

$$Q_{in} = T_H(S_3 - S_2) = T_H(S_1 - S_4)$$

Lost energy.

$$Q_{out} = T_C(S_1 - S_4)$$

Thermal efficiency of the cycle.

$$\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}} = 1 - \frac{T_C}{T_H}$$

Carnot's First Theorem:

The thermal efficiency of an engine operating on the Carnot cycle is independent of the specific properties of the working fluid and is solely determined by the temperatures of the hot and cold sources.

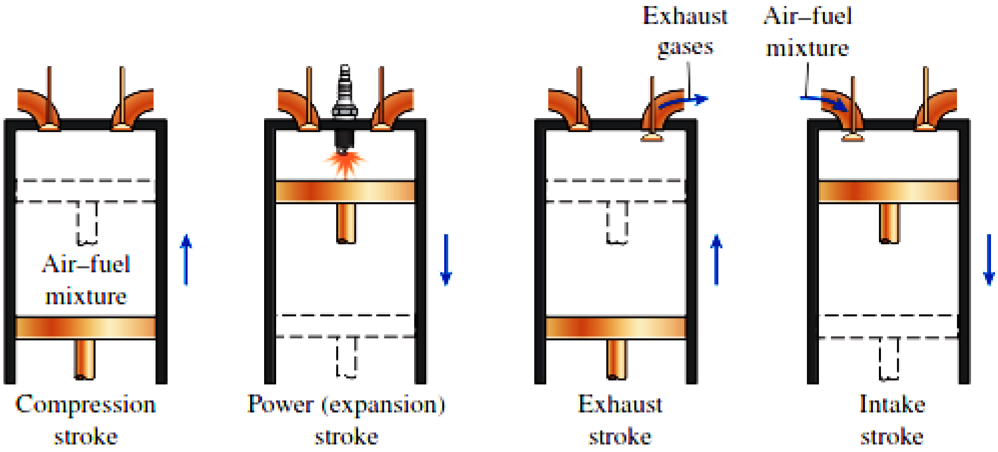
Carnot's Second Theorem:

The efficiency of any real engine is always lower than the Carnot efficiency, which represents the theoretical maximum limit that cannot be exceeded or achieved, regardless of the level of technical perfection of the machine.

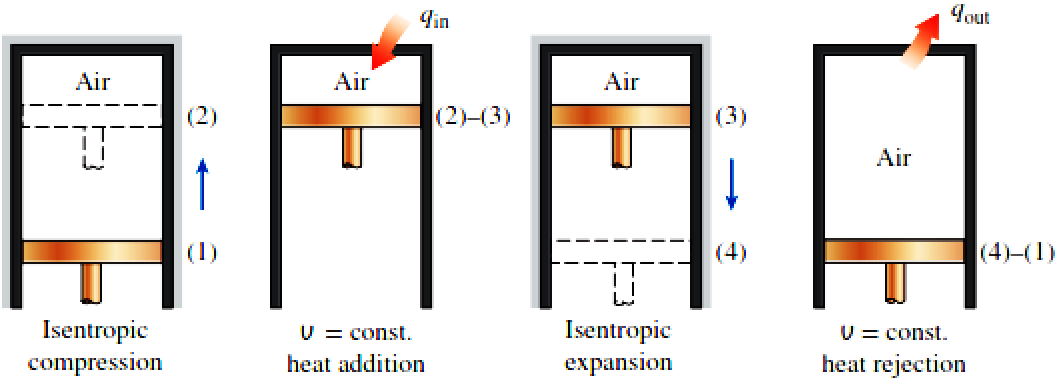
Note : An engine operating on the Carnot cycle has a maximum efficiency but has the disadvantage of being very complicated (and therefore very expensive) to implement, due to technical reasons rather than theoretical ones (difficulties in maintaining isotherms, bulky engine, etc.). For this reason, it is preferred to manufacture engines with lower efficiency but simpler (and cheaper) to implement.

1.4 Air-Standard Otto Cycle.

The Otto cycle represents the ideal thermodynamic cycle for spark-ignition reciprocating engines. It is named after Nikolaus A. Otto, who successfully built a four-stroke engine in Germany in 1876, based on the cycle proposed by the Frenchman Beau de Rochas in 1862. In typical spark-ignition engines, the piston completes four strokes (equivalent to two mechanical cycles) within the cylinder, while the crankshaft completes two revolutions per thermodynamic cycle. These engines are known as four-stroke internal combustion engines. Figure 1.4 provides a schematic representation of each stroke. The Otto cycle is shown on the p-v and T-s Figure 1.5.



(*a*) Actual four-stroke spark-ignition engine



(*b*) Ideal Otto cycle

Figure 1.4 Schematic representation of four-stroke internal combustion engines.

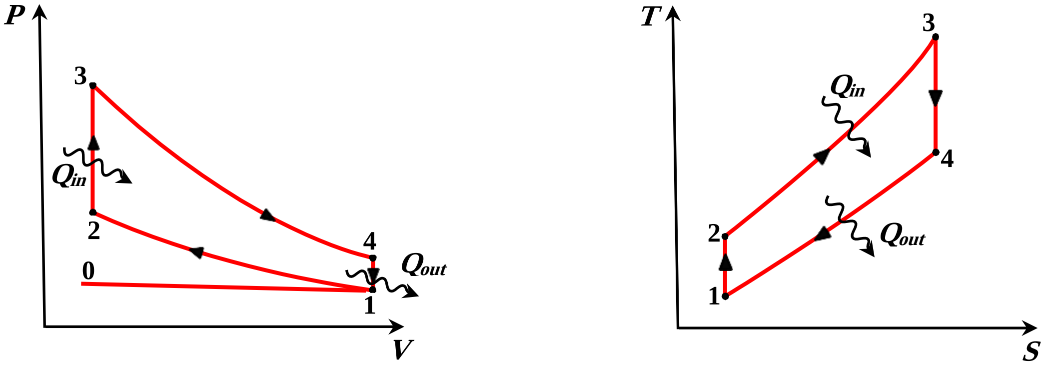


Figure 1.5 P–v and T–s diagrams of the air-standard Otto cycle.

The cycle consists of four internally reversible processes in series:

0-1 Intake: Ingestion of the air-fuel mixture *(this is not a thermodynamic Process)*.

Process 1-2: Isentropic compression of the air as the piston moves from bottom dead center to top dead center, $(Q_{12} = 0)$.

$$W_{is}^{C} = W_{12} = \Delta u_{12} = C_v(T_2 - T_1) > 0$$

$$T_2 = (r^{\gamma-1})T_1$$

Process 2-3: Constant-volume heat transfer to the air from an external source while the piston is at top dead center. This process is intended to represent the ignition of the fuel–air mixture and the subsequent rapid burning.

$$Q_{in} = \Delta u_{23} = C_v(T_3 - T_2) > 0$$

Process 3-4: Isentropic expansion (power stroke), $(Q_{34} = 0)$.

$$W_{is}^{E} = W_{34} = \Delta u_{34} = C_v(T_4 - T_3) < 0$$

$$T_3 = (r^{\gamma-1})T_4$$

Process 4-1: Completes the cycle by a constant-volume process in which heat is rejected from the air while the piston is at bottom dead center.

$$Q_{out} = \Delta u_{41} = C_v(T_1 - T_4) < 0.$$

1-0 Exhaust: evacuation of burnt gases *(this is not a thermodynamic Process).*

γ **:** the adiabatic constant of the working fluid

r : the compression ratio ($r = \frac{V_1}{V_2} = \frac{V_4}{V_3} > 1$), it is a construction parameter of the engine cylinder

Cycle assessment.

Useful energy. $W_{cycle} = W_{12} + W_{34} = -(Q_{in} + Q_{out})$

Costly energy. $Q_{in} = \Delta u_{12} = C_v(T_3 - T_2)$

Lost energy. $Q_{out} = \Delta u_{41} = C_v(T_1 - T_4)$

Thermal efficiency of the cycle.

$$\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}} = 1 - r^{1-\gamma}$$

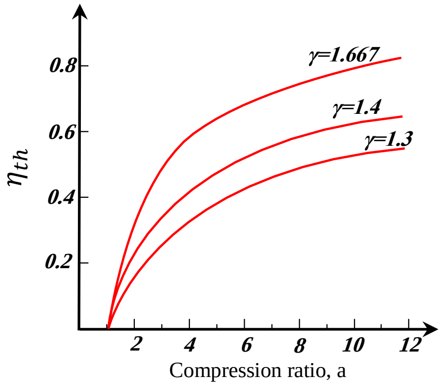


Figure 1.6 The thermal efficiency of the Otto cycle increases with the specific heat ratio γ of the working fluid.

1.5 Diesel cycle the ideal cycle for compression-ignition engines.

The Diesel cycle is widely regarded as the ideal thermodynamic cycle for compression-ignition (CI) reciprocating engines. Rudolph Diesel introduced the concept of the CI engine in the 1890s, which primarily differs from other engines in terms of the combustion initiation method. Spark-ignition engines, also known as gasoline engines, operate by igniting an air-fuel mixture with a spark plug.

However, in compression-ignition engines, commonly referred to as diesel engines, the air is compressed to a temperature exceeding the autoignition temperature of the fuel. Combustion is initiated upon contact as the fuel is injected into this highly compressed and hot air figure 1.7.

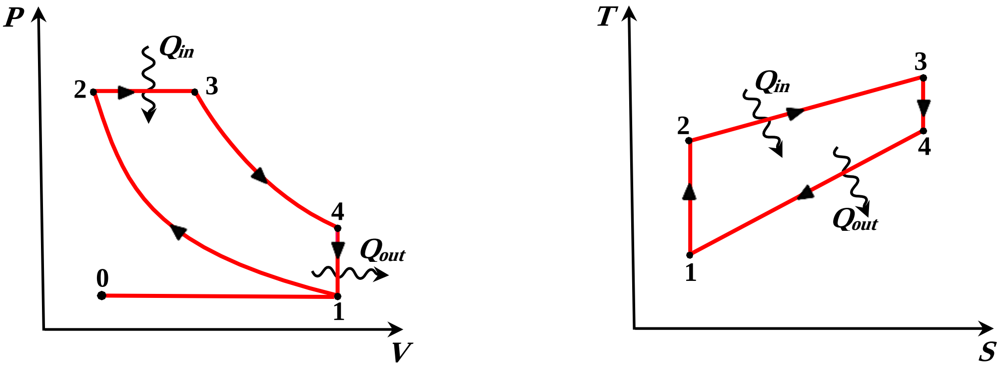


Figure 1.7 P–v and T–s diagrams of the air-standard Diesel cycle.

The cycle consists of four internally reversible processes in series:

0-1 Intake: Ingestion of the fuel *(this is not a thermodynamic Process).*

Process 1-2: Isentropic compression of the air as the piston moves from bottom dead center to top dead center, $(Q_{12} = 0)$.

$$W_{is}^{C} = W_{12} = \Delta u_{12} = C_v(T_2 - T_1) > 0$$

$$T_2 = (r^{\gamma-1})T_1$$

Process 2-3: Constant-pressure heat transfer to the air from an external source while the piston is at top dead center , diesel injection.

$$Q_{in} = \Delta h_{23} = C_p(T_3 - T_2) > 0$$

$$W_{23} = \Delta u_{23} = C_v(T_3 - T_2) < 0$$

Process 3-4: Isentropic expansion (power stroke), $(Q_{34} = 0)$.

$$W_{is}^{E} = W_{34} = \Delta u_{34} = C_v(T_4 - T_3) < 0$$

$$T_3 = \left(\left(\frac{r_c}{r}\right)^{1-\gamma}\right)T_4$$

Process 4-1: Completes the cycle by a constant-volume process in which heat is rejected from the air while the piston is at bottom dead center.

$$Q_{out} = \Delta u_{41} = C_v(T_1 - T_4) < 0.$$

1-0 Exhaust: evacuation of burnt gases *(this is not a thermodynamic Process).*

r : the compression ratio ($r = \frac{V_1}{V_2} > 1$), it is a construction parameter of the engine cylinder

r_c **:** the cutoff ratio ($r_c = \frac{V_3}{V_2} > 1$), it is a construction parameter of the engine cylinder

Cycle assessment.

Useful energy.	$W_{cycle} = W_{12} + W_{23} + W_{34} = -(Q_{in} + Q_{out})$
Costly energy.	$Q_{in} = \Delta u_{12} = C_p(T_3 - T_2)$
Lost energy.	$Q_{out} = \Delta u_{41} = C_v(T_1 - T_4)$
Thermal efficiency of the cycle.	$\eta_{th} = \left\lvert\frac{W_{cycle}}{Q_{in}}\right\rvert = 1 + \frac{Q_{out}}{Q_{in}} = 1 - \frac{1}{\gamma}(r^{1-\gamma})\frac{{r_c}^{\gamma} - 1}{r_c - 1}$

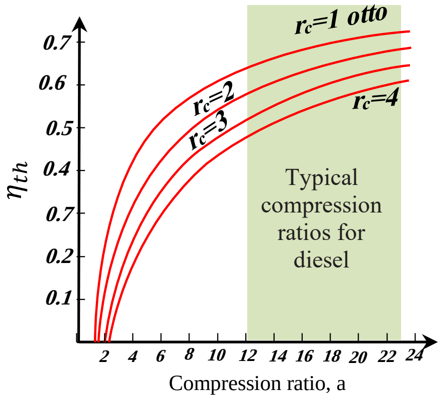


Figure 1.8 Thermal efficiency of the ideal Diesel cycle as a function of compression and cutoff ratios (γ = 1.4).

1.6 Air-Standard Dual Cycle.

The Otto and Diesel cycles do not accurately represent the pressure-volume diagrams of real internal combustion engines. To better approximate the pressure variations, an air-standard cycle known as the air-standard dual cycle is employed. The dual cycle, illustrated in Figure 1.9, follows a similar pattern to the Otto and Diesel cycles.

Just like in the Otto and Diesel cycles, the first process, 1-2, involves an isentropic compression. However, the heat addition takes place in two steps in the dual cycle : process 2-3 is a constant-volume heat addition, while process 3-4 is a constant-pressure heat addition. Additionally, process 3-4 forms the initial part of the power stroke. The remaining part of the power stroke occurs during the isentropic expansion from state 4 to state 5.

Similar to the Otto and Diesel cycles, the cycle is completed by a constant-volume heat rejection process, known as process 5-1. The areas on the temperature-entropy (T-s) and pressure-volume (p-v) diagrams can be interpreted as heat and work, respectively, just as in the case of the Otto and Diesel cycles.

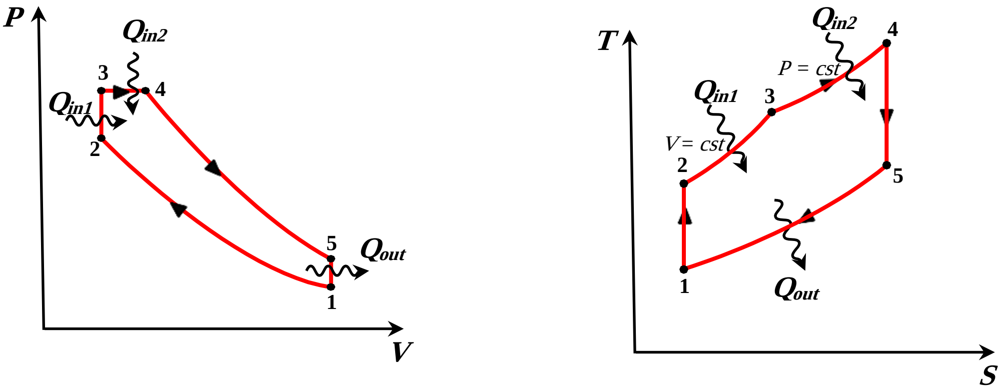


Figure 1.9 P–V and T–s diagrams of the air-standard dual cycle.

The cycle consists of five internally reversible processes in series:

Process 1-2: Isentropic compression of the air as the piston moves from bottom dead center to top dead center, $(Q_{12} = 0)$.

$$W_{is}^{C} = W_{12} = \Delta u_{12} = C_v(T_2 - T_1) > 0$$

$$T_2 = (r^{\gamma-1})T_1$$

Process 2-3: Constant-volume heat transfer to the air from an external source while the piston is at top dead center. This process is intended to represent the ignition of the fuel–air mixture and the subsequent rapid burning.

$$Q_{in1} = \Delta u_{23} = C_v(T_3 - T_2) > 0$$

$$\frac{P_3}{P_2} = \frac{T_3}{T_2} = r_P$$

Process 3-4: Constant-pressure heat transfer to the air from an external source while the piston is at top dead center , diesel injection.

$$Q_{in2} = \Delta h_{34} = C_p(T_4 - T_3) > 0$$

$$W_{34} = \Delta u_{34} = C_v(T_4 - T_3) < 0$$

Process 4-5: Isentropic expansion (power stroke), $(Q_{34} = 0)$.

$$W_{is}^{E} = W_{45} = \Delta u_{45} = C_v(T_5 - T_4) < 0$$

$$T_3 = \left(\left(\frac{r_c}{r}\right)^{1-\gamma}\right)T_4$$

Process 5-1: Completes the cycle by a constant-volume process in which heat is rejected from the air while the piston is at bottom dead center.

$$Q_{out} = \Delta u_{51} = C_v(T_1 - T_5) < 0.$$

$$\frac{T_5}{T_1} = \frac{T_5}{T_1} = r_P . {r_c}^{\gamma}$$

r : the compression ratio $(r = \frac{V_1}{V_2} > 1)$,

r_c **:** the cutoff ratio $(r_c = \frac{V_4}{V_3} > 1)$,

r_P : the pressure ratio $(r_P = \frac{P_3}{P_2} > 1)$,

Cycle assessment.

Useful energy.	$W_{cycle} = W_{12} + W_{34} + W_{45} = -(Q_{in} + Q_{out})$
Costly energy.	$Q_{in} = Q_{in1} + Q_{in2} = \Delta u_{23} + \Delta h_{34} = C_v(T_3 - T_2) + C_p(T_4 - T_3)$
Lost energy.	$Q_{out} = C_v(T_1 - T_5)$
Thermal efficiency of the cycle.	$\eta_{th} = \left\lvert\frac{W_{cycle}}{Q_{in}}\right\rvert = 1 + \frac{Q_{out}}{Q_{in}} = 1 - (r^{1-\gamma})\left[\frac{r_P . r_c{}^{\gamma} - 1}{r_P - 1 + \gamma . r_P(r_c - 1)}\right]$

1.7 Brayton cycle the ideal cycle for gas-turbine engines.

George Brayton introduced the Brayton cycle circa 1870 as a means to power his reciprocating oil-burning engine. Nowadays, this cycle finds application exclusively in gas turbines, where both compression and expansion transpire within rotating equipment. Gas turbines typically operate on an open cycle, as depicted in Figure 1.10. The process commences by drawing in fresh air at ambient conditions, which then undergoes compression to elevate its temperature and pressure. Subsequently, the high-pressure air enters the combustion chamber, where fuel combustion transpires under constant pressure. The resultant high-temperature gases are subsequently directed into the turbine, where they expand to atmospheric pressure while concurrently generating power. Lastly, the exhaust gases exit the turbine without recirculation, thus classifying the cycle as an open cycle.

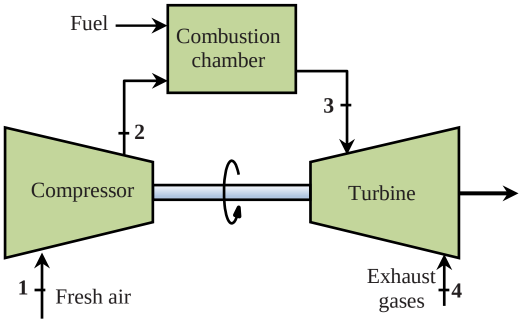


Figure 1.10 An open-cycle gas-turbine engine.

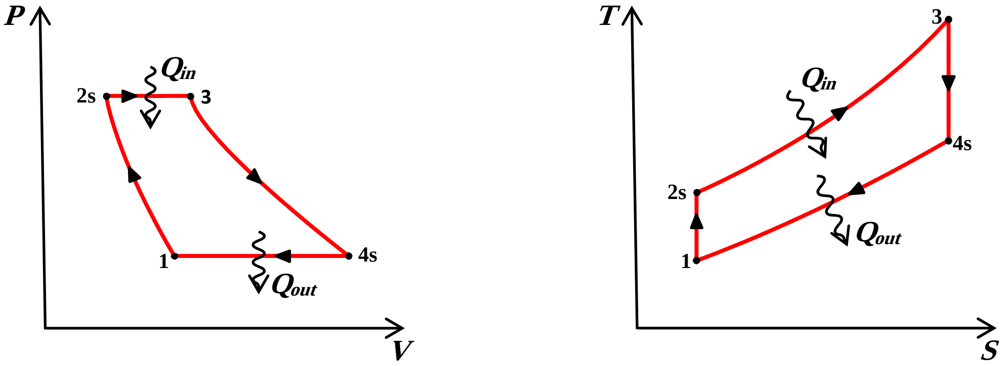


Figure 1.11 T-s and P-v diagrams for the ideal Brayton cycle.

The four processes of the cycle are :

Process 1-2s: Isentropic compression (in a compressor), $(Q_{12} = 0)$.

$$W_{is}^{C} = W_{12s} = \Delta h_{12s} = C_p(T_{2s} - T_1) > 0$$

$$T_{2s} = \left(r_P^{\frac{\gamma-1}{\gamma}}\right) T_1$$

Process 2s-3: Constant-pressure heat addition, $W_{23} = 0$ open system.

$$Q_{in} = Q_{2s3} = \Delta h_{2s3} = C_p(T_3 - T_{2s}) > 0$$

Process 3-4s: Isentropic expansion (in a turbine), $(Q_{34} = 0)$.

$$W_{is}^{E} = W_{34s} = \Delta h_{34s} = C_P(T_{4s} - T_3) < 0$$

$$T_3 = \left(r_P^{\frac{\gamma-1}{\gamma}}\right) T_{4s}$$

Process 4s-1: Constant-pressure heat rejection, $W_{4s1} = 0$ open system.

$$Q_{out} = Q_{4s1} = \Delta h_{4s1} = C_p(T_1 - T_{4s}) < 0.$$

Cycle assessment.

Useful energy. $W_{cycle} = W_{12s} + W_{4s1} = -(Q_{in} + Q_{out})$

Costly energy. $Q_{in} = \Delta h_{2s3} = C_p(T_3 - T_{2s})$

Lost energy. $Q_{out} = \Delta h_{4s1} = C_p(T_1 - T_{4s})$

Thermal efficiency of the cycle.

$$\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}} = 1 - \left(r_p^{\frac{\gamma-1}{\gamma}}\right)$$

Note :

The thermal efficiency of the cycle increases with the pressure ratio.

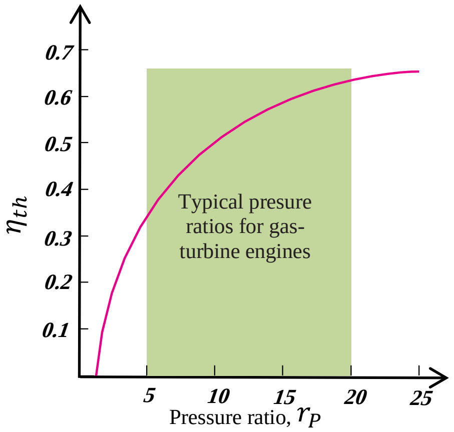


Figure 1.12 Thermal efficiency of the ideal Brayton cycle as a function of the pressure ratio.

1.7.1 Deviation of actual gas-turbine cycles from idealized ones

The actual gas-turbine cycle differs from the ideal Brayton cycle on several accounts. For one thing, some pressure drop during the heat addition and heat compressor is more, and the actual work output from the turbine is less because of irreversibilities. The deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accurately accounted for by utilizing the isentropic efficiencies of the turbine and compressor as :

$$\eta_C = \frac{W_{is}^{C}}{W^{c}} = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{T_{2s}-T_1}{T_2-T_1}$$

$$\eta_T = \frac{W^T}{W{c}_{is}^{T}} = \frac{h_4-h_3}{h_{4s}-h_3} = \frac{T_4-T_3}{T_{4s}-T_3}$$

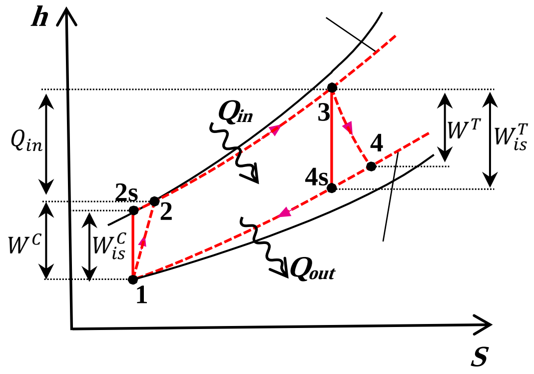


Figure 1.13 The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

The four processes of the cycle are :

Process 1-2s: Adiabatic compression (in a compressor), $(Q_{12} = 0)$.

$$W^C = W_{12} = \Delta h_{12} = C_p(T_2 - T_1) > 0$$

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

Process 2s-3: Constant-pressure heat addition, $W_{23} = 0$ open system.

$$Q_{in} = Q_{23} = \Delta h_{23} = C_p(T_3 - T_2) > 0$$

Process 3-4s: Adiabatic expansion (in a turbine), $(Q_{34} = 0)$.
$$W^E = W_{34} = \Delta h_{34} = C_P(T_4 - T_3) < 0$$

$$T_3 = \left(r_P^{\frac{\gamma-1}{\gamma}}\right) T_4$$

Process 4s-1: Constant-pressure heat rejection, $W_{4s1} = 0$ open system.

$$Q_{out} = Q_{41} = \Delta h_{41} = C_p(T_1 - T_4) < 0.$$

Cycle assessment.

Useful energy. $W_{cycle} = W_{12} + W_{41} = -(Q_{in} + Q_{out})$

Costly energy. $Q_{in} = \Delta h_{23} = C_p(T_3 - T_{2s})$

Lost energy. $Q_{out} = \Delta h_{41} = C_p(T_1 - T_4)$

Thermal efficiency of the cycle.

$$\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}} = 1 - \left(r_P{}^{\frac{\gamma-1}{\gamma}}\right)$$

1.7.2 The brayton cycle with regeneration.

The exhaust temperature of a gas turbine is typically much higher than the surrounding ambient temperature. As a result, the hot turbine exhaust gas possesses a valuable potential for energy utilization (exergy) that would be permanently lost if the gas were directly released into the environment. One way to harness this potential is through a heat exchanger called a regenerator, which allows the air exiting the compressor to be preheated before entering the combustor. This preheating reduces the amount of fuel required to be burned in the combustor.

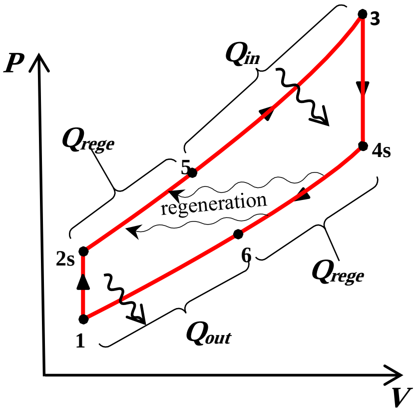


Figure 1.14 T-s diagram of a Brayton cycle with regeneration.

The four processes of the cycle are :

Process 1-2s: Isentropic compression (in a compressor), $(Q_{12} = 0)$.

$$W_{is}^{C} = W_{12s} = \Delta h_{12s} = C_p(T_{2s} - T_1) > 0$$

$$T_{2s} = \left(r_P^{\frac{\gamma-1}{\gamma}}\right) T_1$$

Process 2s-5: Preheating (isobaric) of the air in the regenerator (heat recovery).

$$Q_{rege} = Q_{2s5} = \Delta h_{2s5} = C_p(T_5 - T_{2s}) > 0$$

Process 5-3: Constant-pressure heat addition, $(W_{53} = 0)$ open system.

$$Q_{in} = Q_{53} = \Delta h_{53} = C_p(T_3 - T_5) > 0$$

Process 3-4s: Isentropic expansion (in a turbine), $(Q_{34} = 0)$.

$$W_{is}^{E} = W_{34s} = \Delta h_{34s} = C_P(T_{4s} - T_3) < 0$$

$$T_3 = \left(r_P^{\frac{\gamma-1}{\gamma}}\right) T_{4s}$$

Process 4s-6 Precooling (isobaric) of the gases burned in the regenerator

$$Q_{rege} = Q_{4s6} = \Delta h_{4s6} = C_p(T_6 - T_{4s}) < 0$$

Process 6-1: Constant-pressure heat rejection, $W_{4s1} = 0$ open system.

$$Q_{out} = Q_{61} = \Delta h_{61} = C_p(T_1 - T_6) < 0.$$

Cycle assessment.

Useful energy. $W_{cycle} = W_{12s} + W_{4s1} = -(Q_{in} + Q_{out})$

Costly energy. $Q_{in} = \Delta h_{53} = C_p(T_3 - T_5)$

Lost energy. $Q_{out} = \Delta h_{61} = C_p(T_1 - T_6)$

Thermal efficiency of the cycle.

$$\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}} = 1 - \left(r_P^{\frac{\gamma-1}{\gamma}}\right)$$

1.7.3 Gas turbines with reheat.

For metallurgical reasons,, it is necessary to limit the temperature of the gaseous combustion products entering the turbine. This can be achieved by supplying an excess of air beyond what is required for fuel combustion in the combustor. Consequently, the gases exiting the combustor contain surplus air that can support the combustion of additional fuel. Certain gas turbine power plants capitalize on this excess air by utilizing a multistage turbine with a reheat combustor positioned between the stages. This configuration allows for an increase in the net work output per unit of mass flow. Let us now analyze the concept of reheat from the perspective of an air-standard analysis.

The essential characteristics of a two-stage gas turbine with reheat can be explained by examining a modified ideal air-standard Brayton cycle, as illustrated in Figure 1.15. In this configuration, the gas undergoes expansion from state 3 to state a in the first turbine. It is then reheated at constant pressure from state a to state b. The expansion process is completed in the second turbine, going from state b to state 4s. For comparison, the ideal Brayton cycle without reheat, represented by the path 1-2s-3-4's-1 on the T-s diagram, is also shown.

Due to the slight divergence of lines of constant pressure on a T-s diagram with increasing entropy, the total work output of the two-stage turbine is greater than that of a single expansion from state 3 to state 4's. Consequently, the net work produced by the reheat cycle is greater than that of the cycle without reheat.

However, despite the increase in net work achieved with reheat, the thermal efficiency of the cycle may not necessarily improve. This is because a greater amount of heat input would be required for the reheat cycle. Nevertheless, incorporating reheat results in a higher temperature at the turbine exit compared to the cycle without reheat. This higher temperature enhances the potential for regeneration within the system.

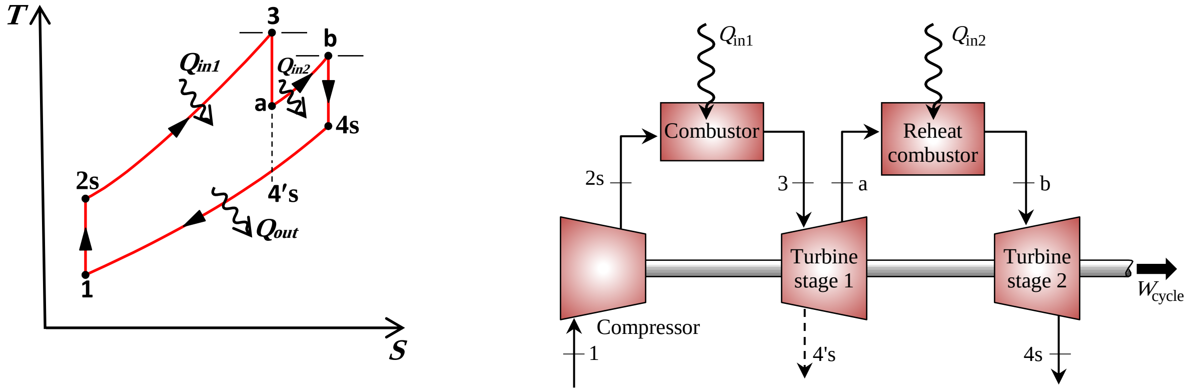


Figure 1.15 Ideal gas turbine with reheat.

The six processes of the cycle are :

Process 1-2s: Isentropic compression (in a compressor), $(Q_{12} = 0)$.

$$W_{is}^{C} = W_{12s} = \Delta h_{12s} = C_p(T_{2s} - T_1) > 0$$

$$T_{2s} = \left(r_P^{\frac{\gamma-1}{\gamma}}\right) T_1$$

Process 2s-3: Constant-pressure heat addition, $W_{23} = 0$ open system.

$$Q_{in1} = Q_{2s3} = \Delta h_{2s3} = C_p(T_3 - T_{2s}) > 0$$

Process 3-a: Isentropic expansion in the first stage of the turbine, $(Q_{3a} = 0)$.

$$W_{is}^{T1} = W_{3a} = \Delta h_{3a} = C_P(T_a - T_3) < 0$$

Process a-b: Constant-pressure heat re-addition, $W_{ab} = 0$ open system.

$$Q_{in2} = Q_{ab} = \Delta h_{ab} = C_p(T_b - T_a) > 0$$

Process b-4s: Isentropic expansion in the second stage of the turbine, $(Q_{b4s} = 0)$.

$$W_{is}^{T2} = W_{bs} = \Delta h_{bs} = C_P(T_{bs} - T_3) < 0$$

Process 4s-1: Constant-pressure heat rejection, $W_{4s1} = 0$ open system.

$$Q_{out} = Q_{4s1} = \Delta h_{4s1} = C_p(T_1 - T_{4s}) < 0.$$

Cycle assessment.

Useful energy. $W_{cycle} = W_{12s} + W_{3a} + W_{bs} = -(Q_{in} + Q_{out})$

Costly energy. $Q_{in} = Q_{in1} + Q_{in2} = C_p(T_3 - T_{2s}) + C_p(T_b - T_a)$

Lost energy. $Q_{out} = \Delta h_{4s1} = C_p(T_1 - T_{4s})$

Thermal efficiency of the cycle. $\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}}$

1.7.4 Compression with Intercooling.

One way to increase the net work output of a gas turbine is by reducing the work required by the compressor. This can be achieved through a technique known as multistage compression with intercooling.

To begin, let's examine the work input to compressors under steady-state conditions, assuming the absence of irreversibilities and negligible changes in kinetic and potential energy between the inlet and exit. Figure 1.16 illustrates a p-v diagram that presents two potential compression paths from a specified state 1 to a predetermined final pressure p2. Path 1-2' represents an adiabatic compression, while path 1-2 corresponds to a compression process involving heat transfer from the working fluid to the surroundings. The area to the left of each curve represents the magnitude of work per unit mass for the respective process. The smaller area to the left of Process 1-2 indicates that the work associated with this process is lower than that of the adiabatic compression from 1 to 2'. This suggests that gas cooling during compression offers advantages in terms of reducing the work input requirement.

However, achieving a heat transfer rate that is sufficiently high to bring about a significant reduction in work is challenging in practical applications. A practical alternative is to separate the work and heat interactions by conducting compression in stages with the aid of heat exchangers, known as intercoolers, which cool the gas between stages. Figure 1.17 demonstrates a two-stage compressor equipped with an intercooler. The accompanying p-v and T-s diagrams depict the states associated with internally reversible processes:

- ✓ Process 1-c denotes an isentropic compression from state 1 to state c, where the pressure is pi.
- ✓ Process c-d represents constant-pressure cooling from temperature Tc to Td.
- ✓ Process d-2 signifies an isentropic compression to reach state 2.

The work input per unit mass flow is represented on the p-v diagram by the shaded area 1-c-d-2-a-b-1. Without intercooling, the gas would undergo an isentropic compression in a single stage from state

1 to state 2', and the work would be represented by the enclosed area 1-2'-a-b-1. The crosshatched area on the p-v diagram illustrates the reduction in work achieved through intercooling.

In certain cases, large compressors utilize multiple stages of compression with intercooling between the stages. Determining the number of stages and the operating conditions of the intercoolers entails an optimization problem. The utilization of multistage compression with intercooling in a gas turbine power plant increases the net work output by decreasing the compression work. However, compression with intercooling alone does not necessarily lead to an increase in the thermal efficiency of a gas turbine because it lowers the temperature of the air entering the combustor (compare temperatures at states 2' and 2 on the T-s diagram in Figure 1.17). A lower temperature at the combustor inlet would necessitate additional heat transfer to attain the desired turbine inlet temperature. Nevertheless, the lower temperature at the compressor exit enhances the potential for regeneration. Therefore, when intercooling is combined with regeneration, a significant increase in thermal efficiency can be achieved.

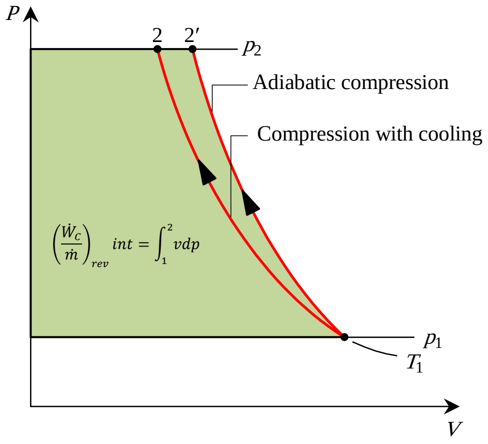


Figure 1.16 Internally reversible compression processes between two fixed pressures

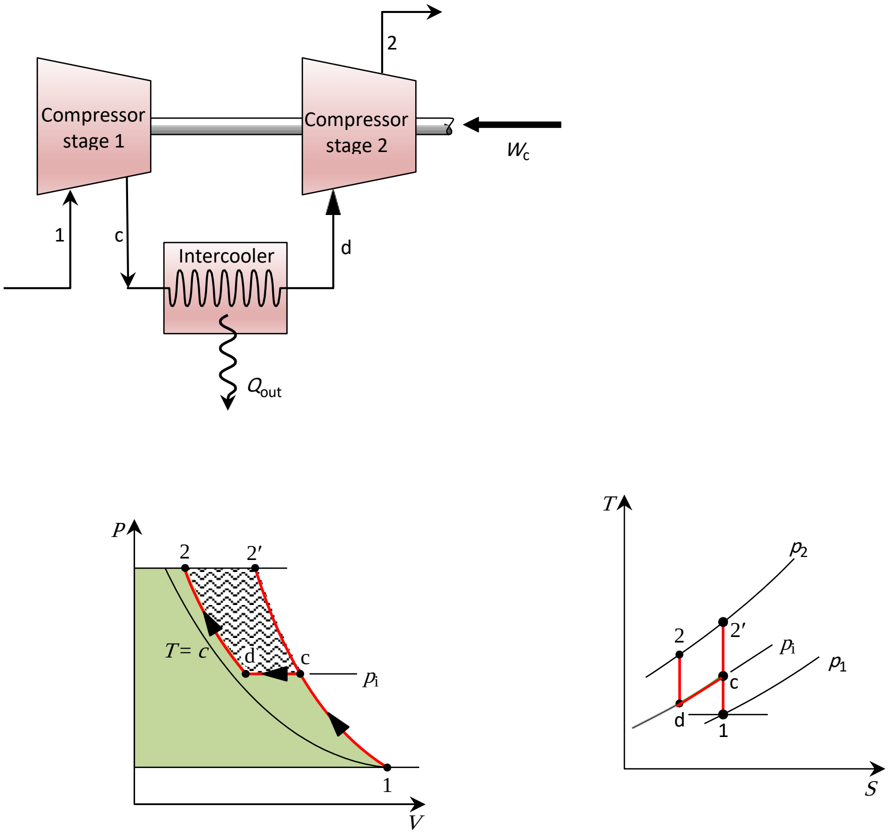


Figure 1.17 Two-stage compression with intercooling.

1.7.5 Reheat and Intercooling

The utilization of reheat between turbine stages and intercooling between compressor stages offers two significant advantages: an increase in net work output and an enhancement of the potential for regeneration. Consequently, when reheat, intercooling, and regeneration are combined, a substantial improvement in performance can be achieved. Figure 1.18 illustrates one configuration that incorporates reheat, intercooling, and regeneration. This particular gas turbine system features two stages of compression and two turbine stages. The accompanying T-s diagram is constructed to represent irreversibilities present in the compressor and turbine stages. However, the diagram does not include the pressure drops that would occur as the working fluid traverses the intercooler, regenerator, and combustors.

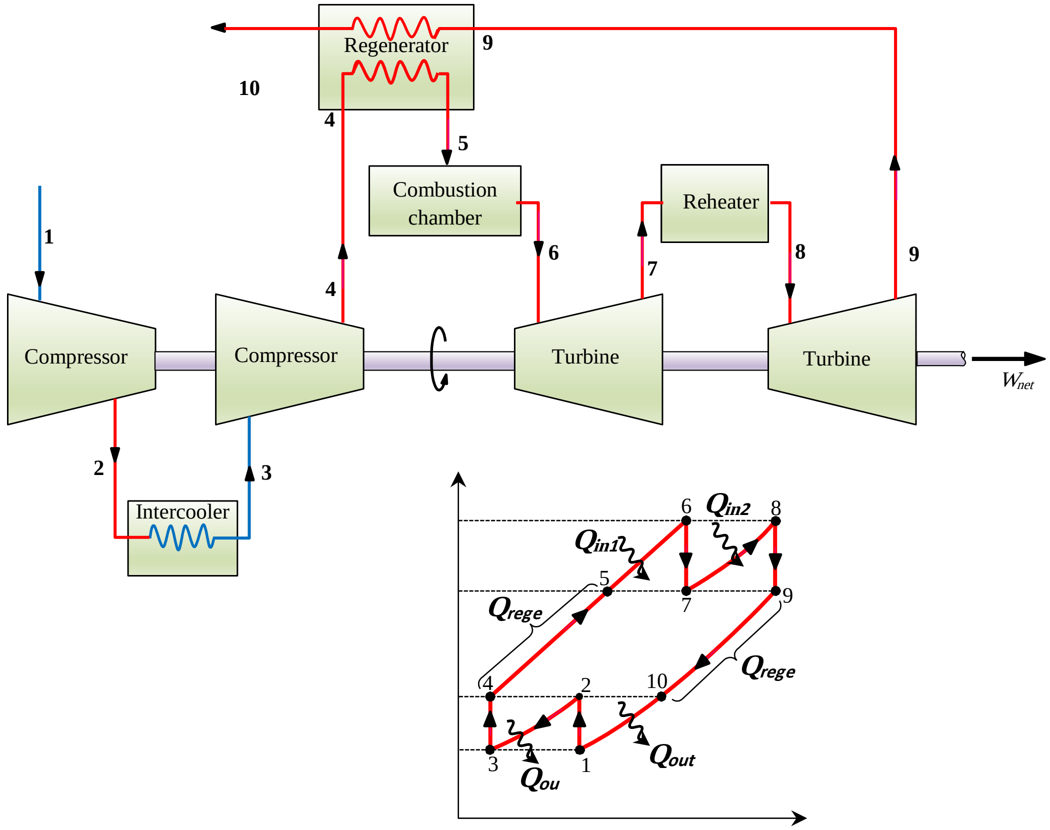


Figure 1.18 Regenerative gas turbine with intercooling and reheat.

1.8 Ericsson and Stirling Cycles.

The Stirling cycle is characterized by four totally reversible processes depicted in Figure 1-19(b):

1-2: Constant temperature expansion (heat addition from the external source)

2-3: Constant volume regeneration (internal heat transfer from the working fluid to the regenerator)

3-4: Constant temperature compression (heat rejection to the external sink)

4-1: Constant volume regeneration (internal heat transfer from the regenerator back to the working fluid)

The ideal Otto and Diesel cycles, which were discussed earlier, are internally reversible cycles composed of internally reversible processes. However, they are not totally reversible because they involve heat transfer through a finite temperature difference during the non-isothermal heat addition and heat rejection processes, which are irreversible. As a result, the thermal efficiency of an Otto or Diesel engine is lower than that of a Carnot engine operating between the same temperature limits.

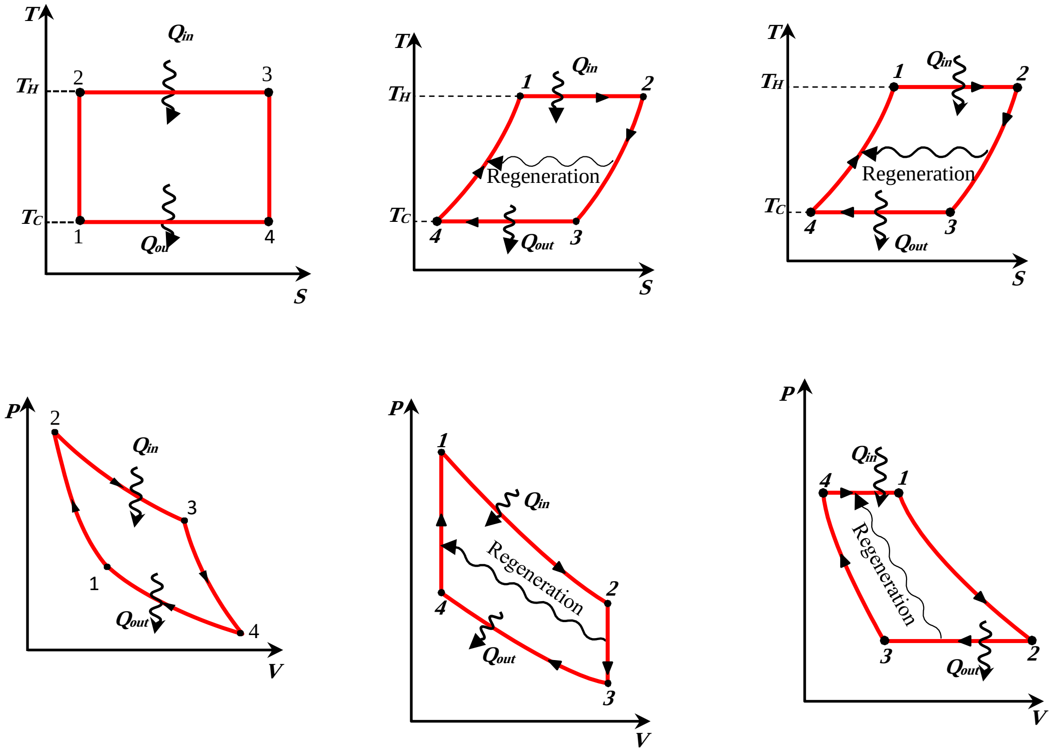
In order for a heat engine cycle to be totally reversible, the temperature difference between the working fluid and the heat source or sink should never exceed a small amount called dT during any heat transfer process. This means that both the heat addition and heat rejection processes must occur isothermally, one at a temperature of T_H (the temperature of the heat source) and the other at a temperature of T_C (the temperature of the heat sink). This is exactly what happens in a Carnot cycle.

There are two other cycles, the Stirling cycle and the Ericsson cycle, that involve an isothermal heat addition process at T_H and an isothermal heat rejection process at T_C. These cycles 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. Both cycles utilize a regenerator, which is a device that stores thermal energy during one part of the cycle and releases it back to the working fluid during another part of the cycle.

The Stirling cycle consists of four totally reversible processes: a constant expansion process at temperature T_H (heat addition), a constant-volume regeneration process, a constant compression process at temperature T_C (heat rejection), and another constant-volume regeneration process. The Stirling cycle requires innovative hardware, and while actual Stirling engines are complex, a hypothetical engine is used to explain the execution of the cycle in a closed system.

The Ericsson cycle is similar to the Stirling cycle, but the two constant-volume processes are replaced by two constant-pressure processes. In an ideal case, the Ericsson cycle is executed in a steady-flow system where the isothermal expansion and compression processes occur in a compressor and a turbine, respectively, and a counter-flow heat exchanger acts as a regenerator.

Both the Stirling cycle and the Ericsson cycle, like the Carnot cycle, are totally reversible. According to the Carnot principle, all three cycles must have the same thermal efficiency when operating between the same temperature limits.



(*a*) Carnot cycle (*b*) Stirling cycle (*c*) Ericsson cycle

Figure 1.19 *T*-*s* and *P*-*v* diagrams of Carnot, Stirling, and Ericsson cycles.

1.9 The different components of a gas thermal power plant:

A thermal power plant is an electricity-generating facility that utilizes a heat source, typically obtained from the combustion of fuels such as natural gas, to produce electricity. The heat derived from the fuel source is used to elevate the temperature of a fluid, often water, causing it to undergo a phase transition from a liquid state to a gaseous state (such as steam). This high-pressure gas is then directed to drive a turbine, which is connected to an alternator responsible for converting the turbine's kinetic energy into electrical energy.

To enable the turbine to operate effectively, it is necessary for the pressure at the turbine's outlet to be lower than at its inlet. This pressure drop is achieved by condensing the gas downstream of the turbine, utilizing a cold source. The condensed fluid, typically recycled as a steam source, undergoes a closed thermodynamic cycle within the system.

Cogeneration, a technique employed in thermal power plants, involves the simultaneous production of both electricity and heat. The heat generated can be utilized for industrial processes or district heating, thereby enhancing the overall efficiency of the power plant.

A gas thermal power plant consists of several key components that work together to generate electricity. These components include:

Combustion Chamber: This is where the fuel, such as natural gas, is burned to produce high-temperature gases. The combustion process releases heat energy.

Gas Turbine : The hot gases from the combustion chamber are directed into a gas turbine. The turbine consists of a rotor with blades that are driven by the high-velocity gases. As the gases expand through the turbine, they transfer their kinetic energy to the rotor, causing it to rotate.

Compressor: The gas turbine is typically equipped with a compressor at its inlet. The compressor compresses the incoming air, increasing its pressure before it enters the combustion chamber. This allows for efficient combustion and higher power output.

Generator: The rotating shaft of the gas turbine is connected to a generator. The generator converts the mechanical energy from the turbine's rotation into electrical energy. It consists of a rotor and stator, where the rotating magnetic field induces an electric current in the stator windings.

Heat Recovery Steam Generator (HRSG): The exhaust gases from the gas turbine still contain a significant amount of heat energy. The HRSG captures this waste heat and uses it to generate steam. The HRSG usually includes a series of heat exchangers to transfer heat from the exhaust gases to water, producing high-pressure and high-temperature steam.

Steam Turbine : The high-pressure steam produced by the HRSG is directed into a steam turbine. Similar to the gas turbine, the steam turbine has blades that are driven by the high-pressure steam. As the steam expands through the turbine, it transfers its energy to the rotor, causing it to rotate.

Condenser : After passing through the steam turbine, the low-pressure steam is directed into a condenser. The condenser cools and condenses the steam back into water by transferring heat to a cooling medium, such as water from a nearby lake or river.

Cooling Tower : The heat absorbed by the cooling medium in the condenser is dissipated to the surroundings through a cooling tower. The cooling tower facilitates the transfer of heat from the cooling medium to the ambient air, allowing the cooling medium to be reused in the condenser.

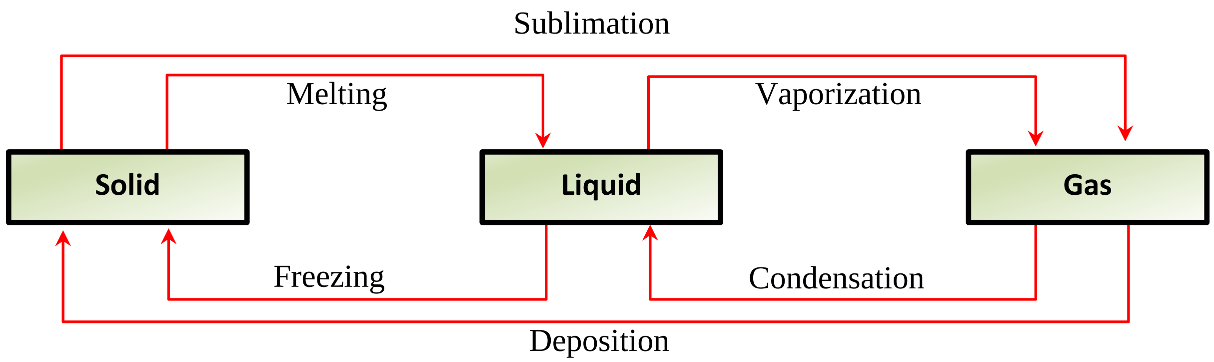
Electrical Grid Connection: The generated electricity from the generator is typically stepped up in voltage through a transformer and then transmitted to the electrical grid for distribution to consumers.

Chapter 2: two-phase power cycles.

2.1 Overview of Phase Change

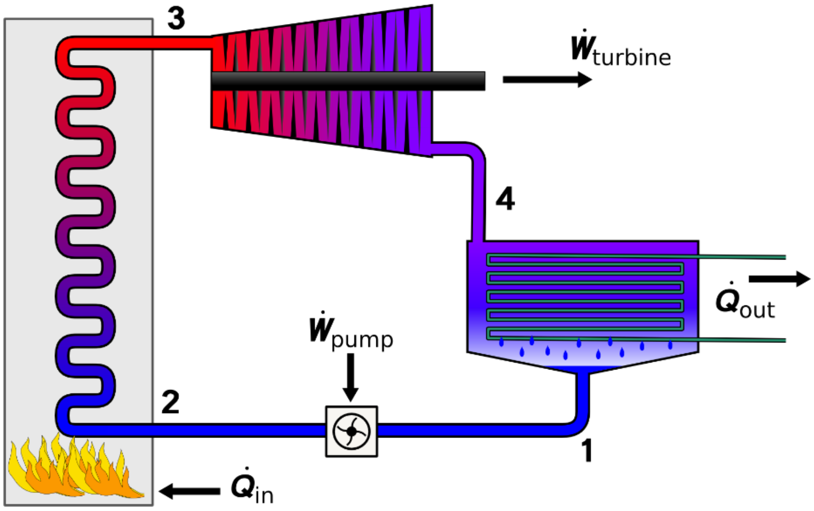
When a pure substance transitions from one equilibrium state to another, under specific conditions, significant and sudden changes in its optical, mechanical, and electrical properties can occur. This phenomenon is referred to as a phase change or phase transition. These phenomena are commonly observed, such as the transformation of liquid water into vapor or ice. However, this definition also encompasses other phenomena, such as the transition of iron from the α phase (body-centered cubic lattice) to the γ phase (face-centered cubic lattice) at a temperature of 1176 K under atmospheric pressure, which is known as an allotropic variation [8].

The following diagram summarizes the possible phase changes of a pure substance and their respective names:



2.2 Ideal Rankine Cycle.

In the absence of irreversibilities, the simple vapor power cycle demonstrates certain idealized characteristics. Specifically, if the working fluid traverses the cycle's components without experiencing frictional pressure drops, the boiler and condenser operate at constant pressure, enabling smooth fluid flow. Furthermore, when irreversibilities and heat transfer with the surroundings are neglected, the processes occurring in the turbine and pump become isentropic, meaning they are reversible and adiabatic. The ideal Rankine cycle, depicted in Figure 2.1, exemplifies a cycle that conforms to these idealizations.



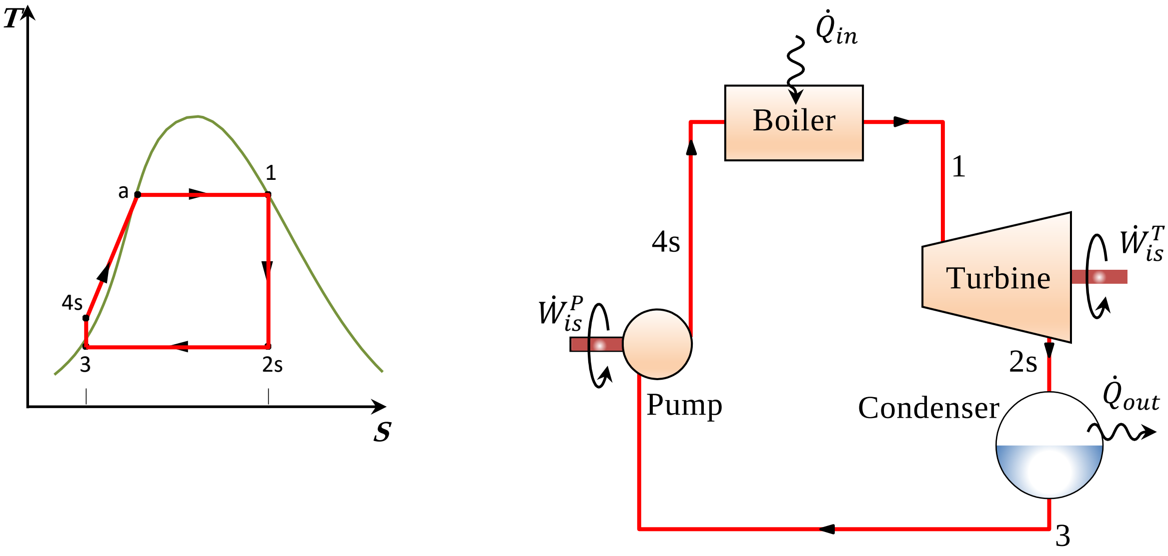


Figure 2.1 The simple ideal Rankine cycle.

Referring to Fig. 8.3, we see that the working fluid undergoes the following series of in- ternally reversible processes:

Process 1-2s: Isentropic expansion in a turbine, $\left(\dot{Q}_{12s} = 0\right)$.

$$\dot{W}_{is}^{T} = \dot{W}_{12s} = \dot{m}\Delta h_{12s} = \dot{m}(h_{2s} - h_1) > 0 \qquad [KW]$$

Process 2s-3: heat rejection in a condenser at constant pressure and temperature, $\left(\dot{W}_{2s3} = 0\right)$

$$\dot{Q}_{out} = \dot{Q}_{2s3} = \dot{m}\Delta h_{2s3} = \dot{m}(h_3 - h_{2s}) > 0 \qquad [KW]$$

Process 3-4s: Isentropic compression in a pump, $(Q_{34s} = 0)$.

$$W_{is}^{P} = W_{34s} = \dot{m}\Delta h_{34s} = \dot{m}(h_{4s} - h_3) < 0 \qquad [KW]$$

$$W_{is}^{P} = \dot{m}\int_{3}^{4s} v dp \approx \dot{m}v(P_{4s} - P_3) \text{ (Incompressible fluid } v = cst)$$

Process 4s-1: Heat transfer to the working fluid as it flows at constant pressure through the boiler to complete the cycle (preheating 4s-a and vaporisation a-1), $\dot{W}_{4s1} = 0$.

$$\dot{Q}_{in} = \dot{Q}_{4s1} = \dot{m}\Delta h_{4s1} = \dot{m}(h_1 - h_{4s}) > 0 \qquad [KW]$$

Cycle assessment.

Useful energy. $\dot{W}_{cycle} = \dot{W}_{12s} + \dot{W}_{34s} = -\left(\dot{Q}_{in} + \dot{Q}_{out}\right)$

Costly energy. $\dot{Q}_{in} = \dot{m}(h_1 - h_{4s})$

Lost energy. $Q_{out} = \dot{m}(h_3 - h_{2s})$

Thermal efficiency of the cycle. $\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}}$

2.2.1 Ideal Rankine Cycle with superheating (HIRN Cycle).

The ideal Rankine cycle also includes the possibility of superheating the vapor, as in Figure 2.2.

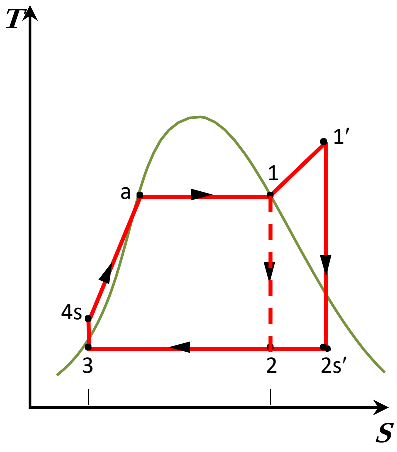


Figure 2.2 The HIRN cycle.

The four processes of the cycle are :

Process 1'-2s': Isentropic expansion in a turbine, $\left(\dot{Q}_{12s} = 0\right)$.

$$\dot{W}_{is}^{T} = \dot{W}_{1'2s'} = \dot{m}\Delta h_{1'2s'} = \dot{m}(h_{2s'} - h_{1'}) > 0 \qquad [KW]$$

Process 2s'-3: heat rejection in a condenser at constant pressure and temperature, $\left(\dot{W}_{2s'3} = 0\right)$

$$\dot{Q}_{out} = \dot{Q}_{2s'3} = \dot{m}\Delta h_{2s'3} = \dot{m}(h_3 - h_{2s'}) > 0 \qquad [KW]$$

Process 3-4s: Isentropic compression in a pump, $(Q_{34s} = 0)$.

$$W_{is}^{P} = W_{34s} = \dot{m}\Delta h_{34s} = \dot{m}(h_{4s} - h_3) < 0 \qquad [KW]$$

$$W_{is}^{P} = \dot{m}\int_{3}^{4s} v dp \approx \dot{m}v(P_{4s} - P_3)$$ (Incompressible fluid $v = cst$)

Process 4s-1: Heat transfer to the working fluid as it flows at constant pressure through the boiler to complete the cycle (preheating 4s-a , vaporisation a-1, superheating 1-1'), $\dot{W}_{4s1'} = 0$.

$$\dot{Q}_{in} = \dot{Q}_{4s1'} = \dot{m}\Delta h_{4s1'} = \dot{m}(h_{1'} - h_{4s}) > 0 \qquad [KW]$$

Cycle assessment.

Useful energy.	$\dot{W}_{cycle} = \dot{W}_{1'2s'} + \dot{W}_{34s} = -\left(\dot{Q}_{in} + \dot{Q}_{out}\right)$
Costly energy.	$\dot{Q}_{in} = \dot{m}(h_{1'} - h_{4s})$
Lost energy.	$Q_{out} = \dot{m}(h_3 - h_{2s'})$
Thermal efficiency of the cycle.	$\eta_{th} = \left\lvert\frac{W_{cycle}}{Q_{in}}\right\rvert = 1 + \frac{Q_{out}}{Q_{in}}$

2.2.2 Effects of Boiler and Condenser Pressures on the Rankine Cycle.

- ✓ **Increase Pressure in the Boiler**

We observed that the thermal efficiency of power cycles tends to increase as the average temperature at which energy is added by heat transfer increases and/or the average temperature at which energy is rejected decreases. Let us apply this idea to study the effects on performance of the ideal Rankine cycle of changes in the boiler and condenser pressures. Although these findings are obtained with reference to the ideal Rankine cycle, they also hold qualitatively for actual vapor power plants.

Figure 2.3 shows two ideal cycles having the same condenser pressure but different boiler pressures. By inspection, the average temperature of heat addition is seen to be greater for the higher-pressure cycle 1'–2s'–3'–4s'–1' than for cycle 1–2s–3–4s–1. It follows that increasing the boiler pressure of the ideal Rankine cycle tends to increase the thermal efficiency. We notice from Figure 2.3 that the cycle deviates to the left, thus increasing the moisture content in the water vapor at the outlet of the turbine, especially if the inlet temperature remains constant. This undesired effect is treated by reheated of water vapor.

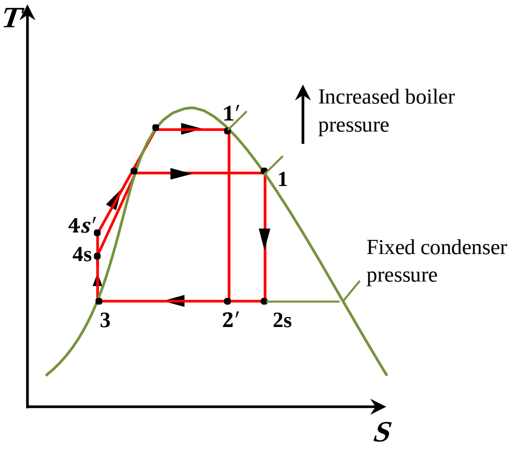


Figure 2.3 Effect of boiler pressure .

✓ Decrease Pressure in the Condenser

Figure 2.4 shows two cycles with the same boiler pressure but two different condenser pressures. One condenser operates at atmospheric pressure and the other at less than atmospheric pressure. The temperature of heat rejection for cycle 1–2s–3–4s–1 condensing at atmospheric pressure is 100°C. The temperature of heat rejection for the lower pressure cycle 1–2s"–3"–4s"–1 is corresponding lower, so this cycle has the greater thermal efficiency. It follows that decreasing the condenser pressure tends to increase the thermal efficiency.

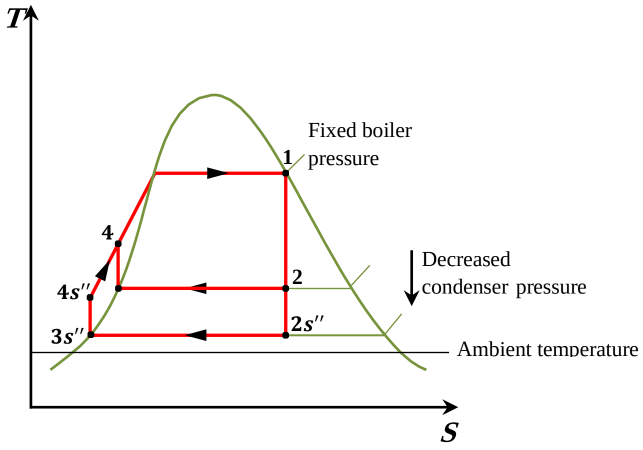


Figure 2.4 Effect of condenser pressure.

2.2.3 Principal Irreversibilities and Losses.

- ✓ Turbine.

The presence of irreversibilities during the adiabatic expansion (1-2) in the turbine, as observed in the actual case, results in a mechanical power output that is lower than the ideal case, accompanied by an increase in entropy (Figure 2.5). The ideal mechanical power corresponds to the reversible adiabatic expansion (1-2s). The ratio of these powers is referred to as the isentropic (or mechanical) efficiency of the turbine.

$$\eta_{is}^{T} = \frac{actual\ power}{ideal\ power} = \frac{\dot{W}^{T}}{\dot{W}_{is}^{T}} = \frac{h_2 - h_1}{h_{2s} - h_1}$$

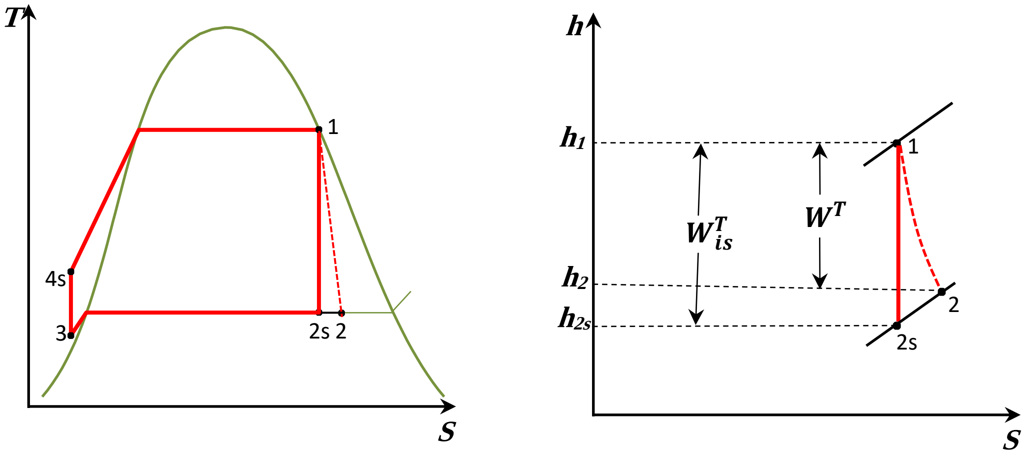


Figure 2.5 Temperature–entropy diagram showing the effects of turbine irreversibility

As irreversibilities decrease the power output of the power plant by reducing the gross power of the turbine, they also decrease the thermal efficiency of the cycle, which is expressed by the following equation:

$$\eta_{th} = \frac{\left|\dot{W}_{Cycle}\right|}{\dot{Q}_{in}} = \frac{\left|\dot{W}^{T} + \dot{W}_{is}^{P}\right|}{\dot{Q}_{in}} = \frac{\left|\eta_{is}^{T}\dot{W}_{is}^{T} + \dot{W}_{is}^{P}\right|}{\dot{Q}_{in}}$$

- ✓ Pump.

In this case as well, irreversibilities result in an increase in entropy (Fig. 3.10). However, the mechanical power obtained in the real case (3-4: irreversible compression) is higher than the mechanical power obtained in the ideal case (3-4s: reversible compression). The ratio of these powers is referred to as the isentropic (or mechanical) efficiency of the pump :

$$\eta_{is}^{P} = \frac{ideal\ power}{actual\ power} = \frac{\dot{W}_{is}^{P}}{\dot{W}^{P}} = \frac{h_{4s} - h_{3}}{h_{4} - h_{3}}$$

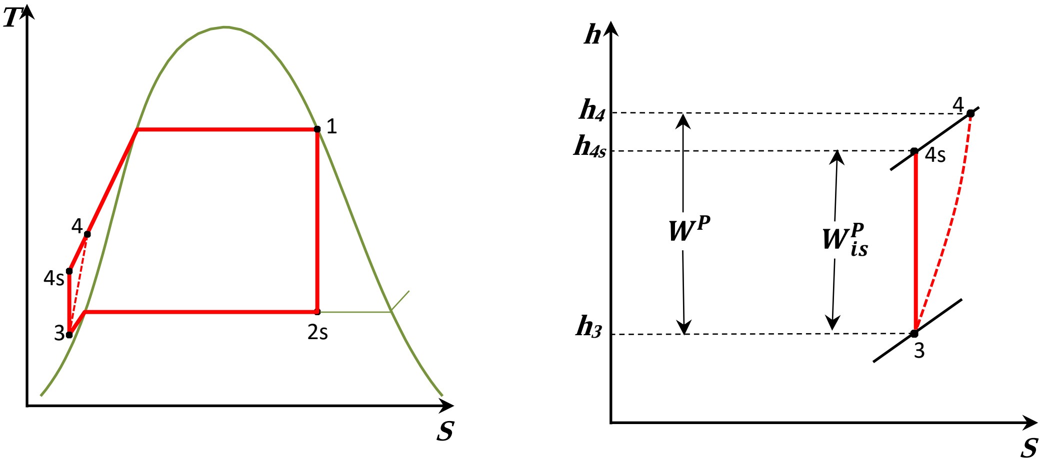


Figure 2.6 Temperature–entropy diagram showing the effects of pump irreversibility

Irreversibilities within the pump decrease the power output of the power plant by increasing the power required by the pump, thereby reducing the thermal efficiency of the cycle.

$$\eta_{th} = \frac{\left|\dot{W}_{Cycle}\right|}{\dot{Q}_{in}} = \frac{\left|\dot{W}^{P} + \dot{W}_{is}^{T}\right|}{\dot{Q}_{in}} = \frac{\left|\frac{\dot{W}_{is}^{P}}{\eta_{is}^{P}} + \dot{W}_{is}^{T}\right|}{\dot{Q}_{in}}$$

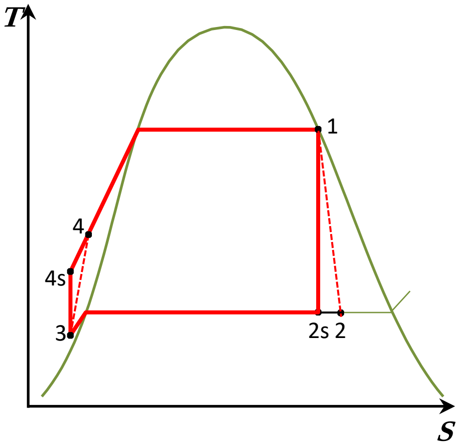


Figure 2.7 Temperature–entropy diagram showing the effects of turbine and pump irreversibilities.

Note: If the irreversibilities in both the turbine and the pump are taken into account, the thermal efficiency of the cycle is given by:

$$\eta_{th} = \frac{\left|\dot{W}_{Cycle}\right|}{\dot{Q}_{in}} = \frac{\left|\dot{W}^{P} + \dot{W}^{T}\right|}{\dot{Q}_{in}} = \frac{\left|\frac{\dot{W}_{is}^{P}}{\eta_{is}^{P}} + \eta_{is}^{T}{}^{\cdot}W_{is}^{T}\right|}{\dot{Q}_{in}}$$

2.2.4 Improving Performance—Superheat and Reheat.

The representations of the vapor power cycle considered thus far do not depict actual vapor power plants faithfully, for various modifications are usually incorporated to improve over- all performance. In this section we consider two cycle modifications known as superheat and reheat. Both features are normally incorporated into vapor power plants.

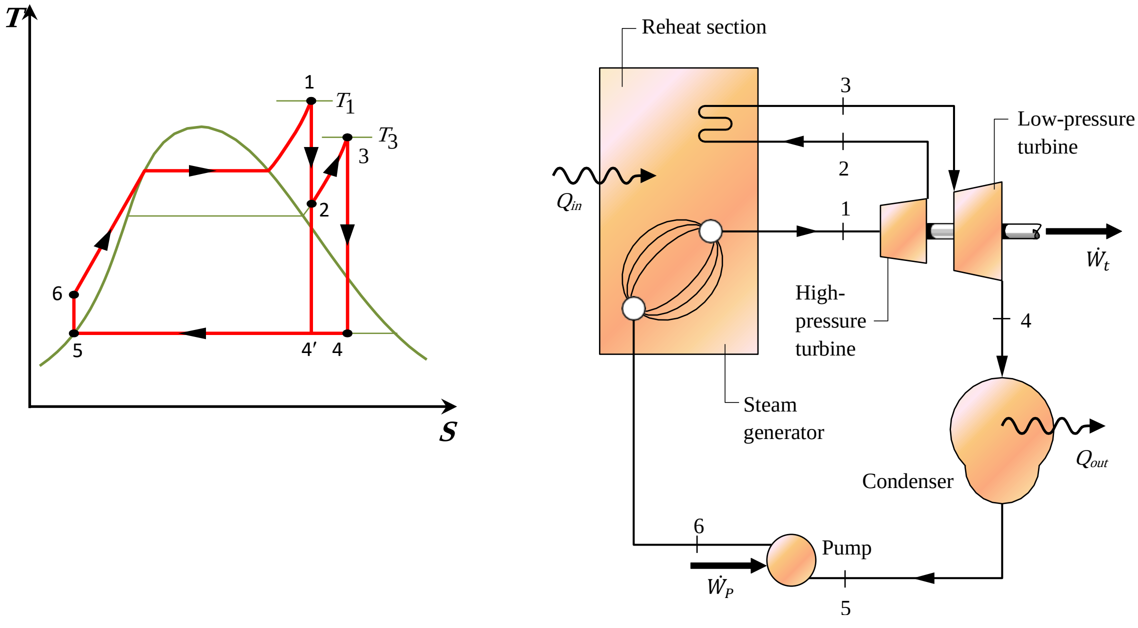


Figure 2.8 Ideal reheat cycle.

Process 1-2s: Isentropic expansion in a first-stage turbine, $\left(\dot{Q}_{12s} = 0\right)$.

$$\dot{W}_{is}^{T1} = \dot{W}_{12s} = \dot{m}\Delta h_{12s} = \dot{m}(h_{2s} - h_1) < 0 \qquad [KW]$$

Process 2s-3: The steam is then reheated in the steam generator at constant pressure and temperature, $\left(\dot{W}_{2s'3} = 0\right)$

$$\dot{Q}_{in2} = \dot{Q}_{2s3} = \dot{m}\Delta h_{2s3} = \dot{m}(h_3 - h_{2s}) > 0 \qquad [KW]$$

Process 3-4: Isentropic expansion in a second-stage turbine, $\left(\dot{Q}_{12s} = 0\right)$.

$$\dot{W}_{is}^{T2} = \dot{W}_{34} = \dot{m}\Delta h_{34} = \dot{m}(h_4 - h_3) < 0 \qquad [KW]$$

Process 4-5: heat rejection in a condenser at constant pressure and temperature, $\left(\dot{W}_{45} = 0\right)$

$$\dot{Q}_{out} = \dot{Q}_{4s5} = \dot{m}\Delta h_{45} = \dot{m}(h_5 - h_{4s}) < 0 \qquad [KW]$$

Process 5-6: Isentropic compression in a pump, $(Q_{34s} = 0)$.

$$W_{is}^{P} = W_{56} = \dot{m}\Delta h_{56} = \dot{m}(h_6 - h_5) < 0 \qquad [KW]$$

$$W_{is}^{P} = \dot{m}\int_5^6 v dp \approx \dot{m}v_5(P_6 - P_5) \text{ (Incompressible fluid } v = cst)$$

Process 6-1: Heat transfer to the working fluid as it flows at constant pressure through the boiler to complete the cycle (preheating 6s-a , vaporisation a-b, superheating b-1), $\left(\dot{W}_{6s1} = 0\right)$.

$$\dot{Q}_{in1} = \dot{Q}_{6s1} = \dot{m}\Delta h_{6s1} = \dot{m}(h_1 - h_{6s}) > 0 \qquad [KW]$$

Cycle assessment.

Useful energy. $\dot{W}_{cycle} = \dot{W}_{12s} + \dot{W}_{34s} + \dot{W}_{56s} = -\left(\dot{Q}_{in} + \dot{Q}_{out}\right)$

Costly energy. $\dot{Q}_{in} = \dot{Q}_{in1}+\dot{Q}_{\imath n2} = \dot{m}(h_1 - h_{6s}) + m(h_3 - h_{2s})$

Lost energy. $Q_{out} = \dot{m}(h_5 - h_{4s})$

Thermal efficiency of the cycle. $\eta_{th} = \left|\frac{W_{cycle}}{Q_{in}}\right| = 1 + \frac{Q_{out}}{Q_{in}}$

2.2.5 Regenerative Vapor Power Cycle.

Another commonly employed technique for enhancing the thermal efficiency of vapor power plants is regenerative open feedwater heating, also known as regeneration (Figure 2.9). This section focuses on the principles underlying regenerative feedwater heating. To introduce this concept, let us revisit Figure 2.9. In the cycle 1-2-3-4-a-1, the working fluid enters the boiler as a compressed liquid at state 4 and is heated while in the liquid phase until it reaches state a. However, with regenerative feedwater heating, the working fluid enters the boiler at a state between 4 and a. As a result, the average temperature of heat addition increases, consequently raising the thermal efficiency.

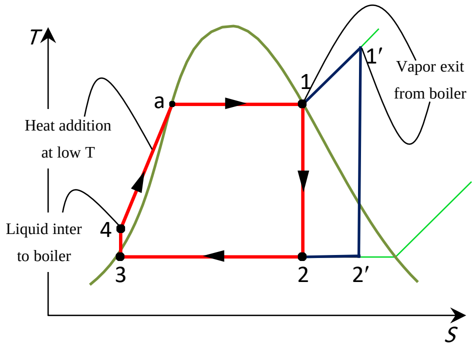


Figure 2.9 Temperature entropy diagram of the ideal Rankine cycle.

2.2.5.1 Open Feedwater Heaters

Figure 2.10 are the schematic diagram and the associated T–s diagram for a regenerative vapor power cycle having one open feedwater heater. For this cycle, the working fluid passes isentropically through the turbine stages and pumps, and the flow through the steam generator, condenser, and feedwater heater takes place with no pressure drop in any of these components. Steam enters the first-stage turbine at state 1 and expands to state 2, where a fraction of the total flow is extracted, or bled, into an open feedwater heater operating at the extraction pressure, p. The rest of the steam expands through the second-stage turbine to state 3. This portion of the total flow is condensed to saturated liquid, state 4, and then pumped to the extraction pressure and introduced into the feedwater heater at state 5. A single mixed stream exits the feedwater heater at state 6.

For the case shown in figure 2.10, the mass flow rates of the streams entering the feedwater heater are chosen so that the stream exiting the feedwater heater is a saturated liquid at the extraction pressure. The liquid at state 6 is then pumped to the steam generator pressure and enters the steam generator at state 7. Finally, the working fluid is heated from state 7 to state1 in the steam generator.

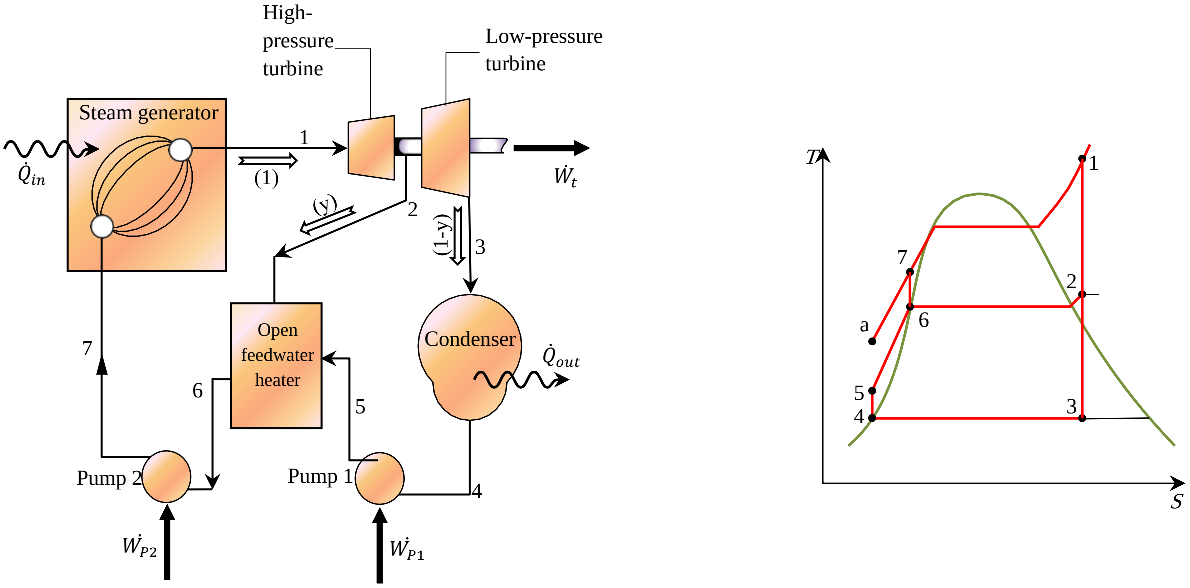


Figure 2.10 Regenerative vapor power cycle with one open feedwater heater.

When examining the T-s diagram of the cycle, it is important to note that heat addition occurs from state 7 to state 1, instead of from state a to state 1, as would be the case without regeneration. As a result, the amount of energy required from the combustion of a fossil fuel or another source to vaporize and superheat the steam is reduced. This reduction in energy consumption is a desired outcome.

It is worth mentioning that only a fraction of the total flow expands through the second-stage turbine (Process 2-3), leading to a decrease in the amount of work developed. However, in practice, operating conditions are carefully selected to ensure that the reduction in heat added more than compensates for the decrease in net work developed. As a result, regenerative power plants achieve an increased thermal efficiency.

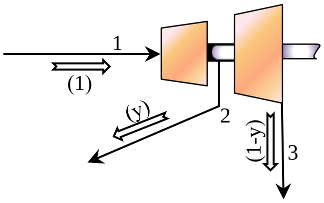


Figure 2.11 two-stage turbine

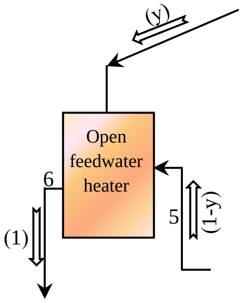


Figure 2.12 Feedwater Heater

- Open Feedwater Heater

Figure 2.12 show the schematic diagram for open feedwater heater. The Stream 1 (water vapor bleed from turbine) are mixed with stream 2 (coming fluid from pump1) to produce stream 3 (saturated fluid from open feed water heater).

Cycle analysis:

Let's now proceed with the thermodynamic analysis of the regenerative cycle depicted in Figure 8.9. An essential starting point for analyzing any regenerative vapor cycle is to evaluate the mass flow rates across each component. By considering a single control volume encompassing both turbine stages, the steady-state mass rate balance can be simplified to the following equation:

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_1$$

The equation can be expressed as follows when divided by $\dot{m}_1$, representing the unit mass passing through the first-stage turbine :

$$\frac{\dot{m}_2}{\dot{m}_1} = \frac{\dot{m}_3}{\dot{m}_1} + 1$$

where $\dot{m}_1$ is the mass flow rate entering the first-stage turbine at state 1, $\dot{m}_2$ is the mass flow rate of the extracted steam exiting at state 2, and $\dot{m}_3$ is the mass flow rate exiting the second-stage turbine at state 3.

Denoting the fraction of the total flow extracted at state 2 as y $\left(\text{where y } = \frac{\dot{m}_2}{\dot{m}_1}\right)$, the fraction of the total flow passing through the second-stage turbine can be expressed as:

$$\frac{\dot{m}_3}{\dot{m}_1} = 1 - y$$

The fractions of the total flow at various locations are indicated on Fig. 8.9. The fraction y can be determined by applying the conservation of mass and conservation of energy principles to a control volume around the feedwater heater. Assuming no heat trans- fer between the feedwater heater and its surroundings and ignoring kinetic and potential en- ergy effects, the mass and energy rate balances reduce at steady state to give.

$$0 = \dot{m}_2 h_2 + \dot{m}_3 h_5 - \dot{m}_1 h_6$$

$$0 = y h_2 + (1 - y) h_5 - h_6$$

Solving for y

$$y = \frac{h_6 - h_5}{h_2 - h_5}$$

Process 1-2: Isentropic expansion in a first-stage turbine, $\left(\dot{Q}_{12} = 0\right)$.

$$\dot{W}_{is}^{T1} = \dot{W}_{12} = \dot{m}_1 \Delta h_{12} = \dot{m}_1 (h_2 - h_1) < 0 \qquad [KW]$$

Process 2-3: Isentropic expansion in a second-stage turbine, $\left(\dot{Q}_{23} = 0\right)$.

$$\dot{W}_{is}^{T2} = \dot{W}_{23} = \dot{m}_3 \Delta h_{23} = \dot{m}_3 (h_3 - h_2) < 0 \qquad [KW]$$

$$\dot{Q}_{in2} = \dot{Q}_{2s3} = \dot{m} \Delta h_{2s3} = \dot{m}(h_3 - h_{2s}) > 0 \qquad [KW]$$

Process 3-4: heat rejection in a condenser at constant pressure and temperature, $\left(\dot{W}_{34} = 0\right)$

$$\dot{Q}_{out} = \dot{Q}_{34} = \dot{m}_3 \Delta h_{34} = \dot{m}_3 (h_4 - h_3) < 0 \qquad [KW]$$

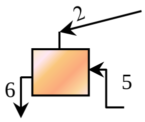
The steam is then reheated in the steam generator at constant pressure and temperature, $(\dot{W}_{2s'3} = 0)$

Process 4-5: Isentropic compression in a pump 1, $(Q_{45} = 0)$.

$$W_{is}^{P1} = W_{45} = \dot{m}_3 \Delta h_{45} = \dot{m}_3 (h_5 - h_4) > 0 \qquad [KW]$$

$$W_{is}^{P1} = \dot{m}_3 \int_4^5 v dp \approx \dot{m}_3 v_4 (P_5 - P_4) \text{ (Incompressible fluid } v = cst)$$

Mixing (y) vapor with (1-y) liquid in Open feedwater heater



$$0 = \dot{m}_2 h_2 + \dot{m}_3 h_5 - \dot{m}_1 h_6$$

Process 6-7: Isentropic compression in a pump 2, $(Q_{67} = 0)$.

$$W_{is}^{P2} = W_{67} = \dot{m}_1 \Delta h_{67} = \dot{m}_1 (h_7 - h_1) > 0 \qquad [KW]$$

$$W_{is}^{P1} = \dot{m}_1 \int_6^7 v dp \approx \dot{m}_1 v_6 (P_7 - P_6) \text{ (Incompressible fluid } v = cst)$$

Process 7-1: Heat transfer to the working fluid as it flows at constant pressure through the boiler to complete the cycle (preheating 7-a , vaporisation a-b, superheating b-1), $\dot{W}_{71} = 0$.

$$\dot{Q}_{in} = \dot{Q}_{71} = \dot{m}_1 \Delta h_{71} = \dot{m}_1 (h_7 - h_1) > 0 \qquad [KW]$$

Cycle assessment.

Useful power.	$\dot{W}_{cycle} = \dot{W}^T + \dot{W}^P = -(\dot{Q}_{in} + \dot{Q}_{out})$
Turbine power	$\dot{W}^T = \dot{m}_1 (h_2 - h_1) + \dot{m}_3 (h_3 - h_2)$
Pump power	$\dot{W}^P = \dot{m}_1 (1 - y)(h_5 - h_4) + \dot{m}_1 (h_7 - h_6)$
Costly power.	$\dot{Q}_{in} = \dot{m}_1 \Delta h_{71} = \dot{m}_1 (h_7 - h_1)$
Lost power.	$Q_{out} = \dot{m}_3 \Delta h_{34} = \dot{m}_3 (h_4 - h_3)$
Thermal efficiency of the cycle.	$\eta_{th} = \left\lvert \frac{W_{cycle}}{Q_{in}} \right\rvert = 1 + \frac{Q_{out}}{Q_{in}} = 1 + \frac{\dot{m}_3 (h_4 - h_3)}{\dot{m}_1 (h_7 - h_1)} = (1 - y) \frac{(h_4 - h_3)}{(h_7 - h_1)}$

2.2.5.2 Closed Feedwater Heaters

Regenerative feedwater heating can also be achieved using closed feedwater heaters. Closed heaters are shell-and-tube-type heat exchangers where the feedwater temperature increases as the extracted steam condenses on the outside of the tubes carrying the feedwater. Since the two streams (steam and feedwater) do not mix, they can be at different pressures. Figure 8.10 illustrates two different schemes for removing the condensate from closed feedwater heaters. In Figure 8.10a, a pump is employed to forward the condensate to a higher-pressure point in the cycle. In Figure 8.10b, the condensate is allowed to pass through a trap into a feedwater heater operating at a lower pressure or into the condenser. A trap is a type of valve that permits only liquid to pass through to a region of lower pressure.

A regenerative vapor power cycle featuring one closed feedwater heater with the condensate trapped into the condenser is presented schematically in Figure 8.11. In this cycle, the working fluid undergoes isentropic expansion through the turbine stages and pumps, with no pressure drops occurring in the other components. The T-s diagram illustrates the principal states of the cycle. The total steam flow expands through the first-stage turbine from state 1 to state 2. At state 2, a fraction of the flow is bled into the closed feedwater heater, where it condenses. Saturated liquid at the extraction pressure exits the feedwater heater at state 7. The condensate is then trapped into the condenser, where it reunites with the portion of the total flow passing through the second-stage turbine. The expansion from state 7 to state 8 through the trap is irreversible, represented by a dashed line on the T-s diagram. The total flow exiting the condenser as saturated liquid at state 4 is pumped to the steam generator pressure and enters the feedwater heater at state 5. The temperature of the feedwater increases as it passes through the feedwater heater, and it exits at state 6. The cycle is completed as the working fluid is heated at constant pressure in the steam generator from state 6 to state 1.

Although the closed heater shown in the figure operates with no pressure drop in either stream, there is a source of irreversibility due to the temperature differences between the two streams.

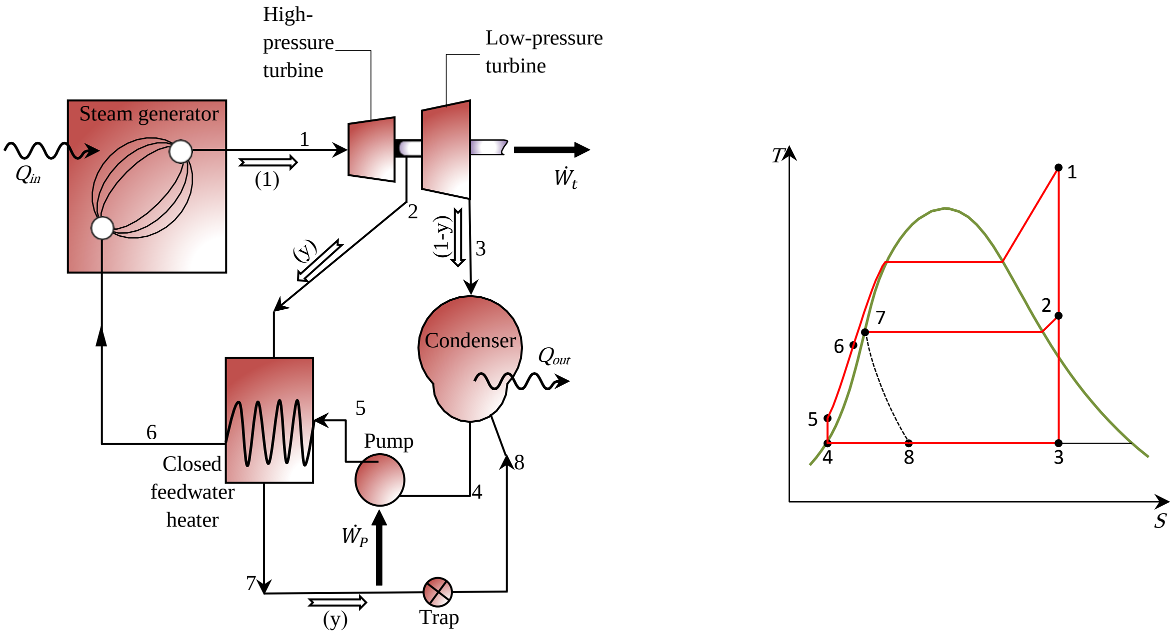


Figure 2.13 Regenerative vapor power cycle with one closed feedwater heater.

Cycle analysis. The schematic diagram of the cycle shown in Fig. 2.13 is labeled with the fractions of the total flow at various locations. This is usually helpful in analyzing such cycles. The fraction of the total flow extracted, y, can be determined by applying the con- servation of mass and conservation of energy principles to a control volume around the closed heater. Assuming no heat transfer between the feedwater heater and its surroundings and ne- glecting kinetic and potential energy effects, the mass and energy rate balances reduce at steady state to give

$$0 = \dot{m}_2(h_2 - h_7) + \dot{m}_1(h_5 - h_6)$$

$$0 = y(h_2 - h_7) + (h_5 - h_6)$$

Solving for y

$$y = \frac{h_6 - h_5}{h_2 - h_7}$$

2.2.5.3 Multiple Feedwater Heaters

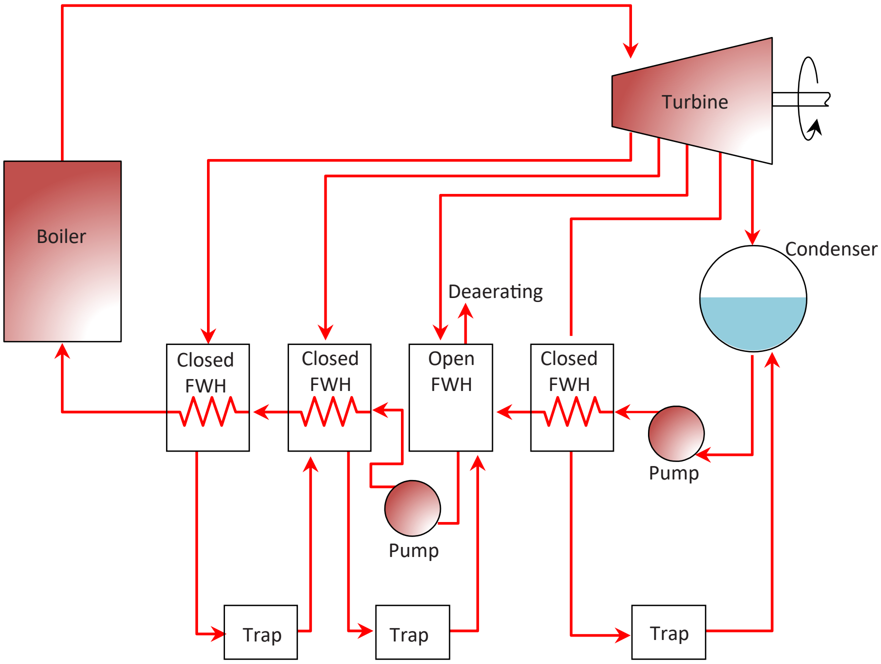


Figure 2.14 A steam power plant with one open and three closed feedwater heaters.

2.3 Combiné Gas Turbine–Vapor Power Cycle

A combined power cycle refers to the coupling of two distinct power cycles in a manner that allows the energy released through heat transfer from one cycle to be utilized partially or entirely as the input for the other cycle. An example of a combined power cycle is the binary vapor cycle, which was introduced in Section 8.5. In this section, we will focus on a specific type of combined power cycle known as the combined gas turbine-vapor power cycle.

In a gas turbine, the stream exiting the turbine is at a high temperature. One way to enhance overall fuel utilization and make use of the potential (exergy) of this high-temperature gas stream is through the implementation of a regenerator. A regenerator allows the exhaust gas from the turbine to preheat the air between the compressor and the combustor (as discussed in Section 9.7).

Another method to harness the potential of the high-temperature gas stream and improve fuel utilization is through the utilization of a combined cycle, as illustrated in Figure 9.23. This combined cycle involves the integration of a gas turbine cycle and a vapor power cycle. These two power cycles are interconnected in such a way that the heat transfer to the vapor cycle is supplied by the gas turbine cycle, which is often referred to as the topping cycle.

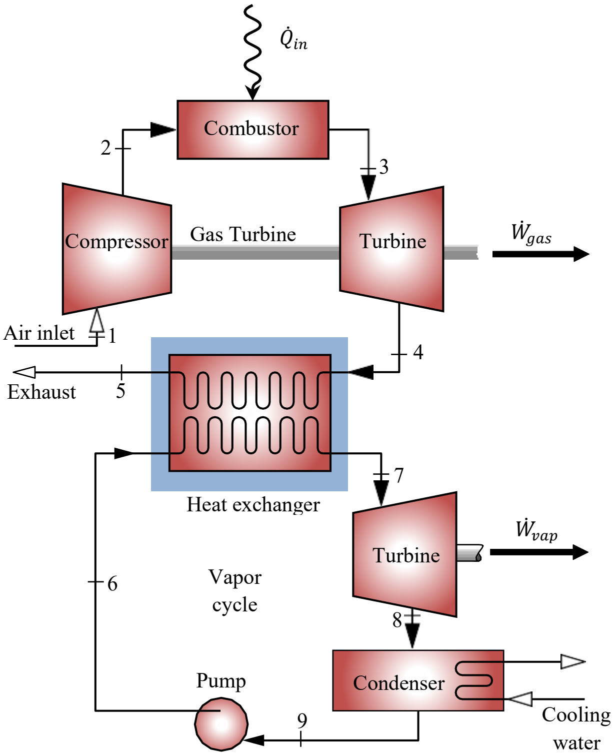


Figure 2.15 Combined gas turbine–vapor power plant

The combined cycle leverages the gas turbine's elevated average heat addition temperature and the vapor cycle's reduced average heat rejection temperature, resulting in a thermal efficiency that surpasses what either cycle could achieve independently. Combined cycles have demonstrated economic feasibility across various applications and are being adopted on a global scale for electric power generation.

With reference to Figure 2.15, the thermal efficiency of the combined cycle is

$$\eta_{th} = \frac{\dot{W}_{gas} + \dot{W}_{Vap}}{\dot{Q}_{in}}$$

The net power developed by the gas turbine is denoted as $\dot{W}_{gas}$, and the net power developed by the vapor cycle is denoted as $\dot{W}_{Vap}$. The symbol represents the total rate of heat transfer to the combined cycle, which includes any additional heat transfer required to superheat the vapor entering the vapor turbine. The evaluation of the quantities mentioned in Equation of the thermal efficiency of the combined cycle, follows the procedures outlined in the sections dedicated to vapor cycles and gas turbines.

The equation governing the energy transferred from the gas cycle to the vapor cycle in the system depicted in Figure 2.15 is derived by applying mass and energy rate balances to a control volume encompassing the heat exchanger. Under the assumptions of steady-state operation, negligible heat transfer with the surroundings, and negligible changes in kinetic and potential energy, the resulting equation is:

$$\dot{m}_v(h_7 - h_6) = \dot{m}_g(h_4 - h_5)$$

where $\dot{m}_g$ and $\dot{m}_v$ are the mass flow rates of the gas and vapor, respectively.

2.4 Hybrid installations (solar-gas)

An Integrated Solar Combined Cycle Power Plant (ISCCS) serves as the foundation for a conventional solar power plant, comprising a solar steam production system that supplies steam to a steam turbine. The power plant operates reliably during nighttime and periods of inadequate solar radiation (such as cloudy days) by utilizing a fossil fuel-based system (natural gas), resulting in the conversion of the power plant into a Combined Cycle (CC) configuration.

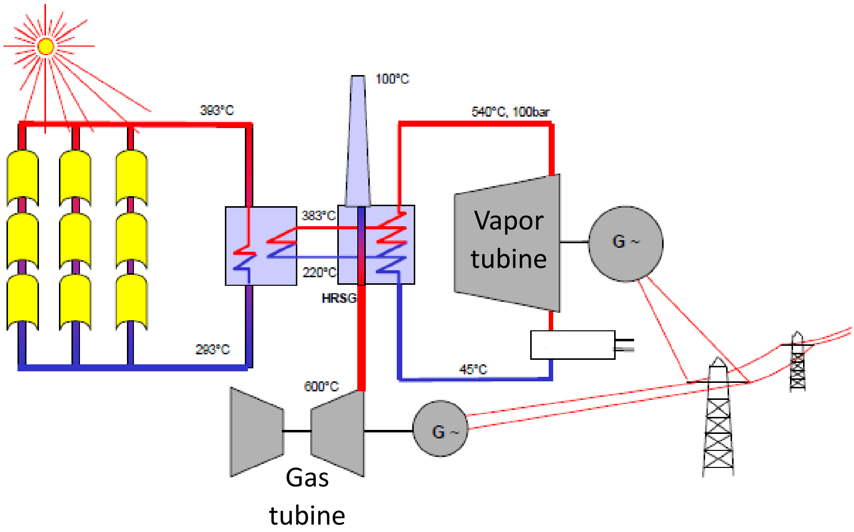


Figure 2.16 Simplified diagram of ISCCS

2.5 Cogeneration

Cogeneration is a process that aims to maximize the utilization of heat energy by producing both useful work and process heat from the same energy source. Traditional energy conversion cycles, such as steam-turbine and gas-turbine cycles, primarily focus on converting heat into work, while the remaining low-grade heat is wasted. However, many industries require process heat in addition to work, and wasting heat is inefficient.

Process heat is crucial in industries like chemical, pulp and paper, oil production and refining, steel making, food processing, and textile industries. Typically, steam is generated through the combustion of fuels like coal, oil, or natural gas in a furnace, and it is supplied to these industries at high pressure and temperature.

In a process-heating plant, all the heat transferred to the steam in the boiler is utilized for process heating. However, from a second-law perspective, the high-quality energy from the furnace is converted to lower-quality steam, resulting in a loss of work potential. It is more efficient to use low-quality energy for tasks that don't require high-quality energy.

Industries that consume large amounts of process heat also have significant electricity demands. It is both economically and technically beneficial to harness the existing work potential to generate electricity instead of wasting it. A cogeneration plant is designed to meet the process-heat requirements of industrial processes while simultaneously producing electricity. Cogeneration involves the production of multiple forms of useful energy from the same energy source, such as process heat and electric power.

A cogeneration plant can use different power cycles, including steam-turbine (Rankine) cycle, gas-turbine (Brayton) cycle, or a combined cycle. In an ideal steam-turbine cogeneration plant, the heat demand for process heat $\dot{Q}_P$ is met by expanding steam in a turbine to generate power. The steam flow rate is adjusted so that saturated liquid steam leaves the process-heating section. The steam is then pumped back to the boiler, where it is heated. The absence of a condenser in this plant allows for the utilization of all the energy transferred to the steam as process heat or electric power. The utilization factor ($\in_u$) of a cogeneration plant represents the ratio of total useful energy output (process heat and electric power) to the energy input (heat supplied to the boiler). A higher utilization factor indicates a more efficient utilization of energy without waste heat

$$\in_u = \frac{Net\ work\ output + \ Process\ heat\ delivered}{Total\ heat\ input} = \frac{\dot{W}_{net} + \dot{Q}_P}{\dot{Q}_{in}}$$

$$\in_u = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}$$

In technical terms, the variable Q represents the heat that is rejected in the condenser. Strictly speaking, Q 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 based on the heating value of the fuel. The utilization factor of the ideal steam-turbine cogeneration plant is obviously 100%. Actual cogeneration plants have utilization factors as high as 80%. 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 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%.

It is important to note that, without the turbine, the heat supply to the steam in the boiler would only be 100 kW instead of 120 kW. The additional 20 kW of heat supplied is converted into work. Therefore, a cogeneration power plant can be considered as a combination of a process-heating plant and a power plant with a thermal efficiency of 100%.

The ideal steam-turbine cogeneration plant described above is not practical because it cannot adapt to variations in power and process-heat loads. A more practical (yet more complex) cogeneration plant is shown in Figure 2.17. During normal operation, some steam is extracted from the turbine at a predetermined intermediate pressure P_6. The remaining 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.

During times of high demand for process heat, all the steam is directed to the process-heating units and none to the condenser ($\dot{m}_7 = 0$). In this mode, there is no waste heat. If this is insufficient, some steam leaving the boiler is throttled by an expansion or pressure-reducing valve (*PRV*) to the extraction pressure P6 and is sent to the process-heating unit. Maximum process heating is achieved when all the steam leaving the boiler passes through the *PRV* ($\dot{m}_5 = \dot{m}_4$). 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 a conventional steam power plant. The rates of heat input, heat rejection, process heat supply, and power production for this cogeneration plant can be expressed as follows:

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

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

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

$$\dot{W}_{Turb} = (\dot{m}_4 - \dot{m}_5)(\dot{h}_4 - h_6) + m_7(h_6 - h_7)$$

Under optimal conditions, a cogeneration plant aims to replicate the ideal cogeneration plant discussed earlier. This means that all the steam expands in the turbine to the extraction pressure and then continues to the process-heating unit. No steam passes through the pressure-reducing valve (*PRV*) or the condenser, resulting in no waste heat being rejected ($\dot{m}_4 = \dot{m}_6$ and $\dot{m}_5 = \dot{m}_7 = 0$). However, achieving this condition in practice can be challenging due to the constant variations in process-heat and power loads. Nevertheless, the plant should be designed in a way that approximates the optimal operating conditions for the majority of the time.

The utilization of cogeneration dates back to the early 20th century when power plants were integrated into communities to provide district heating, including space heating, hot water, and process heating for residential and commercial buildings. However, district heating systems lost popularity in the 1940s due to low fuel prices. Nevertheless, the significant increase in fuel prices during the 1970s led to renewed interest in district heating.

Cogeneration plants have proven to be economically attractive, leading to a growing number of installations in recent years with ongoing installations.

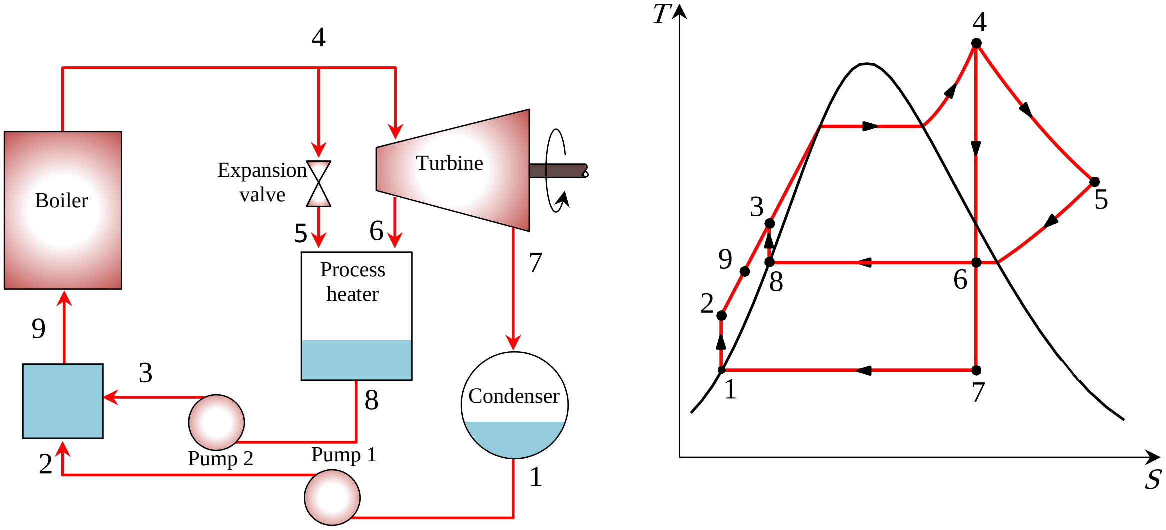


Figure 2.17 A cogeneration plant with adjustable loads

2.6 Kind of Nuclear Power System Cycle

In nuclear plant the origin of the energy is a controlled nuclear reaction taking place in an isolated reactor building. Pressurized water, a liquid metal, or a gas such as helium can be used to transfer energy released in the nuclear reaction to the working fluid in specially designed heat exchangers.

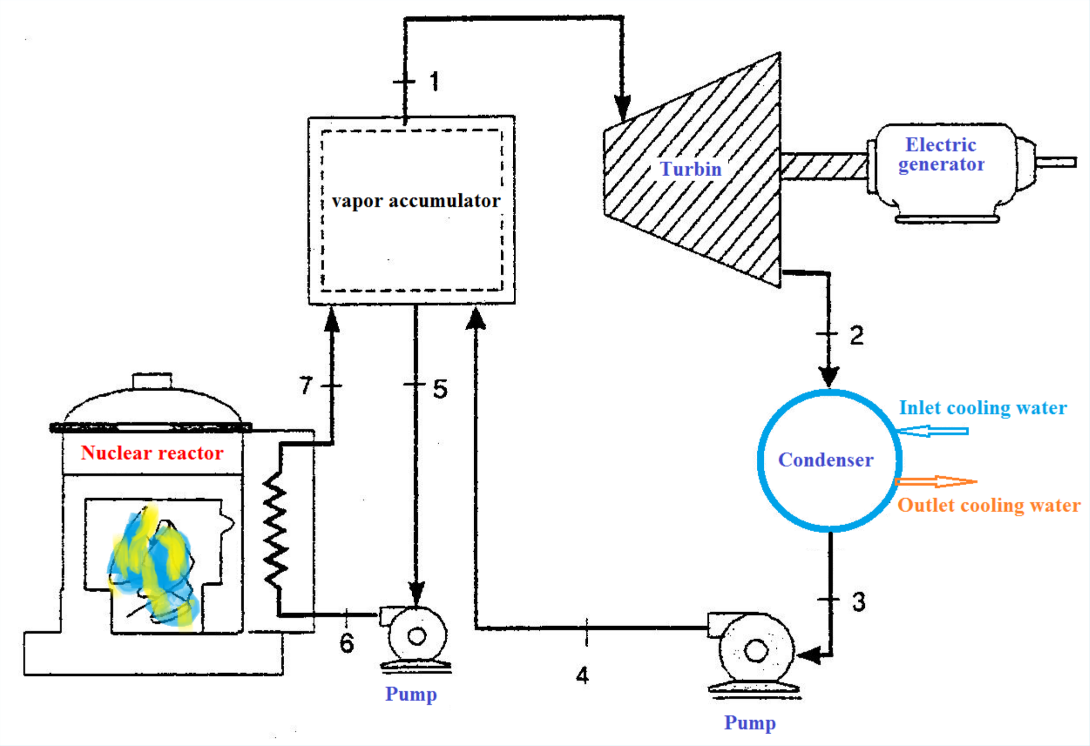


Figure 2.18 Monoframe nuclear power cycle.

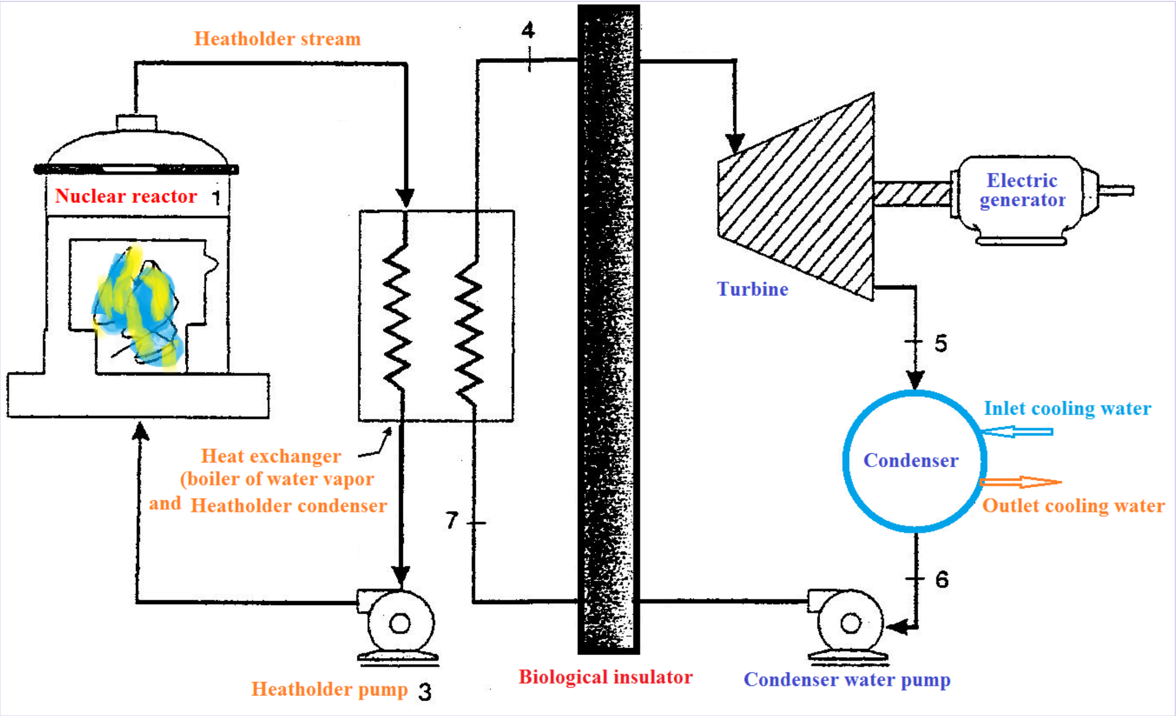


Figure 2.19 Dual frame nuclear power cycle.

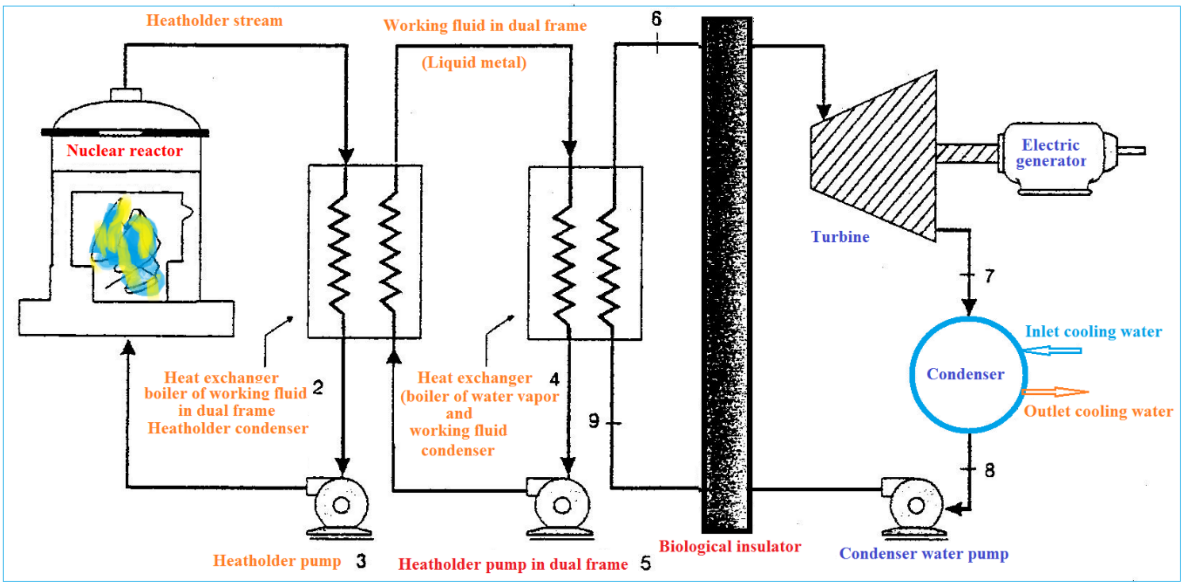


Figure 2.20 Triframe nuclear power cycle.

Chapter 3: Exergy and exergy analysis of thermodynamic systems

1.1 Defining Exergy

The concept of exergy is rooted in the introduction to the second law of thermodynamics. It is established that work can be generated when two systems at different states are brought into contact and allowed to reach equilibrium. The maximum theoretical work that can be obtained during this interaction is referred to as exergy.

To fully define exergy, it is necessary to define the reference environment and explain how numerical values for exergy can be determined. These tasks are interconnected because the numerical value of exergy relies on both the state of the system of interest and the condition of the environment.

1.2.1 Exergy Reference Environment

The exergy reference environment is an essential aspect when calculating exergy. In any system, whether it is a component within a larger system (like a steam turbine in a power plant) or the entire system itself (such as a power plant), it operates within its surroundings. It is important to distinguish between the environment used for calculating exergy and a system’s surroundings.

In strict terms, surroundings refer to everything outside the system. However, when considering the exergy concept, we make a distinction between the immediate surroundings, where intensive properties may change during interactions with the system, and the larger portion of the surroundings at a distance. This larger portion remains unaffected by any processes involving the system and its immediate surroundings. We refer to this larger portion as the environment.

1.2.2 Gas-fired power plants.

The Carnot, Ericsson, and Stirling cycles in their ideal forms are considered fully reversible, meaning they do not encounter any processes that lead to irreversibilities. On the other hand, the ideal Otto, Diesel, and Brayton cycles are only internally reversible, and they may involve irreversibilities that occur outside the system. By conducting a second-law analysis of these cycles, it is possible to identify the points in the system where the largest irreversibilities occur and determine areas for potential improvements.

The exergy destruction in a closed system can be mathematically represented as :

$$X_{dest} = T_0\left(\Delta S_{sys} - S_{in} - S_{out}\right) = T_0\left[(S_2 - S_1)_{sys} - \frac{Q_{in}}{T_{b,in}} - \frac{Q_{out}}{T_{b,out}}\right] \qquad kJ$$

where $T_{b,in}$ and $T_{b,out}$ are the temperatures of the system boundary where heat is transferred into and out of the system, respectively. A similar relation for steady-flow systems can be expressed, in rate form, as

$$\dot{X}_{dest} = T_0\dot{S}_{gen} = T_0\left(\dot{S}_{out} - \dot{S}_{in}\right) = T_0\left[\sum_{out}\dot{m}s - \sum_{in}\dot{m}s - \frac{Q_{in}}{T_{b,in}} - \frac{Q_{out}}{T_{b,out}}\right] \qquad kW$$

or, on a unit–mass basis for a one-inlet, one-exit steady-flow device, as

$$X_{dest} = T_0 S_{gen} = T_0\left(S_e - S_i - \frac{q_{in}}{T_{b,in}} - \frac{q_{out}}{T_{b,out}}\right) \qquad kJ/kg$$

The subscripts "i" and "e" are used to indicate the inlet and exit states, respectively.

The exergy destruction of a cycle is the cumulative sum of the exergy destructions of the individual processes that constitute that cycle. Alternatively, the exergy destruction of a cycle can be determined without tracking the individual processes by considering the entire cycle as a single process and applying one of the aforementioned equations. Entropy is a characteristic property that solely depends on the state of the system. In the case of a cycle, whether it is reversible or actual, the initial and final states are identical, resulting in se = si. Consequently, the exergy destruction of a cycle is influenced by the amount of heat transfer occurring with the high- and low-temperature reservoirs involved, as well as their respective temperatures. It can be expressed on a per unit mass basis as :

$$x_{dest} T_0\left(\sum_{out}\frac{q_{out}}{T_{b,out}} - \sum_{in}\frac{q_{in}}{T_{b,in}}\right) \qquad kJ/kg$$

For a cycle that involves heat transfer only with a source at T_H and a sink at T_C, the exergy destruction becomes

$$x_{dest} T_0\left(\frac{q_{out}}{T_C} - \frac{q_{in}}{T_H}\right) \qquad kJ/kg$$

The exergies of a closed system f and a fluid stream c at any state can be determined from

$$\phi = (u-u_0) - T_0(s-s_0) + P_0(v-v_0) + \frac{V^2}{2} + gz \qquad kJ/kg$$

And

$$\psi = (h-h_0) - T_0(s-s_0) + \frac{V^2}{2} + gz \qquad kJ/kg$$

where subscript “0” denotes the state of the surroundings.

1.2.2 Vapor-fired power plants.

The ideal Carnot cycle is characterized by complete reversibility, meaning it does not encounter any processes that lead to irreversibilities. On the other hand, the ideal Rankine cycles, including simple, reheat, or regenerative configurations, are only internally reversible and may involve irreversibilities external to the system, such as heat transfer occurring across a finite temperature difference. Analyzing these cycles from a second-law perspective allows us to identify the locations of the most significant irreversibilities and determine their magnitudes.

Chapter 8 of the technical literature presents the development of equations that describe exergy and exergy destruction for steady-flow systems. The exergy destruction for a steady-flow system can be mathematically expressed, in the form of rates, as

$$\dot{X}_{dest} = T_0\dot{S}_{gen} = T_0\left(\dot{S}_{out} - \dot{S}_{in}\right) = T_0\left[\sum_{out}\dot{m}s + \frac{\dot{Q_{in}}}{T_{b,in}} - \sum_{in}\dot{m}s + \frac{Q_{out}}{T_{b,out}}\right] \qquad kW$$

or on a unit mass basis for a one-inlet, one-exit, steady-flow device as

$$X_{dest} = T_0S_{gen} = T_0\left(S_e - S_i - \frac{q_{in}}{T_{b,in}} - \frac{q_{out}}{T_{b,out}}\right) \qquad kJ/kg$$

The subscripts "i" and "e" are used to indicate the inlet and exit states, respectively.

The exergy destruction associated with a *cycle* depends on the magnitude of the heat transfer with the high- and low-temperature reservoirs involved, and their temperatures. It can be expressed on a unit mass basis as:

$$x_{dest}T_0\left(\sum_{out}\frac{q_{out}}{T_{b,out}}-\sum_{\imath n}\frac{q_{\imath n}}{T_{b,\imath n}}\right) \qquad kJ/kg$$

For a cycle that involves heat transfer only with a source at T_H and a sink at T_C, the exergy destruction becomes

$$x_{dest}T_0\left(\frac{q_{out}}{T_C}-\frac{q_{in}}{T_H}\right) \qquad kJ/kg$$

The exergies of a closed system f and a fluid stream c at any state can be determined from

$$\phi=(u-u_0)-T_0(s-s_0)+P_0(v-v_0)+\frac{V^2}{2}+gz \qquad kJ/kg$$

And

$$\psi=(h-h_0)-T_0(s-s_0)+\frac{V^2}{2}+gz \qquad kJ/kg$$

where subscript “0” denotes the state of the surroundings.

Chapter 4. Thermodynamics of Combustion

4.1. Mixture Properties

The thermal characteristics of a pure substance are quantified by various parameters, such as internal energy (u), enthalpy (h), specific heat (cp), and others. Combustion systems encompass a variety of gases, and thus, the thermodynamic properties of a mixture arise from the collective behavior of all individual gas species. For gaseous mixtures, the ideal gas law is employed, enabling the application of ideal gas relations to each gas component. Assuming a mixture composed of K distinct gases, the total mass (m) of the system is considered.

$$m = \sum_{i=1}^{k} m_i$$

where mi is the mass of species i. The total number of moles in the system, N, is

$$N = \sum_{i=1}^{k} N_i$$

where Ni is the number of moles of species i in the system. Mass fraction, yi, and mole fraction, xi, describe the relative amount of a given species. Their definitions are given by

$$y_i = \frac{m_i}{m} \quad and \qquad x_i = \frac{N_i}{N}$$

where *i* =1,2,...,K. By definition,

$$\sum_{i=1}^{k} y_i = 1 \qquad and \qquad \sum_{i=1}^{k} x_i$$

With Mi denoting the molecular mass of species i, the average molecular mass, M, of the mixture is determined by

$$M = \frac{m}{N} = \frac{\sum_i N_i M_i}{N} = \sum_i x_i M_i$$

From Dalton's law of additive pressures and Amagat's law of additive volumes along with the ideal gas law, the mole fraction of a species in a mixture can be found from the partial pressure of that species as

$$\frac{P_i}{P}=\frac{N_i}{N}=\frac{V_i}{V}=x_i$$

where P_i is the partial pressure of species *i*, *P* is the total pressure of the gaseous mixture, V_i the partial volume of species *i*, and *V* is the total volume of the mixture. The average intrinsic properties of a mixture can be classified using either a molar base or a mass base. For instance, the internal energy per unit mass of a mixture, *u*, is determined by summing the internal energy per unit mass for each species weighted by the mass fraction of the species.

$$u=\frac{U}{m}=\frac{\sum_i m_i u_i}{m}\sum_i y_i u_i$$

where *U* is the total internal energy of the mixture and ui is the internal energy per mass of species *i*. Similarly, enthalpy per unit mass of mixture is

$$h=\sum_i y_i h_i$$

and specific heat at constant pressure per unit mass of mixture is

$$C_p=\sum_i y_i C_{p,i}$$

A molar base property, often denoted with a ^ over bar, is determined by the sum of the species property per mole for each species weighted by the species mole fraction, such as internal energy per mole of mixture

$$\hat{u}=\sum_i x_i\,\hat{u}_i$$

enthalpy per mole of mixture

$$\hat{h}=\sum_i x_i\,\hat{h}_i$$

and entropy per mole of mixture

$$\hat{s} = \sum_i x_i \, \hat{s}_i$$

Assuming constant specific heats during a thermodynamic process, changes of energy, enthalpy, and entropy of an individual species per unit mass are described as follows:

$$\Delta u_i = C_{v,i}(T_2 - T_1)$$

$$\Delta h_i = C_{p,i}(T_2 - T_1)$$

$$\Delta s_i = C_{p,i} ln \frac{T_2}{T_1} - R_i ln \frac{P_{i,2}}{P_{i,1}}$$

$P_{i,2}$ and $P_{i,1}$ denote the partial pressures of species i at state 1 and state 2, respectively. R_i is the gas constant for species $i \left(R_i = \frac{\hat{R}_u}{M_i} = universal\ gas\ constant/molecular\ mass\ of\ species\ i\right)$. The overall change of entropy for a combustion system is

$$\Delta S = \sum_i m_i \Delta s_i$$

A summary of the thermodynamic properties of mixtures is provided at the end of the chapter.

4.2 Combustion Stoichiometry

For a given combustion device, such as a piston engine, determining the appropriate amounts of fuel and air to be injected for complete combustion can be achieved by balancing the combustion reaction equation for the specific fuel.

A stoichiometric mixture contains the precise quantities of fuel and oxidizer such that, after combustion is complete, all the fuel and oxidizer are consumed to form the products. This ideal mixture approximately yields the maximum flame temperature, as all the energy released from combustion is used to heat the products.

For example, the following reaction equation can be written to balance the combustion of methane and air :

$$CH_4+?\left(O_2+\frac{79}{21}N_2\right)\rightarrow ?\,CO_2+?\,H_2O+?\,N_2$$

Assuming air composed of 21% oxygen (O_2) and 79% nitrogen (N_2), the coefficients associated with each species in the combustion reaction equation are initially unknown. By balancing the atomic abundance on both the reactant and product sides, the coefficient for each species can be determined. For example, to find the coefficient for carbon dioxide (CO_2), we observe that on the reactant side there is 1 mole of carbon (C) atoms. Therefore, the product side must also have 1 mole of C atoms. Thus, the coefficient of CO_2 is unity.

Applying this procedure, the balanced stoichiometric reaction equation for the complete combustion of methane (CH_4) with air can be written as:

$$CH_4\ +\ 2(O_2\ +\ 3{,}76\,N_2) -> \ CO_2\ +\ 2\,H_2O\ +\ 7{,}52\,N_2$$

Note that on the reactant side there are 2·(1 + 3.76) or 9.52 mol of air and its molecular mass is 28.96 kg/kmol. In this text, the reactions are balanced using 1 mol of fuel. This is done here to simplify the calculations of the heat of reaction and flame temperature later in the chapter. Combustion stoichiometry for a general hydrocarbon fuel, $C_\alpha H_\beta O_\gamma$, with air can be expressed as :

$$C_\alpha H_\beta O_\gamma+\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)(O_2+3{,}76N_2)\rightarrow \alpha CO_2+\frac{\beta}{2}H_2O+3{,}76\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)N_2$$

The stoichiometric or theoretical air is the precise amount of air needed to combust a stoichiometric mixture. However, the formula provided is applicable only to fuels composed of a single component. When dealing with fuels comprising multiple components, there are two common approaches. To illustrate, consider a fuel mixture consisting of 95% methane and 5% hydrogen. The first approach involves determining the combustion stoichiometry by employing the fundamental principle of atomic balance, which ensures an equal number of each atom type (C, H, N, O) in both the reactants and the products.

$$0{,}95CH_4+0{,}05H_2+1{,}925(O_2+3{,}76N_2)\rightarrow 0{,}95CO_2+1{,}95H_2O+7{,}238N_2$$

Another method for balancing a fuel mixture involves initially establishing stoichiometry relations specifically for CH_4 (methane) and H_2 (hydrogen) individually.

$$CH_4 + 2(O_2 + 3{,}76N_2) \rightarrow CO_2 + 2H_2O + 2 \times 3{,}76N_2$$
$$H_2 + 0{,}5(O_2 + 3{,}76N_2) \rightarrow H_2O + 0{,}5 \times 3{,}76N_2$$

To account for a fuel mixture composed of multiple components, the individual stoichiometry equations need to be multiplied by the mole fractions of each fuel component. Afterwards, these equations can be added together.

$$0{,}95\{CH_4 + 2(O_2 + 3{,}76N_2) \rightarrow CO_2 + 2H_2O + 2 \times 3{,}76N_2\}$$
$$0{,}05\{H_2 + 0{,}5(O_2 + 3{,}76N_2) \rightarrow H_2O + 0{,}5 \times 3{,}76N_2\}$$
$$\Longrightarrow 0{,}95CH_4 + 0{,}05H_2 + 1{,}95(O_2 + 3{,}76N_2) \rightarrow 0{,}95CO_2 + 1{,}95H_2O + 7{,}238N_2$$

4.1.1 Stoichiometric ratio

Stoichiometry in combustion refers to the ideal composition of the air-fuel mixture required for complete combustion. the stoichiometric air-fuel ratio, denoted as *AFR*, represents the ratio of the masses of air (m_a) and fuel (m_f) present under stoichiometric conditions. This ratio ensures that all the fuel is burned completely, resulting in the most efficient combustion process.

$$AFR = \frac{m_a}{m_f}$$

The inverse ratio is called fuel-air ratio (*FA* or *FAR*) and it’s calculated as:

$$FAR = \frac{1}{AFR} = \frac{m_f}{m_a}$$

In technical terms, the given equation (2.13) is modified for a stoichiometric mixture, where mf and ma represent the masses of the fuel and air, respectively.

$$f_S = \left.\frac{m_f}{m_a}\right|_{stoichiometric} = \frac{M_f}{\left(\alpha + \frac{\beta}{4} + \frac{\gamma}{2}\right).4{,}76.M_{air}}$$

In the table below we can see the stoichiometric air fuel ratio for several fossil fuels.

Fuel	**Chemical formula**	***AFR***
Methanol	CH_3OH	6.47:1
Ethanol	C_2H_5OH	9:1
Butanol	C_4H_9OH	11.2:1
Diesel	$C_{12}H_{23}$	14.5:1
Gasoline	C_8H_{18}	14.7:1
Propane	C_3H_8	15.67:1
Methane	CH_4	17.19:1
Hydrogen	H_2	34.3:1

Table 4.1

where Mf and Mair (~28.84 kg/kmol) are the average masses per mole of fuel and air, respectively. The fuel-air ratio, denoted as *f*, is a value that ranges between zero and 1. In the case of most hydrocarbon fuels, the stoichiometric fuel-air ratio, fs, falls within the range of 0.05 to 0.07. The air-fuel ratio (*AFR*) is another way to describe a combustible mixture and is simply the reciprocal of the fuel-air ratio (*AFR* = 1/*f*).

4.1.2 Equivalence Ratio

The equivalence ratio, denoted as ϕ (phi), is obtained by normalizing the actual fuel-air ratio by the stoichiometric fuel-air ratio.

$$\phi = \frac{f}{f_S} = \frac{m_{as}}{m_a} = \frac{N_{as}}{N_a} = \frac{N_{O2s}}{N_{O2,a}}$$

The subscript *'s'* indicates a value at the stoichiometric condition. When the equivalence ratio $\phi < 1$, it represents a lean mixture. A stoichiometric mixture is indicated when $\phi = 1$, and a rich mixture is denoted when $\phi > 1$. Similar to f, the range of the equivalence ratio is bounded by zero and ∞, corresponding to the limits of pure air and pure fuel, respectively. It's important to note that the equivalence ratio is a normalized quantity that provides information about the composition of the combustion mixture. Combustion engineers often use an alternative variable called lambda (λ), which is based on the air-fuel ratio (*AFR*). Lambda is defined as the ratio of the actual air-fuel ratio to the stoichiometric air-fuel ratio.

$$\lambda = \frac{AFR}{AFR_S} = \frac{1/f}{1/f_s} = \frac{1}{f/f_S} = \frac{1}{\phi}$$

For stoichiometric mixtures $\lambda = 1.0$.

For rich mixtures, $\lambda < 1.0$;

For lean mixtures, $\lambda > 1.0$.

The percent excess air (%EA) is a measure of the amount of air present in excess of the stoichiometric amount. It is calculated as follows:

$$\%EA = 100\frac{m_a - m_{as}}{m_{as}} = 100\left(\frac{m_a}{m_{as}} - 1\right)$$

For example, a mixture with %EA =50 contains 150% of the theoretical (stoichio- metric) amount of air.

Converting between quantification methods: The values of *f*, *AFR*, and *%EA* can be interrelated, and given one of these variables, the other two can be deduced. Table 4.2 provides a summary of these relationships, along with their corresponding graphical representations.

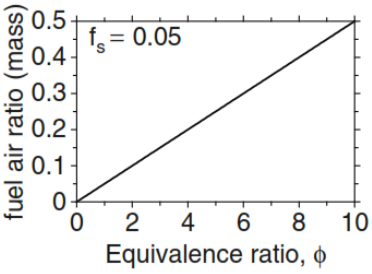
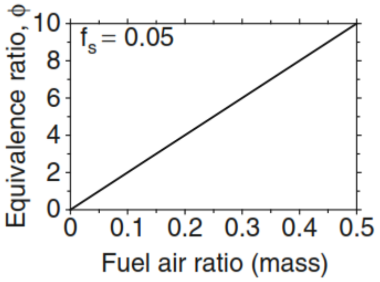
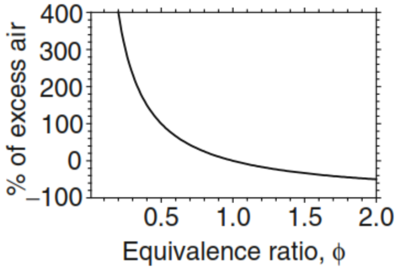
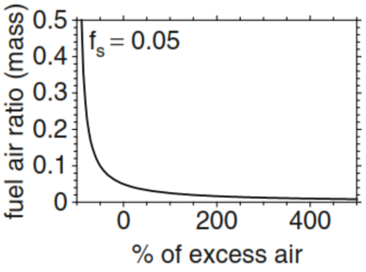
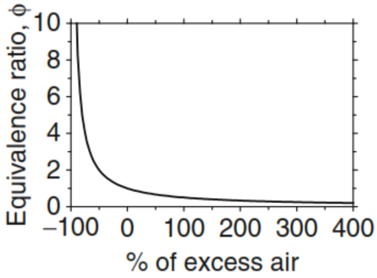
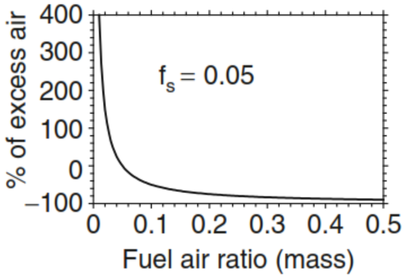
f (fuel air ratio bay mass)	ϕ (equivalence ratio)	$\%EA$ ((% of excess air)
$f = f_s . \emptyset$ $f = \dfrac{100. f_s}{\%EA + 100}$	$\emptyset = \dfrac{f}{f_s}$ $\emptyset = \dfrac{100}{\%EA + 100}$	$\%EA = 100 \dfrac{1 - \emptyset}{\emptyset}$ $\%EA = 100 \dfrac{1 - f/f_s}{f/f_s}$
		
		

Table 4.2 Relations among the three variables for describing reacting mixtures

In general, during combustion, the products consist of various species beyond the major ones such as CO_2, H_2O, N_2, and O_2. Achieving a balanced stoichiometric equation requires considering thermodynamic equilibrium relations. However, when assuming complete combustion and the presence of excess air, the global equation for lean combustion (where $\phi \leq 1$) can be represented as follows:

$$C_\alpha H_\beta O_\gamma + \frac{1}{\emptyset}\left(\alpha + \frac{\beta}{4} - \frac{\gamma}{2}\right)(O_2 + 3{,}76 N_2)$$
$$\rightarrow \alpha CO_2 + \frac{\beta}{2} H_2O + \frac{3{,}76}{\emptyset}\left(\alpha + \frac{\beta}{4} - \frac{\gamma}{2}\right) N_2 + \left(\alpha + \frac{\beta}{4} - \frac{\gamma}{2}\right)\left(\frac{1}{\emptyset} - 1\right) O_2$$

In terms of $\%EA$, we replace $\emptyset = \frac{100}{\%EA+100}$ and the result is

$$C_\alpha H_\beta O_\gamma + \left(\frac{\%EA}{100}+1\right)\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)(O_2+3{,}76N_2)$$
$$\rightarrow \alpha CO_2 + \frac{\beta}{2}H_2O + 3{,}76.\left(\frac{\%EA}{100}+1\right)\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)N_2 + \left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)\frac{\%EA}{100}O_2$$

The amount of excess air can be determined from measurements of exhaust gases. One approach is to calculate the ratio of mole fractions between CO2 and O2, which can provide an indication of the excess air level.

$$\frac{x_{CO_2}}{x_{O_2}} = \frac{\alpha}{\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)\frac{\%EA}{100}} \rightarrow \frac{\%EA}{100} = \frac{\alpha}{\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)\frac{x_{CO_2}}{x_{O_2}}}$$

Or using table 4.2

$$\phi = \frac{100}{\%EA+100} \rightarrow \phi = \frac{1}{1+\frac{\alpha}{\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)\frac{x_{CO_2}}{x_{O_2}}}}$$

For rich combustion $\phi > 1$, the products of combustion may include CO, unburned fuels, and other species formed through the degradation of the fuel. Achieving a complete balance of the chemical reaction often requires additional information about the products. However, if we assume that the products contain only unburned fuel and major combustion products, the corresponding global equation can be written as follows :

$$C_\alpha H_\beta O_\gamma + \frac{1}{\emptyset}\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)(O_2+3{,}76N_2)$$
$$\rightarrow \frac{\alpha}{\emptyset}CO_2 + \frac{\beta}{2\emptyset}H_2O + \frac{3{,}76}{\emptyset}\left(\alpha+\frac{\beta}{4}-\frac{\gamma}{2}\right)N_2 + \left(1-\frac{1}{\emptyset}\right)C_\alpha H_\beta O_\gamma$$

4.2 Heat of formation and calorific powers

The heating values of a fuel, typically expressed in kJ/kg or MJ/kg, are commonly used to quantify the maximum amount of heat generated by combustion with air at standard conditions (25°C and 101.3 kPa). The amount of heat released during fuel combustion depends on the phase of water in the products. If water remains in the gaseous phase in the products, the total heat release value is known as the lower heating value (LHV). However, when water vapor condenses to a liquid phase, additional energy, equal to the latent heat of vaporization, can be extracted. In this case, the total energy release is referred to as the higher heating value (HHV). The LHV can be calculated from the HHV by subtracting the amount of energy released during the phase change of water from vapor to liquid.

$$LHV = HHV - \frac{N_{H_2O,P} M_{H_2O} h_{fg}}{N_{fuel} M_{fuel}} \qquad MJ/kg$$

Here, NH_2O,P represents the number of moles of water in the products. The latent heat for water at standard temperature and pressure (*STP*) is denoted as hfg and has a value of 2.44 MJ/kg or 43.92 MJ/kmol. In combustion literature, the lower heating value (*LHV*) is often referred to as the enthalpy or heat of combustion (Q_C) and is considered a positive quantity.

4.2.1 Determination of the higher heating value *HHV* for Combustion Processes at Constant Pressure

In the context of combustion processes at constant pressure, the determination of the higher heating value (*HHV*) is of interest. The *HHV* represents the total energy released during the combustion of a fuel, including the latent heat of vaporization when water vapor condenses to liquid water.

To determine the *HHV*, a control volume analysis is typically employed. The analysis involves considering a system where reactants, containing a known amount of fuel, enter a control volume at standard conditions. The products exit the system, and the heat release is maximized by cooling the products to the inlet temperature and condensing the water vapor to liquid water.

The conservation of energy principle is applied to the constant pressure reactor. The total enthalpy of the products (H_P) and the total enthalpy of the reactants (H_R) are considered in this analysis. By applying conservation of energy, the *HHV* can be calculated, taking into account the heat released during combustion and the latent heat of vaporization of water.

This determination of *HHV* at constant pressure allows for an understanding of the total energy released during the combustion process, providing valuable information for various applications in engineering and energy systems.

$$-Q_{rxn,p} = H_R - H_P$$

The negative value of Qrxn,p indicates that heat is being transferred out of the system to the surroundings. As mentioned earlier, the heating value of the fuel can be determined by calculating the difference in enthalpies between the reactants and the products. However, in combustion systems, evaluating enthalpies is not straightforward due to the presence of different species entering and exiting the system as a result of chemical reactions.

$Q_{rxn,p}$ is commonly known as the enthalpy of reaction or heat of reaction, with the subscript p indicating that the value was calculated at constant pressure. It is important to note that $Q_{rxn,p}$ is equal to the negative of the enthalpy of combustion (Q_C). In other words, $Q_{rxn,p} = -Q_C$, where Q_C represents the enthalpy of combustion.

4.3.1.1 Enthalpy of Formation

In combustion processes, reactants are consumed and products are formed, resulting in the release of energy. This energy is derived from the rearrangement of chemical bonds in the reactants to form the products. The standard enthalpy of formation, represented as ΔH°, quantifies the amount of chemical bond energy in a chemical species under standard conditions.

The enthalpy of formation of a substance is the energy required to form that substance from its constituent elements at standard temperature and pressure (*STP*) conditions, specifically at 25°C and 1 atmosphere of pressure. The molar base enthalpy of formation, ΔH°, is measured in units of MJ/kmol, while the mass base enthalpy of formation, ΔH°, is measured in units of MJ/kg. Elements in their most stable forms, such as C(graphite), H_2, O_2, and N2, have enthalpies of formation equal to zero. Table 4.3 provides tabulated values of enthalpies of formation for commonly encountered chemical species. When conditions deviate from standard conditions, there is a corresponding change in enthalpy. For thermodynamic systems without chemical reactions, the change in enthalpy of an ideal gas is described by the sensible enthalpy.

Species	$\Delta\hat{h}^0\ (MJ/kmol)$	Species	$\Delta\hat{h}^0\ (MJ/kmol)$
H_2O (g)	241.83	H	+217.99
CO_2	393.52	N	+472.79
CO	110.53	NO	+90.29
CH_4	74.87	NO2	+33.10
C_3H_8	104.71	O	+249.19
C_7H_{16} (g) (n-heptane)	224.23	OH	+39.46
C_8H_{18} (g) (isooctane)	259.25	C (g)	+715.00
CH_3OH (g) (methanol)	201.54	C_2H_2 (acetylene)	+226.73
CH_3OH (l) (methanol)	238.43	C_2H_4 (ethylene)	+52.28
C_2H_6O (g) (ethanol)	235.12	C_2H_6 (ethane)	84.68
C_2H_6O (l) (ethanol)	277.02	C_4H_{10} (n-butane)	126.15

Table 4.3 Enthalpy of formation of common combustion species

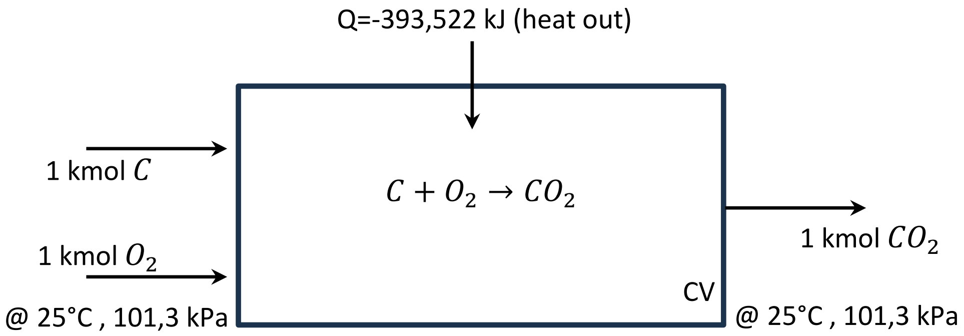


Figure. 4.1 Constant-pressure flow reactor for determining enthalpy of formation

$$\hat{h}_{si} = \int_{T_0}^{T} \hat{C}_P(T)dT$$

The subscript *i* represents a specific species, T_0 represents the standard temperature of 25°C, and the superscript ^ indicates that a quantity is measured per mole. It is important to note that at standard conditions, the sensible enthalpy of any species is zero. The "absolute" or "total" enthalpy, denoted as $\hat{h}_i$, is therefore calculated as the sum of the sensible enthalpy and the enthalpy of formation.

$$\hat{h}_i = \Delta\hat{h}_i^0 + \hat{h}_{si}$$

4.3.1.2 Evaluation of the Heat of Combustion from a Constant-Pressure Reactor

$$-Q_{rxn,p} = H_R - H_P$$

$$= \sum_i N_{i,R}\left(\Delta\hat{h}_{i,R}^0 + \hat{h}_{si,R}\right) - \sum_i N_{i,P}\left(\Delta\hat{h}_{i,P}^0 + \hat{h}_{si,P}\right) = \left[\sum_i N_{i,R}\Delta\hat{h}_{i,R}^0 - \sum_i N_{i,P}\Delta\hat{h}_{i,P}^0\right]$$

$$+ \sum_i N_{i,R}\hat{h}_{si,R} - \sum_i N_{i,P}\hat{h}_{si,P}$$

$$-Q_{rxn,p}^0 = \sum_i N_{i,R}\Delta\hat{h}_{i,R}^0 - \sum_i N_{i,P}\Delta\hat{h}_{i,P}^0$$

Typically, excess air is utilized in combustion tests to ensure complete combustion of the fuel. The amount of excess air employed does not impact the higher heating value (*HHV*) at standard temperature and pressure (*STP*). Unless the reactant mixtures are significantly diluted, the water present in the products at *STP* is usually in the liquid state.

Assuming that the water in the products is in the liquid phase, the *HHV* can be determined.

$$HHV = \frac{-Q_{rxn,p}^0}{N_{fuel}M_{fuel}}$$

The negative sign in front of $Q_{rxn,p}^0$ ensures that *HHV* is positive.

4.3.2 Determination of HHV for Combustion Processes from a Constant-Volume Reactor

A constant-volume reactor is often preferred over a constant-pressure reactor for experimental determination of the higher heating value (*HHV*) of a fuel. In a closed system, the conservation of energy principle can be applied, resulting in the following relationship:

$$-Q_{rxn,v} = U_R - U_P$$

Due to the combustion process, a similar approach is necessary to account for the change in chemical bond energies. The internal energy can be determined by considering its relationship to enthalpy. It is important to note that the relationship $h = u + pv$ is based on mass, whereas the corresponding molar-based relation is $\hat{h} = \hat{u} + \hat{R}_u T$. At standard temperature and pressure (*STP*), where $T = T_0 = 25°\text{C}$, the total internal energy of the reactants, denoted as U_R, within the closed system is evaluated.

$$U_R = H_R - PV = H_R - \sum_i N_{i,R}\hat{R}_u T_0 = \sum_i N_{i,R}\Delta\hat{h}^0_{i,R} - \sum_i N_{i,R}\hat{R}_u T_0$$

The determination of the total internal energy of the products follows a similar procedure.

$$U_P = \sum_i N_{i,P}\Delta\hat{h}^0_{i,P} - \sum_i N_{i,P}\hat{R}_u T_0$$

By utilizing the relationships between internal energy and enthalpy, we can express the heat release at constant volume in terms of enthalpies.

$$-Q^0_{rxn,v} = U_R - U_P = \sum_i N_{i,R}\Delta\hat{h}^0_{i,R} - \sum_i N_{i,R}\hat{R}_u T_0 - \left[\sum_i N_{i,P}\Delta\hat{h}^0_{i,P} - \sum_i N_{i,P}\hat{R}_u T_0\right]$$

$$= \sum_i N_{i,R}\Delta\hat{h}^0_{i,R} - \sum_i N_{i,P}\Delta\hat{h}^0_{i,P} + \left(\sum_i N_{i,P} - \sum_i N_{i,R}\right)\hat{R}_u T_0$$

Therefore, the higher heating value (HHV) for combustion processes is calculated as follows:

$$HHV = \frac{-Q^0_{rxn,v} - \left(\sum_i N_{i,P} - \sum_i N_{i,R}\right)\hat{R}_u T_0}{N_{fuel} M_{fuel}}$$

where N_{fuel} is the number of moles of fuel burned and M_{fuel} is the molar mass of the fuel. The negative sign in front of $Q^0_{rxn,v}$ is used to ensure that the higher heating value (*HHV*) is positive. For a fuel with a general chemical formula of $C_\alpha H_\beta O_\gamma$, the difference between $-Q_{rxn,v}$, and $-Q_{rxn,p}$ is calculated as follows:

$$\left(\sum_i N_{i,P} - \sum_i N_{i,R}\right)\hat{R}_u T_0 = \Delta N \hat{R}_u T_0 = \left(\frac{\beta}{4} + \frac{\gamma}{2} - 1\right)\hat{R}_u T_0$$

and is usually small in comparison to *HHV*; therefore normally no distinction is made between the heat of reaction at constant pressure or constant volume.

4.4 Adiabatic Flame Temperature for reactants at standard conditions

One of the key aspects of a combustion process is the maximum temperature that can be reached by the combustion products. This maximum temperature, also known as the adiabatic flame temperature, is attained when there are no heat losses to the surrounding environment and all of the released energy from combustion is utilized to heat the products. The following sections will outline the methodology employed to calculate this adiabatic flame temperature.

Method 1: Estimate based on average $\hat{C}_p$ values

$$T_P = T_0 + \frac{N_{fuel} M_{fuel} LHV + \sum_i N_{i,R} \hat{h}_{si,R}(T_R)}{\sum_i N_{i,P} \hat{C}_{pi}}$$

$$T_P \approx T_R + \frac{N_{fuel} M_{fuel} LHV}{\sum_i N_{i,P} \hat{C}_{pi}}$$

or if mixture is not stoichiometric: mass-base analysis using *LHV* and ∅

$$\emptyset \leq 1 T_P = T_R + \frac{f.LHV}{(1+f)\bar{C}_P} = T_R + \frac{\emptyset.f.LHV}{(1+\emptyset.f_s)\bar{C}_P}$$

$$\emptyset > 1 T_P = T_R + \frac{f_s.LHV}{(1+f)\bar{C}_P} = T_R + \frac{f_s.LHV}{(1+\emptyset.f_s)\bar{C}_P}$$

Method 2: Enthalpy Balance

$$H_P(T_P) = H_R(T_R)$$

$$H_P(T_P) = \sum_i N_{i,P} \hat{h}_{i,P} = \sum_i N_{i,P}\left[\Delta \hat{h}_{i,P}^0 + \hat{h}_{si,P}(T_P)\right]$$

Trial and error of T_P such that $H_P(T_P)$ matches $H_R(T_R)$

4.5 Chemical Kinetics of Combustion:

The chemical kinetics of combustion describes the transient states of the system during the combustion process and provides information about the rate at which species are consumed and produced, as well as the rate at which the heat of reaction is released.

4.5.1 Elementary Reactions:

A chemical reaction can be described by an overall stoichiometric relation as

$$C_\alpha H_\beta O_\gamma + \left(\alpha + \frac{\beta}{4} - \frac{\gamma}{2}\right)(O_2 + 3{,}76N_2) \rightarrow \alpha CO_2 + \frac{\beta}{2} H_2O + 3{,}76\left(\alpha + \frac{\beta}{4} - \frac{\gamma}{2}\right) N_2$$

However, the actual chemical kinetics involved in combustion rarely follow such a straightforward pattern. In the case of one of the simplest combustion systems, hydrogen with oxygen, the overall stoichiometric relation can be expressed as follows:

$$H_2 + \frac{1}{2} O_2 \rightarrow H_2O$$

In combustion, the chemical reaction between H2 and O2 does not occur directly. Instead, H2 and O2 do not react with each other simultaneously in a single molecular collision. The initiation of the chemical reaction involves the presence of third body molecules (denoted as 'M'), which interact with H2 and O2. This leads to the formation of highly reactive molecules called radicals. These radicals then react with H2 and O2, producing more radicals and resulting in the formation of a radical pool. The overall reaction described earlier is a global reaction, but combustion of hydrogen involves many elementary reactions, each involving only a few species.

The collection of elementary reactions that describe the overall combustion process is called a reaction or combustion mechanism. The complexity of the mechanism depends on the level of detail considered. For example, a detailed combustion mechanism for hydrogen-oxygen combustion may contain several species and reaction steps. Similarly, for hydrocarbon fuels, the mechanism becomes more complex due to the presence of isomers and intermediate species. The number of species and reaction steps can increase significantly with the size of the fuel molecule.

In combustion, there are four main types of elementary reactions that play important roles: chain initiation, chain branching, chain termination or recombination, and chain propagation. These reactions govern the progression of the combustion process.

$$H_2 + M \rightarrow H + H + M \text{ or } O_2 + M \rightarrow O + O + M$$

Fuel	Species
CH_4	53
C_2H_4	75
C_3H_8	176
$n\text{-}C_7H_{16}$	561
$i\text{-}C_8H_{18}$	857
$n\text{-}C_7H_{16}+i\text{-}C_8H_{18}$	1,033

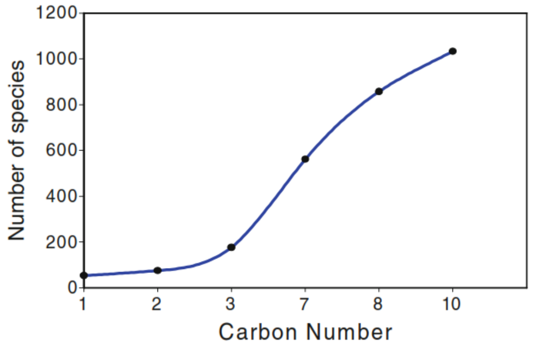


Figure 4.2 Left: typical numbers of species in detailed reaction mechanisms. Right: number of species increases rapidly with the total number of carbon elements in fuels

4.6 Elementary Reaction Rate

4.6.1 Forward Reaction Rate and Rate Constants

The chemical representation of an elementary reaction can be expressed using the following general equation.

$$aA + bB \rightarrow cC + dD$$

The stoichiometric coefficients a, b, c, and d represent the relative quantities of the reactants and products involved in the elementary reaction. Typically, these coefficients have values of one or two, as elementary reactions usually involve no more than two molecules. The rate of reaction progress is commonly described by the empirical form known as the law of mass action, which is expressed as follows:

Rate of reaction progress: $\dot{q}_{RxT} = k[A]^a[B]^b$

The law of mass action states that the rate of a chemical reaction is directly proportional to the concentration of the reactants. This relationship is mathematically represented by the Arrhenius rate constant, denoted as k, which takes the form:

$$k = A_0 exp\left(-\frac{E_a}{\hat{R}_u T}\right) = A_0 exp\left(-\frac{T_a}{T}\right)$$

In the equation, A_0 represents the pre-exponential factor, Ea denotes the activation energy, and $\hat{R}_u$ refers to the universal gas constant (with values of 1.987 cal/mol-K or 8.314 J/mol-K). The ratio $\frac{E_a}{\hat{R}_u}$ has the unit of temperature and is known as the activation temperature (T_a). The pre-exponential factor (A_0) indicates the frequency of collisions between reactant molecules, while the activation energy (E_a) represents the energy barrier required to break the chemical bonds of the molecules during a collision. The exponential term, $exp\left(-\frac{T_a}{T}\right)$, represents the probability of a successful collision leading to the formation of products.

4.7 Simplified Model of Combustion Chemistry

As previously mentioned, the chemical kinetics of practical higher hydrocarbon fuels are complex and require chemical mechanisms that involve hundreds or even thousands of chemical species. The number of species and reaction steps increases exponentially with the number of carbon atoms in the fuel. This makes it impractical for humans to fully comprehend the physical significance of such large mechanisms.

Computers can simulate detailed chemical kinetics in simplified reactors, but engineers often need to understand the behavior of practical, multi-dimensional systems. When coupling large-scale computational fluid dynamics simulations with chemical kinetics calculations, the computational requirements become demanding, especially when tracking hundreds of chemical species and their corresponding reactions at every point in the system.

Therefore, a simplified description of chemical kinetics is highly valuable for practical applications in combustion science and engineering. For simulations involving single-component fuels, a one-step global reaction is often used due to its simplicity.

4.7.1 Global *One-Step Reaction*

For a hydrocarbon fuel with a combustion stoichiometry represented by Equation 3.1, the global rate of progress can be described as follows:

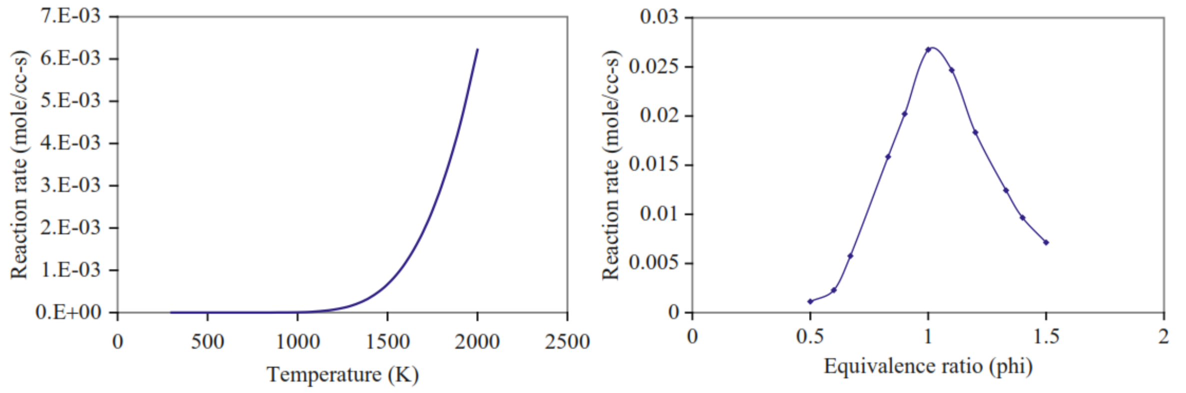


Figure 4.3 Reaction rate for methane as a function of temperature and equivalence ratio

$$\dot{q}_{RxT} = A_0 exp\left(-\frac{E_a}{\hat{R}_u T}\right)[Fuel]^a[O_2]^b$$

The pre-exponential factor, activation energy, and exponents a and b are determined through experimental measurements conducted in flow reactors, as shown in Table 4.2. The concentrations of the fuel and oxidizer are typically expressed in units of mol/cm3, resulting in the rate of progress being measured in mol/cm^3-s.

The pre-exponential factor, A0, is expressed in units of (mol/cm^3)$^{1-(a+b)}$ s^{-1}. The exponential term in the rate equation, which includes the activation energy, indicates that the rate of progress is highly dependent on temperature, as illustrated in Figure 4.3.

Due to this strong temperature dependence, the reaction rate can be significantly influenced by the equivalence ratio of the mixture, which is the ratio of the actual fuel-to-oxidizer ratio to the stoichiometric fuel-to-oxidizer ratio. This sensitivity to equivalence ratio arises from the resulting change in flame temperature, as shown in Figure 4.3.

The rates of fuel and oxygen consumption are affected by these factors.

$$\hat{\dot{r}}_{fuel} = \frac{d[Fuel]}{dt} = -\dot{q}_{RxT,}\ and \qquad \hat{\dot{r}}_{O_2} = \frac{d[O_2]}{dt} = \left(\alpha + \frac{\beta}{4} + \frac{\gamma}{2}\right)\dot{q}_{RxT}$$

The production rates of CO_2 and H_2O are

$$\dot{\hat{r}}_{CO_2} = \frac{d[CO_2]}{dt} = \alpha\dot{q}_{RxT,}\ \ and \qquad \dot{\hat{r}}_{H_2O} = \frac{d[H_2O]}{dt} = \left(\frac{\beta}{2}\right)\dot{q}_{RxT}$$

4.7.2 Pressure Dependence of Rate of Progress

In addition to being strongly temperature dependent, the rate of progress is also pressure dependent through the species concentration.

$$\dot{q}_{RxT} = A_0 exp\left(-\frac{E_a}{\hat{R}_u T}\right)[Fuel]^a[O_2]^b = A_0 exp\left(-\frac{E_a}{\hat{R}_u T}\right) x_{fuel}^a x_{O_2}^b \left(\frac{P}{\hat{R}_u T}\right)^{a+b} \propto P^{a+b}$$

4.7.3 Partial Equilibrium

Due to the challenges associated with measuring radicals at high temperatures (above 1,500 K), estimations of radical concentrations can be made by assuming that, although the combustion process is in a non-equilibrium state, a subset of the combustion reactions are in equilibrium. This condition is referred to as a partial equilibrium state of combustion.

The advantage of assuming partial equilibrium is that it reduces the number of intermediate reactions accordingly. By considering a subset of reactions to be in equilibrium, the complexity of the chemical system is reduced, making it more manageable for analysis and calculation purposes.

4.8 Ignition Phenomena

Ignition is the mechanism that triggers a vigorous combustion reaction and is characterized by a rapid increase in the temperature of the species involved. Understanding ignition is crucial in various combustion applications, ranging from the design of combustion devices to the prevention of unwanted fires. Ignition in combustion systems is often classified into two categories: spontaneous ignition and controlled ignition.

4.8.1 Autoignition and Spontaneous Ignition (Self-ignition, Spontaneous Ignition)

Spontaneous ignition, also known as autoignition, occurs through the self-heating of reactants (e.g., in diesel engines where spontaneous ignition is based on high pressure inside the combustion chamber).

4.8.2. Controlled Ignition

Controlled ignition occurs with the use of an ignition source. In this type of ignition, combustion is initiated by an external energy source that locally increases the temperature of the reactants until ignition occurs (for example in gasoline engines where a spark plug is used to ensure ignition).

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Annex 1

Exercises of chapter 1

Exercise 1.

A cylinder of an engine, operating according to an Otto cycle, has a diameter of 50mm and the piston stroke is equal to 80mm. The clearance volume is equal to 20cm^3.

Calculate the thermal efficiency of the cycle.

Exercise 2.

The temperature and pressure at the inlet of a diesel engine are respectively equal to 15 °C and 1 Bar. If the maximum temperature of the cycle is equal to 1000 °C and the compression ratio is equal to 14/1.

Calculate the thermal efficiency of the cycle.

Exercise 3.

An engine operating on a dual combustion cycle receives air at 1 Bar and 20 °C. The maximum pressure of the cycle is equal to 70 Bar, the maximum temperature is equal to 2000 °C, and the compression ratio is equal to 20/10.

Calculate the thermal efficiency of the cycle.

Exercise 4.

The air enters the turbine at 7 bar and 954 °C. The pressure at the turbine outlet is 1 bar.

Calculate the cycle efficiency and the work ratio.

Exercise 5.

The compression ratio in a gas turbine installation operating on the Joule (Brayton) cycle is 8, and the upper temperature of the cycle (combustion chamber outlet) is t3 = 700 °C, and the air flow rate is 20 kg/s at t1 = 20 °C.

The relative internal efficiencies of the turbine and compressor are 0.85 and 0.83, respectively. Cp = 1.005 kJ/kg·K and $\gamma = 1.4$ are constant throughout the cycle. The installation consists of a single turbine stage coupled to the compressor. Calculate:

1- The actual work produced by the turbine.

2- The actual work consumed by the compressor.

3- The actual useful work (cycle work).

4- The actual useful power output of the turbine (of this installation).

5- The amount of heat supplied to the working fluid during combustion.

6- The thermal efficiency of this cycle.

Exercises of chapter 2

Exercise 1.

A steam power generator operates between a boiler at 50.87 bar and a condenser at 0.04241 bar. Calculate the thermodynamic efficiency, work ratio, and specific steam consumption for:

a) A Carnot cycle entirely located in the two-phase region.

b) An ideal Rankine cycle with dry saturated steam at the turbine inlet.

For the same actual Rankine cycle, where the isentropic efficiency is 80%.

Exercise 2.

Consider the same conditions as in Exercise 1, which means the inlet pressure into the turbine is equal to 50.87 bar and the outlet pressure is 0.04241 bar. However, the steam is superheated to 400 °C before entering the turbine.

Calculate the thermodynamic efficiency, work ratio, and specific steam consumption.

Exercise 3.

A steam turbine power plant operates on a superheated Rankine cycle at T=450 °C and with extraction between the turbine inlet and outlet pressures P=8 MPa (from the boiler) and P=50 kPa (from the condenser).

A fraction of steam y is extracted at P=0.5 MPa and sent to a mixing heat exchanger. It is assumed that the fluid exiting the heat exchanger is in a saturated liquid state at the corresponding pressure. The second pump (main pump) compresses the water to the boiler pressure. The net power output of the power plant is 100 MW.

A. Fill in the entropy table.

B. Determine the fraction of extracted steam y.

C. Calculate the turbine work.

D. Calculate the thermal power released at the boiler.

E. Calculate the thermal power released at the condenser.

F. Calculate the turbine power.

G. Determine the thermal efficiency of the power plant.

H. Calculate the work ratio.

I. Calculate the specific steam consumption.

Exercises of chapter 3

Exercise 1.

The heat exchanger unit in this Example, involves two streams. The first stream consists of water entering as a liquid at a pressure of 8.0 MPa and exiting as a saturated vapor at the same pressure. The second stream consists of gaseous products of combustion, which cool at a constant pressure of 1 atm from 1107 to 547°C. The gaseous stream can be modeled as air, assuming it behaves as an ideal gas. Given T0 = 22°C and p0 = 1 atm,

Determine : **Exercise 1.**

(a) The net rate at which exergy is transferred into the heat exchanger unit by the gas stream, in MW.

(b) The net rate at which exergy is transferred from the heat exchanger by the water stream, in MW.

(c) The rate of exergy destruction, in MW.

(d) The exergetic efficiency.

Exercise 2.

A combined gas turbine–vapor power plant operates with a net power output of 45 MW. The gas turbine section begins with the entry of air into the compressor at 100 kPa and 300 K. The air undergoes compression, reaching a pressure of 1200 kPa. The compressor operates with an isentropic efficiency of 84%. The inlet conditions for the turbine are 1200 kPa and 1400 K. The air expands through the turbine, which has an isentropic efficiency of 88%, until it reaches a pressure of 100 kPa. Subsequently, the air passes through an interconnecting heat exchanger and is ultimately discharged at a temperature of 400 K. In the vapor power cycle section, steam enters the turbine at 8 MPa and 400°C. It undergoes expansion to a condenser pressure of 8 kPa. Water enters the pump as saturated liquid at 8 kPa. The turbine and pump efficiencies for the vapor cycle are 90% and 80%, respectively.

(a) Determine the mass flow rates of the air and the steam, each in kg/s, and the net power developed by the gas turbine and vapor power cycle, each in MW.

(b) Develop a full accounting of the net rate of exergy increase as the air passes through the gas turbine combustor. Discuss. Let T_0 = 300 K, p_0 = 100 kPa.

Exercises of chapter 4

Exercise 1.

Considering a stoichiometric mixture of isooctane and air, determine:

(a) the mole fraction of fuel

(b) the fuel-air ratio

(c) the mole fraction of H_2O in the products

(d) the temperature of products below which H_2O starts to condense into liquid at 101.3 kPa

Exercise 2.

In a model "can-combustor" combustion chamber, n-heptane (C_7H_{16}) is burned under an overall lean condition. Measurements of dry exhaust give mole fractions of CO_2 and O_2 as $x_{CO_2} = 0{,}084$ and $x_{O_2} = 0{,}088$. Determine the $\%EA$, equivalence ratio ϕ, and λ.

Exercise 3.

The heat released by 1 mol of sugar in a bomb calorimeter experiment is 5,648 kJ/mol. Calculate the enthalpy of combustion per mole of sugar.

Exercise 5.

Estimate the adiabatic flame temperature of a constant-pressure reactor burning a stoichiometric mixture of H_2 and air at 101.3 kPa and 25 °C at the inlet.

Exercise 6.

The space shuttle burns liquid hydrogen and oxygen in the main engine. To estimate the maximum flame temperature, consider combustion of 1 mol of gaseous hydrogen with 0.5 mol of gaseous O_2 at 101.3 kPa. Determine the adiabatic flame temperatures using the average c_p method.

Annex 2

Table A.1

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.1

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
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.2

Saturated water—Pressure table

		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
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.2

Saturated water—Pressure table

		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.3 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.3 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 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.3 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 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.3 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 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