

Module 2

**ENERGY EFFICIENCY
IN
THERMAL SYSTEMS**



Sustainable and Renewable
Energy Development Authority

**Sustainable and Renewable Energy Development Authority
Power Division**

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Preamble

In order to ensure energy efficiency and conservation and to determine the future course of action, Sustainable and Renewable Energy Development Authority (SREDA) has developed the Energy Efficiency & Conservation Master Plan up to 2030 in 2016. According to this plan, the target of energy saving has been set 20% per GDP by 2030 which will be achieved by the use of energy efficient machinery and equipment as well as by improving energy management system in the demand side.

In order to achieve the above mentioned target & to ensure the energy efficiency and conservation in industrial & commercial sector, SREDA has formulated the Energy Audit Regulation'2018. Based on this regulation, SREDA will conduct the Energy Auditor Certification Examination to create energy auditors and energy managers in Bangladesh.

SREDA has prepared the following modules as reading material for four paper examinations in cooperation with various National and Foreign partner organizations.

Module No	Examination Paper	Subject
Module 01	Paper 01	Fundamentals of Energy Management and Energy Audit
Module 02	Paper 02	Energy Efficiency in Thermal Systems
Module 03	Paper 03	Energy Efficiency in Electrical Systems
Module 04	Paper 04	Energy Performance Assessment for Equipment and Utility Systems

This module 02 on Energy Efficiency in Thermal Systems is the reading material for the preparation of Paper 02 Examination for prospective candidates.

We hope that these modules will also act as valuable resource for practicing engineers in comprehending and implementing energy efficiency measures in the facilities.

It is the first iteration of these modules. It will be a living document which can be reviewed and revised time to time according to the evolution of the technology and industry. Any suggestion and comments (please email to ad.eaa@sreda.gov.bd) on the contents of those modules will be highly appreciated.

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Chapter 1: Fuels and Combustion

1.1 Introduction to Fuels

Various type of fuels like liquid, solid and gaseous fuels are available for firing in combustion equipment like boilers, furnaces etc. The selection of right type of fuel depends on the various factors such as availability, storage, handling, pollution and landed cost of fuel.

The knowledge of the fuel properties helps in selecting the right fuel for the right purpose and efficient use of the fuel. The following characteristics, determined by laboratory tests, are generally used for assessing the nature and quality of fuels.

1.2 Properties of Liquid Fuels

Liquid fuels like furnace oil, Diesel, LDO, HFO etc. are predominantly used in industrial application. The various properties of liquid fuels are given below.

Density

Density is the ratio of the mass of the fuel to the volume of the fuel at a reference temperature typically 15°C. The knowledge of density is useful for quantity calculations and assessing ignition quality. The unit of density is kg/m³.

Specific gravity

This is defined as the ratio of the weight of a given volume of oil to the weight of the same volume of water at a given temperature. The measurement of specific gravity is generally made by a hydrometer. The density of fuel, relative to water, is called specific gravity. The specific gravity of water is defined as 1. Since specific gravity is a ratio, there is no units.

Higher the specific gravity, higher is the heating value. Its main use is in calculations involving weights and volumes. The specific gravity of various fuel oil is given in Table 1.1.

Table 1.1: Specific gravity of various fuel oils

Fuel Oil	L.D.O Light Diesel Oil	Furnace oil	L.S.H.S Low Sulphur Heavy Stock
Specific Gravity	0.85-0.87	0.89-0.95	0.88-0.98

Viscosity

The viscosity of a fluid is a measure of its internal resistance to flow. Viscosity depends on temperature and decreases as the temperature increases. Any numerical value for viscosity has no meaning unless the temperature is also specified. Viscosity is measured in Stokes / Centistokes. Sometimes viscosity is also quoted in Engler, Saybolt or even

Redwood. Each type of oil has its own temperature-viscosity relationship. The measurement of viscosity is made with an instrument called as viscometer

Viscosity is the most important characteristic in the fuel oil specification. It influences the degree of pre-heat required for handling, storage and satisfactory atomization. If the oil is too viscous it may become difficult to pump, hard to light the burner and operation may become erratic. Poor atomization may result in the formation of carbon deposits on the burner tips or on the walls. Pre-heating is necessary for proper atomization.

Flash Point

The flash point of a fuel is the lowest temperature at which the fuel can be heated so that the vapour gives off flashes momentarily when an open flame is passed over it. Flash point for furnace oil is 660C.

Pour Point

The pour point of a fuel is the lowest temperature at which it will pour or flow when cooled under prescribed conditions. It is a very rough indication of the lowest temperature at which fuel oil is readily pump able

Specific Heat

Specific heat is the amount of kcal needed to raise the temperature of 1 kg of oil by 10⁰C. The unit of specific heat is kCal/kg⁰C. It varies from 0.22 to 0.28 depending on the oil specific gravity. The specific heat determines how much steam or electrical energy it takes to heat oil to a desired temperature. Light oils have a low specific heat, whereas heavier oils have a higher specific heat.

Calorific Value

The calorific value is the measurement of heat or energy produced, and is measured either as gross calorific value or net calorific value. The difference being the latent heat of condensation of the water vapour produced during the combustion process. Gross calorific value assumes all vapour produced during the combustion process is fully condensed. Net calorific value assumes the water leaves with the combustion products without fully being condensed. Fuels should be compared based on the net calorific value.

The calorific value of coal varies considerably depending on the ash, moisture content and the type of coal while calorific value of fuel oil is much more consistent. The typical Gross Calorific Values of some of the commonly used liquid fuels are given below:

Fuel Oil	Calorific Value (kcal/Kg)
Kerosene	-11,100
Diesel Oil	-10,800
L.D.O	-10,700
Furnace Oil	-10,500
LSHS	-10,600

The following conversion formula shows the difference between GCV and NCV.

$$\text{GCV} = \text{NCV} + 584 \left(\frac{9\text{H}_2\% + \text{M}\%}{100} \right)$$

Where,

GCV= Gross calorific value of fuel, kcal/kg

NCV = Net calorific value of fuel, kcal/kg

H₂%= Hydrogen % by weight present in the fuel

M% = Moisture % by weight present in the fuel

584 = Latent heat corresponding to partial pressure of water vapour, kCal/kg

Sulphur

The amount of sulphur in the fuel oil depends mainly on the source of the crude oil and to a lesser extent on the refining process. The normal sulphur content for the residual fuel oil (heavy fuel oil) is in the order of 2-4%.

Typical figures are:

Fuel oil	Percentage of <u>Sulphur</u>
Kerosene	0.05 - 0.2
Diesel Oil	0.3 -1.5
L.D.O	0.5 – 1.8
Heavy Fuel Oil	2.0 – 4.0
LSHS	< 0.5

The main disadvantage of sulphur is the risk of corrosion by sulphuric acid formed during and after combustion, and condensing in cool parts of the chimney or stack, air pre heater and economiser.

Ash Content

The ash value is related to the inorganic material in the fuel oil. The ash levels of distillate fuels are negligible. Residual fuels have more of the ash-forming constituents. These salts may be compounds of sodium, vanadium, calcium magnesium, silicon, iron, aluminium, nickel, etc.

Typically, the ash value is in the range 0.03-0.07%. Excessive ash in liquid fuels can cause fouling deposits in the combustion equipment. Ash has erosive effect on the burner tips, causes damage to the refractories at high temperatures and gives rise to high temperature corrosion and fouling of equipment.

Carbon Residue

Carbon residue indicates the tendency of oil to deposit a carbon acetous solid residue on a hot surface, such as a burner or injection nozzle, when its vaporizable constituents evaporate. Residual oil contain carbon residue ranging from 1 percent or more.

Water Content

Water content of furnace oil when supplied is normally very low as the product at refinery site is handled hot and maximum limit of 1% is specified in the standard.

Water may be present in free or emulsified form and can cause damage to the inside furnace surfaces during combustion especially if it contains dissolved salts. It can also cause spluttering of the flame at the burner tip, possibly extinguishing the flame and reducing the flame temperature or lengthening the flame.

Typical specification of fuel oil is summarized in the Table 1.2.

Table 1.2: Typical specification of fuel

Properties	Fuel Oils		
	Furnace Oil	LS.H.S.	L.D.O.
Density (Approx. g/cc at 15 ^o C)	0.89-0.95	0.88-0.98	0.85-0.87
Flash Point (^o C)	66	93	66
Pour Point (^o C)	20	72	12 (Winter) 18 (Summer)
G.C.V. (kcal/kg)	10,500	10,600	10,700
Sediment, % Wt. Max.	0.25	0.25	0.1
Sulphur Total, % Wt. Max.	Up to 4.0	Up to 0.5	Up to 1.8
Water Content, % Vol. Max.	1.0	1.0	0.25
Ash % Wt. Max.	0.1	0.1	0.02

Storage of Fuel oil

It can be potentially hazardous to store furnace oil in barrels. A better practice is to store it in cylindrical tanks, either above or below the ground. Furnace oil, that is delivered, may contain dust, water and other contaminants.

The sizing of storage tank facility is very important. A recommended storage estimate is to provide for at least 10 days of normal consumption. Industrial heating fuel storage tanks are generally vertical mild steel tanks mounted above ground. It is prudent for safety and environmental reasons to build bund walls around tanks to contain accidental spillages.

As a certain amount of settlement of solids and sludge will occur in tanks over time, cleaning should be carried out at regular intervals-annually for heavy fuels and every two years for light fuels. A little care should be taken when oil is decanted from the tanker to storage tank. All leaks from joints, flanges and pipelines must be attended at the earliest. Fuel oil should be free from possible contaminants such as dirt, sludge and water before it is fed to the combustion system.

Loss of Even One Drop of Oil Every Second Can Cost Over 4000 Litres A Year
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Removal of Contaminants

Furnace oil arrives at the factory site either in tank lorries by road or by rail. Oil is then decanted into the main storage tank. To prevent contaminants such as rags, cotton waste, loose nuts or bolts or screws entering the system and damaging the pump, coarse strainer of 10 mesh size (not more than 3 holes per linear inch) is positioned on the try pipe to the storage tanks.

Progressively finer strainers should be provided at various points in the oil supply system to filter away finer contaminants such as external dust and dirt, sludge or free carbon. It is advisable to provide these filters in duplicate to enable one filter to be cleaned while oil supply is maintained through the other.

The Figure 1.1 gives an illustration of the duplex system of arrangement of strainers.

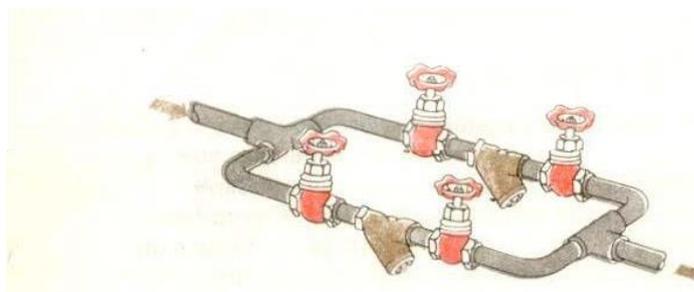


Figure 1.1: Duplex arrangement of strainers in a pipeline

The Table 1.3 gives sizing of strainers at various locations.

Table 1.3: Sizing of strainers at various locations

Location	Strainer Sizes	
	Mesh	Holes / Linear inch
Between rail / tank lorry decanting point and main storage tank	10	3
Between service tank and pre-heater	40	6
Between pre-heater and burner	100	10

Pumping

Heavy fuel oils are best pumped using positive displacement pumps, as they are able to get fuel moving when it is cold. A circulation gear pump running on LDO should give between 7000-10000 hours of service. Diaphragm pump save a shorter service life, but are easier and less expensive to repair. A centrifugal pump is not recommended, because as the oil viscosity increases, the efficiency of the pump drops sharply and the horse power required increases. Light fuels are best pumped with centrifugal or turbine pumps. When higher pressures are required, piston or diaphragm pumps should be used.

Storage Temperature and Pumping Temperature

The viscosity of furnace oil and LSHS increases with decrease in temperature, which makes it difficult to pump the oil. At low ambient temperatures (below 25 °C), furnace oil

is not easily pumpable. To circumvent this, preheating of oil is accomplished in two ways: a) the entire tank may be preheated. In this form of bulk heating, steam coils are placed at the bottom of the tank, which is fully insulated; b) the oil can be heated as it flows out with an outflow heater. To reduce steam requirements, it is advisable to insulate tanks where bulk heating is used.

Bulk heating may be necessary if flow rates are high enough to make outflow heaters of adequate capacity impractical, or when a fuel such as Low Sulphur Heavy Stock (LSHS) is used. In the case of out flow heating, only the oil, which leaves the tank, is heated to the pumping temperature. The out flow heater is essentially a heat exchanger with steam or electricity as the heating medium.

Temperature Control

Thermostatic temperature control of the oil is necessary to prevent overheating, especially when oil flow is reduced or stopped. This is particularly important for electric heaters, since oil may get carbonized when there is no flow and the heater is on. Thermostats should be provided at a region where the oil flows freely into the suction pipe. The temperature at which oil can readily be pumped depends on the grade of oil being handled. Oil should never be stored at a temperature above that necessary for pumping as this leads to higher energy consumption.

1.3 Properties of Coal

Coal Classification

In general there are three main types of coal: anthracite, bituminous, and lignite, but no clear-cut line exists between them and coal is further classed as semi anthracite, semi bituminous, and sub bituminous. Anthracite is the oldest form of coal, geologically speaking. It is a hard coal composed mainly of carbon with little volatile content and practically no moisture. Lignite is the youngest form of coal, composed mainly of volatile matter and moisture content with low fixed carbon content. Fixed carbon refers to carbon in its free state, not combined with other elements. Volatile matter refers to those combustible constituents of coal that vaporize when coal is heated.

The chemical composition of coal has a strong influence on its combustibility. The properties of coal are broadly classified as

- Physical properties
- Chemical properties

Physical Properties

Heating Value

The heating value of coal varies from country to country and even from mine to mine within the same country. The typical GCVs for various coals are given in the Table 14.

Table 1.4: Typical GCV for various coals

Parameter	Lignite	Indian Coal	Indonesian Coal	South African Coal	High Bangladeshi Coal (Bituminous Coal of Barapukuria)
GCV (kcal/kg)	4500 *	4,000	5,500	6,000	6072

*Dry Basis

Analysis of Coal

There are two methods: the ultimate analysis splits up the fuel in to all its component elements, solid or gaseous; and the proximate analysis determines only the fixed carbon, volatile matter, moisture and ash percentages. The ultimate analysis must be carried out in a properly equipped laboratory by a skilled chemist, but proximate analysis can be made with fairly simple apparatus.

Proximate Analysis

Proximate analysis indicates the percentage by weight of the Fixed Carbon, Volatiles, Ash, and Moisture Content in coal. The amounts of fixed carbon and volatile combustible matter directly contribute to the heating value of coal. Fixed carbon acts as a main heat generator during burning. High volatile matter content indicates easy ignition of fuel. The ash content is important in the design of the furnace grate, combustion volume, pollution control equipment and ash handling systems of a furnace. A typical proximate analysis of various coal is given in the Table1.5.

Table 1.5: Typical proximate analysis of various coal (% Weight)

Parameter	Indian Coal	Indonesian Coal	South African Coal	Bangladeshi Coal
Moisture	5.98	9.43	8.5	10.00
Ash	38.56	13.99	18	12.40
Volatile matter	20.70	29.79	23.28	29.20
Fixed Carbon	34.69	46.79	51.22	48.40

Significance of Various Parameters in Proximate Analysis

a) Fixed carbon

Fixed carbon is the solid fuel left in the furnace after volatile matter is distilled off. It consists mostly of carbon but also contains some hydrogen, oxygen, sulphur and nitrogen not driven off with the gases. Fixed carbon gives a rough estimate of heating value of coal.

b) Volatile Matter

Volatile matters are the methane, hydrocarbons, hydrogen and carbon monoxide, and incombustible gases like carbon dioxide and nitrogen found in coal. Thus the volatile

matter is an index of the gaseous fuels present. Typical range of volatile matter is 20 to 35%.

Volatile Matter

- Proportionately increases flame length, and helps in easier ignition of coal.
- Sets minimum limit on the furnace height and volume.
- Influences secondary air requirement and distribution aspects.

c) Ash Content

Ash is an impurity that will not burn. Typical range is 0.5 to 40%.

- Ash reduces handling and burning capacity.
- Increases handling costs
- Affects combustion and boiler efficiency
- Causes clinkering and slagging

d) Moisture Content

Moisture in coal must be transported, handled and stored. Since it replaces combustible matter, it decreases the heat content per kg of coal. Typical range is 0.5 to 10%.

Moisture

- Increases heat loss, due to evaporation and superheating of vapour
- Helps, to a limit, in binding fines.
- Aids radiation heat transfer.

e) Sulphur Content

Typical range is 0.5 to 5%.

Sulphur

- Affects clinkering and slagging tendencies
- Corrodes chimney and other equipment such as air heaters and economisers
- Limits exit flue gas temperature.

Chemical Properties

Ultimate Analysis:

The ultimate analysis indicates the various elemental chemical constituents such as Carbon, Hydrogen, Oxygen, Sulphur, etc. It is useful in determining the quantity of air required for combustion and the volume and composition of the combustion gases. This information is required for the calculation of flame temperature and the flue duct design etc. Typical ultimate analyses of various coals are given in the Table 1.6.

Table 1.6: Ultimate analyses of various coals

Parameter	Lignite,%	Indian Coal,%	Indonesian Coal,%	Bangladeshi Coal,%
Moisture (Dry)	62.01	5.98	9.43	10
Mineral Matter	10.41	38.63	13.99	20.49
Carbon	50	41.11	58.96	61.52
Hydrogen	6.66	2.76	4.16	3.87
Nitrogen	0.60	1.22	1.02	1.52
Sulphur	0.59	0.41	0.56	0.53
Oxygen	19.73	9.89	11.88	12.07

Relationship Between Ultimate Analysis and Proximate Analysis

$$\begin{aligned} \%C &= 0.97C + 0.7(\text{VM} - 0.1A) - M(0.6 - 0.01M) \\ \%H &= 0.036C + 0.086(\text{VM} - 0.1A) - 0.0035M^2(1 - 0.02M) \\ \%N_2 &= 2.10 - 0.020 \text{VM} \end{aligned}$$

Where

$$\begin{aligned} C &= \% \text{ of fixed carbon} \\ A &= \% \text{ of ash} \\ \text{VM} &= \% \text{ of volatile matter} \\ M &= \% \text{ of moisture} \end{aligned}$$

Storage, Handling and Preparation of Coal

Uncertainty in the availability and transportation of fuel necessitates storage and subsequent handling. Stocking of coal has its own disadvantages like build-up of inventory, space constraints, deterioration in quality and potential fire hazards. Other minor losses associated with the storage of coal include oxidation, wind and carpet loss. A 1% oxidation of coal has the same effect as 1% ash in coal, wind losses may account for nearly 0.5-1.0% of the total loss.

The main goal of good coal storage is to minimize carpet loss and the loss due to spontaneous combustion. Formation of a soft carpet, comprising of coal dust and soil causes carpet loss. On the other hand, gradual temperature builds up in a coal heap, on account oxidation may lead to spontaneous combustion of coal in storage.

The measures that would help in reducing the carpet losses are as follows:

1. Preparing a hard ground for coal to be stacked upon.
2. Preparing standard storage bays out of concrete and brick

In process Industry, modes of coal handling range from manual to conveyor systems. It would be advisable to minimize the handling of coal so that further generation of fines and segregation effects are reduced.

Preparation of Coal

Preparation of coal prior to feeding in to the boiler is an important step for achieving good combustion. Large and irregular lumps of coal may cause the following problems:

1. Poor combustion conditions and inadequate furnace temperature.

2. Higher excess air resulting in higher stack loss.
3. Increase of unburnt in the ash.
4. Low thermal efficiency.

Sizing of Coal

Proper coal sizing is one of the key measures to ensure efficient combustion. Proper coal sizing, with specific relevance to the type of firing system, helps towards even burning, reduced ash losses and better combustion efficiency.

Coal is reduced in size by crushing and pulverizing. Pre-crushed coal can be economical for smaller units, especially those which are stoker fired. In a coal handling system, crushing is limited to a top size of 6 or 4 mm. The devices most commonly used for crushing are the rotary breaker, the roll crusher and the hammer mill.

It is necessary to screen the coal before crushing, so that only oversized coal is fed to the crusher. This helps to reduce power consumption in the crusher. Recommended practices in coal crushing are:

1. Incorporation of a screen to separate fines and small particles to avoid extra fine generation in crushing.
2. Incorporation of a magnetic separator to separate iron pieces in coal, which may damage the crusher.

The Table 1.7 gives the proper size of coal for various types of firing systems

Table 1.7: Proper size of coal for various types of firing systems

S. No.	Types of Firing System	Size (in mm)
1	Hand Firing (a) Natural draft (b) Forced draft	25-75 25-40
2	Stoker Firing (a) Chain grate i) Natural draft ii) Forced draft (b) Spreader Stoker	25-40 15-25 15-25
3	Pulverized Fuel Fired	75% below 75 micron*
4	Fluidized bed boiler	< 10 mm

*Micron = 1/1000mm

a) Conditioning of Coal

The fines present in coal create problems in combustion due to segregation effects. Segregation of fines from larger coal pieces can be reduced to a great extent by conditioning coal with water. Water helps fine particles to stick to the bigger lumps due to surface tension of the moisture, thus stopping fines from falling through grate bars or being carried away by the furnace draft. While tempering the coal, care should be taken to ensure that moisture addition is uniform and preferably done in a moving or falling stream of coal.

If the percentage of fines in the coal is very high, wetting of coal can decrease the percentage of unburnt carbon and the excess air level required to be supplied for combustion. Table 1.8 shows the extent of wetting, depending on the percentage of fines in coal.

Table 1.8: Extent of wetting, depending on the percentage of fines in coal

Fines (%)	Surface Moisture (%)
10 - 15	4 - 5
15 - 20	5 - 6
20 - 25	6 - 7
25 - 30	7 - 8

b) Blending of Coal

In case of coal lots having excessive fines, it is advisable to blend the predominantly lumped coal with lots containing excessive fines. Coal blending may thus help to limit the extent of fines in coal being fired to not more than 25%. Blending of different qualities of coal may also help to supply uniform coal feed to the boiler.

The proximate and ultimate analyses of various coals are given in Table 1.9 and 1.10.

Table 1.9: Proximate analysis of various coals

	Lecfines	Lignite	Indian Coal (type 1)	Indian Coal (type 2)	Indonesia n Coal	South African Coal	Bangladeshi Coal
Moisture	9.92	49.79	5.98	4.39	9.43	8.5	10.00
Ash (%)	5.93	10.41	38.65	47.86	13.99	18	12.40
Volatile matter (%)	24.08	47.76	20.70	17.97	29.79	23.28	29.20
Fixed Carbon (%)	60.70	41.83	34.69	29.78	46.79	51.22	48.40

Table 1.10: Ultimate analysis of various coals

	Lecofines	Lignite	Indian Coal (type 2)	Indian Coal (type)	Indonesian Coal	Bangladeshi Coal
Moisture (%)	9.92	Dry basis	5.98	4.39	9.43	10
Mineral matter	5.93	10.41	38.63	47.86	13.99	20.49
Carbon (%)	71.02	62.01	42.11	36.22	58.96	61.52
Hydrogen (%)	3.33	6.66	2.76	2.64	4.16	3.87
Nitrogen (%)	0.96	0.60	1.22	1.09	1.02	1.52
Sulphur (%)	0.55	0.59	0.41	0.55	0.56	0.53
Oxygen (%)	8.29	19.73	9.89	7.25	11.88	12.07
GCV(kCal/kg)	7242	6301	4000	3500	5500	6072

1.4 Properties of Gaseous Fuels

Gaseous fuels in common use are liquefied petroleum gases (LPG), Natural gas, producer gas, blast furnace gas, coke oven gas etc. The calorific value of gaseous fuel is expressed in Kilocalories per cubic meter (kcal/Nm^3) i.e. at normal temperature and pressure.

Calorific Value

Since most gas combustion appliances cannot utilize the heat content of the water vapour, gross calorific value is of little interest. Fuel should be compared based on the net calorific value. This is especially true for natural gas, since increased hydrogen content results in high water formation during combustion.

Typical physical and chemical properties of various gaseous fuels are given in Table 1.11.

Table 1.11: Typical physical and chemical properties of various gaseous fuels

Fuel Gas	Relative Density	Higher Heating Value, kCal/Nm^3	Air/Fuel ratio, m^3 of air to m^3 of Fuel	Flame Temp., $^{\circ}\text{C}$	Flame Speed, m/s
Natural Gas	0.6	9350	10	1954	0.290
Propane	1.52	22200	25	1967	0.460
Butane	1.96	28500	32	1973	0.870

LPG

LPG is a predominant mixture of propane and Butane with a small percentage of unsaturated (Propylene and Butylene) and some lighter C_2 as well as heavier C_5 fractions. Included in the LPG range are propane (C_3H_8), Propylene (C_3H_6), normal and iso-butane (C_4H_{10}) and Butylene (C_4H_8).

LPG may be defined as those hydrocarbons which are gaseous at normal atmospheric pressure, but may be condensed to the liquid state at normal temperature, by the application of moderate pressures. Although they are normally used as gases, they are stored and transported as liquids under pressure for convenience and ease of handling. Liquid LPG evaporates to produce about 250 times volume of gas.

LPG vapour is denser than air: butane is about twice as heavy as air and propane about one and a half times as heavy as air. Consequently, the vapour may flow along the ground and into drains sinking to the lowest level of the surroundings and be ignited at a considerable distance from the source of leakage. In still air vapour will disperse slowly. Escape of even small quantities of the liquefied gas can give rise to large volumes of vapour / air mixture and thus cause considerable hazard. To aid in the detection of atmospheric leaks, all LPG's are required to be odorized. There should be adequate ground level ventilation where LPG is stored. For this very reason, LPG cylinders should not be stored in cellars or basements, which have no ventilation at ground level.

Natural Gas

Methane is the main constituent of Natural gas and accounting for about 97% of the total volume. Other components are: Ethane, Propane, Butane, Pentane, Nitrogen, Carbon

Dioxide, and traces of other gases. Very small amounts of sulphur compounds are also present. Since methane is the largest component of natural gas, generally properties of methane are used when comparing the properties of natural gas to other fuels.

Natural gas is a high calorific value fuel requiring no storage facilities. It mixes with air readily and does not produce smoke or soot. It has no sulphur content. It is lighter than air and disperses into air easily in case of leak. A typical comparison of carbon contents in oil, coal and gas is given in the table 1.12.

Table 1.12: Comparison of carbon contents in oil, coal and gas

	Fuel Oil	Coal	Natural Gas
Carbon	84	41.11	74
Hydrogen	12	2.76	25
Sulphur	3	0.41	-
Oxygen	1	9.89	Trace
Nitrogen	Trace	1.22	0.75
Ash	Trace	38.63	-
Water	Trace	5.98	-

1.5 Properties of Agro Residues

The use of locally available agro residues is on the rise. This includes rice husk, coconut shells, groundnut shells, Coffee husk, Wheat stalk etc. The properties of a few of them are given in the table 1.13.

Table 1.13: Properties of Agro Residues

	De oiled Bran	Paddy Husk	Saw Dust	Coconut Shell
Moisture	7.11	10.79	37.98	13.95
Ash	18.46	16.73	1.63	3.52
Volatile Matter	59.81	56.46	81.22	61.91
Fixed Carbon	14.62	16.02	17.15	20.62
Mineral Matter	19.77	16.73	1.63	3.52
Carbon	36.59	33.95	48.55	44.95
Hydrogen	4.15	5.01	6.99	4.99
Nitrogen	0.82	0.91	0.80	0.56
Sulphur	0.54	0.09	0.10	0.08
Oxygen	31.02	32.52	41.93	31.94
GCV(kCal/kg)	3151	3568	4801	4565

Table 1.14: Proximate Analysis of Typical Agro Residues

	Deoiled Bran	Paddy Husk	Saw Dust	Coconut Shell
Moisture	7.11	10.79	37.98	13.95
Ash	18.46	16.73	1.63	3.52
Volatile Matter	59.81	56.46	81.22	61.91
Fixed Carbon	14.62	16.02	17.15	20.62

Table: 1.15 Ultimate Analysis of Typical Agro Residues

	Deoiled Bran	Paddy Husk	Saw Dust	Coconut Shell
Moisture	7.11	10.79	37.98	13.95
Mineral Matter	19.77	16.73	1.63	3.52
Carbon	36.59	33.95	48.55	44.95
Hydrogen	4.15	5.01	6.99	4.99
Nitrogen	0.82	0.91	0.80	0.56
Sulphur	0.54	0.09	0.10	0.08
Oxygen	31.02	32.52	41.93	31.94
GCV (kcal/kg)	3151	3568	4801	4565

Biomass Storage, Handling and Preparation

Biomass fuels have low bulk density which results in higher transportation cost. The transportation cost constitutes a significant portion of the landed cost of biomass. The low bulk density also requires vast area for storage. A common concern in biomass systems is the difficulty to ensure availability of any particular type of biomass throughout the entire year. A variety of types of biomass necessitate different types of collection and handling equipment. Most of the common biomass fuels, such as Woody biomass, Juliflora, Palm bunches, Jute Sticks, Cotton Stalks, De-oiled bran, Coir pith etc., require special types of handling machines, which add up to additional capital investment.

1.6 Combustion

Principle of Combustion

Combustion refers to the rapid oxidation of fuel accompanied by the production of heat, or heat and light. Complete combustion of a fuel is possible only in the presence of an adequate supply of oxygen.

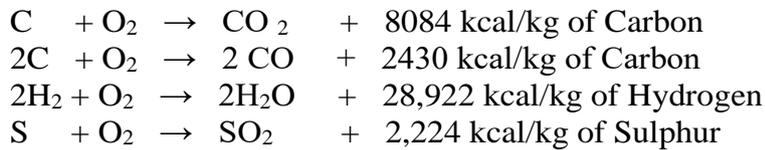
Oxygen (O₂) is one of the most common elements on earth making up 20.9% (by volume) of our air. Rapid fuel oxidation results in large amounts of heat. Solid or liquid fuels must be changed to a gas before they will burn. Usually heat is required to change liquids or solids into gases. Fuel gases will burn in their normal state if enough air is present.

Most of the 79% of air (that is not oxygen) is nitrogen, with traces of other elements. Nitrogen is considered to be a temperature reducing dilute ant that must be present to obtain the oxygen required for combustion.

Nitrogen reduces combustion efficiency by absorbing heat from the combustion of fuels and diluting the flue gases. This reduces the heat available for transfer through the heat exchange surfaces. It also increases the volume of combustion by-products, which then have to travel through the heat exchanger and up the stack faster to allow the introduction of additional fuel air mixture.

This nitrogen also can combine with oxygen (particularly at high flame temperatures) to produce oxides of nitrogen (NO_x), which are toxic pollutants.

Carbon, hydrogen and sulphur in the fuel combine with oxygen in the air to form carbon di oxide, water vapour and sulphur dioxide, releasing 8084, 28922 and 2224 kcal of heat respectively. Under certain conditions, Carbon may also combine with Oxygen to form Carbon Monoxide, which results in the release of a smaller quantity of heat (2430 kcal/ kg of carbon) Carbon burned to CO₂ will produce more heat per pound of fuel than when CO or smoke are produced.



3 T's of Combustion

The objective of good combustion is to release all of the heat in the fuel. This is accomplished by controlling the "three T's" of combustion which are (1) Temperature high enough to ignite and maintain ignition of the fuel, (2) Turbulence or intimate mixing of the fuel and oxygen, and (3) Time sufficient for complete combustion.

Commonly used fuels like natural gas and propane generally consist of carbon and hydrogen. Water vapor is a by-product of burning hydrogen. To maintain its vaporous state, it robs heat from the flue gases, which would otherwise be available for more heat transfer.

Natural gas contains more hydrogen and less carbon per Kg than fuel oils and as such produces more water vapor. Consequently, more heat will be carried away by exhaust while firing natural gas. Too much, or too little fuel with the available combustion air may potentially result in unburned fuel and carbon monoxide generation. Every specific amount of O₂ is needed for perfect combustion and some additional (excess) air is required for ensuring good combustion. However, too much excess air will reduce in heat and efficiency losses.

Not all of the heat in the fuel are converted to heat and absorbed by the steam generation equipment. Usually all of the hydrogen in the fuel is burned and most boiler fuels, allowable with today's air pollution standards, contain little or no sulphur. So the main challenge in combustion efficiency is directed toward unburned carbon (in the ash or in completely burned gas), which forms CO instead of CO₂.

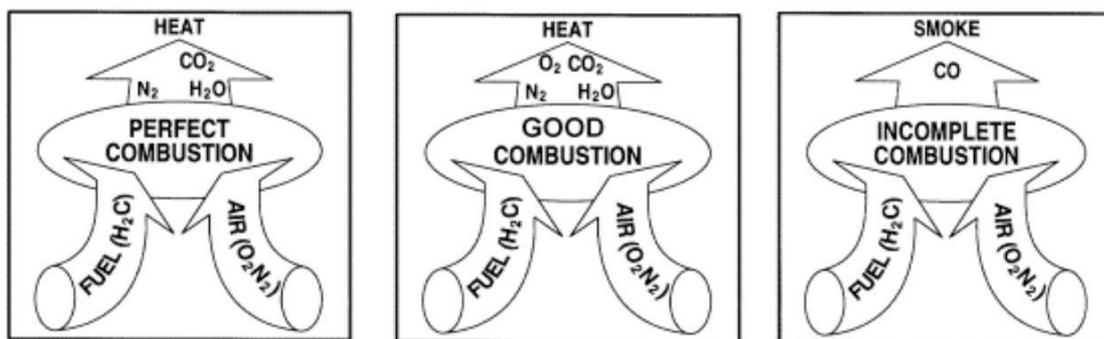


Figure 1.2: Degree of combustion

1.7 Combustion of Oil

Heating Oil to Correct Viscosity

When atomizing oil, it is necessary to heat it enough to get the desired viscosity. This temperature varies slightly for each grade of oil. The lighter oils do not usually require pre-heating. Typical viscosity at the burner tip (for LAP, MAP & HAP burners) for furnace oil should be 100 Redwood seconds⁻¹ which would require heating the oil to about 105⁰C.

Stoichiometric Combustion

The efficiency of a boiler or furnace depends on efficiency of the combustion system. The amount of air required for complete combustion of the fuel depends on the elemental constituents of the fuel that is Carbon, Hydrogen, and Sulphur etc. This amount of air is called **stanchion metric** air. For ideal combustion process for burning one kg of a typical fuel oil containing 86% Carbon, 12% Hydrogen, 2% Sulphur, theoretically required quantity of air is 14.1kg. This is the minimum air that would be required if mixing of fuel and air by the burner and combustion is perfect. The combustion products are primarily Carbon Dioxide (CO₂), water vapour (H₂O) and Sulphur Dioxide (SO₂), which pass through the chimney along with the Nitrogen (N₂) in the air.

Rules for combustion of oil

- Atomise the oil completely to produce a fine spray
- Mix the air and fuel thoroughly
- Introduce enough air for combustion, but limit the excess air to a maximum of 15%
- Keep the burners in good condition

After surrendering useful heat in the heat absorption area of a furnace or boiler, the combustion products or fuel gases leave the system through the chimney, carrying a way a significant quantity of heat with them.

Calculation of Stoichiometric Air

The specification of furnace oil from lab analysis is given below:

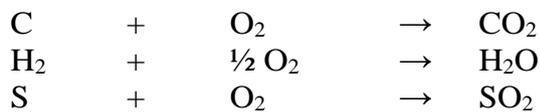
Constituents	% By weight
Carbon	85.9
Hydrogen	12
Oxygen	0.7
Nitrogen	0.5
Sulphur	0.5
H ₂ O	0.35
Ash	0.05

GCV of fuel: 10880 kCal/kg

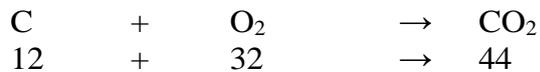
Calculation for Requirement of Theoretical Amount of Air

Considering a sample of 100kg of furnace oil. The chemical reactions are:

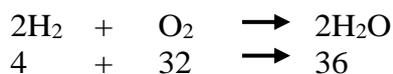
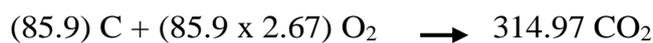
Element	Molecular Weight kg / kg mole
C	12
O ₂	32
H ₂	2
S	32
N ₂	28
CO ₂	44
SO ₂	64
H ₂ O	18



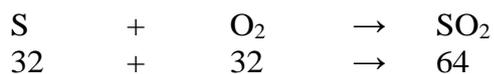
Constituents of fuel



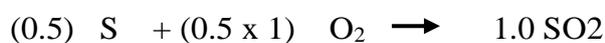
12 kg of carbon requires 32 kg of oxygen to form 44 kg of carbon dioxide therefore 1 kg of carbon requires 32/12 kg i.e 2.67 kg of oxygen



4 kg of hydrogen requires 32 kg of oxygen to form 36 kg of water, therefore 1 kg of hydrogen requires 32/4 kg i.e 8 kg of oxygen



32 kg of sulphur requires 32 kg of oxygen to form 64 kg of sulphur dioxide, therefore 1 kg of sulphur requires 32/32 kg i.e 1 kg of oxygen



$$\begin{aligned}
\text{Total Oxygen required} &= 325.57 \text{ kg} \\
(229.07+96+0.5) & \\
\text{Oxygen already present in} & \\
100 \text{ kg fuel (given)} &= 0.7 \text{ kg} \\
\text{Additional Oxygen Required} &= 325.57 - 0.7 \\
&= 324.87 \text{ kg} \\
\text{Therefore quantity of dry air reqd.} &= (324.87) / 0.23 \\
(\text{air contains 23\% oxygen by wt.}) & \\
&= 1412.45 \text{ kg of air} \\
\text{Theoretical Air required} &= (1412.45) / 100 \\
&= 14.12 \text{ kg of air / kg of fuel}
\end{aligned}$$

Calculation of theoretical CO₂ content in flue gases

$$\begin{aligned}
\text{Nitrogen in flue gas} &= 1412.45 - 324.87 + 0.5 \\
&= 1088.08 \text{ kg}
\end{aligned}$$

Theoretical CO₂% in dry flue gas by volume is calculated as below:

$$\begin{aligned}
\text{Moles of CO}_2 \text{ in flue gas} &= (314.97) / 44 = 7.16 \\
\text{Moles of N}_2 \text{ in flue gas} &= (1088.08) / 28 = 38.86 \\
\text{Moles of SO}_2 \text{ in flue gas} &= 1/64 = 0.016
\end{aligned}$$

$$\begin{aligned}
\text{Theoretical CO}_2\% \text{ by volume} &= \frac{\text{Moles of CO}_2}{\text{Total moles (dry)}} \times 100 \\
&= \frac{7.16}{7.16+38.86+0.016} \times 100 \\
&= 15.6 \%
\end{aligned}$$

Calculation of constituents of flue gas with excess air

$$\% \text{ CO}_2 \text{ measured in flue gas} = 10\% \text{ (measured)}$$

$$\% \text{ Excess air supplied (EA)} = \frac{7900 \times [(CO_2\%)_t - (CO_2\%)_a]}{(CO_2)_a \% \times [100 - (CO_2\%)_t]}$$

$$\begin{aligned}
\% \text{ Excess air supplied (EA)} &= \frac{7900 \times [15.6 - 10]}{10 \times [100 - 15.6]} \\
&= 52.4\%
\end{aligned}$$

$$\begin{aligned}
\text{Theoretical air required for 100 kg} &= 1412.45 \text{ kg} \\
\text{of fuel burnt} &
\end{aligned}$$

$$\begin{aligned}
\text{Total quantity. of air supply required} &= 1412.45 \times 1.524 \\
\text{with } 52.4\% \text{ excess air} &= 2152.57 \text{ kg} \\
\text{Excess air quantity} &= 2152.57 - 1412.45 \\
&= 740.12 \text{ kg.} \\
\text{O}_2 &= 740.12 \times 0.23 \\
&= 170.23 \text{ kg} \\
\text{N}_2 &= 740.12 - 170.23 \\
&= 570 \text{ kg}
\end{aligned}$$

The final constitution of flue gas with 52.4% excess air for every 100 kg fuel.

$$\begin{aligned}
\text{CO}_2 &= 314.97 \text{ kg} \\
\text{H}_2\text{O} &= 108.00 \text{ kg} \\
\text{SO}_2 &= 1 \text{ kg} \\
\text{O}_2 &= 170.23 \text{ kg} \\
\text{N}_2 &= 1088.08 + 570 \\
&= 1658.08 \text{ kg}
\end{aligned}$$

Determination of Actual CO₂% by calculation in dry flue gas by volume

$$\begin{aligned}
\text{Moles of CO}_2 \text{ in flue gas} &= 314.97/44 = 7.16 \\
\text{Moles of SO}_2 \text{ in flue gas} &= 1/64 = 0.016 \\
\text{Moles of O}_2 \text{ in flue gas} &= 170.23 / 32 = 5.32 \\
\text{Moles of N}_2 \text{ in flue gas} &= 1658.08 / 28 = 59.22
\end{aligned}$$

$$\begin{aligned}
\text{Actual CO}_2\% \text{ by volume} &= \frac{\text{Moles of CO}_2}{\text{Total moles (dry)}} \times 100 \\
&= \frac{7.16}{7.16 + 0.016 + 5.32 + 59.22} \times 100 \\
&= \frac{7.16}{71.716} \times 100 = 10\% \\
\text{Actual O}_2\% \text{ by volume} &= \frac{5.32 \times 100}{71.716} = 7.42\%
\end{aligned}$$

Optimizing Excess Air and Combustion

For complete combustion of every one kg of fuel oil 14.1 kg of air is needed. In practice, mixing is never perfect, a certain amount of excess air is needed to complete combustion and ensure that release of the entire heat contained in fuel oil. If too much air than what is required for completing combustion were allowed to enter, additional heat would be lost

in heating the surplus air to the chimney temperature. This would result in increased stack losses. Less air would lead to the incomplete combustion and smoke. Hence, there is an optimum excess air level for each type of fuel.

Control of Air and Analysis of Flue Gas

Thus in actual practice, the amount of combustion air required will be much higher than optimally needed. Therefore some of the air gets heated in the furnace boiler and leaves through the stack without participating in the combustion.

Chemical analysis of the gases is an objective method that helps in achieving finer air control. By measuring carbon dioxide (CO_2) or oxygen (O_2) in flue gases by continuous recording instruments or Orsat apparatus or some cheaper portable instruments, the excess air level as well as stack losses can be estimated with the graph as shown in Figure 1.2 and Figure 1.3. The excess air to be supplied depends on the type of fuel and the firing system. For optimum combustion of fuel oil, the CO_2 or O_2 in flue gases should be maintained at 14-15% in case of CO_2 and 2-3% in case of O_2 measurement.

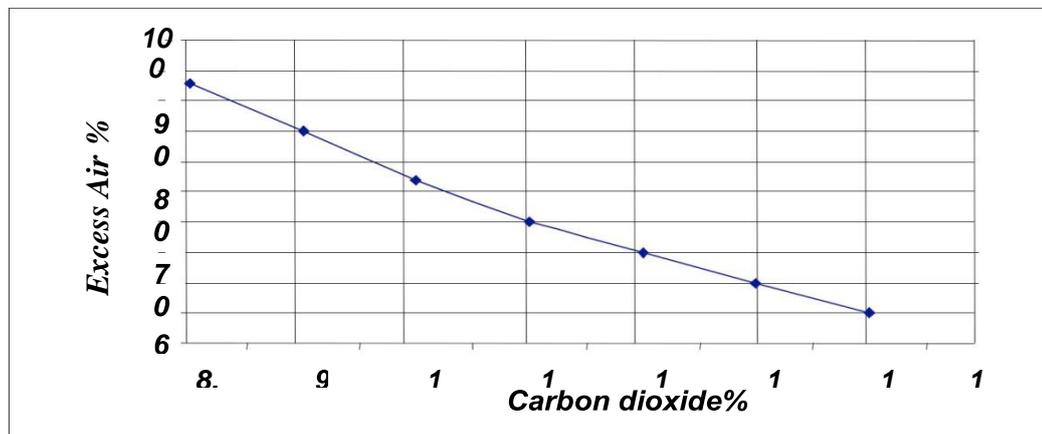


Figure 1.2: Relation between CO_2 and excess air for fuel oil

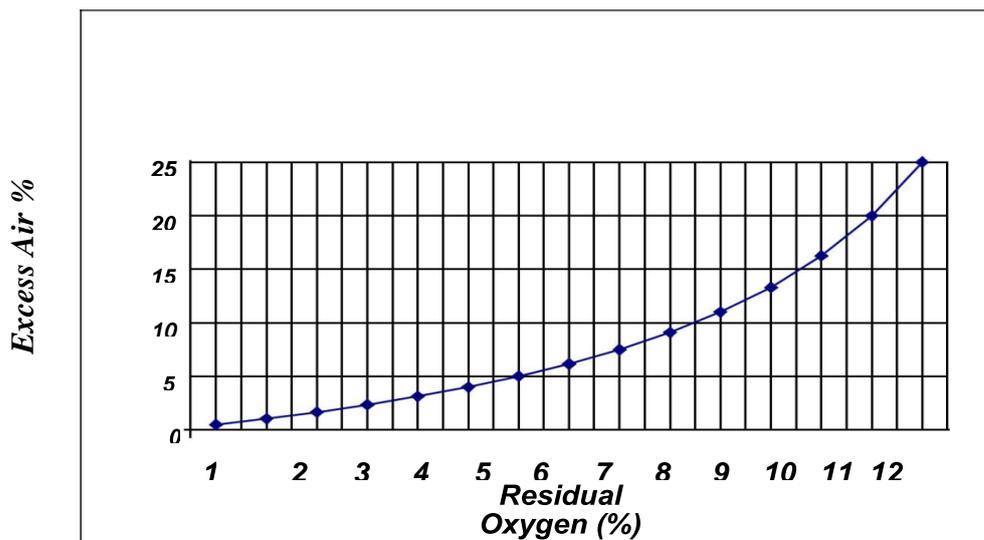


Figure 1.3: Relation between residual oxygen and excess air Oil Firing Burners

The burner is the principal device for the firing of the fuel. The primary function of burner is to atomise fuel to millions of small droplets so that the surface area of the fuel is increased enabling intimate contact with oxygen in air. The finer the fuel drop lets are atomised, more readily will the particles come in contact with the oxygen in the air and burn.

Normally, atomization is carried out by primary air and completion of combustion is ensured by secondary air. Burners for fuel oil can be classified on a basis of the technique to prepare the fuel for burning i.e. atomization.

Figure 1.4 shows a simplified burner head. The air is brought in to the head by means of a forced draft blower or fan. The gas is metered into the head through a series of valves. In order to get proper combustion, the air molecules must be thoroughly mixed with the gas molecules before they actually burn.

The mixing is achieved by burner parts designed to create high turbulence. If insufficient turbulence is produced by the burner, the combustion will be incomplete and samples taken at the stack will reveal carbon monoxide as evidence.

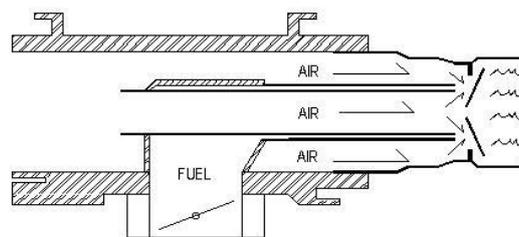


Figure 1.4: Burner head

Since the velocity of air affects the turbulence, it becomes harder and harder to get good fuel and air mixing at higher turn down ratios since the air amount is reduced. Towards the highest turndown ratios of any burner, it becomes necessary to increase the excess air amounts to obtain enough turbulence to get proper mixing. The better burner design will be one that is able to properly mix the air and fuel at the lowest possible air flow or excess air.

An important aspect to be considered in selection of burner is **turn down ratio**. Turn down ratio is the relationship between the maximum and minimum fuel input without affecting the excess air level. For example, a burner whose maximum input is 250,000 kcal and minimum rate is 50,000 kcal, has a 'Turn-Down Ratio' of 5 to 1.

The air in the centre is the primary air which is used for atomization and the one surrounding is the secondary air which ensures complete combustion.

1.8 Combustion of Coal

Features of coal combustion

1kg of coal will typically require 7-8 kg of air depending upon the carbon, hydrogen, nitrogen, oxygen and sulphur content for complete combustion. This air is also known as theoretical or stoichiometric air.

If for any reason the air supplied is inadequate, the combustion will be incomplete. The result is poor generation of heat with some portions of carbon remaining unburnt (black smoke) and forming carbon monoxide instead of carbon dioxides.

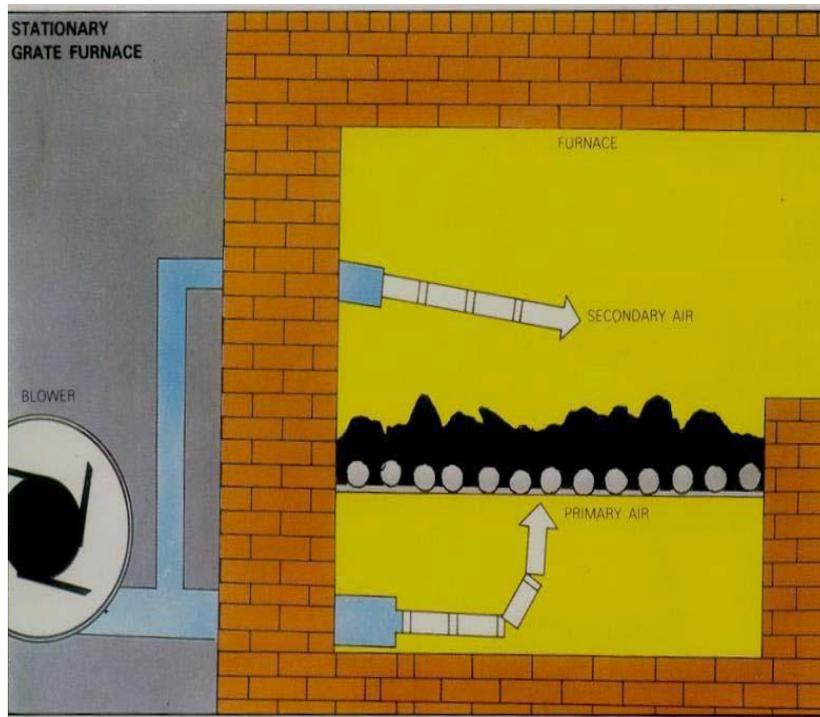


Figure 1.5: Stoker fired boilers

In actual case No fuel can be completely burnt with stoichiometric quantity of air. Complete combustion is not achieved unless an excess of air is supplied.

The excess air required for coal combustion depends on the type of coal firing equipment. Hand fired boilers use large lumps of coal and hence need very high excess air. Stoker fired boilers as shown in the figure 5 use sized coal and hence requires less excess air. Also in these systems primary air is supplied below the grate and secondary air is supplied over the grate to ensure complete combustion.

Fluidized bed combustion in which turbulence is created leads to intimate mixing of air and fuel resulting in further reduction of excess air. The pulverized fuel firing in which powdered coal is fired has the minimum excess air due to high surface area of coal ensuring complete combustion.

Clinker formation

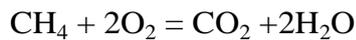
Clinker is a mass of rough, hard, slag-like material formed during combustion of coal due to low fusion temperature of ash present in coal. Presence of silica, calcium oxide, magnesium oxides etc. in ash lead to a low fusion temperature. Typically Bangladeshi coals contain ash fusion temperature as low as 1400°C. Once clinker is formed, it has a tendency to grow. Clinker will stick to a hot surface rather than a cold one and to a rough surface rather than a smooth one.

1.9 Combustion of Gas

Combustion Characteristics of Natural Gas

The stoichiometric ratio for natural gas (and most gaseous fuels) is normally indicated by volume. The air to natural gas (stoichiometric) ratio by volume for complete combustion vary between 9.5:1 to 10:1

Natural gas is essentially pure methane, CH₄. Its combustion can be represented as follows:



So for every 16 Kgs of methane that are consumed, 12 Kgs of carbon are released (as carbon dioxide) and 44 Kgs of carbon dioxide are produced. (Remember that the atomic weights of carbon, oxygen and hydrogen are 12, 16 and 1, respectively.)

Natural gas is primarily composed of methane, CH₄. When mixed with the proper amount of air and heated to the combustion temperature, it burns. **Figure 1.6** shows the process with the amount of air and fuel required for perfect combustion.

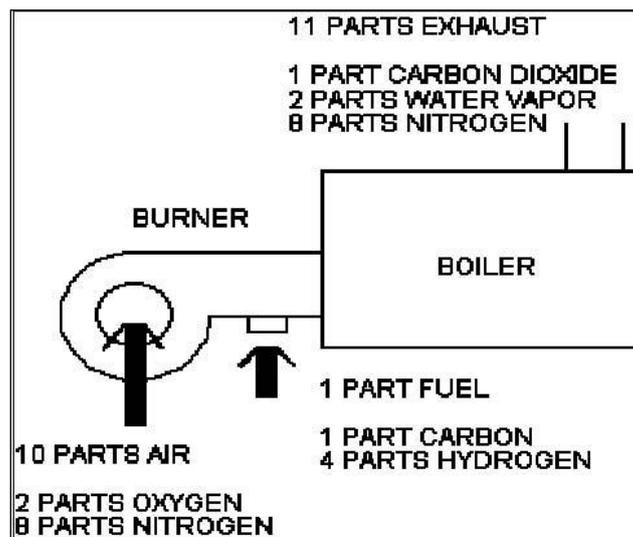


Figure 1.6: Combustion of natural gas

Low-And High-Pressure Gas Burners

The important thing in all gas-burning devices is a correct air-and-gas mixture at the burner tip. Low-pressure burners (figure 1.7), using gas at a pressure less than 0.15 kg/cm² (2 psi), are usually of the multi jet type, in which gas from a manifold is supplied to a number of small single jets, or circular rows of small jets, cantered in or discharging around the inner circumference of circular air openings in a block of some heat-resisting material. The whole is encased in a rectangular cast-iron box, built into the boiler setting and having louver doors front to regulate the air supply. Draft may be natural, induced, or forced.



Figure 1.7: Low pressure burner

In a high-pressure gas mixer (figure 1.8), the energy of the gas jet draws air in to the mixing chamber and delivers a correctly proportioned mixture to the burner. When the regulating valve is opened, gas flows through a small nozzle into a venturine tube (a tube with a contracted section). Entrainment of air with high-velocity gas in the narrow venture section draws air in through large openings in the end. The gas-air mixture is piped to a burner. The gas-burner tip may be in a variety of forms. In a sealed-in tip type, the proper gas-air mixture is piped to the burner, and no additional air is drawn in around the burner tip. Size of the air openings in the ventured been decreased by turning a revolving Sutter, which can be locked in any desired position. Excess air level in natural gas burner is in the order of 5%.

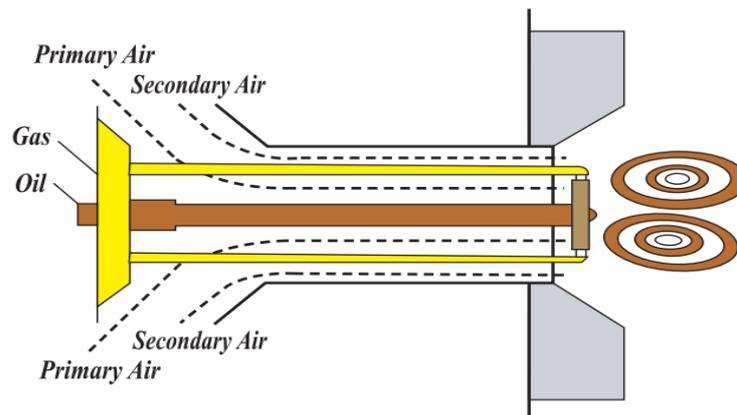


Figure 1.8: High pressure gas mixer

1.10 Combustion of Biomass

Biomass can be converted into energy (heat or electricity) or energy carriers (charcoal, oil, or gas) using both thermochemical and biochemical conversion technologies. Combustion is the most developed and most frequently applied process used for solid biomass fuels because of its low costs and high reliability. During combustion, the biomass first loses its moisture at temperatures up to 100°C, using heat from other particles that release their heat value. As the dried particle heats up, volatile gases containing hydrocarbons, CO, CH₄ and other gaseous components are released. In a combustion process, these gases contribute about 70% of the heating value of the biomass. Finally, char oxidises and ash remains. The combustion installation needs to be properly designed for a specific fuel type in order to guarantee adequate combustion quality and low emissions.

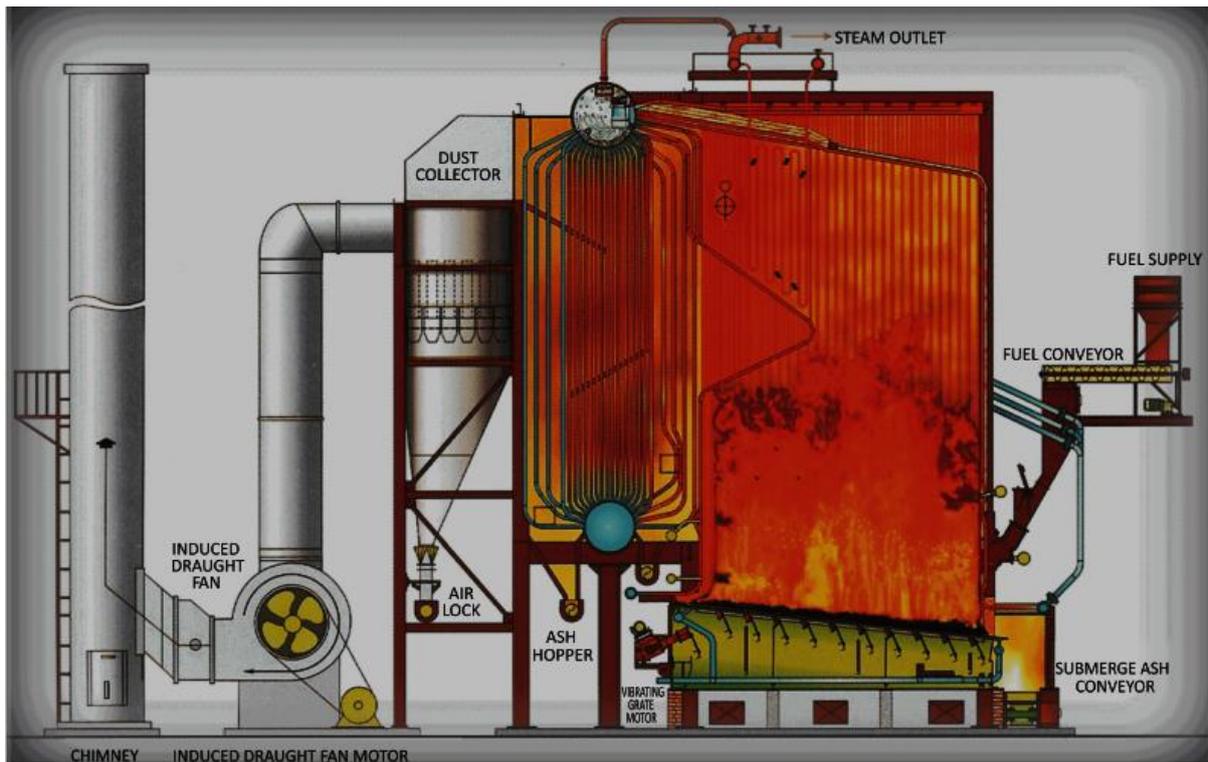


Figure 1.9: Combustion of Biomass in a Boiler

The characteristics and quality of biomass as a fuel depend on the kind of biomass and the pre-treatment technologies applied. For example, the moisture content of the fuel as fed into the furnace may vary from 25 - 55% (on a wet weight basis) for bark and sawmill by-products, and be less than 10% (on a wet weight basis) for pellets. Also, the ash sintering temperatures of biofuels used cover a wide range (800 to 1200°C), as do particle shapes and sizes.

In order to reduce its moisture content, freshly harvested wood is often left outside for a number of weeks before it is chipped and fed to a combustion plant.

Different biomass combustion systems are available for industrial purposes. Broadly, they can be defined as fixed-bed combustion, fluidised bed combustion, and dust combustion.

Fixed-bed combustion

Fixed-bed combustion systems include grate furnaces and underfeed stokers. Primary air passes through a fixed bed, where drying, gasification, and charcoal combustion take place in consecutive stages. The combustible gases are burned in a separate combustion zone using secondary air. Grate furnaces are appropriate for burning biomass fuels with high moisture content, different particle sizes, and high ash content. The design and control of the grate are aimed at guaranteeing smooth transportation and even distribution of the fuel and a homogeneous primary air supply over the whole grate surface. Irregular air supply may cause slagging, and higher amounts of fly ash, and may increase the oxygen needed for complete combustion. Load changes can be achieved more easily and quickly than in grate furnaces because there is better control of the fuel supply.

Fluidised bed combustion

In a fluidised bed, biomass fuel is burned in a self-mixing suspension of gas and solid bed material (usually silica sand and dolomite) in which air for combustion enters from below. Depending on the fluidisation velocity, bubbling and circulating fluidised bed combustion can be distinguished.

The intense heat transfer and mixing provide good conditions for complete combustion with low excess air demand. The low excess air amounts required reduce the flue gas volume flow and increase combustion efficiency. Fluid bed combustion plants are of special interest for large scale applications (normally exceeding 30 MWth). For smaller plants, fixed bed systems are usually more cost-effective. One disadvantage is the high dust loads in the flue gas, which make efficient dust precipitators and boiler cleaning systems necessary. Bed material is also lost with the ash, making it necessary to periodically add new bed material.

Dust combustion

Dust combustion is suitable for fuels available as small, dry particles such as wood dust. A mixture of fuel and primary combustion air is injected into the combustion chamber. Combustion takes place while the fuel is in suspension; the transportation air is used as primary air. Gas burnout is achieved after secondary air addition. An auxiliary burner is used to start the furnace. When the combustion temperature reaches a certain value, biomass injection starts and the auxiliary burner is shut down. Due to the explosion-like gasification process of the biomass particles, careful fuel feeding is essential. Fuel gasification and charcoal combustion take place at the same time because of the small particle size. Therefore, quick load changes and efficient load control can be achieved. Since the fuel and air are well-mixed, only a small amount of excess air is required. This results in high combustion efficiencies.

1.11 Draft

The function of draft is to exhaust the products of combustion into the atmosphere. The draft can be created by natural or artificial means.

Natural Draft

It is the draft produced by a chimney alone. It is caused by the difference in weight between the column of hot gas inside the chimney and column of outside air of the same height and cross section. Being much lighter than outside air, chimney flue gas tends to rise, and the heavier outside airflows in through the ash pit to take its place. It is usually controlled by hand-operated dampers in the chimney and breeching connecting the boiler to the chimney. Here no fans or blowers are used. The products of combustion are discharged at such a height that it will not be a nuisance to the surrounding community.

Mechanical Draft

It is the draft artificially produced by fans. Three basic types of drafts that are applied are:

Balanced Draft: Forced-draft (F-D) fan (blower) pushes air into the furnace and an induced-draft (I-D) fan draws gases into the chimney there by providing draft to remove the gases from the boiler. Here the furnace is maintained at from 0.05 to 0.10 in. of water gauge below atmospheric pressure.

Induced Draft: An induced-draft fan provides enough draft for flow in to the furnace, causing the products of combustion to discharge to atmosphere. Here the furnace is kept at as light negative pressure below the atmospheric pressure so that combustion air flows through the system.

Forced Draft: The Forced draft system uses a fan to deliver the air to the furnace, forcing combustion products to flow through the unit and up the stack.

1.12 Combustion Controls

Combustion controls assist the burner in regulation of fuel supply, air supply, (fuel to air ratio), and removal of gases of combustion to achieve optimum boiler efficiency. The amount of fuel supplied to the burner must be in proportion to the steam pressure and the quantity of steam required. The combustion controls are also necessary as safety device to ensure that the boiler operates safely.

Various types of combustion controls in use are:

On/Off Control

The simplest control, ON/OFF control means that either the burner is firing at full rate or it is OFF. This type of control is limited to small boilers.

High/Low/Off Control

Slightly more complex is HIGH / LOW / OFF system where the burner has two firing rates. The burner operates at slower firing rate and then switches to full firing as needed. Burner can also revert to low firing position at reduced load. This control is fitted to medium sized boilers.

Modulating Control

The modulating control operates on the principle of matching the steam pressure demand by altering the firing rate over the entire operating range of the boiler. Modulating motors use conventional mechanical linkage or electric valves to regulate the primary air, secondary air, and fuel supplied to the burner. Full modulation means that boiler keeps firing, and fuel and air are carefully matched over the whole firing range to maximize thermal efficiency.

Example 1.1

For combustion of 500 kg/hr of natural gas containing 100% methane, calculate the percentage of CO₂ in the flue gas while 15% excess air is supplied.

Solution:



1 mole of Methane requires 2 moles of Oxygen.

16 Kg of Methane requires 64 Kg of Oxygen.

16 Kg of Methane produces 44 Kg of CO₂.

500 Kg/hr of Methane requires 2000 Kg/hr of Oxygen.

500 Kg/hr of Methane produce 1375 Kg/hr of CO₂.

$$\begin{aligned} \text{Theoretical air required for combustion} &= 2000 / 0.23 \\ &= 8696 \text{ Kg/hr} \end{aligned}$$

Considering 15% excess air,

$$\begin{aligned} \text{Actual air supplied for combustion is} &= 8696 * 1.15 \\ &= 10,000.4 \text{ Kg/hr of air} \end{aligned}$$

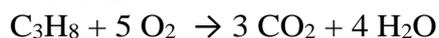
$$\begin{aligned} \text{Flue gas generation with 15\% excess air} &= 500 + 10,000.4 \\ &= 10,500.4 \text{ Kg/hr} \end{aligned}$$

$$\begin{aligned} \text{Percentage CO}_2 \text{ in the flue gas} &= (1375 / 10,500.4) \times 100 \\ &= 13.1\%. \end{aligned}$$

Example 1.2

Propane is used as fuel in heaters for preheating paints. Calculate the Air to Fuel ratio for complete combustion of C₃H₈ (Propane) and CO₂ released per kg of propane, if 15% excess air is supplied to the heater.

Solution:



1 mole of propane requires 5 moles of Oxygen.

Molecular weight of Propane is 44 kg per mole.

44 kg of Propane requires 160 kg of Oxygen.

Theoretical air required for combustion = $160 / 0.23 = 695.6 \text{ Kg/hr}$
Excess air supplied is 15 %,

$$\begin{aligned} \text{Actual air supplied for combustion is} &= 695.6 * 1.15 \\ &= 800 \text{ kg/hr of air} \end{aligned}$$

$$\begin{aligned} \text{Air to Fuel ratio} &= 800 / 44 \\ &= 18.18 \end{aligned}$$

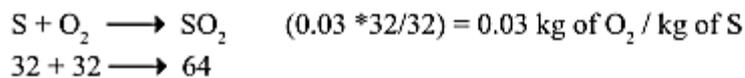
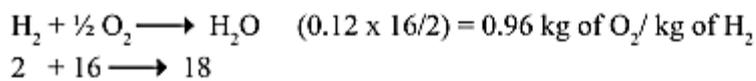
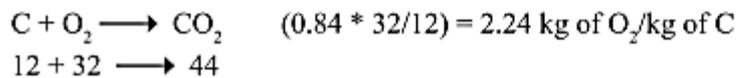
From the above chemical equation,

44 kg of Propane releases 132 kg of CO₂
 3 kg of CO₂ is released from 1 kg of propane.

Example 1.3

For combustion of 500 lit/hr of furnace oil, estimate combustion air quantity per hour with 20% excess air. Specific gravity of furnace oil 0.95. (Fuel analysis: C - 84%, H₂ - 12%, S - 3%, O₂ - 1%).

Ans:



$$\begin{aligned} \text{Total O}_2 \text{ required} &= (2.24 + 0.96 + 0.03 - 0.01) \\ &= 3.22 \text{ kg of O}_2/\text{kg furnace oil} \end{aligned}$$

$$\text{Theoretical Air required} = 3.22 / 0.23 = 14 \text{ kg of air / kg furnace oil}$$

Excess air = 20%

$$\text{Air supplied} = (1 + 0.2) * 14 = 16.8 \text{ kg of air / kg of oil}$$

$$\text{Quantity of furnace oil} = 500 \text{ lit/hr.} = (500 * 0.95) = 475 \text{ kg/hr}$$

$$\text{Total quantity of air required} = 475 * 16.8 = 7980 \text{ kgs of air / hr.}$$

Chapter 2: Boiler

2.1 Introduction

A boiler is an enclosed vessel that provides a means of same anchor combustion heat to be transferred into water until it becomes heated water or a steam. The steam or hot water under pressure is then usable for transfer ring the heat to a process. Water is a useful and cheap medium for transferring heat to a process. When water is boiled in to steam its volume increases about 1,600 times, producing a force that is almost as explosive as gun powder. This causes the boiler to be extremely dangerous equipment that must be treated with utmost respect.

Typical Boiler Specification

Boiler make and year	XYZ & 2003
MCR rating	6 TPH (F & A 100 °C)
Type of boiler	3 Pass Fire tube Package Boiler
Design steam pressure	10.5 kg/cm ² (150 PSIG)Package Boiler
Operating pressure	110-130PSIG
Fuel used	Furnace oil

The process of heating a liquid until it reaches its gaseous state is called evaporation. Heat is transferred from one body to another by means of (1) radiation, which is the transfer of heat from a hot body to a cold body without conveying medium and physical contact, (2) convection, the transfer of heat by a conveying medium, such as air or water and (3) conduction, transfer of heat by actual physical contact, molecule to molecule. The heating surface is any part of the boiler metal that has hot gases of combustion on one side and water on the other. Any part of the boiler metal that actually contributes to making steam is heating surface. The amount of heating surface of a boiler is expressed in square meters. The larger the heating surface a boiler has, the more efficient it becomes. The measurement of the steam produced is generally expressed in tons of water evaporated to steam per hour. Maximum continuous rating is the hourly evaporation that can be maintained for 24 hours. F&A means the amount of steam generated from water at 100°C to saturated steam at 100°C

Bangladeshi Boiler Regulation

The Boilers Act was enacted in 1923 in the undivided India to consolidate and amend the law relating to steam boilers. Bangladesh Boiler Regulation (BBR) was created in 1951 in exercise of the powers conferred by section 28 of the Boilers Act, 1923 and further amended in 2007.

As per ‘The Boilers Act, 1923’ Boiler is defined as, ‘A pressure vessel in which steam is generated for use external to itself by application of heat which is wholly or partly under pressure when steam is shut off but does not include a pressure vessel with capacity less than 22.76 litres (such capacity being measured from the feed check valve to the main steam stop valve).

The following points are important regarding the definition of boiler as per the Boilers Act,

1923.

- i. It must be a 'Closed vessel'
- ii. It must generate steam for 'external use'
- iii. Volume of vessel must be over '22.76 Litres'
- iv. Working pressure more than 1.0 kg/cm^2

'Boiler component' means steam piping, feed piping, economizer, super heater, any mountings or other fitting and any other external or internal part of a boiler which is subjected to a pressure more than 1.0 kg/cm^2 .

'Economiser' means any part of feed-pipe that is wholly or partially exposed to the action of flue gases for the purpose of recovery of waste heat.

'Steam-pipe" means any main pipe exceeding 7.62 cm in internal diameter through which steam passes directly from a boiler to a prime-mover or other first user, and includes any connected fitting of a steam-pipe.

2.2 Boiler Systems

The boiler system comprises of feed water system, steam system and fuel system. The feed water system provides water to the boiler and regulates it automatically to meet the steam demand. Various valves provide access for maintenance and repair. The steam system collects and controls the steam produced in the boiler. Steam is directed through piping to the point of use. Throughout the system, steam pressure is regulated using valves and checked with steam pressure gauges. The fuel system includes all equipment used to provide fuel to generate the necessary heat. The equipment required in the fuel system depends on the type of fuel used in the system. A typical boiler room schematic is shown in Figure 2 .1.

BOILER ROOM SCHEMATIC

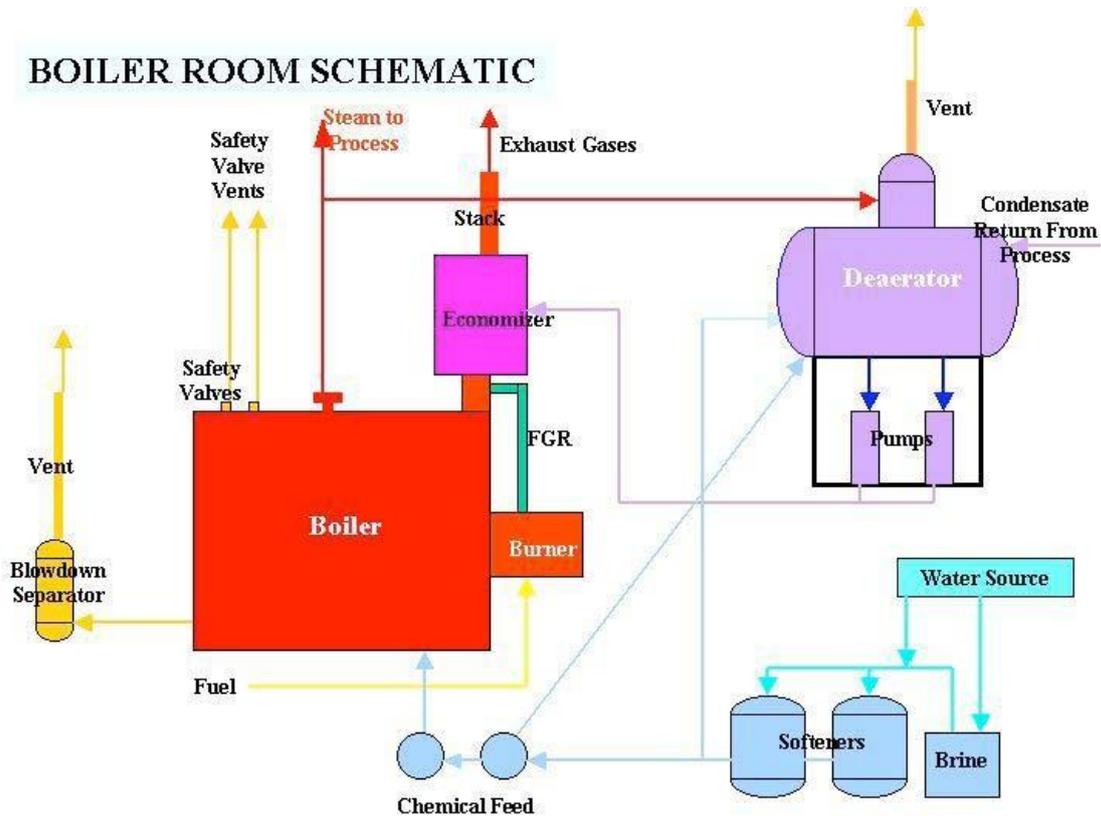


Figure 2.1: Boiler room

The water supplied to the boiler that is converted into steam is called **feed water**. The two sources of feed water are: (1) **Condensate** or condensed steam returned from the processes and (2) **Make up water** (usually raw water) which must come from outside the boiler room and plant processes. For higher boiler efficiencies, the feed water is heated by economizer.

2.3 Boiler Types and Classifications

There are virtually infinite numbers of boiler designs but generally they fit into one of the two categories:

Fire tube boiler: In a fire tube steam boiler, hot gases of combustion pass through the tube surrounded by water (Refer Figure 2.2). Fire tube boilers, typically have a lower initial cost, are more fuel efficient and easier to operate but they are limited generally to capacities of 25 tons/hr and pressures of 17.5 kg/cm². All fire tube boilers have the same basic operating principles. However, fire tube boilers have different designs like 2 pass, 3 pass, and 4 pass based on application and installation considerations.

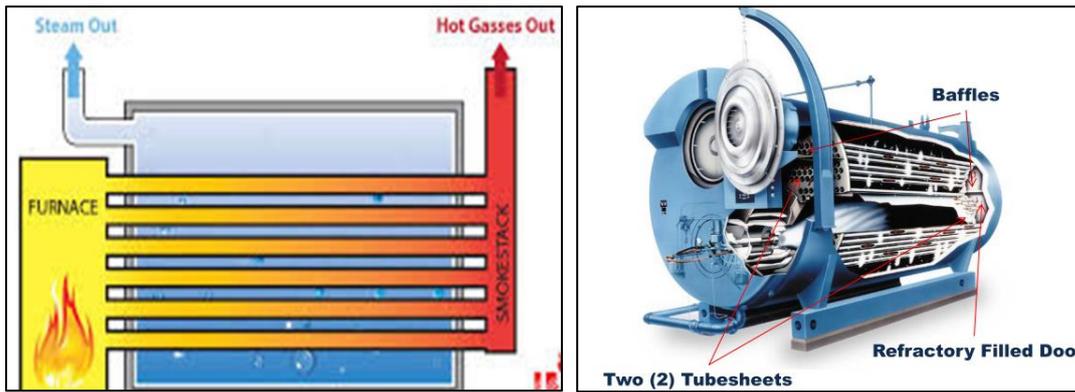


Figure 2.2: Fire tube boiler

Water tube boiler: In a water tube boiler water is inside the tubes and combustion gases pass around the outside of the tubes (Refer figure 2.3). Water-tube boilers are available in sizes far greater than a fire-tube design, up to several million pounds-per-hour of steam and able to handle higher pressures up to 5,000 psig. Disadvantages of water-tube boilers include:

High initial capital cost, cleaning is more difficult due to the design. These boilers can be single or multiple drum type design.

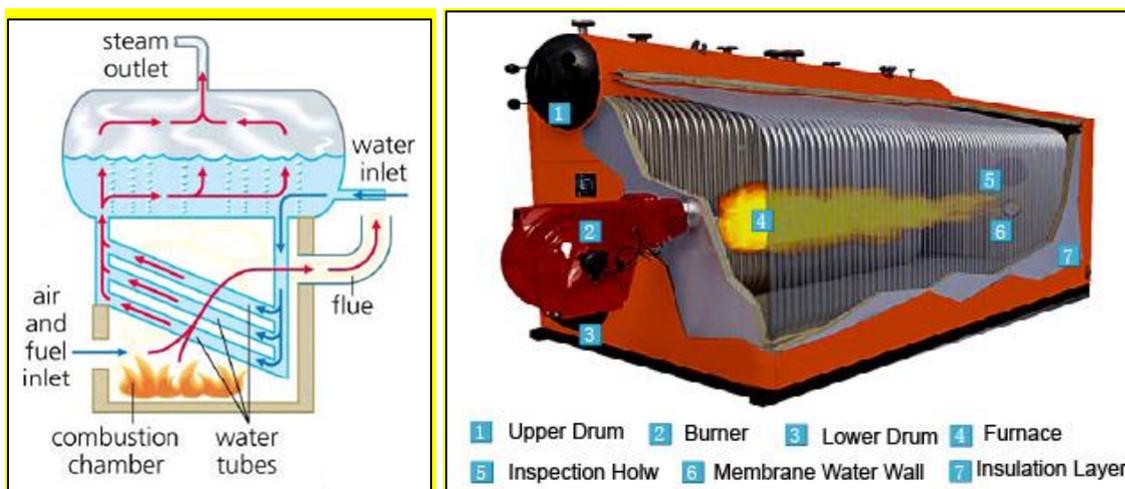


Figure 2.3: Water tube boiler

Once Through Boiler:

Feed water from the bottom header, evaporating it in water tubes, and taking out steams from the top header. In this type of boiler the water flow is only one way.

Traditionally boilers circulate water within them and only a portion gets turned into steam on any one pass. Circulation ratios (the amount of water recirculated versus the amount taken out as steam) can vary from 3 to 20.

A once through boiler passes the water through the tubes only once and there is no circulation ratio. Small once-through boilers will have a mixture of steam and water coming out boiler which is separated and the water is returned to the front of the boiler.

Large scale power plant type once through boilers will have 100% conversion to steam before it comes out of the boiler.

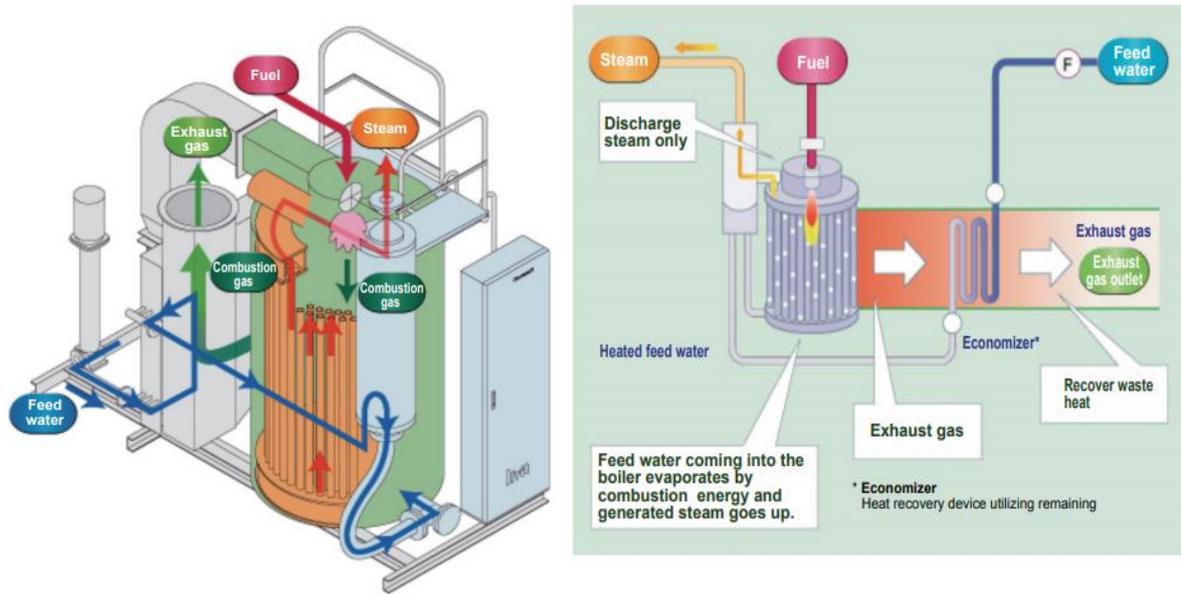


Figure 2.4: Basics of once through boiler

Effects of using once through boiler are;

1. High efficiency: Boiler efficiency of 98%
2. Stable steam pressure: $\pm 0.01\text{MPa}$ with static load
3. Long life: Designed to last for 15 years

Packaged Boiler: The packaged boiler is so called because it comes as a complete package. Once delivered to site it requires only the steam, water pipe work, fuel supply and electrical connections to be made for it to become operational. Package boilers are generally of shell type with fire tube design to achieve high heat transfer rates by both radiation and convection (Refer Figure 2.5).



Figure 2.5: Packaged boiler

The features of package boilers are:

- Small combustion space and high heat release rate resulting in faster evaporation.
- Large number of small diameter tubes leading to good convective heat transfer.

- Forced or induced draft systems resulting in good combustion efficiency.
- Number of passes resulting in better overall heat transfer.
- Higher thermal efficiency levels compared with other boilers.

Packaged boilers can be both water tube and fire tube boilers. Packaged boilers can only be used with oil and gas as fuel without separate preparation devices.

Solid Fuel fired boilers

There are different type's solid fuels available and few common fuels are: Coal, Wood, Rice husk, Baggasse. There are different type furnaces for these fuels. Most used solid fuel is coal. Wood, Rice husk is also used as fuel for many boilers as it is easily available in local area. For rice mill rice husk is a by-product and all boilers in rice mill run by rice husk.

Furnace for firing solid fuel depends on type of fuel and burning rate of fuel per hour. Depending on the design of the furnace, boiler is classified:

Stoker Fired Boiler

Stokers are classified according to the method of feeding fuel to the furnace and by the type of grate. The main classifications are:

- Chain-grate or traveling-grate stoker
- Spreader stoker

Chain-Grate or Traveling-Grate Stoker Boiler

Coal is fed on to one end of a moving steel grate. As grate moves along the length of the furnace, the coal burns before dropping off at the end as ash. Some degree of skill is required, particularly when setting up the grate, air dampers and baffles, to ensure clean combustion leaving the minimum of unburnt carbon in the ash.

The coal-feed hopper runs along the entire coal-feed end of the furnace. A coal grate is used to control the rate at which coal is fed in to the furnace by controlling the thickness of the fuel bed. Coal must be uniform in size as large lumps will not burn out completely by the time they reach the end of the grate (Refer Figure 2.6).

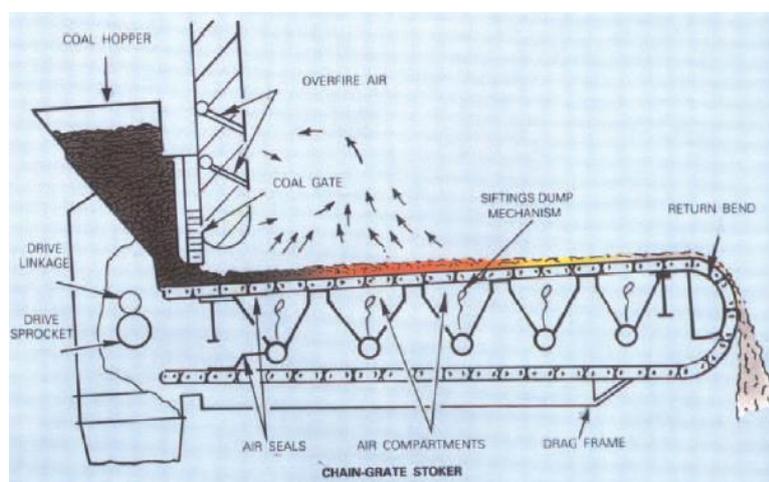


Figure 2.6: Chain-Grate Stoker

Spreader Stoker Boiler

Spreader stokers utilize a combination of suspension burning and grate burning. The coal is continually fed into the furnace above a burning bed of coal. The coal fines are burned in suspension; the larger particles fall to the grate, where they are burned in a thin, fast-burning coal bed. This method of firing provides good flexibility to meet load fluctuations, since ignition is almost instantaneous when firing rate is increased. Hence, the spreader stoker is favored over other types of stokers in many industrial applications.

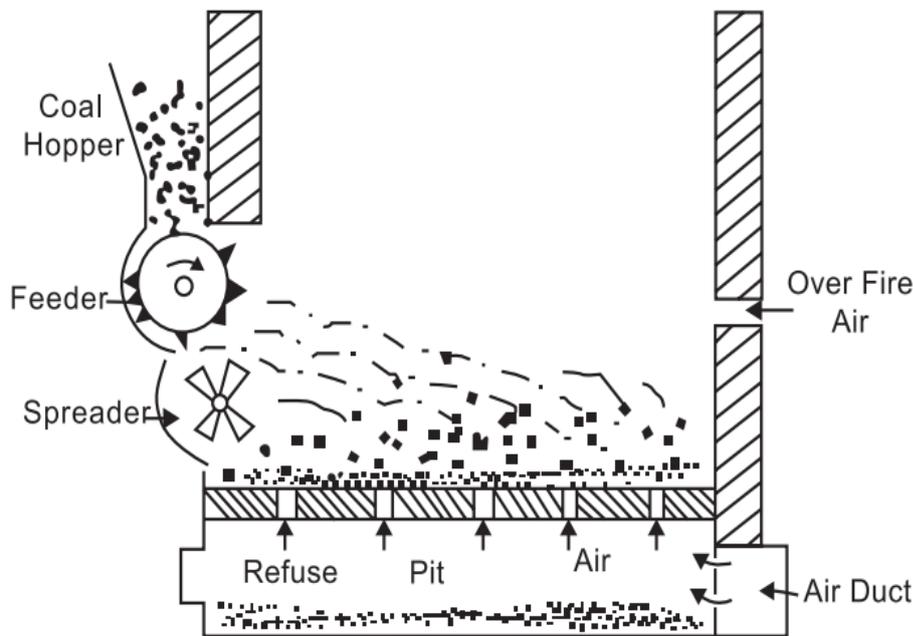


Figure 2.7: Spreader Stoker

Pulverized Fuel Boiler

Pulverized coal firing system is most efficient and modern technology. This is mostly applied in power plants boiler.

In this system coal is made as powder. The coal is ground (pulverized) to a fine powder, so that less than 2% is + 300 micro meter (μm) and 70-75% is below 75 microns, for a bituminous coal. The pulverized coal is blown with part of the combustion air into the boiler plant through a series of burner nozzles. Secondary and tertiary air may also be added. Combustion takes place at temperatures from 1300-1700°C, depending largely on coal rank. Particle residence time in the boiler is typically 2-5 seconds, and the particles must be small enough for complete burn out to have taken place during this time.

This technology is well developed, and there are thousands of units around the world, accounting for well over 90% of coal-fired capacity. Pulverized coal fired boiler can be used to fire a wide variety of coals.

One of the most popular systems for firing pulverized coal is the tangential firing, using four burners corner to corner to create a fire ball at the center of the furnace. See Figure 2.8 below.

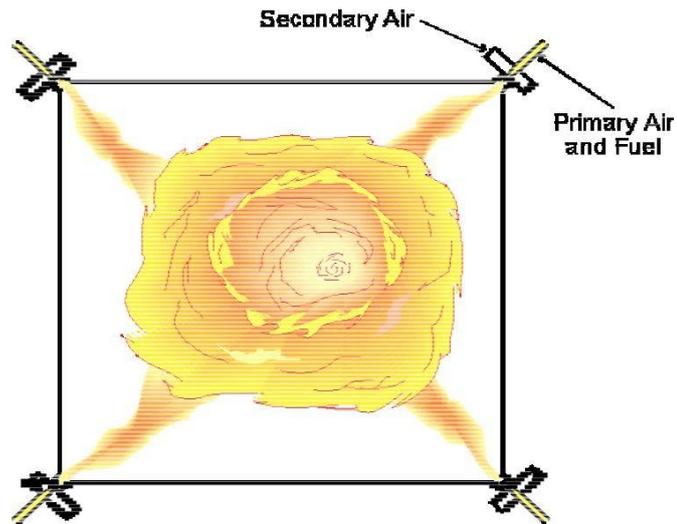


Figure 2.8: Tangential Firing

FBC Boiler

Fluid bed combustion has significant advantages over conventional firing systems and offers multiple benefits namely fuel flexibility, reduced emission of noxious pollutants such as SO_x and NO_x, compact boiler design and higher combustion efficiency.

When an evenly distributed air or gas is passed upward through a finely divided bed of solid particles such as sand supported on a fine mesh, the particles are undisturbed at low velocity. As air velocity is gradually increased, a stage is reached when the individual particles are suspended in the air stream. Further, increase in velocity gives rise to bubble formation, vigorous turbulence and rapid mixing and the bed is said to be fluidized.

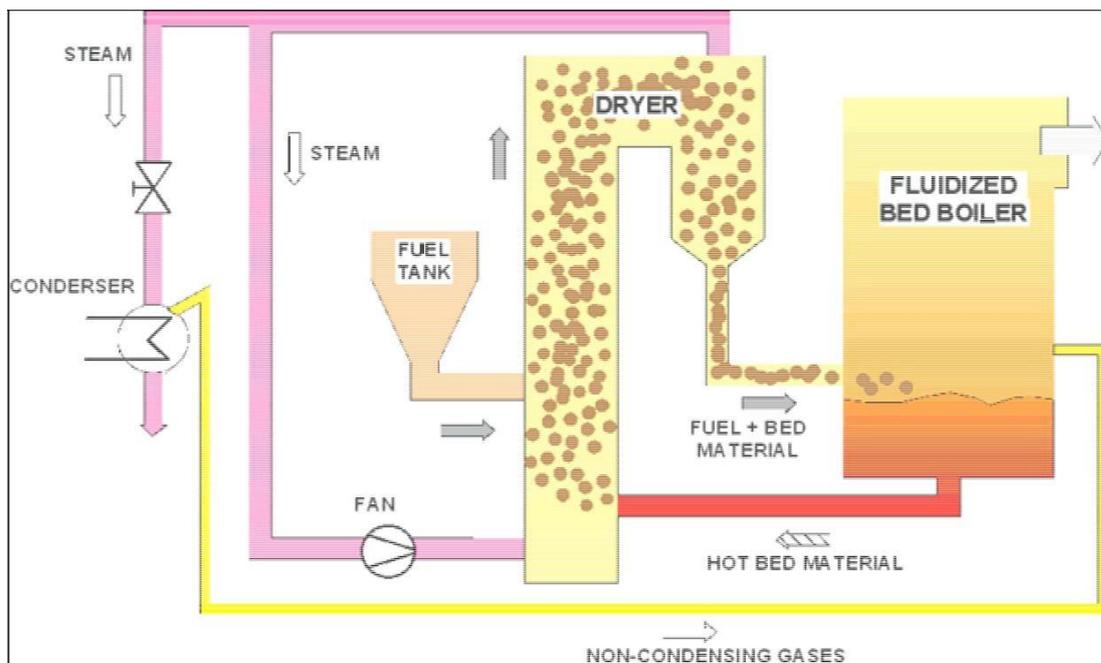


Figure 2.9: Schematic of fluidized bed boiler

Super Critical Boiler

In the temperature entropy diagram of steam, a point is reached where the boiling water and dry saturated steam lines converge and at that point, the latent heat is zero. The critical point corresponds to a pressure of 221.2 bar absolute and a temperature of 374.18 °C.

If water is heated beyond the above condition, steam parameters are referred to as super critical. A boiler producing steam above the critical pressure is called the supercritical boiler. While sub-critical boiler has three distinct sections - economiser, evaporator and super heater the supercritical boiler has only an economiser and super heater. The advantages of super critical boilers are

- Higher heat transfer rate
- More flexible in accepting load variation
- Greater ease of operation
- High thermal efficiency (40-42% of power generating stations)
- The absence of two-phase mixer minimise the problems of erosion and corrosion
- Steadier pressure level

The super critical boilers call for special materials to be used for constituent heat transfer surfaces like drum, water walls, economizer and re-heaters, in order to withstand the elevated pressure & temperature conditions.

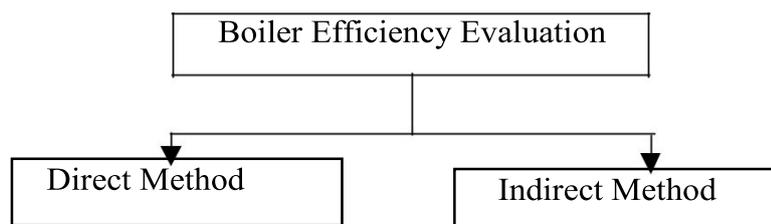
2.4 Performance Evaluation of Boilers

The performance of boiler, like efficiency and evaporation ratio reduces with time due to poor combustion, heat transfer surface fouling and poor operation and maintenance. Deteriorating fuel quality, water quality etc. also leads to poor boiler performance. Efficiency tests help us to find out the deviation of boiler efficiency from the best efficiency and target problems for corrective action.

Boiler Efficiency

Thermal efficiency of boiler is defined as the percentage of heat input that is effectively utilized to generate steam. There are two methods of assessing boiler efficiency.

1. The Direct Method: Where the energy gain of the working fluid (water and steam) is compared with the energy content of the boiler fuel.
2. The Indirect Method: Where the efficiency is the difference between the losses and the energy input.



a. Direct Method

This is also known as ‘input-output method’ due to the fact that it needs only the useful output (steam) and the heat input (i.e. fuel) for evaluating the efficiency. This efficiency can be evaluated using the formula.

$$\text{Boiler Efficiency} = \frac{\text{Heat Output}}{\text{Heat Input}} \times 100$$

Parameters to be monitored for the calculation of boiler efficiency by direct method are:

- Quantity of steam generated per hour (Q) in kg/hr.
- Quantity of fuel used per hour (q) in kg/hr.
- The working pressure (in kg/cm²(g)) and superheat temperature (°C), if any
- The temperature of feed water (°C)
- Type of fuel and gross calorific value of the fuel (GCV) in kcal/kg of fuel

$$\text{Boiler Efficiency } (\eta) = \frac{Q \times (h_g - h_f)}{q \times \text{GCV}} \times 100$$

Where, h_g - Enthalpy of saturated steam in kcal/kg of steam

h_f - Enthalpy of feed water in kcal/kg of water

The fuel calorific value may be gross or net and accordingly, the efficiency reported is referred to as efficiency on GCV or NCV basis.

Example 2.1

Find out the efficiency of the boiler by direct method with the data given below:

Type of boiler	: Coal fired
Quantity of steam (dry) generated	: 8 TPH
Steam pressure / temp	: 10 kg/cm ² (g)/ 180 °C
Quantity of coal consumed	: 1.8 TPH
Feed water temperature	: 85 °C
GCV of coal	: 3200 kCal/kg
Enthalpy of saturated steam at 10 kg/cm ² pressure	: 665 kCal/kg(saturated)
Enthalpy of feed water	: 85 kCal/kg

Solution:

$$\begin{aligned} \text{Boiler Efficiency } (\eta) &= \frac{8 \times (665 - 85)}{1.8 \times 3200} \times 100 \\ &= 80\% \text{ (on GCV basis)} \end{aligned}$$

It should be noted that boiler may not generate 100% saturated dry steam and there may be some amount of wetness in the steam. Since it is practically difficult to measure the dryness fraction, it is assumed the boiler generates 100% saturated steam for calculation purposes. The resulting errors are likely to be in significant.

Advantages:

- Plant people can evaluate quickly the efficiency of boilers
- Requires few parameters for computation
- Needs few instruments for monitoring

Disadvantages:

- Does not give clues to the operator as to why efficiency of system is lower
- Does not calculate various losses accountable for various efficiency levels

b. Indirect Method

Indirect method also called as heat loss method. The efficiency can be arrived at, by subtracting the heat loss fractions from 100. All standard does not includes blow down loss in the efficiency determination process

However, the efficiency calculations are meant for practicing energy managers, simpler calculation procedure is being adopted in industries.

There are reference standards for Boiler Testing at Site using indirect method namely British Standard, BS845:1987 and USA Standard is 'ASME PTC-4.1 Power Test Code Steam Generating Units'.

The principle losses that occur in a boiler are:

- Loss of heat due to dry flue gas
- Loss of heat due to moisture in fuel and combustion air
- Loss of heat due to combustion of hydrogen
- Loss of heat due to radiation
- Loss of heat due to un burnt

In the above, loss due to moisture in fuel and the loss due to combustion of hydrogen cannot be controlled by design and is dependent on the fuel and these two losses are practically zero while computing the efficiency on the basis of net calorific value.

The data required for calculation of boiler efficiency using indirect method are:

Ultimate analysis of fuel (H₂, O₂, S, C, moisture content, ash content)

Percentage of Oxygen or CO₂ in the flue gas

Flue gas temperature in °C (T_f)

Ambient temperature in °C (T_a)

Humidity of air in kg/kg of dry air

GCV of fuel kcal/kg

Percentage combustible in ash (in case of solid fuels)

GCV of ash in kcal/kg (in case of solid fuels)

Solution:

$$\text{Excess Air supplied (EA)} = \frac{O_2 \%}{21 - O_2 \%} \times 100$$

Theoretical air requirement

$$= [(11.6 \times C) + \{34.5 \times (H_2 - O_2/8)\} + (4.35 \times S)]/100 \text{ kg/kg of fuel}$$

Actual mass of air supplied/ kg of fuel (AAS) = $\{1 + EA/100\}$ x theoretical air

i) Percentage heat loss due to dry flue gas = $\frac{m \times C_p \times (T_f - T_a)}{\text{GCV of fuel}} \times 100$

where,

m = mass of dry flue gas in kg/kg of fuel

= (mass of dry products of combustion per kg of fuel) + (mass of N₂ in Fuel on 1kg basis) + (mass of N₂ in actual mass of air supplied).

c_p = specific heat of flue gas (0.24kCal/kg deg. C)

ii) Percentage heat loss due to evaporation of water formed due to H₂ in fuel

$$= \frac{9 \times H_2 \times \{584 + 0.45(T_f - T_a)\}}{\text{GCV of fuel}} \times 100$$

Where,

H₂ = percentage of H₂ in 1 kg of fuel, 0.45 is the specific heat of super-heated steam,

584 is the Latent heat of superheated vapour at partial pressure

iii) Percentage heat loss due to evaporation of moisture present in fuel

$$= \frac{M \times \{584 + 0.45(T_f - T_a)\}}{\text{GCV of fuel}} \times 100$$

Where,

M = percentage moisture in 1kg of fuel

iv) Percentage heat loss due to moisture present in air

$$= \frac{\text{AAS} \times \text{humidity} \times 0.45 \times (T_f - T_a)}{\text{GCV of fuel}} \times 100$$

v) Percentage heat loss due to partial conversion of C to CO

$$= \frac{\%CO \times C}{\%CO + \%CO_2} \times \frac{5654}{\text{GCV of fuel}} \times 100$$

Where,

CO = Volume of CO in flue gas (%)
(1% = 10000 ppm)

CO₂ = Actual Volume of CO₂ in flue gas (%)

C = Carbon content kg / kg of fuel

vi) Percentage heat loss due to un burnt in fly ash

$$= \frac{\text{Total ash collected per kg of fuel burnt} \times \text{G.C.V of fly ash} \times 100}{\text{GCV of fuel}}$$

vii) Percentage heat loss due to un burnt in bottom ash

$$= \frac{\text{Total ash collected per kg of fuel burnt} \times \text{G.C.V of bottom ash} \times 100}{\text{GCV of fuel}}$$

viii) Percentage heat loss due to radiation and other un accounted loss

The actual radiation and convection losses are difficult to assess because of particular emissivity of various surfaces, its inclination, airflow pattern etc. In a relatively small boiler, with a capacity of 10MW, the radiation and unaccounted losses could amount to between 1% and 2% of the gross calorific value of the fuel, while in a 500MW boiler, values between 0.2% to 1% are typical. The loss may be assumed appropriately depending on the surface condition.

$$\text{Efficiency of boiler } (\eta) = 100 - (i + ii + iii + iv + v + vi + vii)$$

Example 2.2 Calculating boiler efficiency by indirect method

- Type of boiler : Oil fired
- Ultimate analysis of Oil:
C - 84.0%; H₂ -12.0%; S- 3.0%; O₂ -1.0%; GCV of Oil- 10200kCal/kg
- Percentage of Oxygen in flue gas :7
- Percentage of CO₂ in flue gas :11
- Flue gas temperature(T_f) : 220⁰C
- Ambient temperature(T_a) : 27 ⁰C
- Humidity of air : 0.018 kg/kg of dry air
- Radiation and other unaccounted losses : 3.5% (estimated)

Solution

1. Excess air supplied(EA) = $(7 \times 100)/(21-7) = 50\%$

2. Theoretical air requirement

$$\begin{aligned} &= [(11.6 \times C) + \{34.5 \times (H_2 - O_2/8)\} + (4.35 \times S)]/100 \text{ kg/kg of fuel} \\ &= [(11.6 \times 84) + \{34.5 \times (12 - 1/8)\} + (4.35 \times 3)]/100 \text{ kg/kg of fuel} \\ &= 974.4 + 409.6875 + 13.05 \\ &= 13.97 \end{aligned}$$

3. Actual mass of air supplied per kg of fuel

$$\begin{aligned}
 &= [1+EA/100] \times \text{Theo. Air (AAS)} \\
 &= [1 + 50/100] \times 13.97 \\
 &= 20.96
 \end{aligned}$$

4. Percentage heat loss due to dry flue gas (L1) = $\frac{m \times C_p \times (T_f - T_a)}{\text{GCV of fuel}} \times 100$

$$\begin{aligned}
 m &= (0.84 \times (44/12)) + (0.03 \times (64/32)) + (20.96 \times 0.77) + ((20.96 - 13.97) \times 0.23) \\
 &= 3.08 + 0.06 + 16.14 + 1.61 = 20.89 \\
 L1 &= 20.89 \times 0.24 \times (220 - 27) \times 100 / 10200 = 9.49\%
 \end{aligned}$$

5. Percentage heat loss due to evaporation of water formed due to H₂ in fuel (L2)

$$\begin{aligned}
 &= \frac{9 \times H_2 \times \{584 + 0.45(T_f - T_a)\}}{\text{GCV of fuel}} \times 100 \\
 L2 &= 9 \times 0.12 \{584 + 0.45(220-27)\} \times 100 / 10200 = 7.10\%
 \end{aligned}$$

6. Percentage heat loss due to moisture present in air (L3)

$$\begin{aligned}
 &= \frac{\text{AAS} \times \text{humidity} \times 0.45 \times (T_f - T_a)}{\text{GCV of fuel}} \times 100 \\
 L3 &= 20.96 \times 0.018 \times 0.45 (220-27) \times 100 / 10200 = 0.31\%
 \end{aligned}$$

7. Percentage heat loss due to radiation and other unaccounted losses (L4) = 3.5% (given)

$$\begin{aligned}
 \text{Boiler Efficiency} &= 100 - (L1 + L2 + L3 + L4) \\
 &= 100 - (9.49 + 7.1 + 0.31 + 3.5) = 79.6\%
 \end{aligned}$$

Boiler Evaporation Ratio

Evaporation ratio means kilogram of steam generated per kilogram of fuel consumed. Evaporation ratio monitoring is best suited for any boiler when its own performance is compared on day to day basis as a performance indicator; given that enthalpy gain in steam and fuel calorific value remain constant. A drop in evaporation ratio indicates a drop in Boiler efficiency.

Typical values of Evaporation ratio for different type of fuels are as follows:

Biomass fired boilers	: 2.0 to 3.0
Coal fired boilers	: 4.0 to 5.5
Oil fired boilers	: 13.5 to 14.5
Gas fired boilers	: 11.0 to 13.0

However, the above ratio will depend upon the type of boiler and associated efficiencies.

2.5 Boiler Blow down

The impurities found in boiler water depend on the untreated feed water quality, the treatment process used and the boiler operating procedures. As a general rule, the higher is the boiler operating pressure, the greater will be the sensitivity to impurities. As the feed water materials evaporate in to steam, dissolved solids concentrate in the boiler either in a dissolved or suspended state. Above a certain level of concentration, these solids encourage foaming and cause carryover of water in to the steam. This leads to scale formation inside the boiler, resulting in localized overheating and ending finally in tube failure.

It is therefore necessary to control the level of concentration of the solid sand this is achieved by the process of 'blowing down', where a certain volume of water is blown off and is automatically replaced by feed water - thus maintaining the optimum level of total dissolved solids (TDS) in the water. Blow down is necessary to protect the surfaces of the heat exchanger in the boiler. However, blow down can be a significant source of heat loss, if improperly carried out. The maximum amount of total dissolved solids (TDS) concentration permissible in various types of boilers is given in Table 2.1.

Table 2.1: Recommended TDS levels for various boilers

	Boiler Type	Maximum TDS (ppm)
1.	Smoke and water tube boilers (12 kg/cm ²)	5,000 ppm
2.	Low pressure Water tube boiler	2000-3000
3.	High Pressure Water tube boiler with super heater etc.	3,000 - 3,500 ppm
4.	Package and economic boilers	3,000 ppm
5.	Coil boilers and steam generators	2000 (in the feed water)

Conductivity as Indicator of Boiler Water Quality

Conductivity is a standard measurement for monitoring the overall total dissolved solids present in the boiler. Arise in conductivity indicates arise in the "contamination" of the boiler water.

Intermittent vs. Continuous Blow down

Conventional methods for blowing down the boiler depend on two kinds of blow down – intermittent and continuous.

Intermittent Blow down

The parameters that are most often monitored to ensure the quality of steam are TDS or conductivity, pH, Silica and Phosphates concentration. The boiler is blown down by manually operating a valve fitted to discharge pipe at the lowest point of boiler shell to reduce these levels and keeping them controlled to a point where the steam quality is not likely to be affected. A substantial amount of heat energy is lost in this process. In intermittent blow down, a large diameter line is opened for a short period of time, the time being based on a thumb rule such as "once a shift for 2 minutes".

Continuous Blow down

There is a steady and constant dispatch of small stream of concentrated boiler water. In a continuous blow down, large quantities of heat are wasted though it is inevitable especially in large high pressure boilers. However, opportunity exists for recovering this heat by blowing in to a flash tank and generating flash steam. This flash steam can be used for pre-heating boiler feed water or for any other purpose.

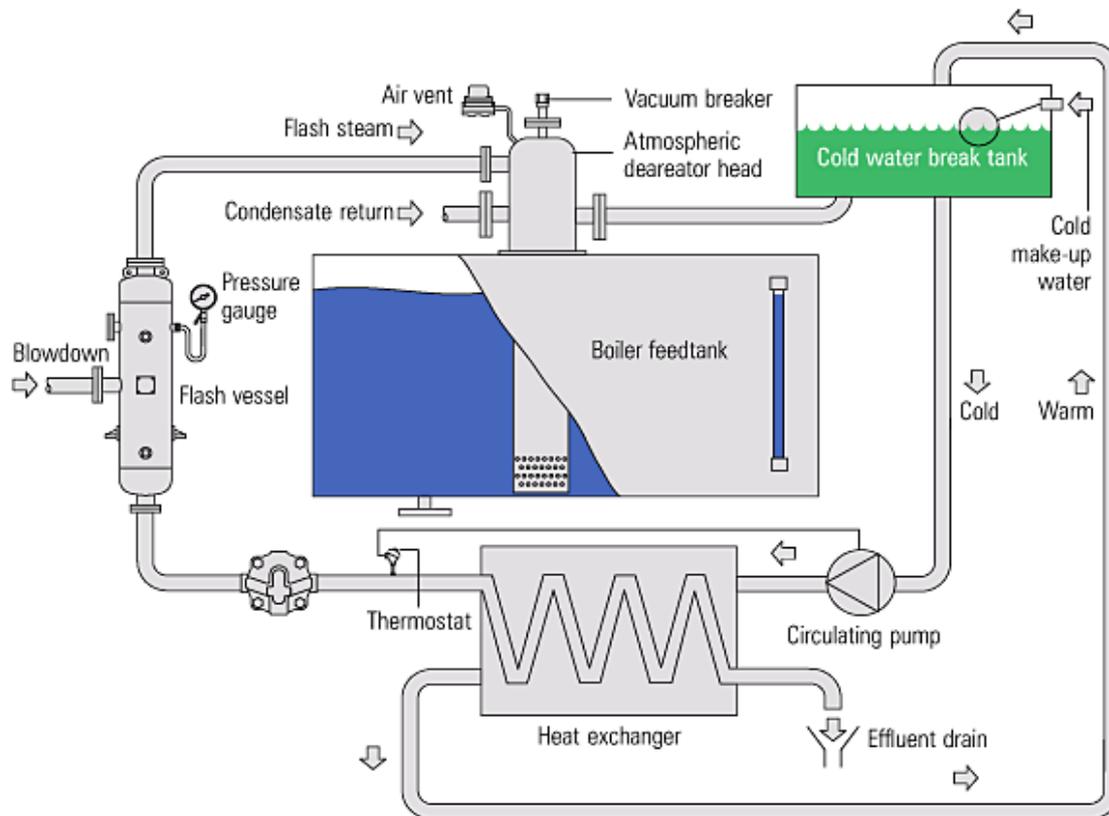


Figure 2.8: Blow down heat recovery system

Benefits of Blow down

Good boiler blow down control can significantly reduce treatment and operation al costs that include:

- Lower pre-treatment costs
- Less make-up water consumption
- Reduced maintenance own time
- Increased boiler life
- Lower consumption of treatment chemicals

The following formula gives the quantity of blow down required:

$$\text{Blow down (\%)} = \frac{[\text{Feed water TDS} \times \% \text{ Make up water}]}{(\text{Maximum Permissible TDS in Boiler water} - \text{Feed water TDS})}$$

If maximum permissible limit of TDS as in a package boiler is 3000ppm, percentage make up water is 10% and TDS in feed water is 300ppm, then the percentage blow down is given as:

$$\begin{aligned} &= (300 \times 10) / (3000 - 300) \\ &= 1.11\% \end{aligned}$$

If boiler evaporation rate is 3000 kg/hr, then required blow down rate is:

$$\begin{aligned} &= 3000 \times 0.0111 \\ &= 33.3\text{kg/hr} \end{aligned}$$

2.6 Boiler Water Treatment

Producing quality steam on demand depends on properly managed water treatment to control steam purity, deposits and corrosion. A boiler is the sump of the boiler system. It ultimately receives all of the pre-boiler contaminants. Boiler performance, efficiency, and service life are direct products of selecting and controlling feed water used in the boiler.

The boiler water must be sufficiently free of deposit forming solids to allow rapid and efficient heat transfer and it must not be corrosive to the boiler metal.

Deposit Control

Deposits in boilers may result from hardness contamination of feed water and corrosion products from the condensate and feed water system. Hardness contamination of the feed water may arise due to deficient softener system. Deposits and corrosion result in efficiency losses and may result in boiler tube failures and inability to produce steam. Deposits act as insulators and slow heat transfer. Large amounts of deposits throughout the boiler could reduce the heat transfer enough to reduce the boiler efficiency significantly. Different type of deposits affects the boiler efficiency differently. Thus it may be useful to analyse the deposits for its characteristics. The insulating effect of deposits causes the boiler metal temperature to rise and may lead to tube-failure by overheating.

Impurities causing deposits

The most important chemicals contained in water that influences the formation of deposits in the boilers are the salts of calcium and magnesium, which are known as hardness salts. Calcium and magnesium bicarbonate dissolve in water to form an alkaline solution and these salts are known as alkaline hardness. They decompose upon heating, releasing carbon dioxide and forming a soft sludge, which settles out. These are called temporary hardness-hardness that can be removed by boiling.

Calcium and magnesium sulphates, chlorides and nitrates, etc. when dissolved in water are chemically neutral and are known as non-alkaline hardness. These are called permanent hardness and form hard scales on boiler surfaces, which are difficult to remove. Non-alkalinity hardness chemicals fall out the solution due to reduction in solubility as the temperature rises, by concentration due to evaporation which takes place within the boiler, or by chemical change to a less soluble compound.

Silica

The presence of silica in boiler water can rise to formation of hard silicate scales. It can also associate with calcium and magnesium salts, forming calcium and magnesium silicates of very low thermal conductivity. Silica can give rise to deposits on steam turbine blades, after been carried over either in droplets of water in steam, or in volatile form in steam at higher pressures.

Two major types of boiler water treatment are: Internal water treatment and External water treatment,

Internal Water Treatment

Internal treatment is carried out by adding chemicals to boiler to prevent the formation of scale by converting the scale-forming compounds to free-flowing sludge's, which can be removed by blow down. This method is limited to boilers, where feed water is low in hardness salts, to low pressures– high TDS content in boiler water is tolerated, and when only small quantity of water is required to be treated. If these conditions are not applied, then high rates of blow down are required to dispose the sludge. They become uneconomical from heat and water loss consideration.

Different waters require different chemicals. Sodium carbonate, sodium aluminate, sodium phosphate, sodium sulphite and compounds of vegetable or inorganic origin are all used for this purpose. Proprietary chemicals are available to suit various water conditions. The specialist must be consulted to determine the most suitable chemicals to use in each case. Internal treatment alone is not recommended.

External Water Treatment

External treatment is used to remove suspended solids, dissolved solids (particularly the calcium and magnesium ions which are major causes of scale formation) and dissolved gases (oxygen and carbon dioxide).

The external treatment processes available are: ion exchange; demineralization; reverse osmosis and de-aeration. Before any of these are used, it is necessary to remove suspended solids and colour from the raw water, because these may foul the resins used in the subsequent treatment sections.

Methods of pre-treatment include simple sedimentation in settling tanks or settling in clarifiers with aid of coagulants and flocculants. Pressure sand filters, with spray aeration to remove carbon dioxide and iron, may be used to remove metal salts from bore well water.

The first stage of treatment is to remove hardness salt and possibly non-hardness salts. Removal of only hardness salts is called softening, while total removal of salts from solution is called demineralization.

The processes are:

Ion-exchange process (Softener Plant)

In ion-exchange process, the hardness is removed as the water passes through bed of natural zeolite or synthetic resin and without the formation of any precipitate. The simplest type is ‘base exchange’ in which calcium and magnesium ions are exchanged for sodium ions. After saturation regeneration is done with sodium chloride. The sodium salts being soluble, do not form scales in boilers. Since base exchanger only replaces the calcium and magnesium with sodium, it does not reduce the TDS content, and blow down quantity. It also does not reduce the alkalinity.

Softening reaction



Regeneration reaction



Demineralization is the complete removal of all salts. This is achieved by using a “cation” resin, which exchanges the cations in the raw water with hydrogen ions, producing hydrochloric, sulphuric and carbonic acid. Carbonic acid is removed in degassing tower in which air is blown through the acid water. Following this, the water passes through an “anion” resin which exchanges anions with the mineral acid (e.g. sulphuric acid) and forms water. Regeneration of cations and anions is necessary at intervals using, typically, mineral acid and caustic soda respectively. The complete removal of silica can be achieved by correct choice of anion resin.

Ion exchange processes can be used for almost total demineralization if required, as is the case in large electric power plant boilers

De-aeration

In de-aeration (Figure 2.9), dissolved gases, such as oxygen and carbon dioxide, are expelled by preheating the feed water before it enters the boiler.

All natural waters contain dissolved gases in solution. Certain gases, such as carbon dioxide and oxygen, greatly increase corrosion. When heated in boiler systems, carbon dioxide (CO_2) and oxygen (O_2) are released as gases and combine with water (H_2O) to form carbonic acid, (H_2CO_3).

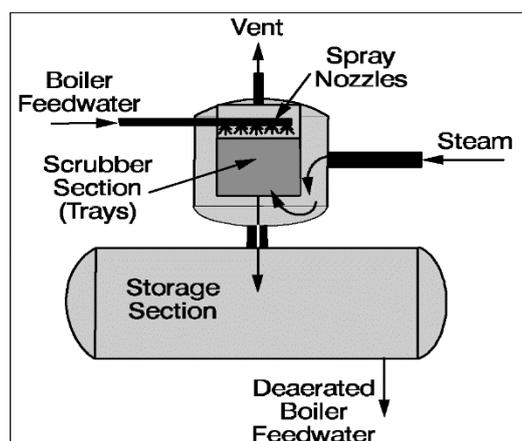


Figure 2.9: Deaerator

Removal of oxygen, carbon dioxide and other non-condensable gases from boiler feed water is vital to boiler equipment longevity as well as safety of operation. Carbonic acid corrodes metal reducing the life of equipment and piping. It also dissolves iron (Fe) which

when returned to the boiler precipitates and causes scaling on the boiler and tubes. This scale not only contributes to reducing the life of the equipment but also increases the amount of energy needed to achieve heat transfer.

De-aeration can be done by mechanical de-aeration, by chemical de-aeration or by both together.

Mechanical de-aeration

Mechanical de-aeration for the removal of these dissolved gases is typically utilized prior to the addition of chemical oxygen scavengers. Mechanical de-aeration is based on Charles' and Henry's laws of physics. Simplified, these laws state that removal of oxygen and carbon dioxide can be accomplished by heating the boiler feed water, which reduces the concentration of oxygen and carbon dioxide in the atmosphere surrounding the feed water. Mechanical de-aeration can be the most economical. They operate at the boiling point of water at the pressure in the de-aerator. They can be of vacuum or pressure type.

The vacuum type of de-aerator operates below atmospheric pressure, at about 82°C , can reduce the oxygen content in water to less than 0.02 mg/litre. Vacuum pumps or steam ejectors are required to maintain the vacuum.

The pressure-type de-aerators operates by allowing steam into the feed water through a pressure control valve to maintain the desired operating pressure, and hence temperature at a minimum of 105°C . The steam raises the water temperature causing the release of O_2 and CO_2 gases that are then vented from the system. This type can reduce the oxygen content to 0.005 mg/litre.

Where excess low-pressure steam is available, the operating pressure can be selected to make use of this steam and hence improve fuel economy. In boiler systems, steam is preferred for de-aeration because:

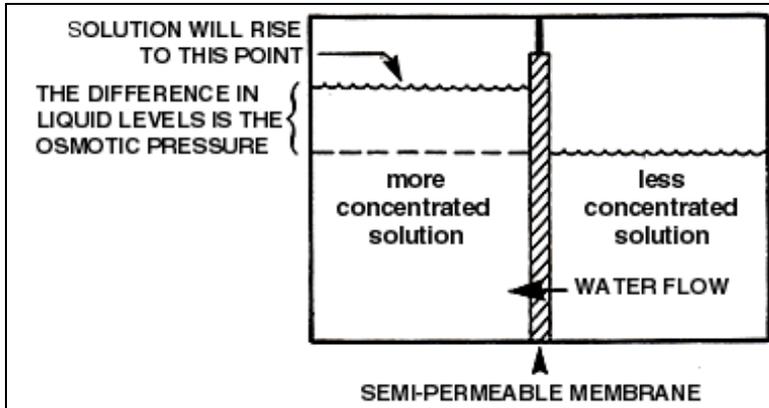
- Steam is essentially free from O_2 and CO_2
- Steam is readily available
- Steam adds the heat required to complete the reaction.

Chemical de-aeration

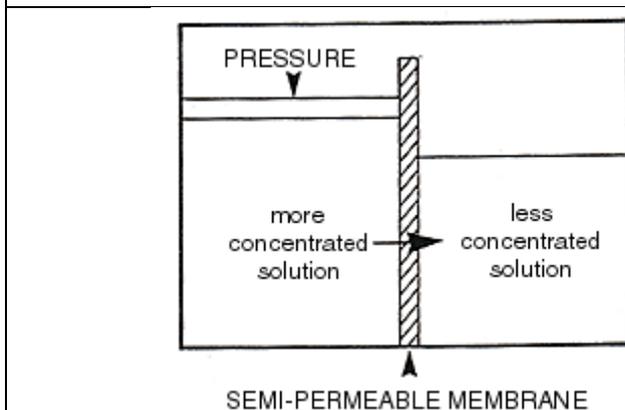
While the most efficient mechanical deaerators reduce oxygen to very low levels (0.005 mg/litre), even trace amounts of oxygen may cause corrosion damage to a system. Consequently, good operating practice requires removal of that trace oxygen with a chemical oxygen scavenger such as sodium sulfite or hydrazine. Sodium sulphite reacts with oxygen to form sodium sulphate, which increases the TDS in the boiler water and hence increases the blow down requirements and make-up water quality. Hydrazine reacts with oxygen to form nitrogen and water. It is invariably used in high pressures boilers when low boiler water solids are necessary, as it does not increase the TDS of the boiler water.

Reverse Osmosis

When solutions of differing concentrations are separated by a semi-permeable membrane, water from less concentrated solution passes through the membrane to dilute the liquid of high concentration, which is called osmosis. If the solution of high concentration is pressurized, the process is reversed and the water from the solution of high concentration flows to the weaker solution. This is known as reverse osmosis.



The semi permeable nature of the membrane allows the water to pass much more readily than the dissolved minerals. Since the water in the less concentrated solution seeks to dilute the more concentrated solution, the water passage through the membrane generates a noticeable head difference between the two solutions. This head difference is a measure of the concentration difference of the two solutions and is referred to as the osmotic pressure difference.



When a pressure is applied to the concentrated solution which is great that the osmotic pressure difference, the direction of water passage through the membrane is reversed and the process that we refer to as reverse osmosis is

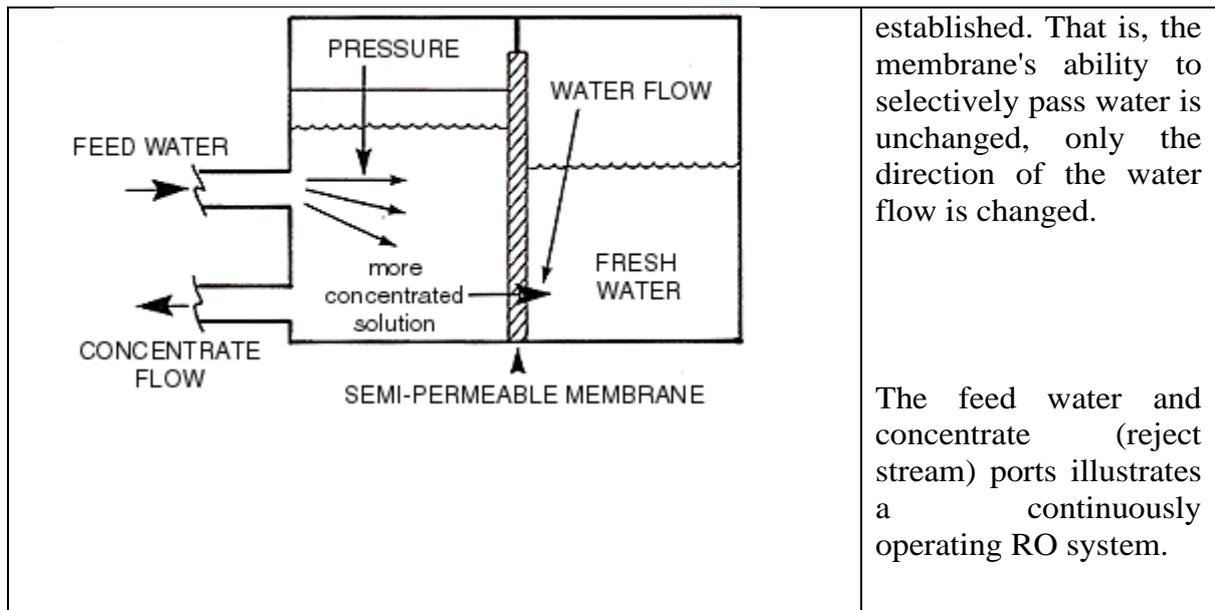


Figure 2.10: Reverse Osmosis

The quality of water produced depends upon the concentration of the solution on the high-pressure side and pressure differential across the membrane. This process is suitable for waters with very high TDS, such as sea water.

Recommended boiler and feed water quality

The impurities found in boiler water depend on the untreated feed water quality, the treatment process used and the boiler operating procedures. As a general rule, the higher the boiler operating pressure, the greater will be the sensitivity to impurities. Recommended feed water and boiler water limits are shown in Table 2.2 and Table 2.3.

Table 2.2: Recommended feed water limits

Factor	Up to 20Kg/cm ²	21 - 40Kg/cm ²	41 - 60Kg/cm ²
Total iron (max)ppm	0.05	0.02	0.01
Total copper (max)ppm	0.01	0.01	0.01
Total silica (max)ppm	1.0	0.3	0.1
Oxygen (max)ppm	0.02	0.02	0.01
Hydrazine residualppm	-	-	-0.02-0.04
0 pH at 25C	8.8-9.2	8.8-9.2	8.2-9.2
Hardness	1.0	0.5	-

Table 2.3: Recommended boiler water limits (IS 10392, Year 1982)

Factor	Upto 20Kg/cm ²	21 - 40Kg/cm ²	41 - 60Kg/cm ²
TDS	3000-3500	1500-2000	500-750
Total iron dissolved solids ppm	500	200	150
Specific electrical conductivity at 25 °C (mho)	1000	400	300
Phosphate residual ppm	20-40	20-40	15-25
pH at 25 °C	10-10.5	10-10.5	9.8-10.2
Silica (max) ppm	25	15	10

2.7 Energy Conservation Opportunities

The various energy efficiency opportunities in boiler system can be related to combustion, heat transfer, avoidable losses, high auxiliary power consumption, and water quality and blow down.

Examining the following factors can indicate if a boiler is being run to maximize its efficiency:

1. Stack Temperature

The stack temperature should be low as possible. However, it should not be so low that water vapor in the exhaust condenses on the stack walls. This is important in fuels containing significant sulphur as low temperature can lead to sulphur dew point corrosion. Stack temperatures greater than 200°C indicates potential for recovery of waste heat. It also indicates the scaling of heat transfer / recovery equipment and hence the urgency of taking an early shutdown for water / flue side cleaning.

2. Feed Water Preheating using Economiser

Typically, the flue gases leaving a modern 3-pass shell boiler are at temperatures of 200 to 300°C. Thus, there is a potential to recover heat from these gases. The flue gas exit temperature from a boiler is usually maintained at a minimum of 200°C, so that the sulphur oxides in the flue gas do not condense and cause corrosion in heat transfer surfaces. When a clean fuel such as natural gas, LPG or gas oil is used, the economy of heat recovery must be worked out, as the flue gas temperature may be well below 200 °C.

The potential for energy saving depends on the type of boiler installed and the fuel used. For a typically older model shell boiler, with a flue gas exit temperature of 260 °C, an economizer could be used to reduce it to 200 °C, increasing the feed water temperature by 15 °C. Increase in overall thermal efficiency would be in the order of 3%. For a modern 3-pass shell boiler firing natural gas with a flue gas exit temperature of 140 °C a condensing economizer would reduce the exit temperature to 65 °C increasing thermal efficiency by 5%.

3. Combustion Air Pre heat

Combustion air preheating is an alternative to feed water heating. In order to improve thermal efficiency by 1%, the combustion air temperature must be raised by 20⁰C. Most gas and oil burners used in a boiler plant are not designed for high air preheat temperatures.

Modern burners can withstand much higher combustion air preheat, so it is possible to consider such units as heat exchangers in the exit flue as an alternative to an economizer, when either space or a high feed water return temperature make it viable.

4. Incomplete Combustion

Incomplete combustion can arise from a shortage of air or surplus of fuel or poor distribution of fuel. It is usually obvious from the colour or smoke, and must be corrected immediately.

In the case of oil and gas fired systems, CO or smoke (for oil fired systems only) with normal or high excess air indicates burner system problems. A more frequent cause of incomplete combustion is the poor mixing of fuel and air at the burner. Poor oil fires can result from improper viscosity, worn tips, carbonization on tips and deterioration of diffusers or spinner plates.

With coal firing, unburned carbon can comprise a big loss. It occurs as grit carry-over or carbon-in-ash may amount to more than 2% of the heat supplied to the boiler. Non uniform fuel size could be one of the reasons for incomplete combustion. In chain grate stokers, large lumps will not burnout completely, while small pieces and fines may block the air passage, thus causing poor air distribution. In sprinkler stokers, stoker grate condition, fuel distributors, wind box air regulation and over-fire systems can affect carbon loss. Increase in the fines in pulverized coal also increases carbon loss.

5. Excess Air Control

Excess air is required in all practical cases to ensure complete combustion, to allow for the normal variations in combustion and to ensure satisfactory stack conditions for some fuels. The optimum excess air level for maximum boiler efficiency occurs when the sum of the losses due to incomplete combustion and loss due to heat in flue gases is minimum. This level varies with furnace design, type of burner, fuel and process variables. It can be determined by conducting tests with different air fuel ratios.

The Table 2.4 gives the theoretical amount of air required for combustion of various types of fuel.

Table 2.4: Theoretical combustion data – common boiler fuels

Fuel	kg of air req./kg of fuel	kg of flue gas/kg of fuel	m ³ of flue/kg of fuel	Theoretical CO ₂ % in dry flue gas	CO ₂ % in flue gas achieved in practice
Solid Fuels					
Bagasse	3.2	3.43	2.61	20.65	10-12
Coal (bituminous)	10.8	11.7	9.40	18.70	10-13
Lignite	8.4	9.10	6.97	19.40	9 -13
Paddy Husk	4.6	5.63	4.58	19.8	14-15
Wood	5.8	6.4	4.79	20.3	11.13
Liquid Fuels					
Furnace Oil	13.90	14.30	11.50	15.0	9-14
LSHS	14.04	14.63	10.79	15.5	9-14

Typical values of excess air supplied for various fuels are given in Table 2.5.

Table 2.5: Excess air supplied for various fuels

Fuel	Type of Furnace or Burners	Excess Air (% by wt.)
Pulverised coal	Completely water-cooled furnace for slag-tap or dry-ash removal	15-20
	Partially water-cooled furnace for dry-ash removal	15-40
Coal	Spreader stoker	30-60
	Water-cooler vibrating-grate stokers	30-60
	Chain-grate and traveling-grate stokers	15-50
	Underfeed stoker	20-50
Fuel oil	Oil burners, register type	15-20
	Multi-fuel burners and flat-flame	20-30
Natural gas	High pressure burner	5-7
Wood	Dutch over (10-23% through grates) and Hoff t type	20-25
Bagasse	All furnaces	25-35
Black liquor	Recovery furnaces for draft and soda-pulping processes	30-40

Controlling excess air to an optimum level always results in reduction in flue gas losses; for every 1% reduction in excess air there is approximately 0.6% rise in efficiency.

Various methods are available to control the excess air:

- Portable oxygen analysers and draft gauges can be used to make periodic readings to guide the operator to manually adjust the flow of air for optimum operation. Excess air reduction up to 20% is feasible.
- The most common method is the continuous oxygen analyzer with a local read out mounted draft gauge, by which the operator can adjust air flow. A further reduction of 10-15% can be achieved over the previous system.

- The same continuous oxygen analyser can have a remote controlled pneumatic damper positioned, by which the readouts are available in a control room. This enables an operator to remotely control a number of firing systems simultaneously.

The most sophisticated system is the automatic stack damper control, whose cost is really justified only for large systems.

6. Radiation and Convection Heat Loss

The external surfaces of a shell boiler are hotter than the surroundings. The surfaces thus lose heat to the surroundings depending on the surface area and the difference in temperature between the surface and the surroundings.

The heat loss from the boiler shell is normally a fixed energy loss, irrespective of the boiler output. With modern boiler designs, this may represent only 1.5% on the gross calorific value at full rating, but will increase to around 6%, if the boiler operates at only 25 percent output.

Repairing or augmenting insulation can reduce heat loss through boiler walls and piping.

7. Automatic Blow down control

Uncontrolled continuous blow down is very wasteful. Automatic blow down controls can be installed that sense and respond to boiler water conductivity and pH. A 10% blow down in a 15 kg/cm² boiler results in 3% efficiency loss.

Case Study:

The following formula gives the quantity of blow down required:

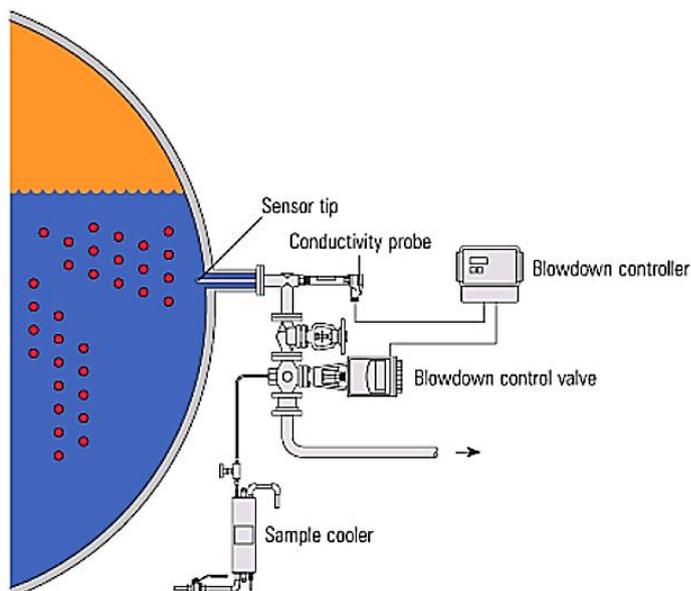


Figure 2.8: Blowdown

$$\text{Blowdown rate} = \frac{F \times S}{(B - F)}$$

Where:

F = Feedwater TDS in parts per million.

B = Required boiler water TDS in parts per million.

S = Steam generation rate in kg/h

The blowdown rate is given in kg/h

A 10 000 kg/h boiler operates at 10 bar g and has a maximum allowable boiler TDS of 2 500 ppm.

Boiler feedwater TDS = 250 ppm.

$$\begin{aligned} \text{Blowdown rate} &= \frac{250 \times 10\,000}{(2\,500 - 250)} \\ &= 1\,111 \text{ kg/h.} \end{aligned}$$

Boiler pressure = 10 bar g

Boiler rating = 10 000 kg/h

Maximum allowable boiler TDS = 2 500 ppm

Feedwater TDS = 250 ppm

Calculated blowdown rate = 1 111 kg/h

To obtain the energy flow in kW:

$$\text{The blowdown rate in kg/s} = \frac{1\,111 \text{ kg/h}}{3\,600}$$

$$\text{The blowdown rate in kg/s} = 0.31 \text{ kg/s}$$

$$\begin{aligned} \text{The amount of energy in each kg} \\ \text{from } h_f^* \text{ at 10 bar g} &= 782 \text{ kJ/kg} \end{aligned}$$

* h_f is the specific enthalpy of water at the saturation temperature - obtained from steam tables.

$$\text{Rate of energy blown down} = 0.31 \text{ kg/s} \times 782 \text{ kJ/kg}$$

$$\text{Rate of energy blown down} = 241 \text{ kW}$$

From the flash vessel ≈ 117 kW

From the heat exchanger = 94 kW

Total energy recovered = 117 kW + 94 kW

Total energy recovered = 211 kW

When energy is recovered from the flash steam and the condensate, 87% of the total energy contained in the original blowdown has been recovered.

In addition, 14% (by mass) of the water has been recovered, making a further contribution to savings.

10.5 Lakh BDT /Year saving is possible. If we invest 20 Lakh tk then Payback period will be 1.9 Year.

8. Reduction of Scaling and Soot losses

In oil and coal-fired boilers, soot build upon tubes acts as an insulator against heat transfer. Any such deposits should be removed on a regular basis. Elevated stack temperatures may indicate excessive soot build-up. Also same result will occur due to scaling on the waterside.

High exit gas temperatures at normal excess air indicate poor heat transfer performance. This condition can result from a gradual build-up of gas-side or waterside deposits. Water side deposits require are view of water treatment procedures and tube cleaning to remove deposits. An estimated 1% efficiency loss occurs with every 4.4 °C increase ion stack temperature.

Stack temperature should be checked and recorded regularly as an indicator of soot deposits. When the flue gas temperature raises about 20°C above the temperature for a newly cleaned boiler, it is time to remove the soot deposits. It is, therefore, recommended to install a dial type thermometer at the base of the stack to monitor the exhaust flue gas temperature.

Every millimetre thickness of soot coating increases the stack temperature by about 55°C. It is also estimated that 3mm of soot can cause an increase in fuel consumption by 2.5%. Periodic off-line cleaning of radiant furnace surfaces, boiler tube banks, economizers and air heaters may be necessary to remove stubborn deposits.

9. Reduction of Boiler Steam Pressure

This is an effective means of reducing fuel consumption, if permissible, by as much as 1 to 2%. Lower steam pressure gives a lowers saturated steam temperature and without stack heat recovery, a similar reduction in the temperature of the flue gas temperature results.

Steam is generated at pressures normally dictated by the highest pressure /temperature requirements for a particular process. In some cases, the process does not operate all the time, and there are periods when the boiler pressure could be reduced. The energy manager should consider pressure reduction carefully, before recommending it. Adverse effects, such as an increase in water carry over from the boiler owing to pressure reduction, may negate any potential saving. Pressure should be reduced in stages, and no more than a 20 percent reduction should be considered.

10. Variable Speed Control for Fans, Blowers and Pumps

Variable speed control is an important means of achieving energy savings. Generally, combustion air control is effected by throttling dampers fitted at forced and induced draft fans. Though dampers are simple means of control, they lack accuracy, giving poor control characteristics at the top and bottom of the operating range. In general, if the load characteristic of the boiler is variable, the possibility of replacing the dampers by a VSD should be evaluated.

11. Effect of Boiler Loading on Efficiency

The maximum efficiency of the boiler does not occur at full load, but at about two-thirds of the full load. If the load on the boiler decreases further, efficiency also tends to decrease. At zero output, the efficiency of the boiler is zero, and any fuel fired is used only to supply the losses. The factors affecting boiler efficiency are:

- As the load falls, so does the value of the mass flow rate of the flue gases through the tubes. This reduction in flow rate for the same heat transfer area reduced the exit flue gas temperatures by a small extent, reducing the sensible heat loss.
- Below half load, most combustion appliances need more excess air to burn the fuel completely. This increases the sensible heat loss.

The net effect of these factors is to produce a load / efficiency curve. It has been generally noticed that the fall in efficiency begins to become serious below about a quarter load, and as far as possible, operation of boilers below this level should be avoided.

12. Proper Boiler Scheduling

Since the optimum efficiency of boilers occurs at 65-85% of full load, it is usually more efficient, on the whole, to operate a fewer number of boilers at higher loads, than to operate a large number at low loads.

13. Boiler Replacement

The potential savings from replacing a boiler depend on the anticipated change in overall efficiency. A change in a boiler can be financially attractive if the existing boiler is:

- old and inefficient
- not capable of firing cheaper substitution fuel
- over or under-sized for present requirements not
- designed for ideal loading conditions

The feasibility study should examine all implications of long-term fuel availability and company growth plans. All financial and engineering factors should be considered. Since boiler plants traditionally have a useful life of well over 25 years, replacement must be carefully studied.

2.8 Case Study

Installing Boiler Economiser

A paper mill retrofitted an economizer to existing boiler. The general specification of the boiler is given below:

Boiler capacity (T/h)	Feed water Temp. ($^{\circ}$ C)	Steam pressure (bar)	Fuel oil
8	110	18	No.6

The thermal efficiency of the boiler was measured and calculated by the indirect method using flue gas analyzer and data logger. The result is summarized below:

Thermal efficiency	:	80.99%
Flue gas temperature	:	315 ⁰ C
CO ₂ %	:	13
CO (PPM)	:	167

The temperature in the flue gas is in the range of 315 to 320⁰C. The waste heat in the flue gas is recovered by installing an economizer, which transfers waste heat from the flue gases to the boiler feed water. This resulted in a rise in feed water temperature by about 26⁰C.

Basic Data

Average quantity of steam generated	:	5.04T/h
Average flue gas temperature	:	315 ⁰ C
Average steam generation / kg of fuel oil	:	16.05kg
Feed water in let temperature	:	110 ⁰ C
Fuel oil supply rate	:	314kg/h
Flue gas quantity	:	17.4 kg/kg of fuel

Cost Economics

Quantity of flue gases	:	5463.6kg/h
Quantity of heat available in the flue gases	:	5463.6 x0.25 x(315-200) =157078kCal/h
Rise in the feed water temperature	:	26 ⁰ C
Heat required for pre-heating the feed water	:	5040X1.065X26 =139557kCal/h
Saving in terms of HSD fuel	:	139557 kCal/h /10000 kCal/kg = 14kg/h
Annual operating hours	:	8600
Annual savings of fuel oil	:	120400kg

Conclusion

Through recovery of waste heat by installation of an economizer, the paper mill was able to save14kg/hr of HSD fuel, which amounts to about 1, 20,400 kg of fuel oil per annum.

Example 2.1

The efficiency of a boiler on GCV basis is 85%. The fuel contains 1.0 % moisture and 12 % hydrogen. The GCV of fuel is 10,500 Kcal/kg. What is the boiler efficiency on the basis of net calorific value?

Solution:

$$\text{NCV} = \text{GCV} - \left[9 \times \frac{\% \text{ of H}_2 \text{ in fuel}}{100} + \frac{\% \text{ of moisture in fuel}}{100} \right] \times 584$$

$$\begin{aligned} \text{NCV} &= 10500 - \left[9 \times \frac{12}{100} + \frac{1.0}{100} \right] \times 584 \\ &= 10500 - [9 \times 0.12 + 0.01] \times 584 \\ &= 10500 - 636.56 \\ &= 9863.44 = 9863 \text{ kCal /kg} \end{aligned}$$

$$\begin{aligned} \text{Boiler efficiency on NCV} &= \frac{85}{9863} \times 10500 \\ &= 90.5\% \end{aligned}$$

Example 2.2

A textile plant is using furnace oil as fuel for firing in the boiler, generating steam on an average of 30T/hr. The unit has decided to switch over to natural gas as fuel to concern for emissions reduction. The boiler feed water temperature is 60°C and the enthalpy of steam is 660 kCal/kg. The other data are as under:

Furnace oil:

GCV of furnace oil	: 10200kCal/kg
%Carbon in furnace oil	: 84
Efficiency of furnace oil	: 82%

Natural Gas:

Gross Calorific value of Natural gas	: 9500kCal/ (Sm ³)
Density of natural gas	: 0.8kg/Sm ³
% carbon in natural gas	: 74
Annual operating hrs.	: 8000
Efficiency of natural gas boiler	: 86%

Calculate the Reduction in GHG emissions?

Solution:

Furnace oil fired boiler:

$$\text{Kg of CO}_2/\text{kg of oil} = 0.84 \times (44/12) = 3.08$$

$$\text{Heat output of boiler} = 30000 \times (660 - 60) = 18 \text{ million kCal/hr.}$$

$$\text{Heat input to boiler} = 18 / 0.82 = 21.95 \text{ million kCal/hr}$$

Furnace oil consumption = $21.95 \times 10^6 / 10200 = 2.152 \text{ T/hr}$.

CO₂ emission with furnace oil = $2,152 \times 3.08 = 6.628 \text{ T/hr}$.

Gas fired Boiler:

Kg of CO₂/kg of gas = $0.74 \times (44/12) = 2.71$

Heat input to boiler = $18/0.86 = 20.93$ million kCal/hr

Natural gas consumption = $20.93 \times 10^6 / 9500 = 2203 \text{ Sm}^3/\text{hr}$

= $2203 \times 0.8 / 1000 = 1.7624 \text{ T/hr}$.

CO₂ emission with natural gas = $1.7624 \times 2.71 = 4.776 \text{ T/hr}$.

Annual CO₂ emission reduction = $(6.628 - 4.776) \times 8000 = 14816 \text{ T/y}$

Example 2.3:

Oil fired Boiler is generating 100 TPH of steam at 85% efficiency, operating 330 days in a year. Management has installed a water treatment plant at Bangladeshi Taka (BDT) 1.16 Crore investment for reducing the TDS in boiler feed from 450 ppm to 150 ppm. The maximum permissible limit of TDS in the boiler is 3000 ppm and make up water is 10%. Temperature of blow down water is 175°C and boiler feed water temperature is 45°C. Calorific value of Fuel oil is 1200 kCal/kg.

Calculate the payback period if the cost of fuel is 23150 BDT / Ton.

Solution:

Blow down % = $(\text{Feed water TDS} \times \% \text{ make up water}) \times 100$
(Maximum permissible TDS in boiler water – Feed water TDS)

Initial blow down = $450 \times 10 / (3000 - 450)$
= 1.76 %

Improved blow down = $150 \times 10 / (3000 - 150)$
= 0.53 %

Reduction in blow down = $1.76 - 0.53$
= 1.24 %

Reduction in blow down = $1.24 \times 100 \times 1000 / 100$
= 1238 kg/hr

Specific heat of water is 1 kCal/kg°C

Heat savings = $m \times C_p \times (T_1 - T_2)$ = $1238 \times 1 \times (175 - 45)$
= 160991 kcal/hr

Fuel oil saving = $160991 / (1200 \times 0.85)$ = 157.83 kg/hr

Annual fuel oil savings = $157.83 \times 24 \times 330 / 1000$
= 1250 MT / annum

Fuel oil cost savings = 1250×23150 = BDT 289.4 Lakh

Investment on water treatment plant	= BDT1.16Crore
Payback period	= 1.16 / 2.894
Payback period	= 0.4 years (or) 5 months

Example 2.4

An oil fired boiler is generating 30 T/hr Steam and operates for 8000 hrs/year. The TDS in boiler feed water was reduced from 500 ppm to 200 ppm. The maximum permissible limit of TDS in the boiler is 3000 ppm and make up water is 10%.

Temperature of the blow down water is 170°C and boiler feed water temperature is 40°C. GCV of fuel is 10000 kcal/kg and efficiency of the boiler is 80%.

Calculate the savings in fuel oil per annum due to reduction in the blow down.

Ans:

Blow down % = Feed water TDS * % make up water * 100 / (maximum permissible TDS in boiler water – Feed water TDS)

Initial blow down = $500 * 10 / (3000 - 500)$

Initial blow down = 2.0 %

Improved blow down = $200 * 10 / (3000 - 200)$

Improved blow down = 0.71 %

Reduction in blow down = 2.0 – 0.71

Reduction in blow down = 1.29 %

Reduction in blow down = $1.29 * 30 * 1000 / 100$

Reduction in blow down = 387 kg/hr

Heat savings = $387 * 1 * (170 - 40)$

Heat savings = 50310 kcal/hr

Fuel Oil saving = $50310 / (10000 * 0.8) = 6.28 \text{ kg/hr}$

= $6.28 * 8000 \text{ hr} = \mathbf{50.24 \text{ MT / annum}}$

Chapter 3: Steam System

3.1 Introduction

Steam has been a popular mode of conveying energy since the industrial revolution. Steam is used for generating power and also used in process industries such as sugar, paper, fertilizer, refineries, petrochemicals, chemical, food, synthetic fibre and textiles. The following characteristics of steam make it so popular and useful to the industry:

- Highest specific heat and latent heat
- Highest heat transfer coefficient
- Easy to control and distribute
- Cheap and inert

3.2 Properties of Steam

Water can exist in the form of solid, liquid and gas as ice, water and steam respectively. If heat energy is added to water, its temperature rises until a value is reached at which the water can no longer exist as a liquid. We call this the "saturation" point and with any further addition of energy, some of the water will boil off as steam. This evaporation requires relatively large amounts of energy and while it is being added, the water and the steam released are both at the same temperature. Equally, if steam is made to release the energy that was added to evaporate it, then the steam will condense and water at same temperature will be formed.

Liquid Enthalpy

Liquid enthalpy is the "Enthalpy"(heat energy) in the water when it has been raised to its boiling point to produce steam, and is measured in kCal/kg, its symbol is h_f . (also known as "Sensible Heat").

The heat required to change the temperature of a substance is called its **sensible heat**. If 1kg of water in a vessel at 25°C i.e. containing heat value of 25 kCal is heated by adding 75 kCal, the water is brought to boiling point of 100°C.

Enthalpy of Evaporation (Heat Content of Steam)

The Enthalpy of evaporation is the heat energy to be added to the water (when it has been raised to its boiling point) in order to change it into steam. There is no change in temperature, the steam produced is at the same temperature as the water from which it is produced, but the heat energy added to the water changes its state from water into steam at the same temperature.

When the steam condenses back in to water, it gives up its enthalpy of evaporation, which it had acquired on changing from water to steam. The enthalpy of evaporation is measured in kCal/kg. Its symbol is h_{fg} . Enthalpy of evaporation is also known as latent heat.

To change the water to steam an additional 540kcal would be required. This quantity of heat required to change a chemical from the liquid to the gaseous state is called latent heat.

For a boiler is operating at a pressure of 8kg/cm^2 , steam saturation temperature is 170°C , and steam enthalpy or total heat of dry saturated steam is given by:

$$h_f + h_{fg} = 171.35 + 489.46 = 660.81 \text{ kCal/kg.}$$

If the same steam contains 4% moisture, the total heat of steam is given by:

$$171.35 + 0.96 \times 489.46 = 641.23 \text{ kCal/kg}$$

The temperature at which water boils, also called as boiling point or saturation temperature increases as the pressure increases. When water under pressure is heated its saturation temperature rises above 100°C . From this, it is evident that as the steam pressure increases, the usable heat energy in the steam (enthalpy of evaporation), which is given up when the steam condenses, actually decreases. The total heat of dry saturated steam or enthalpy of saturated steam is given by sum of the two enthalpies $h_f + h_{fg}$ (Refer Table 3.1 and figure 3.1). When the steam contains moisture, the total heat of steam will be $h_g = h_f + q h_{fg}$ where q is the dryness fraction.

The temperature of saturated steam is the same as the water from which it is generated and corresponds to a fixed and known pressure. Superheat is the addition of heat to dry saturated steam without increase in pressure. The temperature of superheated steam, expressed as degrees above saturation corresponding to the pressure, is referred to as the degrees of superheat.

The Steam Phase Diagram

The data provided in the steam tables can also be expressed in a graphical form. Figure 3.1 illustrates the relationship between the enthalpy and the temperature at various different pressures, and is known as a phase diagram.

As water is heated from 0°C to its saturation temperature, its condition follows the saturated liquid line until it has received all of its liquid enthalpy, h_f , (A - B).

If further heat continues to be added, it then changes phase to saturated steam and continues to increase in enthalpy while remaining at saturation temperature, h_{fg} , (B - C).

As the steam/water mixture increases in dryness, its condition moves from the saturated liquid line to the saturated vapour line. Therefore at a point exactly halfway between these two states, the dryness fraction (X) is 0.5. Similarly, on the saturated vapour line the steam is 100% dry.

Once it has received all of its enthalpy of evaporation, it reaches the saturated vapour line. If it continues to be heated after this point, the temperature of the steam will begin to rise as superheat is imparted (C - D).

The saturated liquid and saturated vapour lines enclose a region in which a steam/water mixture exists - wet steam. In the region to the left of the saturated liquid line only water exists, and in the region to the right of the saturated vapour line only superheated steam exists.

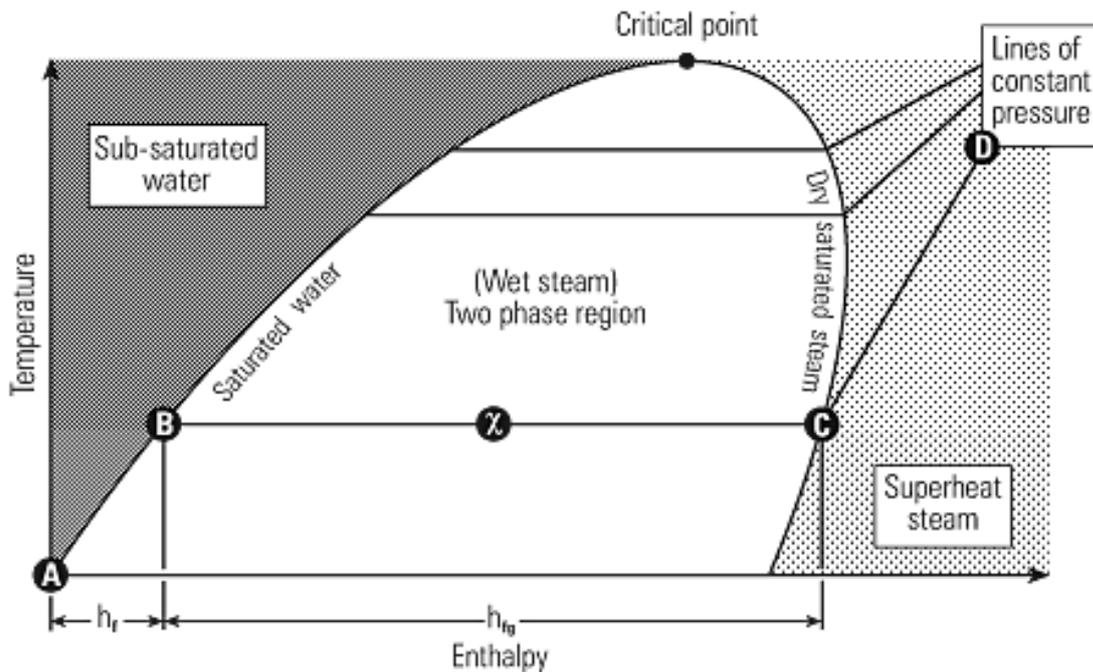


Figure 3.1: Steam Phase Diagram

The point at which the saturated liquid and saturated vapour lines meet is known as the critical point. As the pressure increases towards the critical point the enthalpy of evaporation decreases, until it becomes zero at the critical point. This suggests that water changes directly into saturated steam at the critical point.

Above the critical point only gas may exist. The gaseous state is the most diffuse state in which the molecules have an almost unrestricted motion, and the volume increases without limit as the pressure is reduced.

The critical point is the highest temperature at which liquid can exist. Any compression at constant temperature above the critical point will not produce a phase change. Compression at constant temperature below the critical point however, will result in liquefaction of the vapour as it passes from the superheated region into the wet steam region. The critical point occurs at 374.15°C and 221.2 bar (a) for steam. Above this pressure the steam is termed supercritical and no well-defined boiling point applies.

Table 3.1: Extract From the Steam Tables

Pressure (kg/cm ²)	Temperature °C	Enthalpy in kcal/kg			Specific volume (m ³ /kg)
		Water (h _f)	Evaporation (h _{fg})	Steam (h _g)	
1	100	100.09	539.06	639.15	1.673
2	120	119.92	526.26	646.18	0.901
3	133	133.42	517.15	650.57	0.616
4	143	143.70	509.96	653.66	0.470
5	151	152.13	503.90	656.03	0.381
6	158	159.33	498.59	657.92	0.321
7	164	165.67	493.82	659.49	0.277
8	170	171.35	489.46	660.81	0.244

3.3 Steam Distribution System

The steam distribution system is the essential link between the steam generator and the steam user. Whatever the source, an efficient steam distribution system is essential if steam of the right quality and pressure is to be supplied, in the right quantity, to the steam using equipment. Installation and maintenance of the steam system are important issues, and must be considered at the design stage.

As steam condenses in a process, flow is induced in the supply pipe. Condensate has a very small volume compared to the steam, and this causes a pressure drop, which causes the steam to flow through the pipes. The steam generated in the boiler must be conveyed through pipe work to the point where its heat energy is required. Initially there will be one or more main pipes, or 'steam mains', which carry steam from the boiler in the general direction of the steam using plant. Smaller branch pipes can then carry the steam to the individual pieces of equipment. A typical steam distribution system is shown in Figure 3.2.

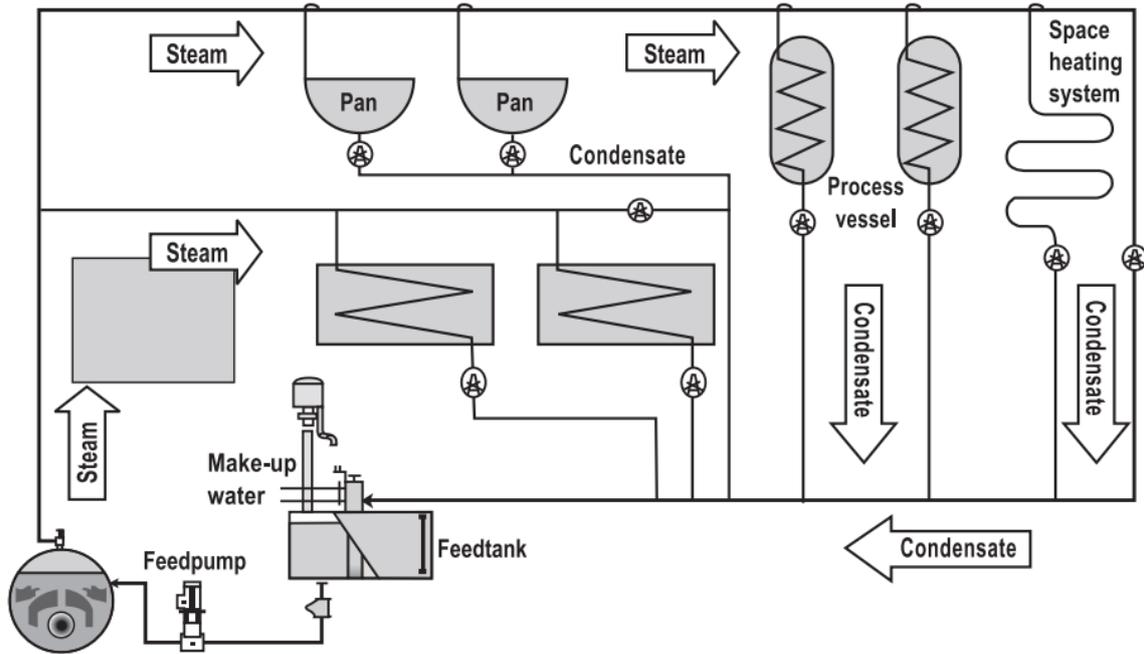


Figure 3.2: Steam distribution System

The working pressure

The distribution pressure of steam is influenced by a number of factors, but is limited by:

- The maximum safe working pressure of the boiler
- The minimum pressure required at the plant

As steam passes through the distribution pipe work, it will inevitably lose pressure due to:

- Frictional resistance within the pipe work
- Condensation within the pipe work as heat is transferred to the environment.

Therefore allowance should be made for this pressure loss when deciding upon the initial distribution pressure. Having determined the working pressure, the following factors have to be considered in the design of a good Steam Distribution System.

The following factors have to be considered in the design of a good Steam Distribution System.

- General Layout
- Pipe sizing and Design
- Pressure Reducing and De-superheating Station
- Air Venting (as given in sub-section 3.7)
- Steam Pipe Insulation (as given in sub-section 3.7)

General Layout

General layout and location of steam consuming equipment is of great importance in efficient distribution of steam. Steam pipes should be laid by the shortest possible distance rather than to follow a building layout or road etc. However, this may come in the way of aesthetic design and architect's plans and a compromise may be necessary while laying new pipes.

Steam Pipe Sizing and Design

Any modification and alteration in the existing steam piping, for supplying higher quality steam at right pressure and quantity must consider the following points:

Pipe Sizing

The objective of the steam distribution system is to supply steam at the correct pressure to the point of use. It follows therefore, that pressure drop through the distribution system is an important feature.

Proper sizing of steam pipelines help in minimizing pressure drop. The velocities for various types of steam are:

Superheated	50-70 m/sec
Saturated	30-40 m/sec
Wet or Exhaust	20-30 m/sec

For fluid flow to occur, there must be more energy at Point 1 than Point 2 (Figure 3.2). The difference in energy is used to overcome frictional resistance between the pipe and the flowing fluid.

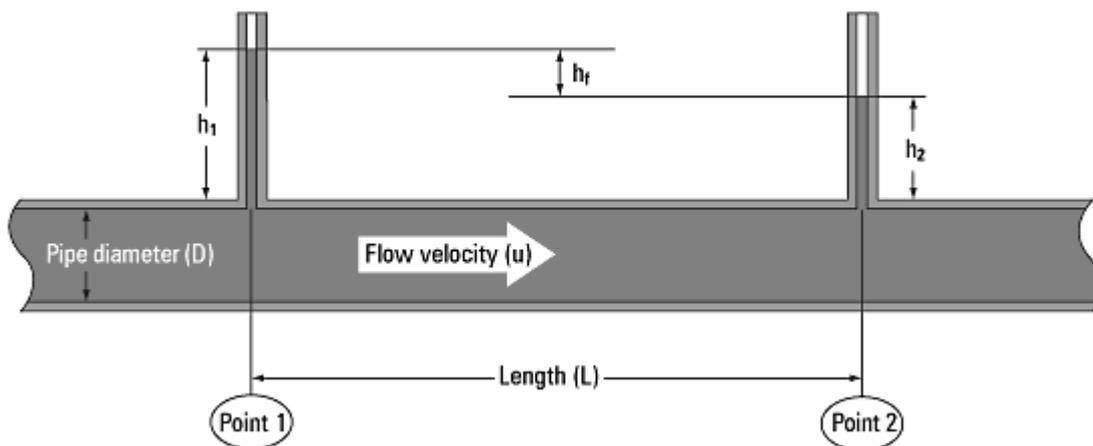


Figure 3.2: Pressure drop in steam pipes

This is illustrated by the equation,

$$h_f = \frac{4 f L u^2}{2 g D}$$

Where,

h_f = Head loss to friction (m)

f = Friction factor (dimensionless), usually obtained from charts

L = Length (m)

u = Flow velocity (m/s)

g = Gravitational constant (9.81 m/s²)

D = Pipe diameter (m)

Example 3.1 Pressure drop calculation

Determine the difference in pressure between two points 1 km apart in a 150 mm bore horizontal pipework system. The water flow rate is 45 m³/h at 15°C and the friction factor for this pipe is taken as 0.005.

$$\text{Velocity (m/s)} = \frac{\text{Volume flowrate (m}^3/\text{s)}}{\text{Cross sectional area (m}^2\text{)}}$$

$$\text{Velocity (m/s)} = \frac{45(\text{m}^3/\text{hr}) \times 4}{3600\text{ s/hr} \times \pi \times 0.15^2}$$

$$\text{Velocity} = 0.71\text{ m/s}$$

$$h_f = \frac{4 f L u^2}{2 g D}$$

$$h_f = \frac{4 \times 0.005 \times 1000 \times 0.71^2}{2 \times 9.81 \times 0.15}$$

$$h_f = 3.43\text{ m} \approx 0.34\text{ bar}$$

Example 3.2 Determining the pipe size

A process requires 5000 kg/h of dry saturated steam at 7 kg/cm² (g). For the flow velocity not to exceed 25 m/s, determine the pipe size.

$$\text{Flow velocity} = 25\text{ m/s}$$

$$\text{Specific volume at } 7\text{ kg/cm}^2 = 0.24\text{ m}^3/\text{kg}$$

$$\text{Mass flowrate} = 5000\text{ kg/hr or } 1.389\text{ kg/s}$$

$$\text{Volumetric flowrate} = 1.389 \times 0.24$$

$$= 0.333\text{ m}^3/\text{s}$$

Therefore, using:

$$\begin{aligned} \text{Cross sectional area} &= \frac{\text{Volumetric flowrate}}{\text{Flowvelocity}} \\ \frac{\Pi \times D^2}{4} &= \frac{\text{Volumetric flowrate}}{\text{Flowvelocity}} \end{aligned}$$

$$D^2 = \frac{4 \times \text{Volumetric flowrate}}{\Pi \times \text{Flowvelocity}}$$

$$D = \sqrt{\frac{4 \times \text{Volumetric flowrate}}{\Pi \times \text{Flowvelocity}}}$$

$$D = \sqrt{\frac{4 \times 0.333}{\Pi \times 25}}$$

$$D = 0.130 \text{ m or } 130 \text{ mm}$$

Since the steam velocity must not exceed 25 m/s, the pipe size must be at least 130 mm; the nearest commercially available size, 150 mm, would be selected.

In practice whether for water pipes or steam pipes, a balance is drawn between pipe size and pressure loss. The steam piping should be sized, based on permissible velocity and the available pressure drop in the line. Selecting a higher pipe size will reduce the pressure drop and thus the energy cost. However, higher pipe size will increase the initial installation cost. By use of smaller pipe size, even though the installation cost can be reduced, the energy cost will increase due to higher-pressure drop. It is to be noted that the pressure drop change will be inversely proportional to the 5th power of diameter change. Hence, care should be taken in selecting the optimum pipe size.

Guide for proper drainage and layout of steam lines:

- 1. The steam mains should be run with a falling slope of not less than 125 mm for every 30 metres length in the direction of the steam flow.*
- 2. Drain points should be provided at intervals of 30-45 metres along the main.*
- 3. Drain points should also be provided at low points in the mains and where the steam main rises. Ideal locations are the bottom of expansion joints and before reduction and stop valves.*
- 4. Drain points in the main lines should be through an equal tee connection only.*
- 5. It is preferable to choose open bucket or TD traps on account of their resilience.*
- 6. The branch lines from the mains should always be connected at the top. Otherwise, the branch line itself will act as a drain for the condensate.*
- 7. Insecure supports as well as an alteration in level can lead to formation of water pockets in steam, leading to wet steam delivery. Providing proper vertical and support hangers helps overcome such eventualities.*
- 8. Expansion loops are required to accommodate the expansion of steam lines while starting from cold.*
- 9. To ensure dry steam in the process equipment and in branch lines, steam separators can be installed as required.*

Pressure Reducing De-superheating Station

A reduction in steam pressure through a pressure reducing valve (PRV) is an isenthalpic process. Saturated steam when reduced to a lower pressure results in superheated steam. Since the process requires only saturated steam, de-superheating is often required, to compensate for superheat gained in PRV application due to isenthalpic expansion. Pressure reduction and de-superheating of steam or conditioning of steam is done in many process industries to suit process requirement. This is due to the fact that steam is produced in a boiler economically at higher pressure and temperature. Generally, the temperature of the steam after de-superheating will be closer to saturation temperature for heat transfer applications.

The de-superheating of steam is done by spraying water through a spray nozzle into a pipe. Normally, the de-superheating is done by automatic control system, using a control device for spraying water, which takes the feedback from the temperature control loop.

Proper Air Venting

When steam is first admitted to a pipe after a period of shutdown, the pipe is full of air. Further, amounts of air and other non-condensable gases will enter with the steam, although the proportions of these gases are normally very small compared with the steam. When the steam condenses, these gases will accumulate in pipes and heat exchangers. Precautions should be taken to discharge them. The consequence of not removing air is a lengthy warming up period, reduction in plant efficiency and process performance. Air in the steam system will also affect the system temperature. Air will exert its own pressure within the system, which will add to the pressure of the steam to give a total pressure. Therefore, the actual steam pressure and temperature of the steam-air mixture will be lower than that shown by pressure gauge.

Automatic air vents for steam systems (which operate on the same principle as thermostatic steam traps) should be fitted above the condensate level so that only air or steam-air mixtures can reach them. The best location for them is at the end of the steam mains as shown in Figure 3.4.

In addition to air venting at the end of a main, air vents should also be fitted:

In parallel with an inverted bucket trap or, in some instances, a thermodynamic trap. These traps are sometimes slow to vent air on start-up.

In awkward steam spaces (such as at the opposite side to where steam enters a jacketed pan).

Where there is a large steam space (such as an autoclave), and a steam/air mixture could affect the process quality.

Example 3.3 Effect of air in steam system

If 20% of air is entrained in a steam system at 5 kg/cm² (g) then the effect of air will be as follows

Steam quality = 80% Steam + 20% Air

Pressure = 0.80 x 5 + 0.20 x 5 = 4 kg/cm² (g) + 1 kg/cm² (g) (steam) (air)

Temp. of steam at 5 kg / cm² (g) = 158 °C

Temp. of vapour mixture is = 152 °C (equivalent to steam at 4 kg/cm² (g))

Steam Pipe Insulation

The insulation of steam conveying pipes and the steam consuming equipment is very essential to retard the flow of heat from the system to the environment. Broadly the purpose of steam pipe insulation is as follows

- Conserve energy by reducing heat loss
- Facilitate temperature control of a process
- Prevent condensation of steam
- Prevent or reduce damage to pipe from exposure to fire or corrosive atmospheres
- Control surface temperature for personal protection and comfort

The following table 3.2 indicates the effect of insulating bare pipes

Table 3.2 Effect of Insulation on Steam Pipes

Pipe Size, inch	Economic Insulation Thickness, mm	Radiation Losses* (kW/m)	
		Insulated	Uninsulated
1/2	15	125	692
2	25	243	1820
4	40	298	2942
12	50	588	7614

* Comparison of Radiation Losses (Pipe Surface Temperature at 150 °C)

Heat can be lost due to radiation from steam pipes. As an example while lagging steam pipes, it is common to see leaving flanges uncovered. An uncovered flange is equivalent to leaving 0.6 metre of pipe line unlagged. If a 0.15 m steam pipe diameter has 5 uncovered flanges, there would be a loss of heat equivalent to wasting 5 tons of coal or 3000 litres of oil a year. This is usually done to facilitate checking the condition of flange but at the cost of considerable heat loss. The remedy is to provide easily detachable insulation covers, which can be easily removed when necessary. The various insulating materials used are cork, Glass wool, Rock wool and Asbestos.

Effect of insulation of flanges: 12 Flanges of 150 mm diameter. Heat loss in the following 2 cases:

Case (I) — Bare flanges

Case (II) — Flanges with 50 mm insulation and aluminium cladding

Table 3.3 Loss Comparison between Case (I) and Case (II)

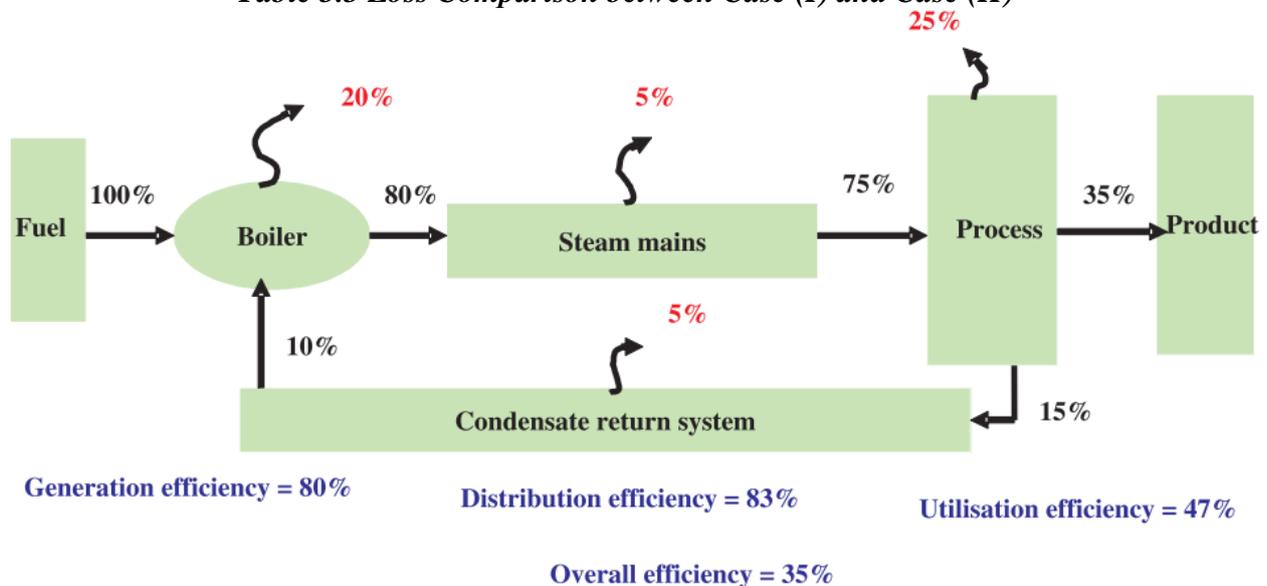


Figure 3.5: Steam System Efficiency

When viewed from the standpoint of being a heat medium, steam has superior properties not offered by other heat mediums. Steam provides for even and rapid heating.

In the case of saturated steam, if the steam pressure is known, then the steam temperature may be determined. Pressure changes instantaneously within a space. When saturated steam condenses, it condenses at the saturation temperature, and the saturated water (condensate) formed is of the same temperature as the saturated steam. This means that if the pressure at the heat transfer surface (the jacket or coil interior of the equipment) is held at a constant, continuous heating will be able to take place at the same temperature at every part of the heat transfer surface.

The amount of the heat transfer is indicated by the heat transfer coefficient (= film coefficient of heat transfer). The unit is [W/m² K].

$W = J/\text{sec}$, so if heat exchange takes place on the same heat transfer surface area and with the same temperature difference, the larger the heat transfer rate, the shorter the time required for heating.

The rough values for the heat transfer rates of hot water and steam are as follows:

- The rate at which heat is transferred to the heat transfer surface of a heat exchanger using hot water as the heat source:
1000 — 6000 [W/m² K]
- The rate at which heat is transferred to the heat transfer surface of a heat exchanger using steam as the heat source:
6000 — 15000 [W/m² K]

In actual heating situations, the heat transfer process will be a combination of the mechanism of heat transfer within the walls of the heat exchanger and the mechanism of heat transfer from the wall surface of the heat exchanger to the product being heated.

Steam provides rapid heating because the transfer of heat caused by the process of condensing. The latent heat contained in steam is released in the instant the steam condenses into liquid phase. The amount of latent heat released is 2 – 5 times greater than the amount of sensible heat in the hot water (saturated water) after condensation. This latent heat is released instantaneously and is transferred by means of a heat exchanger to the product being heated.

In contrast, hot water and oil are used in convective heating, which does not involve a phase change. Instead, the heat medium reduces its own temperature in order to transfer heat to the product being heated. A mainstream in industry is the use of forced-convection by means of equipment such as a pump to create the flow against the heat transfer surface.

Example 3.4 Steam Utilization

A milk evaporator uses a steam jacketed kettle, in which milk is batch-processed at atmospheric pressure. The kettle has a 680 kg per batch capacity. Milk is heated from a temperature of 26 °C to 100 °C, where 25% of its mass is then driven off as vapor. Determine the amount of 1 kg/cm² (g) steam required per batch, not including the heating of the kettle itself. (Specific heat of milk is 0.90 kcal/kg°C). The latent heat of steam at 1 kg/cm² is 525 kcal/kg.

Solution:

$$\begin{aligned}\text{Quantity of water evaporated from milk} &= 680 \times 0.25 \\ &= 170 \text{ kg /batch}\end{aligned}$$

Heat required to raise the temperature of milk from 26 to 100 °C

$$\begin{aligned}&= 680 \times 0.9 \times (100 - 26) \\ &= 45288 \text{ kcal/batch}\end{aligned}$$

$$\begin{aligned}\text{Amount of heat required to evaporate 170 kg of water} \\ &= 170 \times 540\end{aligned}$$

= 91800 kcal/batch

Total heat required = 45288 + 91800
= 137088 kcal/batch

Quantity of steam required = 137088/525
= 261 kg/batch

Thermo compressor

In many of the steam utilization equipment where condensate comes out at high pressure, a major portion of it flashes into low pressure steam which goes wasted. Using a thermo compressor (Figure 3.3) it becomes feasible to compress this low pressure steam by high pressure steam and reuse it as a medium pressure steam in the process. The major energy in steam is in its latent heat value and thus thermo compressing would give a large improvement in waste heat recovery.

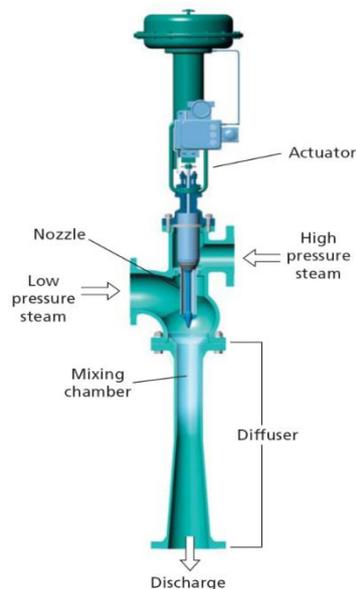


Figure 3.3: Thermo Compressor

Thermo compressors are designed to accurately mix lower-pressure steam with higher-pressure steam. The higher-pressure motive steam entrains the lower pressure steam and increases its pressure. The motive steam is introduced through the nozzle of the thermo compressor. As the nozzle opens, the high velocity motive steam draws the lower-pressure steam into the thermo compressor body. An exchange of momentum occurs as the steam flows are mixed and the mixed flow is accelerated to high velocity with a uniform profile in the mixing chamber of the thermo compressor. As the mixed flow enters the diffuser section, the diffuser flow area gradually increases to allow the velocity of the mixed flow to be reduced. As the velocity is reduced, the steam pressure increases. At the end of the diffuser, the discharge steam pressure is higher than the lower-pressure suction flow entering the thermo compressor. A typical application is in evaporators where the boiling steam is recompressed and used as heating steam.

Advantages of thermo compressors

- No condensation losses take place
- Thermal efficiency of the system is extremely high
- Entrainment of low pressure steam results in substantial savings
- No moving parts and hence maintenance needs are minimal
- No major operational charges
- Low space requirement
- Insensitive to fouling
- High operating reliability

Dryers

Drying is a process by which a liquid (commonly water) is removed from a material. This is usually achieved by applying heat, typically steam and/or the flow of carrier gas (commonly air) through or over the surface of the material (Figure 3.4). The objective of drying is to form a product that meets a water-content specification, so the amount of water removed depends on the desired product.

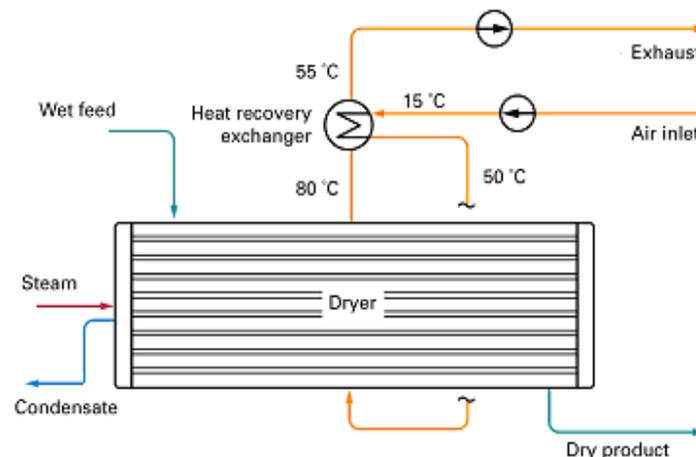


Figure 3.4: Hot Air Dryer Using Steam

The basic drying energy requirement is the latent heat needed to evaporate the water. Clearly, this depends on the amount of water being evaporated. In most cases, the product material, the carrier gas and the equipment also need to be hot. So the total energy required includes:

- Heat leaving the dryer in the exhaust flow. This includes the latent heat of the water evaporated, but the sensible heat of the hot gas can also be significant.
- Heat lost from equipment and ducting. However good the insulation, there is always some heat loss.
- Heat leaving the dryer as hot product.
- Motive power for fans and conveyors.

Common factors resulting in excessive energy use

- Excessive drying load – for example, unnecessarily wet feed material, or off-specification product that needs to be reprocessed
- Excessive airflows
- Unnecessarily hot exhaust flow
- Hot air leaks
- Poor insulation
- Excessive fan power (for example, over specified fans restricted by dampers)
- Steam system inefficiencies

Example 3.5 Heat Energy in Air Drying

A food containing 80% water is to be dried at 100 °C down to moisture content of 10%. If the initial temperature of the food is 21 °C, calculate the quantity of heat energy required per unit weight of the original material, for drying under atmospheric pressure. The latent heat of vaporization of water at 100°C and at standard atmospheric pressure is 2257 kJ/kg. The specific heat capacity of the food is 3.8 kJ/kg°C and of water is 4.186 kJ/kg °C. Find also the energy requirement/kg water removed.

Calculating for 1 kg food

Initial moisture = 80%

800 g moisture is associated with 200 g dry matter.

Final moisture = 10 %,

100 g moisture is associated with 900 g dry matter,

Therefore $(100 \times 200)/900 \text{ g} = 22.2 \text{ g}$ moisture are associated with 200 g dry matter.

1 kg of original matter must lose $(800 - 22) \text{ g}$ moisture = 778 g = 0.778 kg moisture.

Heat energy required for 1 kg original material

= heat energy to raise temperature to 100°C + latent heat to remove water

= $(100 - 21) \times 3.8 + 0.778 \times 2257$

= $300.2 + 1755.9$

= 2056 kJ.

Energy/kg water removed, as 2056 kJ are required to remove 0.778 kg of water

= $2056/0.778$

= 2643 kJ.

Steam is often used to supply heat to air or to surfaces used for drying. In condensing, steam gives up its latent heat of vaporization; in drying, the substance being dried must take up latent heat of vaporization to convert its liquid into vapour, so it might be reasoned that 1 kg of steam condensing will produce 1 kg of vapour, neglecting minor losses.

3.5 Proper Selection, Operation and Maintenance of Steam Traps

The purpose of installing the steam traps is to obtain fast heating of the product and equipment by keeping the steam lines and equipment free of condensate, air and non-

condensable gases. A steam trap is a valve device that discharges condensate and air from the line or piece of equipment without discharging the steam.

Functions of Steam Traps

The three important functions of steam traps are:

- To discharge condensate as soon as it is formed
- Not to allow steam to escape.
- To be capable of discharging air and other incondensable gases.

Types of Steam Traps

The steam traps are classified as follows.

Table 3.4 Steam trap classification

Group	Principle	Sub-group
Mechanical trap	Difference in density between steam and condensate.	Bucket type a) Open bucket b) Inverted bucket (with lever, without lever) c) Float type d) Float with lever e) Free Float
Thermodynamic trap	Difference properties in thermodynamic between steam and condensate	a) Disctype b) Orifice type
Thermostatic trap	Difference in temperature between steam and condensate	a) Bimetallic type b) Metal expansion type.

Some of the important traps in industrial use are explained as follows:

Inverted Bucket

The inverted bucket trap is a mechanically actuated model that uses an upside down bucket as a float. The bucket, connected to an outlet valve through a mechanical linkage, sinks when condensate fills the steam trap, opening the outlet valve. The bucket floats when steam enters the trap, closing the valve (Figure3.5).

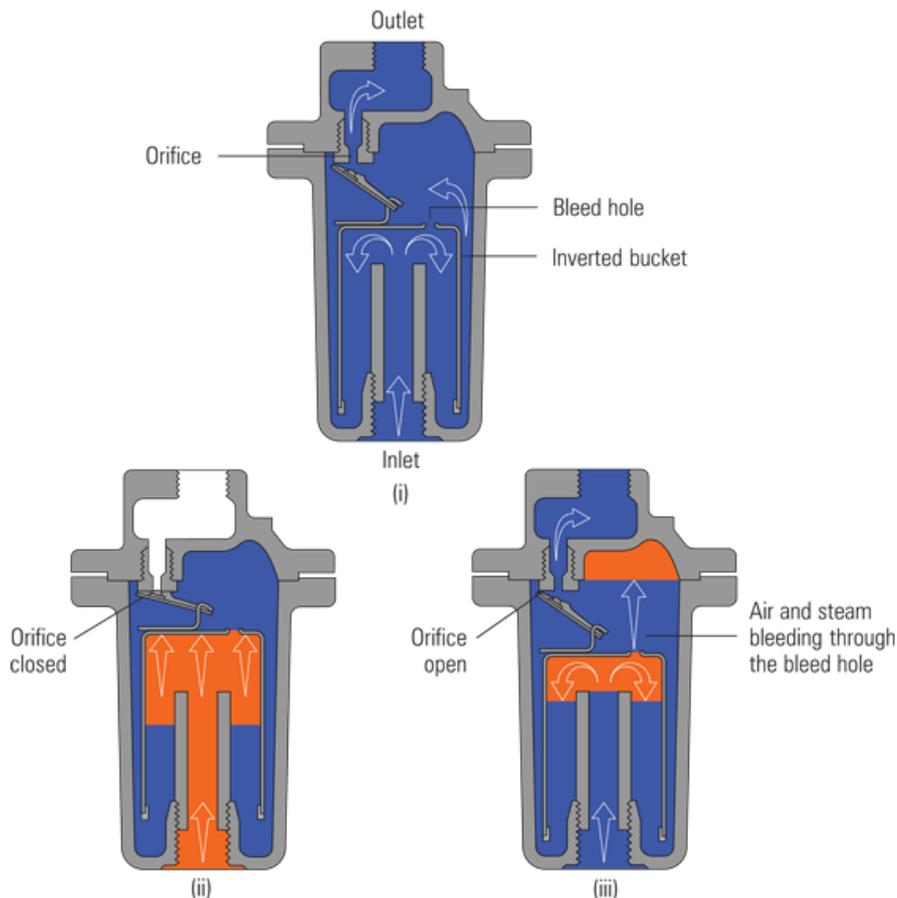


Figure 3.5: Inverted bucket trap

Inverted bucket traps, as a group, are capable of handling a wide range of steam pressures and condensate capacities. However, each specific steam trap handles a very narrow range. An inverted bucket trap designed for 8.5 Kg/cm² service operates at pressures below this; however, its capacity is so diminished that it may "back up" a system with unwanted condensate.

It is important to correlate the pressure rating and size with a specific application. The inverted bucket trap can be a very economical solution for low-to-medium pressures and medium capacity applications such as plant heating and light-duty processes. When handling high pressures and capacities, these traps become large, expensive, and difficult for personnel to handle.

Advantages of the inverted bucket steam trap

- The inverted bucket steam trap can be made to withstand high pressures.
- Like a float-thermostatic steam trap, it has a good tolerance to water hammer conditions.
- Can be used on superheated steam lines with the addition of a check valve on the inlet.
- Failure mode is usually open, so it's safer on those applications that require this feature, for example turbine drains.

Disadvantages of the inverted bucket steam trap

- The small size of the hole in the top of the bucket means that this type of trap can only discharge air very slowly. The hole cannot be enlarged, as steam would pass through too quickly during normal operation.
- There should always be enough water in the trap body to act as a seal around the lip of the bucket. If the trap loses this water seal, steam can be wasted through the outlet valve. This can often happen on applications where there is a sudden drop in steam pressure, causing some of the condensate in the trap body to 'flash' into steam. The bucket loses its buoyancy and sinks, allowing live steam to pass through the trap orifice. Only if sufficient condensate reaches the trap will the water seal form again, and prevent steam wastage.

Float and Thermostatic

The ball float type trap operates by sensing the difference in density between steam and condensate. In the case of the trap shown in Figure 3.6A, condensate reaching the trap will cause the ball float to rise, lifting the valve off its seat and releasing condensate. As can be seen, the valve is always flooded and neither steam nor air will pass through it, so early traps of this kind were vented using a manually operated cock at the top of the body. Modern traps use a thermostatic air vent, as shown in Figure 3.6B. This allows the initial air to pass whilst the trap is also handling condensate.

The automatic air vent uses the same balanced pressure capsule element as a thermostatic steam trap, and is located in the steam space above the condensate level. After releasing the initial air, it remains closed until air or other non-condensable gases accumulate during normal running and cause it to open by reducing the temperature of the air/steam mixture. The thermostatic air vent offers the added benefit of significantly increasing condensate capacity on cold start-up.

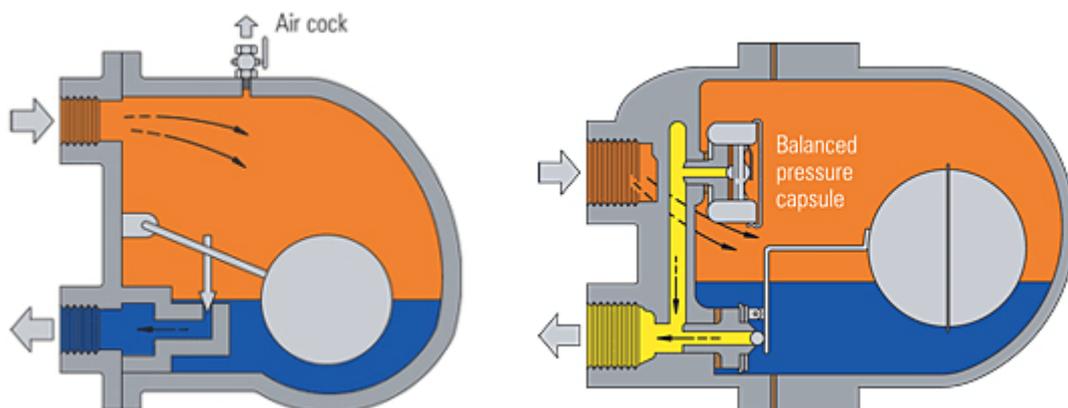


Figure 3.6A *Float trap with air cock* **Figure 3.6B** *Float trap with thermostatic air vent*

In the past, the thermostatic air vent was a point of weakness if water hammer was present in the system. Even the ball could be damaged if the water hammer was severe. However, in modern float traps the air vent is a compact, very robust, all stainless steel capsules,

and the modern welding techniques used on the ball makes the complete float-thermostatic steam trap very robust and reliable in water hammer situations.

In many ways the float-thermostatic trap is the closest to an ideal steam trap. It will discharge condensate as soon as it is formed, regardless of changes in steam pressure. Float and Thermostatic traps are an economical solution for light-to-medium condensate loads and lower pressures

Thermodynamic

The thermodynamic trap is an extremely robust steam trap with a simple mode of operation. The trap operates by means of the dynamic effect of flash steam as it passes through the trap, as depicted in Figure 3.7. The only moving part is the disc above the flat face inside the control chamber or cap.

On start-up, incoming pressure raises the disc, and cool condensate plus air is immediately discharged from the inner ring, under the disc, and out through three peripheral outlets (Figure 3.7, i).

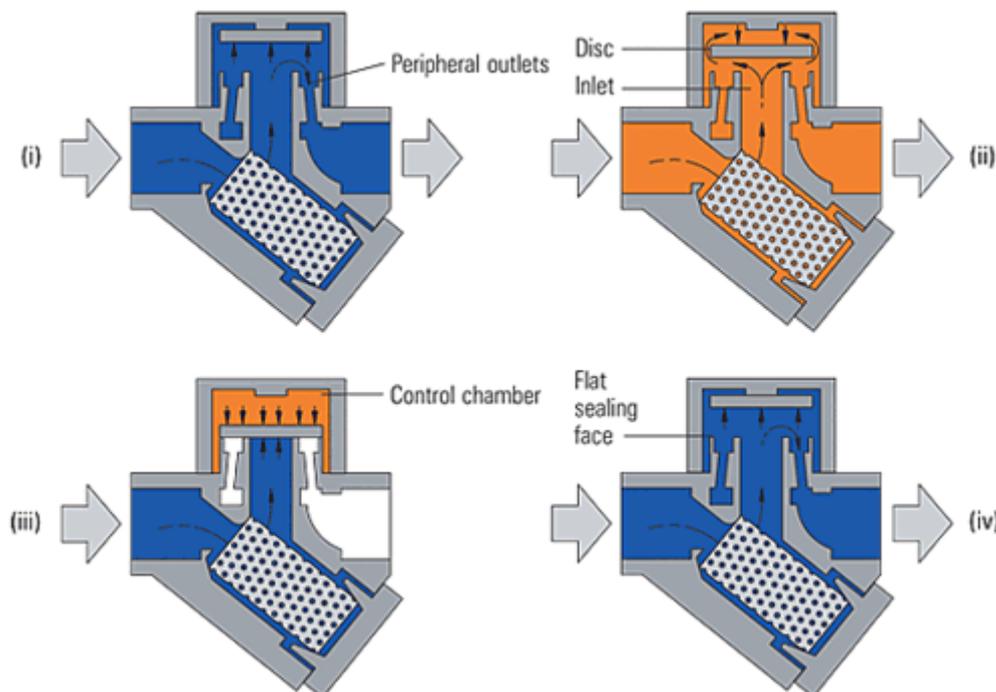


Figure 3.7: Thermodynamic Trap

Hot condensate flowing through the inlet passage into the chamber, under the disc drops in pressure and releases flash steam moving at high velocity. This high velocity creates a low pressure area under the disc, drawing it towards its seat (Figure 3.7, ii).

At the same time, the flash steam pressure builds up inside the chamber above the disc, forcing it down against the incoming condensate until it seats on the inner and outer rings. At this point, the flash steam is trapped in the upper chamber, and the pressure above the disc equals the pressure being applied to the underside of the disc from the inner ring. However, the top of the disc is subject to a greater force than the underside, as it has a greater surface area.

Eventually the trapped pressure in the upper chamber falls as the flash steam condenses. The disc is raised by the now higher condensate pressure and the cycle repeats (Figure 3.7, iv).

Thermostatic

Thermal-element thermostatic traps are temperature actuated. On startup the thermal element is in a contracted position with the valve wide-open, purging condensate, air, and other non-condensable gases. As the system warms up, heat generates pressure in the thermal element, causing it to expand and throttle the flow of hot condensate through the discharge valve.

When steam follows the hot condensate into the trap, the thermal element fully expands, closing the trap. If condensate enters the trap during system operation, it cools the element, contracting it off the seat, and quickly discharging condensate (Figure 3.8).

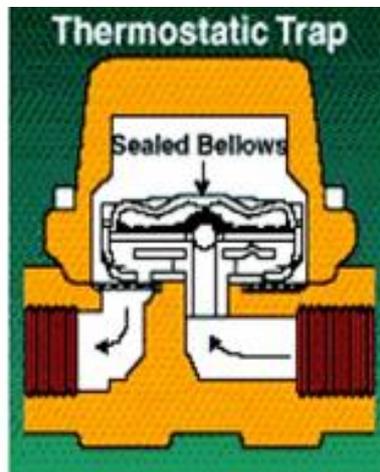


Figure 3.8: Thermostatic Trap

Thermostatic traps are small, lightweight, and compact. One trap operates over extremely broad pressure and capacity ranges. Thermal elements can be selected to operate within a range of steam temperatures. In steam tracing applications, it may be desirable to actually back up hot condensate in the lines to extract its thermal value.

Bimetallic Type

Bimetallic steam traps operate on the same principle as a heating thermostat. A bimetallic strip or wafer connected to a valve bends or distorts when subjected to a change in temperature. When properly calibrated, the valve closes off against a seat when steam is present, and opens when condensate, air, and other non-condensable gases are present (Figure 3.9).

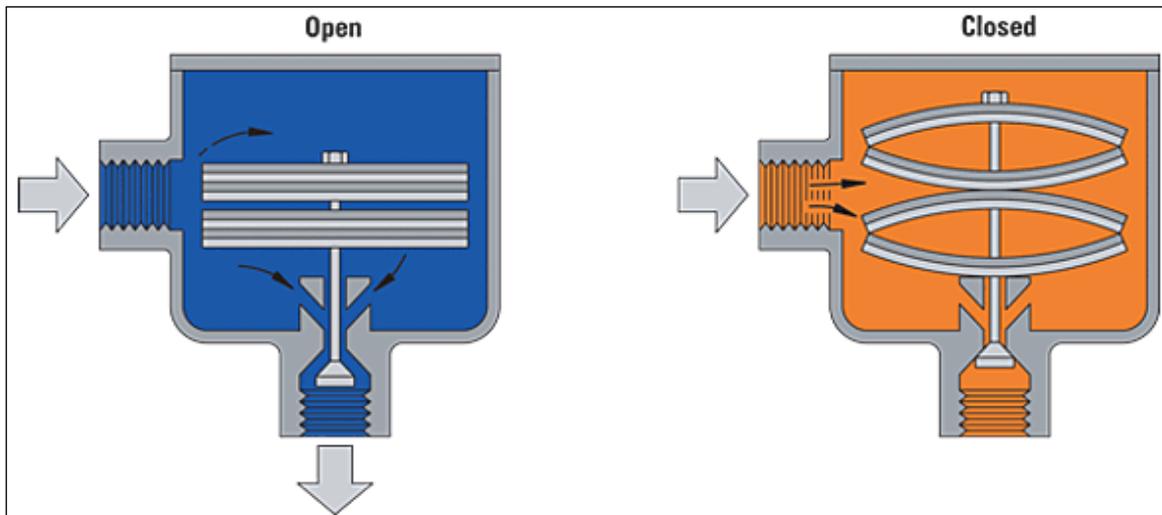


Figure 3.9: Bimetallic Trap

Advantages of the bimetallic steam trap

- Relatively small size for the condensate loads they handle
- Resistance to damage from water hammer

A disadvantage is that they must be set, generally at the plant, for a particular steam operating pressure. If the trap is used for a lower pressure, it may discharge live steam. If used at a higher steam pressure, it can back up condensate into the system.

Thermostatic traps are often considered a universal steam trap; however, they are normally not recommended for extremely high condensate requirements (over 7000 kg/hr). For light-to-moderately high condensate loads, thermostatic steam traps offer advantages in terms of initial cost, long-term energy conservation, reduced inventory, and ease in application and maintenance.

Installation of Steam Traps

In most cases, trapping problems are caused by bad installation rather than by the choice of the wrong type or faulty manufacture. To ensure a trouble-free installation, careful consideration should be given to the drain point, pipe sizing, air venting, steam locking, group trapping vs. individual trapping, dirt, water hammer, lifting of the condensate, etc.

1) Drain Point

The drain point should be so arranged that the condensate can easily flow into the trap. This is not always appreciated. For example, it is useless to provide a 15mm drain hole in the bottom of a 150 mm steam main, because most of the condensate will be carried away by the steam velocity. A proper pocket at the lowest part of the pipe line into which the condensate can drop of at least 100mm diameter is needed in such cases.

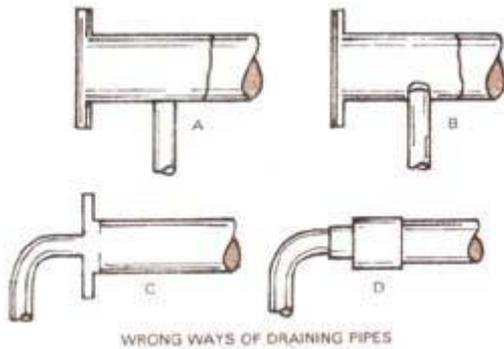


Figure 3.10 A: Wrong ways of Draining Pipes

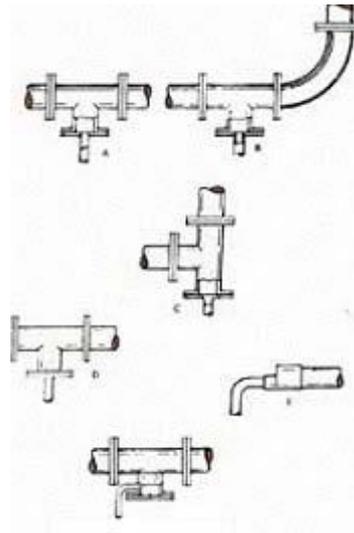


Figure 3.10 B: Right ways of Draining Pipes

Figures 3.10 A and 3.10 B show the wrong and the correct practices in providing the drain points on the steam lines.

2) Pipe Sizing

The pipes leading to and from steam traps should be of adequate size. This is particularly important in the case of thermodynamic traps, because their correct operation can be disturbed by excessive resistance to flow in the condensate pipe work. Pipe fittings such as valves, bends and tees close to the trap will also set up excessive backpressures in certain circumstances.

3) Air Binding

When air is pumped into the trap space by the steam, the trap function ceases. Unless adequate provision is made for removing air either by way of the steam trap or a separate air vent, the plant may take a long time in warming up and may never give its full output.

4) Steam Locking

This is similar to air binding except that the trap is locked shut by steam instead of air. The typical example is a drying cylinder. It is always advisable to use a float trap provided with a steam lock release arrangement.

5) Group Trapping vs. Individual Trapping

It is tempting to try and save money by connecting several units to a common steam trap as shown in Figure 3.11 A. This is known as group trapping. However, it is rarely successful, since it normally causes water-logging and loss of output.

The steam consumption of a number of units is never the same at a moment of time and therefore, the pressure in the various steam spaces will also be different. It follows that the pressure at the drain outlet of a heavily loaded unit will be less than in the case of one that is lightly or properly loaded. Now, if all these units are connected to a common steam trap, the condensate from the heavily loaded and therefore lower pressure steam space finds it difficult to reach the trap as against the higher pressure condensate produced by lightly or partly loaded unit. The only satisfactory arrangement, thus would be to drain each steam space with own trap and then connect the outlets of the various traps to the common condensate return main as shown in above Figure 3.11 B.

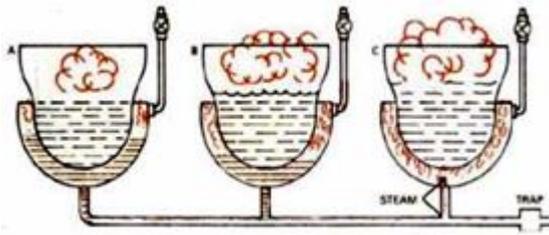


Figure 3.11 A: Group Trapping

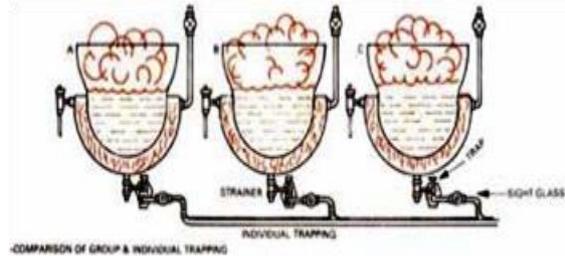


Figure 3.11 B: Individual Trapping

6) Dirt

Dirt is the common enemy of steam traps and the causes of many failures. New steam systems contain scale, castings, weld metal, piece of packing and jointing materials, etc. When the system has been in use for a while, the inside of the pipe work and fittings, which are exposed to corrosive condensate, can get rusted. Thus, rust in the form of a fine brown powder is also likely to be present. All this dirt will be carried through the system by the steam and condensate until it reaches the steam trap. Some of it may pass through the trap into the condensate system without doing any harm, but some dirt will eventually jam the trap mechanism. It is advisable to use a strainer positioned before the steam trap to prevent dirt from passing into the system.

7) Water Hammer

A water hammer in a steam system is caused by condensate collection in the plant or pipe work picked up by the fast moving steam and carried along with it. When this collection hits obstructions such as bends, valves, steam traps or some other pipe fittings, it is likely to cause severe damage to fittings and equipment and result in leaking pipe joints.

The problem of water hammer can be eliminated by positioning the pipes so that there is a continuous slope in the direction of flow. In case of steam mains, a slope of at least 1 m in every 100 meters is necessary, as also an adequate number of drain points every 30 to 50 meters.

8) Lifting the condensate

It is sometimes necessary to lift condensate from a steam trap to a higher level condensate return line (Figure 3.12). The condensate will rise up the lifting pipe work when the steam pressure upstream of the trap is higher than the pressure downstream of the trap.

The pressure downstream of the trap is generally called backpressure, and is made up of any pressure existing in the condensate line plus the static lift caused by condensate in the rising pipe work. The upstream pressure will vary between start-up conditions, when it is at its lowest and running conditions, when it is at its highest.

Backpressure is related to lift by using the following approximate conversion: 1 metre lift in pipe work = 1 m head static pressure or 0.1 bar backpressure. If a head of 5 m produces a backpressure of 0.5 bar, then this reduces the differential pressure available to push condensate through the trap; although under running conditions the reduction in trap capacity is likely to be significant only where low upstream pressures are used.

In steam mains at start-up, the steam pressure is likely to be very low, and it is common for water to back-up before the trap, which can lead to water hammer in the space being drained. To alleviate this problem at start-up, a liquid expansion trap, fitted as shown in Figure 3.12, will discharge any cold condensate formed at this time to waste.

As the steam main is warmed, the condensate temperature rises, causing the liquid expansion trap to close. At the same time, the steam pressure rises, forcing the hot condensate through the 'working' drain trap to the return line.

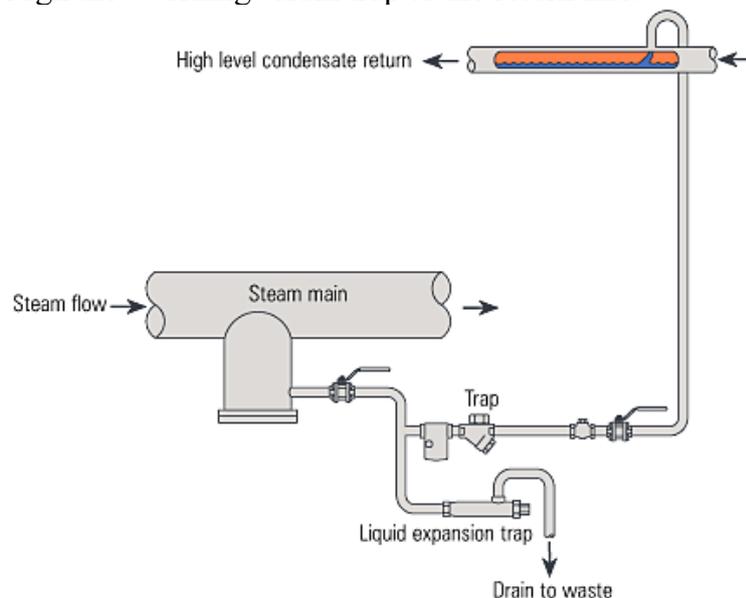


Figure 3.12: Use of a Liquid Expansion Trap

The discharge line from the trap to the overhead return line preferably discharges into the top of the main rather than simply feed to the underside, as shown in Figure 3.12. This assists operation, because although the riser is probably full of water at start-up, it sometimes contains little more than flash steam once hot condensate under pressure passes through. If the discharge line were fitted to the bottom of the return line, it would fill with condensate after each discharge and increase the tendency for water hammer and noise.

It is also recommended that a check valve be fitted after any steam trap from where condensate is lifted, preventing condensate from falling back towards the trap. The above general recommendations apply not just to traps lifting condensate from steam mains, but also to traps draining any type of process running at a constant steam pressure. Temperature controlled processes will often run with low steam pressures. Rising

condensate discharge lines should be avoided at all costs, unless automatic pump-traps are used.

Maintenance of steam traps

Dirt is one of the most common causes of steam traps blowing steam. Dirt and scale are normally found in all steam pipes. Bits of jointing material are also quite common. Since steam traps are connected to the lowest parts of the system, sooner or later this foreign matter finds its way to the trap. Once some of the dirt gets logged in the valve seat, it prevents the valve from shutting down tightly thus allowing steam to escape. The valve seal should therefore be quickly cleaned, to remove this obstruction and thus prevent steam loss.

In order to ensure proper working, steam traps should be kept free of pipe-scale and dirt. The best way to prevent the scale and dirt from getting into the trap is to fit a strainer. Strainer is a detachable, perforated or meshed screen enclosed in a metal body. It should be borne in mind that the strainer collects dirt in the course of time and will therefore need periodic cleaning. It is of course, much easier to clean a strainer than to overhaul a steam trap.

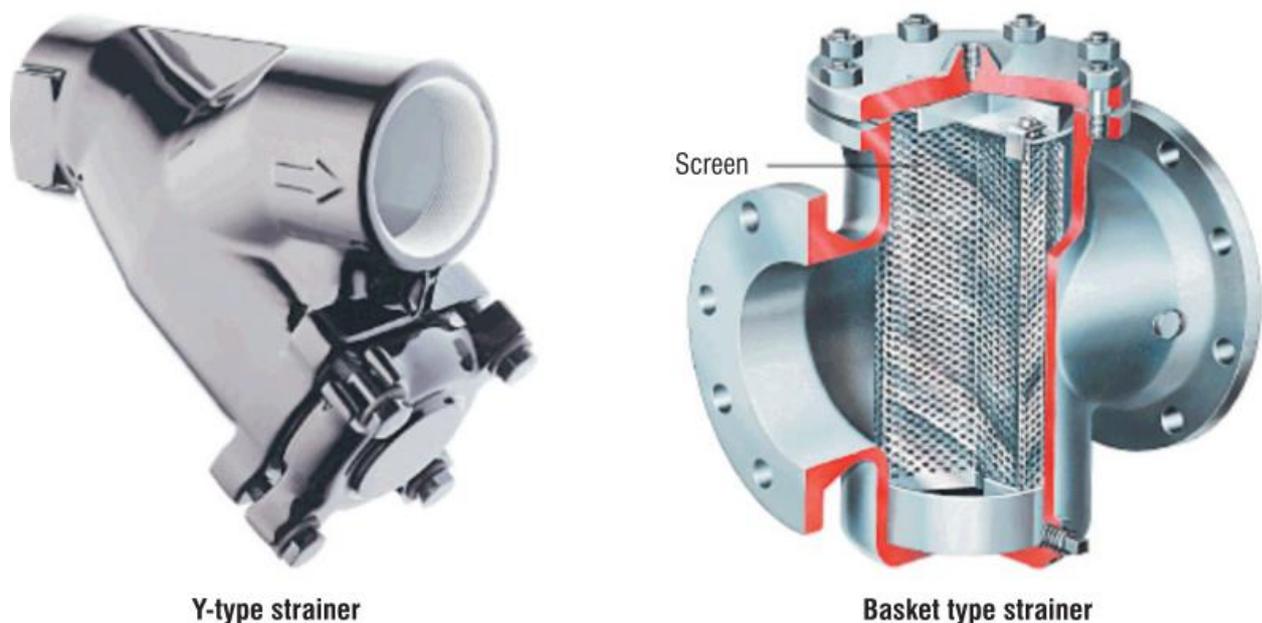


Figure 3.13: Strainers

At this point, we might mention the usefulness of a sight glass fitted just after a steam trap. Sight glasses are useful in ascertaining the proper functioning of traps and in detecting leaking steam traps. In particular, they are of considerable advantage when a number of steam traps are discharging into a common return line. If it is suspected that one of the traps is blowing steam, it can be quickly identified by looking through the sight glass.

In most industries, maintenance of steam traps is not a routine job and is neglected unless it leads to some definite trouble in the plant. In view of their importance as steam savers

and to monitor plant efficiency, the steam traps require considerably more care than is given.

One may consider a periodic maintenance schedule to repair and replace defective traps in the shortest possible time, preferable during regular maintenance shut downs in preference to break down repairs.

Guide to Steam Trap Selection

Actual energy efficiency can be achieved only when

- a) Selection
- b) Installation and
- c) Maintenance of steam traps meet the requirements for the purpose it is installed

The following Table 3.5 gives installation of suitable traps for different process applications.

Table 3.5 Selection of Steam trap

Application	Feature	Suitable trap
Steam mains	<ul style="list-style-type: none"> • Open to atmosphere, small capacity • Frequent change in pressure • Low pressure - high pressure 	Thermodynamic type
Equipment <ul style="list-style-type: none"> • Reboiler • Heater • Dryer • Heat exchanger etc. 	<ul style="list-style-type: none"> • Large capacity • Variation in pressure and temperature is undesirable • Efficiency of the equipment is a problem 	Mechanical trap, Bucket, Inverted bucket, float
<ul style="list-style-type: none"> • Tracer line • Instrumentation 	<ul style="list-style-type: none"> • Reliability with no over heating 	Thermodynamic & Bimetallic

3.6 Performance Assessment Methods for Steam Traps

Steam trap performance assessment is basically concerned with answering the following two questions:

- Is the trap working correctly or not?
- If not, has the trap failed in the open or closed position?

Traps that fail 'open' result in a loss of steam and its energy. Where condensate is not returned, the water is lost as well. The result is significant economic loss, directly via increased boiler plant costs, and potentially indirectly, via decreased steam heating capacity.

Traps that fail 'closed' do not result in energy or water losses, but can result in significantly reduced heating capacity and/or damage to steam heating equipment.

Visual Testing

Visual testing includes traps with open discharge sight glasses (Figure 3.13), sight checks, test tees and three way test valves. In every case, the flow or variation of flow is visually observed. This method works well with traps that cycle on/off, or dribbles on light load. On high flow or process, due to the volume of water and flash steam, this method becomes less viable. If condensate can be diverted ahead of the trap or a secondary flow can be turned off, the load on the trap will drop to zero or a very minimal amount so the visual test will allow in determining the leakage.

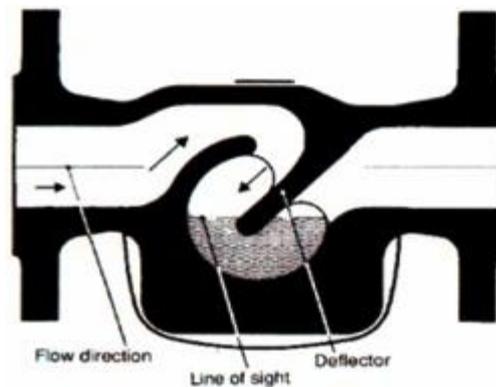


Figure 3.13: Sight Glass

Sound Testing

Sound testing includes ultrasonic leak detectors (Figure 3.14), mechanics stethoscopes, screwdriver or metal rod with a human ear against it. All these use the sound created by flow to determine the trap function like the visual method. This method works best with traps that cycle on/off or dribbles on light load. Traps which have modulating type discharge patterns are hard to check on high flows. (Examples are processes, heat exchangers, air handling coils, etc.). Again by diverting condensate flow ahead of the trap or shutting off a secondary flow as mentioned under visual testing, the noise level will drop to zero or a very low level if the trap is operating correctly. If the trap continues to flow heavily after diversion it would be leaking or blowing through.



Figure 3.14: Ultrasonic Testing

Temperature Testing

Temperature testing includes infrared guns (Figure 3.15), surface pyrometers, temperature tapes, and temperature crayons. Typically they are used to gauge the discharge temperature on the outlet side of the trap. In the case of temperature tapes or crayon, they are set for a predetermined temperature and they indicate when temperature exceeds that level. Infrared guns and surface pyrometer can detect temperatures on both sides of the trap. Both the infrared and surface pyrometers require bare pipe and a clean surface to achieve a reasonable reading. The temperature reading will typically be lower than actual internal pipe temperature due to the fact that steel does have some heat flow resistance. Scale on the inside of the pipe can also affect the heat transfer. Some of the more expensive infrared guns can compensate for wall thickness and material differences. Blocked or turned off traps can easily be detected by infrared guns and surface pyrometers, as they will show low or cold temperatures. They could also pick up traps which may be undersized or backing up large amounts of condensate by detecting low temperature readings.



Figure 3.15: Infra Red Testing

3.7 Energy Saving Opportunities

1. Monitoring Steam Traps

For testing a steam trap, there should be an isolating valve provided in the downstream of the trap and a test valve shall be provided in the trap discharge. When the test valve is opened, the following points have to be observed:

Condensate discharge-Inverted bucket and thermodynamic disc traps should have intermittent condensate discharge. Float and thermostatic traps should have a continuous condensate discharge. Thermostatic traps can have either continuous or intermittent discharge depending upon the load. If inverted bucket traps are used for extremely small load, it will have a continuous condensate discharge.

Flash steam-This shall not be mistaken for a steam leak through the trap. The users sometimes get confused between a flash steam and leaking steam. The flash steam and the leaking steam can be approximately identified as follows:

If steam blows out continuously in a blue stream, it is a leaking steam.

If a steam floats out intermittently in a whitish cloud, it is a flash steam.

2. Continuous steam blow and no flow indicate there is a problem in the trap.

Whenever a trap fails to operate and the reasons are not readily apparent, the discharge from the trap should be observed. A step-by-step analysis has to be carried out mainly with reference to lack of discharge from the trap, steam loss, continuous flow, sluggish heating, to find out whether it is a system problem or the mechanical problem in the steam trap.

3. Avoiding Steam Leakages

Steam leakage is a visible indicator of waste and must be avoided. It has been estimated that a 3 mm diameter hole on a pipeline carrying 7kg/cm^2 steam would waste 33 Kilo Litre of fuel oil per year. Steam leaks on high-pressure mains are prohibitively costlier than on low pressure mains. Any steam leakage must be quickly attended to. In fact, the plant should consider a regular surveillance programme for identifying leaks at pipelines, valves, flanges and joints. Indeed, by plugging all leakages, one may be surprised at the extent of fuel savings, which may reach up to 5% of the steam consumption in a small or medium scale industry or even higher in installations having several process departments.

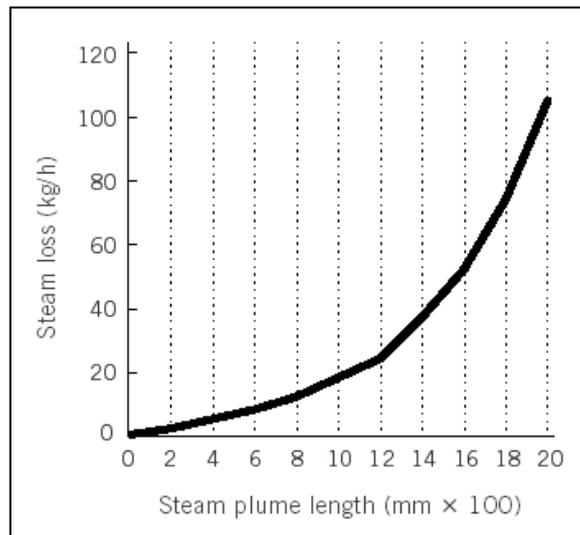


Figure 3.16: Steam Loss vs Plume Length

To avoid leaks it may be worthwhile considering replacement of the flanged joints which are rarely opened in old plants by welded joints. Figure 3.16 provides a quick estimate for steam leakage based on plume length.

Example 3.6

- Plume Length = 700 mm
- Steam loss = 10 kg/h

The following Table 3.6 highlights the significance of loss through steam leaks.

Table: 3.6 Steam Losses due to Leakage

S.No.	Dia. of Leak (in mm)	Annual Steam Loss (Tons/year)	
		At 3.5 kg/cm ²	At 7.0 kg/cm ²
1	1.5	29.1	47.3
2	3.0	116.4	192.7
3	4.5	232.7	432.7
4	6.0	465.4	767.3

4. Providing Dry Steam for Process

The best steam for industrial process heating is the dry saturated steam. Wet steam reduces total heat in the steam. Also water forms a wet film on heat transfer and overloads traps and condensate equipment. Super-heated steam is not desirable for process heating because it gives up heat at a rate slower than the condensation heat transfer of saturated steam.

It must be remembered that a boiler without a super heater cannot deliver perfectly dry saturated steam. At best, it can deliver only 95% dry steam. The dryness fraction of steam depends on various factors, such as the level of water, improper boiler water treatment etc. As steam flows through the pipelines, it undergoes progressive condensation due to the loss of heat to the colder surroundings; the extent of the condensation depends on the effectiveness of the lagging. For example, with poor lagging, the steam can become excessively wet.

Since dry saturated steam is required for process equipment, due attention must be paid to the boiler operation and lagging of the pipelines. The steam produced in a boiler designed to generate saturated steam is inherently wet. Although the dryness fraction will vary according to the type of boiler, most shell type steam boilers will produce steam with a dryness fraction of between 95 and 98%.

Wet steam can reduce plant productivity and product quality, and can cause damage to most items of plant and equipment. The water content of the steam produced by the boiler is further increased if priming and carryover occur.

A steam separator (Figure 3.17) may be installed on the steam main as well as on the branch lines to reduce wetness in steam and improve the quality of the steam going to the units. By change of direction of steam, steam separators causes the entrained water particles to be separated out and delivered to a point where they can be drained away as condensate through a conventional steam trap.

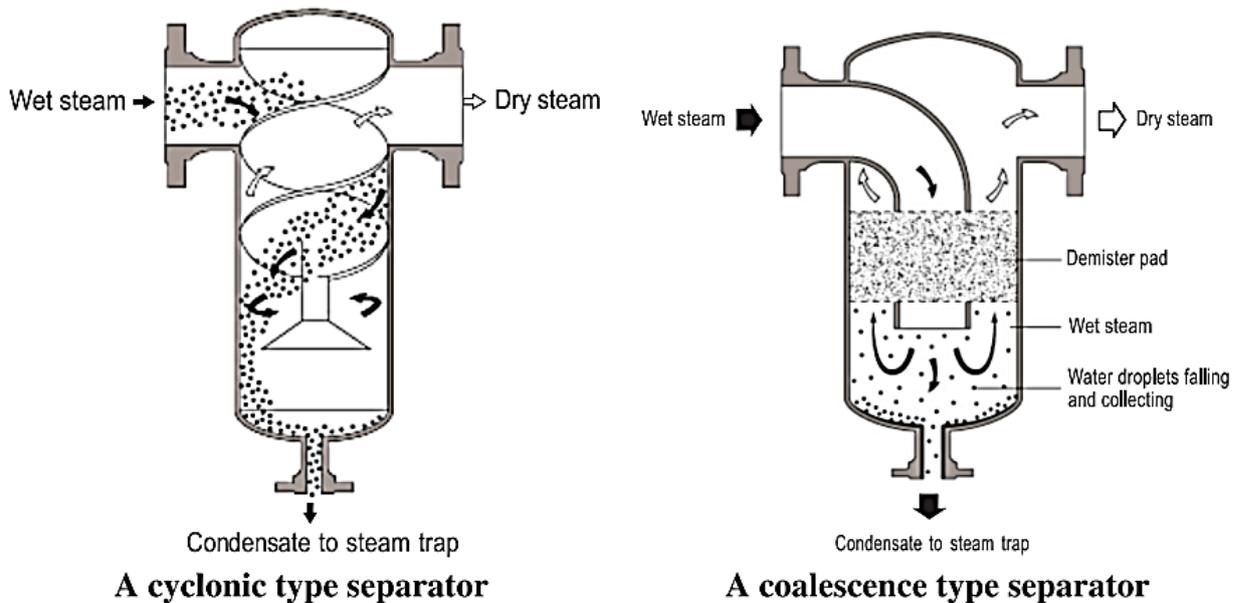


Figure 3.17: Steam Separators

5. Utilising Steam at the Lowest Acceptable Pressure for the Process

A study of the steam tables would indicate that the latent heat in steam reduces as the steam pressure increases. It is only the latent heat of steam, which takes part in the heating process when applied to an indirect heating system. Thus, it is important that its value be kept as high as possible. This can only be achieved if we go in for lower steam pressures. As a guide, the steam should always be generated and distributed at the highest possible pressure, but utilized at a low pressure as possible since it has higher latent heat.

However, it may also be seen from the steam tables that the lower the steam pressure, the lower will be its temperature. Since temperature is the driving force for the transfer of heat at lower steam pressures, the rate of heat transfer will be slower and the processing time greater. In equipment where fixed losses are high (e.g. big drying cylinders), there may even be an increase in steam consumption at lower pressures due to increased processing time. There are however, several equipment in certain industries where one can profitably go in for lower pressures and realize economy in steam consumption without materially affecting production time.

Therefore, there is a limit to the reduction of steam pressure. Depending on the equipment design, the lowest possible steam pressure with which the equipment can work should be selected without sacrificing either on production time or on steam consumption.

6. Proper Utilization of Directly Injected Steam

The heating of a liquid by direct injection of steam is often desirable. The equipment required is relatively simple, cheap and easy to maintain. No condensate recovery system is necessary. The heating is quick, and the sensible heat of the steam is also used up along with the latent heat, making the process thermally efficient. In processes where dilution is not a problem, heating is done by blowing steam into the liquid (i.e.) direct steam injection

is applied. If the dilution of the tank contents and agitation are not acceptable in the process (i.e.) direct steam agitation are not acceptable, indirect steam heating is the only answer.

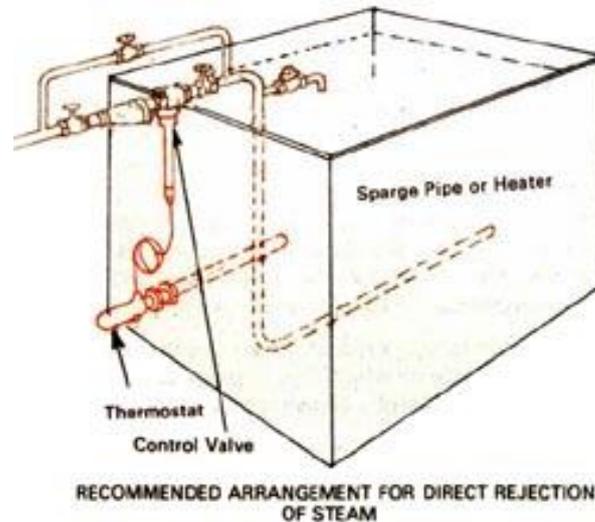


Figure 3.18: Temperature Control for Directly Injected Steam

Ideally, the injected steam should be condensed completely as the bubbles rise through the liquid. This is possible only if the inlet steam pressures are kept very low - around 0.5 kg/cm^2 -and certainly not exceeding 1 kg/cm^2 . If pressures are high, the velocity of the steam bubbles will also be high and they will not get sufficient time to condense before they reach the surface. Figure 3.18 shows a recommended arrangement for direct injection of steam.

A large number of small diameter holes (2 to 5mm), facing downwards, should be drilled on the separate pipe. This will help in dissipating the velocity of bubbles in the liquid. A thermostatic control of steam admitted is highly desirable.

7. Minimising Heat Transfer Barriers

The metal wall may not be the only barrier in a heat transfer process. There is likely to be a film of air, condensate and scale on the steam side. On the product side, there may also be baked-on product or scale, and a stagnant film of product.

Agitation of the product may eliminate the effect of the stagnant film, whilst regular cleaning on the product side should reduce the scale.

Regular cleaning of the surface on the steam side may also increase the rate of heat transfer by reducing the thickness of any layer of scale; however, this may not always be possible. This layer may also be reduced by careful attention to the correct operation of the boiler, and the removal of water droplets carrying impurities from the boiler.

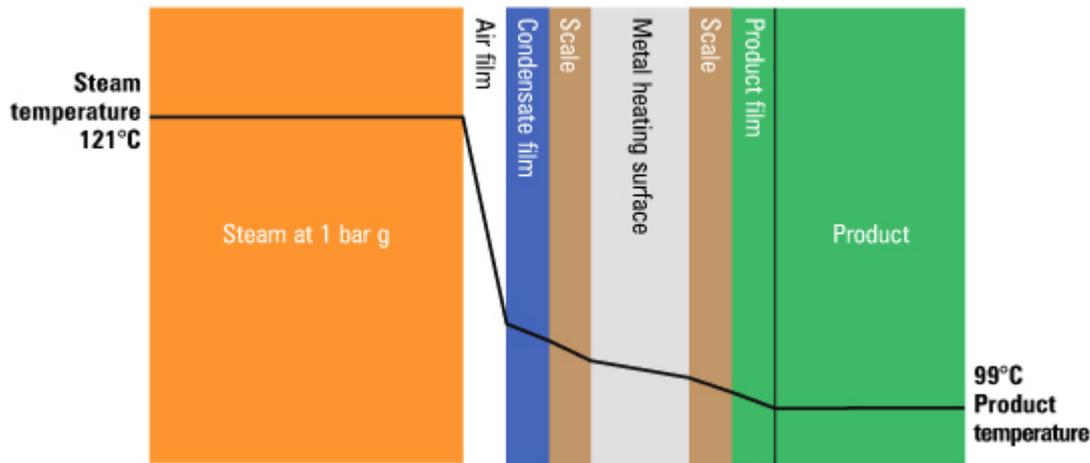


Figure 3.19: Heat Transfer Barriers

The elimination of the condensate film is not quite as simple. As the steam condenses to give up its enthalpy of evaporation, droplets of water may form on the heat transfer surface. These may merge together to form a continuous film of condensate. The condensate film may be between 100 and 150 times more resistant to heat transfer than a steel heating surface, and 500 to 600 times more resistant than copper.

As air is a good insulator, it provides even more resistance to heat transfer. Air may be between 1500 and 3000 times more resistant to heat flow than steel, and 8000 to 16000 more resistant than copper. This means that a film of air only 0.025 mm thick may resist as much heat transfer as a wall of copper 400 mm thick. These comparative relationships depend on the temperature profiles across each layer.

Figure 3.19 illustrates the effect this combination of layers has on the heat transfer process. These barriers to heat transfer not only increase the thickness of the entire conductive layer, but also greatly reduce the mean thermal conductivity of the layer.

The more resistant the layer to heat flow, the larger the temperature gradient is likely to be. This means that to achieve the same desired product temperature, the steam pressure may need to be significantly higher.

To achieve the desired product output and minimise the cost of process steam operations, a high heating performance may be maintained by reducing the thickness of the films on the condensing surface and removal of air from the supply steam.

8. Proper Air Venting

When steam is first admitted to a pipe after a period of shutdown, the pipe is full of air. Further, amounts of air and other non-condensable gases will enter with the steam, although the proportions of these gases are normally very small compared with the steam. When the steam condenses, these gases will accumulate in pipes and heat exchangers. Precautions should be taken to discharge them. The consequence of not removing air is a lengthy warming up period, reduction in plant efficiency and process performance. Air in the steam system will also affect the system temperature. Air will exert its own pressure within the system, which will add to the pressure of the steam to give a total pressure.

A layer of air only 1 mm thick can offer the same resistance to heat as a layer of water 25 μm thick, a layer of iron 2 mm thick or a layer of copper 15 mm thick. It is very important therefore to remove air from any steam system.

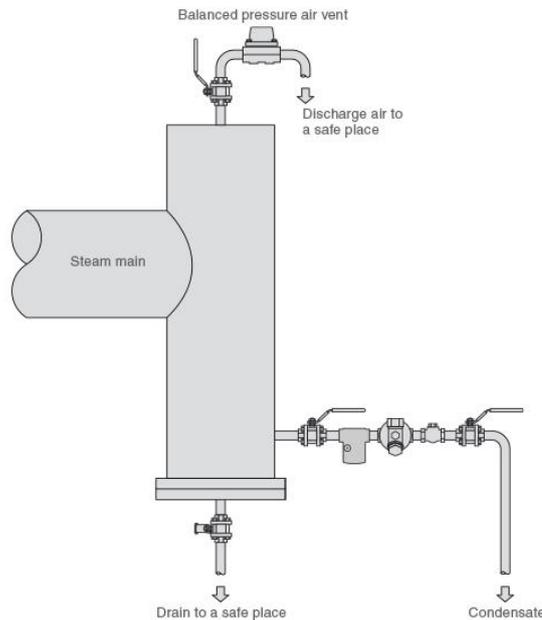


Figure 3.20: Air Vent

Automatic air vents for steam systems (which operate on the same principle as thermostatic steam traps) should be fitted above the condensate level so that only air or steam-air mixtures can reach them. The best location for them is at the end of the steam mains. In addition to air venting at the end of a main, air vents should also be fitted in parallel with an inverted bucket trap or a thermodynamic trap.

9. Condensate Recovery

The steam condenses after giving off its latent heat in the heating coil or the jacket of the process equipment. A sizable portion (about 25%) of the total heat in the steam leaves the process equipment as hot water. If this water is returned to the boiler house, it will reduce the fuel requirements of the boiler. For every 6 $^{\circ}\text{C}$ rise in the feed water temperature, there will be approximately 1% saving of fuel in the boiler. However, in most cases, the boiler water has to be chemically treated to prevent or reduce scale formation, whereas the condensate is almost entirely pure water which needs no treatment. With a good percentage of the condensate returning to the boiler house, the expenses involved for water treatment will be reduced by an appreciable amount.

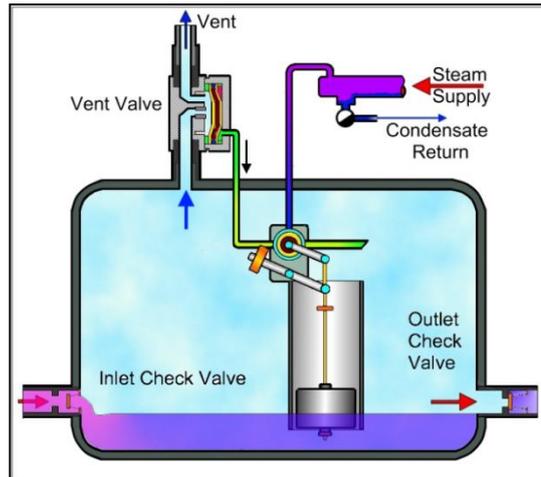


Figure 3.21: Pressure Powered Pump

Use a Steam Driven Pump: A pressure powered pump (Figure 3.21) uses steam pressure to push the condensate from the receiver back to the boiler house. In principle it consists of a receiver which receives condensate from different process/equipment. Once the condensate reaches a set level, the steam valve is opened and the steam pressure pushes the condensate to the boiler room. The operation is cyclic in nature. The advantage is pumping of condensate without losing much heat in the form of flash steam without any cavitation problems.

10. Insulation of Steam Pipelines and Hot Process Equipment

Steam lines including flanges and valves should be insulated to prevent heat loss. The recommended thickness of insulation will mainly depend on surface temperature desired after insulation. The energy and cost savings will depend on the size of the pipe (diameter and length of run), the temperature of steam and the surroundings, heat transfer co-efficient and the number of hours of operation of the plant.

The following Table 3.7 indicates the effect of insulating bare pipes

Table 3.7 Effect of Insulation on Steam Pipes

Pipe Size, inch	Economic Insulation Thickness, mm	Radiation Losses* (kW/m)	
		Insulated	Uninsulated
½	15	125	692
2	25	243	1820
4	40	298	2942
12	50	588	7614

* Comparison of Radiation Losses (Pipe Surface Temperature at 150 °C)

11. Flash Steam Recovery

Flash steam is produced when condensate at a high pressure is released to a lower pressure and can be used for low pressure heating.

The higher the steam pressure and lower the flash steam pressure the greater the quantity of flash steam that can be generated. In many cases, flash steam from high pressure equipment is made use of directly on the low pressure equipment to reduce use of steam through pressure reducing valves.

The flash steam quantity can be calculated by the following formula with the help of a steam table:

$$\text{Flash steam available \%} = \frac{S_1 - S_2}{L_2}$$

Where,

S_1 is the sensible heat of higher pressure condensate.

S_2 is the sensible heat of the steam at lower pressure (at which it has been flashed).

L_2 is the latent heat of flash steam (at lower pressure).

Example 3.7 Calculating the amount of flash steam from condensate

Hot condensate at 7 bar g has a heat content of about 721 kJ/kg. When it is released to atmospheric pressure (0 bar g), each kilogram of water can only retain about 419 kJ of heat. The excess energy in each kilogram of the condensate is therefore $721 - 419 = 302$ kJ. This excess energy is available to evaporate some of the condensate into steam, the amount evaporated being determined by the proportion of excess heat to the amount of heat required to evaporate water at the lower pressure, which in this example, is the enthalpy of evaporation at atmospheric pressure, 2258 kJ/kg.

Therefore, in this example, the percentage of flash steam evaporated = $(302/2258) * 100\%$

Flash steam evaporated = 13.4%

The amount of flash steam in the pipe is the most important factor when sizing trap discharge lines.

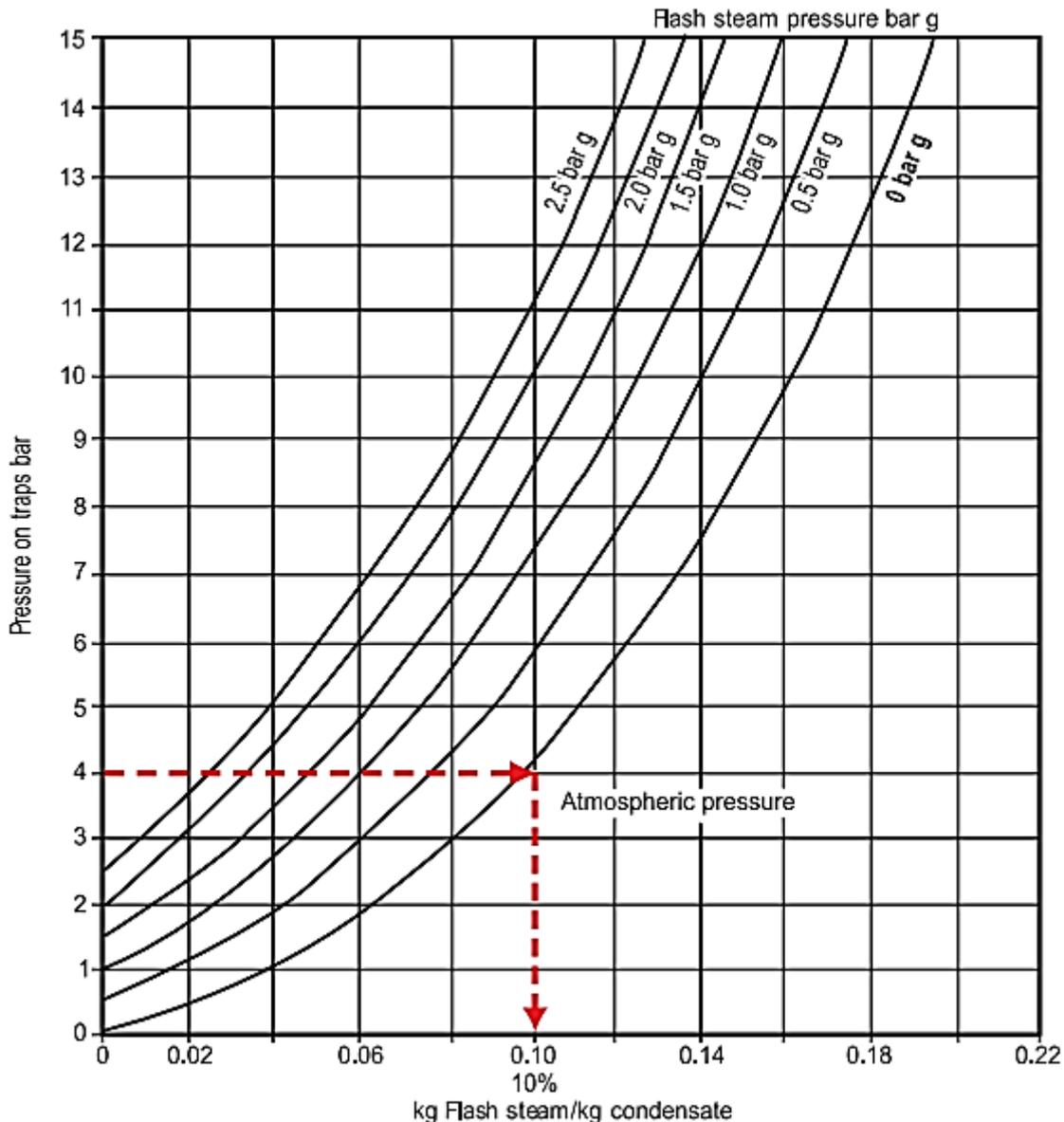


Figure 3.28: Quantity of Flash Steam Graph

Flash steam can be used on low pressure applications like direct injection and can replace an equal quantity of live steam that would be otherwise required. The demand for flash steam should exceed its supply, so that there is no buildup of pressure in the flash vessel and the consequent loss of steam through the safety valve. Generally, the simplest method of using flash steam is to flash from a machine/equipment at a higher pressure to a machine/equipment at a lower pressure, thereby augmenting steam supply to the low pressure equipment.

In general, a flash system should run at the lowest possible pressure so that the maximum amount of flash is available and the backpressure on the high pressure systems is kept as low as possible.

Flash steam from the condensate can be separated in an equipment called the 'flash vessel'. This is a vertical vessel as shown in the Figure 3.22. The diameter of the vessel is such that a considerable drop in velocity allows the condensate to fall to the bottom of the vessel from where it is drained out by a steam trap preferably a float trap. Flash steam itself rises

to leave the vessel at the top. The height of the vessel should be sufficient enough to avoid water being carried over in the flash steam.

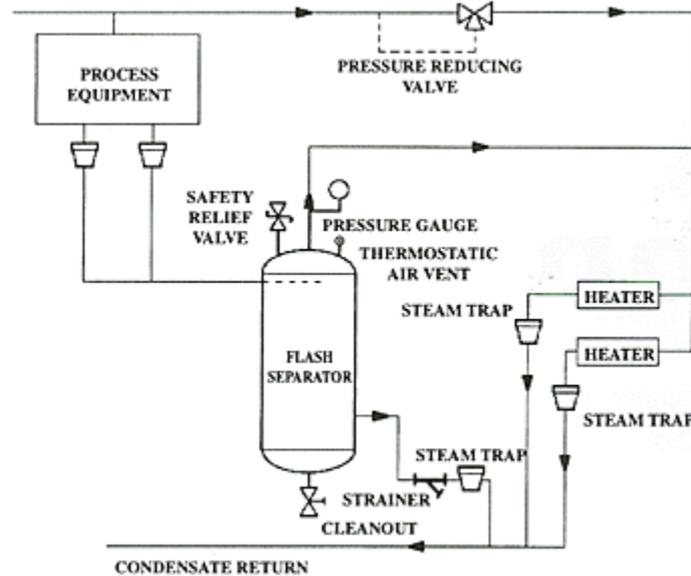


Figure 3.22: Flash Steam Recovery

The condensate from the traps (A) along with some flash steam generated passes through vessel (B). The flash steam is let out through (C) and the residual condensate from (B) goes out through the steam trap (D). The flash vessel is usually fitted with a ‘pressure gauge’ to know the quality of flash steam leaving the vessel. A ‘safety valve’ is also provided to vent out the steam in case of high pressure build up in the vessel.

12. Pipe Redundancy

All redundant (piping which are no longer needed) pipelines must be eliminated, which could be, at times, up to 10-15 % of total length. This would reduce steam distribution losses significantly. The pipe routing shall be made for transmission of steam in the shortest possible way, so as to reduce the pressure drop in the system, thus saving the energy. However, care should be taken that, the pipe routing shall be flexible enough to take thermal expansion and to keep the terminal point loads, within the allowable limit.

13. Reducing the Work to be done by Steam

The equipment should be supplied with steam as dry as possible. The plant should be made efficient. For example, if any product is to be dried such as in a laundry, a press could be used to squeeze as much water as possible before being heated up in a dryer using steam.

When the steam reaches the place where its heat is required, it must be ensured that the steam has no more work to do than is absolutely necessary. Air-heater batteries, for example, which provide hot air for drying, will use the same amount of steam whether the plant is fully or partly loaded. So, if the plant is running only at 50 per cent load, it is wasting twice as much steam (or twice as much fuel) than necessary.

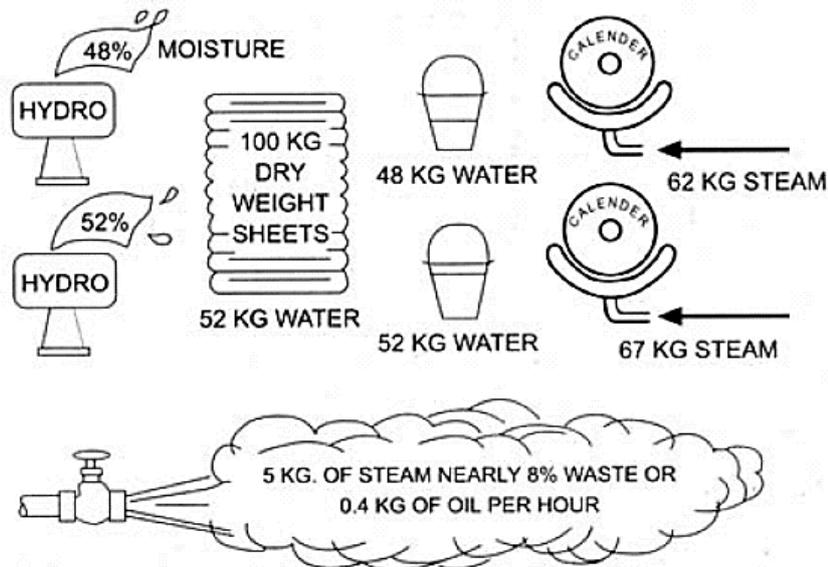


Figure 3.23 Steam Wastage Due to Insufficient Mechanical Drying

The energy saving is affected by following measures:

- Reduction in operating hours
- Reduction in steam quantity required per hour
- Use of more efficient technology
- Minimizing wastage.

Always use the most economical way to removing the bulk of water from the wet material. Steam can then be used to complete the process. For this reason, hydro-extractors, spin dryers, squeeze or calendar rolls, presses, etc. are initially used in many drying processes to remove the mass of water. The efficiency with which this operation is carried out is most important. For example, in a laundry for finishing sheets (100 kg/hr dry weight), the normal moisture content of the sheets as they leave the hydro extractor is 48% by weight.

Thus, the steam heated iron has to evaporate nearly 48 kg of water. This requires 62 kg of steam. If, due to inefficient drying in the hydro-extractor, the steam arrive at the iron with 52% moisture content i.e. 52 kg of water has to be evaporated, requiring about 67 kg of steam. So, for the same quantity of finished product, the steam consumption increases by 8 per cent. This is illustrated in Figure 3.23.

Examples 3.8

In a crude distillation unit of a refinery, 50 MT/hr of crude is heated using saturated steam in a heat exchanger from 35°C to 85°C. Plant is operating for 8000 hrs/annum. Consider specific heat of the crude as 0.631 kCal/kg°C. The plant has two steam headers operating at 3 bar and 8 bar respectively, passing nearby the heat exchanger. Cost of steam is same for both 3 bar and 8 bar @ BDT.4.50/kg.

As an Energy Manager, which of the following options will you recommend to the unit based on the annual cost of steam?

- Utilising 3 bar steam
- Utilising 8 bar steam

Given: Data from steam table:

Steam Pressure, bar	Enthalpy kCal/kg		
	Water	Evaporation	Steam
3.0	133	517	650
8.0	171	489	660

Solution

$$\text{Heat gain in crude} = m * C_p * \Delta T$$

$$\text{Heat gain in crude} = 50 * 1000 * 0.631 * (85 - 35)$$

$$\text{Heat gain in crude} = 1577500 \text{ kCal/hr}$$

$$\text{Heat gain in crude} = \text{heat loss in steam}$$

$$\text{Heat loss in steam} = \text{mass of the steam} * \text{latent heat of steam}$$

Option A:

3 bar pressure steam having 517 kCal/kg of latent heat

$$\text{Mass of the steam} = 1577500 / 517$$

$$\text{Mass of the steam} = 3051 \text{ kg / hr}$$

$$\text{Cost of steam expenditure} = 3051 * 8000 * 4.5$$

$$\text{Cost of steam expenditure} = 10.98 \text{ Crore BDT}$$

Option B:

8 bar pressure steam having 489 Kcal/kg of latent heat

$$\text{Mass of the steam} = 1577500 / 489$$

$$\text{Mass of the steam} = 3225 \text{ kg / hr}$$

$$\text{Cost of steam expenditure} = 3226 * 8000 * 4.5$$

$$\text{Cost of steam expenditure} = 11.61 \text{ Crore BDT}$$

Recommendation: Option A is recommended as it is found to be economical (Steam @ 3 bar pressure) since the expenditure per annum is less when compared to Option B.

Example 3.9

In a process plant, 20 TPH of steam after pressure reduction with pressure reducing valve to 20 kg/cm² gets superheated. The temperature of steam is 280°C. The management wants to install a de-super heater to convert superheated steam into saturated steam at 20 kg/cm² for process use, and its saturation temperature is 210°C.

Calculate quantity of water at 30°C to be injected in de-super heater to get the desired saturated steam using the following data.

$$\text{Specific heat of superheated steam} = 0.45 \text{ kCal/Kg}^\circ\text{C}$$

$$\text{Latent heat of steam at } 20 \text{ kg/cm}^2 = 450 \text{ kCal/kg}$$

Solution

$$\begin{aligned} \text{Quantity of heat available above saturation} &= 20,000 \times 0.45 \times (280-210) \\ &= 6,30,000 \text{ kCal} \end{aligned}$$

Quantity of water required in de-superheater

$$= Q \times \{1 \times (210-30) + 450\} = 630000$$
$$= 1000 \text{ kg/hr}$$

Chapter 4: Industrial Furnaces

A furnace is an equipment to melt metals for casting or heat materials for change of shape (rolling, forging etc.) or change of properties (heat treatment).

4.1 Types and Classification of Different Furnaces

Based on the method of generating heat, furnaces are broadly classified into two types namely combustion type (using fuels) and electric type. The combustion type furnace can be broadly classified as oil fired, coal fired or gas fired.

Based on the mode of charging of material furnaces can be classified as (i) Intermittent or Batch type furnace or Periodical furnace and (ii) Continuous furnace.

Based on mode of waste heat recovery as recuperative and regenerative furnaces.

Another type of furnace classification is made based on mode of heat transfer, mode of charging and mode of heat recovery as shown in the figure 4.1 below.

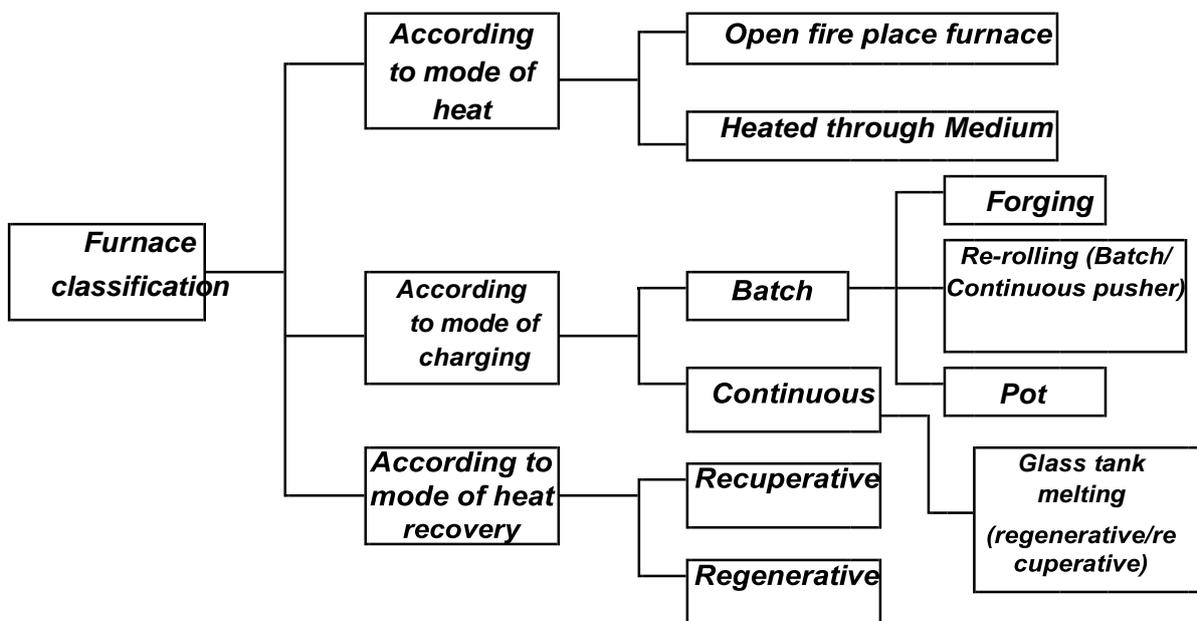


Figure 4.1: Furnace Classification

The electric furnaces can be broadly classified as resistance type for heating and induction and arc furnace for melting of metals.

Characteristics of an Efficient Furnace

Furnace should be designed so that in a given time, as much of material as possible can be heated to a uniform temperature as possible with the least possible fuel and labour. To achieve this end, the following parameters can be considered.

- Determination of the quantity of heat to be imparted to the material or charge.
- Liberation of sufficient heat within the furnace to heat the stock and overcome all heat losses.

- Transfer of available part of that heat from the furnace gases to the surface of the heating stock.
- Equalization of the temperature within the stock.
- Reduction of heat losses from the furnace to the minimum possible extent.

Furnace Energy Supply

Since the products of flue gases directly contact the stock, type of fuel chosen is of importance. For example, some materials will not tolerate sulphur in the fuel. Also use of solid fuels will generate particulate matter, which will interfere the stock placed inside the furnace. Hence, vast majority of the furnaces use liquid fuel, gaseous fuel or electricity as energy input. Electricity is used in induction and arc furnaces for melting steel and cast iron. Non-ferrous melting utilizes oil as fuel.

Oil Fired Furnace

Furnace oil is the major fuel used in oil fired furnaces, especially for reheating and heat treatment of materials. LDO is used in furnaces where presence of sulphur is undesirable. The key to efficient furnace operation lies in complete combustion of fuel with minimum excess air.

Furnaces operate with efficiencies as low as 7% as against upto 90% achievable in other combustion equipment such as boiler. This is because of the high temperature at which the furnaces have to operate to meet the required demand. For example, a furnace heating the stock to 1200°C will have its exhaust gases leaving at least at 1200°C resulting in a huge heat loss through the stack. However, improvements in efficiencies have been brought about by methods such as preheating of stock, preheating of combustion air and other waste heat recovery systems.

Typical Furnace System

i) Forging Furnaces

The forging furnace is used for preheating billets and blooms to attain a 'forge' temperature. The furnace temperature is maintained at around 1200 to 1250 °C. Forging furnaces use an open fireplace system and most of the heat is transmitted by radiation.

The total operating cycle can be divided into (a) heat up time (b) soaking time and (c) forging time.

Normally, large pieces are soaked for 4 to 6 hrs inside the furnace to attain uniform temperature throughout the cross-section of the material. The actual soaking time varies with the type and thickness of the material. The completely soaked material is withdrawn from furnace to the hammer to be forged as required. Larger pieces may have to be reheated several times. A typical forging furnaces is illustrated in Figure 4.2.



Figure 4.2 Forging Furnace

The charging and discharging of the material is done manually and this results in significant heat loss during the forging operation. Specific fuel consumption depends upon the type of material and number of 'reheats' required.

ii) Rerolling Mill Furnace

a) Batch type

A box type furnace is employed for batch type rerolling mill. The furnace is basically used for heating up scrap, small ingots and billets weighing 2 to 20 kg for rerolling. The charging and discharging of the 'material' is done manually and the final product is in the form of rods, strips etc. The operating temperature is about 1200 °C. The total cycle time can be further categorized into heat-up time and rerolling time. During heat-up time the material gets heated upto the required temperature and is removed manually for rerolling. The average output from these furnaces varies from 10 to 15 tonnes / day.

b) Continuous Pusher Type

The process flow and operating cycles of a continuous pusher type is the same as that of the batch furnace. The operating temperature is about 1250 °C. Generally, these furnaces operate 8 to 10 hours with an output of 50 to 100 tons per day. The material or stock recovers a part of the heat in flue gases as it moves down the length of the furnace. Heat absorption by the material in the furnace is slow, steady and uniform throughout the cross-section compared with batch type.

iii) Continuous Steel Reheating Furnaces

The main function of a reheating furnace is to raise the temperature of a piece of steel, typically to between 900 °C and 1250 °C, until it is plastic enough to be pressed or rolled to the desired section, size or shape. The furnace must also meet specific requirements and objectives in terms of stock heating rates for metallurgical and productivity reasons. In continuous reheating, the steel stock forms a continuous flow of material and is heated to the desired temperature as it travels through the furnace.

All furnaces possess the features shown in Figure 4.3

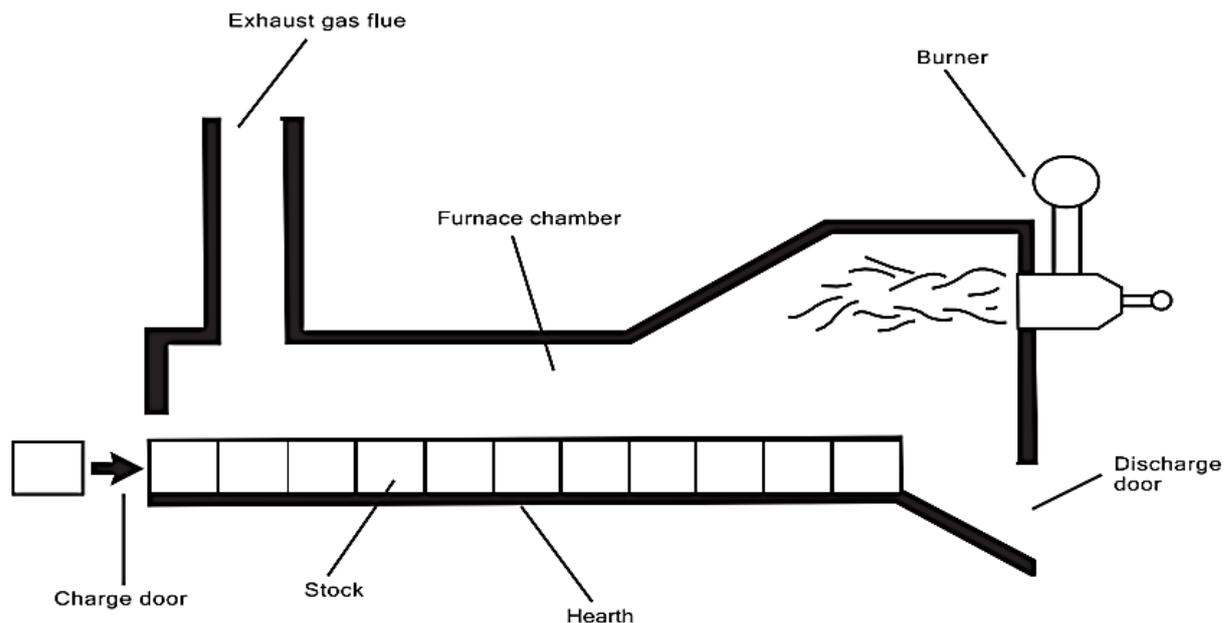


Figure 4.3 Furnace Feature

- A refractory chamber constructed of insulating materials for retaining heat at the high operating temperatures.
- A hearth to support or carry the steel. This can consist of refractory materials or an arrangement of metallic supports that may be water-cooled.
- Burners that use liquid or gaseous fuels to raise and maintain the temperature in the chamber. Coal or electricity can be used for reheating. A method of removing the combustion exhaust gases from the chamber
- A method of introducing and removing the steel from the chamber.
- These facilities depend on the size and type of furnace, the shape and size of the steel being processed, and the general layout of the rolling mill.
- Common systems include roller tables, conveyors, charging machines and furnace pushers.

Heat Transfer in Furnaces

The main ways in which heat is transferred to the steel in a reheating furnace are shown in Figure 4.4. In simple terms, heat is transferred to the stock by:

- ✓ Radiation from the flame, hot combustion products and the furnace walls and roof;

✓ Convection due to the movement of hot gases over the stock surface.

At the high temperatures employed in reheating furnaces, the dominant mode of heat transfer is wall radiation. Heat transfer by gas radiation is dependent on the gas composition (mainly the carbon dioxide and water vapour concentrations), the temperature and the geometry of the furnace.

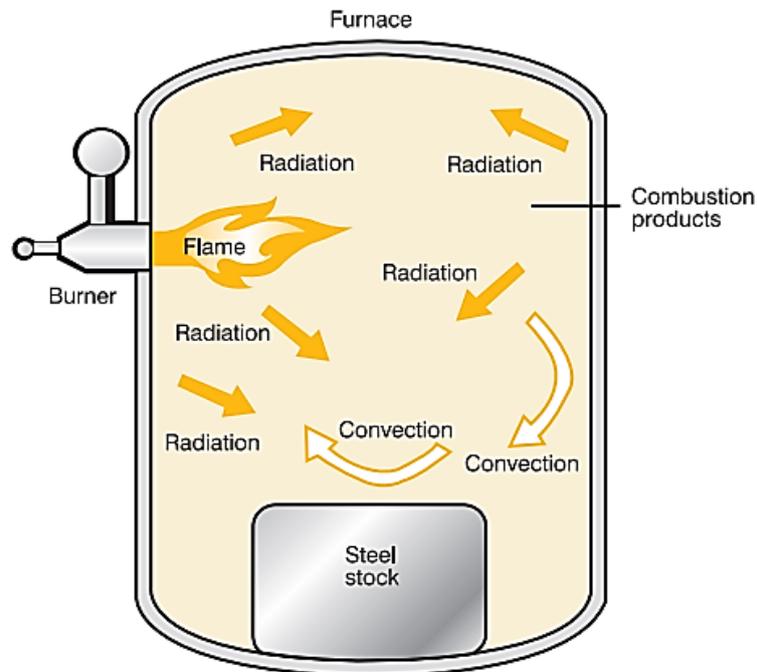


Figure 4.4: Heat Transfer in furnace

Types of Continuous Reheating Furnace

Continuous reheating furnaces are primarily categorised by the method by which stock is transported through the furnace. There are two basic methods:

- Stock is butted together to form a stream of material that is pushed through the furnace. Such furnaces are called pusher type furnaces.
- Stock is placed on a moving hearth or supporting structure which transports the steel through the furnace. Such types include walking beam, walking hearth, rotary hearth and continuous recirculating bogie furnaces.

The major consideration with respect to furnace energy use is that the inlet and outlet apertures should be minimal in size and designed to avoid air infiltration.

i) Pusher Type Furnaces

The pusher type furnace is popular in steel industry. It has relatively low installation and maintenance costs compared to moving hearth furnaces. The furnace may have a solid hearth, but it is also possible to push the stock along skids with water-cooled supports that allow both the top and bottom faces of the stock to be heated. The design of a typical pusher furnace design is shown schematically in Figure 4.5.

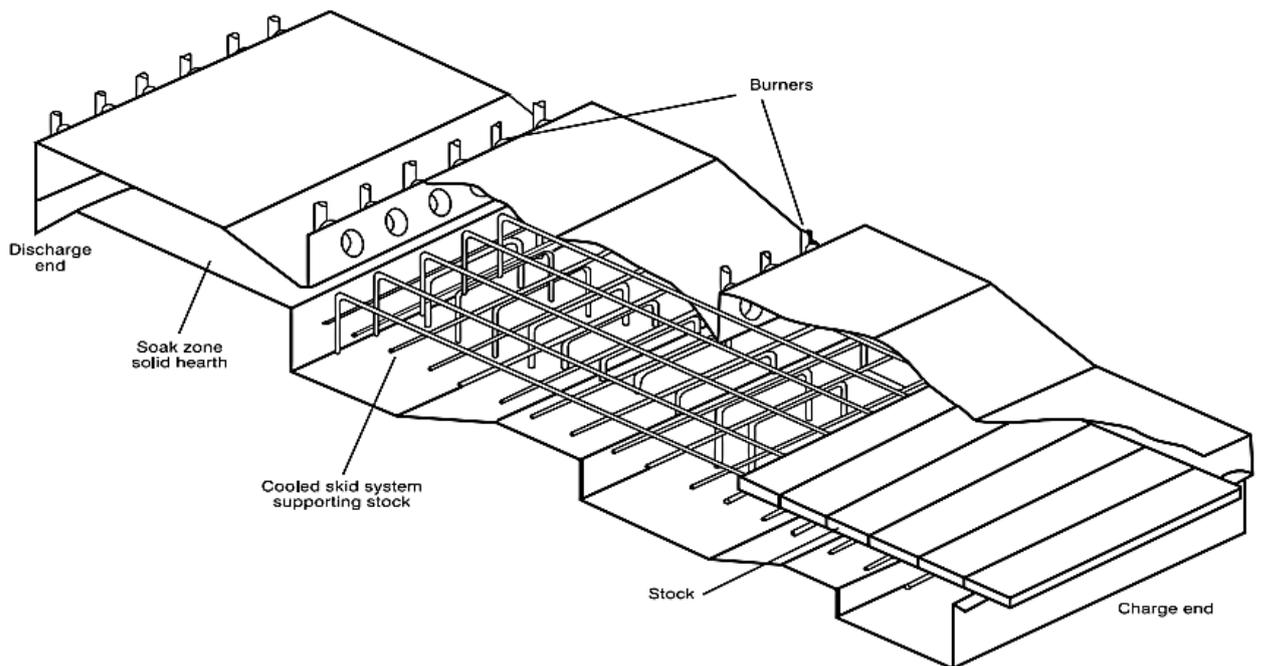


Figure 4.5: Pusher Type Furnace

Pusher type furnaces, however, do have some disadvantages, including:

- ✓ Frequent damage of refractory hearth and skid marks on material
- ✓ Water cooling energy losses from the skids and stock supporting structure in top and bottom fired furnaces have a detrimental effect on energy use;
- ✓ Discharge must be accompanied by charge:
- ✓ Stock sizes and weights and furnace length are limited by friction and the possibility of stock pile-ups.
- ✓ All round heating of the stock is not possible.

ii) Walking Hearth Furnaces

The walking hearth furnace (Figure.4.6) allows the stock to be transported through the furnace in discrete steps. Such furnaces have several attractive features, including: simplicity of design, ease of construction, ability to cater for different stock sizes (within limits), negligible water cooling energy losses and minimal physical marking of the stock. The main disadvantage of walking hearth furnaces is that the bottom face of the stock cannot be heated. This can be alleviated to some extent by maintaining large spaces between pieces of stock. Small spaces between the individual stock pieces limits the heating of the side faces and increases the potential for unacceptable temperature differences within the stock at discharge. Consequently, the stock residence time may be long, possibly several hours; this may have an adverse effect on furnace flexibility and the yield may be affected by scaling.

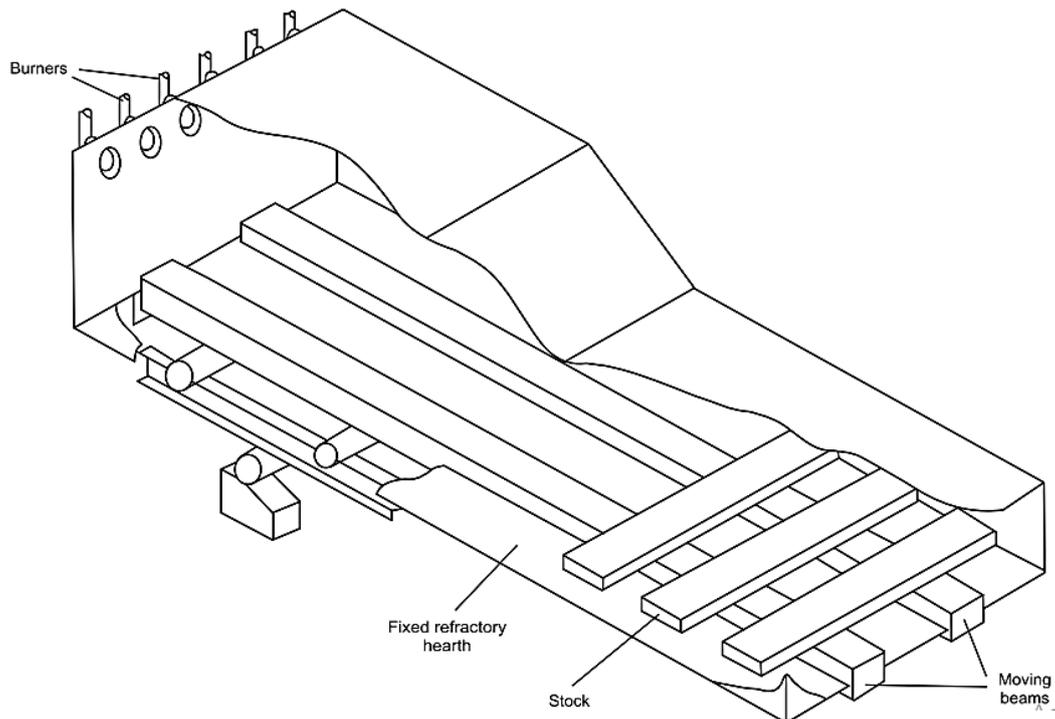


Figure 4.6 Walking Hearth Type Furnace

iii) Rotary hearth furnace

The rotary hearth furnace (Figure 4.7) has tended to supersede the re-circulating bogie type. The heating and cooling effects introduced by the bogies are eliminated, so heat storage losses are less. The rotary hearth has, however a more complex design with an annular shape and revolving hearth.

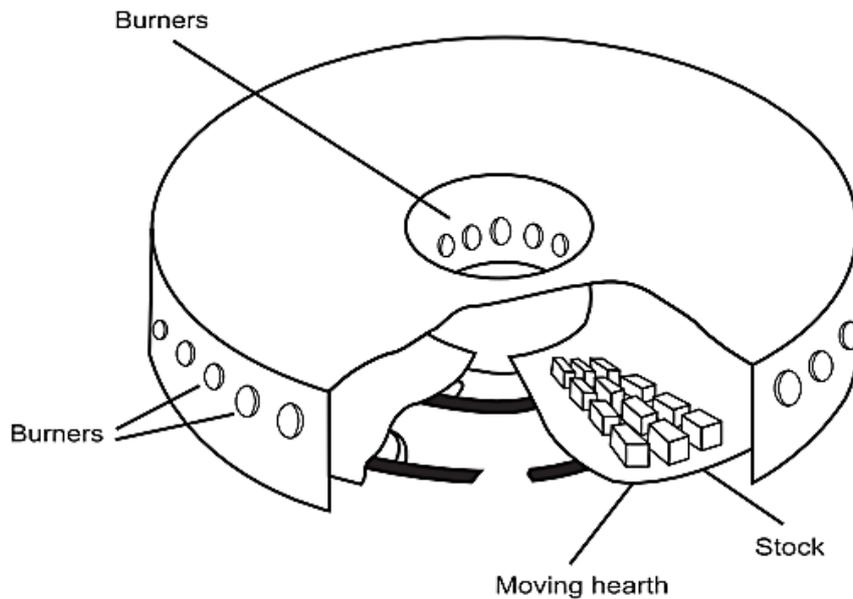


Figure 4.7 Rotary Hearth Furnace

iv) Continuous Recirculating Bogie type Furnaces

These types of moving hearth type furnaces tend to be used for compact stock of variable size and geometry. In bogie furnaces (Figure 4.8), the stock is placed on a bogie with a refractory hearth, which travels through the furnace with others in the form of a train. The entire furnace length is always occupied by bogies. Bogie furnaces tend to be long and narrow and to suffer from problems arising from inadequate sealing of the gap between the bogies and furnace shell, difficulties in removing scale, and difficulties in firing across a narrow hearth width.

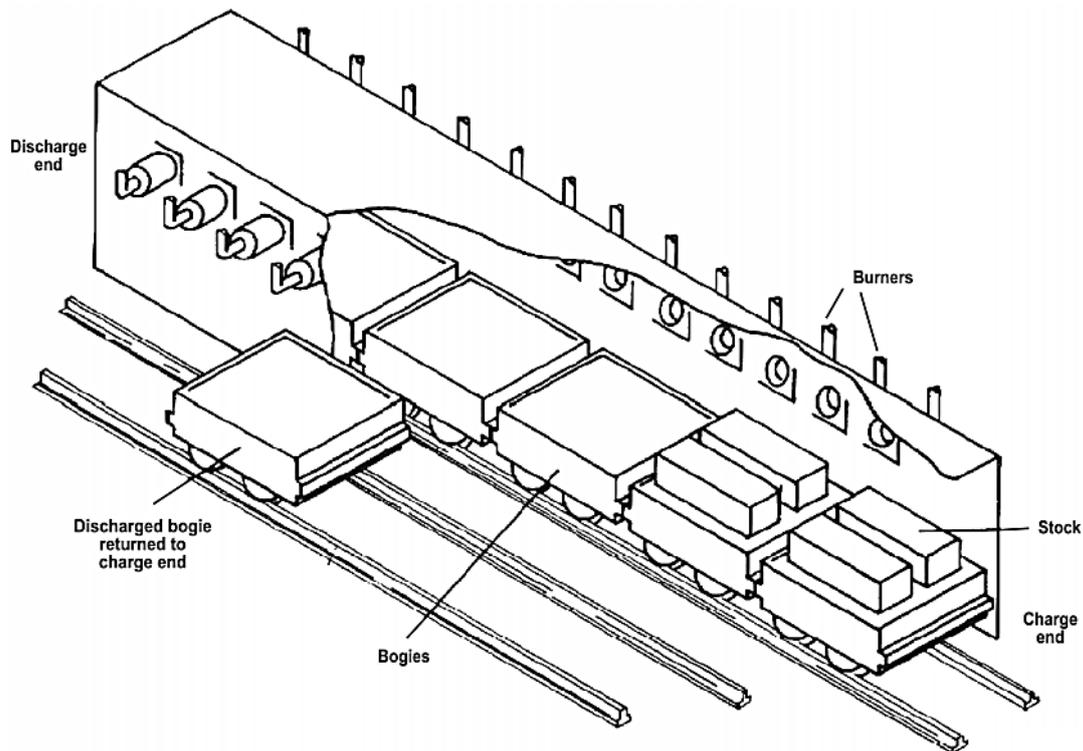


Figure 4.8 Continuous Circulating Bogie Type Furnace

v) Walking Beam Furnaces:

The walking beam furnace (Figure 4.9) overcomes many of the problems of pusher furnaces and permits heating of the bottom face of the stock. This allows shorter stock heating times and furnace lengths and thus better control of heating rates, uniform stock discharge temperatures and operational flexibility. In common with top and bottom fired pusher furnaces, however, much of the furnace is below the level of the mill; this may be a constraint in some applications.

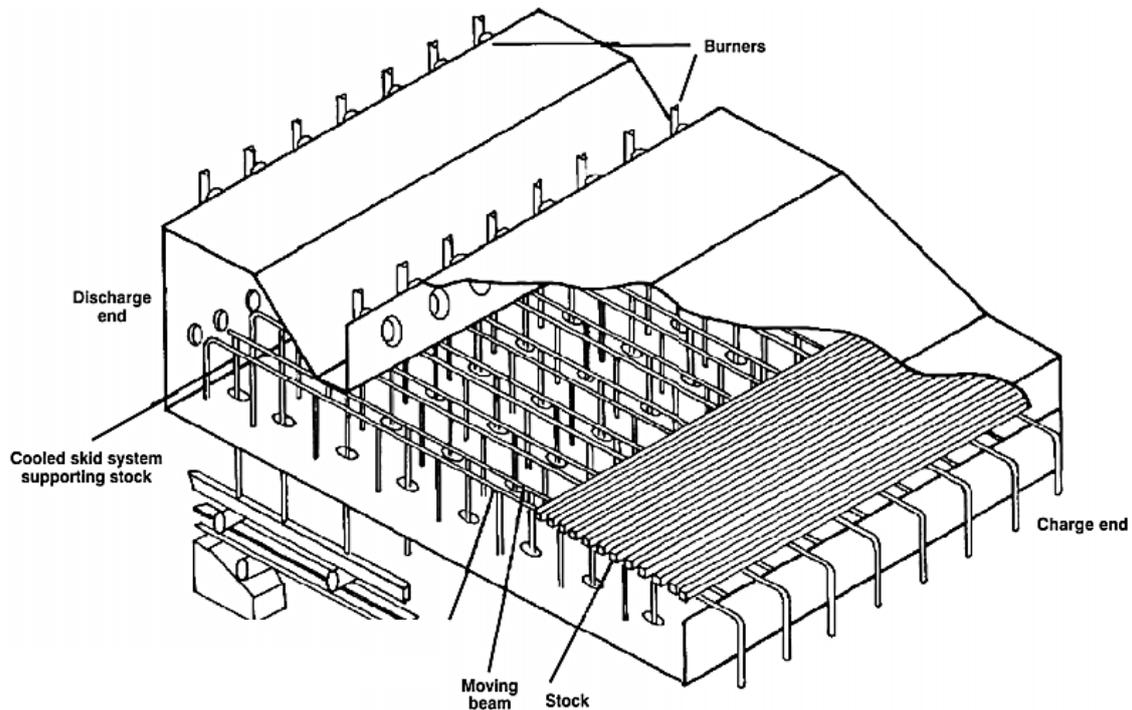


Figure 4.9 Walking Beam Furnace

vi) Cupola Furnace

A cupola furnace (Figure 4.10) is tall, cylindrical cast iron, foundry returns. The charge used in cupola furnace consists of alternate layers of coke, flux and metal (iron). These three components are continuously built into the cupola furnace. The most commonly used iron - to - coke ratio is 8:1. The flux may be limestone (CaCO_3), fluorspar, sodium carbonate or calcium carbide. Limestone is the commonly employed flux. The total weight of the flux will be approximately 1/5th the weight of the coke charge. Sufficient air is passed through the tuyeres for proper combustion of coke.

Melting in Cupola Furnace

Cupola furnace works on the counter current principle. As the combustion takes place, the charge materials (coke, flux and metal) will be descending downwards, while the hot gases due to combustion will be ascending upwards. Heat exchange takes place between the rising hot gases and the descending charge thereby melting the metal. The liquid metal drops down, while the coke floats up on top of it.

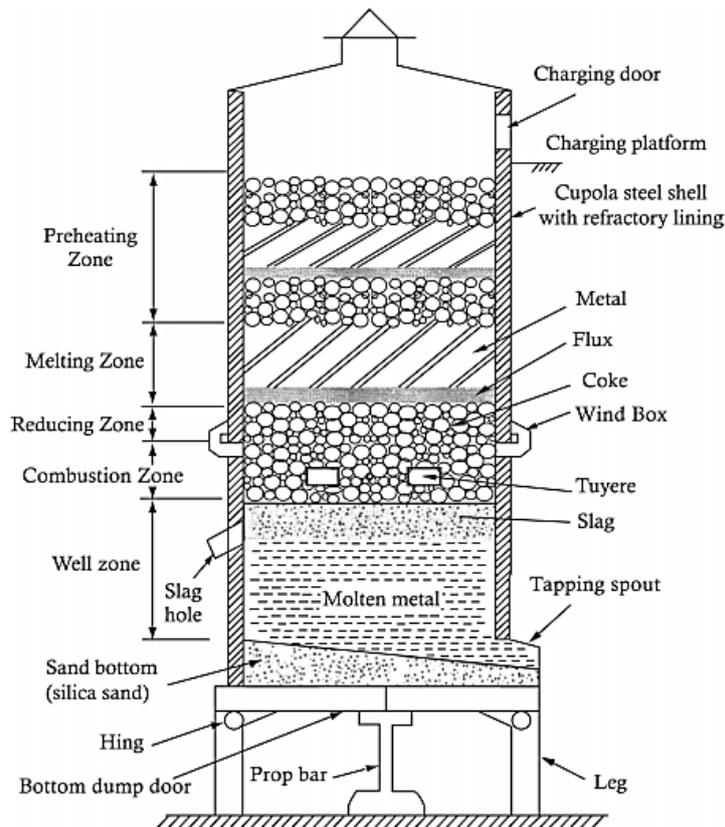


Figure 4.10 Cupola Furnace

The flux also melts and reacts with the impurities of the molten metal forming a slag. The slag floats on the surface of the molten metal thereby preventing oxidation of the metal.

Zones of Cupola Furnace

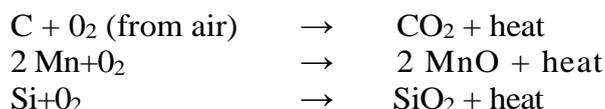
Well zone in Cupola Furnace

Well zone in the portion situated between the rammed sand bottom and just below the bottom edge of the tuyeres. The molten metal is occupied in this zone.

Combustion Zone in Cupola Furnace

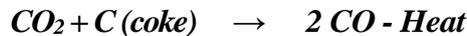
The combustion zone or oxidizing zone is situated normally 15-30 cm from the bottom edge of the tuyeres. It is in this zone where rapid combustion of coke takes place due to which a lot of heat is generated in the furnace. The combustion is rapid due to the supply of blast air through the tuyeres.

Oxidation of manganese and silicon evolve still more heat. The reactions which take place in this zone are:



The temperature in this zone varies from 1541 °C -1870 °C

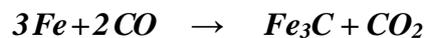
Reducing zone or protection zone is the portion located from the top of the combustion zone to the top of the coke bed. In this zone, some of the hot CO₂ gas moving upward through the hot coke gets reduced to CO. In other words, reduction of CO₂ to CO occurs in this zone. Due to the reducing atmosphere, the charge is protected from oxidation. The reaction taking place in this zone is given by.



Due to the reduction, the temperature reduces to around 1200 °C in this zone.

Melting Zone in Cupola Furnace

The portion located just above the coke bed to the top of the metal (iron) is called the melting zone. The metal starts melting in this zone and trickles down through the coke bed to the well zone. The molten iron while passing down through the reducing zone picks up carbon and the reaction is given by:



Preheating zone in Cupola Furnace

The portion occupied from the top surface of the melting zone to the charging door is called 'Preheating zone'. The hot gases rising upwards from the combustion and reducing zone gives its heat to the charge before passing out of the furnace. Thus, the charge is preheated before descending downwards

Hot Blast Cupola

The temperature of exhaust gas of a cupola is as high as 800 °C, making it possible to preheat blast air up to as high as 400 °C by heat exchange. In addition, both the sensible and latent heat of exhaust gas can be recycled for preheating blast air by combustion of CO gas included in exhaust gas. When blast air is preheated to 300 °C or higher, the sensible heat of blast air is added to heat input, activating combustion of coke, leading to the rise in combustion temperature.

vii) Induction Furnaces

Induction furnaces are ideal for melting and alloying a wide variety of metals with minimum melt losses, however, little refining of the metal is possible. There are two main types of induction furnace: coreless and channel, the principle of operation of which are the same.

Coreless Induction Furnace

Coreless induction furnace (Figure 4.11) consists of: a water cooled helical coil made of a copper tube, a crucible installed within the coil and supporting shell equipped with trunnions on which the furnace may tilt. Alternating current passing through the coil induces alternating currents in the metal charge loaded to the crucible. These induced currents heat the charge.

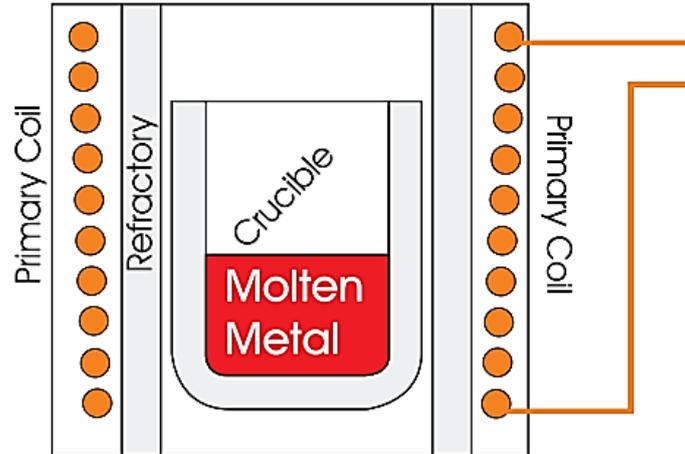


Figure 4.11 Induction Furnace

When the charge is molten, electromagnetic field produced by the coil interacts with the electromagnetic field produced by the induced current. The resulted force causes stirring effect helping homogenizing the melt composition and the temperature.

The frequency of the alternating current used in induction furnaces may vary from the line frequency (50 Hz or 60 Hz) to high frequency 10,000 Hz .

The total absolute energy required to melt one tonne of different metals at different molten temperature is given in table 4.1.

Table 4.1: Energy Required to Melt One Ton of Different Metals

Type of metal	Specific heat require	Latent heat require	Total require kWh/Ton
Mild steel @ 1650 °C Melting temp.	$1000 \times 0.682 \times 1620 \text{ }^\circ\text{C} \div 3600 \text{ kWh}$ $\Delta T = 1650^\circ\text{C} - 30^\circ\text{C}$ Specific heat = 0.9 kJ/kg °C = 307 kWh	$272 \times 1000 \div 3600 \text{ kWh}$ Latent Heat = 272 kJ/kg = 76 kWh	$307 \text{ kWh} + 76 \text{ kWh}$ = 376 kWh
Aluminum @ 710 °C Melting temp.	$1000 \times 0.9 \times 680 \text{ }^\circ\text{C} \div 3600 \text{ kWh}$ $\Delta T = 710 \text{ }^\circ\text{C} - 30 \text{ }^\circ\text{C}$ Specific heat = 0.9 kJ/kg °C = 170 kWh	$396.9 \times 1000 \div 3600 \text{ kWh}$ Latent Heat = 396.9 kJ/kg = 110 kWh	$170 \text{ kWh} + 110 \text{ kWh}$ = 180 kWh
Copper @ 1130 °C Melting temp.	$1000 \times 0.386 \times 1100 \text{ }^\circ\text{C} \div 3600 \text{ kWh}$ $\Delta T = 1130 \text{ }^\circ\text{C} - 30 \text{ }^\circ\text{C}$ Specific heat = 0.386 kJ/kg °C = 118 kWh	$212 \times 1000 \div 3600 \text{ kWh}$ Latent Heat = 212 kJ/kg = 59 kWh	$118 \text{ kWh} + 59 \text{ kWh}$ = 117 kWh
Gold @ 1130 °C Melting temp.	$1000 \times 0.131 \times 1130 \text{ }^\circ\text{C} \div 3600 \text{ kWh}$ $\Delta T = 1130 \text{ }^\circ\text{C} - 30 \text{ }^\circ\text{C}$ Specific heat = 0.131 kJ/kg °C = 36.38 kWh	$67.62 \times 1000 \div 3600 \text{ kWh}$ Latent Heat = 67.62 kJ/kg = 18.78 kWh	$36.38 \text{ kWh} + 18.78 \text{ kWh}$ = 56 kWh

Furnace Efficiency

$$\text{Efficiency, \%} = \frac{\text{Theoretical total heat required for melting } (H_T), \text{ kWh}}{\text{Actual Electricity consumed for melting } (H_A), \text{ kWh}} \times 100$$

Theoretical heat required for melting (H_T)

$$\text{Heat required for melting metal, } H_1, \text{ kWh} = \frac{W_m \times (C_p \times (T_2 - T_1) + h)}{3600}$$

(3600 kJ = 1 kWh)

$$\text{Heat required for melting slag, } H_2, \text{ kWh} = \frac{1.65 \times W_s}{3.6}$$

(3.6 MJ = 1 kWh)

Where,

- W_m - Weight of the metal, kg
- W_s - Weight of the slag, kg
- C_p - Specific heat of metal, kJ/kg °C
- T_2 - **Final temperature of the metal**
- T_1 - Initial or charge temperature of the metal

Total theoretical heat required for melting, $H_T = H_1 + H_2$

Actual electricity consumed for melting (H_A)

The actual consumption of electricity for melting can be measured from the input busbar to the furnace. the difference between Actual and theoretical values will be loss due to conduction, radiation and other losses.

Example 4.1

Calculate the furnace efficiency from the data given below

Specific heat	- 0.682 kJ/kg °C
Latent heat	- 272 kJ/kg
Melting temperature	- 1650 °C
Charge temperature	- 30 °C
Quantity of metal	- 1000 kg
Quantity of slag	- 25 kg
Electricity consumed	- 625 kWh

Solution

$$\begin{aligned} \text{Heat required for melting metal, } H_1, \text{ kWh} &= \frac{1000 \times (0.682 \times (1650 - 30) + 272)}{3600} \\ &= 382.45 \text{ kWh} \end{aligned}$$

$$\begin{aligned} \text{Heat required for melting slag, } H_2, \text{ kWh} &= \frac{1.65 \times 25}{3.6} \\ &= 11.45 \text{ kWh} \end{aligned}$$

$$\begin{aligned} \text{Total theoretical heat required for melting, } H_T &= 382.45 + 11.45 \\ &= 394 \text{ kWh} \end{aligned}$$

$$\begin{aligned} \text{Efficiency, \%} &= \frac{394, \text{ kWh}}{625, \text{ kWh}} \times 100 \\ &= 63 \% \end{aligned}$$

Distribution of losses in induction furnace

Losses in induction furnace

The theoretical energy require to melt one Ton of steel is 385 To 400 kWh/Ton. However in actual practice, the specific energy consumption is remarkably higher to 550 - 950 kWh/ton.

1. Power loss in generator / panel = 2 - 4 %
2. Power loss in capacitor Bank = 1.0 - 3 %
3. Power loss in Crucible = 18 - 25 % (Water cooled cables, Bus bar, and change over switches)
4. Radiation loss = 7 - 9%

Factors affecting the furnace efficiency

The factors, which affect the furnace efficiency, are as under;

- Due to poor maintenance the total production stops sometimes. Higher breakdown results in increasing the ' cost of production
- Due to low supply voltage the furnace draws less power, causes slow melting and inefficient operation resulting increase in production cost.
- Sometimes lining material selection is wrong with respect to metal. Basic lining is better conductor of heat compared to acidic lining. Wrong lining selection increases breakdown, Furnace down time and furnace losses, resulting in inefficient operation.
- Poor coordination between melting staff & contractor
- Absence of material handling equipment
- Poor molding efficiency, so furnace on hold
- Poor quality scrap, reducing lining life, takes more time to melt.
- Absence of thermal insulation between lining & coil.

Hot Air Generator

Hot air at a wide range of temperatures and pressures is produced for applications like foundry sand drying, shell sand coating, core drying rooms or ovens, drying and processing of ores and minerals, food processing, tea drying, seed drying and paint drying. Furnace oil, HSD, LDO, LPG or natural gas may be used.

Hot air generators supply air heated to elevated temperature by mixing it with products of combustion from a burner. The hot air generator consists of an inner refractory chamber, venturi section where hot products of combustion mix with dilution air, and outlet section.

The combustion chamber is lined with firebrick, while the mixing chamber and the outlet section are lined with insulation brick. Air and oil pipelines required for combustion are included. The combustion equipment comprises of burner with burner block, mounting plate, isolating valve, and oil pumping and heating unit.

4.2 Performance Evaluation of a Fuel Fired Furnace

The fuel required for combustion is cleaned, preheated and burnt in the combustion zone of the furnace. Thermal efficiency of the furnaces is the ratio of heat delivered to a material stock and heat supplied to the heating equipment. The purpose of a heating process is to introduce a certain amount of thermal energy into a material stock / product, raising it to a certain temperature to prepare it for additional processing or change its properties.

This results in energy losses in different areas and forms as shown in san key diagram (Figure 4.2).

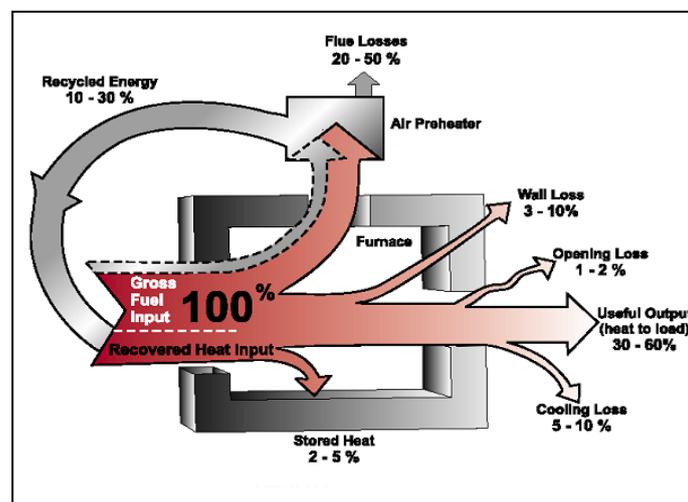


Figure 4.12 Fuel Fired Furnace

The major losses that occur in the fuel fired furnaces are listed below.

1. Heat lost through exhaust gases either as sensible heat or as incomplete combustion
2. Heat loss through furnace walls and hearth
3. Heat loss to the surroundings by radiation and convection from the outer surface of the walls
4. Heat loss through gases leaking through cracks, openings and doors.

Economy in fuel can be achieved if the total heat that can be passed on to the stock is as large as possible.

Furnace Efficiency

Thermal efficiency of a furnace is determined either by direct or indirect method of evaluation.

Direct method

The efficiency of furnace can be assessed by measuring the amount of heat added to the stock and the heat in the fuel consumed, on a batch/day basis as relevant.

$$\text{Thermal efficiency of the furnace} = \frac{\text{Heat in the stock}}{\text{Heat in the fuel consumed for heating the stock}} \times 100$$

The quantity of heat to be imparted (Q_s) to the stock can be found from the following relation;

$$Q_s = m \times C_p \times (t_1 - t_2)$$

where,

Q_s	=	Quantity of heat imparted to the stock in kCal
m	=	Weight of the stock in kg
C_p	=	Mean specific heat of stock in kCal/kg $^{\circ}$ C
t_1	=	Final temperature of stock desired, $^{\circ}$ C
t_2	=	Initial temperature of the stock before it enters the furnace, $^{\circ}$ C

Heat in fuel = Quantity of fuel (q) in kg/h X GCV in kCal/kg

Example 4.2

The following are the operating parameters of rerolling mill furnace

Weight of input material	- 10 T/hr
Furnace oil consumption	- 600 litres/hr
Specific gravity of oil	- 0.92
Final material temperature	- 1200 $^{\circ}$ C
Initial material temperature	- 40 $^{\circ}$ C
Outlet flue gas temperature	- 650 $^{\circ}$ C
Specific heat of the material	- 0.12 kcal/kg $^{\circ}$ C
GCV of oil	- 10,000 kCal/kg
Percentage yield	- 92 %

- Calculate furnace efficiency by direct method
- Calculate Specific fuel consumption on finished product basis

Solution

a) Furnace efficiency by direct method

Heat input	$600 \text{ lit/hr} \times 0.92 \times 10000 = 55,20,000 \text{ kcal/hr}$
Heat output	$10,000 \times 0.12 \times (1200 - 40) = 1,39,2000 \text{ kcal/hr}$
Efficiency	$1,39,2000 / 55,20,000 = 25.2 \%$

b) Specific fuel consumption on finished product basis

Weight of finished products	$10 \times 0.92 = 9.2 \text{ T/hr}$
Furnace oil consumption	600 litres/hr
Specific fuel consumption	$600/9.2 = 65.2 \text{ litres/ton}$

Indirect Method

Similar to the method of evaluating boiler efficiency by indirect method, furnace efficiency can also be calculated by indirect methods. Furnace efficiency is calculated after subtracting sensible heat loss in flue gas, loss due to moisture in flue gas, heat loss due to openings in furnace, heat loss through furnace skin and other unaccounted losses.

In order to find out furnace efficiency using indirect method, various parameters that are required are hourly furnace oil consumption, material output, excess air quantity, temperature of flue gas, temperature of furnace at various zones, skin temperature and hot combustion air temperature. Instruments like infrared thermometer, fuel consumption monitor, surface thermocouple and other measuring devices are required to measure the above parameters.

Typical thermal efficiencies for common industrial furnaces are given in Table 4.1.

Table 4.2: Thermal efficiencies for common industrial furnaces

Furnace Type	Typical thermal efficiency (%)
1) Low Temperature furnaces	
a. 540 – 980 °C (Batch type)	20-30
b. 540 – 980 °C (Continuous type)	15-25
c. Coil Anneal (Bell) radiant type	4-7
d. Strip Anneal Muffle	7-12
2) High temperature furnaces	
Slot forge	5-12
b. Pusher, Roll down or Rotary	7-14
c. Batch forge	5-10
d. Car Bottom	7-12
3) Continuous Kiln	
a. Hoffman	25-93
b. Tunnel	21-82
c. Transverse-arch Annular	26-96
4) Ovens	
a. Indirect fired ovens (20°C-370°C)	35-40
b. Direct fired ovens (20°C-370°C)	35-40

Energy Balance in a Typical Reheating Furnace

The heat inputs and outputs are calculated (as per JIS GO702) on the basis of per tonne of stock or product output and simplified.

Example 2 Furnace Efficiency Calculation (Indirect Method)

An oil-fired reheating furnace has an operating temperature of around 1340°C. Average fuel consumption is 400 litres/hour. The flue gas exit temperature after air pre heater is 650 °C. Air is preheated from ambient temperature of 40°C to 190°C through an air pre-heater. The furnace has 460 mm thick wall (x) on the billet extraction outlet side, which is 1 m high (D) and 1 m wide. Evaluate the heat balance to identify heat losses, efficiency and specific fuel consumption. The other data are as follows.

Specific gravity of oil	=	0.92
Average fuel oil consumption	=	400 Litres / hr
	=	400 x 0.92 = 368 kg/hr
Theoretical air	=	14.12 kg air /kg oil
O ₂ in flue gas	=	12%
CO ₂ in flue gas	=	6.5%
CO in flue gas	=	50 ppm
Weight of stock/billet	=	6000 kg/hr
Abs. humidity	=	0.03437 kg/kg dry air
Preheated oil temperature	=	100 °C
Specific heat of oil	=	0.5 kcal/kg°C
Specific heat of billet	=	0.12 kcal/kg°C
Specific heat of flue gas in kcal/kg °C	=	0.26 kcal/kg °C
C _p of super-heated vapour	=	0.47 kcal/kg °C
Surface temperature of ceiling	=	85 °C
Surface temperature of side walls	=	100 °C
Surface temperature of flue duct	=	64 °C
Area of ceiling	=	15 m ²
Area of side walls	=	36 m ²
Area of flue duct	=	10.3 m ²
Diameter of flue duct	=	0.4 m

Furnace oil constituents (% by weight)

Carbon - 85.9 %, Hydrogen -12%, Oxygen - 0.7%, Nitrogen - 0.5%, Sulphur - 0.5%, Moisture - 0.35%, Ash - 0.05%, GCV- 10,000 kcal/kg

Solution

1) Calculation of air quantity and specific fuel consumption:

$$\begin{aligned}\% \text{ Excess Air supplied} &= \frac{\text{O}_2\%}{21 - \text{O}_2\%} \times 100 \quad [\text{from flue gas analysis}] \\ &= 12 / (21 - 12) \\ &= 133.3 \%\end{aligned}$$

$$\begin{aligned}
\text{Total mass of air supplied/ kg of fuel (AAS)} &= \{1 + EA/100\} \times \text{theoretical air} \\
&= \{1 + (133.3/100)\} \times 14.12 \\
&= 32.94 \text{ kg of air / kg of fuel}
\end{aligned}$$

$$\begin{aligned}
\text{Amount of dry flue gas (m)} &= \text{Amount of wet flue gas} - \text{Amount of water vapour in flue gas} \\
&= (\text{AAS} + 1 \text{ kg of fuel}) - (\text{Moisture} + 9 \text{ Hydrogen}) \\
&= (32.94 + 1) - [(0.35/100) + 9 \times (12/100)] \\
&= 33.94 - 1.084 \\
&= 32.86 \text{ kg dry flue gas /kg of fuel}
\end{aligned}$$

$$\begin{aligned}
\text{Specific fuel consumption (F)} &= \text{Amount of fuel consumed (kg/hr)} / \text{Amount of billet (t/hr)} \\
&= 368 / 6 \\
&= 61.33 \text{ kg of fuel / tonne of billet}
\end{aligned}$$

2) Calculation of Heat Input:

$$\begin{aligned}
\text{Combustion heat of fuel (Q}_1\text{)} &= \text{Amount of fuel consumed per tonne of billet} \times \text{GCV of fuel} \\
&= 61.3 \times 10,000 \\
&= 6,13,300 \text{ kcal/tonne of billet}
\end{aligned}$$

$$\begin{aligned}
\text{Sensible heat of fuel (Q}_2\text{)} &= F \times C_{p \text{ fuel}} [t_f - t_a] \\
&= 61.33 \times 0.5 \times (100 - 40) \\
&= 1840 \text{ kcal/tonne of billet}
\end{aligned}$$

$$\begin{aligned}
\text{Total heat input} &= Q_1 + Q_2 \\
&= 6,13,300 + 1840 \\
&= 6,15,140 \text{ kcal/tonne of billet}
\end{aligned}$$

3) Calculation of Heat Output:

$$\begin{aligned}
\text{Heat carried away by 1 tonne of billet (Q}_3\text{)} &= 1000 \text{ kg/t} \times C_p \times (T_0 - T_i) \\
&= 1000 \times 0.12 \times (1340 - 40) \\
&= 1,56,000 \text{ kcal/tonne of billet}
\end{aligned}$$

$$\begin{aligned}
\text{Heat loss in dry flue gas per tonne of billet (Q}_4\text{)} &= F \times m \times C_{pfg} [t_1 - t_a] \\
&= 61.33 \times 32.86 \times 0.26 \times [650 - 40] \\
&= 3,19,627 \text{ kcal/tonne of billet}
\end{aligned}$$

$$\begin{aligned}
\text{Heat loss due to formation of water vapour from fuel per tonne of billet (Q}_5\text{)} &= F \times M \times \{584 + C_{p \text{ of super heated vapour}} \times [t_1 - t_a]\}
\end{aligned}$$

$$= 61.33 \times 1.084 \times \{584 + 0.47 \times (650 - 40)\}$$

$$= 57,886 \text{ kcal/tonne of billet}$$

Heat loss due to moisture in combustion air (Q_6) = $F \times AAS \times \text{Humidity of air} \times C_p \text{ of super heated vapour} \times [t_1 - t_a]$

$$= 61.33 \times 32.94 \times 0.03437 \times 0.47 \times (650 - 40)$$

$$= 19,907 \text{ kcal/tonne of billet}$$

% Heat loss due to partial conversion of C to CO (Q_7) = $F \times \frac{\%CO \times C}{\%CO + \%CO_2} \times 5654$

$$= 61.33 \times \frac{0.005 \times 0.859}{0.005 + 6.5} \times 5654$$

$$= 229 \text{ kcal/tonne of billet}$$

Amount of heat loss from the furnace body and other sections (Q_8) = $(q_1 + q_2 + q_3 + q_4)$, kcal/hr / Amount of billet (t/hr)

where,

q_1 , Heat loss from the furnace body ceiling surface (horizontal surface facing upward) = $\{h \times \Delta t^{1.25} \times A_i\} + \{4.88 \times \epsilon \times [(T_w/100)^4 - (T_a/100)^4] \times A_i\}$

where,

H = Natural convection heat transfer rate 2.8 (kcal/m² h °C)

Δt = $T_w - T_a = 358 - 313 = 45$

E = Emissivity of the furnace body surface (0.75)

q_1 = $\{2.8 \times 45^{1.25} \times 15\} + \{4.88 \times 0.75 \times [(358/100)^4 - (313/100)^4] \times 15\}$

$$= 8644 \text{ kcal/hr}$$

q_2 , Heat loss from the furnace body sidewall surfaces (vertical surfacing sideways) = $\{h \times \Delta t^{1.25} \times A_i\} + \{4.88 \times \epsilon \times [(T_w/100)^4 - (T_a/100)^4] \times A_i\}$

where,

H = Natural convection heat transfer rate 2.2 (kcal/m² h °C)

Δt = $T_w - T_a = 373 - 313 = 60$

q_2 = $\{2.2 \times 60^{1.25} \times 36\} + \{4.88 \times 0.75 \times [(373/100)^4 - (313/100)^4] \times 36\}$

$$= 26084 \text{ kcal/hr}$$

q_3 , Bottom surface (horizontal surface facing downward) = As the bottom surface is not exposed to the atmosphere, q_3 is ignored in this calculation

q_4 , Heat loss from the flue gas duct between the furnace exit and the air preheater (including heat loss from the external surface of the air preheater)

$$= \{h \times \Delta t^{1.25} / D^{0.25} \times A_i\} + \{4.88 \times \epsilon \times [(T_w/100)^4 - (T_a/100)^4] \times A_i\}$$

where,

H = Natural convection heat transfer rate 1.1 (kcal/m² h °C)

Δt = $T_w - T_a = 337 - 313 = 24$ °K

E = Emissivity of the furnace body surface (0.75)

$$q_4 = [1.1 \times 24^{1.25} / 0.4^{0.25} \times 10.3] + \{4.88 \times 0.75 \times [(337/100)^4 - (313/100)^4] \times 10.3\}$$

$$= 2001 \text{ kcal/hr}$$

$$Q_8 = (q_1 + q_2 + q_3 + q_4), \text{ kcal/hr} / \text{Amount of billet (t/hr)}$$

$$= (8644 + 26084 + 0 + 2001)/6$$

$$= 6,122 \text{ kcal/t}$$

Radiation heat loss through furnace openings (Q_9)

Where,

Hr = Open time during the period of heat balancing

A = Area of an opening in m² = 1 m²

ϕ (from graph) = Co-efficient based on the profile of furnace openings

= Diameter (or) the shortest side / wall thickness = 1 / 0.46 = 2.17

= 0.70 (value corresponding to 2.17 and square shape from graph (Figure 4.3))

$$Q_9 = 1 \times 1 \times 0.70 \times 4.88 \times [(1613/100)^4 - (313/100)^4] / 6$$

$$= 38,485 \text{ kcal/t}$$

Other unaccounted heat losses (Q_{10})

- = Other heat loss will include the following,
- Heat carried away by cooling water in the flue damper
- Heat carried away by cooling water at the furnace access door
- Radiation from the furnace bottom
- Heat accumulated by refractory
- Instrumental error and measuring error & Others

$$Q_{10} \text{ (unaccounted losses)} = (Q_1 + Q_2) - (Q_3 + Q_4 + Q_5 + Q_6 + Q_7 + Q_8 + Q_9)$$

$$= (6,15,140) - (1,56,000 + 3,19,627 + 57886 + 19,907 + 229 + 6,122 + 38,485)$$

$$= 16,884 \text{ kcal/t}$$

$$\text{Total heat output} = Q_3 + Q_4 + Q_5 + Q_6 + Q_7 + Q_8 + Q_9 + Q_{10}$$

$$\begin{aligned}
& 1,56,000 + 3,19,627 + 57886 + 19,907 + 229 + 6,122 + \\
& 38,485 + 16,884 \\
& = 6,15,140 \text{ kcal/t}
\end{aligned}$$

4) Heat balance table

Heat Input			Heat output		
Item	kcal/t	%	Item	kcal/t	%
Combustion heat of fuel (Q ₁)	6,13,300	99.70	Heat carried away by 1 tonne of billet (Q ₃)	1,56,000	25.4
Sensible heat of fuel (Q ₂)	1,840	0.30	Heat loss in dry flue gas per tonne of billet (Q ₄)	3,19,627	52.0
			Heat loss due to formation of water vapour from fuel per tonne of billet (Q ₅)	57,886	9.4
			Heat loss due to moisture in combustion air (Q ₆)	19,907	3.2
			% Heat loss due to partial conversion of C to CO (Q ₇)	229	0.04
			Amount of heat loss from the furnace body and other sections (Q ₈)	6,122	1.0
			Radiation heat loss through furnace openings (Q ₉)	38,485	6.26
			Unaccounted losses (Q ₁₀)	16,884	2.7
Total	6,15,140	100		6,15,140	100

5. Efficiency of furnace (by direct method)

$$\eta_{\text{furnace}} = \frac{\text{Heat carried away by billet, kCal/hr}}{\text{Combustion heat of fuel, kCal/hr}} \times 100$$

$$= \frac{[(6000 \text{ kg/hr} \times 0.12 \text{ kCal/kg } ^\circ\text{C} \times (1340 - 40) ^\circ\text{C}]}{368 \text{ kg/hr} \times 10000 \text{ kCal/kg}} \times 100$$

$$= 25.4\%$$

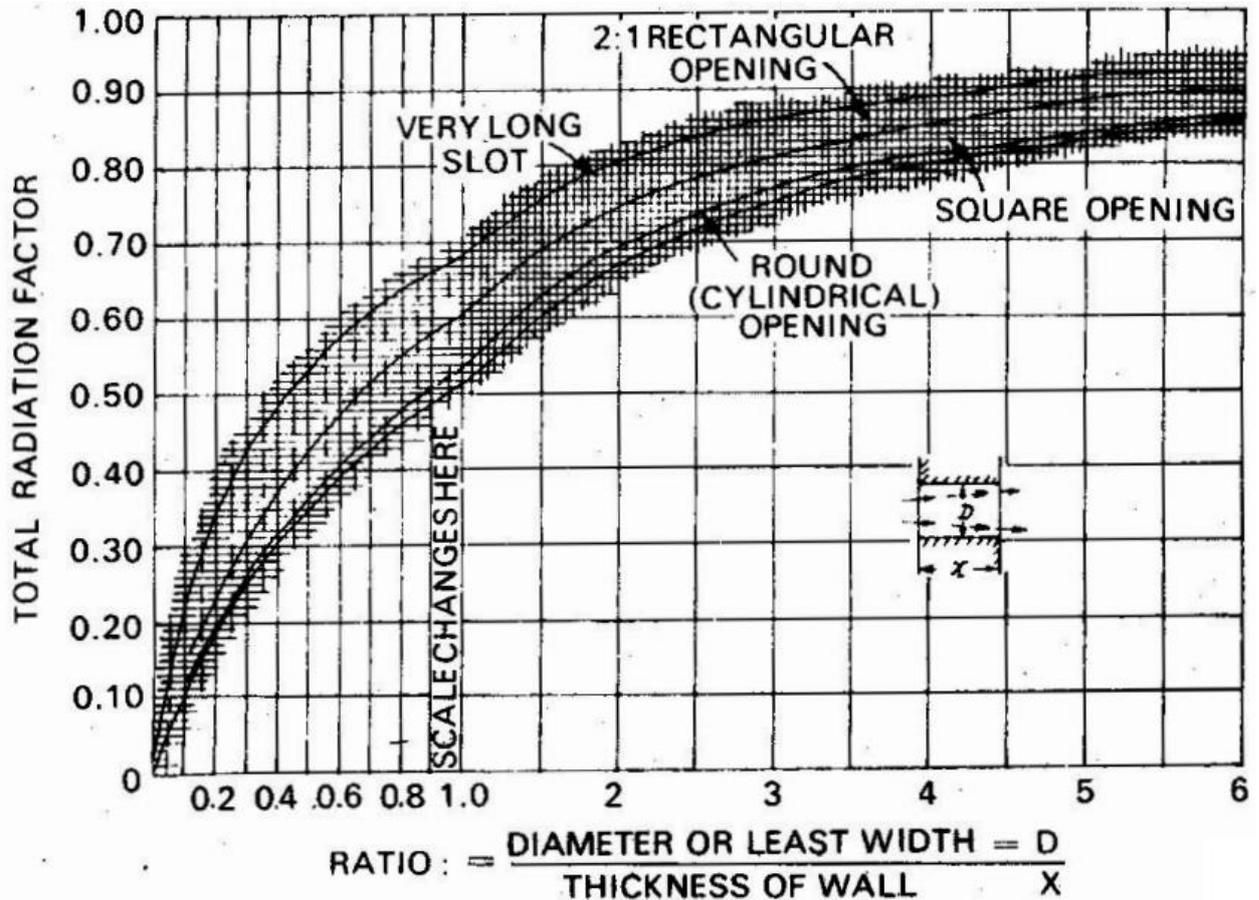


Figure 4.13: Factor for determining the equivalent of heat release from openings to the quality of heat release from perfect black body

The instruments required for carrying out performance evaluation in a furnace is given in the Table 4.2.

Table 4.3 Furnace Instrumentation

Sl. No.	Measuring Parameters	Location of Measurement	Instrument Required	Required Value
1.	Furnace soaking zone temperature (reheating furnaces)	Soaking zone side wall	Pt/Pt-Rh thermocouple with indicator and recorder	1200-1300°C
2.	Flue gas	Flue gas exit from furnace and entry to recuperator	Chromel Alummel Thermocouple with indicator	700°C max.
3.	Flue gas	After recuperator	Hg in steel thermometer	300°C (max)
4.	Furnace hearth pressure in the heating zone	Near charging end side wall over hearth level	Low pressure ring gauge	+0.1 mm of Wg

5.	Flue gas analyser	Near charging end side wall	Fuel efficiency monitor for oxygen & temperature.	O ₂ % = 5 t = 700°C (max)
6.	Billet temperature	Portable	Infrared Pyrometer or optical pyrometer	----

4.3 General Fuel Economy Measures in Furnaces

Typical energy efficiency measures for an industry with furnace are:

1. Complete combustion with minimum excess air
2. Correct heat distribution
3. Operating at the desired temperature
4. Reducing heat losses from furnace openings
5. Maintaining correct amount of furnace draught
6. Optimum capacity utilization
7. Waste heat recovery from the flue gases
8. Minimum refractory losses
9. Use of Ceramic Coatings

1. Complete Combustion with Minimum Excess Air:

The amount of heat lost in the flue gases (stack losses) depends upon amount of excess air. In the case of a furnace carrying away flue gases at 900°C, % heat lost is shown in table 4.3.

Table 4.3: Heat Loss in Flue Gas Based on Excess Air Level

Excess Air	% of total heat in the fuel carried away by waste gases (flue gas temp. 900 °C)
25	48
50	55
75	63
100	71

To obtain complete combustion of fuel with the minimum amount of air, it is necessary to control air infiltration, maintain pressure of combustion air, fuel quality and excess air monitoring.

Higher excess air will reduce flame temperature, furnace temperature and heating rate. On the other hand, if the excess air is less, then unburnt components in flue gases will increase and would be carried away in the flue gases through stack. The figure 4.14 also indicates relation between air ratio and exhaust gas loss.

The optimization of combustion air is the most attractive and economical measure for energy conservation. The impact of this measure is higher when the temperature of furnace is high. Air ratio is the value that is given by dividing the actual air amount by the theoretical combustion air amount, and it represents the extent of excess of air.

If a reheating furnace is not equipped with an automatic air/fuel ratio controller, it is necessary to periodically sample gas in the furnace and measure its oxygen contents by a gas analyzer. The Figure 4.15 shows a typical example of a reheating furnace equipped with an automatic air/fuel ratio controller.

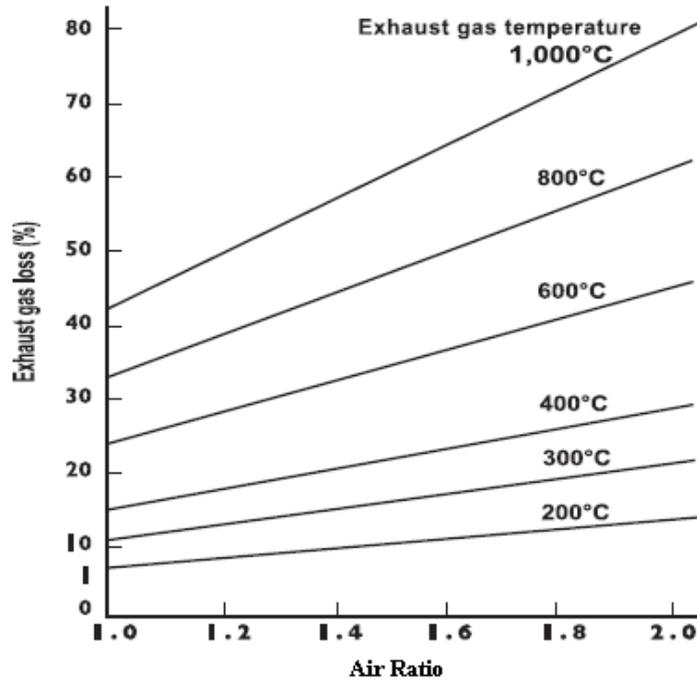


Figure 4.14: Relation between air ratio and exhaust gas loss

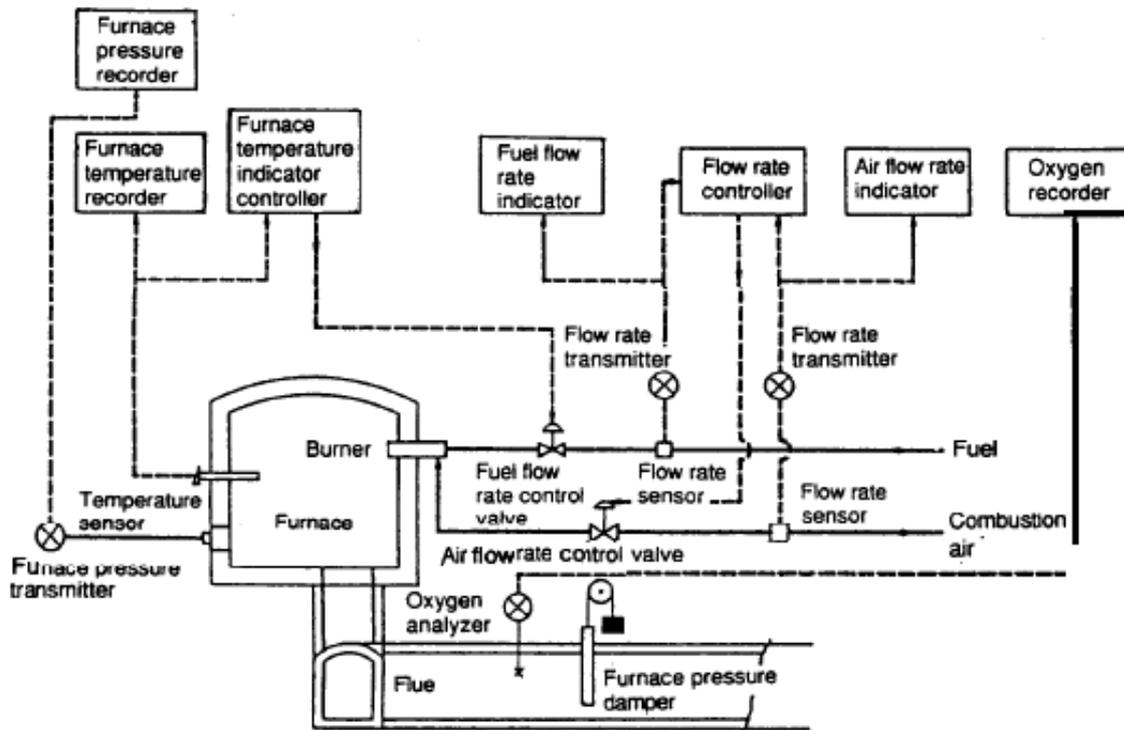


Figure 4.15: Air/fuel ratio control system with flow rate controller

More excess air also means more scale losses, which is equally a big loss in terms of money.

2. Proper Heat Distribution:

Furnace design should be such that in a given time, as much of the stock could be heated uniformly to a desired temperature with minimum fuel firing rate.

Following care should be taken when using burners, for proper heat distribution:

- i. The flame should not touch any solid object and should propagate clear of any solid object. Any obstruction will de atomise the fuel particles thus affecting combustion and create black smoke. If flame impinges on the stock, there would be increase in scale losses (Refer Figures 4.16 and 4.17).
- ii. If the flames impinge on refractories, the incomplete combustion products can settle and react with the refractory constituents at high flame temperatures.
- iii. The flames of different burners in the furnace should stay clear of each other. If they intersect, inefficient combustion would occur. It is desirable to stagger the burners on the opposite sides.
- iv. The burner flame has a tendency to travel freely in the combustion space just above the material. In small furnaces, the axis of the burner is never placed parallel to the hearth but always at an upward angle. Flame should not hit the roof.

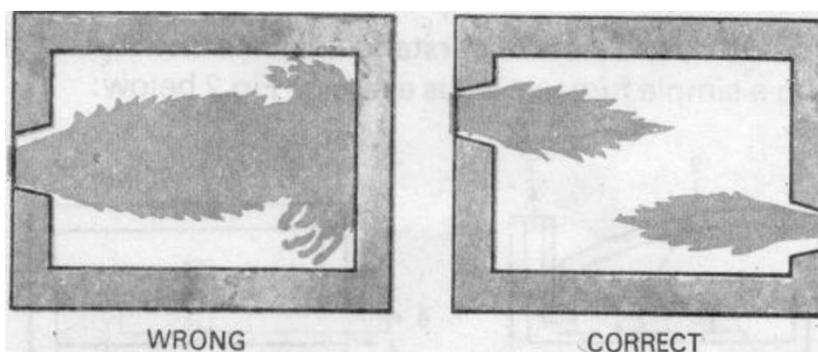


Figure 4.16: Heat Distribution in Furnace

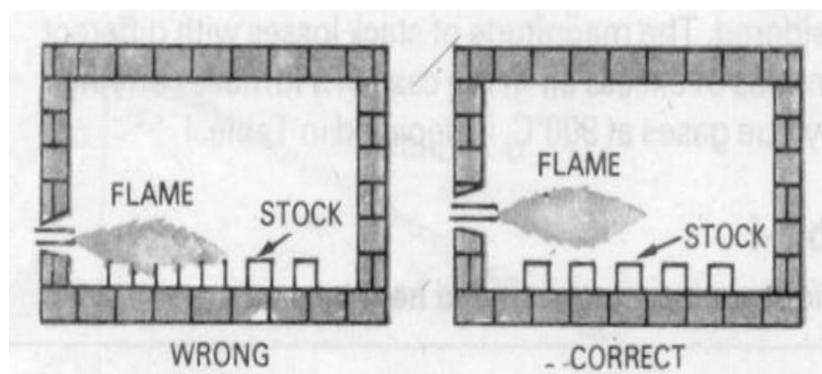


Figure 4.17: Alignment of Burners in Furnace

- v. The larger burners produce a long flame, which may be difficult to contain within the furnace walls. More burners of less capacity give better heat distribution in the furnace and also increase furnace life.
- vi. For small furnaces, it is desirable to have a long flame with golden yellow colour while firing furnace oil for uniform heating. The flame should not be too long that it enters the chimney or comes out through the furnace top or through doors. In such cases, major portion of additional fuel is carried away from the furnace.

3. Maintaining Optimum Operating Temperature of Furnace:

It is important to operate the furnace at optimum temperature. The operating temperatures of various furnaces are given in table 4.4.

Table 4.4 Operating Temperature of Various Furnaces

Slab Reheating furnaces	1200 °C
Rolling Mill furnaces	1200 °C
Bar furnace for Sheet Mill	800 °C
Bogey type annealing furnaces	650 °C -750 °C

Operating at too high temperatures than optimum causes heat loss, excessive oxidation, de-carbonization as well as over-stressing of the refractories. These controls are normally left to operator judgment, which is not desirable. To avoid human error, on/off controls should be provided.

4. Prevention of Heat Loss through Openings:

Heat loss through openings consists of the heat loss by direct radiation through openings and the heat loss caused by combustion gas that leaks through openings.

The heat loss from an opening can also be calculated using the following formula:

$$Q = 4.88x \left(\frac{T}{100} \right)^4 \times a \times A \times H$$

where,

T: absolute temperature (K)

a: factor for total radiation

A: area of opening, m²

H: time (Hr)

This is explained by an example as follows.

Example 4.3

A reheating furnace with walls 460 mm thick (X) has a billet extraction outlet, which is 1 m high (D) and 1 m wide. When the furnace temperature is 1,340°C the quantity (Q) of radiation heat loss from this opening is evaluated as follows.

The shape of opening is square, and $D/X = 1/0.46 = 2.17$. Thus, the factor for total radiation is 0.71 (refer Figure 4.13) and we get,

$$Q = 4.88 \times ((1340 + 273)/100)^4 \times 0.71 \times 1 = 234,500 \text{ kcal/hr.}$$

If the furnace pressure is slightly higher than outside air pressure (as in case of reheating furnace) during its operation, the combustion gas inside may blow off through openings and heat is lost with that. But damage is more, if outside air intrudes into the furnace, making temperature distribution uneven and oxidizing billets. This heat loss is about 1% of the total quantity of heat generated in the furnace, if furnace pressure is controlled properly.

Control of furnace draft:

If negative pressures exist in the furnace, air infiltration is liable to occur through the cracks and openings thereby affecting air-fuel ratio control. Tests conducted on apparently airtight furnaces have shown air infiltration up to the extent of 40%.

Neglecting furnaces pressure could mean problems of cold metal and non-uniform metal temperatures, which could affect subsequent operations like forging and rolling and result in increased fuel consumption. For optimum fuel consumption, slight positive pressure should be maintained in the furnace as shown in Figure 4.18.

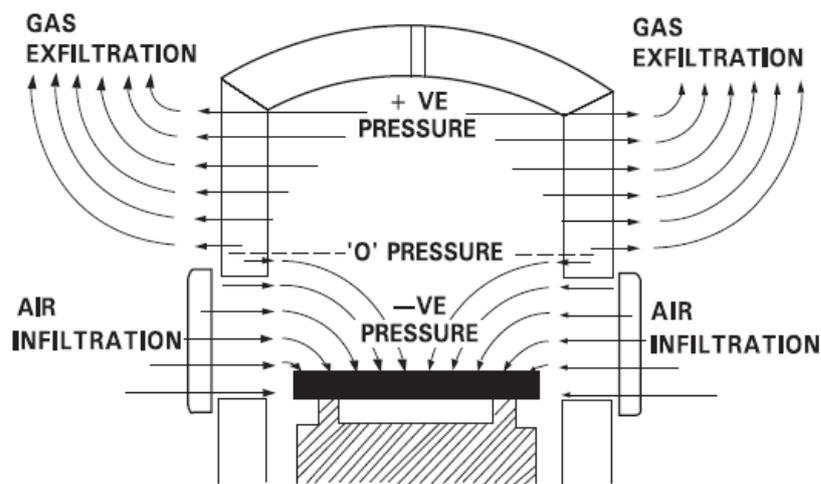


Figure 4.18: Effect of pressure on the location of zero level and infiltration of air

Ex-filtration is less serious than infiltration. Some of the associated problems with ex-filtration are leaping out of flames, overheating of the furnace refractories leading to reduced brick life, increased furnace maintenance, burning out of ducts and equipment attached to the furnace, etc.

In addition to the proper control on furnace pressure, it is important to keep the openings as small as possible and to seal them in order to prevent the release of high temperature gas and intrusion of outside air through openings such as the charging inlet, extracting outlet and peephole on furnace walls or the ceiling.

6. Optimum Capacity Utilization:

One of the most vital factors affecting efficiency is loading. There is a particular loading at which the furnace will operate at maximum thermal efficiency. If the furnace is under loaded a smaller fraction of the available heat in the working chamber will be taken up by the load and therefore efficiency will be low.

The best method of loading is generally obtained by trial-noting the weight of material put in at each charge, the time it takes to reach temperature and the amount of fuel used. Every endeavour should be made to load a furnace at the rate associated with optimum efficiency although it must be realised that limitations to achieving this are sometimes imposed by work availability or other factors beyond control.

The loading of the charge on the furnace hearth should be arranged so that,

- It receives the maximum amount of radiation from the hot surfaces of the heating chambers and the flames produced.
- The hot gases are efficiently circulated around the heat receiving surfaces

Stock should not be placed in the following position

- In the direct path of the burners or where impingement of flame is likely to occur.
- In an area which is likely to cause a blockage or restriction of the flue system of the furnace.
- Close to any door openings where cold spots are likely to develop.

The other reason for not operating the furnace at optimum loading is the mismatching of furnace dimension with respect to charge and production schedule.

In the interests of economy and work quality the materials comprising the load should only remain in the furnace for the minimum time to obtain the required physical and metallurgical requirements. When the materials attain these properties they should be removed from the furnace to avoid damage and fuel wastage. The higher the working temperature, higher is the loss per unit time. The effect on the materials by excessive residence time will be an increase in surface defects due to oxidation. The rate of oxidation is dependent upon time, temperature, as well as free oxygen content. The possible increase in surface defects can lead to rejection of the product. It is therefore essential that coordination between the furnace operator, production and planning personnel be maintained.

Optimum utilization of furnace can be planned at design stage. Correct furnace for the jobs should be selected considering whether continuous or batch type furnace would be more suitable. For a continuous type furnace, the overall efficiency will increase with heat recuperation from the waste gas stream. If only batch type furnace is used, careful planning of the loads is important. Furnace should be recharged as soon as possible to enable use of residual furnace heat.

7. Waste Heat Recovery from Furnace Flue Gases:

In any industrial furnace the products of combustion leave the furnace at a temperature higher than the stock temperature. Sensible heat losses in the flue gases, while leaving the

chimney, carry 35 to 55 percent of the heat input to the furnace. The higher the quantum of excess air and flue gas temperature, the higher would be the waste heat availability.

Waste heat recovery should be considered after all other energy conservation measures have been taken. Minimizing the generation of waste heat should be the primary objective.

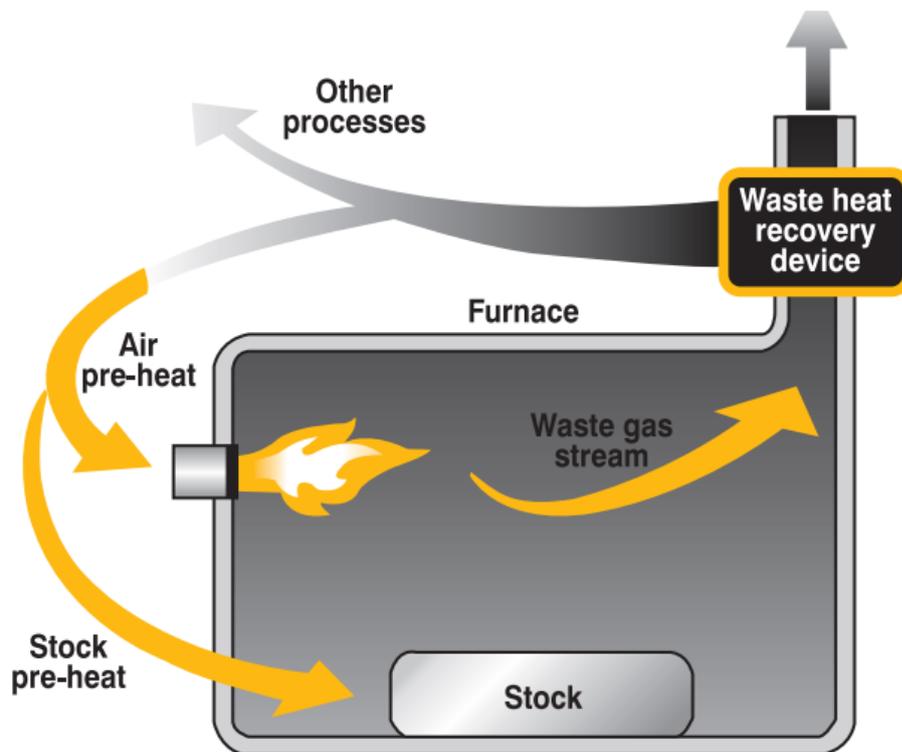


Figure 4.19: Waste Heat Recovery

The sensible heat in flue gases can be generally salvaged by the following methods:

- Charge preheating
- Preheating of combustion air
- Utilizing waste heat for other process (to generate steam or hot water by a waste heat boiler)

Charge Pre-heating

When raw materials are preheated by exhaust gases before being placed in a heating furnace, the amount of fuel necessary to heat them in the furnace is reduced. Since raw materials are usually at room temperature, they can be heated sufficiently using high-temperature gas to reduce fuel consumption rate.

Preheating of Combustion Air

For a longtime, the preheating of combustion air using heat from exhaust gas was not used except for large boilers, metal-heating furnaces and high-temperature kilns. This method is now being employed in compact boilers and compact industrial furnaces as well. (Refer Figure 4.20).

A recuperator is a device that recovers heat from exhaust gas exhausted from a furnace. A metallic recuperator has heat transfer surface made of metal, and a ceramic recuperator has heat transfer surface made of ceramics. When the exhaust gas temperature is lower than $1,000^{\circ}\text{C}$ and air for combustion is preheated, a metallic recuperator is used in general.

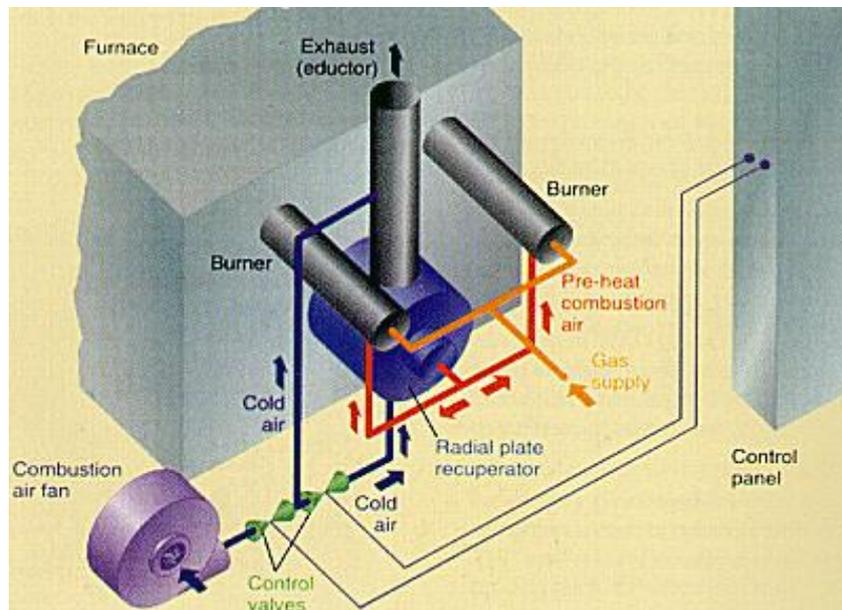


Figure 4.20: Preheating the air for combustion by a recuperator

External Recuperators

There are two main types of external recuperators:

- ✓ radiation recuperators;
- ✓ convection recuperators

Radiation recuperators generally take the form of concentric cylinders, in which the combustion air passes through the annulus and the exhaust gases from the furnace pass through the centre, see Figure 4.21 (a). The simple construction means that such recuperators are suitable for use with dirty gases, have a negligible resistance to flow, and can replace the flue or chimney if space is limited. The annulus can be replaced by a ring of vertical tubes, but this design is more difficult to install and maintain. Radiation recuperators rely on radiation from high temperature exhaust gases and should not be employed with exhaust gases at less than about 800°C .

Convection recuperators consist essentially of bundles of drawn or cast tubes; see Figure 4.21 (b). Internal and/or external fins can be added to assist heat transfer. The combustion air normally passes through the tubes and the exhaust gases outside the tubes, but there are some applications where this is reversed. For example, with dirty gases, it is easier to keep the tubes clean if the air flows on the outside. Design variations include 'IP tube and double pass systems. Convection recuperators are more suitable for exhaust gas temperatures of less than about 900°C .

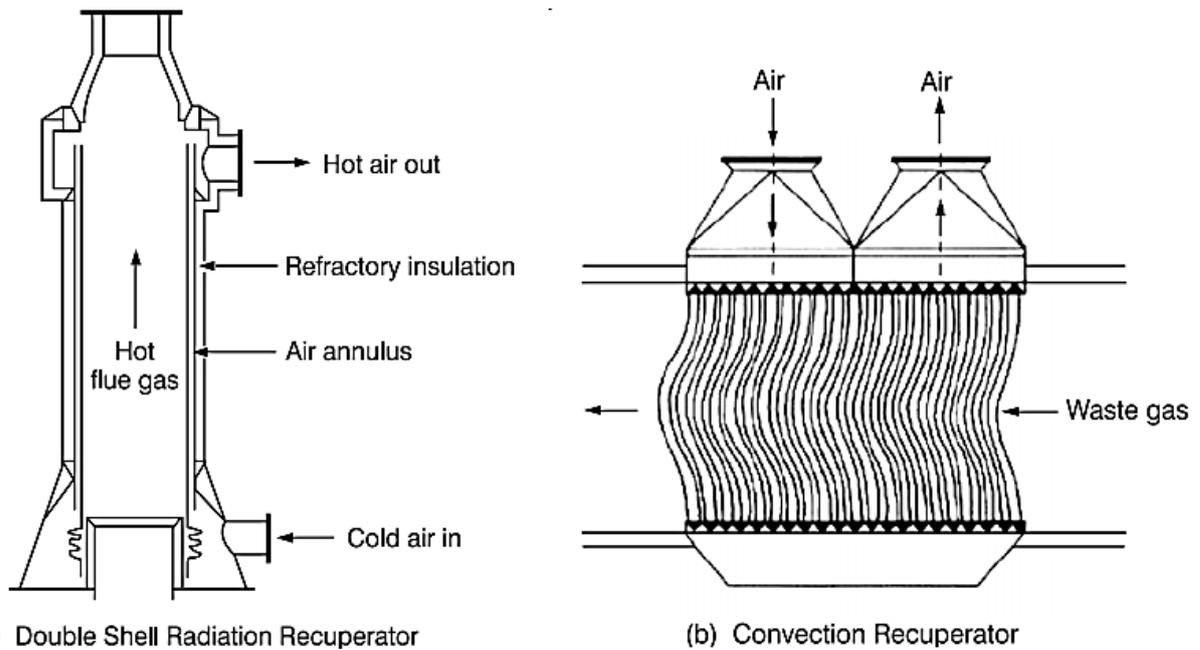


Figure 4.21: Metallic Recuperators

Self-Recuperative Burners

Self-recuperative burners (SRBs) are based on traditional heat recovery techniques in that the products of combustion are drawn through a concentric tube recuperator around the burner body and used to pre-heat the combustion air (Figure 4.22)

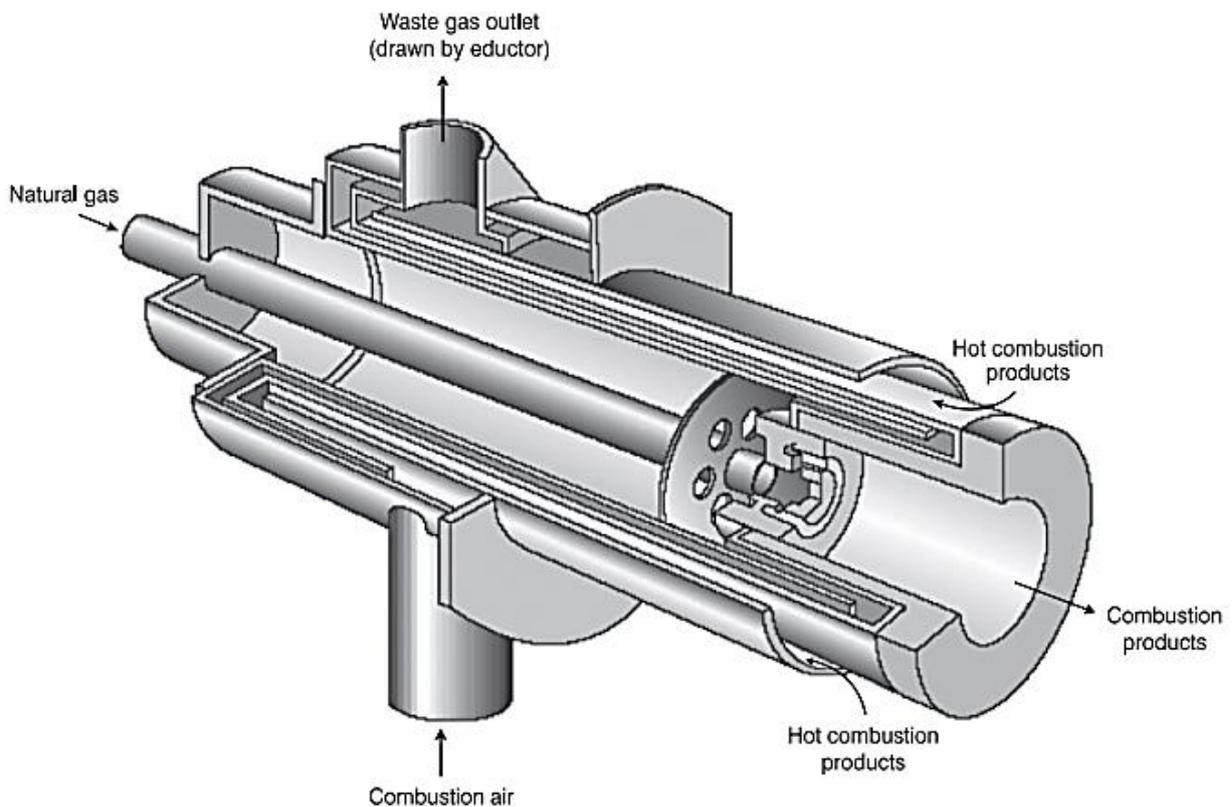


Figure 4.22: Self-Recuperative Burners

A major advantage of this type of system is that it can be retro-fitted to an existing furnace structure to increase production capability without having to alter the existing exhaust gas ducting arrangements. SRBs are generally more suited to heat treatment furnaces where exhaust gas temperatures are lower and there are no stock recuperation facilities.

Estimation of fuel savings

By using preheated air for combustion, fuel can be saved. The fuel saving rate is given by the following formula:

$$S = \frac{P}{F + P - Q} \times 100(\%)$$

Where,

- S: Fuel saving rate
- F: Calorific value of fuel (kcal/kgfuel)
- P: Quantity of heat brought in by preheated air (kcal/kgfuel)
- Q: Quantity of heat taken away by exhaust gas (kcal/kgfuel)

By this formula, fuel saving rates for heavy oil and natural gas were calculated for various temperatures of exhaust gas and preheated air. The results are shown in the following Figure 4.23 and Figure 4.24.

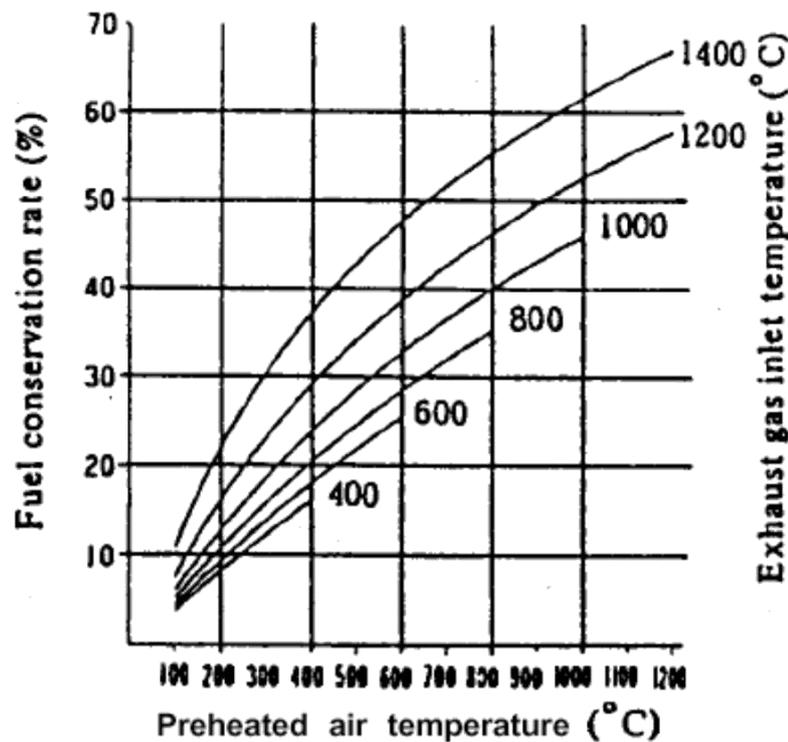


Figure 4.23: Fuel conservation rate when oil is used

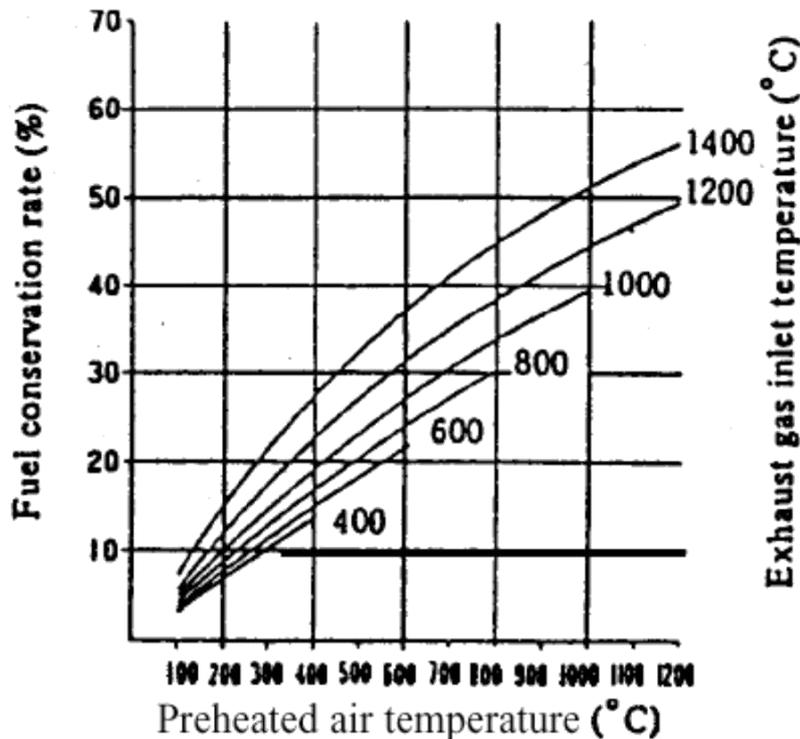


Figure 4.24: Fuel conservation rate when natural gas is used

For example, when combustion air for heavy oil is preheated to 400°C by a heat exchanger with an inlet temperature of 800°C, the fuel conservation rate is estimated to be about 20 percent. When installing a recuperator in a continuous steel reheating furnace, it is important to choose a preheated air temperature that will balance the fuel saving effect and the invested cost for the equipment.

Also, the following points should be checked:

- Draft of exhaust gas: When exhaust gas goes through a recuperator, its draft resistance usually causes a pressure loss of 5-10 mm H₂O. Thus, the draft of stack should be checked.
- Air blower for combustion air: While the air for combustion goes through a recuperator, usually 100-200 mm H₂O pressure is lost. Thus, the discharge pressure of air blower should be checked, and the necessary pressure should be provided by burners.

Since the volume of air is increased owing to its preheating, it is necessary to be careful about the modification of air-duct diameters and blowers. As for the use of combustion gases resulting from high-density oils with a high sulphur content, care must be taken to avoid problems such as clogging with dust or sulphides, corrosion or increases in nitrogen oxides.

Table 4.5 gives details of various air preheaters. In addition, heat-pipe-type heat exchangers and high temperature gas or gas-plate heat exchangers can serve as air preheaters.

Table 4.5: Details of air preheaters

Type			Exhaust Gas Temperature, °C	Preheated Air Temperature, °C	Object Furnace	
Recuperative	Metallic recuperator	Flue installation	Convective: multitubular, other	1000 below or 300 – 600	Heating furnace, heat treatment furnace and other industrial furnaces	
		Chimney installation	Radiative and convective	1000 – 1300	-do-	
	Ceramic (tile) recuperator			1200 – 1400	400 – 700	Soaking pit and glass
Regenerative	General			1000 – 1600	600 – 1300	Coke oven, hot blast stove & glass kiln
	Rotary regenerative			600	100 - 300	Boiler, hot blast stove

Utilizing Waste Heat as a Heat Source for Other Processes

The temperature of heating-furnace exhaust gas can be as high as 400 –600°C, even after heat has been recovered from it.

When a large amount of steam or hot water is needed in a plant, installing a waste heat boiler to produce the steam or hot water using the exhaust gas heat is preferred. If the exhaust gas heat is suitable for equipment in terms of heat quantity, temperature range, operation time etc., the fuel consumption can be greatly reduced. In one case, exhaust gas from a quenching furnace was used as a heat source in a tempering furnace so as to obviate the need to use fuel for the tempering furnace itself.

8. Minimizing Wall Losses

About 30-40% of the fuel input to the furnace generally goes to make up for heat losses in intermittent or continuous furnaces. The appropriate choice of refractory and insulation materials goes a long way in achieving fairly high fuel savings in industrial furnaces.

The heat losses from furnace walls affect the fuel economy considerably. The extent of wall losses depends on:

- Emissivity of wall
- Thermal conductivity of refractories
- Wall thickness
- Whether furnace is operated continuously or intermittently

Heat losses can be reduced by increasing the wall thickness, or through the application of insulating bricks. Outside wall temperatures and heat losses of a composite wall of a certain thickness of fire brick and insulation brick are much lower, due to lesser conductivity of insulating brick as compared to a refractory brick of similar thickness. In the actual operation in most of the small furnaces the operating periods alternate with the idle periods. During the off period, the heat stored in the refractories during the on period is gradually dissipated, mainly through radiation and convection from the cold face. In addition, some heat is abstracted by air flowing through the furnace. Dissipation of stored heat is a loss, because the lost heat is again imparted to the refractories during the heat “on” period, thus consuming extra fuel to generate that heat. If a furnace is operated 24 hours, every third day, practically all the heat stored in the refractories is lost. But if the furnace is operated 8 hours per day all the heat stored in the refractories is not dissipated. For a furnace with a fire brick wall of 350mm thickness, it is estimated that 55percent of the heats to red in the refractories is dissipated from the cold surface during the 16 hours idle period. Furnace walls built of insulating refractories and cased in a shell reduce the flow of heat to the surroundings.

Prevention of Radiation Heat Loss from Surface of Furnace

The quantity of heat release from surface of furnace body is the sum of natural convection and thermal radiation. This quantity can be calculated from surface temperatures of furnace. The temperatures on furnace surface should be measured at as many points as possible, and their average should be used. If the number of measuring points is too small, the error becomes large.

The quantity (Q)of heat release from are heating furnace is calculated with the following formula:

$$Q = a \times (t_1 - t_2)^{5/4} + 4.88 E \times \left(\left(\frac{t_1 + 273}{100} \right)^4 - \left(\frac{t_2 + 273}{100} \right)^4 \right)$$

Where,

- Q : Quantity of heat released (kcal/hr/m²)
- a : Factor regarding direction of the surface of natural convection ceiling = 2.8, side walls = 2.2, hearth = 1.5
- t₁ : Temperature of external wall surface of the furnace (°C)
- t₂ : Temperature of air around the furnace (°C)
- E : Emissivity of external wall surface of the furnace

The first term of the formula above represents the quantity of heat release by natural convection, and the second term represents the quantity of heat release by radiation. The following Figure 4.25 shows the relation between the temperature of external wall surface and the quantity of heat release calculated with this formula.

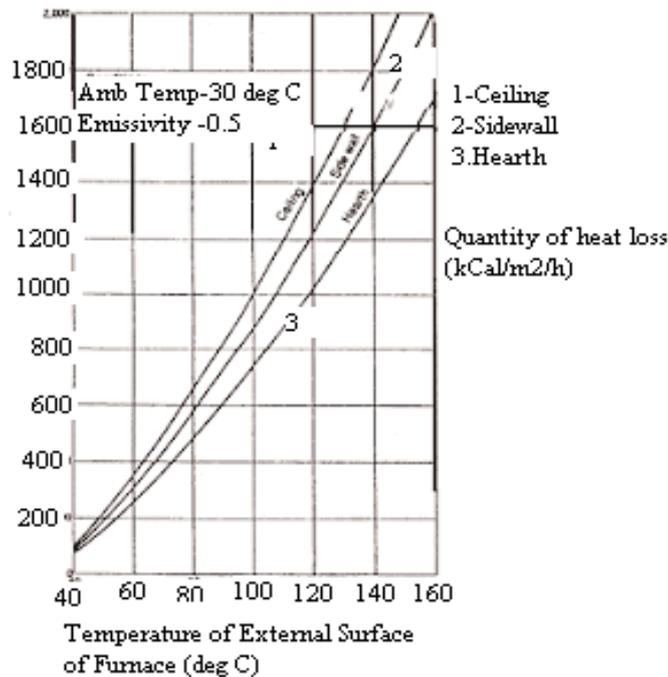


Figure 4.25: Quantity of heat release at various temperature

This is explained with an example as follows:

Example 4.4

There is a reheating furnace whose ceiling, side walls and hearth has 20 m², 50 m², and 20 m² of surface area respectively. Their surface temperatures are measured and the averages are 80°C, 90°C and 100°C respectively. Evaluate the quantity of heat release from the whole surface of the furnace.

From the above figure 4.25, the quantities of heat release from ceiling, side walls and hearth per unit area are respectively 650 kcal/m²h, 720 kcal/m²h, and 730 kcal/m²h.

Therefore, the total quantity of heat release is,

$$Q = (650 \times 20) + (720 \times 50) + (730 \times 20) = 13000 + 36000 + 14600 = 63,600 \text{ kcal/hr.}$$

Use of Ceramic Fibre

Ceramic fibre is a low thermal mass refractory used in the hot face of the furnace and fastened to the refractory walls. Due to its low thermal mass the storage losses are minimized. This results in faster heating up of furnace and also faster cooling. Energy savings by this application is possible only in intermittent furnaces.

9. Use of Ceramic Coatings

Ceramic coatings in furnace chamber promote rapid and efficient transfer of heat, uniform heating and extended life of refractories. The emissivity of conventional refractories decreases with increase in temperature whereas for ceramic coatings it increases. This outstanding property has been exploited for use in hot face insulation.

Ceramic coatings are high emissivity coatings which when applied has a long life at temperatures up to 1350°C. The coatings fall into two general categories-those used for coating metal substrates, and those used for coating refractory substrates. The coatings are non-toxic, non-flammable and water based. Applied at room temperatures, they are sprayed and air dried in less than five minutes. The coatings allow the substrate to maintain its designed metallurgical properties and mechanical strength. Installation is quick and can be completed during shut down. Energy savings of the order of 8-20% have been reported depending on the type of furnace and operating conditions.

Fish Bone Diagram for Energy Conservation Analysis in Furnaces

All the possible measures discussed can be incorporated in furnace design and operation. The figure 4.26 shows characteristics diagram of energy conservation for a fuel-fired furnace.

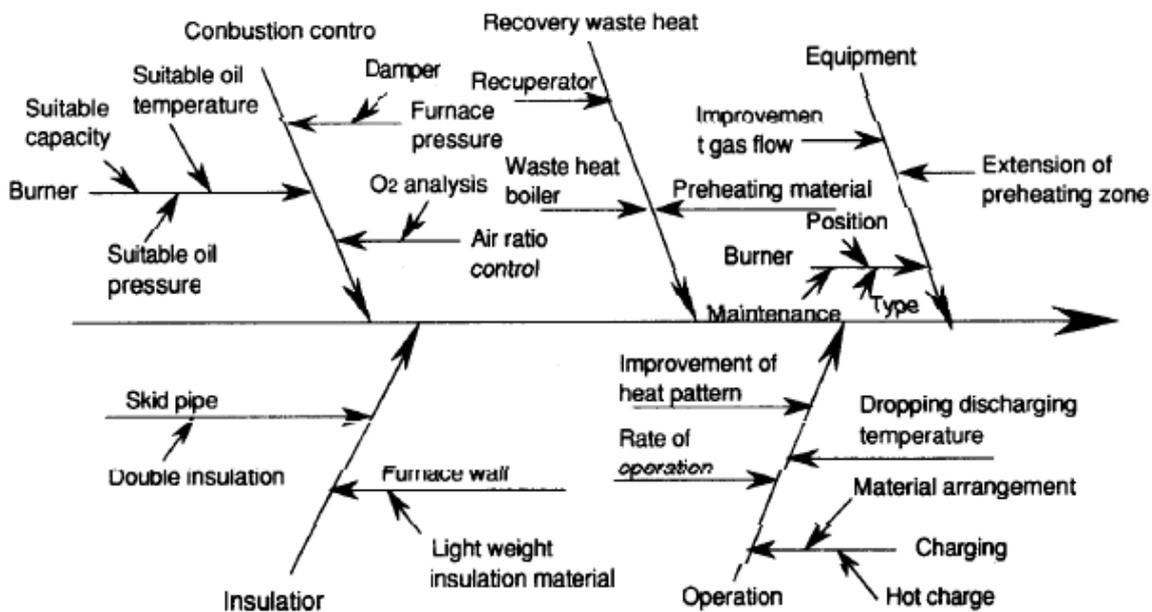


Figure 4.26 Characteristic Diagram of Energy Conservation for Reheating Furnace

Case Study 4.1

In a rolling mill, following energy conservation measures have been implemented and savings achieved are explained below:

Saving by Installing a Recuperator

This plant had a continuous pusher type billet reheating furnace. The furnace consists of two burners at the heating zone. The furnace is having a length of 40 ft. Annual furnace oil consumption is 620 KL. The furnace did not have any waste heat recovery device. The flue gas temperature is found to be 650 °C. To tap this potential heat, the unit has installed a recuperator device. It was possible to preheat the combustion air to 325 °C. By resorting to this measure, there was 15% fuel saving which is 93 KL of oil per annum.

Judicious Use of Combustion Equipment in two Zone Furnaces

During start up the furnace is already filled up with billets of previous day and has a temperature of about 700°C. During initial starting of furnace only the billets in the soaking zone are heated up as there is no movement of billets. Further the flow of heat is from soaking zone to heating zone. Therefore, during starting the soaking zone burners are switched on first and heating zone burners are started only after the mill starts operating.

Similarly, prior to mill stoppage billets lying in soaking zone have already attained re-rolling temperature and the incoming billets from heating zone will not be rolled and hence there is no need to heat those billets. As a result the burners in heating zone are stopped at least 30 to 60 minutes before the stoppage of mill.

The company has saved about 32.4 kL of furnace oil per annum.

Calculations:

- a. During Starting:
Savings by operating 2 nos. of LAP 4 A Wesman burners
30 minutes after starting up = 54 litres
- b. During Shutdown:
Savings by switching off 2 Nos. of LAP Wesman burners
Before 30 minutes of furnace shut down = 54 litres

Total savings per day = 108 litres

Annual Savings = 32.4 KL

Example 4.5

Calculate the induction melting furnace efficiency from the following melt cycle data

Mild steel (MS) scrap charged	: 1500 kg
Specific heat of MS	: 0.682 kJ/kg °C
Latent heat of MS	: 272 kJ/kg
MS melting temperature	: 1650 °C
Inlet MS charge temperature	: 40 °C
Electricity consumed during cycle	: 1020 kWh

Solution

Theoretical energy required for melting = $1500 (0.682 \times (1650 - 40) + 272) / 3600$ kWh
= 570.8 kWh

Actual input = 1020 kWh

Furnace efficiency = $570.8 \times 100 / 1020$
= 56%

Example 4.6

In a crude distillation unit of a refinery, furnace is operated to heat 500 m³/hr of crude oil from 255°C to 360°C by firing 3.4 tons/hr of fuel oil having GCV of 9850 kcal/kg. As an energy conservation measure, the management has installed an air pre heater (APH) to reduce the flue gas heat loss. The APH is designed to pre-heat 57 tonnes/hr of combustion air to 195°C.

Calculate the efficiency of the furnace before & after the installation of APH.

Consider the following data:

- Specific heat of crude oil = 0.6 kcal/kg°C
- Specific heat of air = 0.24 kcal/kg°C
- Specific gravity of Crude oil = 0.85
- Ambient temperature = 28°C.

Solution

Before the installation of APH

$$\begin{aligned}\text{Heat gain by the crude} &= 500 \times 1000 \times 0.85 \times 0.6 \times (360-255) \\ &= 26775000 \text{ kCal/hr}\end{aligned}$$

$$\begin{aligned}\text{Heat input to the furnace} &= 3.4 \times 1000 \times 9850 \\ &= 33490000 \text{ kCal/hr}\end{aligned}$$

$$\begin{aligned}\text{Efficiency of the furnace} &= 26775000 / 33490000 \\ &= 80 \%\end{aligned}$$

After the installation of APH

$$\begin{aligned}\text{Heat gain by the crude} &= 500 \times 1000 \times 0.85 \times 0.6 \times (360-255) \\ &= 26775000 \text{ kCal/hr}\end{aligned}$$

$$\begin{aligned}\text{Heat gain by Air-preheater} &= 57 \times 1000 \times 0.24 \times (195-28) \\ &= 2284560 \text{ kCal/hr}\end{aligned}$$

Heat reduction in the furnace input = Heat gain by Air-preheater

$$\begin{aligned}\text{New Heat input to the furnace} &= 33490000 - 2284560 \\ &= 31,205,440 \text{ kCal/hr}\end{aligned}$$

$$\begin{aligned}\text{Efficiency of furnace after installation of APH} &= 26775000 / 31,205,440 \\ &= 85.8 \%\end{aligned}$$

Chapter 5: Insulation and Refractories

5.1 Insulation

5.1.1 Purpose of Insulation

Insulation is used to prevent heat loss or heat gain to enable less energy consumption while meeting the demands of heating and cooling. A thermal insulator is a poor conductor of heat and has a low thermal conductivity and hence is used in buildings and in manufacturing processes to prevent heat loss or heat gain. Thermal insulation delivers the following benefits:

- Reduces over-all energy consumption
- Offers better process control by maintaining process temperature.
- Prevents corrosion by keeping the exposed surface of a refrigerated system above dew point
- Provides fire protection to equipment
- Absorbs vibration
-

Insulating materials are porous, containing large number of dormant air cells. Insulation for heating system should be fire proof, be vermin proof, have lasting quality, be mechanically strong, be compact and be light in weight.

5.1.2 Types and Application

The Insulation can be classified into three groups according to the temperature ranges for which they are used. They are

Low Temperature Insulations (up to 90°C)

This range covers insulating materials for refrigerators, cold and hot water systems, storage tanks, etc. The commonly used materials are Cork, Wood, 85% magnesia, Mineral Fibers, Polyurethane and expanded Polystyrene, etc

Medium Temperature Insulations (90 - 325°C)

Insulators in this range are used in low temperature, heating and steam raising equipment, steam lines, flue ducts etc. The types of materials used in this temperatures range include 85% Magnesia, Asbestos, Calcium Silicate and Mineral Fibers etc.

High Temperature Insulations (325° C - above)

Typical uses of such materials are super-heated steam system, oven dryer and furnaces etc. The most extensively used materials in this range are Asbestos, Calcium Silicate, Mineral Fibre, Mica and Vermiculite based insulation, Fireclay or Silica based insulation and Ceramic Fibre.

The Table 5.1 describes the characteristics and applications of various insulating materials.

Table 5.1: Characteristics & Applications of various insulating materials

S. No.	Type of Insulation	Application	Characteristics
1	Polystyrene An organic form made by polymerizing styrene	Suitable for low temperatures (-167 °C to 82 °C). Mainly used in <ul style="list-style-type: none"> • Cool rooms, • refrigeration piping and • concrete retaining structures 	Rigid and light weight. Combustible, has a low melting point, is UV degradable, and susceptible to attacks by solvents
2	Polyurethane Made by reacting isocyanates and alcohols. Made in continuous slab or foamed in situ.	Suitable for low temperatures (-178°C to 4 °C). Mainly used in <ul style="list-style-type: none"> • cool rooms, • refrigerated transports, • deep freezing cabinets, • refrigeration piping, • floor and foundation insulation. 	Closed cell structure, low density and high mechanical strength Combustible, produces toxic vapours and has a tendency to smoulder.
3	Rockwool (mineral fibre) Manufactured by melting basalt and coke in a cupola at about 1500 °C. Phenolic binders used.	Suitable for temperatures up to 820 °C. Mainly used to insulate <ul style="list-style-type: none"> • industrial ovens, • heat exchangers, • driers, • boilers and high temperature pipes 	Has a wide density range and is available in matts, blankets, loose form or preformed for pipe insulation. It is chemically inert, non-corrosive and maintains Mechanical strength during handling
4	Fibre Glass Formed by bonding long glass fibres with a thermo setting resin to form blankets and batts, semi-rigid boards, high density rigid boards and preformed pipe sections	Suitable for temperatures up to 540 °C. Mainly used to insulate <ul style="list-style-type: none"> • industrial ovens, • heat exchangers, • driers, boilers and pipe works	Will not settle or disintegrate with ageing. Fibre glass products are slightly alkaline pH9 (neutral is pH7). It should not promote or accelerate the corrosion of steel. Hence, it is protected from external contamination.
5	Calcium Silicate Made from anhydrous calcium silicate material reinforced with a non-asbestos binder. Available in slab form of various sizes.	Suitable for temperatures up to 1050 °C. Mainly used to <ul style="list-style-type: none"> • insulate furnace walls, • fire boxes, • back-up refractory, 	Has a minute air cell structure, has a low thermal conductivity and will retain its size and shape in its useable temperature range. It is light weight, but has good structural strength

		<ul style="list-style-type: none"> • flue lining and boilers 	so it can withstand mechanical abrasion. It will not burn or rot, is moisture resistant and non-corrosive.
6	<p>Ceramic Fibre Made from high purity alumina and silica grains, melted in an electric furnace and blasted by high velocity gases into light fluffy fibres. Made in a variety of forms</p>	<p>Suitable for temperatures up to 1430 °C. Mainly used to insulate</p> <ul style="list-style-type: none"> • furnace and kiln back-up refractory, • fire boxes, • glass feeder bowls, • furnace repair, induction • coil insulation, • high temperature gaskets and • wrapping material 	<p>Suitable for many applications because of the variety of forms. These include cloth, felt, tape, coating cements and variform castable (fire brick)</p>

Classification of insulating materials based on their forms

- **Powdered:** It is mixed with water used to cover odd shapes such as pipe unions, elbows, valves etc.
- **Sheet:** applied to flat areas such as ceilings, ducts, etc.
- **Block and brick:** used to cover outside surfaces of boilers.
- **Blanket:** used to cover ducts, pipes.
- **Tube:** used to insulate steam and hot water pipes.
- **Roll:** used to cover cold and warm-air ducts for furnace castings in hot-air heating systems.
- **Custom Moulded Insulation:** Lagging materials can be obtained in bulk, in the form of moulded sections; semi - cylindrical for pipes, slabs for vessels, flanges, valves etc. The main advantage of the moulded sections is the ease of application and replacement when undertaking repairs for damaged lagging.

5.1.3 Thermal Resistance (R)

The effectiveness of insulation is measured in terms of thermal resistance, called R-value, which indicates the resistance of heat flow. The higher the R-value, the greater the insulating power. The actual R-value of thermal insulation depends on the type of material, its thickness and density. In calculating the R-value of insulation, the R-values of the individual layers are added. The amount of insulation depends on:

- Climate
- Type of fuel used
- The section the requires insulation

Once the type of insulation material that is required is finalized based on temperature of process and the average ambient temperature, the thickness of insulating material required needs to be evaluated.

5.1.4 Economic Thickness of Insulation (ETI)

The effectiveness of insulation follows the law of decreasing returns. Hence, there is a definite economic limit to the amount of insulation, which is justified. An increased thickness is uneconomical and cannot be recovered through small heat savings. This limiting value is termed as economic thickness of insulation. An illustrative case is given in Figure 5.1. Each industry has different fuel cost and boiler efficiency. These values can be used for calculating economic thickness of insulation. This shows that thickness for a given set of circumstances results in the lowest overall cost of insulation and heat loss combined over a given period of time. The following figure 5.1 and 5.2 illustrates the principle of economic thickness of insulation.

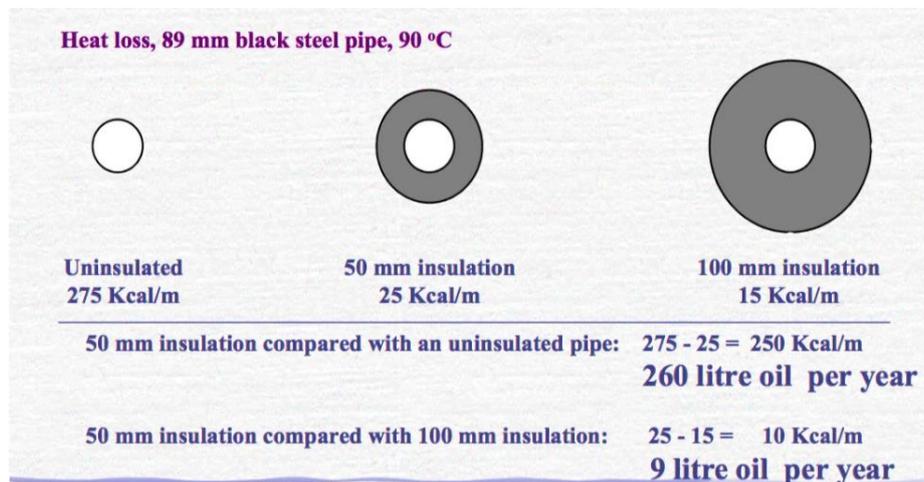
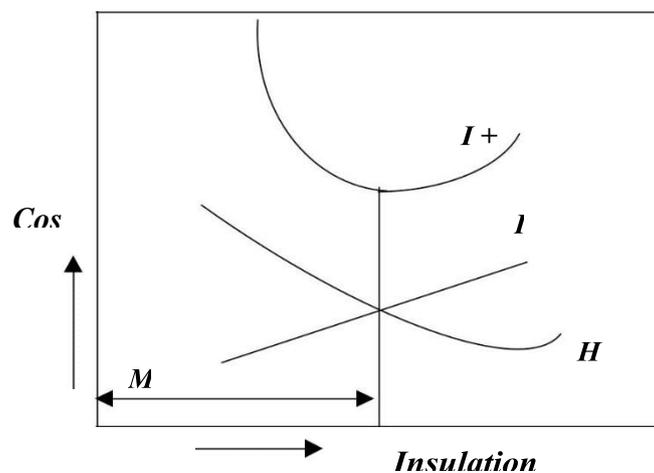


Figure 5.1: Illustration of optimal insulation



I: Cost of Insulation H: Cost of Heat Loss
 I + H: Total Cost M: Economic Thickness

Figure 5.2: Determination of economic thickness of insulation

Heat loss from a surface is expressed as

$$H = h \times A \times (T_h - T_a)$$

Where,

h is Heat transfer coefficient, W/m² K (2000 W/ m² K)

H is Heat loss, Watts

T_a is Average ambient temperature, °C

T_h is Hot surface temperature (for hot fluid piping) in °C & Cold surface temperature (for cold fluids piping) in °C

The point where the amount of insulation gives the greatest return on investment is called as “**economic thickness of insulation (ETI)**”. The determination of economic thickness requires the attention to the following factors.

- Cost of fuel
- Annual hours of operation
- Heat content of fuel
- Boiler efficiency
- Operating surface temperature
- Pipe diameter/thickness of surface
- Estimated cost of insulation.
- Average exposure ambient still air temperature.

The cost of insulation depends not only on the thickness of insulating material but also on the material itself. Materials with higher thermal resistance will require insulating material of less thickness compared to insulating material with lower thermal resistance. At the same time cost of higher thermal resistance for same thickness may be higher compare to insulating material with lower thermal resistance.

5.1.5 Heat Savings and Application Criteria

Various charts, graphs and references are available for heat loss computation. The surface heat loss can be computed with the help of a simple relation as given below. This equation can be used up to 200 °C surface temperature. Factors like wind velocities, conductivity of insulating material etc. has not been considered in the equation.

$$S = \left[10 + \frac{T_s - T_a}{20} \right] (T_s - T_a)$$

Where

S =Surface heat loss in Kcal/hr m²

T_s =Hot surface temperature in °C

T_a =Ambient temperature in °C

$$\text{Total heat loss/hr (Hs)} = S \times A$$

Where,

A = Surface area in m²

Based on the cost of heat energy, the quantification of heat loss in BDT can be worked out as under:

$$\text{Equivalent fuel loss (Hf) } \left(\frac{\text{kg}}{\text{yr}} \right) = \frac{H_s \times \text{Yearly hours of operation}}{\text{GCV} \times \eta_b}$$

$$\text{Annual heat loss in monetary terms (BDT)} = \text{Hf} \times \text{Fuel cost (BDT/kg)}$$

Where

GCV = Gross Calorific value of fuel Kcal/Kg

η_b = Boiler efficiency in %

Example 5.1

Consider a 50m pipe of 150 mm outer diameter carrying process fluid at 150°C. The ambient temperature is 20 °C. The heat is supplied by boiler operating at 80% efficiency and the cost of fuel which is HSD with GCV of 10000 Kcal/ lit is BDT 50/litter. It is considered to insulate this pipe with semi-cylindrical pre-fabricated glass mineral fibre with a life of 5 years. The cost of glass mineral fiber of various thickness and the surface temperature achievable after insulation for calculation purpose is given below.

Thickness, mm	Cost, BDT/m	Surface Temperature. after insulation, °C
25	450	65
50	600	48
75	750	43
100	900	40

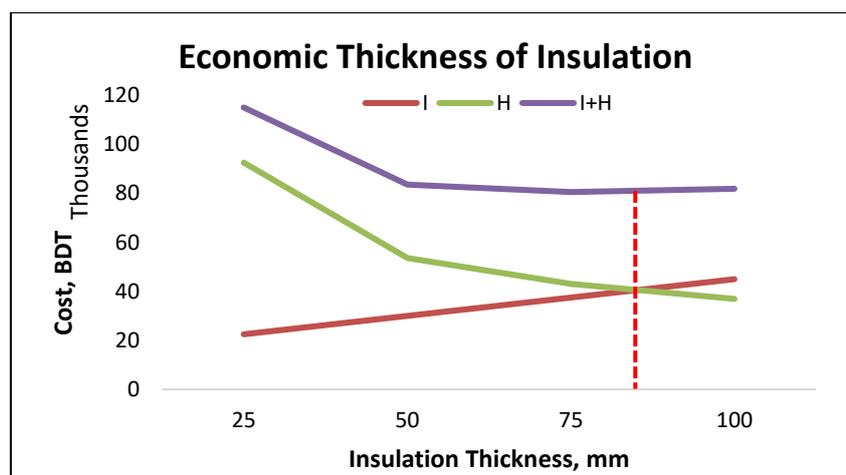
Calculate fuel saved by using above thickness levels of insulation. Draw a graph of Costs of insulation vs thickness and identify the economic thickness of insulation required.

Methodology:

- Step 1: Calculate the surface area of pipe
- Step 2: Calculate unit heat loss and using surface area calculate total surface heat loss with and without insulation.
- Step 3: Convert heat loss to fuel loss using calorific value of fuel used in boiler. Adjust fuel usage to boiler efficiency. Convert hourly loss to yearly loss.
- Step 4: Using unit fuel cost data estimate fuel loss in cost terms for the entire life of insulating material i.e. 5 years.
- Step 5: Using given insulating material costs estimate cost of insulation for varying thickness of the insulating material.
- Step 6: Calculate the total cost of insulation and fuel loss cost.
- Step 7: Plot graph of Insulation thickness v/s Cost of insulating material, Cost of fuel loss and total cost.
- Step 8: Identify the economic thickness of insulating material by drawing a vertical line at the point of intersection of cost of insulating material and cost of fuel loss lines.

Economic thickness of insulation for 150 mm dia pipe					
Step 1					
	<i>Unit</i>				
<i>Insulation Thickness</i>	<i>mm</i>	25	50	75	100
<i>Length of pipe, l</i>	<i>m</i>	50			
<i>Bare pipe outer diameter, d</i>	<i>mm</i>	150			
Surface Area = $\pi \times d \times l$					
<i>Bare pipe surface area</i>	<i>m²</i>	23.56	23.56	23.56	23.56
Step 2					
<i>Ambient Temperature</i>	<i>°C</i>	20			
<i>Surface Temperature</i>					
<i>-before insulation</i>	<i>°C</i>	150			
<i>-after insulation</i>	<i>°C</i>	65	48	43	40
<i>Surface heat loss</i>		$S = \left[10 + \frac{T_s - T_a}{20} \right] (T_s - T_a)$			
<i>-without insulation</i>	<i>Kcal/hr.m²</i>	2145			
<i>-after insulation</i>	<i>Kcal/hr.m²</i>	551.25	319.2	256.45	220
<i>Heat loss reduction</i>	<i>Kcal/hr.m²</i>	1594	1826	1888	1925
Step 3					
<i>Total energy saved (H_s)</i>	<i>Kcal/year</i>	13387500	15336720	15863820	16170000
<i>Yearly hours of operation</i>	<i>hrs</i>	8400			
<i>GCV</i>	<i>Kcal/Lit</i>	10000			
<i>Boiler Efficiency</i>	<i>%</i>	80			
$\text{Equivalent fuel loss (H}_f) \left(\frac{\text{Lit}}{\text{yr}} \right) = \frac{H_s \times \text{Yearly hours of operation}}{\text{GCV} \times \eta_b}$					
<i>Total fuel energy lost per year</i>	<i>Lit/year</i>	370	215	172	148
Fuel Saving = Fuel Loss without insulation – Fuel loss with insulation					
<i>Yearly fuel savings</i>	<i>Lit/year</i>	1071	1227	1269	1294
Step 4					
<i>Life cycle heat loss in monetary terms (BDT)</i> $= H_f \times \text{Life of insulation} \times \text{Fuel cost (BDT/lit)}$					
<i>Fuel Cost</i>	<i>BDT/Lit</i>	50			
Cost of fuel loss					
<i>-without insulation</i>	<i>BDT/5 yrs</i>	360360			
<i>-with insulation [H]</i>	<i>BDT/ 5yrs</i>	92610	53626	43084	36960
Step 5					
<i>Cost of insulation</i>	<i>BDT/m</i>	450	600	750	900
<i>Total cost of insulation [I]</i>	<i>BDT</i>	22500	30000	37500	45000
Step 6					
<i>Total Cost (insulation & fuel loss) [I+H]</i>	<i>BDT</i>	115110	83626	80584	81960

Step 7:



Step 8: From the above graph it can be seen that economic thickness of insulation is 80mm.

5.1.6 Cold Insulation

Cold Insulation Features

Cold Insulation should be considered and where operating temperature are below ambient where protection is required against heat gain, condensation or freezing. Condensation will occur whenever moist air comes into contact with the surface that is at a temperature lower than the dew point of the vapour. In addition, heat gained by uninsulated chilled water lines can adversely affect the efficiency of the cooling system. The most important characteristics of a suitable Cold insulation material have following features:

- Low thermal conductivity
- High water resistance, and
- Durability at low temperature

Other properties like easy workability, negligible capillary absorption should also be taken into consideration while making a selection. The insulation system is only as good as its vapour barrier and the care with which it is installed.

Material Selection for Cold Insulation

Selection of insulation materials should be carefully considered where the possibility of steam purging of the equipment is required or for other reasons which may cause the temperature to be increased to a level that exceeds the maximum limiting temperature of the insulation materials, i.e., material then deteriorate. Examples of cold insulation include Urethane Foam, Expanded Polystyrene, Resin bonded glass wool, Resin Bonded Glass wool, and Phenolic Foam.

Economics of Cold Insulation:

Unlike hot insulation system, the concern area in Cold Insulation is the heat gain into the refrigerated space, which leads to increase in the refrigeration load (TR) & energy consumption as a consequence. The cost of heat gain can thus be assessed & evaluated

against cost of additional cold insulation thickness, to optimize overall energy consumption & cost in refrigeration system.

5.2 Refractories

Refractories are non-metallic materials have insulating and other chemical and physical properties that make them able to contain the heat generated by burning of the fuel in the furnace and to minimise heat losses, withstand high temperatures and make them applicable for structures, or as components of systems, that are exposed to environments above 1,000 °F (811 K; 538 °C) and should not contaminate the material with which it is in contact. Refractory materials are used in furnaces, kilns, incinerators, and reactors.

The various combinations of operating conditions under which refractories are used, make it necessary to manufacture a range of refractory materials with different properties and accordingly are made in varying combinations and shapes for different applications. A furnace designer should have a clear idea of the service conditions of different refractory and for which the refractory is being used and needs to consider the following points, before selecting a refractory.

- Area of application
- Working temperatures
- Extent of abrasion and impact
- Structural load of the furnace
- Stress due to temperature gradient in the structures and temperatures fluctuations
- Chemical compatibility with the furnace environment
- Heat transfer and fuel conservation
- Cost consideration

Properties of Refractories

Some of the important properties of refractories are:

Size: The size and shape of the refractories is a part of the design feature. It is an important feature in design since it affects the stability of any structure. Accuracy and size is extremely important to enable proper fitting of the refractory shape and to minimize the thickness and joints in construction.

Bulk density: A useful property of refractories is bulk density, which defines the material present in a given volume. An increase in bulk density of a given refractory increases its volume stability, its heat capacity, as well as resistance to slag penetration.

Porosity: The apparent porosity is a measure of the volume of the open pores, into which a liquid can penetrate, as a percentage of the total volume. This is an important property in cases where the refractory is in contact with molten charge and slags. A low apparent porosity is desirable since it would prevent easy penetration of the refractory size and continuity of pores will have important influences on refractory behaviour. A large number of small pores is generally preferable to an equivalent number of large pores.

Cold crushing strength: The cold crushing strength, which is considered by some to be of doubtful relevance as a useful property, other than that it reveals little more than the

ability to withstand the rigors of transport, can be used as a useful indicator to the adequacy of firing and abrasion resistance in consonance with other properties such as bulk density and porosity.

Pyrometric cone equivalent (PCE): Temperature at which a refractory will deform under its own weight is known as its softening temperature which is indicated by PCE. Refractories, due to their chemical complexity, melt progressively over a range of temperature. Hence refractoriness or fusion point is ideally assessed by the cone fusion method. The equivalent standard cone which melts to the same extent as the test cone is known as the pyrometric cone equivalent.

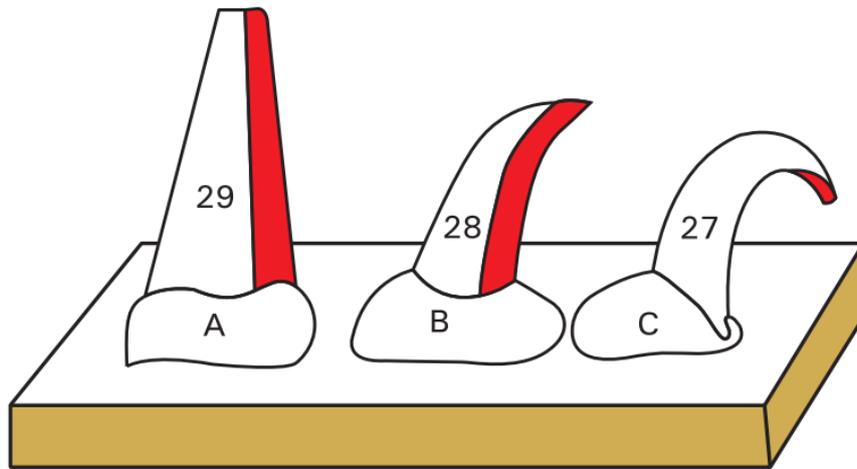


Figure 5.3: Pyrometric Cones

Thus in the Figure 5.3 refractoriness of Sample A is much higher than B and C. The pyrometric cone equivalent indicates only the softening temperature. But, in service the refractory is subjected to loads which would deform the refractory at a much lower temperature than that indicated by PCE. With change in the environmental conditions, such as reducing atmosphere, the P.C.E. value changes drastically.

Refractoriness under load (RUL): The refractoriness under load test (RUL test) gives an indication of the temperature at which the bricks will collapse, in service conditions with similar load.

Creep at high temperature: Creep is a time dependent property which determines the deformation in a given time and at a given temperature by a material under stress.

Volume stability, expansion, and shrinkage at high temperatures: The contraction or expansion of the refractories can take place during service. Such permanent changes in dimensions may be due to several operational factors.

Reversible thermal expansion: Any material when heated, expands, and contracts on cooling. The reversible thermal expansion is a reflection on the phase transformations that occur during heating and cooling.

Thermal conductivity: Thermal conductivity depends upon the chemical and mineralogical compositions as well as the glassy phase contained in the refractory and the application temperature. The conductivity usually changes with rise in temperature. In cases where

heat transfer is required though the brick work, for example in recuperators, regenerators, muffles, etc. the refractory should have high conductivity. Low thermal conductivity is desirable for conservation of heat by providing adequate insulation.

The provisions for back-up insulation, conserves heat but at the same time it increases the hot face temperature and hence the demand on the refractory quality increases.

Light weight refractories of low thermal conductivity find wider applications in the moderately low temperature heat treatment furnaces, where its primary function is usually conservation of energy. It is more so in case of batch type furnaces where the low heat capacity of the refractory structure would minimize the heat storage during the intermittent heating and cooling cycles.

5.2.1 Selection of Refractories

The selection of refractories for any particular application is made with a view to achieve the best performance of the equipment furnace, kiln or boiler and depends on their properties. Further, the choice of a refractory material for a given application will be determined by the type of furnace or heating unit and the prevailing conditions e.g. the gaseous atmosphere, the presence of slags, the type metal charges etc. It is, therefore, clear that temperature is by no means the only criterion for selection of refractories. Important physical properties of some insulating refractories that are considered in selecting them are shown in the following Table 5.2.

Table 5.2: Physical properties of some insulating refractories

Type/grade	Thermal Conductivity at 400 °C	Max. safe Temp °C	Cold Crushing Strength Kg/cm ²	Porosity %	Bulk Density Kg/m ²
Diatomite Solid	0.025	1000	270	52	1090
Diatomite Porous	0.014	800	110	77	540
Clay	0.030	1500	260	68	560
High Alumina	0.028	1500-1600	300	66	910
Silica	0.040	1400	400	65	830

Beside the physical properties chemical properties and the working atmosphere where these are used also influence their selection. Considering these factors refractories are classified based on the refractory constituents, behaviour of the refractory constituents in their working atmosphere, and the form of refractory application

5.2.2 Classification of Refractories

Refractories are classified based on their refractoriness, based on chemical composition of refractories, their form and operating temperatures.

Classification based on refractoriness

Based on the property of refractoriness, they are usually classified into four classes using their PCE¹ (pyrometric cone equivalent) value as given in table 5.3.

Table 5.3: Classification of Refractories based on PCE Value

Class of Refractory	PCE value
Super duty refractories	33 - 38
High duty refractories	30 - 33
Intermediate duty refractories	28 - 30
Low duty refractories	19 - 28

Classification based on composition of Refractories

Refractories can be classified on the basis of chemical composition and method of manufacture. The constituents, characteristics and applications of these refractories are summarized in table 5.4.

Table 5.4: Classification of Refractories based on Materials, their characteristics and applications

Refractory Type	General Characteristics	Application
Acid Bricks: Used in areas where both slag and atmosphere are acidic and are attacked by alkalis (basic slags).		
Silica	High strength at high temperatures, residual expansion, low specific gravity, high expansion coefficient at low temperatures, low expansion coefficient at high temperatures	Glass tank crown, copper refining furnace, electric arc furnace roof
Fused silica	Low thermal expansion coefficient, high thermal shock resistance, low thermal conductivity, low specific gravity, low specific heat	Coke over, hot stove, soaking pit, glass tank crown
Fireclay (Chamotte)	Consists of kaolinite ($\text{Al}_2\text{O}_3 \cdot 2\text{SiO}_2 \cdot 2\text{H}_2\text{O}$) and minor amounts of other clay materials. Low thermal expansion coefficient. Low thermal conductivity, low specific gravity, low specific heat, low strength at high temperatures, less slag penetration	Ladle, runner, sleeve, coke oven, annealing furnace, blast furnace hot stove, reheating furnace, soaking pit
Alumina	High refractoriness, high mechanical strength, high slag resistance, high specific gravity, relatively high thermal conductivity	Hot stove, stopper head, sleeve, soaking pit cover, reheating furnace, glass tank, high-temperature kiln.
High alumina	Composed of bauxite or other raw materials that contain 50 to 87.5 percent alumina. High refractoriness, high	Slide gate, aluminium melting furnace, skid rail, ladle,

¹The pyrometric cone is "A pyramid with a triangular base and of a defined shape and size; the "cone" is shaped from a carefully proportioned and uniformly mixed batch of ceramic materials so that when it is heated under stated conditions, it will bend due to softening, the tip of the cone becoming level with the base at a definitive temperature. Pyrometric cones are made in series, the temperature interval between the successive cones usually being 20 degrees Celsius. The number of that standard pyrometric cone whose tip would touch the supporting plaque simultaneously with a cone of the refractory material being investigated when tested in accordance with ASTM Test Method C-24.

Refractory Type	General Characteristics	Application
	mechanical strength, high slag resistance, high specific gravity, relatively high thermal conductivity	incinerator, reheating furnace hearth.
Roseki	Low thermal expansion coefficient, high thermal shock resistance, low thermal conductivity, low specific gravity, low specific heat	Ladle, runner, sleeve, coke oven, hot stove, soaking pit, annealing, blast and reheating furnace.
Zircon	Containing Zirconium silicate (ZrO_2SiO_2). Maintains good volume stability for extended periods, has high thermal shock resistance, high slag resistance, high specific gravity	Ladle, nozzle, stopper head, sleeve
Zirconia	High melting point, low wet ability against molten metal, low thermal conductivity, high corrosion resistance, high specific gravity	Nozzle for continuous casting, glass tank, high-temperature furnace, crucible.
Alumina zirconia silica	High slag resistance, high corrosion resistance against molten glass	Glass tank, incinerator, ladle, nozzle for continuous casting
Mullite	Made from kyanite, sillimanite, andalusite, bauxite or mixtures of alumina silicate materials; mullite refractories are about 70% alumina. They maintain a low level of impurities and high resistance to loading in high temperatures and offers good thermal shock resistance, excellent thermal stability, resistance to most chemical attack, resistance to abrasion and good electrical resistivity.	Steel ladles, lances, reheat furnaces and slide gates are examples of mullite aggregate based products with various alumina contents. In kiln areas such as kiln setter slabs and posts for supporting ceramic ware during firing
Basic Bricks: Are stable to alkaline slags, dusts and fumes at high temperatures and are attacked by acid slags.		
Lime	High slag resistance, low hydration resistance	Special refining surface
Magnesia	High refractoriness, relatively low strength at high temperature, high basic slag resistance, low thermal shock resistance, low durability at high humidity	Hot-metal mixer, secondary refining vessel, rotary kiln, checker chamber of glass tank, electric arc furnace
Magnesia-chrome	High refractoriness, High refractoriness under load, high basic slag resistance, relatively good thermal shock resistance (low MgO bricks), high strength at high temperature (direct bonded and fusion cast)	Hot-metal mixer, electric arc furnace, secondary refining vessel, nonferrous refining furnace, rotary cement kiln, lime and dolomite kiln, copper furnace, ladle, checker chamber of glass tank, slag line of electric arc furnace, degasser for copper, non-ferrous smelter.

Refractory Type	General Characteristics	Application
Dolomite	High refractoriness under load, high basic slag resistance, low durability in high humidity, high thermal expansion coefficient	Basic oxygen furnace, electric arc furnace, secondary refining vessel, rotary cement kiln.
Spinel	High thermal shock resistance, high strength at high temperatures, high slag resistance	Rotary cement kiln, ladle.
Neutral Non-oxide Bricks : are chemically stable to both acids and bases and are used in areas where slag and atmosphere are either acidic or basic		
Chrome	High refractoriness, low strength at high temperature, low thermal resistance	Buffer brick between acid and base brick
Silicon carbide	They are produced by the reaction of sand and coke in an electric furnace. High refractoriness, high strength at high temperature, high thermal conductivity, high thermal shock resistance, reduced oxidation resistance at high temperature, high slag resistance	Kiln furniture, incinerator, blast furnace
Silicon carbide-graphite	High refractoriness, high strength at high temperature, high thermal conductivity, high thermal shock resistance	Incinerator
Silicon nitride	High strength, high thermal shock resistance, relatively high oxidation resistance	Kiln furniture, blast furnace
Composite		
Silicon carbide	Containing High corrosion resistance against low iron oxide, high strength at high temperatures, high thermal shock resistance	Ladle, blast furnace, electric arc, torpedo ladle, iron ladle
Magnesia-carbon	High slag resistance, high thermal shock resistance	Basic oxygen furnace, electric arc furnace, ladle
Alumina-carbon	High refractoriness, high thermal shock resistance, high corrosion resistance	Submerged entry nozzle, slide gate

Classification based on Form of Refractories

Refractories and refractory materials are used in various forms depending on the requirements. Table 5.5 gives different forms in which these are produced.

Table 5.5: Forms of Refractories

Kind	Definition
Shaped Refractories	
Bricks	Refractories that have shapes and are used to line furnaces, kilns, glass tanks, incinerators, etc.
Insulating firebrick	Low thermal conductivity firebrick.
Unshaped Refractories (Monolithic)	

Mortar	Materials for bonding bricks in a lining. The three types of mortar – heat-setting; air-setting; and hydraulic-setting – have different setting mechanisms.
Castables	Refractory materials are mixed with hydraulic-setting cement (either Portland or a high-alumina cement) and casted. Used to line furnaces, kilns, formation of the bases of tunnel kiln cars used in the ceramic industry etc.
Plastics	Refractories in which raw materials and plastic materials are mixed with water. Plastic refractories are roughly formed, sometimes with chemical additives.
Gunning mixes	Refractories that are sprayed on the surface by a gun.
Ramming mixes	Granular refractories that are strengthened by gunning formulation of a ceramic bond after heating. Ramming mixes have less plasticity and are installed by air rammer.
Slinger mixes	Refractories installed by a slinger machine.
Patching materials/ coating materials	Refractories with properties similar to refractory mortar. However, patching materials have controlled grain size for easy patching or coating.
Light weight castables	These are porous lightweight materials which are mixed with hydraulic cement and water and formed by casting. Lightweight castables are used to line furnaces, kilns, etc.
Fibrous Materials	
Ceramic fiber	Ceramic fibre is analumino silicate or ZrO ₂ added alumino silicate material manufactured by blending and melting alumina and silica at temperature of 1800–2000 °C and fibre made by blowing compressed air or dropping the melt on spinning disc. There are produced as in blanket, felt, module, vacuum, rope, or loose fiber form.

Recommended operating temperatures for continuous operations are given in the Table 5.6.

Table 5.6: Recommended operating temperature for Continuous Operation

	<i>Al₂O₃ %</i>	<i>SiO₂ %</i>	<i>ZrO₂ %</i>
1150 °C	43-47	53-57	-
1250 °C	52-56	44-48	-
1325 °C	33-35	47-50	17-20

5.3 High Emissivity Coatings

Emissivity is the measure of a material’s ability to both absorb and radiate heat. The high emissivity materials when coated increases the surface emissivity of materials, with resultant benefits in heat transfer efficiency and in the service life of heat transfer components like refractories and metallic components such as radiant tubes and heating elements. High emissivity coatings are applied in the interior surface of furnaces or where rapid heating is required. The use of such coatings was found to reduce fuel or power to tune of 25-45%. The Figure 5.3 shows emissivity of various insulating materials including high emissivity coatings. High emissivity coating shows a constant value over varying process temperatures.

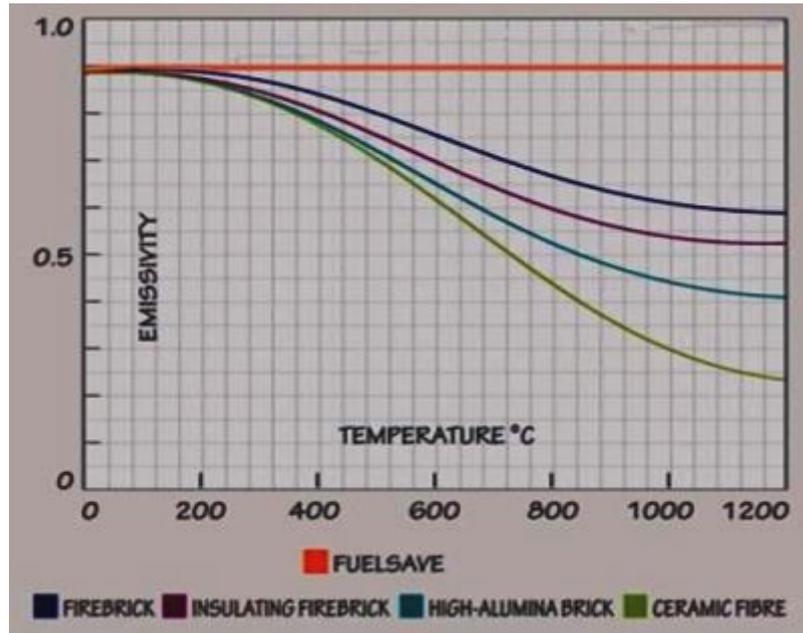


Figure 5.4: Emissivity of refractory materials at different temperatures

Furnaces, which operate at high temperature, have emissivity of 0.3. By using high emissivity coatings this can go up to 0.8 thus effectively increasing the radiative heat transfer.

Chapter 6: FBC Boilers

6.1 Introduction

The traditional grate fuel firing systems have got limitations and are techno-economically unviable to meet the challenges of future. Fluidized bed combustion has significant advantages over conventional firing systems and offers multiple benefits. Since its introduction in the 1970s the technology has gained acceptance in various industrial applications.

It is known for its ability to burn low-grade fuels with low calorific value, high ash content and high moisture content. The fuels being burnt in these boilers include coal, washery rejects, rice husk, bagasse & other agricultural wastes, and their capacities range from 0.5 T/hr. to 74T/hr.

Other advantages are fuel flexibility, emission performance, re-use of non-hazardous by-products (e.g. gypsum) and the possibility of the technology to be implemented in an existing plant (retrofit).

6.2 Fundamentals of FBC

A fluidized bed consists of a stream of gas flowing upward through a bed of solid particles such as ash or sand. At low gas flow rates, the gas permeates through the bed without disturbing the particles and is called a **packed bed**. As the gas flow rate increases, the force exerted on the particles becomes greater until eventually the gas stream supports the particles and the bed becomes "**fluidized**." This causes the particles to separate and the bed to expand. The gas velocity at this point is termed **the minimum fluidizing velocity**. As the gas velocity is increased further, bubbles form and rise through the bed. Bubbles passing through the bed cause a highly turbulent mixing of the particles and give the bed the appearance of a boiling fluid. At this point, a bed surface or the boundary separating the bed material and the space above it is visible. This bed is called a "**bubbling bed**." As the gas velocity is further increased, the smaller particles become entrained in the gas stream and are transported from the bed. If the velocity is increased sufficiently, a condition would be reached where all the particles would be transported from the bed and a distinct bed surface is no longer apparent. A system (at this velocity) with a collection device to separate the gas and return the particles to the bed area is called a "**circulating bed**."

This phenomenon of fluidization with increasing gas velocity is illustrated in figure 6.1.

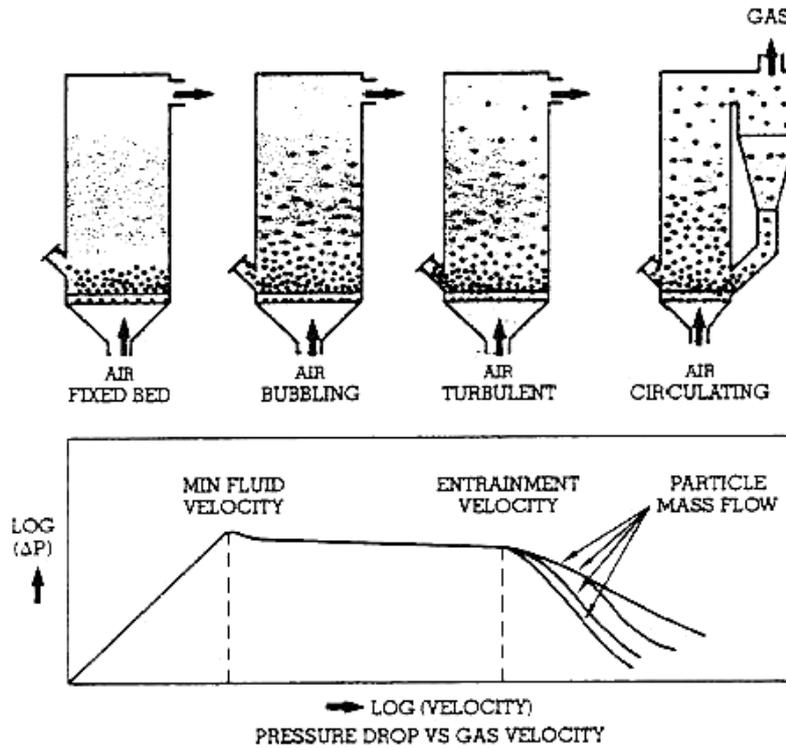


Figure 6.1: Principle of fluidization

With higher air velocities, the bed particles leave the combustion with the flue gases so that solids recirculation is necessary to maintain circulating fluidized bed. The mean solids velocity increases at a slower rate than does the gas velocity, as illustrated in Figure 6.2. Therefore, a maximum slip velocity between the solids and the gas can be achieved resulting in good heat transfer and contact time between solids and gas.

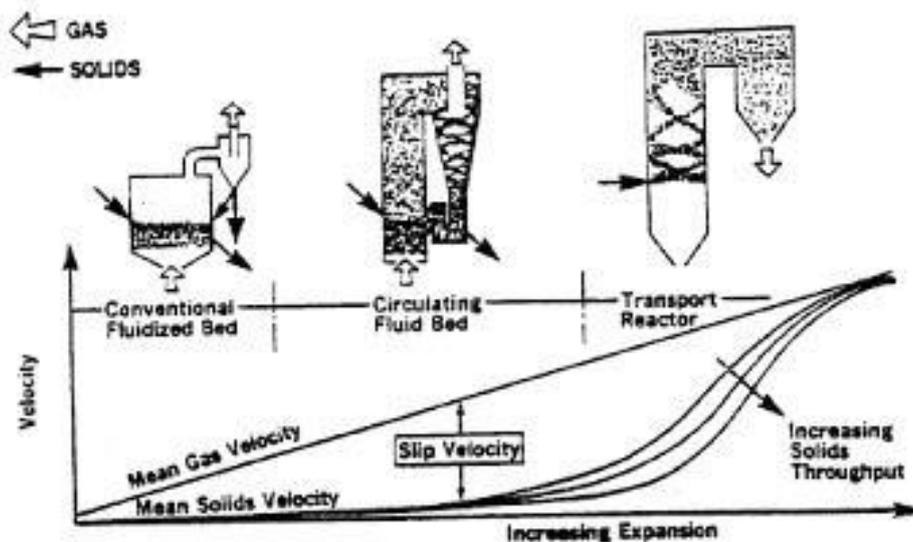


Figure 6.2: Relation between gas velocity and solid velocity

A fluidized bed combustion boiler is one where combustion takes place over a fluidized bed. If the sand in a fluidized state is heated to the ignition temperature of the coal and the

coal is injected continuously into the bed, the coal will burn rapidly and the bed attains a uniform temperature due to effective mixing. This, in essence, is fluidized bed combustion.

The furnace combustion takes place at about 840°C to 950°C. To start a cold boiler, the bed is first preheated to around 540°C by passing through the combustion products from an auxiliary heater. At this temperature solid fuel could be ignited. Fuel is then introduced either from the base or the top of the bed. Heated fluidizing air is blown through the distributor plate to supply the primary combustion air. Combustion is continued in the freeboard space where secondary air is supplied for completing the combustion.

The combustion of fuel with air can produce flame temperatures in excess of 1650°C, which can lead to catastrophic material failure. To prevent such problems in bed operations, the temperature must be kept below 1100 °C, temperature much below the ash fusion temperature, and is never allowed to reach adiabatic combustion temperature to avoid melting of ash. This is achieved usually by using in-bed boiler tubes (a heat sink) in combination with controlling the fuel content in the bed to, for example, less than 5%.

A typical FBC boiler plant is shown schematically in figure 6.3

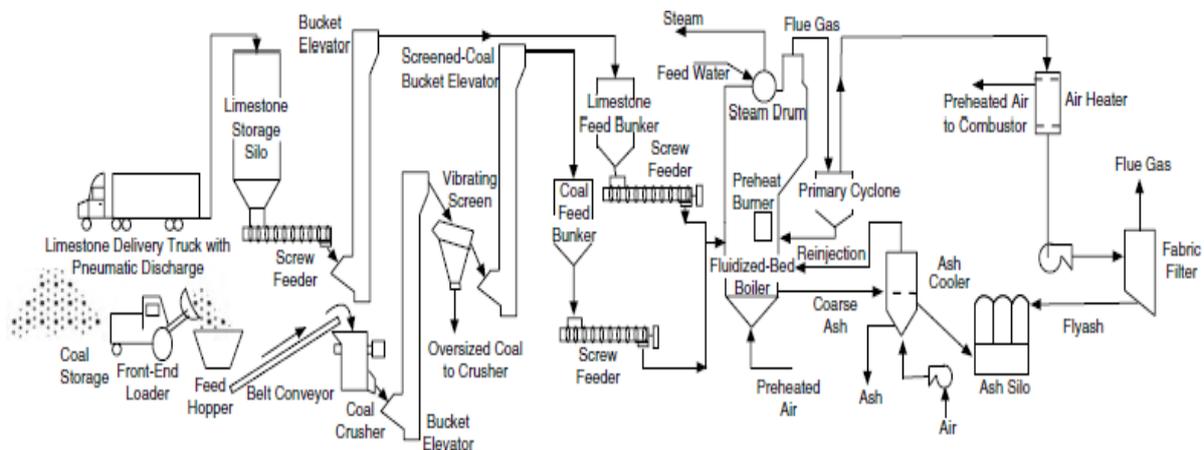


Figure 6.3: Typical fluidized bed boiler plant firing solid fuels

6.2.1 Components of FBC Boiler

A FBC boiler normally will have the following seven principal components as shown in figure 6.3.

1. Feeder for Fuels and Sulphur Sorbents.
2. Air Movers.
3. Air Distributor.
4. Plenum Chamber.
5. Combustion Chamber.
6. Solid Withdrawal System.
- 7.

6.2.1.1 Fuel Feeders

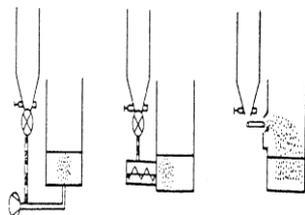
For feeding fuel and sorbent like limestone or dolomite, usually two methods are followed as explained below:

Under Bed Pneumatic Feeding: If the fuel is coal, it is crushed to 1-6mm size and pneumatically transported from feed hopper to the combustor through a feed pipe piercing the distributor. Based on the capacity of the boiler, the number of feed points increases, as it is necessary to distribute the fuel into the bed uniformly.

Over-Bed Feeding: The crushed coal, 6-10 mm size is conveyed from coal bunker to a spreader by a screw conveyor.

The spreader distributes the coal over the surface of the bed uniformly. This type of fuel feeding system accepts over size fuel also and eliminates transport lines, when compared to under-bed feeding system.

Under-bed



Over-bed

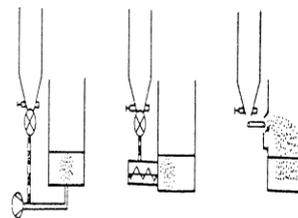


Figure 6.4: Types of Fuel Feeder

There are broadly four types of industrial feeders: gravity chute, screw feeder, spreader, and pneumatic feeders.

6.2.1.2 Air Movers

Air movers supply air for combustion and fluidization of the burning fluidized bed and pneumatic transportation of solid matters such as coal, limestone, and ash.

6.2.1.3 Air Distributor

An essential function of the distributor is to introduce the fluidizing air evenly through the bed cross section to keep the solid particles in constant motion, and prevent formation of defluidization zones within the bed. The distributor is normally constructed from metal plate with a number of perforations, in a definite geometric pattern. The perforations may be located in simple nozzles or nozzles with bubble caps, which serve to prevent solid particles from flowing back into the space below the distributor.

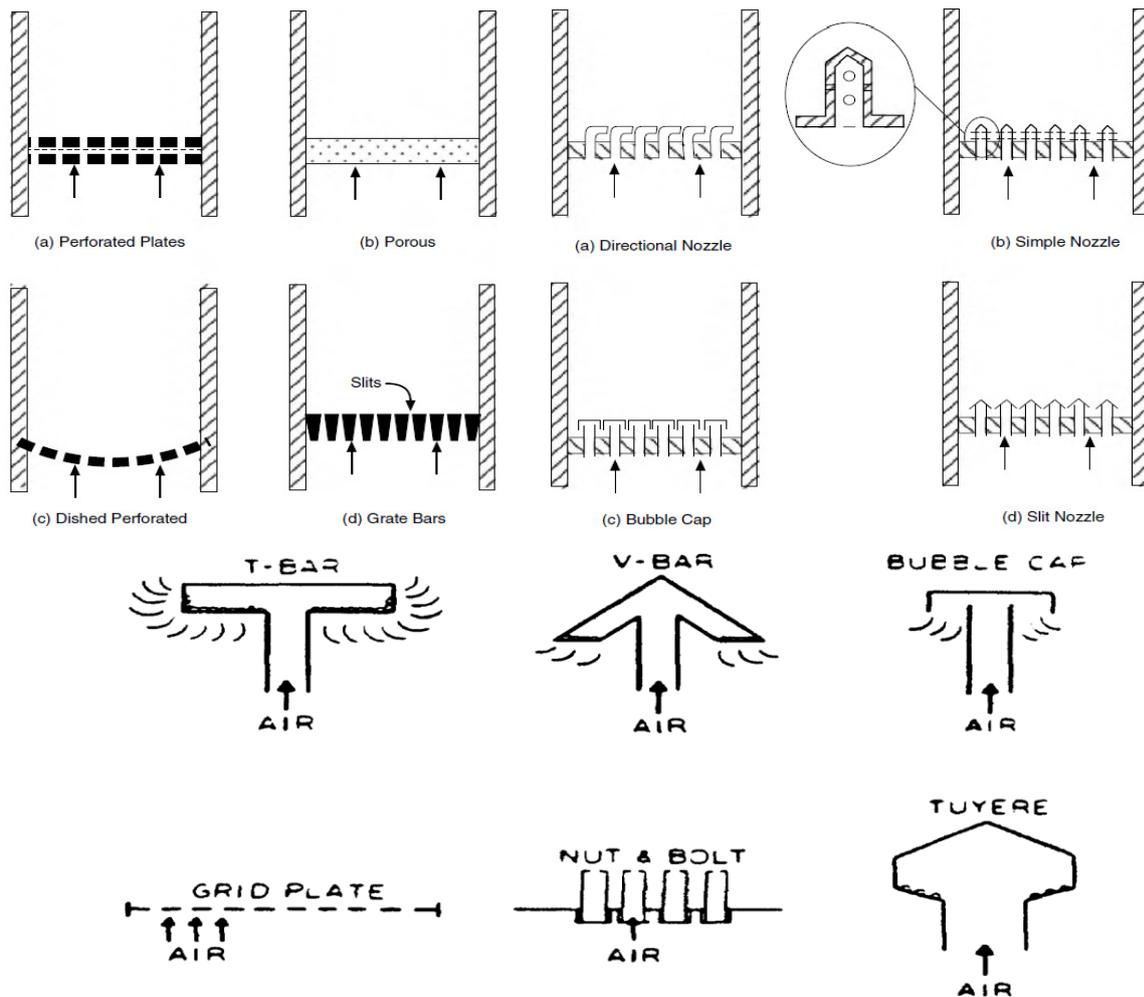


Figure 6.5: Air Distributors and Nozzles

The distributor plate forms the furnace floor and this is protected from high temperature of the furnace by:

- Refractory Lining
- A Static Layer of the Bed Material or
- Water Cooled Tubes.

6.2.1.4 Plenum Chamber

The plenum chamber is located directly underneath the distributor. Fluidizing air enters the distributor by way of the plenum chamber. The plenum chamber serves to minimize air pressure surges, and to contain the spent materials that drifted (weeping) through the distributor. The function of the air box is to distribute the air under the grid as uniformly as possible.

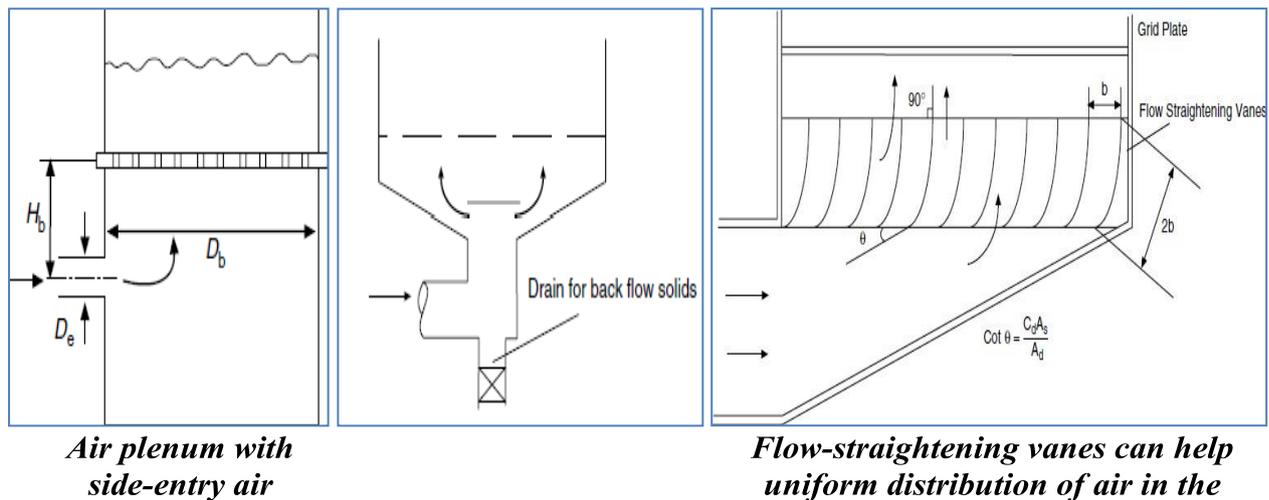


Figure 6.6: Types of Plenum Chamber

6.2.1.5 Bed & In - Bed Heat Transfer Surface

Bed Heat Transfer Surface: Depending on the bed height these are of two types with average bed particle size of about 1 mm:

- Shallow bed and
- Deep bed.

At the same fluidizing velocity, the two ends fluidize differently, thus affecting the heat transfer to an immersed heat transfer surfaces. A shallow bed offers a lower bed resistance and hence a lower pressure drop and lower fan power consumption. In the case of deep bed, the pressure drop is more and this increases the effective gas velocity and also the fan power.

In - Bed Heat Transfer Surface: In a fluidized in-bed heat transfer process it is necessary to transfer heat between the bed material and an immersed surface, which could be a tube bundle, or a coil. The heat exchanger orientation can be horizontal, vertical or inclined. From a pressure drop point of view, a horizontal bundle in a shallow bed is more attractive than a vertical bundle in a deep bed. Also, the heat transfer in the bed depends on number of parameters like

- (i) Bed pressure
- (ii) Bed temperature
- (iii) superficial gas velocity
- (iv) particle size,
- (v) Heat exchanger design and
- (vi) Gas distributor plate design.

6.2.1.6 Ash Removal System

Bed ash removal: In the FBC boilers, the bottom ash constitutes roughly 30 - 40 % of the total ash, the rest being the fly ash. The bed ash is removed by both continuous over flow, to maintain bed height, and also by intermittently from the bottom to remove over size particles to avoid accumulation and consequent defluidization. While

firing high ash coal such as washery rejects, the bed ash overflow drain quantity is considerable so special care has to be taken.

Fly ash removal: The amount of fly ash to be handled in FBC boiler is relatively very high, when compared to conventional boilers. This is due to elutriation of particles at high velocities. Fly ash carried away by the flue gas is removed in number of stages. Firstly in convection section, then from the bottom of air pre heater/economizer, and a major portion in dust collectors.

The type of dust collectors used are cyclone, bag filters, electrostatic precipitators (ESP's) or some combination of all of these. To increase the combustion efficiency, recycling of fly ash is practiced in some of the units.

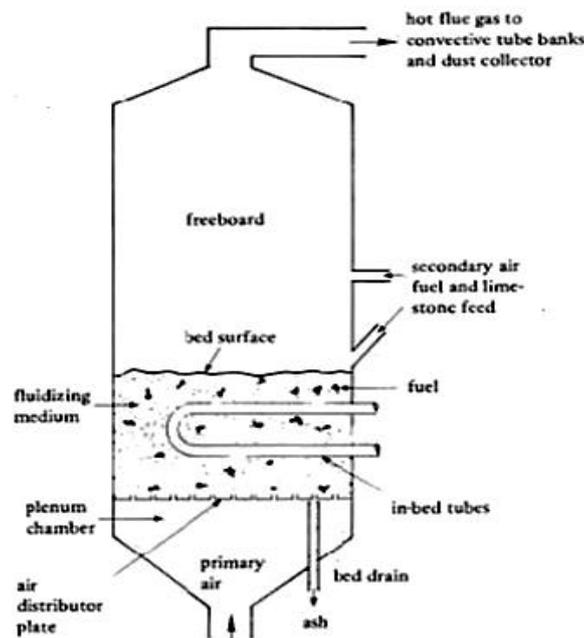


Figure 6.7: Conventional or Bubbling FBC showing components of FBC

6.3 Advantages of FBC Boilers

The criteria that an industrial boiler user applies to select a boiler are: fuel flexibility, operational reliability, environmental acceptability, and economic viability. On each of this criteria FBC boilers have advantage over conventional boilers as described below.

6.3.1 Fuel flexibility

Most FBC boilers can practically burn all combustible material.

The high thermal inertia and latent heat stored in the bed material allow newly added fuel to ignite quickly and evenly. Wet or low-quality fuels can also be burned efficiently. However, the degree to ease of combustibility varies. The fuels that can be used in FBC along with the ease of usage is shown in figure 6.8.

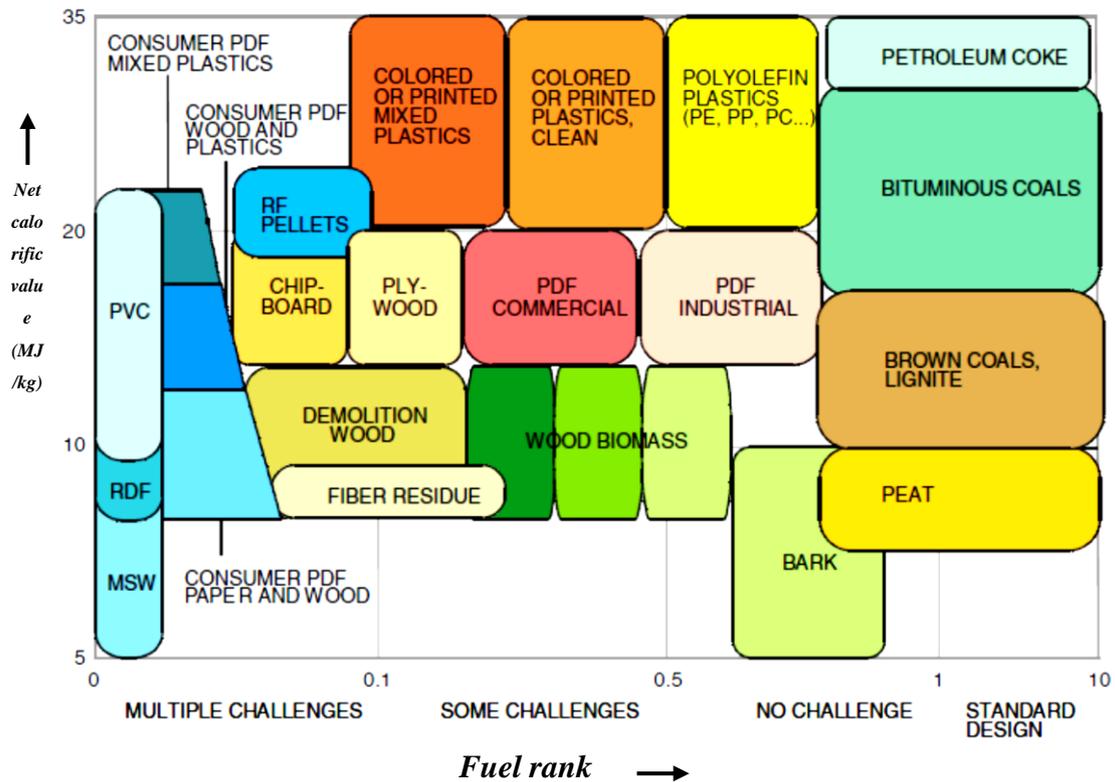


Figure 6.8: Fuels for FBC boiler and their Ease of Use

6.3.2 Operational Reliability

A boiler's operational reliability depends on many factors namely, the auxiliary equipment that pre-treats the fuel (e.g., crushing, screening, feeding); the boiler feed water treatment unit that prepares the water to meet boiler requirements; and the systems that handle combustion flue gas and refuse.

FBC are more reliable than conventional boilers due to following reasons.

- Non usage of pneumatic feeding system
- Operation at lower and more uniform temperature than conventional boilers. Because of the turbulent mixing and the efficient transfer of heat, temperatures within the fluidized bed are uniform which minimizes hot spots. These lower temperatures are below the melting points of most ashes which avoids the slagging and fouling of heat transfer surfaces with melted ash, one of the major problems encountered in solid-fuel fired boilers.
- Having no moving parts that need continuous and frequent maintenance.

6.3.3 Environmental Acceptability

Fluidized bed combustion is more environmentally friendly as it suppresses sulphur dioxide at the time of combustion rather than removing it from flue gases later with expensive and sometimes difficult-to-operate, post-combustive ("scrubbing") devices. By virtue of low combustion temperature, the FBC boiler exhibits attractive sulphur retention

characteristics and low nitrogen oxides emission thus enabling a unit to meet the Bangladesh boiler emissions norms as given in table 6.1 in cost effective terms.

Table 6.1: Standards for Gaseous emissions for Boiler of Industrial Unit

S. No.	Parameters	Standards (mg/Nm³)
Power plant		
1.	Particulate matter	
a.	Capacity ≥200 MW	150
b.	Capacity <200 MW	350
2.	Oxides of nitrogen for Gas fuel based power plant	
a.	≥ 500 MW	50 ppm
b.	200 – 500 MW	40 ppm
c.	< 200 MW	30 ppm
Industrial Boiler		
1.	Soot and particulate (fuel based)	
a.	Coal	500
b.	Gas	100
c.	Oil	300
2.	Oxides of Nitrogen (fuel based)	
a.	Coal	600
b.	Gas	150
c.	Oil	300
Stack height for Power and Industrial boiler		
a.	Steam per hour upto 15 tons	11
b.	Steam per hour more than 15 tons	14Q^{0.3}
Where, Q = Emission of sulphur dioxide in kg/hr.		
Boiler using bagasse		
1.	Particulate matter	
a.	Step grade	250
b.	Pulsating	500
c.	Horse shoe spreader stocker	800

6.3.4 Economic Viability

The unique feature of FBC to burn inexpensive, low-grade, and high sulphur coals indicate a favourable cost differential in using FBC.

As they yield a rate of heat transfer five to ten times more efficient than the rate of heat transfer achieved through conventional coal-firing boilers it leads to reducing the size of boiler substantially, thus reducing costs.

Table 6.2: Summary of Advantages and Disadvantages of FBC Boilers

Advantages	Disadvantages
<ul style="list-style-type: none">• Fuel flexibility• In situ SO₂ removal• Low NO_x emission• Good system availability• No slagging• Low corrosion rate• Easy sizing of fuel	<ul style="list-style-type: none">• Higher power of air fan• Larger cross-section of a furnace• Higher surface loss of heat• Higher carbon-in-ash level• Higher erosion rate

The criteria for considering FBC boilers is given in Figure 6.9.

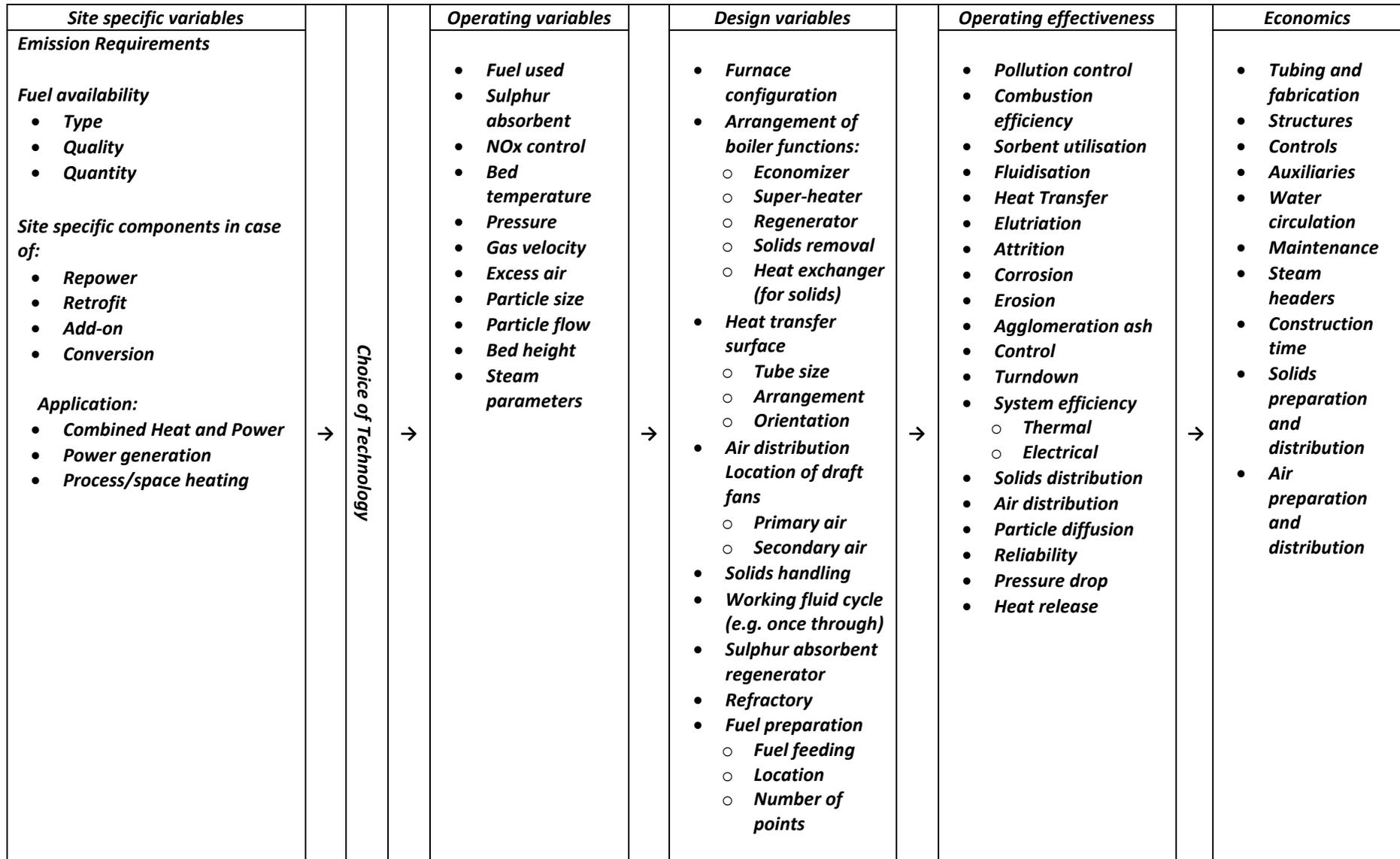


Figure 6.9: Overview of Variables influencing economic and technological performance

6.4 Types of Fluidized Bed Combustion Boilers

There are three variants of FBC technology.

- The bubbling fluidized bed (BFB) was the first version of FBC technology also called Atmospheric Classic Fluidised Bed Combustion System (AFBC). BFB technology is well suited for utilisation of ‘difficult’ fuels such as high moisture fuels (e.g. wastes and sludge’s), high-ash fuels (e.g. some types of municipal solid waste and refuse-derived fuel) and low volatile fuels (including anthracite, culm and petroleum coke) as well as for smaller industrial applications.
- The circulating fluidized bed (CFB) also called Atmospheric Circulating (fast) Fluidised Bed Combustion system (CFBC) is the second variant and is derived from the BFB technology and surpasses its predecessor in terms of sulphur removal, efficiency and scale. The basic difference between BFB and its successor CFB is the fluidisation velocity, which is higher for CFB compared to BFB.
- The third variant is a hybrid type of the BFB and CFB and is called Pressurised Fluidised Bed Combustion System (PFBC). It was developed to combine the advantages of both BFB and CFB and thus found its application to be in medium-scaled (industrial) capacity range.

6.4.1 AFBC / Bubbling Bed

In this type of FBC coal is crushed to a size of 1 – 10 mm depending on the rank of coal, type of fuel feeding and fed to the combustion chamber. The atmospheric air, which acts as both the fluidization air and combustion air, is delivered at a pressure, and flows through the bed after being preheated by the exhaust flue gases. The in-bed tubes carrying water generally act as the evaporator.

Fluidized bed combustion is to be done in a relatively narrow temperature range within which the bed must be operated. With coal, there is risk of clinker formation in the bed if the temperature exceeds 950°C and combustion efficiency declines below 800°C. For efficient sulphur retention the temperature should be in the range of 800 - 850° C.

The combustion gases pass over the super heater sections of the boiler, flow past the economizer, the dust collectors and the air pre heaters before being exhausted to atmosphere.

Almost all atmospheric bubbling bed boilers (AFBC) use the in-bed evaporator tubes for extracting the heat from the bed to maintain the bed temperature. Typical fluidized bed combustors of this type are shown in Figures 6.10 and 6.11.

The bubbling bed has heat transfer tubes in the bed of limestone, sand and fuel. The velocity of fluidizing air is in the range of 1.2 to 3.7 m/sec. About 2 to 4 Kgs of solids are recycled per kg of fuel burned.

The bed depth is usually 0.9 m to 1.5 m deep and the pressure drop averages about 1 inch of water per inch of bed depth. The bulk of the bed consists of limestone, sand, ash, or other material and a small amount of fuel. The rate at which air is blown through the bed determines the amount of fuel that can be combusted.

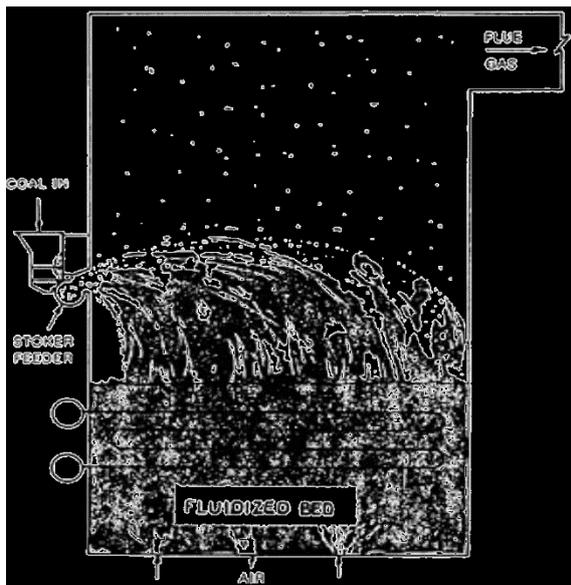
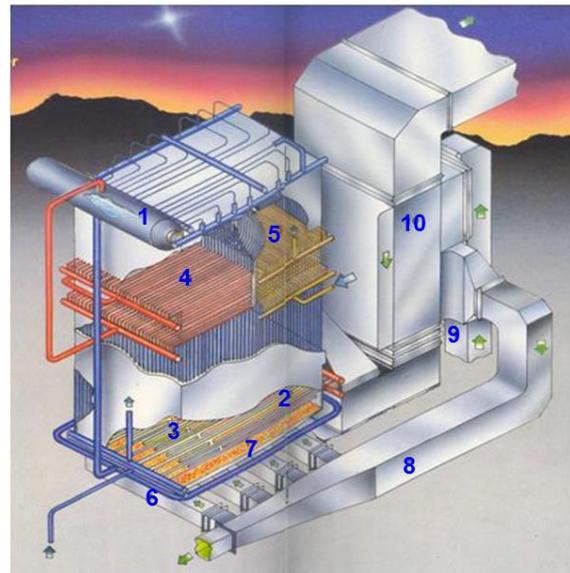


Figure 6.10: Bubbling bed boiler-1



1 Steam Drum ,2 Bed Superheater, 3 Bed Evaporator, 4 Convection Superheater, 5 Economiser, 6 Windbox, 7 Air Distributor Plate, 8 Hot Air Duct, 9 Cold Air Duct, 10 Air Preheater

Figure 6.11: Bubbling bed boiler-2

6.4.2 Circulating Fluidised Bed Combustion (CFBC)

This CFB technology utilizes the fluidized bed principle in which crushed (6–12 mm size) fuel and limestone are injected into the furnace or combustor. The particles are suspended in a stream of upwardly flowing air (60-70% of the total air), which enters the bottom of the furnace through air distribution nozzles. The balance of combustion air is admitted above the bottom of the furnace as secondary air. While combustion takes place at 840-900°C, the fine particles (<450 microns) are elutriated out of the furnace with flue gas velocity of 4-6 m/s. The particles are then collected by the solids separators and circulated back into the furnace as shown in Figure 6.12.

The particles circulation provides efficient heat transfer to the furnace walls and longer residence time for carbon and limestone utilization. The controlling parameters in the CFB combustion process are temperature, residence time and turbulence.

In a circulating system the bed parameters are so maintained as to promote solids elutriation from the bed. They are lifted in a relatively dilute phase in a solids raiser, and a down-comer with a cyclone provides a return path for the solids. There are no steam generating tubes immersed in the bed. Generation and super heating of steam takes place in the convection section, water walls, at the exit of the riser.

CFBC boilers are generally claimed to be more economical than AFBC boilers for industrial application requiring more than 75 – 100 T/hr of steam. For large units, the taller furnace characteristics of CFBC boiler offers better space utilization, greater fuel particle and sorbent residence time for efficient combustion and SO₂ capture, and easier application of staged combustion techniques for NO_x control than AFBC generators.

The circulating bed is designed to move solids out of the furnace area and to achieve most of the heat transfer outside the combustion zone. Some circulating bed units even have external heat exchangers. The fluidizing velocity in circulating beds ranges from 3.7 to 9 m/sec and solid recycle is 50 to 100 kg per kg of fuel burnt.

Furnace temperature is roughly the same in both AFBC and CFBC, but the circulating bed is said to achieve better calcium to sulphur utilization – 1.5 to 1 vs. 3.2 to 1 for the bubbling bed.

CFBC requires mechanical cyclones to capture and recycle the bed material and the requirement of a tall boiler.

A CFB could be good choice if the following conditions are met.

- Capacity of boiler is large to medium
- Sulphur emission and NO_x control is important
- The boiler is required to fire low grade fuel or fuel with fluctuating quality.

Major performance features of the circulating bed system are as follows:

- It has a high processing capacity because of the high gas velocity through the system.
- The temperature of about 870°C is reasonably constant throughout the process because of the high turbulence and circulation of solids. The low combustion temperature also results in minimal NO_x formation.
- Sulphur present in the fuel is retained in the circulating solids in the form of calcium sulphate is removed in solid form. The use of limestone or dolomite sorbents allows a higher sulphur retention rate.
- The combustion air is supplied at 1.5 to 2 psig rather than 3-5 psig as required by bubbling bed combustors.
- It has high combustion efficiency.
- It has a better turndown ratio than bubbling bed systems.
- In a bubbling bed system, the surface generally is perpendicular to the flow.

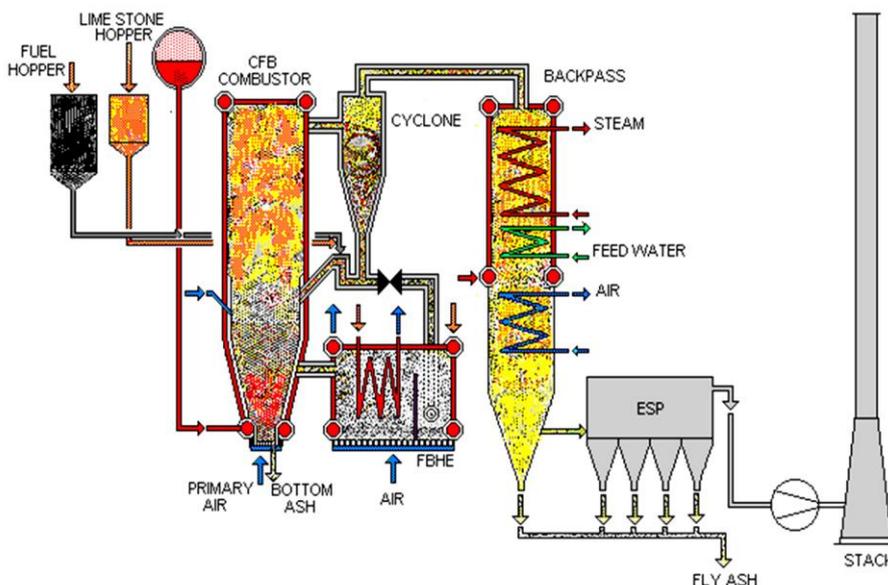


Figure 6.12: Circulating bed boiler design

Table 6.3: Comparison of BFBC and CFBC

Parameter	Units	BFBC	CFBC
Combustion Temperature	°C	760 - 870	800 - 900
Fuel particle size	mm	0 - 50	0 - 25
Fluidization Velocities	m/s	1 - 3	03-Oct
Solids Circulation		No	Yes
Particle Concentration		High in bottom, low in free board	Gradually decreases with furnace height
Limestone (a) particle size	mm	0.3 - 0.5	0.1 - 0.2
Average steam parameters			
Steam flow range	Kg/s	36 (13 - 139)	60 (12 - 360)
Steam temperature range	°C	466 (150 - 543)	506 (180 - 580)
Steam Pressure range	bar	72 (10 - 160)	103 (10 - 275)
Size	T/hr	All sizes	45-680
Combustion Efficiency	%		2 - 3 % Better than BFBC
Sorbent use		~100% more than CFBC	
Bed Area		2.5 times of CFBC	
Fuel Feed		Over-bed/Under-bed	In-bed
Heat recovery		In-bed tubes	No in-bed tubes
Boiler controls		Conventional	Conventional
Material handling		Conventional	Conventional
Start-up	Hrs.	4	8
O & M			Lower than BFBC
Plant power Auxiliary			Similar if BFBC is over-bed and more if BFBC is under- bed feed system
(a): Applicable in case when limestone is used for in-bed sulphur removal.			
Source:			
Development of fluidized bed combustion—An overview of trends, performance and cost Joris Koornneef , Martin Junginger, Andre´ Faaij, in Progress in Energy and Combustion Science 33 (2007) 19–55			
Comparison of Bubbling and Circulating Fluidized Bed Industrial Steam Generation by R N Gaglia and A Hall in Proceedings of the 1987 international Conference on FBC			

6.4.3 Pressurized Fluid Bed Combustion

Pressurised Fluidized Bed Combustion (PFBC) is a variation of fluid bed technology that is meant for large scale coal burning applications. In PFBC, the bed vessel is operated at pressure up to 16 atm (16 kg/cm²).

The off-gas from the fluidized bed combustor drives the gas turbine. The steam turbine is driven by steam raised in tubes immersed in the fluidized bed. The condensate from the steam turbine is pre-heated using waste heat from gas turbine exhaust and is then taken as feed water for steam generation.

The PFBC system can be used for cogeneration or combined cycle power generation. By combining the gas and steam turbines in this way, electricity is generated more efficiently than in conventional system. The overall conversion efficiency is higher by 5% to 8%. (Refer Figure 6.13).

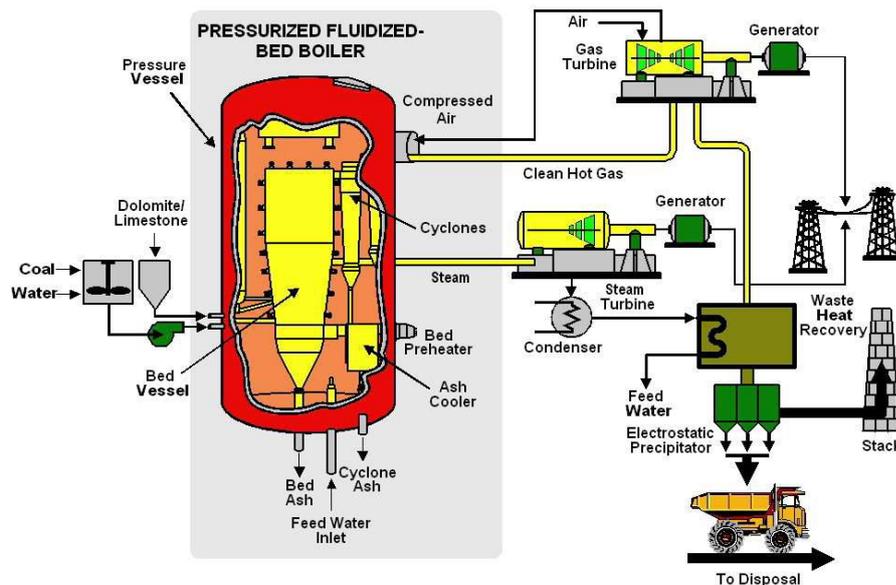


Figure 6.13: PFBC Boiler for Cogeneration

At elevated pressure, the potential reduction in boiler size is considerable due to increased amount of combustion in pressurized mode and high heat flux through in-bed tubes. A comparison of size of a typical 250 MW PFBC boiler versus conventional pulverized fuel-fired boiler is shown in the Figure 6.14.

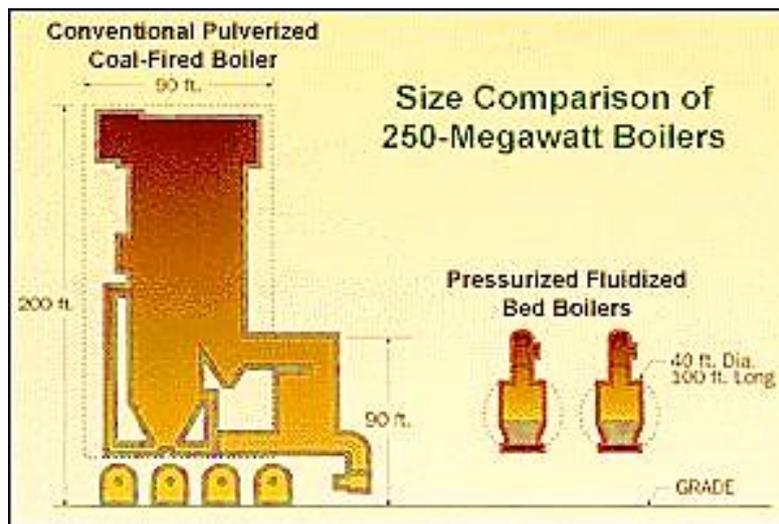


Figure 6.14: Comparison of PFBC boiler versus pulverized fuel boiler

6.5 Retrofitting of FBC Systems to Conventional Boilers

Retrofitting fluidized bed coal fired combustion systems to conventional boilers has been carried out successfully in many countries. The important aspects to be considered in retrofit projects are:

- Water/steam circulation design
- Furnace bottom-grate clearance
- Type of particulate control device
- Fan capacity
- Availability of space.

Retrofitting of a fluidized bed combustor to a conventional stoker fired water tube boiler may involve:

- The replacement of grate by a distributor plate with short stand pipes for admitting air from the wind box located under neath.
- Installation of stand pipes to remove ash from the bed.
- Provision of horizontal hair pin tubes in the bed with a pump for forced circulation from the boiler drum.
- Modification of crusher to size the coal/limestone mixture for pneumatic under-bed injection of the mixture.

Conversion of a conventional coal fired system to a fluidized bed combustion system can be accomplished without effecting major changes, after making a cost-benefit analysis. Oil fired boilers can also be converted to coal fired fluidized bed combustion systems. However, it has to be examined on a case to case basis.

6.6 Advantages of Fluidized Bed Combustion Boilers

1. High Efficiency

FBC boilers can burn fuel with a combustion efficiency of over 95% irrespective of ash content. FBC boilers can operate with overall efficiency of 84% (plus or minus 2%).

2. Reduction in Boiler Size

High heat transfer rate over a small heat transfer area immersed in the bed result in overall size reduction of the boiler.

3. Fuel Flexibility

FBC boilers can be operated efficiently with a variety of fuels. Even fuels like flotation slimes, washer rejects, agro waste can be burnt efficiently. These can be fed either independently or in combination with coal into the same furnace.

4. Ability to Burn Low Grade Fuel

FBC boilers would give the rated output even with inferior quality fuel. The boilers can fire coals with ash content as high as 62% and having calorific value as low as 2,500 kcal/kg. Even carbon content of only 1% by weight can sustain the fluidized bed combustion.

5. Ability to Burn Fines

Coal containing fines below 6 mm can be burnt efficiently in FBC boiler, which is very difficult to achieve in conventional firing system.

6. Pollution Control

SO₂ formation can be greatly minimized by addition of limestone or dolomite for high sulphur coals. 3% limestone is required for every 1% sulphur in the coal feed. Low combustion temperature eliminates NO_x formation.

7. Low Corrosion and Erosion

The corrosion and erosion effects are less due to lower combustion temperature, softness of ash and low particle velocity (of the order of 1 m/sec).

8. Easier Ash Removal — No Clinker Formation

Since the temperature of the furnace is in the range of 750 — 900 °C in FBC boilers, even coal of low ash fusion temperature can be burnt without clinker formation. Ash removal is easier as the ash flows like liquid from the combustion chamber. Hence less manpower is required for ash handling.

9. Less Excess Air — Higher CO₂ in Flue Gas

The CO₂ in the flue gases will be of the order of 14 — 15% at full load. Hence, the FBC boiler can operate at low excess air - only 20 — 25%.

10. Simple Operation, Quick Start-Up

High turbulence of the bed facilitates quick start up and shut down. Full automation of start up and operation using reliable equipment is possible.

11. Fast Response to Load Fluctuations

Inherent high thermal storage characteristics can easily absorb fluctuation in fuel feed rates. Response to changing load is comparable to that of oil fired boilers.

12. No Slagging in the Furnace-No Soot Blowing

In FBC boilers, volatilization of alkali components in ash does not take place and the ash is non-sticky. This means that there is no slagging or soot blowing.

13. Provisions of Automatic Coal and Ash Handling System

Automatic systems for coal and ash handling can be incorporated, making the plant easy to operate comparable to oil or gas fired installation.

14. Provision of Automatic Ignition System

Control systems using micro-processors and automatic ignition equipment give excellent control with minimum manual supervision.

15. High Reliability

The absence of moving parts in the combustion zone results in a high degree of reliability and low maintenance costs.

16. Reduced Maintenance

Routine overhauls are infrequent and high efficiency is maintained for long periods.

17. Quick Responses to Changing Demand

A fluidized bed combustor can respond to changing heat demands more easily than stoker fired systems. This makes it very suitable for applications such as thermal fluid heaters, which require rapid responses.

18. High Efficiency of Power Generation

By operating the fluidized bed at elevated pressure, it can be used to generate hot pressurized gases to power a gas turbine. This can be combined with a conventional steam turbine to improve the efficiency of electricity generation and give a potential fuel savings of at least 4%.

6.7 Application Considerations with Biomass FBC Boilers

Biomass is a clean source of energy and is gaining huge popularity for FBC application in Industry. Used judiciously, entrepreneurs can reduce the cost of steam generation and at the same time reduce greenhouse gas emissions. With biomass fed boilers, the following areas need attention:

- Uneven spreading of biomass fuel on boiler grate can lead to secondary combustion in the super-heater zone, resulting in overheating of super heater tubes and fluctuations in steam pressure.
- Frequent erosion of super-heater and economizer coils can occur, due to high silica content in the biomass, especially in rice husk.
- High extraneous matter in biomass (sand and mud) causes tube fouling and fluidized bed to be drained more frequently, with resultant heat loss.
- Carbon and dust coating of boiler tubes results in lowering of steam temperatures, especially during soot blowing.
- Presence of Pesticides (used during farming) adds to tube failure frequencies; mainly due to potassium constituents.
- Corrosive constituents in biomass adversely affect boiler internals, especially the super-heater tubes. Chloride content in certain types of biomass (like cotton stalk, 8-9%) can combine with sodium and potassium in high temperature regime to aggravate the corrosion process.
- Some boilers which use Red Gram husk/twigs as fuel pose corrosion problems at the cold end (i.e., secondary super-heater and economizer tubes), due to the sulphur content.
- The biomass fuel mix fed to the boiler, in quite a few cases, contains a combination of 6 to 7 biomass types. Each biomass has a separate air-to fuel ratio, and it is difficult to set a workable air—fuel ratio.
- High moisture content in the biomass causes frequent jamming of the fuel in feeders, leading to fluctuations in steam pressure and temperature.
- High moisture content in the biomass also leads to plugging and choking of closely spaced heating surfaces. This situation is further aggravated by the super-heater tube coil with very close spacing, often the result of a desire to achieve a compact design.

- Due to biomass fuel size variation, occurrence of unburnts in flue gases and bottom ash is high, resulting in lower efficiency and also variation in steam pressure and temperature.
- Absence of biomass feed rate measurement mechanism leaves little room for accurate assessment of heat rate/efficiency. Providing a weighing mechanism is difficult on account of different biomass fuel combinations being used, with different (and low) bulk densities.
- Degradation of biomass during storage in exposed ambient wet atmosphere leads to loss of heat value. Loss of material due to windage and carpet loss, coupled with loss of heat value on account of decay (inherent biomass characteristics), can cause an error in assessment of input fuel energy (as the input heat is customarily evaluated based on received biomass quantities and GCV).

Chapter 7: Cogeneration

7.1 Introduction– Definition and Need

In conventional power plant when steam or gas expands through a turbine, nearly 60 to 70% of the input energy escapes with the exhaust steam or gas yielding only 30-40% efficiency. Also further losses of around 10-15% are associated with the transmission and distribution of electricity in the electrical grid. These losses are greatest when electricity is delivered to the smallest consumers.

If this energy in the exhaust steam or gas is utilized for meeting the process heat requirements, the efficiency of utilization of the fuel will increase and corresponding GHG emissions will reduce. Such an application, where the electrical power and process heat requirements are met from the fuel, is termed as “Cogeneration” or combined heat and power (CHP). The concept of CHP is illustrated in figure 7.1. Since, most of the industries need both heat and electrical energy, cogeneration can be a sensible investment for industries.

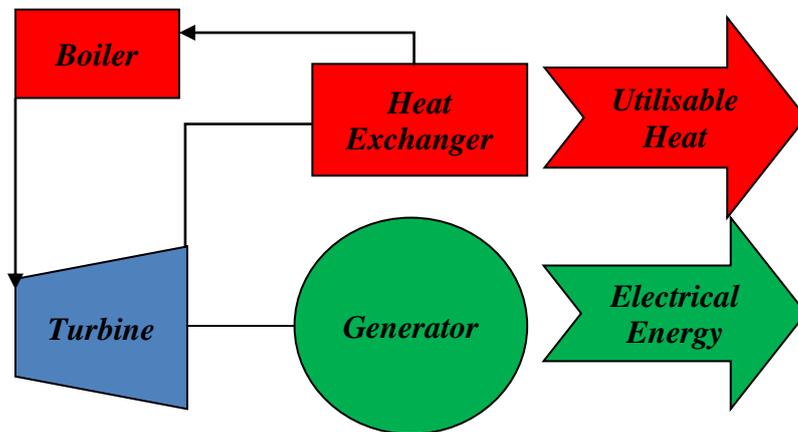
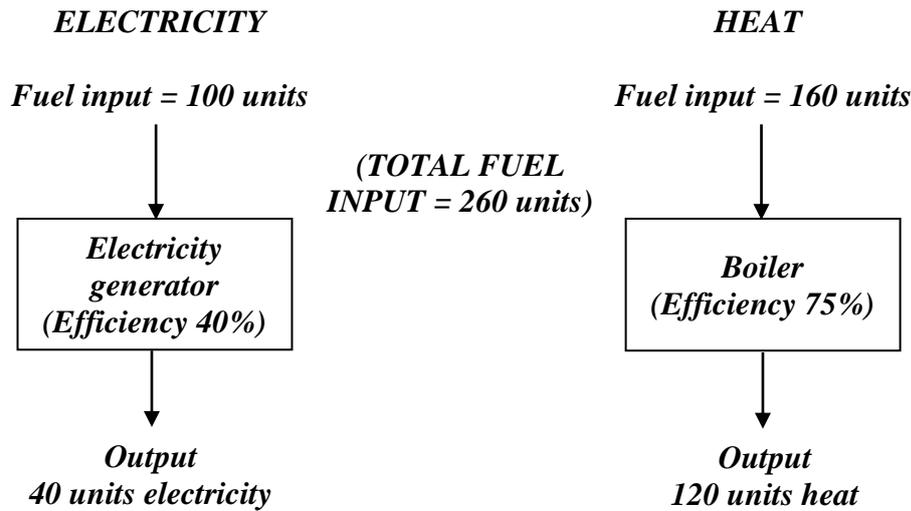


Figure 7.1: Cogeneration System

In cogeneration system efficiencies can go upto 90% and above providing energy savings ranging between 15-40% when compared against the supply of electricity and heat from conventional power stations and boilers. Since, electricity generated by cogeneration plant is normally used locally the transmission and distribution losses are negligible.

As an illustrative case, a plant needs 40 units of electric power and 120 units of thermal energy for its operation. Initially the plant met its requirement by having separate source for the electric power and thermal energy. In this process the total input required is 260 units. After installing a cogeneration system to meet both the loads, the plant is able to increase the overall efficiency of the system and bring down the input from 260 units to 200 units. Figure 7.2 gives comparison between Separate Heat and Power and Cogeneration System.

Case A: Separate Heat and Power



Case B: Combined Heat and Power

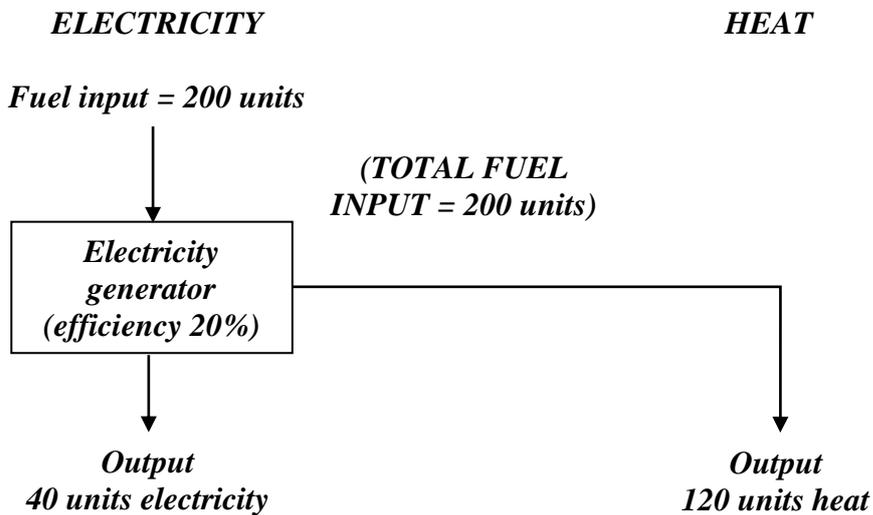


Figure 7.2: Comparison between energy inputs without and with cogeneration

In a cogeneration system, if less electricity is generated than needed, it will be necessary to buy extra. However, when the scheme is sized according to the heat demand, normally, more electricity than needed is generated. The surplus electricity can be sold to the grid. It can also be supplied to another customer via the distribution system, which is called the wheeling of power. As a rough guide, Cogeneration is likely to be suitable if there is constant heat demand for at least 4,500 hours per year.

7.2 Classification of Cogeneration Systems

There are two main types of cogeneration concepts: “Topping Cycle” plants, and “Bottoming Cycle” plants. A topping cycle plant generates electricity or mechanical power first whereas a bottoming cycle plant generates heat first.

7.2.1 Topping Cycle

There are four types of topping cycle cogeneration systems. They are:

Topping cycle cogeneration systems are of four types:

- 1) Fuel can be burned directly in either a gas turbine or a diesel engine to produce electrical or mechanical power and the exhaust is used to provide process heat or process steam. This is called a combined- cycle topping system. The process is shown in figure 7.3.

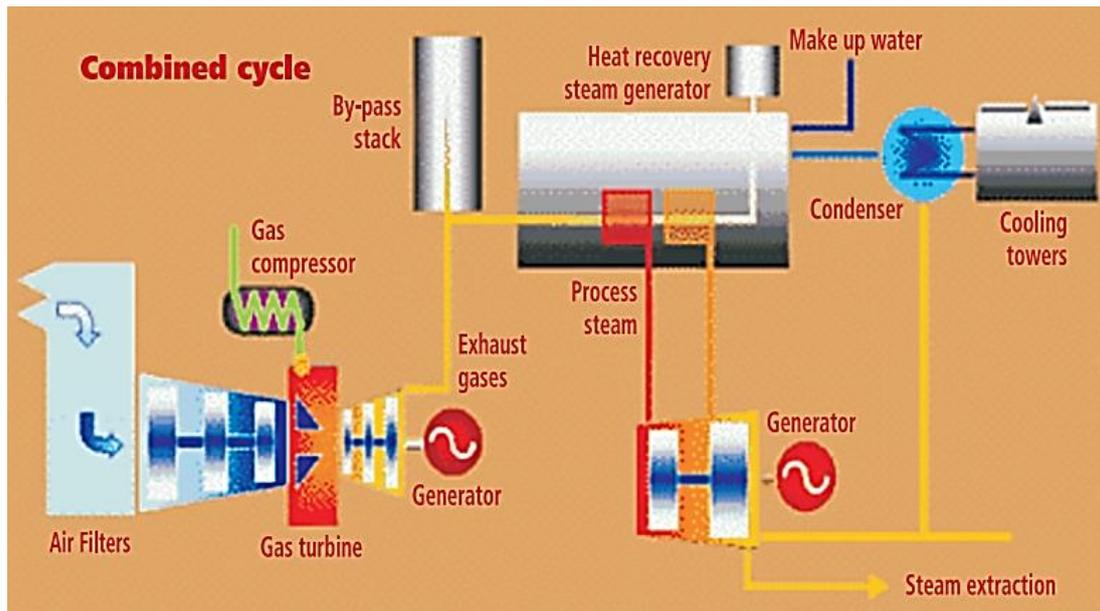


Figure 7.3: Steam turbine topping systems

- 2) Fuel can be burned initially to produce high-pressure steam which is then passed through a steam turbine to produce power, and the exhaust is used as process steam. A gas turbine or diesel engine producing electrical or mechanical power followed by a heat recovery boiler to create steam to drive a secondary steam turbine. This is a steam-turbine topping system.

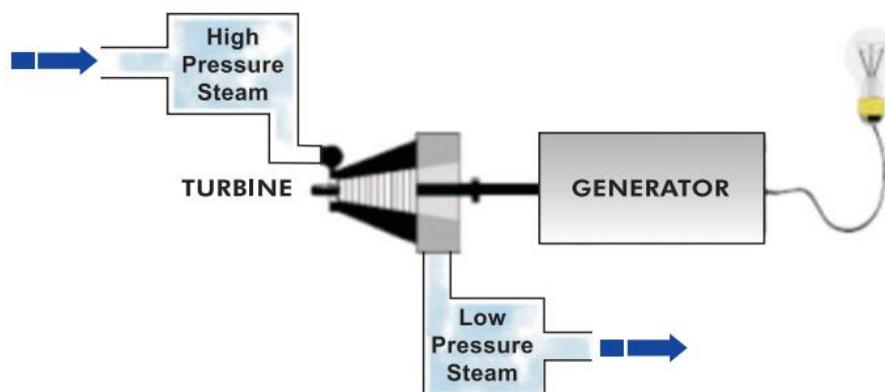


Figure 7.4: Steam turbine topping systems

- 3) A third type employs hot water from an engine jacket cooling system flowing to a heat recovery boiler, where it is converted to process steam and hot water for space heating.

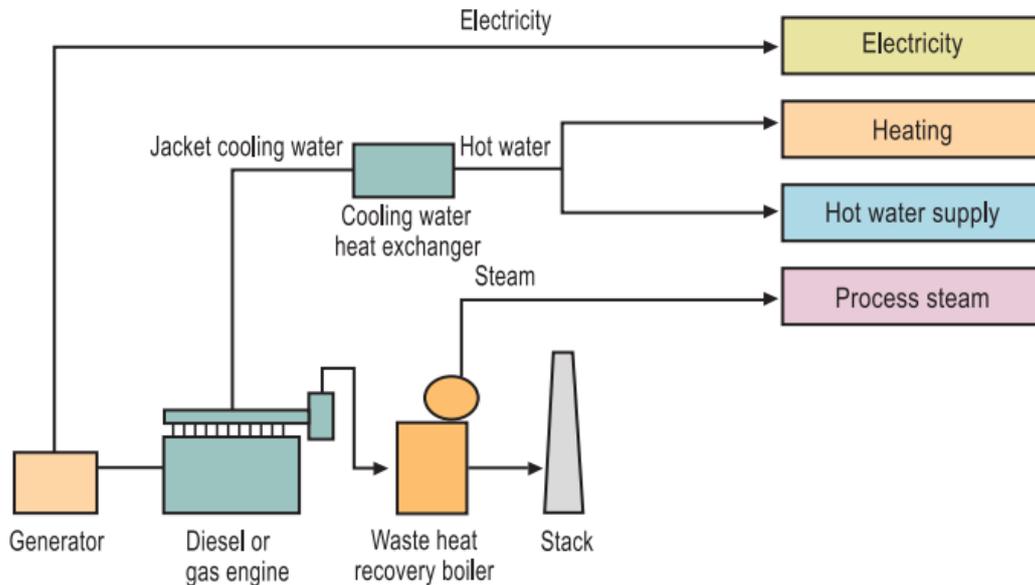


Figure 7.5: Diesel or gas engine topping systems

- 4) The fourth type is a gas-turbine topping system. A natural gas turbine drives a generator. The exhaust gas goes to a heat recovery boiler that makes process steam and process heat.

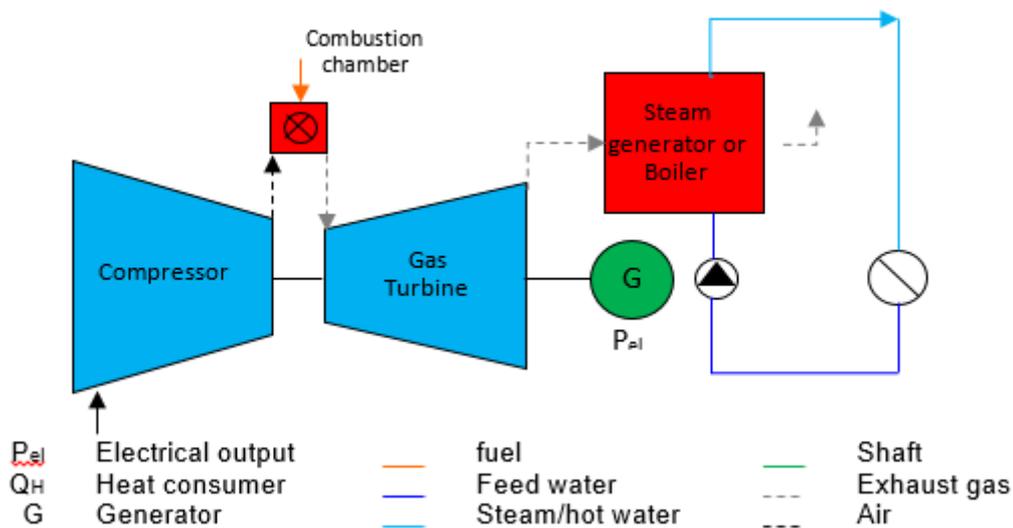


Figure 7.6: Gas turbine topping systems

7.2.2 Bottoming Cycle:

Bottoming cycle plants are much less common than topping cycle plants. These plants exist in heavy industries such as glass or metals manufacturing where very high temperature furnaces are used. The Figure 7.7 illustrates the bottoming cycle where fuel is burnt in a furnace to produce synthetic rutile. The waste gases coming out of the furnace is utilized in a boiler to generate steam, which drives the turbine to produce electricity.

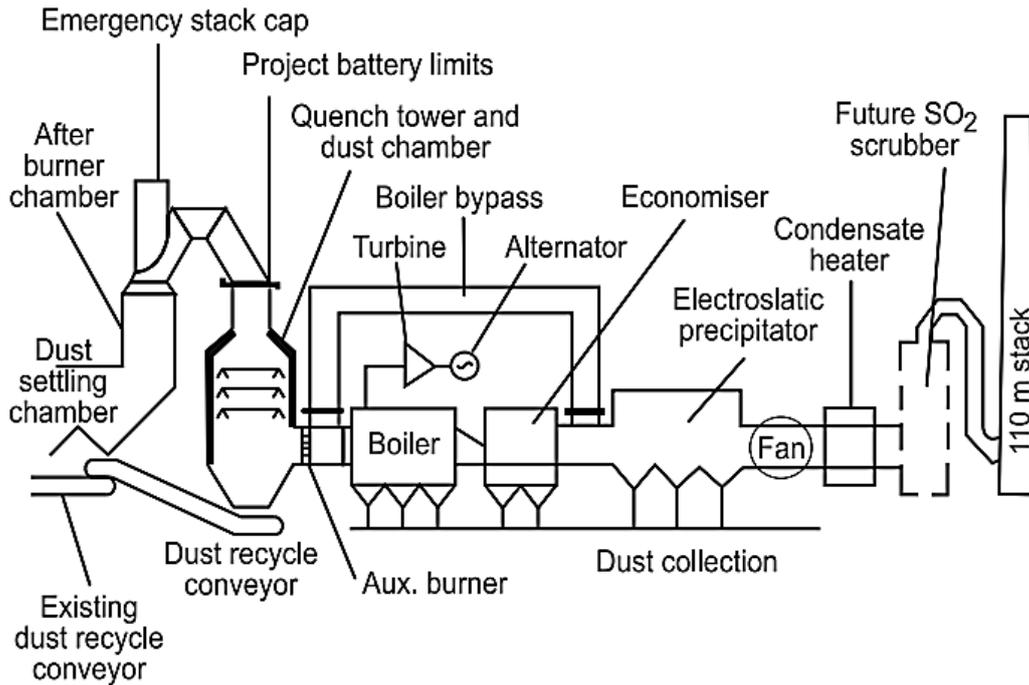


Figure 7.7: Bottoming cycle

7.3 Types of Cogeneration Systems

- Steam turbine
- Gas turbine
- Diesel engine

7.3.1 Steam Turbine

Steam turbines are the most commonly employed prime movers for cogeneration applications. In the steam turbine, high pressure steam generated in a boiler or heat recovery steam generator (HRSG) is expanded to a lower pressure level, converting the thermal energy of high pressure steam to kinetic energy through nozzles and then to mechanical power through rotating blades. Boiler fuels can include fossil fuels such as coal, oil and natural gas or renewable fuels like wood or municipal waste.

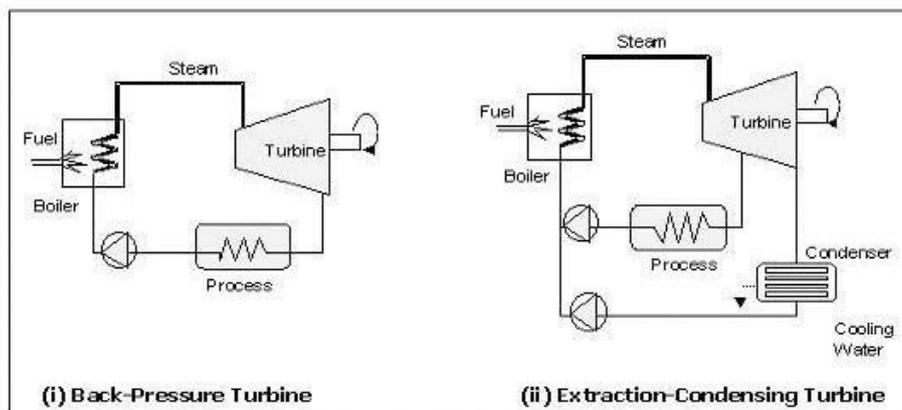


Figure 7.8: Steam turbine cogeneration system

The thermodynamic cycle for the steam turbine is the Rankine cycle, although a number of different cycles are also used, such as the Reheat, the Regenerative and the combined cycle. The Rankine cycle is the basis for conventional power generating stations and consists of a heat source (boiler) that converts water to high-pressure steam. The steam flows through the turbine to produce power and may be wet, dry saturated or superheated. Depending on the pressure (or temperature) levels at which process steam is required, back-pressure steam turbines can have different configurations. In extraction and double extraction back-pressure turbines, some amount of steam is extracted from the turbine after being expanded to a certain pressure level. The extracted steam meets the heat demands at pressure levels higher than the exhaust pressure of the steam turbine.

The efficiency of a back-pressure steam turbine cogeneration system is the highest. In cases where 100 per cent back-pressure exhaust steam is used, the only inefficiencies are gear drive and electric generator losses, and the inefficiency of steam generation. Therefore, with an efficient boiler, the overall thermal efficiency of the system could reach as much as 90 percent.

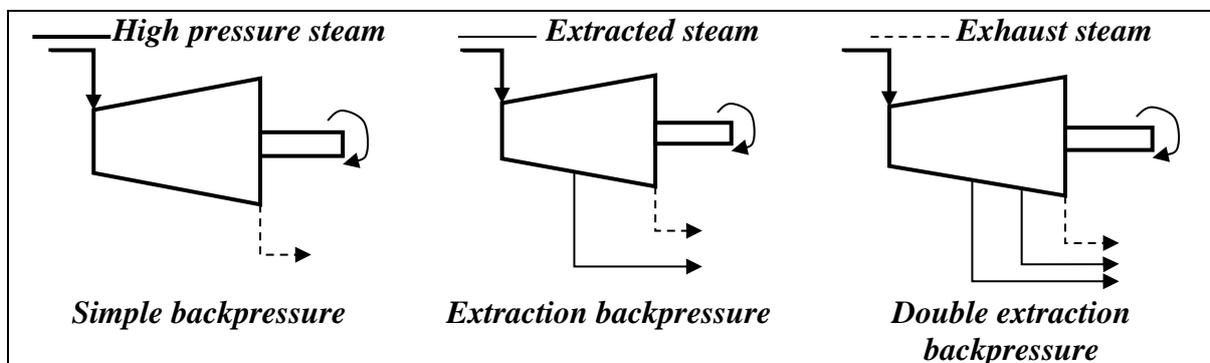


Figure 7.9: Configurations for back pressure steam turbines

Back-Pressure turbine: In this type steam enters the turbine chamber at **High Pressure** and expands to **Low or Medium Pressure**. Enthalpy difference is used for generating power /work.

Total Condensing turbine: In this type, steam entering at High / Medium Pressure condenses in a surface condenser and work is done till it reaches the Condensing pressure (vacuum).

Extraction cum Condensing steam turbine: In this high Pressure steam enters the turbine and passes out from the turbine chamber in stages. The Figure 7.9 shows a two stage extraction cum condensing turbine. In this MP steam and LP steam pass out to meet the process needs. Balance quantity condenses in the surface condenser. The Energy difference is used for generating Power. This configuration meets the heat-power requirement of the process.

The extraction condensing turbines: These turbines have higher power to heat ratio in comparison with back-pressure turbines. Although condensing systems need more auxiliary equipment such as the condenser and cooling towers, better matching of electrical power and heat demand can be obtained where electricity demand is much higher than the steam demand and the load patterns are highly fluctuating.

The overall thermal efficiency of an extraction condensing turbine cogeneration system is lower than that of back pressure turbine system, basically because the exhaust heat cannot be utilized (it is normally lost in the cooling water circuit). However, extraction condensing cogeneration systems have higher electricity generation efficiencies.

7.3.2 Gas Turbine

In gas turbines fuel is burnt in a pressurized combustion chamber using combustion air supplied by a compressor. These hot gases expand through the blades on the turbine rotor causing them to move generating mechanical energy.

In conventional Gas turbine, gases enter the turbine at 900 to 1000°C and leave at 400 to 500 °C. Residual energy in the form of hot exhaust gases can be used to generate wholly or partly, the thermal (steam) demand of the site.

The available mechanical energy can be applied in the following ways:

- to produce electricity with a generator (most applications);
- to drive pumps, compressors, blowers, etc.

A gas turbine operates under exacting conditions of high speed and high temperature. The hot gases supplied to it must be free of particulates which would erode the blades and contain no more than minimal amounts of contaminants, which would cause corrosion under operating conditions. High-premium fuels are therefore most often used, particularly natural gas. Distillate oils such as gas oil LPGs and Naphtha are suitable.

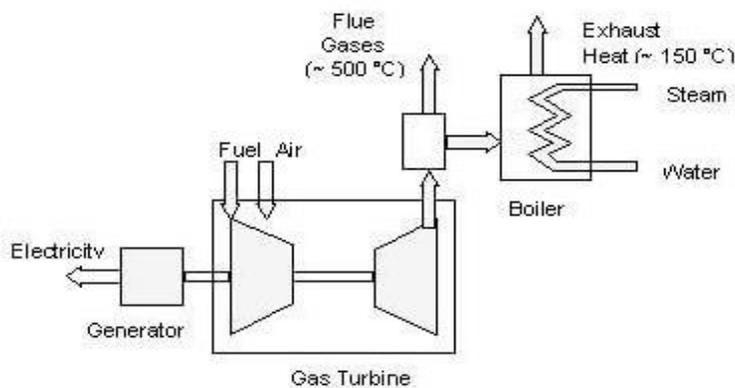


Figure 7.10: Gas turbine

7.3.2.1 Gas Turbine Efficiency

Turbine Efficiency is the ratio of actual work output of the turbine to the net input energy supplied in the form of fuel. For stand-alone Gas Turbines, without any heat recovery system the efficiency will be as low as 35 to 40%. This is attributed to the blade efficiency of the rotor, leakage through clearance spaces, friction, irreversible turbulence etc.

7.3.2.2 Increasing Overall Efficiency

Since Exhaust gas from the Gas Turbine is high, it is possible to recover energy from the hot gas by a Heat Recovery Steam Generator and use the steam for process.

7.3.2.3 Net Turbine Efficiency

Above efficiency figures did not include the energy consumed by air compressors, fuel pump and other auxiliaries.

Air compressor alone consumes about 50 to 60 % of energy generated by the turbine. Hence net turbine efficiency, which is the actual energy output available will be less than what has been calculated. In most Gas Turbine plants, air compressor is an integral part of Turbine plant.

7.3.3 Diesel Engine Systems

This system provides process heat or steam from engine exhaust. The engine jacket cooling water heat exchanger and lube oil cooler may also be used to provide hot water or hot air. There are, however, limited applications for this. As these engines can use only fuels like HSD, distillate, residual oils, natural gas, LPG etc. and as they are not economically better than steam/gas turbine, their use is not widespread for co-generation. One more reason for this is the engine maintenance requirement.

7.4 Methods for Calculating CHP System Efficiency

Two efficiency metrics namely Total system efficiency and Effective electric efficiency are used to compare CHP systems with SHP systems:

Total system efficiency refers to the sum of the useful power output (in MWh expressed in Btu/hr) and useful thermal outputs (in Btu/hr) divided by the total fuel input (in Btu/hr) and is the more commonly cited efficiency metric.

Effective electric efficiency refers to the electricity output divided by the additional fuel the CHP system uses over and above what would have been used by a conventional system to meet the facility's thermal energy load.

Both efficiency metrics consider all the outputs of CHP systems and reflect the benefits of CHP. Since each metric measures a different performance characteristic, the purpose and calculated value of each type of efficiency metric differs. For example, the total system efficiency is typically most appropriate for comparing CHP system energy efficiency with the efficiency of a site's SHP options. The effective electric efficiency is typically used to compare the CHP system with conventional electricity production (i.e., the grid).

In general, a CHP system's total system efficiency differs from its effective electric efficiency by 5% to 15%.

7.5 Typical Cogeneration Performance Parameters

The following Table 7.1 gives typical Cogeneration Performance Parameters for different Cogeneration Packages giving heat rate, overall efficiencies etc.

Table 7.1: Typical Cogeneration Performance Parameters

Prime Mover in Cogen. Package	Nominal Range (Electrical)	Electrical Generation to Heat Rate (kCal / kWh)	Efficiencies, %		
			Electrical Conversion	Thermal Recovery	Overall Cogeneration
Smaller Reciprocating Engines	10 – 500 kW	2650 - 6300	20-32	50	74-82
Larger Reciprocating Engines	500 – 3000 kW	2400 - 3275	26-36	50	76-86
Diesel Engines	10-3000 kW	2770 - 3775	23-38	50	73-88
Smaller Gas Turbines	800-10000 kW	2770-3525	24-31	50	74-81
Larger Gas Turbines	10-20 MW	2770-3275	26-31	50	78-81
Steam Turbines	10-100 MW	2520-5040	17-34	-	-

Note: Adapted from Cogeneration Handbook California Energy Commission, 1982

7.6 Heat: Power Ratio

Cogeneration is likely to be most attractive when the demand for both steam and power is balanced i.e. consistent with the range of steam (Heat) to power output ratios that can be obtained from a suitable cogeneration plant.

Heat-to-power ratio is defined as the ratio of thermal energy to electricity required by the energy consuming facility and expressed in different units such as Btu/kWh, kcal/kWh, kW/kW, etc. It is one of the most important technical parameters influencing the selection of the type of cogeneration system. The proportions of heat and power needed (heat to power ratio) vary from site to site, so the type of plant selected should match demands as closely as possible. The plant may therefore be set up to supply part or all of the site heat and electricity loads, or an excess of either may be exported if a suitable customer is available.

Heat-to-power ratios of different cogeneration systems and for certain energy intensive industries are shown in Table 7.2 & 7.3.

Energy Utilisation Factor

Energy Utilisation Factor (EUF) is the percentage of useful output i.e. a combination of electrical output and thermal output to fuel heat input. The formula for EUF is given below.

$$\text{Energy Utilisation Factor, EUF} = \frac{\text{Electrical output, } Q_e + \text{Thermal output, } Q_{th}}{\text{Fuel heat input, } Q_f}$$

The electrical output is the useful power generation from the turbine and the thermal output is the useful extraction from turbine for process heating. The fuel heat input is the fuel flow rate multiplied by its calorific value.

Table 7.2: Heat-to-power ratios & key parameters of cogeneration systems

Cogeneration System	Heat-to- Power ratio (KW/KW)	Power Output (as % of fuel input)	Overall efficiency %
Back-pressure steam turbine	4.0-14.3	14-28	84-92
Extraction-condensing steam turbine	2.0-10	22-40	60-80
Gas turbine	1.3-2.0	24-35	70-85
Combined Cycle	1.0-1.7	34-40	69-83
Reciprocating Engine	1.1-2.5	33-53	75-85

Table 7.3: Typical Heat: Power ratios for certain energy intensive industries

Industry	Minimum	Maximum	Average
Breweries	1.1	4.5	3.1
Pharmaceuticals	1.5	2.5	2.0
Fertilizer	0.8	3.0	2.0
Food	0.8	2.5	1.2
Paper	1.5	2.5	1.9

7.7 Factors for selection of cogeneration system

Following factors should be given due consideration in selecting the most appropriate cogeneration system for a particular industry.

- Normal as well as maximum/minimum power load and steam load in the plant, and duration for which the process can tolerate without these utilities, i.e. criticality and essentiality of inputs.
- What is more critical - whether power or steam, to decide about emergency back-up availability of power or steam.
- Anticipated fluctuations in power and steam load and pattern of fluctuation, sudden rise and fall in demand with their time duration and response time required to meet the same.
- Under normal process conditions, the step by step rate of increase in drawl of power and steam as the process picks up - whether the rise in demand of one utility is rapid than the other, same or vice-versa.
- Type of fuel available - whether clean fuel like natural gas, naphtha or high speed diesel or high ash bearing fuels like furnace oil, LSHS, etc or worst fuels like coal, lignite, etc., long term availability of fuels and fuel pricing.
- Commercial availability of various system alternatives, life span of various systems and corresponding outlay for maintenance.
- In general, simultaneous demands for heat and power must be present for at least 4,500 h/year, although there are applications where CHP systems may be cost effective with fewer hours. For example, when electricity rates are high or when the local power provider offers incentives, this operating period could be as low as 2,200 h/year.

- Power-to-heat ratio for the plant should not fluctuate more than 10%.
- Influence exerted by local conditions at plant site, i.e. space available, soil conditions, raw water availability, infrastructure and environment.
- Project completion time.
- Project cost and long term benefits.

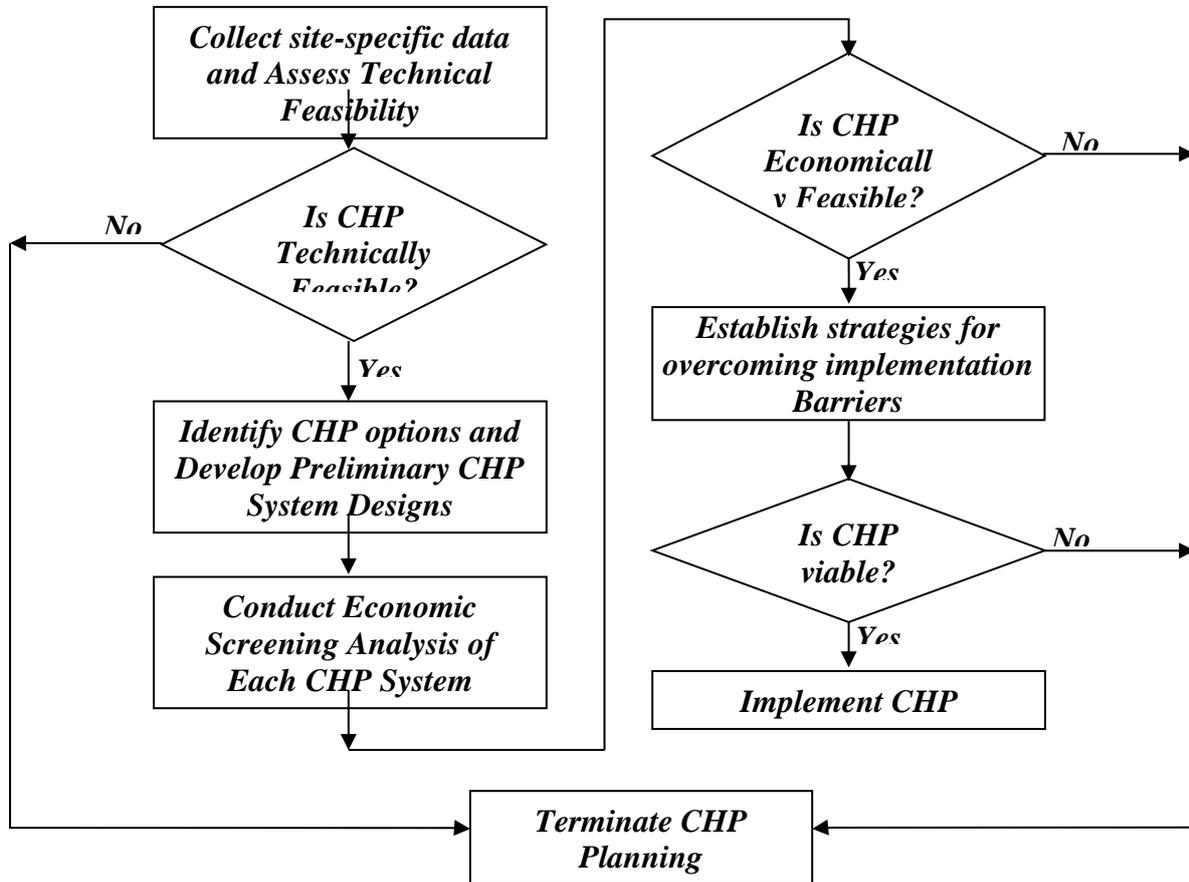


Figure 7.11: Framework for Evaluating CHP viability

7.8 Operating Strategies for Cogeneration Plant

For cogeneration plant there are three main operating regimes:

- The unit is operated to provide base load electrical and thermal output. Any shortfall is supplemented with electricity from external public/private utility and heat from stand-by boilers or boost (thermic fluid) heaters;
- The unit is operated to provide electricity in excess of the site’s requirements, for export, whilst all the thermal output is used on site;
- The unit is operated to provide electricity for site, with or without export, and the heat produced is used on site with the surplus being exported to off-site customers.

Table 7.4: Characteristics of Prime Movers for CHP Applications ^a

Prime Mover Characteristic	Steam Turbine	Gas turbine	Micro turbine	Reciprocating Engine	
				Compression Ignition	Spark Ignition
Capacity, MW	0.05 to > 250	0.5 to 250	0.03-0.35	0.03 to 4	0.05 to 5
Power-to-heat ratio	0.05 to 0.2	0.5 to 2	0.4 to 0.7	0.5 to 1	0.5 to 1
Fuels	All types of fuel can be burned to produce steam	Natural gas, bio gas, propane and distillate fuel oil	Natural gas, Waste and sour gases, gasoline, kerosene, diesel and distillate fuel oil	Natural gas, diesel and residual oil	Natural gas, bio gas, propane, land fill gas and gasoline
Installed cost, \$/KW	200 -1000	400 -1800	1300 -2500	900 - 1500	901 - 1500
Maintenance cost, \$/KWh	≤ 0.002	0.003 - 0.01	≤ 0.018	0.005 - 0.015	0.007 - 0.02
Overhaul period, h	> 50000	12000 - 50000	5000 - 40000	25000 - 30000	24000 - 60000
Start-up time	Hours	Minutes	Minutes	Seconds	Seconds
Total CHP efficiency (HHV) ^b	70 - 85 %	70 - 75 %	65 - 75 %	70 - 80 %	70 - 80 %
CHP electrical efficiency (HHV) ^c	20 - 40 %	22 - 36 %	18 - 29 %	27 - 45 %	22 - 40 %
Availability	Nearly 100%	90 - 98 %	90 - 98 %	90 - 95 %	92 - 97 %
Noise	High	High	High	Moderate	High
Service life	30 years or more	30000 - 100000 hrs	40000 - 80000 hrs	15 - 25 years	15 - 25 years
Part-load operation	Good	Poor	Satisfactory	Good	Satisfactory
NOx control options	Unnecessary but may be required as part of steam supply system.	Steam or water injection, lean premixed combustion, SCR,	Lean premixed combustion, SCR, SNCR and SCONOXTM	Lean air-fuel mixture, SCR, SNCR and SCONOXTM	Lean air-fuel mixture, staged ignition, catalytic 3-way conversion(TWC),

Prime Mover Characteristic	Steam Turbine	Gas turbine	Micro turbine	Reciprocating Engine	
				Compression Ignition	Spark Ignition
		SNCR and SCONOXTM			SCR, SNCR and SCONOXTM
Preferred uses for recovered heat	Process heat, hot water and low-pressure to high pressure steam	Process heat, hot water and low-pressure to high pressure steam	Process heat, hot water and low-pressure steam	Hot water and low-pressure steam	Hot water and low-pressure steam
Temperature of rejected heat, °F	Varies depending on extraction conditions	500 -1100	400 - 600	180 - 900	180 - 1200
Operating mode	Load-tracking and continuous base-loaded operation	Base-loaded, load-tracking, and peak shaving operations	Base-loaded, load-tracking, and peak shaving operations	Base-loaded, load-tracking, emergency and peak shaving operations	Base-loaded, load-tracking, emergency and peak shaving operations
Potential applications	Topping-cycle, bottoming-cycle, combined-cycle, and trigeneration CHP systems	Topping-cycle, combined-cycle, and trigeneration CHP systems	Topping-cycle, combined-cycle, and trigeneration CHP systems	Topping-cycle, combined-cycle, and trigeneration CHP systems	Topping-cycle, combined-cycle, and trigeneration CHP systems
a Based on CHP systems that operate at least 8000 hrs/year					
b Total CHP efficiency is a measure of (the net electricity generated plus the net heat supplied to the process), divided by the total fuel input					
c CHP electrical efficiency is a function of the net electricity generated, divided by the total fuel input					

Table 7.5: Sources of Information for decision making

<i>Required information</i>	<i>Sources of Information</i>								
	<i>Energy Audit</i>	<i>Utility Company</i>	<i>Fuel Suppliers^a</i>	<i>Equipment Suppliers</i>	<i>Management</i>	<i>Legal authorities</i>	<i>Air -water quality authority</i>	<i>Special authorities^b</i>	<i>Financial Institutions</i>
<i>Total thermal loads, magnitude and profile</i>	X								
<i>Total electrical loads, magnitude and profile</i>	X								
<i>Cooling Loads</i>	X								
<i>Major load centers and energy consumers</i>	X								
<i>Anticipated load changers, Mission or Function changes</i>	X								
<i>Waste heat sources</i>	X								
<i>Waste fuel sources</i>	X								
<i>Complimentary off-site loads</i>	X								
<i>Present Electrical Energy Costs and Rate Formats</i>	X	X							
<i>Projected Electrical Energy Rate structure</i>		X							
<i>Policies towards parallel generation</i>		X							
<i>Ownership/Operation policies and preferences</i>		X			X				
<i>Cogenerator rate structure, Standby charges, Reliability Requirements, Payment for Power to Grid</i>		X							
<i>Environmental and Siting Constraint Overview</i>		X							
<i>Tax and Investment Incentives</i>		X				X			
<i>Regulations Related to Cogeneration</i>		X				X			
<i>Present Fuel Costs</i>	X		X						
<i>Projected fuel costs</i>		X	X						
<i>Projected fuel availability</i>		X	X						
<i>Fuel Characteristics and Properties</i>			X						
<i>Performance Data, Design and Off Design Conditions</i>				X					
<i>Fuel Consumption</i>				X					
<i>Fuel Flexibility</i>				X					
<i>Emissions Data and Specifications</i>				X					
<i>Air Emission Regulations</i>				X			X		
<i>Other Emission Regulations</i>				X				X	
<i>Fiscal Policies</i>					X				
<i>Funding Sources</i>					X				X
<i>Cost and Conditions of financing</i>									X
<i>Siting restrictions</i>						X	X	X	

a Oil, Natural gas or Coal

b For example, Zoning, airport, Coastal

(Source : Cogeneration Systems- Technical Report by E E Cooper 1980)

7.9 Relative Merits of Co-Generation Systems

The following Table 7.6 gives the advantages and disadvantages of various co-generation systems:

Table 7.6: Advantages and disadvantages of various cogeneration systems

Variant	Advantages	Disadvantages
Back-pressure	- High fuel efficiency rating	Little flexibility in design and operation
Steam turbine & fuel firing in boiler	<ul style="list-style-type: none"> • Simple plant • Well-suited to low quality fuels 	<ul style="list-style-type: none"> • More capital investment • Low fuel efficiency rating • High cooling water demand • More impact on environment • High civil const. cost due to complicated foundations
Gas turbine with waste heat recovery boiler	<ul style="list-style-type: none"> • Good fuel efficiency • Simple plant • Low civil const. Cost • Less delivery period • Less impact on environment • High flexibility in operation 	<ul style="list-style-type: none"> • Moderate part load efficiency • Limited suitability for low quality fuels
Combined gas & steam turbine with waste heat recovery boiler	<ul style="list-style-type: none"> • Optimum fuel efficiency rating • Low relative capital cost • Less gestation period • Quick start up & stoppage • Less impact on environment • High flexibility in operation 	<ul style="list-style-type: none"> • Average to moderate part-load efficiency • Limited suitability for low quality fuels
Diesel Engine & waste heat recovery Boiler & cooling water heat exchanger	<ul style="list-style-type: none"> • Low civil const. Cost due to block foundations & least no. of auxiliaries • High Power efficiency • Better suitability as standby power source 	<ul style="list-style-type: none"> • Low overall efficiency • Limited suitability for low quality fuels • Availability of low temperature steam • Highly maintenance prone.

7.10 Case Study

Economics of a Gas Turbine based Cogeneration System

Alternative I: Gas Turbine based Cogeneration.		
Gas Turbine Parameters	Units	Quantity
Capacity of gas turbine generator	KW	4000
Plant operating hours per annum	hrs/ year	8000
Plant load factor (PLF)	%	90%
Heat rate as per standard given by gas turbine suppliers	KCal/KWH	3049.77
Waste heat boiler parameters- unfired steam output	TPH	10
Steam temperature	°C	200
Steam pressure	Kg/ Sq.cm	8.5
Steam enthalpy	Kcal/Kg	676.44
Fuel used		Natural Gas
Calorific Value - LCV	Kcal/ S Cu.m	9500
Price of gas	BDT/1000 S Cu.m	3000
Capital investment for Cogeneration plant	BDT	1300000
Cost Estimation of Power & Steam from Cogeneration Plant		
Power generated = PLF x Plant Capacity x Operating hrs	KWH/ Year	28800000
Heat input = Power generated x Heat rate given by turbine supplier	Kcal	87833376000
Natural gas(NG) required per annum = Heat input/ LCV of NG	S cu.m	9245618.526
Annual cost of fuel = Gas consumed x price	BDT	27736855.58
Annualised Cost of capital and operation charges	BDT	29863000
Overall cost of power from cogeneration plant (Alternative -I Cost)	BDT	57599855.58
Cost of power	BDT/KWH	1.999994985

Alternative-II Electric Power from State Grid & Steam from Natural Gas (NG) Fired Boiler		
Boiler installed in plant	TPH	10
Cost of electric power from grid	BDT/KWH	3
Capital investment for 10TPH, 8.5 kg/cm² @ 200°C NG fired fire tube boiler & all auxiliaries	BDT	8000000
Hours of operation	hrs/ year	8000
Cost Estimation of Power & Steam from Grid and Steam from direct Conventional fired boiler		
Cost of power from state grid for 28800000 KWH	BDT/ year	86400000
Fuel cost for steam by separate boiler		
Heat output in form of 10TPH steam per annum = Boiler capacity x steam Enthalpy x Hrs of operation	Kcal/ year	54115200000
Heat input required to generate 10TPH steam per annum @ 90% efficiency	Kcal/ year	60128000000
Natural gas(NG) required per annum = Heat input/ LCV of NG	S cu.m/ year	6329263.158
Annual cost of fuel = Gas consumed x price	BDT	18987789.47
Total cost for alternative II= Cost of grid power + fuel cost for steam	BDT	105387789.5
Alternative -I total cost	BDT	57599855.58
Alternative -II total cost	BDT	105387789.5
Differential cost	BDT	47787933.89

(Note: In case of alternative II, there will be some additional impact on cost of steam due to capital cost required for a separate boiler)

In the above case it can be seen that Alternative I is economical compared to Alternative II.

7.11 Trigeneration

Trigeneration refers to simultaneous generation of steam (heat), power and refrigeration through integrated systems. Industries requiring electricity, steam and cooling such as food processing and cold storages find the concepts of tri-generation very attractive.

A combined cycle trigeneration plant could typically consist of a gas turbine generator, waste heat recovery boiler, steam turbine, generator, and absorption chiller, to meet 100% of the facility energy needs. Whenever the power is surplus, it is sold to the grid.

An illustrative trigeneration system schematic is presented in Figure 7.12.

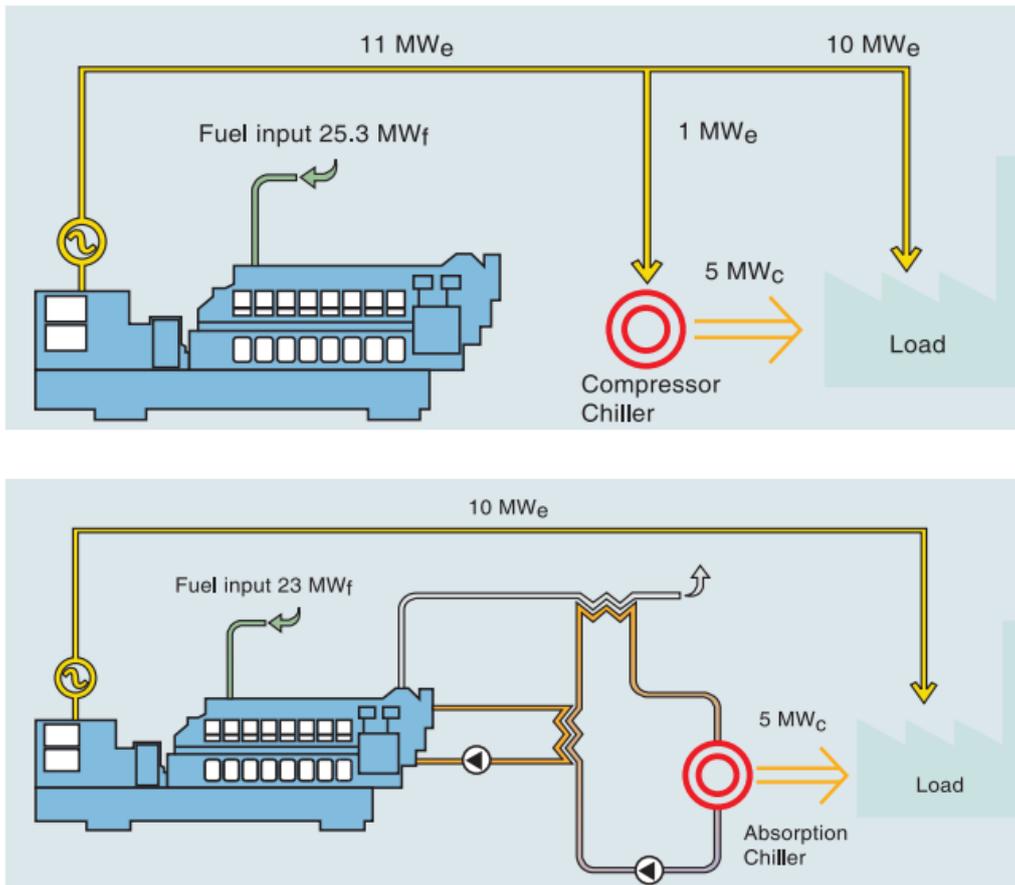


Figure 7.12: Trigeneration: Power, Heating and Cooling

7.13 Microturbine

The Micro steam Turbine Power System is a compact, efficient power system that generates electricity from pressure energy previously wasted in the steam pressure reducing valves.

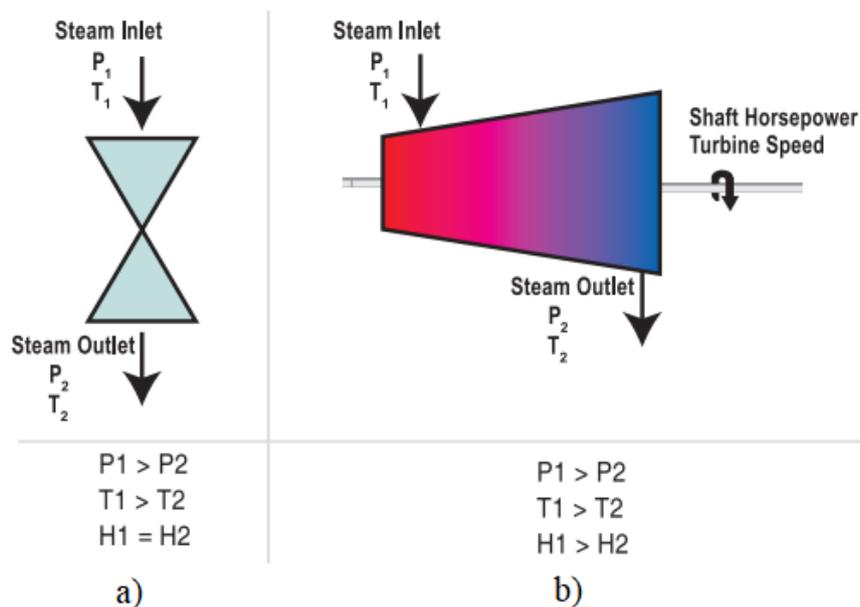


Figure 7.13 a) PRV, b) Micro turbine

In a typical steam system, a Pressure Reducing Valve (PRV) reduces the steam pressure from P_1 to P_2 (Figure 7.13a). This pressure reduction happens in such a manner that the total energy content (enthalpy) does not change ($H_1 = H_2$) and no shaft work is done.

On the other hand, when steam goes through a steam turbine, it expands and the steam pressure reduces from P_1 to P_2 (Figure 7.13b). The steam turbine produces shaft horsepower and as a result, the steam exit energy content (enthalpy) is lower when compared to the PRV case.

Steam is used for heating purposes in the plant. The process heat duty is fixed by the plant demand. Since the steam supplied to the process has a lower enthalpy, an additional amount of steam is required to ensure the same available heat duty. This additional amount of steam has an associated cost. Hence, power generated from a backpressure steam turbine is not free.

Nevertheless, using a backpressure steam turbine can improve the overall plant and global energy efficiency and more importantly, it can reduce total operating costs.

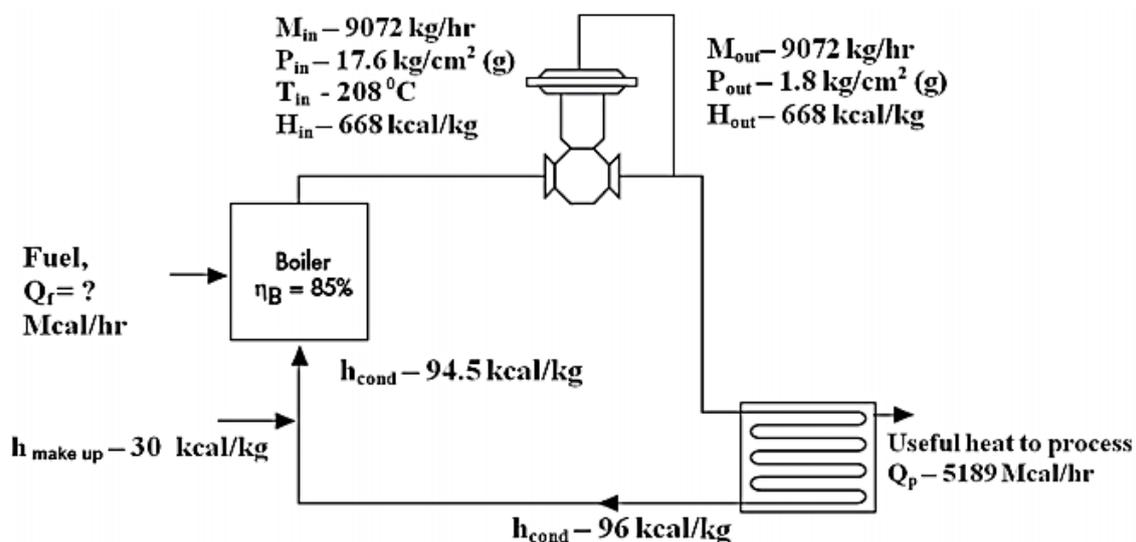
Example 7.1 Pressure reducing valve versus back pressure turbine

A chemical plant generates steam at 17.6 kg/cm^2 (g) which is further reduced to 1.8 kg/cm^2 (g) through a pressure reducing valve, for utilization in the process. The process requires 5189 MCal/hr of heat. The plant has planned to replace the PRV with a back pressure steam turbine.

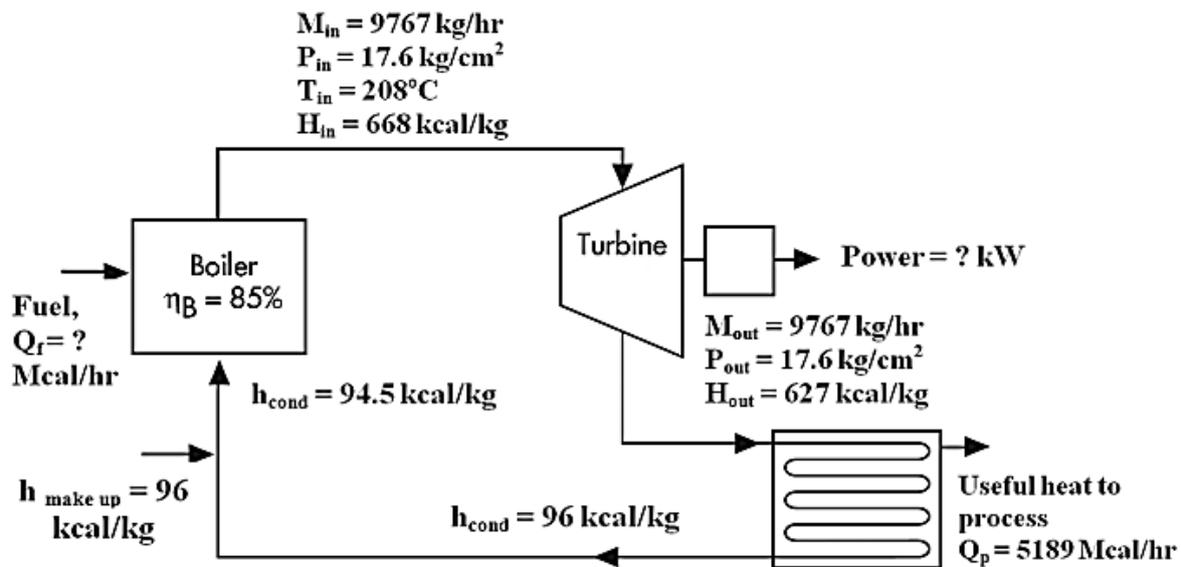
Calculate the net savings per annum if the plant operates 8000 hrs/annum with the following data and the data provided in the schematic diagrams.

Calorific value of fuel	- 10,000 kcal/kg,
Fuel cost	- BDT 25,000/Ton,
Grid cost of electricity charge	- BDT 6/kWh.
Gear box efficiency	- 98%
Alternator efficiency	- 98%
Mechanical efficiency of turbine	- 97%

a) Boiler and PRV operation



b) Boiler and Micro turbine operation:



The schematic of a back pressure steam turbine cogeneration system of process plant operating round the clock with operating data is depicted below.

If the steam requirement of the process is to be increased to 44TPH which can be met by the existing boiler through the back pressure turbine,

a) Find out the reduction in cost of electrical energy drawn from the grid per day due to additional power generation, assuming the same steam to power recovery as in the existing case and at a grid electricity cost of BDT 7/kWh, Aux power remains the same.

b) Also find out the additional coal requirement per day?

Solution

- a) Present steam to power recovery ratio = 40,000 kg/hr / 3,200 kWh
 = 12.5 kg/kWh
 Additional steam generation = 4TPH
 Additional power generation potential = 4000 kg/hr / 12.5 kg/kWh
 = 320 kW
 Daily saving due to additional power generation = 320 kW x 24 hr x BDT 7
 = BDT 53,760
 Additional coal requirement per hour = 4000 x (780-135) / (0.8 x 4300)
 = 750 kg/hr
 b) Additional coal requirement per day = 750 kg/hr x 24
 = 18,000 kg/day
 = 18 Ton/day

Chapter 8: Waste Heat Recovery

8.1 Introduction

Industrial waste heat refers to the heat energy that is generated but not fully utilized in the industrial processes and leaves the boundaries of a plant or building into the surrounding environment. Waste heat is generally associated with waste streams of air, exhaust gases, water, or other liquids. Waste heat losses arise both from equipment inefficiencies and due to equipment and process requirements. These losses can be reduced by improving equipment efficiency and installing waste heat recovery technologies.

Waste heat recovery (WHR) equipment is defined as any mechanical apparatus which usefully recovers thermal energy from process waste streams that are above ambient temperatures. The recovered heat is used to produce heat, generate power and in some cases for cooling applications. The aim of this chapter is to provide an understanding of why, when and how to recover waste heat and will cover the following aspects of WHR.

- Opportunities and benefits of WHR
- factors effecting waste heat recovery
- applications of waste heat
- WHR methods and technologies
- Industry/sector specific applications of WHR technologies
- Formulae for calculating heat losses
- Case examples of WHR implemented

8.2 Opportunities and benefits of WHR

The best waste heat-recovery opportunities are those that have the following characteristics:

- The waste heat supply is constant.
- The need is co-located with the waste heat supply.
- The need is synchronized with the available waste heat.
- The waste heat supply is higher in temperature than the need or
- The need and the waste heat stream temperatures match the capabilities of available heat-pumping equipment.
- The size of the waste heat stream and the need are large enough to justify the custom engineering required.

Benefits of 'waste heat recovery' can be broadly classified in two categories:

8.2.1 Direct Benefits:

Recovery of waste heat has a direct effect on the efficiency of the process. This is reflected by reduction in the utility consumption and process cost.

8.2.2 Indirect Benefits:

a) Reduction in equipment sizes: Waste heat recovery reduces the fuel consumption, which leads to reduction in the flue gas produced. If WHR systems are considered and

incorporated at design stage it may result in reduction in equipment sizes of all flue gas handling equipments such as fans, stacks, ducts, burners, etc. In other modification and retrofit cases possibility exists to reduce the size of equipment.

b) Reduction in auxiliary energy consumption: Reduction in equipment sizes gives additional benefits in the form of reduction in auxiliary energy consumption like electricity for fans, pumps etc.

c) Reduction in pollution: A number of toxic combustible wastes such as carbon monoxide gas, sour gas, carbon black off gases, oil sludge, Acrylonitrile and other plastic chemicals etc., when combusted/burnt in the thermal oxidation or incinerators serves dual purpose of heat recovery and mitigation of the environmental pollution levels.

8.3 Factors affecting waste heat recovery

Before considering waste heat recovery it is important to consider why, when and how to recover this waste heat to gain benefits. The following factors influence the selection of a heat exchanger.

- Source of the waste heat stream.

In few cases it is difficult to access and recover heat from unconventional sources such as hot solid product streams (e.g., ingots) and hot equipment surfaces (e.g., sidewalls of primary aluminum cells).

- Amount of waste heat available

Once you find suitable waste heat source, it is important to establish that the source is capable of supplying sufficient ‘quantity’ of heat, and that the heat is of a good enough ‘quality’ (i.e. temperature) to promote good heat transfer.

- Characteristics of the waste heat stream

Essential considerations in making optional choice of waste heat recovery device:

- 1) Temperature of waste heat. (Temperature is a measure of quality of waste heat)
- 2) Flow rate of the fluid
- 3) Chemical composition of waste fluid
- 4) Properties of waste fluid (C_p , μ , ρ , κ)
- 5) Allowable pressure drop
- 6) Minimum temperature to which waste heat can be cooled
- 7) Corrosive elements in the exhaust fluid
- 8) Temperature to which the designed fluid is to be heated

The quality of the heat is based on the waste stream temperature and it is divided into three grades as mentioned in table 8.1 below. Higher the grade, the greater the potential value for heat recovery.

Table 8.1: Different grades of heat

High Grade	Medium Grade	Low Grade
1100°F – 3000°F (593°C – 1649°C)	400°F – 1100°F (204°C – 593°C)	80°F – 400°F (27°C – 204°C)

- Waste heat use.

It is important to have a use for any waste heat which may be recovered. In many applications there may be no demand for the heat that is available, with the result the excess heat energy is dumped into the environment. In other situations there may be a long time lag between waste heat production and the demand for heat. Waste heat therefore cannot be used unless there is some use of waste heat and/or some form of thermal storage is available.

Waste heat can be used in various ways, but the major uses include the following:

- 1) preheating combustion air
- 2) generating electrical and mechanical power
- 3) generating process steam
- 4) preheating boiler water
- 5) heating general process liquids and solids
- 6) heating viscous, corrosive, and difficult liquids
- 7) heating, ventilation and refrigeration applications.

The source of heat for applications 1 through 4 is usually hot gases, most frequently from combustion processes. The source of heat for applications 5 through 7 is usually process steam, process liquids/solids or exhaust air.

Table 8.2: Major Applications of Waste Heat Recovery

Application Heat Exchanger Type	Pre-heat combustion air	Generate power	Generate Process steam	Pre-heat boiler water	Heat process liquids	Corrosive and viscous fluids	Heating and ventilation
Tube in tube	✓				✓		
Shell in tube		✓	✓	✓	✓		
Direct Contact	✓						
Solid plate fin	✓						✓
Heat pipes							✓
Runaround coils					✓	✓	✓
Spiral				✓	✓	✓	
Coils					✓		✓
Plate and frame					✓	✓	✓
Finned tube							✓
Panel coil					✓		
Cartridge					✓	✓	
Screw conveyor					✓	✓	

- Adequate space availability to install heat recovery system

Many facilities have limited physical space to access waste heat streams and to install waste heat recovery systems (e.g., limited floor or overhead space). In such cases it may not be feasible to install the heat recovery system.

- Primary energy saving possibility

Often the insertion of a heat exchange system increases the resistance of the fluid streams, resulting in higher fan or pump energy consumption. Heat energy is therefore replaced by electrical energy with no net energy savings.

- Economic viability or cost effectiveness of WHR technology

Heat recovery devices can be expensive to install. It is therefore essential that the economic payback period be determined before any investment is undertaken. If proper planning and analysis is not carried out at the concept and design stage, impact of installation of a waste heat recovery device is minimal or may even increase energy cost.

WHR technologies are similar to heat exchangers. The heat exchangers considered unviable are now being considered for heat recovery for the following reasons

- Heat exchange equipment costs came down making them viable
- Fuel costs increased making an economic case for waste heat recovery
- Regulatory requirements might have made it mandatory to install the same.

8.4 Overview of Waste Heat Recovery Methods and Technologies

Once the need and feasibility to install WHR equipment is established the next logical step is to select the appropriate method for utilization of waste heat and the equipment required. The common methods by which waste heat is utilized are as follows.

- Direct utilization (eg: for drying or process heating)
- Energy Cascading (Using energy for high temperature application first followed by using the rejected heat for low temperature applications. For example after using high temperature and pressure steam for power generation low temperature and pressure steam can be used for other processes or space heating).
- Cogeneration (Producing electrical power and process heat simultaneously)
- Recuperators (Shell and tube, plate, coil, spiral heat exchangers)
- Regenerators (stationary or rotating type)
- Waste heat boilers

WHR can be classified based on the maximum outlet temperature of recovered waste heat. Accordingly based on waste heat stream temperatures WHR equipment is classified as follows.

- 1) Gas-to-gas heat exchanger (Graphite heat exchangers, stack-type recuperators, direct contact recuperator, plate fin (ceramic and metal) heat exchangers and ceramic tubes)
- 2) Gas-to-liquid heat exchanger (waste heat boilers, economizers and power generators)
- 3) Liquid-to-liquid heat exchanger (shell-and-tube, spiral, coil, finned-tube, plate-and-frame (plate), and run-around heat exchangers)
- 4) Other low-temperature WHR equipment (heat pumps, and heat pipes)

At higher temperature, only gas-to-gas heat transfer is used because of the difficulties encountered at these temperatures. At moderate temperatures (upto $\sim 1000^{\circ}\text{F}$), gas-to-liquid transfer is used (steam boilers). At lower temperatures, the dominant mode is liquid-to-liquid heat recovery.

A brief description of commonly encountered heat exchangers is given in sections below.

8.4.1 Gas-to-gas Heat Exchanger

8.4.1.1 Graphite heat exchangers

Graphite heat exchangers have high thermal conductivity and frequently used for heating or cooling of ultra-corrosive liquid chemicals. This specific design allows for heat recovery between two ultra-corrosive fluids.



Figure 8.1: Graphite heat exchangers

(Source : GAB Neumann's Annular Grove graphite heat exchangers)

8.4.1.2 Recuperators

This is the most common type of equipment used for waste heat recovery. In this heat transfer is affected by bringing hot and cold streams adjacently where in heat from hot fluid is transferred through the fluid separation barrier by means of convection and conduction. The radiation recuperator gets its name from the fact that a substantial portion of the heat transfer from the hot gases to the surface of the inner tube takes place by radiative heat transfer. The cold air in the annulars, however, is almost transparent to infrared radiation so that only convection heat transfer takes place to the incoming air.

Based on type of heat transfer they are classified as stack-type or direct contact recuperators. Based on material of construction the direct contact recuperators are further classified as metallic recuperator (used to recover heat from gases at about 1000°C) and ceramic recuperator (used to recover heat from gases at about 1550°C).

Stack-type Recuperators

Stack type recuperators are used when the stack gas temperatures are high and heat transfer dominantly occurs by radiation rather than convection. These are generally co-current heat exchangers and are used for air preheating and is considered cost effective. It has an advantage that hot gases do not have to be rerouted and fans are not required to produce a

draft. A major disadvantage is it required substantial heat difference between exhaust gas and preheat air.

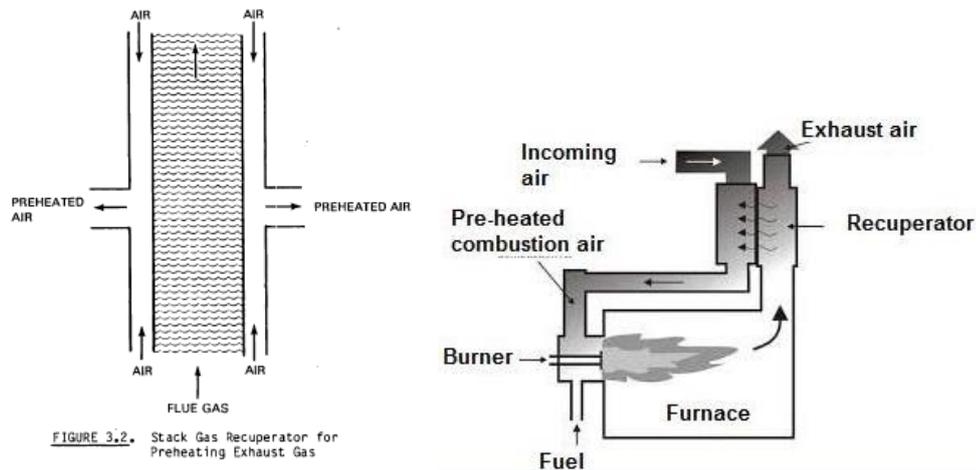


Figure 8.2: Preheating Combustion Air using a Recuperator in Furnace

Direct contact recuperator (ceramic and metal)heat exchangers

Direct contact recuperators provide highest air preheat temperature. These recuperators are also called “checkers”. In this heat exchanger exhaust and air streams are alternated between two sets of checkers. The exhaust stream heats the checkers in one set while the air is being heated by the other. These are considered expensive and require considerable space for duct work and damper arrangements.

Metallic recuperator

The simplest configuration for a recuperator is the metallic radiation recuperator, which consists of two concentric lengths of metal tubing as shown in Figure 8.3

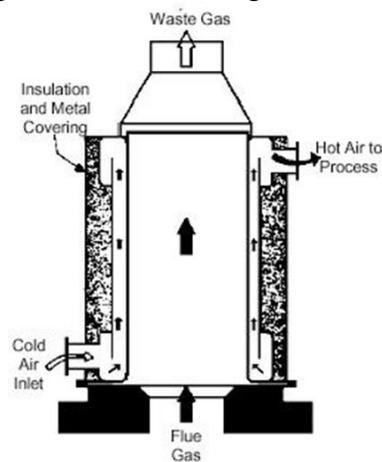


Figure 8.3: Metallic radiation recuperator

The inner tube carries the hot exhaust gases while the external annulus carries the ambient air from the atmosphere which recovers the heat from hot exhaust gases. The heated ambient air is supplied via air inlets of the furnace burners. This equivalent energy of hot combustion air does not have to be supplied by the fuel. So, less fuel is burned for a given

furnace loading. The saving in fuel also means a decrease in combustion air requirements and therefore lesser quantities of exhaust gas.

The principal limitation of metal recuperators is the reduced life of the liner at inlet temperatures exceeding 1100°C.

Ceramic Recuperator

In order to overcome the temperature limitation of metallic recuperators, ceramic tube recuperators have been developed. The materials of ceramic recuperators allow operation on the gas side up to 1550°C and on the preheated air side up to 850°C. Early ceramic recuperators were built of tile and joined with furnace cement, and thermal cycling caused cracking of joints and rapid deterioration of the tubes. Later developments introduced various kinds of short silicon carbide tubes which can be joined by flexible seals located in the air headers.

Although the design of this heat exchanger may change with its particular application, three types are widely used.

Flat plate recuperator

Flat plate recuperator (Figure 8.4) consists of a series of metal (usually aluminium) plates separating 'hot' and 'cold' air or gas flows sandwiched in a box-like structure. The plates are sealed in order to prevent intermixing of the two fluid flows. They are often used in ducted air-conditioning installations to reclaim heat from the exhaust air stream without cross contamination.

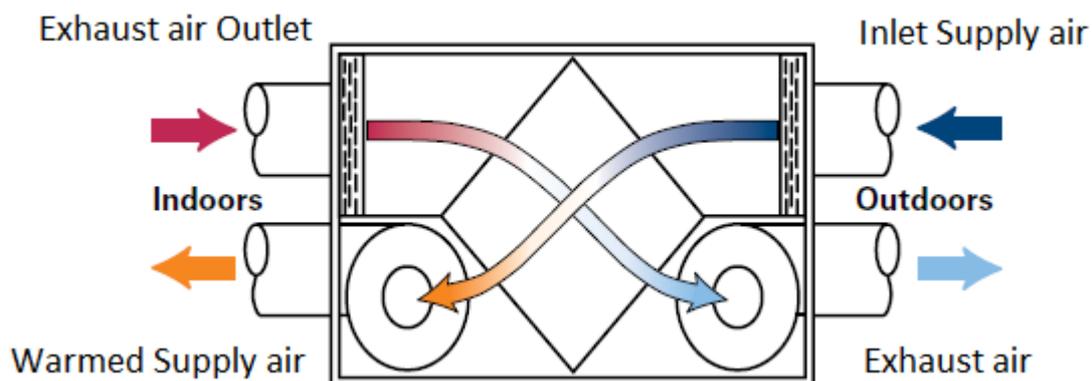


Figure 8.4: Flat Plate Recuperator

8.4.1.3 Plate fin (ceramic and metal) heat exchangers

Plate fin heat exchangers overcome the problems associated with direct contact types and for high temperature applications ($> 1360^{\circ}\text{F}$ discharge air and $> 1500^{\circ}\text{F}$ for flue gas) are available in both metal and ceramic designs

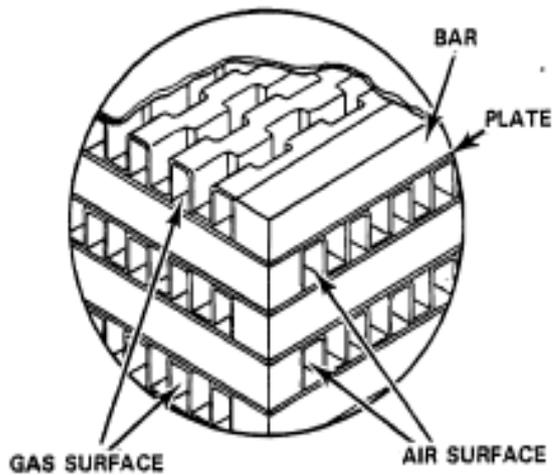


Figure 8.5: Plate Fin Heat Exchanger

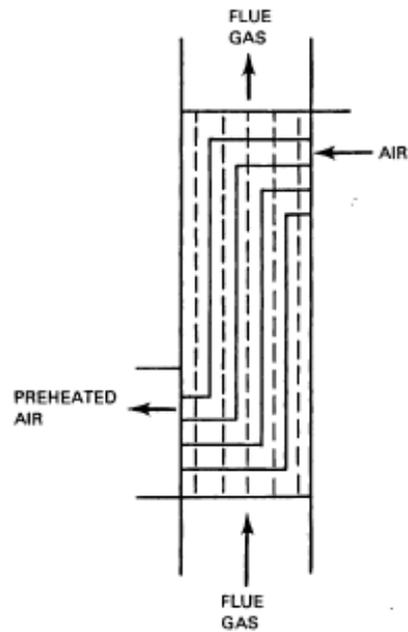


Figure 8.6: Metallic Recuperator with cross-counter flow design

8.4.1.4 Ceramic tubes

These are tubular heat exchangers made of ceramic tubes. They have problems of differential thermal expansion of the ceramic tubes and related hardware which can cause cracking and sealing problems.

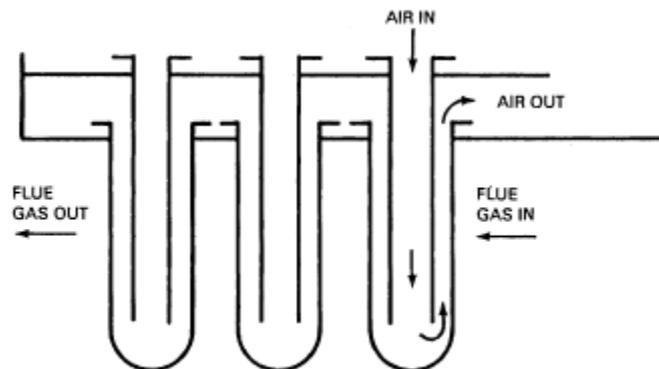


Figure 8.7: Ceramic Re-entrant Tube Heat Exchanger

8.4.2 Gas-to-liquid heat exchanger

8.4.2.1 Waste Heat Boilers

Waste heat boilers are available with fire-tube or water-tube designs in both extended surface and smooth tube models.

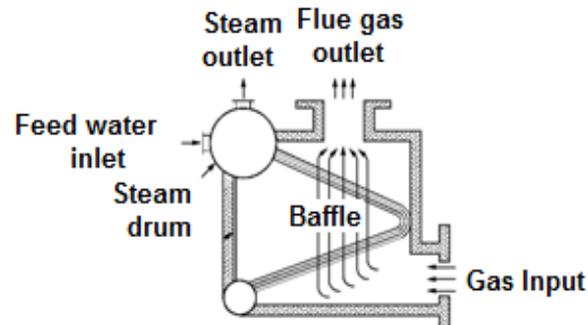


Figure 8.8: Generating Steam using Waste Heat Boiler

8.4.2.2 Economizers

Economizers operate on exhaust gas that already passed through either recuperators or waste heat boilers and are intended to operate waste heat from other heat utilization or recovery equipment. The dew point condensation of exhaust gases sets the lower operating limit for conventional economisers. Direct contact economisers using liquid spray etc. captures the latent heat (~ 1000 BTU/lb of water vapor contained) as well as the remaining sensible heat.

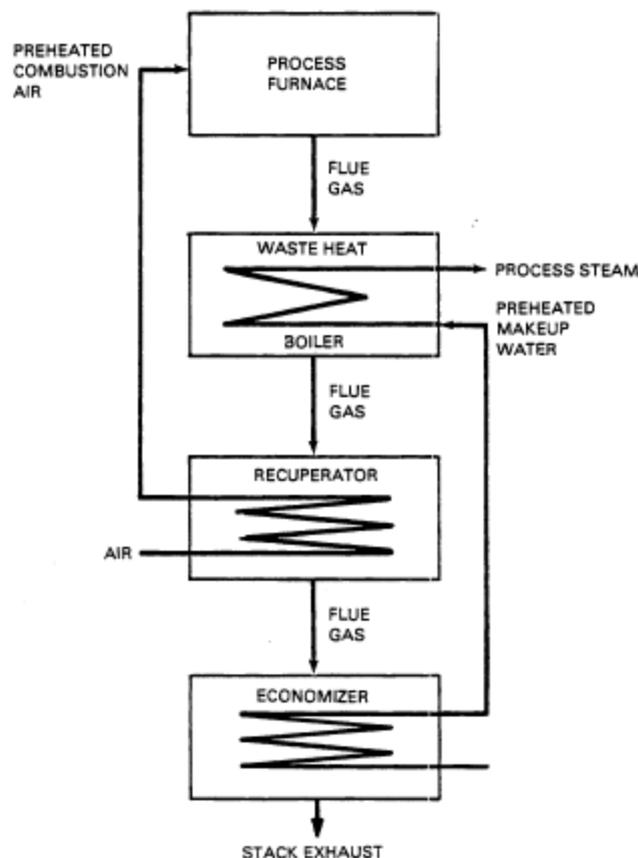


Figure 8.9: Schematic Diagram of Waste Heat Boiler Recuperator and Economiser

8.4.2.3 Power Generators

Cogeneration where process steam and power are jointly generated is used to reduce heat wastage. Organic rankine cycle is used where waste heat temperatures are lower compared to conventional steam cycle systems.

Organic Rankine Cycle (ORC)

ORC unit is a power generation plant which on a mini-scale is typically in the range of 10–250 kW. Unlike the traditional power plant where working fluid is water, evaporated gas is steam, and engine is steam turbine, the ORC system uses working fluids which boil at much lower temperatures and pressures than water. Typical organic fluids used include R234fa, R134, pentane, cyclopentane, n-heptane, hexane, and toluene. The ORC systems can even work on low temperature heat sources (90–300 °C) for heat recovery.

The schematic of ORC system is shown in Figure 8.10. The ORC system is based on the principle whereby organic fluid is heated causing it to evaporate, and the resulting gas is used to turn an organic vapour turbine (expander) which is coupled to a generator producing power. The exhaust vapour is subsequently condensed in water or air-cooled condenser and is recycled to the vaporiser by a liquid pump.

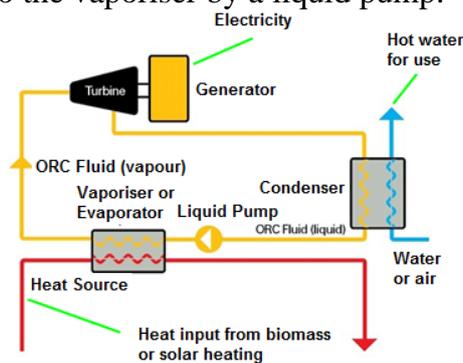


Figure 8.10: Schematic of a Small Scale Organic Rankine Cycle

8.5 Liquid-to-liquid heat exchanger:

8.5.1.1 Shell-and-tube

The most frequently used industrial heat exchanger is the shell-and-tube heat exchanger. It can be made from any material or combination of materials including metal, graphite and glass. The configuration of this heat exchanger is multiple parallel, small-diameter tubes are mounted inside a single, large-diameter tube wherein one fluid flows on the inside of the tubes, while the other fluid is forced through the shell and over the outside of the tube. To ensure that the shell-side fluid will flow across the tubes, and thus induce higher heat transfer, baffles are placed in the shell as shown in figure 8.11. Depending on the head arrangement at the end of the exchanger, one or more tube passes may be used.

The disadvantage of this exchanger is difficulty in cleaning the outside of the tube bundle because of which it is normally used for fluids where significant fouling is not expected.



Figure 8.11: Shell-tube heat exchanger with one pass

8.5.1.2 Plate heat exchanger

Plate heat exchanger also called compact heat exchangers (have surface areas greater than $650 \text{ m}^2/\text{m}^3$), (Figure 8.12) are primarily used in gas-flow systems where the overall heat-transfer coefficient are low and it is desirable to achieve a large surface area in a small volume. They can easily be expanded or contracted to accommodate future system modification. These heat exchangers consists of a large number of thin metal plates (usually stainless steel, titanium, or nickel), which are clamped tightly together and sealed with gaskets. The thin plates are profiled so that ‘flow ways’ are created between the plates when they are packed together. This leads to formation of large surface area, across which heat transfer can take place. Ports located at the corners of the individual plate separate the ‘hot’ and ‘cold’ fluid flows and direct them to alternate passages so that no intermixing of hot and cold fluid occurs. The whole exchanger experiences a counter-flow pattern.

The maximum operating temperature is usually about 130°C if rubber sealing gaskets are fitted, but the operating temperature can be extended to 200°C if compressed asbestos fibre seals are used.

The advantage of these heat exchangers are it is easier to clean, and is only one-fourth the size of the shell-and tube heat exchangers, less prone to fouling and less costly to operate in the long-term than shell-and-tube heat exchangers. For similar applications, plate heat exchangers may be smaller, more efficient, have less internal volume, and cost less than shell-and-tube heat exchangers.

The high-efficiency, low-approach temperatures, counter-flow design, relative ease of fabrication from exotic materials, and clean ability are making the plate heat exchanger attractive for some very difficult liquid-to-liquid applications, such as recovering heat from geothermal brines and manufacturing of chemicals and pharmaceuticals.

Although plate heat exchangers are typically meant for liquid to liquid heat exchange, they are now available for even gas to liquid and even for gases laden with tar with provision of cleaning system.

8.5.1.3 Spiral heat exchanger

Spiral heat exchangers use a double spiral of strip material sandwiched between two plates providing separate spiral flow paths for the fluids as shown in figure 8.13. They can be dismantled easily for cleaning and inspection.

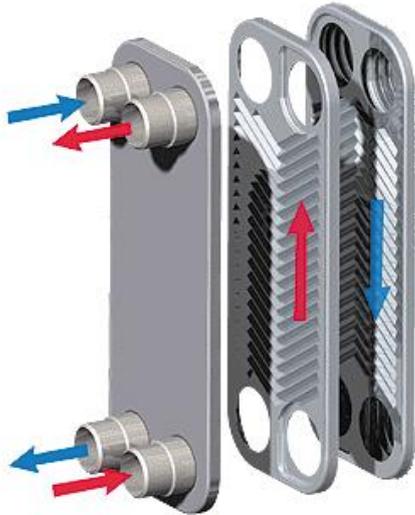


Figure 8.12: Plate Heat Exchanger

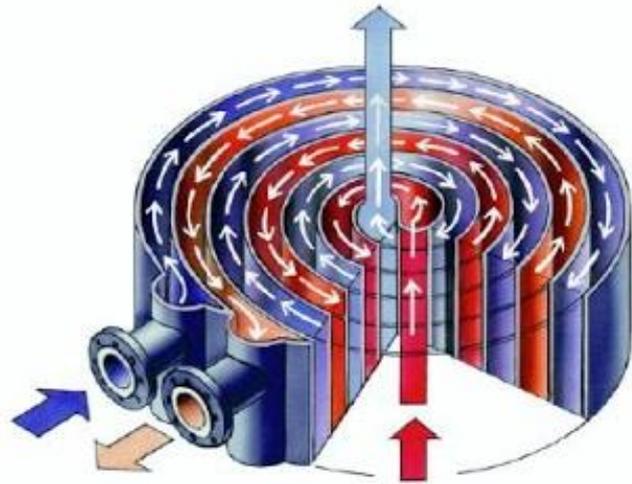


Figure 8.13: Spiral Heat Exchanger

8.5.1.4 Coil heat exchangers

Coils are often used for cooling small amounts of fluid keeping a critical mechanical equipment cool. Heat is transferred between the fluid in the coil and a fluid bath.

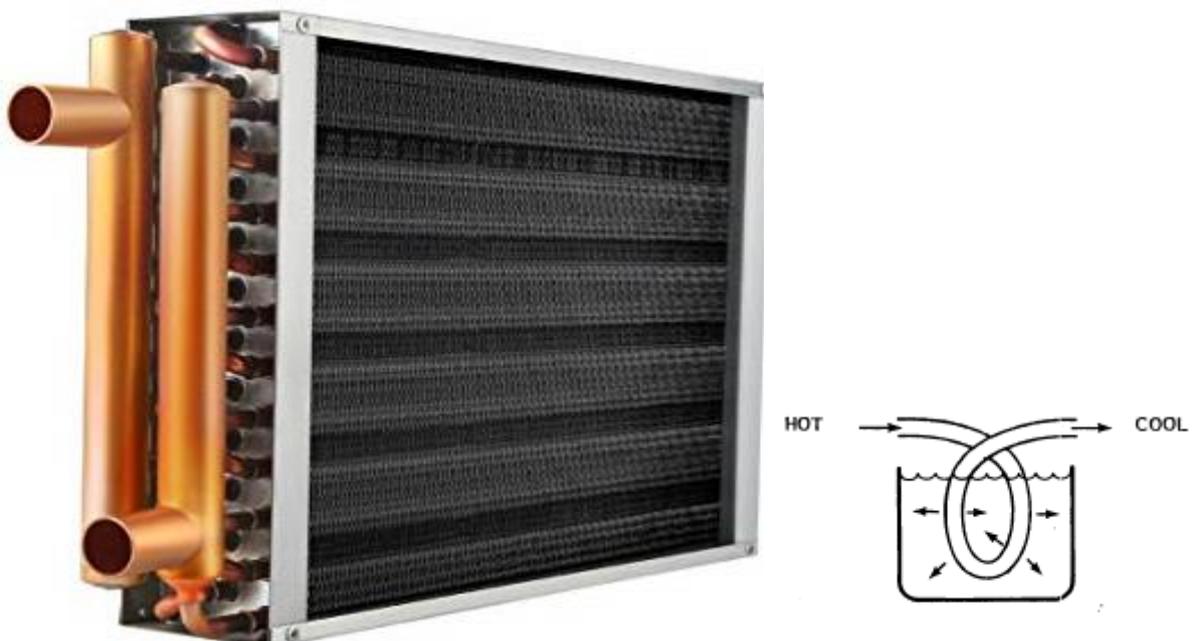


Figure 8.14: Coil Heat Exchanger

8.5.1.5 Finned-tube heat exchangers

Tube-type heat exchangers often use fins on the heat exchanger tubing. Fins can increase the heat transfer rate by increasing the effective heat transfer area. The fins typically run circumferentially around the outside of the tube, although longitudinal fins and internal fins are also used. The fins may be solid or segmented. (Segmented fins are lighter and increase the heat transfer rate by increasing turbulence.)

8.5.1.6 Run-around Coils

When two recuperative heat exchangers are linked together by a secondary fluid which transports heat between them, the system is known as a run-around coil. The cooling system in an automobile where the engine is the source of heat and the air is the sink and the heat is transferred using coolant liquid is an example of run-around coil type heat exchanger. Run-around coils are often employed to recover waste heat from exhaust air streams and to preheat incoming supply air. This will thereby help avoid the risk of cross-contamination between the two air streams. Such a system is shown in Figure 8.15.

Run-around coils have the advantage that they can be used in applications where the two fluid streams are physically far apart to use a recuperative heat exchanger. While this feature is usually considered advantageous, it can increase energy consumption since a pump is introduced into the system. It may also result in heat loss from the secondary fluid. This makes it important to insulate the pipe work circuit; otherwise, the overall effectiveness of the system will be reduced. Run-around coils are relatively inexpensive to install since they utilise standard air/water heating coils.

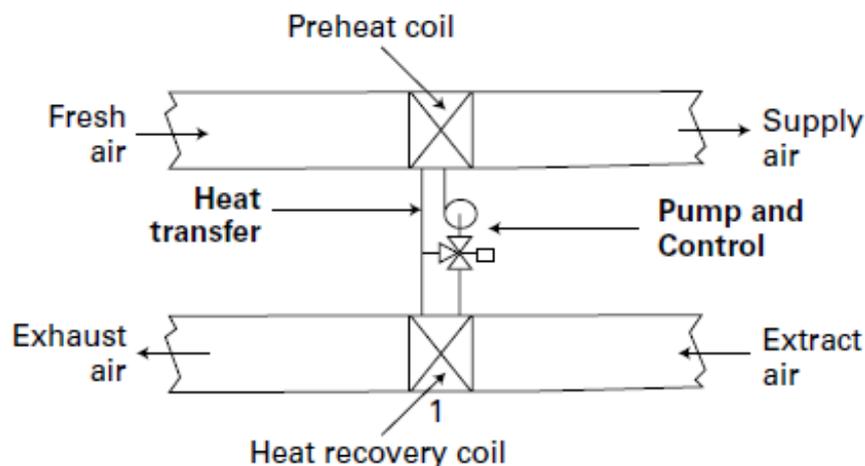


Figure 8.15: Run-around Coil Heat Recovery System

8.5.2 Regenerative Heat Exchangers

In a regenerative heat exchanger a matrix of material is alternately passed from a hot fluid to a cold fluid, so that heat is transferred between the two in a cyclical process. The most commonly type of regenerative heat exchanger is thermal wheel (Figure 8.16) which has a matrix of material mounted on a wheel rotating at about 10 rpm, through hot and cold

fluid streams alternatively. The major advantage of the thermal wheel is the large surface area to volume ratio which results in a relatively low cost per unit surface area.

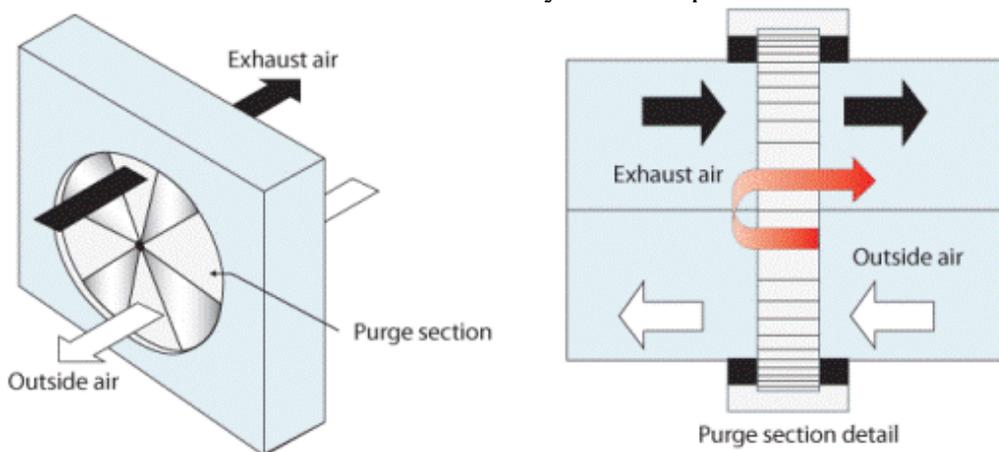


Figure 8.16: Thermal Wheel

The matrix material in the thermal wheel is usually an open structured metal made of knitted stainless steel or aluminium wire, or corrugated sheet aluminium or steel. For use at higher temperatures honeycomb ceramic materials are used. Although thermal wheels are usually used solely to recover sensible heat, it is possible to reclaim the enthalpy of vaporisation of the moisture in the ‘hot’ stream passing through the thermal wheel. This is achieved by coating a non-metallic matrix with a hygroscopic or desiccant material such as lithium chloride.

The main disadvantage of thermal wheels is that there is the possibility of cross-contamination between the air streams. This can be reduced considerably by ensuring that the cleaner of the two fluids is maintained at the highest pressure and with a use of purging device.

Case 8.1

A rotary heat regenerator was installed on a two colour printing press to recover some of the heat, which had been previously dissipated to the atmosphere, and used for drying stage of the process. The outlet exhaust temperature before heat recovery was often in excess of 100°C. After heat recovery the temperature was 35°C. Percentage heat recovery was 55% and pay back on the investment was estimated to be about 18 months. Cross contamination of the fresh air from the solvent in the exhaust gases was at a very acceptable level.

8.5.3 Heat Pump

A heat pump is essentially a vapour compression refrigeration machine which takes heat from low temperature source such as air or water and upgrades it to be used at higher temperature. Unlike a conventional refrigeration machine, using heat pump the heat produced at the condenser is used and not wasted to the atmosphere. The Figure 8.17 shows the operating principle of simple vapour compression heat pump.

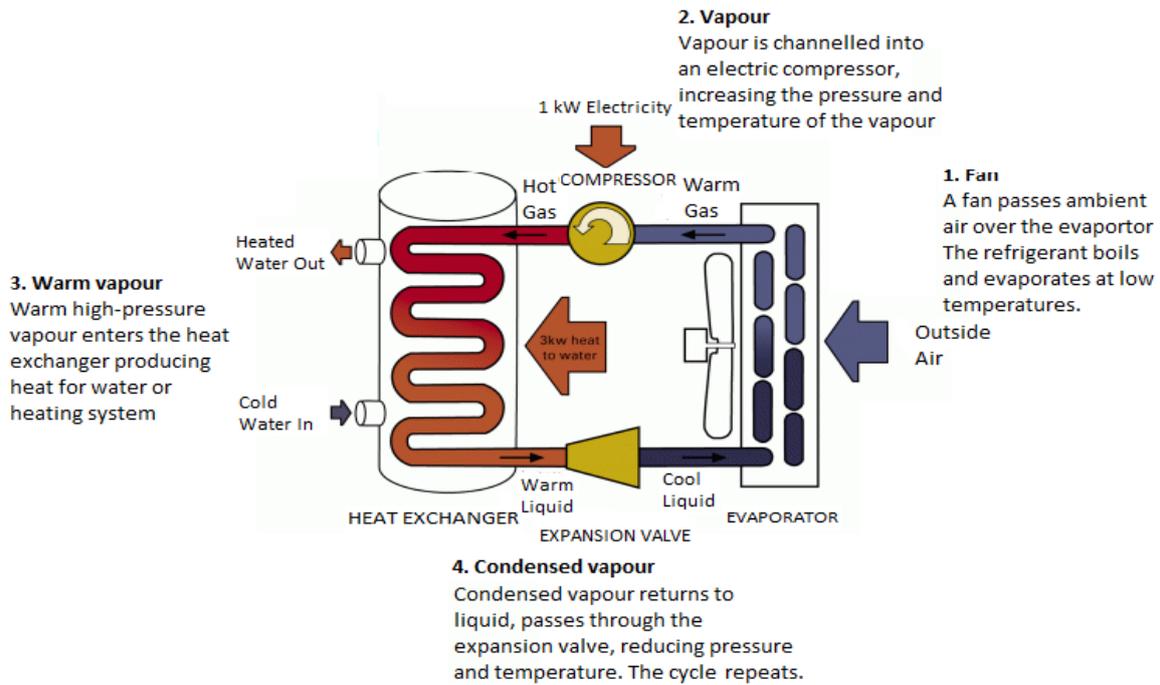


Figure 8.17: Vapor Compression Heat Pump

The performance of the vapour compression refrigeration cycle is quantified by the coefficient of performance (COP), which can be expressed as follows:

For a refrigeration machine:
$$COP_{Ref} = \frac{\text{Useful Refrigeration Output}}{\text{Net Work Input}}$$

For a heat pump:
$$COP_{Hp} = \frac{\text{Useful Heat Rejected Cycle}}{\text{Net Work Input}}$$

Heat pumps are well suited to applications where the evaporating and condensing temperatures are close together such as in the cases when recovering heat from exhaust air in heating and air conditioning applications. As a result, heat pumps are often used in air conditioning applications.

8.5.4 Heat Pipe

Heat pipes are devices which can transfer 1000 times more thermal energy than copper. It can be used in traditionally difficult heat exchange environments such as high particulate gases, dirty liquids, corrosive environments, low temperature gradients.

Heat pipe is basically a metal and metal alloy tube that is sealed on both ends and with an internal wick or mesh along the interior of the pipe. The Heat Pipe comprises of three elements – a sealed container, a capillary wick structure and a working fluid. The capillary wick structure is integrally fabricated into the interior surface of the container tube and sealed under vacuum. Thermal energy applied to the external surface of the heat pipe is in equilibrium with its own vapour as the container tube is sealed under vacuum and causes the working fluid near the surface to evaporate instantaneously. Vapour thus formed absorbs the latent heat of vaporization and this part of the heat pipe becomes an evaporator region. The vapour then travels to the other end the pipe where the thermal energy is removed causing the vapour to condense into liquid again, thereby giving up the latent

heat of the condensation. This part of the heat pipe works as the condenser region. The condensed liquid then flows back to the evaporated region. (Figure 8.18) is Heat pipe has a working fluid within a vacuum and typical working fluids used include liquid nitrogen, methanol, water, and sodium. The temperature range along with fluid type and compatible metals tubes are given the table 8.3 below.

Table 8.3: Temperature Ranges for Heat-Transfer Fluids Used in Heat Pipes

Fluid	Temperature range (°C)	Compatible Metals
Nitrogen	-180 to 80	Stainless steel
Ammonia	-70 to 60	Nickel, Aluminum, Stainless steel
Methanol	-45 to 115	Nickel, Copper, Stainless steel
Water	5 to 215	Nickel, Copper
Mercury	190 to 535	Stainless steel
Sodium	510 to 870	Nickel, Stainless steel
Lithium	870 to 1480	Alloy of Niobium and Zirconium
Silver	1480 to 1980	Alloy of Tantalum and Tungsten

The performance (amount of heat that can be transferred) of a heat pipe is a function of its length, diameter, wick structure, and overall shape. The larger the diameter, the more power that can be transported, but longer the length, the less capable is the performance.

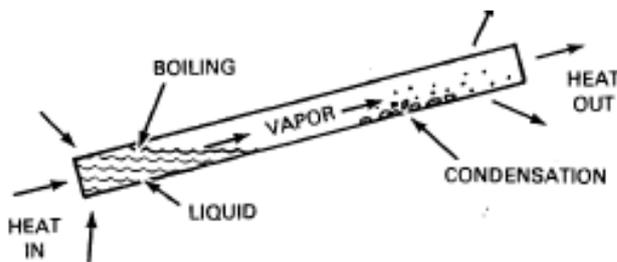


Figure 8.18: Heat Pipe

The heat pipes are used in following industrial applications:

a. Process to Space Heating:

The heat pipe heat exchanger transfers the thermal energy from process exhaust for building heating. The preheated air can be blended if required. The requirement of additional heating equipment to deliver heated make up air is drastically reduced or eliminated.

b. Process to Process:

The heat pipe heat exchangers recover waste thermal energy from the process exhaust and transfer this energy to the incoming process air. The incoming air thus become warm and can be used for the same process/other processes and reduces process energy consumption.

c. HVAC Applications:

Cooling: Heat pipe heat exchangers pre-cools the building make up air in summer and thus reduces the total tons of refrigeration, apart from the operational saving of the cooling system. Thermal energy is supply recovered from the cool exhaust and transferred to the hot supply make up air.

Advantages of heat pipe system

- Heat pipes operates independently so are not vulnerable to a single pipe failure.
- No cross contamination occurs as hot and cold sides are separated by a splitter plate.
- No wear and tear occurs as there are no moving parts inside the heat pipe.
- No additional power is required to run the system

Industrial applications:

- Preheating of boiler combustion air
- Recovery of Waste heat from furnaces
- Reheating of fresh air for hot air driers
- Recovery of waste heat from catalytic deodorizing equipment
- Reuse of Furnace waste heat as heat source for other oven
- Cooling of closed rooms with outside air
- Preheating of boiler feed water with waste heat recovery from flue gases in the heat pipe economizers.
- Drying, curing and baking ovens
- Waste steam reclamation
- Brick kilns (secondary recovery)
- Reverberatory furnaces (secondary recovery)
- Heating, ventilating and air-conditioning systems

Case 8.2

In a hospital, the HVAC system exhausts 140 m³/min of air which has a heat recovery potential of 28225kcal/hr. Calculate the savings and payback period of heat recovery system to be installed at a cost of BDT 1,12,000.

Solution:

Savings in Hospital Cooling Systems

	Units	Value
Volume of exhaust	m ³ /min	140
Heat recovery potential	kcal/hr	28225
Conversion: 1 kcal/hr	TR	0.00033069
Electricity required	KW/TR	0.8
Plant capacity reduction	TR	9.33
Electricity cost(operation)	BDT/Million kcal (based on 0.8KW/TR)	268
Plant capacity reduction cost (Capital)	BDT/TR	12,000

Capital cost savings	BDT	1,12,000/-
Payback period	hours	16570

8.5.5 Thermo-compressor

In many cases, very low pressure steam are reused as water after condensation for lack of any better option of reuse. In many cases it becomes feasible to compress this low pressure steam by very high pressure steam and reuse it as a medium pressure steam. The major energy in steam, is in its latent heat value and thus thermo compressing would give a large improvement in waste heat recovery.

The thermo-compressor is a simple equipment with a nozzle where HP steam is accelerated into a high velocity fluid. This entrains the LP steam by momentum transfer and then recompresses in a divergent venturi. A figure of thermo-compressor is shown in Figure 8.19.

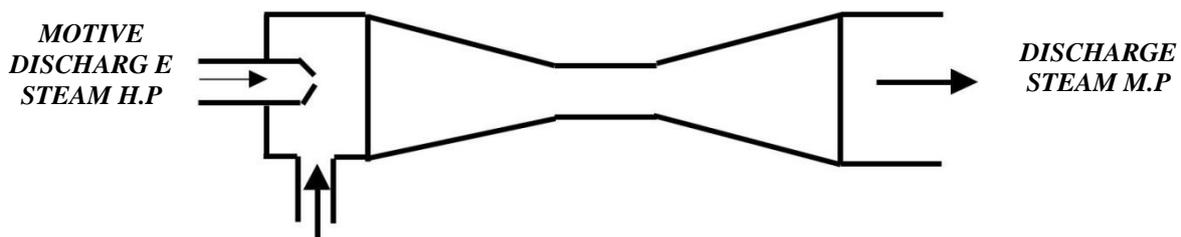


Figure 8.19: Thermo-compressor

It is typically used in evaporators where the boiling steam is recompressed and used as heating steam

Case 8.3

Exhaust steam from evaporator in a fruit juice concentrator plant was condensed in a pre-condenser operation on cooling water upstream of a steam jet vacuum ejector

Equipment Suggested	Alt-1 Thermo compressor Alt-2 shell & tube exchanger
Cost of thermo compressor	BDT 1.5Lakhs
Savings of jacket steam due to recompression of vapour	BDT 5.0 Lakhs per annum
Cost of shell & tube exchanger to pre heat boiler feed water	BDT75,000/-
Savings in fuel cost	BDT 4.5 Lakhs per annum

The recovery technologies for different heat source are given below

Table 8.4: Waste of Heat Recovery Technologies for High range heat sources

High range Heat Source	Temperature range (°C)	Recovery method	Typical uses	Type of heat exchanger (Gas-Gas, Gas-Liquid)	Large temperature differentials permitted	No Cross - contamination
Nickel refining furnace	1370 –1650	RR, CR	1	G-G	x	x
Aluminium refining furnace	650-760	CHW,CR, WHB	1,2,3,5	G-G, G-L	x	x
Zinc refining furnace	760-1100	RR, CR	1	G-G	x	x
Copper refining furnace	760- 815	CHW,CR, WHB	1,2,3,5	G-G, G-L	x	x
Steel heating furnaces	925-1050	RR, CR	1	G-G	x	x
Copper reverberatory furnace	900-1100	RR, CR	1	G-G	x	x
Open hearth furnace	650-700	CHW,CR	1,2,3,5	G-G, G-L	x	x
Cement kiln (Dry process)	620- 730	CHW,CR	1,2,3,5	G-G, G-L	x	x
Glass melting furnace	1000-1550	RR, CR	1	G-G	x	x
Hydrogen plants	650-1000	CHW,CR, WHB	1,2,3,5	G-G, G-L	x	x
Solid waste incinerators	650-1000	CHW,CR, WHB	1,2,3,5	G-G, G-L	x	x
Fume incinerators	650-1450	CHW,CR, WHB	1,2,3,5	G-G, G-L	x	x

Recovery Methods

RR- Radiation Recuperator; CR - Convection Recuperator; MHW - Metallic Heat Wheel;CHW - Ceramic Heat Wheel WHB - Waste-heat Boilers

Typical Uses

1. Combustion air preheat 2. Space preheat 3. Boiler makeup water preheat 4. Boiler feed water preheat
5. Domestic hot water 6. Hot water or steam generation. 7. Liquid feed flows requiring heating

(Source: W. Turner Energy Management Handbook, 2007; Waste Heat Recovery: Technology and Opportunities in U.S Industry,2008.)

Table 8.5: Waste Heat Recovery Technologies for medium range heat sources

Heat Source	Temperature range (°C)	Recovery method	Typical uses	Type of heat exchanger (Gas-Gas, Gas-Liquid)	Large temperature differentials permitted	No Cross - contamination
Steam boiler exhausts	230-480	MHW, HHW, PHE	4,6,7	G-G, G-L	-	x
Gas turbine exhausts	370-540	MHW, HHW, PHE	4,6,7	G-G, G-L	-	x
Reciprocating engine exhausts	315-600	CR, HP, WHB, CHW	1	G-G	x	x
Reciprocating engine exhausts (turbo charged)	230- 370	MHW, HHW, PHE	4,6,7	G-G, G-L	-	x
Heat treating furnaces	425 - 650	CR, HP, WHB, CHW	1	G-G	x	x
Drying and baking ovens	230 - 600	CR, HP, WHB, CHW	1	G-G	x	x
Catalytic crackers	425 - 650	CR, HP, WHB, CHW	1	G-G	x	x
Annealing furnace cooling systems	425 - 650	CR, HP, WHB, CHW	1	G-G	x	x

Recovery Methods

CR - Convection Recuperator; CHW - Ceramic Heat Wheel; PHE – Plate type heat exchanger ; HHW - Hygroscopic Heat Wheel; MHW - Metallic Heat Wheel; FHE- Finned-tube Heat exchanger; ST- shell and tube exchanger; WHB - Waste-heat Boilers; HP - Heat Pipe

Typical Uses

1. Combustion air preheat. 2. Space preheat. 3. Boiler makeup water preheat. 4. Boiler feed water preheat. 5. Domestic hot water 6. Hot water or steam generation. 7. Liquid feed flows requiring heating.

(Source: W. Turner Energy Management Handbook, 2007; Waste Heat Recovery: Technology and Opportunities in U.S Industry,2008.)

Table 8.6: Waste Heat Recovery Technologies for low range heat sources

Heat Source	Temperature range (°C)	Recovery method	Typical uses	Type of heat exchanger (Gas-Gas, Gas-Liquid)	Large temperature differentials permitted	No Cross - contamination
Process steam condensate	55-88	PHE,MHW, HHW	2	G-L	-	x
Cooling water from:						
Furnace doors	32-55	PHE,MHW, HHW	2	G-L	-	x
Bearings, Welding machines, Injection molding machines.	32-88	PHE,MHW, HHW	2	G-L	-	x
Annealing furnaces	66-230	FHE, WHB, ST	3,6,7	G-G,G-L	x	x
Forming dies, pumps	27-88	PHE,MHW, HHW	2	G-L	-	x
Air compressors	27-50	PHE,MHW, HHW	2	G-L	-	x
Air conditioning and refrigeration condensers	32-43	PHE,MHW, HHW	2	G-L	-	x
Drying, baking and curing ovens	93-230	FHE, WHB, ST	2	G-L		x
Hot processed liquids	32-232	FHE, WHB, ST	3,6,7	G-G,G-L	x	x

Recovery Methods

MHW - Metallic Heat Wheel; HHW - Hygroscopic Heat Wheel; PHE – Plate type heat exchanger ; FHE Finned-tube Heat exchanger; ST- Shell and tube exchanger; WHB - Waste-heat Boilers.

Typical Uses

1. Combustion air preheat. 2. Space preheat. 3. Boiler makeup water preheat. 4. Boiler feed water pre heat. 5. Domestic hot water. 6. Hot water or steam generation. 7. Liquid feed flows requiring heating.

(Source: W. Turner Energy Management Handbook, 2007; Waste Heat Recovery: Technology and Opportunities in U.S. Industry, 2008)

8.6 Heat exchanger configurations

The two most commonly used heat exchanger flow configurations are *counter flow* and *parallel flow*. These flow patterns are represented in Figures 8.20 and 8.21 respectively, along with their characteristic temperature profiles.

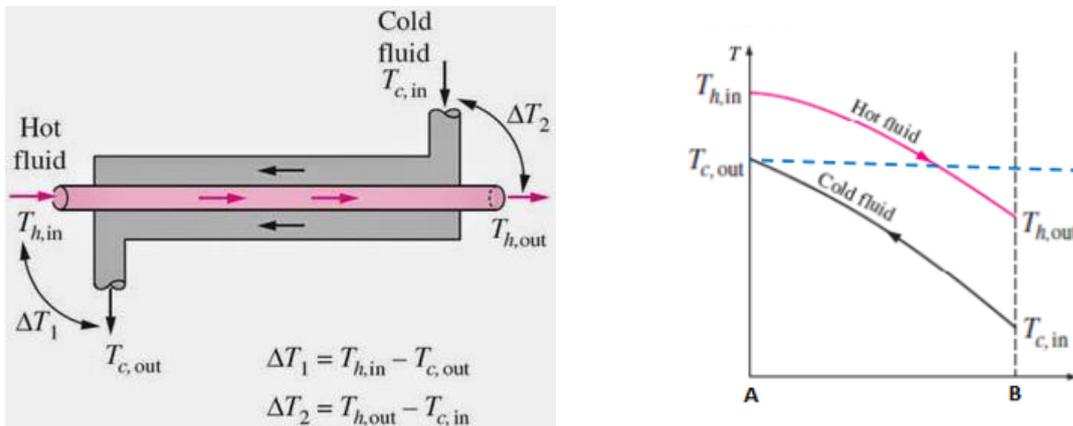


Figure 8.20: Counter Flow Heat Exchanger

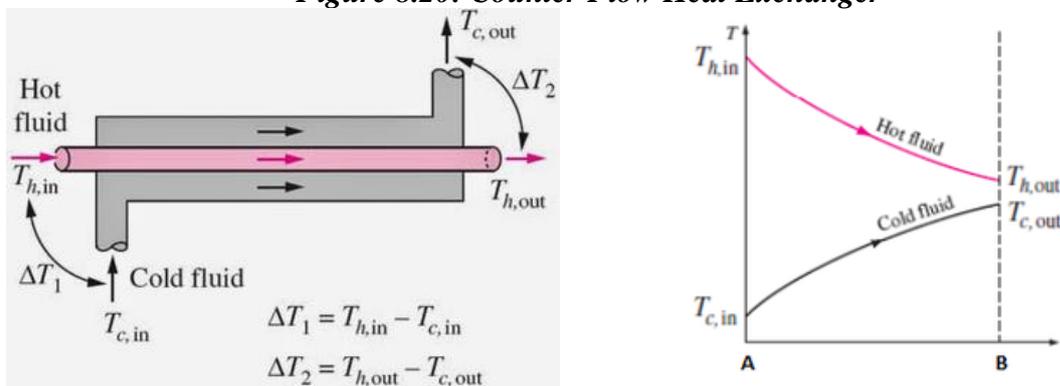


Figure 8.21: Parallel Flow Heat Exchanger

It should be noted that with the parallel flow configuration the ‘hot’ stream is always warmer than the ‘cold’ stream. With the counter flow configuration it is possible for the outlet temperature of the cold fluid to be higher than the outlet temperature of the hot fluid. The general equations which govern the heat transfer in recuperative heat exchangers are as follows:

$$Q = m_h C_h (t_{h.in} - t_{h.out}) = m_c C_c (t_{c.in} - t_{c.out})$$

and

$$Q = U A_o (\text{LMTD}) K$$

Where,

Q is the rate of heat transfer (W),
 m_h is the mass flow rate of hot fluid (kg/s),
 m_c is the mass flow rate of cold fluid (kg/s),
 C_h is the specific heat of hot fluid (J/kgK),
 C_c is the specific heat of cold fluid (J/kgK),
 $t_{h,in}$ and $t_{h,out}$ are the inlet and outlet temperatures of hot fluid
 $t_{c,in}$ and $t_{c,out}$ are the inlet and outlet temperatures of cold fluid

U is the overall heat transfer coefficient (i.e. U value) (W/m²K),

A_o is the outside surface area of heat exchanger (m²),

LMTD is the logarithmic mean temperature difference

K is the constant which is dependent on the type of flow through the heat exchanger (K = 1 for counter-flow and parallel flow, and is therefore often ignored).

The logarithmic mean temperature difference (LMTD) can be determined by:

$$\text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

where,

$\Delta T_1 = t_{h.in} - t_{c.out}$ and $\Delta T_2 = t_{h.out} - t_{c.in}$ for Counter Flow Heat Exchanger

$\Delta T_1 = t_{h.in} - t_{c.in}$ and $\Delta T_2 = t_{h.out} - t_{c.out}$ for Parallel Flow Heat Exchanger

Case 8.4: Heat recovery from heat treatment furnace

In a heat treatment furnace, the exhaust gases are leaving the furnace at 900°C at the rate of 2100 m³/hour. The total heat recoverable at 180°C final exhaust can be calculated as

$$Q = V \times \rho \times C_p \times \Delta T$$

Q is the heat content in kCal

V is the flow rate of the substance in m³/hr

ρ is density of the flue gas in kg/m³

C_p is the specific heat of the substance in kCal/kg °C

ΔT is the temperature difference in °C

C_p (Specific heat of flue gas) = 0.24 kCal/kg/°C

Heat available (Q) = 2100 x 1.19 x 0.24 x (900-180) = 4,31,827 kCal/hr

By installing a recuperator, this heat can be recovered to pre-heat the combustion air. The fuel savings would be 33% (@ 1% fuel reduction for every 22°C reduction in temperature of flue gas).

8.7 Heat Transfer Augmentation

Principles of heat transfer for waste heat recovery are similar to any heat exchange mechanisms. To promote the heat transfer co-efficient h, three different methods are used. They are

- Active method
- Passive Method
- Combined active and passive method

Active Method employs supplementary power supply. Passive method includes twisted tapes, ribbons, wire coils, indentation of heat transfer surface, etc. to break the thermal boundary layer to promote the heat transfer coefficient. In some cases, both the active and passive methods may be combined to promote the heat transfer coefficient.