Yanbu Industrial College

Department of Mechanical Engineering Technology

POWER PLANT ENGINEERING and ECONOMY

Power Plant Engineering

2010

© Dr. Rahim K. Jassim

PhD, MEMgt, BSc(hons) , P.E., MASME, MIPENZ, Reg. Eng

YANBU INDUSTRIAL COLLEGE

COURSE SYLLABUS

DEPARTMENT: MECHANICAL ENGINEERING TECHNOLOGY

COURSE NAME: MET-401 POWER PLANT ENGINEERING LT:3 LB:3 CR :4

CATALOG DESCRIPTION:

This course is designed to provide the students with application of thermal engineering and mechanics to different thermo-fluid systems. Types, construction, working principles performance of; boilers, heat exchangers, turbines, power plants and internal combustion engines are covered. Overall plant performance, load curves and economics of power plants is introduced. This course is supported by tutorials, laboratory experiments and field visits.

PREREQUISITE: MET 303 - Applied Thermodynamics and Heat Transfer
MET 306 - Applied Fluid Mechanics

PREREQUISITE BY TOPIC:

1. Basic thermodynamics of power cycles and combustion processes
2. Conduction, convection and radiation heat transfer

CO-REQUISITE: None
COMPUTER USAGE: Use of computer programs in power plant calculations

TEXT AND MATERIALS:

Textbook:

1. Power Plant Engineering. by *Nag, P.K.*, Tata-McGraw Hill. Higher Education, 3nd edition, 2008.

References:

1. Powerplant Technology, by *EL-Wakil, M.M.,* McGraw Hill, 1st Edition, 1984
2. Modem Power Plant Engineering, by *Weisman, J.,* and *Eckert, L.,* Prentice Hall, 1st edition. 1985
3. Power Plant System Design, by *Kam W. Li* and *A. Paul Priddy*, John Wiley, 1st edition, 1985.
4. The Exergy Method of Thermal Plant Analysis, by *Kotas, T. J*. reprinted, Malabar, Florida, USA: Krieger, 1995.
5. Fundamentals of Engineering Thermodynamics, by *Moran J. M*. and *Shapiro, H. N*., 5th edition., John Wiley & Sons 2004.

COURSE OBJECTIVES:

Upon successful completion of the course, the student will demonstrate competency by being able to:

1. Describe sources of energy and types of power plants.
2. Analyze different types of steam cycles and estimate efficiencies in a steam power plant.
3. Describe basic working principles of gas turbine and diesel engine power plants Define the performance characteristics and components of such power plants.
4. List the principal components and types of nuclear reactors.
5. Evaluate cycle efficiency and performance of a gas cooled reactor power plant.
6. Classify different types of coupled vapor cycles and list the advantages of combined cycles power plant.
7. List different types of fuels used in power plants and estimate their heating values.
8. List types, principles of operations, components and applications of steam turbines, steam generators, condensers, feed water and circulating water systems. Estimate different efficiencies associated with such systems.
9. Define terms and factors associated with power plant economics. Calculate present worth depreciation and cost of different types of power plants. Estimate the cost of producing power per kW.

COURSE OUTLINE:

Chapter	TOPIC	WEEK
I	**Introduction**	**1**
	1. Energy sources 2. Types of power plants 3. Thermodynamics review	
II	**Analysis of Steam Cycles**	**2-3**
	1. Carnot and Rankine cycles 2. Reheating of steam 3. Regenerative feedwater heaters 4. Efficiencies in a steam power plant	
III	**Fuels and Combustion**	**4**
	1. Types of fuels 2. Air-fuel ratio 3. Heat of combustion 4. Heating values	

IV Steam Generators 5

1. Basic types of steam generators
2. The economizers, superheaters and reheaters
3. Boiler circulation
4. Boiler efficiency

V Steam Turbines 6-7

1. Flow through nozzles
2. Types of steam turbines
3. Velocity diagrams
4. Turbine governing and control
5. Power and efficiency in steam turbines

VI Diesel Engine and Gas Turbine Power Plants 8-9

1. Types of diesel plant
2. Advantages and disadvantages of diesel engine power plant
3. Performance characteristic of diesel engine power plant
4. Gas turbine power plant
5. Components of gas turbine power plants
6. Gas turbine fuels and materials

VII Combined Cycle power plant 10

1. Binary vapor cycles
2. Coupled cycles
3. Combined gas turbine - steam turbine power plants
4. Combined cycle plants for co-generation

VIII Nuclear Power Plants 11

1. Principle components of a nuclear reactor
2. Types of nuclear reactors
3. Gas cooled reactor power plant
4. Cycle efficiency

IX Condensers, Feedwater and Circulating Water Systems 12-13

1. Direct contact condensers
2. Surface condensers
3. Feed water heaters
4. Circulating water systems used in steam power plant
5. Cooling towers
6. Cooling towers calculations

X Economics of Power Generation 14-15

1. Terms and factors

2. Load duration curves
3. Power plant economics (present worth, depreciation and cost)

XI	**Review**	**16**
XII	**Final Examination**	**17**

INSTRUCTIONAL OBJECTIVES *:

Upon successful completion of the course, the student will be able to:

1.00 Introduction

1.01 List and explain different sources of energy
1.02 Identify different types of power plants and recognize their components
1.03 Define basic terms and properties used in thermodynamics and state first and law of thermodynamics and apply it to different thermodynamics systems
1.04 Solve problems on first and second law of thermodynamics

2.00 Analysis of Steam Cycles

2.01 Represent different types of steam cycles on pressure-volume and temperature diagrams
2.02 List the methods and advantages of reheating of steam
2.03 Explain the advantages of regeneration
2.04 Discuss different types of Feed water heaters and their applications
2.05 Calculate different type of efficiencies in a steam power plant
2.06 Solve problems on a simple Rankine and reheating cycles

3.00 Fuels and Combustion

3.01 Describe various types of fuels, properties and application
3.02 Calculate stoichiometric and actual fuel-air ratio
3.03 Apply the first law of thermodynamics for calculation of heat of combustion in open and closed systems
3.04 Estimate the higher and lower heating values of different types of fuels

4.00 Steam Generators

4.01 List types, applications and describe working principles of most commonly boilers
4.02 Describe types, advantages and operating principles of auxiliary systems use steam generators
4.03 Differentiate between natural and forced circulation in boilers
4.04 Calculate boiler efficiency
4.05 Evaluate the operation performance of a steam boiler
4.06 Solve problems on steam generators

5.00 Steam Turbines

5.01 List types, application and working principles of nozzles

5.02 Describe the construction and working principles of impulse and reaction turbine:
5.03 Construct velocity diagram in impulse and reaction bleeding
5.04 List the function, construction and working principles of different turbine governors and control equipment
5.05 Analyze the performance of a steam turbine unit and calculate the efficiency power produced.

6.00 Diesel Engine and Gas Turbine Power Plants

6.01 Describe basic principles and types of diesel plants
6.02 List the advantages and disadvantages of diesel plants
6.03 Recognize and evaluate the performance characteristics of diesel plants
6.04 Evaluate the operation performance of a diesel engine
6.05 Describe the construction and working principles of gas turbine power plants
6.06 Evaluate the performance of a gas turbine power plant
6.07 Name basic components and auxiliary systems used in gas turbine power plan
6.08 Identify different types of fuels and materials used in gas turbine power plants
6.09 Solve problems on diesel engine and gas turbine power plant cycles

7.00 Combined Cycle Power Plants

7.01 Describe the working principle of a binary vapor cycle
7.02 Describe the working principle of different types coupled cycles
7.03 Describe construction, working principles and advantages of a combined gas turbine - steam turbine power plant
7.04 List the advantages of combined cycles co-generation plant
7.05 Solve problems on combined cycle power plants

8:00 Nuclear Power Plants

8.01 Describe the working principles and construction of a nuclear reactor
8.02 List different types of nuclear reactors
8.03 Explain the construction and working principles of a gas cooled reactor power plant
8.04 Calculate the cycle efficiency of a power plant

9.00 Condensers, Feed Water and Circulating Water System

9.01 List types, construction and working principles of the direct contact type and surface condensers
9.02 Draw schematic flow diagrams and their corresponding T-S diagrams of direct contact type and surface condensers
9.03 List the functions and types of feed water heaters
9.04 List the functions and types of circulating water system used in power plant
9.05 List the functions, types and working principles of cooling towers
9.06 Calculate make-up water in cooling towers
9.07 Solve problems on cooling towers

10.00 Economics of Power Generation

10.01 Define load factor, demand factor and methods of plant selection
10.02 Make a load-duration curve analysis of a plant
10.03 Estimate fixed and running cost of a plant and its depreciation

** Conditions and standards for all enabling objectives are implied. They are "after completing this lesson" and "by correctly answering the corresponding test item".*

EVALUATION:

Evaluation will be competency-based and student grades will be based on the following factors;

Course work (Class quizzes and Major tests)	50%
Final Examination	50%
	100%

Letter grades will be awarded to each student according to his numeric standing as outline below:

95-100	=	A+	-	4.00
90-94	=	A	-	3.75
85-89	=	B+	-	3.50
80-84	=	B	-	3.00
75-79	=	C+	-	2.50
70-74	=	C	-	2.00
65-69	=	D+	-	1.50
60-64	=	D	-	1.00
0-59	=	F	-	0.00

LABORATORY OUTLINE:

	Laboratory Exercises	Week
1.	Tutorial 1-First and Second Law of Thermodynamics Calculations	1
2.	Tutorial 2 - Calculations of Simple Rankine and Reheating Cycles	2
3.	Field Visit to a Steam Power Plant	3
4.	Trial and Operation of Boilers	4-5
5.	Tutorial 3 - Boiler Efficiency Calculations	6
5.	Tutorial 4 - Diesel Engine and a Gas Turbine Power Plants Cycle's Calculations	7
6.	Operating and Testing the Performance of Diesel Engines	8
8.	Operating and Testing the Performance of Gas Turbine Power Plants	9
9.	Field Visit to a Diesel Engine and Gas Turbine Power Plants	10
10.	Tutorial 5 - Combined Cycle Power Plant Calculations	11
11.	Performance Test of Cooling Towers	12-13
12.	Review	14
13.	Final Examination	15

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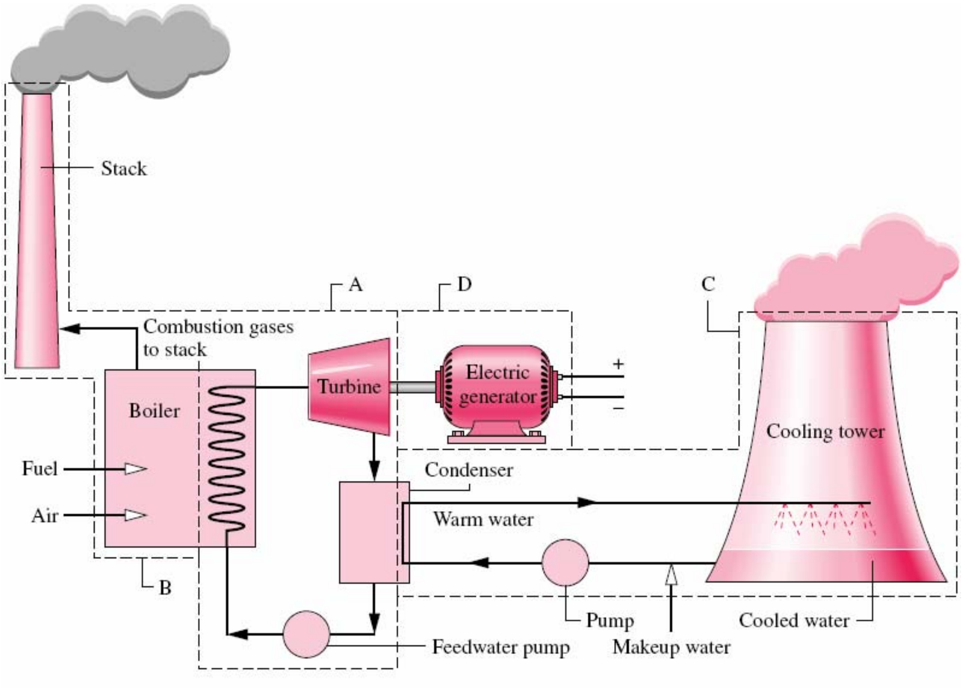
MET401

Department of Mechanical Engineering Technology



Chapter I- Introduction

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Chapter (I)

Introduction

1.1- Energy sources

The energy sources can be of two types:

1. Renewable or non conventional energy resources
2. Non-renewable or conventional energy resources

1. Renewable energy resources

- Solar energy
- Wind energy
- Tidal energy
- Ocean tides and waves energy
- Flowing of stream of water

2. Non-renewable energy resources

- Coal, coke, etc
- Petroleum
- Natural gas
- Nuclear power

1.2-Fuels

Fuel is defined as any material which when burnt will produce heat. Various fuels commonly used are as follows:

- Solid fuels (Wood, peat, lignite, bituminous coal and anthracite coal)
- Liquid fuels (Petroleum and its derivatives)
- Gaseous fuels (CNG, LPG, etc)

1.3- Types of Power Plants

1. Steam Power Plant
2. Gas turbine Power Plants
3. Combined Cycle Power Generation
4. Nuclear Power Plants
5. Hydroelectric Power plant

1.4- Steam Tables

Steam tables commonly consist of two sets of tables of the energy transfer properties of water and steam: saturated steam tables and superheated steam tables. Both sets of tables are tabulations of pressure (P), temperature (T), specific volume (v), specific internal energy (u), specific enthalpy (h), and specific entropy (s). The following notation is used in steam tables.

T : temperature (^{o}C)
P : pressure (kPa)
v : specific volume (m^3/kg)
u: specific internal energy (kJ/kg)
h: specific enthalpy (kJ/kg)
s: specific entropy (kJ/kg K)

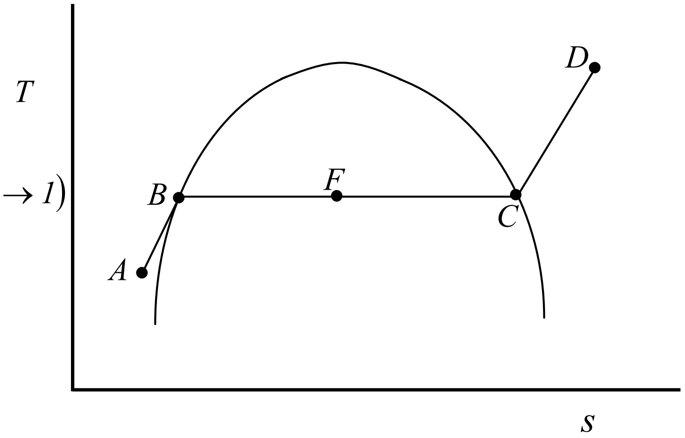
Definitions

Subcooled or compressed
Saturated liquid
Wet (mixture)
Saturated vapor
Superheated
Quality (x)

<u>Water</u>

Subcooled water (*A*)
Saturated liquid (*B)* ($x = 0$)
Evaporation of saturated liquid (*B-C*) $(x = 0 \rightarrow 1)$
Wet steam (*F*) $(x = ------)$
Dry Saturated steam (*C*) *)* ($x = 1$)
Superheated steam (D)

$$x = \frac{vapour\ \ mass}{total\ \ mass}$$



1.5- The enthalpy-entropy diagram

The Mollier diagram, shown in Figure 1.1, is a chart on which enthalpy (h) versus entropy (s) is plotted. It is sometimes known as the h-s diagram and has an entirely different shape from the T-s diagrams. The chart contains a series of constant temperature lines, a series of constant pressure lines, a series of constant quality lines, and a series of constant superheat lines. The Mollier diagram is used only when quality is greater than 50% and for superheated steam.

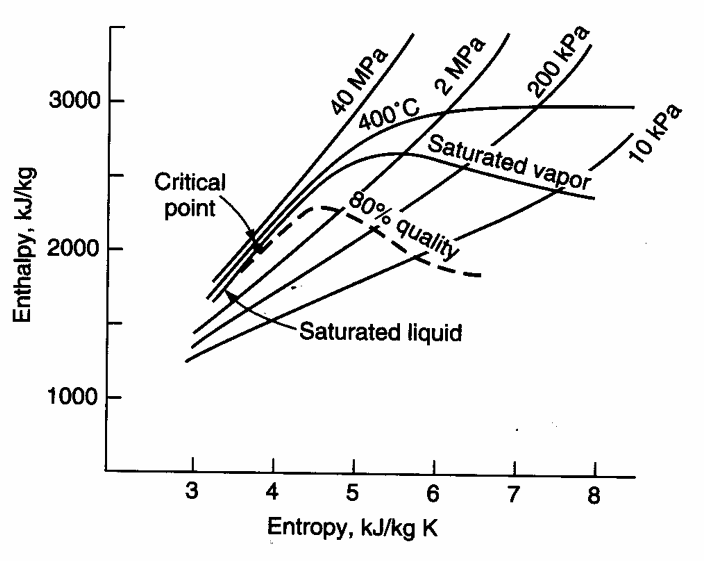


Figure 1.1

Example 1.1

Heat is transferred to a dry saturated steam at 193 °C until it's superheated by 167 °C, while the pressure remains constant, determine Δh

Example 1.2

Steam at 300 °C and 0.5 MPa is expanded in a reversible adiabatic process (isentropic) to 0.01 MPa, determine Δh and x.

1.6- Thermodynamics review

We have already discussed the first law of thermodynamics: control volume in applied thermodynamics and heat transfer subject.

1.6.1- Conservation of mass and the control volume

The conservation of mass is one of the most fundamental principles in nature. We are all familiar with this principles, and it is not difficul to understand. For closed systems, the conservation of mass principles is implicitly used by requiring that the mass of the system remains constant during a process. For control volumes (see Figure 1.2), however, mass can cross the boundaries, and so we must keep track of the amount of the mass entering and leaving the control volume (see Figure 1.2). The conservation of mass principle for the control volume (CV) undergoing a process can be expressed as

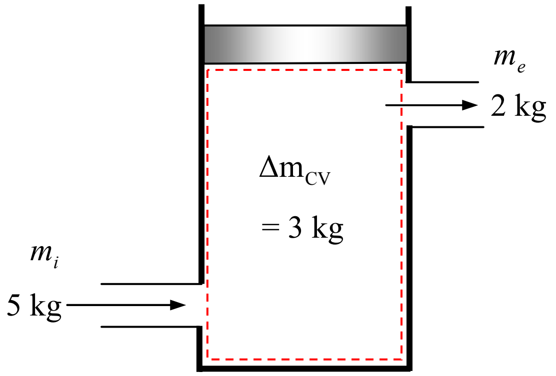
$$\Delta \mathrm{m}_{\mathrm{CV}} = \sum \mathrm{m}_{\mathrm{i}} - \sum \mathrm{m}_{\mathrm{e}}$$

The average rate of change of mass within the control volume

$$\frac{dm_{CV}}{dt} = \sum \dot{m}_i - \sum \dot{m}_e$$

$$\begin{pmatrix}\text{net change}\\ \text{in mass within}\\ \text{control volume}\end{pmatrix} = \begin{pmatrix}\text{total}\\ \text{mass entering}\\ \text{control volume}\end{pmatrix} - \begin{pmatrix}\text{total}\\ \text{mass leaving}\\ \text{control volume}\end{pmatrix} \quad (1.1)$$

Figure 1.2



Equation (1.1) for the conservation of mass is commonly termed the *Continuity Equation*. While this form of the equation is sufficient for the majority of applications in thermodynamics, it is frequently rewitten in terms of local fluid properties in the study of fluid dynamics and heat transfer.

Let us consider one further aspect of mass flow across a control volume. For simplicity, we assume that a fluid is flowing uniformaly inside a pipe or duct, as shown in Figure 1.3.

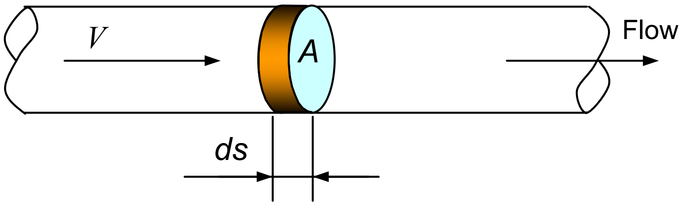


Figure (1.3)

We wish to examine the flow in terms of the amount of mass crossing the surface *A* during the time interval δt. As seen from the figure above, the fluid moves a distance *ds* during this time, and therefore the volume of fluid crossing surface *A* is *A ds.* Consequently, the mass crossing surface A is given by

$$\delta \mathrm{m} = \frac{A\ ds}{v}$$

If both sides of the above expression is devided by δt and take the limit $\delta t \to 0$, the result is

$$\dot{\mathrm{m}} = \frac{A\ V}{v} \qquad (1.2)$$

Where V is the fluid velocity m/s

A cross sectional area m^2

1.6.2-The First law of Thermodynamics for a Control Volume

The first law of thermodynaocs for a control mass, which consists of a fixed quantity of mass is expressed as

$${}_1Q_2 = E_2 - E_1 + {}_1W_2$$

This expression can be written as a rate equation

$$\boxed{\dot{Q}_{c.v.} + \sum \dot{m}_i \left(h_i + \left(\frac{V_i^2}{2} + g\,Z_i \right) \times 10^{-3} \right) = \frac{dE_{c.v}}{dt} + \sum \dot{m}_e \left(h_e + \left(\frac{V_e^2}{2} + g\,Z_e \right) \times 10^{-3} \right) + \dot{W}_{c.v.} \quad (1.3)}$$

The first application of the first law of thermodynamics for the control volume will be to develop a suitable analytical model for SSSF operation of devices such as turbines, compressors, nozzles, boilers, heat excangers, etc.

SSSF assumptions

$$\frac{dm_{c.v.}}{dt} = 0 \quad \text{and} \quad \frac{dE_{c.v.}}{dt} = 0$$

Then, equations (1.1 and 1.3) can be written,

$$\sum \dot{\mathrm{m}}_{\mathrm{i}} = \sum \dot{\mathrm{m}}_{\mathrm{e}}$$

$$\dot{Q}_{c.v.} + \sum \dot{m}_i \left(h_i + \left(\frac{V_i^2}{2} + g\,Z_i \right) \times 10^{-3} \right) = \sum \dot{m}_e \left(h_e + \left(\frac{V_e^2}{2} + g\,Z_e \right) \times 10^{-3} \right) + \dot{W}_{c.v.}$$

In many applications of SSSF model have only one flow stream entering and one leaving the control volume. Then

$$\dot{m}_i = \dot{m}_e = \dot{m} \tag{1.4}$$

$$\dot{Q}_{c.v.} + \dot{m} \left(h_i + \left(\frac{V_i^2}{2} + g\,Z_i \right) \times 10^{-3} \right) = \dot{m} \left(h_e + \left(\frac{V_e^2}{2} + g\,Z_e \right) \times 10^{-3} \right) + \dot{W}_{c.v.} \tag{1.5}$$

Rearranging this equation, we have

$$q + h_i + \left(\frac{V_i^2}{2} + g\,Z_i \right) \times 10^{-3} = h_e + \left(\frac{V_e^2}{2} + g\,Z_e \right) \times 10^{-3} + w \tag{1.6}$$

where, by definition,

$$q = \frac{\dot{Q}_{c.v}}{\dot{m}} \quad \text{and} \quad w = \frac{\dot{W}_{c.v}}{\dot{m}} \tag{1.7}$$

1.6.3- Power Cycles

Two important area of application for SSSF or cylinder/piston boundary work processes of thermodynamics are ***power generation*** and ***refrigeration*** systems. Both systems are accomplished by operating on a thermodynamic cycles. These cycles are generally divided into two categories: ***power cycles*** and ***refrigeration cycles.***

The systems used to produce a net power output are often called engines (i.e. steam power plant, internal combustion engine and gas turbine) and the thermodynamic cycles they operate on are called power cycles. Whereas, the systems used to produce refrigeration are called refrigerators, air conditioners, or heat pumps, and the thermodynamic cycles they operate on are called refrigeration cycles.

Thermodynamic cycles can be categories to: Closed cycles and open cycles. In closed cycles, the working fluid is returned to the initial state at the end of the cycle and is recirculated (i.e. steam power plant, refrigerators) . In open cycle, the working fluid is renewed at the end of each cycle (i.e. internal combustion engine, gas turbine).

In this course, the focus will be only on power cycles

In gas power cycle, the working fluid remains a gas throughout the entire cycle. Internal combustion engine, gas turbines are familiar examples of devices that operate on gas cycles. Analysis of these cycles are rather complex. To simplify the analysis to a manageable level, we utilize the following assumptions, commonly known as the air – standard assumptions.

1. The working fluid is air that continuously circulates in a closed loop and always behaves as an *ideal gas*.

Some important relations

$$du = C_{vo} dT \tag{1.8 a}$$

$$dh = C_{po} dT \tag{1.8 b}$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k \qquad \text{where } k = \frac{C_{po}}{C_{vo}} \tag{1.9 a}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = \left(\frac{V_1}{V_2}\right)^{k-1} \tag{1.9 b}$$

Also, the specific heat of air is constant and is evaluated at 25°C

2. All the processes that make up the cycle are internally reversible
3. The combustion process is replaced by a heat addition process from external an external source (Figure 1.4).
4. The exhaust process is replaced by a heat rejection process that restores the working fluid to its initial state.

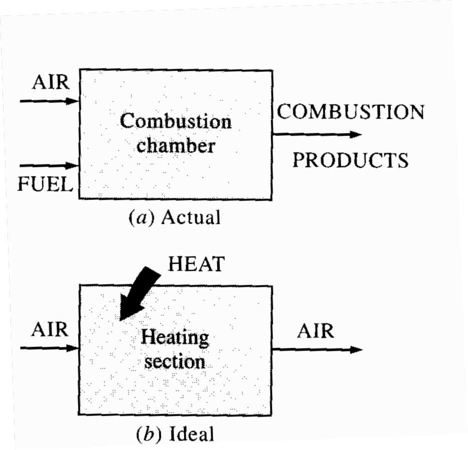


Figure (1.4)

1.6.4 - Reciprocating engines

The basic components of a reciprocating engine are shown in Figure 1.5. The piston reciprocates in the cylinder between two fixed positions called the ***top dead center*** (*TDC*)—the position of the piston when it forms the smallest volume in the cylinder—and the ***bottom dead center*** (*BDC*)—the position of the piston when it forms the largest volume in the cylinder. The distance between the *TDC* and the BDC is the largest distance that the piston can travel in one direction, and it is called the ***stroke*** of the engine. The diameter of the piston is called the ***bore***. The air or air-fuel mixture is drawn into the cylinder through the ***intake valve*****,** and the combustion products are expelled from the cylinder through the ***exhaust valve***.

The minimum volume formed in the cylinder when the piston is at *TDC* is called the ***clearance volume*** (Figure 1.6). The volume displaced by the piston as it moves between *TDC* and *BDC* is called the ***displacement volume***. The ratio of the maximum volume formed in the cylinder to the minimum clearance volume is called the ***compression ratio*** (*r*) of the engine:

$$r = \frac{V_{\max}}{V_{\min}} = \frac{V_{BDC}}{V_{TDC}} \qquad (1.10)$$

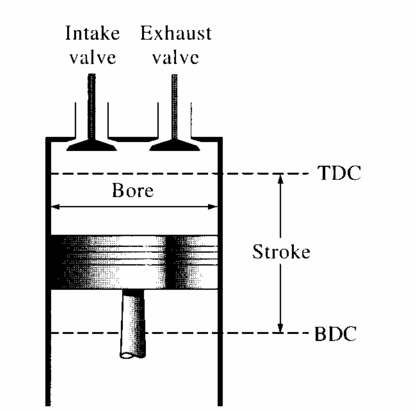


Figure (1.5)

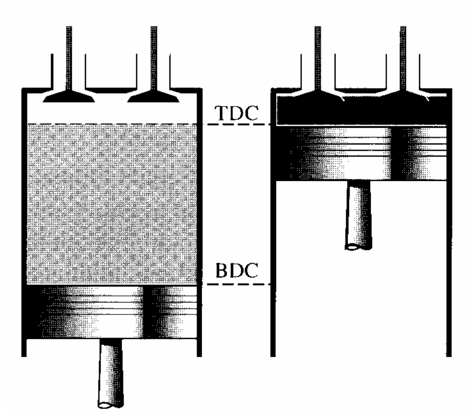


Figure (1.6)

Notice that the compression ratio is a volume ratio and should not be confused with the pressure ratio.

Another term frequently used in conjunction with reciprocating engines is the ***<u>Mean Effective Pressure</u>*** (*MEP*). It is a fictitious pressure which, if it acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle (Figure 1.7). That is,

W_{net}., = *MEP* X piston area X stroke = *MEP* X displacement volume

$$MEP = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}} \qquad (1.11)$$

The mean effective pressure can be used as a parameter to compare the performances of reciprocating engines of equal size. The engine that has a larger value of MEP will deliver more net work per cycle and thus will perform better.

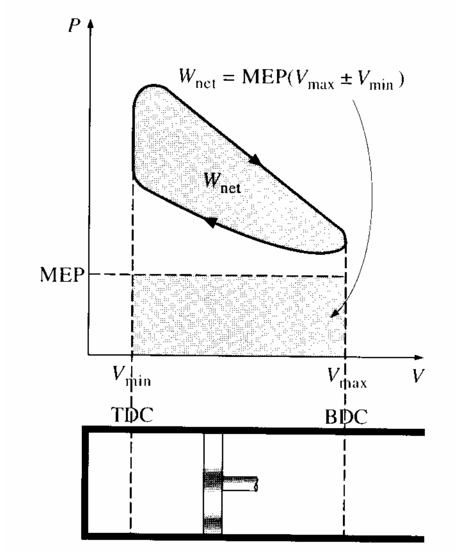


Figure (1.7)

1.6.4.1- Otto Cycle

The air standard Otto cycle is an ideal cycle approximates a spark-ignition internal combustion engine (IC). This cycle is shown on *P-v* and *T-s* diagrams of Figure (1.8).

Process 1-2 is an isentropic compression *(s=c)* of the air as the piston moves from BDC to TDC.

Process 2-3 heat is added at *constant volume* while the piston is momentarily at rest at the TDC point. (This process corresponds to the ignition of the fuel air mixture by the spark and subsequent burning in the engine).

Process 3-4 is an isentropic expansion *(s=c),* and

Process 4-1 is the heat rejection of heat from the air while the piston is at BDC (*v=c*).

The thermal efficiency of this cycle is found as follows,

$$\eta_{th,Otto} = \frac{W_{net}}{Q_H} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{m\,C_{vo}\left(T_4 - T_1\right)}{m\,C_{vo}\left(T_3 - T_2\right)} = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} \tag{1.12}$$

Processes 1-2 and 3-4 are isentropic, and v_2=v_3 and v_4=v_1. Thus

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = \left(\frac{V_4}{V_3}\right)^{k-1} = \frac{T_3}{T_4} \tag{1.13}$$

Therefore,

$$\frac{T_3}{T_2} = \frac{T_4}{T_1} \tag{1.14}$$

and

$$\eta_{th,Otto} = 1 - \frac{T_1}{T_2} = 1 - \left(r\right)^{1-k} = 1 - \frac{1}{\left(r\right)^{k-1}} \tag{1.15}$$

where

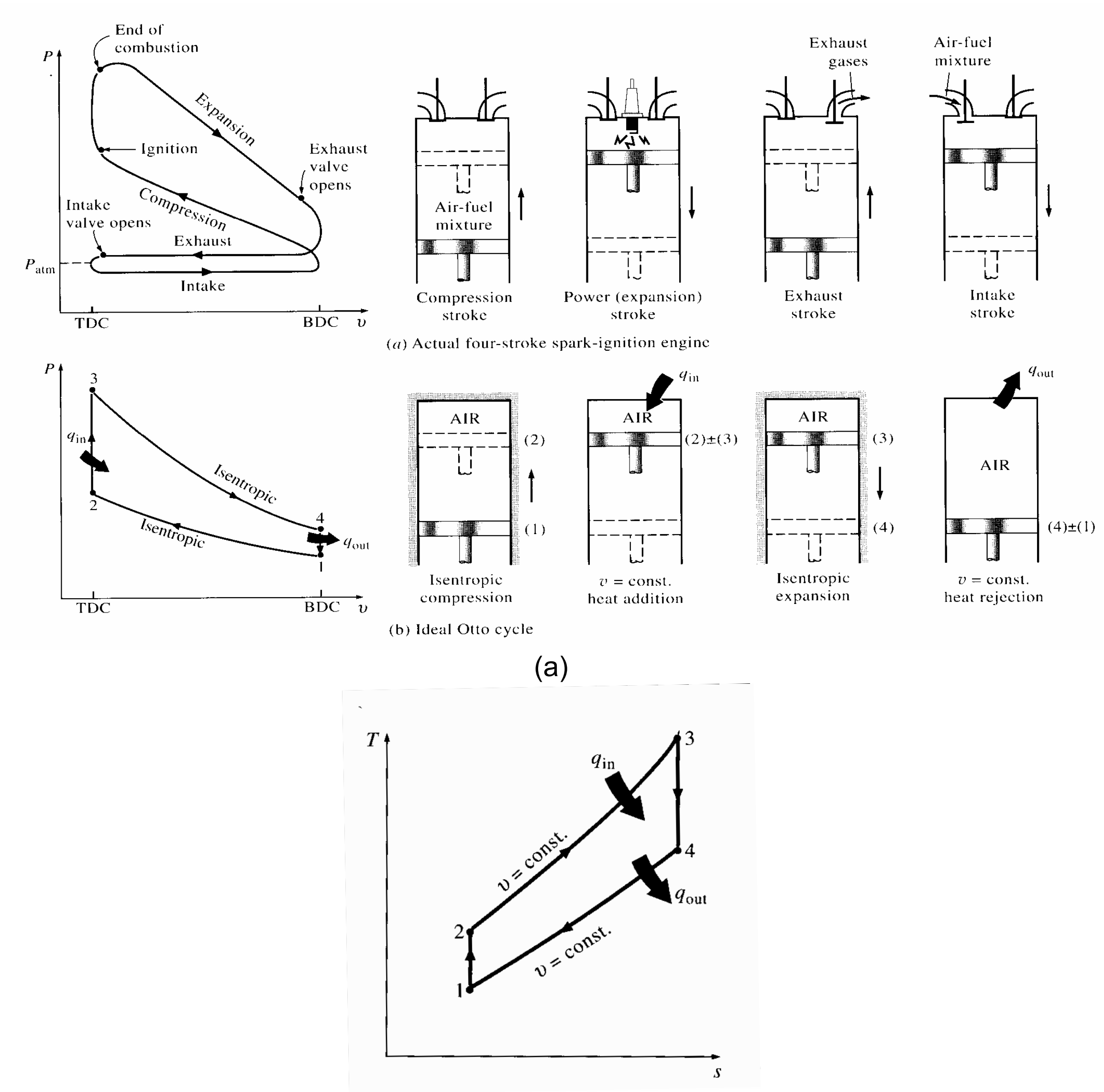
$$r = \frac{V_1}{V_2} = \frac{V_4}{V_3} \tag{1.16}$$

Example 1.3

An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, the air is at 100 kPa and 17 ^{o}C, and 1800 kJ/kg of heat is transferred to air during the constant-volume heat addition process. Determine

a- the maximum temperature and pressure that occur during the cycle.
b- the net work output
c- the thermal efficiency, and
d- the mean effective pressure for the cycle

Solution



(a)

(b)

Figure (1.8)

1.6.4.2- The Diesel Cycle

The air-standard Diesel cycle is shown in Figure 1.9. This is the ideal cycle for the diesel engine, which is also called the compression-ignition engine.

The Diesel cycle is the ideal cycle for CI reciprocating engines. The CI engine, first proposed by Rudolph Diesel in the 1890s, is very similar to the SI engine discussed in the last section, differing mainly in the method of initiating combustion. In spark-ignition engines (also known as gasoline engines), the air-fuel mixture is compressed to a temperature that is below the autoignition temperature of the fuel, and the combustion process is initiated by firing a spark plug. In CI engines (also known as diesel engines), the air is compressed to a temperature which is above the autoignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engines (Figure1.9).

In gasoline engines, a mixture of air and fuel is compressed during the compression stroke, and the compression ratios are limited by the onset of autoignition or engine knock. In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition. Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 and 24. Not having to deal with the problem of autoignition has another benefit: many of the stringent requirements placed on the gasoline can now be removed, and fuels that are less refined (thus less expensive) can be used in diesel engines.

The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke. Therefore, the combustion process in these engines takes place over a longer interval. Because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a *constant-pressure* addition process.

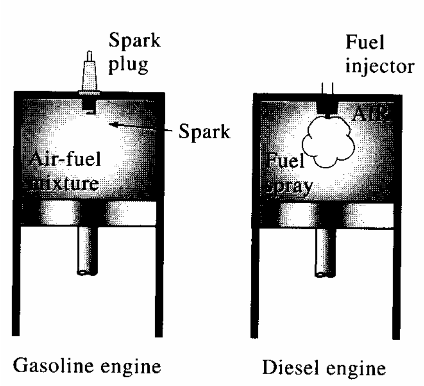


Figure (1.9)

In fact, this is the only process where the Otto and Diesel cycles differ. The remaining three processes are the same for ideal cycles. That is, process 1-2 is isentropic compression, 3-4 isentropic expansion, and 4-1 is constant-volume heat rejection. The similarity between the two cycles is also apparent from the *P-v* and *T-s* diagrams of the Diesel cycle, shown in Figure 1.10.

A measure of performance for any power cycle is it's thermal efficiency. Below we develop a relation for the thermal efficiency of a Diesel cycle, utilizing the cold-air-standard assumptions. Such a relation will enable us to examine the effects of major parameters on performance of diesel engines.

The Diesel style cycle, like the Otto cycle, is executed in a piston-cylinder device, which forms a closed system. Therefore, equations developed for closed systems should be used in the analysis of individual processes. Under the cold-air-standard assumptions, the amount of heat added to the working fluid at *constant pressure* and rejected from it at *constant volume* can be expressed as

$$ {}_2q_3 = {}_2w_3 + \Delta_2 u_3 = P_2(v_3 - v_2) + (u_3 - u_2) = (h_3 - h_2) = C_{po}(T_3 - T_2) \qquad (1.17) $$

and

$$ {}_4q_1 = C_{vo}\left(T_4 - T_1\right) \tag{1.18}$$

Then the thermal of the Diesel cycle is given by the relation

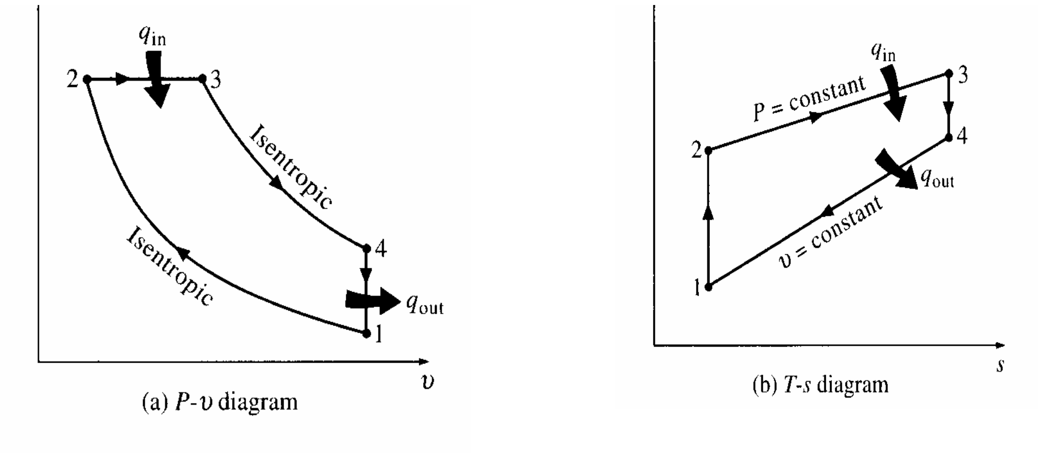
$$\eta_{th,Diesel} = \frac{W_{net}}{Q_H} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{m\,C_{vo}\left(T_4 - T_1\right)}{m\,C_{po}\left(T_3 - T_2\right)} = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{k\,T_2\left(\frac{T_3}{T_2} - 1\right)} \tag{1.19}$$

We know define a new quality, the *cutoff ratio* r_c, as the ratio of the cylinder volumes after and before the combustion process

$$r_c = \frac{V_3}{V_2} = \frac{v_3}{v_2} \tag{1.20}$$

Utilizing this definition and the isentropic ideal gas relations for processes 1-2 and 3-4, then the thermal efficiency relation reduces to

$$\eta_{th,Diesel} = 1 - \frac{1}{(r)^{k-1}}\left[\frac{r_c^k - 1}{k\left(r_c - 1\right)}\right] \tag{1.21}$$



(a) P-υ diagram

(b) T-s diagram

Figure (1.10)

Example 1.4

An ideal Diesel cycle with air as the working fluid has a compression ratio of 18 and cutoff ratio of 2. At the beginning of the compression process, the working fluid is at 100 kPa and 300 K, and 0.002 m^3. Utilizing the cold-air standard assumptions, determine

a- the temperature and pressure of the air at the end of each process.

b- The net output and the thermal efficiency, and

c- The mean effective pressure.

Solution

1.6.4.3- The Brayton Cycle

In discussing idealized four-SSSF-process power cycles in previous sections, a cycle involving two constant-pressure and two isentropic processes. This cycle used with a condensing working fluid is the Rankine cycle, but when used with a single-phase, gaseous working fluid it is termed the Brayton cycle. The air-standard Brayton cycle is the ideal cycle for the *simple gas turbine*. The simple open cycle gas turbine utilizing an internal-combustion process and the simple closed-cycle gas turbine, which utilizes heat-transfer processes, are both shown schematically in Figure 1.11. The air-standard Brayton cycle is shown on the P-v and T-s diagrams of Figure 1.12.

The efficiency of the air-standard Brayton cycle is found as follows:

$$\eta_{th,Brayton} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_{po}\left(T_4 - T_1\right)}{C_{po}\left(T_3 - T_2\right)} = 1 - \frac{T_1\left(T_4/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right)} \quad (1.22)$$

Processes 1-2 and 3-4 are isentropic, and P_2=P_3 and P_4=P_1. Thus

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} = \frac{T_3}{T_4} \quad (1.23)$$

Substituting equation 1.23 into the thermal efficiency relation (eq. 1.22) and simplifying give

$$\eta_{th,Brayton} = 1 - \frac{1}{r_p^{(k-1)/k}} \quad (1.24)$$

where $r_p = \frac{P_2}{P_1}$ (Pressure ratio)

We now define a new ratio, the ***back work ratio*** ***(bwr)***, as the ratio of the compressor work to the turbine work.

$$w_{comp} = C_{po}\left(T_2 - T_1\right) \quad (1.25\text{ a})$$

$$w_{turb} = C_{po}\left(T_3 - T_4\right) \quad (1.25\text{ b})$$

$$bwr = \frac{w_{comp}}{w_{turb}} = \frac{T_2 - T_1}{T_3 - T_4} \quad (1.25\text{ c})$$

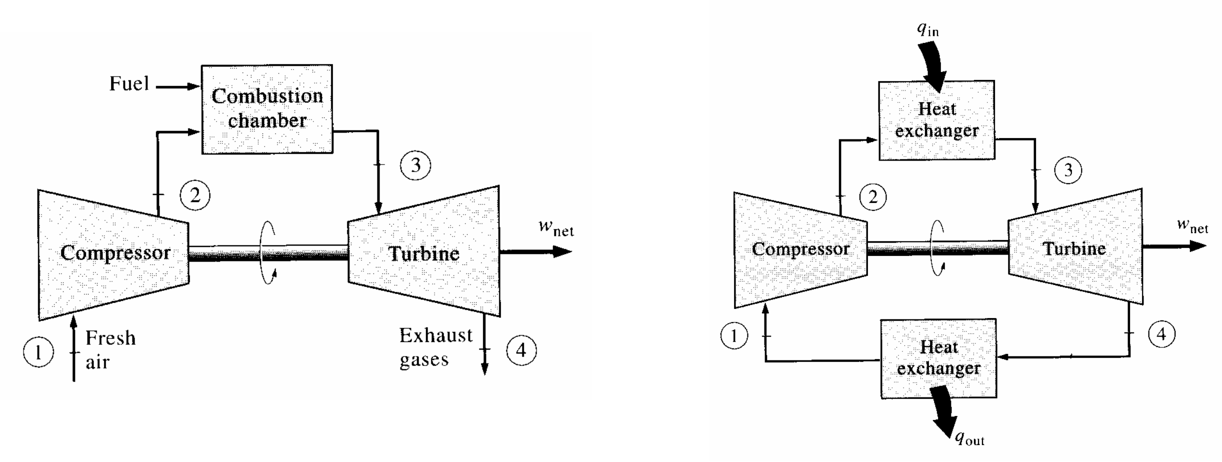
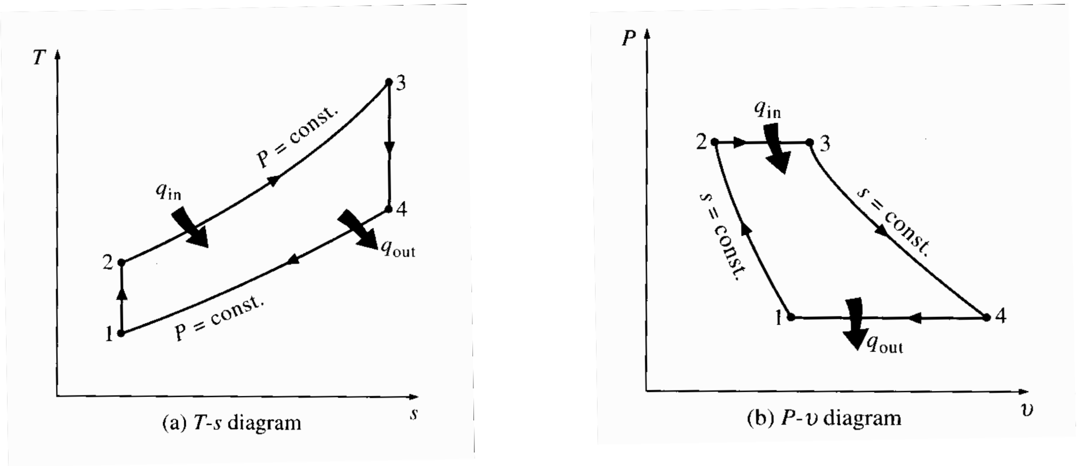


Figure (1.11)



(a) T-s diagram

(b) P-v diagram

Figure (1.12)

Example 1.5

A stationary power plant operating on an ideal Brayton cycle has a pressure ratio of 8. The gas temperature is 300 K at the compressor inlet and 1300 K at the turbine inlet. Utilizing the cold-air standard assumptions, determine

a- the gas temperature at the exits of the compressor and turbine.

b- the back work ratio.

c- the thermal efficiency.

Solution

1.6.4.5- The Rankine Cycle

The Rankine cycle is an idealized four SSSF process for simple vapor power plants as shown in Figure 1.13a, in which state 1 is saturated liquid and state 3 either saturated vapor or superheated vapor. It is convenient to show the states and processes on the T-s diagram, as given in Figure 1.13b. The four processes are,

1-2: Reversible adiabatic (Isentropic) pumping process in the pump.

2-3: Constant pressure transfer of heat in the boiler

3-4: Reversible adiabatic (Isentropic) expansion in the turbine

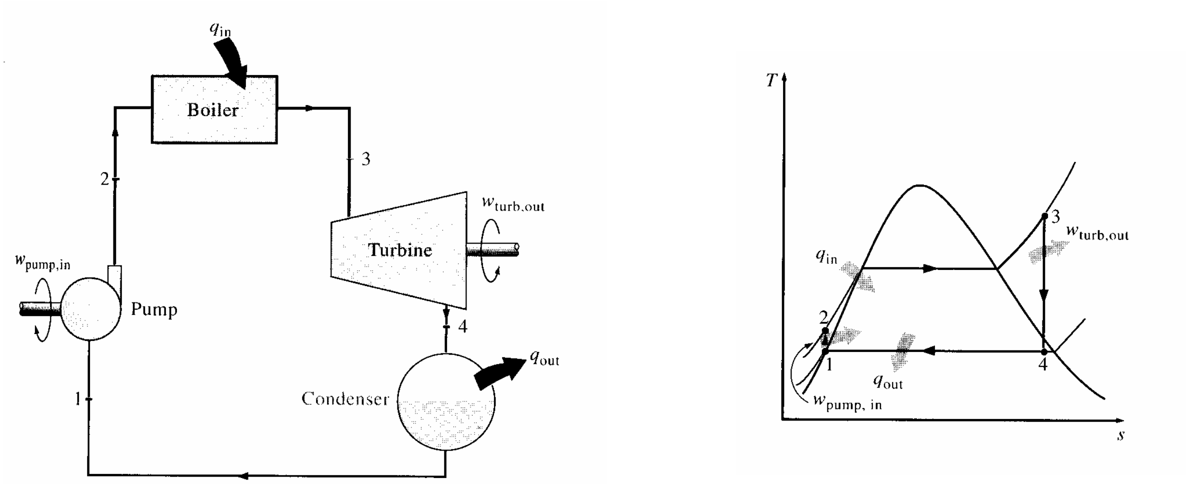
4-1: Constant pressure transfer of heat in the condenser.

As mentioned earlier that, all four components associated with the Rankine cycle are SSSF processes. The kinetic and potential energy changes of the steam are relatively small to the work and heat transfer terms and are therefore usually neglected. Then the SSSF energy equations per unit mass of steam are as follows,

Pump	$w_{pump,in} = h_2 - h_1$, or $w_{pump,in} = v(P_2 - P_1)$, $v = v_f$ at P_1
Boiler	${}_2q_3 = q_{in} = h_3 - h_2$
Turbine	$w_{turb,out} = h_3 - h_4$
condenser	${}_4q_1 = q_{out} = h_4 - h_1$

The thermal efficiency is defined by the relation

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{h_4 - h_1}{h_3 - h_2} \tag{1.26}$$



(b)

(a)

Figure (1.13)

Another parameter which is similar to Brayton cycle used to describe power plant performance is the ***back work ratio*** (*bwr*), in Rankine cycle is defined as the ratio of the pump work input to the work developed by the turbine.

$$bwr = \frac{h_2 - h_1}{h_3 - h_4} \qquad (1.27)$$

Example 1.6

Determine the missing properties and the phase description in the following table for water

	T , °C	*P, kPa*	*h, kJ/kg*	*x*	*Phase*
(a)		200		0.6	
(b)	125		1600		
(c)		1000	2950		
(d)	75	500			
(e)		850		0.0	

Example 1.7

Consider a steam power plant operating on a simple ideal Rankine cycle (see Figure 1.13). The steam enters the turbine at 3 MPa and 350 ^{o}C and is considered in the condenser at a pressure of 75 kPa. Determine the thermal efficiency of this cycle.

Solution

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(End of Thermodynamics review)

Revision Questions

1.1-The compression ratio in an air standard Otto cycle is 8. At the beginning of the compression stroke the pressure is 100 kPa and the temperature is 25 °C. The heat transfer to the air per cycle is 1700 kJ/kg air. Determine

- The pressure and temperature at the end of each process of the cycle. (T_2, P_2, v_2, T_3, P_3, T_4, P_4).
- The heat rejection, ${}_4q_1$.
- The thermal efficiency ($\eta_{th,Otto}$)
- The mean effective pressure (*MEP*)

Ans: 684.6 K, 1838 kPa, 0.107 m3/kg, 3052.3 K,8194.4 kPa, 1328.6 K, 446 kPa
740 kJ/kg, 56.5 %, 1275 kPa

1.2- An air-standard Diesel cycle has a compression ratio of 18, and the heat transferred to the working fluid per cycles is 1800 kJ/kg. At the beginning of the compression process the pressure is 0.1 MPa and the temperature is 25 °C. Calculate

- The pressure and temperature at the end of each process of the cycle.
- The thermal efficiency ($\eta_{th,Diesel}$)
- The mean effective pressure (*MEP)*

Ans: 947 K, 5720 kPa, 2738 K, 440 kPa, 1316 K, 59.4%, 1322.6 kPa

1.3- Determine the efficiency of a Rankine cycle using steam as the working fluid in which the condenser pressure is 10 kPa. The boiler pressure is 2 MPa. The steam leaves the boiler as saturated vapor.

Ans: 30.3%

1.4- In a Brayton cycle the air enters the compressor at 0.1 MPa, 25 °C. The pressure leaving the compressor is 1.0 MPa, and the maximum temperature in the cycle is 1100 °C. Determine

- The pressure and temperature at each point in the cycle
- The compressor work, turbine work, and cycle efficiency.

Ans: 575.6 K, 711 K 279 kJ/kg, 665 kJ/kg, 48.2 %

1.5-Steam is the working fluid in an ideal Rankine cycle. Saturated vapor enters the turbine at 8 MPa and saturated liquid exits the condenser at a pressure of 0.008 MPa. The net power output of the cycle is 100 MW. Determine:

	Ans
• The thermal efficiency	37.1%
• The back work ratio	0.84%
• The mass flow rate of the steam in kg/h	3.77 x 10^5 kg/h
• The rate of heat transfer, $\dot{Q}_{in}$ in MW	269.77 MW
• The mass flow rate of condenser cooling water, in kg/h, if the cooling water enters the condenser at 25 °C and exits at 35 °C.	14.27x10^6 kg/h

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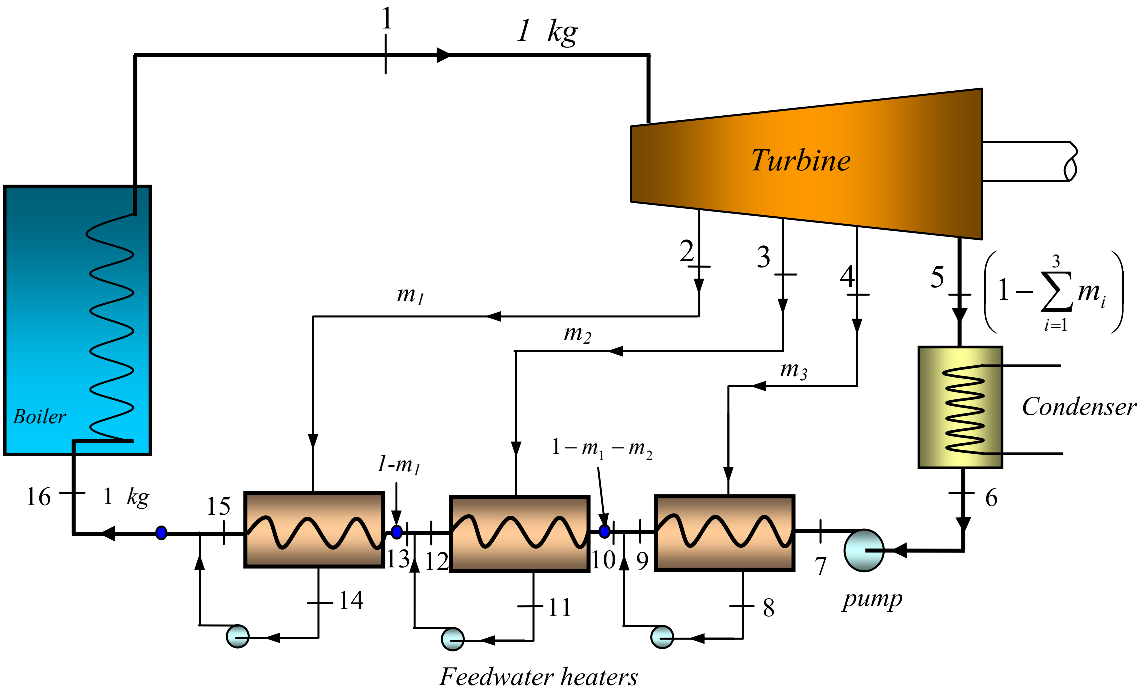
Power Plant Engineering and economy

MET401

Department of Mechanical Engineering Technology



Chapter II- Analysis of Steam Cycles

2010

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PhD, MEMgt, BSc(hons) , P.E., MASME, MIPENZ, Reg. Eng

(Chapter II)

Analysis of Steam Cycles

2.1- Analysis of Carnot Cycle

Having defined the reversible process and considered some factors that make processes irreversible. Let us pose the question. If the efficiency of all heat engine is less than 100%, what is the maximum (most) efficiency cycle we can have.

The important point to be made here is that the Carnot cycle, regardless of what the working substance may be, always has the same four basic processes. These processes are,

1-2 Reversible isothermal process in which heat is transferred to or from the high temperature reservoir.

2-3 Reversible adiabatic process in which the temperature of the working fluid decreases from the high temperature to the low temperature.

3-4 Reversible isothermal process in which heat is transferred to or from the low temperature reservoir.

4-1 Reversible adiabatic process in which the temperature of the working fluid increases from the low temperature to the high temperature.

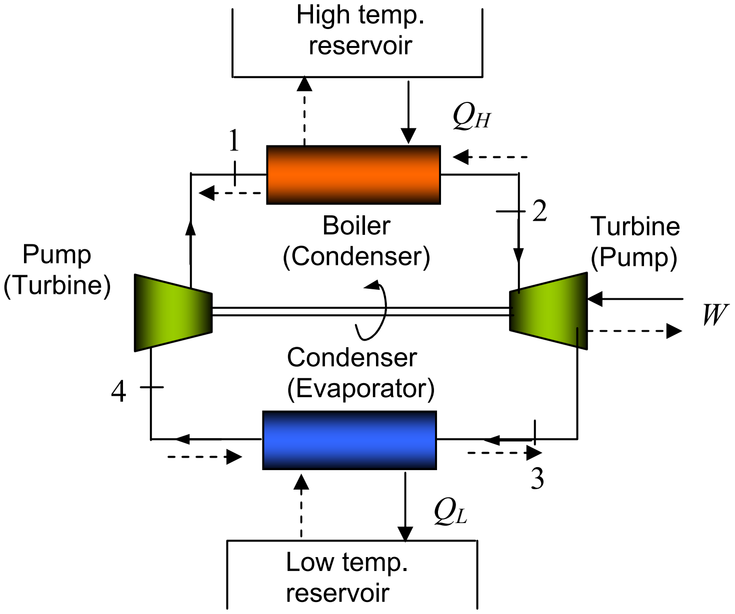
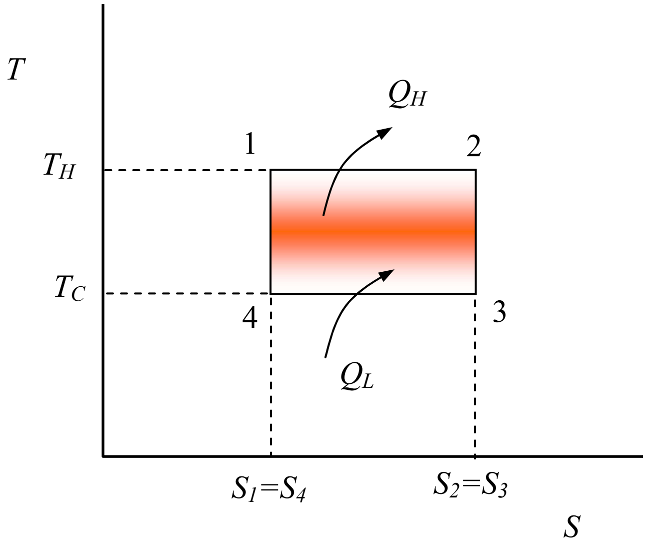


Figure (2.1 a)

Figure (2.1b)



The efficiency of the Carnot engine is

$$\eta_{rev}=\frac{\dot{W}_{net}}{\dot{Q}_{in}}=\frac{T_H-T_L}{T_H}=\left(1-\frac{T_L}{T_H}\right) \tag{2.1}$$

Where T_H and T_L are the absolute temperatures (K).

Carnot cycle is regarded as totally reversible cycle (see Figures 2.1a and b), and the efficiency of Carnot cycle is the highest efficiency of a cyclic device can have, also is used as a reference point to evaluate the real cycle performance.

2.2- Rankine Cycle

Rankine cycle is shown in more details in Figure (2.2)

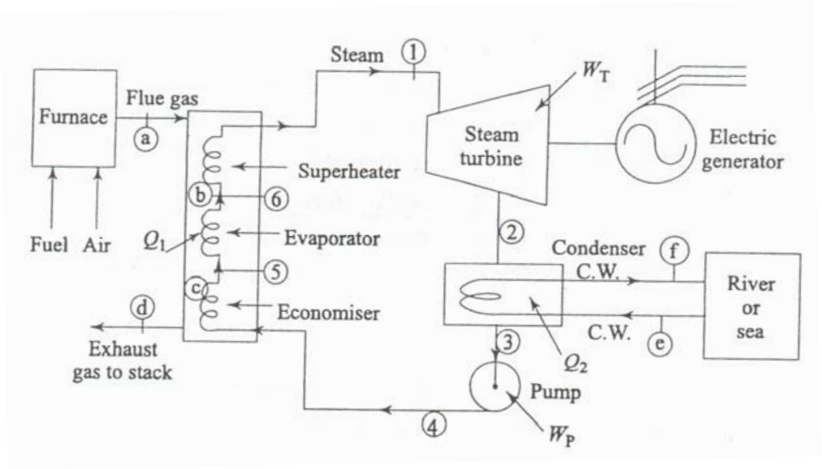
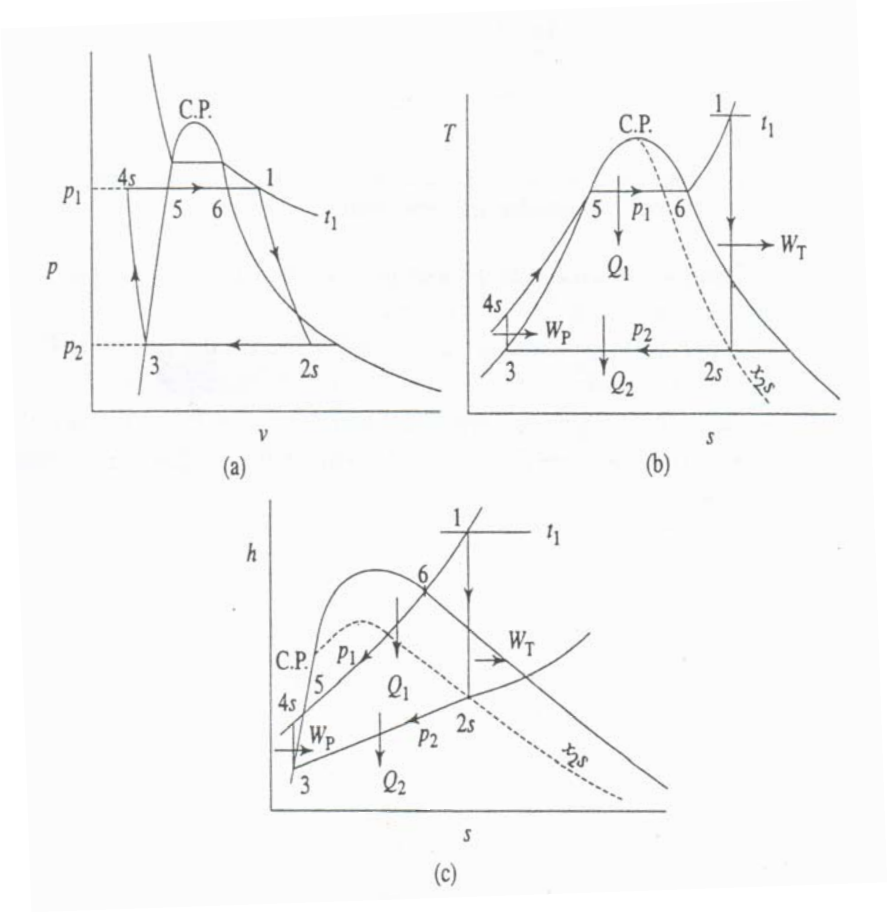


Figure (2.2) A simple steam power plant



Figures (2.3) Rankine cycle on *P-v, T-s and h-s* coordinates *(a,b,c)*

Rankine Cycle efficiency

$$\eta_{Rank} = \frac{\dot{W}_{net}}{\dot{Q}_B} = \frac{\dot{W}_t - \dot{W}_P}{\dot{Q}_B} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} \tag{2.2}$$

The enthalpy at point (4) can be calculated as follows,

$$h_4 = h_3 + w_p \tag{2.3}$$

$$w_p = v_3(P_4 - P_3) \tag{2.4}$$

$v_3 = v_f$ at P_3

2.3- Analysis of Rankine cycle

a- Cycle efficiency

The performance of the power plant is expressed in terms of the thermal or cycle efficiency which defined as the ratio of the net work output to the heat input this, thus

$$\eta_{cy} = \frac{\dot{W}_{net}}{\dot{Q}_B} \tag{2.5}$$

b- Net positive Heat Rate (NPHR)

An alternative and widely used performance parameter for the power plant is called "**N**et **P**ositive **H**eat **R**ate" or **P**lant **N**et **H**eat **R**ate (*PNHR*) which is the *rate of heat supplied per unit rate of net work output*. The $\dot{Q}_{in}$ and $\dot{W}_{net}$ are both expressed in the same units, the heat rate is simply the *reciprocal* of η_{cy} . In the SI units however, the heat rate was usually expressed as so many kJ of heat input per kWh of net work out in these units

$$\text{Net Positive Heat Rate} = \frac{3600}{\eta_{cy}} \qquad \left(\frac{kJ}{kW\ h}\right) \tag{2.6}$$

c- Boiler efficiency

$$\eta_B = \frac{\dot{Q}_B}{\dot{m}_f \ \ HHV} \tag{2.7}$$

where

$\dot{m}_f$: fuel mass flow rate (kg/s)

HHV: fuel Higher Heating Value (kJ/kg), which can be defined as the heat liberated in kJ by complete combustion of 1 kg of fuel (see Chapter III-section 3.1.2.2).

d- Overall efficiency

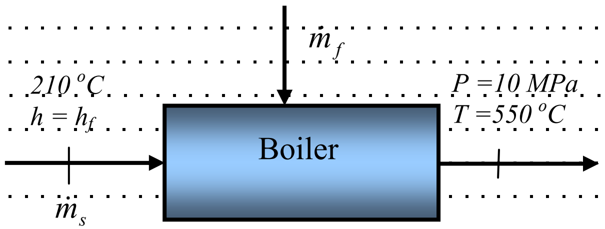
$$\eta_o = \frac{\dot{W}_{net}}{\dot{m}_f \ HHV} \tag{2.8}$$

Example 2.1

A cyclic steam power plant is designed to supply steam from the boiler at 10 MPa and 550 ^oC when the boiler is supplied with feed water at a temperature of 210 ^oC. It is estimated that the thermal efficiency of the cycle will be 38.4% when the net power output is 100MW. Calculate the steam consumption rate ($\dot{m}_s$). The enthalpy of the feed water may be taken as being equal to the enthalpy of saturated water at the same temperature.

The boiler efficiency is 87% and the *HHV* of the fuel is 42500 kJ/kg. Calculate the rate of fuel consumption in kg/s ($\dot{m}_f$) and ton/min.

Solution



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2.4-The effect of irreversibility

a- Frictional pressure drop in the steam pipe and across the governing throttle value between boiler and turbine.

b- Frictional effects in flow through the turbine and pump.

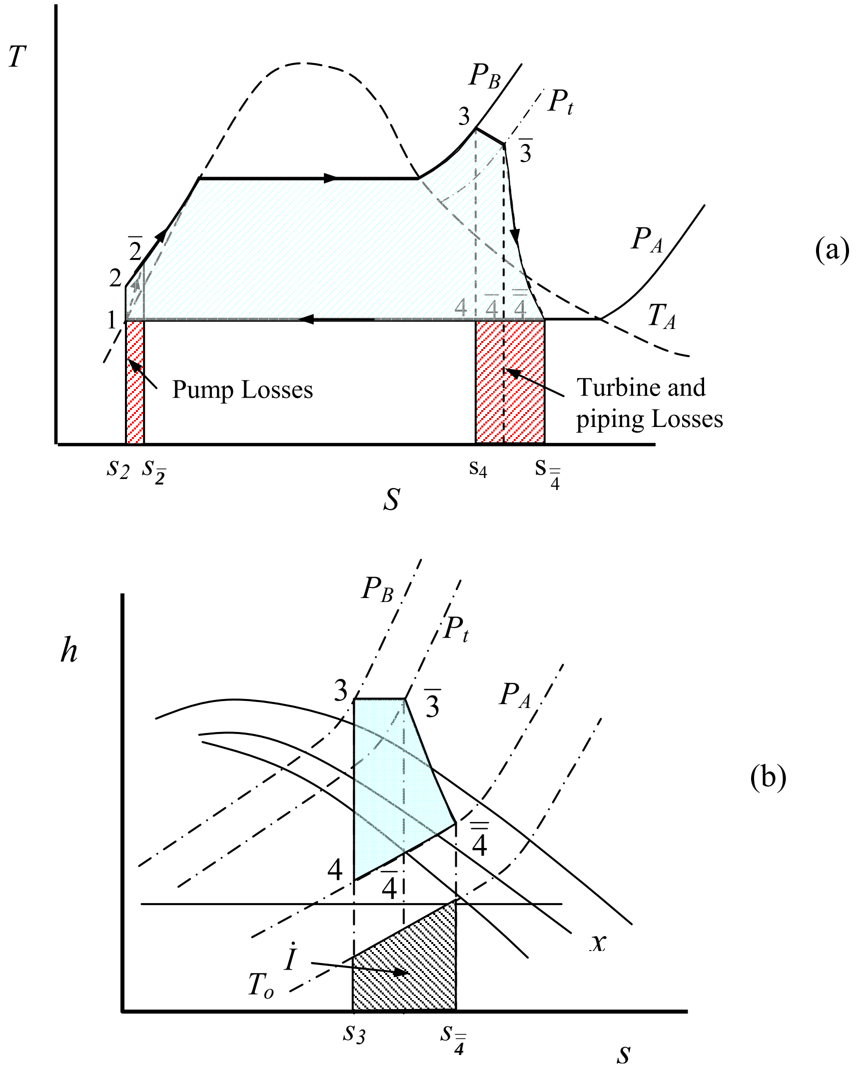

Figure (1.4)

2.5-Lost work due to irreversibility

If the condenser temperature T_A is taken to be the same as the temperature T_o of the environment, this is a special case of a more general and important theorem in the study of the availability (exergy).

$$\text{Lost Work} = \text{Increase in } \dot{Q}_A = T_A\left(\Delta S\right) \tag{2.9}$$

2.6-Alternative expressions for Rankine cycle efficiency in Terms of available energy (exergy)

In practice, the exhaust steam entering the condenser is always wet. The steam is then condensed at constant temperature in the ideal condenser. In these circumstances an alternative expression for the exact Rankine cycle efficiency may be written down from a study of availability (exergy). If , for purpose of analysis, the environment temperature T_o is taken as being the same the condenser temperature T_A, then the turbine, condenser and feed pump of the ideal Rankine cycle together constitute an ideal non-cycle open-circuit steady flow work-producing device exchanging heat reversibly with the environment at temperature T_A. In this device, all internal processes are reversible and the heat exchange with the environment is reversible since there is zero temperature difference between the fluid and the environment in the heat exchange process.

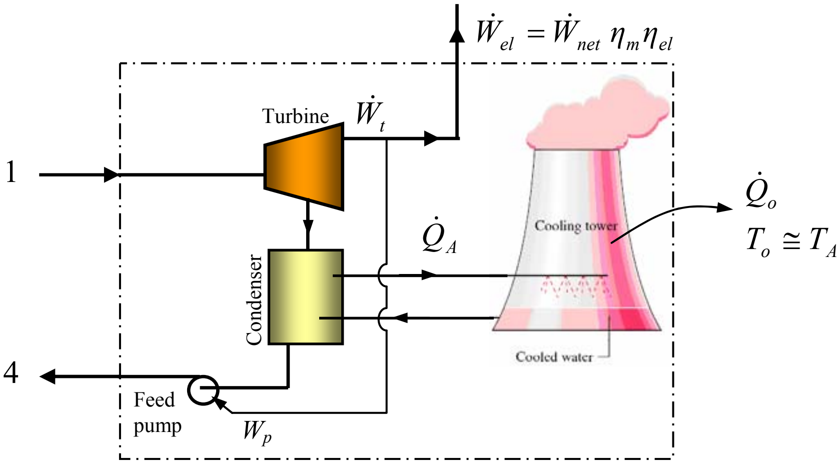

Figure (1.5)

The alternative Rankine efficiency expression is given by

$$\left(\eta_{Rank}\right)_{exact} = \frac{\left(\dot{W}_{net}\right)_{ideal}}{\dot{Q}_B} = \frac{\varepsilon_1 - \varepsilon_4}{h_1 - h_4} \tag{2.10}$$

where

$$\varepsilon = h - T_o s \tag{2.11}$$

$$\left(\eta_{Rank}\right)_{exact} = \frac{\dot{W}_{net}}{\dot{Q}_B} = \frac{\left(h_1 - h_4\right) - T_o\left(s_1 - s_4\right)}{h_1 - h_4} \tag{2.12}$$

2.7-Rational Efficiency (exergetic efficiency)(ψ)

The *rational efficiency* can be expressed in the following forms

$$\psi = \frac{\dot{E}_{out}}{\dot{E}_{in}} = \frac{\dot{W}_{el}}{\dot{m}_f \, \varepsilon_f} = \frac{\dot{W}_{el} \;\; \eta_B}{\dot{m}_s \left(h_1 - h_4 \right) \varphi} = \frac{\dot{W}_{net} \;\; \eta_B \eta_m \eta_{el}}{\dot{m}_s \left(h_1 - h_4 \right) \varphi} \tag{2.13}$$

where φ is the ratio of fuel chemical exergy (ε_o) to Net Calorific Value (NCV). Table (2.1) shows a typical values of φ for some industrial fuels and other combustible substances.

Fuel	φ
Different types of coal	1.06-1.1
Wood	1.15-1.3
Different fuel oils and petrol	1.04-1.08
Natural gas	1.04 ± 5 %
Hydrogen	0.985
Carbon Monoxide	0.973

Table (2.1)

$$\psi = \frac{Actual \;\; Work}{ideal \; Work} = \frac{Actual \; Work}{\varepsilon_1 - \varepsilon_4} = \frac{Actual \; Work}{\left(h_1 \text{ - } h_4 \right) - T_o \left(s_1 - s_4 \right)} \tag{2.14}$$

Example 2.2

Calculate the thermal efficiency of a Rankine cycle in which the steam is initially dry saturated at 2MPa and the condenser pressure is 7 kPa. Express this efficiency as a percentage of the Cannot cycle efficiency for The same limits of temp.

Assuming that, the feed pump work input = zero.

Solution:

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2.8- Specific Fuel and Steam Consumption

a. specific fuel consumption = $\dfrac{\dot{m}_f}{\dot{W}_{el}} = \left(\dfrac{kg}{kW\ h}\right)$ (2.15)

where $\dot{W}_{el} = \dot{W}_{net}\eta_m\eta_{el}$

and η_m mechanical efficiency , η_{el} electrical efficiency

b. specific steam consumption = $\dfrac{\dot{m}_s}{\dot{W}_{el}} = \left(\dfrac{kg}{kW\ h}\right)$ (2.16)

Example 2.3

In a steam power plant, the measured rate of steam supply was 7.1 kg/s when the net rate of work output (Actual $\dot{W}_{net}$) was 5000 kW. The feed water was supplied to the boiler at a temperature of 40 ºC, and the superheated steam leaving the boiler was at 1.4 MPa and 300 ºC. Calculate the thermal efficiency of the cycle, the corresponding net positive heat rate, the rational efficiency, and the turbine irreversibility.

Solution:

Example 2.4 (exergy)

Calculate the irreversibility of each control volume (A,B,C and D) and rational efficiency of the power plant that shown in Figure (2.6).

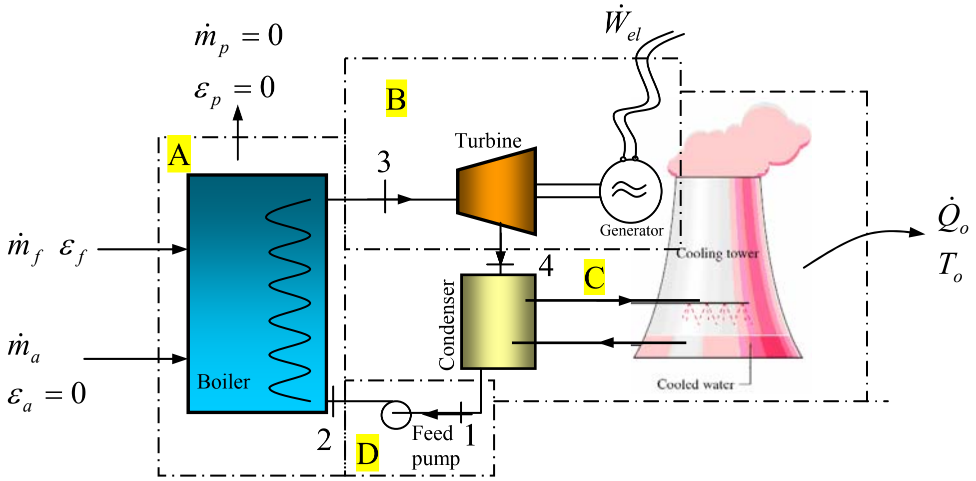


Figure 2.6

Solution:

2.9- The Reheat Rankine Cycle

In the last section we noted that the efficiency of the Rankine cycle could be increased by increasing the pressure during the addition of heat. However, the increase in pressure also increases the moisture content of the steam in the low pressure end of the turbine. The reheat cycle has been developed to take advantage of the increased efficiency with higher pressures, and yet avoid excessive moisture in the low pressure stages of the turbine. This cycle is shown schematically and on a *T-s* diagram in Figure 1.7. The unique feature of this cycle is that the steam is expanded to some intermediate pressure in the turbine and is then reheated in the boiler, after which it expands in the turbine to the exhaust pressure. Thus, the total heat input and total turbine work output for a reheat cycle become,

$$q_{in} = (h_3 - h_2) + (h_5 - h_4) \qquad (2.17)$$

$$w_{turb,out} = (h_3 - h_4) + (h_5 - h_6) \qquad (2.18)$$

It is evident from the *T-s* diagram that there is very little gain in efficiency from reheating the steam, because the average temperature at which heat is supplied is not greatly changed. The chief advantage is in decreasing to a safe value the moisture content in the low pressure stages of the turbine. If metals could be found that would enable us to superheat the steam to 5, the simple Rankine cycle would be more efficient than the reheat cycle, and there would be no need for the reheat cycle.

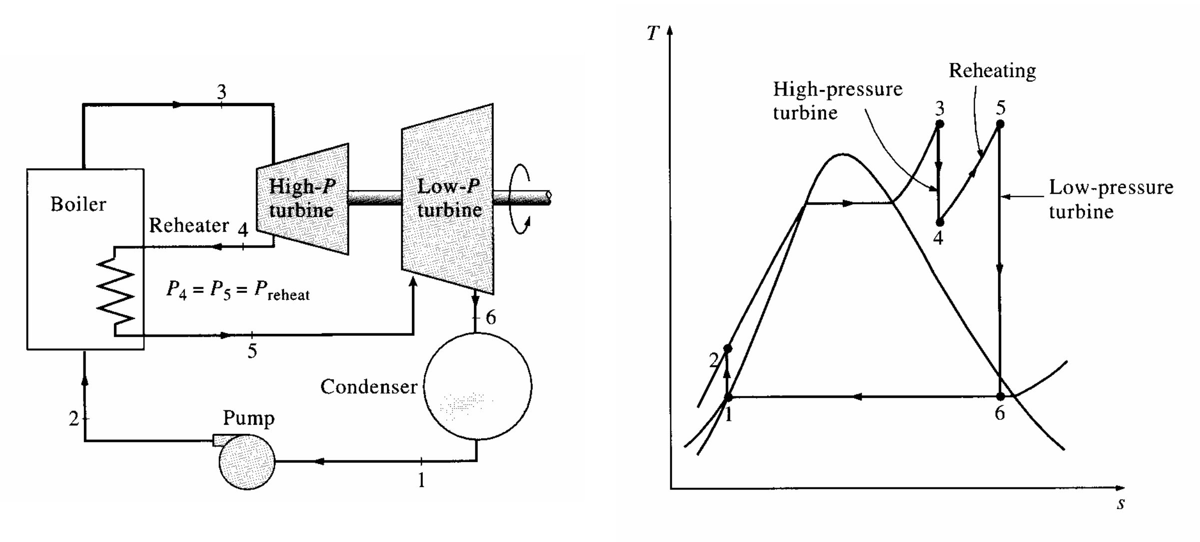


Figure (2.7 a)

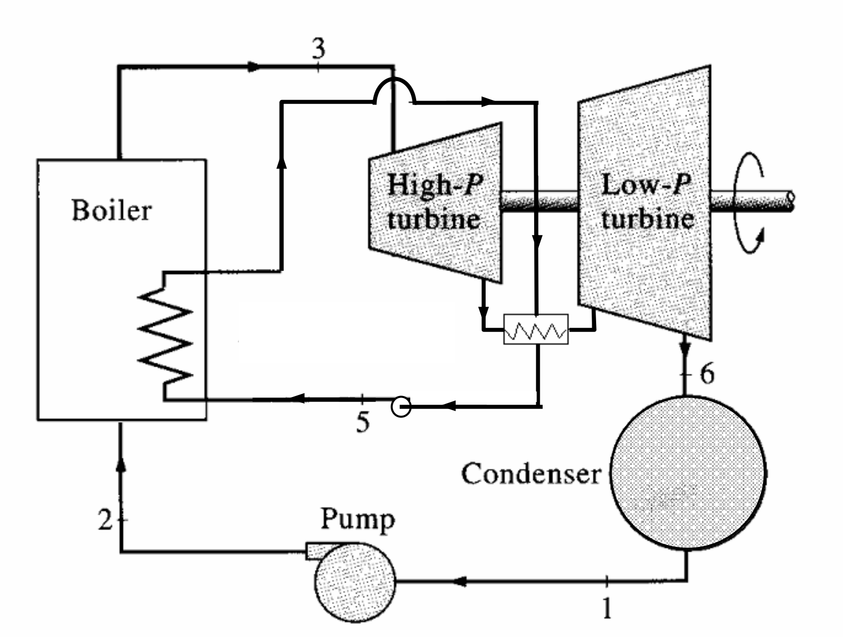


Figure (2.7 b)

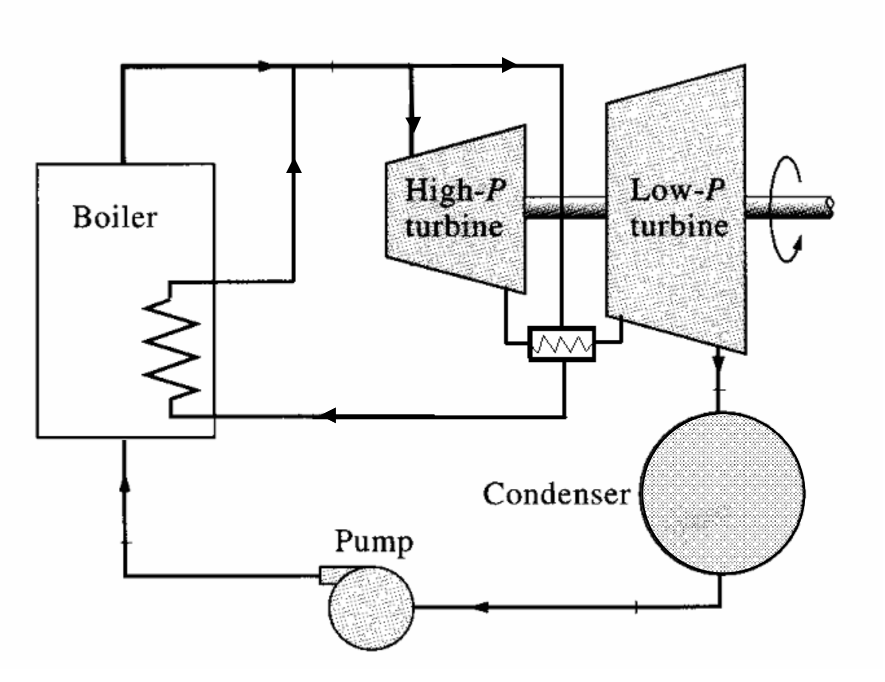


Figure (2.7 c)

Example 2.5

Consider a steam power plant operating on the ideal reheat Rankine cycle (see Figure 2.7a). The steam enters the high-pressure turbine at 15 MPa and 600 ^{o}C and is condensed in the condenser at a pressure of 10 kPa. If the moisture content of the steam at the exit of the low –pressure turbine is not exceed 10.4 percent, **determine** (a) the pressure at which the steam should be reheated and (b) the thermal efficiency of the cycle. Assume the steam is reheated to the inlet temperature of the high-pressure turbine.

Solution:

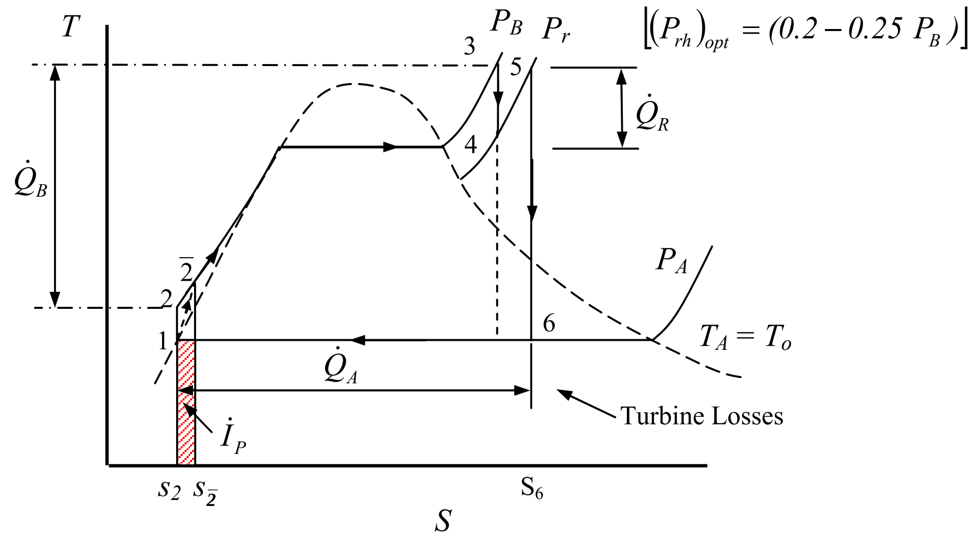


Figure (2.8)

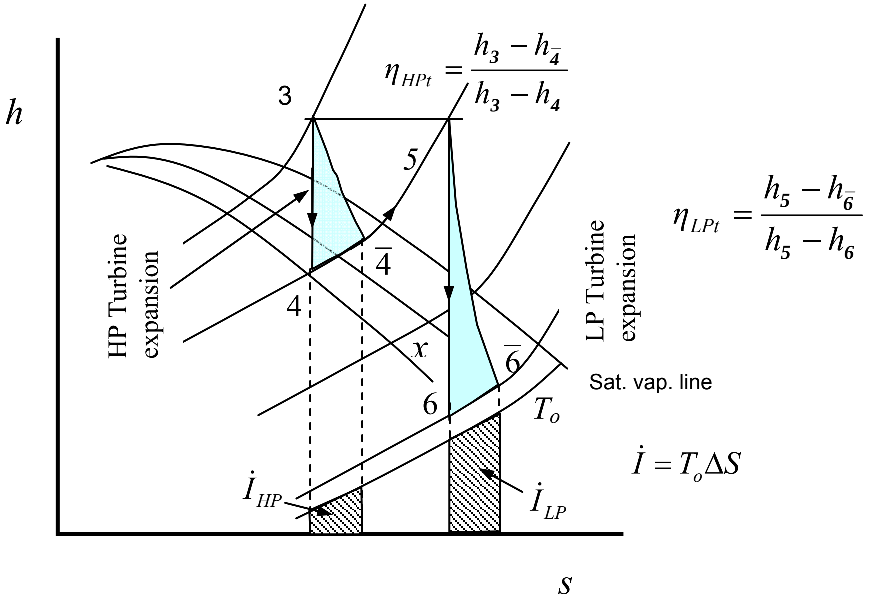


Figure (2.9)

Figure (2.9) show the expansion is not isentropic; in this case the cycle thermal efficiency can be expressed

$$\eta_{th,Rank} = \frac{\left(h_3 - h_{\bar{4}}\right) + \left(h_5 - h_{\bar{6}}\right)}{\left(h_3 - h_2\right) + \left(h_5 - h_4\right)} \tag{2.19}$$

Example 2.6

In a stream power plant, the steam mass flow rate is 80 ton/h, the steam inter the turbine at 6 MPa and 480 ºC, condenser pressure = 0.06 MPa, mechanical and electrical efficiencies are 0.96 & 0.95 respectively. Find the specific steam consumption and the power output.

Solution

Example 2.7

Find the cycle efficiency and specific steam consumption of a reheat cycle (Fig. 2.7a) operating between pressures of 3 MPa and 0.004 MPa, with superheat temp of 450 ºC. Assume that the first expansion is carried out to the point where the steam is dry saturated and then the steam is reheated to the original superheat temp. The feed pump term may be neglected.

Solution

1.10- The Regenerative Rankine Cycle

Another important variation from the Rankine cycle is the regenerative cycle which uses *feedwater heaters* as shown in Figure (2.10). In this cycle the coming water from the condenser before entering the boiler is heated by steam extracted or bled from the turbine at different stages and pushed to feedwater heater/s. In doing so, the thermal efficiency of the cycle will be improved significantly, reduces the steam flow to the condenser (needing smaller condenser) and reduces the temperature difference between the condenser (reduces the operating cost).

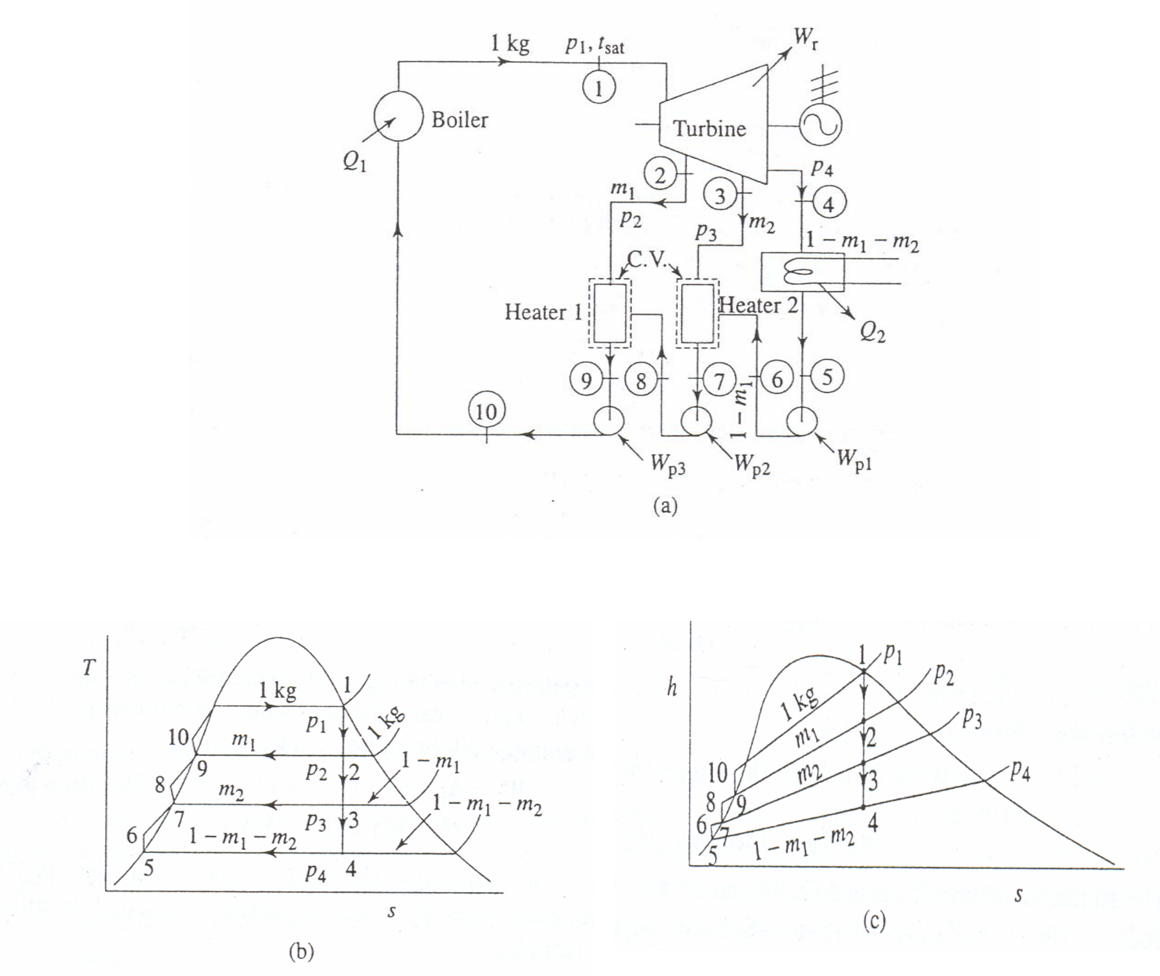


Figure (2.10) Regenerative cycle with two direct contact feedwater heaters

Feedwater heaters are two types; open (contact) heaters and closed heaters. In an open type heater, the extracted steam is allowed to mix with feedwater and both leave the heater at a common temperature (Figure 2.10 a). In a closed heater, the fluids are kept separate and are not allowed to mix together (Figure 2.11).

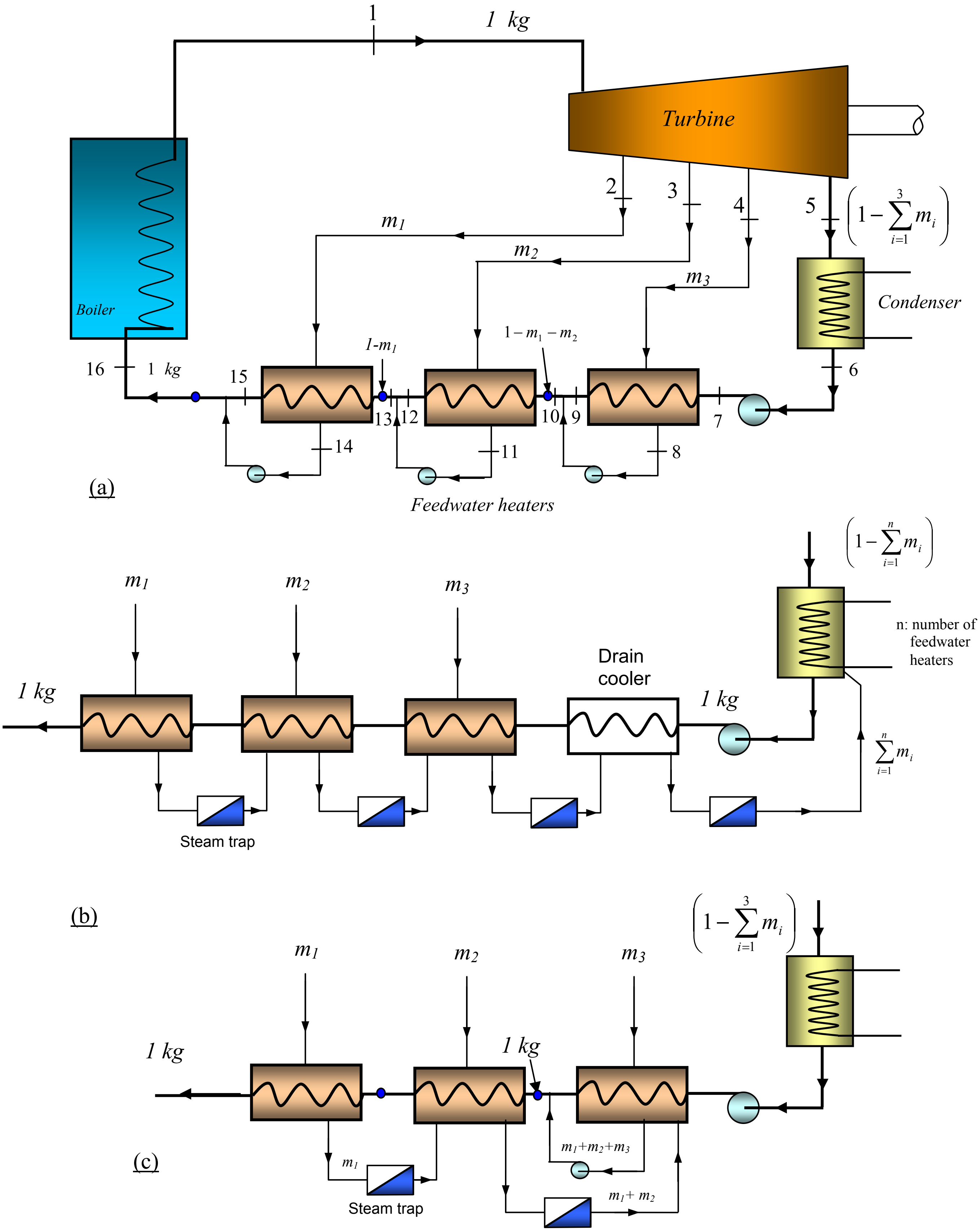
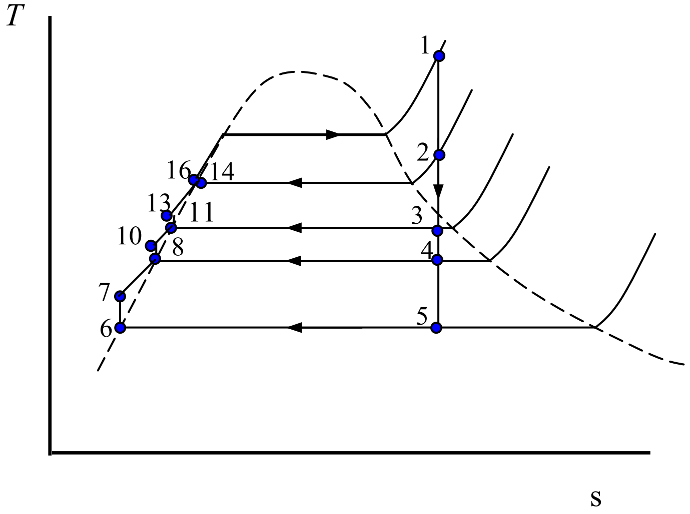

1
1 kg
Turbine
Boiler
2
3
4
5
$\left(1-\sum_{i=1}^{3} m_i\right)$
m1
m2
m3
Condenser
16
1 kg
15
1-m1
$1-m_1-m_2$
13
12
10
9
7
6
14
11
8
(a)
Feedwater heaters
$\left(1-\sum_{i=1}^{n} m_i\right)$
m1
m2
m3
n: number of feedwater heaters
Drain cooler
1 kg
1 kg
$\sum_{i=1}^{n} m_i$
Steam trap
(b)
$\left(1-\sum_{i=1}^{3} m_i\right)$
m1
m2
m3
1 kg
1 kg
(c)
m1
Steam trap
m1+m2+m3
m1+ m2

Figure (2.11)


T
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6
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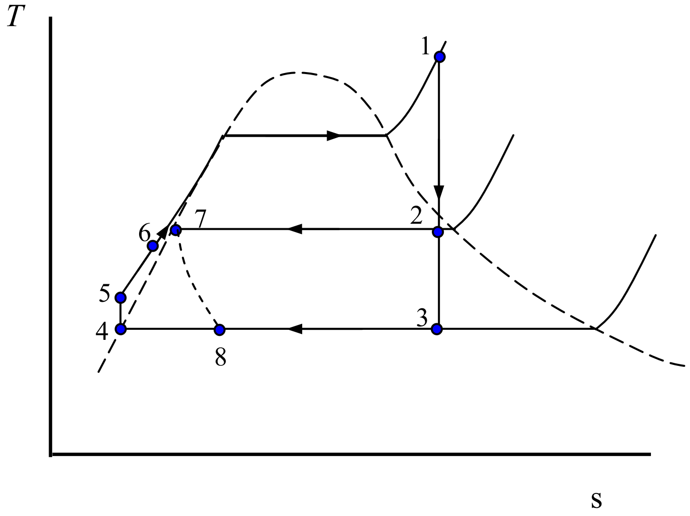
T
1
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8
s

Cycle efficiency

Refer to Figure (2.10 (a))

Pumps work

$$\dot{W}_p = \sum_{i=1}^{3} W_{pi} = (1 - m_1 - m_2)(h_6 - h_5) + [(1 - m_1)(h_8 - h_7)] + [1(h_{10} - h_9)] \tag{2.20}$$

Total turbine work

$$\dot{W}_t = 1(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4) \tag{2.21}$$

Total heat input

$$\dot{Q}_B = 1(h_1 - h_{10}) \tag{2.22}$$

$$\eta_{cy} = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_B} \tag{2.23}$$

Steam extracted

(1) the energy balance for heater 1 gives

$$m_1 h_2 + (1 - m_1) h_8 = 1 h_9 \tag{2.24}$$

$$m_1 = \frac{h_9 - h_8}{h_2 - h_8} \tag{2.25}$$

(2) the energy balance for heater 2 gives

$$m_2 h_3 + (1 - m_1 - m_2) h_6 = (1 - m_1) h_7 \tag{2.26}$$

$$m_2 = \frac{(1 - m_1)(h_7 - h_6)}{h_3 - h_6} \tag{2.27}$$

(see Figure 2.11 (a))

$$\dot{W}_p = \sum_{i=1}^{4} W_{pi} = \left(1 - \sum_{i=1}^{n} m_i\right)(h_7 - h_6) + [m_3(h_9 - h_8)] + [m_2(h_{12} - h_{11})] + [m_1(h_{15} - h_{14})] \tag{2.28}$$

$$\dot{W}_t = 1(h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4) + (1 - m_1 - m_2 - m_3)(h_4 - h_5) \tag{2.29}$$

$$\dot{Q}_B = 1(h_1 - h_{16}) \tag{2.30}$$

$$\eta_{cy} = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_B} \tag{2.31}$$

Example 2.8

Consider a regenerative cycle using steam as the working fluid (see Figure 2.12). Steam leaves the boiler and enters the turbine at 4 MPa, 400 ^{o}C. After expansion to 400 kPa, some of the steam is extracted from the turbine for the purpose of heating the feedwater in an open feedwater heater. The pressure in the feedwater heater is 400 kPa and the water leaving it is saturated liquid at 400 kPa. The steam not extracted expands to 10 kPa. Determine the cycle efficiency.

Solution

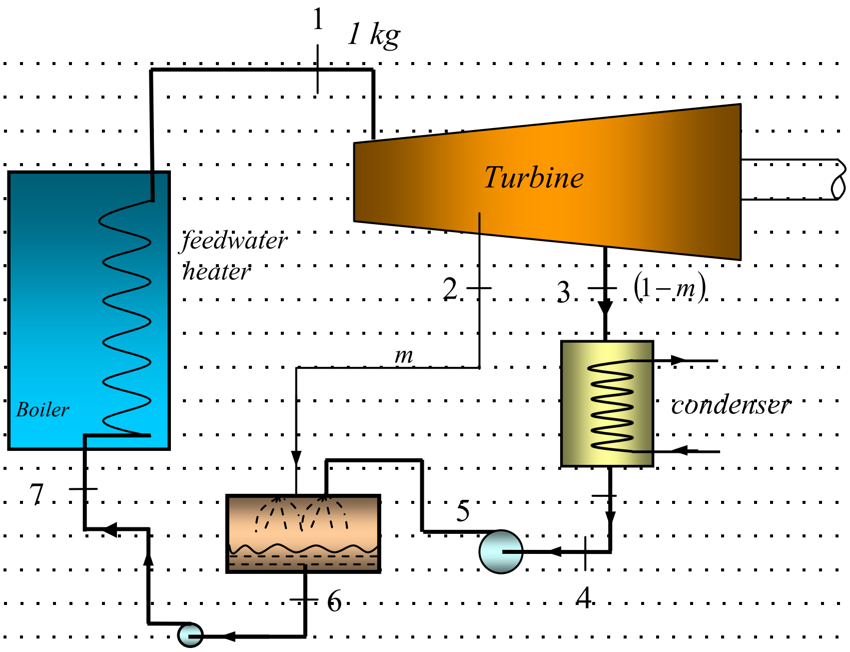


Figure (2.12).

Example 2.9

Steam power plant has the following data:

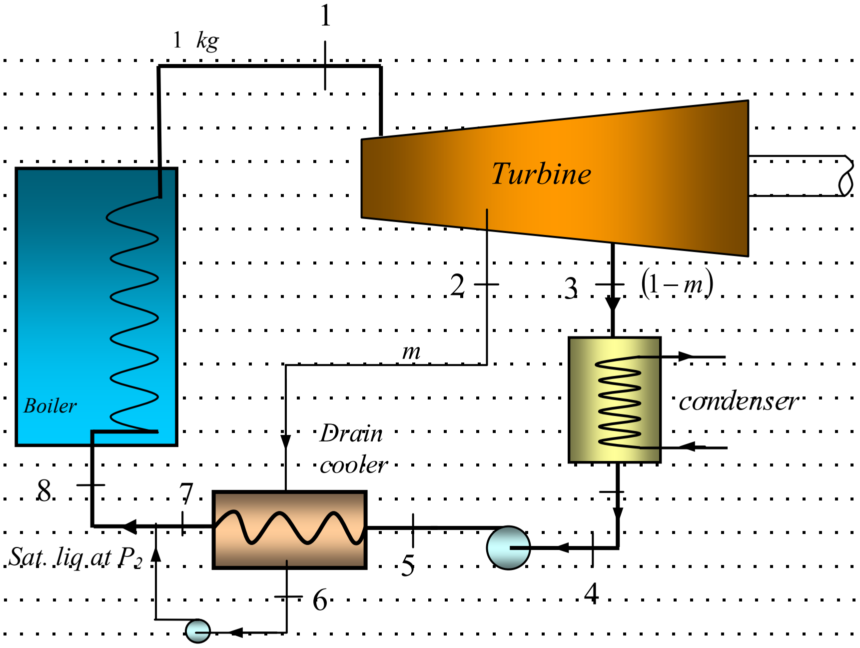
- Steam condition at turbine inlet 6 MPa, 450 ºC.
- Condenser pressure 0.008 MPa.
- Turbine efficiency 0.83.
- Mechanical efficiency 0.92.
- Electrical efficiency 0.96.

The plant is fitted with one closed feedwater heater using bled steam and having its drains pumped into the free line after the heater. Steam may be bled at 0.8 MPa.

Sketch the cycle and calculate the following:

a- The amount of bled steam per kg of exhaust steam.
b- Cycle efficiency.
c- The percentage improvement in efficiency due to feed heating.

Solution



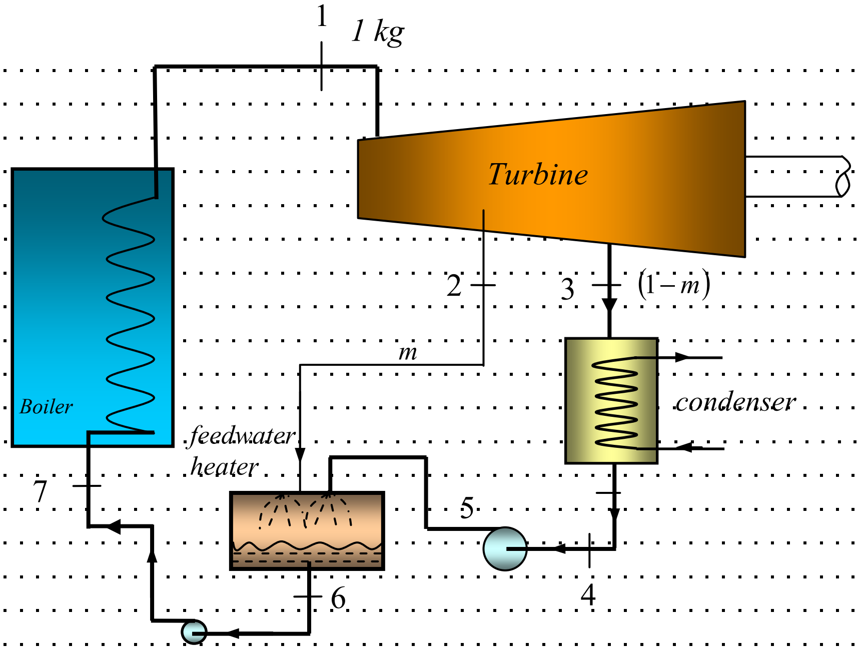
Example 2.10

In a hypothetical cyclic steam power plant incorporating a single direct-contact feedwater heater, the steam leaves the boiler at 1 MPa and 400 ºC and the condenser pressure is 4 kPa. Expansion in the turbine is reversible and adiabatic. The feedwater heater takes steam bled from the turbine at pressure of 75 kPa and heats the feedwater to the corresponding saturation temperature. Temperature and enthalpy changes of the fluid in passing through any pumps may be neglected.

Calculate:

a. The mass of steam bled from the turbine per kg of steam leaving the boiler.
b. The cycle efficiency.
c. The improvement in thermal efficiency due to the introduction of this single stage of feed heating, expressed as a percentage of the Rankine cycle efficiency.

Solution:



Example 2.11

During a test on steam turbine equipped with a single stage of regenerative feed heating, the following results were obtained.

Steam pressure at stop value = 2MPa.
Steam temperature at stop value = 315 °C.
Steam pressure in nozzle box = 1.65 MPa.
Bled steam pressure = 0.27 MPa.
Exhaust pressure = 0.01 MPa.
Temperature of condensate = 38 °C.
Temperature of drain leaving drain cooler = 44 °C.
Temperature of feed water leaving feedwater heater =126 °C.
Steam entering turbine = 25000 kg/h.

<u>Calculate</u>

The quantity of steam bled off, the power output if the work done by the steam which completes the expansion of 630 kJ/kg, and the thermal efficiency.

Solution:

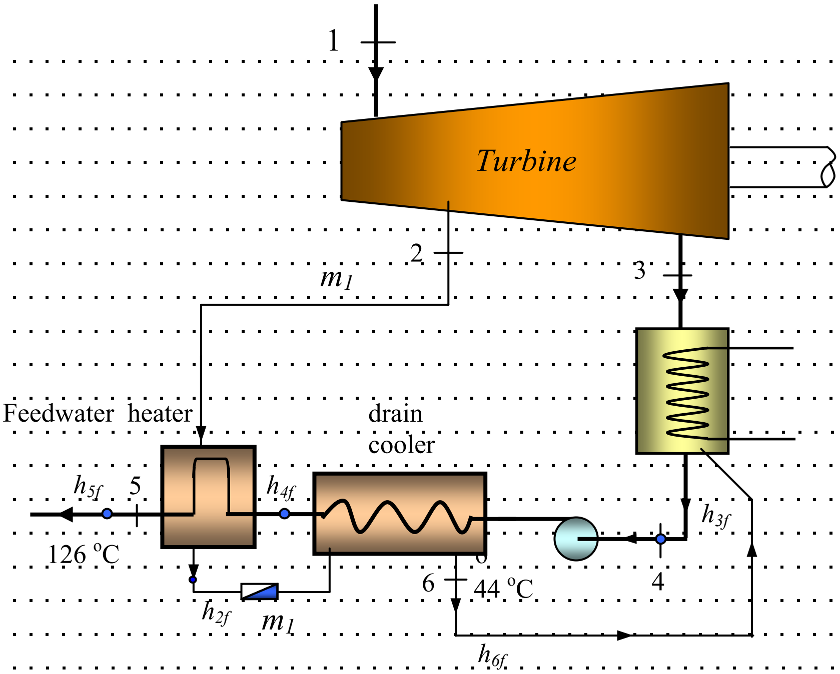
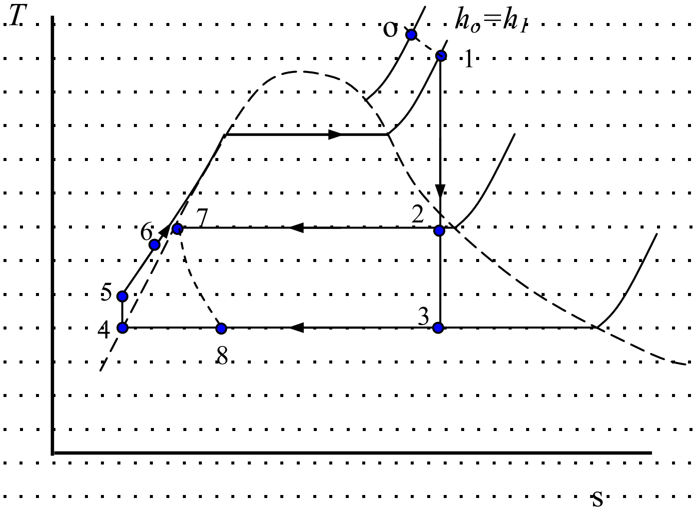


Figure 2.15


T
o
$h_o=h_1$
1
2
3
4
5
6
7
8
s

Example 2.12

A small steam power plant uses a single, closed type regenerative feedwater heater as shown in Figure (2.13). Steam bled from the turbine enters the feedwater heater at a pressure of 0.6 MPa as saturated vapor and condenses there and leaving as saturated liquid. Feed water enters the feed pump at a pressure of 0.005 MPa in the saturated liquid state. On leaving the feed pump it is heated in the feedwater heater to a temperature of 7°C below the saturation temperature of the bled steam. The heat lost from the feedwater heater to the environment amounts to 3 percent of the enthalpy change of the bled steam in the feedwater heater. Neglecting the feed pump work and the pressure losses of both steams of water substance in the feedwater heater.

Calculate:

a- The fraction of steam bled from the turbine for feed heating and

b- The irreversibility in the feedwater heater per unit mass of steam generated in the boiler (see appendix A equation 11).

The enthalpy and the entropy of compressed liquid may be taken to be equal to the saturated liquid values of these properties at the same temperature. T_o=298.15K"

Solution:

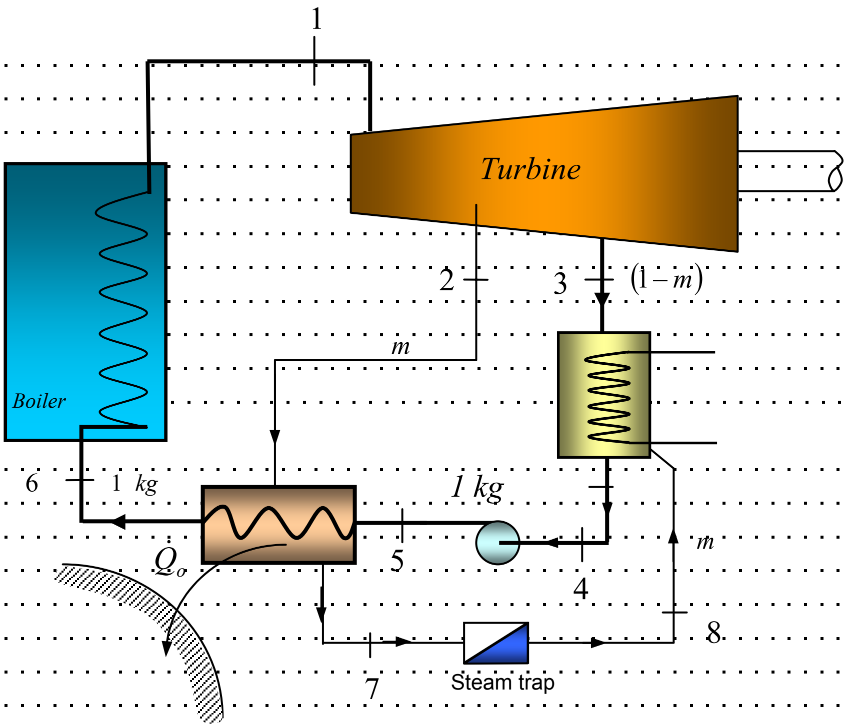


Figure 2.13 a

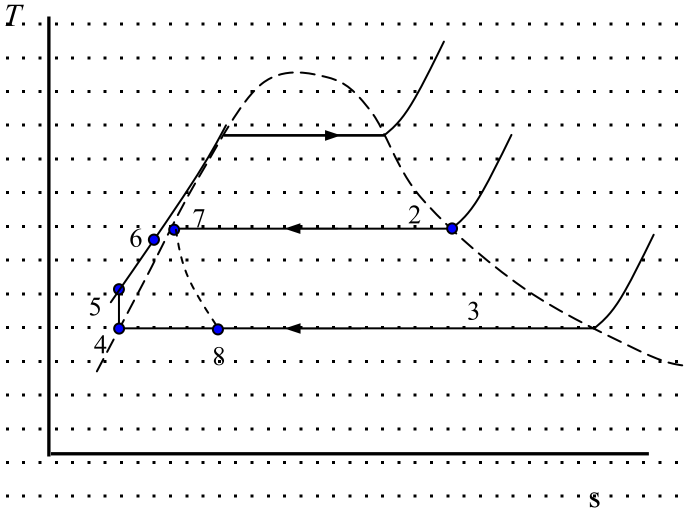


Figure 2.13 b

Revision Questions

2.1- Consider a reheat cycle utilizing steam. Steam leaves the boiler and enters the turbine at 4 MPa, 400 ^{o}C. After expansion in the turbine to 400 kPa, the steam is reheated to 400^{o}C and then expanded in the low pressure turbine to 10 kPa. Determine the cycle efficiency.

Ans: 35.9%

2.2 The inlet pressure to the adiabatic and revisable turbine is 4 MPa. The entropy of the steam at inlet is 6.9362 kJ/kg K, and the turbine exhausts at a pressure is 7 kPa. If the quality of the steam at the exhaust is 0.91, calculate the lost work due to irreversibility per kg of steam flowing through the turbine, what is the Rational efficiency of the turbine.

Ans: 201.7 kJ/kg , 82.9 %

2.3- A Steam Power plant is equipped with a single regenerative feedwater heater operating under the following conditions,

Turbine inlet = 2 MPa .
Turbine inlet Temp = 100 superheat.
Extraction pressure = 0.2 MPa.
Exhaust pressure = 0.00 MPa.

Compare the regenerative and non – regenerative cycle with respect to:

a - Cycle thermal efficiency.
b - Specific steam consumption in kg / kW h.

Assume isentropic expansion and no temperature difference between the heating steam and the feed water.

Ans: 0.346, 4.07 kg / kWh

2.4- Steam is supplied to a turbine 2.7 MPa and 380 ^{o}C, the steam is expanded isentropically to 0.0035 MPa. At a stage of the turbine where the pressure is 0.27 MPa, a connection is made to a surface feedwater heater in which the feed water is heated by bled steam to 125 ^{o}C. The condensed steam from feed water heater is cooled in a drain cooler to heat the main condensate before entering the feedwater heater. The cooled drain water combines with the condensate in the condenser. Calculate

a - The mass of bled steam per kg of steam entering the turbine Assume the temp of drain leaving drain cooler = 50 C

b – Thermal efficiency of the cycle.

Ans: 0.1673, 0.38.3

2.5- A steam power plant has *four* regenerative heating stages with:

Output at generator terminal = 25000 kW
Steam consumption at boiler stop value = 98 ton/h
Steam entering turbine at 6 MPa / 490 ^{o}C
Steam leaving turbine at 0.0035 MPa / $h_i =$ 2282 kJ/kg
Temperature of feed water entering first heater = 30 ^{o}C
Temperature of feed water leaving last heater = 200 ^{o}C
Pressure at bleeding points 2, 0.7, 0.19, 0.05 MPa

Assume equal rise of temperature in all heaters, condensate is pumped directly after each heater to the main stream. Calculate

a – Bled steam to each heater.
b – Thermal efficiency of the plant, given that $\eta_{mech}\ \eta_{el} = 0.95$.
c – Fuel consumption per hour when running at full load give that the calorific value of the fuel is 41868 kJ /kg

Ans: a : 0.108 , 0.099 , 0.0922 and 0.0856 kg/ kg condensate
b : 32.3 %
c : 2.1492 Ton/h

2.6- 15000 kW steam turbine plant consist of one high and one low pressure turbine. The steam leaves the boiler at 3 MPa and 500 °C. The steam does not go directly to high pressure turbine but first passes through a steam reheater in which it gives up heat to reheat the steam coming out of high pressure turbine before it inters the low pressure turbine. The line steam enters high pressure turbine at 3 MPa and 370 °C. The pressure of steam leaving the high pressure turbine is 0.7 MPa, the condenser pressure is 0.007 MPa. The efficiency of the high pressure turbine is 0.79 and that of the low pressure turbine is 0.76. Find;

a- The conditions of steam at which at leaving the high pressure turbine and at which enters the low pressure turbine.
b- The thermal efficiency of the cycle.
c- The heat rate in kJ/kW h.

Ans: a- 2888.5 kJ/kg, 3178.5 kJ/kg, 2545 kJ/kg
b- 0.276, c- 13043.478 kJ/kW h

2.7- A simplified diagram of a steam power plant and the operating pressures and boiler exit temperature are shown in Figure (2.14). The steam bled from the turbine for feed heating is in saturated vapor state whilst that discharged by the turbine into the condenser has a quality of 0.8. Both the steam entering the condenser as well as that bled into the feedwater heater condense to the saturated liquid states, whilst the feed water is preheated to the saturation temperature of the bled steam. Assume that.

(i) the compression process in the feed pump is isentropic .
(ii) heat losses and pressure losses are negligible .
(iii) the enthalpy and entropy of compressed liquor may be taken to be equal to those of the saturated liquid at the same temperature and T_o = 290 K.

Calculate per mass of steam generated in the boiler

a- The fraction of steam bled for feed heating.
b- The heat transferred from the cooling tower to the surroundings.
c- The feed pump work
d- The irreversibility's[◆] occurring in the regions A to F marked in Figure (2.14)
e- η_{cy}
f- $\eta_{ideal\ cycle}$, and g- Rational efficiency

[◆] $\dot{I} = T_o\left(s_e - s_i - \frac{\sum \dot{Q}_o}{T_o}\right)$ see appendix A

Ans: a- 0.304, b-1544 kJ/kg, c-11.495 kJ/kg, d-89, 0, 90, 33, 124.3, 817 kJ/kg e- 0.37, f- 0.5, g- 0.73

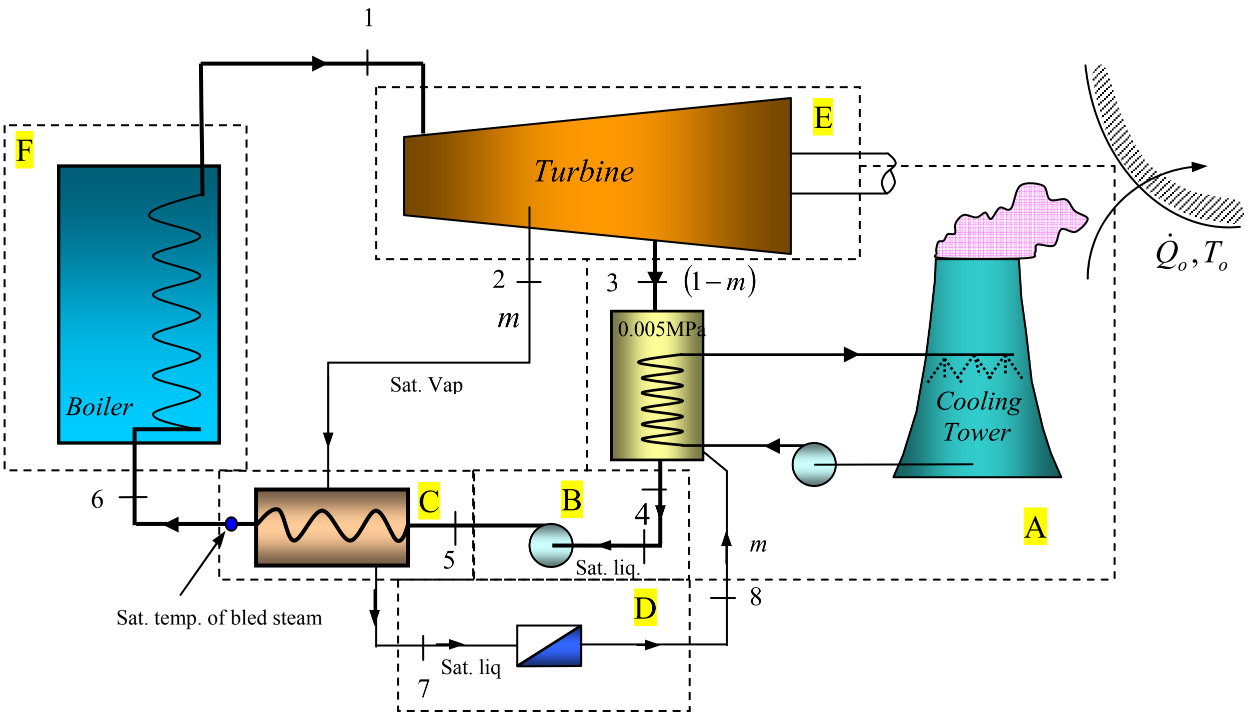


Figure 2.14

2.8- Steam is being bled off from two stages of a steam turbine for feed heating purposes which operates under the following conditions:

Initial steam pressure = 2.65MPa
Initial steam temp = 370 °C
Steam pressure in first heater = 0.53 MPa
Steam pressure in second heater = 0.1 MPa
Exhaust pressure = 0.006 MPa
Condensate temp = 35 °C
Feed temp after second heater = 95 °C
Feed temp after first heater = 150 °C

The bled steam is condensed in the feedwater heater and there is no cooling of the condensate. The drains from the first heater are passed through a steam trap in the second heater and the combined drains from the second heater are pumped by a drain pump into the feed pipe after the second heater. If the internal power generated by the turbine is 12 MW.

Find the required capacity of heater drain pump. The efficiency of the turbine is 0.8 and the condition curve may be assumed to be a straight line (see Figure (2.15)) .

Ans.
m_1 = 0.1035 kJ/kg of entering steam
m_2 = 0.0758
Capacity of drain pump = 2.852 kg/s

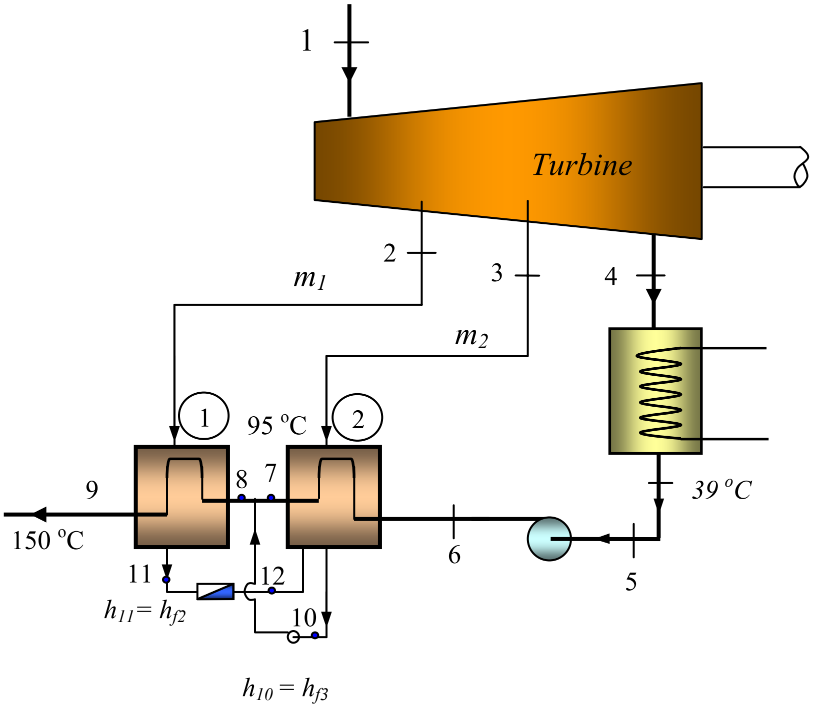


Figure 2.15

2.9- In an ideal, reversible non-regenerative steam cycle the initial steam pressure and temperature are 15MPa and 500 °C respectively and the condenser pressure is 4 kPa. If the work input to the feed pump may not be neglected, calculate the percentage exhaust quality and the cycle efficiency.

a- when there is no reheating , and
b- when the steam is reheated to 500 °C and 6 MPa .

Ans: a- 73.6% , 43.59%, b- 80.2% , 44.94%

2.10-A small steam power plant uses a single, closed type regenerative feed water heater as shown in Figure (2.16). Steam bled from the turbine enters the feed water heater at a pressure of 0.8 MPa where it condenses and get subcooled to a temperature of 90 °C. The water then leaves the condenser and enters the feed pump at a pressure of 0.005 MPa in the saturated liquid state. On leaving the feed pump it is heated in the feedwater heater to a temperature of 155.5 °C. The stray heat transfer from the feedwater heater to the environment amounts to 4 percent of the enthalpy change of the bled steam in the feedwater heater. Neglecting the feed pump work and the pressure losses of both steams of water substance in the feedwater heater.

Calculate:

a- The fraction of steam bled from the turbine for feed heating and
b- The irreversibility in the feedwater heater per unit mass of steam generated in the boiler.
c- The irreversibility of throttling process, assuming it to be adiabatic.

The enthalpy and the entropy of compressed liquid may be taken to be equal to the saturated liquid values of these properties at the same temperature. T_o=298 K"

Ans: 0.2256 kg/kg steam, 76.98 kJ/kg steam, 4.529 kJ/kg steam

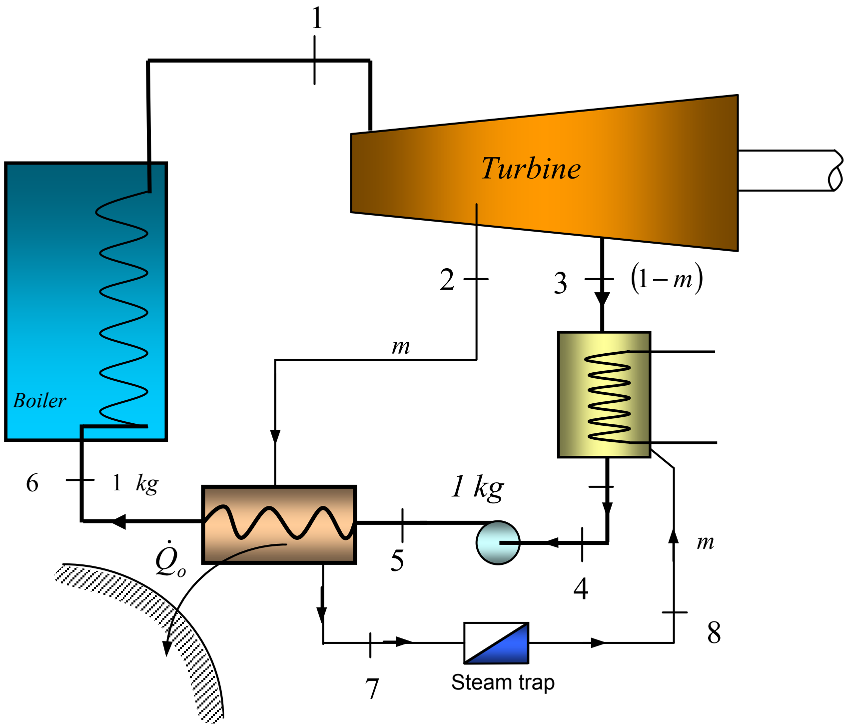


Figure 2.16 a

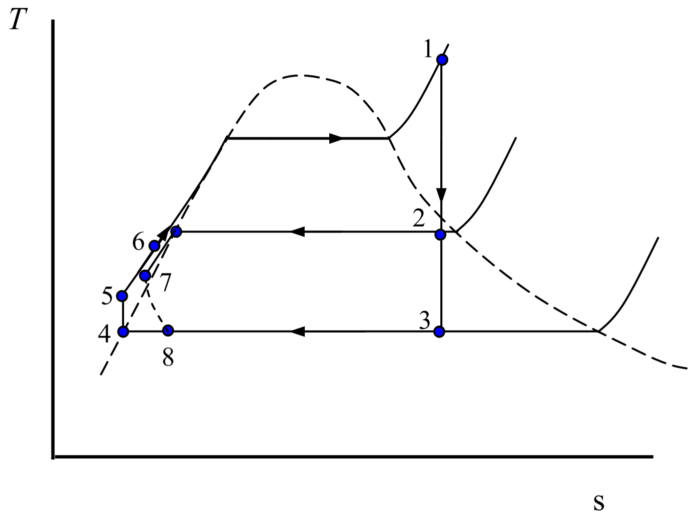


Figure 2.16 b

2.11- Consider a power plant with 550 °C and 14 MPa throttle, reheat at 1.4 MPa back to 550 °C and 10 kPa condenser pressure (see Figure 2.17). The plant has two closed feed water heaters, both with terminal temperature differences of 10 °C ($\left[\left(T_{14}-T_{12}\right) and \left(T_{11}-T_{9}\right)\right]$. The high pressure (HP) heater condensate is throttled into the low pressure (LP) heater, which in turn drains into the condenser. The efficiency of HP turbine, LP turbine and the feed water pump are 0.88, 0.9 and 0.8 respectively. Evaluate the feed water heater mass fractions, cycle efficiency and net work. (Assume $T_9 = \frac{T_{sl}-T_7}{3}+T_7$ and $T_{12} = \frac{T_{sl}-T_7}{3}+T_9$

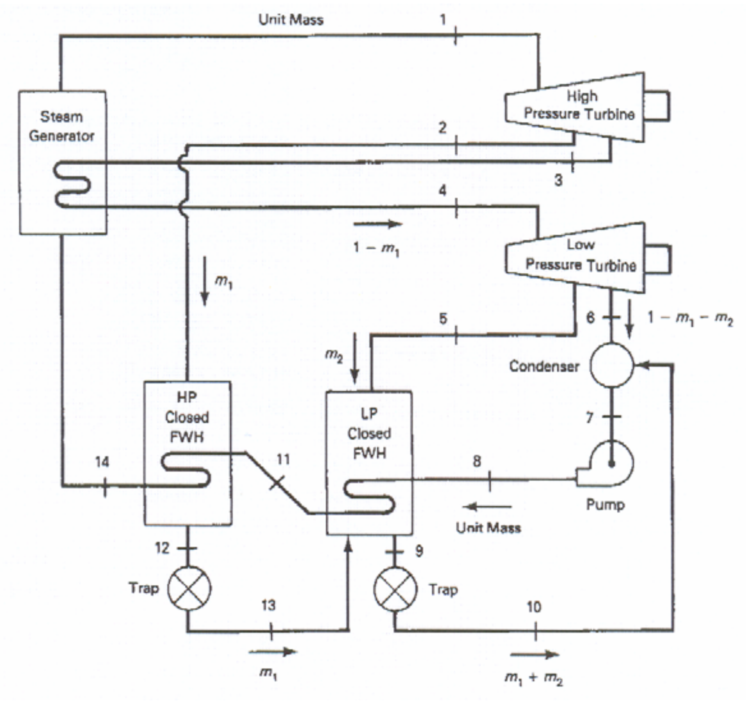


Figure 2.17 a

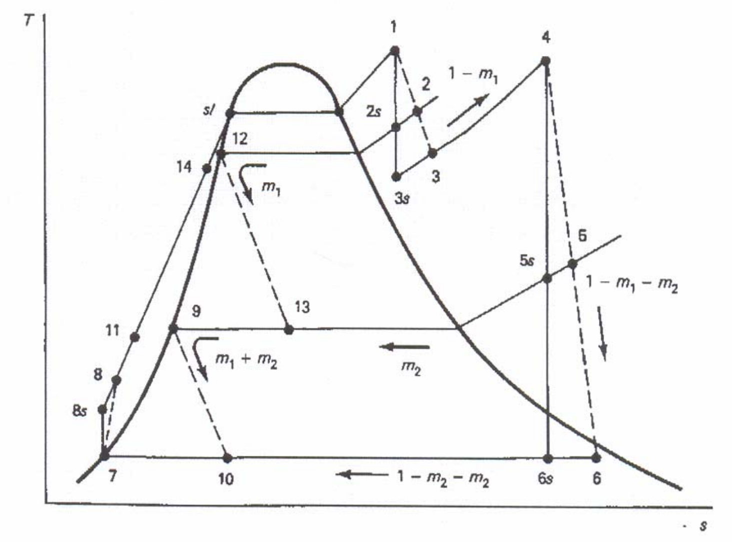


Figure 2.17 b

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Power Plant Engineering and economy

MET401

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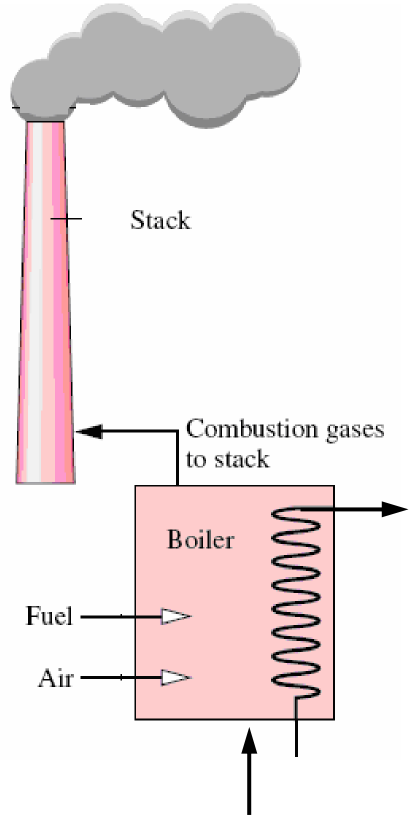

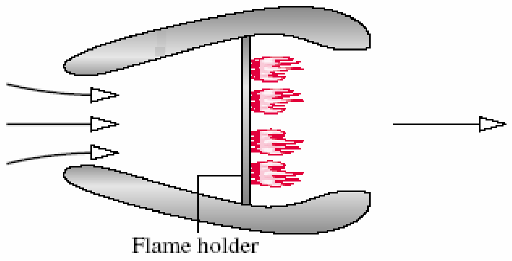

Chapter III- Fuels and Combustion

2010

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Chapter III

Fuels and Combustion

3.1-Fuels

Fuel is simply a combustible substance. In this Chapter emphasis is on hydrocarbon fuel which contains hydrogen and carbon. Sulfur and other chemical substances also may be presented. Hydrocarbon fuels can exist as liquids, gases, and solids.

Liquid hydrocarbon fuels are commonly derived from crude oil through distillation and cracking processes. Examples are gasoline, diesel fuel, kerosene, and other types of fuel oil. Most liquid fuels are mixtures of hydrocarbons for which compositions are usually given in terms of mass fractions. For simplicity in combustion calculations, gasoline is often modeled as octane, C_8H_{18}, and diesel fuel as dodecane, $C_{12}H_{26}$.

Gaseous hydrocarbon fuels are obtained from natural gas wells or are produced in certain chemical processes. Natural gas normally consists of several different hydrocarbons, with the major constituent being methane, CH_4. The compositions of gaseous fuels are usually given in terms of mole fractions. Both gaseous and liquid hydrocarbon fuels can be synthesized from coal, oil shale, and tar sands.

Coal is a familiar solid fuel. Its composition varies considerably with the location from which it is mined. For combustion calculations, the composition of coal is usually expressed as an ultimate analysis. The ***ultimate analysis*** gives the composition on a mass basis in terms of the relative amounts of chemical elements (carbon, sulfur, hydrogen, nitrogen, oxygen) and ash.

3.1.1-Fuel requirements

- Heating Value (HV) or Calorific Value (CV)
- Price-It should be cheaper
- Burning efficiency
- Minimum ash
- Handling and storage cost

3.1.2- Coal

Coal is the principal energy source, particularly in Europe, China and India because of its large deposits and availability. Coal is originated from vegetable matter which grew millions of years ago. Trees and Plants falling into water decayed and later produced peat bogs. Huge geological upheavals buried these bogs under layers of silt. Subterranean heat, soil pressure and hardened it to form brown coal or lignite.

3.1.2.1- Coal ultimate analysis

The ultimate analysis gives the chemical elements that comprise the coal substance, together with ash and moisture. The coal substance consists of organic compounds of carbon, hydrogen, and oxygen derived from the original vegetable matter. The analysis shows the following components on *mass basis*: carbon (C), hydrogen (H), oxygen (O), nitrogen (N), sulfur (S), moisture (M) and ash (A). Therefore

$$C + H + O + N + S + M + A = 100\% \text{ by mass} \tag{3.1}$$

The ***dry*** and ***ash*** free analysis on combustible basis is obtained on dividing C,H,O,N and S by the fraction $\left(1-\frac{M+A}{100}\right)$.

3.1.2.2- Heating Value or Calorific Value

It is defined as the heat liberated in ***kJ*** by complete combustion of 1 ***kg*** of fuel (solid or liquid). For gaseous fuels, the normal practice is to express calorific value in ***kJ/m³***.

Calorific value is further classified as (i) Higher Heating Value (HHV) and (ii) Lower Heating Value (LHV).

HHV: It is the total amount of heat produced when unit quantity of fuel is completely burnt and the products of combustion are cooled to room temperature (25 °C).

LHV: It is the net amount of heat produced when unit quantity of fuel is completely burnt and the products of combustion are ***NOT*** cooled to room temperature but are allowed to escape.

The ***H****eating* ***V****alue* or ***C****alorific* ***V****alue* of coal is a property of fundamental importance. It may be determined on as-received, dry, or dry-and- ash-free basis. It is the heat transferred when the products of complete combustion of a sample of coal (or other fuel) are cooled to the initial temperature of air and fuel. It is normally determined in a standard test in a bomb calorimeter, where a coal sample of known mass is burnt with pure oxygen supply completely in a stainless steel bomb or vessel surrounded by a known mass of water, and the rise in water temperature is noted. Two different heating values are cited for coal. The higher heating value (HHV) assumes that the water vapor in the products condenses and thus includes the latent heat of vaporization of the water vapor formed by combustion. The lower heating value (LHV) assumes that the water vapor formed by combustion leaves as vapor itself. Therefore,

$$LHV = HHV - m_w h_{fg} \tag{3.2}$$

where m_w is the mass of water vapor formed given by

$$m_w = M + 9H + m_a W \tag{3.3}$$

where M and H are the mass fractions of moisture and hydrogen in the coal, W is the specific humidity of atmospheric air and m_a is the actual amount of air supplied per kg of coal. For energy balance and efficiency calculations of steam generators, HHV of fuel is considered in the USA, whereas LHV is the standard used in European practice.

If the ultimate analysis is known, the HHV of anthracite and bituminous coals can be determined approximately by using ***Dulong and Petit*** formula as given below:

$$HHV = \left(33.83C + 144.45\left(H - \frac{O}{8}\right) + 9.38\,S\right) \times 10^3 \quad \text{in kJ/kg} \tag{3.4}$$

where C H O and S are mass fractions of carbon, hydrogen, oxygen and sulphur in coal. Assuming the latent heat of vaporization h_{fg} at the partial pressure of water vapor in the combustion products as 2442.3 kJ/kg, the lower heating value of coal from equation 3.2 is given by

$$LHV = HHV - 2442.3\,m_w \tag{3.5}$$

If the moisture (M) in the coal fuel and water vapor of atmospheric air $\left(m_a W\right)$in the supplied air both are zero, then lower heating value can be expressed as

$$LHV = HHV - 2442.3\left(9\,H\right) \tag{3.6}$$

3.1.3- Fuel Oil

Petroleum is believed to have been formed during past geological ages from decayed marine life, both vegetable and animal. Dead marine animals and vegetable matter accumulated for millions of years ultimately got transformed into oil, mainly in sedimentary rocks, by pressure and heat. Oil deposits accumulated in the rocks and sands below the earth's crust. Oil generally has a body of water below and pressurized natural gas above. Fairly thick and dense earth strata (caprock) cover most deposits preventing seepage (Fig. 3.1). Oil wells drilled through this layer penetrate the deposits. The pressure forces the gas and oil to the surface. After the pressure has diminished, the oil must be pumped.

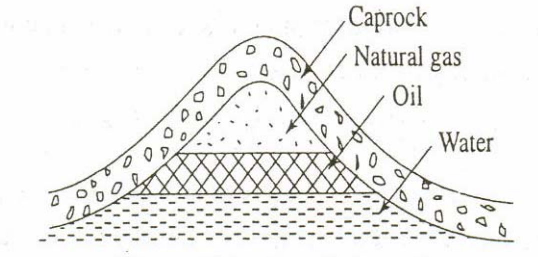

Figure 3.1 An oil deposit [ ]

Liquid fuels are an excellent energy source. They are easy to handle, easy to store and easy to burn. They have nearly constant heating values. They are primarily a mixture of hydrocarbon compounds, which may also contain nitrogen, oxygen and sulphur. The bulk of the hydrocarbons belong to the ***paraffin series, C_nH_{2n+2}*** like methane (CH_4), ethane (C_2H_6), propane (C_3H_8) and butane (C_4H_{10}) which are gaseous, and pentane (C_5H_{12}), hexane (C_6H_{14}) and octane (C_8H_{18}) which are liquid at STP. In addition, there can be isoparaffins, cycloparaffins and aromatic compounds. The proportion of various hydrocarbon groups varies widely with geographic location. The ultimate analysis of oil indicates the mass fractions of carbon, hydrogen, oxygen, nitrogen and sulphur. Regardless of the crude oil source, its composition is fairly uniform within close limits, as given below

Carbon 83-87%, Hydrogen 11-16%,

Oxygen + Nitrogen 0 -7%, Sulphur 0 - 4%

There can also be some moisture and sediment.

Crude oil is seldom used as such: in the refining process it is distilled into a number of fractions. The lighter fractions (having lower boiling point) like gasoline, aviation fuel, kerosene, light diesel oil, heavy diesel oil, lubrication oil and so on are principally transportation and machine fuels. The heavier fractions are used for boiler fuels and chemical production.

The required physical properties of fuel oil are specific gravity, viscosity, pour point, flash point and heating value. The ***pour point*** is the lowest temperature at which the oil will flow under standard pressure conditions. The ***flash point*** is the minimum temperature at which the oil may be ignited.

3.1.4-Natural and petroleum gas

Natural gas was formed millions of years ago from decaying vegetable matter generally along with petroleum. Oil wells drilled into the geologic formation containing the gas and trapped oil release gas predominantly. In some fields, the natural gas is flared or burned at the wellhead because of lack of facilities to transport the gas. In regions close to the source, natural gas has been used as a power plant fuel.

Transportation of natural gas is made through pipelines. Natural gas is the cleanest of all fossil fuels. It is free from ash and mixes well with air to undergo complete combustion producing very little smoke. It consists of a mixture of the most volatile paraffins-methane to pentane. It has high hydrogen content and produces a considerable amount of water vapor when burned. The heat of combustion varies from 33.5 to 40 MJ/m^3. The specific gravity is 0.63 relative to air.

Since the major constituent of all natural gases is methane (critical temperature - 83 °C), cryogenic temperatures are required to maintain the gas as a liquid at moderate pressures (e.g. -100 °C at 36 bar). Liquid natural gas (LNG) is transported by special tankers and stored in spherical pressure vessels to be used when needed, particularly during peak load.

Compressed natural gas (CNG) is now being considered as the alternative fuel for automobiles. Liquid petroleum gas (LPG) refers to hydrocarbons, such as propane, propylene, butane,.butylene and so on, which are liquefied under moderate pressures and at normal temperatures. It is used widely as domestic fuel and also to supplement natural gas flow.

3.2- Combustion

When a chemical reaction occurs, the bonds within molecules of the reactants are broken; and atoms and electrons rearrange to form products. In combustion reactions, rapid oxidation of combustible elements of the fuel results in energy release as combustion products are formed. The three major combustible chemical elements in most common fuels are carbon, hydrogen, and sulfur. Sulfur is usually a relatively unimportant contributor in the energy released, but it can be a significant cause of pollution and corrosion problems. Combustion is ***complete*** *when all the carbon present in the fuel is burned to carbon dioxide all the hydrogen is burned to water*, *all the sulfur is burned to sulfur dioxide, and all other combustible elements are fully oxidized*. When these conditions are not fulfilled, combustion is ***incomplete***.

In this Chapter, we deal with combustion reactions expressed by chemical equations of the form

$$\text{reactants} \rightarrow \text{products}$$

or

$$\text{fuel} + \text{oxidizer} \rightarrow \text{products}$$

When dealing with chemical reactions, it is necessary to remember that mass is conserved, so the mass of the products equals the mass of the reactants. The total mass of each chemical element must be the same on both sides of the equation, even though the elements exist in different chemical compounds in the reactants and products. However, the number of moles of products may differ from the number of moles of reactants.

For example consider the complete combustion of hydrogen with oxygen

$$1H_2 + \frac{1}{2}O_2 \rightarrow 1H_2O \tag{3.7}$$

In this case, the reactants are hydrogen and oxygen. Hydrogen is the fuel and oxygen is the oxidizer. Water is the only product of the reaction. The numerical coefficients in the equation, which precede the chemical symbols to give equal amounts of each chemical element on both sides of the equation, are called ***stoichiometric coefficients***. In words, Eq. 3.7 states

$$1\ kmol\ H_2 + (1/2)\ kmol\ O_2 \rightarrow 1\ kmol\ H_2O$$

Note that the total numbers of moles on the left and right sides of Eq. 3.7 are not equal However, because mass is conserved; the total mass of reactants must equal the total mass of products. Since 1 $kmol$ of H, equals 2 kg, $(1/2)kmol$ of O_2 equals 16 kg, and 1 $kmol$ of H_2O equals 18 kg, equation 3.7 can be interpreted as stating

$$2\ kg\ H_2 + 16\ kg\ O_2 \rightarrow 18\ kg\ H_2O$$

3.2.1- Combustion Reactions

Combustion is the high temperature oxidation of the combustible elements of a fuel with heat release. The combustible elements in coal and fuel oil are carbon, hydrogen and sulphur. The basic chemical equations for complete combustion are

$$C + O_2 \rightarrow CO_2$$

$$2H_2 + O_2 \rightarrow 2H_2O \tag{3.8}$$

$$S + O_2 \rightarrow SO_2$$

When insufficiently oxygen is present, the carbon will be burned incompletely with the formulation of carbon monoxide.

$$2C + O_2 \rightarrow 2CO \tag{3.9}$$

In order to burn a fuel completely, four basic conditions must be fulfilled:

1. Supply *enough* air for complete combustion of fuel.
2. Secure enough *turbulence* for thorough mixing of fuel and air.
3. Maintain a furnace *temperature* high enough to ignite the incoming fuel air mixture.
4. Provide a furnace volume large enough to allow *time* for combustion to be completed.

Apart from adequate air supply, the three *T's*, *viz*., time, temperature and turbulence have to be kept in mind while designing a furnace. Combustion may be said to hinge upon the word *MATT* - *M* is for *mixture* (turbulence), so that the fuel molecules meet the oxygen molecules, *A* is for the proper *air-fuel ratio* in order to support combustion, one *T* is for *temperature*, and the other *T* is for *time*. Since the complete mixing of the fuel and air is virtually impossible, excess air must be supplied to ensure complete combustion. The greater is the rate of mixing or turbulence, the lower would be the excess air required.

3.2.2- Theoretical or Stoichiometric Air

The ultimate analysis of the fuel is given by

$$C + H + O + N + S + M + A = 1.0$$

Oxygen needed for the oxidation processes can be calculated as follows:

1- C	+ O_2	$\rightarrow CO_2$
$12\,kg$	$32\,kg$	$44kg$
$1\,kg$	$2.67\,kg$	$3.67\,kg$
$C\,kg$	$\underline{2.67\,C\;kg}$	$3.67\,C\,kg$

2- $2H_2$	+ O_2	$\rightarrow 2H_2O$
$4\,kg$	$32\,kg$	$36\,kg$
$1\,kg$	$8\,kg$	$9\,kg$
$H\,kg$	$\underline{8\,H\;kg}$	$9\,H\,kg$

3- S	+ O_2	$\rightarrow SO_2$
$32\,kg$	$32\,kg$	$64kg$
$1\,kg$	$1\;kg$	$2\,kg$
$S\,kg$	$\underline{S\,kg}$	$2\,S\,kg$

4- Oxygen in the fuel, $\underline{O}$

Oxygen required for complete combustion of 1 kg fuel is

$$M_{O_2} = 2.67\,C + 8H + S - O$$

Air contains 23.2% oxygen by mass. Therefore, theoretically *air required for complete combustion of 1 kg fuel is*

$$M_T = \frac{M_{O_2}}{0.232} = \frac{2.67\,C}{0.232} + \frac{8\,H}{0.232} + \frac{S}{0.232} - \frac{O}{0.232}$$

or,
$$M_T = 11.5\,C + 34.5\left(H - \frac{O}{8}\right) + 4.3\,S \tag{3.10}$$

where *C, H, O* and *S* are the mass fractions of carbon, hydrogen, oxygen and sulphur in the fuel as given by the ultimate analysis.

Normally the amount of air supplied is either greater or less than the theoretical amount. The amount of air actually supplied is commonly expressed in terms of the *percent of* *theoretical air*. For example, 150% of theoretical air means that the air actually supplied is 1.5 times the theoretical amount of air. The amount of air supplied can be expressed alternatively as a percent excess. Thus, 150% of theoretical air is equivalent to 50% ***excess air***, and 80% of theoretical air is the same as a 20% ***deficiency*** of air. Excess air is always needed for complete combustion. It is expressed as a percentage or by the use of a dilution coefficient. The percent excess air supplied is

$$\%\,excess\ air = \frac{M_A - M_T}{M_T} \times 100 \tag{3.11}$$

where M_A is the actual amount of air supplied for complete combustion of 1 kg fuel. The dilution coefficient, d, is given by,

$$d = \frac{M_A}{M_T} \tag{3.12}$$

The percentage of excess air varies between 15 and 30% for most large utility boilers.

In the combustion of methane,

$$\underbrace{CH_4 + 2O_2}_{reac\,tan\,ts} \rightarrow \underbrace{CO_2 + 2H_2O}_{products} \tag{3.13}$$

Atmospheric air contains 21% oxygen, 78% nitrogen, and 1% argon by volume. In combustion calculations, however, the argon is usually neglected, and air is assumed to consist of 21% oxygen and 79% nitrogen by volume (or molar basis). On a mass basis, air contains 23.2% oxygen and 76.8% nitrogen.

For each mole of oxygen taking part in a combustion reaction, there are 0.79/0.21 or 3.76 moles of nitrogen. So, for the complete combustion of methane, the reaction can be written as

$$CH_4 + a(O_2 + 3.76N_2) \rightarrow bCO_2 + cH_2O + d\,N_2 \tag{3.14}$$

where a,b,c, and d represent the numbers of moles of oxygen, carbon dioxide, water and nitrogen. Applying the conservation of mass principles, results in four equations among the four unknowns

C: $b=1$

H: $2c=4$

O: $2a=2b+c$

N: $2 \times 3.76a=2d \rightarrow 3.76a=d$

Solving these equations, the combustion equation can thus be written as:

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52\,N_2 \tag{3.15a}$$

With 150% theoretical air or 50% excess air, the methane combustion reaction can be written as

$$CH_4 + 2(1.5)O_2 + 2(3.76)(1.5)N_2 \rightarrow CO_2 + 2H_2O + O_2 + 11.28\,N_2 \tag{3.15b}$$

Example 3.1

A coal has the following ultimate analysis:

C=71.9%, N=1.3%, H=4.8%, O=6.3%, S=3.4%, A=9.1%, and M=3.5%

Calculate the total air required for complete combustion per kg of coal. Assuming the excess air is 30%.

Solution

The theoretical air is calculated by equation 3.10. It gives

$$M_T = 11.5 \times 0.719 + 34.5\left(0.048 - \frac{0.063}{8}\right) + 4.3 \times 0.034 = 9.78 \quad kg/kg\ coal$$

Equation 3.11 is used for calculation of total air. That is,

$$0.3 = \frac{M_A - 9.78}{9.78}$$

$$M_A = 12.714\ kg/kg\ coal$$

3.2.3- Air Fuel Ratio, AF

Two parameters that are frequently used to quantify the amounts of fuel and air in a particular combustion process are the air-fuel ratio and its reciprocal, the fuel-air ratio. The *air-fuel ratio* is simply the ratio of the amount of air in a reaction to the amount of fuel. The ratio can be written on a molar basis (moles of air divided by moles of fuel) or on a mass basis (mass of air divided by mass of fuel). Conversion between these values is accomplished using the molecular weights of the air, M_{air} and fuel, M_{fuel}

$$\frac{mass\ of\ air}{mass\ of\ fuel} = \frac{moles\ of\ air \times M_{air}}{moles\ of\ fuel \times M_{fuel}}$$

or

$$AF = \overline{AF}\left(\frac{M_{air}}{M_{fuel}}\right) \tag{3.16}$$

where $\overline{AF}$ is the air-fuel ratio on a molar basis and AF is the ratio on a mass basis. For the combustion calculations of this subject the molecular weight of air is taken as 28.97.

Tables 3.1 provide the molecular weights of several important hydrocarbons. Since AF is a ratio, it has the same value whether the quantities of air and fuel are expressed in SI units or English units.

Let us determine the AF for complete combustion of methane, Eq, 3.14. The coefficient 2 before the term $(O_2 + 3.76\,N_2)$ is the number of moles of oxygen in the combustion air, per mole of fuel, and not the amount of air. The amount of combustion air is 2 moles of oxygen plus 2 x 3.76 moles of nitrogen, giving a total of 9.52 moles of air per mole of fuel. Thus, for the reaction given by Eq. 3.14 the air fuel ratio on a molar basis is 9.52. To calculate the air fuel ratio on a mass basis, use Eq. 3.16 to write

$$AF = \overline{AF}\left(\frac{M_{air}}{M_{fuel}}\right) = 9.52\left(\frac{28.97}{16.04}\right) = 17.19$$

3.2.4- Combustion Equation

Let us consider a coal having the following ultimate analysis (by mass):

$$C = 60\%, H = 4\%, S = 3.2\%, O = 4.8\%, N = 2\%, M = 5\%, \text{ and } A = 21\%$$

The exhaust gas has the following volumetric analysis:

$$CO_2 + SO_2 = 12\%, CO = 2\%, O_2 = 4\% \text{ and } N_2 = 82\%$$

Let a moles of oxygen be supplied for 100 kg fuel. Then, the combustion equation can be written as

$$\frac{60}{12}C + \frac{4}{2}H_2\frac{3.2}{32}S + \frac{4.8}{32}O_2 + a\left(O_2 + 3.76\,N_2\right) \rightarrow b\,CO_2 + d\,CO + e\,SO_2 + fO_2 + g\,N_2 + h\,H_2O$$

By equating the coefficients,

$$Carbon: \frac{60}{12} = b + d = 5, Hydrogen\ \frac{4}{2} = h = 2, Sulphur: \frac{3.2}{32} = e = 0.1$$

$$Oxygen: \frac{9.6}{32} + 2a = 2b + d + 2e + 2f + h\ ,\ Nitrogen: 3.76\,a = g$$

From the exhaust gas analysis,

$$\frac{b+e}{b+d+e+f+g} = 0.12,\ \frac{d}{b+d+e+f+g} = 0.02,\ \frac{f}{b+d+e+f+g} = 0.04$$

$$\frac{g}{b+d+e+f+g} = 0.82$$

By solving these equations, the coefficients, a, b, d, e, f, g and h are determined and given to be $a = 7.045, b = 4.27, d = 0.73\,e = 0.1, f = 1.46\,, g = 26.49, and\ h = 2$.

Actual air supplied per kg coal $\left(M_A\right)$ = $\frac{32\,a}{0.232 \times 100} = 9.72\,kg$

Example 3.2

Determine the air fuel ratio on both a molar and mass basis for the complete combustion of octane, C_8H_{18}, with (a) the theoretical amount of air (b) 50% excess air.

Solution

(a) It is conveniently to conduct the calculation on the basis of 100 kg of dry product.

$$C_8H_{18} + a\left(O_2 + 3.76N_2\right) \rightarrow bCO_2 + cH_2O + d\,N_2$$

C: $b=8$

H: $2c=18$

O: $2a=2b+c$

N: $2 \times 3.76a=2d \rightarrow 3.76a=d$

Solving these equations, $a = 12.5, b=8, c=9 , d=47$. The balanced chemical equation is,

$$C_8H_{18} + (12.5)(O_2 + 3.76)N_2 \rightarrow 8CO_2 + 9H_2O + 47\,N_2$$

The air fuel ratio on a molar mass is

$$\overline{AF} = \frac{12.5 + 12.5 \times 3.76}{1} = 59.5\,\frac{kmol(air)}{kmol\,(fuel)}$$

The air fuel ratio expressed on a mass basis is

$$AF = \overline{AF}\left(\frac{M_{air}}{M_{fuel}}\right) = 59.5\left(\frac{28.97}{114.22}\right) = 15.1\,\frac{kg\,(air)}{kg\,(fuel)}$$

(b)

$$C_8H_{18} + 1.5(12.5)(O_2 + 3.76\,N_2) \rightarrow bCO_2 + cH_2O + d\,N_2 + eO_2$$

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Table 3.1 Atomic of Molecular Weights of selected elements and compounds

Substance	Chemical Formula	M (kg/kmol)
Acetylene	C_2H_2	26.04
Air	-	28.97
Ammonia	NH_3	17.03
Argon	Ar	39.94
Benzine	C_6H_6	78.11
Butane	C_4H_{10}	58.12
Carbon	C	12.01
Carbon dioxide	CO_2	44.01
Carbon monoxide	CO	28.01
Copper	Cu	63.54
Ethane	C_2H_6	30.07
Ethyl alcohol	C_2H_5OH	46.07
Ethylene	C_2H_4	28.05
Helium	He	4.003
Hydrogen	H_2	2.016
Methane	CH_4	16.04
Methyl alcohol	CH_3OH	32.04
Nitrogen	N_2	28.01
Octane	C_8H_{18}	114.22
Oxygen	O_2	32.00
Propane	C_3H_8	44.09
Propylene	C_3H_6	42.08
Sulfur dioxide	SO_2	64.06
Water	H_2O	18.02

Example 3.3

Determine the molar analysis of the products of combustion when octane, C_8H_{18} is burned with 200% theoretical air, and determine the dew point[♣] of the products if the pressure is 100 kPa.

[♣] The partial pressure of water is (the product pressure x %H_2O) and then the dew point is the saturation temperature corresponding to this pressure)

Example 3.4

Methane, CH_4, is burned with dry air. The molar analysis of the products on a dry basis is CO_2, 9.7%, CO, 0.5%, O_2, 2.95% and N_2, 86.85%. Determine (a) the air fuel ratio on both molar and a mass basis, (b) the percent theoretical air.

Solution

(a)

$$aCH_4 + b(O_2 + 3.76)N_2 \rightarrow 9.7CO_2 + 0.5CO + 2.95\,O_2 + 86.85\,N_2 + cH_2O$$

C: $9.7+0.5=a$

H: $2c=4a$

O: $2b=(9.7)(2)+0.5+2(2.95)+c$

Solving this set of equations gives $a=10.2$, $b=23.1$, $c=20.4$. The balanced chemical equation is

$$10.2\,CH_4 + 23.1(O_2 + 3.76)N_2 \rightarrow 9.7CO_2 + 0.5CO + 2.95\,O_2 + 86.85\,N_2 + 20.4H_2O$$

The air fuel ratio on a molar mass is

$$\overline{AF} = \frac{23.1 + 23.1\times 3.76}{10.2} = 10.78\frac{kmol(air)}{kmol\,(fuel)}$$

The air fuel ratio expressed on a mass basis is

$$AF = \overline{AF}\left(\frac{M_{air}}{M_{fuel}}\right) = 10.78\left(\frac{28.97}{16.04}\right) = 19.47\frac{kg\,(air)}{kg\,(fuel)}$$

(b)

The balanced chemical equation for the complete combustion of methane with the theoretical amount of air is

$$CH_4 + a(O_2 + 3.76)N_2 \rightarrow bCO_2 + cH_2O + d\,N_2$$

Applying the conservation of mass principles, results in four equations among the four unknowns

C: $b=1$

H: $2c=4$

O: $2a=2b+c$

N: $2 \times 3.76a=2d \rightarrow 3.76a=d$

Solving these equations, the balanced chemical equation is

$$CH_4 + 2(O_2 + 3.76)N_2 \rightarrow CO_2 + 2H_2O + 7.52\,N_2$$

The theoretical air fuel ratio on a molar basis is

$$\overline{AF}_{theo} = \frac{2 + 2 \times 3.76}{1} = 9.52 \frac{kmol(air)}{kmol\,(fuel)}$$

The percent theoretical air is then found from

$$\%\ \text{theoretical air} = \frac{\overline{AF}}{\overline{AF}_{theo}} = \frac{10.78}{9.52} = 1.13\ (113\%)$$

Example 3.5 (4.1)

In an oil fired boiler the fuel had an analysis by mass; carbon 84%, hydrogen 10%, sulphur 3.2%, oxygen 1.6%, remained incombustible. The analysis of dry flue gas by volume gave: combined CO_2+SO_2 15.72%, O_2 1%, there being no CO or SO_3. Calculate per kg of fuel (a) mass of air supplied, (b) percentage excess air supplied, (c) mass of dry flue gas formed, and (d) mass of water vapor formed.

Solution

Example 3.6 (4.3)

An unknown hydrocarbon fuel, C_xH_y, was allowed to react with air. The analysis of product gases by volume gave: CO_2 12.1%, O_2 3.8% and CO 0.9%. Determine (a) the chemical equation for the actual reaction, (b) the composition of the fuel, (c) the air fuel ratio during the test, and (d) the excess or deficiency of air used.

Solution

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3.3-Thermodynamic analysis

The steam generator system comprises the fuel preparation and firing system. Heat released by the burning of fuel is transferred to the two steam circuits for main steam and reheat steam. Any energy remaining in the flue gas at the exit of the air heater is lost.

An important measure of performance for a power plant is its **N**et **Pl**ant **H**eat **R**ate ($NPHR$) or **Pl**ant **N**et **H**eat **R**ate ($PNHR$) defined as the amount of fuel energy or boiler heat input required to generate one kWh and deliver it to the transmission lines. For a coal-fueled steam electric generating unit, the net plant heat rate is given by

$$NPHR = \frac{\dot{Q}_B}{\dot{W}_{net}} = \frac{\dot{m}_f \, HHV}{\dot{W}_t - \dot{W}_p - AP} \tag{3.17}$$

where $\dot{Q}_B$ = heat input to boiler from fuel (kW), $\dot{m}_f$ = fuel burning rate (kg/s), $\dot{W}_t$ = turbine output (kW), $\dot{W}_p$ = pump input (kW), AP = auxiliary power (kW), HHV = higher heating value (kJ/kg).

the boiler efficiency, $\eta_B = \dfrac{\dot{m}_s(\Delta h)}{\dot{m}_f \, HHV}$ (3.18)

The net turbine heat rate $NTHR = \dfrac{\dot{m}_s(\Delta h)}{W_t - W_p} = \dfrac{1}{\eta_{cycle}}$ (3.19)

$$NPHR = \frac{NTHR}{\eta_B\left[1 - \dfrac{AP}{W_t - W_p}\right]} \quad \frac{kJ}{kW\ h} \tag{3.20}$$

The overall efficiency of the plant is then

$$\eta_o = \frac{3600}{NPHR} \tag{3.21}$$

3.4 Draft system

Large amounts of air are needed for combustion of the fuel. The gaseous combustion products in huge quantity have also to be removed continuously from the boiler furnace. To produce the required flow of either air or combustion gas, a pressure differential is needed. The term "draught" or "draft" is used to define the static pressure in the furnace, in the various ducts, and the stack. The function of the draft system is basically two-fold:

1. To supply to the furnace the required quantity of air for complete combustion of fuel.
2. To remove the gaseous products of combustion from the furnace and throw these through chimney or stack to the atmosphere. There are two ways of producing draft: (a) natural draft and (b) mechanical draft.

3.4.1-Natural Draft

The natural draft is produced by a chimney or a stack. It is caused by the density difference between the atmospheric air and the hot gas in the stack. For a chimney of height H meters (Figure 3.2) the draft or pressure difference (N/m^2) produced is given by

$$\Delta P = gH\left(\rho_a - \rho_g\right) \tag{3.22}$$

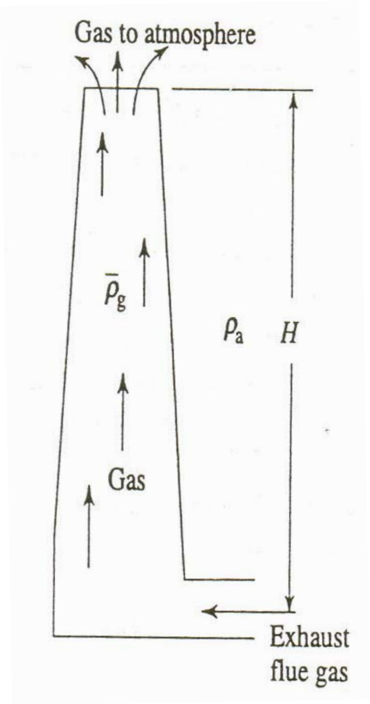

Figure (3.2) Draft produced by a chimney

where ρ_a = density of atmospheric air, kg/m^3

ρ_g = average gas density in the chimney, kg/m^3

g = acceleration due to gravity, 9.81 m/s^2

Assuming both air and gas as ideal gases,

$$\Delta P = g\,H\left[\frac{P_a}{R_a T_a} - \frac{P_g}{R_g T_g}\right] \tag{3.23}$$

where $P_a = P_g$ 101.325 kPa

$R_a = R_g$ = 0.287 kJ/kg K

T_a = ambient temperature, K

T_g = average gas temperature in the chimney, K$= \left(T_o + T_H\right)/2$

T_o= gas temperature at inlet to the chimney, K

T_H= gas temperature at exit from the chimney, K.

$$\Delta P = \frac{g\,H\,P_a}{R_a}\left[\frac{1}{T_a} - \frac{1}{T_g}\right] \tag{3.24}$$

For given H and T_a the natural draft depends on the average gas temperature T_g. Higher is the T_g higher is the draft produced. But a high T_g means a large exhaust loss through chimney resulting in a lower boiler efficiency. With an optimum T_g the amount of draft produced by density difference is thus limited.

Tall and conspicuous from a distance, chimneys or stacks are used in all power plants. In early steam generators, all the needed draft used to be produced by stacks alone to overcome all the pressure losses at the required gas flow rates. In modem boilers, the fuel burning rate is high, and so the rate of air supply as well as the rate of flue gas removal is high. There are also various heat exchangers like superheaters, reheater, economizer, and air preheater on the way to cause large pressure losses for which stacks alone are insufficient, and fans are added for producing mechanical draft. Stacks have thus two functions:

1. To assist the fans in overcoming pressure losses.
2. To help disperse the gas effluent into the atmosphere at a sufficient height to cause minimum atmospheric pollution.

Stacks introduce pressure losses of their own. These pressure losses are caused by wall friction and the pressure equivalent to the kinetic energy of the gases leaving the stack. Therefore, the actual draft produced is less than the theoretical value ΔP given by Eq. (3.24), so that

$$\Delta P_{act} = \Delta P - \rho_g \frac{\overline{V}^2}{2}\left[1 + \frac{f\,H}{D}\right] \tag{3.25}$$

where f = friction factor, dimensionless

H = stack height, m
D = stack inside diameter, m
V= stack exit velocity, m/s
ρ_g = average gas density in the stack, kg/m^3

Dispersion of the flue gases into the atmosphere is defined as the movement f the fine gases horizontally as well as vertically and their dilution by the atmosphere. Existing wind causes the horizontal motion of the flue gases. The vertical motion of fine gases results from the upward motion of high-velocity arm stack-exit gases to much higher elevations. The exit velocity of flue gases t stack exit results in a plume rise ΔH above the actual stack (Fig. 3.3). The gases bend in the direction of wind flow. The plume height ΔH is the height of a virtual point source above the stack obtained by extending the lines of dispersion backward. The effective stack height ff„ is, thus, given by

$$H_e = H + \Delta H \tag{3.26}$$

Higher is the velocity of flue gases at stack exit, higher will be the plume rise ΔH and greater will be the dilution of exiting gases. There are several analytical methods of estimating ΔH. Most of these utilize a momentum term that accounts for the vertical momentum of the gas caused by the stack exit velocity and a buoyancy term that accounts for the difference between stack gas and atmospheric densities, pertinent to the prevailing situation of mixed convection in the stack.

1. Correlation of Carson and Moses

$$\Delta H = 2.62 \frac{\dot{Q}_e^{0.5}}{V_w} - 0.029 \frac{V_s\, D}{V_w} \tag{3.27}$$

where, V_s = stack gas exit velocity, m/s
D = stack diameter, m
V_w = wind velocity at stack exit, m/s
$\dot{Q}_e$ = heat emission, (*J/s*)= $\dot{m}\, c_p \left(T_s - T_a\right)$
$\dot{m}$ = gas mass flow rate, kg/s
c_p = specific heat of gas = 1.005 kJ/kg K for dry air at low temperature
T_s = s temperature at stack exit, K
T_a = r temperature at stack exit, K

2. Correlation of Briggs

$$\Delta H = \frac{114\, C\, F^{1/3}}{V_w} \tag{3.28}$$

where C= dimensionless temperature gradient parameter=I.5S -41.4
$\frac{\Delta\theta}{\Delta z}$ = air temperature gradient, K/m which is zero for neutral atmospheric stability conditions
F = buoyancy flux = $\frac{g\, V_s\, D^2 \left(T_s - T_a\right)}{4\, T_a}$ m^4/s^2
g = acceleration due to gravity = 9.81 m/s^2

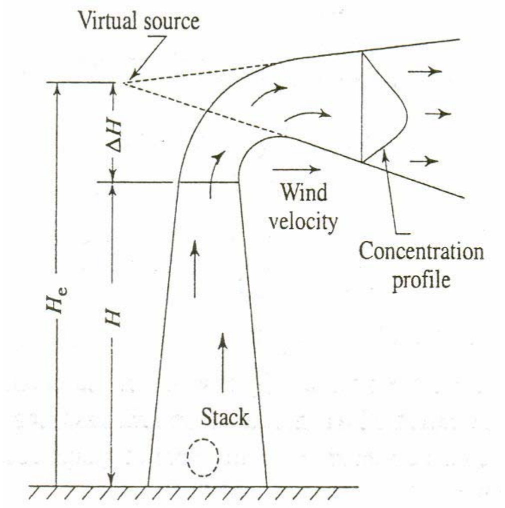

Figure (3.3) Dispersion of flue gases at stack exit

Example 3.7 (4.12)

A 200 m high 4 m diameter stack emits 1000 kg/s of 100 °C gases into 5 °C air. The prevailing wind velocity is 50 km/h. The atmosphere is in a condition of neutral stability. Calculate the height of the gas plume using the correlation of Carson and Moses

Solution

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3.4.2 Mechanical Draft

Mechanical draft is produced by fans. There are two types of fans in use today: forced draft (*FD*) and induced draft (*ID*) fans. When either one is used alone, it should overcome the total air and gas pressure losses within the steam generator.

Forced draft fans are installed at inlet to the air preheater. They handle cold air. So they have less maintenance problems, consume less power (since cold air has low specific volume, and work input per unit mass flow rate is given by $\int v\,dP$) and therefore, their capital and operating costs are lower. If $\dot{m}_f$ is the fuel burning rate, v_a is the specific volume of inlet air and ΔP_{FD} is the pressure head developed by the fan to overcome all the pressure losses, then the power required $\left(\dot{W}_{P_{FD}}\right)$ to drive the *FD* fan is given by

$$\dot{W}_{P_{FD}} = \frac{\dot{V}_a \times \Delta P}{\eta_{FD}} = \frac{\dot{m}_f \times AF \times v_a \times \Delta P}{\eta_{FD}} \tag{3.29}$$

where ΔP is the pressure difference across the fan stages and generally measured in terms of millimeters of water gauge (mmWG) = (kg/m^2), (kg/m^2) x g = (N/m^2) , AF is the air-fuel ratio and η_{FD} is the efficiency of the forced draft fan. For good reliability two forced draft fans operating in parallel are normally used, each capable of undertaking at least 60% of full load air flow when the other is out of service.

The forced draft fan if used alone, as in many large steam generators and almost all marine applications, maintains the entire system up to the stack entrance under positive gauge pressure. The furnace is then said to be pressurized. Utmost care is taken so that noxious gases do not leak out to the atmosphere. It thus needs a gas-tight furnace construction and properly designed ignition openings and inspection doors. The stack in such a situation is shorter and meant only for disposal of flue gases.

Induced draft fans are normally located at the foot of the stack. They handle hot combustion gases. Their power requirements are, therefore, greater than forced draft fans. In addition, they must cope with corrosive combustion products and fly ash. Induced draft fans are seldom used alone. They discharge essentially at atmospheric pressure and place the system upstream under negative gauge pressure. If v_g is the specific volume of flue gases handled by the ID fan and ΔP_{ID} is the pressure head developed, then the power needed $\left(\dot{W}_{P_{ID}}\right)$ to drive the ID fan is given by

$$\dot{W}_{P_{ID}} = \frac{\left(\dot{m}_f + \dot{m}_a\right) \times v_g \times \Delta P_{ID}}{\eta_{ID}} = \frac{\dot{m}_f \left(1 + AF\right) \times v_g \times \Delta P_{ID}}{\eta_{ID}} \tag{3.30}$$

When both forced and induced draft fans are used in a steam generator, the FD fans push atmospheric air through the air preheater, dampers, various air ducts, and burners into the furnace, and the ID fan sucks out the flue gases through the heat transfer surfaces in the superheaters, reheaters, economizer, gas-side air preheater and dust collectors and discharge into the stack (Fig. 3.4). The stack because of its height, adds a natural driving pressure of its own. In such a case the furnace is said to operate with balanced draft, meaning that the pressure in it is approximately atmospheric. .Actually, it is maintained at a slightly negative gauge pressure to ensure that any leakage would be inward. Modern boilers are mostly designed with balanced draft firing.

A typical large power plant (600MWe) may required two 1320 kW forced draft fans and two 3000 kW induced draft fans.

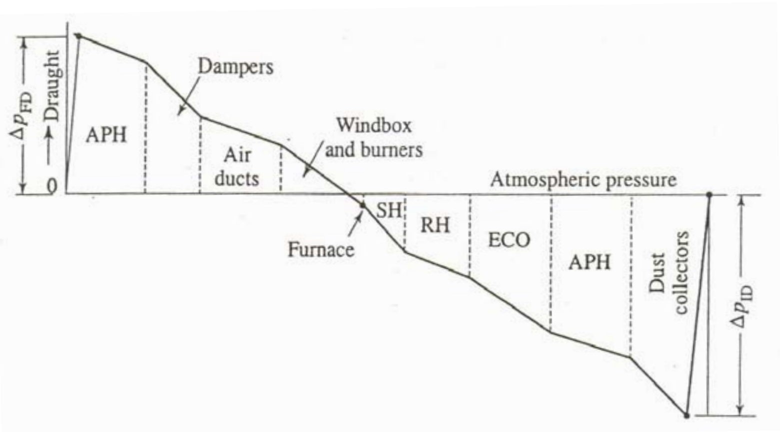

Figure (3.4) Balanced draft system in a modern boiler

Example 3.8 (4.8)

A textile factory has a battery of 6 Lancashire boilers, each supplying 6 t/h of steam at 16 bar, 250 °C from feedwater at 30 °C. The boilers burn fuel oil of calorific value 43960 kJ/kg with an overall efficiency of 75%. For efficient combustion 16 kg of air per kg fuel is required for which a draft of 20 mm of water gauge is required at the base of the chimney. The flue gases leave the boilers at 320 °C. The average temperature of the gases in the stack may be taken to be 300 °C. The atmosphere is at 30 °C. Assuming the velocity of gases at stack exit to be negligible, determine the height of the stack and the diameter at its base.

Solution

Example 3.9 (4.9 and 4.10)

(a) Compute the required motor capacity needed for the FD fan under the following conditions:

Coal rate	10 tonnes/h
Coal analysis	*C=78.0%, H=3.0%,O=3.0%, S=1%, M=7%, A=8%*
Excess air	30%
Plenum chamber pressure	180 mm water gauge
Mechanical efficiency of the fan	60%
Room temperature	30 °C

(b) If the draft produced by the ID fan is 250 mmWG, the temperature is 180 ^{o}C and the fan efficiency is 52%, find the motor capacity of the ID fan.

Solution

Revision questions

1- Methane (CH_4) is burned with atmospheric air. The molar analysis of the products on a dry basis is CO_2, 9.7% ; O_2 , 2.95%; CO, 0.5%; N_2 , 86.85%. (a) Calculate the air-fuel ratio on both a molar and a mass basis, (b) the percent of theoretical air, and (c) the dew point temperature of the product if the pressure is100 kPa.

2- A natural gas has the following molar analysis: CH_4, 80.62%, C_2H_6, 5.41%, C_3H_8, 1.87%; C_4H_{10}, 1.6%, N_2, 10.5%. The gas is burned with dry air, giving products having a molar analysis on a dry basis CO_2,7.8%; CO, 0.2%; O_2,7%; N_2, 85%. (a) Determine the air fuel ratio on a molar basis. (b) Assuming ideal gas behavior for the fuel mixture, determine the amount of products in kmol that would be formed from 100 m^3 of fuel mixture at 300 K and 100 kPa. (c) Determine the percent of theoretical air.

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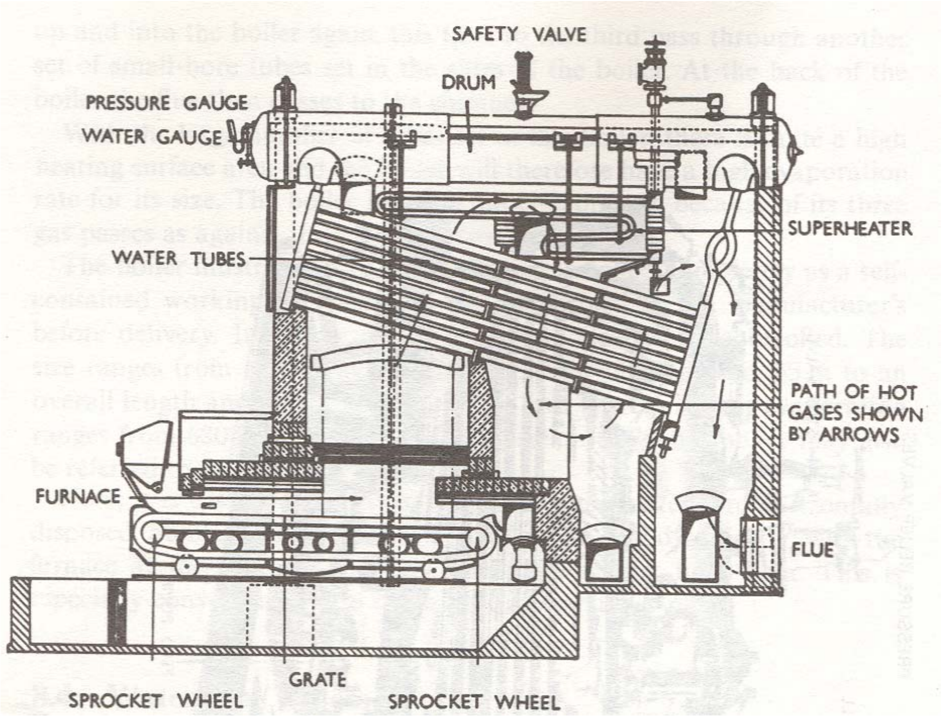
Power Plant Engineering and economy

MET401

Department of Mechanical Engineering Technology



Chapter IV- Steam Generators

2010

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Chapter IV

Steam Generators

4.1-Introduction

Steam generators are used to generate steam at desired rate and at the desired pressure and temperature by burning fuel in its furnace. Steam generators are used in both fossil-and nuclear-fuel electric generating power plants. Steam generator is a complex combination of the following components

a- Economizer
b- Boiler
c- Superheater
d- Reheater
e- Air heater
f- Auxiliaries (ash handling equipment, control equipment, fans, burners, etc).

Steam generators are classified in different ways. They may, for example classified as either

1. Utility steam generators
2. Industrial steam generators
3. Marine steam generators

4.2. Boiler

A boiler is the device in which steam is generated. Generally, then, it must consist of a water container and some heating device. There are many designs of boilers but they can be basically divided up into two types. These are the fire-tube boiler and the water-tube boiler. Before describing various boiler designs, however, it will be useful to discuss the formation of steam in a boiler, and the methods employed to improve its thermal efficiency.

Now, whatever the type of boiler, steam will leave the water at its surface and it passes into what is called the steam space. This is the space in the water container directly above the water. Steam formed above the surface of water is always wet and will remain wet so long as there is water present. This is because the steam, rising from the surface of the turbulent boiling water, will carry away with it some minute droplets of water

Since the water container will, and must, always contain water, then the steam in the steam space is always wet. If wet steam is all that is required, then the steam, as it is formed, is piped out of the steam space away into the steam main.

If, however, superheated steam is required, then the wet steam is removed from the steam space and is piped into a ***superheater***. This consists of a long tube or series of tubes which are suspended across the path of the hot gases from the furnace. As the wet steam progresses through the tube or tubes it is gradually dried out and eventually superheated. From the superheater it passes to the steam main. If a control of the degree of superheat is required, as in some of the larger boilers, then an attemperator is fitted. The control of the degree of superheat is obtained by the injection of water or steam into the superheated steam.

If an attemperator is fitted, the superheater is generally divided into two parts. The first part is called the primary superheater. Then comes the attemperator followed by the second part of the superheater called the secondary superheater.

The energy balance of the superheater is given by

$$\dot{Q}_{sh} = \dot{m}_g \, c_{pg} \left(t_{g,i} - t_{g,e}\right) \tag{4.1a}$$

$$= \dot{m}_s \left(\Delta h\right) \tag{4.1b}$$

$$= U_o \, A_o \, \Delta T_m \tag{4.1c}$$

where

$$\frac{1}{U_o A_o} = \frac{1}{h_{sc} \, A_i} + \frac{1}{h_i \, A_i} + \frac{x_w}{k_w A_{\ell m}} + \frac{1}{h_{fo} \, A_o} + \frac{1}{h_o \, A_o} \tag{4.2}$$

$$\Delta T_m = Log \;\; Mean \, Temperature \, Difference = \frac{\Delta T_{max} - \Delta T_{min}}{\ell n\left(\dfrac{\Delta T_{max}}{\Delta T_{min}}\right)} \tag{4.3}$$

$$A_{\ell m} = \frac{A_o - A_i}{\ell n\left({}^{A_o}\!/\!_{A_i} \right)} \tag{4.4}$$

where

x_w: thickness of the tube wall
k_w: thermal conductivity of wall material
h_{sc}: heat transfer coefficient due to scale formed on the inside of the tube.
h_{fo}: heat transfer coefficient due to solids (ash and soot) formed on the outside surface of tubes.

Now, the flue gases, having passed through the main boiler and then the superheater, will still be hot. The energy in these flue gases (370 °C-540 °C) can be used to improve the thermal efficiency of the boiler. To achieve this thermal efficiency improvement, the flue gases are firstly passed through an economizer. The ***economizer*** is really a heat exchanger in which the feedwater being pumped into the boiler is heated. The feedwater thus arrives in the boiler at a higher temperature than would be the case if no economizer were fitted. Hence, less energy is required to raise the steam, or if the same energy is supplied, then the more steam is raised. This results in a higher thermal efficiency.

The rate of heat transfer from the flue gases to the feedwater is given by

$$\dot{Q}_{econ} = \dot{m}_g \, c_{pg} \left(t_{g,i} - t_{g,e}\right) \tag{4.5a}$$

$$= \dot{m}_{fw} \, c_{pfw} \left(t_{sat} - t_{fw}\right) \tag{4.5b}$$

$$= U_o \, A_o \, \Delta T_m \tag{4.5c}$$

$$\dot{m}_{fw} = \left(n \frac{\pi}{4} d_i^2 \right) \frac{V_{fw}}{v_f} \tag{4.6}$$

$$A_o = n \pi d_o \ell \tag{4.7a}$$

$$A_i = n \pi d_i \ell \tag{4.7b}$$

ℓ; length of the coil, n; number of tubes
d_i: tube inner diameter, m^2
d_o: tube inner diameter, m^2
V_{fw}: velocity of fluid
v_f: specific volume of saturated water
n: number of tubes

Normally both gas and liquid flow under forced convection, then, Nusselt number is

$$Nu = f(Re, Pr)$$

If the flows are turbulent, the heat transfer coefficients can be calculated using Dittus-Boelter equation

$$Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^n \tag{4.8}$$

where

n = 0.3 for cooling and 0.4 for heating

Having passed through the economizer the flue gases are still moderately hot. Further thermal efficiency improvement can be obtained by passing them through an air heater (see Figure 4.1). This is, again, a heat exchanger in which the air being ducted to the boiler furnace is heated. The air thus arrives at the furnace hotter than if the air heater were not fitted. This results in a higher furnace temperature which thus increases the furnace potential for steam raising. Thermal efficiency improvement results

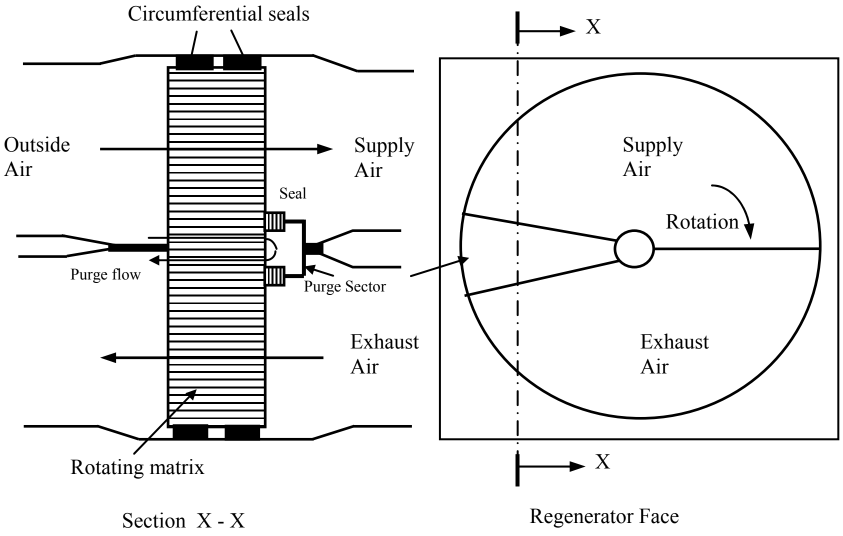


Figure 4.1

A still further improvement of thermal efficiency of the boiler is obtained by the installation of a reheater. The reheater will often appear in the flue gas path before the economizer. In some of the larger steam turbines in power stations for example, steam is removed from the turbine after partial expansion. This steam is fed back to the boiler to the reheater.

Here it is reheated to a higher temperature and then passed back to the turbine where it completes its expansion in the latter stages. The object of reheating steam in a turbine plant is to preserve the steam quality in the low pressure stages of the turbine, If there was no reheat then the steam in the low pressure stages would become too wet. Wet steam has an erosive and corrosive effect on the turbine blades. By returning the steam to the boiler, after partial expansion, the quality of the steam is improved and wet steam in the low pressure stages is therefore largely avoided. By reheating there is also greater potential work output from the steam in the low pressure stages. There may also be slight improvement in thermal efficiency by the process of reheating.

As a point of interest in the discussion on reheater, in the past, in power stations, it has usually been the practice to cross-connect all boilers and turbines. Thus, any boiler could run any turbine. The installation of reheaters in this case was difficult and hence was not often adopted.
Modern boilers, however, have become much more reliable and new installations are such that there is one boiler connected to one turbine making a single boiler-turbine unit. This being the case, reheaters are installed since there is no cross-coupling.

The location of the superheater, reheater, economizer and air heater are illustrated in Figure 4.2. After the air heater the flue gas passes to the exhaust chimney.

Other auxiliary equipment fitted to all boilers will be:

(a) A pressure gauge. This will record the gauge pressure of the saturated steam formed in the steam space.

(b) A water gauge glass. This will record the water level in the boiler. Often, two are fitted in case one breaks.

(c) A pressure relief valve. This is fitted as a safety precaution and is set to blow-off at a particular pressure. Often, two are fitted as an added precaution in case one sticks. They are either of the dead-weight or spring loaded types.

4.2.1- Fire-tube boilers

The basic documents of the fire-tube type of boiler can be illustrated by using the Lancashire boiler as an example. The Lancashire boiler is one of the older type of boiler designs, having been developed originally around 1850.

A cross-section of a modem Lancashire boiler installation is shown in Figure 4.3

The basic elements of the boiler are a large steel shell through which pass two large-bore cylinders called flues. Part of each flue is corrugated to take up the expansion when the boiler gets hot and to prevent collapse when under pressure. A section of one of these flues is shown. At the entrance of each flue, which is the front end of the boiler, a furnace is installed. The furnace can be arranged to burn gas, oil or coal. In the case of coal, the furnace can be hand fired or it can be provided with a mechanical stoker as illustrated. This mechanical stoker is sometimes referred to as a chain grate stroker. It will be noted that it is really like an endless belt. Coal is fed from the coal hopper on the left on to the front of the stoker. The rate of feed of coal to the stoker can be controlled by the chain-operated grate shown. The chain grate is rotated slowly and the coal on the grate progresses toward the back of the furnace. As it does so, it burns and the rate of grate feed is so adjusted, together with the necessary air supply, such that the coal is completely burnt by the time it reaches the back of the furnace. This will leave only ash, which falls over the back from where it is manually removed or it is automatically removed as illustrated. This automatic removal of ash is again accomplished by means of an enless belt device which brings the ash forward and out of the boiler to be deposited in an ash pit ready for removal from the boiler house.

The hot gaseous products of combustion, or flue gas, pass from the furnace through the large-bore corrugated flues. The water in the boiler surrounds these flues and thus heat from the hot flue gases transfers through the tubes into the water.

Now the boiler is set in a brickwork foundation called a **setting** and this setting so arranged that it is the means of improving the thermal efficiency of the boiler. The hot flue gas, somewhat reduced in temperature but still quite hot, after leaving the back end of the boiler, is diverted downward and under the boiler through a brickwork duct which is part of the boiler setting. The hot flue gas thus passes forward and under the boiler thus transferring heat through the bottom of the boiler shell to the water in the boiler. At the front of the boiler the hot flue gas stream is divided and the two streams are diverted up to pass back round the sides of the boiler. This is accomplished by two ducts provided round the sides of the boiler, built into the setting. These two side ducts meet at the back of the boiler and feed into the chimney.

Thus this boiler is provided with three gas passes.

The first pass is through the boiler. The second pass is back under the boiler. The third pass is round the sides of the boiler.

These passes are an attempt to extract the maximum amount of energy from the hot flue gas before they are released to atmosphere. An increased steaming capacity results and hence the thermal efficiency of the boiler is increased.

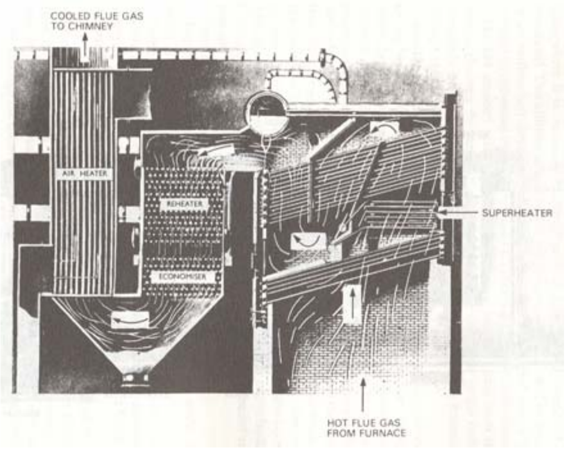


Figure 4.2

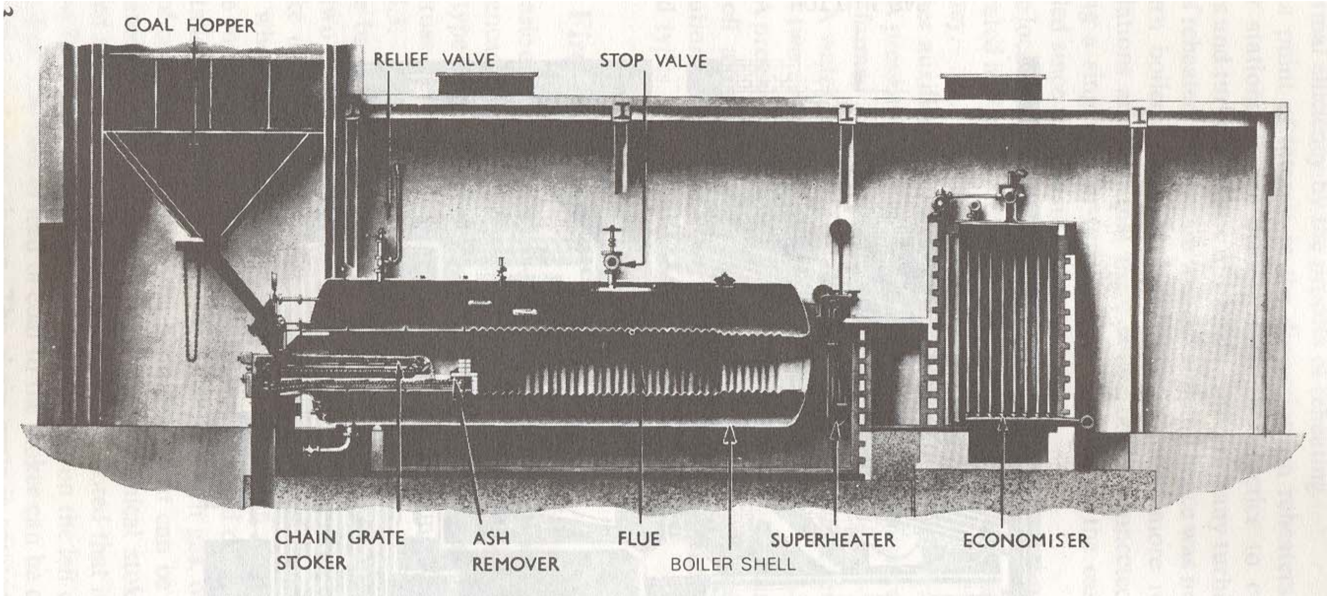


Figure 4.3.

The boiler illustrated also shows the installation of a superheater at the back of the boiler. This is introduced in the flue gas stream after the first pass. An economizer is also shown as a series of vertical straight tubes at the rear of the boiler installation. The flue gas stream, after the third pass, passes through the economizer into the chimney. Both the superheater and the economizer into the chimney. Both the superheater and the economizer result in a thermal efficiency improvement.

Various sizes of Lancashire boiler can be obtained. The smallest size has a boiler shell of about 5.5m long by some 2m diameter. The largest is about 10m long by 3.5m diameter.

The quantity of steam produced by a boiler in one hour is referred to as its evaporative capacity and this evaporative capacity for ease of working is often converted into what is known as the equivalent evaporation from and at 100°C.

The evaporative capacity of a boiler will depend upon its design, the type of fuel and furnace and on the quality of the fuel. For the larger sizes of Lancashire boiler it is possible to obtain an equivalent evaporation of some 6500kg of steam/h. The smallest will have an equivalent evaporation of about 1400 kg of steam/h. The Lancashire boiler, like other shell type boilers, can be worked up to a pressure of about 1.7 MPa (17 bar). There is a large water bulk and so it can meet sudden steam demand without a great reduction in water level. It also has a lower initial cost than some other types of boiler.

As a point of interest, the Lancashire boiler is really a development of the Cornish boiler. The Cornish boiler is similar to the Lancashire boiler except that it has a single furnace and a single flue through the boiler. One of the disadvantages of the Lancashire boiler is that repeated heating and cooling of the boiler, with the resultant expansion and contraction that occur, upsets the brickwork setting and ducting. This can result in the infiltration of air which upsets the furnace draught.

An improvement on the Lancashire boiler came with the economic boiler. This is, again, a fire-tube boiler and is illustrated in Figure 4.4. It has a cylindrical outer shell and contains two large-bore flues into which are set the furnaces. The one illustrated has a mechanical stoker and ash remover. The hot flue gases pass out of the furnace flues at the back of the boiler into a brickwork setting which deflects them back to pass through a number of small-bore tubes arranged above the large-bore furnace flues. These small bore tubes break up the water bulk in the boiler and present a large heating surface to the water. The flue gases pass out of the boiler at the front and into an induced draught fan which passes them into the chimney.

The economic boiler is only about half the size of an equivalent Lancashire boiler and it has a higher thermal efficiency. The general range of sizes of the economic boiler are from the small size about 3 m long and 1.6m diameter to the large size about 6.5 m long and 4m diameter. Equivalent evaporation ranges from about 900kg steam/h to about 14000 kg steam/h.

Another type of economic boiler is illustrated in Figure 4.5. This type is called a super-economic boiler. The boiler illustrated is oil fired through a central large-bore corrugated flue. This gives the flue gas its first pass. At the rear of the boiler the flue gas is deflected down and back to pass through a number of small bore tubes set in the bottom of the boiler. This gives the flue gas its second pass. At the front of the boiler the fire gas is deflected up and into the boiler again, this time to the third pass through another set of small bore tubes set in the sides of the boiler. At the back of the boiler the flue then the chimney.

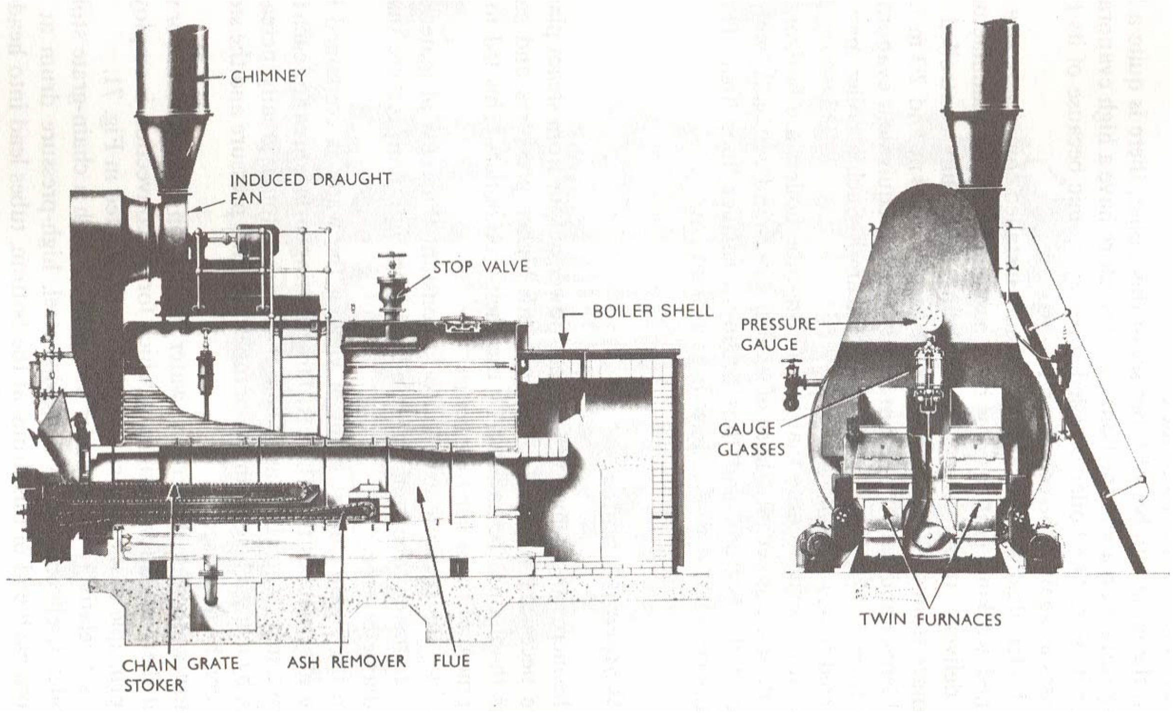


Figure 4.4.

With the large number of tubes set in this boiler, there is quite a high heating surface area and the boiler will therefore have a high evaporation rate for its size. The boiler is called super-economic because of its three gas passes as against the economic two passes. The boiler illustrated is capable of being installed completely as a self-contained working unit, having been assembled at the manufacturer's before delivery. It is fully automatic, being electrically controlled. The size ranges from an overall length and height of 3.4m and 2.3 m to an overall length and height of6.1 m and 3.7 m. The equivalent evaporation ranges from 680 kg steam/h to 8000 kg steam/h. Such a boiler may also be referred to as a package boiler.

Here, it is useful to note that not all fire-tube boilers are horizontally disposed. Some especially those of smaller size, stand vertically, with the furnace at the bottom, and thus occupy a smaller floor area. This is especially convenient where space is at a premium.

4.2.2- Water-tube boilers

With the increasing demand for higher power output from steam plant it became necessary to develop boilers with higher pressures and steam .outputs than could be handled by the shell-type boilers. This led to the development of the water-tube boiler.

Consider the pressure aspect. For a constant thickness of material, a smaller diameter tube can withstand a higher internal pressure than a larger diameter tube.

From the steam output point of view, if the water is contained in a large number of tubes, then there is a large heating surface area and each tube contains a small water bulk and hence the steam output is greatly increased.

Thus, by using a large number of tubes both the pressure and the steam output can be increased.

In most water-tube boilers the water circulation is by natural convection but there arc a few designs in which forced convection is employed. The principle of the water-tube boiler is illustrated in Figure 4.6.

This is a relatively small boiler and is fitted with a chain-grate stoker. The boiler itself consists of a welded-steel, high-pressure drum at the top. From each end of this drum, at the bottom, tubes lead into headers. These headers are connected by a large number of water tubes. Between the water tubes and the drum is fitted a superheater. Baffles are introduced across the water tubes to act as deflectors to the flue gas stream.
water tubes are fitted with a positive scope to give assistance to the formation of convection currents in the water. The high end of the tubes is at the hottest end of the boiler and the low end of the tubes is at the cool end of the boiler. Thus the hottest water and steam rise up the tubes and into the front of the drum while cool water falls from the rear of the drum into the rear of the tubes. Steam is removed from the drum and passed into the; superheater. Note that there are three flue gas passes across the water tube before the flue gas passes out of the boiler.

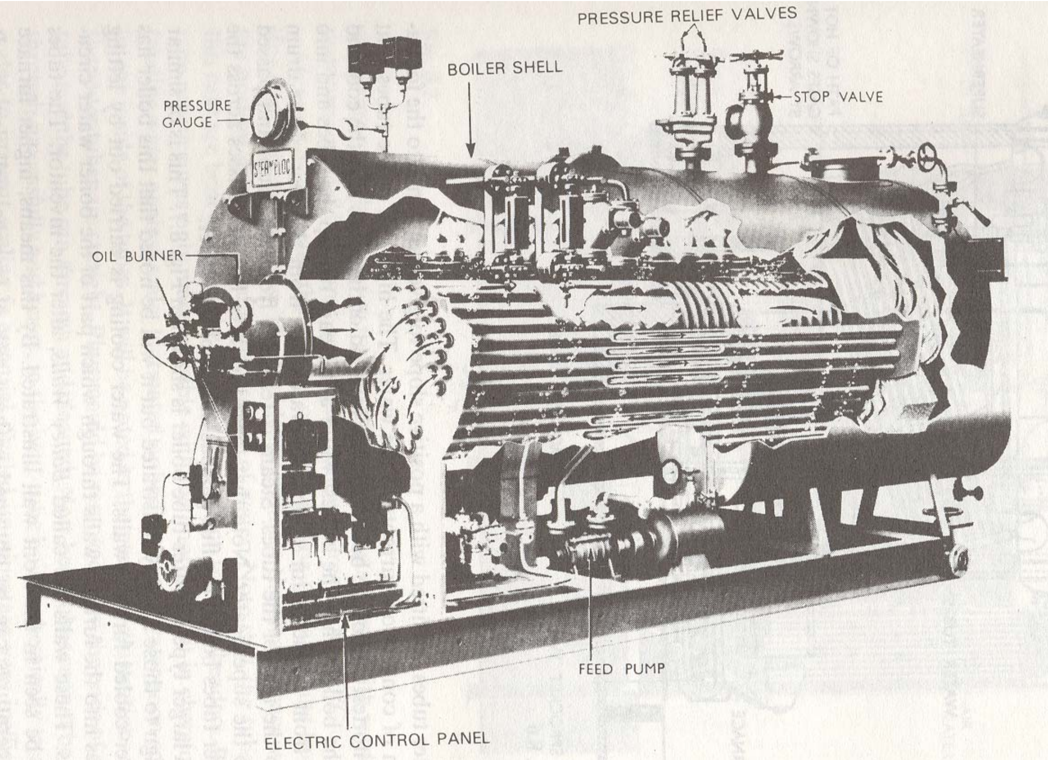


Figure 4.5

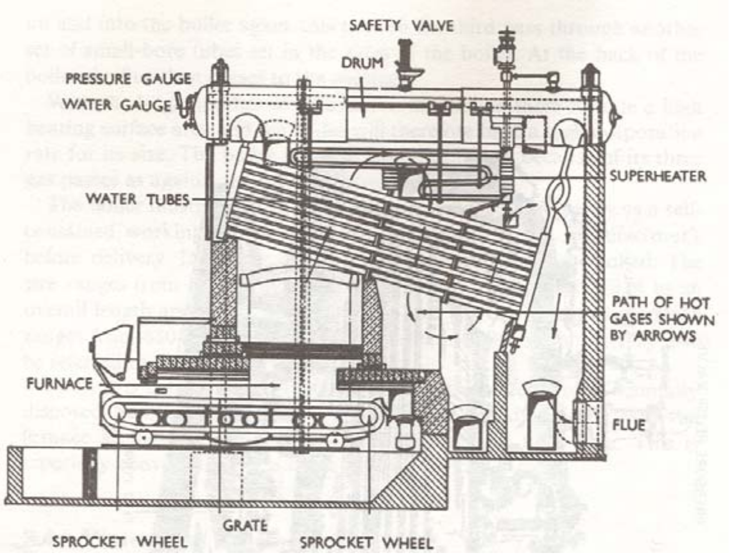


Figure 4.6.

A larger type of water-tube boiler is shown in Figure 4.7. This is of similar design to those already illustrated but it will be noted that this boiler has water-cooled furnace walls. The water cooling is carried out by letting tubes into the furnace walls through which part of the boiler water circulates. These walls are called **Bailey Walls**, after the inventor. The tubes can be seen in the front wall illustrated. By this means, higher furnace temperatures can be obtained without risk of furnace wall damage. It also assists the steam raising properties of the boiler and hence helps improve the boiler efficiency. The drum, in this case, is mounted across the boiler. An economizer and a tubular air; heater are fitted.

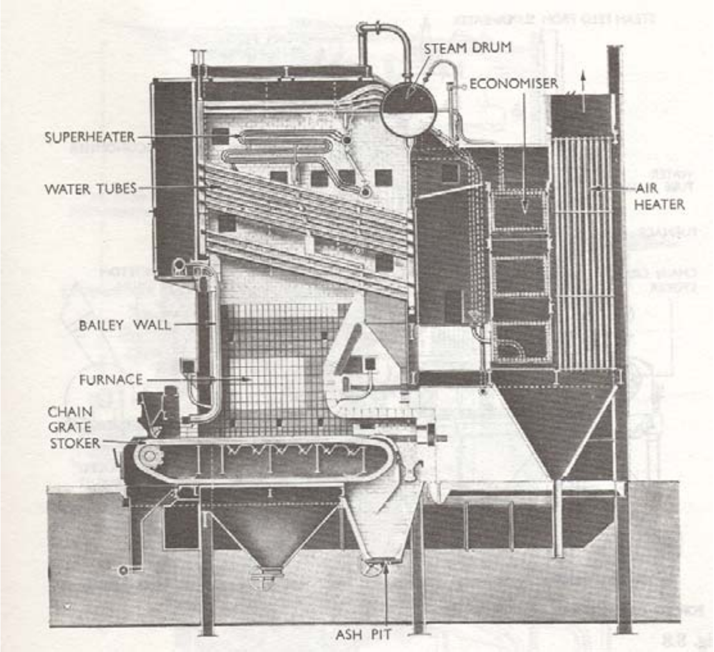


Figure 4.7.

Boilers of this design are capable of raising from 9000 to 45000 kg steam/h with working pressures up to 7.0 MN/m^2 and steam temperatures up to 450° C

Figure 4.8 illustrates the section of another type of water-tube boiler, this time with two drums. The top drum contains the steam space while the bottom drum and the tubes are full of water. Note that the tubes in this case are bent and that the drums are, in effect, the tube headers. Bentatube practice was only made possible with the improvement in chemically treating water against scale. Untreated or badly treated water soon scales up the inside of the tubes. It will be noted in the previous illustrations that the water tubes are straight. Plugs could be removed from the ends of each tube and the tube run through and cleaned. Now bent tubes cannot be run through successfully and hence, in the bent-tube boiler, chemically treated water to prevent scaling is essential. Scaled-up pipes are very inefficient since scale is a very poor conductor of heat. Also, as a result of scale, local corrosion may occur and also local overheating which may eventually result in a local blow-out. However, present-day water treatment is very successful in preventing scaling and bent-tube boilers are very common.

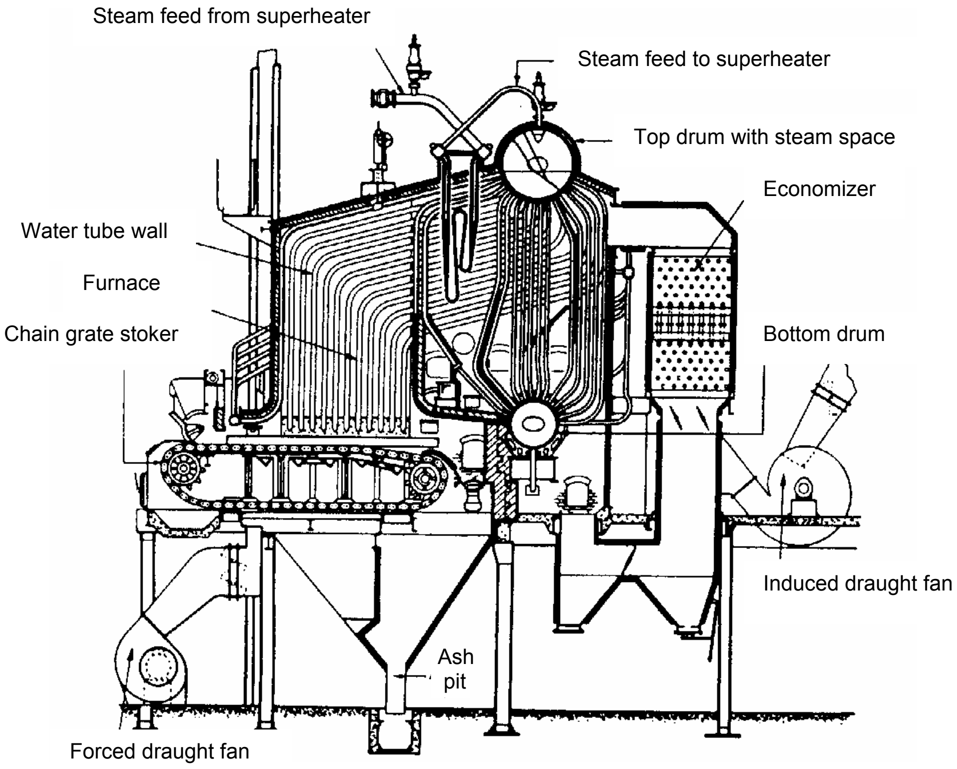


Figure 4.8.

Note the water tubes fitted up the furnace wall and the economizer. This illustration also shows clearly the steam feed from the steam space of the top drum into the superheater and from the superheater to the steam main. This boiler is also fitted with a forced-draught fan before the furnace and an induced draught fan after the economizer. Again the boiler is fitted with a chain-grate stoker.

A very large water-tube boiler of modern design is illustrated in Figure 4.9. It is called a radiant heat boiler.

The boiler is fired with pulverized coal. The coal-pulverizing mills are shown at the bottom. A section of one of these pulverizing mills is shown in Figure 4.10. Coal is fed into the mill through the feeder on the right. It drops into the mill via a chute and it is crushed to a powder by the rotating balls. Primary air is blown into the mill through the duct on the left. The air picks up the pulverized coal and the air-coal mixture leaves the mill through the top to be led into ducting feeding the boiler burners. The mill is driven by means of an electric motor through the shaft showing at the bottom right.

The pulverized fuel burners are set in the side of the boiler. A section of a type of pulverized fuel burner is shown in Figure 4.11 the fuel and primary air blow through the center of the burner and spray into the boiler furnace. To assist burning, secondary air is fed round the delivery of the fuel and primary air supply. The amount of secondary air can be controlled by the air register control gear. An oil lighting-up burner tube for igniting the burner is the above the burner.

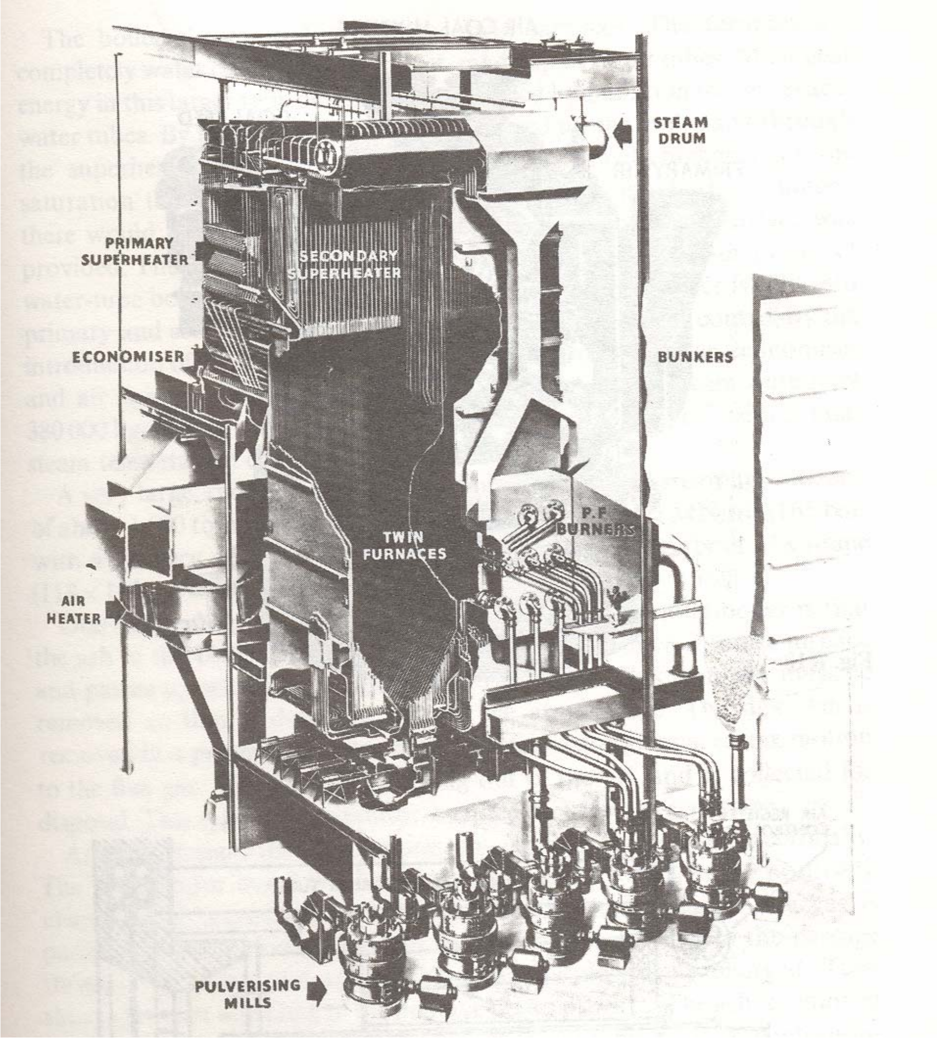

STEAM DRUM
PRIMARY SUPERHEATER
SECONDARY SUPERHEATER
ECONOMISER
BUNKERS
P.F BURNERS
TWIN FURNACES
AIR HEATER
PULVERISING MILLS

Figure 4.9

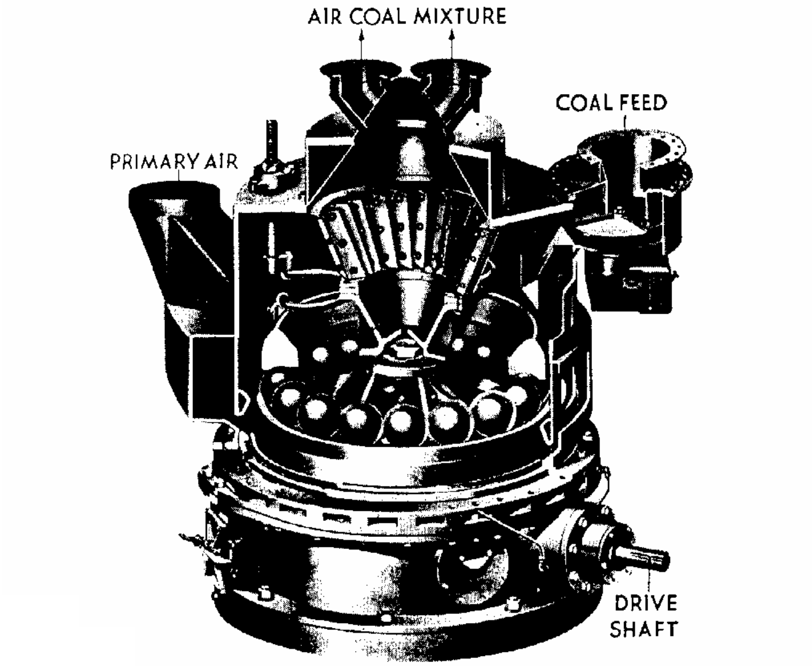


Figure 4.10

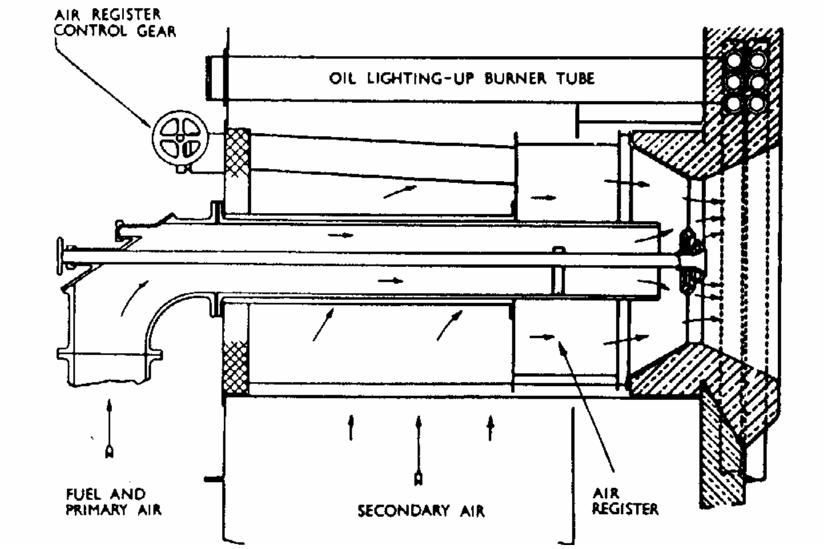


Figure 4.11

The boiler shown in Figure 4.9 has twin furnaces. The furnaces are completely water cooled, being in fact, made up of water tubes. Most heat energy in this larger type of boiler is transferred by radiation to the vertical water tubes. By the time the flue gas has passed up the boiler and through the superheater, the temperature is not very much higher than the saturation temperature of the steam drum. Under these circumstances, there would be little steaming improvement if a convection surface was provided. Thus, convection water tubes, as illustrated in the other types of water-tube boiler, are not included. Note that the superheater is split into primary and secondary sections so as to provide superheat control by the introduction of an attemperator. Note also the provision of an economizer and air heater. Such a boiler as that illustrated has a steam output of 380 000 kg/h. The superheater outlet pressure is $11 MN/m^2$ with a final steam temperature of 570°C.

A very large, radiant heat, oil-fired boiler can have a steaming capacity of about 1600 tonne/h (1.6×10^6kg) at a pressure of 16.5 MPa (165 bar) with a delivery steam temperature of 550°C. It will use about 118 tonne (118×10^3 kg) of fuel oil/h.

One of the difficulties of a pulverized fuel (P.F) burning boiler is that the ash in the coal is also pulverized and hence is blown into the furnace and passes up with the flue gas out of the boiler. This dust ash must be remove so that it does not pollute the atmosphere. The dust ash is removed in a precipitator. One type of precipitator gives a vortex motion to the flue gas. The dust is thus flung out of the gas and is collected for disposal. This type of precipitator is called a cyclone precipitator.

Another design, the electrostatic precipitator, is operated electrically. The precipitator consist of a bank of plates or wires, some positively charged and the others negatively charged. The ash-laden flue gas is through the plates the ash particles become negatively charged. They thus move over and cling to the positive plates or wires. The ash is removed by rapping the plates or wires with mechanical rappers. It falls into hoppers from which it is removed.

Another effect of the combustion of coal, and also the combustion of hydro-carbon fuels such as oil and petrol, is that the combustion can produce undesirable flue and exhaust products which have a pollutant effect in the atmosphere into which they are discharged. Such pollutants have an unfortunate and undesirable effect not only locally but also, at times, at some distance from their original source, having been carried away in the atmosphere. Two of the main pollutants appear to be the sulphur oxides (written generally as SO_x), and the nitrogen oxides (written generally as NO_x). These can form acids and compounds which have a corrosive effect on surroundings and buildings and also have a contributory effect on the formation of what is generally referred to as 'acid rain'. The acid rain appears to have a marked effect on some flora and fauna of forests and lakes. Because of these undesirable effects, much effort is increasingly being made to 'scrub' the flue gas of pollutants before they are discharged to atmosphere.

4.2.3- Once-through boilers

In this type of boiler, water is force-circulated in a single passage through the boiler which consists of a number of tubes in parallel. The pressure in the boiler can be above the critical pressure for steam (approximately 22.09MPa). Thus these boilers will operate at pressures of some 22 MPa to about 34 MPa. Steam temperature will be of the order of 600°C. Radiant heat furnaces are employed and general sizes vary from industrial plant up to large power stations. Advantages claimed with this type of boiler are that the welded construction avoids expansion troubles due to starting up and shutting down. Starting up and shutting down can be accomplished more rapidly. The boiler can be operated at any pressure and temperature over its load range. Steam can be supplied at gradually increased superheat temperature which assists turbine starting.

Small, once through, sub-critical boilers are also manufactured.

4.3- Coal firing

Historically, coal was the fuel most generally used for boiler firing in the industrial nations of the world. Nevertheless, during the recent past, there was some considerable switch to the use of oil firing of boilers due to the then availability of relatively cheap oil. However, more recently, supplies of oil have become more unreliable and more expensive.

Thus, there has become a renewed interest in the development and utilization of coal. It further appears that, generally, there is much more coal reserve naturally available than that of oil. Hence, many boiler plants that were converted from the use of coal to oil are being converted back to the use of coal once again. New boiler plant in the future will, no doubt, largely concentrate on the use of coal as fuel. In the preceding discussion on boilers, the use of coal-burning equipment such as the chain grate and the pulverized fuel burner were indicated and described. A further method of utilizing coal is by means of fluidized bed combustion.

A diagrammatic illustration of a fluidized bed combustor is shown in Figure 4.12.

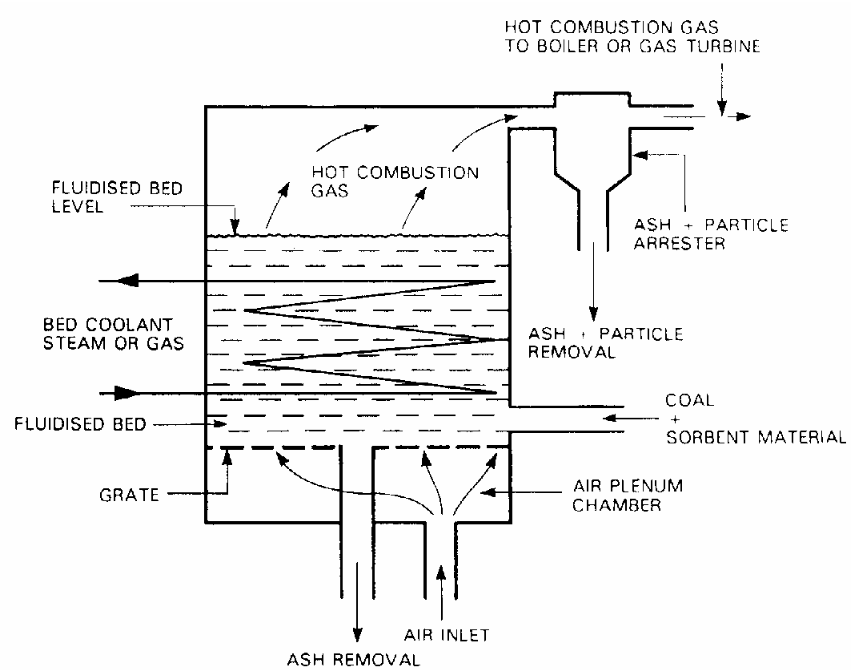


Figure 4.12.

The bed consists of a thick layer of fine inert is articles of such natural substances as sand or limestone. If such a bed has air blown through it, the air being evenly distributed through a grate at its base by means of a device such as a plenum chamber, then, at a particular velocity and mass flow of air, the bed will begin to behave like a fluid, or, in other words, it becomes a fluidized bed. In fact, a hollow vessel, such as a ball, would float on its surface in the same way as it would float on the surface of a fluid. If particles of coal are added to the fluidized bed, they become well mixed throughout the bed. If the temperature of the bed is high enough, then the coal will chemically combine with the oxygen in the air passing through the bed and it will bum. The fluid nature of the bed will ensure an even heat transfer through the bed and also to any coolant device immersed in it. Such coolant devices could be required to produce superheated steam in a boiler or hot gas for use in a gas turbine, as examples. Hot gas from the combustion process will leave the bed at its top surface. The hot gas can be arranged to proceed through a boiler or a gas turbine or industrial process plant.

Note that a device is required for the removal of the coal ash at the base of the bed through the bed grate. A further device, sometimes referred to as an arrester, is located at the top of the combustor to intercept any small ash particles (fly-ash), and any particles of bed material which leave the bed with the heat combustion gas. This leaves a cleaner hot gas to proceed to the plant which follows. Particles collected in the arrester are removed from its base.

4.4- Oil Firing

Many boilers, including some installed in ships, employ oil firing equipment. A typical oil burner is illustrated in Figure 4.13 Such an oil burner is fitted into the side of the boiler furnace. A heater is often necessary to bring the oil to the correct viscosity and the oil is then pumped to a pressure of up to 2 MPa (20 bar) and delivered to the burner via the tube shown at the bottom. The oil then passes through an atomizer in which it is broken up into very fine particles and sprayed as a cone, into the furnace. The spray nozzle can be seen, in section, at the centre front of the burner.

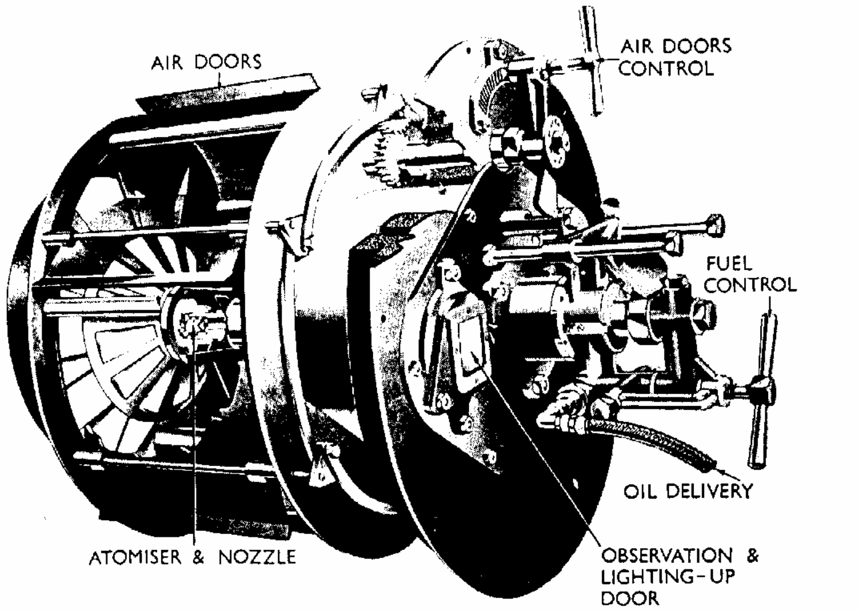


Figure 4.13

Combustion air is controlled by means of what is called a register. The air enters the register via air doors mounted round the burner. The amount of air can be controlled by varying the degree of opening of the air doors. The air doors are adjusted by means of the handle at the top which moves over position notches. An observation and lighting-up door is mounted; centre left, on the front. The front of the burner is heavily lagged and metal covered.

A large boiler, as in a power station, can employ as many as 32 oil burners.

4.5- Gas firing

Many of the smaller boilers of the shell type are gas fired. In these boilers burning gas jets are fed into tubes which pass through the water space in the boiler much in the same way as illustrated in the economic boiler.

Gas-fired boilers can be either horizontal or vertical.

4.6- Boiler Calculations

a - heat transfer required to form steam

Refer to Figure 7.14, let h_2 : specific enthalpy of steam formed (kJ /kg) .

h_1 : specific liquid enthalpy of feed water (kJ/ kg) .

Then, since the steam is formed at constant pressure, then the heat transfer that required to form *1kg* of steam in the boiler

$$=1 \times (h_2 - h_1)\ kJ \tag{4.8}$$

b- energy received from the fuel $\dot{Q}_f$

$$\dot{Q}_f = \dot{m}_f\ HHV = \dot{Q}_{in} \tag{4.9}$$

c- boiler thermal efficiency

$$\eta_B = \frac{Energy\ to\ steam}{Energy\ from\ fuel}$$

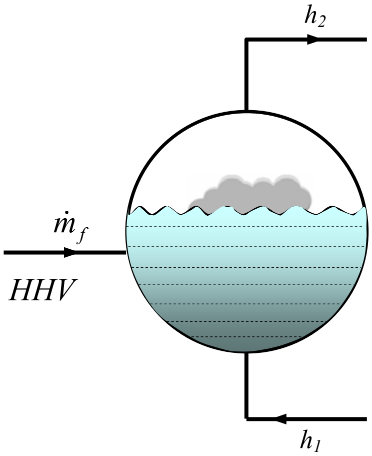


Figure 7.14

$$\eta_B = \frac{\dot{Q}_B}{\dot{Q}_f} = \frac{\dot{m}_s (h_2 - h_1)}{\dot{m}_f\ \ HHV} \tag{4.10}$$

where

$\dot{m}_s$: mass of steam raised in a given time.

$\dot{m}_f$: mass of fuel used in a given time.

d- equivalent evaporation of a boiler

Since some boilers can be operated under many different running conditions then, for these boilers, it is necessary to have some standard upon which to base, and compare their respective evaporation capacities. The standard commonly adopted is that the equivalent evaporation of a boiler from and at 100 ^{o}C.

$$\text{Energy received by steam} = \dot{m}_s\left(h_2 - h_1\right) \tag{4.11}$$

From this, the amount of water at 100 ^{o}C which could be evaporated into dry saturated steam at 100 ^{o}C, if supplied with amount of energy is then determined. This is then called the $\varepsilon quivalent\,\varepsilon vaporation\ (\varepsilon\varepsilon)$ of the boiler from and at 100 ^{o}C.

The specific enthalpy of evaporation at 100 ^{o}C = h_{fg} = 2257 kJ/kg

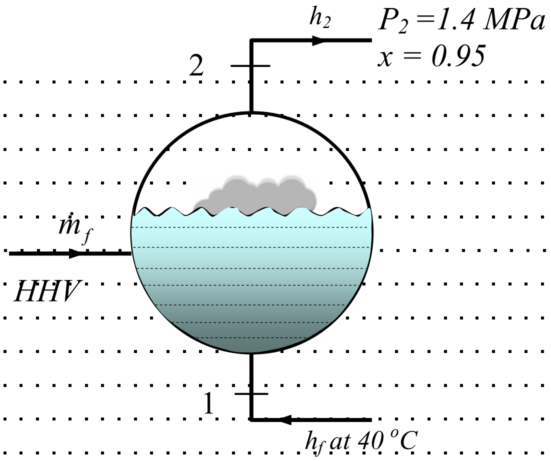
Then,

$$\varepsilon\varepsilon = \frac{\dot{m}_s\left(h_2 - h_1\right)}{\dot{m}_f\left(2257\right)} \tag{4.12}$$

Example 4.1

A boiler working at a pressure of 1.4 MPa evaporates 8 kg of water per kg of coal from feed water entering at 40 ^{o}C. The steam at the stop value is 0.95 quality. Determine the equivalent evaporation from and at 100 ^{o}C.

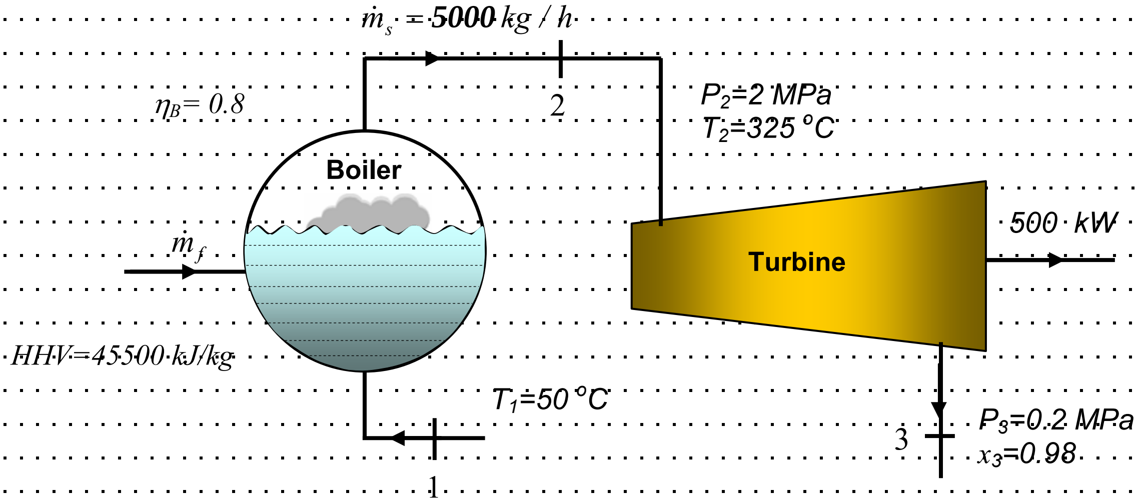
Solution:



Example 4.2

A boiler generates 5000 kg of steam per hour at 2 MPa the steam temperature is 325 °C and the feed water temperature is 50°C. The efficiency of the boiler is 80% when using oil of Higher Heating Value 45500 kJ/kg. The steam generated is supplied to a turbine which develops 500 kW and exhausts at 0.2 MPa, the quality of the steam being 0.98. Estimate the mass of oil used per hour and the fraction of the enthalpy drop through the turbine which is converted into useful work (turbine efficiency). If the turbine exhaust is used for process heating, find the heat transfer available per kg of exhaust steam above 50 °C.

Solution:



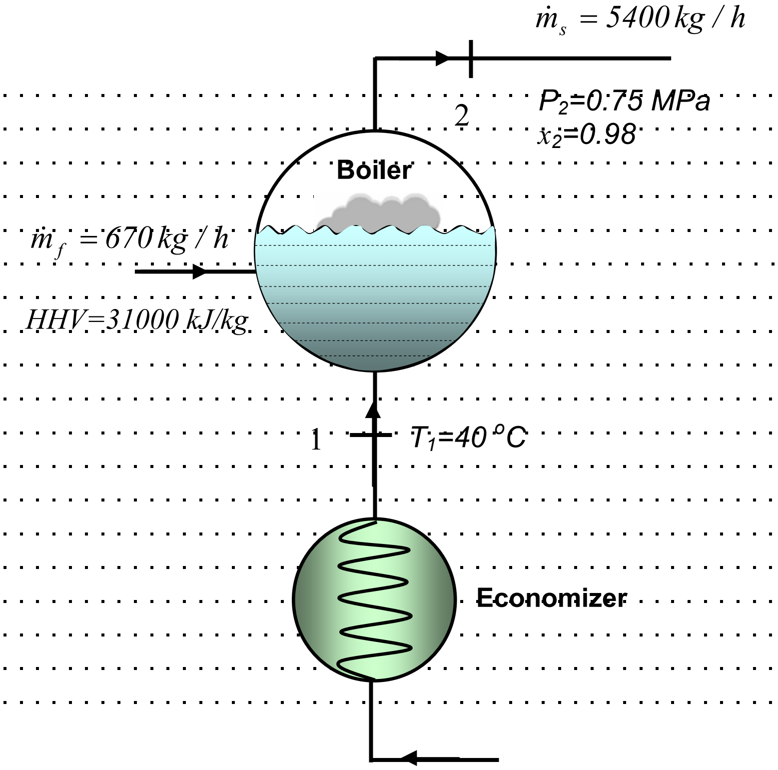
Example 4.3

A boiler plant supplies 5400 kg of steam/h at 0.75 MPa and 0.89 dry from feed water at 40 °C, when using 670 kg of coal/h having a Higher Heating Value of 31000 kJ/kg. Determine

a) - *i* - The efficiency of the boiler
   ii - The equivalent of evaporation from and at 100 °C.

b) Find the saving in coal/h if an economizer is fitted and it is estimated that the feed water could be raised to 100 °C, assuming (*i*) other conditions remained unaltered, (*ii*) the efficiency of the boiler increases by 5%.

Solution:



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4.7- Boiler circulation

The flow of water and steam within the boiler circuit is called circulation. Adequate circulation must be provided to carry away the heat from the furnace. If circulation is caused by density difference, the boiler is said to have *natural* circulation. If it is caused by a pump, it has *controlled* or *forced* circulation.

The natural circulation driving forces are explained with the simplified flow diagram of Figure (4.14). The driving pressure caused by natural convection. *ΔP*, is given by

$$\Delta P = gH\left(\rho_D - \rho_m\right) \tag{4.13}$$

where H is the height of the riser, ρ_D is the density of saturated water in the downcomer, ρ_m is the mean density of steam-water mixture in the riser and g is the gravitational acceleration (m/s^2).

$$\rho_m = \frac{\rho_{bottom} + \rho_{top}}{2} \tag{4.14}$$

Now, $\rho_{bottom} = \rho_D$ and $\rho_{top} = {}^{1}\!/_{v_{top}}$ $, v_{top} = v_f + x_{top}\, v_{fg}$

where x_{top} is the quality of the mixture at the top of the riser and is often referred as the top dryness fraction (TDF). The top dryness fraction is the reciprocal of the circulation ratio (CR).

$$CR = \frac{m}{m_g} = \frac{m_g + m_\ell}{m_g} = \frac{1}{TDF} \tag{4.15}$$

The most difficult of the parameters of equation (4.13) to obtain is the ρ_m. It is a function of the void fraction, α (volumetric quality) distribution along the riser height. The void fraction of the two phase mixture is defined as

$$\alpha = \frac{1}{1 + [(1+x)/x]\psi} \tag{4.16}$$

and $$x = \frac{1}{1 + [(1+\alpha)/\alpha]\left(\frac{1}{\psi}\right)} \tag{4.17}$$

where $\psi = \frac{v_f}{v_g} S$

where v_f and v_g are the specific volumes of the saturated liquid and vapor, respectively, at the pressure of the system, and $S = \left(\bar{V}_g \middle/ \bar{V}_f \right)$ is the slip ratio of the two phase mixture.

The two phases of the mixture do not travel at the same speed. Instead there is a slip between them, which causes the vapor to move faster than liquid. S is dimensionless number and found to vary between 1 and 10, approaching 1 at higher pressure (where the liquid and vapor densities approach each other). As the pressure increases, the slip ratio decreases.

The mixture density distribution ρ can be found from

$$\rho = (1 - \alpha)\rho_f + \alpha\,\rho_g \tag{4.18}$$

The average mixture density in the riser is obtained from

$$\rho_m = \frac{1}{H}\int_0^H \rho(z)\,dz \tag{4.19}$$

where z is the axial distance from the bottom of the riser.

In case of uniform axial heating, the mean density, ρ_m is solved by (El-Wakil, 1981)[*] to yield

$$\rho_m = \rho_f - \frac{\rho_f - \rho_g}{1-\psi}\left\{1 - \left[\frac{1}{\alpha_e(1-\psi)} - 1\right] \ell n \frac{1}{1-\alpha_e(1-\psi)}\right\} \tag{4.20}$$

where α_e is the void fraction at the riser exit.

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[*] El-Wakil M.M. “ Nuclear Heat transport” American Nuclear Society, LaGrange Park, I11, 1981.

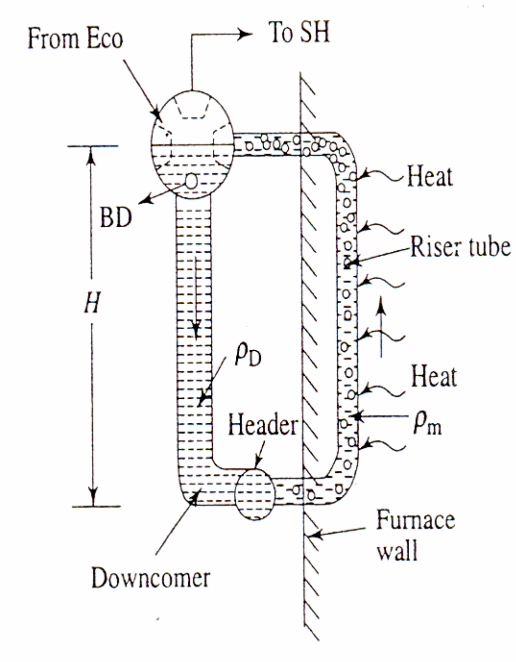

Figure 4.14

Mass flow rate of saturated water $\left(\dot{m}_f\right)$ entering the riser

$$\dot{m}_f = \rho_f A_c V, \quad A_c = \frac{\pi D_i^2 n}{4} \tag{4.21}$$

Rate of steam formulation in the riser tube,

$$\dot{m}_s = x_{top} \dot{m}_f \tag{4.22}$$

The Heat Transfer rate per unit *projected area*

$$\dot{Q} = \frac{\dot{m}_s h_{fg}}{A_p} = \frac{x_{top} \dot{m}_f h_{fg}}{D_O H} \tag{4.23}$$

Example 4.4 (Page 417)

A furnace wall riser, 18 m long, 76.2 mm OD and 6.1 mm thick receives saturated water at 80 bar and 1.5 m/s velocity. Assuming a CR of 12.5 and S of 1.2, determine (a) the pressure head developed, (b) the void fraction at riser exit, and (c) the heat transfer rate per unit projected area of the riser tube.

Solution:

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Example 4.5

A furnace wall riser, 15.25 m long, D_o = 75 mm and 5.6 mm thick receives saturated water at 7 MPa and 1.5 m/s velocity. Determine the energy absorption rate $\left(\dot{Q}/A_p\right)$ in the tube, if the maximum vapor fraction at the riser exit (V_{vap}) is 83%.

Solution:

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4.8- Boiler furnace heat transfer

The absorption factor can be presented by Hudson-Orrok empirical equation

$$F_{ab} = \frac{1}{1+\left[C\,\frac{\dot{m}_g}{\dot{E}_r}\left(\frac{\dot{E}_r}{A_p}\right)^{0.5}\right]} \qquad (4.24)$$

where

F_{ab} : absorption factor

C : furnace correction factor which depends on the furnace shape and used fuel properties.

$\dot{m}_g$: flue gases flow rate (kg/h)

$\dot{E}_r$: emitted energy rate (MJ/h)

A_p : tube projected area (m^2)

1- Energy released from the *fuel* can be expressed as

$$E_r = LHV - F\left(HHV\right) + AF\,c_{pa}\left(T_a - T_o\right) \qquad \left(kJ/kg\ fuel\right) \qquad (4.25)$$

$$LHV = HHV - 9 \times H\left(h_{fg}\right)_{at\ 25\,^oC} \qquad (4.26)$$

(see Chapter III equation 3.6)

where

F : unburned fuel fraction
AF: air fuel ratio (kg/kg fuel)
h_{fg} saturated water tables at 25 °C = 2442.3 kJ/kg
T_a combustion chamber inlet air temperature

2- Energy released rate, $\dot{E}_r$

$$\dot{E}_r = \text{Furnace volume}\left(\dot{V}_f\right) \text{ x fuel firing rate } (kJ/h\ m^3) \qquad (4.28)$$

3- Fuel mass flow rate

$$\dot{m}_f = \frac{\dot{E}_r}{E_r} = \frac{kJ/h}{kJ/kg\ fuel} = \frac{kg\ fuel}{h} \qquad (4.29)$$

4- Gas mass flow rate

$$\dot{m}_g = \dot{m}_f \frac{m_g}{m_f} = \frac{kg}{h} \qquad (4.30)$$

$$m_g = m_f + m_a - m_{ash} - m_c (kg) \qquad (4.31)$$

$$\frac{m_g}{m_f} = 1 + AF - \frac{m_{ash}}{m_f} - \frac{m_C}{m_f} \quad \left(\frac{kg}{kg\ fuel}\right) \qquad (4.32)$$

5- Moisture content in gases $(m_w) = 9 \times m_{H_2}$ (4.33)

percentage of water vapor in gases = $\frac{m_w}{m_g}$ (4.34)

6- Gas energy at furnace outlet $\left(\dot{E}_g\right)$

$$\dot{E}_g = (1 - F_{ab})\dot{E}_r \qquad (4.35)$$

energy rate per *kg* of outlet gases

$$E_g = \frac{\dot{E}_g}{\dot{m}_g} \quad (kJ/kg) \qquad (4.36)$$

energy released per *kg* of outlet gases

$$E_r = \frac{\dot{E}_r}{\dot{m}_g} \quad (kJ/kg) \qquad (4.37)$$

7- Absorbed energy can be expressed as:

$$E_{ab} = F_{ab} \times E_r \ (kJ/kg) \qquad (4.38)$$

Example 4.6

Furnace is designed to burn coal fuel and has the dimensions of 29 m height, 9.2 m width and 7.6 m depth. The following data are given:

HHV=31670 kJ/kg
Hydrogen percentage in the fuel=0.0501
Carbon percentage = 1%
Ash percentage = 0.082
Air fuel ratio, AF = 11.85 kg/kg fuel
Fuel firing rate = 950 MJ/h m^3
T_a = 367 ^oC
T_o= 25 ^oC
Unburned fuel fraction, F = 1%
Furnace correction factor, C = 0.0849
c_{pa} = 1.023 kJ/kg K

Calculate; Absorbed energy, E_{ab}, and the gas temperature at furnace outlet.

Solution:

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The energy in the gas is equivalent to the enthalpy of the gas leaving furnace above 25 ^{o}C . then from Table (4.1) at 1550 kJ/kg of gas and 3.5% moisture content , the gas temp , can be found .

$T_g \approx$ 1600 K

Temp (K)	Percent by mass of water vapor						
	0	2	4	6	8	10	12
300	2	2	2	2	2	2	2
400	103	105	106	108	110	112	113
500	206	21	213	217	220	224	228
600	312	317	323	328	334	339	344
700	421	428	435	443	450	457	465
800	532	542	551	650	569	579	588
900	646	658	669	681	692	703	715
1000	763	776	790	804	817	834	844
1100	882	897	913	929	945	963	977
1200	1002	1021	1039	1057	1075	1093	1112
1300	1125	1145	1166	1187	1207	1228	1249
1400	1248	1272	1295	1318	1342	1365	1388
1500	1374	1399	1425	1451	1477	1503	1529
1600	1500	1528	1557	1586	1614	1643	1672
1700	1627	1658	1690	1712	1753	178	1816
1800	1755	1789	1824	1858	1892	1927	1961
1900	1884	1921	1958	1996	2033	2070	2108
2000	2013	2053	2094	2134	2175	2215	2256

Table (4.1 enthalpy of flue gas, above 25 ^{o}C in kJ/kg

4.9- Energy balance of a steam generator

The boiler energy balance based on the higher heating value *(HHV*) of the fuel, consists of eight items that represent the energy absorbed and seven loss factors.

The steam generator losses results principally from incomplete combustion of the fuel, and from an outlet gas temp. that exceeds the inlet air temp.

The steam generator energy balance (see Figure 4.15) consists of the following items:

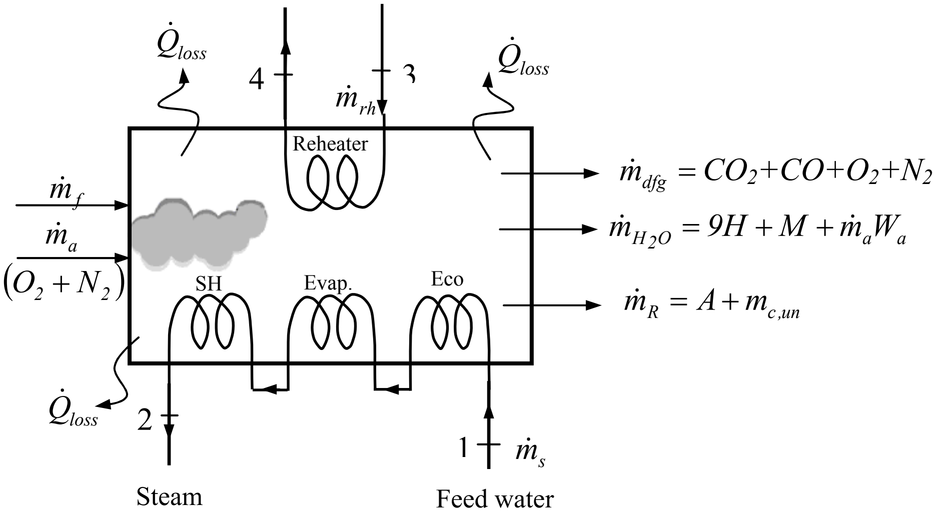

Figure (4.15)

1- Energy absorbed by the water (Useful energy)

$$e_1 = \frac{\dot{m}_s (h_s - h_w) + \dot{m}_{rh} \, \Delta h_{rh}}{\dot{m}_f} \quad \left(\frac{kJ}{kg\ fuel}\right) \tag{4.39}$$

where:

$\dot{m}_s$: steam flow rate at the superheater outlet (kg/h)

$\dot{m}_{rh}$: steam flow rate at the reheater outlet (kg/h)

$\dot{m}_f$: fuel firing rate (kg/h)

h_s : enthalpy of the steam at the superheater outlet (kJ/kg).

h_w : enthalpy of the entering feed water (kJ/kg).

Δh_{rh}: enthalpy change of the steam flowing through the reheater (kJ/kg).

2- Energy loss due to dry stack gas

$$e_2 = m_{dg} \, c_{pg} \left(T_g - T_a\right) \tag{3.40}$$

where

m_{dg}: mass of dry gas kg/kg fuel.

c_{pg}: specific heat of the dry gas = [1.004 kJ/kg.K]

T_g : outlet or stack gas temperature ^oC.

T_a : inlet air temperature °C.

3- Energy in the stack gas attributed to the moisture in the fuel.

$$e_3 = m_m(h_v - h_f) \tag{4.41}$$

where

m_m: mass of water in the fuel (kg/kg fuel).
h_v: enthalpy of superheated water vapor in the stack gas at T_g and the partial pressure of the water vapor (P_p = 10 kPa) (kJ/kg)
h_f : enthalpy of the liquid water at T_a (kJ/kg).

Note :

The partial pressure of the water vapor in the composition of gas mixture is low, then h_v is virtually independent of pressure and can be evaluated at the representative pressure of 10 KPa and T_g .

4- Energy in the stack gas attributed to the water vapor formed by combustion of the hydrogen in the fuel.

$$e_4 = 9 \times (m_H)_{net}(h_v - h_f) \tag{4.42}$$

where

$$(m_H)_{\text{net}} \text{ (kg/kg fuel)} = \left[(m_{H_2})_{fuel} - \frac{m_m}{9} \right]$$

5- Energy in the stack gas attributed to superheating of water vapor in the combustion air

$$e_5 = m_a\ \omega\ c_{pv}\ (T_g - T_a) \text{ (kg/kg fuel)} \tag{4.43}$$

where

m_a : mass of dry combustion air (kg/kg fuel)
ω : specific humidity (moisture content) of the air (kg vapor/ kg dry air)
c_{pv} : specific heat of water vapor, usually taken as (1.88) (kg/kg.K) .
T_g : outlet or stack gas temperature, °C
T_a : inlet air temp. °C

6- Losses attributed to incomplete combustion and indicated by the carbon monoxide in the stack gas.

$$e_6 = \frac{CO}{CO_2 + CO} \times m_{c,b}(23560) \tag{4.44}$$

where

CO : carbon monoxide in the stack gas (mol/100 mol of dry gas)
CO_2 : carbon dioxide in the stack gas (mol/100 mol of dry gas)
m_{cb} : mass of carbon burned (kg/kg fuel)

7 – Losses attributed to unburned carbon

$$e_7 = m_{ucb}\,(32750\,) \tag{4.45}$$

where

m_{ucb} : mass of unburned carbon (kg/kg fuel)
32750 : higher heating value of carbon (kJ/kg)

8 – R and U [Radiation and uncountable losses]

$$e_8 = HHV \text{ of the fuel } - \sum_{i=1}^{7} e_i \quad (\text{ kJ/ kg fuel }) \tag{4.46}$$

Steam generator efficiency

The efficiency of the steam generator is equal to the ratio of the energy absorbed by the water to the higher value of the fuel . Thus

$$\eta_B = \frac{\dot{m}_s\left(h_s - h_w\right) + \dot{m}_{rh}\,\Delta h_{rh}}{\dot{m}_f\,HHV} = \frac{e_1}{HHV} \tag{4.47}$$

All energy losses at a particular load can be measured, and from Eq. 4.47 the efficiency of the steam generator can be estimated. Then the fuel consumption rate can be determined.

Example 4.7

Calculate the thermal energy and efficiency of a boiler operating with the following data: (Assume there is no reheat process)

$\dot{m}_s$ =200000 (kg/h), P_e= 4MPa, T=405 °C
$HHV=33330$ kJ/kg
$\dot{m}_f$ =17650 kg/h
Refuses =1382 kg/h
m_a=14 kg dry air/kg fuel
$\omega=0.013$ kg/kg dry air
T_a= 27 °C
T_g = 167 °C
$P_{atm}=96.5$ kPa
Entering feed water at 4.6 MPa and 162°C
m_m= 0.033 kg/kg fuel

Fuel ultimate analysis		*Flue gas analysis*
C=0.8165	N=0.0127	CO_2 =14%

H=0.0446 S=0.0104 CO=0.2%
O=0.0525 A=0.0633 O_2=4.8%

Solution:

4.10 Boiler Draft loss

In order to calculate the fans power, the pressure loss of the flue gases inside the boiler should be calculated. The pressure loss due to the friction, ΔP_f, can be evaluated from the following empirical equation;

$$\Delta P_f = \frac{f \times N \times G_{max}^2}{\left(6.34 \times 10^8\right)\rho_g} \qquad \left[cm\ H_2O\right] \qquad (4.48)$$

where

f: friction factor and it's a function of Reynolds Number (Re) and depends on the tubes diameter and arrangements.
N: number of transverse rows
G: mass velocity, kg/m^2 s= ρu_{max}=$\dfrac{\dot{m}_g}{A_{min}}$
ρ_g: flue gas density (kg/m^3)

The empirical friction factor is given by

a- for gas flow through in-line tube rows (see Figure 4.16a)

$$f = \left\{0.25 + \frac{0.118}{\left[\left(S_n - d_o\right)/d_o\right]^{1.08}}\right\} Re_{max}^{-0.16} \qquad (4.49)$$

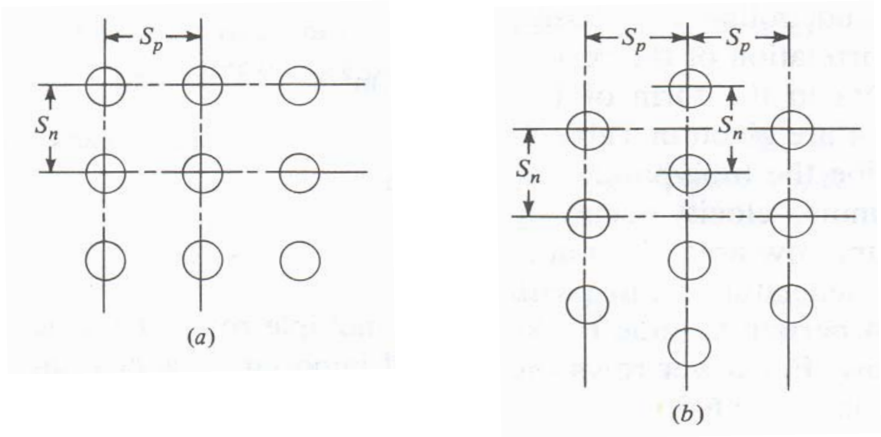

Figure (4.16)

b- for gas flow through staggered tube rows (see Figure 4.16b)

$$f = \left\{0.044 + \frac{0.08\,S_p/d_o}{\left[\left(S_n - d_o\right)/d_o\right]^{0.43+1.13d_o/S_p}}\right\} Re_{max}^{-0.15} \qquad (4.50)$$

The maximum velocity is

$$u_{max} = u_{\infty}\left(\frac{S_n}{S_n - d_o}\right) \tag{4.51}$$

and $Re_{\text{max}} = \dfrac{\rho\, u_{max}\, d_o}{\mu} = \dfrac{G_{max}\, d_o}{\mu}$ (4.52)

When the flue gases flow vertically through the bank of tubes, a stack effect correction factor has to be added to the total pressure losses. The stack effect can be calculated as

$$\Delta P_s = \frac{L\, g}{97.76}\left(\rho_a - \rho_g\right) \qquad \left[cm\, H_2O\right] \tag{4.53}$$

where L is the vertical distance (m) and g is the gravitational acceleration (m/s^2)

Then, the total flue gases pressure losses through the boiler is

$$\Delta P = \Delta P_f \pm \Delta P_s$$

The + sign for downward flow and – for upward flow.

Revision Questions

4.1- A boiler with superheater generates 6000 kg/h of steam at 1.5 MPa, 0.98 dry at exit from the boiler and at a temp of 300 ^oC on leaving the superheater. If the feed water temp is 80 ^oC and the overall efficiency combined boiler and superheater is 85% Find.

a – the amount of coal of HHV=30000 kJ/kg used per hour.
b – the equivalent evaporation from and at 100 ^oC for the combined unit.
c – the heating surface required in the superheater if the heat transmission may be taken as 450000 kJ/m^2 of heating surface per hour.

Ans : a- 646 kg/h, b-11.3 kg steam/kg fuel, c- 4.461m^2

4.2- The feed water to a boiler enters an economizer at 32^oC and leaves at 120 ^oC, being feed into the boiler at this temp, the steam leaves the boiler 0.95 dry at 2 MPa and passes through a superheater where its temperature is raised to 250 ^oC without change of pressure, the steam output is 8.2 kg/kg of coal burned and the HHV of the coal is 28000 kJ/kg. Calculated the energy picked up per kg of water and steam in

a - the economizer.
b – the boiler.
c – the superheater.

Take the specific heat of superheated steam =2.093 kJ/kg K, c_{pw} = 4.187 kJ/kg.K and expressing the answering as percentages of the energy supplied by the coal.

Ans. a - 10.78%, b – 64.44%, c – 5.78%

4.3-Furnace is designed to burn coal fuel, the heat transfer area (A$_p$) is 1250 m^2, the emitted heat transfer ($\dot{E}_r$) is 2.525 x10^6 MJ/h, absorption factor, F_{ab}= 0.42, the furnace constant, C = 0.085, and the water vapor percentage in the burnt gases is 3%. Determine the gas flow rate ($\dot{m}_g$) and the gas temperature

Ans: 912736.62 kg/h,1647 K.

4.4-A boiler furnace has the dimensions of 8 m with 6 m depth and 20 m height, the lower heating value LHV of the fuel =27000 kJ/kg. The temp of the air entering the furnace =300 ^oC the temp of ambient air =20 ^oC. The mass of air ($\dot{m}_a$) =10 kg/kg fuel, and

$F = 0.0, \quad \dot{m}_f = 40000$ kg/h, $\quad \dot{m}_a = 400000$ kg/h, $\; m_{ash} = 0.0$

$m_c = 0.0, m_H = 0.1, F_{ab} = 0.53$

Find, $\dot{E}_r$, $\dot{m}_g$, $\dot{m}_w$, $\dot{E}_g$ & $\dot{E}_{ab}$

Ans. 119476 MJ/h, 440000 kg/h , 8.2%, 561450.7 MJ/h,1439 kJ/kg

4.5- A boiler furnace has an average energy absorbing surface (A$_p$) of 1150 m^2. Fuel fired rate $\dot{m}_f$ = 6530 kg/h. LHV = 29811 kJ/kg. The unburned fuel fraction (F) = 0.

The air fuel ratio is 11.73 kg/kg fuel and flue gas flow $\left(m_g / m_f\right)$ is 12.6 kg/kg fuel, the water vapor in flue gas is 4 % and the furnace correction factor (C) is 0.085 calculate.

F_{ab}, E_r, E_{ab} and E_g . Assume c_{pa} = 1.023 kJ/kg K

Ans: 0.418, 2672 kJ/kg, 1117.3 kJ/kg, 1555 kJ/kg

4.6-A steam generator produces 300000 kg steam per hour the gas flow is 365000 kg/h. the water enters and leaves economizer of 105 and 140 ^oC respectively. the inlet and outlet gas temp are 436 ^oC and 330 ^oC the tube outside diameter D_o =50 mm, the tube length 3.2 m, the tube are arranged in an in line manner so that the spilling in both the normal and paralleled dimensions to the flow is 75mm, U_0 =75 W/m^2 K and the number of tubes per horizontal row in 129 tubes, If μ=3.03 x 10^{-5} N.s/m^2, R =0.286 kJ/kg K. Find the number of tubes in the vertical direction and draft loss.

Ans; 8 rows, 3.0472 cm H_2O

4.7- A power plant boiler has the following data:

Steam flow rate =100000 kg/h, steam conditions 3.5 MPa and 350 ^oC, feed water enthalpy =600 kJ/kg, fuel firing rate =12000 kg/h, fuel HHV = 26000 kJ/kg, water vapor in combustion air =0.015 kg/kg dry air, free moisture in the fuel =0, unburned refuses =0, combustion air 15 kg/kg fuel, c_{pv} =1.88 kJ/kg K, c_{pg} =1.004 kJ/kg K, stack gas temperature = 150 ^oC, combustion air temperature and fuel temperature = 25 ^oC.

Fuel ultimate analysis: C=0.85, H= 0.05, Ash =0.02,
Flue gas analysis: CO_2 =15% CO=0.1%
Calculate the heat balance and boiler efficiency

Ans:
e_1=20866.67 kJ/kg fuel, e_2=1951.5 kJ/kg fuel
e_3=0 kJ/kg fuel, e_4=1205.15 kJ/kg fuel
e_5=52.875 kJ/kg fuel, e_6=132.6 kJ/kg fuel
e_7=0 kJ/kg fuel, e_8=1791.6 kJ/kg fuel
η_B=80%

4.8-A boiler generates 180000 kg/h steam. At superheater outlet, the steam pressure and temperature are 7 MPa and 560 ^oC. The steam enters the superheater at 7.1 MPa with quality, x, =1. The gas temperature entering and leaving the superheater are respectively 1045 ^oC and 860 ^oC. U_o=50 W/m^2 K. Calculate the superheater heat transfer is for;

(a) counter flow of steam and gas
(b) Parallel flow of steam and gas

Ans: 1141.85 m^2, 1193.97 m^2

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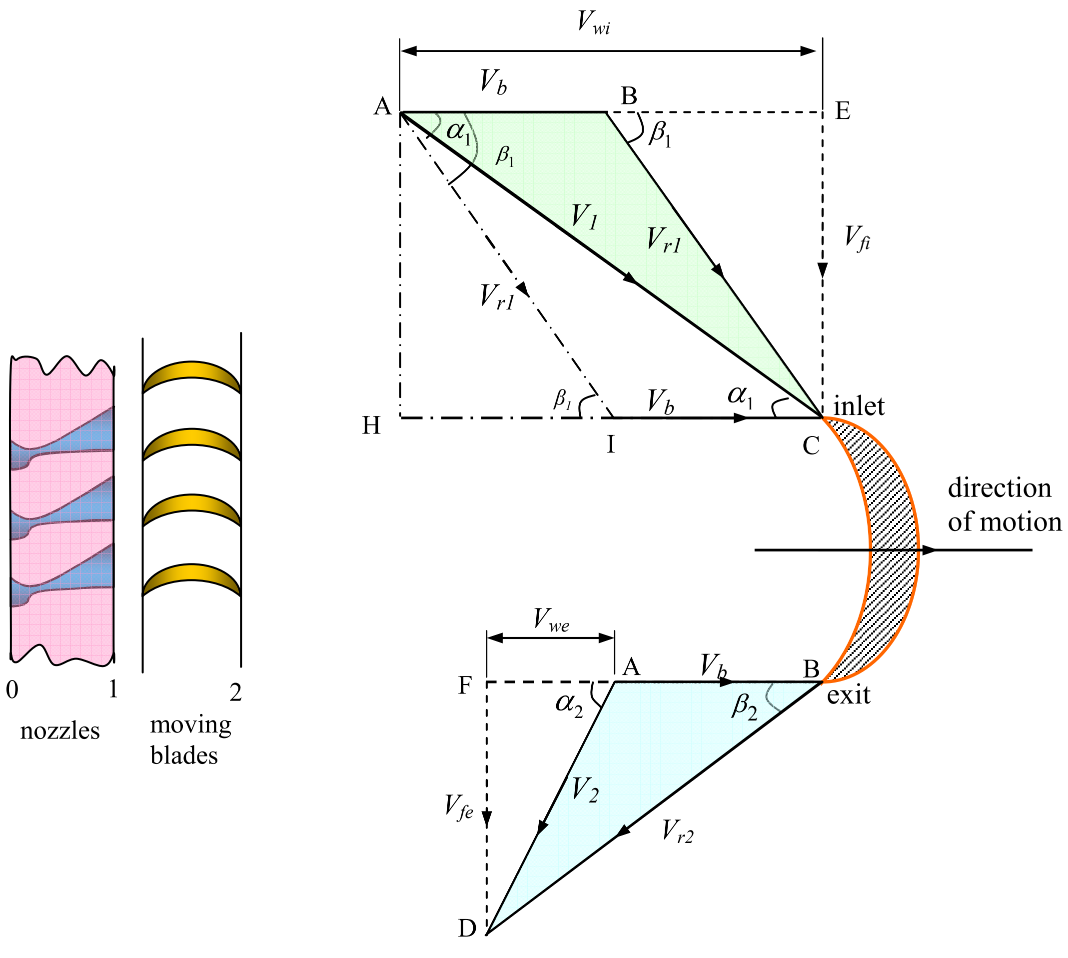
Power Plant Engineering and economy

MET401

Department of Mechanical Engineering Technology



Chapter V- Steam Turbines

2010

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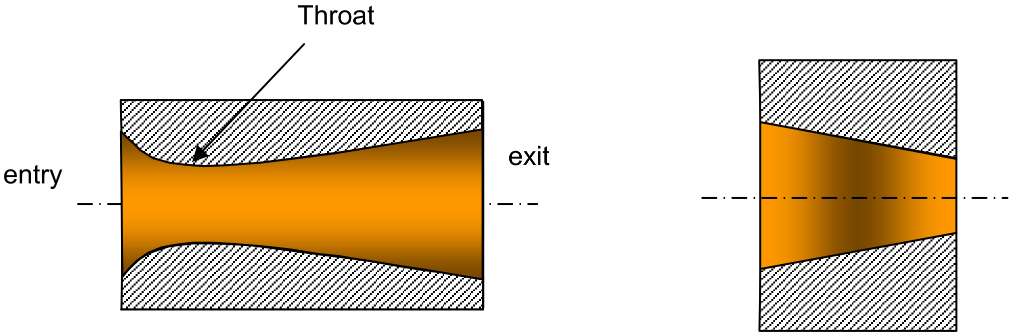
Chapter V

Steam Turbines

5.1. Flow through nozzles

A nozzle is a duct of varying cross-sectional area so designed that a drop in pressure from inlet to outlet accelerates the flow. A duct which decreases the velocity of a fluid and causes a corresponding increase in pressure is called diffuser. The same duct may be either a nozzle or diffuser depending upon the end conditions across it. If the cross-section of a duct decreases continuously from inlet to exit, the duct is said to be ***convergent***, and if it increases from inlet to exit, the duct is said to be ***divergent***. If the cross-section initially decreases and then increases, the duct is called convergent-divergent. The minimum cross section of this duct is referred to as throat.



Convergent-divergent

Convergent

Figure (5.1)

The flow through a nozzle usually occurs at very high speed, and there is little time for the fluid to gain or lose energy by a flow of heat through the walls of the nozzle as the fluid passes though it. The process is therefore always assumed to be adiabatic. Also, no work crosses the boundary during the process and neglecting the potential energy change. The continuity equation and SSSF energy equation become

$$\dot{m}_i = \dot{m}_e = \dot{m} = \frac{A\,V}{v} \tag{5.1}$$

$$\cancelto{0}{Q_{c.v.}} + \dot{m}\left(h_i + \left(\frac{V_i^2}{2} + \cancelto{0}{g Z_i} \right) \times 10^{-3} \right) = \dot{m}\left(h_e + \left(\frac{V_e^2}{2} + \cancelto{0}{g Z_e} \right) \times 10^{-3} \right) + \cancelto{0}{\dot{W}_{c.v.}}$$

$$h_i + \frac{V_i^2}{2000} = h_e + \frac{V_e^2}{2000}$$

$$V_e = \sqrt{2000\,(h_i - h_e) + V_i^2} \tag{5.2}$$

If the V_i is very small, then

$$V_e = \sqrt{2000\left(h_i - h_e\right)} = 44.72\left(h_i - h_e\right)^{1/2} \quad (m/s) \tag{5.3}$$

5.1.1- Pressure variation along the nozzle

1-Convergent nozzle

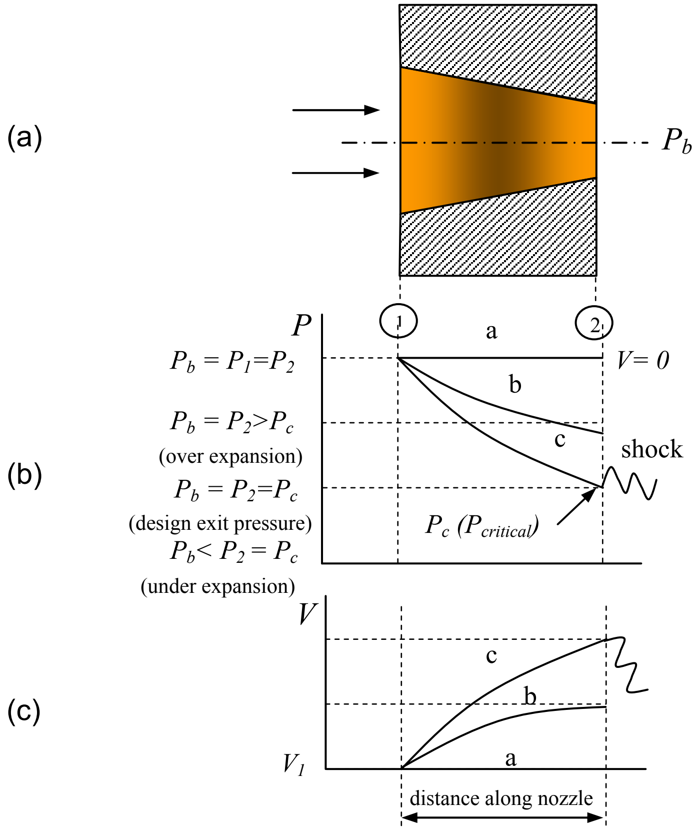


Figure (5.2)

2-Convergent-Divergent nozzle

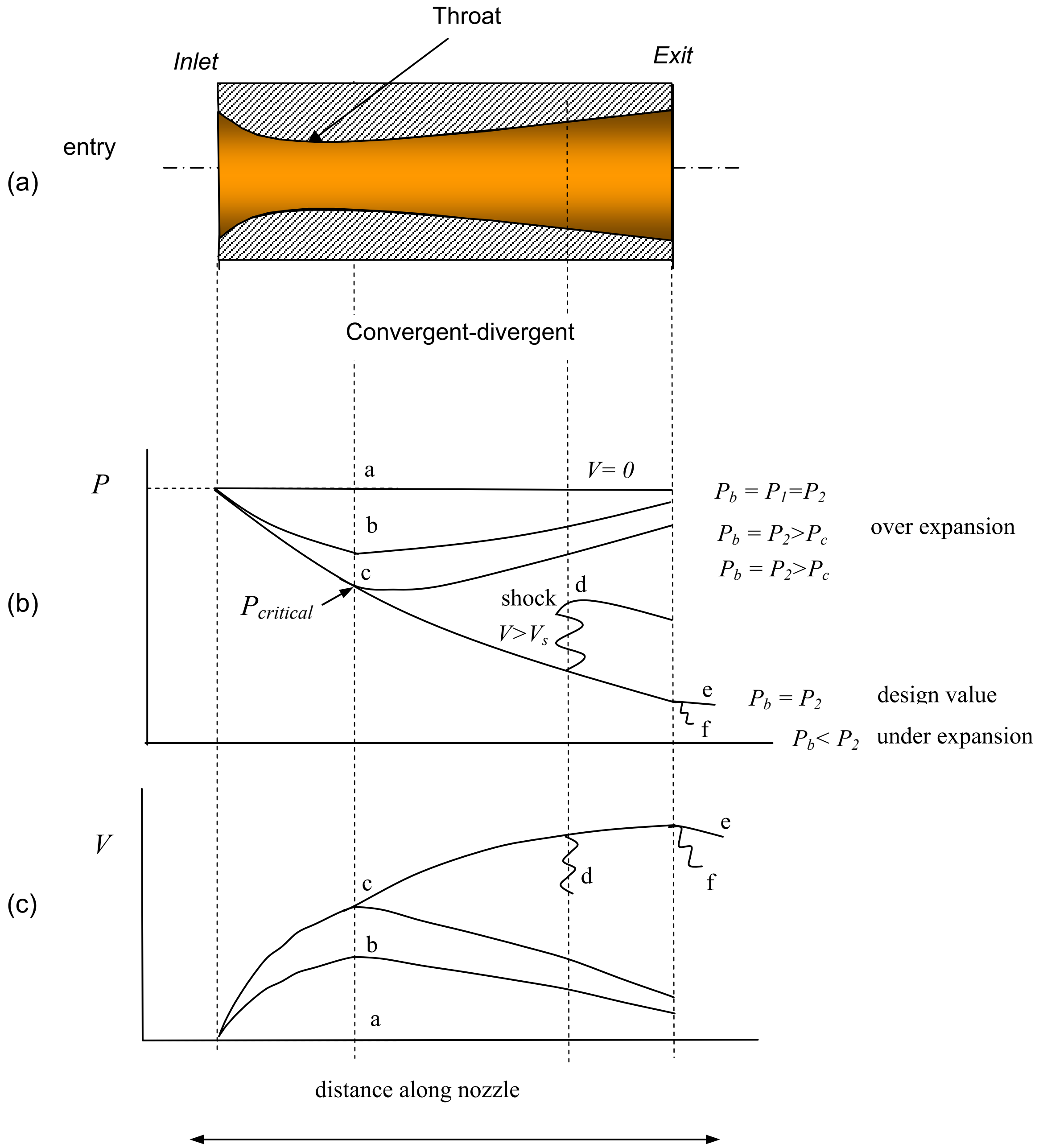


Figure (5.3)

5.1.2- Effect of Friction on nozzle performance

1. friction between fluid and walls of the nozzle
2. friction between fluid layers
3. bumps inside the nozzle

4. expansion inside the nozzle
    - thermal descent (10% → 15%).
    - Increasing the quality at the nozzle exit, which leads to increase the steam volume?

5.1.3-Nozzle efficiency

In nozzle design, the usual practice is to base all calculations on isentropic flow and then make an allowance for friction using a coefficient or efficiency.

The nozzle efficiency, η_N, is defined as the ratio of the actual enthalpy drop to the isentropic enthalpy drop (see Figure 5.4)

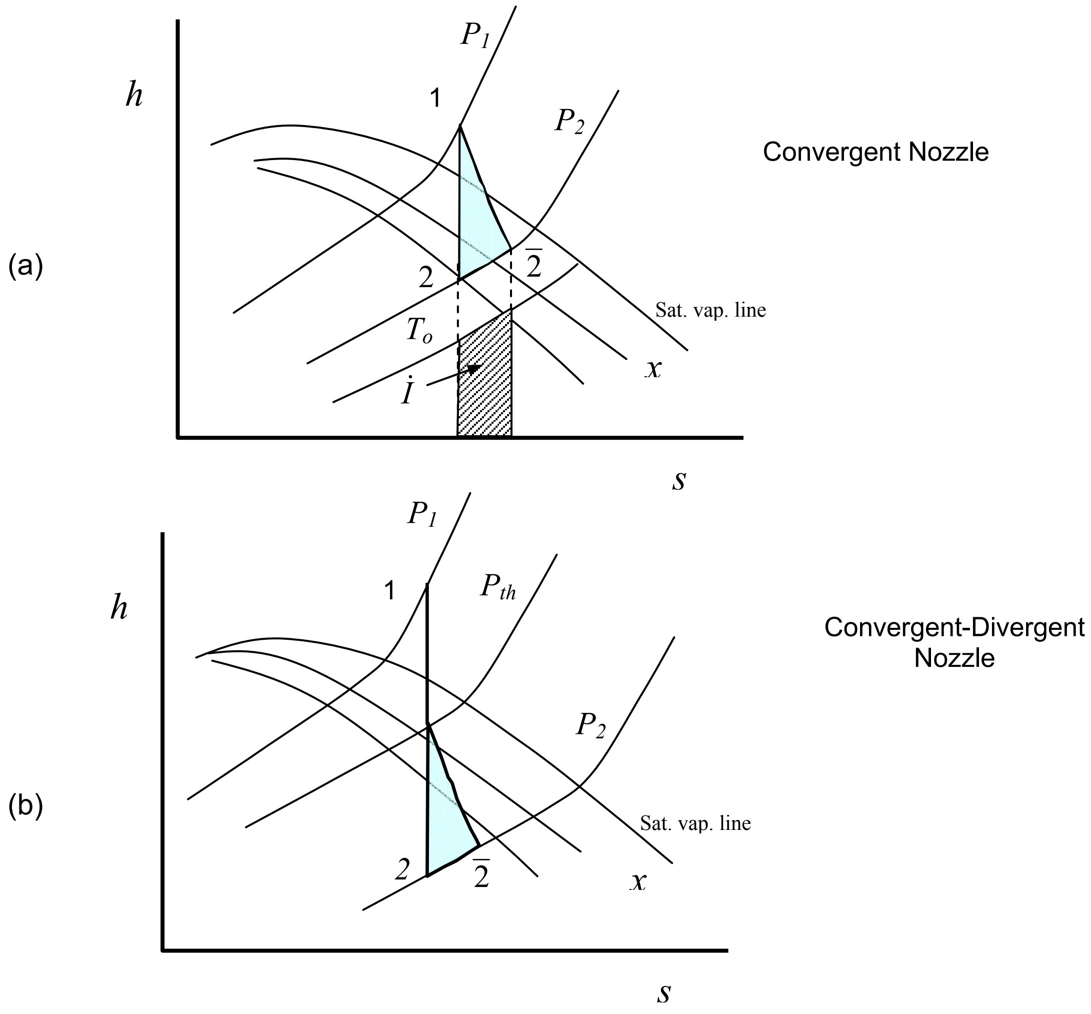


Figure (5.4)

$$\eta_N = \frac{\text{Actual enthalpy drop}}{\text{Isentopic enthalpy drop}} = \frac{h_1 - h_{\bar{2}}}{h_1 - h_2} \qquad (5.4)$$

The nozzle exit velocity can be expressed for an ***ideal gas***

$$\frac{V_e^2}{2} = (h_i - h_e) = C_p(T_o - T) \tag{5.5}$$

and $C_p = k\,C_v$, $C_v = C_p - R$, $C_p = \dfrac{k\,R}{k-1}$ (5.6)

Then $\dfrac{V_e^2}{2} = \dfrac{k\,R}{k-1}(T_o - T)$

$$\frac{T_o}{T} = \frac{V_e^2(k-1)}{2kRT} + 1 \tag{5.7}$$

The sonic velocity is given, by

$C^2 = \dfrac{dP}{d\rho}$ and $P = \rho RT$ for an ideal gas $C = \sqrt{k\,R\,T}$ (5.8)

The Mach number is defined as the ratio of the actual velocity V to the sonic velocity C.

$$M = \frac{V_e}{C} \tag{5.9}$$

The temperature ratio

$$\frac{T_o}{T} = \frac{M^2(k-1)}{2} + 1 \tag{5.10a}$$

By substituting $M=1$ in the above equation

$\dfrac{T_o}{T} = \dfrac{(k+1)}{2}$, and (5.10b)

$$\left(\frac{T_o}{T}\right)^{\frac{k}{k-1}} = \frac{P_o}{P} \tag{5.10c}$$

$$\frac{P}{P_o} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{5.11}$$

For steam, an approximation can be obtained if it is assumed that steam expands isentropically in the nozzle following the law $PV^n=C$, where *n* is the isentropically index and it is not a ratio of specific heats as is *k*. Then, the throat pressure can be calculated as follows

$$\frac{P_{th}}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \quad = \quad P_{th} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} P_1 \tag{5.12}$$

For dry and **saturated steam**, $n = (1.035 + 0.1\,x)$, when $x=1$, then $n = 1.135$, and

$$P_{th} = 0.577\ P_1 \tag{5.13a}$$

and when the steam entering the nozzle is **superheated**, n = 1.3, then

$$P_{th} = 0.5457\ P_1 \tag{5.13b}$$

Example 5.1

Steam at 0.7 MPa and 250 ^{o}C expands in a nozzle to 0.1 MPa. Determine the throat and nozzle exit cross section areas if the steam flow rate is 2.5 kg/s and neglecting the loss between the entrance of the nozzle and the throat. Assuming that, the nozzle efficiency between the throat and the nozzle exit is 0.9.

Solution:

Example 5.2

Steam at 1.8 MPa and 275 ^{o}C expands through a convergent-divergent nozzle to 0.1 MPa. If there is 10 % loss in energy between the throat and the nozzle exit and the vapor flow rate is 0.5 kg/s, determine the cross-sectional area of the throat and the nozzle exit.

Solution:

Revision Questions

5.1.1- A steam at a pressure of 1.47 MPa and 250 ^{o}C enters a nozzle with a mass flow rate of 1 kg /s and expands to 0.395 MPa. Determine the throat section and the nozzle exit areas if:

a- the expansion happens without a loss.
b- the loss is 10% of the total thermal descent

Ans: a: 4.8 cm^2, 6.18 cm^2
b: 5.4 cm^2 , 6.6 cm^2

5.1.2- A steam at a flow rate of 4 kg/s expands in a nozzle from 0.8 MPa and 200 ^{o}C to 0.2 MPa, and the expansion from the beginning of the nozzle to the throat was without a loss and the loss was 10 % from the throat to the end of the nozzle. Calculate the throat section area and the area of the nozzle exit if the velocity of the steam at the entrance of the nozzle is 70 m/s.

Ans: 33.84 cm^2 , 50 cm^2

5.1.3- A steam turbine generates 1200 kW by passing a steam at 2 MPa, 300 ^{o}C and a flow rate of 8.4 kg / kWh. This vapor expands to 0.02 MPa where the throat diameter of each nozzle in the turbine is 1.2 cm and the loss between the throat and the nozzle end is 12% of the thermal descent. Determine the number of nozzles and the diameter of the nozzle exit.

Ans: 10 , 4.4 cm

5.1.4- A nozzle delivered steam at 0.7 MPa and 300 ^{o}C.

a- Determine the temperature and velocity of the steam at the throat region.
b- If the throat diameter is 6mm and the length between the throat and the nozzle exit is 50 mm. Determine also the divergence angle so that the pressure at the end of nozzle is 0.1 MPa, assuming the loss in the divergence part is 15 % of the thermal descent.

Ans: a: 225 ^{o}C , b: 547.7 m/s , 1.3^{o}

5.1.5- A vapor enters a nozzle at 0.7 MPa, 200 ^{o}C and with a velocity of 150 m/s, and exits at 0.07 MPa and a quality of 94%. Determine the exit steam velocity and also the area of the inlet and exit nozzle areas when the mass flow rate is 4.5 kg

Ans: 814 m/s , 89.4 cm^2 , 122.6cm^2

5.1.6- A vapor exits from a nozzle to the atmospheric pressure with a velocity of 490 m/s and a temperature of 150 °C. The nozzle efficiency and the entrance velocity are 0.93 and 150 m/s respectively. Calculate the pressure and the temperature of the steam at the entrance of the nozzle.

Ans: 0.175 MPa, 208 °C

5.1.7- A vapor at 1.7 MPa, 250 °C and with a velocity of 110 m/s enters a group of twelve (12) nozzles in a steam turbine. The pressure inside the turbine at the nozzles exits is 1 MPa and the nozzle efficiency is 0.9. Determine the exit area of each nozzle if the total flow rate is 280 kg/min and then calculate the percentage of increase in the steam mass flow rate.

Ans : 1.716 cm^2, 2.8%

5.1.8- A vapor at a flow rate of 3 kg/s enters a group of eight (8) nozzles at 1.5 MPa, 240 °C and a velocity of 480 m/s, and then expands to 0.07 MPa. If the efficiency of each nozzle is 0.95 and the cross-sectional area of each nozzle is rectangular with sides' length ratio is 1:2.5. Determine:

a- the steam exit velocity for each nozzle and the dimensions of the nozzle section at its entrance and its exit.
b- the steam exit velocity in case of the neglecting the steam entrance velocity.

Ans : 1011.16 m/s , 1.75cm , 4.38cm ,1.65cm , 4.14cm , 1007.96 m/s

5.1.9- A vapor enters a group of convergent-divergent nozzles at 3 MPa and 300 °C and expands to 0.5 MPa. If the exit velocity and the steam circulation rate are 800 m/s and 14 kg/s respectively, determine.

a- the nozzle efficiency.
b- the total cross-sectional area of the nozzles.
c-the velocity at the throat.

Ans: a-0.8
b-6120 mm^2
c-530 m/s

5.2- Steam Turbines

Steam turbine is a prime mover in which rotary motion is obtained by the gradual change of momentum of the steam. In the steam turbine rotary motion is imparted directly to the shaft by means of high velocity steam jets striking the blades fixed on the rim of a wheel which is fastened to the shaft. The turbine is much simpler in mechanical construction and it utilizes the kinetic energy of the steam instead of pressure only. The expansive property of the steam is almost utilized in the turbine either in the admission nozzles or in the turbine blading.

The turbine is an ideal prime mover and has got a variety of uses. In large sizes, it is used for driving electric generators and ship propellers. In smaller sizes it is adaptable for driving pumps, fans, compressors etc. In general the turbine is well adapted for work which requires high rotative and constant speeds even with widely fluctuating loads.

Definition: A turbine may he defined as, ***"the turbine is a prime mover in which a rotary motion is obtained by centrifugal force brought into action by changing the direction of a jet or a fluid escaping from a nozzle at high velocity."***

In a steam turbine, steam is passed through nozzle or fixed blades where the heat drop takes place, increasing the velocity of steam. This high velocity steam impinges on the curved vanes, which causes the direction of the steam to be changed. Due to this change of momentum motive force is exerted on the moving blades and power is obtained, Figures (5.5) and (5.6).

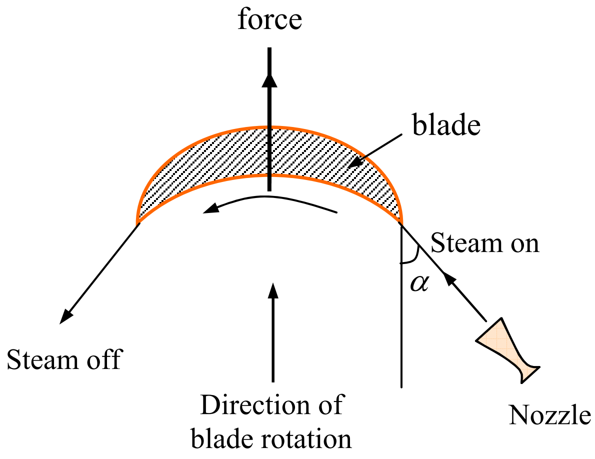


Figure 5.5

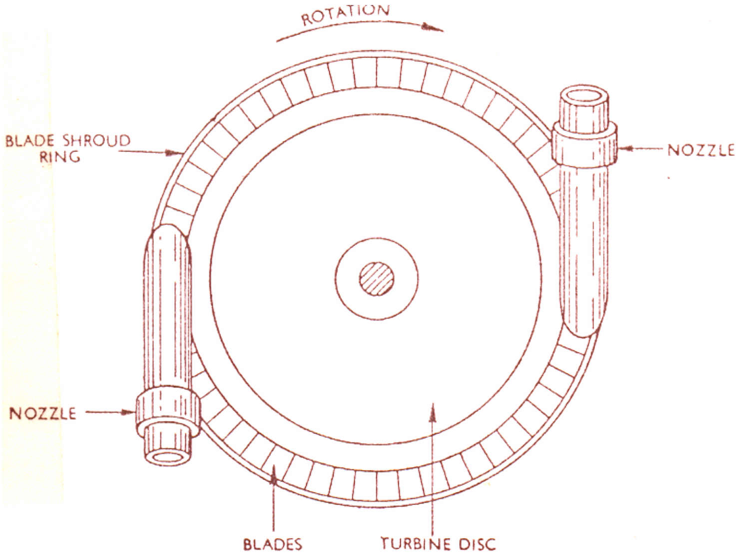


Figure (5.6 a)

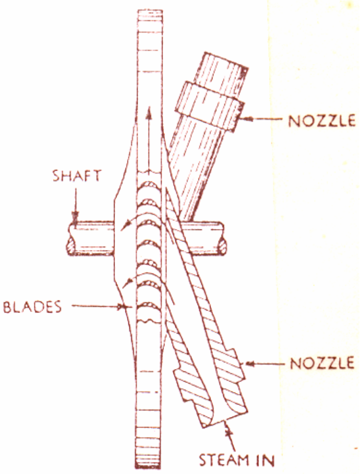


Figure (5.6 b)

5.3- Classification of Steam Turbines

There are several ways in which the steam turbines may be classified. The most important and common division being with respect to the action of steam as:

1. Impulse
2. Reaction
3. Combination of impulse and reaction

Other classifications are:

1. With respect to whether there is one or more revolving vane discs, separated by stationary reversing waves.
2. With respect to the number of step reductions, as:
   - (a) Single stage
   - (b) Multi stage
3. With respect to the direction of steam flow, as
   - (a) Axial
   - (b) Radial
   - (c) Tangential
   - (d) Mixed
4. With respect to the terminal pressures as:
   - (a) Condensing
   - (b) Non-condensing

5. With respect to the pressure of steam, as:
   (a) High pressure
   (b) Medium pressure
   (c) Low pressure
6. With respect to the source of the steam, as:
   (a) Extraction
   (b) Accumulator.

5.3.1- Impulse Turbine

An impulse turbine, as the name indicates is a turbine, which runs by the impulse of steam jet. In an impulse design of steam turbine, there is an attempt to attain no expansion of steam within the moving blades and as such its pressure remains constant while passing over the blades (Figure 5.7). The expansion of steam, however takes place in a set of stationary nozzle which lowers its pressure but impact a high velocity to it. The velocity attained in a properly designed nozzle depends upon the pressure drop through it.

Impulse turbine designs may be classified as single stage, multiple pressure stage, Multiple Velocity stage, and combination of the later two. The term pressure stage refers to a unit of nozzles combined with moving blades in which the energy of the steam resulting from each pressure drop is utilised. Velocity stage refers to the number of moving blades in which the energy of the steam in each pressure drop is absorbed.

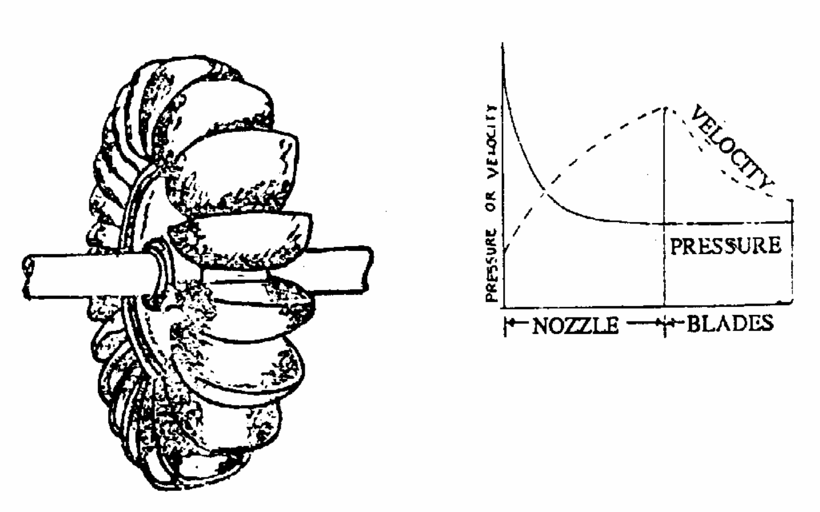


Figure 5.7

<u>De-Laval Impulse Turbine</u>

A De-Laval is the simplest type of impulse steam turbine. The De Laval turbine consists of a single rotor or disk to which are attached turbine blades. As a rule, the nozzles do not extend entirely around the periphery of impulse turbines: hence, at a particular instant, not all the blades are acted upon by steam.

A diagrammatic representation of a single impulse stage is seen in Figure 7 where we note that the entire pressure drop occurs in the stationary nozzle, the velocity and kinetic energy are a maximum at the exit section of the nozzle, velocity and kinetic energy decrease as energy leaves the fluid in the form of work during its flow across the blades. The element in which energy is converted into work are called blades or buckets. The surface of the blades is made very smooth minimise the frictional losses The blades are generally made of special steel alloys. In most of the cases the blades are bolted to runner disk. But sometimes the blades and disc are cast as a single unit.

The use of this type of turbine is not so common on account of its following disadvantages:

1. Since all the kinetic energy is to be absorbed in moving blades only, so velocity of the wheel is too, high (25000 to 30000 r p m.) for practical purposes.

2. The velocity of steam at exit is sufficiently high which means that there is a considerable loss of kinetic energy. In an actual turbine this loss is 10% to 12%.

The above mentioned disadvantages can be overcome by arranging the expansion of steam or the utilisation of kinetic energy or both in several rings. This is known as compounding. The various method of compounding are explained as follows:

(a) Pressure Compounded Turbine

In this case as shown in Figure 5.8 rings of fixed nozzles are interposed between the rings moving blades which are keyed to the shaft in series. The partition between the stages is by means of a diaphragm fixed on the casing and having fine clearance with the shaft. The ring of fixed nozzles is made in the diaphragm itself. The steam from the boiler enters the first ring of nozzles where its pressure is partially reduced and the velocity is increased. It then passes over the first moving blade ring where nearly all of its velocity is absorbed. The exit angle of the nozzle is so made that the steam enters the moving blades without shock. The exhaust from the first moving blade ring enters the next nozzle ring and is again partially expanded and its velocity is again increased which is absorbed in the second ring of moving blades and so on.

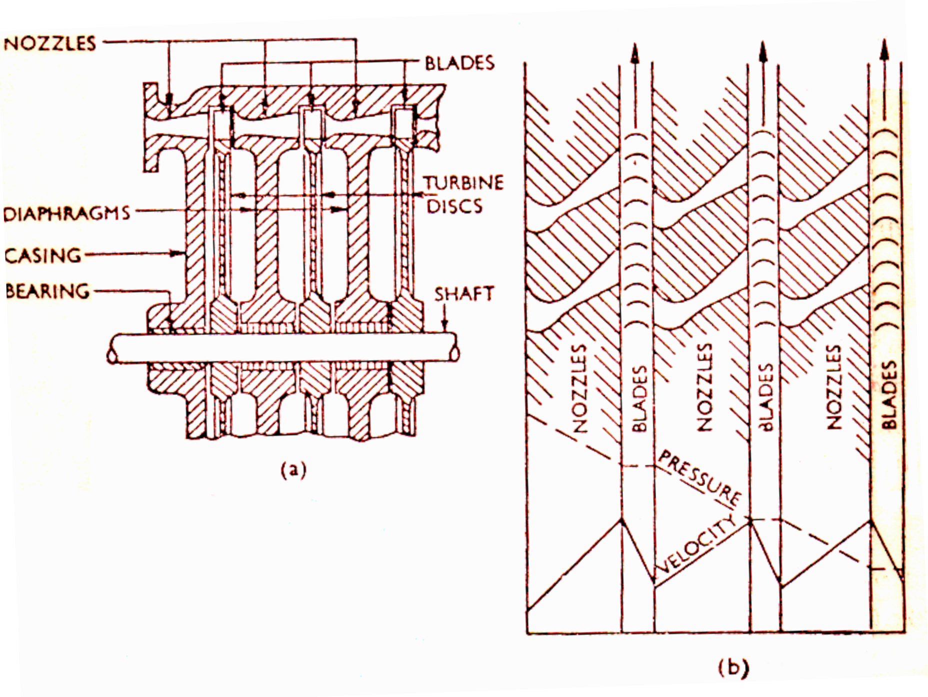


Figure 5.8

Since only part of the pressure drop occurs in each stage, the steam velocities will not be very high and hence the turbine velocity will be restricted. It may be noted that all pressure drop occurs in the nozzle the pressure remaining constant over each ring of moving blades.

This is the most efficient type of turbine, because the ratio of blade velocity to steam velocity remains constant, but it has disadvantage of large number of stages and hence it is most expensive. Therefore pressure-compounded turbine has become out dated.

This method of pressure-compounding is used in the Rateau and Zoelly turbine.

(b) Velocity compounding

There are a number of moving blades, separated by rings of fixed blades keyed in series on a common shaft, (Figure 5.9). The steam from the boiler is passed through nozzles where in expanding to, condenser pressure, it acquires very high velocity as shown in the diagram. The high velocity is then passed through a number of moving and fixed blade rings. A fall in velocity occurs every time, when the steam passes over a ring of moving blades. The steam thus leaves the turbine with a low velocity. It may however be noted that the function of the fixed blades is to change the direction of jet without appreciably altering the velocity. Since there is no fall in pressure as the steam flows over the moving blades. Only part of the velocity of steam is used in each ring of blades, the speed of the turbine will be restricted. It may be noted that all pressure drop occurs in the nozzles, the pressure remaining constant over each ring of moving and fixed blades.

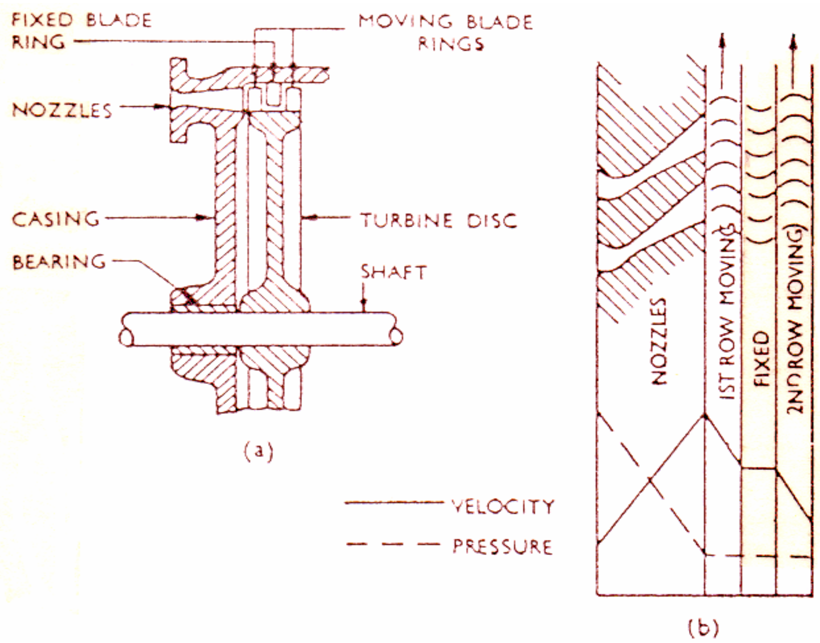


Figure 5.9. Pressure compounder – diagrammatic arrangement of impulse turbine

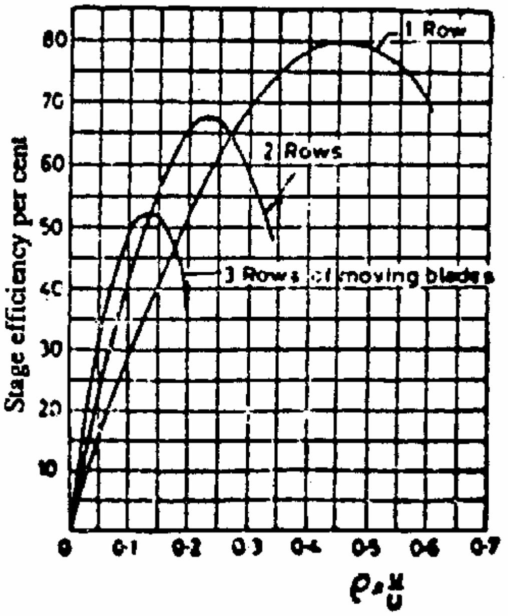


Figure 5.10. Efficiency or velocity compounded turbine

The advantages of velocity-compounded turbine is relatively fewer number of stages and hence less initial cost. However, the disadvantage is low efficiency as the ratio of blade velocity to steam velocity is not optimum for all the wheels. Therefore, efficiency goes on decreasing for subsequent stages (Figure 5.10). Hence, large velocity compounded turbines are obsolete. This method of velocity compounding is used in the Curtis turbine.

(c) Pressure-Velocity compounding

Pressure-velocity compounding is a combination of the above two methods as shown in Figure 5.11. The total drop in steam pressure is carried out in two stages and the velocity obtained in each stage is also compounded. Turbine employing this method of compounding may be said to combine many of the advantages of both pressure and velocity staging. The total pressure drop of the steam is divided into stages and the velocity obtained in each stage is also compounded. It results in bigger pressure drop in each stage and hence less number of stages which means a more compact turbine than a pressure compounded turbine. However the efficiency is lower than a pure pressure compounded turbine. Figure 5.11 shows the diameter of the pressure velocity compounded turbine is increased at each stage to allow for the increasing volume of the steam at low pressures. Each stage has a ring of nozzle followed by moving, and fixed blades.

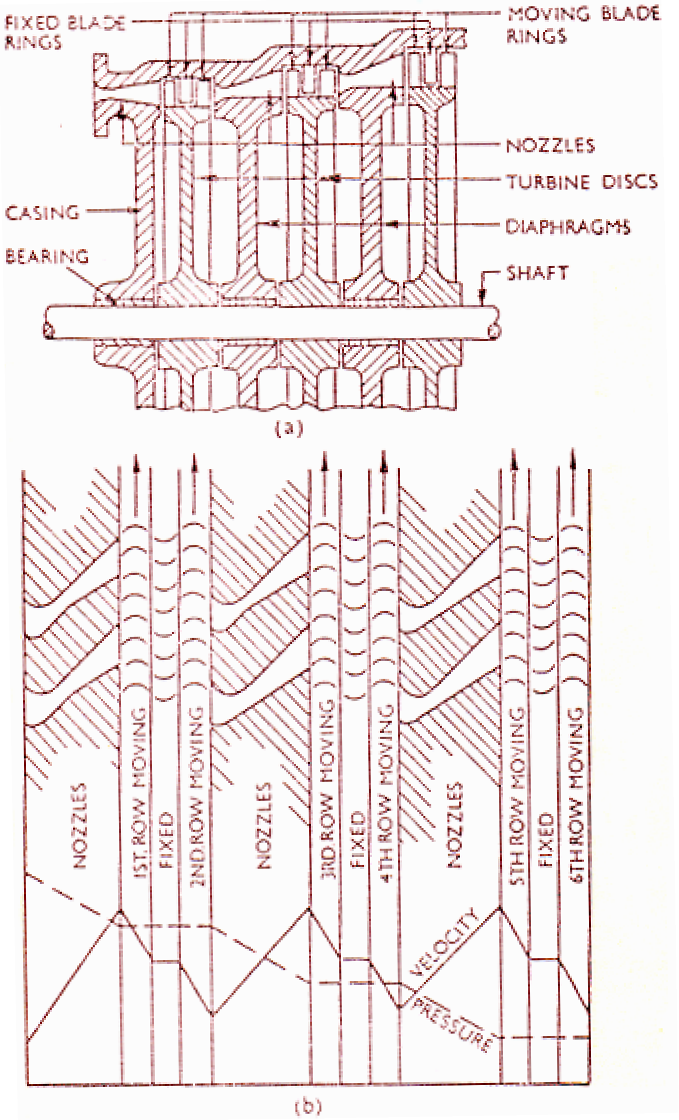


Figure 5.11

The pressure-velocity-compounded turbine is an impulse turbine because the pressure is constant during each stage. This method of pressure-velocity compounding is used in the Moore and Curtis turbines.

Internal Losses in Turbines

There are several factors, which effect the performance of a steam turbine. All these factors which reduce the output of the turbine are known as internal losses. Though there are many internal losses in a steam turbine, yet the following are important from the subject point of view:

(a) Nozzle loss

It is an important loss in impulse turbine, which occurs when the steam flows through the nozzle. This loss takes place due to friction in the nozzle.

(b) Blade friction loss

This loss takes place due to friction of the surface of blades. As a result of the blade friction, the relative velocity of the steam is reduced while gliding over the blade.

(c) Wheel friction loss

This loss takes place due to the resistance offered by the steam to the moving turbine wheel or disc. As a result of this loss, the turbine wheel rotates at a lower speed.

(d) Mechanical friction loss

This loss takes place due to friction between the shaft and wheel bearings as well as regulating the valves. With the help of lubricating this loss can be reduced.

(e) Leakage loss

Which occurs due to leakage of the steam at each stage of the turbine, blade tips and glands.

(f) Governing loss

Which occurs due to throttling of the steam at the main stop valve of the governor.

5.3.2- Reaction Turbine

In case of reaction turbine, the steam expands as it flows over the blades which, therefore act as nozzles. It consists of large number of stages, each consisting of fixed and moving blades. The heat drop takes place throughout in both fixed and moving blades. Unlike the impulse turbine no nozzles as such are provided in a reaction turbine. The fixed blades act both as nozzles in which the velocity of the steam is increased and as the means of directing the steam so that it enters the ring of moving blades without shock. The turbine derives its name of reaction because the steam expands over the moving blades also giving a reaction to the moving blades. This reaction force is not there in impulse turbines. The steam velocity in the reaction turbine is not very high due to division of pressure drop over fixed and moving blades. Because the pressure drop takes place both in the fixed and moving blades, all blades are nozzle shaped.

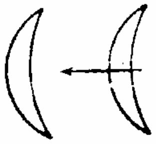


Figure 5.12 (a)

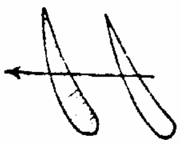


Figure 5.12 (b) turbine blades

The blade arrangements for impulse and impulse reaction turbines are shown in Figure 5.12 (a). The variation of pressure and velocity of steam passing through the turbine is also shown in Figure 5.12 (b).

Impulse Vs. Impulse Reaction Turbine

Following are the few points of comparison between an impulse turbine and a reaction turbine:

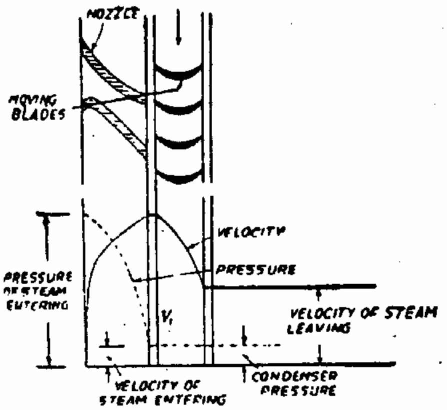
1. In impulse turbine the steam completely expands in the nozzle and its pressure remains constant during its flow through the blade passages. In impulse reaction turbine, steam expands partially in the nozzle and further expansion takes place in the rotor blades.

2. The impulse turbine blades have symmetrical profile where the reaction turbine blades have aerofoil section as shown in Figure 5.12. The area of flow changes along the blade passage like that in a nozzle.
3. The pressure on both ends of the moving blade of an impulse turbine is same whereas different pressure exists on two ends of the moving blade of a reaction turbine.

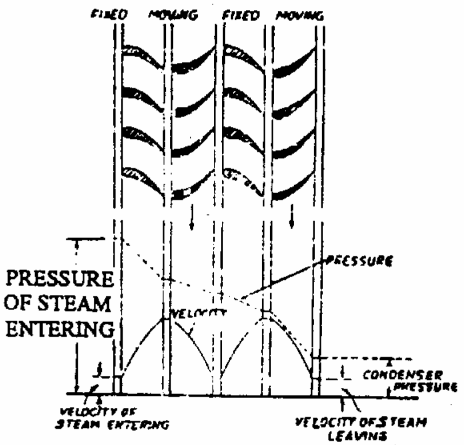
4. The blade efficiency curve for the reaction turbine is more flat compared with the efficiency curve of impulse turbine. This is an added advantage of reaction turbine, leading to part load economy.

5. The steam velocity in a reaction turbine is not very high and hence the speed of the turbine is relatively low.

6. The number of stages required for reaction turbine are more compared with impulse turbine for the same power developed as the Pressure drop in each stage is small.



(a) An impulse Turbine



(b) A reaction Turbine.

Figure 5.13

This section describes the way in which work is transferred in a steam turbine. In the turbine the working fluid enters at a high pressure and acquires increased kinetic energy as its expands to a lower pressure in a ring of fixed nozzles. The stream of the fluid then undergoes a change of momentum as it flows through passages between blades attached to the turbine disc (rotor wheel), and the component in a direction tangential to the circle of rotation produces the output torque at the shaft.

5.4- Velocity Diagram

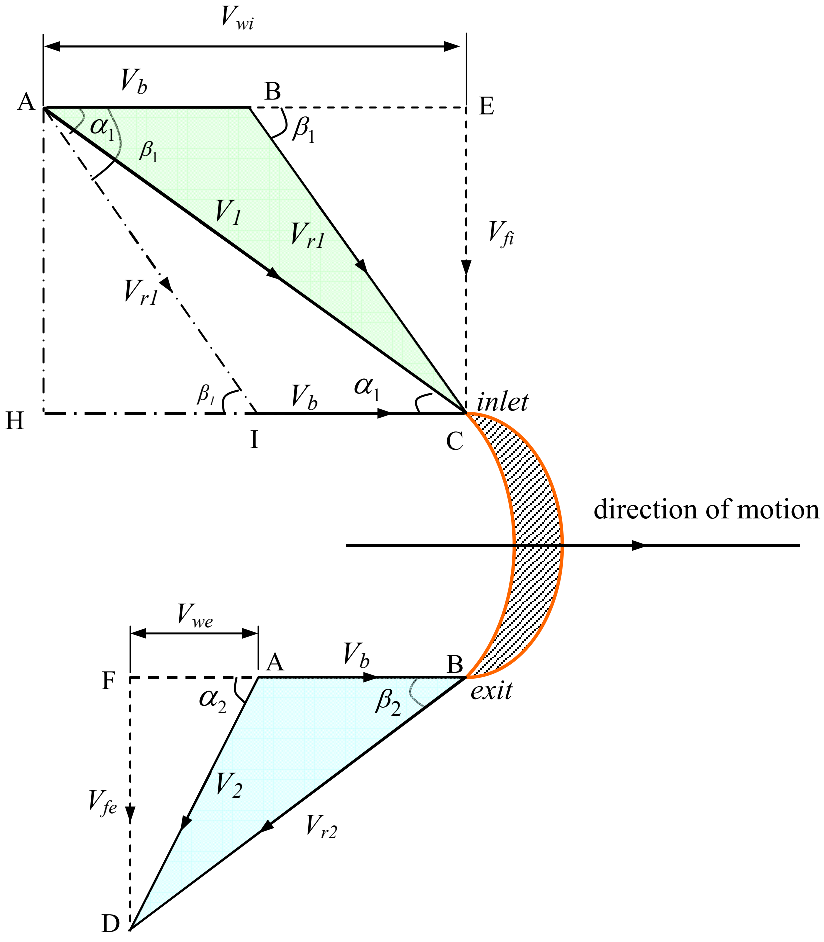


Figure 5.14

α : Angle of absolute velocity measured to the direction of blade motion.

α_1: Angle of absolute velocity measured to the direction of blade motion at inlet "nozzle angle"

α_2: Angle of absolute velocity measured to the direction of blade motion at exit.

β: Angle of relative velocity measured to the direction of blade motion.

β_1: Angle of relative velocity measured to the direction of blade motion at the entrance of moving blade "blade inlet angle".

β_2: Angle of relative velocity measured to the direction of blade motion at the outlet of moving blade "blade outlet angle".

V_b : linear velocity of the blade "mean blade speed"= $\frac{\pi DN}{60}$, $N\ (rpm)$

V_1: absolute velocity at inlet to moving blade = $V_{\text{leaving nozzle}}$

V_2: absolute velocity at outlet from moving blade.

V_{r1}: relative velocity at inlet to moving blade .

V_{r2}: relative velocity at outlet from moving blade.

V_{fi}: velocity of flow at inlet = $V_1 \sin\alpha_1$ (see Figure (5-14)).

V_{fe}: velocity of flow at exit = $V_2 \sin\alpha_2$ (see Figure (5-14)).

V_{wi}: velocity of whirl at inlet = $V_1 \cos\alpha_1$ (see Figure (5-14)).

V_{we}: velocity of whirl at exit = $V_2 \cos\alpha_2$ (see Figure (5-14)).

$\dot{m}$: mass flow rate (kg/s).

K : blade velocity coefficient = $\dfrac{V_{r2}}{V_{r1}} < 1$

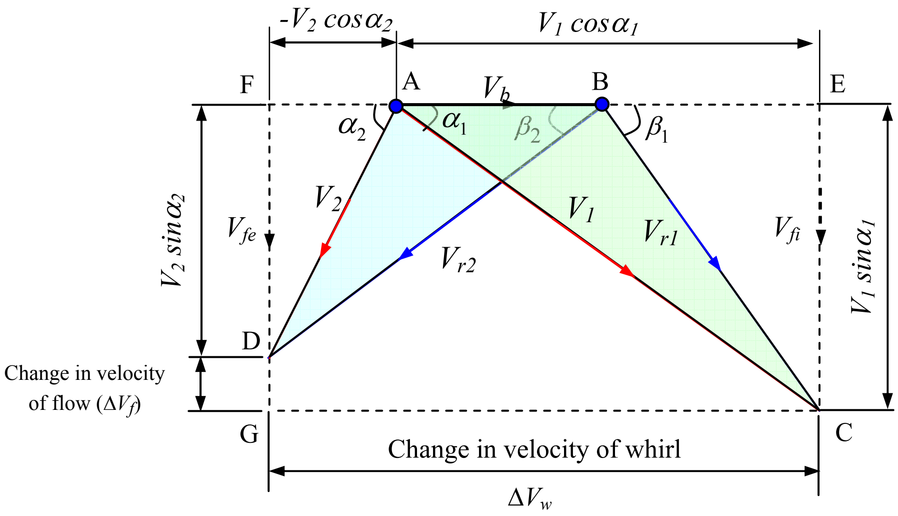


Figure (5.15)

Notes

The absolute velocities started from point A where as the relative velocities started from point B.

In case of no friction, the $V_{r2} = V_{r1}$ and the difference equals the loss caused by friction

5.4.1. Work done on the blades

The force that exerted by the steam on the blade is proportional to the change in the momentum.

Force = Rate of change of momentum (5.14)

= mass x change of velocity (5.15)

Change in velocity of whirl:

$$= -\left(V_2 \cos\alpha_2 + V_1 \cos\alpha_1\right) \tag{5.16}$$

Then the force is,

$$\text{Force} = -\,\dot m\left(V_2 \cos\alpha_2 + V_1 \cos\alpha_1\right) \tag{5.17}$$

The negative sign denotes the force acts in the opposite direction of the rotation. Using Newton's Law

Driving force on the rotor, *F* = Force

$$F = \dot m\left(V_2 \cos\alpha_2 + V_1 \cos\alpha_1\right) \tag{5.18}$$

$$= \dot m\left(V_{r2} \cos\beta_2 + Vr_1 \cos\beta_1\right) \tag{5.19}$$

The work done on the blades is equal to

$$W = Force \times Distance \tag{5.20}$$

Hence (refer to Figure 5.15)

$$\text{Power input } = \dot W = \frac{W}{s} = \dot m V_b\left(\Delta V_w\right)\left(\frac{kg}{s}\times\frac{m}{s}\right)\times\left(\frac{m}{s}\right) = \frac{N\times m}{s} = \frac{J}{s} = (Watt) \tag{5.21}$$

Rewrite equation (5.19)

$$\dot W = \dot m\, V_b\left(V_{r2} \cos\beta_2 + V_{r1} \cos\beta_1\right) \tag{5.22}$$

$$= \dot m\, V_b\, V_{r1} \cos\beta_1\left(\frac{V_{r2} \cos\beta_2}{V_{r1} \cos\beta_1} + 1\right) \tag{5.23}$$

$$= \dot m\, V_b\, V_{r1} \cos\beta_1\left(K\,C + 1\right) \tag{5.24}$$

where

$$C = \frac{\cos\beta_2}{\cos\beta_1}$$

The supplied steam kinetic energy is

$$KE_s = \frac{V_1^2}{2}\left(\frac{W}{kg\ \ steam}\right) \tag{5.25}$$

Then the blade or diagram efficiency can be expressed as

$$\eta_{blade\ \ or\ diagram} = \frac{work\ done\ by\ blade/kg\ steam}{Energy\ Supplied/kg\ \ steam} = \frac{2V_b\,\Delta V_\omega}{V_1^2} \tag{5.26}$$

5.4.2. Axial Thrust

The axial thrust, i.e. thrust produced by the difference in the axial components of the velocities (along the axis of the shaft), hence

Force= mass x change in velocity

Axial thrust $= \dot{m}\left(V_{fe} - V_{fi}\right) \; (N)$ (5.27 a)

$$\dot{m}\left(V_2 \sin\alpha_2 - V_1 \sin\alpha_1\right) = \dot{m}\,\Delta V_f \qquad (5.27\ b)$$

The magnitudes of V_{fe} and V_{fi} can be deduced from the velocity diagram. The axial thrust direction will be in the direction of flow if the valve of V_{fe} is smaller than the value of V_{fi} and vis-versa.

5.4.3. Stage efficiency

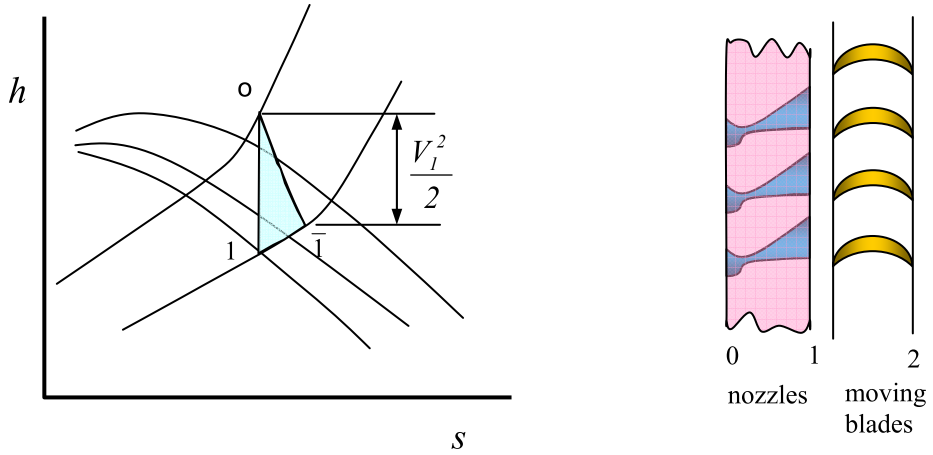


Figure (5.16)

$$\eta_{nozzle} = \frac{h_o - h_{\bar{1}}}{h_o - h_1} \qquad (5.28)$$

From Figure (5.16)

$$\eta_{nozzle} = \frac{V_1^2/2}{h_o - h_1} \qquad (5.29)$$

$$\eta_{stage} = \frac{\text{work done in stage}}{\text{enthalpy drop in stage}} = \frac{\dot{m}\,V_b\,\Delta V_w}{\dot{m}\left(h_o - h_1\right)} = \frac{V_b\left(V_{wi} + V_{we}\right)}{h_o - h_i} \qquad (5.30)$$

From equations (5.26) and (5.29), we have

$$\eta_{stage} = \eta_{nozzle} \times \eta_{diagram} \tag{5.31}$$

If the mechanical losses is considered , then equation (5.31) can be written as,

$$\eta_{net} = \eta_{nozzle} \times \eta_{diagram} \times \eta_{mech} \tag{5.32}$$

$$= \frac{\text{net work delivered to shaft}}{\text{total energy supplied}}$$

Example 5.3

Steam at a velocity of 500 m/s enters an impulse wheel having a nozzle angle (α_1) of 25°. The exit angle of the moving blade (β_1) is 20°. The relative velocity of the steam may assumed constant ($V_{r1}=V_{r2}$) over the moving blades. If the blade speed is 200 m/s, find the moving blade angle at inlet (β_1), the exit velocity and direction of steam (V_2) and the work done/kg of steam. If the turbine is supplied with 2 kg/s of steam, find the axial thrust.

$V_1 = 500$ m/s , $\alpha_1 = 25\,^{o}$, $\beta_2 = 20\,^{o}$, $V_{r1} = V_{r2}$

$V_b = 200$ m/s , $\dot{m} = 2$ kg/s

Solution:

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5.5- Velocity diagram for velocity compounded impulse turbine

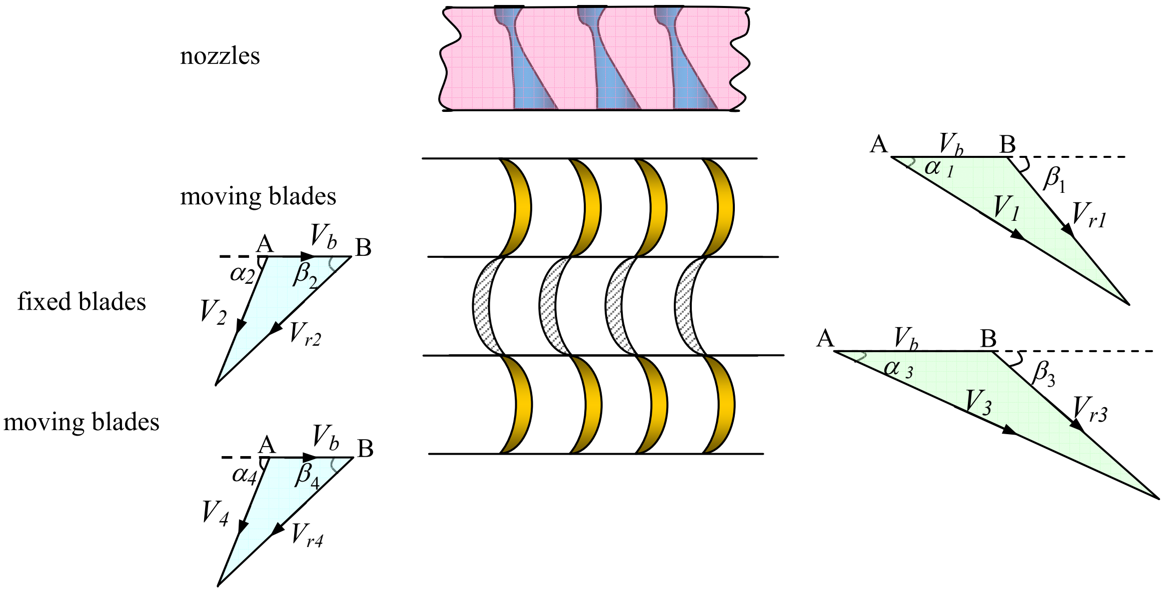

Figure (5.17)

The work done on the *first row of moving blades* is equal to

$$\dot{W}_{\mathbf{1}} = \dot{m} \times V_b \times change\ of\ velocity\ of\ whirl\ of\ first\ row\ of\ moving\ blades$$

$$= \dot{m} V_b \, \Delta V_{w\mathbf{1}} \qquad (5.33)$$

and he work done on the *second row of moving blades* is equal to

$$\dot{W}_{\mathbf{2}} = \dot{m} V_b \, \Delta V_{w\mathbf{2}} \qquad (5.34)$$

The total work

$$\dot{W}_t = \dot{W}_1 + \dot{W}_2 \qquad (5.35)$$

$$\dot{W}_t = \dot{m} V_b \left(\Delta V_{w_{\mathbf{1}}} + \Delta V_{w\mathbf{2}} \right) \qquad (5.36)$$

and

$$\eta_{diagram} = \frac{2 V_b \left(\Delta V_{w1} + \Delta V_{w2} \right)}{V_1^2} \qquad (5.37)$$

The axial thrust,

$$\text{Axial thrust} = \dot{m} \left(\Delta V_{f1} + \Delta V_{f2} \right) \qquad (5.38)$$

where

$\Delta V_{f1} = \left(V_{fe1} - V_{fi1} \right)$ "change in velocity of flow of row 1"

$\Delta V_{f2} = \left(V_{fe2} - V_{fi2}\right)$ "change in velocity of flow of row 2"

Blades velocity coefficient

$$K_{moving\ blade_1} = \frac{V_{r2}}{V_{r1}}$$

$$K_{fixed\ blade_1} = \frac{V_3}{V_2}$$

$$K_{moving\ blade_2} = \frac{V_{r4}}{V_{r3}}$$

Example 5.4

For a two-velocity stage impulse turbine, outlet angle of fixed blade 24°, outlet angles of 1st row and 2nd row of moving blades =22° and 35° respectively. Mean blades velocity =150 m/s, steam velocity from nozzles =800m/s. Coefficient of the blades velocity = 0.75. If the flow is 3kg/s steam. Find the work done in the turbine if nozzle angle is 20° .

α_3 =24° , β_2=22° , β_4=35° , V_b=150 m/s , V_1=800m/s

K=0.75 , $\dot{m}$ =3 kg/s , α_1=20° , W_t=?

Solution:

5.6- Velocity diagram for reaction turbine stage

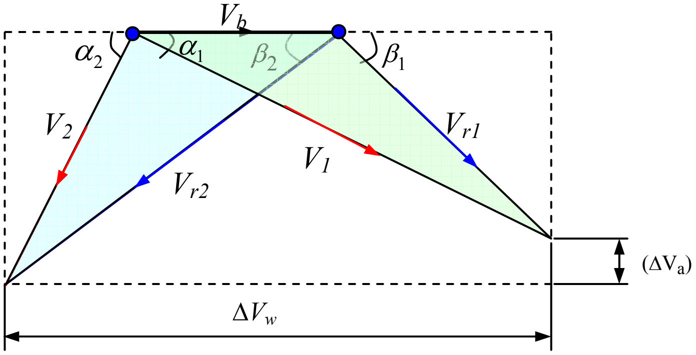

Figure (5.18)

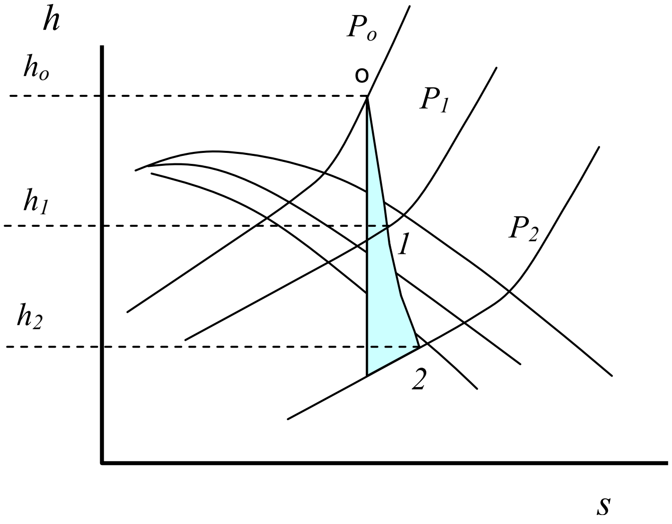

Figure (5.19)

Refer to figure (5.19)

Degree of Reaction $= \dfrac{Heat\ drop\ in\ rotor\ blades}{Total\ heat\ drop\ in\ the\ stage} = \dfrac{h_1 - h_2}{h_o - h_2}$ (5.39)

5.7- Blade height

a- Blade height for impulse turbine

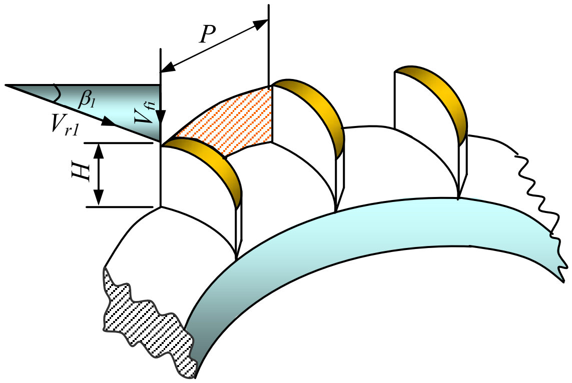

Figure (5.20)

$$\dot{m}\,v = A\,V_{fi} \tag{5.40}$$

$$A = N_b \times H \times P$$

where

A: the area through which the steam is passing, m^2

N_b: number of blades.

H: Blade height, m

P: Pitch of blades, m

β_1: Blade inlet angle

V_{fi}: Velocity of flow at inlet, m/s

v: specific volume of steam, m^3/kg

Then

$$\dot{m}\,v = N_b \times H \times P \times V_{fi}$$

$$H = \frac{\dot{m}\,v}{N_b \times P \times V_{fi}} \tag{5.41}$$

$N_b \times P$ is equal to the circumference at the mean blade diameter = πD

D: mean blade diameter, m

b-Blade height for reaction turbine

$$H = \frac{\dot{m}\,v}{\pi \times D \times V_{fi}} \tag{5.42}$$

Example 5.5

A two velocity stage impulse steam turbine receives steam from the nozzle at an angle of 20° and a velocity of 500 m/s. The mean blades speed is 140 m/s. Outlet angle of 1st moving blades =25°, of the fixed blades = 30°, and of 2nd row moving blades = 45°. Assume a linear decrease of axial velocity component equals axial width of blades and a height ratio of 2[*] from inlet to outlet. Draw the combined velocity diagram and find the work done and the axial thrust /kg of steam /s

Solution:

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[*] *N* x *P and v are constant.*

$$V_{fi1}H_1 = V_{fe2}H_2 \rightarrow \frac{V_{fi1}}{V_{fe2}} = \frac{H_2}{H_1} = 2$$ [1,2 denote the first row and second row respectively]

Example 5.6

An impulse turbine is supplied with steam at a pressure of 1.5 MPa and temperature of 250 °C. The back pressure is 0.012 MPa. Nozzle efficiency is 0.9. Blade velocity coefficient (*K*) =0.8, mechanical efficiency =0.9, nozzle angle =20°. Symmetrical blades with angle = 30°. Find the revolution per minute if the mean diameter of wheel is 75 cm, the blade efficiency, axial thrust /kg steam, exit loss and the stage efficiency.

Solution:

Revision Questions

5.2.1. At a particular stage of a reaction steam turbine, the steam blade speed is 60 m/s and the steam is at a pressure of 350 kPa with a temperature of 175^{o}C. Fixed and moving blades at this stage have inlet angles of 30^{o} and exit angles of 20$^{o.}$ Determine

a- the blade height at this stage of the blade height is 1/10 the mean blade ring diameter and the steam flow is 13.5 kg/s.

b- the power developed by a pair of fixed and moving blade rings at this stage .

c- the specific enthalpy drop in kJ/kg at the stage if the stage efficiency is 85%.

Ans: a-65 mm, 218.7 kW, 19.058 kJ/kg

5.2.2. A nozzle in an impulse turbine delivers 1 kg/s steam to a set of blades moving at 200 m/s . The nozzles are inclined at an angle of 16^{o} to the plane of the wheel. The blade velocity coefficient is 0.72, calculate the blade efficiency, work done and the energy lost in the blades/s[♣].Take the blade angles at inlet and outlet =25^{o}.

Ans: 76%,105 kW, 27.072 kW

5.2.3. The outlet angles of the nozzle and blades of a two row impulse turbine are in succession 16^{o}, 18^{o}, 20^{o} and 30^{o} and the corresponding nozzle and blade velocity coefficients are 0.92, 0.72, 0.78 and 0.84. The flow through the nozzle is at the rate of 4 kg/s, and the supply is at 1.2 MPa and 50 ^{o}C superheat. The chamber pressure is 0.5 MPa and the blade speed is 1/5 that of the jet speed leaving the nozzle. Determine the work done and the blade efficiency.

Ans: 391.67 kW, 61.13 %

5.2.4. An impulse turbine with blade speed =150 m/s, velocity of steam reaching nozzle = 90 m/s, nozzle efficiency =0.85, absolute velocity of steam at blade exit = 85 m/s at an angle of 80^{o}, blade velocity coefficient =0.82, mass flow rate =2 kg/s. If the blade is equiangular, find the blade angles, the absolute velocity of steam at entrance, the axial thrust and the heat drop in the nozzle.

Ans: 27^{o}, 27^{o}, 37^{o} m/s, 64.4 kJ/kg

5.2.5. An impulse turbine has a number of pressure stages (pressure compounded turbine). The nozzle angle in the first stage is 20^{o} and the blade exit angle =30^{o}. The blade speed =120 m/s and the velocity of steam leaving nozzle =300 m/s. If the blade velocity coefficient =0.8 and the nozzle efficiency =0.85, find the work done per kg steam and the stage efficiency.

If the steam supplied to the first stage is at 2 MPa and 260 ^{o}C and the condenser pressure =0.007 MPa estimate the number of stage required.

Ans: 36 kJ/kg, 68 %, $\cong 4$

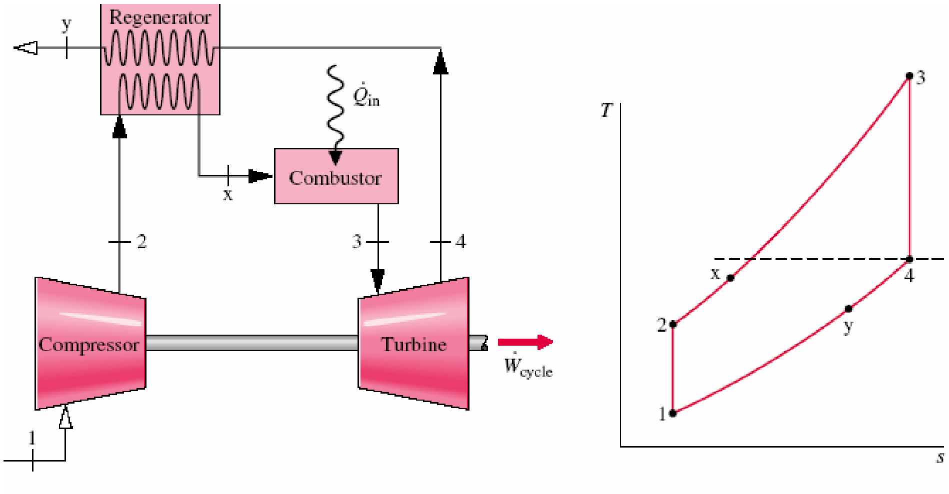
[♣] Energy lost in the blades/s = $\dot{m}\left(\frac{V_{r1}^2}{2}-\frac{V_{r2}^2}{2}\right)$

YANBU INDUSTRIAL COLLEGE

Power Plant Engineering and economy

MET401

Department of Mechanical Engineering Technology



Chapter VI- Gas Turbine Power Plants

2010

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Chapter VI

Gas Turbine Power Plants

6.1- Introduction

Gas turbine power plant is quite attractive due to its low capital cost and high reliability and flexibility. Gas turbine power plants may operate on either an open or closed basis. The open mode shown in Fig. 6.1a is more common. This is an engine in which atmospheric air is continuously drawn into the compressor, where it is compressed to a high pressure. The air then enters a combustion chamber, or combustor, where it is mixed with fuel and combustion occurs, resulting in combustion products at an elevated temperature. The combustion products expand through the turbine and are subsequently discharged to the surroundings. Part of the turbine work developed is used to drive the compressor; the remainder is available to generate electricity, to propel a vehicle, or for other purposes. In the system shown in Fig. 6.1b, the working fluid receives an energy input by heat transfer from an external source, for example a gas-cooled nuclear reactor. The gas exiting the turbine is passed through a heat exchanger, where it is cooled prior to reentering the compressor.

An idealization often used in the study of open gas turbine power plants is that of an air-standard analysis. In an air-standard analysis two assumptions are always made:

1. The working fluid is air, which behaves as an ideal gas.
2. The temperature rise that would be brought about by combustion is accomplished by a heat transfer from an external source.

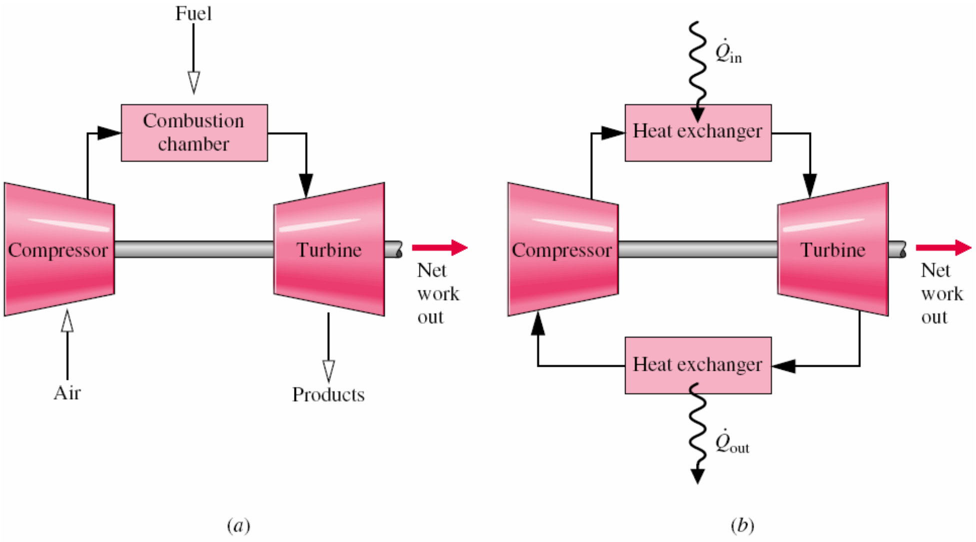
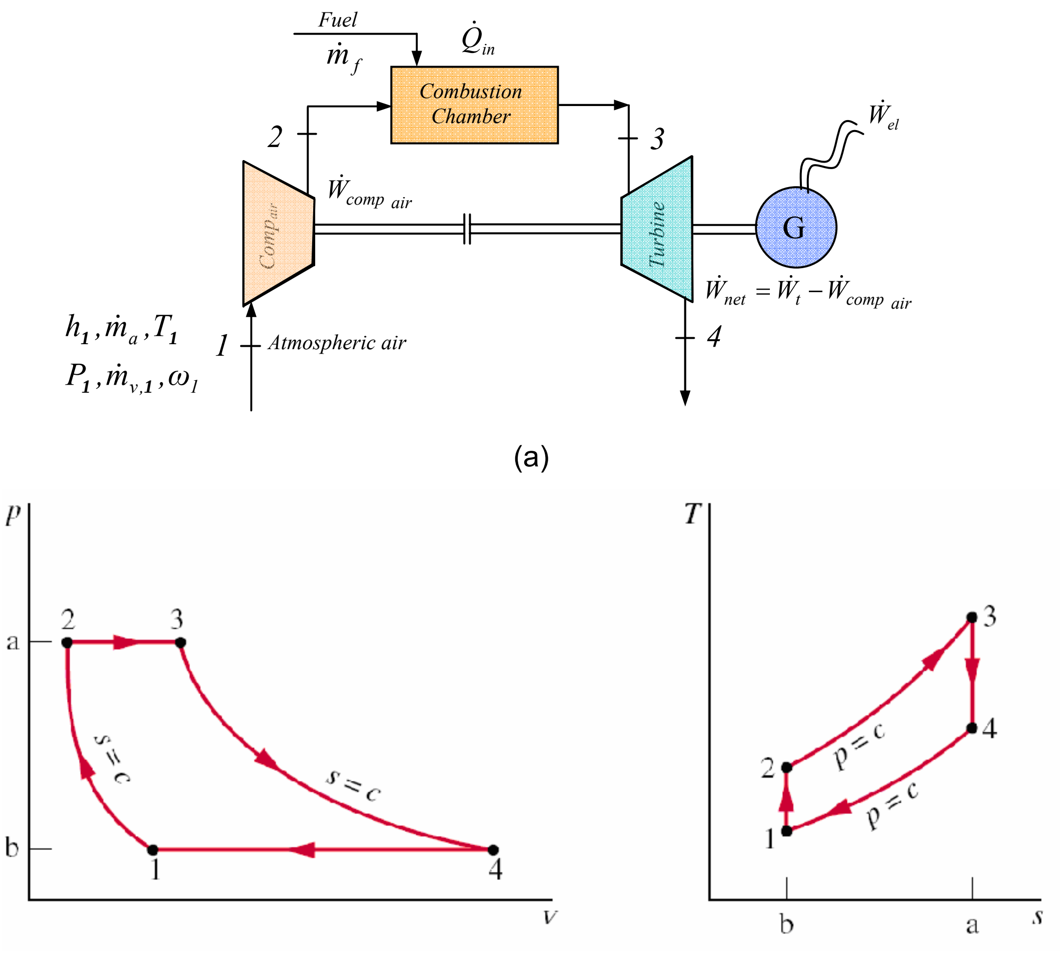


Figure 6.1 Simple gas turbine (a) open to the atmosphere (b) closed

6.2- Air-Standard Brayton Cycle

A schematic diagram of an air-standard gas turbine is shown in Figure 6.2. The directions of the principal energy transfers are indicated on this figure by arrows. In accordance with the assumptions of an air-standard analysis, the temperature rise that would be achieved in the combustion process is brought about by a heat transfer to the working fluid from an external source and the working fluid is considered to be air as an ideal gas. With the air standard idealizations, air would be drawn into the compressor at state 1 from the surroundings and later returned to the surroundings at state 4 with a temperature greater than the ambient temperature. After interacting with the surroundings, each unit mass of discharged air would eventually return to the same state as the air entering the compressor, so we may think of the air passing through the components of the gas turbine as undergoing a thermodynamic cycle. A simplified representation of the states visited by the air in such a cycle can be devised by regarding the turbine exhaust air as restored to the compressor inlet state by passing through a heat exchanger where heat rejection to the surroundings occurs. The cycle that results with this further idealization is called the air-standard Brayton cycle.



(a)

(b) (c)

Figure 6.2 Air standard gas turbine cycle.

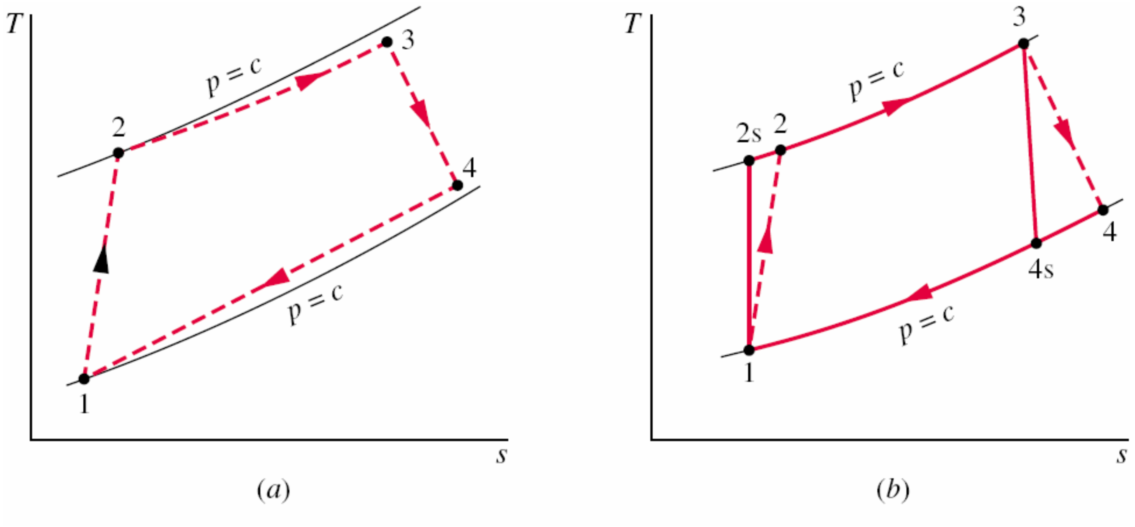
6.3- Analysis of the Actual Gas Turbine Plant

Consider an irreversible gas turbine cycle as shown in Figure 6.3, processes 1-2 and 3-4 are irreversible and processes 2-3 and 4-1 are isobaric heat addition and rejection respectively. Processes 1-2s and 3-4s are isentropic presenting the process in an ideal cycle.

For the isentropic processes 1-2 and 3-4, we have

$$\frac{T_{2s}}{T_1} = \frac{T_3}{T_{4s}} = \left[\frac{P_2}{P_1}\right]^{\frac{k-1}{k}} = PR^{\frac{k-1}{k}} \qquad (6.1)$$

where PR is the pressure ratio and k is the specific heats ratio.



(a) (b)

Figure 6.3 T-s diagram of an open type gas turbine cycle

6.3.1- Turbine

The compressor extracts its power from the turbine output as shown in Figure 6.2 a, the thermal efficiency of the cycle is

$$\eta_{cy} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{\dot{W}_t - \dot{W}_{comp_{air}}}{\dot{Q}_{in}} \qquad (6.2)$$

Applying the first law of thermodynamics to the gas turbine (neglect the potential and kinetic energy terms), the power produced by the turbine is

$$\dot{W}_t = \dot{m}_t\, c_{pg}\, \eta_t \left(T_3 - T_{4s}\right) \qquad (6.3)$$

where $\eta_t = \dfrac{T_3 - T_4}{T_3 - T_{4s}}$

and $\dot{m}_t$ is the total gases mass flow rate at the turbine inlet given as

$$\dot{m}_t = \dot{m}_a + \dot{m}_v + \dot{m}_f = \dot{m}_a\left(1 + \omega_1 + FA\right) \qquad (6.4)$$

and ω_1 is the humidity ratio at state 1, the fuel air ratio $AF = \dot{m}_f / \dot{m}_a$, $\dot{m}_v$ mass of water vapor at state 1.

Substituting for T_{4s} and $\dot{m}_t$ from Eqs.6.1 and 6.4 respectively in Eq. 6.3 yields

$$\dot{W}_t = \dot{m}_a\big(1+\omega_1 + AF\big)\big)c_{pg}\,\eta_t\,T_3\left(1-\frac{1}{PR^{\frac{k-1}{k}}}\right) \tag{6.5}$$

The turbine isentropic efficiency can be estimated using the practical relations recommended by Korakianitis and Wilson [ ] as:

$$\eta_t = 1-\left(0.03+\frac{PR-1}{180}\right) \tag{6.6}$$

The gas specific heat (c_{pg}) is evaluated as

$$\begin{aligned} c_{pg} &= 1.0887572\times10^3 - 1.4158834\times10^{-1}\,T + 1.9160159\times10^{-3}\,T^2 \\ &\quad - 1.2400934\times10^{-6}\,T^3 + 3.0669459\times10^{-10}\,T^4 - 2.6117109\times10^{-14}\,T^5 \end{aligned}$$

The gas turbine is almost constant volume machine at a specific rotating speed, and then the inlet air volumetric flow rate, $\dot{V}_a$ is fixed regardless of the ambient air conditions. As the air temperature rises in hot summer days, its density falls but the volumetric flow rate remains constant. Therefore, the mass flow rate reduces and consequently the power output decreases. Equation 6.5 can be written in terms of the volumetric flow rate at the compressor inlet state as:

$$\dot{W}_t = \dot{V}_a\rho_a\big(1+\omega_1 + FA\big)c_{pg}\,\eta_t\,T_3\left(1-\frac{1}{PR^{\frac{k-1}{k}}}\right) \tag{6.7}$$

where ρ_a is the moist air density which is a function of the temperature T_1 and the humidity ratio ω_1 and can be calculated using the ***Engineering Equation Solver*** (***EES***) software. The effect of the air pressure drop across chilling coils is small and can be neglected, hence $P_1 \cong P_0$. The air density will vary significantly with humidity ratio change $\omega_o \rightarrow \omega_1$ and decrease in the air temperature $T_o \rightarrow T_1$.

6.3.2- Air compressor

For humid air, the compression power can be estimated from:

$$\dot{W}_{compair} = \dot{m}_a\,c_{pa}\big(T_2 - T_1\big) + \dot{m}_v\big(h_{g2} - h_{g1}\big) \tag{6.8}$$

where h_{g2} and h_{g1} are the enthalpies of saturated water vapor at the compressor exit and inlet states respectively, $\dot{m}_v$ is the mass of water vapor = $\dot{m}_a\omega_1$

Relating the compressor isentropic efficiency to the changes in temperature of the dry air and assuming that the compression of water vapor behaves as an ideal gas then

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{6.9}$$

from which T_2 is expressed in terms of T_1 and the pressure ratio PR as

$$T_2 = T_1\left[\frac{PR^{\frac{k-1}{k}} - 1}{\eta_c} + 1\right] \tag{6.10}$$

Substituting for T_2 into Eq. 6.8 gives the actual compressor power as:

$$\dot{W}_{comp_{air}} = \dot{m}_a\left[c_{pa}\frac{T_1}{\eta_c}\left(PR^{\frac{k-1}{k}} - 1\right) + \omega_1\left(h_{g2} - h_{g1}\right)\right] \tag{6.11}$$

Where η_c can be evaluated using the following empirical relation,

$$\eta_c = 1 - \left(0.04 + \frac{PR - 1}{150}\right) \tag{6.12}$$

6.3.3- Combustion Chamber

Heat balance on the combustion chamber gives the heat rate supplied to the gas turbine cycle as:

$$\dot{Q}_{in} = \dot{m}_f\ CV = \left(\dot{m}_a + \dot{m}_f\right)c_{pg}T_3 - \dot{m}_a c_{pa}T_2 + \dot{m}_v\left(h_{v3} - h_{v2}\right) \tag{6.13}$$

Introducing the fuel air ratio $FA = \dot{m}_f / \dot{m}_a$ and substituting for T_2 in terms of T_1 from Eq. 6.10 gives the cycle heat rate as:

$$\dot{Q}_{in} = \dot{m}_a\, T_1\left[\left(1 + FA\right)c_{pg}\frac{T_3}{T_1} - c_{pa}\left(\frac{PR^{\frac{k-1}{k}} - 1}{\eta_c} + 1\right) + \frac{\omega_1}{T_1}\left(h_{v3} - h_{v2}\right)\right] \tag{6.14}$$

where FA, as expressed in, is:

$$FA = \frac{c_{pg}\left(T_3 - 298\right) - c_{pa}\left(T_2 - 298\right) + \omega_1\left(h_{v3} - h_{v2}\right)}{CV - c_{pg}\left(T_3 - 298\right)} \tag{6.15}$$

h_{v2} and h_{v3} and are the enthalpies of water vapor at the combustion chamber inlet and exit states respectively and can be calculated from:

$$h_{v,j} = 2501.3 + 1.8723\ T_j \qquad j = 2 \text{ or } 3 \tag{6.16}$$

It is seen that the three terms of the gas turbine efficiency in Eq. 6.2 ($\dot{W}_t, \dot{W}_{comp,air}$ and $\dot{Q}_h$) depend on the air temperature and relative humidity at the compressor inlet whose values are affected by the atmospheric conditions.

The cycle thermal efficiency with a cooling system, η_{cy} in terms of the air properties at the compressor intake, the fuel air ratio and the refrigeration machine characteristics is

$$\eta_{cy} = \left\{ \left(1+\omega_1+FA\right)c_{pg}\,\eta_t\,\frac{T_3}{T_1}\left(1-\frac{1}{PR^{\frac{k-1}{k}}}\right) - \left[c_{pa}\,\frac{1}{\eta_c}\left(PR^{\frac{k-1}{k}}-1\right)+\frac{\omega_1}{T_1}\left(h_{g2}-h_{g1}\right)\right]\right\} \Bigg/ \left[\left(1+FA\right)c_{pg}\,\frac{T_3}{T_1}-c_{pa}\left(\frac{PR^{\frac{k-1}{k}}-1}{\eta_c}+1\right)+\frac{\omega_1}{T_1}\left(h_{v3}-h_{v2}\right)\right] \qquad (6.17)$$

Figure 6.4 shows the thermal efficiency variation with pressure ratio for different turbine inlet absolute temperature. As the maximum temperature T_3 is kept constant (1373 K), an increase in pressure ratio will increase the cycle efficiency. However, there is a limit to the pressure ratio. The ratio will reach the maximum limit when the air temperature at the compressor outlet is equal to the design turbine inlet temperature. In this case, the cycle net work tends towards zero, and the cycle efficiency approaches the reversible efficiency. In the present Chapter where the outlet air temperature is cooled from 50 °C to wet bulb temperature (37.12 °C) increases the term T_3/T_1 from 4.25 to 4.425. Consequently, the thermal efficiency increases as seen by the trends in Figure 6.4.

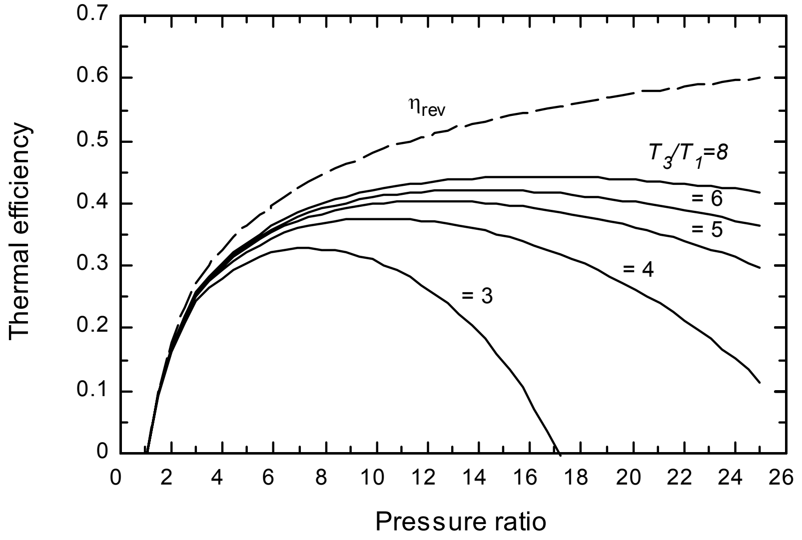


Figure 6.4 The thermal efficiency of a simple gas turbine cycle

Example 6.1

Consider a simple gas turbine power plant with air entering the compressor at 1 bar and 27 °C and exhausting at 10 bar, with volumetric flow rate of 5 m^3/s. The maximum cycle temperature is 1127 °C at the turbine inlet. Calculate the cycle efficiency and net work of air, using equations 6.6 and 6.12 for calculating the turbine and compressor efficiency. The pressure drops in the combustion chamber, compressor and turbine are assumed to be negligible.

Solution

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6.4- Effect of regeneration

The temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the thermal efficiency of the gas turbine cycle may be improved by introducing a regenerator. The regeneration process transfers the portion of energy of the exhaust gases that is normally rejected to the surroundings to preheat the air entering the combustion chamber. This, in turn, decreases the heat input (fuel) requirements for the same net work output.

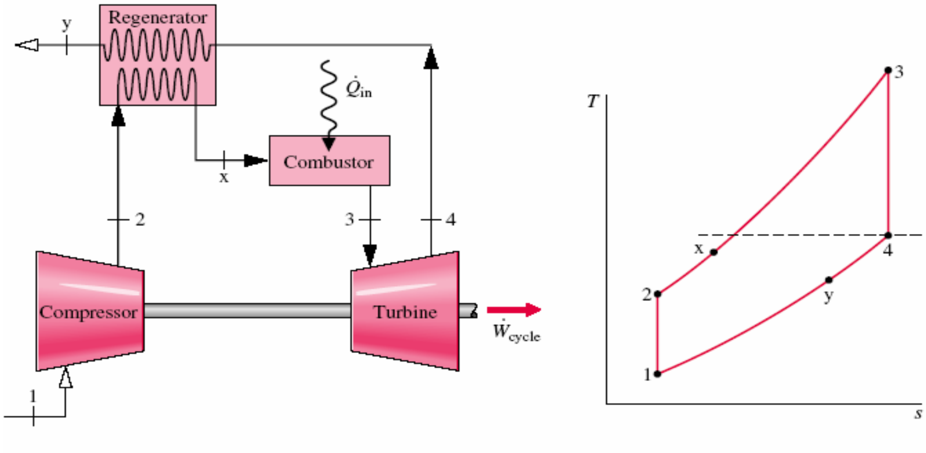

Figure (6.5) A regenerative gas turbine system

Determining the efficiency of the ideal cycle with a regenerator

The efficiency of this cycle with regeneration is found as follows, where the states are shown in Figures (6.5).

$$\eta_{th,Brayton,reg} = \frac{w_{net}}{q_{in}} = \frac{w_t - w_c}{q_{in}} \tag{6.18}$$

$$q_{in} = c_{pa}\left(T_3 - T_x\right)$$

$$w_t = c_{pa}\left(T_3 - T_4\right)$$

But for an ideal regenerator, $T_4 = T_x$, and therefore $q_{in} = w_t$, consequently

$$\eta_{th,Brayton,reg} = 1 - \frac{w_c}{q_{in}} = 1 - \frac{c_{pa}\left(T_2 - T_1\right)}{c_{pa}\left(T_3 - T_4\right)} = 1 - \frac{T_1\left({T_2}/{T_1} - 1\right)}{T_3\left(1 - {T_4}/{T_3}\right)}$$

$$= 1 - \frac{T_1\left(\left({P_2}/{P_1}\right)^{\left({(k-1)}/{k}\right)} - 1\right)}{T_3\left(1 - \left({P_1}/{P_2}\right)^{\left({(k-1)}/{k}\right)}\right)} = 1 - \frac{T_1}{T_3}\left(\frac{P_2}{P_1}\right)^{{(k-1)}/{k}} \tag{6.19}$$

The effectiveness of a regenerator can be defined by

$$\varepsilon_{reg} = \frac{\dot{Q}_{actual}}{\dot{Q}_{max}} = \frac{h_x - h_2}{h_4 - h_2} \qquad (6.20a)$$

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

$$\varepsilon_{reg} = \frac{T_x - T_2}{T_4 - T_2} \qquad (6.20b)$$

Example 6.2

A stationary power plant operating on an ideal Brayton cycle has a pressure ratio of 8. The gas temperature is 300 K at the compressor inlet and 1300 K at the turbine inlet. Utilizing the cold-air standard assumptions, determine

- the gas temperature at the exits of the compressor and turbine.
- the back work ratio.
- the thermal efficiency.
- determine the thermal efficiency of the gas-turbine power plant, if a regenerator having an effectiveness of 80% is installed.

Solution

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6.5- Effect of intercooling

The net work *output of a gas turbine* can be increased by reducing the compressor work input. This can be accomplished by means of multistage compression with intercooling. Let us first consider the work input to compressors at steady state, assuming that irreversibilities are absent and changes in kinetic and potential energy from inlet to exit are negligible. The P-v diagram of Fig. 6.6 shows two possible compression paths from a specified state 1 to a specified final pressure P_2. Path 1-2' is for an adiabatic compression. Path 1-2 corresponds to a compression with heat transfer from the working fluid to the surroundings. The area to the left of each curve equals the magnitude of the work per unit mass of the respective process. The smaller area to the left of Process 1-2 indicates that the work of this process is less than for the adiabatic compression from 1 to 2'. This suggests that cooling a gas during compression is advantageous in terms of the work input requirement.

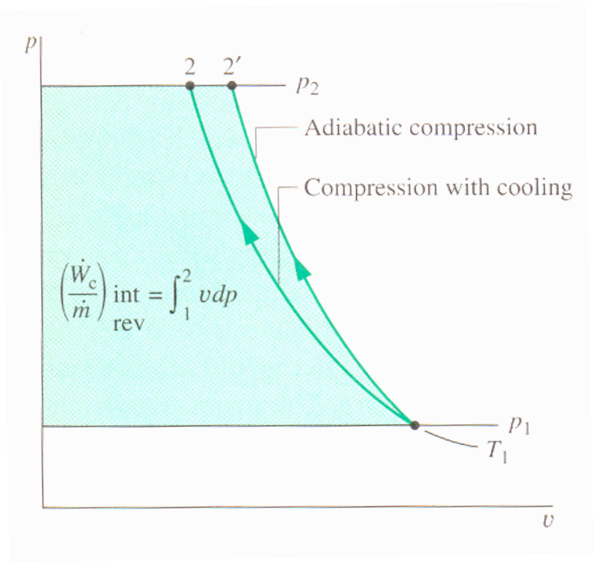


Figure 6.6 Compression process between two fixed pressures

Although cooling a gas as it is compressed would reduce the work, a heat transfer rate high enough to effect a significant reduction in work is difficult to achieve in practice. A practical alternative is to separate the work and heat interactions into separate processes by letting compression take place in stages with heat exchangers, called ***intercoolers***, cooling the gas between stages. Figure 6.7 illustrates a two-stage compressor with an intercooler. The accompanying P-v and T-s diagrams show the states for internally reversible processes:

- Process 1-c is an isentropic compression from state I to state c where the pressure is P_i
- Process c-d is constant-pressure cooling from temperature T_c, to T_d.
- Process d-2 is an isentropic compression to state 2.

The work input per unit of mass flow is represented on the P-v diagram by shaded area 1-c-d-2-a-b-I. Without intercooling the gas would be compressed isentropically in a single stage from state I to state 2' and the work would be represented by enclosed area 1-2'-ab-1. The crosshatched area on the P-v diagram represents the reduction in work that would be achieved with intercooling.

Some large compressors have several stages of compression with intercooling between stages. The determination of the number of stages and the conditions at which to operate the various intercoolers is a problem in optimization. The use of multistage compression with intercooling in a gas turbine power plant increases the net work developed by reducing the compression work. By itself, though, compression with intercooling would not necessarily increase the thermal efficiency of a gas turbine because the temperature of the air entering the combustor would be reduced (compare temperatures at states 2' and 2 on the T-s diagram of Fig. 6.7). A lower temperature at the combustor inlet would require additional heat transfer to achieve the desired turbine inlet temperature. The lower temperature at the compressor exit enhances the potential for regeneration, however, so when intercooling is used in conjunction with regeneration, an appreciable increase in thermal efficiency can result.

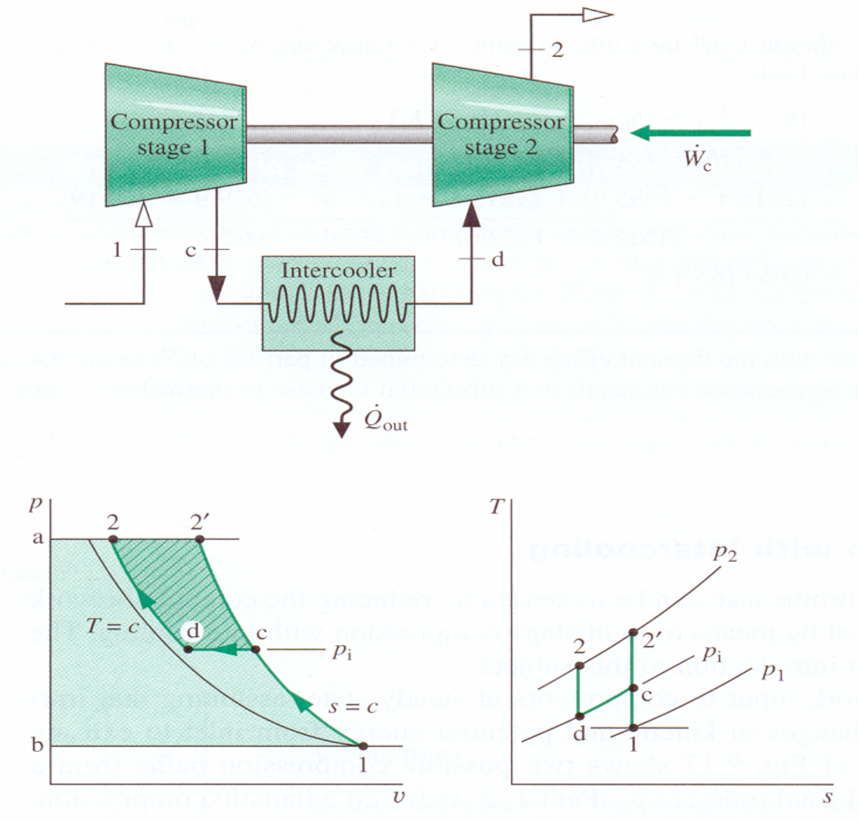


Figure 6.7 Two stage compression with intercooling

The efficiency of this cycle with intercooling can be found as follows, where the states are shown in Figures (6.7).

$$\eta_{th,Brayton,int} = \frac{w_{net}}{q_{in}} = \frac{w_t - \left[c_{pa}\left(T_c - T_1\right) + c_{pa}\left(T_2 - T_d\right)\right]}{q_{in}} \qquad (6.21)$$

Minimum work of compression

The total compressor work input per unit of mass flow is

$$w_c = (h_c - h_1) + (h_2 - h_d) \tag{6.22a}$$

$$w_c = c_{pa}(T_c - T_1) + c_{pa}(T_2 - T_d) \tag{6.22b}$$

Assuming $T_d = T_1$

$$w_c = c_{pa}\, T_1\left(\frac{T_c}{T_1} + \frac{T_2}{T_1} - 2\right) \tag{6.22c}$$

The pressure and temperature ratios across the compressor stages are

$$\frac{T_c}{T_1} = \left[\frac{P_i}{P_1}\right]^{\frac{k-1}{k}} \text{ and } \frac{T_2}{T_d} = \left[\frac{P_2}{P_i}\right]^{\frac{k-1}{k}} \tag{6.23}$$

Substituting equation 6.23 into equation 6.22c yields

$$w_c = c_{pa}\, T_1\left(\left(\frac{P_i}{P_1}\right)^{\frac{k-1}{k}} + \left(\frac{P_2}{P_i}\right)^{\frac{k-1}{k}} - 2\right) \tag{6.24}$$

Hence, for specified values of T_1, P_1,P_2, and c_p, the values of the total compressor work input varies with the intercooler pressure only. To determine the pressure P_i, that minimizes the total work, from the derivative

$$\frac{\partial w_c}{\partial P_i} = \frac{\partial}{\partial P_i}\left\{c_{pa}T_1\left[\left(\frac{P_i}{P_1}\right)^{\frac{k-1}{k}} + \left(\frac{P_2}{P_i}\right)^{\frac{k-1}{k}} - 2\right]\right\} \tag{6.25a}$$

$$= c_{pa}T_1\left(\frac{k-1}{k}\right)\left[\left(\frac{P_i}{P_1}\right)^{\frac{-1}{k}}\left(\frac{1}{P_1}\right) + \left(\frac{P_2}{P_i}\right)^{\frac{-1}{k}}\left(-\frac{P^2}{P_i^2}\right)\right] \tag{6.25b}$$

$$= c_{pa}T_1\left(\frac{k-1}{k}\right)\left(\frac{1}{P_i}\right)\left[\left(\frac{P_i}{P_1}\right)^{\frac{k-1}{k}} - \left(\frac{P_2}{P_i}\right)^{\frac{k-1}{k}}\right] \tag{6.25c}$$

When the partial derivative is set to zero, the desired relationship is obtained

$$\frac{P_i}{P_1} = \frac{P_2}{P_i} \tag{6.26}$$

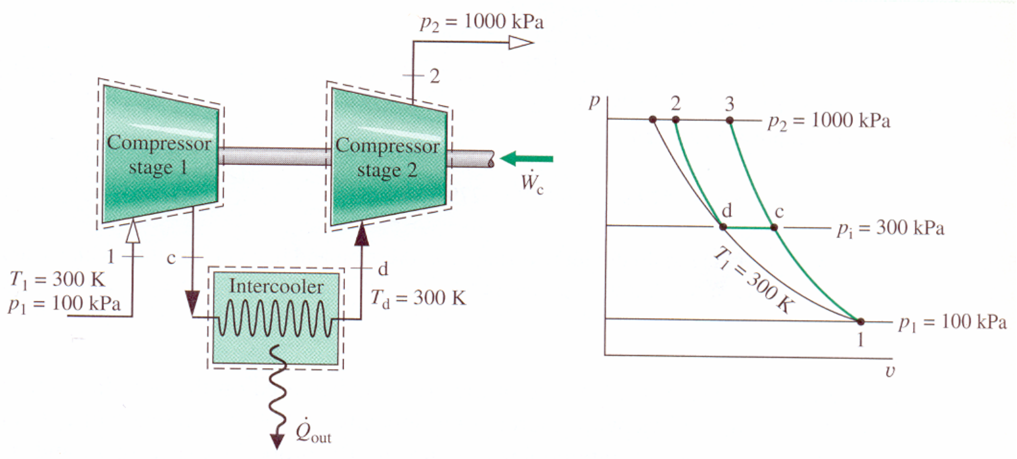
$$P_i = \sqrt{P_1 \times P_2} = (P_1 \times P_2)^{0.5} \tag{6.27}$$

By checking the sign of the second derivative, it can be verified that the total compressor work is minimum.

Example 6.3

Air is compressed from 100 kPa, 300 K to 1000 kPa in a two-stage compressor with intercooling between stages. The intercooler pressure is 300 kPa. The air is cooled back to 300 K in the intercooler before entering the second compressor stage. Each compressor stage is isentropic. For steady-state operation and negligible changes in kinetic and potential energy from inlet to exit, determine (a) the temperature at the exit of the second compressor stage and (b) the total compressor work input per unit of mass flow. (c) Repeat for a single stage of compression from the given inlet state to the final pressure.

Solution



6.6- Effect of Reheating

For metallurgical reasons, the temperature of the gaseous combustion products entering the turbine must be limited. This temperature can be controlled by providing air in excess of the amount required to burn the fuel in the combustor (see Chapter 6). As a consequence, the gases exiting the combustor contain sufficient air to support the combustion of additional fuel. Some gas turbine power plants take advantage of the excess air by means of a multistage turbine with a reheat combustor between the stages. With this arrangement the net work per unit of mass flow can be increased. Let us consider reheat from the vantage point of an air-standard analysis.

The basic features of a two-stage gas turbine with reheat are brought out by considering an ideal air-standard Brayton cycle modified as shown in Figure 6.8. After expansion from state 3 to state a in the first turbine, the gas is reheated at constant pressure from state a to state b. The expansion is then completed in the second turbine from state b to state 4. The ideal Brayton cycle without reheat, 1-2-3-4'-1, is shown on the same T-s diagram for comparison. Because lines of constant pressure on a T-s diagram diverge slightly with increasing entropy, the total work of the two-stage turbine is greater than that of a single expansion from state 3 to state 4'. Thus, the net work for the reheat cycle is greater than that of the cycle without reheat. Despite the increase in net work with reheat, the cycle thermal efficiency would not necessarily increase because a greater total heat addition would be required. However, the temperature at the exit of the turbine is higher with reheat than without reheat, so the potential for regeneration is enhanced.

When reheat and regeneration are used together, the thermal efficiency can increase significantly.

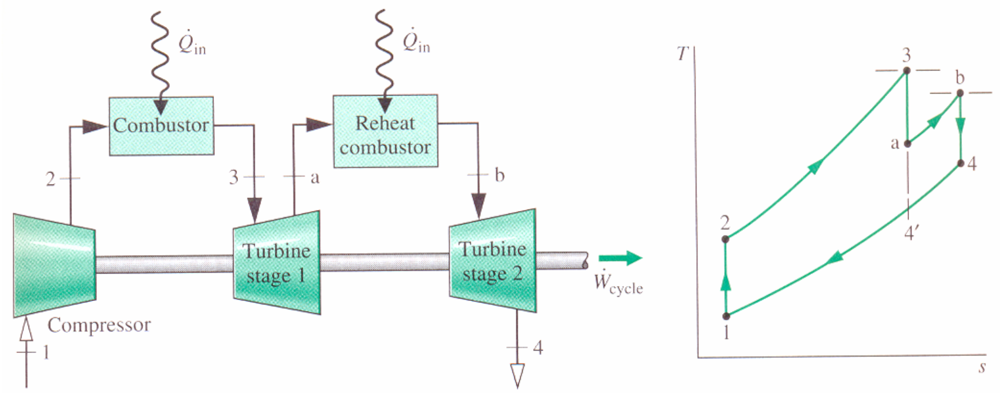


Figure 6.8 Gas turbine with reheat

The efficiency of this cycle with reheating is found as follows, where the states are shown in Figures (6.8).

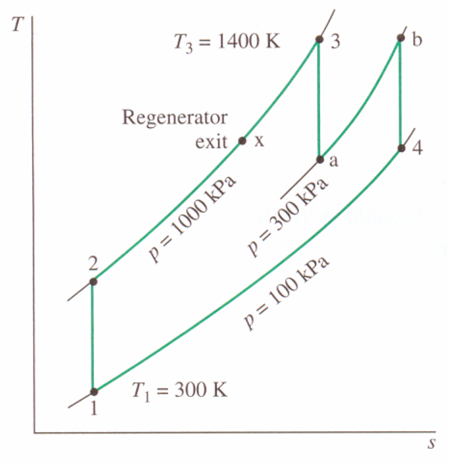
$$\eta_{th,Brayton,reh}=\frac{w_{net}}{q_{in}}=\frac{c_{pa}\left(T_3-T_a\right)+c_{pa}\left(T_b-T_4\right)-c_{pa}\left(T_2-T_1\right)}{c_{pa}\left(T_3-T_2\right)+c_{pa}\left(T_b-T_a\right)} \tag{6.28}$$

For an ideal cycle with regeneration $T_2= T_x$

Example 6.4

Consider a modification of the cycle of example 6.1 involving reheat and regeneration. Air enters the compressor at 100 kPa, 300 K and is compressed to 1000 kPa. The temperature at the inlet to the first turbine stage is 1400 K. The expansion takes place isentropically in two stages, with reheat to 1400 K between the stages at a constant pressure of 300 kPa. A regenerator having an effectiveness of 90% is also incorporated in the cycle. Determine the thermal efficiency.

Solution



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6.7- Effect of Reheating and Intercooling

Reheat between turbine stages and intercooling between compressor stages provide two important advantages: The net work output is increased, and the potential for regeneration is enhanced. Accordingly, when reheat and intercooling are used together with regeneration a substantial improvement in performance can be realized. One arrangement incorporating reheat, intercooling, and regeneration is shown in Figure 6.9. This gas turbine has two stages of compression and two turbine stages. The accompanying T-s diagram is drawn to indicate irreversibilities in the compressor and turbine stages. The pressure drops that would occur as the working fluid passes through the intercooler, regenerator, and combustors are not shown.

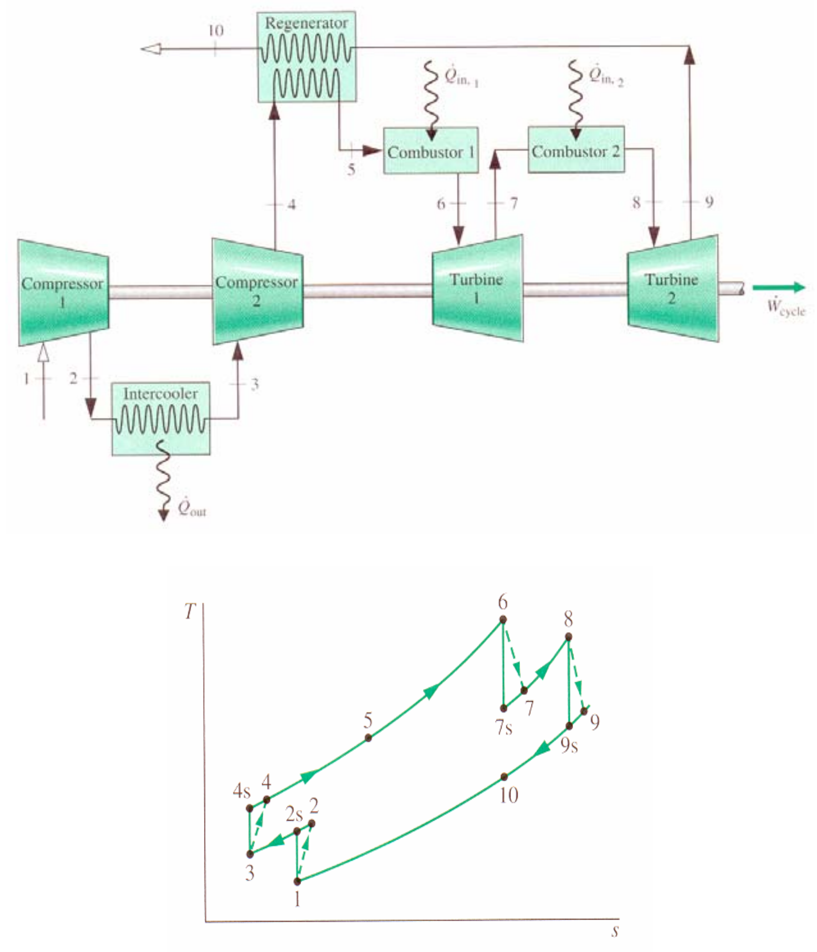

Figure 6.9. Regenerative gas turbine with reheating and intercooling

The efficiency of this cycle with reheating is found as follows, where the states are shown in Figures (6.9).

$$\eta_{th,Brayton,reh}=\frac{w_{net}}{q_{in}}=\frac{c_{pa}(T_6-T_7)+c_{pa}(T_8-T_9)-\left[c_{pa}(T_2-T_1)+c_{pa}(T_4-T_3)\right]}{c_{pa}(T_6-T_5)+c_{pa}(T_8-T_7)} \tag{6.29}$$

For an ideal cycle without regeneration $T_5 = T_4$

Example 6.5

A regenerative gas turbine with intercooling and reheat as shown in Figure 6.9 operates at steady state. Air enters the compressor at 100 kPa, 300 K with a mass flow rate of 5.807 kg/s. The pressure ratio across the two-stage compressor is 10. The pressure ratio across the two-stage turbine is also 10. The intercooler and reheater each operate at 300 kPa. At the inlets to the turbine stages, the temperature is 1400 K. The temperature at the inlet to the second compressor stage is 300 K. The isentropic efficiency of each compressor and turbine stage is 80%. The regenerator effectiveness is 80%. Determine (a) the thermal efficiency, (b) the back work ratio, (c) the net power developed, in kW.

Solution

6.8-Thermodynamic analysis

The network of a Gas Turbine power plant is given by

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c \quad (6.30a)$$

$$\dot{W}_{net} = \left(\dot{m}_a + \dot{m}_f\right)c_{pg}\left(T_3 - T_4\right) - \dot{m}_a c_{pa}\left(T_2 - T_1\right) \quad (6.30b)$$

And the heat supply is

$$\dot{Q}_{in} = \dot{m}_f \ HHV \quad (6.31)$$

where CV = Calorific Value of the fuel (kJ/kg), $\dot{m}_f$ = fuel burning rate (kg/s),

Therefore, the overall plant efficiency,

$$\eta_o = \frac{\dot{W}_{net}}{\dot{m}_f \ HHV} \quad (6.32)$$

6.9-Components of gas turbine plant

The construction and operation of the components of a gas turbine plant are necessary for proper understanding and design.

6.9.1-Compressor

The high flow rates of air through the turbine and the relatively moderate pressure ratios necessitate the use of rotary compressors. The types of compressor commonly used are the following.

1. Centrifugal compressors
2. Axial flow compressors

A centrifugal compressor consists of an impeller with a series of curved radial vanes as shown in Fig. 6.10. Air is sucked in near the hub, called the impeller eye and is whirled round at high speed by the vanes on the impeller rotating at high rpm. The static pressure of air increases from the eye to the tip of the impeller. Air leaving the impeller tip flows through diffuser passages (scroll) which convert the kinetic energy to pressure energy (Fig. 6.11). The compressors may have single inlet or double inlet. In a double inlet impeller having an eye on either side, air is drawn in on both sides (Fig. 6.12). The impeller is subjected to approximately equal forces in the axial direction. About half the pressure rise occurs in the impeller vanes and half in the diffuser passages.

If the air flow into the impeller eye is in the axial direction (Fig. 6.10), the blade velocity diagram at inlet is shown in Fig. 6.13 (a). By using fixed guide blades, the inlet velocity to the impeller eye is inclined at an angle, known as pre-whirl (Fig. 6.13 (b)).

At exit from the impeller the flow is in the radial direction and the blade velocity V_b, is larger, since the radius of the impeller is larger at outlet. The blade velocity diagram is shown in Fig. 6.14 (a) being the case of radially inclined blades and (b) being that of blades inclined backwards at an angle β_2.

The inertia of the air trapped between the impeller blades, however, causes the actual whirl velocity V'_{w2} to be less than V_{w2}. It is known as slip.

Slip factor = $\frac{V'_w}{V_w}$ (6.33)

Power input = $\dot{m}\left(V_{b2}V'_{w2} - V_{b1}V'_{w1}\right)$ (6.34)

For low pressure ratios (less than 4/1) the centrifugal compressor is lighter and is able to operate effectively over a wider range of mass flows at any speed. Using titanium alloys pressure ratios above eight have now been achieved.

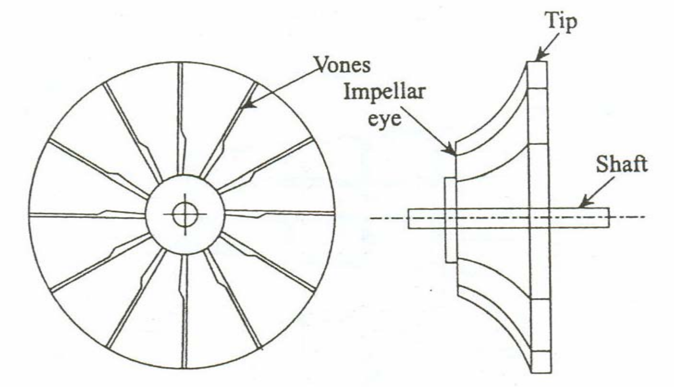

Figure 6.10 Centrifugal compressor impeller

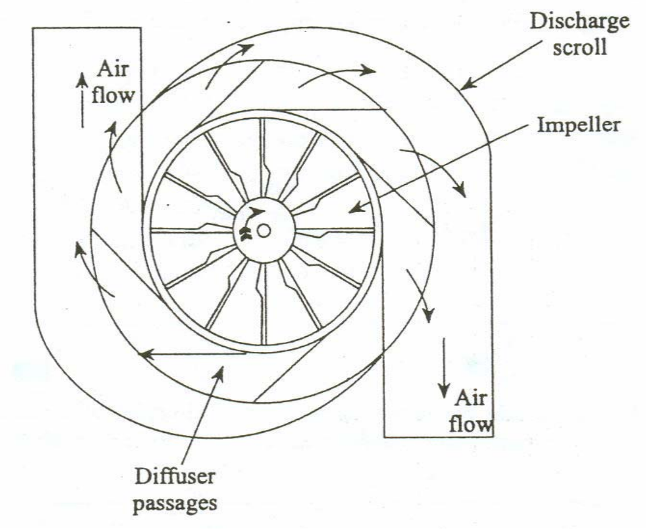

Figure 6.11 Centrifugal compressor showing discharge scroll

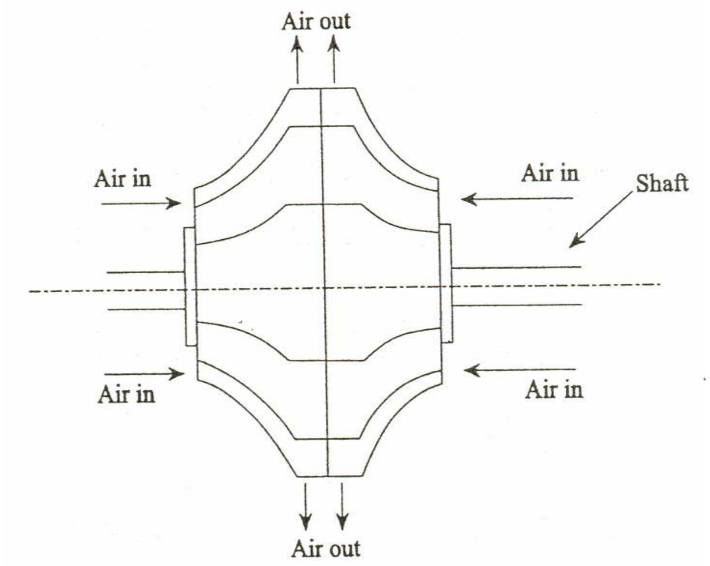

Figure 6.12 Double sided impeller of a centrifugal compressor

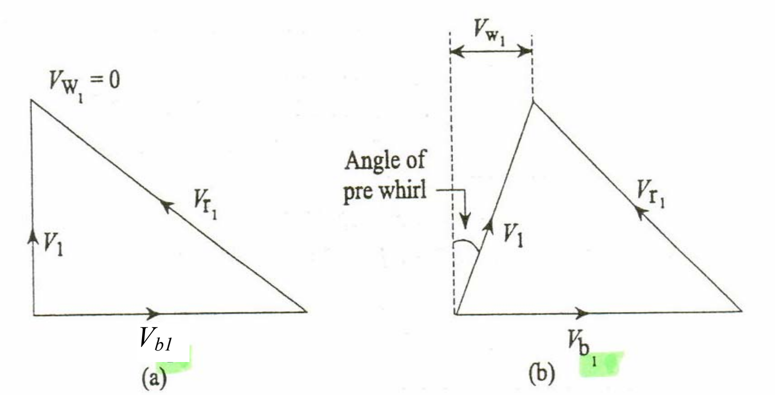

Figure 6.13 Blade velocity diagrams at blade inlet of a centrifugal compressor (a) without and (b) with pre-whirl

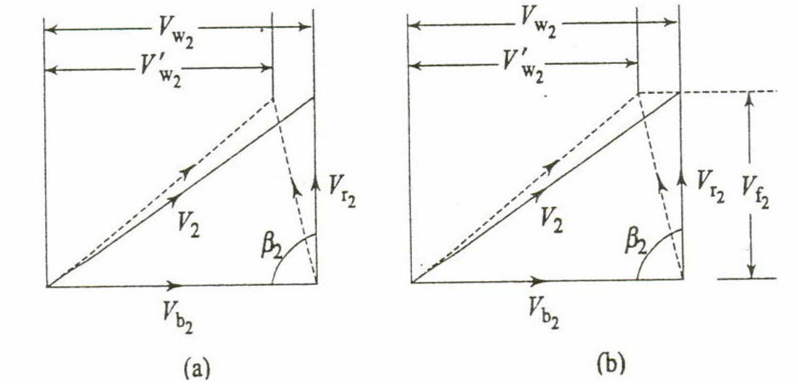

Figure 6.14 Blade velocity diagrams at blade outlet of a centrifugal compressor for (a) Radially inclined and (b) Backward inclined blading

For larger units with higher pressure ratios the axial-flow compressor is more efficient and is usually preferred. For industrial and large marine .gas turbine plants axial compressors are normally used, although some units may employ two or more centrifugal compressors with intercooling between stages. Centrifugal compressors are cheaper to produce, more robust and have a wider operating range than the axial-flow type.

An axial-flow compressor is similar to an axial-flow turbine with a succession of moving blades on the rotor shaft and fixed blades arranged around the stator (casing). Air flows axially through the moving and fixed blades, with diffusers passages throughout which continuously increase the pressure and decrease the velocity. Stationary guide vanes are provided at entry to the first row of moving blades (Fig. 6.15). The work input to the rotor shaft is transferred by the moving blades to the air, thus accelerating it. The spaces between the blades as well as the stator blades from diffusing passages decreasing velocity and increasing pressure. There can be a large number of stages (5 to 14) with a constant work input per stage.

An equal temperature rise in the moving and fixed blades is usually maintained. The axial velocity of air is also kept constant throughout the compressor. A diffusing flow is less stable than a converging flow as in a turbine and for this reason the blade shape and profile are more important for a compressor than for a reaction turbine.

Typical blade sections of an axial-flow compressor are shown in Fig.6.16 (a) and the corresponding velocity diagrams in Fig.6.16 (b).

Power input, $$\dot{W} = \dot{m} V_b \left(\Delta V_w\right) \tag{6.35}$$

From the geometry of the diagram (refer to Figure 6.16 b)

$$\Delta V_w = V_{r1} \sin\beta_1 - V_{r2} \sin\beta_2 = V_f\left(\tan\beta_1 - \tan\beta_2\right) \tag{6.36}$$

Degree of reaction,

$$R = \frac{Enthalpy\ rise\ in\ rotor}{Enthalpy\ rise\ in\ the\ stage}$$

$$= \frac{h_1 - h_0}{h_2 - h_0} = \frac{V_{r1}^2 - V_{r2}^2}{2V_b\left(\Delta V_w\right)}$$

By re-arrangement,

$$R = \frac{V_f^2}{2V_b}\left(\tan\beta_1 - \tan\beta_2\right) \tag{6.37}$$

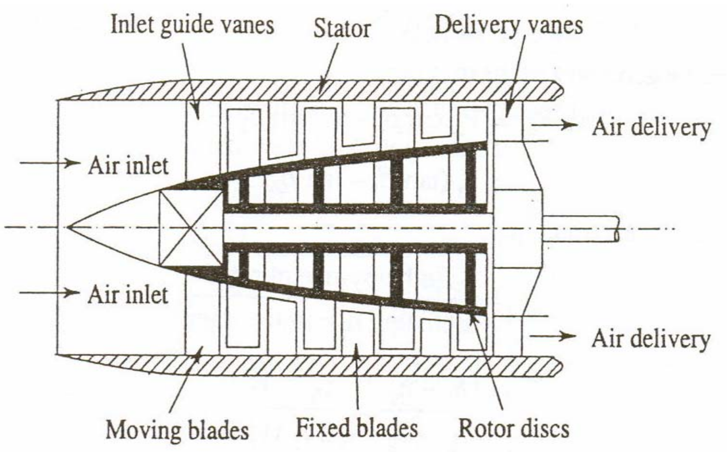

Figure 6.15 Axial flow compressor

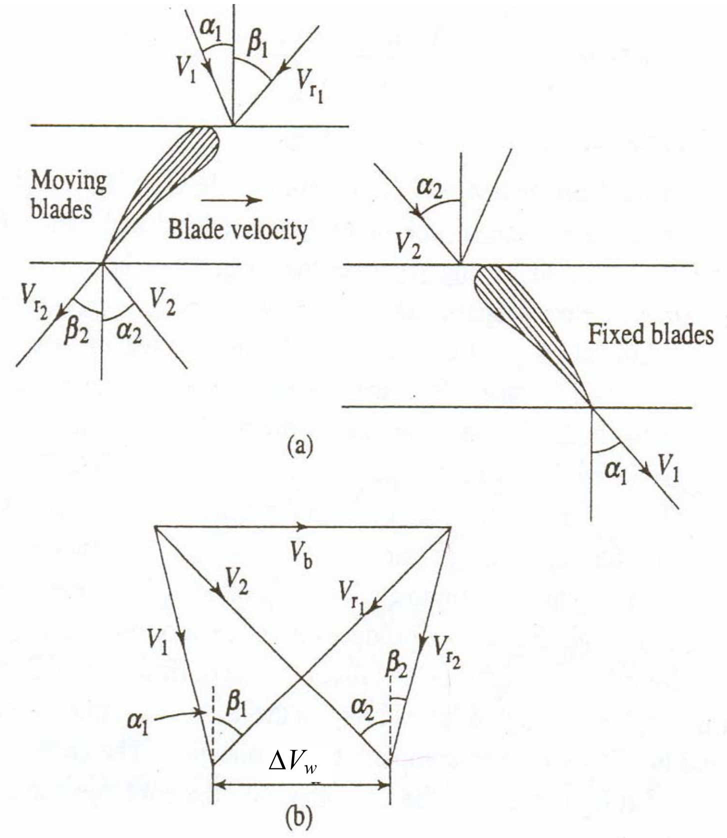

Figure 6.16 (a) Typical blade sections (b) Blade velocity diagrams for an axial flow compressor

6.9.2-Combustion chamber

In an open cycle GT plant combustion may be arranged to take place in one or two large cylindrical can-type combustion chambers (CC) with dueling to convey the hot gases to the turbine. Combustion is initiated by an electric spark and once the fuel starts burning, the flame is required to be stabilized. A pilot or recirculated zone is created in the main flow to establish a stable flame which helps to sustain combustion continuously. The common methods of flame stabilization are by swirl flow and by bluff body.

Figure 6.17 shows a can-type combustor with swirl flow flame stabilization. About 20 per cent of the total air from the compressor is directly fed through a swirler to the burner as primary air, to provide a rich fuel-air mixture in the primary zone, which continuously burns, producing high temperature gases. Air flowing through the swirler produces a vortex motion creating a low pressure zone along the axis of the CC to cause reversal of flow. About 30 per cent of total air is supplied through dilution holes in the secondary zone through the annulus round the flame tube to complete the combustion. The secondary air must be admitted at right points in the CC, otherwise the cold injected air may chill the flame locally thereby reducing he rate of reaction. The secondary air not only helps to complete the combustion process but also helps to cool the flame tube. The remaining 50 per cent of air is mixed with burnt gases in the tertiary zone to cool the gases down to the temperature suited to the turbine blade materials.

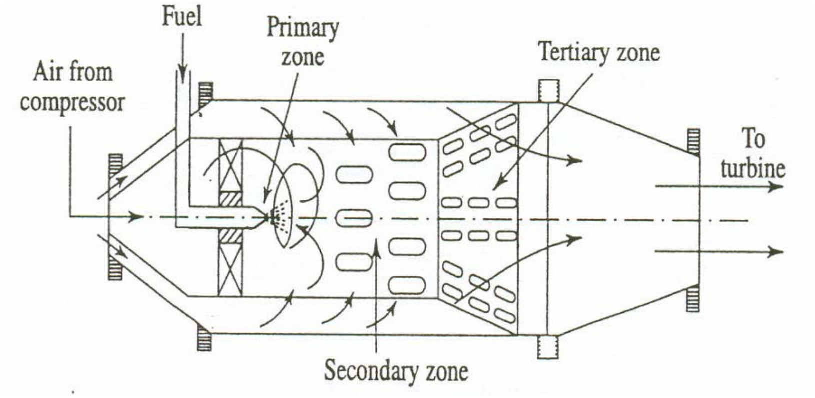

Figure 6.17 Can-Type combustor with swirl flow flame stabilizer

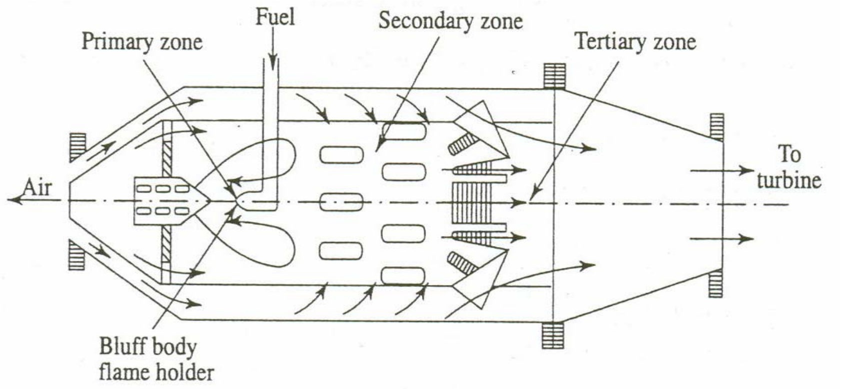

Figure 6.18 Can-type combustor with bluff-body flame stabilizer

Figure 6.18 shows a can-type combustor with a bluff body stabilizing the flame. The fuel is injected upstream into the air flow and a sheet metal cone and perforated baffle plate ensure the necessary mixing of fuel and air. The low pressure zone created downstream side causes the reversal of flow along the axis of the CC to stabilize the flame. Sufficient turbulence is produced in all three zones of the CC for uniform mixing and good combustion.

The air-fuel ratio in a GT plant varies from 60/1 to 120/1 and the air velocity at entry to the CC is usually not more than 75 m/s. There is a rich and a weak limit of flame stability and the limit is usually taken at flame blowout. Instability of the flame results in rough running with consequent effect on the life of the CC.

Because of the high air-fuel ratio used, the gases entering the HP turbine contain a high percentage of oxygen and therefore if reheating is performed, the additional fuel can be burned satisfactorily in HP turbine exhaust, without needing further air for oxygen.

A term "*combustion efficiency*" is often used in this regard, which is defined as follows.

$$Combustion\, efficiency = \frac{Theoritical\ AF\ ratio\ for\ actual\, temperature\, rise}{Actual\ AF\ ratio\ for\ actual\, temperature\, rise} \tag{6.38}$$

Theoretical temperature rise depends on the calorific value of the fuel used, the fuel-air ratio and the initial temperature of air. To evaluate the combustion efficiency, the inlet and outlet temperatures and the fuel and air mass flow rate are measured. The fuel used in aircraft gas turbine is a light petroleum distillate or kerosene of gross calorific value of 46.4 MJ/kg. For gas turbines used in power production or in cogeneration plants, the fuel used can be natural gas.

In order to give a comparison of combustion chambers operating under different ambient conditions, a combustion intensity is defined as the following.

$$Combustion\, intensity = \frac{Heat\, release\, rate}{Volume\, of\ CC \times inlet\ pressure} \tag{6.39}$$

The lower the combustion intensity, the better the design. In aircraft a figure about 2 kW (m^3 atm) is normal, whereas in large industrial plants it is about 0.2 kW (m^3 atm).

6.9.3- Gas turbines

Like steam turbines, gas turbines are also of the axial-flow type (Fig. 6.19). The basic requirements of the turbines are light weight, high efficiency, reliability in operation and long working life. Large work output can be obtained per stage with high blade speeds when the blades are designed to sustain higher stresses. More stages are always preferred in gas turbine power plants, because it helps to reduce the stresses in the blades and increases the overall life of the turbine. The cooling of gas turbine blades is essential for long life as it is continuously subjected to high temperature gases.

Blade angles of gas turbines follow the axial-flow compressor blading (Fig. 6.16 (a)), where the degree of reaction is not 50 per cent. It is usually assumed for any stage that the absolute velocity at inlet to each stage (V_2) is equal to the absolute velocity at exit from the moving blades (i.e. V_2) and that the same flow velocity V_f is constant throughout the turbine.

The degree of reaction, R, as defined for a steam turbine, is valid for gas turbines also. It is the ratio of the enthalpy drop in the moving blades to the enthalpy drop in the stage. As shown in Fig. 6.16 (a), we have

$$= R = \frac{V_{r2}^2 - V_{r1}^2}{2V_b\left(\Delta V_w\right)} = \frac{V_f^2}{2V_b}\left(tan\,\beta_2 - tan\,\beta_1\right) \text{ (see equation 6.37)} \qquad (6.40)$$

Putting R =0.5 in equation 6.40, we get

$$V_b = V_f\left(tan\,\beta_2 - tan\,\beta_1\right)$$

$$\alpha_2 = \beta_1$$

It also follows that $\alpha_1 = \beta_2$.The fixed and moving blades have the same cross-section and the diagram is symmetrical.

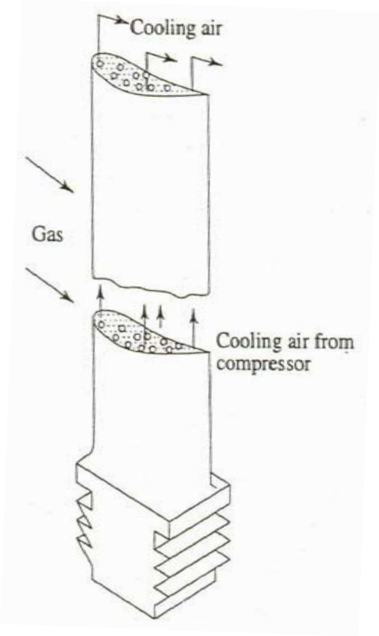

Figure 6.19 Typical air cooled gas turbine blade

Vortex bidding is the name given to the twisted blades which are designed by using three dimensional flow equations with a view to decrease fluid flow losses. A radial equilibrium equation can be derived and it can be shown that one set of conditions which satisfies this equation isas follows.

a-Constant axial velocity along the blades, i.e.

V_f = constant,

b-Constant specific work over the annulus, i.e.

$V_b\,\Delta V_w$„ = constant.

c-Free vortex at entry to the moving blades, i.e. V_{w1} x r = constant, where r is the blade radius at any point.

Since the specific work output is constant over the annulus, it can be calculated at the mean radius, and multiplied by the mass flow rate it becomes the power for the stage.

Since the fluid density varies along the blade height, the density at the mean radius can be used, so that $\dot{m} = \rho_m V_f A$, where A is the blade annular area.

d- *Duct work:* The duct work consists of ducts between the compressor and the combustion chamber, combustion chamber to the turbine, and the exhaust duct. The ducts must be sized to minimize the pressure losses, as the loss in pressure directly reduces the capacity of the plant.

Ducts should be supported from the floor to reduce vibration. Expansion joints must be provided to allow for dimensional changes due to temperature variation.

Example 6.6

A centrifugal compressor running at 16000 rpm takes in air at 17°C and compresses it through a pressure ratio of 4 with an isentropic efficiency of 82 percent. The blades are radially inclined and the slip factor is 0.85. Guide vanes at inlet give the air an angle of pre-whirl of 20° to the axial direction. The mean diameter of the impeller eye is 200 mm and the absolute air velocity at inlet is 120 m/s. Calculate the impeller tip diameter.

Solution

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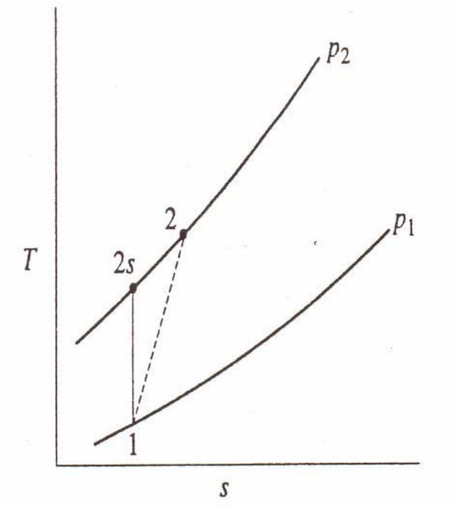

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Example 6.7

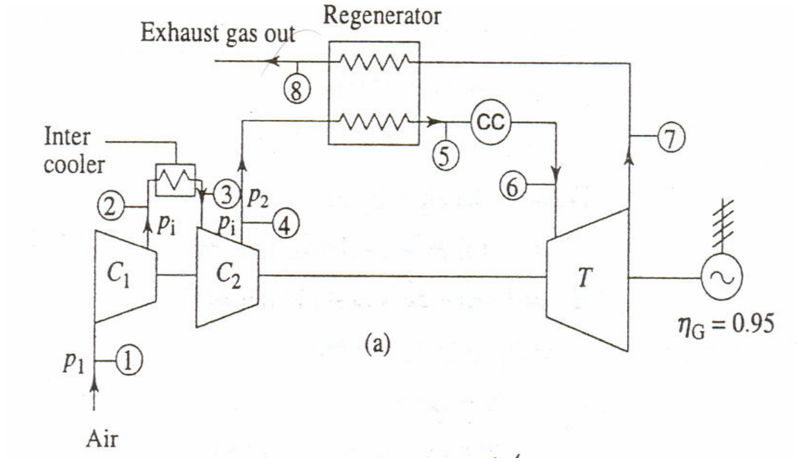
Air enters an axial flow compressor at 25°C and undergoes a pressure increase 6 times that at inlet. The mean velocity of rotor blades is 220 m/s. The inlet and exit angles of both the moving and fixed blades are 45° and 15° respectively. The degree of reaction at the mean diameter is 50 per cent and there are 10 stages in the compressor. If the isentropic efficiency of the compressor is 83 per cent and the axial velocity is taken constant throughout, find the work done factor of the compressor.

Solution

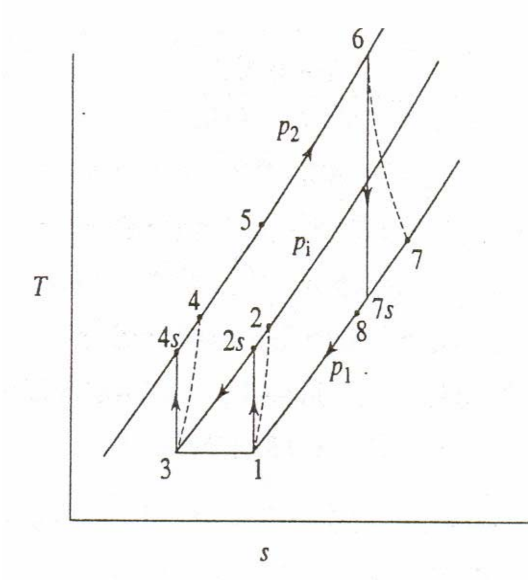
Example 6.8

A gas turbine power plant consists of a two stage compressor with intercooling and a single stage turbine with a regenerator. Air enters the compressor at I bar, 20°C. The maximum temperature of the cycle is limited to 900 °C and the maximum pressure ratio is 6. The effectiveness of the regenerator is 0.7. The rate of air flow through the plant is 210 kg/s and the calorific value of fuel used is 40.8 MJ/kg. The isentropic efficiency of both the compressors is 0.82, the isentropic efficiency of the turbine 0.92, the combustion efficiency is 0.95, the mechanical efficiency is 0.96 and the generator efficiency is 0.95. Take for gases c_p = 1.08 kJ/kg K and k = 1.33.

Assuming perfect intercooling and neglecting pressure and heat losses, estimate (a) the air-fuel ratio (b) the cycle efficiency (c) the power supplied by the plant and (d) the specific fuel consumption of the plant and the fuel consumption per hour.



(a)



(b)

Solution

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Revision questions

1- A regenerative gas turbine power plant is shown in Figure 1. Air enters the compressor at 100 kPa and 300 K with mass flow rate of 0.65 kg/s and is compressed to 800 kPa. The isentropic efficiency of the compressor is 80%, and the regenerator effectiveness is 90%. All the power developed by the high-pressure turbine is used to run the compressor. The low-pressure turbine provides the net power output. Each turbine has an isentropic efficiency of 85% and temperature at the inlet to the high-pressure turbine is 1300K as shown in Figure 1. Determine

a. The net power output in kW
b. The thermal efficiency
c. The temperature of the air at states 2, 3, 5, 6, and 7.

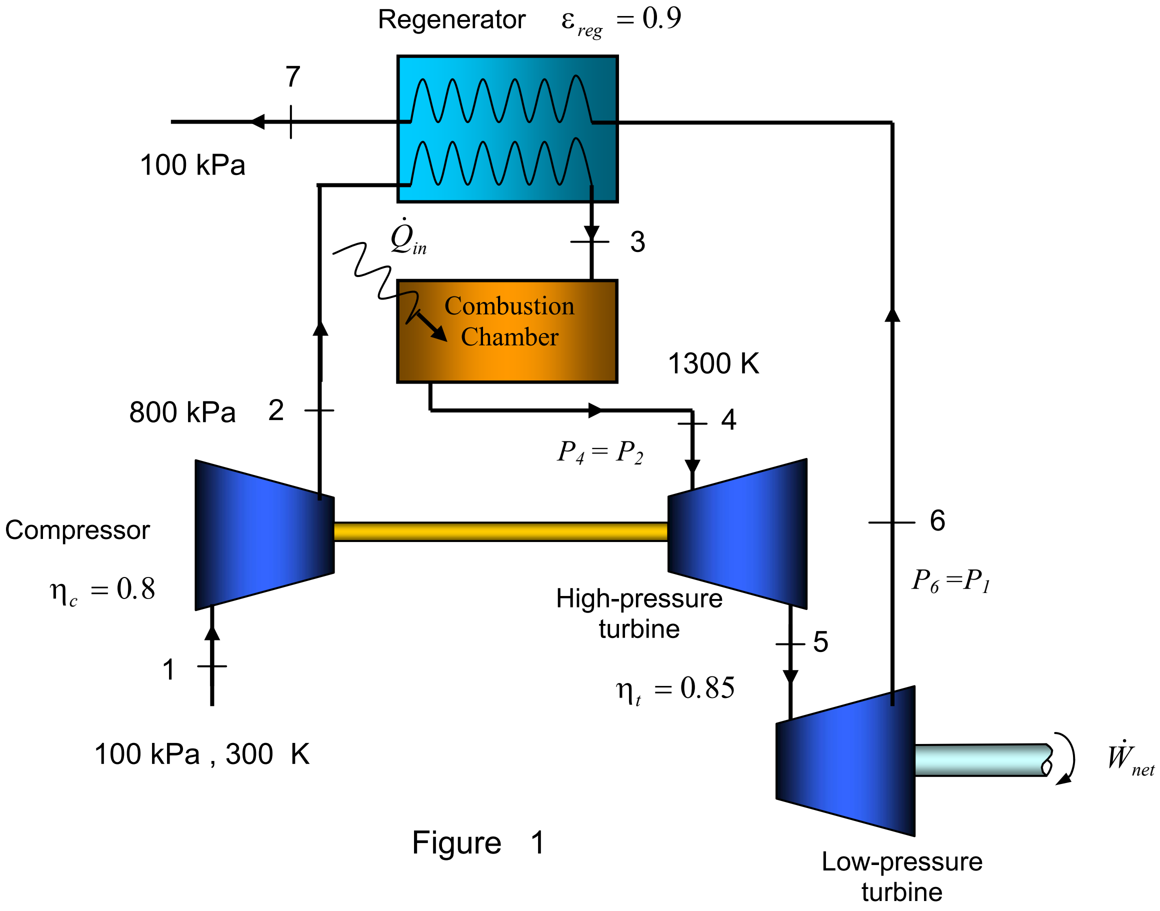

Figure 1

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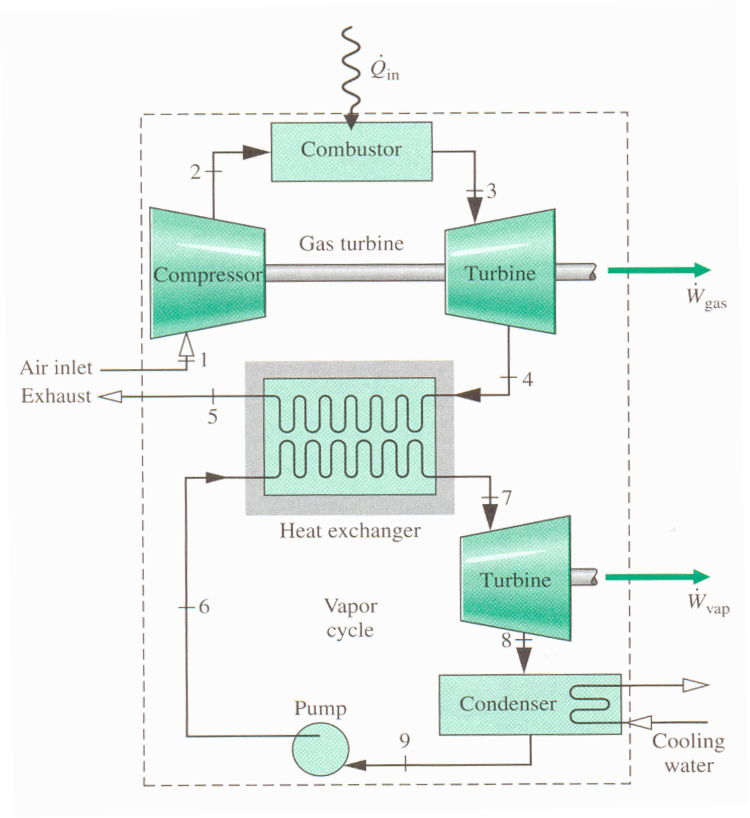

Chapter VII- Combined Cycle Power Plant

2010

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Chapter VII

Combined Cycle Power Plant

7.1- Introduction

Water is used as the working fluid in the vast majority of vapor power systems because it is plentiful and low in cost, nontoxic, chemically stable, and relatively non-corrosive. In addition, water has a relatively large change in specific enthalpy when it vaporizes at ordinary steam generator pressures, which tends to limit the mass flow rate for a desired power plant output. The properties of liquid water and water vapor are also such that the back work ratios achieved are characteristically quite low, and the techniques of superheat, reheat, and regeneration can be effective for increasing power plant efficiencies.

Water is less satisfactory insofar as some other desirable working fluid characteristics are concerned. For example, the critical temperature of water is only 374.14 ^{o}C, which is about 225°C below the maximum allowable turbine inlet temperatures. Accordingly, to achieve a high average temperature of heat addition and realize the attendant higher thermal efficiency, it may be necessary for the steam generator to operate at supercritical pressures. This requires costly piping and heat exchanger tubes capable of withstanding great stresses. Another undesirable characteristic of water is that its saturation pressure at ordinary condenser temperatures is well below atmospheric pressure. As a result, air can leak into the system, necessitating the use of special ejector pumps attached to the condenser or deaerating feedwater heaters to remove the air.

Although water has some shortcomings as a working fluid, no other single working fluid has been found that is more satisfactory overall for large electrical generating plants. Still, vapor power cycles intended for special uses may employ working fluids that are better matched to the application at hand than water. Cycles that operate at relatively low temperatures may perform best with a refrigerant such as ammonia as the working fluid. Power systems for high-temperature applications may employ substances having desirable performance characteristics at these temperatures. Moreover, water may be used together with some other substance in a binary vapor cycle to achieve better overall performance than could be realized with water alone.

7.2- Binary Vapor Cycles

In a binary vapor power cycle two working fluids are used, one with good high-temperature characteristics and another with good characteristics at the lower temperature end of the operating range. Fig. 7.1 shows a schematic diagram and an accompanying T-s diagram of a binary vapor cycle using water and a mercury, with each substance in both the liquid and vapor phases. In this arrangement, two ideal Rankine cycles are combined, with the heat rejection from the high-temperature cycle (the topping cycle) being used as the energy input for the low temperature cycle. This energy transfer is accomplished in an interconnecting heat exchanger, which serves as the condenser for the mercury cycle and the boiler for the water cycle. Since the increase in the specific enthalpy of the water as it passes through the heat exchanger is typically several times the magnitude of the specific enthalpy decrease of the mercury, several units of mass of mercury must circulate in the topping cycle for each unit of mass of water in the other cycle.

Binary vapor power cycles can operate with higher average temperatures of heat addition than conventional cycles using water only and thus can attain higher thermal efficiencies. However, the higher efficiencies achieved in this manner must justify the increased costs related to the construction and operation of the more complex cycle arrangement.

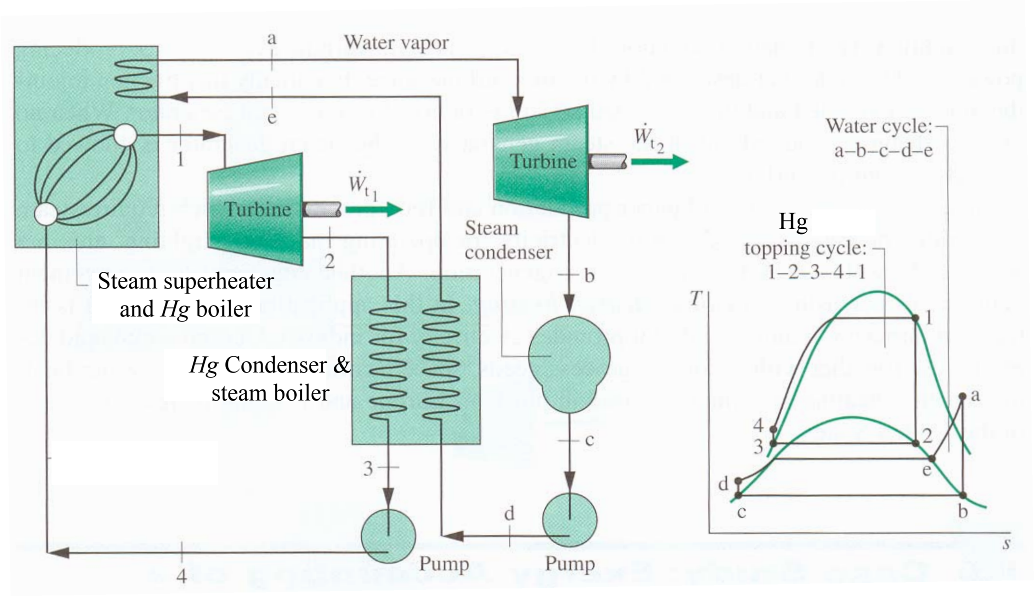

Figure 7.1 Water-metal binary

Let $\dot{m}_m$ represent the flow rate of mercury in the mercury cycle per $\dot{m}_s$ of steam circulating in the steam cycle. Then,

$$\dot{Q}_{in} = \dot{m}_m(h_1 - h_4) + \dot{m}_s(h_a - h_e) \tag{7.1}$$

$$\dot{Q}_{out} = \dot{m}_s(h_b - h_c) \tag{7.2}$$

$$\dot{W}_t = \dot{m}_m(h_1 - h_2) + \dot{m}_s(h_a - h_b) \tag{7.3}$$

$$\dot{W}_p = \dot{m}_m(h_4 - h_3) + \dot{m}_s(h_d - h_c) \tag{7.4}$$

The cycle efficiency

$$\eta_{cycle} = \frac{\dot{Q}_{in} - \dot{Q}_{out}}{\dot{Q}_{in}} = \frac{\dot{W}_t - \dot{W}_p}{\dot{Q}_{in}} \tag{7.5}$$

and, the steam rate $= \dfrac{3600}{\dot{W}_t - \dot{W}_p} \quad (kg / kW\,h)$ (7.6)

The energy balance of mercury condenser steam boiler gives

$$\dot{m}_m(h_2 - h_3) = \dot{m}_s(h_e - h_d)$$

$$\frac{\dot{m}_m}{\dot{m}_s} = \frac{h_e - h_d}{h_2 - h_3} \quad (kg\,Hg / kg\,H_2O) \tag{7.7}$$

Example 7.1

Calculate the efficiency of a binary vapor cycle shown in figure (7.1). The steam cycle operates between pressures of 3MPa and 0.004MPa, and uses a superheat temperature of 450 ^{O}C . The mercury cycle works between pressures of 1.4 MPa and 0.01MPa , the mercury entering the turbine in a dry saturated condition.

Solution

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Table 7.1 **Mercury-Hg**

P bar	T_s ^oC	v_g m^3/kg	h_f kJ/kg	h_{fg} kJ/kg	h_g kJ/kg	s_f kJ/kgK	s_{fg} kJ/kg K	s_g kJ/kg K
0.0006	109.2	259.6	15.13	297.20	312.33	0.0466	0.7774	0.8240
0.0007	112.3	224.3	15.55	297.14	312.69	0.0477	0.7709	0.8186
0.0008	115.0	197.7	15.93	297.09	313.02	0.0487	0.7654	0.8141
0.0009	117.5	176.8	16.27	297.04	313.31	0.0496	0.7604	0.8100
0.0010	119.7	160.1	16.58	297.00	313.58	0.0503	0.7560	0.8063
0.002	134.9	83.31	18.67	296.71	315.38	0.0556	0.7271	0.7827
0.004	151.5	43.29	20.93	296.40	317.33	0.0610	0.6981	0.7591
0.006	161.8	29.57	22.33	296.21	318.54	0.0643	0.6811	0.7454
0.008	169.4	22.57	23.37	296.06	319.43	0.0666	0.6690	0.7356
0.010	175.5	18.31	24.21	295.95	320.16	0.0685	0.6596	0.7281
0.02	195.6	9.570	26.94	295.57	322.51	0.0744	0.6305	0.7049
0.04	217.7	5.013	29.92	295.15	325.07	0.0806	0.6013	0.6819
0.06	231.6	3.438	31.81	294.89	326.70	0.0843	0.5842	0.6685
0.08	242.0	2.632	33.21	294.70	327.91	0.0870	0.5721	0.6591
0.10	250.3	2.140	34.33	294.54	328.87	0.0892	0.5627	0.6519
0.2	278.1	1.128	38.05	294.02	332.07	0.0961	0.5334	0.6295
0.4	309.1	0.5942	42.21	293.43	335.64	0.1034	0.5039	0.6073
0.6	329.0	0.4113	44.85	293.06	337.91	0.1078	0.4869	0.5947
0.8	343.9	0.3163	46.84	292.78	339.62	0.1110	0.4745	0.5855
1	356.1	0.2581	48.45	292.55	341.00	0.1136	0.4649	0.5785
2	397.1	0.1375	53.87	291.77	345.64	0.1218	0.4353	0.5571
3	423.8	0.09551	57.38	291.27	348.65	0.1268	0.4179	0.5447
4	444.1	0.07378	60.03	290.89	350.92	0.1305	0.4056	0.5361
5	460.7	0.06044	62.20	290.58	352.78	0.1334	0.3960	0.5294
6	474.9	0.05137	64.06	290.31	354.37	0.1359	0.3881	0.5240
7	487.3	0.04479	65.66	290.08	355.74	0.1380	0.3815	0.5195
8	498.4	0.03978	67.11	289.87	356.98	0.1398	0.3757	0.5155
9	508.5	0.03584	68.42	289.68	358.10	0.1415	0.3706	0.5121
10	517.8	0.03266	69.61	289.50	359.11	0.1429	0.3660	0.5089
12	534.4	0.02781	71.75	289.19	360.94	0.1455	0.3581	0.5036
14	549.0	0.02429	73.63	288.92	362.55	0.1478	0.3514	0.4992
16	562.0	0.02161	75.37	288.67	364.04	0.1498	0.3456	0.4954
18	574.0	0.01949	76..83	288.45	362.28	0.1515	0.3405	0.4920
20	584.9	0.01778	78.23	288.24	366.47	0.1531	0.3359	0.4890
22	595.1	0.01637	79.54	288.05	367.59	0.1546	0.3318	0.4864
24	604.6	0.01518	80.75	287.87	368.62	0.1559	0.3280	0.4839
26	613.5	0.01416	81.89	287.70	369.59	0.1571	0.3245	0.4816
28	622.0	0.01329	82.96	287.54	370.50	0.1583	0.3212	0.4795
30	630.0	0.01252	83.97	287.39	371.36	0.1594	0.3182	0.4776
35	648.5	0.01096	86.33	287.04	373.37	0.1619	0.3115	0.4734
40	665.1	0.00978	88.43	286.73	375.16	0.1641	0.3056	0.4697
45	680.3	0.00885	90.35	286.44	376.97	0.1660	0.3004	0.4664
50	694.4	0.00809	92.11	286.18	378.29	0.1678	0.2958	0.4636
55	707.4	0.00746	93.76	285.93	379.69	0.1694	0.2916	0.4610
60	719.7	0.00693	95.30	285.70	381.00	0.1709	0.2878	0.4587
65	731.3	0.00648	96.75	285.48	382.23	0.1723	0.2842	0.4565
70	742.3	0.00609	98.11	285.28	383.40	0.1736	0.2809	0.4545
75	752.7	0.00575	99.42	285.08	384.50	0.1748	0.2779	0.4527

Example 7.2

In a mercury – steam binary vapor cycle, the mercury leaves the mercury boiler dry – saturated at 1 MPa and is condensed in the mercury condenser at 0.02 MPa. dry saturated steam leaves the steam boiler at 4 MPa , and is heated to 400 ^{o}C in a superheated placed in the flue – gas passes of the mercury boiler .

The steam is supplied to a regenerative steam cycle in which the condenser pressure is 0.004 MPa. The feed water from the condenser is hated to the corresponding saturation heating stages, and is then farther heated to the steam saturation temperature by being passed through an mercury boiler. The isentropic efficiency of the mercury and the steam turbines are 75% and 80% respectively the feed pump work may be neglected.

Calculate

a. the ratio of the mass flow rates of mercury and steam through the condenser – boiler
b. the η_{cy} of the binary cycle .

Solution

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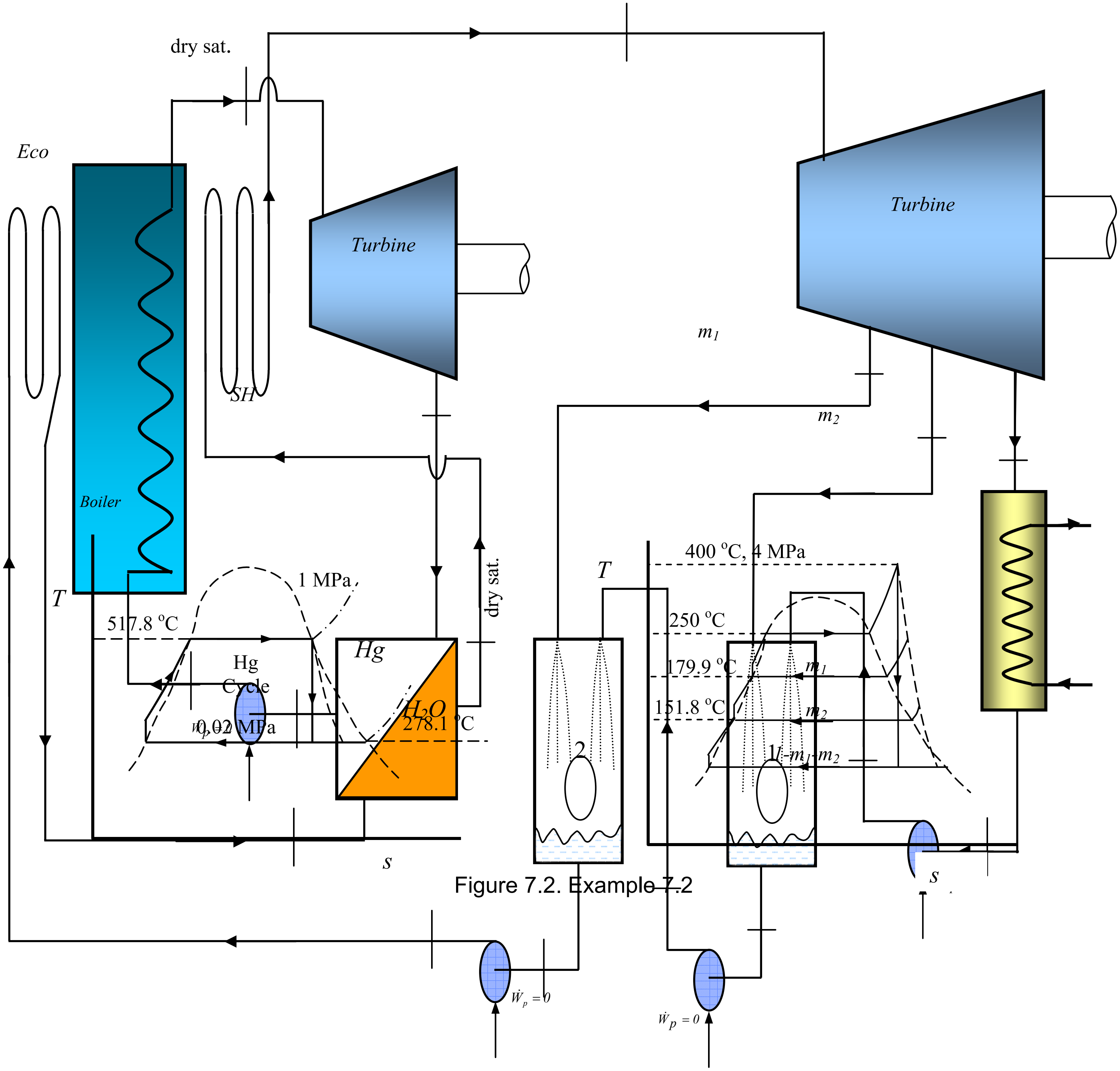

Figure 7.2. Example 7.2

7.3-Coupled cycles

The mercury steam cycle represents the two-fluid cycles where two Rankine cycles have been coupled in series. The mercury cycle is called the topping cycle and the steam cycle is called the bottoming cycle. If a sulphur dioxide cycle is added to it in the low temperature range, so that the heat released during the condensation of steam is utilized in forming sulphur dioxide vapor which expands in another turbine, then the mercury-steam-sulphur dioxide cycle is a three fluid or tertiary cycle. Similarly, other liquid metals, apart from mercury, like sodium or potassium, may be considered for a working fluid in the topping cycle. Apart from SO_2 other refrigerants (ammonia, freons etc.) may be considered as working fluids for the bottoming cycle. Figure 7.3 shows a sodium-mercury-steam tertiary cycle on T-s coordinates, having thus three cycles coupled in series.

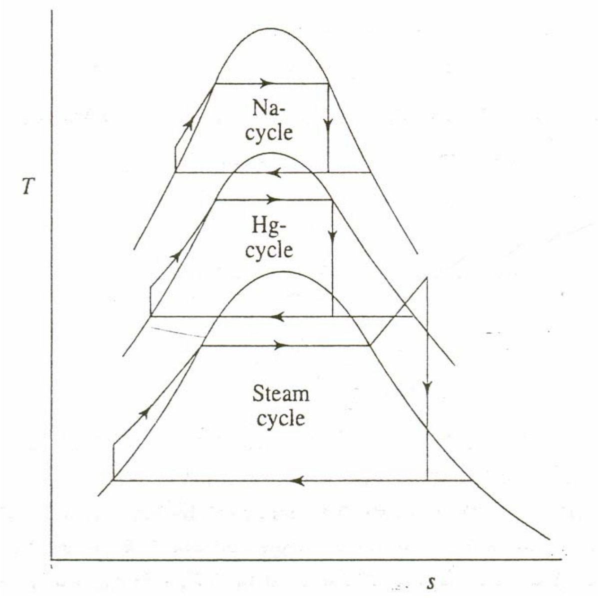

Figure 7.3 Sodium-mercury-steam cycle

Thermodynamics of Rankine-Rankine Coupled Cycles

If two Rankine cycles with two different working fluids are coupled in series, the heat lost by one is absorbed by the other (Fig. 7.4), as in the mercury-steam binary cycle. Let η_1 and η_2 be the efficiencies of the topping and bottom cycles, respectively, and η be the overall efficiency of the combined cycle.

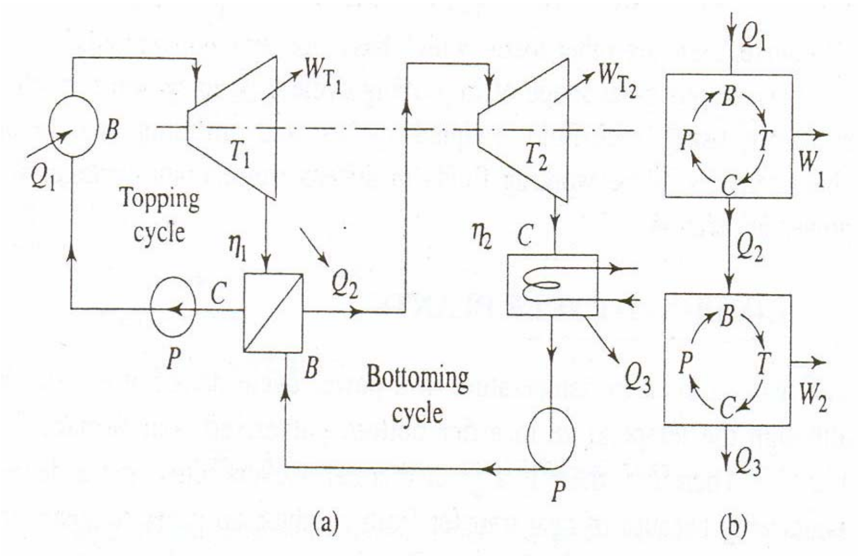

Figure 7.4 Two vapor power cycles coupled in series

$$\eta_1 = 1 - \frac{\dot{Q}_2}{\dot{Q}_1} \;\; and \;\; \eta_1 = 1 - \frac{\dot{Q}_3}{\dot{Q}_2}$$

or $$\dot{Q}_2 = \dot{Q}_1 (1 - \eta_1) \;\; and \;\; \dot{Q}_3 = \dot{Q}_2 (1 - \eta_1)$$

Now, $$\eta_1 = 1 - \frac{\dot{Q}_3}{\dot{Q}_2} = 1 - \frac{\dot{Q}_2 (1 - \eta_2)}{\dot{Q}_1} = \frac{\dot{Q}_1 (1 - \eta_1)(1 - \eta_2)}{\dot{Q}_1}$$

$\therefore$ $$\eta = 1 - (1 - \eta_1)(1 - \eta_2) \tag{7.8}$$

or $$1 - \eta = (1 - \eta_1)(1 - \eta_2) \tag{7.9}$$

For n cycles coupled in series, the overall efficiency would be given by

$$1 - \eta = (1 - \eta_1)(1 - \eta_2)......(1 - \eta_n)$$

or $$1 - \eta = \overset{n}{\underset{i=1}{\Pi}} (1 - \eta_i) \tag{7.10}$$

$\therefore$ Total loss = product of losses in all the cycles in series

For two-cycles coupled in series, from Eq. (7.8),

$$\eta = 1 - (1 - \eta_1 - \eta_2 + \eta_1 \eta_2)$$

$$\eta = \eta_1 + \eta_2 - \eta_1 \eta_2 \tag{7.11}$$

Thus, the overall efficiency of two cycles coupled in series equals the sum of the individual efficiencies minus their product.

By combining two cycles in series, even if individual efficiencies are low, it is possible to achieve a fairly high combined efficiency, which cannot be attained by a single cycle. For example, if η_1 = 0.50 and η_2 = 0.40.

$$\eta = 0.5 + 0.4 - 0.5 \times 0.4 = 0.7$$

It is almost impossible to achieve such a high efficiency in a single cycle.

For sodium-mercury-steam tertiary cycle, if η_1 = 0.5. η_2 = 0.4 and η_3 = 0.4, from equation (7.8),

$$\eta = 1 - (1 - \eta_1)(1 - \eta_2)(1 - \eta_3) = 1 - 0.5 \times 0.6 \times 0.6 = 0.82$$

Of course, there are other losses which have not been considered. There is thus great scope of improving cycle efficiency and achieving fuel economy using multi-fluid coupled cycles and utilizing the favorable characteristics of the working fluids in different temperature ranges for bulk power generation.

Example 7.3 (p. 165)

For a mercury-steam-cycle sulphur dioxide cycle, the heat rejected in the mercury cycle is given to the steam cycle and the heat rejected in the steam cycle is utilized in the SO2 cycle. If the efficiencies of the mercury, steam and SO_2 cycles are 0.5, 0.4 and 0.25, respectively, find the overall efficiency of the composite cycle.

Solution

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7.4-Combined Gas Turbine-Vapor Power Cycle

A combined power cycle couples two power cycles such that the energy discharged by heat transfer from one cycle is used partly or wholly as the input for the other cycle. In the present section, a combined gas turbine-vapor power cycle is considered.

The stream exiting the turbine of a gas turbine is at a high temperature. One way the potential (exergy) of this high-temperature gas stream can be used, thereby improving overall fuel utilization, is by a regenerator that allows the turbine exhaust gas to preheat the air between the compressor and combustor (Chapter III). Another method is provided by the combined cycle shown in Fig. 7.5, involving a gas turbine cycle and a vapor power cycle. The two power cycles are coupled so that the heat transfer to the vapor cycle is provided by the gas turbine cycle, which may be called the ***topping cycle***.

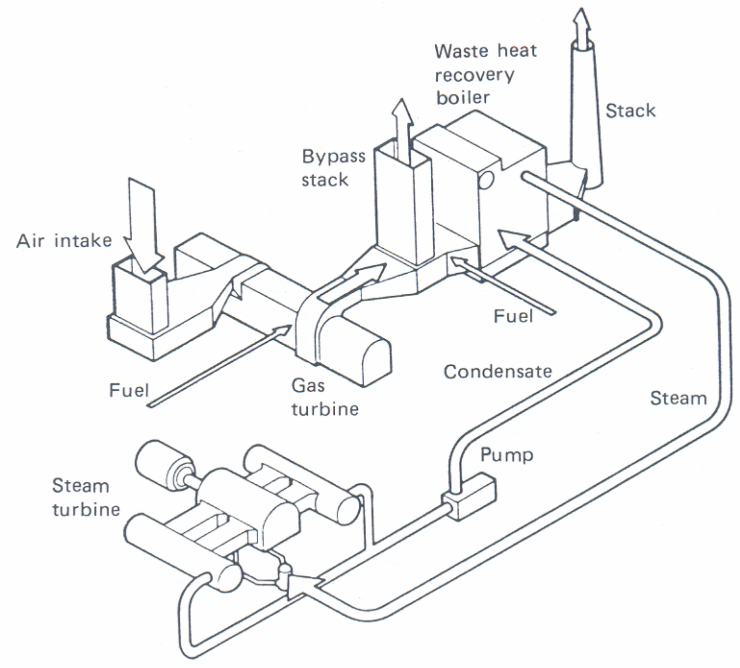

Figure 7.5 A simplified diagram for a combined-cycle plant

The combined cycle has the gas turbine's high average temperature of heat addition and the vapor cycle's low average temperature of heat rejection, and thus a thermal efficiency greater than either cycle would have individually. For many applications combined cycles are economical, and they are increasingly being used worldwide for electric power generation.

With reference to Fig. 7.6, the thermal efficiency of the combined cycle is

$$\eta = \frac{\dot{W}_{gas} + \dot{W}_{vap}}{\dot{Q}_{in}} \qquad (7.12)$$

where $\dot{W}_{gas}$ is the net power developed by the gas turbine and w is the net power developed by the vapor cycle. e,, denotes the total rate of heat transfer to the combined cycle, including additional heat transfer, if any, to superheat the vapor entering the vapor turbine. The evaluation of the quantities appearing in Eq. 7.12 follows the procedures described in the sections on vapor cycles and gas turbines.

The relation for the energy transferred from the gas cycle to the vapor cycle for the system of Fig. 7.6 is obtained by applying the mass and energy rate balances to a control volume enclosing the heat exchanger. For steady-state operation, negligible heat transfer with the surroundings, and no significant changes in kinetic and potential energy, the result is

$$\dot{m}_v\left(h_7 - h_6\right) = \dot{m}_g\left(h_4 - h_5\right) \tag{7.13}$$

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

As witnessed by relations such as Eqs. 7.12 and 7.13, combined cycle performance can

be analyzed using mass and energy balances. To complete the analysis, however, the second law is required to assess the impact of irreversibilities and the true magnitudes of losses. Among the irreversibilities, the most significant is the exergy destroyed by combustion. About 30% of the exergy entering the combustor with the fuel is destroyed by combustion irreversibility. An analysis of the gas turbine on an air-standard basis does not allow this exergy destruction to be evaluated, however, and means introduced in new editions of thermodynamics text books must be applied for this purpose.

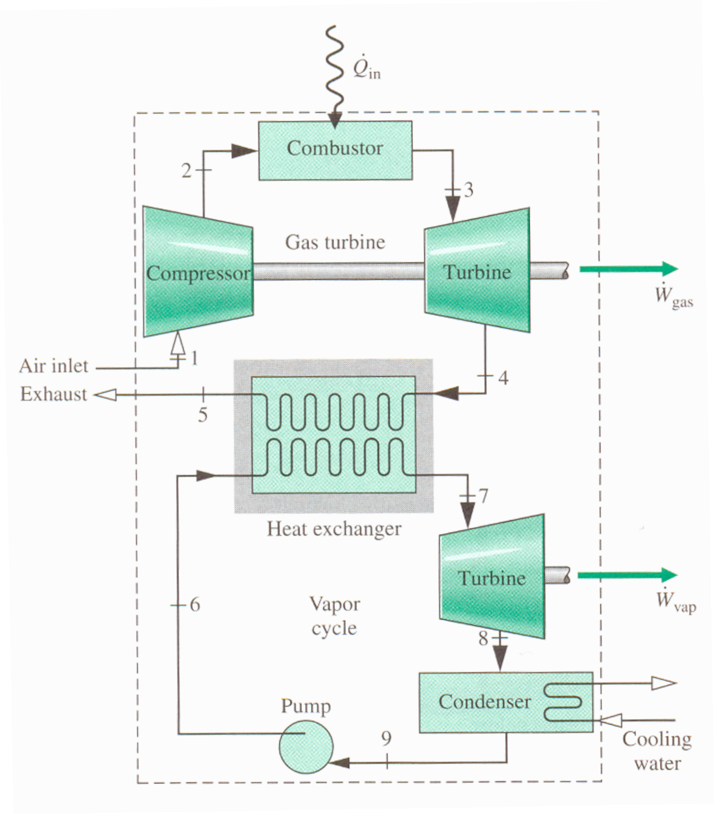

Figure 7.6 Combined Gas turbine-Vapor power plant

Example 7.4

A combined gas turbine-vapor power plant as shown in Fig. 7.6 has a net power output of 45 MW. Air enters the compressor of the gas turbine at 100 kPa, 300 K, and is compressed to 1200 kPa. The isentropic efficiency of the compressor is 84%. The condition at the inlet to the turbine is 1200 kPa, 1400 K. Air expands through the turbine, which has an isentropic efficiency of 88%, to a pressure of 100 kPa. The air then passes through the interconnecting heat exchanger and is finally discharged at 400 K. Steam enters the turbine of the vapor power cycle at 8 MPa, 400°C, and expands to the condenser pressure of 8 kPa. Water enters the pump as saturated liquid at 8 kPa. The turbine and pump of the vapor cycle have isentropic efficiencies of 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_o = 300 K, P_o = 100 kPa.

Solution

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7.5- Exergy analysis of Combined Cycles

The exergy of a system is the maximum work obtainable as the system comes to equilibrium with the surroundings (Kotas, 1995), The higher the value of exergy, more is the work obtainable from the system. The First Law of Thermodynamics makes only an energy balance of a system or a control volume. It does not make any distinction of different forms of energy, particularly between work and heat, or heat (internal energy) available at different temperatures. It is the Second Law which asserts that from engineering viewpoint, all forms of energy are not of the same quality. Energies of two systems may be quantitatively equal, but qualitatively they may be different. When steam is adiabatically throttled, its energy does not change, but its quality degrades. Exergy is a measure of energy quality and exergetic (or second law) efficiency is a measure of the perfectness of a thermal system. While energy of a system in any process remains constant, a part of its exergy is always destroyed.

For dry solid fossil fuels the ratio φ of chemical exergy ε_o to the net calorific value (NCV) with a mass ratio of oxygen to carbon (*o/c*) less than 0.667 is given by Kotas, 1995.

$$\frac{\varepsilon_o}{NCV} = \varphi = 1.0437 + 0.1882\left(\frac{h}{c}\right) + 0.061\left(\frac{o}{c}\right) + 0.0404\left(\frac{n}{c}\right) \qquad (7.14)$$

where c, h, o and n are the mass fractions of carbon, hydrogen, oxygen and nitrogen, respectively.

For *solid fuels* with the mass ratio 2.67 > $\left(\frac{o}{c}\right)$ > 0.667,

$$\varphi = \frac{1.0438 + 0.1882\left(\frac{h}{c}\right) - 0.2509\left\{1 + 0.7256\left(\frac{h}{c}\right)\right\} + 0.0383\left(\frac{n}{c}\right)}{1 - 0.3035\left(\frac{o}{c}\right)} \qquad (7.15)$$

For *liquid fuels* the effect of sulphur was included in the correlation

$$\varphi = 1.0401 + 0.1728\left(\frac{h}{c}\right) + 0.0432\left(\frac{o}{c}\right) + 0.2169\left(\frac{s}{c}\right)\left\{1 - 2.0628\left(\frac{h}{c}\right)\right\} \qquad (7.16)$$

For *gaseous fuels* of a given composition, the chemical exergy can be calculated from

$$\varepsilon_o = -\Delta h_o + T_o \Delta s_o + RT_o\left[x_o \,\ell n \frac{P_{o2}}{P_o} - \sum x_k \,\ell n \frac{P_k}{P_o}\right] \qquad (7.17)$$

where the subscript k refers to the components of the products of composition. Typical values of φ for some industrial fuels are given in Chapter II, Table 2.1.

Now, for a chemically reactive system

$\varepsilon_o = -\Delta g_o$ = the change in the standard Gibbs function change (kJ/k mol)

and $\Delta h = -(NCV)$ = the change in the enthalpy of formation (kJ/k mol). Now, from equation (7.14)

$$\varphi = \frac{-\Delta G_o}{-\Delta H_o}$$

and $\Delta H_o = \dot{m}_f \times NCV \qquad (kW)$ (7.18)

where $\dot{m}_f$ is the fuel burning rate (kg/s).

Therefore, ΔG_o can be estimated.

Again,

$$\Delta G_o = \Delta H - T_o \Delta S_o$$

$T_o \Delta S_o$ = rate of exergy loss in combustion or reaction

$$= -\dot{m}_f \times NCV + \varphi\left(\dot{m}_f \times NCV\right) = \dot{m}_f \times NCV(\varphi - 1) \qquad (7.19)$$

Various amounts of lost work or exergy loss due to irreversibility can be estimated for each component in the plant as given below. The combined cycle can be assumed as given in Fig. 7.7

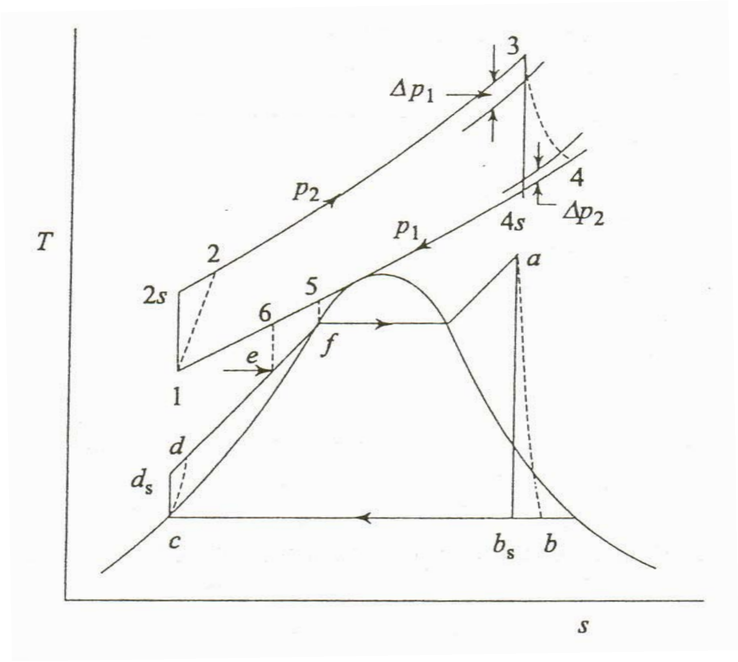


Figure 7.7 Combined GT-ST cycle for exergy analysis

***Compressor*:** Rate of energy dissipation or irreversibility in the compressor,

$$\dot{I}_{comp} = \dot{m}_a \, T_o \left(s_2 - s_1\right)$$

$$= \dot{m}_a \, T_o \left[c_{pa} \, \ell n \left(\frac{T_2}{T_1} \right) - R_a \, \ell n \left(\frac{P_2}{P_1} \right) \right] \qquad (7.20)$$

where $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{(k \eta_C)}}$, η_c the compressor isentropic efficiency

Combustor. The rate of energy dissipation due to combustion,

$$\dot{I}_{comb} = T_o \left[\left(S_p\right)_3 - \left(S_R\right)_2 \right] \qquad (7.21)$$

where $\left(S_R\right)_2 = \left(S_A\right)_2 + \left(S_F\right)_2$ and the subscripts P, R, A, and F represent products, reactants, air and fuel, respectively.

$$\therefore \qquad \dot{I}_{comb} = T_o \left[\left(S_p\right)_3 - \left(\left(S_A\right)_2 + \left(S_F\right)\right)_2 \right]$$

$$\dot{I}_{comb} = T_o \left\{ \left[\left(S_p\right)_3 - \left(S_p\right)_o \right] + \left(S_p\right)_o - \left[\left(S_A\right)_2 - \left(S_A\right)_o + \left(S_F\right)_o + \left(S_A\right)_o \right] \right\}$$

Now,

$$\Delta S_o = \left(S_p\right)_o - \left[\left(S_F\right)_o + \left(S_A\right)_o \right]$$

$$\dot{I}_{comb} = T_o \left\{ \left[\left(S_p\right)_3 - \left(S_p\right)_o \right] - \left[\left(S_A\right)_2 - \left(S_A\right)_o + \Delta S_o \right] \right\}$$

$$\dot{I}_{comb} = T_o \left\{ \left[\dot{m}_g \, c_{pg} \, \ell n \left(\frac{T_3}{T_o} \right) - \dot{m}_g \, R_g \, \ell n \left(\frac{P_3}{P_o} \right) \right] - \dot{m}_a \, c_{pa} \, \ell n \left(\frac{T_2}{T_o} \right) - \dot{m}_a \, R_a \, \ell n \left(\frac{P_2}{P_o} \right) + \Delta S_o \right\} \qquad (7.22)$$

where $T_o \Delta S_o$ is given in equation 7.19.

Gas Turbine: rate of exergy loss in the gas turbine

$$\dot{I}_{gt} = \dot{m}_g \, T_o \left(s_4 - s_3\right)$$

$$= \dot{m}_g \, T_o \left[c_{pg} \, \ell n \left(\frac{T_4}{T_3} \right) - R_g \, \ell n \left(\frac{P_4}{P_3} \right) \right] \qquad (7.23)$$

where $\frac{T_4}{T_3} = \left(\frac{P_4}{P_3} \right)^{\frac{(k-1)\eta_t}{k}}$, η_t the turbine isentropic efficiency

HRSG: Rate of exergy loss in the heat recovery steam generator,

$$\dot{I}_{HRSG} = T_o \left[\dot{m}_s \left(s_a - s_e\right) + \dot{m}_g \left(s_6 - s_4\right) \right]$$

$$= T_o \left[\dot{m}_s (s_a - s_e) + \dot{m}_g \left(c_{pg} \, \ell n \frac{T_6}{T_4} - R_g \ell n \frac{P_6}{P_4} \right) \right] \quad (7.24)$$

Exhaust loss: Rate of exergy loss due to exhaust flue gases.

$$\dot{I}_{exhaust} = \int_{T_6}^{T_O} \left(1 - \frac{T_o}{T} \right) dQ$$

$$= \dot{m}_g \, c_{pg} \left[(T_6 - T_o) - T_o \, \ell n \frac{T_6}{T_o} \right] \quad (7.25)$$

Steam turbine: Rate of exergy loss in the steam turbine,

$$\dot{I}_{st} = \dot{m}_s \, T_o \, (s_b - s_a) \quad (7.26)$$

Condenser: Rate of exergy loss in the condenser,

$$\dot{I}_{cond} = \dot{m}_s \, T_o (s_c - s_b) + \frac{T_o}{T_2} \dot{Q}_2 + \dot{Q}_o \quad (7.27)$$

where $\dot{Q}_2$, is the rate of heat rejection from the condenser at temperature T_2 and $\dot{Q}_o$ is the rate of heat loss to the environment.

Feed pump: Rate of exergy loss in the feed pump,

$$\dot{I}_{FP} = \dot{m}_s \, T_o \, (s_d - s_c) \quad (7.28)$$

Rate of exergy losses in the feedwater heaters, cooling towers and so on can similarly be estimated. Total rate of exergy losses in the plant,

$$\sum \dot{I} = \dot{I}_{comp} + \dot{I}_{comb} + \dot{I}_{gt} + \dot{I}_{HRSG} + \dot{I}_{exhaust} + \dot{I}_{st} + \dot{I}_{cond} + \dot{I}_{FP} \quad (7.29)$$

Total rate of exergy input = $-\Delta G_o = -\Delta H_o \times \varphi$

Total rate of power output = $W_{gt} + W_{st}$

Total rate of exergy input =Total rate of exergy output (power output) + total rate of exergy loss = ΔG_o

Exergetic or second law efficiency = $\dfrac{W_{st} + W_{gt}}{\Delta G_o}$

The advantage of exergy analysis is that it provides the locations where the exergy losses or irreversibilities are more. Ways and means can then be explored, say, by thermodynamically optimizing the operating or geometrical parameters, so as to reduce these losses to the practical minimum values.

Example 7.5

The following data refer to a combined cycle power plant

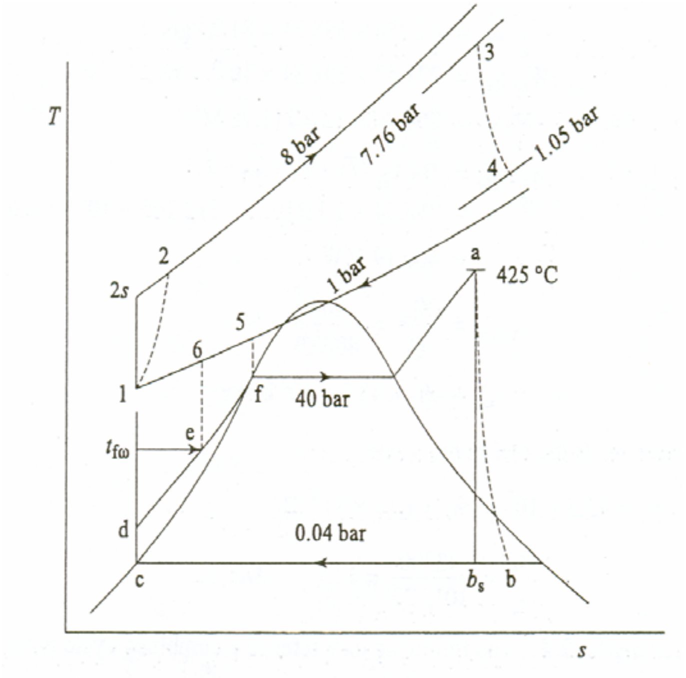
Inlet condition of air to the compressor	1 bar, 25 °C
Pressure ratio of the compressor	8
Maximum gas temperature at inlet to the gas turbine	900 °C
Pressure drop in the combustion chamber	3%
Efficiency of the compressor	0.88
Efficiency of the gas turbine	0.88
Calorific value of liquid octane (C_8H_{18}) used as fuel	44430 kJ/kg
Specific heat of air	1.006 kJ/kg K
Specific heat of gas	1.148 kJ/kg K
Specific heat ratio of gas	1.333
Specific heat ratio of air	1.4
Condition of steam at inlet to the steam turbine	40 bar, 425 °C
Condenser pressure	0.04 bar
Feed water temperature to the $HRSG$	170.4 °C
Efficiency of the steam turbine	0.82
Pressure drop of gas in the $HRSG$	5kPa
Steam flow rate	29.235 kg/s

Assume $\varphi = \dfrac{\Delta G_o}{\Delta H_o} = 1.0401 + 0.1728\left(\dfrac{h}{c}\right)$

where $\left(\dfrac{h}{c}\right)$ is the mass ratio of hydrogen to carbon in the fuel.

(a) Determine the total power output and the overall efficiency of the combined cycle plant and estimate the "lost heat" coefficient x_l.

(b) Give an energy balance of the plant and estimate the exergetic efficiency.



Solution

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Revision Questions

YANBU INDUSTRIAL COLLEGE

Power Plant Engineering and economy

MET401

Department of Mechanical Engineering Technology

Chapter VIII- Nuclear Power Plants

2010

PhD, MEMgt, BSc(hons) , P.E., MASME, MIPENZ, Reg. Eng

Chapter VIII

Nuclear Power Plants

8.1- Introduction

The unit cost per kWh of electricity generation in nuclear power plants is now comparable to or even lower than the unit cost in coal-fired power plants in most parts of the world. In addition, the problems associated with environmental pollution, mine safety, fuel transportation and so on are much less severe in nuclear power stations. Nuclear power utilization can help save a considerable amount of fossil fuels which can be used in other areas of utility.

In recent years, a strong public opinion has grown against the use of nuclear energy for power generation due to the problems related to nuclear safety, radioactive waste disposal and nuclear weapons proliferation. Despite these difficulties, the future of large capacity electricity generation includes nuclear energy as one of the main sources. In many countries like France, Japan, the U.K. and Russia, the bulk of the electricity is produced in nuclear power plants.

8.2- Structure of the atom

All matter is composed of unit particles called atoms. An atom consists of a relatively heavy, positively charged nucleus and a number of much lighter negatively charged, electrons orbiting around the nucleus. The nucleus consists of protons and neutrons, which together are called nucleons. Protons are positively charged, while the neutrons are electrically neutral. The electric charge on the proton is equal in magnitude but opposite in sign to that on an electron. The atom as a whole is electrically neutral, since the number of protons is equal to the number of electrons in orbit.

The number of protons in the nucleus is called the atomic number, Z. The total number of nucleons in the nucleus is called the mass number, A which is the integer nearest the mass of the nucleus. Nuclear symbols are written conventionally as ${}_{Z}X^{A}$.

where X is the usual chemical symbol. Most of the weight of an atom is concentrated in the nucleus. The radius of a nucleus is of the order of 10^{-16} m and that of a atom is 10^{-11} m. Figure 8.1 illustrates the atomic structure of some simple atoms. The masses of the three primary atomic sub-particles are:

Neutron mass, m_n = 1.008665 amu = 1.674 x 10^{-27} kg

Proton mass, m_p = 1.007277 amu = 1.673 x 10^{-27} kg

Electron mass, m_e = 0.0005486 amu = 9.109 x 10^{-31} kg

The *atomic mass unit*, amu, is a unit of mass approximately equal to 1.66 x 10^{-27} kg.

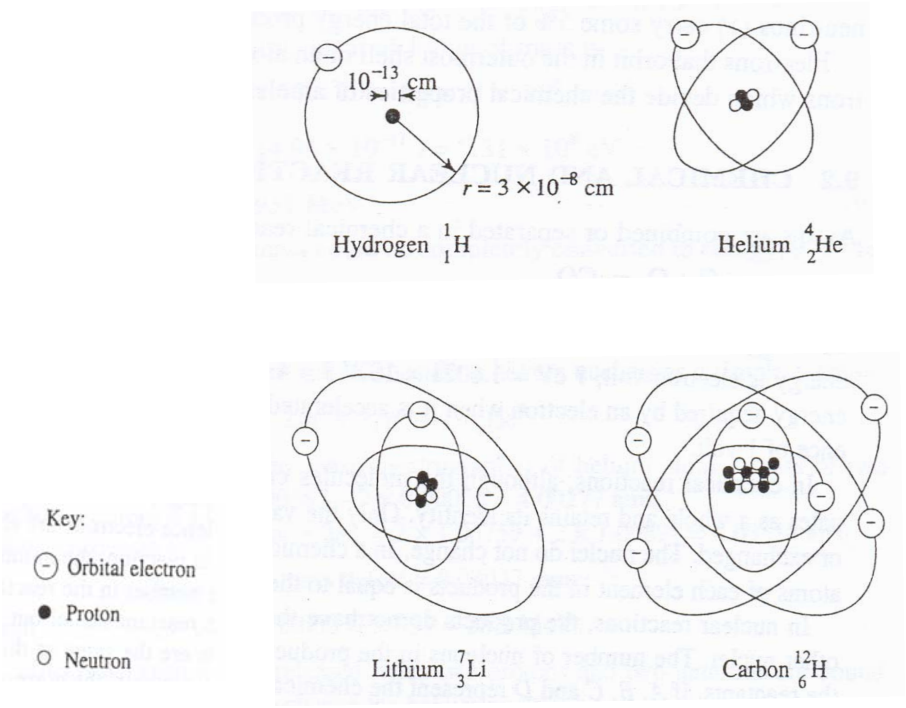

Figure 8.1 Structure of the atom

Hydrogen $\left({}_1H^1\right)$ has a nucleus composed of one proton, no neutron, and one orbital electron (Z = 1, A =1). It is the only atom that has no neutron. Deuterium, frequently called *heavy hydrogen* $\left({}_1H^2\right)$ has one proton and one neutron in its nucleus and one orbital electron ($Z = 1, A = 2$). Helium $\left({}_2He^4\right)$ has two protons, two neutrons and two electrons ($Z= 2, A= 4$).

Atoms with nuclei having the same number of protons have similar chemical and physical properties and differ mainly in their masses. They are called isotopes. For example, deuterium, often called heavy hydrogen, is an isotope of hydrogen. When combined with oxygen, ordinary hydrogen and deuterium form ordinary water (H_2O) and heavy water (D_2O) respectively.

Natural uranium is composed of 99.282 % U^{238}, 0.712 % U^{235} and 0.006 % U^{234}, the atomic number being 92 in all cases. Many isotopes do not appear in nature and are synthesized in the laboratory or in nuclear reactors.

Two other particles are of importance, *viz*., the *positron* and *neutrino*. The positron is a positively charged electron having the symbols ${}_{+1}e^0, e^+$ or β^+, the symbol for electron being ${}_{-1}e^0, e^-$ or β^-. The neutrino is a tiny, electrically neutral particle, ejected along b particle during nuclear fission. The ejected neutrinos (V) carry 5% of the total energy produced in fission.

Electrons that orbit in the outermost shell of an atom are called valence electrons which decide the chemical properties of an element.

8.3-Nuclear Fusion and Fission

Nuclear energy of importance in energy production are *fission, fusion* and *radioactivity*. Radioactivity is an important source of energy for small devices and a source of radiation for use in research, industry, medicine, and a wide variety of applications, as well as an environmental concern. In fusion, two or more light nuclei fuse to form a heavier nucleus. In fission, a heavy nucleus is split into two or more lighter nuclei.

8.3.1-Nuclear Fission

Fission can be caused by bombarding with high energy neutrons. Neutrons are most suitable for fission because they are electrically neutral and thus require no high K.E. to overcome electrical repulsion from positively charged nuclei. Two or three neutrons are usually released for each one absorbed in fission, and thus keep the reaction going as shown in Figure 8.2. The speed of the released neutrons is about 1600 km/s and with an initial K.E. of 2 MeV. To slow down the neutron, a *moderator* can be used. The ${}_{92}U^{238}$ atom having 92 protons and 146 neutrons, where as ${}_{92}U^{235}$ atom having 92 protons and 143 neutrons can be fissioned even by low energy neutrons. Isotopes like U-233, U-235 and Pu-239 can be fissioned by neutrons of all energies, whereas isotopes U-238, Th-232 and Pu-240 are fissionable by high energy (14 MeV[*]) only.

$$ {}_{92}U^{235} + {}_{0}n^{1} \rightarrow {}_{54}Xe^{140} + {}_{38}Sr^{94} + 2\,{}_{0}n^{1} \qquad (8.1)$$

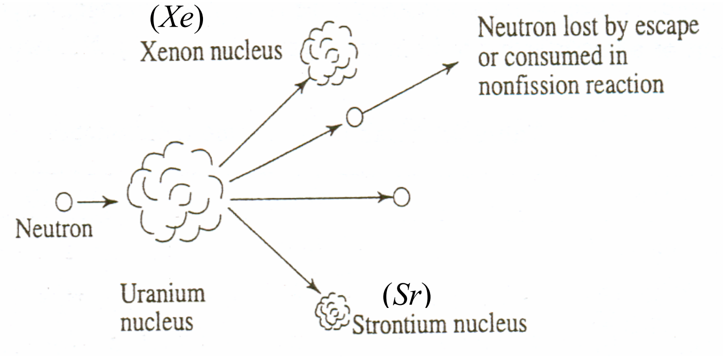

Figure 8.2. A fission reaction

The immediate (prompt) products of a fission reaction, such as Xe^{140} and Sr^{94} are called fission fragments, which along with other decay products (α, β, γ : etc.) are called *fission products.*

When a neutron collides with and is absorbed by a fissionable nucleus, the latter is transformed into a compound nucleus in an excited state, e.g.,

$$ {}_{92}U^{235} + {}_{0}n^{1} \rightarrow \left[{}_{92}U^{236}\right] \rightarrow fission $$

---

[*] MeV :- Million electron Volts
1 ev = 1.6021 x 10^{-19} J = 4.44 x10^{-26} KWh

which then undergoes fission. If the excitation energy is not sufficiently large the nucleus may not undergo fission and may emit only γ radiation or eject a particle. Such absorption of a neutron in a non-fission reaction occurs about 16% of the time in all neutron absorptions by U-235.

8.3.2-Nuclear Fusion

Energy is produced in the sun and stars by continuous fusion reactions in which four nuclei of hydrogen fuse in a series of reactions involving other particles that continually appear and disappear in the course of the reactions, such as He^3, nitrogen, carbon, and other nuclei, but culminating in one nucleus of helium and two positrons

$$4\left({}_{1}H^{1}\right)\rightarrow {}_{2}He^{4} + 2\,{}_{+1}e^{0} \qquad (8.2)$$

resulting in a decrease in mass of about 0.0276 amu, corresponding to 25.7 MeV. The heat produced in these reactions maintains temperatures of the order of several million degrees in their cores and serves to trigger and sustain succeeding reactions.

On earth, although fission preceded fusion in both weapons and power generation, the basic fusion reaction was discovered first, in the 1920s, during research on particle accelerators. Artificially produced fusion may be accomplished when two light atoms fuse into a larger one as there is a much greater probability of two particles colliding than of four. The 4-hydrogen reaction requires, on an average, billions of years for completion, whereas the deuterium-deuterium reaction requires a fraction of a second.

To cause fusion, it is necessary to accelerate the positively charged nuclei to high kinetic energies, in order to overcome electrical repulsive forces, by raising their temperature to hundreds of millions of degrees resulting in a plasma. The plasma must be prevented from contacting the walls of the container, and must be confined for a period of time (of the order of a second) at a minimum density. Fusion reactions are called *thermonuclear* because very high temperatures are required to trigger and sustain them.

Many problems have to be solved before an artificially made fusion reactor be comes a reality. The most important of these are the difficulty in generating and maintaining high temperatures and the instabilities in the medium (plasma), the conversion of fusion energy to electricity, and many other problems of an operational nature.

8.4- Energy from nuclear reaction

The energy corresponding to the change in mass in a nuclear reaction is calculated from Einstein's law

$$\Delta E = \Delta m\,C^{2} \qquad (8.3)$$

Where E =energy, J, C = velocity of light = 3 x 10^8 m/s, and Δm = mass defect, kg

The energy associated with the mass defect is known as the binding energy (BE) of the nucleus. It acts as "glue" which binds the protons and neutrons together in the nucleus. The energy equivalent of 1 g of mass is

$$\Delta E = 1\times 10^{-3}\ \text{kg} \times (3\times10^{8}\ \text{m/s})^{2} = 9\times 10^{13}\ \text{J}$$

Similarly, the energy equivalent of 1 amu of mass is (see section 8.2)

ΔE = 1.66 x 10^{-27} kg x $(3\text{x}10^8 \text{ m/s})^2$ =14.94 x 10^{-11} J

= 9.31 x 10^8 eV= **931 MeV** (8.4)

Therefore, if 1 amu of mass could be completely converted to energy, 931 MeV.

8.5- Energy from fission

There are many fission reactions which release different amounts of energy, For the reaction,

$$ {}_{92}U^{235} + {}_{0}n^{1} \rightarrow {}_{56}Ba^{137} + {}_{36}Kr^{97} + 2\,{}_{0}n^{1} \qquad (8.5)$$

Ba: Barium, Kr: Krypton

has the mass balance

235.0439 + 1.00867 $\rightarrow$ 138.9061 + 96.9212 + 2 x 1.00867
or, 236.0526 $\rightarrow$ 235.8446 amu

There is thus a reduction in mass, which appears in the form of energy (exothermic).The mass defect is

Δm = 235.8446 - 236.0526 = - 0.2080 amu

Therefore,

ΔE = (-2080) x 931= -193.6 MeV

The fission of U-235 yields on an average about 193 MeV. which is the same for the fission of U^{233} and Pn^{239}. This amount of energy is prompt, i.e. released during the fission process. More energy is, however, produced due to (i) the slow decay of the fission fragments, and (ii) the non-fission capture of excess neutrons. The total energy released per fission reaction is about 200 MeV. The complete fission of 1 g of U-235 nuclei thus produces

$$\frac{Avogadro\ Constant}{U^{235}\ Isotope\ mass} \times 200\,MeV = \frac{6.023\times 10^{23}}{235.0439} \times 200\,MeV = 5.126\times 10^{23}\ MeV$$
$$= 8.19\times 10^{10}\,J = 2.276\times 10^{4}\ kWh = 0.948\,MW.day$$

Thus, a reactor burning **1g** of U^{235} generates nearly **1MW-day** of energy. This is referred to by the term "*fuel burnup*", which is the amount of energy in MW-day's produced of each metric ton of fuel.

The complete fission of all U^{235} nuclei in a fuel mass is impossible, since many of the fission products capture neutrons in a non-fission reaction. In time, the number of neutrons so captured becomes great enough because of the accumulation of products, and the fission chain can no longer be sustained.

Depending upon fuel enrichment, this happens when only a small percentage, often less than 1% of the fissionable nuclei in the fuel has been consumed. Further use of this poisoned fuel can only be made by removing the fission products and reprocessing.

8.6- Neutron energies

Because neutrons are essential to the fission process, this and subsequent sections will deal with them and their interactions. As any other body, the kinetic energy of a neutron, KE_n is given by

$$KE_n = \frac{1}{2} m_n V_n^2 \qquad \text{(J)} \qquad (8.6)$$

where $m_{n,}$ kg is the mass of neutron (1.008665) (see section 8.2), and V_n (m/s) the speed of neutron (see table 4.1).

$$KE_n = \frac{1}{2}\left(1.674 \times 10^{-27}\right) \times V^2 \qquad \text{(J)}$$

1 ev = 1.6021 x 10^{-19} J

Thus

$$KE_n = 5.227 \times 10^{-9}\, V^2 \qquad \text{(eV)} \qquad (8.7)$$

The newly born fission neutrons have energies varying between 0.075 to 17 MeV. As these neutrons travel through matter, they collide with other nuclei and get slowed down. This process is called *scattering*. The neutron gives up some of its energy with each successive collision.

Table 8.1 Neutron three general velocity categories

Classification	Neutron energy	Corresponding velocity (m/s)
Fast	> 10^5	> 4.4 x 10^6
Intermediate	1-10^5	(1.38-4.4) x 10^6
Slow	< 1	< 1.38 x 10^4

Newly born fission neutrons carry, on an average, about 2% of a reactor fission energy in the form of KE. As stated earlier, they can be:

(1) Prompt neutrons, emitted within 10^{-14} s after fission occurs from the fission fragments.
(2) Delayed neutrons, produced in radioactive decay reactions of the fission fragments and their products. Through the energies of delayed neutrons are relatively small, they play a vita role in nuclear reactor control.

8.7-The Principle Components of the Nuclear Reactor

I- The Core and the Fuel Elements

The nuclear fuel material must be suitable for a self-sustaining neutron chain reaction. Some nuclear fuels are U^{235}, U^{233} and plutonium 239. The solid fuel is fabricated into various small shapes, such as plates, pins, pellets, etc; they are assembled to form fuel elements. These elements are arranged in fixed geometrical patterns to form the reactor core.

II- Moderator

This is a material which slows down fast neutrons quickly without describing them. This process is desirable because slow-moving neutrons are more effective in triggering fission than last neutrons. The moderator material is well-distributed in the reactor core or mixed with the fuel. ***Graphite, Hydrogen, Helium, Beryllium, Lithium, borom and Heavy water*** are used as a moderator.

III- Reflector

A reflector is a material which has good neutron scattering properties coupled with a low absorbing power. Therefore, when the reactor core is surrounded by this material, it reflects the neutrons back into the active core. This reduces the neutron leakage. As a result, less fuel is required to produce sufficient neutrons to sustain a chain reaction.

IV- Coolant

Coolant is a substance (liquid or gas), which transfers the heat produced in the reactor core (see Fig 8.3) to another fluid in a heat exchanger.
The coolant keeps the core temperature within the prescribed limits and prevents overheating. Some coolants are salt fuel water, liquid sodium, elements carbon dioxide and helium.

V- Control System

The control mechanism in a nuclear reactor consists of the sensing elements and automatic devices which control the position of the control rods. These rods may be shaped like the fuel rods themselves and are interspersed throughout the core. Instead of containing fuel, they contain a *neutron absorber*, also called poison, such as **boron**, **cadmium,** or **indium.** They have high neutron-absorption cross sections and do not contribute to neutron multiplication. The sensing element measures the density of the neutron flux in the reactor and actuates the control rods to absorb excess neutrons. For a uniform rate of power generation, the rate of fission must be maintained constant. Boron and cadmium rods are used for controlling fission. A control system, besides regulating the power output is also required to shutdown the reactor in the event of an emergency therefore there are two sets of control rods-one for routine power regulation and the other for emergency shut down .

VI- Radiation shield

A radiation shield as shown in Figure (8-3) surrounds the reactor to protect the personnel from radiations emitted during the fission process. The materials used will vary from system to system and many combinations are effective in reducing gamma and neutron radiation.

8.8- Uranium Enrichment

- Gaseous diffusion

- Gaseous centrifugation
- Laser enrichment
- Jet nozzle process

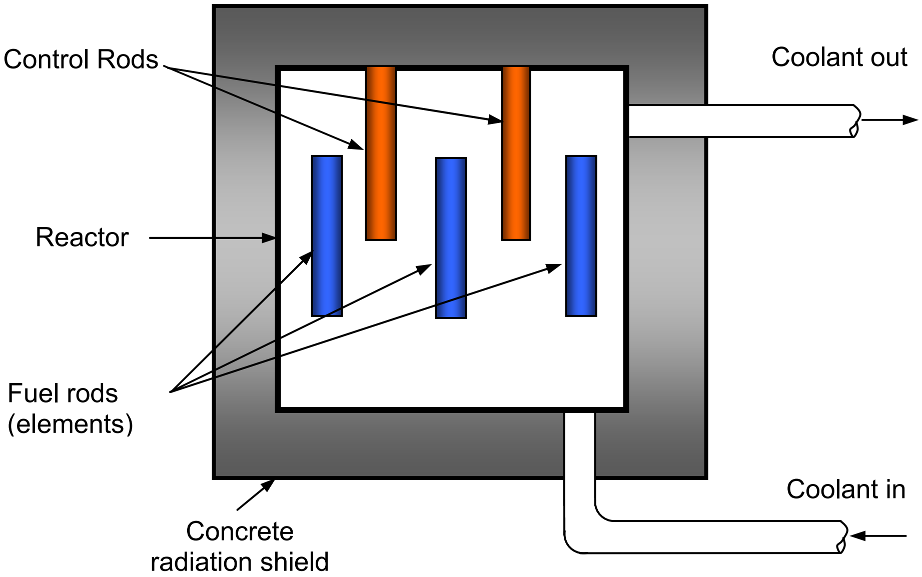

Figure (8.3)

8.9- Reactor control

There are several methods to control a nuclear reactor, i.e., to start, increase, decrease, and turn off its power. The most common method is the use of *control rods* (see Figure 8.3). These rods may be shaped like the fuel rods themselves and are interspersed throughout the core. Instead of containing fuel, they contain a *neutron absorber*, also called poison, such as **boron**, **cadmium,** or **indium.** They have high neutron-absorption cross sections and do not contribute to neutron multiplication.

Such control rods are designed with sufficient absorber to change the neutron balance in the core so that less than one neutron is left for fission for each neutron engaging in fission, a situation that would lead to a power decrease and eventual shutdown. The control rods are operated by control-rod drives that can move them in and out of the core around a power equilibrium position which is usually a partially inserted position. The rods are moved out to increase power at a prescribed safe rate or are moved in to decrease power. In either case, when the required power is attained the rods are returned to the equilibrium position.

The rods can also be used to adjust power levels selectively within the core to help even out the radial power distribution. Fuel zoning, i.e., the use of variable fuel enrichment within the core (usually three roughly concentric zones are used) is also used to attain the same purpose.

Control rods can be suddenly and completely inserted in the core to shut it down in case of an emergency. Such an operation is called a *scram*. It can be done automatically or manually by a visible colored button on the reactor console in the control room. A number of rods may be built into the system which are usually fully withdrawn from the core during normal operation and which have the sole function of shutdown by becoming fully inserted upon demand.

The equilibrium position varies with the life of the core. As more fuel is depleted, i.e., is fissioned with time, fewer neutrons are produced and more neutrons are absorbed in nonfission reactions. The rods are then slowly moved to less-inserted positions to compensate for this loss of neutrons and to keep the core in equilibrium.

The rods are said to possess a reactivity worth that should be adequate to cover operational control during the life of the core, fuel depletion, safety shutdown, and such other effects as Doppler, samarium, and xenon poisoning

Another method of control in pressurized-water reactors, called *chemical shim*, is used in addition to, not in lieu of, control rods. Chemical shim is the use of a soluble absorber, usually boric acid, in the moderator coolant. The concentration of this absorber in the moderator coolant is decreased slowly during the core lifetime to overcome the effect of fuel depletion. The concentrations are sufficient to permit operating the core almost unrodded, i.e., with all control rods nearly fully withdrawn.

It should be mentioned here that boiling-water reactors do not use chemical shim, and that the core of a BWR can be in equilibrium at several positions of the control rods because of the strong effect of the steam voids on reactivity.

Another control system is the use of *reflectors*. These are mechanically operated devices, situated just outside the core, that contain material that reflects some of the neutrons escaping the core back into it. The reflectors are swung away or toward or are axially moved with respect to the core to increase or decrease power. This method is used only for small-power reactors and in special cases, such as the SNAP 10A reactor which was operated in space.

Another method of control of lesser use yet is the use of some movable fuel rods. These are withdrawn from the core to decrease power and inserted to increase it, the opposite of the poison control rods.

8.10- Multiplication factor, k_{eff}

The ratio of the number of neutrons at the end of one generation to the number at the beginning of that generation is called the effective multiplication factor, , k_{eff} so that

$$k_{eff} = \frac{neutrons\ produced}{neutrons\ absorbed + neutrons\ escaped} \tag{8.8}$$

If $k_{eff} = 1$, the condition for a steady and stable chain reaction is satisfied and the reactor is said to be critical. If $k_{eff} > 1$, the reactor is supercritical and a divergent chain reaction exists in which the neutron density and fission rate increase at an explosive rate as in an atom bomb. If $k_{eff} < 1$, the reactor is subcritical and the chain reaction decreases and eventually dies out.

In a critical reactor, the rate at which neutrons are used up or lost must be exactly equal to the rate at which they are produced by fission. The theory of nuclear reactors is concerned with the analysis of all the processes which take place in the core of a reactor, and in particular with the slowing down, diffusion and absorption of neutrons.

Only by analyzing these processes in detail can accurate calculations be made to determine the multiplication factor, critical mass of fuel or size of a reactor (permitting a certain amount of neutron escape).

The percentage increase in the multiplication factor is called “reactivity” which is defined as

$$\rho = \frac{k_{eff} - 1}{k_{eff}} \tag{8.9}$$

If k_{eff} = 1, the chain reaction is steady and stable and ρ = zero.

The neutron life time, τ^*, is defined as the average time between successive neutron generations (prompt neutrons). The number of neutrons existing in a unit volume is n. Then, the time rate of change of the neutron density is

$$\frac{dn}{d\tau} = \frac{n\rho}{\tau^*} \tag{8.10a}$$

or

$$\frac{dn}{n} = \frac{\rho d\tau}{\tau^*} \tag{8.10b}$$

or

$$\ell n \frac{n}{n_o} = \frac{\rho\,\tau}{\tau^*} \tag{8.10c}$$

where n and n_o represent the neutron density at time rand τ = 0. It can also be expressed as

$$n = n_o\, e^z \tag{8.11}$$

where

$$z = \frac{\rho}{\tau^*}\tau \tag{8.12}$$

Example 8.1

A reactor is operating at a low power of 1 W. It then becomes supercritical with k_{eff} =1.0015. The average neutron life is 0.0001 s for prompt neutrons. Determine the reactor power level at the end of 1 s.

Solution

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Example 8.2

Calculate the time required for the reactor power to double for (a) assumed fission with prompt time $\left(\tau^{*} = 0.0001\,s\right)$ and (b) actual fission with prompt and delay neutrons $\left(\tau^{*} = 0.1\,s\right)$. Take k_{eff} =1.002

Solution

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8.11- Reactor energy

Energy is released by fission within the fuel and is transferred by heat conduction to the surface of the fuel and through the cladding. From the surface of the cladding heat is transferred by convection to the coolant, which passes from the core to the external heat exchangers in which steam is generated to operate on a power cycle.

The fuel elements are usually long cylindrical rods or rectangular plates of uranium (or thorium) enclosed by cladding. The uranium may be in the pure metallic form, in the form of a compound such as *uranium dioxide*, UO_2 or in the form of an alloy with another metal such as aluminium or zirconium. The desirable properties of a fuel, which must be fissionable, are high thermal conductivity, good corrosion resistance, good mechanical strength at high temperatures and a high limiting temperature for operation. The cladding serves three functions.

1. To provide structural support and strength for the fuel and prevent distortion.
2. To prevent the release of radioactive fission products into the coolant stream.
3. In certain types of reactors (mainly gas-cooled) to provide extended surfaces in the form of fins to promote more heat transfer to the coolant.

Materials suitable for cladding should have low neutron capture cross-section, high thermal conductivity, good mechanical strength at high temperatures and chemical compatibility with the fuel and coolant. The most common cladding materials are aluminium, magnesium alloys (Magnox), stainless steel and alloys of zirconium (zircaloy).

The specific rate of heat release by fission per unit volume of fuel is called the volumetric thermal source strength, $\dot{q}_G$, given by

$$\dot{q}_G = G N \sigma_f \varphi \qquad MeV/m^3\ s \tag{8.13}$$

where G = energy per fission ($\cong$ 180 -190 MeV)
N = number of fissionable nuclei/m^3
σ_f = effective microscopic fission cross-section of that fuel, m^2, and
φ = neutron flux per m^2 s.

Example 8.3 (p 648)

The fuel density N for a uranium oxide fuel is given by
$N=2.373 \times f \times 10^{22}\ U^{235}$ nuclei/cm^3
where f is the mass fraction of U-235 in the fuel
Determine for a light water moderated uranium reactor the specific energy release rate in W/cm^3 for the following conditions
G = 180 MeV
σ_f = 577 bans = (577×10^{-24} cm^2)
φ = neutron flux =10^{13}/cm^2 s.
f = 3.5 % U^{235} enrichment

Solution

8.12- Types of Nuclear reactors

Nuclear power reactors are commonly classified into five groups, Four of which have names derived from one or more of the maternal used in their construction, and one whose name derives from the neutrino process involves the groups are : -

- i- Pressurized water reactor (PWR)
- ii- Boiling water reactors (BWR)
- iii- Gas Cooled Reactor (GCR)
- iv- Heavy water reactors (HWR)
- v- Fast breeder reactor (FBR)

within each of these categories are to be found considerable variation in size , both in terms of physical dimensions and power output , and in the types and arrangements of material, these variations will be considered under the appropriate heading. Nuclear reactors are used for a verity of purposes, including materials testing , neutron research and the production of plutonium, but discussion will restricted here to the types of reactor whose function of electricity .

(i) – Pressurized water reactor (PWR)

In a PWR, the coolant pressure (155 bar) is higher than the saturation pressure (85.93 bar) corresponding to the maximum coolant temperature (300 °C) in the reactor, so that no coolant boiling takes place. A PWR power plant is composed of two loops in series, the coolant loop, called the *primary* loop, and the water-steam or *working fluid* loop as shown in Figure (8.4), the coolant picks up reactor heat and transfers it to the working fluid in the steam generator. The steam is then used in a Rankine cycle to generate electricity.

Table 8.2 shows the Characteristics of PWR reactor

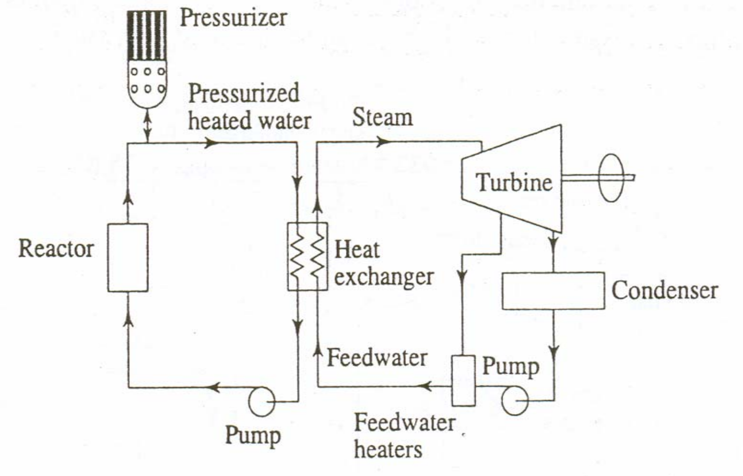

Figure (8.4) Schematic arrangement of a PWR power plant

Table 8.2

Characteristics of a typed PWR System	
Core thermal power	3820 MW (Th)
Plant efficiency	34%
Plant electricity	1300 MW (e)
Core diameter Output	335 cm
Core active Length	365 cm
Cladding material	Zircaloy – 4
Cladding thickness	0.62 mm
Fuel material	UO_2
Assembly array of fuel pins	17 X 17
Number of fuel assemblies	193
Control rod type	Sliver – indium – cadmium
Number of control rod assemblies	57 full length , 8 part length
Number of control rods per assembly	24
Control flow rate	65.45 103 t/h
Control pressure	15.5Mpa
Control inlet temp	292 C
Control outlet temp	325 C
Refueling sequence	1/3 of fuel per year
Refueling Time	17 day (minimum)

(ii) – Boiling Water Reactor (BWR)

A BWR differs from the PWR in that the steam flowing to the turbine is produced directly in the reactor core. Steam is separated and dried by mechanical devices located in the upper part of the pressure vessel assembly. The dried steam is sent directly to the high pressure turbine thus eliminating the need for steam generators (Fig. 8.5). The coolant thus serves the triple function of coolant, moderator and working fluid. Since the coolant boils in the reactor itself, its pressure is much less than that in a PWR and it is maintained at about 70 bar with steam temperature around 285 °C. However, an increase in the boiling rate displaces water (moderator) in the core and reduces the ability of the moderator to thermalize neutrons and hence, reduces the reactor power level. At power levels above 60% of the nominal, the fraction of steam in the core can be kept nearly constant by varying the coolant circulation rate.

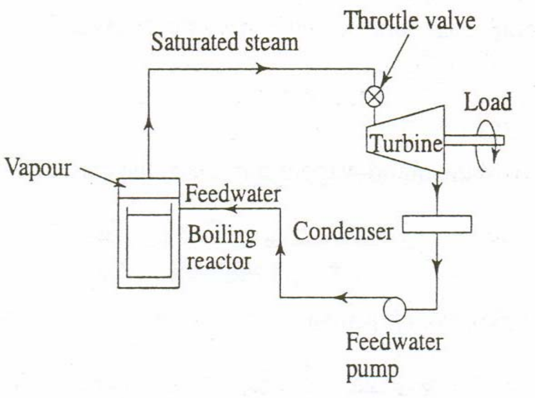

Figure 8.5 Schematic of a direct cycle BWR plant

The saturated liquid that separates from the vapor at the top of the reactor in a steam separator flows downward either internally within the reactor or externally outside the reactor and mixes with the return condensate (Fig.8.6). This recirculating coolant again either flows naturally due to density difference or by a forced circulation pump. The ratio of the recirculated coolant to the saturated vapor produced is called the circulation (or recirculation) ratio (as defined earlier in Chapter #). It is a function of the core average exit quality. The BWR core exit quality varies from 10 to 14%, so that circulation ratio is of the range 6-10. This is necessary to avoid large void fractions in the core, which would reduce the moderating power of the coolant resulting in low heat transfer coefficient or vapor blanketing and burnout.

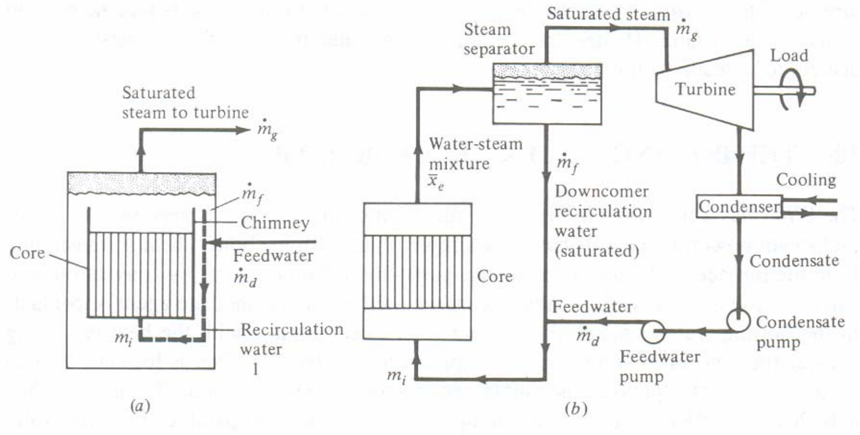

Figure (8.6) A BWR system with (a) internal and (b) external circulation

A slightly subcooled liquid enters the core bottom at a rate of $\dot{m}_i$ and rises through the core and chimney, if any. The chimney is an unheated section above the core that helps increase the pressure for natural circulation. The resulting vapor separates and goes to the turbine at a rate of $\dot{m}_g$ while saturated liquid recirculates via the downcomer at the rate of $\dot{m}_f$. There it mixes with the cold return feedwater $\dot{m}_d$ to form the subcooled inlet liquid $\dot{m}_i$.

An overall mass balance in the reactor core gives

$$\dot{m}_d = \dot{m}_g$$

$$\dot{m}_g + \dot{m}_f = \dot{m}_i$$

The quality of the liquid-vapor mixture at the core exit x_e, is given by

$$x_e = \frac{m_g}{m_f + m_g} = \frac{m_d}{m_f + m_d} = \frac{m_d}{m_i}$$

The circulation ratio R is then

$$R = \frac{\dot{m}_f}{\dot{m}_g} = \frac{1 - x_e}{x_e} \qquad (8.14)$$

Neglecting any heat loss and KE and PE changes, an energy balance of the core gives,

$$\dot{m}_i h_i = \dot{m}_f h_f + \dot{m}_d h_d$$

where h_i, h_f and h_d are the enthalpies of the inlet, recirculated and incoming feedwater, respectively. On rearrangement

$$h_i = (1 - x_e) h_f + x_e h_d$$

or $$x_e = \frac{h_f - h_i}{h_f - h_d} \qquad (8.15)$$

The condition of the liquid entering the bottom of the core is given by the enthalpy of subcooling

$$\Delta h_{sub} = h_f - h_i = x_e (h_f - h_d)$$

or the degree of subcooling is given by

$$\Delta t_{sub} = t_f - t_i$$

The *total heat generation* $\dot{Q}_t$ can be obtained from a heat balance on the core as a system or on the reactor as a system. The two relationships, which yield identical results, are

$$\dot{Q}_t = \dot{m}_i \left[(h_f + x_e h_{fg}) - h_i \right] = \dot{m}_g (h_g - h_d) \qquad (8.16)$$

Table 8.3 shows the Characteristics of BWR reactor

Table 8.3

Character of a Typical BWR System	
Core thermal power	3579 MW (th)
Plant efficiency	34%
Plant efficiency output	1220 MW (e)
Core diameter	460 CM
Core active length	388.6 CM
Cladding material	Zircalay –2
Cladding thickness	0.813
Fuel material	UO2
Assembly array of fuel pins	8X8
Number of fuel assemblies	748
Control rod type	Cruciform blades (boron Carbide)
Number of control rate	177
Coolant flow rate	4702 *10 t/h
Coolant pressure	717 Mpa
Coolant inlet temp	215 C
Coolant outlet temp	288 C
Refueling sequence	18 mths 1/3of fuel every
Refueling time	1 week

(iii)- Gas cooled reactor (GCR)

The first gas cooled reactors with CO_2 gas (at a pressure 16 bar) as coolant and graphite as moderator were developed in Britain during 1956-1969. The fuel was natural uranium, clad with an alloy of magnesium called Magnox (Figure 8.7)

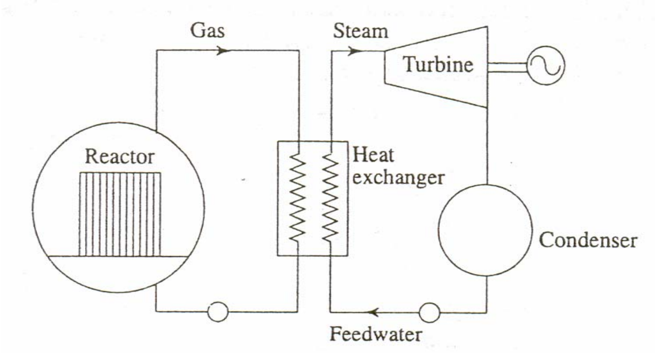

Fig. 8.7 Schematic of a gas-cooled reactor plant

Several types of gas-cooled reactors have been designed and built, with England developing an advanced gas-cooled reactor (AGR) system, and Germany and the USA developing helium-cooled, graphite-moderated systems (HTGR). The AGR uses UO_2 as the fuel clad in stainless steel tubes with CO_2 gas as coolant and graphite as moderator.

The graphite moderated helium-cooled HTGR is designed to use U-233 as the fissile material and thorium as fertile material. Initially, the system would have to be fuelled with U-235, until sufficient U-233 is available for makeup fuel. Because of the very high melting point of graphite, these fuel elements can operate at very high temperatures, and it is possible to generate steam at conditions equivalent to those in modem coal-fired power plant. The basic fuel forms are small spheres of fissile and fertile material as carbides, UC_2 or ThC_2. The fissile spheres are 0.35 to 0.50 mm in diameter and the fertile spheres are 0.6 to 0.7 mm in diameter. Each sphere is coated with two to three layers of carbon and silicon carbide to prevent fission products from escaping from the particles. Helium is a suitable coolant in the sense that it is chemically inert, has good heat transfer characteristics and low neutron absorption. Being a monatomic gas, it can produce more power for given temperatures in the Brayton cycle and higher efficiency.

A direct cycle HTGR gas turbine plant is shown in Fig. 8.8. It incorporates a regenerator and multi-stage compression with intercooling. Typical figures for such a cycle are: pressure ratio 4, turbine inlet pressure 50 bar, turbine inlet (reactor outlet) temperature 900 °C, compressor inlet temperature 50 °C. The temperatures at other points are shown in the figure, With U-233/Th-232 fuel, the HTGR functions as a thermal breeder reactor.

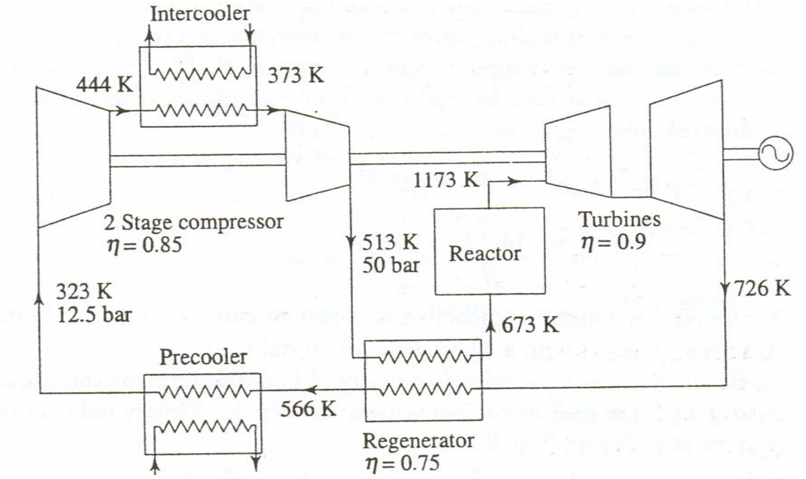

Figure 8.8 A HTGR direct cycle gas turbine plant using helium

(iv) – Heavy water reactor (HWR)

Heavy water is an alternative reactor moderator material, having good slowing down properties and very low thermal neutron absorption, which means that, as with graphite, a critical chain reactor can be sustained with natural uranium fuel, the reactivity of a heavy water/natural uranium lattice is in feet higher than for a comparable graphite/natural uranium lattice. The main advances in heavy water reactor technology have been in Canada where the CANADU system (Canadian Deuterium Uranium) has been developed, initially using heavy water as both moderator and coolant. The heavy water as moderator is used with a light water as a coolant known as the steam Generating heavy water Reactor (SGHWR) a - CANDU

Table 8.4

Characters of a CANDU Reactor system	
Core thermal power	2140 MW (th)
Plant efficiency	28 %
Plant electrical output	600 MW (e)
Core diameter	6.3 M
Core active length	5.9 M
Cladding material	Zircaloy
Cladding thickness	0.04 mm
Fuel material	Natural uranium oxide
Number of fuel pins per bundle	37
Number of fuel bundles per channel	12
Number of fuel channel	380
Coolant type	(D_2O) heavy water
Coolant pressure (mean)	10.7
Coolant inlet temperature	267 C
Coolant outlet temperature	312 C
Moderator type	Heavy water
Moderator pressure	Apprex. Atmospheric
Moderator inlet temp	43 C
Moderator outlet temp	71C
Refueling Sequence	Continuous on-line

b – SGHWR

Table 8.5

Characteristic of a proposed commercial SGHWR design	
Core thermal power	2026 MW (th)
Plant efficiency	30.02
Plant electrical output	660 MW (e)
Cladding material	Zircaloy
Cladding thickness	0.69 mm
Fuel material	UO_2
Fuel enrichment	1.8% U^{235}
Number of fuel pin per bundle	36 or (60 in advanced design)
Coolant type	Light water
Steam drum operating pressure	6.48 MPa
Moderator outlet temp	199 C

(v) – Fast breeder reactors (FBR)

Fast breeder reactors are designed to create or breed new fissile material, while producing useful electric power. Most produce fissile plutonium from fertile uranium 238. The fuel rods in the core region thus contain a mixture of fissile Pu-239 and U-238. The active core region is surrounded by a blanket of fertile U-238. This blanket region captures neutrons that would otherwise be lost through leakage, thus producing additional fissile material. FBR power reactor is commonly classified into four types.

a – Liquid Metal Fast Breeder Reactor (LMFBR)
b – Gas Cooled Fast Breeder Reactor (GCFBR)
c – Melting Salt Breeding Reactor (MSBR)
d – Light Water Breeding Reactor (LWBR)

8.15- GCR Power plant

8.15.1- Introduction

8.15.2- The simple dual – pressure cycle

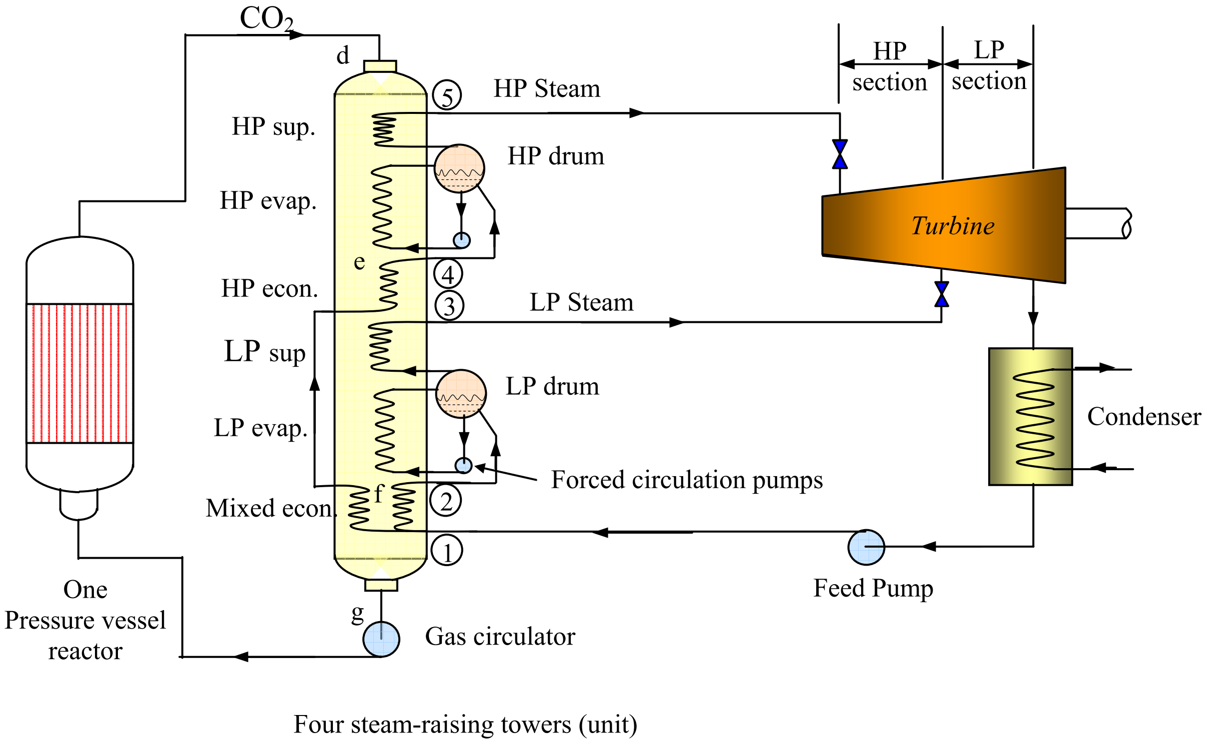

Figure (8.9)

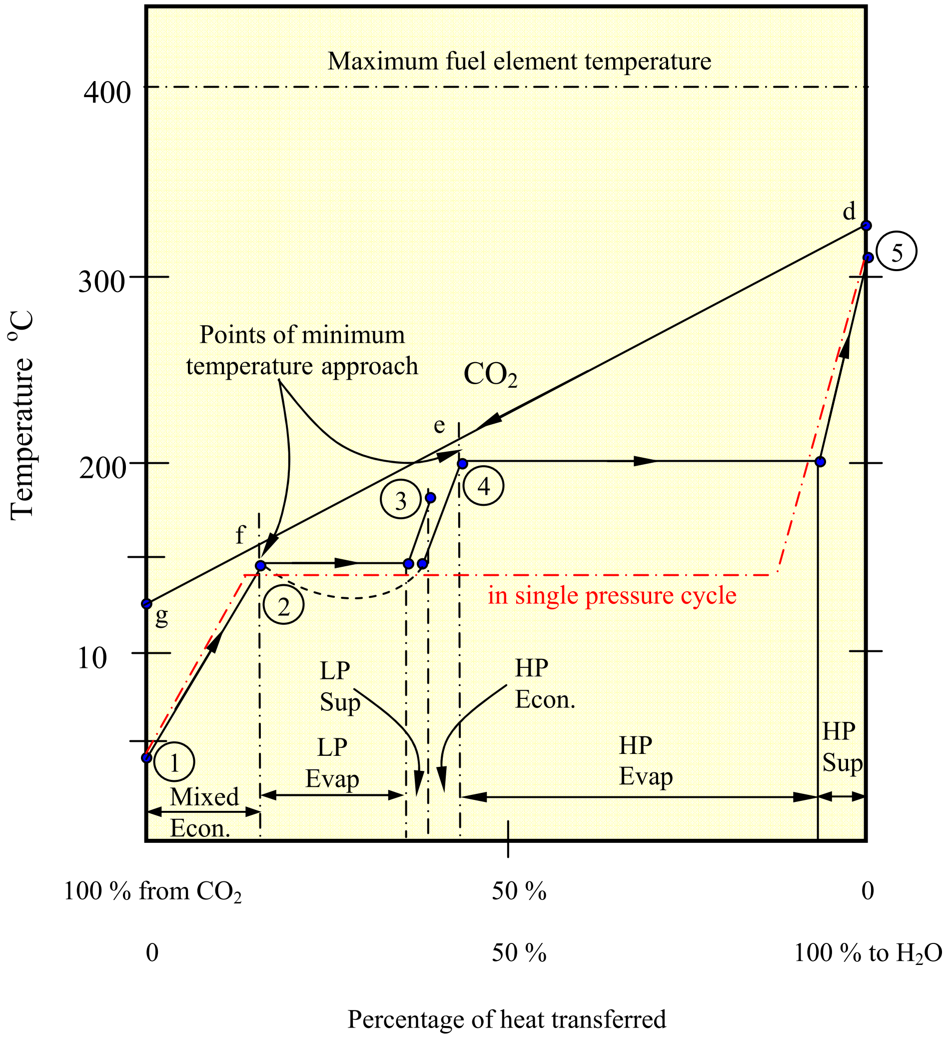

Figure (8.10)

8.15.3- Calculation of HP & LP Steam Flows and Cycle Efficiency

Referring, to Figure 8.9, the mass flow rates for the HP $(\dot{m}_H)$and LP $(\dot{m}_L)$ with respect to the CO$_2$ mass flow rate can be calculated, If the CO$_2$ entering temperature (T_d) and the minimum temp approach values at two pinch – points e & f have also been specified. The smaller the temp approach the optimum is the required surface area of the heat exchanger from thermoeconomic pointview, a value of 15 K being chosen as an appropriate economic figure.

Treating the CO$_2$ as a perfect gas having a constant specific heat c_p equal to the mean value between T_d and T_g.

Section d-e

$T_e = T_4 + 15$

and

$$\dot{m}_H(h_5-h_4)=\dot{m}_{CO_2}\,c_p(T_d-T_e)$$

or $$\frac{\dot{m}_H}{\dot{m}_{CO_2}}=\frac{c_p(T_d-T_e)}{(h_5-h_4)} \qquad (8.17)$$

Section e-f

$T_f = T_2 + 15$

and

$$\dot{m}_H(h_4-h_2)+\dot{m}_L(h_3-h_2)=\dot{m}_{CO_2}\,c_p(T_e-T_f)$$

or $$\frac{\dot{m}_L}{\dot{m}_{CO_2}}=\frac{c_p(T_e-T_f)}{(h_3-h_2)}-\frac{\dot{m}_H}{\dot{m}_{CO_2}}\frac{(h_4-h_2)}{(h_3-h_2)} \qquad (8.18)$$

Section f-g

$$(\dot{m}_H+\dot{m}_L)(h_2-h_1)=\dot{m}_{CO_2}\,c_p(T_f-T_g)$$

$$T_g=T_f-\frac{(\dot{m}_H+\dot{m}_L)(h_2-h_1)}{\dot{m}_{CO_2}\,c_p} \qquad (8.19)$$

In Figure (8.7) it has been assumed that, in the mixed economizer both the HP and LP fluid stream are raised to the LP saturation temperature, and the HP fluid stream leaves the HP economizer at the HP saturation temperatures, so that the values of T_2 and T_4 required in the above calculations may be read from the steam tables.

Given the isentropic efficiencies of the HP and LP portions of the turbine, and assuming perfect mixing of HP exhaust steam and the LP steam at LP turbine inlet, the state of the steam at turbine exhaust may be calculated and hence the heat rejected in the cycle. Finally calculation of the heat input enables the cycle efficiency to be determined.

8.15.4- Efficiency of the corresponding ideal dual pressure cycle .

The efficiency of the corresponding ideal dual reversible cycle may be calculated from considerations of available energy (exergy).The network output from this ideal device would be equal to the energy.

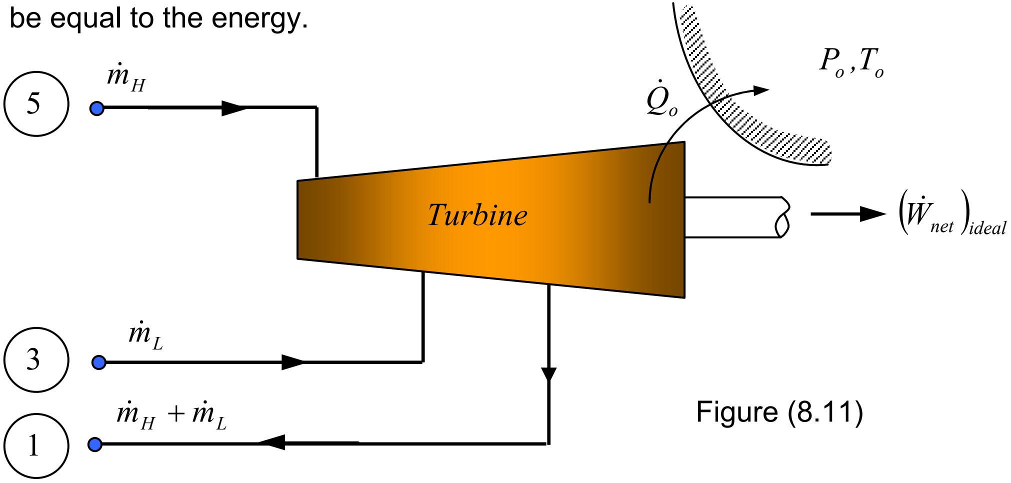

Figure (8.11)

$$\left(\dot{W}_{net}\right)_{ideal} = \left(\dot{E}_5 - \dot{E}_1\right) + \left(\dot{E}_3 - \dot{E}_1\right) \tag{8-20a}$$

$$\left(\dot{W}_{net}\right)_{ideal} = \dot{m}_H\left(\boldsymbol{\varepsilon}_{\boldsymbol{5}} - \boldsymbol{\varepsilon}_{\boldsymbol{1}}\right) + \dot{m}_L\left(\boldsymbol{\varepsilon}_{\boldsymbol{3}} - \boldsymbol{\varepsilon}_{\boldsymbol{1}}\right) \tag{8-20b}$$

$$\left(\dot{W}_{net}\right)_{ideal} = \dot{m}_H\left[\left(h_5 - h_1\right) - T_o\left(s_5 - s_1\right)\right] + \dot{m}_L\left[\left(h_3 - h_1\right) - T_o\left(s_3 - s_1\right)\right] \tag{8-20c}$$

The ideal efficiency of the cycle is given by

$$\left(\eta_{cycle}\right)_{ideal} = \frac{\left(\dot{W}_{net}\right)_{ideal}}{\dot{m}_H\left(h_5 - h_1\right) + \dot{m}_L\left(h_3 - h_1\right)} \tag{8.21}$$

Maximum work

The reversible work of HX between CO_2 and the steam is given by

$$\dot{W}_{rev.HX} = \dot{Q} - \dot{Q}_o \tag{8.22}$$

$$\frac{\dot{Q}}{T} = \frac{\dot{Q}_o}{T_o} \quad \left(dS_{system} = dS_{surr}\right)$$

$$\frac{T_o}{T} = \frac{\dot{Q}_o}{\dot{Q}}$$

Then

$$d\dot{W}_{rev.HX} = d\dot{Q}\left(1 - \frac{T_o}{T}\right) \qquad d\dot{Q} = \dot{m}_{CO_2}\, c_p\, dT$$

$$\dot{W}_{rev.HX} = -\dot{m}_{CO_2}\, c_p \int_d^g \left(1 - \frac{T_o}{T}\right) dT = \left(\dot{W}_{max}\right)_{ideal} \tag{8.23}$$

Then, the maximum work obtained from the heat transferred from the *unit mass* of CO_2 is given by;

$$\left(W_{max}\right)_{ideal} = -c_p \int_d^g \left(1 - \frac{T_o}{T}\right) dT$$

$$\left(W_{max}\right)_{ideal} = c_p\left[\left(T_d - T_g\right) - T_o\, \ell n\left(\frac{T_d}{T_g}\right)\right] \tag{8.24}$$

The corresponding cycle efficiency is then

$$\left(\eta_{max}\right)_{ideal} = \frac{\left(W_{net}\right)_{ideal}}{c_p\left(T_d - T_g\right)} = \frac{c_p\left(T_d - T_g\right) - T_o\, \ell n\left(\frac{T_d}{T_g}\right)}{c_p\left(T_d - T_g\right)} = 1 - T_o \frac{\ell n\left(\frac{T_d}{T_g}\right)}{\left(T_d - T_g\right)} \tag{8.25}$$

Example 8.4

At the design load the feed water is heated just to the saturation temperature of the LP steam in each section of the mixed economizer, the minimum temperature approach between the two fluids is 17 K in both the HP and LP heat exchangers, and the conditions in the steam raising towers as fallows:

Temperature of entering CO_2 = 337 ^oC

Temperature of entering feed water = 38 ^oC

HP steam condition at exit = 1.45 MPa, 316 ^oC

LP steam condition at exit = 0.435MPa, 177 ^oC

Treating CO_2 as a perfect gas with c_p = 1.017 kJ/kg.K and neglecting pressure drops and external heat losses in the heat exchanges, make the following calculations for the plant per kg of CO_2 circulated:

1– Calculated the masses of HP and LP steam produced, the temperature of the CO_2 at exit from the towers, and the heat transferred in the steam generators.

2 – Taking as the environment temperature that of the circulating water at inlet to the condenser, which is at 24 ^oC, calculate the exergy for the production of work from (a) the heat transferred from the CO_2 in its passage through the steam generators, and (b) the H_2O. In each case express the exergy as a percentage of the heat transferred in the steam generators.

3 – Calculate the reduction in the exergy of the steam if the environment temperature is taken as being the saturation temperature of the steam in the condenser, in which the pressure is 6 KPa. Express the new exergy as a percentage of the heat transferred in the steam generators.

4 – If the dual pressure cycle were to be replaced by a single – pressure cycle, producing steam at 316 ^oC from feed water at 38 ^oC, with the inlet and exit temps of the CO_2 the minimum temperature approach between the two fluids the same as in the dual pressure cycle, determine (a) the steam pressure, and the mass of steam produced, (b) the available exergy for an environment temp of 24 ^oC, again expressing it as a percentage of the heat transformed in the steam generators (c) cycle efficiency .

Solution

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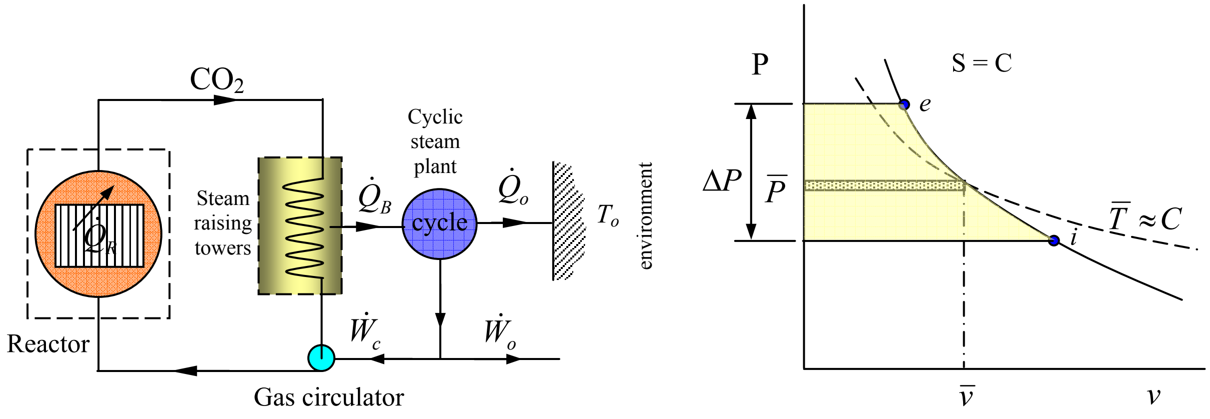
8.15.5- The effect of circulator power on the plant efficiency

The circulating power can be expressed as;

$$\dot{W}_c = -\dot{m}_{CO_2}\int v\,dP \quad \text{(for reversible isothermal and adiabatic)} \qquad (8.26)$$



(a) (b)

Figure (8.12)

$$\dot{W}_c = -\dot{m}_{CO_2}\int_i^e v\,dP = \left(h_i - h_e\right)_{s=c\ or\ \bar{T}\cong C}$$

$$\bar{v} = \frac{R\bar{T}}{\bar{P}}$$

where $\bar{v},\ \bar{T}\ and\ \bar{P}$ mean values

$$\dot{W}_c = -\dot{m}_{CO_2} R\int_i^e \frac{\bar{T}}{\bar{P}}dP = -\dot{m}_{CO_2} R\frac{\bar{T}}{\bar{P}}\left(P_e - P_i\right) = \Delta h_S \qquad (8.27)$$

For given $\dot{m}_{CO_2}$ and ΔP, $\dot{W}_c$ will increase with $\bar{T}$ but decrease with $\bar{P}$.

Referring to figure (8.12), the power required to drive the CO_2 circulators is by no means negligible and so leads to a significant reduction in overall station efficiency. However this power is not lost in its entirety for it is fed into the circulating CO_2, so that the heat transferred in the heat exchanger $\dot{Q}_B$ is that amount greater than the reactor thermal output $\dot{Q}_R$. A certain proportion of the circulator power is consequently recovered in the work output from the cycle, as the following calculating shows, the overall station efficiency is given by

$$\eta_o = \frac{\dot{W}_o}{\dot{Q}_R} = \frac{\dot{W}_{net} - \dot{W}_c}{\dot{Q}_R} \qquad (8.28)$$

But

$$\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_B} \tag{8.29}$$

Then

$$\dot{W}_{net} = \eta_{cycle} \times \dot{Q}_B$$

Neglecting stray heat loss and external losses in the circulators and their driving motors

$$\dot{Q}_B = \dot{Q}_R + \dot{W}_c \tag{8.30}$$

Hence

$$\eta_o = \eta_{cycle} - \left(1 - \eta_{cycle}\right)\frac{\dot{W}_c}{\dot{Q}_R} \tag{8.31}$$

Example 8.5

The pressure of the CO_2 at inlet to the gas circulators is 0.8 MPa and T_i=130.5 °C (see example 8.4); they provide a pressure rise of 28 kPa and their internal isentropic efficiency is 80%. Estimate the temperature rise of the CO_2 in its passage through the circulators, calculate the thermal output of the reactor per kg of CO_2 circulated, and overall efficiency of η_{cycle} = 35 %.

Solution

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Example 8.6

The reactor uses natural uranium fuel, eash tonne of which contains 7 kg of the fissile U^{235}, of which half can be consumed before the fuel elements have to half can be consumed before the fuel elements have to be withdrawn from the reactor. The U^{235} is consumed at the rate of 1.3 g per megawatt – day of reactor thermal output. What steady rate of fuel replacement would be required if the reactor operated continuously at the design load.

If the heat transferred in the steam generators were supplied in a boiler of 85% efficiency and burning coal having a higher calorific value of 25.6 MJ/ kg, what would be the rate of coal consumption at the design load when,

The mass flow rate CO_2 is 880 kg/s
Temperature of entering CO_2 is 337 ^{o}C
Temperature at reactor inlet or leaving CO_2 is 130.5 ^{o}C
Actual temp. rise due to circulation 3.24 ^{o}C
c_p of CO_2 = 1.017 kJ/ kg . K

Solution

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8.15.6- The effect of regenerative feed heating

The effect of regenerative feedwater heater have already dealt with extensively in Chapter II. The following example (8.9) and Figure (8.11) is incorporate regenerative feed heating to improve cycle efficiency by increasing the feed temperature.

Example 8.7

A nuclear, steam power plant with a gas cooled reactor operates on a single pressure cycle. The reactor coolant, CO_2 enters the steam raising unit at 560 ^oC . steam enters the turbine at a pressure of 12 MPa and atemp of 500 ^oC where it expands to the condenser pressure of 0.005MPa steam is bled from the turbine at 1MPa into a closed-type regenerative feedwater heater where it condensers to the saturated liquid state – this condensate is then throttled through an adiabatic value into the condenser, the feed water is pre-heated to the saturation temperature of the bled steam. assuming the temp .difference at the pinch to be 30°c the expansion process in the turbine to be frictionless and adiabatic and neglecting the feed pump work , calculate.

- a- the temp CO_2 as it leaves the steam – raising unit .
- b- the rate of the mass flow rate of H_2O to that of CO_2
- c- the fraction of the total steam flow which must be bled for feed heating .
- d- cycle efficiency (i) with feed heating (ii) without feed heating take c_p for CO_2 = 1.017 kJ/kg.k at all temps .

Solution

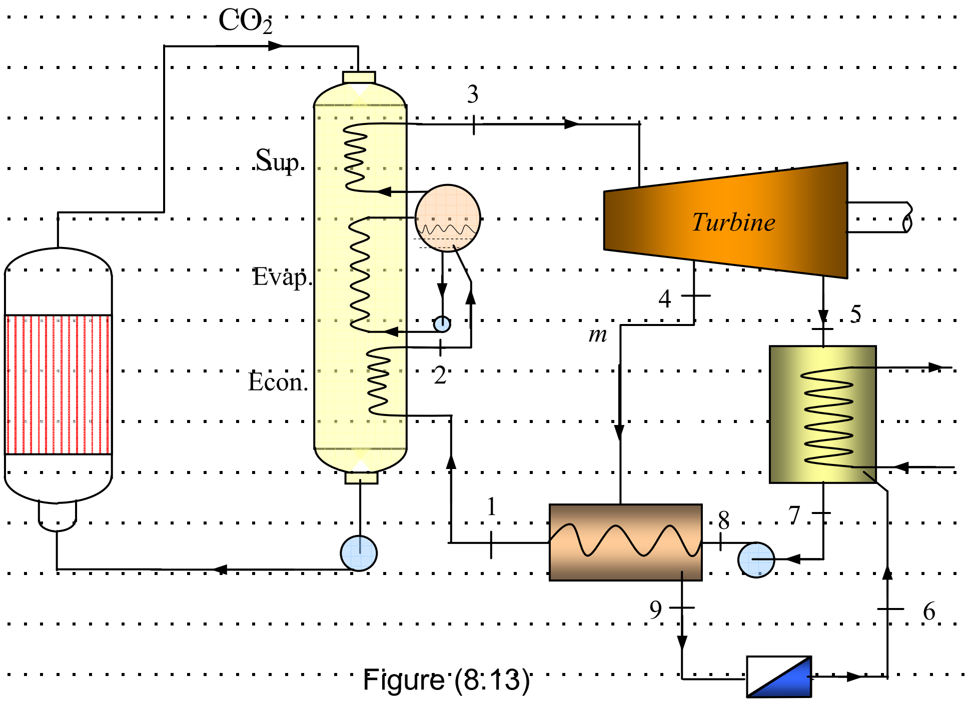

Figure (8:13)

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Revision Question

8.1- The CO_2 of a gas cooled reactor enters the steam raising unit at 550 °C where steam is generated at 15 MPa and superheated to 500 °C. The steam is then expanded in a turbine which exhausts at a pressure of 0.005 MPa into the condenser (see Figure 8.14). A fraction of the total steam flow is bled from the turbine at a pressure of 0.6 MPa into an open type of feedwater heater where it units with the feed water from the condenser. The feed water leaves the feedwater heater in a saturated liquid state, corresponding to the pressure of the bled steam, and is then returned to the steam raising unit.

Assuming the temperature difference at the pinch point to be 30 °C the expansion in the turbine to be reversible and adiabatic and neglecting feed pump work, calculate .

a. The ratio of mass flow rates of H_2O to that of CO_2.
b. The temp. of CO_2 as it leaves that steam raising unit.
c. The fraction of the total steam flow bled into the feedwater heater.
d. The cycle efficiency.

Take c_p for CO_2 = 1.017 kJ/ kg K at all temps

Ans: a-0.106, b – 278 °C, c- 0.218, d- 0.467

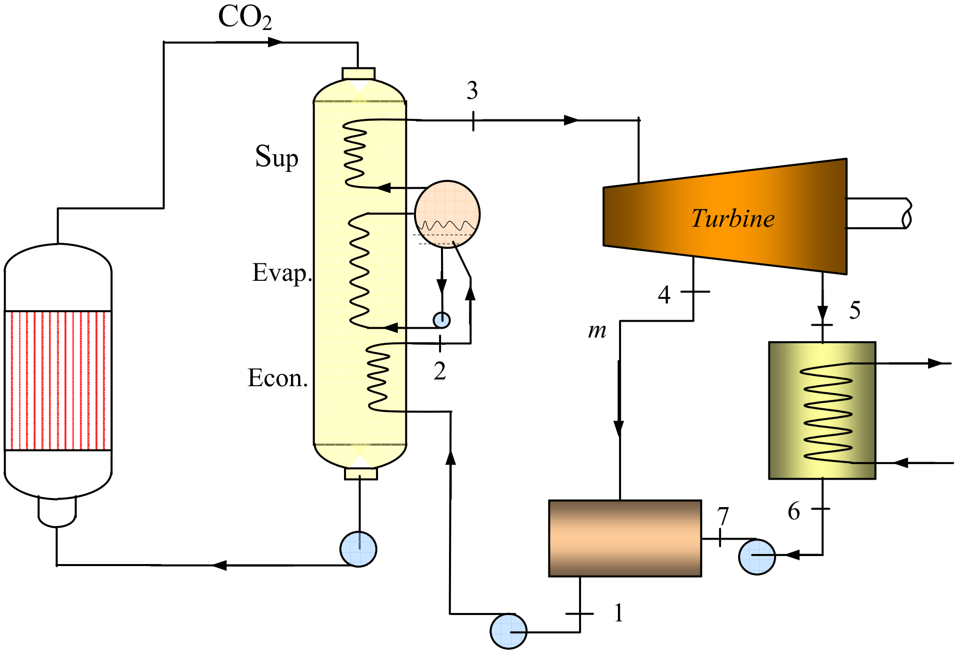


Figure (8.14

8.2- The plant layout of a proposed (PWR) power plant and the operating pressures temps are shown in Figure (8.15). Superheated steam is generated on the secondary side of the heat exchanger. The water substance leaving the condenser is in the saturated liquid state whilst the feed water is heated to the saturation temp of the bled steam. Assume that:

a- the expansion process in the turbine is reversible and adiabatic.
b- Pressure losses due to friction and heat losses are negligible.
c- Pump work may be neglected.

Calculate

(i) The ratio of the mass flow rates of the water substance in the primary circuit to that of secondary circuit.
(ii) The fraction of steam bled for feed heating , and
(iii) The ideal cycle efficiency

The enthalpy of compressed liquid may be taken to be equal to that of the saturated liquid at the same temp.

Ans: (i)-18, (ii)- 0.212, (iii)- 0.412

8.3- The plant layout of a proposed (PWR) power plant and the pressures and exits conditions shown in Figure (8.16) calculate m_1, m_2 , and η_{cycle}

Assume , $h_8 = h_7$, $h_{10} = h_9$, $h_{11} = h_{12}$

Ans: 0.191 , 0.098 , 40%

8.4- Describe to types of graphite moderated reactors, and discuses the main differences between them.

8.5- The power output of a reactor is to be doubled a value of 1.005 is prescribed for the effective neutron multiplreation factor (k_{eff}) find the time period (τ) when the average neutron life time $\tau^* = 0.1$ see.

Ans. 13.9 s

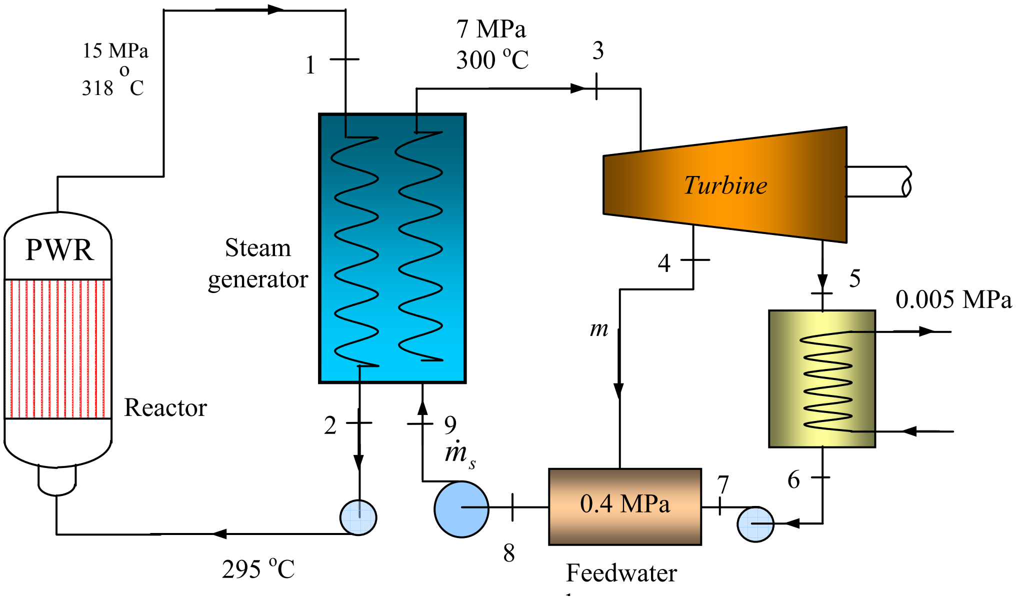
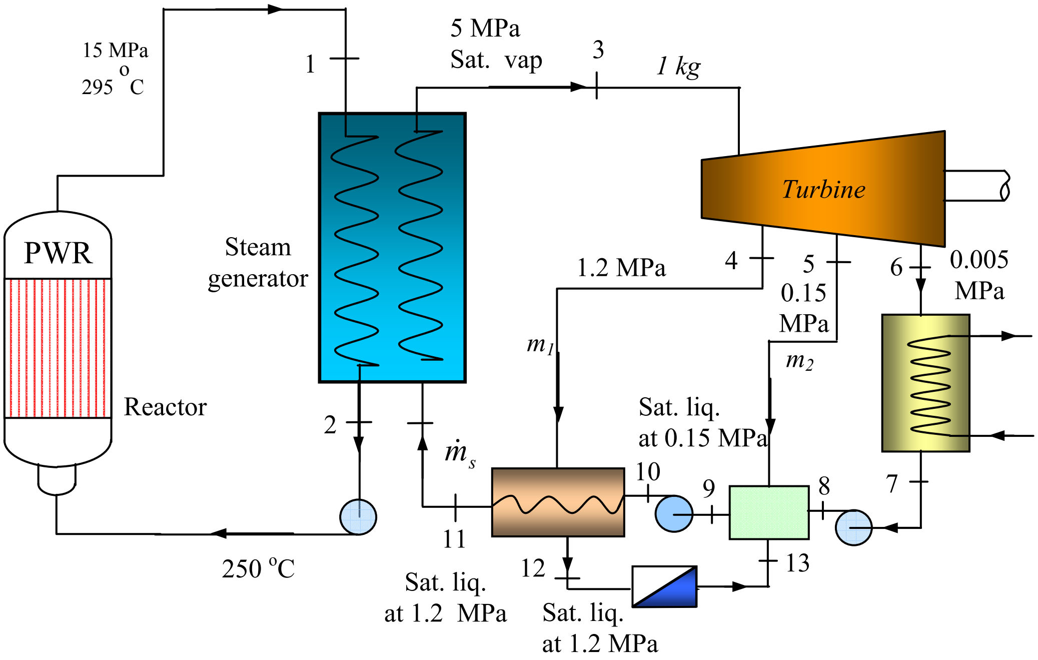


Figure (8.15)


15 MPa
295 °C
1
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Sat. vap
3
1 kg
PWR
Steam
generator
Turbine
Reactor
1.2 MPa
4
5
0.15
MPa
6
0.005
MPa
m_1
m_2
Sat. liq.
at 0.15 MPa
2
$\dot{m}_s$
10
9
8
7
11
12
13
250 °C
Sat. liq.
at 1.2 MPa
Sat. liq.
at 1.2 MPa

Figure (8.16)

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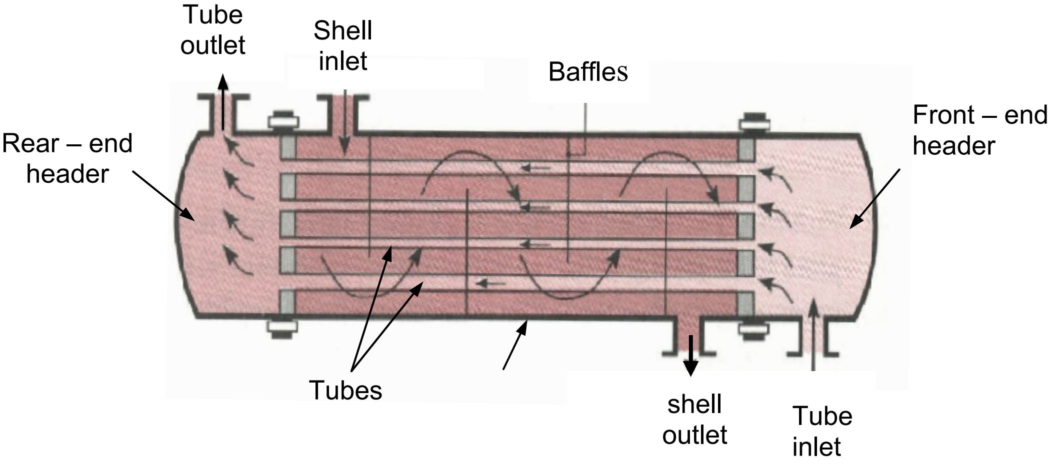

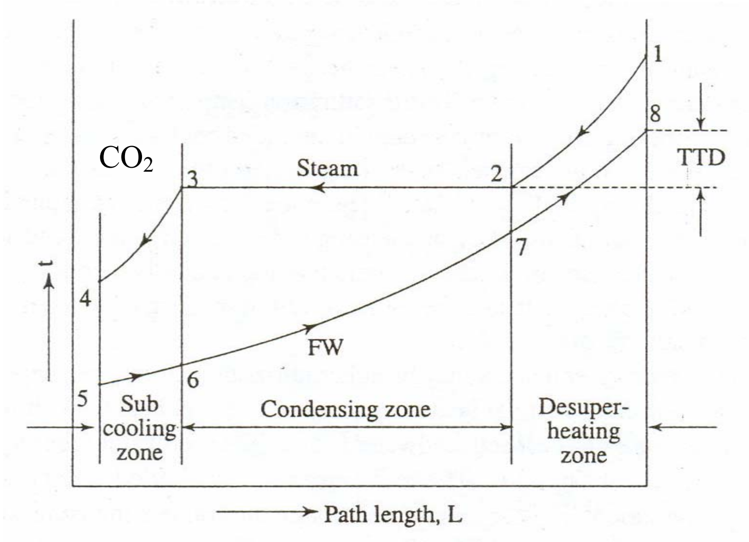

Chapter IX- Condensers, Feedwater and Circulating Water system

2010

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Chapter IX

Condensers, Feedwater and Circulating Water system

9.1-Introduction

A condenser where the exhaust steam from the turbine is condensed operates at a pressure lower than atmosphere. There are two objects of using a condenser in a steam plant:

1. To reduce the turbine exhaust pressure so as to increase the specific output of the turbine (Fig, 9.1). If the circulating cooling water temperature in a condenser is low enough (say 30°C), it creates a low back pressure (vacuum) for the turbine. This pressure is equal to the saturation pressure corresponding to the condensing steam temperature (say 0.074 bar at 40°C, Fig. 9.1), which, in turn, is a function of the cooling water temperature. It is known that the enthalpy drop or turbine work per unit pressure drop is much greater at the low pressure end than at the high pressure end of a turbine. A condenser by lowering the back pressure, say, from 1.013 to 0.074 bar, thus increases the plant efficiency and reduces the steam flow for a given output. The lower the pressure, the greater the output and efficiency. Hence, it is important to use the lowest possible cooling water temperature. This restricts the temperature rise of cooling water in the condenser tubes to 5-8 °C so that the tube outer surface temperature remains low and consequently, the condensing steam temperature is low and vacuum is high.

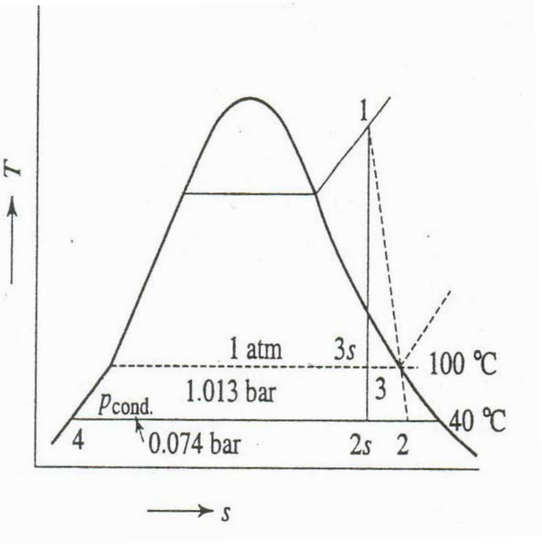


Figure 9.1 The use of condenser increases the specific work output of turbine from (h_1-h_3) to (h_1-h_2)

2. To recover high quality feedwater in the form of condensate and feed it back to the steam generator without any further treatment.

As a result, only the makeup water to replenish the water losses in the cyclic plant needs be treated.

9.1.1-Types

There are two broad classes of condensers:

(a) Direct contact type condensers, where the condensate and cooling water directly mix and come out as a single stream,

(b) Surface condensers, which are shell-and-tube heat exchangers where the two fluids do not come in direct contact and the heat released by the condensation of steam is transferred through the walls of the tubes into the cooling water continuously circulating inside them.

9.2- Direct contact condensers

These can be of three types:

- Spray condenser
- Barometric condenser
- Jet condenser

In a spray condenser, the cooling water is sprayed into the steam. Steam by mixing directly with cold water gets condensed. The exhaust steam from the turbine at state 2 mixes with cooling water at state 5 to produce saturated water at state 3, which is pumped to state 4 (Fig. 9.2).

Part of the condensate ($\dot{m}_4$), equal to the turbine exhaust flow ($\dot{m}_2$) is sent back to the plant as feedwater. The remainder is cooled in a dry cooling tower to state 5, and is then sprayed on to the turbine exhaust. Since the cooling water mixes with the steam and part of the condensate is used as feedwater, the water must be of high purity.

In a geothermal or ocean thermal energy conversion (OTEC) plant, only vacuum is required to be maintained in the condenser and no feedwater is needed. Hence, the mixture at state 4 is discarded.

For the spray condenser (Figure 9.2), a mass balance and an energy balance give the following equations

$$\dot{m}_2 = \dot{m}_4 \quad , \quad \dot{m}_3 = \dot{m}_2 + \dot{m}_5$$

$$\dot{m}_2 h_2 + \dot{m}_5 h_5 = \dot{m}_3 h_3$$

From which

$$\frac{\dot{m}_5}{\dot{m}_2} = \frac{h_2 - h_3}{h_3 - h_5} \tag{9.1}$$

Since h_2-h_3, is much greater than h_3 -h_5, the circulating water flow ($\dot{m}_5$) is much larger than the steam flow ($\dot{m}_2$).

In a barometric condenser (Fig. 9.3a), the cooling water is made to fall in a series of baffles to expose large surface area for the steam fed from below to come in direct contact. The steam condenses and the mixture falls in a tail pipe to the hot well below. By virtue of its static head, the tail pipe compresses the mixture to atmospheric pressure. Thus,

$$P_{atm} - P_{cond} + \Delta P_f = \rho g H \qquad (9.2)$$

where ρ = density of mixture, H = height of tail pipe, and ΔP_f is the pressure drop due to friction.

For low values of ΔP_f, H is around 9.5 m. Higher is the value of H, higher is the friction. Friction is lowered by increasing the tail pipe diameter, which results in a tall and heavy system.

In the jet condenser (Fig, 9.3b), the height of the tail pipe is reduced by replacing it with a diffuser. The diffuser helps raising the pressure in a shot distance than a tail pipe.

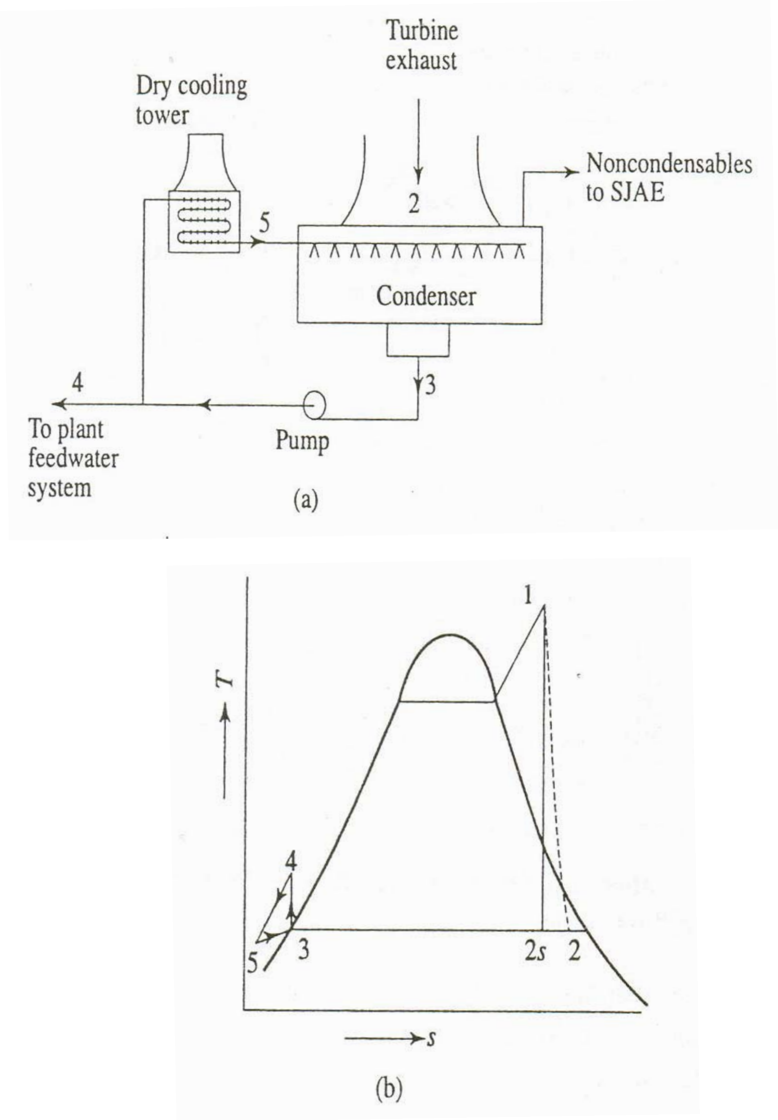


Figure 9.2 Schematic flow diagram of a direct-contact spray condenser (a) and the corresponding T-S diagram (b)

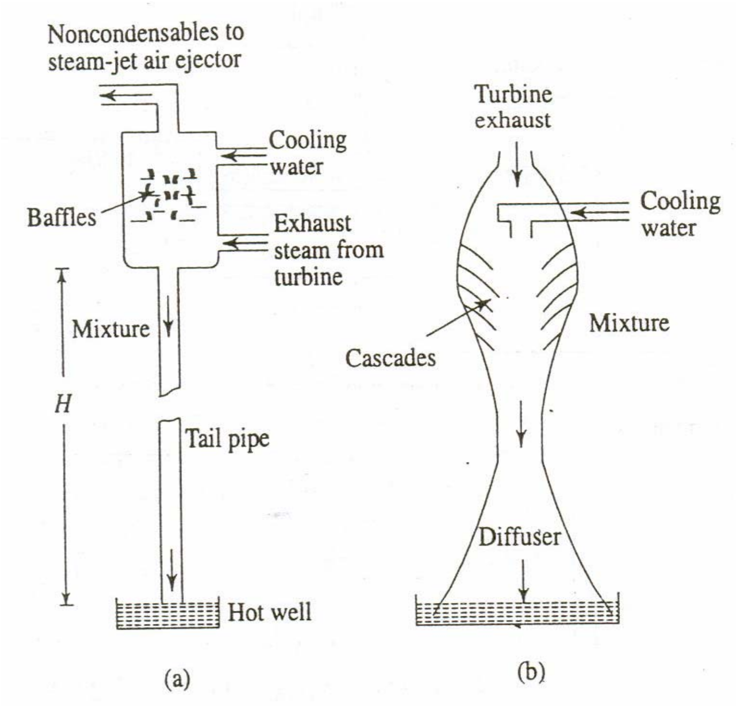


Figure 9.3 Schematics of direct-contact condensers: (a) barometric, (b) diffuser or jet

In spray type condensers, the non-condensable gases are usually removed with a steam jet air ejector (SJAE), as discussed later.

9.3- Surface Condensers

Surface condensers are mostly used in power plants. They are essentially shell-and-tube heat exchangers. For the convenience of cleaning and maintenance, cooling water flows through the tubes and steam condenses outside the tubes. Figure 9.4 shows a surface condenser with two passes on the water side. It consists of a steel shell with water boxes on each side. The right water box is divided to allow for two water passes. At each end there are tube sheets into which the water tubes are rolled. This prevents leakage of circulating water into the steam. An expansion joint allows for the different rates of expansion between the tubes and shell. There are vertical plates at intermediate points between the two tube sheets to provide support to the long tubes and to prevent tube vibration. The hot well acts as a reservoir of the condensate with a capacity equal to the total condensate flow for a certain period of time, say 5 min.

In a single pass condenser (Fig. 9.5), cooling water flows through the tubes once, from one end to the other. Compared to a two-pass condenser, a single- pass condenser with the same number and size of the tubes and with the same water velocity requires twice as much water flow but results in half the water temperature rise and therefore, lower condenser pressure. Thus, a single-pass condenser is good for overall plant efficiency and reduces thermal pollution, but requires more than twice the water flow and hence, four times the pumping power.

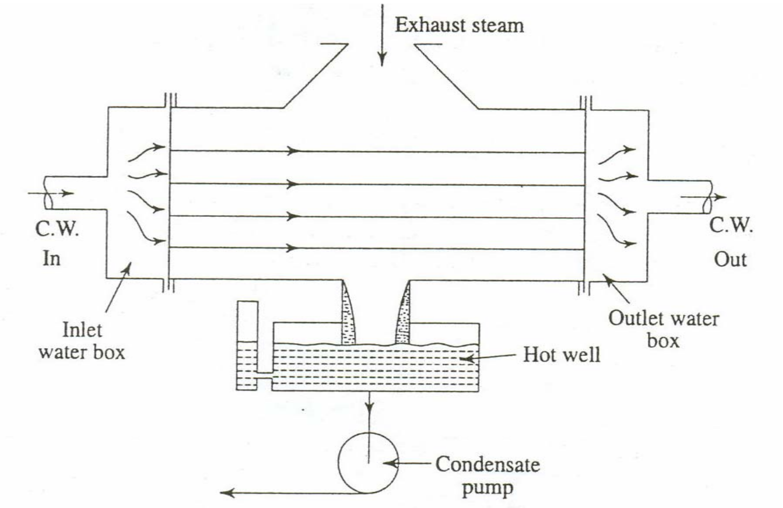


Figure 9.4 Schematics of a two-pass surface condenser

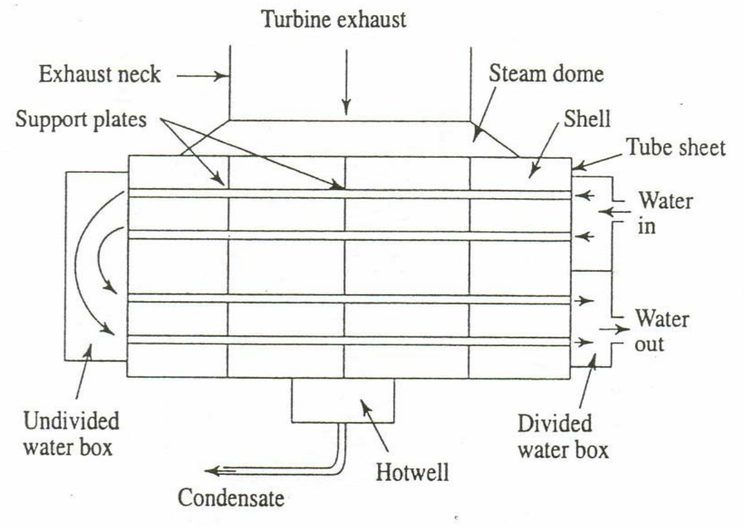


Figure 9.5 Single-pass condenser with divided water boxes

Apart from the division required to allow for a number of water passes, the water boxes are often further divided. For example, in a single-pass condenser, both the inlet and outlet water boxes are divided by a partition. This permits to take off half the condensing surface out of service for cleaning while water flows through the other half to keep the unit running at half load (Figure 9.5).

The tube arrangement in the condenser shell controls its effectiveness. Older designs tended to crowd a maximum number of tubes inside the shell to make available as much surface area as possible. Modern designs have steam lanes between tube banks to get maximum steam flow with least pressure drop and uniform distribution of steam in the shell. Tube lengths of 9-15 m are now quite common. Such long tubes result in a large rise in temperature of cooling water and the condensing ability decreases at the exit ends of the tubes. Thus, the tubes are closer at the cold end than at the hot end. To minimize the unequal distribution of steam flow from the turbine exhaust duct to condenser tubes, a well-tapered steam dome is added above the tube bundles. An expansion joint provided between the turbine exhaust and the condenser inlet permits the condenser to be rigidly mounted on the floor.

Figure 9.6 shows a typical modern two-pass surface condenser for a large steam power plant. Steam enters the tube bundles in two separate sections from the top, sides and bottom, and flows toward the centre of the tube nest in each section. At that point most of steam has condensed leaving only air and other non-condensable gases which are cooled and removed by SJAE, as explained later.

The tube material can be (a) cupronickel (70% copper, 30% nickel) (b) aluminium brass (76% copper, 22% zinc and 2% aluminium), (c) aluminium bronze (95% copper and 5% aluminium), (d) muntz metal (60% copper, 40% zinc), (e) admiralty alloy (71% copper, 28% zinc and 1% tin), or (f) stainless steel. The outside diameter of tubes is either 22 mm, 23 mm or 25.4 mm. The length varies from 9 to 15m.

9.3.1- Design Calculations

When wet steam comes in contact with a cold surface, the temperature of which is below the saturation temperature at the exhaust pressure of steam, it cannot but condense rejecting the latent heat of condensation (Figure 9.7). For filmwise condensation, the average heat transfer coefficient for a horizontal tube as given by Nusselt is

$$h_{av} = 0.725\left[\frac{k_f^3\,\rho_f^2\,g\,h_{fg}}{N_{tube}\,\mu_f\,d_o\left(t_{sat}-t_w\right)}\right]^{0.25} \tag{9.3}$$

The bulk temperature of the condensate lies between t_w and t_{sat}, So, it is less than t_{sat} and subcooled. This subcooling is not desired as it increases the heat supply in the boiler. By making some exhaust steam flows upward through the condensate its temperature is increased. Nusselt's equation gives only a conservative value for the condensing film coefficient of heat transfer, which is also influenced by vapor superheat, vapor velocity, turbulence, and the inside air.

The inside heat transfer coefficient on the water side may be obtained with the help or Dittus-Boelter equation

$$Nu_d = 0.023\,Re_d^{0.8}\,Pr^{0.4} \tag{9.4}$$

Where Re_d, is the Reynolds number $= \dfrac{\rho V d}{\mu} = \dfrac{V d}{\nu}$ ° and Pr is the prandtl number $= \dfrac{C_P \mu}{k}$

For water, $h_i \propto V^{0.8}$, V being the water velocity. Higher water velocity will improve heat transfer but increases pumping power also. The optimum water velocity varies between 2.0 to 2.5 m/s.

The overall heat transfer coefficient for a condenser tube is

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{1}{h_{si} A_i} + \frac{x_w}{k_w A_{\ell m}} + \frac{1}{h_o A_o} \tag{9.5}$$

Since condenser tubes are thin and made of good thermal conductivity, the tube wall resistance can be ignored, then

$$\frac{1}{U_o} = \frac{1}{h_i} + \frac{1}{h_{si}} + \frac{1}{h_o} \tag{9.6}$$

Here h_o, is much larger than h_i, and U_o mainly depends on water velocity as given by

$$\frac{1}{U_O} = A + B\frac{1}{V^{0.8}} \tag{9.7}$$

where $A = \frac{1}{h_{si}} + \frac{1}{h_o}$ and $A = \dfrac{1}{0.023\dfrac{k_f\, Pr^{0.4}}{d_i^{0.2}\, v_f^{0.8}}}$

By estimating the overall heat transfer coefficient at different water velocities, a plot (Wilson plot) can be made on log-log coordinates from which the intercept A and slope B can be obtained (Fig. 9.8). The rate of heat transfer from the condensing vapor to the cooling water is

$$\dot{Q} = \dot{m}_s(h_2 - h_3) = \dot{m}_c\, c_{pc}(t_{c2} - t_{c1}) = U_o A_o \Delta t_{\ell m} \tag{9.8}$$

where $\Delta t_{\ell m} = (\Delta t_i - \Delta t_e) \Big/ \ell n\left(\frac{\Delta t_i}{\Delta t_e}\right)$ is the logarithmic mean temperature difference (Fig. 9.9). The temperature difference at exit, Δt_e, is the terminal temperature difference (TTD) of the condenser. A small TTD results in a large condenser but reduced water flow and higher exit water temperature. It increases the capital cost but reduces the operating cost. The cooling water inlet temperature should be sufficiently low to have a good vacuum in the condenser shell. It is usually recommended that Δt_i should lie between 11 to 17 °C and that Δt_e or TTD should not be less than 3 °C.

From equation 9.8, the water mass flow rate is,

$$\dot{m}_c = \frac{\dot{m}_s(h_2 - h_3)}{c_{pc}(t_{c2} - t_{c1})} \tag{9.9}$$

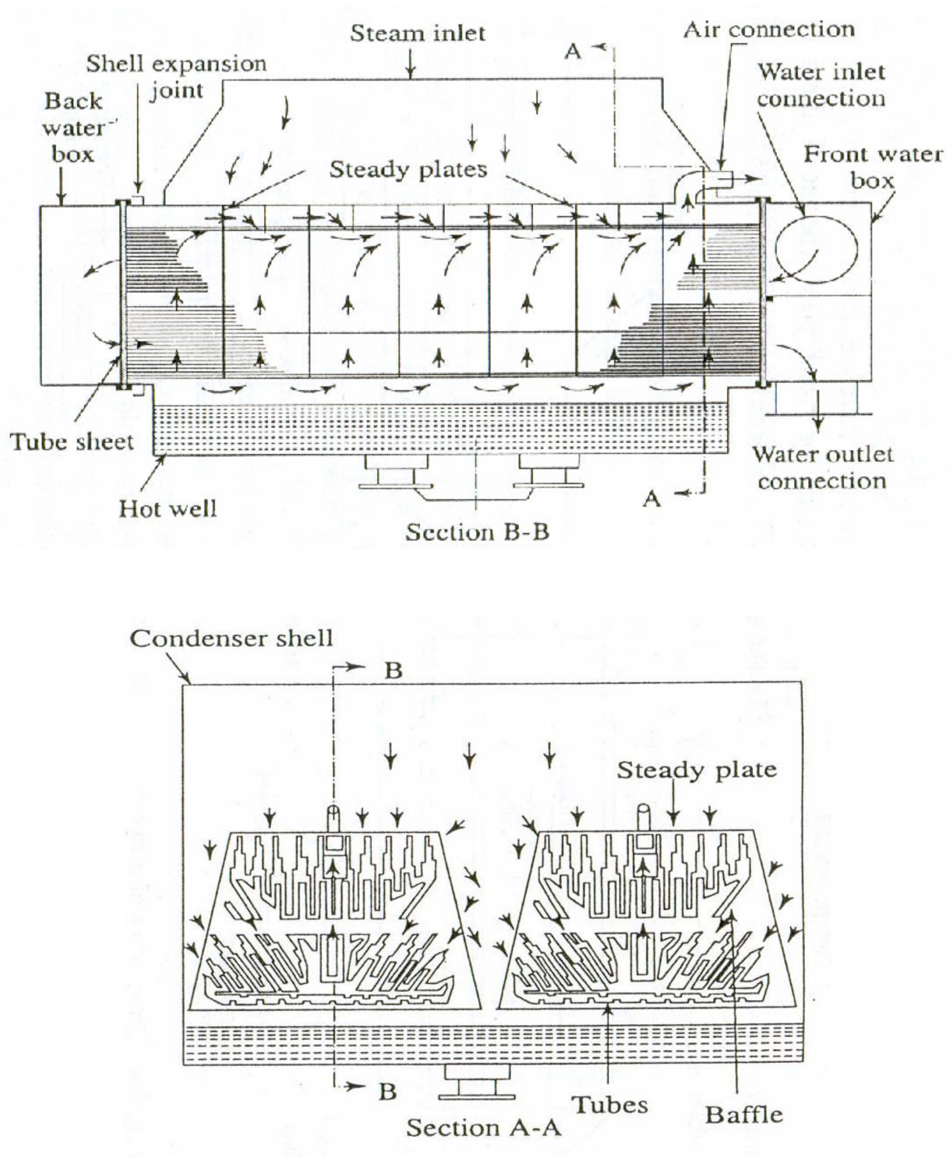


Figure 9.6 Sections through a typical two-pass surface condenser for a large steam power plant.

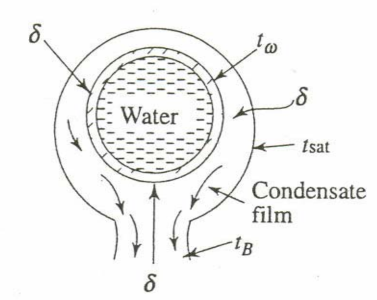


Figure 9.7 Film condensation of steam on a horizontal tube

The rise in water temperature is limited to about 8-10 °C. So, for every kg of steam condensed, 75 to 100 kg of cooling water is required. Thus, to meet the huge water demand, the power plant is located where water is available in plenty. The surface area needed by the condenser is obtained from Eq. (9.8).

$$A_o = \frac{\dot{m}_s (h_2 - h_3)}{U_o \, \Delta t_{\ell m}} = n \pi d_o \ell \tag{9.10}$$

where n = number of tubes, and ℓ = length of one tube (for a single-pass condenser). The water flows through the tubes. Therefore,

$$\dot{m}_s = \left(n \, \frac{\pi}{4} d_i \right) \rho V \tag{9.11}$$

where p is the density of water (1000 kg/m^3) and V is the water velocity (1.8 - 2.5 m/s). Therefore, the length and number of tubes can be estimated from the above two equations. For a modem condenser of a utility plant the number of tubes may be as high as 50,000 (22-23 mm $o.d.$).

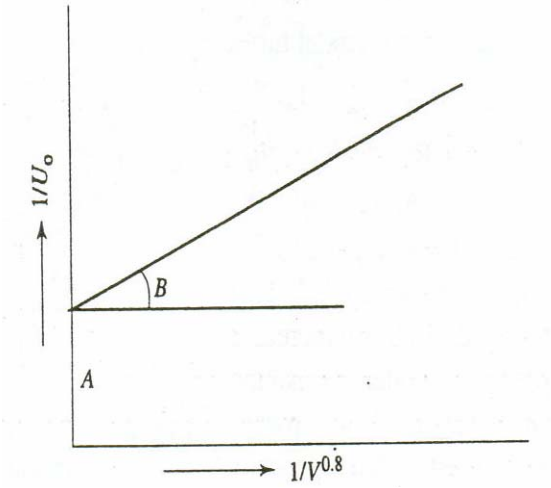


Figure 9.8 Plot of $\frac{1}{U_o} vs \frac{1}{V^{0.8}}$

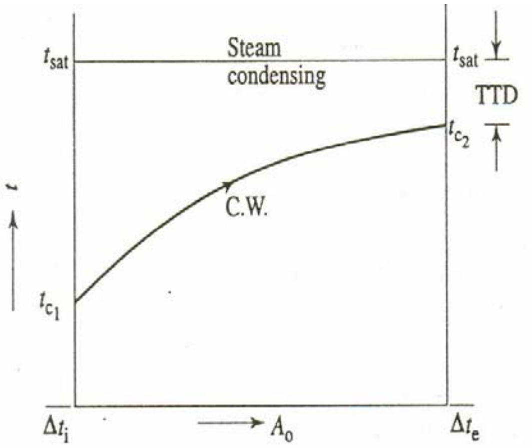


Figure 9.9 Temperature profiles in a condenser

The pressure drop in the condenser consists of (1) the pressure drop in the water boxes and (2) the pressure drop due to friction in the tubes $\left(\frac{f\,L}{d}\times\frac{\rho V^2}{2}\right)$. The pumping power required is then given by

$$P = \dot{m}_c\, v\,\Delta P = \frac{\dot{m}_c \Delta P}{\rho} \tag{9.12}$$

Where ΔP is the total pressure head to be developed by the pump in overcoming the losses.

9.4 Feedwater Heaters

As discussed in Chapter II regenerative feedwater heaters are always used in steam power plants to improve the cycle efficiency. They raise the temperature of the feedwater before it enters the economizer. Both open and closed type heaters are used. In small industrial plants, only one open feedwater heater may be used. But in large industrial and utility plants, five to seven closed heaters and one open heater are used. The open heater acts as a deaerator.

9.4.1 Closed Feedwater Heater

Closed feedwater heaters are shell-and-tube heat exchangers. They are basically small condensers which operate at higher pressures than the main condenser because bled steam is condensed on the shell side, whereas the feedwater, acting like circulating cooling water in the condenser, is heated on the tube side.

It was shown in Chapter that the temperature rise in each heater and economizer is equal for maximum cycle efficiency. Thus the heaters receive bled steam from the turbine at pressures determined roughly by equal temperature rise from the condenser to the boiler saturation temperature. They are classified as low pressure (LP) and high pressure (HP) heaters depending upon their locations in the cycle. The LP heaters are usually located between the condensate pump and the deaerator, which is followed by the boiler feed pump (BFP). The HP heaters are located between the BFP and the economizer.

When bled steam entering a feedwater heater is superheated, as in a HP heater, the heater includes a desuperheating zone where steam is cooled to its saturation temperature. It is followed by a condensing zone where the steam is condensed to a saturated liquid rejecting the latent heat of condensation. This liquid, called heater drain, is then cooled below its saturation temperature in a subcooling zone or a drain cooling zone before the drain is cascaded backward or pumped forward.

Figure 9.10 shows the schematic diagram and the temperature profiles of a three-zone closed feedwater heater. There are, however, two-zone heaters that include a desuperheating and a condensing zone or a condensing and a subcooling zone. There are also single-zone heaters that include only a condensing zone. A drain-cooling zone, instead of being a part of the shell, may be located outside it. It is then called a drain cooler.

Closed feedwater heaters may be either horizontal or vertical, depending upon space availability. Vertical heaters occupy less space. Fig. 9.11 shows a typical horizontal three-zone closed feedwater heater. The feedwater tubes are usually in the form of U-tube bundles. The feedwater enters a divided water box and flows through the subcooling zone and then through the condensing zone, and leaves to the water box through the desuperheating zone. The bled steam first flows through the desuperheating zone separated by a shroud. The vertical baffles provide good heat transfer and tube support. The condensing zone is the major portion of the heater. The subcooling zone is separated from the rest of the heater by an end plate. Pressure drops of the feedwater in heaters are usually large due to friction in long small diameter tubes of the heaters.

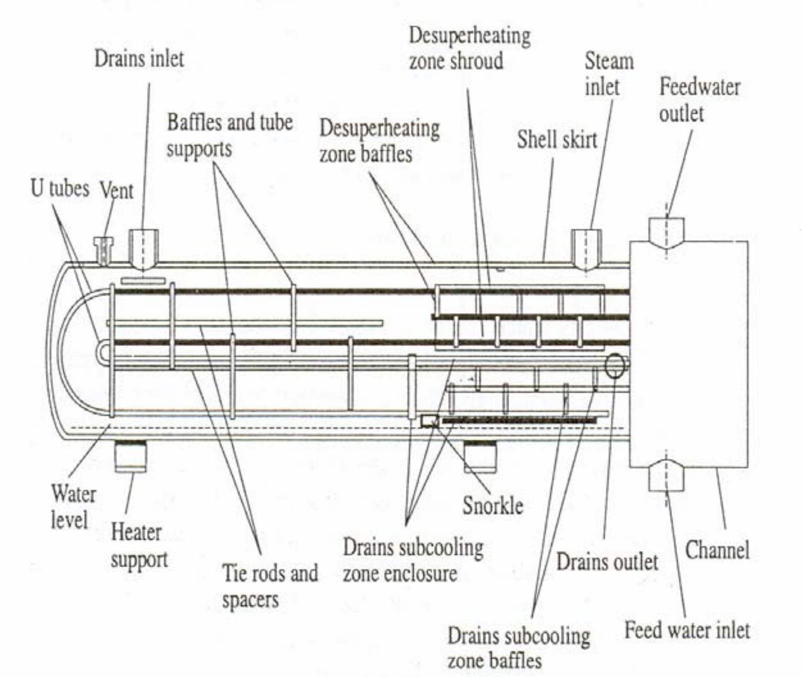


Fig. 9.10 (a) A three-zone horizontal closed-type feedwater heater

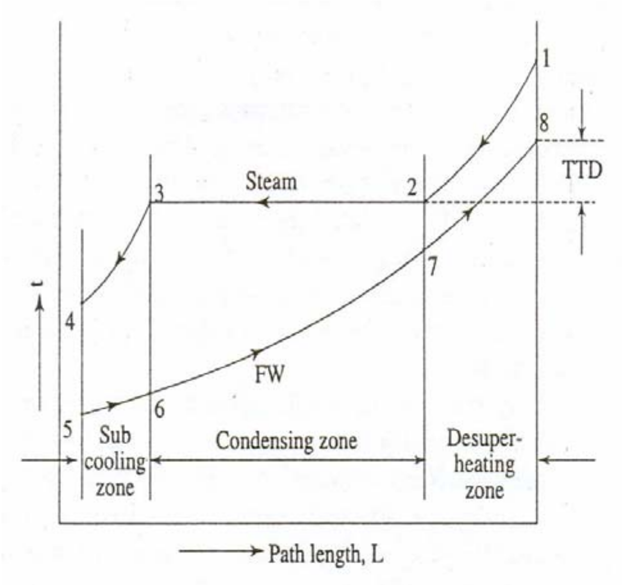


Fig. 9.10 (b) Temperature profiles along the path length in a three-zone feedwater heater.

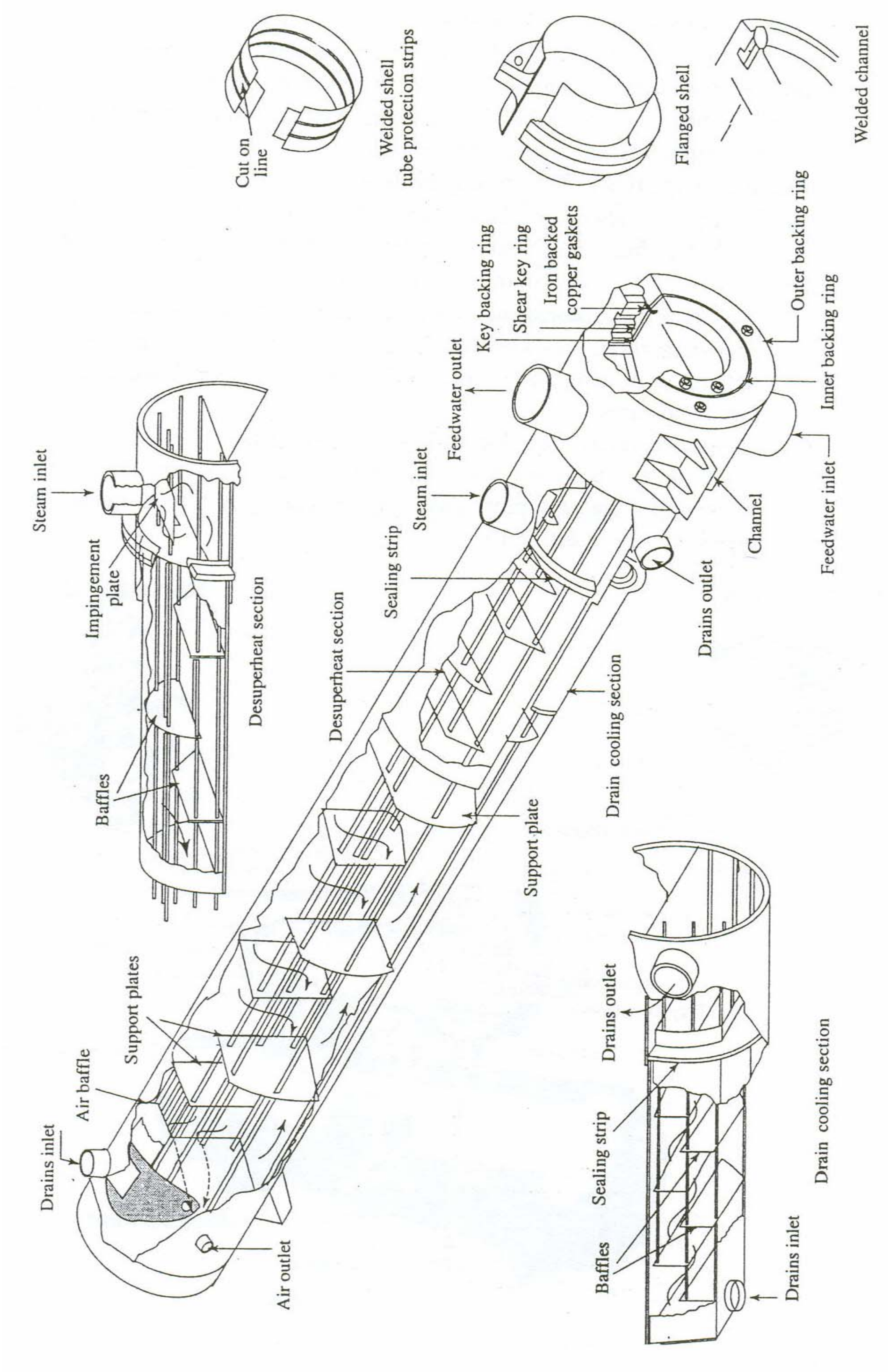


Figure 9.11 High temperature horizontal FW heater in a large steam power plant

9.2- Open Feedwater Heater

An open feedwater heater or deaerator is one in which the feedwater is heated by direct mixing with the steam bled from the turbine. It is used to remove dissolved gases in feedwater (Chapter). It is located at a sufficient height (20-25 m) above the boiler feed pump so that the suction pressure does not fall below saturation pressure to prevent cavitations.

There are three types of deaerating heaters.

a. Spray-type deaerators: Here, feedwater is sprayed through nozzles into the heater from the top and bled steam is fed from the bottom. Water is heated and scrubbed to release the dissolved gases.

b. Tray-type deaerators: Feedwater here falls through a series of cascading horizontal trays. As water falls from tray to tray, it comes in direct contact ith the up flowing bled steam, and gets heated and scrubbed to release the dissolved gases.

c. Combined spray-tray deaerators'. Feedwater is first sprayed and then made to cascade down a series of trays and bled steam flows upward. These types are now preferred in power plants. A typical heater of this type is shown in Fig. 9.12.

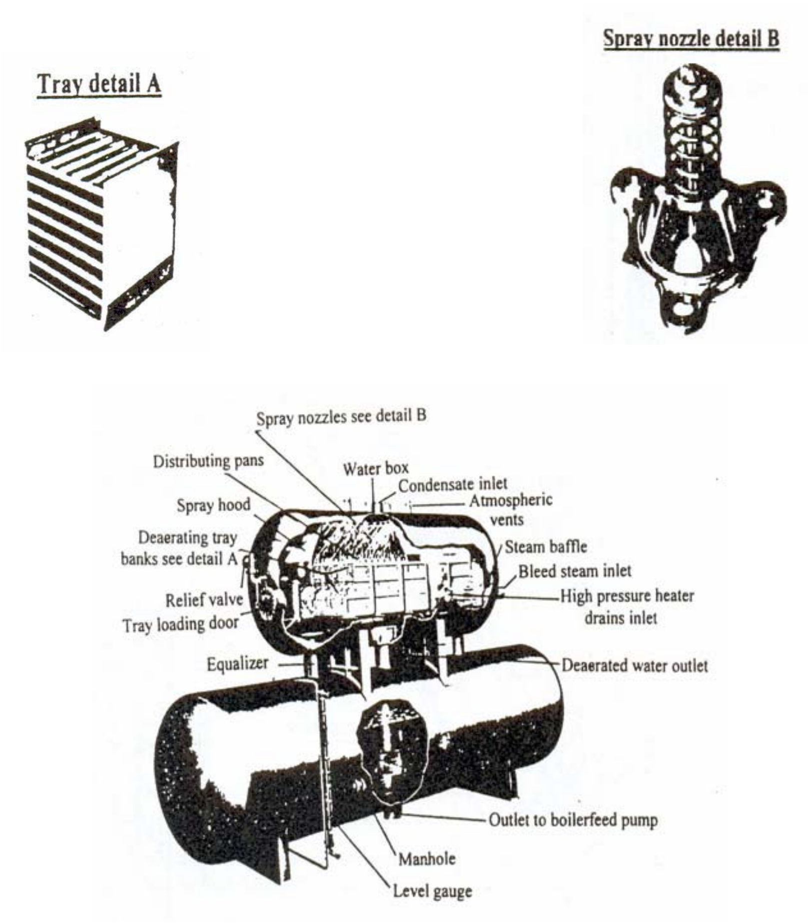


Figure 9.12 A typical open-type deaerating feedwater heater

9.5- Circulating Water System

The circulating water system supplies cooling water to the turbine condensers and thus acts as a medium through which heat is rejected from the steam cycle to the environment. Cooling water can flow through the condenser in two methods: (1) Once-through system, and (2) Closed loop system.

Once-through system (Fig. 9.13) is used when there is a large source of water available. Water is taken from a natural body of water like a lake, river, or ocean and pumped through the condenser, where it is heated, and then discharged back to the source.

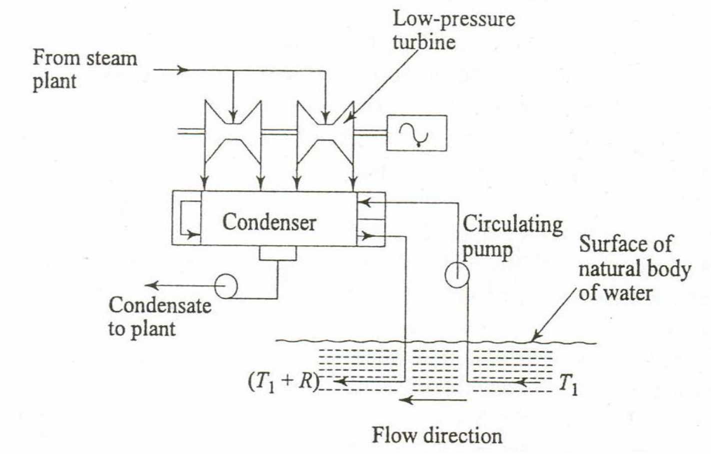


Figure 9.13 Schematic of a once-through circulating water system

In closed loop systems, warm water from the condenser is passed through a cooling device like a cooling tower or a spray pond and the cooled water is then pumped back for condenser circulation (Fig. 9.14). However, a natural body of water is still necessary nearby to supply the makeup water to replace the loss due to evaporation, blowdown and so on.

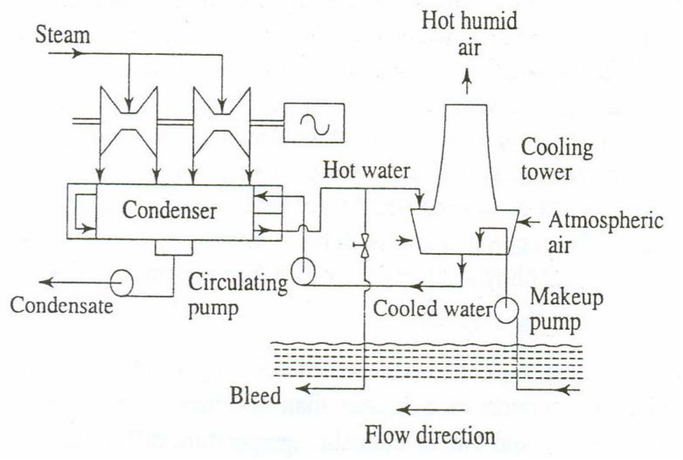


Figure 9.14 Schematic usual view of a wet cooling tower operating in the closed mode

The once-through system, though more efficient, causes thermal pollution. In addition, availability of huge quantity of water is shrinking. Closed loop systems are now almost universally preferred.

9.6- Cooling Towers

Cooling towers cool the warm water discharged from the condenser and feed the cooled water back to the condenser. They, thus, reduce the cooling water demand in the power plant. They can be either wet type or dry type.

9.6.1- Wet Cooling Towers

Wet cooling towers have a hot water distribution system that showers or sprays water evenly over a lattice of horizontal slats or bars called fill or packing (Fig. 9.15). The fill thoroughly mixes the falling water with air moving through the fill as the water splashes down from one fill level to another by gravity. Outside air enters the tower through louvres on the side of the tower. Intimate mixing of water and air enhances heat and mass transfer (evaporation), which cools the water.

More the water evaporates, more will be the cooling since the latent heat of evaporation is taken from water itself (evaporative cooling). Cold water is collected in a concrete basin at the bottom of the tower, from where it is pumped back to the condenser. Hot and moist air leaves the tower from the top.

Air entering the tower is unsaturated and as it comes in contact with the water spray, water continues to evaporate till the air becomes saturated. So, *the minimum temperature to which water can be cooled is the adiabatic saturation or wet bulb temperature of the ambient air*. All this temperature (WBT), air is 100% saturated and cannot absorb any more water vapor. Hence will be no further evaporation and cooling. The humid air while moving up comes in contact with warm water spray and so me air temperature rises.

A cooling tower is specified by (a) approach, (b) range, and (c) cooling efficiency. The **approach** (A) is defined as *the difference between the exit temperature of cooling water and the wet bulb temperature of the ambient air*, or

$$A = t_{c2} - t_{wb} \qquad (9\text{-}13)$$

Warm water from the condenser enters the cooling tower at temperature t_{c1} and is cooled to temperature t_{c2}, higher than the minimum value, the wet bulb temperature, t_{wb} and this unattainable temperature difference is the approach. The approach varies from 6 °C to 8 °C.

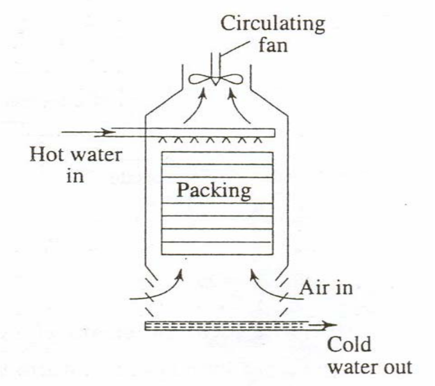


Figure 9.15 Packing or fill in a wet cooling tower

The **cooling range** or simple range (R) is defined as *the difference in temperatures of the incoming warm water (t_{c1}) and the exiting cooled water (t_{c2})*, or

$$R = t_{c1} - t_{c2} \qquad (9\text{-}14)$$

It is the range by which warm water from the condenser is cooled. The range varies from 6 °C to 10 °C.

The cooling efficiency is defined as the ratio of the actual cooling of water to the maximum cooling possible, or

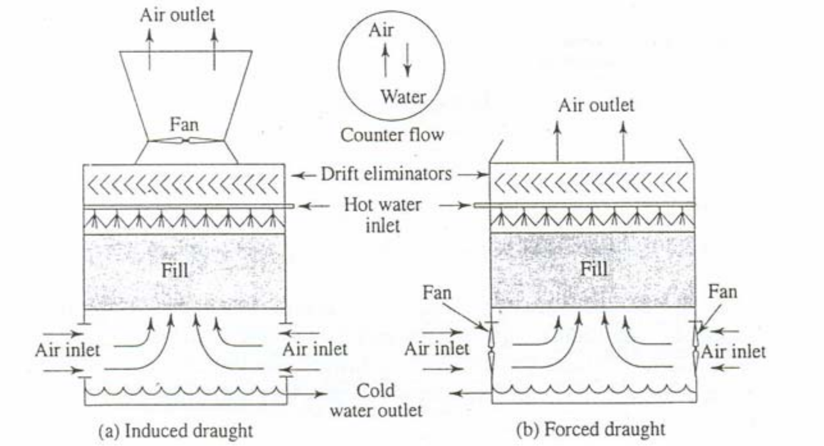
$$\eta_{cooling} = \frac{actual\,cooling}{maximum\ cooling\ possible} = \frac{t_{c1} - t_{c2}}{t_{c1} - t_{wb}} \tag{9.15}$$

The approach, range and cooling efficiency are the performance parameters of cooling towers.

Wet cooling towers can be either mechanical draught or natural draught cooling towers. In mechanical draught cooling towers, air is moved through the fill by one or more fans driven by motors. As in steam generators, the fans could be of the forced draught (FD) type or induced draught (ID) type. The FD fan is mounted on the lower side of the tower (Fig. 9.16). Since it operates on cooler air, it consumes less power. However, it has the disadvantages of (a) air distribution problems in the fill, often causing channeling of air flowing through paths of less flow resistance, (b) leakage and (c) recirculation of the hot and moist air back to the tower.

Most of the mechanical draught cooling towers for utility applications are of the induced draught type. The ID fan is located at the top of the tower Fig. 9.16 (a). Air enters the sides of the tower through large openings at low velocity and passes through the fill. Hot humid air is exhausted by the fan at the top to the atmosphere. It maintains the tower at a negative pressure thereby reducing leakage. The ID fans are large, having 0.6 to 10m in diameter. They are driven by electric motors at low speeds through reduction gearing. They are of the propeller type which deliver large volume flow of air at lower static pressures.

The blades are usually made of cast aluminium, stainless steel, or fibre glass so as to protect them from corrosion.



(a) Induced draught (b) Forced draught

Figure 9.16 Induced draught counterflow cooling towers

The air flow into the tower is more or less horizontal. However, in the fill the flow can be horizontal or vertical, in which case it is either a cross-flow or a counter-flow cooling tower. The main advantages of mechanical draught cooling towers are:

1. Low capital and construction costs.

2. Assured supply of the required quantity of air at all loads and climatic conditions,
3. Small physical structure.

In natural draught cooling towers, the flow of air occurs due the natural pressure head caused by the difference in density between the cold outside air and the hot humid air inside (Fig. 9.17(a) and (b)). Thus, the pressure head developed is

$$\Delta P_d = (\rho_o - \rho_i) g H \tag{9.16}$$

where H = height of the tower above the fill, ρ_o = density of outside air, and ρ_i = density of inside air.

Because of relatively small density difference, $\rho_o - \rho_i$ H must the large so as to result in the desired ΔP_d , which must balance the air pressure losses in the tower. Natural draught cooling towers are, therefore, very tall. The towerbody, above the water distribution system and the fill, is an empty shell of circular cross-section, but with a hyperbolic vertical profile. The hyperbolic profile offers superior strength and the greatest resistance to outside wind loading compared to other forms. Natural draught cooling towers are, therefore, often termed as hyperbolic towers. Made of reinforced concrete, they are an imposing sight and are conspicuous from a distance.

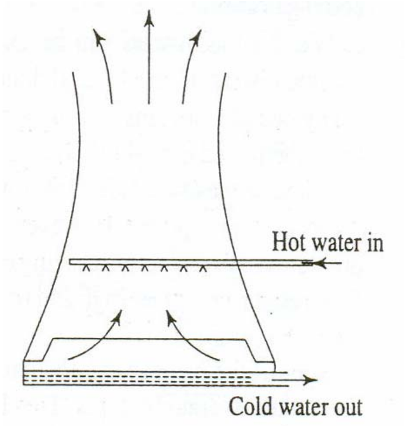


Figure 9.17 a Natural draught cooling tower

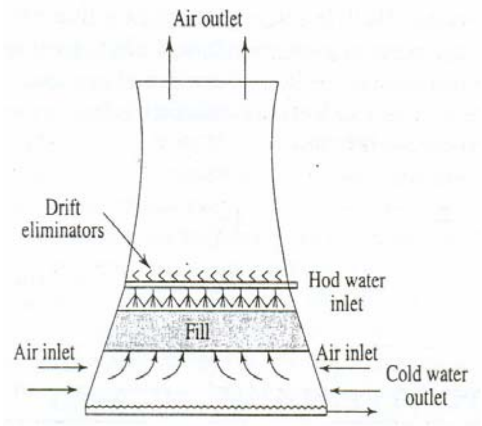


Figure 9.17 b Counterflow hyperbolic natural draught cooling tower

Mechanical draught cooling towers are preferred when the approach is low and a broad range of water flow is expected. The broad range is possible since they are made of multicell units with a variable air-flow fan. These towers are, therefore, more versatile and respond readily to changes in cooling parameters and demands.

Natural draught cooling towers are chosen (1) in cool, humid climates (low wet bulb temperature and high relative humidity), (2) when the wet bulb temperature is low and high condenser water inlet and outlet temperature, i.e. a broad range and a long approach, or (3) in heavy winter loads. However, their initial capital cost is high and occupy more space.

There is always some water loss in the cooling tower due to

a. evaporation

b. drift

c. blowdown

Water that evaporates leaves the tower along with air in the form of water vapor. The evaporation loss rate is 1-1.5 per cent of the total circulating water flow rate, Drift is fine water droplets entrained and carried by the air. This water is thus lost to the circulating water system. Drift eliminators are provided at exit to minimize the drift loss (Fig. 9.18). The baffles force the air to make a sudden change in direction. Heavier water particles separate out by gravity. Thus the drift loss is much less, about 0.03 per cent. To maintain a certain solid concentration, blowdown is necessary from the cold water basin at the bottom of the tower. The blowdown loss of water is also 1-1.5 per cent of the total water flow. To replenish these losses, makeup water (2-2.2% of water flow) is added.

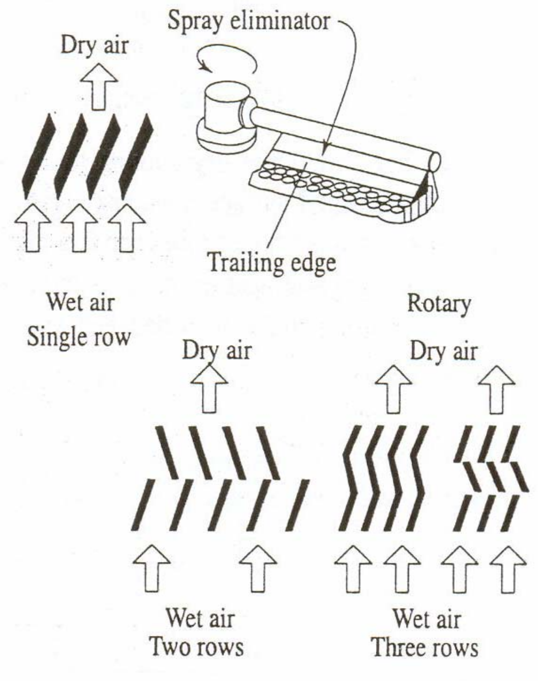


Figure 9.18 Types of drift eliminators

9.6.2- Dry Cooling Towers

Dry cooling towers are employed where cooling water is not available in plenty, even for the use of makeup. In a dry tower, warm water from the condenser flows through finned tubes over which the cooling air is passed. Heat is rejected to the air as water is cooled.

There are two basic types of dry cooling towers: direct and indirect. In a direct dry cooling tower (Fig. 9.19), turbine exhaust steam flows into a large steam header and then to a large number of finned tubes which are cooled by atmospheric air blown over them by an FD fan.

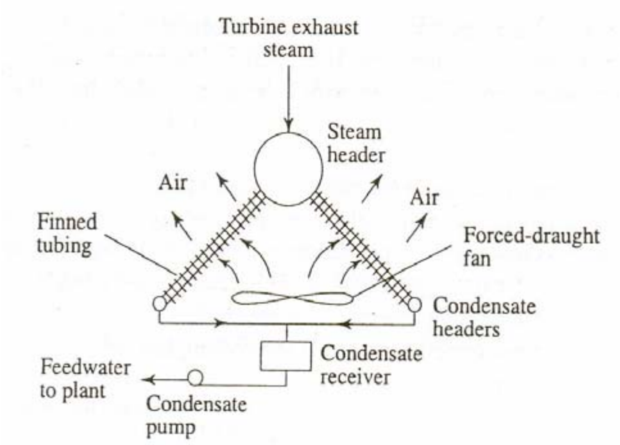


Figure 9.19 Schematic of a direct dry-cooling tower

For an indirect dry cooling tower there are three designs. The first uses a conventional surface condenser (Fig. 9.20). Warm water from the condenser flows through Finned tubes and is cooled by atmospheric air blown over the tubes. There are two heat exchangers in series, one between steam and water in the condenser and the other between water and air in the tower.

The second design uses a direct contact spray condenser (Fig. 9.21). The turbine exhaust steam enters the open condenser and the cold circulating water is sprayed into the steam for intimate mixing. The condensate falls into a bottom

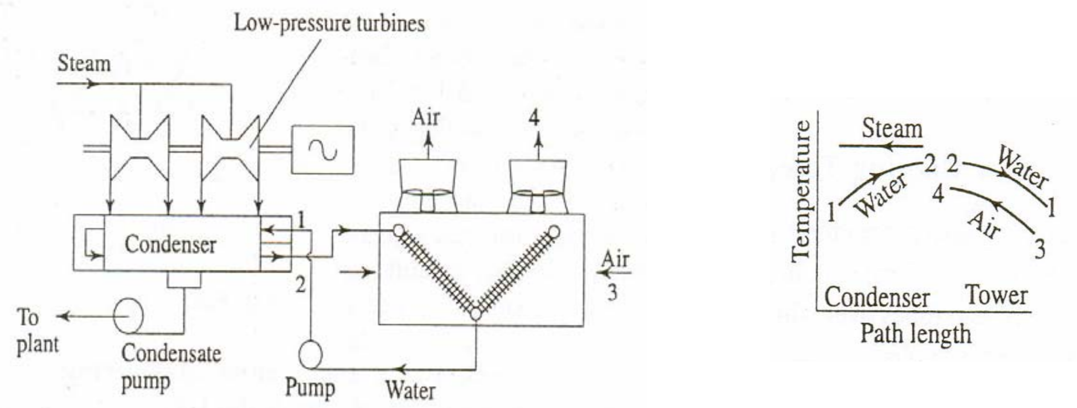


Figure 9.20 Schematic of an indirect dry-cooling tower with a conventional surface condenser

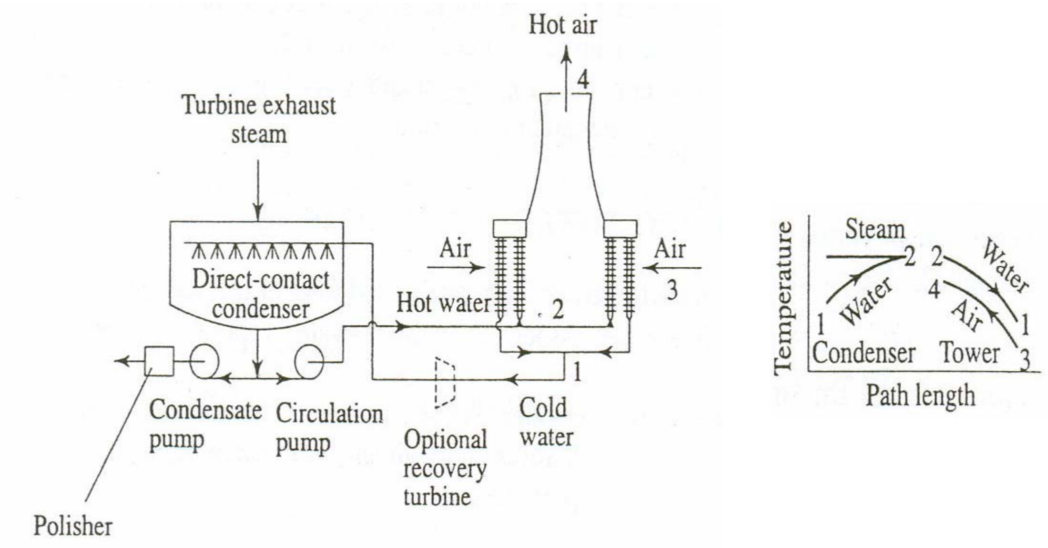


Fig. 9.21 Schematic of an indirect dry-cooling tower with an open-type condenser

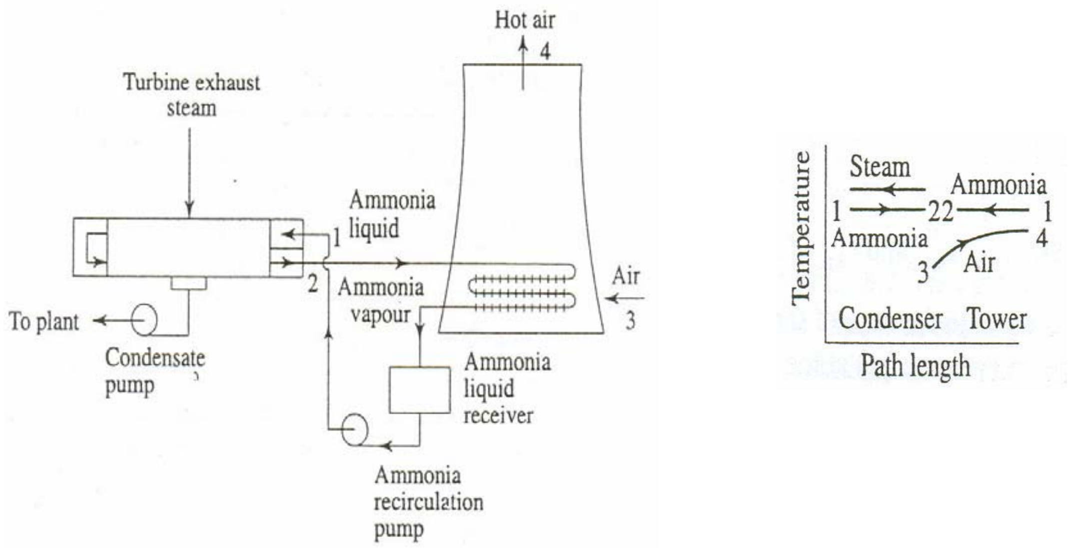


Figure 9.22 Schematic of an indirect dry-cooling tower with a surface condenser having ammonia as the coolant

Dry cooling towers have attracted much attention now-a-days. Plants can be erected without regard for large supplies of cooling water. Typical sites are near the sources of abundant fuel where there is no sufficient water. Their disadvantages are that they are not so efficient as evaporative cooling, and that there is an increase in turbine exhaust pressure and a decrease in cycle efficiency. However, as power plants grow bigger and water sources are dwindling, they are likely to receive greater attention in future.

9.7 Cooling Tower Calculations

In a wet cooling tower, ambient air is used to cool the warm water exiting the condenser. Properties associated with air-water vapor mixture may now be discussed.

Atmospheric air is considered to be a mixture of dry air and water vapor, If P_a and P_w are the partial pressures of dry air and water vapor, respectively, then by Dalton's law of partial pressures,

$$P_a + P_w = P$$

where P_a is the atmospheric pressure. Since P_w is very small, the saturation temperature of water vapor at P_w is less than atmospheric temperature, t_{atm} (Fig. 9.23). So, the water vapor in air exists in the superheated state and air is said to be unsaturated. Saturated air holds the maximum water vapor at the given temperature. If the temperature is increased, then only it can accept more water vapor (till it is again saturated). If the temperature of saturated air is decreased, some water vapor will condense and the new cooler air would also be saturated. At 15 °C, the partial pressure of water vapor in saturated air (P_s) is equal to 1.705 kPa from steam tables. So, the partial pressure of dry air (P_a) is (101.325 - 1.705) or 99.62 kPa.

Relative Humidity (RH, is equal to the ratio of the partial pressure of water vapor (P_w) in air to the saturation pressure (P_s) at the air temperature, or

$$RH = \frac{P_w}{P_s} \tag{9.17}$$

For air at 15°C and 50% RH

P_w= 0.5 x 1.705 = 0.853 kPa

P_a=101.325-0.853 = 100.472 kPa

Specific humidity or humidity ratio (W) or moisture content is the mass of water vapor per unit mass of dry air. If $\dot{m}_a$ = mass of dry air and $\dot{m}_v$ = mass of water vapor, then

$$W = \frac{m_v}{m_a} = \frac{\dot{m}_v}{\dot{m}_a} \ (kg/kg\ d.a) \tag{9.18a}$$

Specific humidity is maximum when air is saturated at the given temperature t, or

$$W_{max} = W_s = \frac{\dot{m}_{v,s}}{\dot{m}_a} \tag{9.18b}$$

If dry air and water vapor behave as ideal gases,

$$P_w V = m_v R_w T$$

$$P_a V = m_a R_a T$$

$$W = \frac{m_v}{m_a} = \frac{P_w}{P_a} \times \frac{R_a}{R_w} = \frac{P_w}{P - P_w} \times \frac{8.3145/28.96}{8.3145/18} = 0.622 \frac{P_w}{P - P_w} \tag{9.20}$$

The degree of saturation (μ) is the ratio of actual specific humidity to the saturated specific humidity, both at the same temperature T.

$$\mu = \frac{W}{W_s} = \frac{P_w}{P_s} \times \frac{P - P_s}{P - P_w} \tag{9.20}$$

Dew point temperature (dpt, t_{dp}) is the saturation temperature at the partial pressure of water vapor. When air is cooled at constant pressure, the temperature at which water vapor starts condensing is called the dew point temperature (Fig. 9.23).

Dry bulb temperature (dbt, t_{db},) is the temperature recorded by a thermometer with a dry bulb,

Wet bulb temperature (*wbt*, t_{wb}) is the temperature recorded by a thermometer when the bulb is enveloped by a cotton wick saturated with water. A psychrometric measures both *dbt* and *wbt*. If air flowing over the bulb is relatively dry, water in the wick would evaporate at a rapid rate, cooling the bulb and resulting in a much lower reading than if the bulb were dry. If the air is humid, the evaporation rate is slow, the lowest temperature recorded by the moistened bulb is the *wbt*. If air is saturated, i.e. *RH* = 100%, *wbt = dbt*. The *wbt* is also called adiabatic saturation temperature.

Psychrometric chart (Fig. 9.24) is a graphical plot with specific humidity and partial pressure of water vapor as ordinates and *dbt* as abscissa. The volume of the mixture, *wbt*, relative humidity and enthalpy of the mixture appear as parameters. Any two of these parameters fix the condition of the mixture. The chart is plotted for one barometric pressure, say 760 mm Hg. The constant *wbt* line represents the adiabatic saturation process and coincides with the constant enthalpy line.

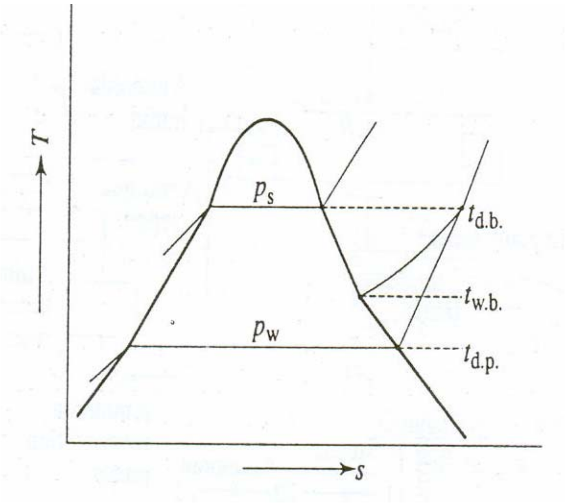


Figure 9.23 $t_{d,b}$, $t_{w,b}$ of air

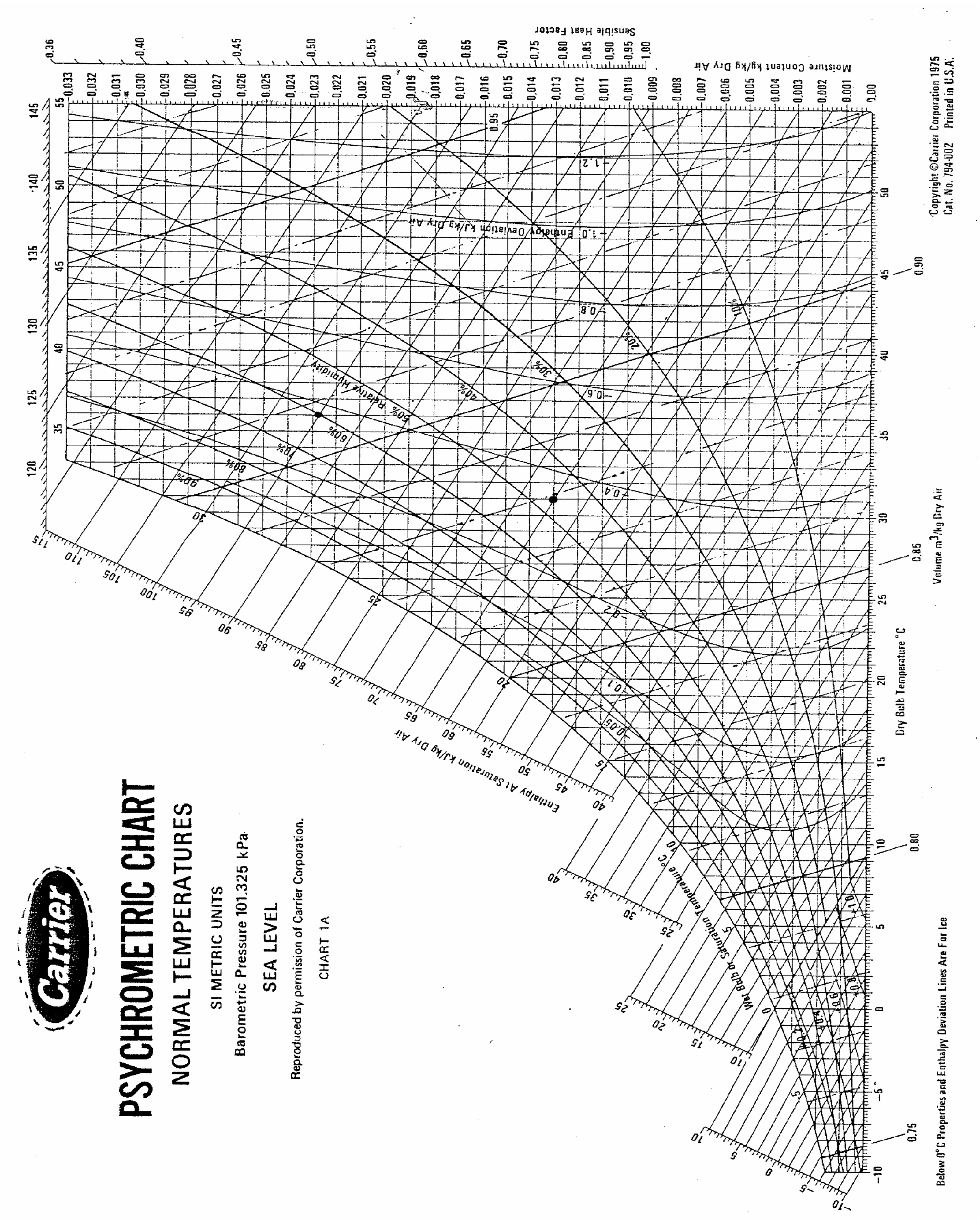


Figure 9.24 Psychrometric chart

The cooling tower utilizes the phenomenon of evaporative cooling to cool the warm water below the *dbt* of air. However, water never reaches the minimum temperature, *i.e.*, the *wbt* since an excessively large cooling tower would then be required. Also, since warm water is continuously introduced to the tower (Fig. 9.25), the equilibrium conditions are not achieved, and the *dbt* of air is increased. Hence, while the water is cooled, the air is heated and humidified.

if $\dot{m}_{mu}$ is the make-up water supplied to replenish the evaporation loss (Fig. 9.25), then

$$\dot{m}_{mu} = \dot{m}_a\left(W_2 - W_1\right) \tag{9.21}$$

where $\dot{m}_a$ = the mass flow rate of dry air, kg/s, W = specific humidity, kg water vapor per kg dry air.

By energy balance,

$$\dot{m}_a h_1 + \dot{m}_{c3} h_{w3} + \dot{m}_{mu} h_w = \dot{m}_a h_2 + \dot{m}_{c4} h_{w4}$$

$$\dot{m}_c\left(h_{w3} - h_{w4}\right) = \dot{m}_a\left(h_2 - h_1\right) - \dot{m}_a\left(W_2 - W_1\right)h_w \tag{9.22}$$

$\dot{m}_{c3} = \dot{m}_{c4}$ = circulating water flow rate, kg/s, h_w = enthalpy of water, kJ/kg ; and h = enthalpy of dry air, kJ/kg

$$\text{Range}, R = t_{w3} - t_{w4} = \frac{\dot{m}_a}{\dot{m}_c c_{pw}}\left[\left(h_2 - h_1\right) - \left(W_2 - W_1\right)h_w\right] \tag{9.23}$$

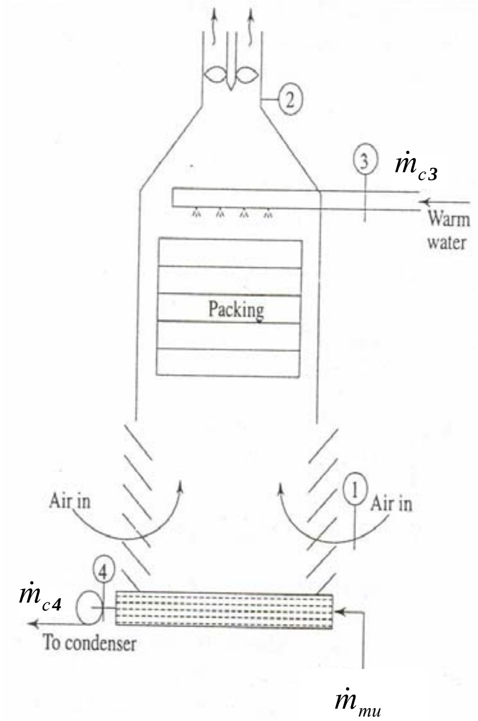


Figure 9.25 Cooling tower calculation

$$\text{Approach } (A) = t_{w3} - t_{w4} \tag{9.24}$$

Most towers are rated in practice for wet-bulb approach between 6 °C and 8 °C Values of 6 to 10 °C are common for the cooling range. The relationships among t_{c1}, t_{c2}, R and A are shown in Fig. 9.26. Vaporization, hence cooling, takes place as long as the partial pressure of water vapor at the cooling water surface exceeds the partial pressure of water vapor in the bulk air stream.

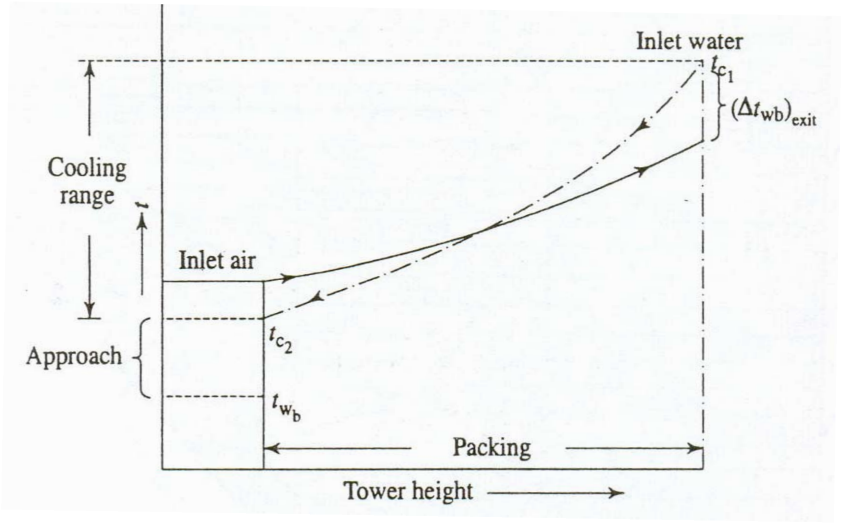


Figure 9.26 Temperature relationship in a counterflow cooling tower

Example 9.1

A surface condenser receives 250 *t/h* of steam at 40 °C with 12% moisture. The cooling water enters at 32 °C and leaves at 38 °C. The pressure inside the condenser is found to be 0.078 bar. The velocity of circulating water is 1.8 m/s. The condenser tubes are of 25.4 mm od and 1.25 mm thickness. Taking the overall heat transfer coefficient as 2600 W/m^{2}K, determine (a) the rate of flow of cooling water, (b) the rate of air leakage into the condenser shell, (c) the length of tubes, and (d) the number of tubes.

Solution

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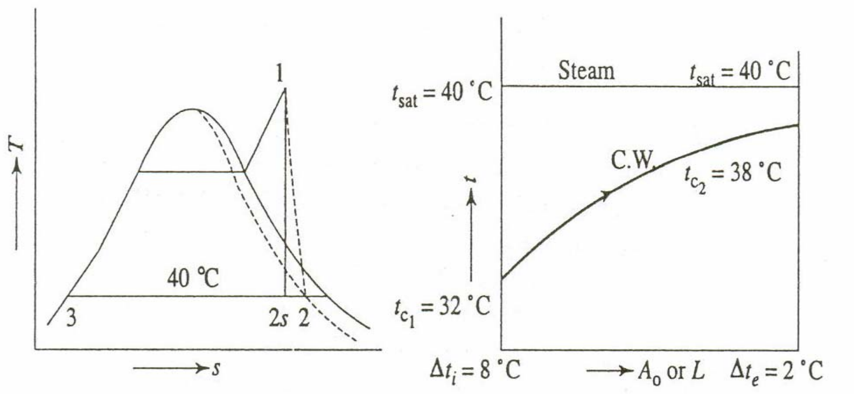
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$t_{sat} = 40$ °C
C.W.
$t_{c_2} = 38$ °C
t
$t_{c_1} = 32$ °C
$\Delta t_i = 8$ °C
A_0 or L
$\Delta t_e = 2$ °C

(a) (b)

Example 9.2

Water at 30 °C flows into a cooling tower at the rate of 1.15 kg per kg air. Air enters the tower at the *dbt* of 20 °C and a relative humidity of 60 % and leaves it at a *dbt* of 28 °C and 90% relative humidity. Makeup water is supplied at 20 °C. Determine (a) the temperature of water leaving the tower, (b) the fraction of water evaporated, and (c) the approach and range of the cooling tower.

Solution

Example 9.3

Warm water at 45 °C enters a cooling tower at the rate of 6 kg/s. An ID fan draws 10 m^3/s of air through the tower and absorbs 4.90 kW. The air entering the tower is at 20 °C *dbt* and 60% relative humidity. The air leaving the tower is assumed to be saturated and its temperature is 26 °C. Calculate the final temperature of the water and the amount of makeup water required per second. Assume that the pressure remains constant throughout the tower at 1.013 bar.

Solution

Revision questions

9.1-A condenser for a steam power plant receives 185 t/h of steam at 40 °C, 92% quality. Cooling water enters at 33 °C and leaves at 37 °C. The condensate leaves at 39 °C. The pressure inside the condenser is found to be 0.077 bar. Find (a) the cooling water flow required in m^3/s, and (b) the air leakage into the condenser in kg/h.

9.2- A 4000 kW marine steam turbine has a condenser flow at full load of 12000 kg/h. The quality of steam at the exhaust of the turbine is 0.85. The pressure maintained in the condenser is 0.07 bar. Seawater used for circulation is at 21 °C. The terminal temperature difference between steam and outgoing water is 5.7 °C. The overall heat transfer coefficient between condensing steam and cooling water is 3257 W/m^2 K. The circulating water velocity is maintained at 1.7 m/s. The condenser tubes are of 13 mm id and 1.22 mm thickness. Determine (a) the rate of flow of cooling water, (b) the length of tubes, and (c) the number of tubes in the condenser.

9.3- A surface condenser receives 200 t/h of steam at 40 °C with 10 % moisture. The cooling water enters at 33 °C and leaves at 38 °C. The pressure inside the condenser is found to be 0.078 bar. Neglecting any subcooling, determine (a) the condenser cooling water flow in kg/s, (b) air ejector capacity in kg/s, and (c) the surface area of the condenser. Take U_o = 2970 W/m^2K

Ans. (a) 5748 kg/s, (b) 4.623 kg/s, (c) 10, 150 m^2

9.4- A surface condenser receives 20 t/h of dry saturated steam at 36.2 °C. The condensate leaves at 34.6 °C. The volume of air and vapor leaving the air cooler section at 29 °C is 3.8 m^3/min. Assuming constant pressure throughout the condenser and neglecting the partial pressure of air at inlet to the condenser, calculate (a) the mass of air removed per 10000 kg of steam, (b) the mass of steam condensed in the air cooler per minute, and (c) the rate of heat removal by the cooling water.

Ans. (a) 2.63 kg, (b) 0.492 kg, (c) 13451 kW

9.5- Water at 60 °C leaving the condenser at the rate of 22.5 kg/s is sprayed into a natural draught cooling tower and leaves it at 27 °C. Air enters the tower at 1,013 bar, 13 °C and 50% relative humidity and leaves it at 38 °C, 1.013 bar and saturated, Calculate (a) the air flow rate required in m^3/s, and (b) the makeup water required in kg/s.

Ans. (a) 21 m^3/s, (b) 1 kg/s

9.6- Water from a cooling system is itself to be cooled in a cooling tower at a rate of 2.78 kg/s. The water enters the tower at 65 °C and leaves a collecting tank at the base at 30 °C. Air flows through the tower, entering the base at 15 °C, 0.1 MPa, 55% RH and leaving the top at 35 °C, 0.1MPa, saturated. Makeup water enters the collecting tank at 14 °C. Determine the air flow rate into the tower in m^3/s and the makeup water flow rate in kg/s.

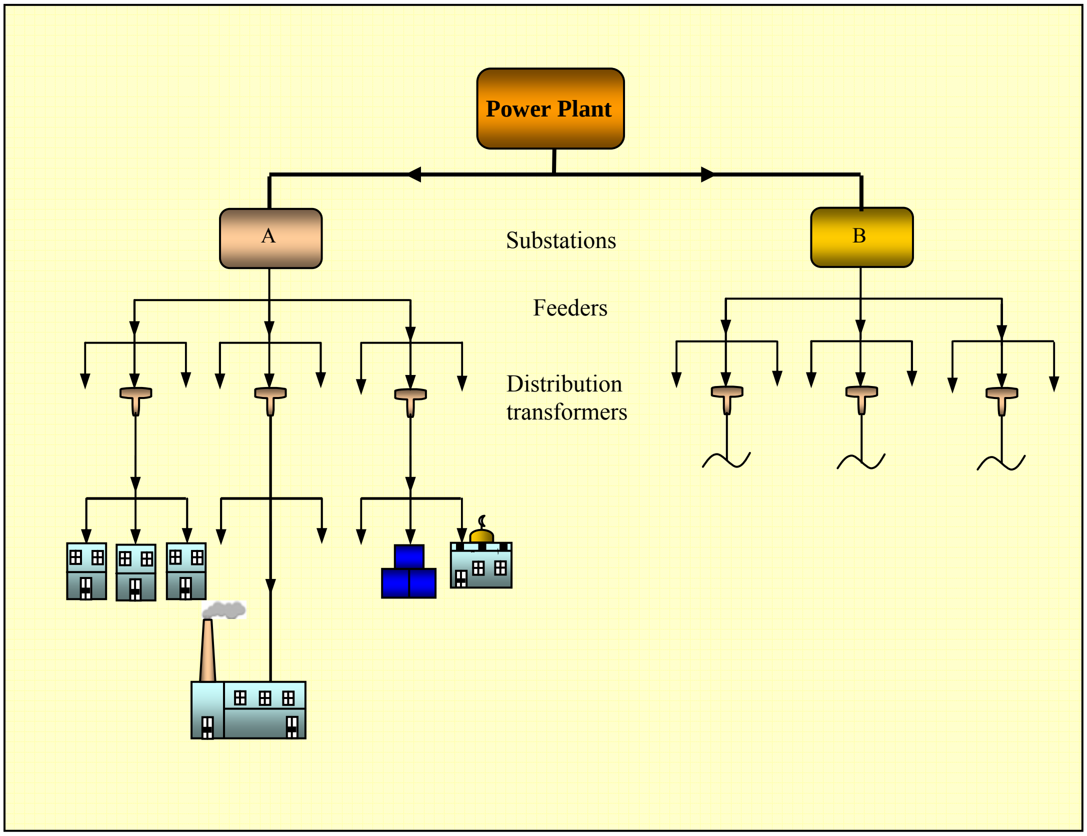
Ans. 3.438 m^3/s, 0.129 kg/s

YANBU INDUSTRIAL COLLEGE

Power Plant Engineering and economy

MET401

Department of Mechanical Engineering Technology



Chapter X – Economics of power generation

2010

PhD, MEMgt, BSc(hons) , P.E., MASME, MIPENZ, Reg. Eng

Chapter X

Economics of power generation

10.1-Terms and Factors

The manner in which the energy output of a single power plant is subdivided and sent to thousands of individual customers is diagrammed in Figure (10-1). First the energy is sent to ***substations*** which are located at the ends of the primary distribution system. ***Feeder*****s** which lead out from the substations, may each supply several distributions. ***Transformers*** only one of which (per feeder) is actually included in the diagram. Each transformer is connected by short-voltage lines to the systems of one or more customers.

There are number of terms which need definition. The basic information is, of course, the operating data of demanded load plotted against time sequence. This is commonly referred to as a *Load curve*. See Figure (10-2).

The load duration curve, Figure (10-3) is obtained from the daily curve. The co-ordinates of this curve may extend from 0 to maximum demand in kW or from 0 to 100% maximum demand. The co-ordinates reach from 0 hour to 8760 hour. The number of hours during which 1000 kW, 2000 kW, 3000 kW, etc. is demanded is recorded from the daily load curves, then totaled from the year and plotted as the *Loa d duration curve*. Hence the interpolation of point A (see Figure 10-3) is 12000 kW were demanded for 4380 hour of the year, or that thought 50% of the year, not more than 40% of the maximum demand was required. If a certain standby unit of the station had been called upon to carry all loads above 18000 kW, the load factor for that unit could be found by dividing the area *abc* expressed (in kW h) by that units maximum demand (12000 kW) times its hours of operation, that is, the length *bc* measured in hours (1576.8)=(0.18 x 8760).

Load factor

The daily, monthly, or annual load factor *is the average load over the time specified divided by the maximum peak.* This peak is seldom taken as the maximum instantaneous value but rather as maximum 15 min, half hour or hour - long peak.

$$\text{Load factor} = \frac{\text{average Load}\left(L_{av}\right)}{\text{maximum Load}\left(L_{max}\right)} \tag{10-1}$$

$$\text{Average Load} = \frac{\text{area}}{\text{length}} = \frac{\text{kWh}}{\text{h}} \ \left(\text{kW}\right)$$

Now each customer has a *connected load* and also there is a maximum demand of the customer. If all the devices run to their fullest extent simultaneously, the maximum demand then would equal the connected load.

However, experience demonstrates that the actual maximum demand of a consumer will be less than his connected load since all the devices never run at full load at the same time.

Demand factor

The relationship of maximum demand and connected load is measured by,

$$\text{Demand factor } (d) = \frac{\text{maximum demand}}{\text{connected load}} < 1 \tag{10-2}$$

Demand factor depends upon the nature and activities of the consumer and to some extent upon the location of the power station.

Group Diversity factor

The time distribution of maximum demands for similar types of consumers is measured by the diversity factor

$$\text{Group diversity factor } (D) = \frac{\text{sum of individual max demands}}{\text{actual max demand of consumer group}} > 1 \qquad (10\text{-}3)$$

Peak Diversity factor

The peak demand of a system is made up of the individual demands of the devices that happen to be functioning at the time of the system peak demand, the demand of a particular group of similar consumers is seldom at the maximum value that it may reach at some other time of the year. This diversity is measured by,

$$\text{Peak diversity factor } (r) = \frac{\text{atual max demand of consumer group}}{\text{demand of consumer at time of system peak demand}}$$

$$r = \frac{\sum Peak}{Peak\ Load} > 1 \qquad (10\text{-}4)$$

System maximum demand

If there are more than one group

Then, $L_1+L_2+L_3+ \dots\dots +L_n$

where (n) number of the groups.

L demand of each type of consumer at time of system maximum demand

Then

$$L_{max} = \frac{\sum c\ell . \times d}{r \times D} \qquad (10\text{-}5)$$

where $c\ell$ connected load of the group.

And $$L_1 = \frac{\sum c\ell_1 \times d_1}{r_1 \times D_1} \qquad (10\text{-}6a)$$

$$L_2 = \frac{\sum c\ell_2 \times d_2}{r_2 \times D_2} \qquad (10\text{-}6b)$$

Where

$c\ell_1$: Connected loads of group 1

$c\ell_2$: Connected loads of group 2

d_1: demand factors of group 1

d_2: demand factors of group 2

r_1: Peak diversity factor of group 1

r_2: Peak diversity factor of group 2

D_1: group diversity factors among consumers in group 1

D_2: group diversity factors among consumers in group 2

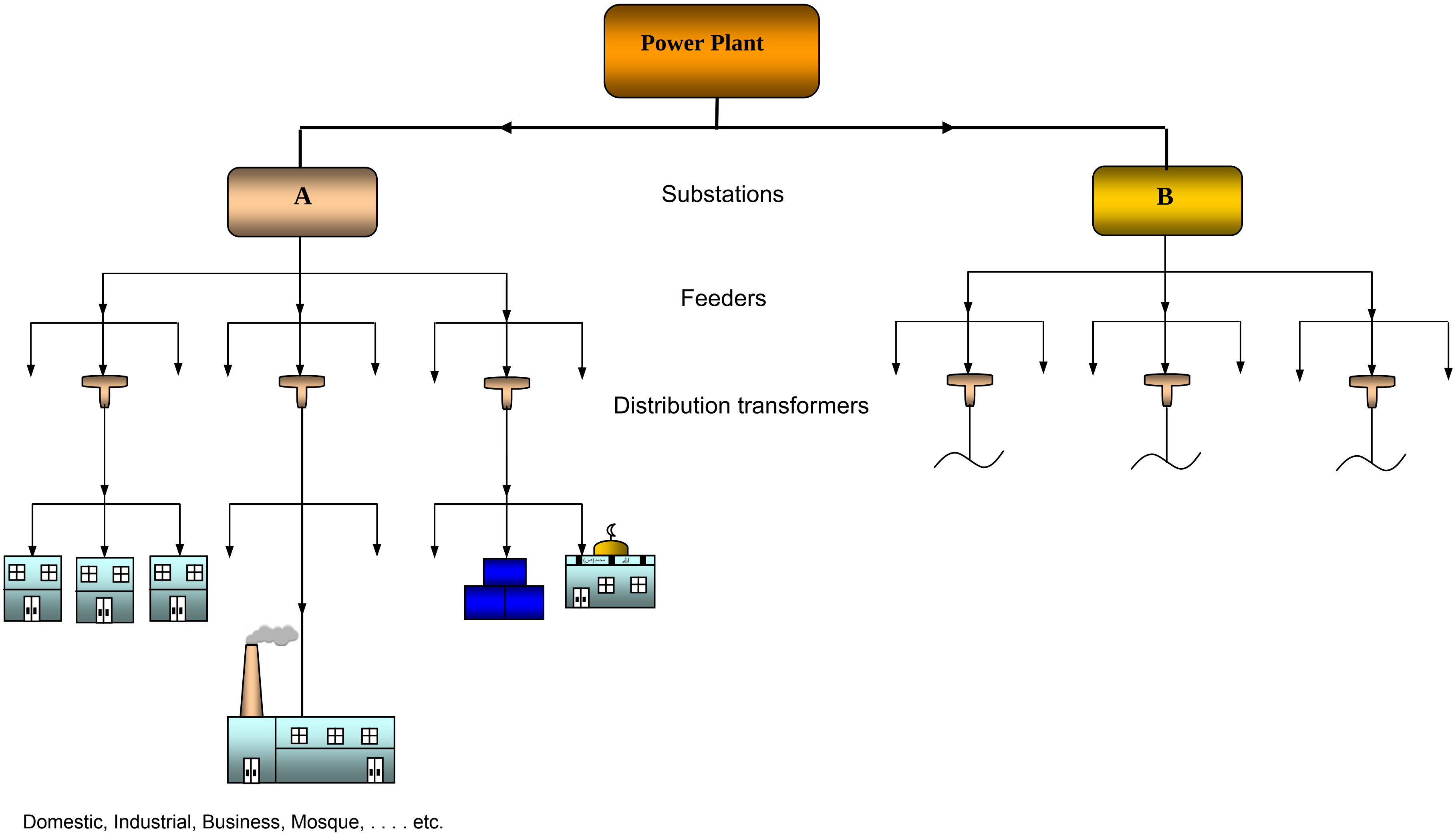


Figure (10-1) Elements of electrical power system

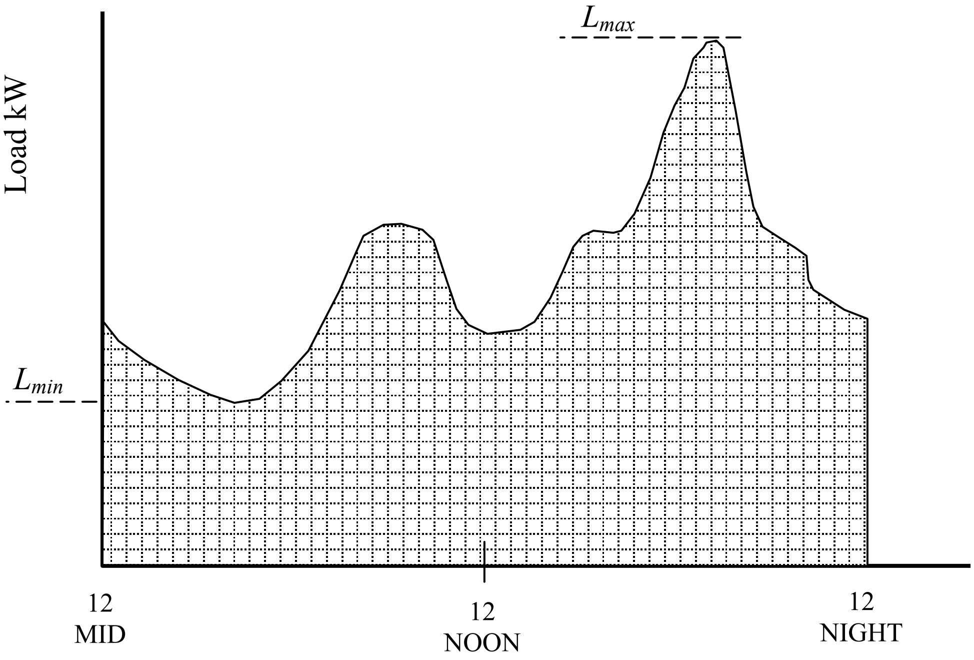


Figure (10-2) daily load curve

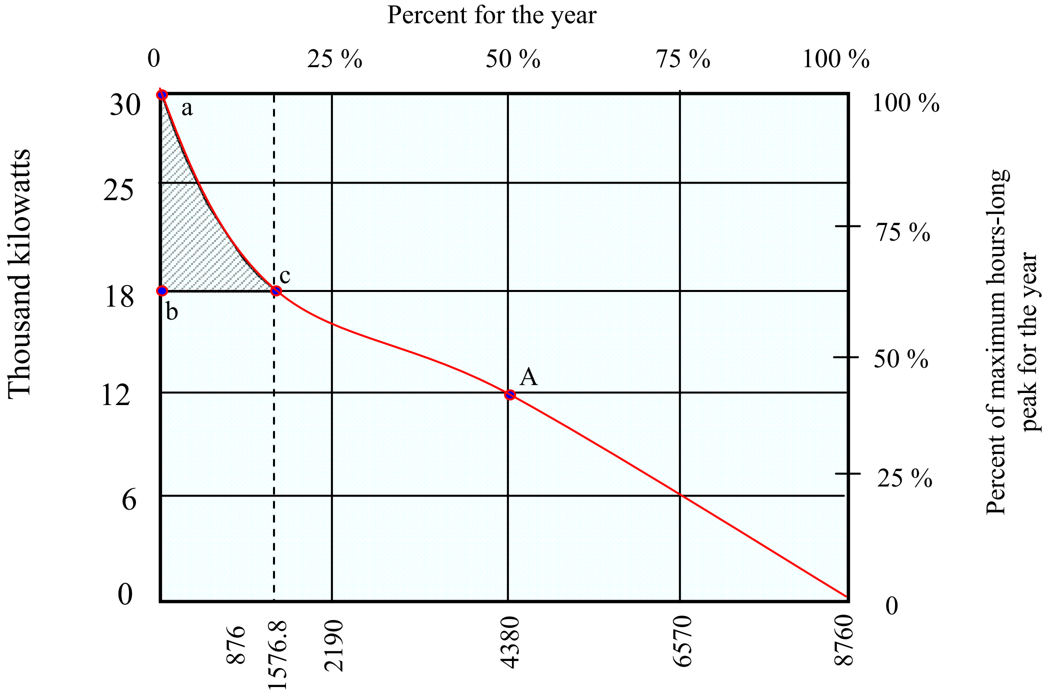


Figure (10-3) Load duration curve

Example 10.1

A new housing development is to be added to the lines of a public utility. There are 1000 apartments each having a connected load of 4 kW, also stores and services are included of the characteristics shown in the following tabulation.

Store or service	Connected load kW	Demand factor (d) %
1- laundry	20	0.68
2-mosques	10 each	0.56
1-resturant	60	0.52
1-bookstore	5	0.66
1-dry-goods store	7	0.76
2-drugsstores	10 each	0.79
2-grocery stores	5 each	0.73
1-shoe store	2	0.67
1-cloting store	4	0.53
1-theater	100	0.49

The demand factor of the apartments is 45%. The group diversity factor of the residential load for this system is 3.5, and the peak diversity factor is 1.4, and the commercial – load group diversity factor is 1.5 and the peak diversity factor is 1.1. Find the increase in peak demand on the total system delivery from the station bus resulting from addition of this development on the distribution system. Assume line losses at 7.5% of delivered energy.

Solution

Max. demand
kW
13.6
11.2
31.2
3.3
5.3
15.8
7.3
1.3
2.1
49

140.1 kW

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Example 10.2

A daily load curve which exhibited a 15 min peak of 6520 kW is drawn to scales of 1 cm = 4h, and 1cm = 1000 kW. The area under the curve is measured by plainmeter and found to be 12.17 cm^2. Find the load factor based on the 15 min peak.

Solution

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Capacity factor

Another term which is useful in variable load studies is the capacity factor. Since the load and diversity factors are not involved with the reserve capacity of the plant or system, a factor is needed which will measure the reserve, likewise the degree of utilization of the installed equipment. This need is met by the Capacity factor which is defined as the actual energy production divided by the maximum possible energy that might have been produced during the same period. Thus the annual capacity factor would be the annual kilowatt hours produced divided by the kilowatts of plant capacity times hours of the year.

Then, Capacity factor = $\frac{L_{av}}{C_{ap}}$ (10-7) where C_{ap} Capacity of the plant

Reserve factor

The difference between load and capacity factors is an indication of reserve capacity. And when its divided by the capacity factor will introduce a new factor which called *Reserve factor*.

$$\text{Reserve factor} = \frac{C_{ap} - L_{max}}{L_{max}} \tag{10-8}$$

Load factor become identical when the peak load is equal to the capacity of the plant.

The relationship between the two factors is evidently

$$\text{Capacity factor} = \frac{L_{max}}{C_{ap}} \times Load\ Factor \tag{10-9}$$

Utilization factor

The factor that measures the use made of the total installed capacity of the plant is called the Utilization factor.

$$\text{Utilization factor} = \frac{L_{max}}{C_{ap}} \tag{10-10}$$

A modification of capacity factor is obtained by using only the actual number of hours the plant was in operation. This is the plant use factor defined as follows, the annual kilowatt hours production divided by the kilowatts of capacity times number of hours plant was in operation.

Example 10.3

The load shown by the duration curve Figure (10.3) is carried by a base load station having a capacity of 18000 kW and a station having a capacity of 20000 kW the annual load factors use factors , and capacity factors of these two power stations will be evacuated .
The shaded area of the Figure (10.3) represents the annual output of the standby station. The remaining area under duration curve is the base load station output. These area have to be measured and interpreted in terms of kW h. This being done , it is found that

Annual standby station output = - - - - - z - - - kWh

Annual base – load station output = - - zz- - - kWh

Peak load on the standby station is shown to be 12000KW hours that the standby station was in use during the year are measured from the figure and found to be 1576.8 h.

Standby station factor

$$\text{Load factor} = \frac{z}{12000 \times 1576.8}$$

$$\text{Use factor} = \frac{z}{20000 \times 1576.8}$$

Capacity factor = $\frac{z}{20000\times 8760}$

Base-load station factors

Load factor = $\frac{zz}{18000\times 8760}$

Since this base-load station has no reserves above peak load , and is on continuous use , its use and capacity factors are also the same value of load factor

10.2- Load - Curve analysis

The area under the curve of a daily chronological load curve Figure (10-4) measures the total energy consumed by the load during the day. This energy is evaluated by $\int_0^{24} kW\, dt$, the unit being the kilowatt hour. Fundamentally the load duration curve is nothing more than a rearrangement of all the load elements of a chronological curve in the order of descending magnitude Figure (10-5). The areas under the load duration and corresponding chronological curves are equal. Since it is important to determine the equation of either load curve, the area or energy is determined graphically.

Figures (10-4 &10-5) show the graphical method of constructing the load - duration curve from the chronological curve. The co-ordinate of the duration curve is laid to equal the number of hours in the chronological curve, in this case 24 hours. The criterion of plotting the duration curve makes the co-ordinate at any load ordinate equal to the length of the co-ordinate intercepted by that load ordinate on the chronological curve. Thus, at the maximum demand or peak load, the intercept is one point which will be plotted at 0 hour. At load *a* the intercept is *a1* hours is plotted at *a* hours on the duration curve see Figure(10-4 & 10-5). At load *b* the intercept is a total of (*b1*+*b2*) hours and is plotted accordingly. At minimum load *c* the intercept covers the entire period of 24 hours of 24 hours. Any point of the load-duration curve is a measure of the number of hours in a given period during which the given load and higher loads have prevailed. If the chronological curve indicated a constant demand during the entire day, it would be of rectangular shape and the load-duration curve would be an exact duplicate. The load duration curve can be extended to include any period of time. By laying out the coordinate to include 8760 hours. The variation and distribution in demand for an entire year can be summarized in one curve. Knowing the amount of energy between given demand levels on the load curve, can be most easily found by plotting a load -energy curve Figure (10-6) which derived from either the chronological or the duration curve. The load energy curve plots the cumulative integration of the area under the load curve starting at zero load vs the particular load. Expressed mathematically, it is a plot of $\int_0^{L} \tau\, dL$ versus L, where τ is the hours of duration. Referring to the Figures (10-4,10-5 & 10-6), the minimum load for the load curve is *c* kW. Between 0 and *c* kW successive integrations are directly proportional to the load-energy curve plots as a straight line between these limits. For the next element of load above the minimum *c* kW, the elementary area is less than the elementary area at *c* kW. Hence the load-energy curve will deviate from a straight line and bend upward.

Since the elementary areas subsequently decrease constantly as the kilowatt load increase, the curve has a continually increasing slope up to the peak load. Obviously the energy corresponding to the peak load will equal the total energy in the load curve. To find the energy included between the load values of *a* kW and *b* kW on the load curves, find the corresponding energies at *a* and *b* kW hour. The energy quantity will then be (a - b) kW hour. Extending the straight line portion to interest the total kilowatt hour co-ordinate determines the average load L_{av}. This intersection represents the load value that the chronological and duration curves would have if they were of rectangular shape and contained the same total energy.

From Figure (8.4)

$$L_{av} = \frac{Area}{24} \qquad (10\text{-}11)$$

$$Area = \int_{0}^{24} L\,dt \qquad (10\text{-}12$$

If the relation between the Load and time is linear, then

$$L_{av} = \frac{1}{2}\left(L_{max} - L_{min}\right) \qquad (10\text{-}13)$$

From Equation (10.1)

$$\text{Load factor} = \frac{L_{av}}{L_{max}} = \frac{E/h}{L_{max}} \qquad (10\text{-}14)$$

where

L_{max}: Peak load for period, either instantaneous or average, for periods of 1 hour or less.

E: total energy in load curve for period.

h: Total number of hours in period

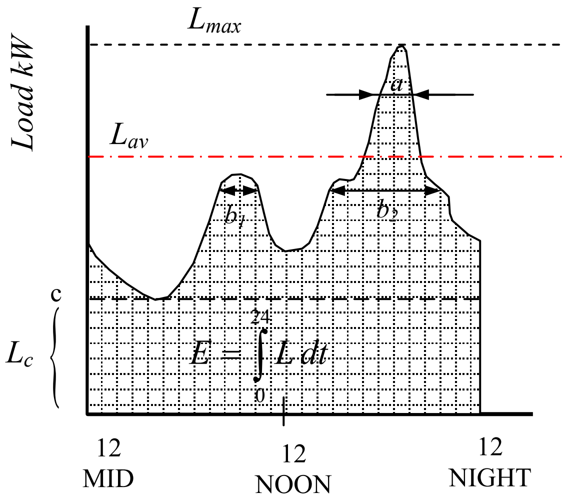

Figure (10-4)

Chronological Load Curve

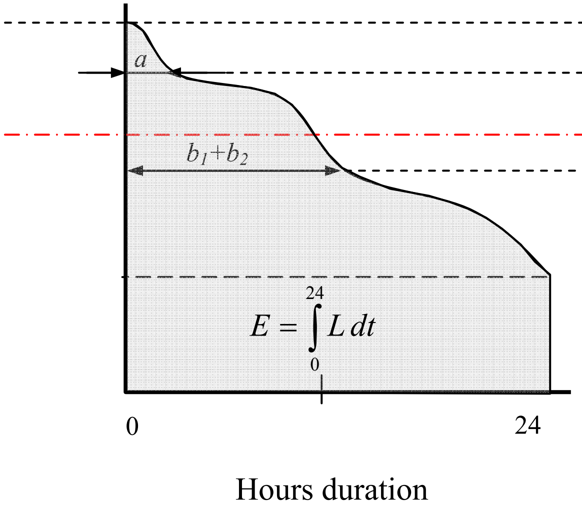

Figure (10-5)

Load duration curve

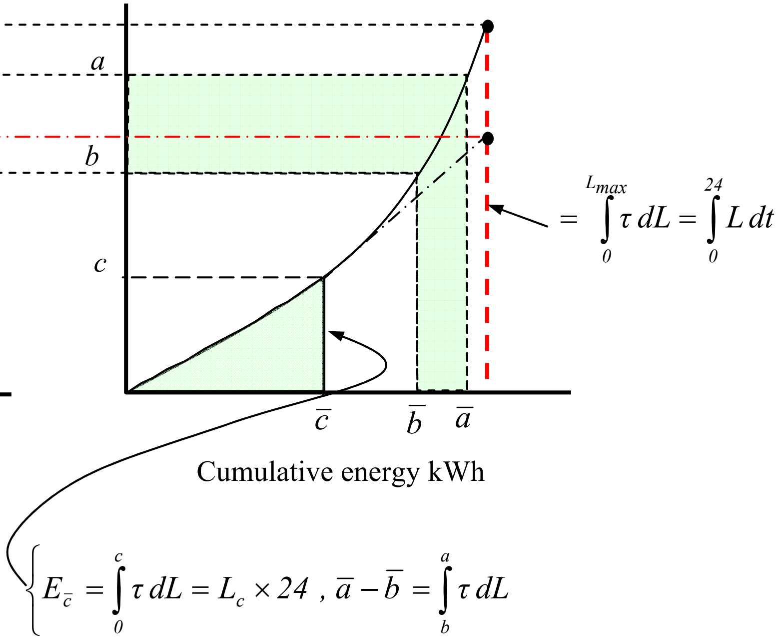

Figure (10-6)

Energy load curve

10.3- Station Performance Characteristics

Performance of generating plants is compared by their average thermal efficiencies over a period of time.

$$\eta_{th} = \frac{\text{useful energy output during the period}}{\text{total energy input for the period}} \qquad (8\text{-}15)$$

This measure of performance varies with uncontrollable conditions such as cooling-water temperature, shape of load curve, total output and quality of fuel; hence it's not a satisfactory standard of comparison unless all plant performances are corrected to the same controlling conditions.

Plant performance is most precisely described by the following curves.

1- Input – Output curve.
2- Heat rate curve (*HR*).
3- Incremental rate curve (*IR*).
4-

Figure (10–7) represents the relationship between input and output, the trend of such a curve is defined by.

$$I = a + bL + cL^2 + dL^3 + \dots\dots\dots + nL^r \qquad (8\text{-}16)$$

When the load (output) is equal to zero ($L = 0$), there is an input which is (a) represents the amount of energy required to keep the apparatus functioning. This energy dissipates as functional and heat losses.

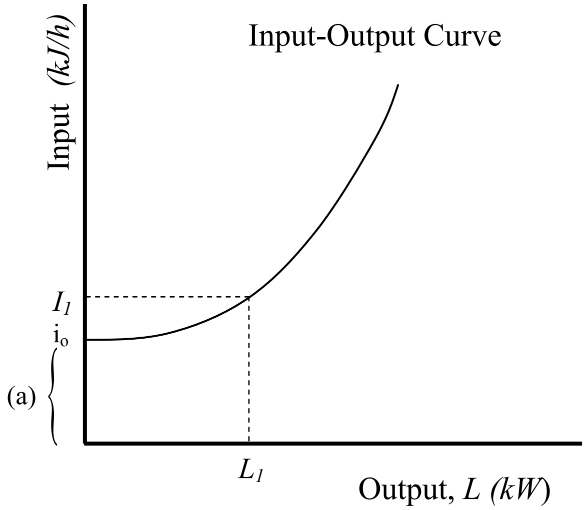

Figure (10-7)

The thermal efficiency can be represented in terms of input and output.

$$\eta_{th} = \frac{3600\ L}{I}\left(\frac{kW\ h}{kJ}\right) \qquad (8\text{-}17)$$

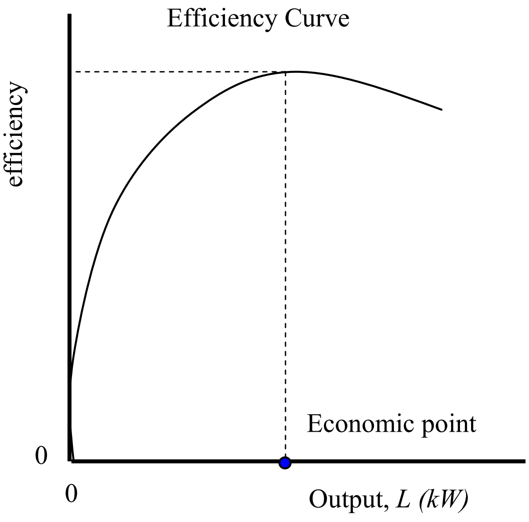

Figure (10-8)

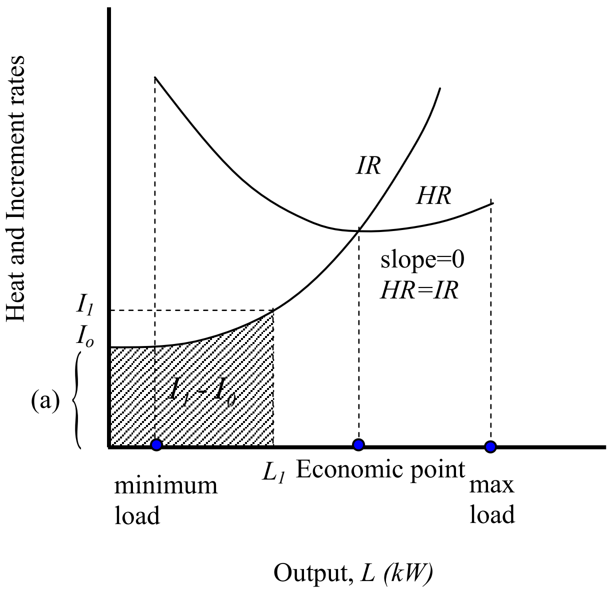

Figure (10-9)

The Heat Rate curve Figure (10.9) is derived by taking at each load the corresponding input, then

$$HR = \frac{I}{L} \qquad (10\text{-}18)$$

Then from Eqs. (10–17) and (10–18)

$$HR = \frac{3600}{\eta_{th}} \qquad (10\text{-}19)$$

The heat rate curve can be expressed also mathematically, thus, if the input – output curve is defined by

$$I = a + bL + cL^2 + dL^3$$

Then

$$HR = \frac{I}{L} = \frac{a}{L} + b + cL + dL^2 \qquad (10\text{-}20)$$

The incremental – rate curve Figure (10–9) is derived from the input – output curve by finding at any load the additional (or differential) input required incremental rate defined as,

$$IR = \frac{dI}{dL}\left(slope\right) \qquad (10\text{-}21)$$

Mathematically the IR is the slope of the input – output curve at the given load. Physically the IR expresses the amount of additional energy required to produce an added unit of output at any given load.

From the definition of an incremental rate, the areas under the IR curve measure the additional input required to increase output from $L1$ to $L2$.

since $dI = IR\,dL$ (10-22)

$$\left(I_2 - I_1\right) = \int_{L_1}^{L_2} IR\,dL \tag{10-23}$$

If the input – output curve is defined by

$$I = a + bL + cL^2 + dL^3$$

Then , the corresponding incremental – rate curve is defined by

$$IR = \frac{dI}{dL} = b + 2cL + 3dL^2 \tag{10-24}$$

Example 10.4

A 20 MW station has an input – output curve defined by

$I = 30 + 0.5 L + 0.65 L2 + 0.01 L3$

Find the increase in input required to increase the station output from 7 to 9 MW, HR and IR.

Solution

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Example 10.5

A power station has an input – output curve defined by : $I = a + bL + cL2$
where (a) is the input without load (output), and is estimated about 10% of the input (I_{max}) at the maximum load. Assuming the economical Load (L_{eco}) and the maximum load (L_{max}) are 20 MW and 30 MW respectively. The Heat Rate at the economical load (HR_{eco}) is 10000 $\left({}^{kJ}\!/_{kW\,h} \right)$ and (HR_{max}) is 11000 $\left({}^{kJ}\!/_{kW\,h} \right)$ at the maximum load. Find the values of the constants (a, b and c)

Solution

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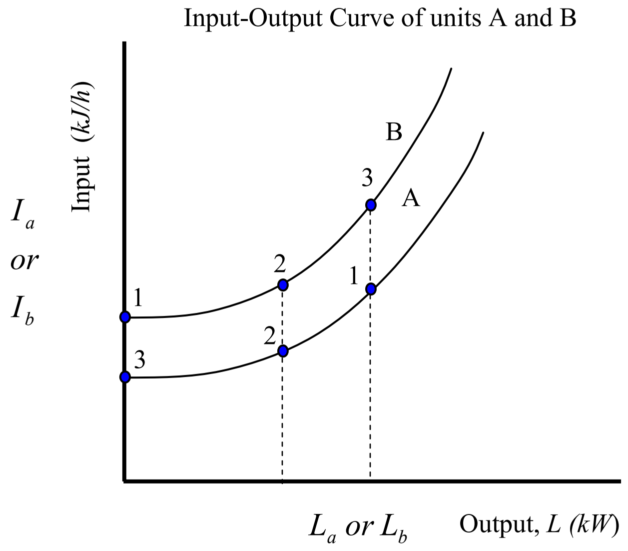
10-5- Load division

In a utility system with many generating units, decision is often needed to determine the order in which the generating units are dispatched into operation. In practice, these are usually done according to the unit relative efficiency.

For example, a hypothetical utility system has two units, A and B. The load division is attacked as follows;

Plotting the sum of the inputs to A and B against the load on A or the load on B for a given constant load on the two units as shown in the following Figs (10-10 and 10-11) and tables (10-1) and (10-2).



Input-Output Curve of units A and B

Figure (10-10)

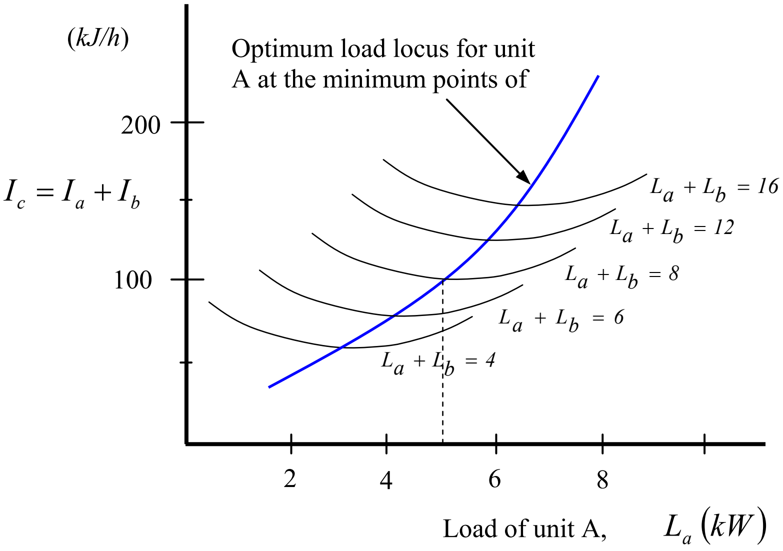


Figure (10-11)

For a total load of 8MW on the two units, in placing zero load on A and 8MW on B, the total combined input is 148 MJ/h. As the load is shifted from B to A, maintaining the combined unit output constant, the total input decreases unit A carries 5MW and B carries 3MW shifting more load onto A then increases the combined input. For maximum economy this load should be divided so as to require the minimum combined input. By plotting a series of constant combined output loci it would be possible to determine the division of load for the entire range of capacity of the two units, as shown in Fig (10-11).

This method is cumbersome and becomes unwieldy when more than two units run in parallel. The basic principle of economic load division demonstrated in the following, loads to a simple solution for the general case. Taking any combined constant-output locus at the point of minimum input.

$$\frac{dI_c}{dL_a} = 0 \qquad (10\text{-}25)$$

Where $I_c = I_a + I_b$ (10-26)

$$L_c = L_a + L_b \qquad (10\text{-}27)$$

Then

$$\frac{dI_c}{dL_a} = \frac{dI_a}{dL_a} + \frac{dI_b}{dL_a} = 0 \qquad (10\text{-}28)$$

$$\frac{dI_a}{dL_a} = -\frac{dI_b}{dL_a} \qquad (10\text{-}29)$$

But

$$\frac{dI_b}{dL_a} = \frac{dI_b}{dL_b} \times \frac{dL_b}{dL_a} \qquad (10\text{-}30)$$

From equation (10-27)

$$\frac{dL_b}{dL_a} = \frac{dL_c}{dL_a} - \frac{dL_a}{dL_a} \qquad (10\text{-}31)$$

Since Lc is constant

$$\frac{dL_b}{dL_a} = -1 \qquad (10\text{-}32)$$

Substituting equation (10-32) into equation (10-30)

$$\frac{dI_b}{dL_a} = -\frac{dI_b}{dL_b} \qquad (10\text{-}33)$$

Substituting equation (10-33) into equation (10-29)

$$\frac{dI_a}{dL_a} = \frac{dI_b}{dL_b} \qquad (10\text{-}34)$$

$L_c=L_a+L_b=4$				
I_a	I_b	I_a+I_b	L_a	L_b
20	70	90	0	4
33	46	79	2	2
52	26	76	4	0

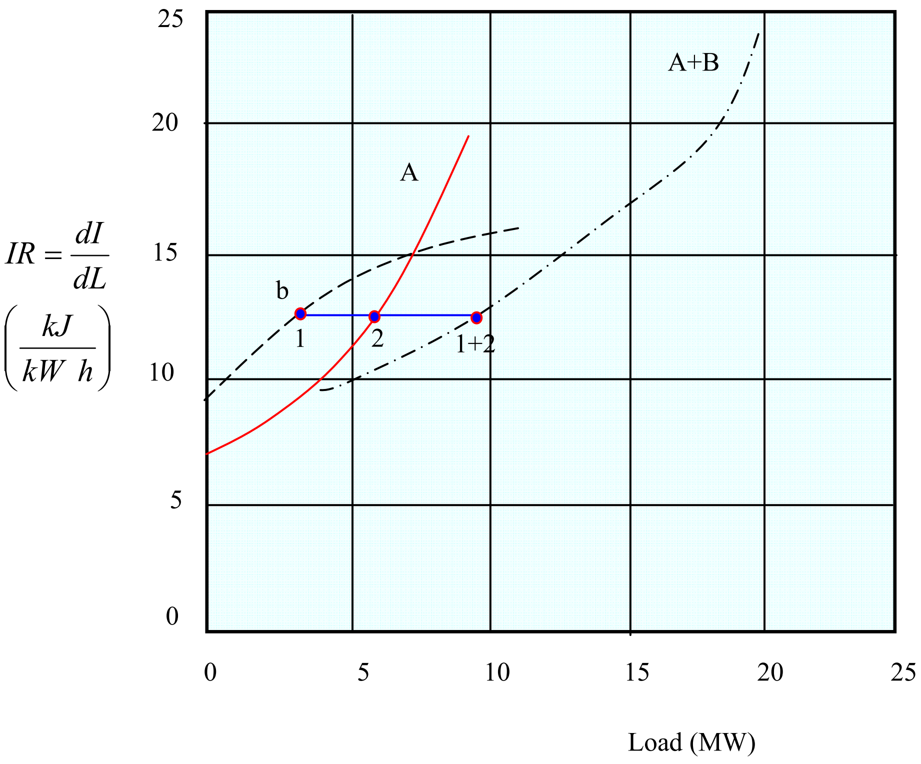
Table (10-1)

$L_c=L_a+L_b=8$				
I_a	I_b	I_a+I_b	L_a	L_b
20	128	148	0	8
33	98	131	2	6
52	70	122	4	4
76	46	122	6	2
110	26	136	8	0

Table (10-2)

The same thing for L_c = 12 & 16 (KW) or, for minimum combined input to carry a given combined output the slopes of the input – output curves for each unit must be equal. But the slopes of these curves are the incremental rates. Then by plotting the incremental rate curves Figure (10-12) for the units (A & B) all that is necessary to divide the load economically is to maintain the incremental rates equal and select the loads accordingly.

To expedite load division for varying total loads, the combined incremental – rate curve is plotted by adding up the loads at each incremental rate and plotting the sums against the particular incremental rate. The curve A + B is such a combined curve as in Figure (10 - 12).



$IR = \frac{dI}{dL}$
$\left(\frac{kJ}{kW\ h}\right)$
25
20
15
10
5
0
0
5
10
15
20
25
Load (MW)
A
A+B
b
1
2
1+2

Figure (10-12)

10.6 - Economics

10.6.1-Introduction

The basic of most engineering decisions is economics. Designing and building a device or system that functions property is only a part of the engineers' task. The device or system must, in addition, be economical it must show an adequate return on the investment.

10.6.2-Interest

Interest is the rental charge for the use of money. Borrowers of money pay the interest for the use of the money at stated periods through the term of the loan - for example every 6 month (semi-annually) or every year (annually). A series of equal payment at regular periods of time is called *Annuity*.

The are seven frequently encountered interest paying situations, as follows,

1-Simple interest.
2-Compound interest.
3-Present worth.
4-Series compound amount.
5-Sinking fund.
6-Series present amount.
7-Capital recovery.

1-Simple interest

Assume Mohammad has borrowed SR15000 from Ahmed and both agreed that the money will be return after three years and the interest is 5%.

At the end of 3 Mohammad will brings to his friend Ahmed

$$15000 + 15000 \times (0.05) \times (3) = 17250 \qquad (10\text{-}35)$$

The SR15000 is called the *principal* or *present amount*, and the rate of interest is the interest earned per unit present amount per unit time. As Mohammad has made the calculation, the interest is called *simple interest*.

The entire amount, S, to be repaid at the future time is,

$$S = P + Pni = P(1 + ni) \qquad (10\text{-}36)$$

where S future amount
P present amount
n number of periods (years)
i rate of interest per period

2-Compound interest

The agreement which conducted between Mohammad and Ahmed was Mohammad suppose to pay the money after three years and the interest is 5%, that means the interest is compound interest, therefore,

At the end of the first year

$$15000 + 15000 \times (0.05) = 15750$$

At the end of the second year

$$15750 + 15750 \times (0.05) = 16537.5$$

Now, at the end of the third year

$$165375 \; + \; 16537.5 \times (0.05) = 17364.3$$

The future amount ,S, is,

<u>for the first year</u>

$$S = P + Pi = P(1+i)$$

<u>for the second year</u>

$$S = P(1+i) + P(1+i)i$$

$$= P(1+i)(1+i) = P(1+i)^2$$

for the third year

$$S = P(1+i)^2 + P(1+i)^2 i$$

$$= P(1+i)^2 (1+i) = P(1+i)^3$$

Then in general form, the future amount

$$S = P(1+i)^n = P(CAF) \tag{10-37}$$

where CAF is compound amount factor

3-Present worth

A companion calculation to that of compound interest is the *present worth* calculation. Present worth is the value of a sum of money at the present time with compounded interest, will have a specified value of a certain time in the future. In equation (10-37), therefore S is known and P is to be determined.

$$P = \frac{S}{(1+i)^n} = (PWF)S \tag{10-38}$$

where PWF present worth factor $= \dfrac{1}{(1+i)^n}$

PWF is simply the reciprocal of the CAF for given values of i and n.

4-Series compound amount

In many financial situations, equal payments are made at regular intervals in order to accumulate an amount some time in the future. The sum earns interest on the progressively increasing amount.

Suppose that SR15000 is invested at the *end* of every year. At the end of the first period the accumulated sum S is only the 15000 payment which has just been made(*Annuity*).

At the end of two years,

$$S = 15000(1+i) + 15000 = 15000[(1+i)+1] \tag{10-39}$$

At the end of n years,

$$S = 15000\left[(1+i)^{n-1} + (1+i)^{n-2} + \; .\,.\,.\,.\,.\,.\,.\,.\,.\,. + (1+i) + 1\right] \tag{10-40}$$

The series in the brackets,

$$\sum_{j=0}^{n-1}(1+i)^{j} \tag{10-41}$$

Multiplying both sides of Equation (10-41) by $(1+i)$,

$$S(1+i) = 15000\left[(1+i)^{n} + (1+i)^{n-1} + \; .\,.\,.\,.\,.\,.\,.\,.\,.\,. + (1+i)^{2} + (1+i)\right] \tag{10-42}$$

Subtracting equation (10-41) from (10-42), we have

$iS = A\left[(1+i)^{n} - 1\right]$ where $A = SR\ 15000$

$$S = A\left[\frac{(1+i)^{n} - 1}{i}\right] \tag{10-43}$$

Then

$$S = A\ (SCAF) \tag{10-44}$$

where $SCAF$ series compound amount factor

5-Sinking fund

The companion calculation to the series compound- amount factor is the sinking fund factor. If the amount that is to be accumulated by some specific time in the future has been specified, the sinking fund calculation determines the regular payments that must be made in order to accumulate the amount. In equation form,

$$A = \frac{S}{SCAF} = S\left[\frac{i}{(1+i)^{n} - 1}\right] = S(SFF) \tag{10-45}$$

where SFF sinking fund factor

6-Series present worth

The reverse procedure is where an amount exists at the start and equal withdrawals are made periodically so that the amount is exactly used up after a specified time. Interest is earned on the amount that has not yet been withdrawn. Let P be the original, with an amount A being withdrawn at the end of the first and subsequent periods. The amount P earns interest at the rate of i percent during the first period.

After one period the current value is

$$P(1+i) - A \tag{10-46}$$

After two periods, the current value is

$$\left[P(1+i) - A\right](1+i) - A = P(1+i)^{2} - A\left[(1+i) + 1\right] \tag{10-47}$$

If the amount is to be exhausted at the end of n periods,

$$P(1+i)^{n} - A\left[(1+i)^{n-1} + \,.\,.\,.\,.\,.\,.\,.\,. + 1\right] = 0 \tag{10-48}$$

The series in the brackets has already been evaluated and is expressed in equations (10-40 and 10-43). Therefore,

$$P = A\left[\frac{(1+i)^n - 1}{i(1+i)^n}\right] = A\left[\frac{1-(1+i)^{-n}}{i}\right] = A(SPWF) \qquad (10\text{-}49)$$

where $SPWF$ series present worth factor.

7-Capital recovery factor

The companion factor to the $SPWF$ is the capital recovery factor CRF which when multiplied by the original amount P gives the regular withdrawals amount that exhausts the money after a specified time.

$$A = \frac{P}{SPWF} = P\left[\frac{i(1+i)^n}{(1+i)^n - 1}\right] = P(CRF) \qquad (10\text{-}50)$$

A summary of the interest and annuity formulas discussed so far is presented in Table (10-3).

Symbol	Name	Equation	Situation
SIF	Simple Interest Factor	$(1+ni)$	$S=P(1+ni)$
CAF	Compound Amount Factor	$(1+i)^n$	$S=P(CAF)$
PWF	Present Worth Factor	$\frac{1}{(1+i)^n}$	$P=(PWF)\ S$
$SCAF$	Series Compound Amount Factor	$\left[\frac{(1+i)^n-1}{i}\right]$	$S=(SCAF)\ A$
SFF	Sinking Fund Factor	$\left[\frac{i}{(1+i)^n-1}\right]$	$A=(SFF)\ S$
$SPWF$	Series Present Worth Factor	$\left[\frac{(1+i)^n-1}{i(1+i)^n}\right]$	$P=(SPWF)A$
CRF	Capital Recovery Factor	$\left[\frac{i(1+i)^n}{(1+i)^n-1}\right]$	$A=(CRF)P$

Table (10-3)

S : future amount , A : Annuity , n : Number of period , P : Present amount , i : Interest rate per period

10.6.3- Depreciation

Depreciation is an amount that is listed as an annual expense in the tax calculation to allow for replacement of the facility at the end of its life. There are numerous methods of computing depreciation which are permitted by the interval revenue service, such as straight line, sum-of the year's digits (SYD) and double-rate declining balance. The second method only will be explained in this section.

The sum-of-the-year's digits (SYD) method for a given year is represented by the formula

$$\text{Depreciation } (SR) = \frac{2(N - t + 1)}{N(N+1)}(P - S) \tag{10-51}$$

where, N : tax life, years.

t : year in question.

P : first cost, *SR*

S : salvage value, *SR*

If the tax life is 15 years, for example, the depreciation is as follows:

$\frac{15}{120}$ $(P\text{-}S)$ for the first year

$\frac{14}{120}$ $(P\text{-}S)$ for the second year

$\frac{1}{120}$ $(P\text{-}S)$ for the fifteenth year

10.6.4- Depreciation rate

The ratio of the annual depreciation to the capital invested is the depreciation rate. From the formulations already developed as the sinking fund method which is (see table 10-3)

$$\frac{A}{S} = \left[\frac{i}{(1+i)^n - 1}\right] \tag{10-52}$$

where:

A: annual depreciation changed against income

S: value of capital funds that must be accumulated

n: expected life of capital equipment or business, year

i: earning rate or interest that investment of A will yield

Example 10-6

Find the depreciation rate for power plant equipment is expanding and is earning 5% on its invested capital if the expected life of the equipment is 20 years.

Solution

$$\frac{A}{S}=\left[\frac{i}{(1+i)^n-1}\right]=\frac{0.05}{(1+0.05)^{20}-1}=0.03024$$

If at the end of each year an amount equal to 3.024 percent of the investment is reinvested in the power plant, the original capital will be recovered at the time the equipment is worn out. The depreciation rate can also be determined from eq. (10-49)

$$\frac{A}{P}=\left[\frac{i}{1-(1+i)^{-n}}\right]=\frac{0.05}{1-(1+0.05)^{-20}}=0.08024$$

In this formulation A represents the annuity that will both pay the earnings at i interest rate and repay the original investment P during the life of n years. In this case, since i is 5 %, the rate of repayment of original investment or the depreciation must be = 0.08024 – 0.05 = 0.03024 or 3.024 percent, the same determined by the sinking fund formula.

10.6.5-Power plant life time

The power plant life time period can be divided into,

1-Production period (hypothetical age)

2-Construction period

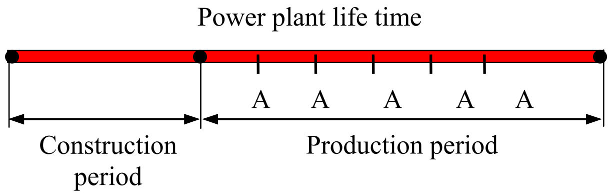


Figure (10-13)

Figure (10.13) shows the construction period and the hypothetical period and the Annuity.

10.6.6-Production cost of electricity

10.6.6.1-Fixed cost

$$\text{Fixed Cost}/kW\,h = \frac{Rate\,of\ return\ charge(A)}{E_{year}} = \left[\frac{SR}{kWh}\right] \quad (10\text{-}53)$$

where

$$E_{year} = L_{av} \times 24 \times \text{Number of working day per year} \left[\frac{kW\,h}{Year}\right] \quad (10\text{-}54)$$

10.6.6.2-Variable cost:

- Annual fuel cost = $E_{year}\left[\frac{kW\,h}{Year}\right]$ x HR $\left[\frac{kJ}{kW\,h}\right]$ x Specific fuel cost $\left[\frac{SR}{10^6\ kJ}\right]$ (10-55)

  Fuel cost / $kW\,h(f)$ = HR x Specific fuel cost

- Operation and maintenance cost:

Operation and maintenance (O&M) expenses include operating labor, materials, and tools for plant maintenance on both a routine and emergency basis. These expenses are neither a function of plant capital nor plant engineering capacity. They vary from year to year and generally become higher as the plant becomes older.

10.6.6.3-Construction cost *(CC)*

Construction cost = Total capacity (C_{apt})x Specific construction cost (scc)

$$CC = Cap_t[kW] \times scc\left[\frac{SR}{kW}\right] \quad (10\text{-}56)$$

10.6.6.4- Other costs

These costs cover administrative and general cost, Insurance cost, Income tax and Taxes other than income taxes.

10.6.7-Levelizing Factor

$$F_{lev} = \frac{\sum_{i=1}^{n} f_i (PWF)_i}{\sum_{i=1}^{n} (PWF)_i} \qquad \left[\frac{SR}{kJ}\right] \quad (10\text{-}57)$$

$$F_{lev} = \frac{\sum_{i=1}^{n} f_i (PWF)_i}{SPWF}$$

where

f fuel cost $\left[\frac{SR}{10^6\ kJ}\right]$

If f is escalated by k = constant

Then

$$F_{lev} = \frac{f \sum_{i=1}^{n} (1+k)^{i-1} (PWF)_i}{SPWF} \tag{10-58}$$

$$F_{lev} = \frac{f \left(\frac{1-\zeta^n}{r-k} \right)}{SPWF} \tag{10-59}$$

where

$$\zeta = \left(\frac{1+k}{1+\mathbf{r}} \right) \tag{10-60}$$

$\mathbf{r}$: rate of return or interest rate

k: escalation rate

$$SPWF = \frac{P}{A} \Rightarrow \frac{A}{P} = \left[\frac{i}{1-(1+i)^{-n}} \right]$$

10.7: Thermoeconomic optimization techniques

10.7.1: Introduction

Practical optimizing calculations for power plant always involve consideration of economic as well as thermodynamic factors. Thermoeconomic optimizing is a technique which combines thermodynamic analysis and economic optimizing of thermal system.
The purpose of thermoeconomic optimization is to achieve subject to physical constraints a minimum cost of the product or introduce or suggest ways of improving efficiency. The advantage of using thermoeconomic optimization is that the various parameters of power plant element can be optimized on their own to determine the optimal design.

10.7.2: Thermoeconomic optimization of thermal system

There are two different methods of thermoeconomical optimization. The first method is called structural method and the second method is called autonomous method.

1- The structural method

The application of this method in the power plant is to determine for the power plant elements the capital cost corresponding the minimum annual operating cost, and by implication, to the minimum unit cost of the product. The changes in the power plant parameter (y) causes changes in the irreversibility rates, and necessitates changes in the capital cost.

$$\dot{I}(y) = \dot{E}_{in}(y) - \dot{E}_{out} \qquad (10\text{-}61)$$

where:

$\dot{E}_{in}$: input available energy (exergy) rate = $\Delta H - To\,\Delta S$

$\dot{E}_{out}$: output available energy (exergy)rate

In this type of method it is essential to keep the total exergy output $\dot{E}_{out}$ independent of y. The irreversibility rate $\dot{I}(y)$ may be taken as the consumption of exergy in the system, necessary to generate the product exergy $\dot{E}_{out}$. For this optimization, take the objective function to be the *annual cost* plant operation, C, which can be expressed as:

$$C(y) = t_{op}\, C_{in}^{\varepsilon}\, \dot{E}_{in}(y) + CRF \sum_{i=1}^{m} C_{\ell}^{c}(y) + b^{c} \qquad (10\text{-}62)$$

where:

top : period of operation per year

C_{in}^{ε} : unit cost of input exergy to the system.

CRF : Capital Recovery Factor, which when multiplied by the total investment gives the annual repayment (Annuity) to pay back gives the investment after a specified period.

C_{ℓ}^{c} : Capital cost of ℓ-th element of the system consisting of m element.

b^{c} : the part of the annual cost which is not affected by the optimization.

The first term of the right hand side of equation (10-62) can be written as

$$t_{op}\, C_{in}^{\varepsilon}\, \dot{E}_{in}(y) = E_{year} \text{ x } HR \text{ x Specific fuel cost}$$

$$= L_{av}\left(y\right) \text{ x } t_{op} \text{ x } HR \text{ x Specific fuel cost} \quad (10\text{-}63)$$

Subject to the usual mathematical conditions being fulfilled, the objective function will be differential with respect to y.

$$\frac{\partial \dot{E}_{in}}{\partial y} = \frac{\partial \dot{I}}{\partial y} \quad (10\text{-}64)$$

$$\frac{\partial C}{\partial y} = t_{op}\, C^{\varepsilon}_{in}\, \frac{\partial \dot{I}}{\partial y} + CRF \sum_{i=1}^{m} \frac{\partial C^{c}_{\ell}}{\partial y} \quad (10\text{-}65)$$

The optimum condition corresponds to

$$C\left(y\right) \to \min \quad (10\text{-}66)$$

and is given by:

$$\left(\frac{\partial \dot{I}}{\partial y}\right)_{opt} = -\frac{CRF}{t_{op} C^{\varepsilon}_{in}} \sum_{i=1}^{m} \frac{\partial C^{c}_{\ell}}{\partial y} \quad (10\text{-}67)$$

Accordingly, the values of İ and y corresponding to a min un the annual cost function can be found by locating the point on a graph of y versus I where the slope is equal to eq. (10-67).

2- The autonomous method

In this method the plant components can be optimized on its own using local unit cost of energy fluxes entering and leaving the control volume. By using the exergy analysis in optimization, the basic plants components e.g gas turbine plant, liquefaction plant, refrigeration plant, etc, can be broken into exergy dissipative zones, with each zone containing one component of the system, and optimized individually, thereby producing through iteration the optimum design of the plant as a whole. The objective of this section is to demonstrate the thermoeconomic optimization technique of the components of the plant. Therefore the plant has been considered as a number of exergy dissipative zone, and can be optimized by minimizing the annual expenditure for a fixed output. This cost results from capital cost of the equipment,, time of operation per year and capital recovery factor, and the cost of the exergy dissipation or irreversibilities of the zone. From these parametric equations the combined optimum design of the cycle can be obtained.
The capital investment is given by:

$$\dot{Z} = \frac{CRF}{t_{op}} C^{c}_{\ell} \quad (10\text{-}68)$$

In terms of costs, the zonal exergy for the incoming flows is $C^{\varepsilon}_{in}\, \dot{E}_{in}$ and the zonal outgoing exergy cost is $C^{\varepsilon}_{out}\, \dot{E}_{out}$, where C^{ε}_{in} and C^{ε}_{out} are the unit cost of incoming and outgoing exergy flow respectively.

If the unit cost of outgoing exergy is assumed to be equal to the unit cost of incoming exergy plus the capital cost of investment, then

$$\dot{E}_{out}\, C^{\varepsilon}_{out} = \dot{E}_{in}\, C^{\varepsilon}_{in} + \dot{Z}_{1} + \dot{Z}_{2} + \dot{Z}_{3} + \ldots\ldots . + \dot{Z}_{m} \quad (10\text{-}69)$$

Example 10-7

An electrical generating system connected to 5 groups of consumers with the following data:

Group	Connected load kW	Demand factor % ‘*d*’	Group diversity Factor ‘*D*’	Peak diversity factor ‘*r*’
1	1500	70	1.5	1.10
2	7000	60	3.0	1.10
3	15000	80	2.0	1.05
4	10000	60	1.1	1.00
5	13000	40	1.5	1.20

The daily load factor for this system is 0.6, the load- duration curve is assumed to be linear, and utilization factor is 0.9. Find the minimum load on the system and the capacity factor. The system is supplied by a power plant of capacity 25% more than maximum load. The specific construction cost of the plant is 4500 SR/kW and the construction period = 5 years, the construction cost has to be paid through 5 payments as follows,

- Beginning of 1st year 10%.
- Beginning of 2nd year 10%.
- Beginning of 3rd year 30%.
- Beginning of 4th year 30%.
- Beginning of 5th year 20%.

Through interest rate of 0.12 annually. Fuel specific cost of 2 SR/10^6 kJ. Plant average thermal efficiency = 0.35, working days = 300 day/year.
The plant hypothetical age 25 years. Find fixed and variable cost.

Solution

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Example 10.8

A power plant operates for three years as follows:

Year	Average load MW	HR kJ/kWh	Time of operation hours	Specific fuel cost SR/kJ
1st	12	10100	7000	1.25×10^{-6}
2nd	10	10000	7500	1.25×10^{-6}
3rd	11	10050	6500	1.25×10^{-6}

The interest rate (i) = 8%.
Calculate the present worth cost of the fund through the three years.

Solution

Example 10.9

An electrical generating system supplies three consumers indicated in the following table:

Consumer	Total connected load MW	Demand factor %	Group diversity factor	Peak diversity factor
I	80	70	1.5	1.0
II	50	50	2	1.2
III	40	41	3	1.1

The minimum demand of the system is 50% of the max demand (L_{max}) and the daily variation of the load is linear while the utilization factor of the plant is 0.8 and the capacity factor is 0.6. Determine; maximum demand, min demand, average load, load factor, reverse factor, plant capacity and daily generated electrical energy.

Solution

Example 10.10

a- A power system consists of the following consumers

Items, class	No.	Connected load kW	Demand factor (d) %	Group diversity factor (D)	Peak diversity factor (r)
Factories	10	2000	40	1.2	1.5
Houses	1000	10	30	1.5	1.3
Services station	5	500	35	1.4	1.6

Calculate

a- the maximum demand, if the min. load of the system is 0.3 of the maximum load and the relation between max. load and the min. load on load-duration curve is linear, the capacity factor is 0.5, and calculate the required capacity of the plant (C_{ap}), load factor, and utilization factor.

Assume the year of 8760 hours and the annual average heat rate HR = 10000 kJ/kW h, specific fuel cost is 2 SR/10^6 kJ, the interest rate (i) = 0.1, calculate the present worth of total fuel cost per year for the first year of operation, also calculate the fuel cost/kW h.

Additionally, assume that the fuel cost/kW h is 0.5 of the fixed cost/kWh of this system, calculate the annuity (A) , the specific construction cost/kW, assuming plant life time is 25 years and construction period is 2 years and at the beginning of this construction period the construction cost will be paid.

b- Calculate finally the levelized fuel cost of the escalation rate (k) =0.08 and interest rate of return (***r***)=0.09 and the fuel specific cost 3 SR/10^6 kJ through 5 years.

Solution

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Example 10.11

The following data to a power generating system supplies a group of consumers consists of 1000 flats.

- Demand factor = 0.7
- Group of diversity factor = 2
- Peak diversity factor = 1.5
- The fixed cost per kW h produced = 0.01 SR
- The specific construction cost (s_{cc}) = 4500 SR/kW
- The daily energy produced = 720000 kWh
- The working day per year = 300 days
- The reserve factor = 0.2
- The plant life time is 15 years and the construction period is 3 years.
- The total construction cost must be paid at the beginning of the construction period.
- The interest rate = 0.1
- The annual average heat rate (HR) =10000 kJ/kW h
- Specific fuel cost = 1.2 SR/10^6 kJ
- The operating and maintenance cost is taken as 25 % of fixed fuel cost/ kWh.

Calculate

1. Max load of the system and average load.
2. Connected load for each flat.
3. Load factor, capacity factor, and utilization factor.
4. Fuel cost per year.
5. Production cost per kW h

Solution

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Revision Questions

10-1. An electrical system experiences linear changes in load such that its daily load curve is described in the following Table

Time	MW
12 pm	20
2 am	10
6	10
8	50
12 Mid	50
12.30 pm	40
1	50
5	50
6	70
12	20

a. Plot the chronological and load curve.

b. Plot the load- energy curve of the system.

c. Find the average load.

d. Find the load factor.

e. What is the utilization factor of the plant serving this load if its capacity is 100 MW.

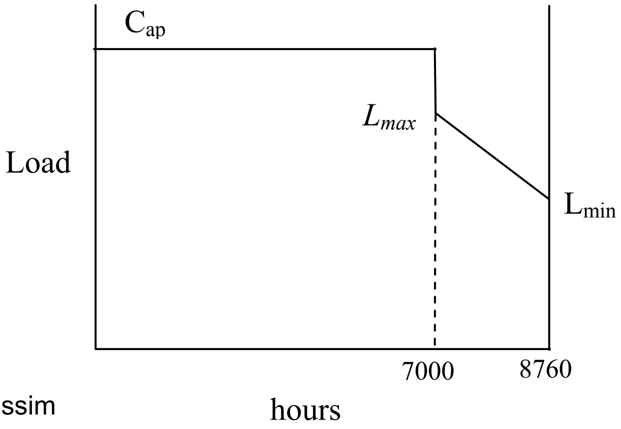
10-2. The following data related to a power generating system.

- Fuel cost/kWh = 0.02 SR
- Specific fuel cost =2 SR/10^6 kJ
- Escalation rate (k) = 0.08
- Fuel cost per year = 1000000 SR
- The working hours per year at C_{ap} = 7000 h
- The plant capacity is 20 % more than max load.
- L_{min} = 2000 kW
- The relation between L_{max} and L_{min} on load-duration curve is linear.
- Demand factor (d) = 0.4
- Group diversity factor (D) = 1.5
- Peak diversity factor (r) = 2
- Connected load of each consumer is 3 kW
- Fixed cost is 0.03 SR/kW h
- Plant life time (n) = 10 years
- Interest rate (i) = 0.1

Calculate through the first year

HR, E_{year}, C_{ap}, L_{max}, F_{lev}, Cap factor, Utilization factor, No. of consumers, Annuity, P (present worth)

<u>Ans:</u> 10000 kJ/kWh, 5 x 107 kWh, 6237.9 kW, 5198.3 kW, 0.692, 0.5769, 0.8334, 12996, 1.5 x 10^6 SR, 9.216 x 10^6 SR



10.3. An electric power plant system consists of three groups of consumers I, II, III groups. Group III has a connected load equals to the connected loads of group I + group II. Group I connected load equals to 0.75 of connected load of group II. The load data of these groups are as follows:

Group	Demand factor	Group diversity factor	Peak diversity factor
I	0.4	2	1.5
II	0.3	1.5	1.2
III	0.5	1.8	2

The capacity of the power plant supplying this system is 50 MW, the specific construction cost (cc) = 4500 SR/kW, the interest rate (i) = 0.1(neglect the effect of construction period). The life time of the plant (n) is 15 years. The daily average load on the system=20 MW, the load factor ($L.\ F$) is 0.5. The average annual heat rate, HR, =10000 kJ/kW h. The specific fuel cost is 3 SR/10^6 kJ. Assume the working day per year is 300 days. Calculate

The annuity (A), the annual electric energy produced, the fixed cost per kWh, L_{max}, *Cap factor* , utilization factor, L_{min} if the relation between L_{max} and L_{min} is linear, reverse factor, also calculate the annual fuel cost and fuel cost per/kWh. If the fuel cost has as escalated by (k) =0.08 and (r) = 0.1, calculate the levelized heat cost over the first 5 years of the plant life and the connected loads for each group of consumers.

Ans:

A =3286844.4 SR, E_{year} = 144x10^6 kWh, Fixed cost/kWh = 0.0228 SR, L_{max} = 40 MW

$C_{ap}\,factor$=0.4, Utilization factor = 0.8, L_{min} =0, reverse factor =0.25

Fuel cost/kWh =0.03 SR, Fuel cost /year = 4.32 x 10^6 SR, F_{lev}=3.468 SR/10^6 kJ

Connected load of group I = 58.87 MW

Connected load of group II = 78.5 MW

Connected load of group III = 137.37 MW

10-4. An electric power system consists of 1000 consumers each has a connected load of 100 kW. The demand factor is 0.8 while group diversity factor and peak diversity factor are equals. The load factor is 0.7. the average load is 35 MW, reverse factor is 0.1. Find group or peak diversity factor, capacity factor and utilization factor. The specific construction cost is 4500 SR/kW. The construction period is 3 years, the interest rate = 10%, the plant life time is 15 years. Also, find the fixed cost per kWh if the working day per year is 300. If the total cost per kWh is 0.05 SR. Find the fuel cost per kWh and the total annual fuel cost and the average heat rate if the specific fuel cost is 2 SR/10^6 kJ.

Ans: D=r=1.265, Capacity factor =0.636, Utilization factor =0.909, fixed cost/kWh = 0.0191 SR, fuel cost/kWh = 0.0309, HR =15455 kJ/kWh, Annual fuel cost =7.8 x 10^6 SR.

10.5- Three groups of electrical power consumers A,B and C. The ratios of the connected loads between them are 3,4 and 5 respectively. These consumers have the following data:

Factor	A	B	C
Demand factor	0.5	0.7	0.6
Group diversity factor	2.0	2.5	2.0
Peak diversity factor	1.0	1.5	1.6

The load factor on the system is 0.6 and the average load is 48 MW. Calculate the connected load for each group when the power plant supplies this system has a reverse factor of 0.25. The average heat rate through the year of the plant is 10000 kJ/kWh, the specific fuel cost is 1 SR/10^6 kJ. The specific construction cost is 4500 SR/kW. The construction cost has to be paid at the beginning of the construction period of 4 years and the interest rate of 0.1. Assume the present worth calculation taking the year of commissioning as a reference year. The life time of the plant is 20 years. Calculate the fixed and variable costs per kWh through a year. Also calculate the capacity factor and utilization factor and calculate the series present worth factor (SPWF) over the plant life time.

Ans:

$c\ell_A$ =98.596 MW, $c\ell_B$ =131.46 MW, $c\ell_C$ =164.327 MW, Fixed cost/kWh = 0.01323 SR, Variable cost/kWh = 0.01 SR, Capacity factor =0.48, utilization factor =0.8, SPWF=8.984

10.6- An electrical generating system connected to 6 groups of consumers with the following data:

Group	Store or service	No. of consumers	Connected load kW	Demand factor % 'd'	Group diversity Factor 'D'	Peak diversity factor 'r'
1	Houses	5000	5	0.4	1.5	1.5
2	Factories	100	300	0.5	1.7	1.3
3	Stores	500	50	0.35	2.0	1.5
4	Lighting	300	50	0.6	1.5	1.2
5	Farmers	100	100	0.5	1.3	1.5
6	mosques	5	20	0.5	1.2	1.4

The capacity of the power plant supplying this system is 30 MW, the specific construction cost is 4500 SR/kW, the construction period is 4 years, the construction cost has to be paid through 4 payments as follows,

- Beginning of 1st year 20%
- Beginning of 2nd year 20%
- Beginning of 3rd year 30%
- Beginning of 4th year 30%

If the total cost (fixed and variable costs) is 0.042 SR/kWh and the annual average heat rate HR=11000 kJ/kWh. Assume the working day per year is 300 day. The plant life time is 25 years, the interest rate is 0.1.the relation between the maximum and minimum loads on load duration curve is linear. Use in your calculation the present worth method. Calculate, L_{av}, L_{max}, fuel cost/kWh when the specific fuel cost is 2 SR/10^6 kJ and load factor, capacity factor, utilization factor, L_{min} and reverse factor.

Ans:

L_{av}=11939.74 kW, L_{max} = 21742.3 kW, Fuel cost/kWh =0.022 SR, Load factor=0.549, Capacity factor =0.4, utilization factor=0.724, L_{min}=2137.2 kW.

Appendix A

Exergy Analysis of some example of irreversible processes

The exergy of a system is the maximum work obtainable as the system comes to equilibrium with the surroundings. The First Law of Thermodynamics as shown in section 2 makes only an energy balance of a system or a control volume. It does not make any distinction of different forms of energy, particularly between work and heat, or heat available at different temperatures. It is the Second Law which asserts that from engineering viewpoint, all forms of energy are not of the same quality. Energies of two systems may be quantitatively equal, but qualitatively they may be different. Therefore, exergy method is a technique based on both the first and second law of thermodynamics; it provides an alternative to the traditional methods of thermodynamic analysis

The advantage of exergy analysis is that it provides the locations where the exergy destructions or irreversibilities are more. Ways and means can then be explored, say, by thermodynamically optimizing the operating or geometrical parameters of each sub-region, so as to reduce these destructions to the practical minimum values.

The exergy destruction is a measure of thermodynamic imperfection of a process and is expressed in terms of lost work potential. In general, the expression for exergy destruction can be expressed as.

$$\dot{I} = T_o \left[\left(\dot{S}_{out} - \dot{S}_{in} \right) - \sum_{i=1}^{n} \frac{\dot{Q}_i}{T_i} \right] = T_o \dot{\Pi} \geq 0 \qquad (1)$$

and the exergy balance,

$$\dot{E}_{in} + \dot{E}^{Q} = \dot{E}_{out} + \dot{W}_x + \dot{I} \qquad (2)$$

Various amounts of lost work or exergy loss due to irreversibility can be estimated for each process of typical thermal plants and wherever possible the irreversibility rate will be represented graphically on the T - S diagram.

1. Adiabatic turbine (Figure 1)

The expansion process (Fig. 1b] involves an increase in entropy, which in the absence of any heat transfer $\left(\sum \dot{Q}_i = 0\right)$ consists all of entropy production $\left(\dot{\Pi}\right)$. Thus from (1) and with $\sum \dot{Q}_i = 0$, we have

$$\dot{I}_t = T_o \left(\dot{S}_2 - \dot{S}_1 \right) \qquad (3)$$

This is represented by the shaded rectangular area on the T - S diagram.

From (2) with $\sum \dot{Q}_i = 0$ (hence $\dot{E}^Q = 0$) and $\dot{W}_x = \left(\dot{W}_x\right)_t$

$$\dot{E}_1 - \dot{E}_2 = \left(\dot{W}_x\right)_t + \dot{I}_t \qquad (4)$$

Expression (4) shows that the drop in exergy between the given states 1 and 2 is made up of the turbine power output and the irreversibility rate.

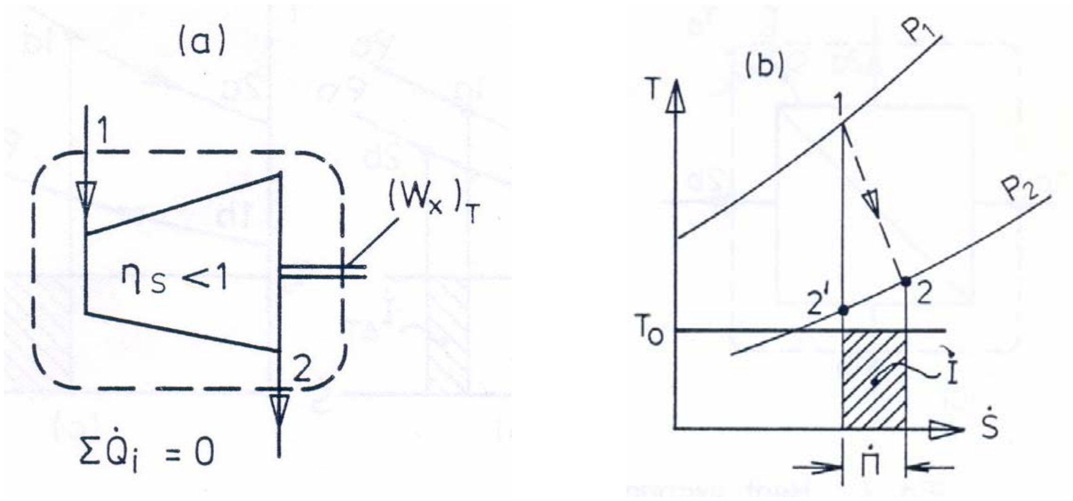


Figure 1. Adiabatic turbine

2. Adiabatic compressor (Figure 2)

In this case we have as before $\sum \dot{Q}_i = 0$ and also $\dot{W}_x = \left(\dot{W}_x\right)_c$. The expression for irreversibility rate therefore becomes

$$\dot{I}_c = T_o\left(\dot{S}_2 - \dot{S}_1\right) \tag{5}$$

This is represented by the cross-hatched area in Fig,2b. The exergy balance can be written in this case as follows;

$$\left(\dot{W}_x\right)_c = \dot{E}_2 - \dot{E}_1 + \dot{I}_c \tag{6}$$

The interpretation o-f this is that the power input to the compressor is turned into an increase in the exergy of the gas and some of it is dissipated .through irreversibility.

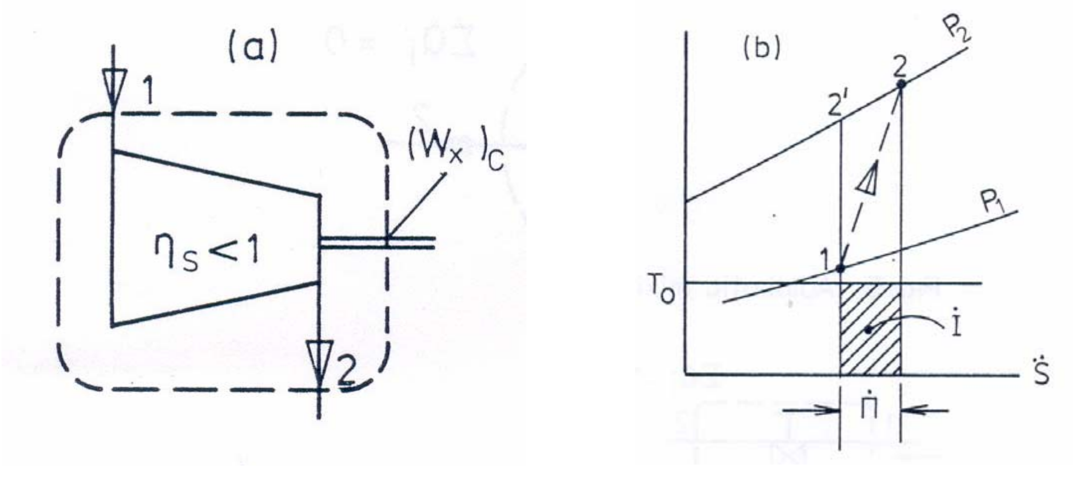


Figure 2. Adiabatic compressor

3. Frictionless compressor with heat transfer to environment (Figure 3)

The compression process is of polytropic type $\left(PV^n = C\right)$ with as shown in Figure 3. Since the heat transfer is to the environment we have $\dot{Q}_i = -\dot{Q}_o$ and $T_i = T_o$, hence $\dot{E}^Q = 0$.

Thus we have

$$\dot{I}_c = \dot{Q}_o - T_o\left(\dot{S}_1 - \dot{S}_2\right) \quad (7)$$

And

$$\left(\dot{W}_x\right)_c = \dot{E}_2 - \dot{E}_1 + \dot{I}_c \quad (8)$$

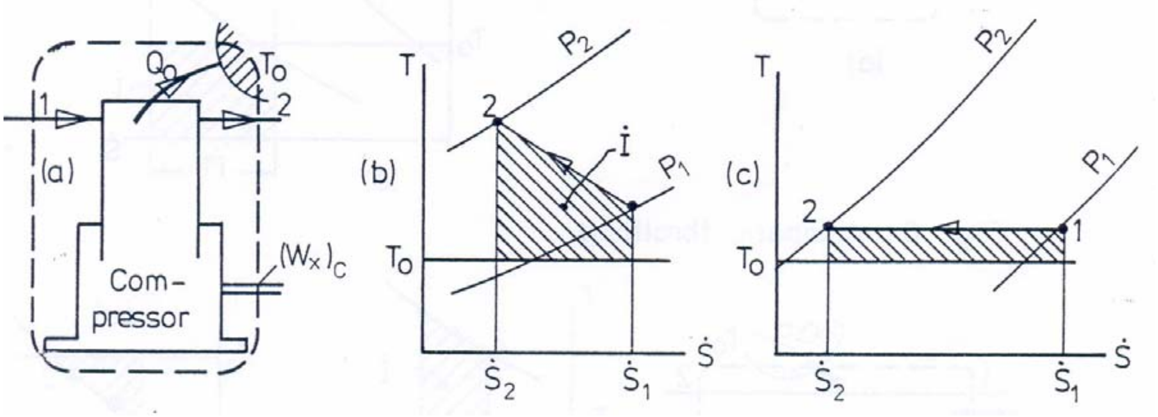


Figure 3. Frictionless compressor with heat transfer to environment

The irreversibility rate is represented in Fig. 3b by the cross-hatched area and is due to degradation of the thermal energy rejected to the environment by the compressed gas. The interpretation o-F expression (8) is the same as for (6).

In Fig. 3c the special case when n = 1 (i.e. an isothermal compression is shown. If the compression, process takes place at a temperature very close to T_o, the irreversibility of the process (represented by the cross-hatched area) becomes very small. This may be seen as a justification for using an isothermal process as an ideal model for comparing with real polytropic compression processes.

Hence we have the isothermal efficiency.

4. Heat exchanger (Figure 4)

Two types of heat exchangers are considered here, counter flow and parallel flow type. In the case of operation above ambient temperature there is likely to be some "heat loss" from the heat exchanger to the environment, i.e. $\dot{Q}_i = -\dot{Q}_o$ and $T_i = T_o$, hence $\dot{E}^Q = 0$.. Hence, we can write

$$\dot{I}_{hx} = T_o\left(\dot{S}_2 - \dot{S}_1\right) + \dot{Q}_o \quad (9)$$

where

$$\begin{aligned} \dot{S}_1 &= \dot{S}_{1a} + \dot{S}_{1b} \\ \dot{S}_2 &= \dot{S}_{2a} + \dot{S}_{2b} \end{aligned} \quad (10)$$

and so,

$$\dot{I}_{hx} = T_o\left[\left(\dot{S}_{2b} - \dot{S}_{1b}\right) - \left(\dot{S}_{1a} - \dot{S}_{2a}\right)\right] + \dot{Q}_o \tag{11}$$

The exergy balance becomes

$$\dot{E}_1 - \dot{E}_2 = \dot{I}_{hx} \tag{12}$$

where

$$\begin{aligned} \dot{E}_1 &= \dot{E}_{1a} + \dot{E}_{1b} \\ \dot{E}_2 &= \dot{E}_{2a} + \dot{E}_{2b} \end{aligned} \tag{13}$$

so that on substitution of (13) in (12) we get

$$\left(\dot{E}_{1a} - \dot{E}_{2a}\right) - \left(\dot{E}_{2b} + \dot{E}_{1b}\right) = \dot{I}_{hx} \tag{14}$$

The irreversibility rate o-f a heat exchanger is in general due to three. principal causes:

(a) heat transfer over a finite difference in temperature between the two streams.

(b) frictional dissipation (resulting in pressure losses).

(c) "heat losses" to the environment. The "heat loss" $\dot{Q}_o$ may in general be partly from stream a and .partly -from stream *b*. In the case of heat exchangers operating at temperatures below ambient temperature it is the heat gain rather than heat loss that is the cause of this type of irreversibility.

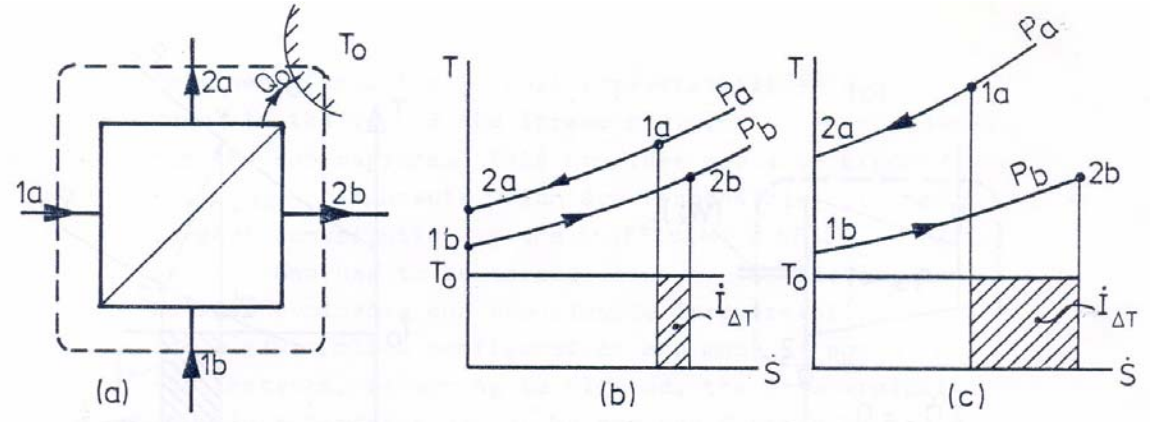


Figure 4. Heat exchanger

In the special case when $\dot{Q}_o = 0$ and frictional losses are negligible the cross-hatched area in Figure 4b and 4c represent irreversibilities due to the heat transfer over a finite temperature difference $\dot{I}^{\Delta T}$ for the counterflow and the parallel flow heat exchangers respectively.

Since the later case we must have $T_{2a} > T_{2b}$, it is clear that for the same minimum temperature difference between the streams the counterflow heat exchanger involves smaller irreversibility of the two and with the same heat capacities $\left(\dot{m}_a c_{pa} = \dot{m}_b c_{pb}\right)$ may in the limit $\left(as\, \Delta T \rightarrow 0\right)$ become reversible $\left(i.e\, \dot{I}^{\Delta T} = 0\right)$.

Referring to expression (14) it will be seen that in the case of a heat exchange operating above ambient temperature the loss of exergy $\left(\dot{E}_{1a} - \dot{E}_{2a}\right)$of the hotter stream is greater than the gain of available energy $\left(\dot{E}_{2b}-\dot{E}_{1b}\right)$ of the cooler streams and the difference between the two is equal to the irreversibility rate $\dot{I}_{hx}$.

The relative values of $\left(\dot{E}_{1a}-\dot{E}_{2a}\right)$ and $\left(\dot{E}_{2b}-\dot{E}_{1b}\right)$ are reversed in the case when the heat exchanger operates at temperatures below the ambient temperature.

5. Adiabatic mixing chamber (Figure 5)

Since in this case $\dot{W}_x = 0$ and $\sum \dot{Q}_i = 0$ the irreversibility rate is given by

$$\dot{I}_{mc} = T_o\left(\dot{S}_2 - \dot{S}_1\right) \qquad (15)$$

where

$$\begin{aligned}\dot{S}_1 &= \left(\dot{m}_{1a}\, s_{1a} + \dot{m}_{1b}\, s_{1b}\right)\\ \dot{S}_2 &= \left(\dot{m}_{1a} + \dot{m}_{1b}\right)s_2\end{aligned} \qquad (16)$$

and the exergy balance becomes

$$\dot{E}_1 - \dot{E}_2 = \dot{I}_{mc} \qquad (17)$$

where

$$\begin{aligned}\dot{E}_1 &= \dot{m}_{1a}\varepsilon_{1a} + \dot{m}_{1b}\varepsilon_{1b}\\ \dot{E}_2 &= \left(\dot{m}_{1a} + \dot{m}_{1b}\right)\varepsilon_2\end{aligned} \qquad (18)$$

The process of mixing two or more substances of different chemical species and/or in different thermodynamic states is inherently irreversible.

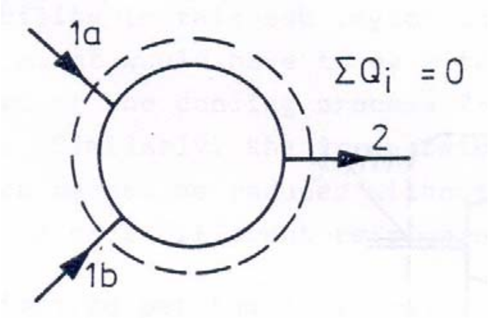


Figure 5. Adiabatic mixing chamber

6. Adiabatic throttling (Figure 6)

This process may occur where a flow of a fluid takes place through a restricted passage such as a partially opened, well lagged valve (Fig. 6a). If the changes in kinetic energy may be taken to be negligible the initial and final enthalpies are equal i.e. $h_1 = h_2$.

With $\dot{W}_x = 0$ and $\sum \dot{Q}_i = 0$, the irreversibility rate is

$$\dot{I}_{th} = T_o\left(\dot{S}_2 - \dot{S}_1\right) \qquad (19)$$

and the exergy balance is

$$\dot{E}_1 - \dot{E}_2 = \dot{I}_{th} \tag{20}$$

This process is also inherently irreversible and the irreversibility is represented by the cross-hatched area in Fig. 6 b. The above considerations apply also to pressure loss through viscous friction in an adiabatic pipe flow.

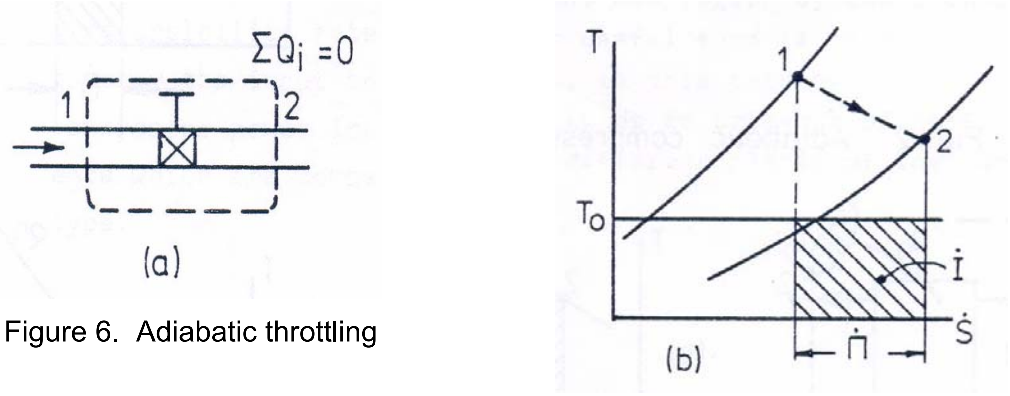


Figure 6. Adiabatic throttling

7. Thermal interaction with the environment (Figure 7)

When there is "heat loss", say, from a hot fluid flowing through a pipe (Fig. 7a) the fluid suffers a loss of exergy which is equal to the irreversibility rate. This is shown by adapting expressions (1) and (2) to the present case. Thus, with $\dot{Q}_i = -\dot{Q}_o$, $T_i = T_o$ and $\dot{W}_x = 0$ we get for irreversibility rate

$$\dot{I}_{ti} = T_o\left(\dot{S}_2 - \dot{S}_1\right) + \dot{Q}_o \tag{21}$$

and the exergy balance

$$\dot{E}_1 - \dot{E}_2 = \dot{I}_{ti} \tag{22}$$

Expression (21) is interpreted graphically in Fig. 7b. The irreversibility rate is equal to the loss of maximum potential work which could theoretically be obtained from the heat lost by the hot -fluid.

The case o-f the fluid carried by the pipe being colder than the surroundings (e.g. in cryogenics and refrigeration) is shown in Fig. 7c. In this case $\dot{Q}_i = \dot{Q}_o$ (i.e. the fluid gains heat from the environment) and $S_2 > S_1$.

Hence, the irreversibility rate is given by

$$\dot{I}_{ti} = T_o\left(\dot{S}_2 - \dot{S}_1\right) - \dot{Q}_o \tag{23}$$

and is represented by the cross hatched area. The exergy balance is given by expression (22), although it will be noted that higher exergy-is associated with lower temperature.

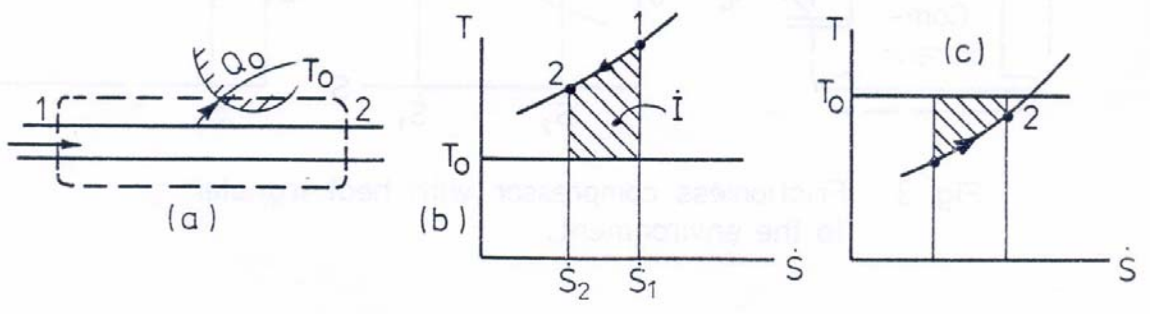


Figure 7. Thermal interaction with the environment

8. Vapor compression refrigeration cycle (Figure 8)

Expressions (1) and (2) which have been derived for an open system operating under steady conditions may be adapted for use with a steadily (or cyclically) operating heat engine, or reversed heat engine. If the control surface is drawn around the plant so that no streams of matter cross it (as shown for the refrigerator in Fig. 8a) then

$$\dot{E}_1 = 0\ ,\ \dot{E}_2 = 0 \tag{24}$$

and

$$\dot{S}_1 = 0\ ,\ \dot{S}_2 = 0 \tag{25}$$

By combining (1) and (25) the irreversibility rate for such a control region is given by

$$\dot{I} = -T_o \sum_{0}^{n} \left[\frac{\dot{Q}_i}{T_i} \right] \tag{26}$$

Considering the heat transfers and the temperature at the control surface where they take place the irreversibility rate of the whole plant is

$$\dot{I} = T_o \left(\frac{\dot{Q}_o}{T_o} - \frac{\dot{Q}_i}{T_i} \right) \tag{27}$$

As a consequence of (24), the exergy balance for this type of control region is

$$\dot{E}^Q = \dot{W}_x + \dot{I} \tag{28}$$

but

$$\dot{E}^Q = \dot{E}_i^Q + \dot{E}_o^Q$$

where

$$\dot{E}_i^Q = \dot{Q}_i \left(\frac{T_i - T_o}{T_i} \right) = -\dot{Q}_i \left(\frac{T_o - T_i}{T_i} \right)$$

and $\dot{Q}_o = 0$

Also $\dot{W}_x = -\left(\dot{W}_x\right)_c$

Hence, for this particular plant the exergy balance is

$$\left(\dot{W}_x\right)_c = \dot{Q}_i\left(\frac{T_o - T_i}{T_i}\right) + \dot{I} \tag{29}$$

On the LHS o-f expression (29) we have the actual power input to the plant. On the RHS the first term is the minimum power[*] -for transferring heat at the rate $\dot{Q}_i$ from temperature T_r to T_i ,which is the primary function of the plant. The second term on the RHS, the reversibility rate, represents the difference between the actual and the minimum power input. The plant can be further analyzed by dividing it into four sub-regions, each containing one of the principal components of the plant. Each sub-region is an open system operating under steady conditions (note that (24) and (25) will not apply to the sub-regions).

<u>Sub-region A</u> contains the adiabatic compressor. Using the same method of analysis as in Section 2, above, the expressions for the irreversibility rate and the exergy balance are

$$\dot{I}_A = T_o\left(\dot{S}_2 - \dot{S}_1\right) \tag{30}$$

$$\left(\dot{W}_x\right)_c = \dot{E}_2 - \dot{E}_1 + \dot{I}_A \tag{31}$$

The cross-hatched area in Fig. 8e represents irreversibility of the process.

<u>Sub-region B</u> contains the condenser. The function of a condenser in a refrigerator is to reject thermal energy to the environment. In this respect it is like the system illustrated in Section 7. Thus the expression for the irreversibility rate is

$$\dot{I}_A = \dot{Q}_o - T_o\left(\dot{S}_2 - \dot{S}_1\right) \tag{32}$$

and is represented by the cross-hatched area in Fig. 8d. The exergy balance can be written in the following form

$$\dot{E}_2 - \dot{E}_3 = \dot{I}_B \tag{33}$$

It follows from this expression that as the refrigerant passes through the condenser its exergy is dissipated through irreversibility.

<u>Sub-region C</u> contains the throttling valve. The method of analysis of an adiabatic throttling valve has been dealt with in Section 5, above. The irreversibility rate is

$$\dot{I}_C = T_o\left(\dot{S}_4 - \dot{S}_3\right) \tag{34}$$

and is given by the cross-hatched area in Fig. 8e. The exergy balance is

$$\dot{E}_3 - \dot{E}_4 = \dot{I}_C \tag{35}$$

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[*] Note that $\dot{W}_{x\,min} = \dot{Q}_i \Big/ \left(\frac{T_i}{T_o - T_i}\right) = \dot{Q}_i \big/ (COP)_{carnot}$, as for a plant operating on the Carnot cycle between T_i and T_o (see Figure 8 b).

Sub-region D contains the evaporator which operates over a range o-f temperatures below that of the environment. Heat from the lower temperature reservoir (cold store) at temperature Ti is transferred to the evaporating refrigerant. As this happened the exergy of the cold store increases and that of the refrigerant decreases ($E_i<E_4$). The irreversibility for this sub-region is given by

$$\dot{I}_D = T_o\left[\left(\dot{S}_1 - \dot{S}_2\right) - \frac{\dot{Q}_i}{T_i}\right] \tag{36}$$

The exergy balance can be conveniently written as follows.

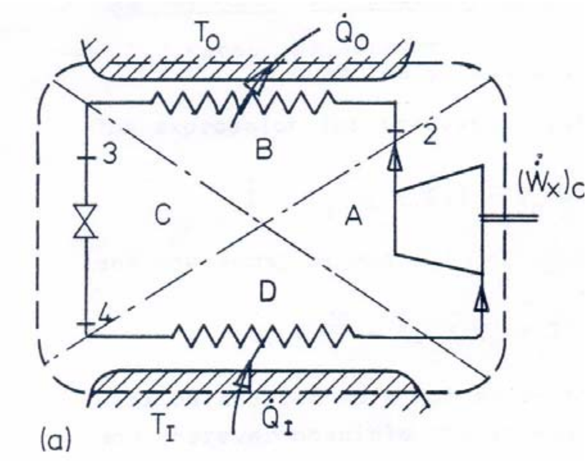
$$\dot{E}_4 - \dot{E}_1 = -\dot{E}^Q + \dot{I}_D \tag{37}$$

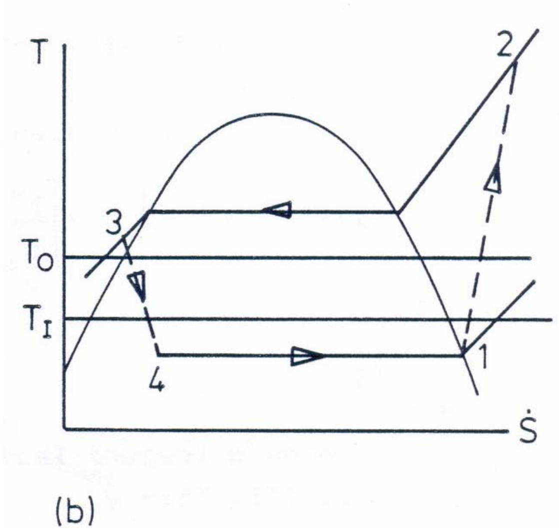
where $-\dot{E}^Q = -\dot{E}_i^Q = \dot{Q}_i \dfrac{T_o - T_i}{T_i}$ (38)

hence,

$$\dot{E}_4 - \dot{E}_1 = \dot{Q}_i \frac{T_o - T_i}{T_i} + \dot{I}_D \tag{39}$$

The first terra an the RHS of the above equation represent the gain of exergy by the cold store. These processes are shown in Fig. 8f in which the effects of frictional pressure losses and "heat gain" -From the environment have been .neglected. The two cross-hatched areas under these conditions are equal and this leads to (using (36)) the value of $\dot{\Pi}^{\Delta T}$ i.e. entropy production rate due to heat transfer over a finite temperature difference. The corresponding irreversibility rate for this sub-region is then given by the dotted area.





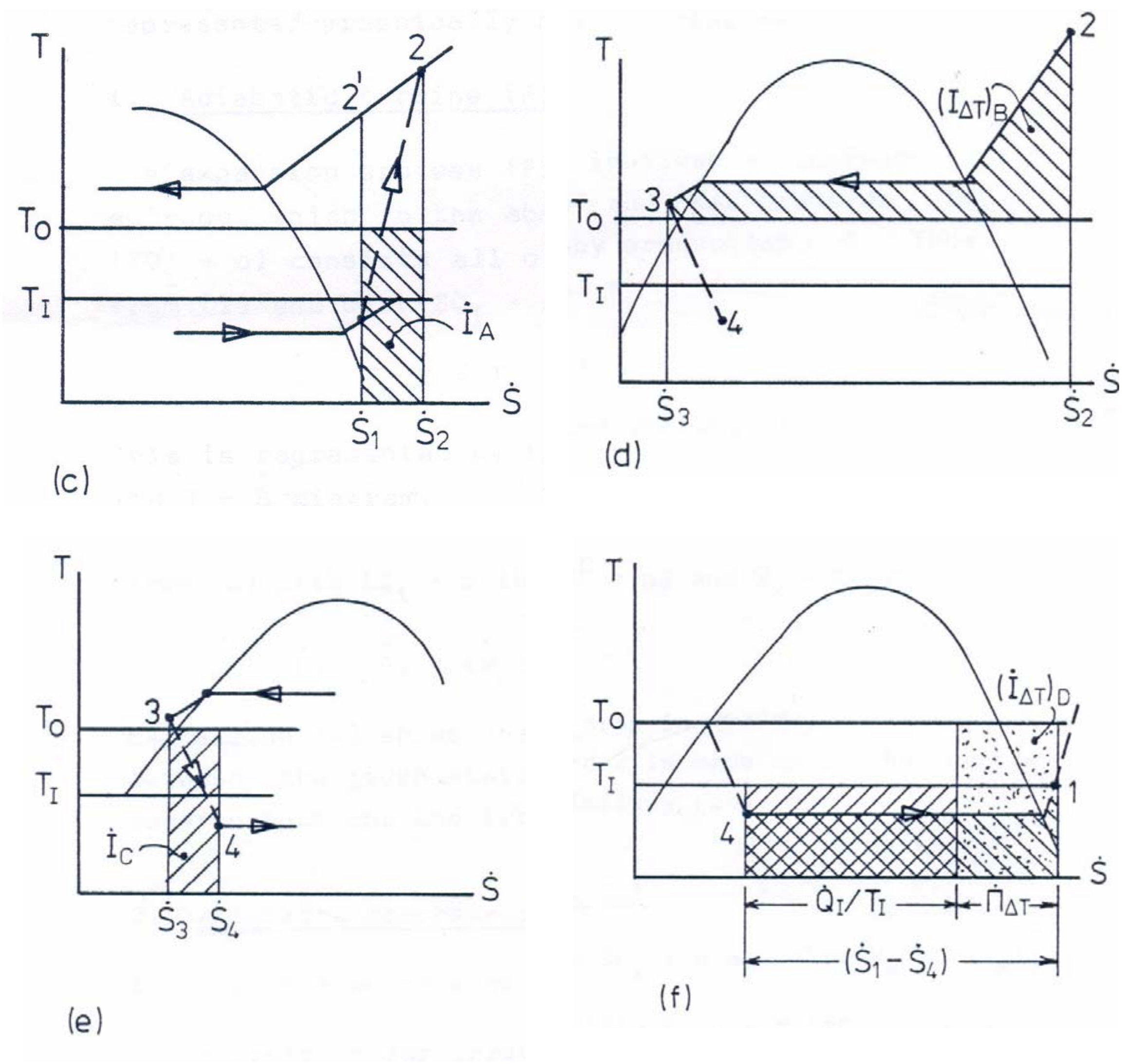


Figure 8. Vapor compression refrigerator

9. Comparison of irreversibility losses in the various sub-regions of the plant

As is well-known entropy is an extensive property, Hence we can write for the plant under consideration.

$$\dot{I} = T_o\,\dot{\Pi} = T_o\left(\dot{\Pi}_A + \dot{\Pi}_B + \dot{\Pi}_C + \dot{\Pi}_D\right) = \dot{I}_A + \dot{I}_B + \dot{I}_C + \dot{I}_D \tag{40}$$

This means that the overall irreversibility rate is equal to the sum of the irreversibility rates evaluated for the sub-regions. This provides means of pinpointing those plant components which are responsible for the largest contribution to the inefficiency of the plant. One has to be careful however, to distinguish between avoidable and unavoidable irreversibilities for a given plant configuration and mode of operation.

For instance, referring to Fig. 8d, the irreversibility of this sub-region cannot be reduced further beyond a certain point even if the condensation occurs at a temperature very close to T_o. To obtain further reduction of the irreversibility in this sub-region the mode of operation of the plant would have to be altered, namely, the superheat part of the cooling process 2-3 would have to be eliminated. Similarly, the irreversibility of the throttling process cannot be reduced without a modification of the plant or use of a different refrigerant.

It .may be convenient to put the irreversibilities in a dimensionless form. One way of doing this is to divide the irreversibility rates of the sub-region by the overall irreversibility rate. Another useful form is to divide them by the input to the plant, in this case by the compressor power input. This leads to criteria of performance which are comparable for different plants of the same type.

10. Gas turbine power plant with inlet air cooler (Figure 9)

Various amounts of lost work or exergy loss due to irreversibility can be estimated for each component in the gas turbine plant and the proposed air cooler system as given below.

10.1- Compressor

Rate of exergy destruction or irreversibility in the compressor,

$$\dot{I}_{comp} = \dot{m}_o T_o (s_2 - s_1) + \alpha\left(\frac{\dot{m}_o}{1-\alpha}\right) T_o (s_6 - s_1)$$

$$\dot{I}_{comp} = \dot{m}_a (1+\omega_1) T_o \left[c_{pa}\, \ell n\left(\frac{T_2}{T_1}\right) - R_a\, \ell n\left(\frac{P_2}{P_1}\right) + \left(\frac{\alpha}{1-\alpha}\right)\left(c_{pa}\, \ell n\left(\frac{T_6}{T_1}\right) - R_a\, \ell n\left(\frac{P_6}{P_1}\right)\right)\right] \quad (41)$$

where $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{(\gamma\eta_{cr})}}$

10.2- Combustion Chamber

Combustion process is often accomplished by heat transfer as well as fluid friction and mixing so there is usually more than one form of irreversibility present. In principle it is impossible to evaluate in this case what part of the combustion total irreversibility is due to any particular cause. The process of combustion can be examined, however by assuming that it takes place under adiabatic conditions and that irreversibilities due to friction and mixing are negligible.

One important concept of fuel combustion is the chemical exergy ε_o which has been extensively dealt with by Kotas, 1995. Therefore, the only relevant aspects that used in this paper are explained briefly.

The chemical exergy of a combustible substance can be obtained from (Kotas, 1995)

$$\varepsilon_o = -\Delta h_o + T_o \Delta s_o + RT_o\left[x_{O_2}\, \ell n \frac{P_{O_2}}{P_o} - \sum x_k\, \ell n \frac{P_k}{P_o} \right] \quad (42)$$

where the subscript *k* refers to the components of the products of composition.

Equation (42) may be applied to the calculation of chemical exergy of *gaseous fuels* for which the chemical composition can be determined and the thermochemical data for the components can be obtained. The ratio of chemical exergy ε_o to the net calorific value (*NCV*) can be by φ, ie:

$$\varphi = \frac{\varepsilon_o}{NCV} \tag{43}$$

For *dry solid fossil fuels* the ratio φ with a mass ratio of oxygen to carbon (*o/c*) less than 0.667 is given by:

$$\varphi = 1.0437 + 0.1882\left(\frac{h}{c}\right) + 0.061\left(\frac{o}{c}\right) + 0.0404\left(\frac{n}{c}\right) \tag{44}$$

where *c, h, o* and *n* are the mass fractions of carbon, hydrogen, oxygen and nitrogen, respectively.

For *solid fuels* with the mass ratio 2.67 > $\left(\frac{o}{c}\right)$ > 0.667, which, in particular includes wood.

$$\varphi = \frac{1.0438 + 0.1882\left(\frac{h}{c}\right) - 0.2509\left(1 + 0.7256\left(\frac{h}{c}\right)\right) + 0.0383\left(\frac{n}{c}\right)}{1 - 0.3035\left(\frac{o}{c}\right)} \tag{45}$$

For *liquid fuels* the effect of sulphur was included in the correlation giving the expression:

$$\varphi = 1.0401 + 0.1728\left(\frac{h}{c}\right) + 0.0432\left(\frac{o}{c}\right) + 0.2169\left(\frac{s}{c}\right)\left\{1 - 2.0628\left(\frac{h}{c}\right)\right\} \tag{46}$$

Typical values of φ for some industrial fuels are given in Table A1.

Fuel	$\varphi = \varepsilon_o / NCV$
Different types of coal	1.06-1.1
Wood	1.15-1.3
Different fuel oils and petrol	1.04-1.08
Natural gas	1.04 ± 5 %
Hydrogen	0.985
Carbon Monoxide	0.973

Table (A1)

Now, for a chemically reactive system

$\varepsilon_o = -\Delta g_o =$ the change in the standard *Gibbs function* change (*kJ/kmol*)

and $\Delta h = -(NCV)$= the change in the enthalpy of formation (*kJ/kmol*). Now, from equation (35),

$$\varphi = \frac{-\Delta G_o}{-\Delta H_o}$$

and $\Delta H_o = \dot{m}_f \times NCV \qquad (kW)$ (47)

where $\dot{m}_f$ is the fuel burning rate (*kg/s*).

Therefore, ΔG_o can be estimated.

Again,

$$\Delta G_o = \Delta H - T_o \Delta S_o$$

$T_o \Delta S_o$ = rate of exergy loss in combustion or reaction

$$= -\dot{m}_f \times NCV + \varphi\left(\dot{m}_f \times NCV\right) = \dot{m}_f \times NCV(\varphi - 1) = \dot{m}_a \times f \times NCV\,(\varphi - 1) \quad (48)$$

The rate of exergy destruction due to combustion,

$$\dot{I}_{comb} = T_o\left[\left(S_p\right)_3 - \left(S_R\right)_2\right] \quad (49)$$

where $\left(S_R\right)_2 = \left(S_A\right)_2 + \left(S_F\right)_2$ and the subscripts $P, R, A,$ and F represent *products*, *reactants*, *air* and *fuel*, respectively (Nag, 2002). Then

$$\dot{I}_{comb} = T_o\left[\left(S_p\right)_3 - \left(\left(S_A\right)_2 + \left(S_F\right)_2\right)\right]$$

$$\dot{I}_{comb} = T_o\left\{\left[\left(S_p\right)_3 - \left(S_p\right)_o\right] + \left(S_p\right)_o - \left[\left(S_A\right)_2 - \left(S_A\right)_o + \left(S_F\right)_o + \left(S_A\right)_o\right]\right\}$$

Now,

$$\Delta S_o = \left(S_p\right)_o - \left[\left(S_F\right)_o + \left(S_A\right)_o\right]$$

$$\dot{I}_{comb} = T_o\left\{\left[\left(S_p\right)_3 - \left(S_p\right)_o\right] - \left[\left(S_A\right)_2 - \left(S_A\right)_o + \Delta S_o\right]\right\}$$

$$\dot{I}_{comb} = \dot{m}_a T_o\left\{\left(1 + f + \omega_1\right)\left[c_{pg}\,\ell n\left(\frac{T_3}{T_o}\right) - R_g\,\ell n\left(\frac{P_3}{P_o}\right)\right] - \left(1 + \omega_1\right)\left[c_{pa}\,\ell n\left(\frac{T_2}{T_o}\right) - R_a\,\ell n\left(\frac{P_2}{P_o}\right)\right]\right\} + T_o \Delta S_o \quad (50)$$

where $T_o \Delta S_o$ is given in equation (48)

and $R_g = \dfrac{c_{pg}(\gamma - 1)}{\gamma}$

10.3-Gas Turbine

The rate of exergy destruction in the gas turbine is,

$$\dot{I}_t = \dot{m}_a\left(1 + f + \omega_1\right)T_o\left(s_4 - s_3\right)$$

$$= \dot{m}_a\left(1 + f + \omega_1\right)T_o\left[c_{pg}\,\ell n\left(\frac{T_4}{T_3}\right) - R_g\,\ell n\left(\frac{P_4}{P_3}\right)\right] \quad (51)$$

where $\dfrac{T_4}{T_3} = \left(\dfrac{P_4}{P_3}\right)^{\frac{(\gamma - 1)\eta_t}{\gamma}}$

10.4- Exhaust loss

The rate of exergy loss due to exhaust flue gases is,

$$\dot{I}_{exhaust} = \int_{T_e}^{T_o} \left(1 - \frac{T_o}{T} \right) dQ$$

$$= \dot{m}_a \left(1 + f + \omega_1 \right) c_{pg} \left[\left(T_e - T_o \right) - T_o \, \ell n \left(\frac{T_e}{T_o} \right) \right] \qquad (52)$$

In this study the exhaust loss is not calculated at T_4, because T_4 can be reduced sufficiently by using either a regenerator in gas turbine cycle or a heat exchanger steam generator *(HESG)* in combined cycle power generation. Therefore the exist temperature is limited to sulphur erosion limit which is 130 ^oC (403.15 K) (Huang, et al.; 2000).

10.5-Heat Exchanger

The rate of exergy destruction in the heat exchanger is,

$$\dot{I}_{HX} = \alpha \dot{m}_1 T_o \left(s_6 - s_7 \right) + \frac{T_o}{T_6} \dot{Q}_{out} \qquad (53a)$$

$$\dot{I}_{HX} = \alpha \left(\frac{\dot{m}_a}{1 - \alpha} \right) \left(1 + \omega_1 \right) T_o \left(s_6 - s_7 \right) + \frac{T_o}{T_6} \dot{Q}_{out} \qquad (53b)$$

where $\dot{Q}_{out}$, is the rate of heat rejection from the condenser at temperature T_6.

10.6-Expander

The rate of exergy destruction in the expander can be expressed as,

$$\dot{I}_{exp} = \alpha \dot{m}_1 T_o \left(s_8 - s_7 \right)$$

$$= \alpha \left(\frac{\dot{m}_a}{1 - \alpha} \right) \left(1 + \omega_1 \right) T_o \left[c_{pa} \, \ell n \left(\frac{T_8}{T_7} \right) - R_a \, \ell n \left(\frac{P_8}{P_7} \right) \right] \qquad (54)$$

where $\frac{T_8}{T_7} = \left(\frac{P_8}{P_7} \right)^{\frac{(\gamma - 1)\eta_{exp}}{\gamma}}$, η_{exp} the expander isentropic efficiency.

10.7-Mixing chamber

Jassim, *et al* 2004 have dealt extensively with various aspects of heat transfer processes of mixing including determination of the optimum balance between *physical* and *chemical* irreversibilities.

The rate of exergy destruction in the mixing chamber is completely different then the above expressions, therefore the following subsections will deal with it in more details.

10.7.1-Exergy Based Property Tables

The most general definition of physical exergy can be expressed in molar form as (Jassim *et.al* 2004)

$$\tilde{\varepsilon}_{ph} = \left(\tilde{h} - \tilde{h}_o \right) - T_o \left(\tilde{s} - \tilde{s}_o \right) \qquad (55a)$$

This equation can be used with suitable property tables (Kotas, 1995). For an ideal gas, Eq. (47a) may be, conveniently expressed in terms of thermal component $\tilde{\varepsilon}^{\Delta T}$ and the pressure component $\tilde{\varepsilon}^{\Delta P}$ as,

$$\tilde{\varepsilon}_{ph} = \tilde{\varepsilon}^{\Delta T} + \tilde{\varepsilon}^{\Delta P} = \tilde{c}_p^{\varepsilon}\left(T - T_o\right) + \tilde{R}\,T_o\,ln\left(\frac{P}{P_o}\right) \tag{55b}$$

where

$$\tilde{c}_p^{\varepsilon} = \frac{\tilde{\varepsilon}^{\Delta T}}{T - T_o} = \frac{1}{T - T_o}\left[\int_{T_o}^{T}\tilde{c}_p dT - T_o\int_{T_o}^{T}\frac{\tilde{c}_p dT}{T}\right] \tag{56}$$

$\tilde{c}_p^{\varepsilon}$ may be looked upon as*, mean, molar isobaric exergy capacity.*

Two forms of mean molar isobaric heat capacities may be defined one for enthalpy and the other for entropy evaluation. They are respectively,

$$\tilde{c}_p^{h} = \frac{1}{T - T_o}\int_{T_o}^{T}\tilde{c}_p\,dT \tag{57}$$

$$\tilde{c}_p^{s} = \frac{1}{\ell n\left(T/T_o\right)}\int_{T_o}^{T}\frac{\tilde{c}_p\,dT}{T} \tag{58}$$

where $\tilde{c}_p$ is the molar isobaric specific heat capacity.

Using a polynomial expression $\tilde{c}_p$ both $\tilde{c}_p^{h}$ and $\tilde{c}_p^{s}$ can be evaluated (Kotas, 1995). Hence $\left(\tilde{c}_p^{\varepsilon}\right)$ can be calculated from,

$$\tilde{c}_p^{\varepsilon} = \tilde{c}_p^{h} - T_o\tilde{c}_p^{s}\,\frac{\ell n\left(T/T_o\right)}{T - T_o} \tag{59}$$

Values of $\tilde{c}_p^{h}$, $\tilde{c}_p^{s}$ and $\tilde{c}_p^{\varepsilon}$ have been tabulated for a number of common gases over a range of temperatures with T_o = 298.15 K,

By using tables of $\tilde{c}_p^{h}$, $\tilde{c}_p^{s}$ and $\tilde{c}_p^{\varepsilon}$ which are tabulated in (Kotas, 1995) appendix D, energy and exergy balances for physical and chemical processes can be carried out speedily.

Substituting Eq. (59) into Eq. (55), we have

$$\tilde{\varepsilon}_{ph} = \tilde{c}_p^{h}\left(T - T_o\right) - T_o\tilde{c}_p^{s}\,\ell n\left(\frac{T}{T_o}\right) + \tilde{R}\,T_o\,\ell n\left(\frac{P}{P_o}\right) \tag{60}$$

10.7.2-Mixture of ideal gases

The *physical exergy* of a mixture of N components can be evaluated as,

$$\left(\tilde{\varepsilon}_{ph}\right)_M = \sum_{j=1}^{N} x_j\,\tilde{\varepsilon}_j^{\Delta T} + \tilde{R}\,T_o\,\ell n\left(\frac{P}{P_o}\right) \tag{61}$$

where P is the total pressure of the mixture

Using tabulated values of, Eq. (61) may be written in alternative forms,

$$\left(\tilde{\varepsilon}_{ph}\right)_M = \sum_{j=1}^{N} x_j \left[\tilde{c}_p^{\,h}\left(T - T_o\right) - T_o\,\tilde{c}_p^{\,s}\,\ell n\left(\frac{T}{T_o}\right)\right]_j + \tilde{R}\,T_o\,\ell n\left(\frac{P}{P_o}\right) \qquad (62)$$

Substituting "Eq. (59)" into "Eq. (62)" we have

$$\left(\tilde{\varepsilon}_{ph}\right)_M = \left(T - T_o\right)\sum_{j=1}^{N} x_j \tilde{c}_{p,j}^{\,\varepsilon} + \tilde{R}\,T_o\,\ell n\left(\frac{P}{P_o}\right) \qquad \textit{(kJ/kmol)} \qquad (63)$$

10.7.3-Standard Chemical Exergy (ε_o)

Exergy balance could be more easily compiled if the value of the chemical exergy were tabulated. (Kotas, 1995 and Szargut, *et al* 1988) have included the notion of standard chemical exergy. Values of standard chemical exergy for various chemical elements have been computed and tabulated (Kotas, 1995 and Szargut, *et al* 1988). The use of such tables can be found to greatly facilitate and speed up the calculation of exergy balance for chemical processes.

In this case study the standard chemical exergy of the CO_2, H_2O, O_2, N_2 are calculated from the following expression,

$$\tilde{\varepsilon}_o = \tilde{R}\,T_o\,\ell n\left(\frac{P_o}{P_{oo}}\right) \qquad \textit{(kJ/kmol)} \qquad (64)$$

where P_{oo} is the substance partial pressure

The standard chemical exergy of the common constituents are calculated from the above equation and tabulated in Table A2.

Chemical Symbol	Mole Fraction in dry air	Standard partial pressure in the environment (kPa)	Molar chemical Exergy (*kJ/kmol*) eq. (54)
CO_2	0.000345	0.0335	19867.56
H_2O	--	0.88	11765.45
O_2	0.2099	20.40	3973.24
N_2	0.7803	75.78	720.13

Table A2 Standard of chemical elements (T_o = 298.15 K and P_o =101.325 kPa)

10.7.4-Chemical Exergy of a Mixture

The chemical exergy of a mixture is given by the following expression (Kotas, 1995)

$$\tilde{\varepsilon}_{oM} = \sum_j x_j\,\tilde{\varepsilon}_{oj} + \tilde{R}\,T_o \sum_j x_j\ \ell n\ x_j \qquad \textit{(kJ/kmol)} \qquad (65)$$

where x_j is the mole fraction of j th component in the mixture

$\tilde{\varepsilon}_o$ is the molar chemical exergy (*kJ/kmol*)

10.7.5- Mixing irreversibility

Consider an adiabatic mixing process involving one cold fluid and one hot fluid as shown in Figure 1. The exergy balance with $\dot{Q}_o$ = 0 is,

$$\dot{I}_{mc} = \dot{E}_8 + \dot{E}_o - \dot{E}_1 \tag{66}$$

or

$$\dot{I}_{mc} = \alpha \dot{m}_1 \, \varepsilon_8 + \dot{m}_o \, \varepsilon_o - \dot{m}_1 \, \varepsilon_1 \tag{67}$$

In general, the specific exergy components are,

$$\varepsilon = \varepsilon_o + \varepsilon_{ph} \tag{68}$$

Hence

$$\dot{I}_{mc} = \alpha \dot{m}_1 \left(\varepsilon_{o,8} + \varepsilon_{ph,8} \right) / \tilde{m}_{\mathbf{8}} + \dot{m}_o \left(\varepsilon_{o,o} + \varepsilon_{ph,o} \right) / \tilde{m}_o - \dot{m}_1 \left(\varepsilon_{o,1} + \varepsilon_{ph,1} \right) / \tilde{m}_{\mathbf{1}} \tag{69a}$$

Rearranging equation (69a), yields:

$$= \dot{m}_a \left(1 + \omega_1 \right) \left[\left(\frac{\alpha}{1 - \alpha} \right) \frac{\left(\varepsilon_{o,8} + \varepsilon_{ph,8} \right)}{\tilde{m}_{\mathbf{8}}} + \frac{\left(\varepsilon_{o,o} + \varepsilon_{ph,o} \right)}{\tilde{m}_o} - \left(\frac{1}{1 - \alpha} \right) \frac{\left(\varepsilon_{o,1} + \varepsilon_{ph,1} \right)}{\tilde{m}_{\mathbf{1}}} \right] \tag{61b}$$

The values of ε_{ph} and ε_o can be calculated from Eqs. (61) and (63), respectively,

Total rate of exergy losses in the plant,

$$\sum \dot{I} = \dot{I}_{comp} + \dot{I}_{comb} + \dot{I}_t + \dot{I}_{exhaust} + \dot{I}_{HX} + \dot{I}_{exp} + \dot{I}_{mc} \tag{70}$$

The calculation procedure for mole fraction x_j of equations 61 and 63 have been presented in the following Tables A3, A4 and A5 for ($\alpha = 0.2$, x =3).

a-Outside stream

The molar mass of the outside stream is,

$$\tilde{m} = \sum_j x_j \tilde{m}_j = 28.76 \text{ kg/kmol}$$

Table A3 cold side calculation

Composition	Mass fraction $\mu_j = x_i \frac{\tilde{m}_j}{\tilde{m}}$	Mass of composition $m_j = \dot{m}_o \mu_j$ (kg/s)	Number of moles $n_j = \frac{m_j}{\tilde{m}_j}$ (kmol/s)	Mole fraction
CO_2	0.0005279	0.0006005	0.00001365	0.000345
H_2O	0.005923	0.006737	0.000374	0.009455
O_2	0.2335	0.2657	0.008302	0.2099
N_2	0.76	0.8645	0.03086	0.7803
Total number of moles			0.03955	

b-Cold side stream

$$\tilde{m} = \sum_j x_j \tilde{m}_j = 28.59 \text{ kg/kmol}$$

Table A4 cold side calculation

Composition	Mass fraction $\mu_j = x_i \frac{\tilde{m}_j}{\tilde{m}}$	Mass of composition $m_j = \alpha \dot{m}_1 \mu_j$ (kg/s)	Number of moles $n_j = \frac{m_j}{\tilde{m}_j}$ (kmol/s)	Mole fraction
CO_2	0.0005279	0.0001493	0.000003393	0.000345
H_2O	0.0	0.0	0.0	0.0
O_2	0.2335	0.06606	0.002065	0.2099
N_2	0.76	0.215	0.007675	0.7803
Total number of moles			0.009743	

(c) mixture

Table A5. Mixture calculation

Composition	Mass of composition $(m_j)_{cold} + (m_j)_{hot}$ (kg/s)	Number of moles $n_j = \frac{m_j}{\tilde{m}_j}$ (kmol/s)	Mole fraction
CO_2	0.0007508	0.00001706	0.0003457
H_2O	0.006737	0.000374	0.007577
O_2	0.3321	0.01038	0.2103
N_2	1.081	0.03858	0.7818
Total number of moles		0.04935	

The physical and chemical exergies for the mixing streams are calculated using equations 61 and 63. The values of mean molar isobaric exergy capacity, $\tilde{c}^{\varepsilon}_{p,j}$ for equation 61 have been evaluated from Table D3 (Kotas, 1995) and the values are linearly extrapolated and interpolated according to corresponding temperatures.

11. Exergetic efficiency

Exergetic efficiency is a criterion of performance which can be formulated for the gas turbine coupled to an air expansion cycle or its components for which the output is expressible in terms of exergy. Defining the exergetic efficiency η_{ex}, as a ratio of total rate of exergy output $\left(\dot{E}_{out}\right)$ to total rate of exergy input $\left(\dot{E}_{in}\right)$.

$$\eta_{ex} = \frac{\dot{E}_{out}}{\dot{E}_{in}} \tag{71}$$

The effective power output of the turbine

$$\dot{W}_{eff,t} = \dot{W}_t - \dot{I}_t \tag{72a}$$

The effective power output of the expander

$$\dot{W}_{eff,exp} = \dot{W}_{exp} - \dot{I}_{exp} \tag{72b}$$

The gross of power input to the air compressor is

$$\dot{W}_{eff,comp} = \dot{W}_{comp} + \dot{I}_{comp} + \dot{I}_{mc} \tag{72c}$$

The gross of power input to the combustion chamber is

$$\dot{Q}_{eff,cc} = \dot{Q}_{cc} + \dot{I}_{cc} \tag{73a}$$

The gross of power input to the *HX* is

$$\dot{Q}_{eff,HX} = \dot{Q}_{HX} + \dot{I}_{HX} \tag{73b}$$

The exergy balance for the gas turbine and air expansion cycle shown in Figure (1) can be expressed in the following form,

$$\dot{E}_{out} = \dot{W}_{eff,t} + \dot{W}_{eff,exp} - \dot{W}_{eff,comp} \tag{74}$$

The total rate of exergy input

$$\dot{E}_{in} = \dot{Q}_{eff,cc} - \dot{Q}_{eff,HX} \tag{75}$$

Then

$$\eta_{ex} = \frac{\dot{W}_{eff,t} + \dot{W}_{eff,exp} - \dot{W}_{eff,comp}}{\dot{Q}_{eff,cc} - \dot{Q}_{eff,HX}} \tag{76}$$

The irreversibilities of heat exchangers $\left(\dot{I}_{HX}\right)$, expanders $\left(\dot{I}_{exp}\right)$ and mixing chamber $\left(\dot{I}_{mc}\right)$ are all equal to zero when the GT cycle with no cooling system.

For the present parametric analysis let us define dimensionless terms that give the advantage of using a Brayton cycle as a cooling system as the *exergetic power gain ratio* (PGR_{ex}) and *exergetic thermal efficiency change* (TEC_{ex})

$$PGR_{ex} = \frac{\left(\dot{E}_{out}\right)_c - \left(\dot{E}_{out}\right)_{nc}}{\left(\dot{E}_{out}\right)_{nc}} \times 100\% \tag{77}$$

$$TEC_{ex} = \frac{\eta_{ex,c} - \eta_{ex,nc}}{\eta_{ex,nc}} \times 100\% \tag{78}$$

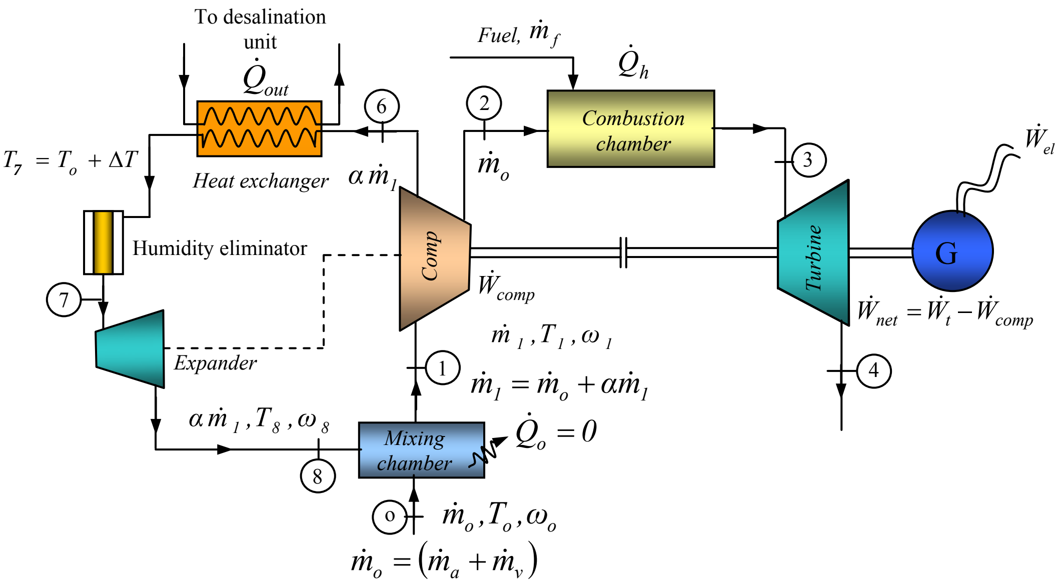


Figure 9 a. A schematic of a gas turbine with an air cooler cycle

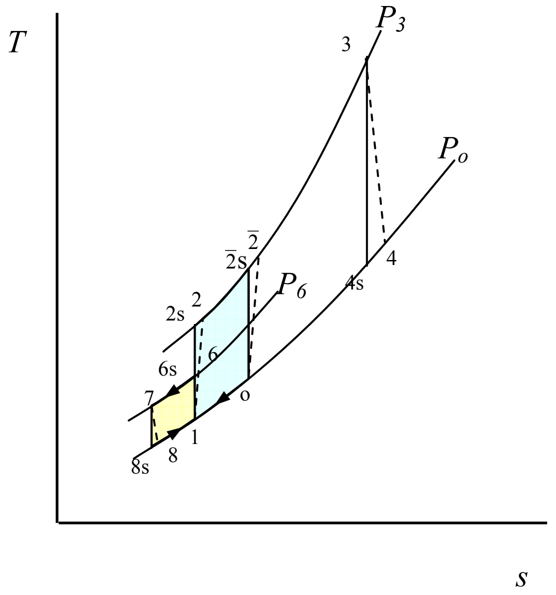


Figure 9 b.. *T-S* diagram for the proposed cycle

References of appendix A

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Dr. Jassim is an associate professor of mechanical engineering technology, Yanbu Industrial College, Kingdom of Saudi Arabia. He received his ***BSc (Hons)*** in Mechanical Engineering and Technical Education from University of Technology, Baghdad, (Iraq) in July 1987. He obtained his ***PhD*** degree in Mechanical Engineering from University of London, QMW College (UK) in August 1992. Dr. Jassim was actively involved in project management, human resource management and QA management with *APV* New Zealand Ltd and *APV* Australia Pty Ltd. In April 1999, he was awarded a Master degree in engineering management ***(MEMgt***) from The University of Auckland (NZ) where he specialized in Project Management, Human Resource Management, Operation Management, and Financial Management. His research work covers thermoeconomic optimisation of thermal systems, project management and HRM. He has published over 50 reviewed technical papers and technical reports, 2 contributed chapters in HRM books and 5 books. Dr. Jassim is a member of ASME and a member of IPENZ and holder for two titles, ***PE*** (Professional Engineer) and ***Reg. Eng.*** (Registered Engineer). Finally, his biography cited in the Annual Who's Who in Science and Engineering, Marquis, USA.