25
Gas and Steam Turbines Combined Cycles

25.1. INTRODUCTION

Gas turbine plants are necessarily used as peak load plants, emergency stand-by unit or hydro-station stand-by unit and base load plant under specific conditions mentioned earlier. The quick starting and good response characteristics of the gas turbine plant make the gas turbine as desirable peak load and essential stand-by plant. The non-availability of cooling water will not hamper gas turbine plant while ruling out steam turbine plant. The gas turbine plant can be used as base load plant where the gas turbine fuel is relatively cheap.

The temperature of the exhaust gases of a simple gas turbine plant lies between 400 to 500°C and contains about 16% oxygen compared with 21% in atmospheric air. A large quantity of energy (70% of initial) is also carried away by the exhaust gases with large quantity of O_2 without use.

With the use of preheater in the cycle, the heat carried away by the exhaust gases is reduced from 70% to 60%. The air preheaters with their large gas and air piping make the plant considerably more costly and do not increase the power output for a given air flow, they merely improve the efficiency. The pressure losses in the air preheater and piping have a detrimental effect on the output. In addition, the maximum output does not coincide with the pressure ratio for optimum thermal efficiency. The use of preherater only increases thermal efficiency but O_2 still is carried unused with exhaust gases.

An electrical utility industry has launched an effort to recover the heat energy of the exhaust gases by coupling a steam plant with a gas turbine installation. This combined cycle recovers much of exhaust energy by passing high temperature exhaust gases to heat recovery boiler to generate steam which can be further used to drive a steam turbine.

Increased power and higher thermal efficiency obtained from this, the concept of combined cycle reduces the cost of the additional equipment and lowers the generating cost if the number of operating hours per year substantially increased.

The combined cycle plant could become alternative to conventional base load plants in the intermediate load range with an additional advantage of reduced emission of heat to the atmosphere and reduced requirements of cooling water.

As one looks at the present energy situation, it is hard to predict the course of petroleum fuel prices, other than to say that the long term is upward. But no matter what happens, a reliable combined cycle designed with maximum fuel flexibility would be a key candidate for replacement of retired equipment and industrial cogeneration.

The large quantity of heat available with exhaust gases can be used for power generation or process steam or process heat depending upon the requirements and economics. Figure 25.1 shows the arrangement of gas turbine plant with exhaust heat recovery system.

Laying of HBJ pipe line has set the pace for co-generation in India. NTPC has planned for co-generation plants at Kawas in Gujrat, Anta in Rajasthan and in U.P. with a total capacity of 1500 MW.

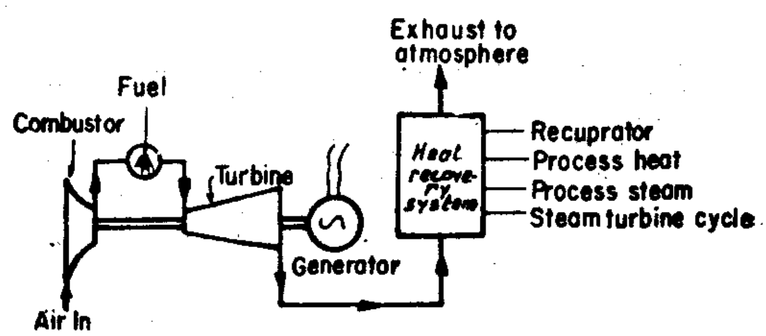


Fig. 25.1. Gas turbine plant with exhaust heat recovery systems.

The heat in exhaust either can be used to heat the air before coming to the combustion chamber or to generate the steam in a separate cycle. The former can be performed effecitively for moderate pressure ratio plants because of temperature rise during compression, whereas the latter can be done even for high pressure ratios.

When the heat in the exhaust gases of the gas turbine is used for power generation using steam as working fluid, the cycle of power generation is known as combined cycle.

The economic advantage of a combined plant compared with steam or gas turbine power plant can be in the first cost as well as in running cost. Whatever may be the fuel used to run the gas turbine plant, important saving can be only achieved if combined cycle plant is installed instead of single cycle gas turbine plant.

The power generation cost is determined by initital investment, fuel (running) cost and the cost of maintenance and operation.

Figure 25.2 exhibits capital cost for different power plants on the basis of 1985 costs. It can be seen that large differences exist between the various types. However, the least expensive plant is not necessarily most economical as its running cost may be high.

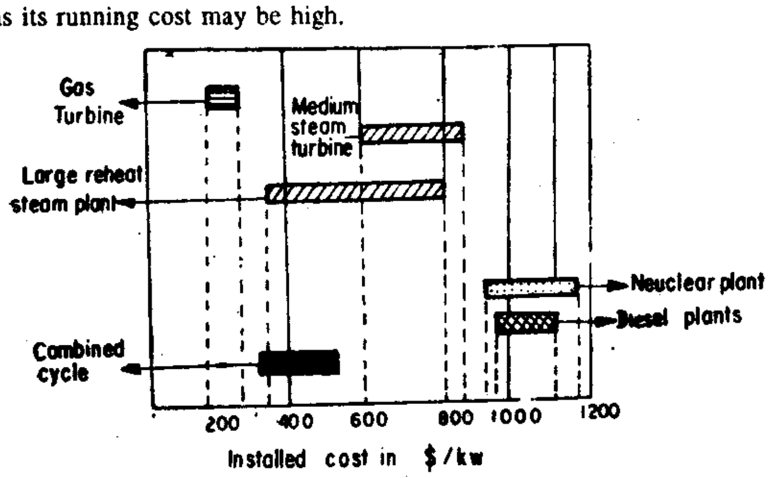


Fig. 25.2. Capital cost of different power plants.

The cost of power (running cost) depends on type and quality of fuel used and to a large extent the specific fuel consumption of the power plant. All the fuels cannot be used in all the plants. Solid fuels are restricted in their applications due to many difficulities experienced. Such restrictions also exist for low quality residual oil. Only steam plants can use these fuels.

However, if a fuel is suitable for gas turbine plant, advantages can be drawn from remarkably high efficiency of a combined cycle. Fig. 25.3 represents the ranges of the achievable efficiencies for different plants and it is obvious that the best efficiency is obtained with combined cycle. The combined cycle is very competitive in first cost and running cost, therefore, all utility plants in future will be converted to combined plants if proper technology is developed to use all types of fuels including coal to municipal waste.

Fig. 25.4 shows a comparison of a number of fossil fueled plants used to generate electricity. The combined cycle is shown to be more efficient than other commonly used systems.

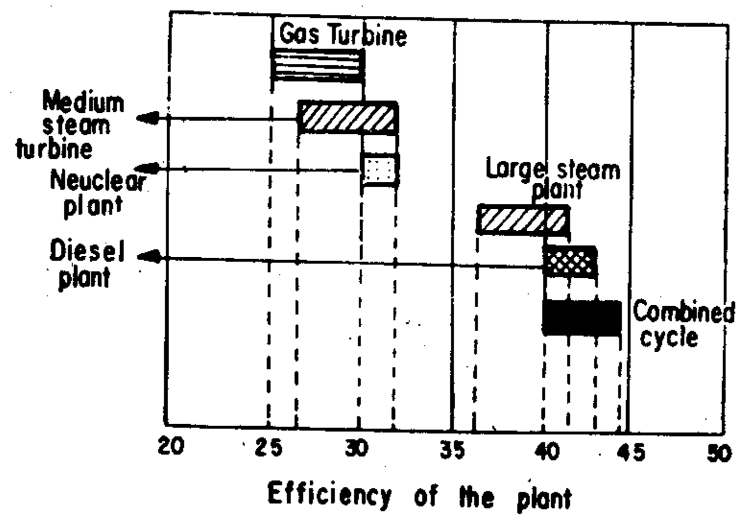


Fig. 25.3. Efficiency of different plants means running cost.

The design of the heat recovery boiler is always critical in the combined cycle because if more heat is recovered from the gas turbine exhaust, the more expensive will be the heat recovery boiler.

In conventional regenerative steam turbine cycle, extreme efforts are made for regenerative feed heating by steam bleeding from the turbine, which reduces the heat rejection to the condenser but also reduces the output of the steam trubine. But in the combined cycle system, low grade heat of gas turbine exhaust gases is used for feed water heating and here the rate of heat recovery boiler is to maximise the steam production for use in the turbine.

The pressure of the steam generated in the heat recovery boiler is the most important single factor in determining the amount of steam produced in conjunction with the pinch point (defference in temperature between gas temperature leaving the evaporation section of the boiler and steam saturation temperature.)* The lower the saturation pressure and temperature, the more of the lower grade heat can be used. On the other hand, the lower the pressure, the lower the heat available for conversion to work in the turbine. A lower pressure requires a larger volume flow through the steam turbine and therefore a bigger turbine size. Therefore, an economic compromise has to be made between the output advantage of the larger steam flow and extra cost of the turbine.

There is no advantage for generating high pressure steam in heat recovery boiler (which is contrary to conventional steam turbine practice) and recommended value for optimum gain is 25 to 30 bar. Optimum results are obtained when low pressure generated steam is passed into the turbine and the feed heating is achieved entirely by feed water recirculation through the economiser. By such means, an efficiency

*See the chapter on Waste Heat Recovery System for further explanation of the statement.

of 47% can be achieved when using gas as fuel in gas turbines with 20% increase in the size of conventional steam plant.

The development of gas turbine technology for power generation has accelerated the advanced power plant concepts. Combined cycle power plant is presently at forefront because of its high efficiency.

Demand for combined cycle plants began to accelerate sharply in the late 1980s. Between 1990 and 1993, 80,000 MW of combined cycle plants were ordered worldwide about 30% in Western Europe and 25% in U.S.A. and remaining by Far East countries as shown in Fig. 25.5.

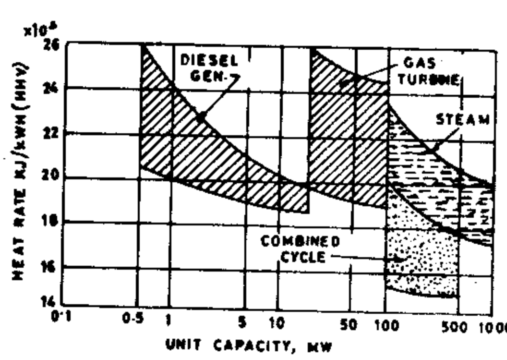


Fig. 25.4. Comparison of heat rates for fossil-fueled energy system. A combined-cycle installation can compete except in maximum size with a steam-only simple-cycle plant. It has the lowest heat rate of any of the four most common fossil systems.

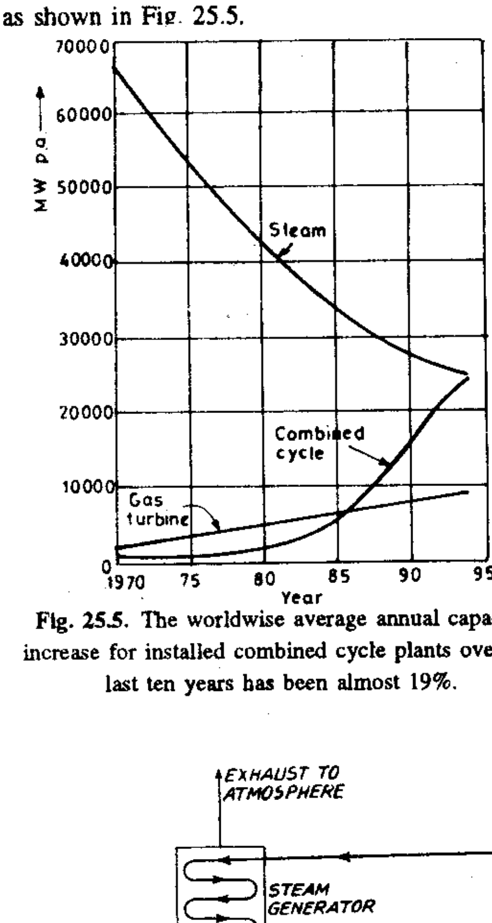


Fig. 25.5. The worldwise average annual capacity increase for installed combined cycle plants over the last ten years has been almost 19%.

The average annual capacity increase of installed combined cycle plant over last 10 years is 19% while fossil fueled power plants have experienced only 2.5% increase per annum. This clearly indicates the combined cycle plants are the focal point of the power plant market. (700 MW Rye House combined cycle plant in U.K. is biggest in the world).

25.2. ARRANGEMENTS OF COMBINED CYCLE

The heat in the exhaust can be used by using one of the arrangements of combined cycle mentioned below :

1. The heat in the exhaust gases of a simple gas turbine plant can be used to generate steam in waste-heat boiler and the generated steam is used in the steam turbine for power generation. The arrangement of the system is

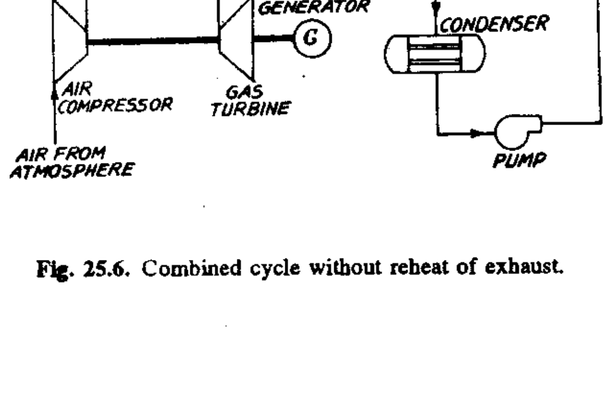


Fig. 25.6. Combined cycle without reheat of exhaust.

shown in Fig. 25.6. In this arrangement, the regenerator is replaced by a waste heat boiler without any use of oxygen carried by the exhaust gases.

2. The heat in the exhaust and O_2 carried with the exhast gases, both are used in the waste heat boiler by supplying the fuel in the secondary combustion chamber. The steam generated is used with steam power plant. The complete arrangement of the system using liquid fuel is shown in Fig. 25.15. The steam feed heaters are not shown in the figure to avoid complicated circuitry.

A 2600 MW combined cycle plant of this type constructed by Tokyo Electric Power Co. is the largest in the world.

The solid fuel can be used in secondary combustion chamber for generating steam economically.

The temperature of the exhaust coming out of the steam boiler depends on the type of fuel used in the gas turbine plant and steam generator. For the natural gas, the temperature of flue gas from the steam generator can be low as natural gas contains low SO_2 and there is hardly any danger of low temperature corrosion in the steam generator. But, if liquid fuels like LDO and HSD which contain higher sulphur are used, then exhaust gas temperature should be sufficiently above the acid dew point of oxides of sulphur to avoid low temperature corrosion. The sulphur content of the fuel determines the minimum feed water temperature to the steam generator and, therefore, one of the factors that governs the type of basic cycle to be adopted for the steam cycle. Therefore, the steam generator manufacturers, during the design of the equipment, optimises the pinch point (temperature difference between the gas and steam in the boiler) for economical design taking into account the optimum exhaust gas temperature from steam generator.

Sizing of the Units and Plant Layout

In case of conventional steam power plant, large turbine-generator units upto 1200 MW capacity are presently available. However, this is not so in case of gas turbine units. The maximum size of the gas turbine unit presently available is about 120 MW which is much lower compared to the largest single steam turbine unit. It is always advantageous to split the plant capacity into smaller capacity gas turbine units coupled with one or two steam turbine units. This would not only provide the flexibility in the operation of the plant but also ensure higher plant availability. *More important is the arrangement that would provide higher plant efficiency over a wider plant loading.* This aspect of higher plant efficiency for a combined cycle plant over a wide load range is shown in Fig. 25.7. The partial load efficiency of the combined cycle plant is nevertheless higher than for the conventional power plants with the same rating.

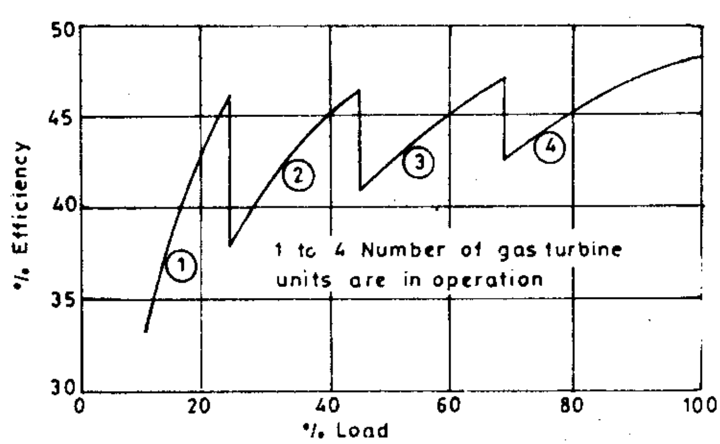


Fig. 25.7. Effect of load variation on efficiency in a combined cycle power plant.

Presently, with special blade air cooling system, the inlet temperature of 1300°C could be safely used. Special coating is generally provided for the first turbine blade rows to resist high temperature corrosion attack.

Some manufacturers have designed aeroderivative type gas turbine units of 35 MW capacity which gives 36% open cycle efficiency. These units operate at high inlet temperature and higher air compression ratio (upto 12) and require good quality fuels.

Plant layout requires special attention during design of a gas turbine combined cycle plant. The relative location of gas turbine and steam turbine generators shall be selected to provide easy operation and maintenance of the plant and to reduce the cost of piping and cables. The location of fuel receipt, treatment and supply system with respect to the main gas plant shall be optimum from the viewpoint of piping layout and effluent disposal.

The location of electric generator at the cold end of gas turbine allows an axial exhaust duct to be connected straight into the steam generator leading to a simple layout for the combined cycle plant.

Another important aspect is the provision of material handling equipment for facilitating easy maintenance. The control room should be so located and insulated to reduce the noise level to minimum.

The gas turbine needs more maintenance and frequency is further increased if heavy fuel oil is used as fuel compared with steam turbines. Therefore, this aspect should be kept in mind while planning the equipment layout for the gas turbine unit and its associated auxiliaries.

It is advisable to locate gas turbine generator unit inside a building like steam turbine, even it can be located outdoor. This arrangement not only provides weather protection for gas turbine unit but helps for easy maintenance of the equipments. Indoor installation improves sound proofing as well as reduces space requirements.

A gas turbine is designed to operate at a fixed exhaust temperature. If this temperature is either increased or decreased, the thermal efficiency of the gas turbine cycle suffers. That is why, a gas turbine is coupled with heat recovery boiler. It lowers the temperature of the gases and increases the overall efficiency of the system.

Usually, the heat recovery system is required to supply a certain steam flow at a given pressure and variable steam quantity as per load on steam plant. It is not always possible to balance this output with fixed exhaust gas temperature of the gas turbine. In this case, exhaust temperature is increased by fixing with supplementary fuel as shown in Fig. 25.7.

The beneficial effect of supplementary firing is the increase in heat recovery capability as the fixed input is increased. The effect of input temperature for steam generator on cost and heat transfer area is shown in Fig. 25.8.

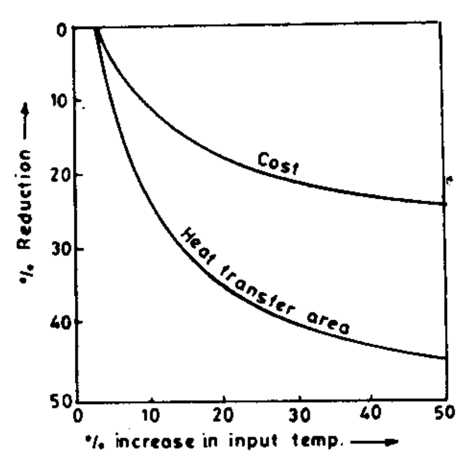


Fig. 25.8. Percent increase in input temperature.

The overall efficiency of combined power plant with secondary firing is given by

$$\eta = \eta_2 + \frac{Q_1\eta_1(1-\eta_2)}{Q_1+Q_2}$$

where Q_1 is the heat input in gas turbine combustor and Q_2 is the heat input in the secondary firing and η_1 and η_2 are the efficiencies of gas cycle and steam cycle separately for the given parameters for each cycle.

If $Q_2 = 0$, then

$$\eta = \eta_2 + \eta_1(1-\eta_2)$$

This is the efficiency of a combined cycle without secondary firing.

(3) The third type arrangement is shown in Fig. 25.9. This type of cogeneration is most economically used in process industries such as chemical, steel, paper and petroleum refining. Each is a consumer of high level of energy, usually on a round-the-clock basis and all require large quantities of steam and electric power.

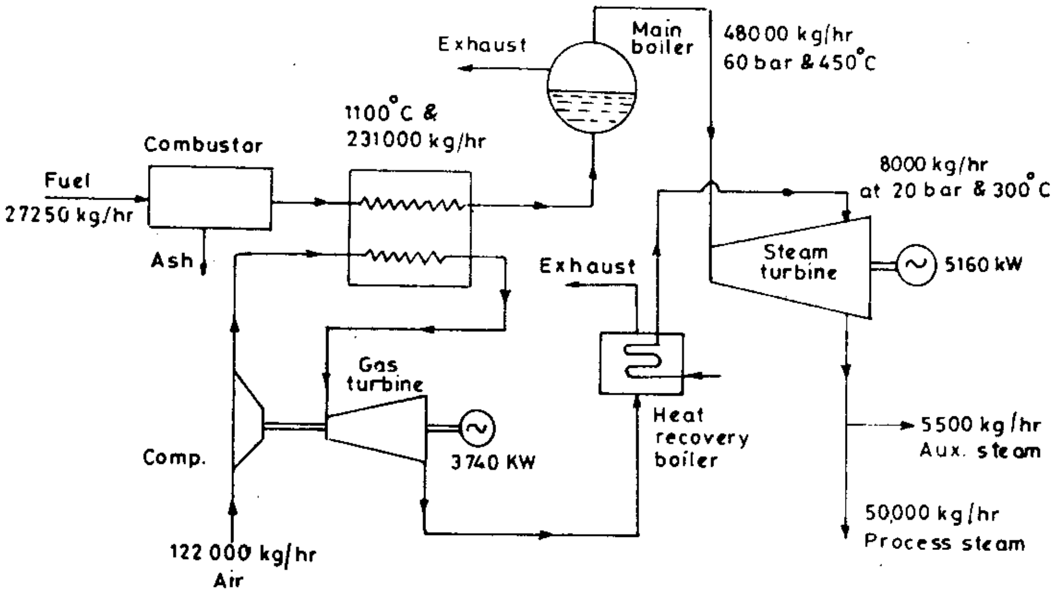


Fig. 25.9.

This arrangement has a considerable advantage of not needing a steam condensing system with its attendant costs and requirements to minimize the impact of heated discharges to the environment. Also, the turbine acts as a pressure reducing device.

25.3. REPOWERING SYSTEM

Before the advent of large capacity power stations, there were many built with steam turbine units of 30, 40 and 60 MW size throughout the world which are still in excellent conditions. For cycle efficiency reasons, they are operated as stand-by units. It is possible to upgrade such situations by adding gas turbines and heat recovery boilers to substitute for the existing boilers and operate the plant in combined cycle. This has been done in America, Europe and Hong Kong.

Repowering of the existing small capacity steam plant conserves energy in the steam plants by adding gas turbine, heat recovery boilers and

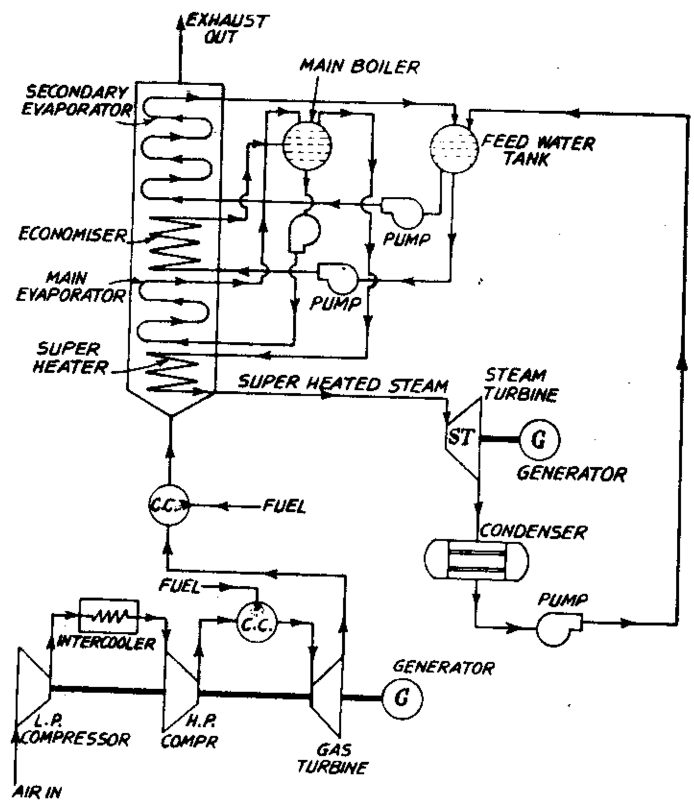


Fig. 25.10. Combined cycle using liquid or solid fuel.

economisers in conjunction with existing steam equipment, Repowering is significant in areas where utilities have been limited by environmental or economic considerations to using oil or gas in steam generators.

Both repeat and non-repeat plants are used for combined cycle repowering. The repowering cost can be as Rs. 800 – 1000/kW. The plant efficiency can be improved as much as 40% with a corresponding plant capacity to 100%.

These are mainly two repowering systems commonly used in practice.

1. **Steam Turbine Repowering.** In this system, gas turbine generates power and new heat recovery steam generators replace the old steam generators. The steam output from the new boilers drives existing steam turbine. The arrangement of the system is shown in Fig. 25.11.

2. **Boiler Repowering.** In this arrangement, gas turbine generates power and acts as a forced drought fan for the existing boiler, supplying hot exhaust gases as combustion air. Increased thermal efficiency is achieved when an economiser is installed instead of regenerative feed heaters. The arrangement is shown in Fig. 25.12.

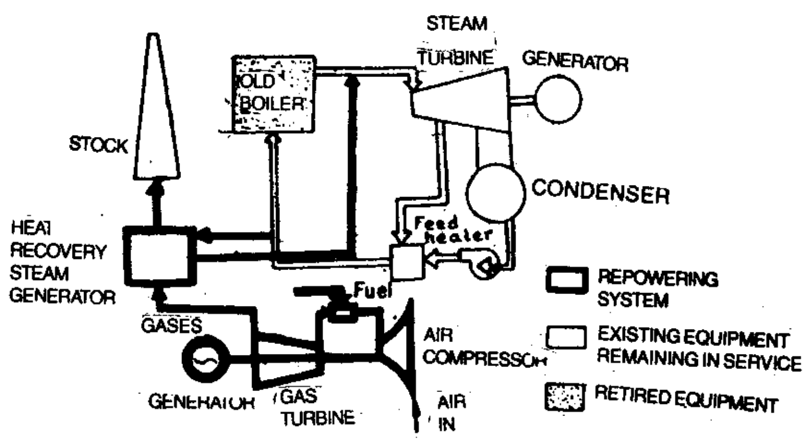


Fig. 25.11. Steam Turbine Repowering.
(Old steam generator is replaced by gas turbine plant and heat recovery steam generator).

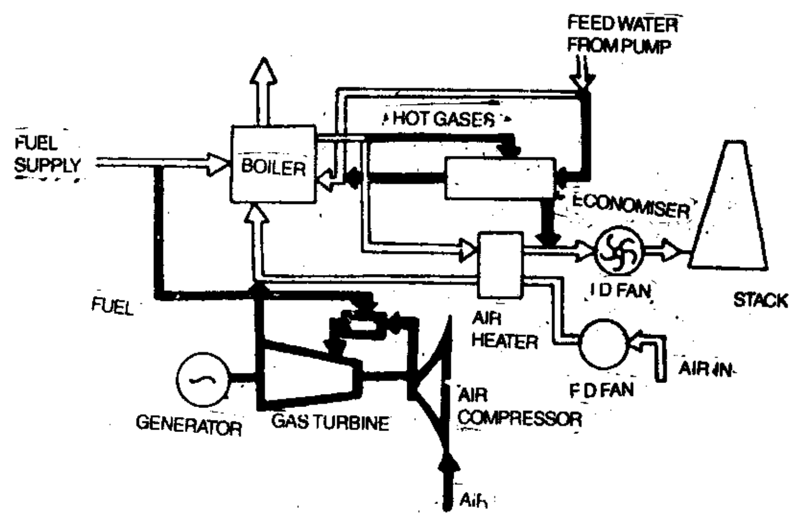


Fig. 25.12. Boiler Repowering.
(Replacement of FD-fan and Air-heater and addition of Gas Turbine plant and Economiser).

In both the cases, substantial boiler modifications are necessary to accommodate the exhaust gases.

Repowering can also be used to pulverized coal-fired plants but it poses some other difficulties, when exhaust gases from gas turbine are used as a source of O_2 in the pulverized boiler, the flow (velocity) of exhaust gases through the boiler is increased. Pulverized coal boiler basically uses less O_2 to reduce flyash erosion, therefore, increased exhaust gas flow through the existing boiler is not possible.

Advantages of Repowering Cycle. Repowering offers a means of increasing capacity and efficiency of the existing power stations in the cheapest possible way. Every 30 MW existing steam set can be upgraded to 111 MW by addition of gas turbine. There are 200 such sets of 10,000 MW capacity presently in U.K. It is thought by the Government to repower few of them instead of building new nuclear plants which the public does not want.

The advantages of repowering system in addition to the above are listed below :

* Use is made of existing sites and civil works such as water-cooling channels, cooling towers, buildings and offices.
* The requirement of cooling water is changed little even though the output is trebled.
* The construction of the combined cycle can be completed just within a couple of years.
* Fuel used can be gas, distillate or blended residual. Extra handling of coal can be eliminated which reduces maintenance cost considerably.
* Automatic control of gas turbine helps for peak load requirement and operating staff can be reduced to one-third of requirements for a coal-fired station.
* The smoke and grit emission are reduced considerably which maintain the environment more clean.

Belle Isle station (U.S.A.) is the first repowering installation in the power industry commissioned in 1949. A 2 × 4 MW gas turbines were added as the boiler was unable to supply required quantity of steam to the existing steam turbine. Plant capacity was increased by 9 MW while the plant heat rate was improved by 2.5%.

Another notable repowering unit is at Hong Kong, where 5 × 21 MW gas turbine units and a heat recovery boiler capable of generating 36 tons of steam per hour were added in the original plant capacity of 10 MW.

Reverton Station of Empire District Electric Co., Riverbank station of duke power Co, Long Beach Generating Station of Southern California Edison Co., are some of the notable repowered combined cycle plants in U.S.A.

There are 44 combined cycle plants presently working in USA with total capacity of 6890 MW with a largest unit of 600 MW and smallest of 12 MW.

25.4. COMBINED CYCLE WITH GAS PRODUCTION FROM COAL

Electricity generation, like all forms of energy conversion, is being affected by the scarcity and escalating costs of petroleum fuels and by added costs of protecting the environment against pollution. To overcome this difficulty, it has become very necessary to use the coal as fuel instead of petroleum as coal resources are sufficiently large. The demand for power is continuously increasing whereas the supply of natural gas and oil is continuously decreasing. The scientists are presently trying to develop improved process to permit the use of coal as an environmentally acceptable fuel.

Coal is probably the most abundant fossil energy of the world. At present, consumption rates already known, reserves alone will last for several hundred years. Another great advantage of coal is that, reserves are more evenly spread over the world than oil and gas. Thus, many countries on five continents possess large reserves which they are also prepared to offer on the world market. This leads to relatively stable coal prices with positive effects on the economics of the exporting and importing countries. By using indigenous coal, the spending of hard currency for energy imports is avoided and local employment is provided. But coal is also burdened with disadvantages as it contains a number of undesired constituents such as ash and sulphur. By applying gasification, coal can continue to play an important role and at the same time meet present and future more stringent environmental regulations.

Especially in Electric power plant, the unique situation arises that emissions can be reduced significantly at no extra or even lower cost of electricity. The key to this is coal gasification. It allows to exploit the great potential of combined cycle plants.

Gasification offers the inherent advantages that not only can heat be generated from coal but as well feed gas for a wide variety of syntheses for chemicals. At the same time, the gas can be used as fuel for gas turbines.

Another advantage of coal gasification is that the gas is purified at elevated pressure and prior to combustion as in electric power generation in combined cycle plant. This leads to much reduced volumes (down to 1%) when compared to conventional power station flue gases. Using purified gas as fuel, environmental protection can thus be pushed to its ultimate limits without economic penalty.

If the gas turbine power plant is to continue the transition from a low capacity factor (0.1—0.2) peaking device to intermediate or base load plant, it must sustain its capability to burn non-petroleum or synfuels (fuels produced from coal) derived from available coal reserves.

To produce an appreciable impact on electric power generating efficiency, combined cycle plants would have to be installed to serve an increasing share of base load. If coal derived fuels are to be used in gas turbine plant, the combined cycle efficiency must be very high to compensate for the losses in the fuel conversion process and still provide overall thermal efficiency higher than that available from a high pressure steam cycle with stack gas clean up.

To reach this goal, two parallel developments are necessary. One is an efficient fuel conversion process that will handle a wide range of coals, including high and low sulphur contents and with heavy caking and smelling characteristics and other is a high temperature gas turbine that can use coal derived fuels.

Integrated gasification combined cycle plant represents a proper step in this direction. The cost of such plant is one of the major considerations in its development but its cost would not be appreciably different than that of a large supercritical steam plant with scrubber.

The possible fuels in future for combined cycle plants are gasified and liquefied coals. Several liquification plants are underway presently in USA and it will expand in future. But gasification still has the greatest emphasis as its technology is more simple than liquification.

Many gasification devices as moving bed, or known as Lurgi technology, FBC gasifier and entrained flow gasifier have been proposed on a very large scale recently in USA.

Coal gasification is the first step towards the transformation of coal into useful gas turbine fuel. Large amounts of O_2 and therefore large amount of air is required to operate the coal gasifier. Compressors are necessary to deal with air, O_2 and N_2 as a by-product.

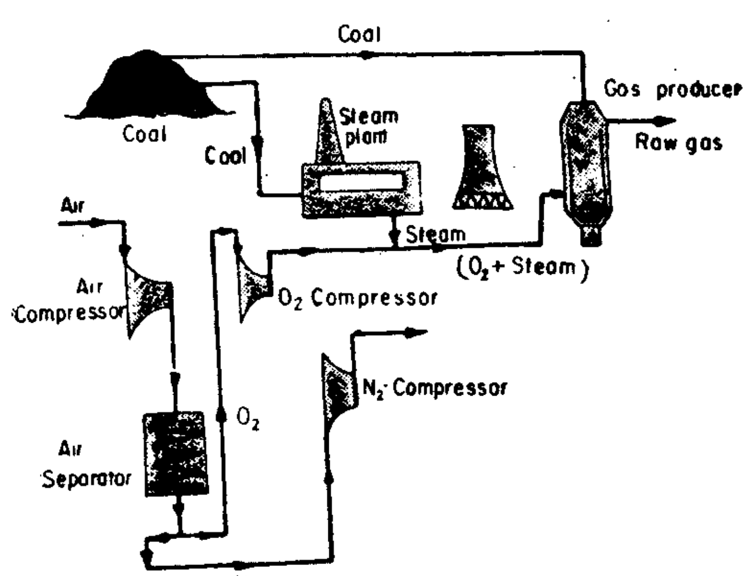


Fig. 25.13. Coal Gasifier—Gas contains 25 – 45% H_2, 20 – 55% CO and 10 – 30% CO_2 and 0 – 20% CH_4

Note—Coal is efficiently gasified in the presence of high pressure steam and oxygen.

A typical gas-producing plant using coal as fuel is shown in Fig. 25.13. Oxygen is produced by separating from air and then compressed oxygen and steam from thermal plant is supplied to the gasifier. The power required to run air and oxygen compressors is supplied from the thermal power plant. Presently, air compressors in capacity range of 35 to 600 thousand m^3/hr are available for the purpose.

The energy in the gas coming out of gasifier is 78% of the energy in the coal.

There are mainly two types of gasifiers, one using air and other O_2 separated from air. O_2 blown gasifier gives higher gas yield and consumes 7% less compressor power compared with air blown gasifier. But the simplification of the air blown system due to omission of the air separation plant, specific capital cost of air blown system (Rs./kW) would be lower than oxygen blown system. Therefore, presently, air blown gasifiers appear to offer the preferred choice for coal gasification combined cycles with the potential for better efficiency and lower capital cost. It is estimated that integrated system with air gasifier is about 4% more efficient than integrated plant with oxygen blown gasifier and can achieve a coal to electricity with an efficiency of about 32% of a typical fossil plant using scrubbers.

The presently known arrangement of power plants with coal gasification shows that the gasification and power-producing portions are integrated with interchange of electric power, compressed air, steam and gas fuel between the systems. The benefits of integration are improved efficiency, reduced capital cost and operational simplicity.

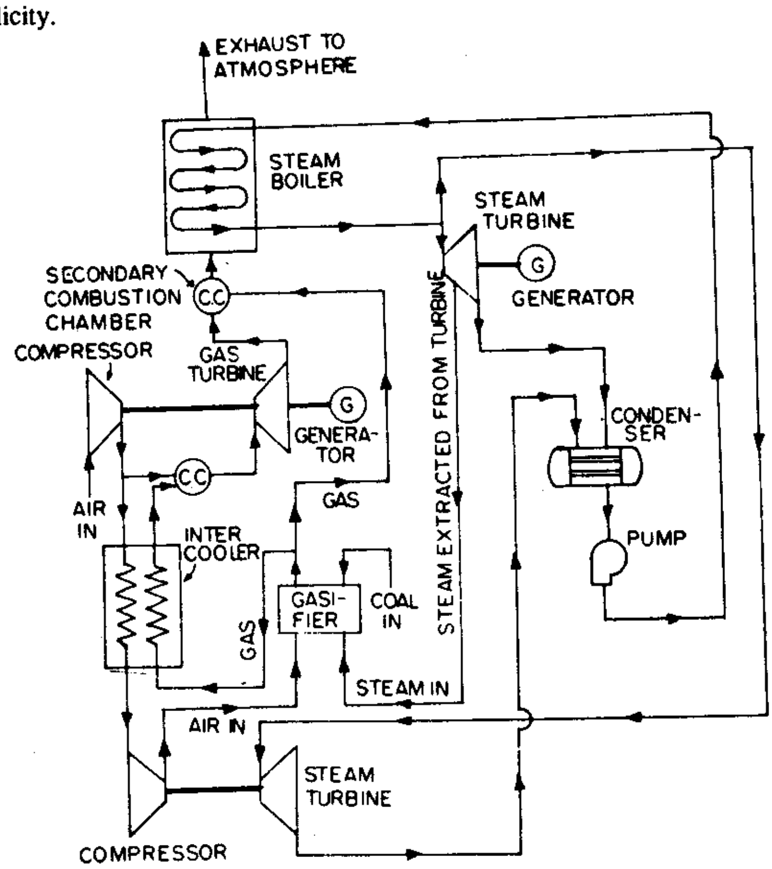


Fig. 25.14. Combined cycle using producer gas as fuel where producer gas is generated by using steam and compressed air from the plant and coal from outside.

Two typical arrangements using heat and O_2 in the exhaust gases of gas turbine plant for generating the steam and the plants are inter-connected for gasification of coal are shown in Fig. 25.14 and Fig 25.15. This type of plant is currently planned for the cool-water station of Southern California Edison Co.

The performance of gasification and power generation plants of a combined system cannot be separately expressed. Combined coal to electricity is about 38%. The integration of such plants has reduced the coal consumption by 11%. It is estimated that a 1000 MW plant can save about seven million dollars per year if integrated plant is used instead of steam plant of the same capacity. Integrated plant is more economical compared with separate gas and power plants even though it costs $ 39 million more for 1000 MW plant as it produces lower cost electrical power.

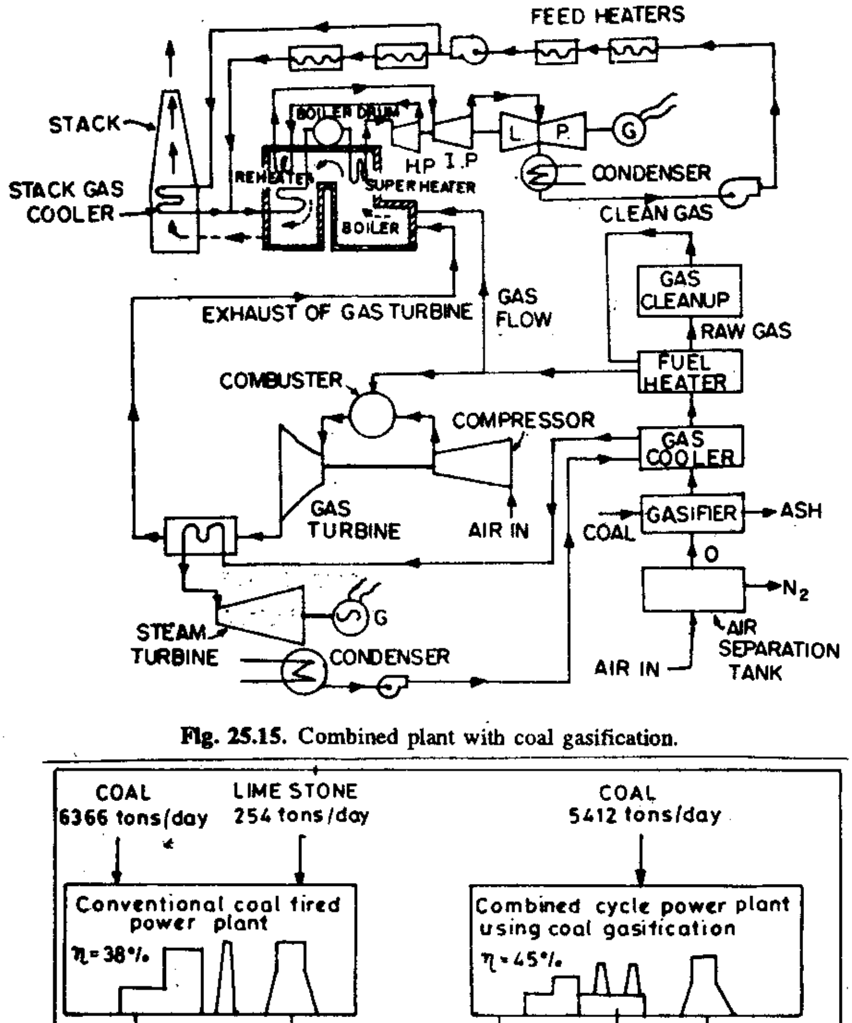


Fig. 25.15. Combined plant with coal gasification.

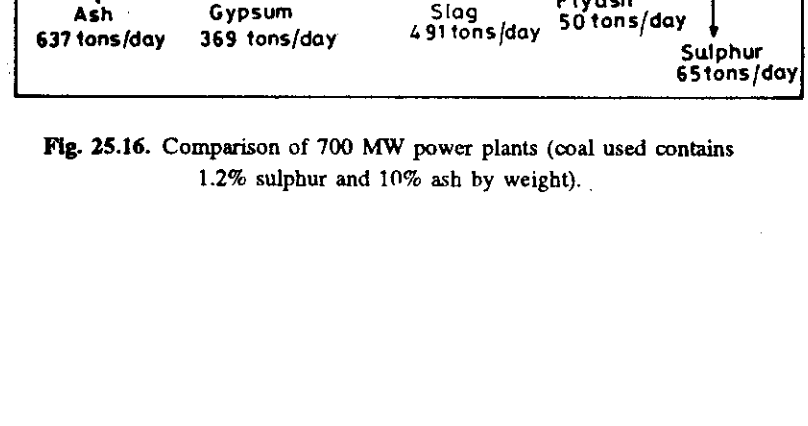


Fig. 25.16. Comparison of 700 MW power plants (coal used contains 1.2% sulphur and 10% ash by weight).

Additional to the financial advantages of the integration system there are sociological and environmental advantages of conservation of energy and reduced emissions. It offers the potential for removing more sulphur coal at less cost than is possible with the type of scrubbers used in power plants today—by using the proven stretford process. Commercial scale coal gasification combined cycle plants have been successfully operated in North America.

Fig. 25.16 shows the comparison between coal fired power plant combined cycle power plant using coal gasification as Inputs and exhausts are concerned.

25.5. COMBINED CYCLE USING PFBC-SYSTEM

Pressurised fluidized bed combustion (PFBC) system is another successful approach for using coal as a fuel for running gas turbine plant and using exhaust heat of gas turbine to run a steam power plant as discussed earlier. PFBC combined cycle system can boost overall conversion efficiency by 5% over a conventional pulverised coal-fired plant equipped with wet scrubber. PFBC can burn any type of coal (any % of sulphur and ash content) and reduces 95% of SO_2 and NO_x emissions well below all required standards.

PFBC enjoys all the advantages of conventional FBC systems discussed earlier. But it is the elevated pressure of PFBC that is the source of additional benefits more important to large scale utility units. This feature allows a PFBC unit to be integrated into a combined cycle power plant. Because, once part of the combustion heat has been transferred to a fluid, the combustion gas is still at a sufficient pressure to power a gas turbine and generate additional power.

The world's first PFBC-combined cycle plant of 135 MW electric power and 225 MW equivalent to district heating with overall efficiency of 90% is operating at Vartan in Stockholm, Sweden.

There are two basic systems used with PFBC.

(1) **Air-Heating System.** Curtiss Wright have developed such a system. In this system, coal is burned directly in a PFBC and the air passing through the tubes in the bed at high pressure is heated to high temperature (1000°C). The air stream from the compressor splits into two parts with 65% going through the tubes and 35% through the bed for combustion purposes. Only the combustion gases have to pass through hot gas clean-up process, so that the hot air that went through the tubes bypasses this process. The recombined flow of hot air and hot gases then passes through the gas turbine and then through the waste heat boiler. In this cycle, about 70% of the power output is provided by the gas turbine and 30% by the steam turbine generator. The overall efficiency of this system is about 40-41%. The arrangement of the system is shown in Fig. 25.17.

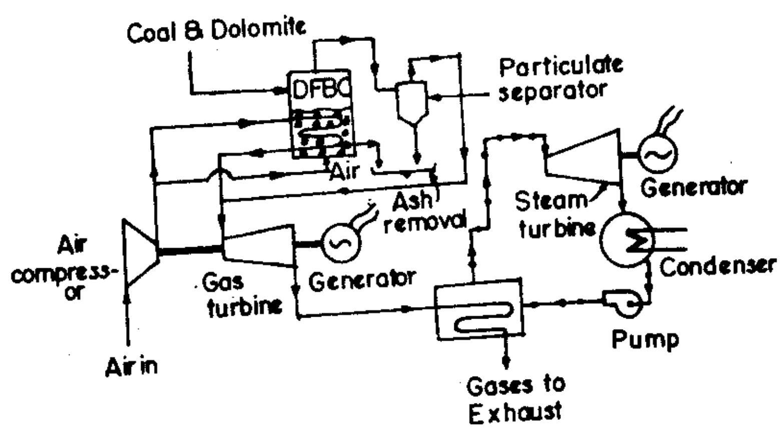


Fig. 25.17. Air-cycle System.

(2) **Steam Generating System.** General Electrical Co. has developed this type of system. In this system, the generation mix would be just opposite that in the Curtiss-Wright design. The combustion gases, from

PFBC, would drive the gas turbine while steam produced in the boiler in the PFBC drives the steam turbine. 30% of the total power of the combined system is generated by gas turbine and 70% by the steam turbine. The arrangement of the system is shown in Fig. 25.18.

An outstanding feature of PBFC system, pressured operation reduces the space required for combustion vessel since volume is reduced in proportionate to the operating pressure. A bed area for atmospheric unit is 2 m^2/MW at fluidizing velocity of 2 m/sec whereas for pressurized unit requires 0.2 m^2/ MW at 10 bar.

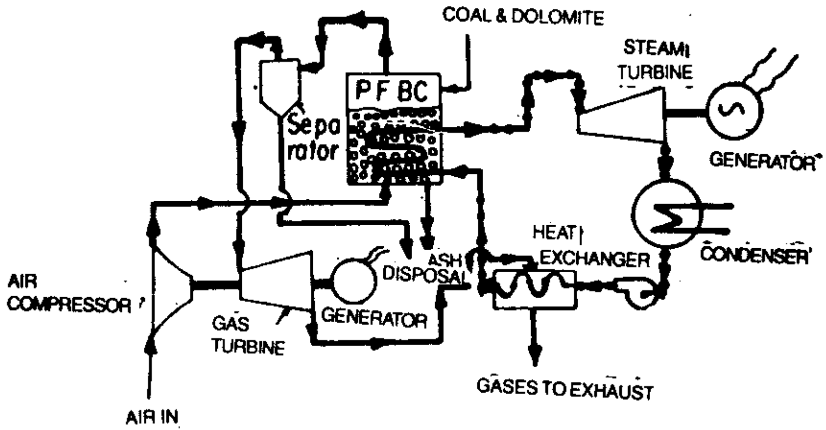


Fig. 25.18. Steam cycle system.

The detailed constructructional features of PBFC used in both the systems are shown in Fig. 25.19.

Steam Versus Air Cycle. The steam cycle offers the following advantages :

(1) It requires upto tenfold less area of tubes in the bed to achieve the same heat transfer. Therefore, it also needs fewer tubes and smaller combustion shell on kW-basis.

(2) The steam cycle offers slightly higher (1%) net efficiency than the air cycle.

(3) A steam cooled, PFBC plant uses about 30% of the air required by an airoperated plant for an equivalent output, although the combustion air is same in each case. Thus the gas turbine will be larger in size in case of the air-cycle.

The advantages of air cycle are listed below :

(1) The pressure difference between bed side and air side is almost zero. Thus tube crack would not result.

(2) Reduced water consumption as steam power generation is less.

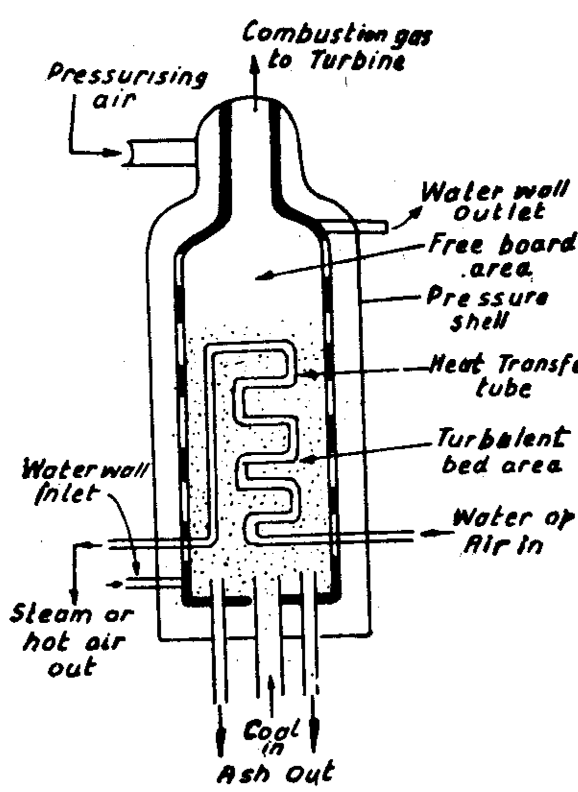


Fig. 25.19. The components of PBFC.

(3) The combustor is air-cooled and independent of the steam cycle, so the gas turbines can be operated on their own.

(4) Air cycle system is simple to control as diverting cool air from the bed heat exchanger inlet to the gas turbine inlet lowers the temperature and reduces the turbine load. One limitation of this system is gas turbine pressure ratio. With high pressure ratio (about 8), the turbine exhaust temperature is too low to produce the steam of desired quality.

Technical Hurdles in the Development of PFBC. The maximum temperature in PFBC is limited to 1000°C which gives maximum efficiency of the combined cycle of 42%. This temperature is limited by the ash softening temperature, sulphur capture efficiency and NO_x formation point. In addition to this, the system must be protested from alkali volatization as they are highly corrosive. This problem is avoided by maintaining the temperature in the bed below 900°C that is below volatization temperature of all alkalis.

The major hurdle faced by the system is the cleaning of the gas at elevated temperature to extend the life of the turbine. The already developed highly efficient combined cycles accept clean fuels but cannot tolerate high contaminant level of gas and maintain reasonable parts-lives. If the present designs are to be used, product gas clean up technique must be developed that reduces corrosive and erosive constituents to the level found in today's petroleum based distillate oils and natural gas.

The present gas clean up systems are developed upto 100°C but gas clean-up at high temperature is most difficult problem faced by the engineers. The equipment cost, to clean up the gas at high temperature, is considerably high and there are considerable heat losses during clean up process which reduces overall efficiency of the system.

The life of the turbine blades depends upon particulate loading in the gas in addition to the temperature of the gas. The erosive action on blades depends upon particulate loading, its composition, and their relative velocities with blade. The corrosive action on blades depends upon the presence of vanadium and alkali compounds and metallic vapour in the gas.

Present acceptable contamination levels are 0.029 scf for particulate matter and 1 ppm vanadium and 1 ppm for all alkali compounds (sodium, potassium and lead). At present, there is no full scale combustion gas clean up system that reduces the contaminant to acceptable levels. The hot particulate clean up is in its infancy and is generally limited to cyclones in the 800-1100°C temperature range, although the sophisticated granular bed filter (GBF) technique and augmented cyclone have exhibited promise in the removal of fine particles less than 5μ, acceptable for the present gas turbines.

The system will be more viable to use coal, peat, waste wood municipal; waste, sewage sludge and garbage as future fuels provided efficient and economical methods are developed to remove particulates to the acceptable level at high temperatures.

Particulate Control in PFBC by Laser. It has been already mentioned that direct coal-fired gas turbines need efficient high temperature particulate control to be an attractive option for efficient conversion of coal to electrical energy. The particulates in the range of 3 to 10 μm are difficult to remove. Below the size of 10 μm bagfillers or ESP might be applied. But the feasibility of these methods for particulate removal at high temperature is not yet established.

A recent method of using Laser to control the size of particulate in the gas before the entry in the turbine is under study. It is proposed that the particulates < 10 μm in the gas which are not removed by conventional methods will be fragmented into a smaller size by the bombardment of Laser which will be below the threshold size for turbine blade erosion.

Fragmentation can be accomplished by shock heating the particulates. Rapid energy deposition in order to shock heat may be accomplished with a pulse CO_2 Laser beam. CO_2 lasers are considered more feasible as CO_2 laser providing greater than 1·kJ/pulse are commercially available.

A PFBC combined cycle system of 1000 MW capacity using Laser ash (LASH) fragmentation method for particulate removal is shown in Fig. 25.20. It is also estimated that the LASH system for 1000 MW capacity plant will cost $30 million as the present Laser cost is $200/J. Nearly 99% of the energy supplied to LASH can be again recovered in the form of heat as shown in Fig. 25.20.

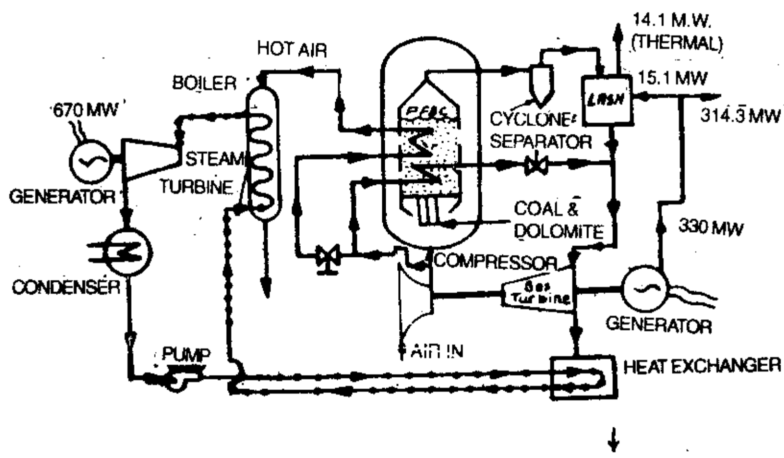


Fig. 25.20. Combined cycle plant of 1000 MW capacity with LASH particulate control.

25.6. COMBINED CYCLE WITH ORGANIC FLUIDS

A combined cycle is a synergistic combination of cycle operating at different temperatures, each can operate independently. The heat rejected by the higher temperature cycle is recovered and used by a lower temperature cycle to produce additional power and gives higher overall efficiency. Combined cycles which have been proposed are gas-steam, steam-organic fluid and gas organic fluid. Gas-steam cycles area already discussed and steam-organic fluid is discussed in the chapter on waste heat utilization. In this article, gas-organic fluid will be discussed. Under certain circumstances, gas-organic fluid works more efficiently and for greater period than gas-steam cycle.

The higher temperature cycle is known as topping cycle whereas lower temperature cycle is known as bottoming cycle. The topping cycles include Diesel, Brighten and Rankine but bottoming cycle is basically only Rankine cycle.

All commercial bottoming cycles use water as working fluid as steam is less costly, chemically stable, has high specific heat and high heat transfer rates. The disadvantages of the steam include **high latent heat and high critical pressure.*

The amount of sensible heat which can be re-utilised from the exhaust gas of gas turbine plant depends on the temperature and mass flow rate of the exhaust gases and allowable stack temperature which is generally fixed on the basis of technological and economical requirements. The stack gas temperature should always be higher than their dew point temperature to avoid the corrosion of the heat exchanger and stacks. The large excess air used in gas turbine plant results in a small partial pressure of sulphurated compounds which leads to reduced dew point temperature of the exhaust gases.

The thermodynamic difficulties encountered in utilizing steam as working fluid in the bottoming cycle are represented in Fig. 25.21. The temperature of exhaust gases from the turbine is say 350°C, steam at 10 bar with 300°C temperature is generated by cooling the gases to 110°C which is the limit set by dew point temperature of the exhaust gases. If the generated steam is used for power generation in steam turbine, it adds 4.9 MW in the original 15 MW generated by gas turbine. In this case, only 70% of the heat in the exhaust gases is used and gives only 16% efficiency.

If organic fluid methylene chloride is used in the bottoming cycle instead of steam as working fluid, the vapour at 25 bar and 220°C can be generated utilizing all the thermal energy available above 110°C. This can develop the power 7.5 MW instead of 4.9 MW with steam cycle giving 24.5% efficiency.

*See the Chapter on waste heat utilization.

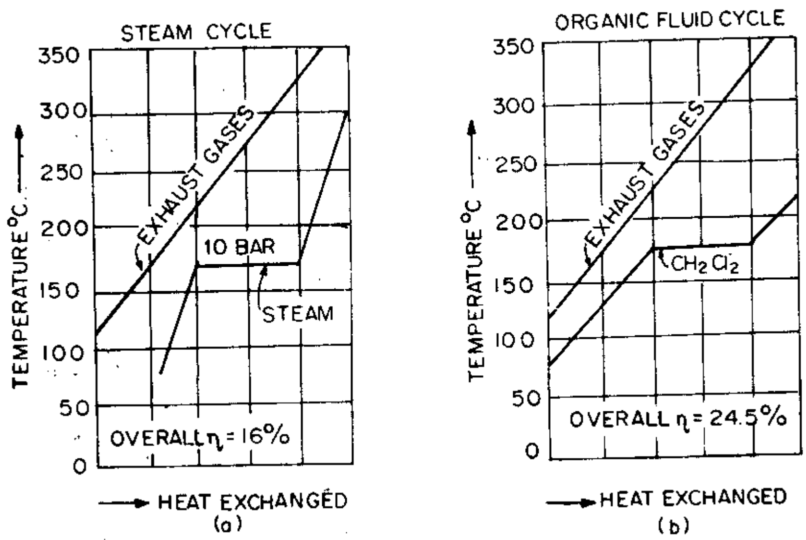


Fig. 25.21. Heat recovery from low temperature exhaust gas by steam cycle and organic fluid cycle.

It is obvious from the above illustration that when the maximum gas temperature and dew point temperature, both are low, steam exhibits thermodynamic instability to extract heat from the exhaust gases. Therefore, under such condition, heat recovery by means of organic fluid cycle becomes more efficient. But organic fluids, owing to their limited thermal stability, can utilize efficiently only low temperature heat. When the exhaust gas temperature is high, the better use of the thermal energy can be made by first using the heat to heat the compressed air with the help of recuperator to reduce the gas temperature and then by subsequent recovery of the sensible heat by the organic fluid as mentioned above. The combined cycle arrangement using air preheater and exhaust heat recovery cycle using organic fluid are shown in Fig. 25.22 (*a*) and Fig. 25.22 (*b*).

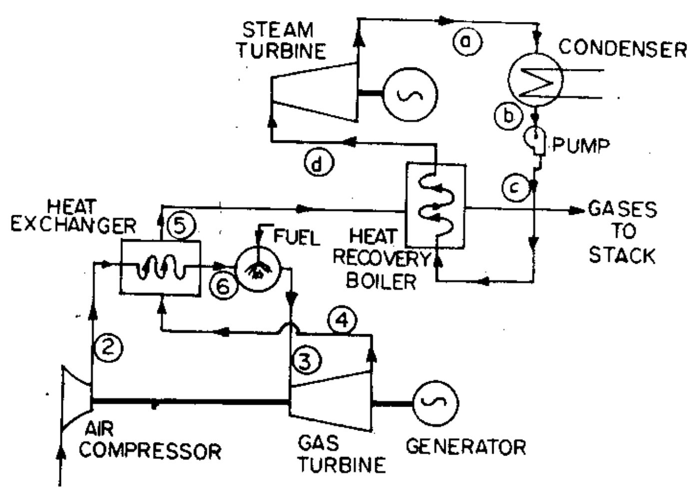


Fig. 25.22. (*a*) Open cycle gas turbine plant with air-preheater and heat recovery cycle using organic fluid.

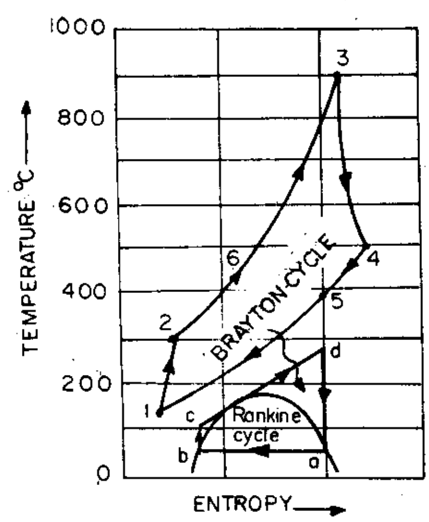


Fig. 25.22. (*b*) Representation of cycles on T-s diagram.

Closed Cycle Gas Turbine with combined Cycle. The following characteristics of a closed cycle gas turbine play an important roal in the design and performance of power recovery equipment.

(1) If a better thermodynamic fluid like He is used in a closed cycle gas turbine plant which has better heat transfer properties, a compact heat exchanger with higher regenerator effectiveness can be achieved in practice. For this reason, regenerative preheating of the working fluid is preferable even with heat recovery cycle.

(2) The sensible heat of the working gas (He) at the turbine exhaust can be utilized to a temperature much lower than that which is dictated by dew point temperature in open cycle more easy.

(3) In addition to this, an organic fluid condenser makes the problem of heat rejection in the bottoming cycle more easy.

(4) The combined cycle capacity of lowering the gas temperature in the primary is of particular value in the case of closed cycles, as the design of the high temperature heater, which is most costly component of the power plant, is favourably affected by the reduction of the gas inlet temperature.

25.7. CONSIDERATION FOR BEST MIX

The next problem in the design of combined cycle is the proportions of the total output of the plant to be supplied by the gas and steam turbines.

One designer claims that only 20% of the total station output should be supplied by gas turbine and the remaining by steam turbine for maximum operating flexibility in serving middle range loads. This would require additional fuel to be used in the waste heat boiler for generating large amount of steam needed in the steam portion of the plant. This distribution could possibly reduce the generating cost due to reduced maintenance cost associated with gas turbine plants.

Another designer claims that as much as 80% of the total energy must be supplied by the gas turbine and remaining by the steam turbine. This distribution of generation does not require any additional fuel be burned in the boiler and any output from the steam portion of the plant has zero fuel cost.

In a combined plant, the total energy rejected to the environment is the energy carried by the exhaust gases and energy given to the cooling water in the condenser. The efficiency of the combined cycle would be maximum when the total heat rejected by the combined plant to the environment is minimum.

At this point, the heat rate per kW-hr. energy generated would be minimum. Fig. 25.23 (*a*) shows the variation of heat rate per kW-hr. energy generated versus the percentage of total energy generated by the gas turbine plant. Fig. 25.23 (*b*) shows the variation of percentage reduction in total heat rejected by the combined plant to the environment versus the percentage of total energy generated by the gas turbine plant.

It is obvious from Fig. 25.23 (*a*) that the steam turbine plant would operate independently of the gas turbine plant with net heat rate of 2700 kcal/kW-hr. The heat rate per kW-hr. drops rapidly with the increasing gas turbine usage till the gas turbine power is 45% of total*. This reduction is caused due to the recovery of heat from the gases coming out from gas turbine plant. At this point, the ratio of heat supplied in the waste heat boiler to the heat supplied in the gas turbine combustion chamber per kW-hr energy generated is maximum. As the percentage of power developed by turbine increases above 45%, the ratio mentioned earlier decreases. The fuel supplied in the waste heat boiler per kW-hr. energy generated also decreases. A point reaches, where steam-generators approach unfired waste heat recovery boilers. This happens when the power generated by the gas turbine plant is 77.5% of the total. Between the points of maximum boiler firing and no boiler firing, the heat rate passes through a minimum value and this is the point of minimum heat rate or maximum economic running of the combined power plant. It is not possible to run the combined

*Percentage reduction in rejected energy = $[E_1 - E_2/(E_1)] \times 100$ where E_1 is the energy rejected by the steam plant when the required power is only supplied by steam plant and E_2 is the energy rejected by the combined plant when the required power is supplied by the combined plant. This ratio becomes negative when the power supplied by the gas turbine exceeds 75% of the total power.

power plant at this point as the percentage power to be developed by the gas turbine plant depends upon the peak load requirements. This point also changes with the change of load on the power plant.

25.8. OPTIMUM DESIGN OF GAS TURBINE UNIT FOR COMBINED CYCLE PLANT

The performance of a combined cycle plant is also affected by the pressure ratio of the gas turbine and the present analysis aims to find out the optimum pressure ratio for the most economical running of the plant.

The best thermal efficiency in a single stage gas turbine process without air-preheater is obtained with the following compressor pressure ratio :

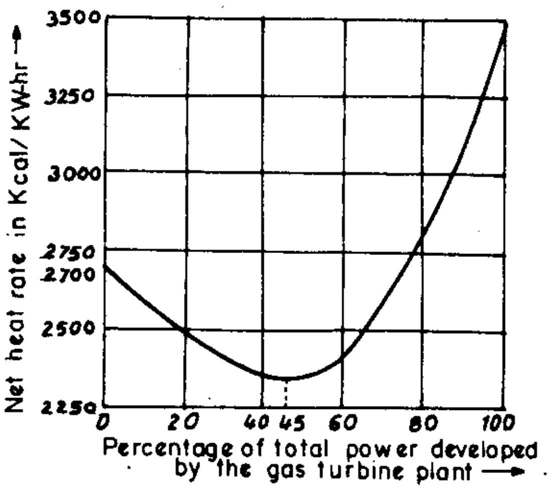


Fig. 25.23. (*a*)

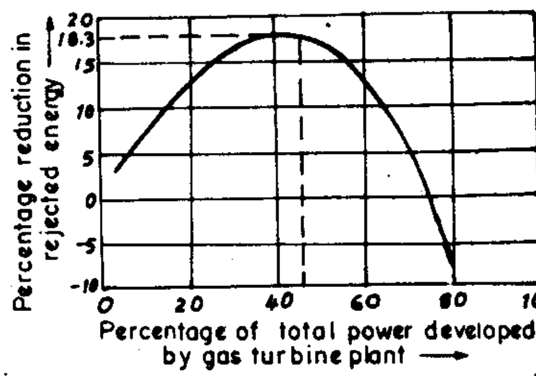


Fig. 25.23 (*b*)

$$\left(\frac{p_2}{p_1}\right)_{opt.}^{(a+b)} = \frac{T_3}{T_1}\,\frac{\eta_e\,\eta_t}{(1-\eta_{thg})} \qquad ...(a)$$

The compressor pressure ratio for the maximum output is given by an expression

$$\left(\frac{p_2}{p_1}\right)_{opt.}^{(a+b)} = \frac{T_3}{T_1}\,\eta_c\,\eta_t \qquad ...(b)$$

With a gas turbine cycle using only waste heat boiler, the optimum pressure ratio for the combined processes is found from :

$$\left(\frac{p_2}{p_1}\right)_{opt.}^{(a+b)} = \frac{T_3}{T_1}\,\eta_c\,\eta_t\left[\frac{1-\eta_{ths}}{1-\eta_{theom}}\right] \qquad ...(c)$$

If $$\eta_{tgh} = \frac{W_{gt.}}{Q} \qquad ...(d)$$

and $$\eta_{theom} = \frac{W_{gt}\times W_{st.}}{Q} \qquad ...(e)$$

We can write,

$$\eta_{theom} = \eta_{thg}\times\frac{W_{st.}}{Q} = \eta_{thg} + \frac{W_{st}}{1}\times\frac{\eta_{thg}}{W_{gt.}} \qquad ...(f)$$

$$= \eta_{thg}\left(1+\frac{W_s}{W_{gt.}}\right) \qquad ...(g)$$

Substituting the value of η_{theom} from equation (g) into equation (c), we get

$$\left(\frac{p_2}{p_1}\right)_{opt.}^{(a+b)} = \frac{T_3}{T_1}\,\eta_c\,.\,\eta_t\left\{\frac{1-\eta_{ths}}{1-\eta_{thg}\left(1+\dfrac{W_{st.}}{W_{gt.}}\right)}\right\} \qquad ...(h)$$

where T_1 = Air temperature at the inlet of compressor.

T_3 = Temperature of gases at the inlet of turbine.

η_c = Polytropic efficiency of compressor including pressure losses.

η_t = Polytropic efficiency of turbine including pressure losses.

η_{ths} = Thermal efficiency of steam cycle.

η_{thg} = Thermal efficiency of gas cycle.

η_{thcom} = Thermal efficiency of the combined cycle.

Q = Heat input at gas turbine combustion chamber.

W_{st} = Net output of steam turbine.

W_{gt} = Net output of gas turbine.

$a = \dfrac{n-1}{n}\,\dfrac{1}{\eta_c}$ and $b = \dfrac{n-1}{n}\,\eta_t$

n = Index of compression and expansion.

Comparing the equations (a) and (c), it is obvious that the optimum pressure ratio of combined cycle is smaller than a single gas turbine cycle. In a combined cycle, the optimum pressure ratio lies somewhere between that for maximum output and that for optimum thermal efficiency of a purely gas turbine cycle.

25.9. ADVANTAGES OF COMBINED CYCLE

(1) The efficiency of the combined cycle plant is better than simple turbine cycle or steam cycle. With the current state of technology, the overall η of 47% for the overall plant. It is possible to attain 52% efficiency with certain new generation gas turbines.

(2) The capital cost of combined plant with supplementary firing is slightly higher than a simple gas turbine plant and much below those of a classical steam plant of the same power capacity.

(3) The combined plant is more suitable for rapid start and shutdown than steam plant.

(4) The cooling water requirement of a combined cycle is much lower than a pure steam plant having the same output. Because 2/3 power is produced in gas turbine plant where no cooling water is needed. Conventional cooling water system (once through) or recycle system may be adopted. In case of lack of sufficient water at certain sites, dry cooling system can also be adopted.

(5) The combined system offers self-sustaining feature. If, unfortunately, power station is down due to some fault, the gas turbine offers to start the station from the cold condition. No outside power source is required. Gas turbine is always equipped with a diesel engine to start from cold. The power required for running the auxiliaries and diesel set for start up is supplied from the D.C. system of the station.

(6) Nuclear generation and heavy duty gas turbines represent two highly complementary approaches to growing worldwide power generation needs. The two are working from opposite ends of the load duration curve and continue to converse to converge. Therefore, it is clear that the mid-range area requires specialized generation in itself. The large units if used to take variable load, cannot maintain high capacity factor during light load operations and, therefore, running charges are high for peak load demand. The combined cycle is the unique solution for the above-mentioned requirements.

(7) Many utilities throughout the world are planning and installing simple gas turbine units which

will later be converted into combined cycle operation using gasified coal. The two phase development requires short installation time for peaking power plus the future capability for efficient operation for base load generation. Only the burners on gas turbines need to be replaced to accomplish the fuel conversion.

(8) The present trend to increase the thermal efficiency of gas turbine plant is to increase the turbine inlet temperature. Higher turbine inlet temperature reduces the heat rate and fuel cost and generation cost. The present combined-cycle efficiency is 40% and it is anticipated that the thermal efficiency may reach to 50% by the end of 2000's when better turbine materials would be made available.

(9) The environmental standards of many old fossil fuel plants are not acceptable and they are likely to be closed. These can be renovated by replacing the old boiler with a gas turbine unit and heat recovery boiler. With these modifications, exhaust emissions can be reduced and thermal efficiency and generation capacity can be increased. The combined cycle can take advantage of continuing evolutionary advances in gas turbine technology, particularly higher firing temperature and compression ratio. The economic and ecological benefits of fixing premium fuels in combined cycle are so overriding that conventional oil/gas fired power plants are hardly an acceptable option today.

(10) It gives high ratio of power output to occupied ground space.

(11) It provides more flexibility of operation due to multiple units.

(12) Low operating manpower.

(13) Less down time for maintenance.

(14) Whether natural gasified coal is the fuel, combined cycle greatly reduces CO_2 emissions. Also important, the amount of heat rejected to the environment is less than for competing systems, because of the high overall efficiency.

Further development of dry low NO_x combustors for large gas turbines should allow these machines to meet limits as low as 9 ppm. When coal gasification is employed, upto 98% of the sulphur in the coal is recovered. Thus, the SO_2 emissions are an order of magnitude lower than those from conventional coal-fired power plants, even with large flue-gas desulphurization (FED) plants.

(15) Regarding solid waste, the quantity of ash that results from burning coal equals the amount of slag produced from coal gasifiers. However, on a per unit electricity basis, the amount of solid waste from coal gasification combined cycle (CGCC) is less. FED consumes great quantities of limestone and water and produces great amount of sludge or gypsum and waste water. CGCC plants produce relatively small quantities of sulphur which has a higher value than gypsum.

(16) Combined cycle plants minimize visual impact on the environment, too. They have a lower profile than plants with tall furnaces and generally require less land area than conventional plants.

(17) The superior efficiency of the combined cycle plant over conventional reheat cycle not only conserves fossil fuels but slashes emissions and wastes per unit of electricity generated.

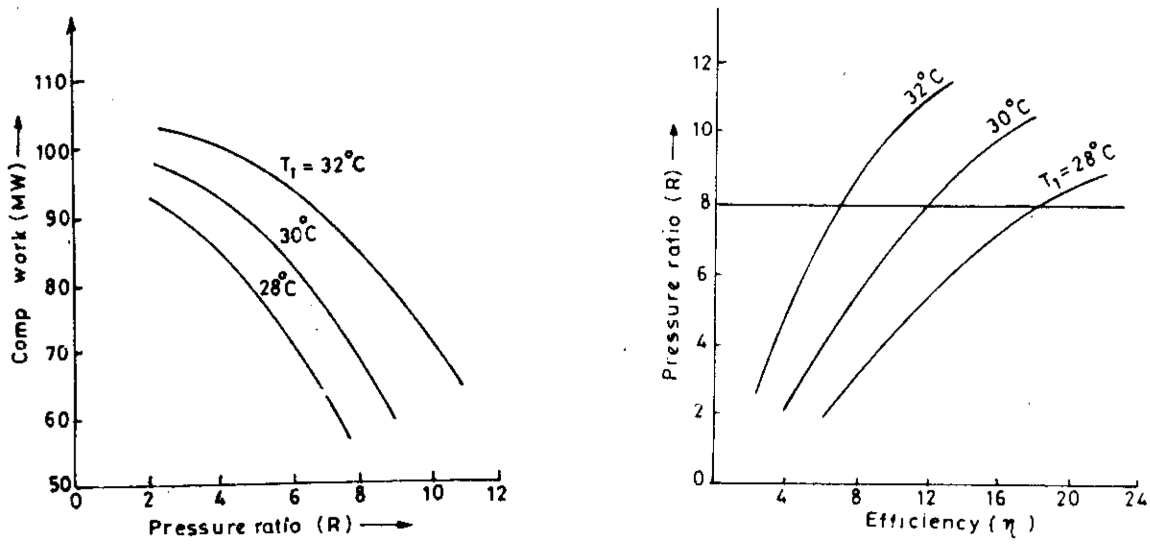
25.10. PARAMETERS AFFECTING THERMODYNAMIC EFFICIENCY OF SIMPLE GT-CYCLE AND COMBINED CYCLE

The most important parameters which affect the efficiency and specific output of the simple gas turbine plant are the compressor pressure ratio and turbine inlet temperatures.

The effects of pressure ratio on the performance of the compressor are shown in Fig 25.24 (*a*) and Fig. 25.24 (*b*) and the effects of the pressure ratio and gas turbine inlet temperature on the performance or the gas turbine are shown in Fig. 25.25 (*a*) and Fig. 25.25 (*b*).

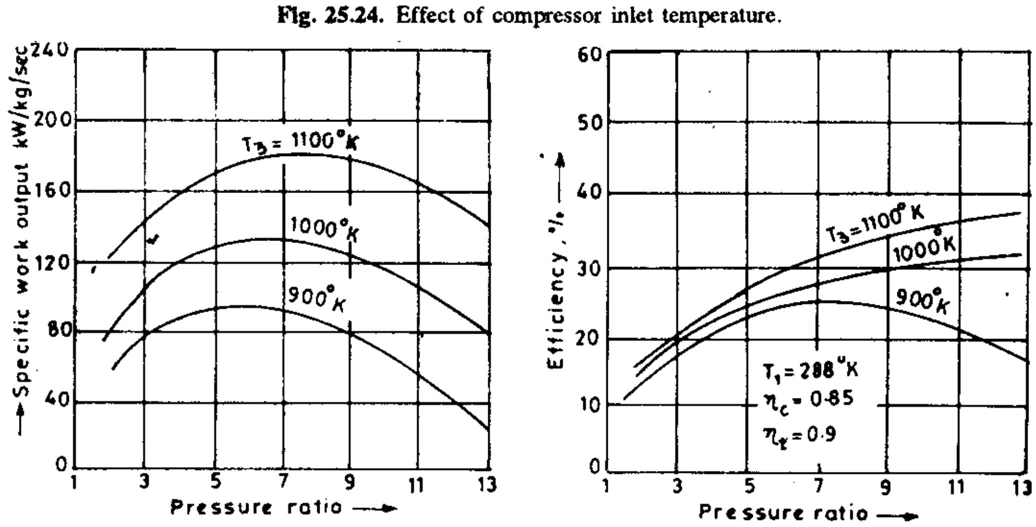
It* is obvious from Fig. 25.24 (*a*), the optimum pressure ratio with respect to specific output is not the optimum one for thermal efficiency. Designers choose a pressure ratio which maximizes overall economy of the plant. Presently, the inlet temperature of the turbine as high as 1300°C with a pressure ratio of 16 are used in modern gas turbine plants. Modern gas turbine components depend upon a combination

*For mathematical analysis, the students are advised to see the book on *Thermal Engineering* by the same author.



(a) (b)

Fig. 25.24. Effect of compressor inlet temperature.



(a) (b)

Fig. 25.25. Effect of turbine inlet temperature.

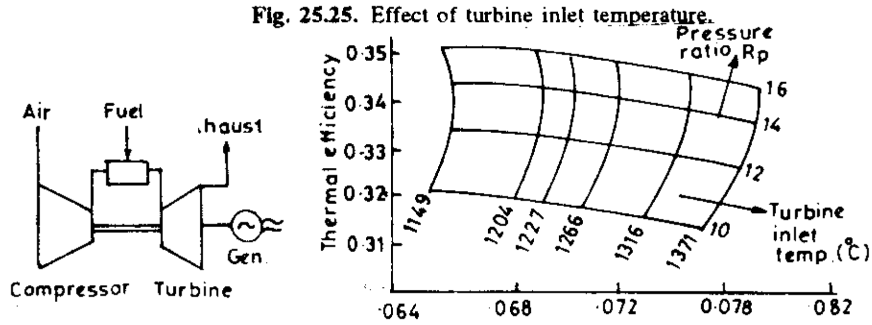


Fig. 25.26. (a) Simple gas turbine cycle and effects of thermodynamics parameters (R_p and T_3) on its performance.

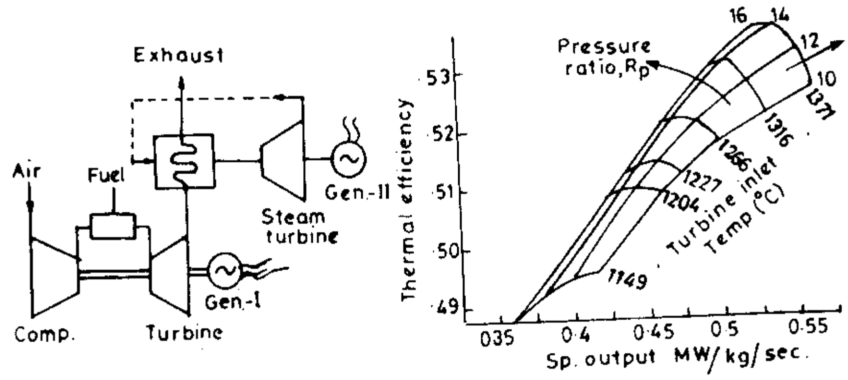


Fig. 25.26. (*b*) Combined cycle plant without secondary fuel and effects of thermodynamic parameters (R_p and T_3) on its performance.

of improved materials, design modifications and new machining techniques to withstand these temperatures. An improved class of super alloys is now permitting the design of metal components that can operate at temperatures only 315°C below their melting point for thousands of hours—even under severe centrifugal, thermal and vibratory stresses provided they are properly cooled.

The effect of pressure ratio and turbine inlet temperature on the performance of single gas turbine cycle and combined cycle are shown in Fig. 25.26 (*a*) and Fig. 25.26 (*b*).

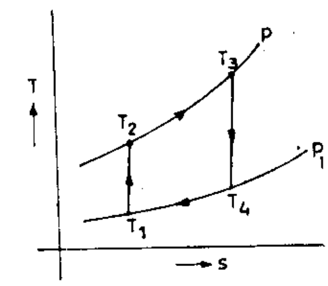


Fig. 25.27.

Performance of combined Cycle Power Plant

(1) **Compressor Inlet Temperature.** The parameters which affect the performance of the combined cycle are compressor inlet temperature, pressure ratio, turbine inlet and exit temperatures and compressor and turbine efficiencies.

Increasing T_1, the compressor work increases but at the same time, heat input is reduced as T_2 increases with an increase in T_1. Work output of the turbine is not affected by T_1. The rate of increase in compressor work is greater than the rate of reduction in heat supply. Therefore, the net effect of increasing T_1 is a decrease in the gas turbine cycle efficiency and that also decreases the overall efficiency of the combined plant.

(2) **Gas Turbine Inlet Temperature.** Increasing T_3 increases the turbine work output and increases the thermal efficiency at a given pressure ratio. But at the same time, heat supply is more with an increase in T_3. The rate of increase in turbine work is higher than the rate of increase in heat supply to the combustion chamber for all pressure ratios. Therefore, increase in T_3 increases the cycle efficiency.

(3) **Pressure Ratio.** With increase in pressure ratio, the efficiency of the cycle increases at a given

T_3. The efficiency increases upto a maximum value and then starts decreasing, indicating that there is optimum pressure ratio for a given value of T_3.

(4) **Gas Turbine Exit Temperature.** The exit temperature of the gas (T_4) which transfers its heat for generating steam in heat recovery boiler decides the efficiency of the steam cycle. Higher the exit gas temperature, higher will be the efficiency of steam cycle as superheat temperature of the steam increases as shown in Fig. 25.29.

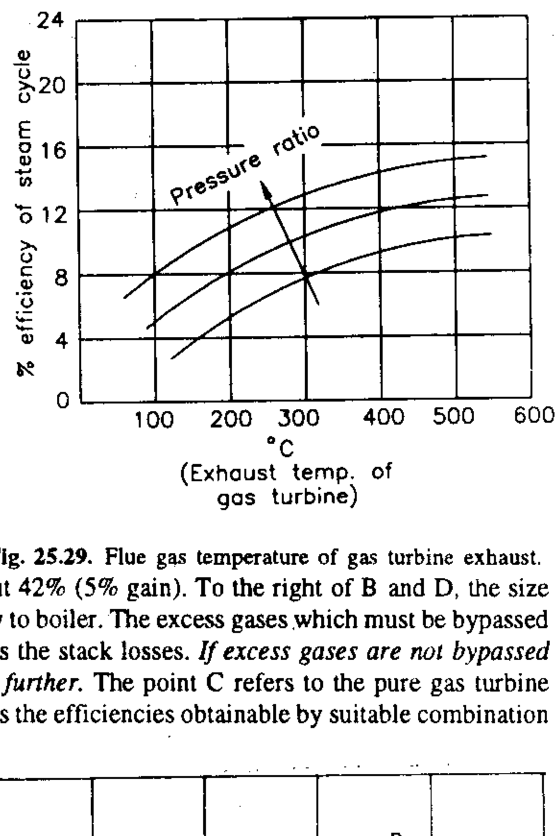


Fig. 25.29. Flue gas temperature of gas turbine exhaust.

(5) **Combined Cycle Efficiency Versus Ratio or Work Output of Gas Turbine and Steam Turbine.** Fig. 25.30 shows the possibilities of selecting a proper combination of both cycles. Point 'A' where the gas turbine output is zero represents pure steam cycle with an efficiency of 37% when steam is supplied at 135 bar and 550°C. Point 'B' is with gas turbine to steam turbine output ratio of about 13% denotes higher combined efficiency of about 42% (5% gain). To the right of B and D, the size of the gas turbine gives a good matching gas turbine flow to boiler. The excess gases which must be bypassed after some point to the right of B otherwise it increases the stack losses. *If excess gases are not bypassed and instead an economizer is used, the efficiency falls further.* The point C refers to the pure gas turbine with an efficiency of 26%. The dotted line BC represents the efficiencies obtainable by suitable combination of gas turbine and steam turbine in parallel.

(6) **Part Load Performance.** Considering the part load performance of a combined cycle with supplementry fuel suply, upto 75% load, the load is controlled by varying the turbine inlet temperature whereas after 75% load, the supplementary firing controls the plants load. Hence load shared by steam plant is instantly increased and gas turbine load nearly remains constant. In pure gas turbine plant, the efficiency goes down as the load is reduced, whereas in combined plant, if the load is reduced from 100% to 60%, the supplementary firing is turned down to reduce the output but the efficiency increases as shown in Fig. 25.30. It shows that the overall efficiency is about 42% throughout the load range from 100% to 60%. The part load behaviour is the principal reason for high economy of combined plant.

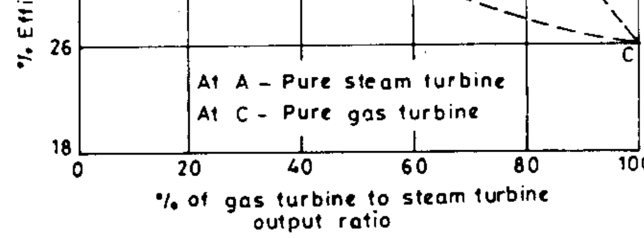


Fig. 25.30. Percentage of gas turbine to steam turbine output ratio.

25.11. PERFORMANCE OF COMBINED CYCLE

The part load behaviour of a combined cycle with supplementary fuel supply as shown in Fig. 25.31 is discussed herewith.

Figure 25.32 explains the part load behaviour of the plant. Up to 75% load, the load is controlled by varying the turbine inlet temperature. Whereas after 75% load, the supplementary firing controls the plant

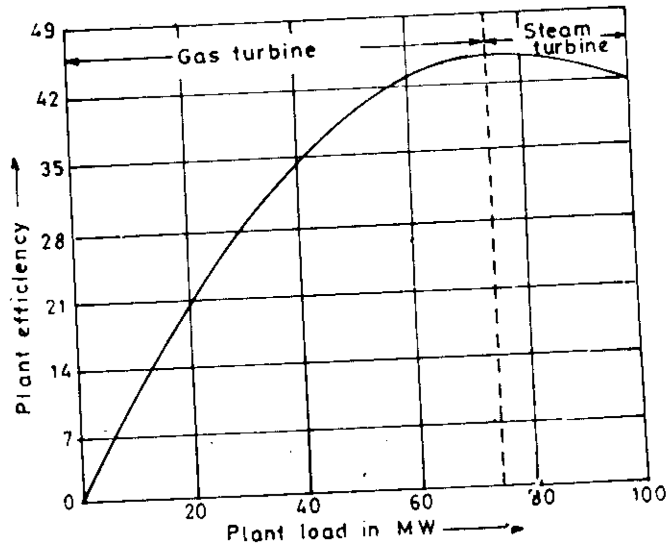


Fig. 25.31. Plant load in MW.

load. Hence the load shared by steam plant is instantly increased and the gas turbine load remains constant. In a gas turbine cycle, the efficiency goes down as the load is reduced. The combined plant behaves much better in this respect. In a combined plant, if the load is reduced from 100% to 60%, the supplementary firing is turned down to reduce to output but the efficiency increases as shown in Fig. 25.30. The part load behaviour is a principal reason for high economy of combined plant.

Capital cost considerations. The addition of gas turbine to an existing steam plant can be considered as one way of increasing the plant capacity. The increase in plant capacity can be achieved by increasing the capacity of the existing steam plant. Therefore, while adopting the gas turbine (combined plant), the cost of gas turbine must be compared with the incremental cost of conventional plant.

The capital cost of gas turbine plant (Rs. per kW) is reducing day by day as a result of increase in available size and better performance Fig. 25.34 are Fig. 25.35 show the reduction in capital cost of power (Rs. per kW) with an increase in plant capacity for a steam plant.

The capital cost of the gas turbine was Rs. 670/kW in 1960 whereas it is reduced to Rs. 425/kW in 1974. Therefore, to get the power at a capital cost of Rs. 425/kW, the minimum steam plant capacity required is 400 MW. Therefore, upto 400 MW capacity, the addition of gas turbine is more economical than increasing the capacity of existing steam plant.

This shows how gas turbine plants progressively become competitive in first cost for larger and larger plants. But, unfortunately, the gas turbines are not yet competitive for largest power plants.

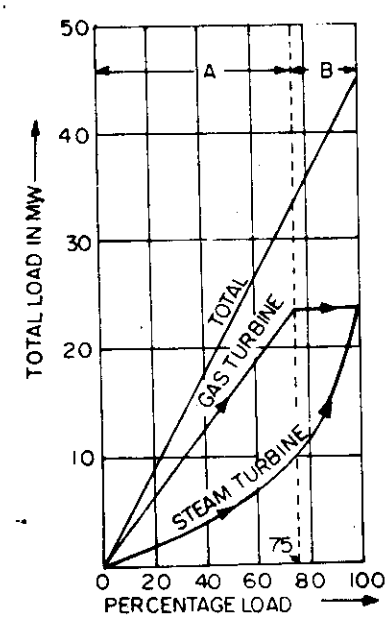


Fig. 25.32. Power distribution in a combined cycle plant.

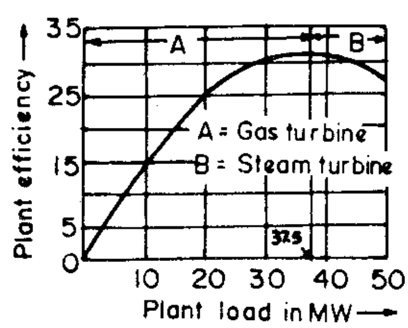


Fig. 25.33. Load versus thermal efficiency.

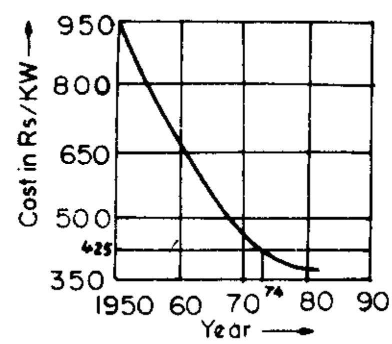


Fig. 25.34.

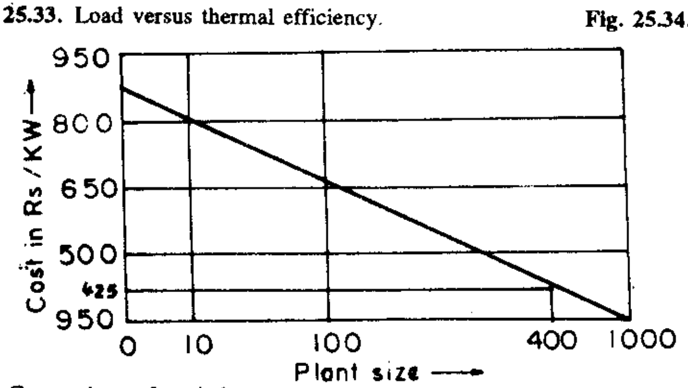


Fig. 25.35. Comparison of capital costs of gas turbine and steam turbine power plants.

25.12. ECONOMICS OF COMBINED CYCLE PLANTS

Figures 25.36 (*a*, *b* and *c*) show the generating costs versus operating hours per year for gas turbine and combined cycle plant when the total output of 250 MW capacity is same for different fuel prices. The following conclusion can be drawn from the figures.

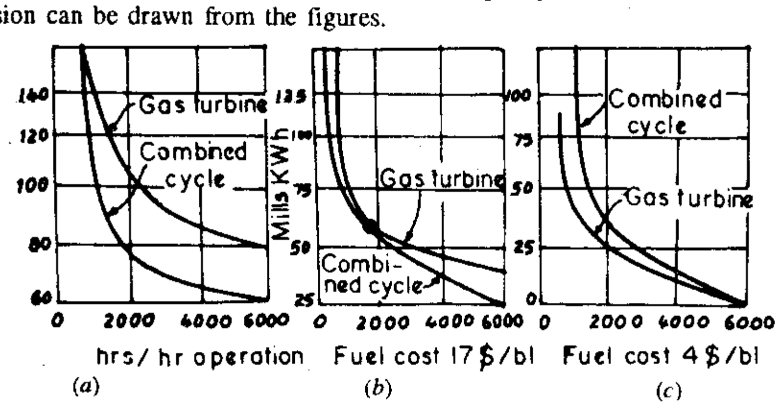


Fig. 25.36.

* The fuel price, plant η and annual operating hours have a large influence on the electricity production cost.
* The combind cycle is always more economical than gas turbine for base and medium load plants at high oil prices.
* Gas turbine plant is economical only beolw 1500 equivalent utilization hours *i.e.* peak load duty operated power plant and in the case of low fuel prices as shown in Fig. 25.36 (*c*).

The electricity production cost and efficiency improvement does not clearly indicate the actual amount of the saving. The accumulated savings over the operation period are plotted in Fig. 25.36 (*d*), for fuel price of 34 $/bL.

The amount can be considered as a profit as direct fuel saving or the quantity of oil can be sold at the world market prices. In latter case, oil-producing countries can conserve their own petroleum resources.

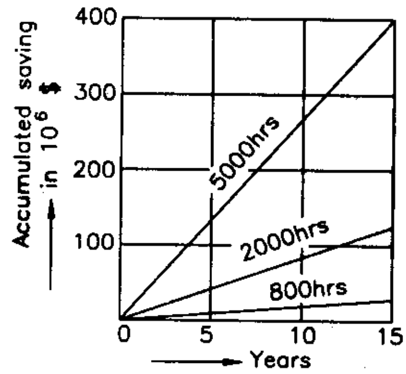


Fig. 25.36. (*d*)

25.13. THE FUTURE OF COMBINED CYCLE

The heavy duty gas turbines will find increasing worldwide use in electric power generation. It had been predicted that by 1980, 10% of electrical power generated worldwide should come from fossil fuel fired gas turbines, with half of that being generated by highly efficient combined cycle plants.

It is certain that gas turbines will play a major role in the further development of power industry as prospects for coal gasification and liquification are extremely good. From 1980, the combined cycle plants using gasified coal being used as mid-range or base load plants of 500 MW to 1000 MW capacity.

Higher efficiencies, lower first cost, short delivery time and minimum environmental effects are among the factors which will influence system planners and utility management to select heavy duty gas turbine plants over other forms of generation.

The long term growth of combined cycle lies in the current and planned development programmes to gasify or liquify the coal. Presently, we are witnessing a growth in the use of gas and oil-fired regenerative and combined cycle plants for mid-load generation. The coal burned combined cycle plants appeared by 1980 for mid and base load generation.

It is universally known fact that the gas turbine efficiency increases directly with an increase in inlet gas temperature. The present material technology available limits the inlet gas temperature to 1200°C. Therefore, the future trend in developing the combined cycle is also to increase the gas inlet temperature. The effect of gas inlet temperature on the overall efficiency of the combined plants which will be used in future is shown in Fig. 25.37 assuming steam cycle efficiency of 35%. A 40°C increase in inlet temperature, an

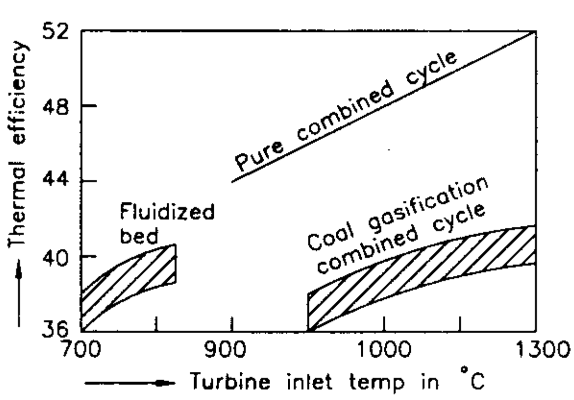


Fig. 25.37. Future trend of combined cycle.

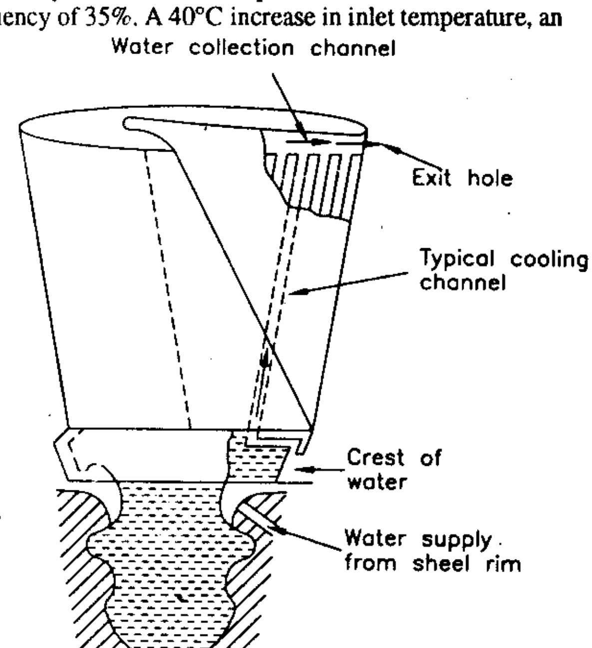


Fig. 25.38 (*a*).

efficiency increases nearly by 1% for presently available equipment with air-cooled turbine blades. Therefore, the future trend will be to find out better methods of blade cooling so the blade temperature can be maintained at lower value even the inlet gas temperature is increased considerably.

The different methods which are under progress for cooling the gas turbine blades are water cooling steam cooling and transportation cooling.

In water-cooling system, water is ejected from the tips of the blades by centrifugal force, as shown in Fig. 25.38 (*a*) and carries mineral deposits with it and prevents clogging of the channel. The water is collected in a stationary shroud from which it flows to a heat exhanger where it is cooled and recirculated.

Steam cooling is used by feeding the steam from low pressure point along the steam turbine. Fig. 25.38 (*b*) exhibits a view of a possible way to achieve blade heat protection by using steam cooling. Steam is fed to the blade at points A and B. Steam nozzles at the leading edge direct a blanket of steam to cover the blade surfaces as shown in figure.

A transpiration cooling used by Curtiss Wright is shown in Fig. 25.38 (*c*). In this arrangement, the load is carried by a span and aero-foil does not carry any load. The aero-foil in the form of mesh is made from higher % of chromium which can easily resist high temperature corrosion. The blanket of the air on the surface of the blade protects the blade from impingement of high temperature gas molecules. It is anticipated that with 1400°C gas temperature, the blade skin temperature will not exceed 800°C which is below the the threshold for hot corrosion.

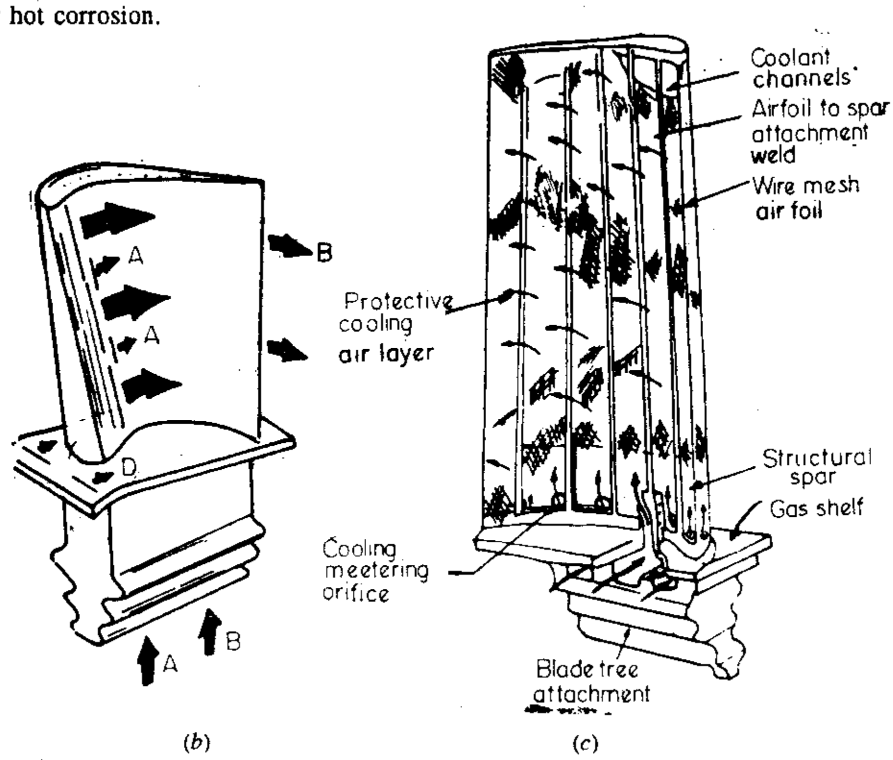


Fig. 25.38.

The cooling systems provide a much more fuel flexibility. Higher ash fuels can be used if the blade temperature is maintained at a low level. With water cooled blades, it is possible to use liquid coal as future fuel for combined cycle.

It is predicted that the gas turbine plant not only will be capable of operating on any type of fuel as solid, liquid, gaseous or nuclear but it will be able to meet the complete requirements of worldwide electric power generation systems ranging from reserve to base load plants.

25.14. FUELS FOR GAS TURBINES AND FUTURE FOR COMBINED CYCLE IN INDIA

The ideal fuel for the gas turbine units is natural gas from the viewpoint of efficient energy conversion operation and pollution control. This fuel being clean would not foul the gas turbine blades and as such, the availability of the unit would be highest. It is always advisable to use a gas turbine unit designed for multifuels (gaseous as well as liquid fuels). This would facilitate unhindered operation of the power station in case of non-availability of one type of the fuel due to some reason. The light distillates, such as light diesel oil (LDO), high speed diesel (HSD). naptha etc. would be the next preferred gas turbine fuels. These fuels can be used without treatment if the contaminants are minimal. The gas turbines can also use heavy residual fuel oils as furnace oil and low sulphur heavy stock (LSHS). The furnace oil has a viscosity of about 170 mm^2/sec at 50°C with a maximum sulphur content of 4.5% whereas LHSH is waxy in nature and has a viscosity of about 500 mm^2/sec at 50°C with a maximum sulphur content of 1.5% and it has a pour point of 72°C. In view of its high pour point, LSHS oil needs to be handled hot at all points of storage and handling from the refining point to the consuming centre. The LSHS has higher calorific value about 3.5% more compared to furnace oil, is cheaper by about 15% and it has lower asphaltenses, ash and carbon residue. Both the furnace oil and LSHS, if pretreated properly to reduce contaminants within the acceptable limits, could be used in the gas turbines.

The gas turbine blades are very much vulnerable to presence of harmful elements like sodium, potassium, vanadium and nickel in the fuel oils. It is, therefore, essential to restrict these harmful elements to level stipulated, beyond which, there is a danger of excessive corrosion of the blades due to harmful deposits of these elements. The fuel treatment plan shall limit the impurities to the following before the oils are used as fuels in the gas turbines :

Sodium + potassium = 1 ppm (max) by weight
Calcium = 10 ppm (max) by weight
Water = 1% (max) by weight

Basically centrifugal type and electrostatic type fuel treatment plants are used in practice.

Large reserves of natural gas have been discovered in Bombay off-shore, the Krishna- Godavari basin, Tripura and to some extent in Rajasthan and the Andamans. Prospects for more in these basins as also in Jammu & Kashmir and Himachal Pradesh are very bright. Already, certain large capacity gas turbine combined cycle plants (500 MW – 600 MW) are being installed at Kawas in Gujarat, Anta in Rajasthan and Auraiya in U.P. All these plants will use natural gas as fuel. It is expected that with the finding of large gas reserves, the country would go in for more and more gas turbine combined cycle plants in future for efficient use of the fuel.

In addition to above, the Govt. of India has decided to give the power generation Industry to the private sector. Keeping this in view and considering the higher efficiency of the combined cycle, Indian Govt. has decided to run such plants on gas as a fuel. As indigeneous gas fuel is not available, India has signed an agreement with Oman on long-term gas supply contract under which, the Sultanate will export gas to India through an ambitious five billion doller sub-sea pipeline system. Two countries had signed a historic memorandum of understanding (MOU) in March 1993 under which, Oman offered to design, construct and operate a sub-sea pipeline system to transport natural gas from Sultanate to India.

The agreement on price also provides that the variation in the price of gas would be linked to the formula based on a basket of crude oil and other petroleum products. This will insulate the users from violent fluctuations in the international prices of crude and also make the gas price competitive with other fuels.

Apart from the price, the other principal terms agreed to include the period of contract, quantity, the delivery points, the quality of gas.

The Oman Oil Company (OOC) has already completed the feasibility study for the system that will carry 56.6×106 m^3 of gas per day.

The project envisages, among other things, the laying of two 60 cm sub-sea pipelines over a distance of 1135 km which traverses the Arabian Sea between Ra's AL Jifan in Oman to Bhaghau in Gujarat. Each pipeline will be able to supply one billion cubic feet of gas per day. At its deepest, the pipe line will be 3500 m below sea level which will be the deepest sub-sea pipeline in the world. The present deepest is 800 metres below sea level.

The first of the two pipelines designed to carry 28.3 million m^3 of gas per day, is scheduled to be commissioned in July 1999 and second equal capacity will be commissioned by the end of 2001. The gas will be mainly used for power generation and will also be supplied to fertiliser plants and other industrial units. The whole project would cost $ 5 billion.

Sufficient demand has been identified to absorb the imported natural gas and consuming units will be setup in time to synchronise with the arrival of the imported gas.

In the above system, India will have to invest around Rs.40,000 crores in ónland pipelines and down-stream facilities for the utilization of the gas. The gas will be sufficient to generate 10,000 MW of power in the country for the lifetime of the power plants (25-years).

The layout of the pipeline is shown in Fig. 25.39. The pipeline will be carried through Iran and Pakistan. The said pipeline (200 km) passes under sea at four points in Pakistan and fifth part runs over the ground of Pakistan. Presently (Aug. 1995), the Pakistan Govt, has objected for this on the ground of political issue.

The Rs. 827 crore Jegurupadu gas based power plant in Andhra Pradesh, one of the eight fast track projects identified by Union Govt. will be cleared within a month.

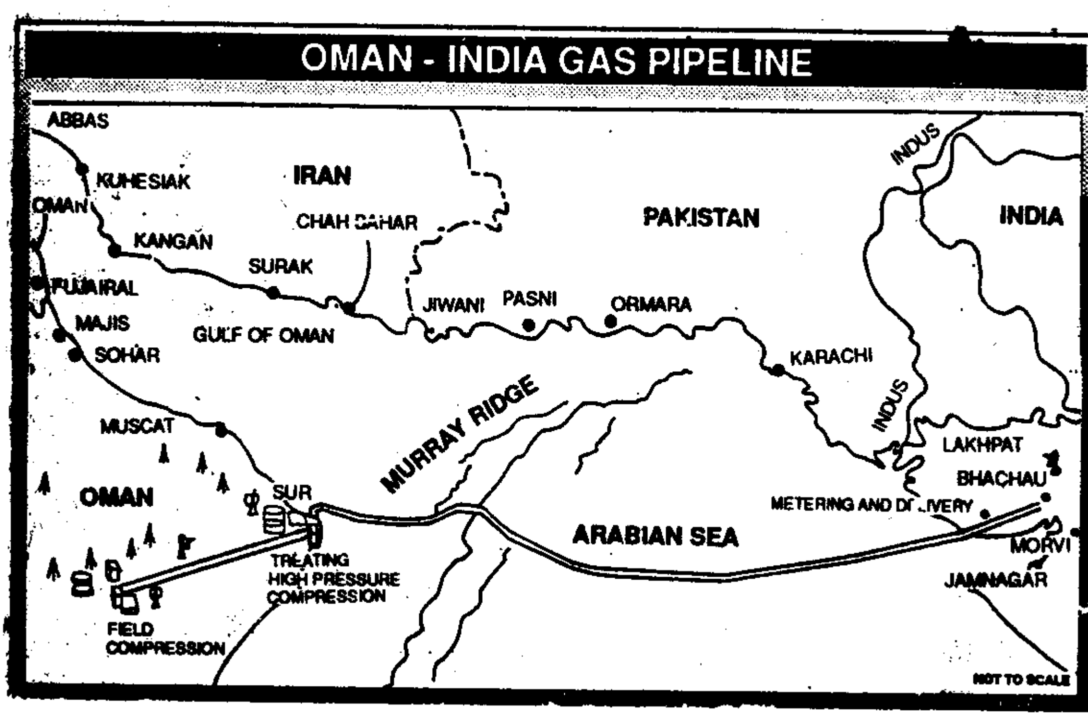


Fig. 25.39.

25.15. COMBINED CYCLE PLANTS IN INDIA

Many measures taken by Govt. of India will accelerate the power development in India. While capacity addition is a welcome strategy pursued by Govt. of India sustaining the growth of power, cogeneration deserves closer attention for supplementing the country's electricity infrastructure.

A cogeneration in India power industry plays very important role on account of

(1) Energy saving potential – when we compare 0.7 kg/kWh coal in conventional plant against 0.4 kg/kWh in a cogeneration plant.

(2) Higher plant load factor of 0.75 against 0.6 for conventional plant.

(3) Capital cost of cogeneration is Rs.1.5 crore per MW against Rs. 3 crore per MW in conventional plant.

(4) Transmission and distribution losses are very less.

(5) Return on investment is 10% against 6% in public utilities.

Though there are 700 cogenerators already operating in the country with s· .e from 500 kW to 50 MW, over 10,000 MW of untapped cogeneration power potential still exists as per recent estimates.

A few major combined power plants which are in operation in India are listed in the following table :

Cogeneration Parameters

Sl. No.	Customer	Power (MW)	Steam Parameters Pressure bar	Temp. °C	Flow Tons/hr	Heat to Power Ratio
1.	ONGC, Uran	22	13.5	Sat	75	2.64
2.	BPCL, Bombay	22	21.5	330	62.4	2.52
3.	CRL	22	41/	374/	60/	2.68
			5.5	Sat	8.6	
4.	HPCL, Vizag	8.4	12.5	256	37.7	3.67
5.	IOC, Baroda	32	15	215	115.6	2.99
6.	FCI, Talcher	22	41/	400/	55/	2.57
			2	Sat	9.9	
7.	Chambal FCL, Kota	22	42	400	70	2.84

The details of some of the plants are listed below.

(A) Cogeneration Plant for Cochin Refinery

It is 21.9 MW plant used for oil refinery at Ambalamugal, Karnakulam, Kerala. It serves dual purpose of generating power and supplying process steam.

Other deails of the plant are listed below :

(1) H.P. steam output – 60 Tons/hour at 40 bar and 374°C.

(2) L.P. steam output – 8.6 Tons/hr at 4.5 bar and saturated.

(3) Heat input to gas turbine – 83.44 MW through refinery fuel gas.

(4) Heat input for supplementary firing – 7.3 MW.

(5) Overall efficiency of generation plant – 79.2%.

(B) Cogeneration Plant at ONGC Hazira

This plant of 38.4 MW capacity is used to meet the power and steam requirement of Hazira Gas Processing Complex located at Surat. The details are listed below :

(1) H.P. steam output – 37 tons/hr at 37 bar and saturated.

(2) L.P. steam output – 210 tons/hr at 9 bar and saturated.

(3) Condensate temperature – 90°C.

(C) Vijjeswaram cogeneration Plant

It is 100 MW capacity plant in South India, first of its kind. It consists of two 33 MW gas turbine generators and two waste heat recovery boilers and one 34 MW steam turbine generator operating in combined mode. It is also estimated that it will generate 600 million units per year when operating at full load.

The other details of the plant are listed below :

(*i*) Gas Temp. at inlet and outlet of turbine – 1050°C and 559°C.

(*ii*) C.V. of the gas used = 35280 kJ/kg.

(*iii*) Pressure of the gas supplied = 22 bar.

(*iv*) Heat rate for Gas Turbine = 13000 kJ/kWh.

(*v*) Gas requirement at full load = 0.5 million m^3/day.

PFBC – Pressurised Fluidised Bed Combustion.

CGCC – Coal Gasification combined Cycle.

HRSG – Heat Recovery Steam Generator.

HRSB – Heat Recovery Steam Boiler.

Nuclear-Steam Combined Cycle

The HTGRs (high temperature gas cooled reactor) were preferred and developed in America, Britian and France as they offer special advantages as high conversion ratio, good fuel cycle economics, high specific power and high fuel brun-up when Helium is used as working fluid. The only disadvantage experienced was, the high temperature range in the cycle and low thermodynamic efficiency.

This can be avoided if the nuclear gas-cooled plant is coupled with conventional steam plant where the steam is generated by using the heat of the gas coming out of the turbine.

The arrangement of such combined cycle plant is shown in Fig. 25.40. The hot gas (He) coming out of nuclear reactor is used to develop power in gas turbine.

The He is compressed in the compressor and then preheated in regenerator as shown in Fig. 25.40 and then it is further heated in HTGR. It expands in the gas turbine and enters in the regenerator. The energy left in He after coming out of the regenerator is used to generate the steam with the help of extra fuel supply as shown in figure. The He gas then again enters the compressor to repeat the cycle.

The steam cycle is the conventional working on Rankine cycle.

The heat generated in HTGR is completely used in both cycles and that heat is rejected only in the steam condenser.

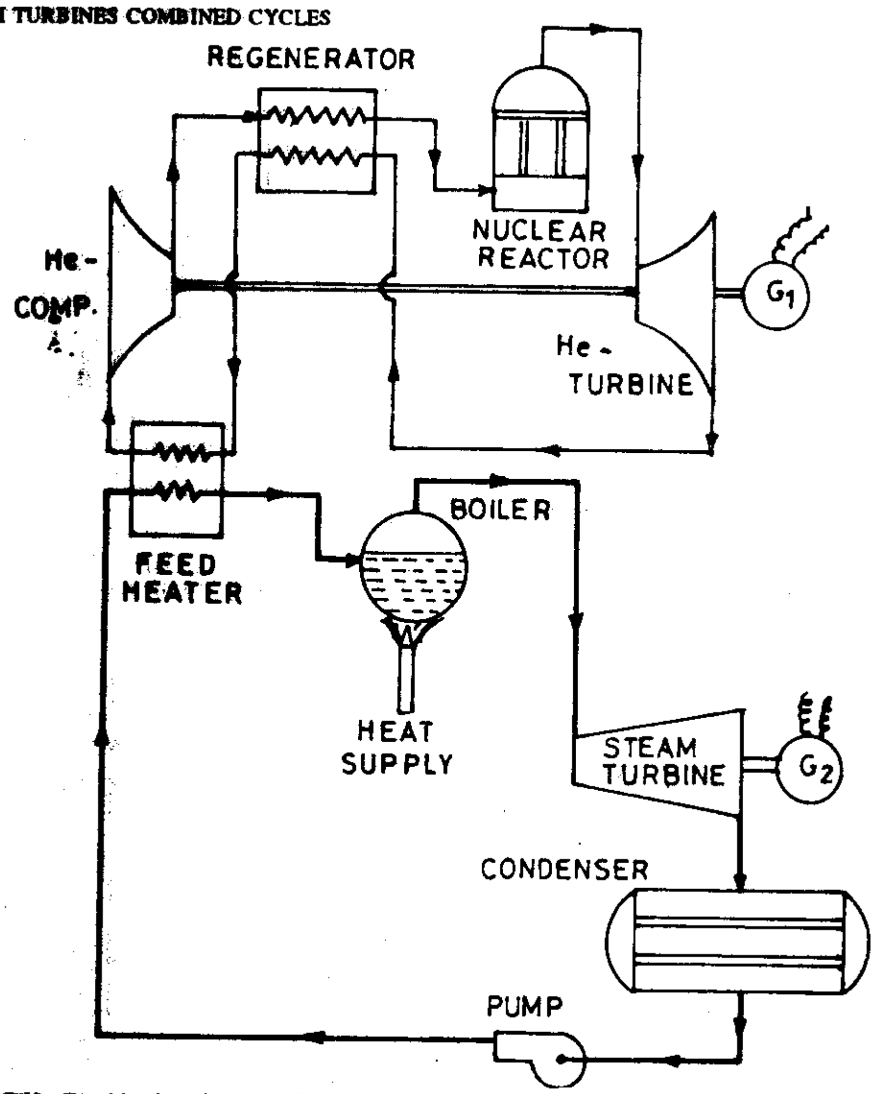


Fig. 25.40. Combined cycle using Nuclear Gas Turbine and fossil fuel fired steam turbine.

SOLVED PROBLEMS

Problem 25.1. *In a combined cycle power plant, the air is supplied at a rate of 2000 tons/hr and at temperature 20°C. The pressure ratio is 7 : 1. Inlet pressure to compressor and outlet pressure from the turbine may be taken as 1 bar. The temperature in the gas turbine cycle is limited to 1000°C. The isentropic efficiency for compressor = 80% and for turbine = 85%. The C.V. of oil used = 45 × 10³ kJ/kg.*

The data for steam turbine is given below. The temperature of gas used for steam generation is increased to 1200°C by burning the fuel in the exhaust coming out from gas turbine. The condition of steam generated in the boiler is 50 bar and 500°C. The condenser pressure = 0.1 bar. The temperature of gas going to stack = 200°C. Find out the following :

(i) *Total power generating capacity of the plant.*

(ii) *Overall efficiency of the plant.*

(iii) *Mass of fuel used per hr.*

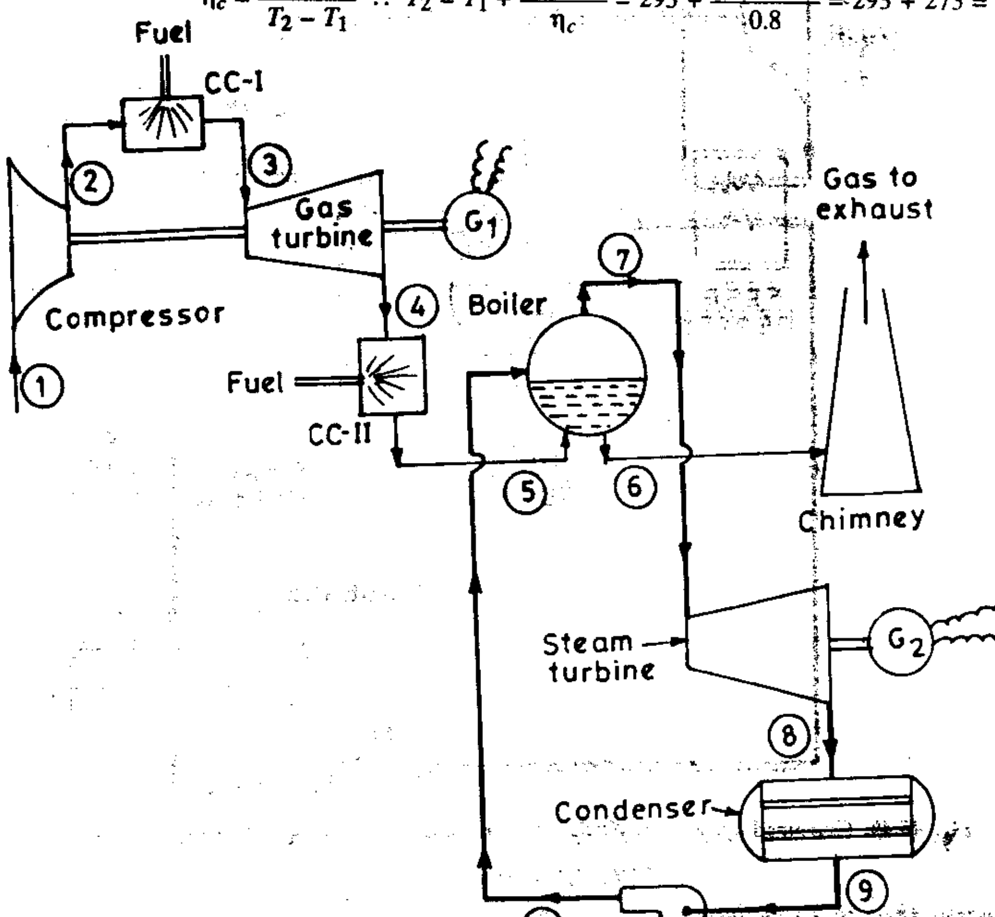
Take $C_{pa} = 1$ *kJ/kg-K,* $\gamma = 1.4$ *for air and* $C_{pg} = 1.1$ *kJ/kg-K,* $\gamma = 1.33$ *for gas*

Do not neglect the fuel.

Solution. The arrangements of the components and corresponding processes in the cycles are shown in Fig. 25.1 (*a*) and Fig. 25.1 (*b*).

$$T_2' = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (20 + 273)\,(7)^{\frac{0.4}{1.4}} = 293 \times 1.75 = 511 \text{ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} \quad \therefore \quad T_2 = T_1 + \frac{T_2' - T_1}{\eta_c} = 293 + \frac{511 - 293}{0.8} = 293 + 273 = 566 \text{ K}$$



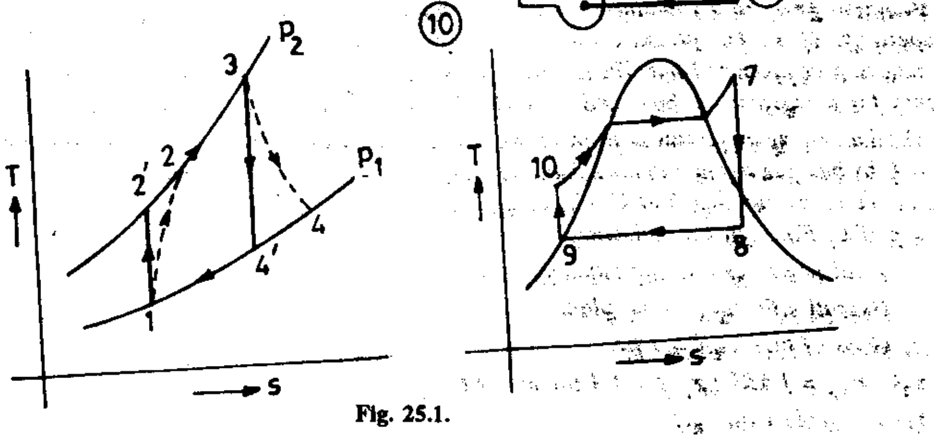


Fig. 25.1.

$$\frac{T_3}{T_4'} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (7)^{\frac{0.33}{1.33}} = 1.62 \quad \therefore \quad T_4' = \frac{T_3}{1.62} = \frac{1000+273}{1.62} \approx \mathbf{786\ K}$$

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$\therefore \quad T_4 = T_3 - \eta_t (T_3 - T_4') = 1273 - 0.85(1273 - 786) = 1273 - 414 = \mathbf{859\ K}$$

Considering heat balance in CC-I

$$m_{f1} \, . CV = (m_{a1} + m_{f1})\, C_{pg}\, (T_3 - T_2)$$

$$\therefore \quad CV = \left(\frac{m_{a1}2}{m_{f1}} + 1\right) \times 1.1\ (T_3 - T_2)$$

$$\frac{m_{a1}}{m_{f1}} = \frac{CV}{1.1(T_3 - T_2)} - 1 = \frac{45 \times 10^3}{1.1(1273 - 566)} - 1 = \mathbf{56.86}$$

$$m_{a1} = \frac{2000 \times 1000}{3600} = \mathbf{555.5\ kg/sec}$$

$$\therefore \quad m_{f1} = \frac{m_{a1}}{56.86} = \frac{555.5}{56.86} = \mathbf{9.8\ kg/sec}$$

W_g (power developed in gas-turbine plant)

$$= (m_{a1} + m_{f1})\, C_{pg}\, (T_3 - T_4) - m_{a1}\, C_{pa}\, (T_2 - T_1)$$
$$= (555.5 + 9.8) \times 1.1\ (1273 - 859) - 555.5 \times 1\ (566 - 293)$$
$$= 257437.5 - 151651.5 = 105786 \text{ kW} \approx \mathbf{106\ MW}$$

Heat generated in CC-II = Heat given to the gas passing through CC-II

$$m_{f2} \, . \, CV = (m_{a1} + m_{f1} + m_{f2})\, C_{pg}\, (T_5 - T_4)$$

$$\therefore \quad m_{f2} \times 45 \times 10^3 = (555.5 + 9.8 + m_{f2}) \times 1.1\ (1473 - 859) = (563.3 + m_{f2}) \times 675.4$$

$$\therefore \quad m_{f2} = \mathbf{6.8\ kg/sec}$$

Considering the heat balance in the boiler.

$$m_s\, (h_7 - h_{10}) = (m_{a1} + m_{f1} + m_{f2})\, C_{pg}\, (T_5 - T_6)$$

If pump work is neglected, then $h_{10} = h_9$.

$$\therefore \quad m_s\, (h_7 - h_9) = (m_{a1} + m_{f1} + m_{f2})\, C_{pg}\, (T_5 - T_6)$$

$h_7 = 3400$ (from $h - s$ chart)

$h_9 = 45.5$ (from steam table)

Now substituting the values in the above equation

$$\therefore \quad m_s\, (3400 - 45.5) = (555.5 + 9.8 + 6.8) \times 1.1\ (1000 - 200)$$

$$\therefore \quad m_s = \mathbf{150\ kg/sec}$$

W_s (power developed in steam turbine)

$$= m_s\, (h_7 - h_8)$$
$= 150\ (3400 - 2220)$ where $h_8 = 2220$ from h-s chart
$$= 177000 \text{ kW} = \mathbf{177\ MW}$$

$\therefore$ Total power generated is given by

$$W_t = 106 + 177 = \mathbf{283\ MW}$$

(ii) $$\eta = \frac{W_t}{Q_s} = \frac{W_t}{(m_{f1} + m_{f2}).CV} = \frac{283 \times 10^3}{(9.8 + 6.8) \times 45 \times 10^3} = 0.4 = \mathbf{40\%}$$

(iii) Mass of fuel supplied per hour $= \dfrac{(m_{f1} + m_{f2}) \times 3600}{1000}$ tons/hr

$$= \frac{(9.8 + 6.8) \times 3600}{1000} = \textbf{59.67 tons/hr}$$

(*iv*) Specific fuel consumption $= \frac{59.67 \times 1000}{283 \times 1000} = \textbf{0.211 kg/kWh}$

Problem 25.2. *In a Gas and steam turbine combined cycle, the maximum temperature in the cycle is limited to 1100°C. The inlet temperature of air to compressor is 20°C and pressure ratio is 8. The amount of air supplied to gas turbine is 400% of theoretical air. In a supplementary firing, the gas temperature is raised to 1000°C before entering into the heat recovery boiler. The gas leaves the steam generator at 300°C. The steam is generated at 80 bar and 600°C from the feed water supplied at 150°C. Calculate the fuel added in supplementary firing per kg of air and the mass ratio of air flow to steam flow.*

Also find out power generating capacity of the plant if the air flow through compressor = 1.5 kg/sec.

Take condenser pressure = 0.05 bar.

Fuel used is CH_4

C.V of fuel used = 61600 kJ/kg

C_p (for gas and air) = 1 kJ/kg-K and $\gamma = 1.4$.

Isentropic η of compressor and turbine may be taken as 100%.

Solution. The arrangement of the components of the combined cycle is shown in Fig. 25.2 (*a*) and corresponding cycles are shown in Fig. 25.2 (*b*).

The combustion reaction taking place in CC-I is given by

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$
$$16 + 64 = 44 + 36$$

$\therefore$ Amount of O_2 required = 64/16 = 4 kg per kg of CH_4.

$\therefore$ Amount of air required $= \frac{100}{23} \times 4 = 17.4$ kg/kg of fuel as air contains 23% of O_2 by weight.

As 400% excess is supplied as given in the problem.

$\therefore$ Actual air supplied = 17.4 × 5 = **87 kg/kg of fuel.**

$\therefore$ (m_{f1}) Amount of fuel supplied per kg of air flow through CC-I $= \frac{1}{87} = \textbf{0.0115 kg}$

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 293(8)^{0.286} = \textbf{531 K}$$

$$T_4 = \frac{T_3}{\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}} = \frac{1373}{1.81} = \textbf{758.6 K}$$

In CC-II

Heat generated by burning the fuel = Heat gained by the air

$$m_{f2} \times C.V = m_a\, C_{pa}\, (T_5 - T_4)$$
$$m_{f2} \times 61600 = 1 \times 1\, (1273 - 758.6)$$

$\therefore$ m_{f2} = **0.00835 kg per kg of air flow**

Amount of heat given by the gas in the boiler = Heat gained by the steam

$$1 \times C_{pg}\, (T_5 - T_6') = m_s\, (h_7 - h_{11})$$
$$h_7 = 3510 \text{ kJ/kg (from } h\text{-}s \text{ chart)}$$

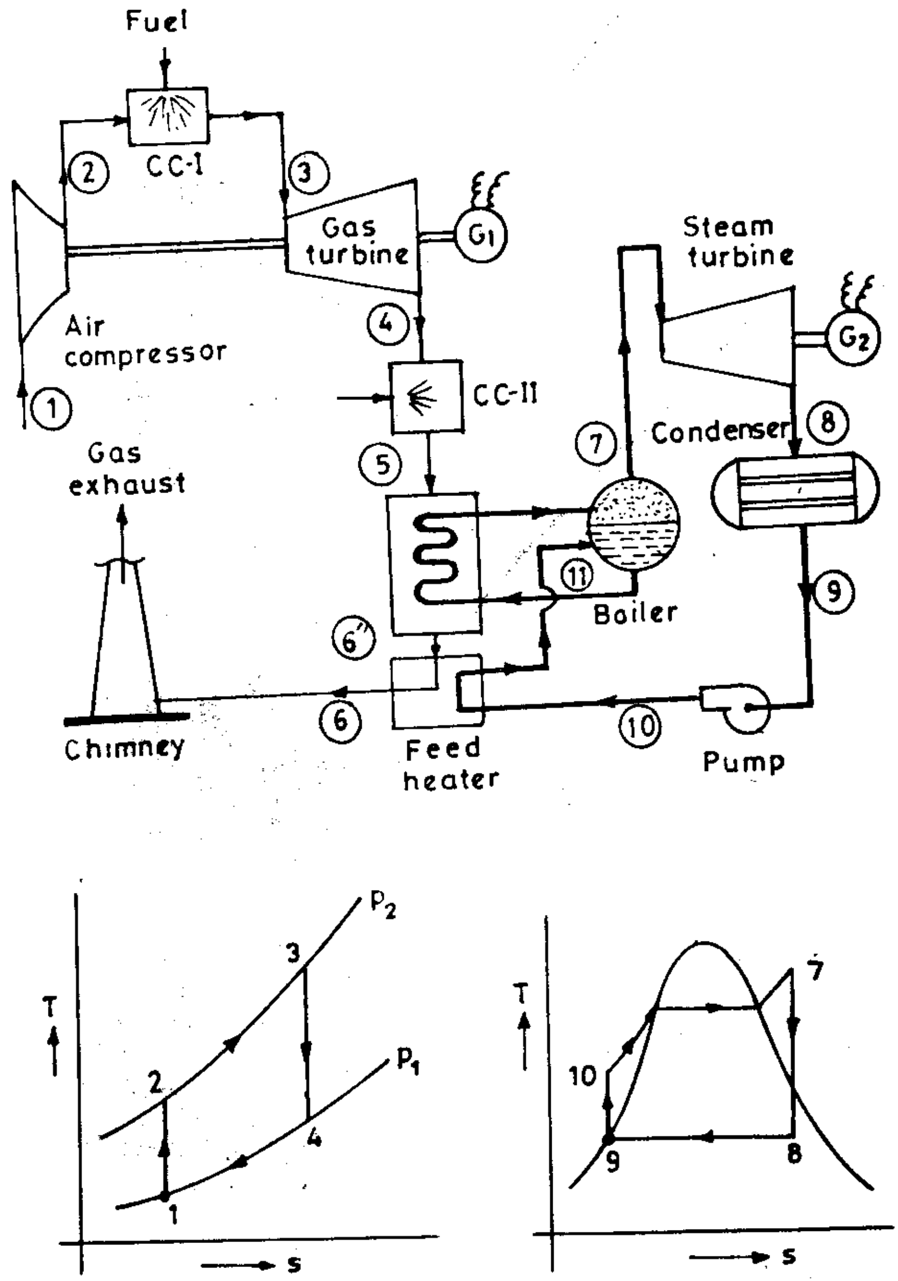


Fig. Prob. 25.2.

$$h_{10} = C_{pw} T_{11} = 4.2 \times 150 = 630 \text{ kJ/kg}$$

$\therefore \quad 1 \times 1\,(1273 - 573) = m_s\,(3510 - 630)$

$\therefore \quad m_s = \dfrac{700}{2880} = \mathbf{0.243\ kg/kg\ of\ air}$

$$\therefore \quad \frac{m_a}{m_s} = \frac{1}{0.243} = \mathbf{4.115}$$

Generating capacity of the plant is given by

$$W_t = W_g + W_s$$
$$= C_{pa}\, m_a\, [(T_3 - T_4) - (T_2 - T_1)] + m_s\,(h_7 - h_8)$$

If $m_a = 1.5$ kg/sec (given) then $m_s = 0.243 \times 1.5 = \mathbf{0.365}$ **kg/sec**

$h_8 = 2080$ kJ/kg (from *h-s* chart)

$$\therefore \quad W_t = 1 \times 1.5\,[(1373 - 758.6) - (531 - 293)] + 0.365\,(3510 - 2080)$$
$$= 565 + 522 = \mathbf{1087\ kW}$$

Total heat supplied is given by

$$Q_s = (m_{f1} + m_{f2})\, m_a \times \text{C.V}$$

where m_{f1} and m_{f2} are the amounts of fuels supplied in CC-I and CC-II per kg of air flow.

$$Q_s = (0.0115 + 0.00835) \times 1.5 \times 61600 = \mathbf{1834\ kW}$$

$$\eta_{(plant)} = \frac{W_t}{Q_s} = \frac{1087}{1834} = 0.593 = \mathbf{59.3\%}$$

m_f (fuel used per hour) $= (m_{f1} + m_{f2})\, m_a \times 3600$ kg/hr

$$= (0.0115 + 0.00835) \times 1.5 \times 3600 = \mathbf{107.2\ kg/hr}$$

$\therefore$ Specific fuel consumption

$$= \frac{107.2}{1087} = \mathbf{0.099\ kg/kW\text{-}hr}$$

Problem 25.3. *A combined power plant consists of a gas turbine unit and a steam turbine unit, the exhaust gas from the gas turbine is supplied to the steam generator at which further supply of fuel is burned in the gas. The pressure ratio for the gas turbine is 8 : 1 and the inlet air temperature is 15°C. The maximum cycle temperature is limited to 800°C.*

The data for steam turbine is given below, Flue gas temperature in furnace = 800°C.

C.V. of fuel = 40×10^3 kJ/kg, chimney gas temperature = 200°C, steam supply condition to turbine = 60 bar and 500°C, condenser pressure = 0.05 bar.

The total power output of the combined plant is 190 MW. Assuming ideal cycles for both the units, calculate (a) thermal efficiency of the combined plant (b) power generated in each unit of the cycle and boiler capacity. Assume C_p (gases) = 1.11 kJ/kg-K and $\gamma = 1.33$ and C_{pa} 1 kJ/kg-K. Neglect the fuel mass and (c) Mass of fuel supplied in tons per hour. *(P.U. Winter 1995)*

Solution. The arrangement of the plant components is shown in Fig. 25.3 (*a*) and corresponding cycles are shown in Fig. 25.3 (*b*).

Assuming W_g is the power generated in gas turbine and W_s is the power generated in steam turbine, we can write

$$W_t = W_g + W_s = 190 \times 10^3 \text{ kW} \quad \text{...}(i)$$

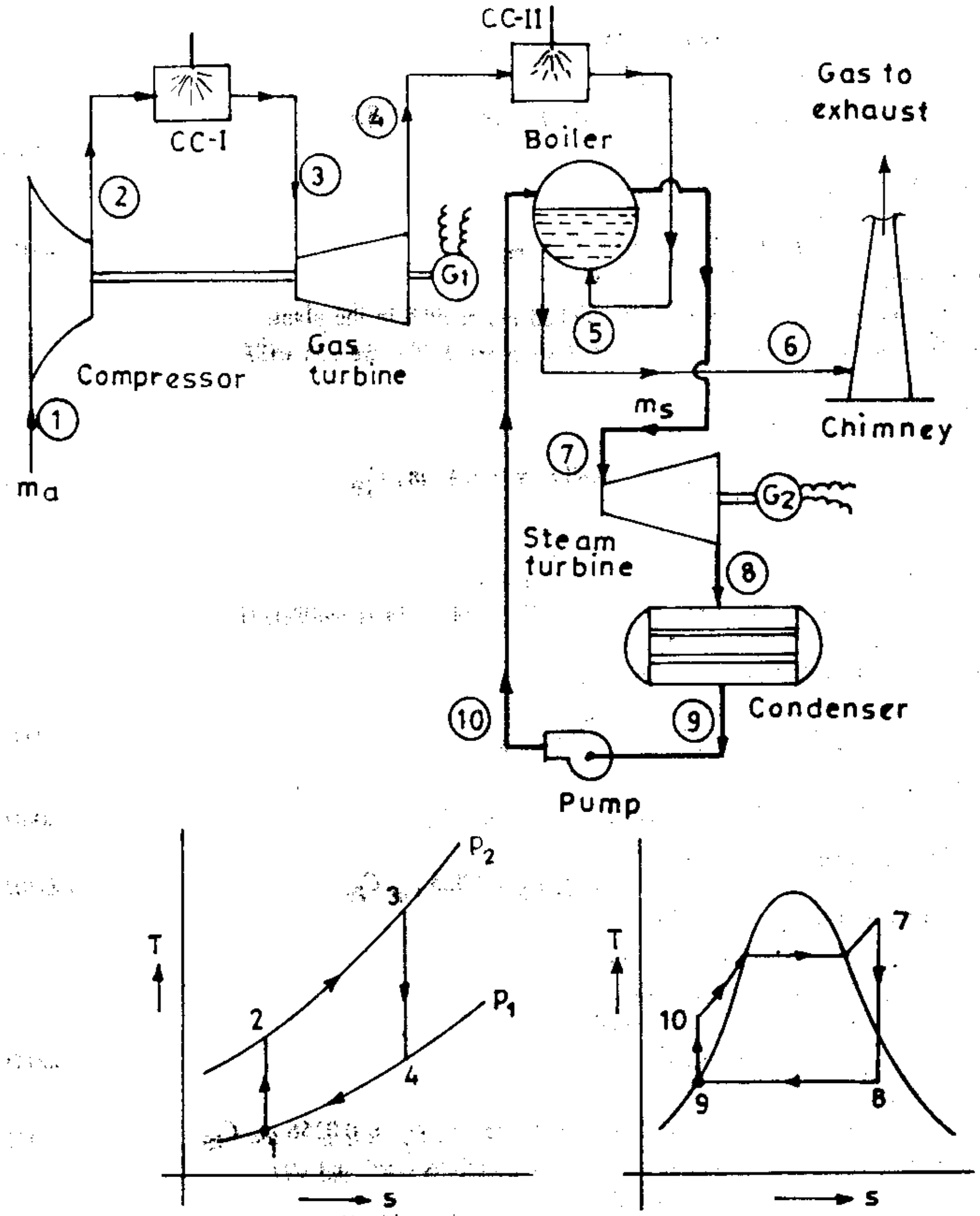
The efficiency of the plant is given by

$$\eta = \frac{W_t}{m_{ft}.\text{CV}} \quad \text{...}(ii)$$

where $m_{ft} = m_{f1} + m_{f2}$ where m_{f1} and m_{f2} are the fuels supplied in CC-I and CC-II.

Neglecting the amount of fuels supplied in the gas cycle, we can write,

$$W_g = m_a \,.\, C_{pa}\,[(T_3 - T_4) - (T_2 - T_1)]$$



Gas turbine cycle Steam power cycle

Fig. Prob. 25.3.

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (8)^{\frac{1.33-1}{1.33}} = (8)^{0.25} = 1.68$$

$\therefore$ $T_2 = 1.68\ T_1 = 1.68\ (15 + 273) = \mathbf{483.8\ K}$

Similarly, $$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 1.68$$

$$T_4 = \frac{T_4}{1.68} = \frac{(800 + 273)}{1.68} = \mathbf{638.7\ K}$$

$$\therefore \quad W_g = m_a \times C_{pa} [(1073 - 638.7) - (483.8 - 288)]$$
$$= 238.5\ m_a\ C_{pa} \qquad \text{...(iii)}$$

Neglecting the pump-work

$$W_s = m_s\ (h_7 - h_8)$$
$$= m_s\ (3370 - 2080) = 1290\ m_s \qquad \text{...(iv)}$$

where h_7 and h_8 values are taken from h-s chart.

The m_a, m_{f1}, m_{f2} and m_s are the masses supplied per second in the plant.

Assuming no losses and efficiencies of combustion are 100%, we can write.

For CC-I

$$m_a\ C_{pa}\ (T_3 - T_2) = m_{f1} \times \text{C.V}$$

$$\therefore \quad m_{f1} = \frac{m_a\ C_{pa}}{40 \times 10^3} (1073 - 483.8) = 0.0147\ m_a\ C_{pa} \qquad \text{...(v)}$$

For CC-II

Heat given by flue gases = Heat gained by steam in boiler

$m_a\ .\ C_{pa}\ (T_5 - T_6) = m_s\ (h_7 - h_9)$ as $h_{10} = h_9$ (if pump work is neglected)

$$m_a\ C_{pa}\ (800 - 200) = m_s\ (3370 - 32.6)$$

where $h_9 = 32.6$ is taken from stable table at 0.05 bar.

$$\therefore \quad m_s = \frac{m_a\ C_{pa} \times 600}{3337.4} = 0.178\ m_a\ C_{pa} \qquad \text{...(vi)}$$

Substituting this value in equation (*iv*)

$$W_s = 1290 \times 0.178\ m_a\ C_{pa} = 232\ m_a\ C_{pa} \qquad \text{...(vii)}$$

Now adding the equations (*iii*) and (*vii*)

$$W_t = 238.5\ m_a\ C_{pa} + 232\ m_a\ C_p = 470.5\ m_a\ C_{pa} \qquad \text{...(viii)}$$

The fuel burned in CC-II is given by

Heat given in CC-II = Heat gained by air

$$\therefore \quad m_{f2}\ CV = m_a\ C_{pa}\ (T_5 - T_4)$$
$$m_{f2} \times 40 \times 10^3 = m_a\ C_{pa}\ (1073 - 638.7)$$
$$\therefore \quad m_{f2} = 0.01085\ m_a\ C_{pa} \qquad \text{...(ix)}$$

Adding the equations (*v*) and (*ix*)

we get $$m_{ft} = m_{f1} + m_{f2} = (0.0147 + 0.01085)\ m_a\ C_{pa} = 0.0256\ m_a\ C_{pa} \qquad \text{...(x)}$$

Now substituting the values in equation (*ii*) from equations (*viii*) and (*ix*).

$$\eta = \frac{470.5\ m_a\ C_{pa}}{0.0256\ m_a\ C_{pa} \times 40 \times 10^3} = 0.46 = \mathbf{46.6\%}$$

Now equating the equations (*i*) and (*viii*)

$$470.5\ m_a\ C_{pa} = 190 \times 10^3$$

$$\therefore \quad m_a = \frac{190 \times 10^3}{470.5 \times 1.11} = \mathbf{363.8\ kg/sec}$$

From equation (*iii*)

$$W_g = \frac{238.5 \times 363.8 \times 1.11}{1000} = \mathbf{96.3\ MW}$$

$$W_s = P_t - P_g = 190 - 96.3 = \mathbf{93.7\ MW}$$

The steam generating rate is given by equation (*vi*)

$$m_s = 0.178 \times 363.8 \times 1.0 = \mathbf{71.9\ kg/sec}$$

Total mass of fuel supplied is given by

$$m_{ft} = m_{f1} + m_{f2}$$
$$= 0.0256\ m_a\ C_{pa} \text{ as per equation } (x)$$
$$= 0.0256 \times 363.8 \times 1.0 \text{ kg/sec}$$
$$= \frac{0.0256 \times 363.8 \times 1.0 \times 3600}{1000} \text{ tons/hr} = \mathbf{33.53\ tons/hr}$$

Problem 25.4. *A combined cycle power plant is to be designed to develop 100 MW power. 60% power is to be developed in gas turbine plant and remaining in steam plant. The air-pressure and temperature at the inlet of compressor are 1 bar and 300 K. The pressure ratio is limited to 8 and the maximum temperature any where should not exceed 1000°C. The steam is to be generated at 50 bar and 600°C by using supplementary fuel and condenser pressure is to be maintained at 0.05 bar. Taking the following data, find out :*

(*i*) *Overall efficiency of the plant.*

(*ii*) *Air flow to steam flow ratio.*

(*iii*) *Generating cost of power if fuel cost is Rs. 2500 per ton*

$\eta_c = 85\%$, $\eta_t = 90\%$, $\eta_{com} = 95\%$,

$C_{pa} = 1$ kJ/kg-K, $\gamma_a = 1.4$, $C_{pg} = 1.1$ kJ/kg-K, $\gamma_g = 1.33$.

The fuel used is naptha whose C.V is 40×10^3 kJ/kg. The gas temperature going to chimney = 200°C. Don't neglect the fuel and consider the power required to run the pump.

(*iv*) *Quantity of cooling water required in the condenser per hour if the rise in temperature is limited to 10°C.*

Solution. The arrangement of the components of the plant is shown in Fig. 25.4 (*a*) and corresponding cycles are shown in Fig. 25.4 (*b*).

Considering compressor

$$T_2' = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 300\ (8)^{\frac{1.4-1}{1.4}} = 300\ (8)^{0.286} = \mathbf{544\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} \quad \therefore \quad T_2 = 300 + \frac{544 - 300}{0.85} = \mathbf{587\ K}$$

Considering turbine

$$\frac{T_3}{T_4'} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (8)^{\frac{1.33-1}{1.33}} = (8)^{0.248} = 1.68$$

$$\therefore \quad T_4' = \frac{1000 + 273}{1.68} = \mathbf{758\ K}$$

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$\therefore \quad T_4 = T_3 - \eta_t\ (T_3 - T_4') = 1273 - 0.9(1273 - 758) = \mathbf{809.5\ K}$$

Considering heat balance in CC-I

$$m_{f1}\ .\ CV \times \eta_{com} = (m_{a1} + m_{f1})\ C_{pg}\ (T_3 - T_2)$$

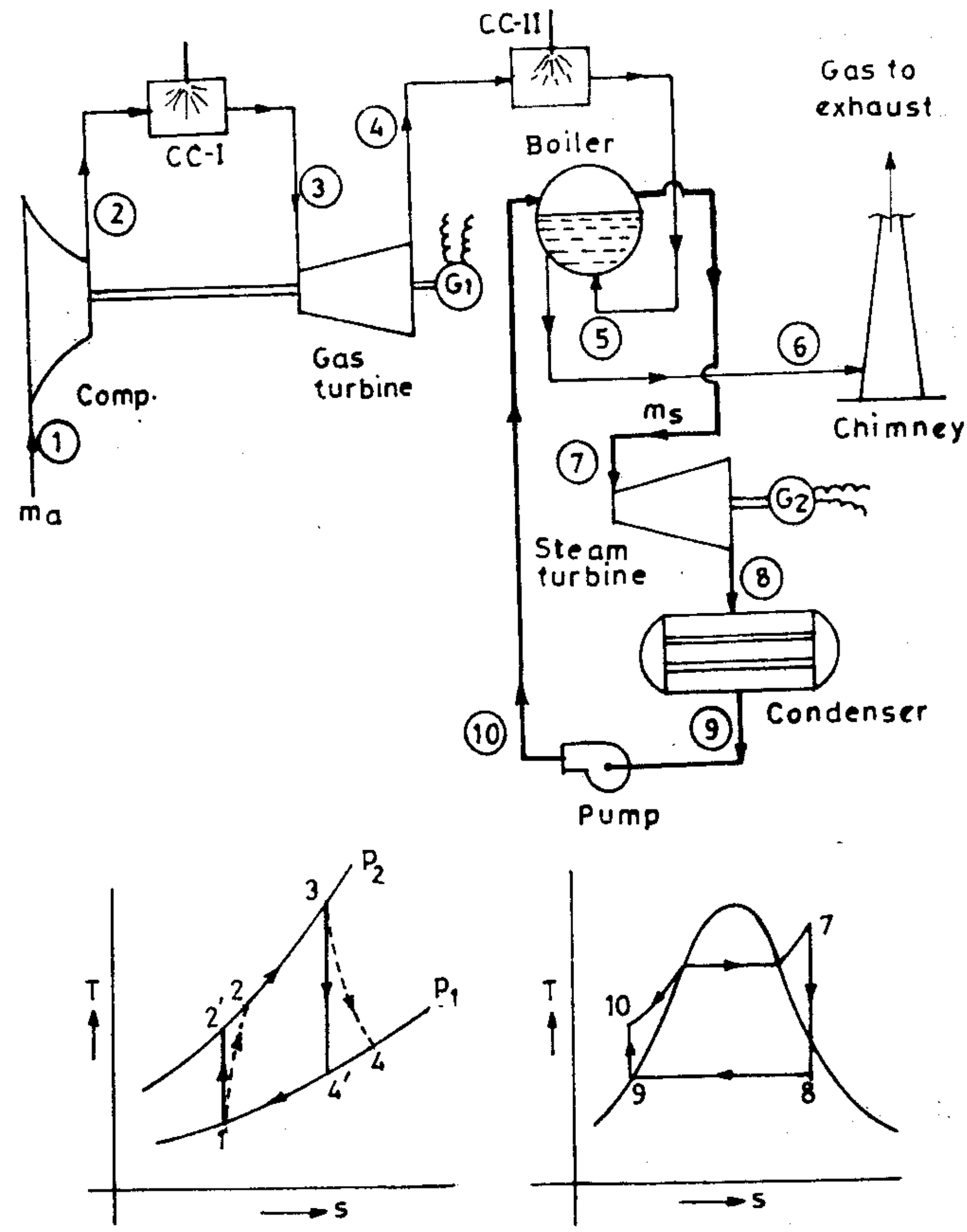


Fig. Prob. 25.4.

$$\therefore \qquad CV.\eta_{com} = \left(\frac{m_{a1}}{m_{f1}} + 1\right) C_{pg}\,(T_3 - T_2)$$

$$\therefore \quad 40 \times 10^3 \times 0.95 = \left(\frac{m_{a1}}{m_{f1}} + 1\right) \times 1.1 \times (1273 - 587)$$

$$\therefore \qquad \frac{m_{a1}}{m_{f1}} = \frac{40000 \times 0.95}{1.1 \times 686} - 1 = \mathbf{49.4}$$

Work developed in gas turbine plant = 60 MW

$\therefore \quad W_g = 60000 = (m_{a1} + m_{f1})\, C_{pg}\,(T_3 - T_4) - m_{a1}\, C_{pa}\,(T_2 - T_1)$

where m_{a1} is the mass flow in kg/sec

$$\therefore \quad 60{,}000 = m_{a1}\left[\left(1 + \frac{m_{f1}}{m_{a1}}\right) C_{pg}\,(T_3 - T_4) - C_{pa}\,(T_2 - T_1)\right]$$

$$= m_{a1}\left[\left(1 + \frac{1}{49.4}\right) \times 1.1\,(1273 - 809.5) - 1 \times (587 - 300)\right]$$

$$= m_{a1}\,[520.2 - 287]$$

$$\therefore \quad m_{a1} = \frac{60000}{233.2} = \mathbf{257.3\ kg/sec}$$

$$m_{f1} = \frac{m_{a1}}{49.4} = \frac{257.3}{49.4} = \mathbf{5.2\ kg/sec}$$

Heat generated in CC-II = Heat given to gas passing through CC-II

$\therefore \quad m_{f2}.CV.\eta_{com} = (m_{a1} + m_{f1} + m_{f2}).C_{pg}\,(T_5 - T_4)$

$m_{f2} \times 40 \times 10^3 \times 0.95 = (257.3 + 5.2 + m_{f2}) \times 1.1 \times (1273 - 809.5)$

$\therefore \quad m_{f2} = \mathbf{3.25\ kg/sec}$

$\therefore \quad m_f = m_{f1} + m_{f2} = 5.2 + 3.25 = \mathbf{8.45\ kg/sec}$

(*i*) η (efficiency of the plant)

$$= \frac{W_t}{Q_s} = \frac{W_t}{m_f.\mathrm{CV}} = \frac{100 \times 1000}{8.45 \times 40 \times 1000} = 0.296 = \mathbf{29.6\%}$$

Considering heat balance in the boiler

Heat gained by the steam = Heat lost by the gas

$\therefore \quad m_s\,(h_7 - h_9) = (m_{a1} + m_{f1} + m_{f2})\, C_{pg}\,(T_5 - T_6)$

$= (257.3 + 5.2 + 3.25) \times 1.1\,(1273 - 473) = 233860$

Now substituting the values of h_7 from h-s chart and h_9 from steam table

$m_s\,(3610 - 32.6) = 233860$

$\therefore \quad m_s = 65.4$ kg/sec

$$(ii) \quad \frac{m_a}{m_s} = \frac{257.3}{65.4} = \mathbf{3.93}$$

Cost of fuel per hour $= \dfrac{8.45 \times 3600}{1000} \times 2500 = 76050$ rupees

Energy generated per hour $= 100 \times 10^3$ kWh

(*iii*) Cost of generation $= \dfrac{76050}{100 \times 10^3} \approx \mathbf{0.76\ rupees/kWh}$

(*iv*) Quantity of cooling water required

$$= \frac{m_s \times 3600 \times (h_8 - h_9)}{C_{pw}\,(\Delta T)} \quad \text{where } h_8 = 2220 \text{ kJ/kg taken from } h\text{-}s \text{ chart}$$

$$= \frac{65.4 \times 3600\,(2220 - 32.6)}{4.2 \times 10 \times 1000} \text{ tons/hr} = \mathbf{12262\ tons/hr}$$

Problem 25.5. *A combined cycle plant is designed to develop 200 MW power. The air is taken by the compressor at 300 K and 1 bar pressure. The maximum temperature of the gas turbine cycle is limited to 800°C. The pressure ratio is 8.*

The data for the steam plant is given below. The exhaust coming from gas turbine is heated further to 800°C before entering into the boiler furnace. The steam is generated at 50 bar and 600°C. The exhaust gas temperature is limited to 200°C to avoid the condensation of corrosive gases. The condenser pressure is 0.05 bar.

Assuming the isentropic efficiencies of compressor and both turbines to be 100%, find the following :

(i) Thermal efficiency of each plant and combined plant.

(ii) Ratio of air supplied to the compressor to steam generated in the boiler.

Neglect the pump work and consider the fuel masses in the plant.

Take C_p *(air or gas)* $= 1$ *kJ/kg-K*

γ *(air or gas)* $= 1.4$

C.V. of fuel used $= 42 \times 10^3$ *kJ/kg*

Solution. The arrangement of the components of the plant is shown in Fig. 25.5 (*a*) and the cycles of operation are shown in Fig. 25.5 (*b*).

Considering the gas turbine cycle

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 300(8)^{0.286} = \mathbf{544\ K}$$

$$T_4 = \frac{T_3}{(8)^{0.286}} = \frac{1073}{1.81} = \mathbf{593\ K}$$

Heat generated in CC-I = Heat supplied to gas

$$m_{f1}.CV = (m_{a1} + m_{f1})\, C_{pg}\, (T_3 - T_2)$$

$$m_{f1} \times 42 \times 10^3 = (m_{a1} + m_{f1}) \times 1\ (1073 - 544) = 529\ (m_{a1} + m_{f1})$$

$$\therefore \quad m_{f1} = 0.0128\ m_{a1} \qquad ...(a)$$

The power developed in the gas turbine plant is given by

$$W_g = (m_{a1} + m_{f1})\, C_{pg}\, (T_3 - T_4) - m_{a1}\, C_{pa}\, (T_2 - T_1)$$

$$= 1.0128\ m_{a1} \times 1\ (1073 - 593) - m_{a1} \times 1\ (544 - 300)$$

$$= m_{a1}\ (486.1 - 244) = 242.1\ m_{a1} \qquad ...(b)$$

Considering CC-II

Heat generated in CC-II = Heat given to gases passing through CC-II

$$m_{f2}.CV = (m_{a1} + m_{f1} + m_{f2})\, C_{pg}\, (T_5 - T_4)$$

$$= (m_{a1} + 0.0128\ m_{a1} + m_{f2}) \times 1\ (1073 - 593)$$

$$m_{f2} \times 42 \times 10^3 = 480\ (1.0128\ m_{a1} + m_{f2})$$

$$\therefore \quad m_{f2} = 0.0117\ m_{a1} \qquad ...(c)$$

Considering the steam-cycle boiler

Heat gained by the steam = Heat given by the gas

$$m_s\ (h_7 - h_9) = (m_{a1} + m_{f1} + m_{f2})\, C_{pa}\, (T_5 - T_6)$$

$h_7 = 3620$ kJ/kg (from *h-s* chart)

$h_9 = 32.6$ kJ/kg (from steam table)

$$\therefore \quad m_s\ (3620 - 32.6) = (m_{a1} + 0.0128\ m_{a1} + 0.0117\ m_{a1}) \times 1 \times (800 - 200)$$

$$3587.4\ m_s = 514.7\ m_{a1}$$

$$\therefore \quad m_s = \frac{614.7}{3587.4}\ m_{a1} = 0.1714\ m_{a1} \qquad ...(d)$$

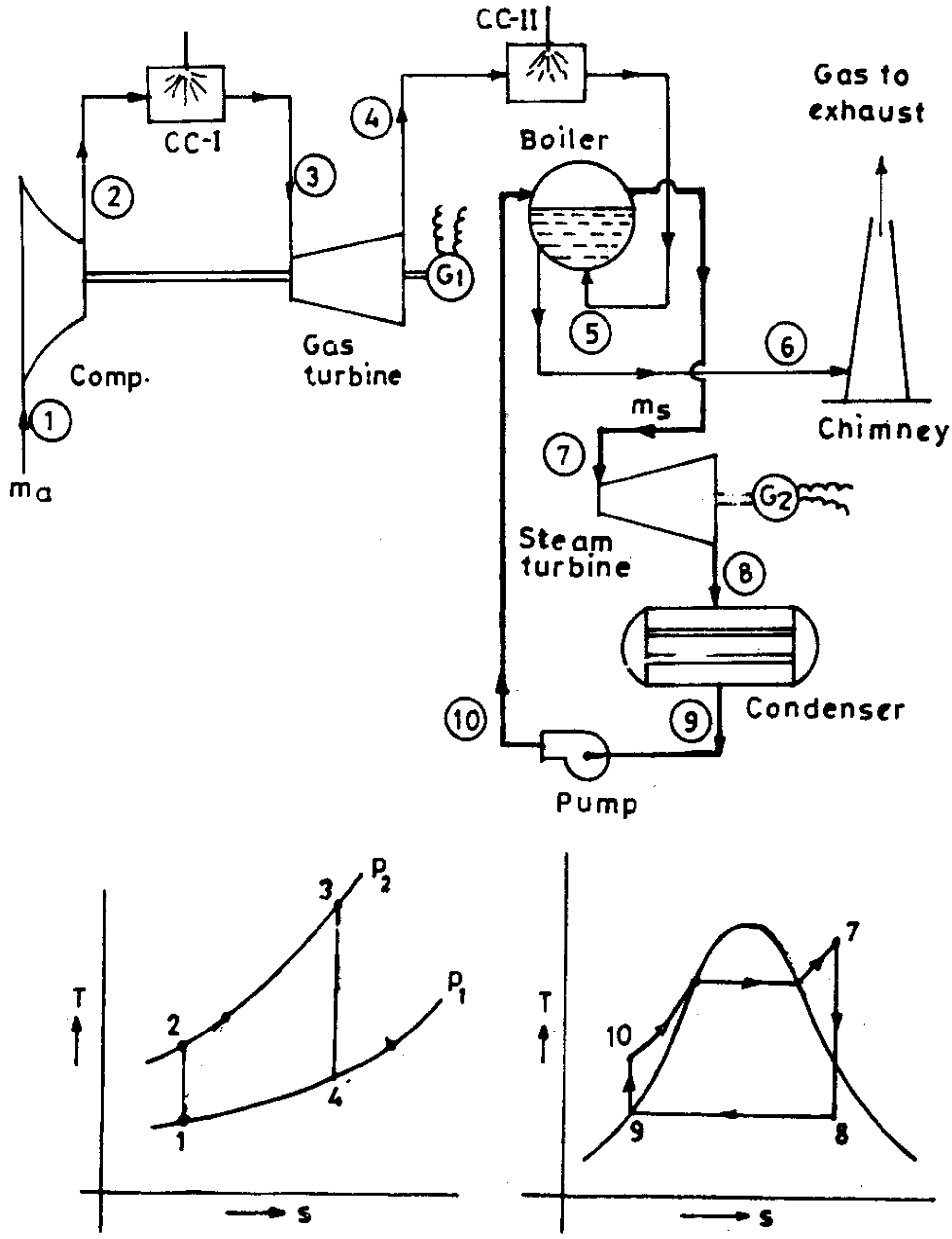


Fig. Prob. Prob. 25.5.

Power developed in the steam turbine is given by

$$W_s = m_s\,(h_7 - h_8)$$
$$= 0.1714\ m_{a1}\ (3620 - 2220) \text{ as } h_8 = 2220 \text{ kJ/kg (taken from } h\text{-}s \text{ chart)}$$
$$\approx 240\ m_{a1} \qquad \text{...}(e)$$

Now from the equations (*b*) and (*c*), we can write

$$W_g + W_s = 200 \times 10^3 = 242.1\ m_{a1} + 240\ m_{a1}$$

$$\therefore \quad m_{a1} = \frac{200 \times 10^3}{482.1} = \textbf{415 kg/sec}$$

$$m_{f1} = 0.0128\ m_{a1} = 0.0128 \times 415 = \textbf{5.312 kg/sec}$$

$$m_{f2} = 0.0117\ m_{a1} = 0.0117 \times 415 = \textbf{4.856 kg/sec}$$

$$\therefore \quad m_f = m_{f1} + m_{f2} = 5.312 + 4.8566 = \textbf{10.168 kg/sec}$$

(*ii*) From equation (*d*)

$$\frac{m_a}{m_s} = \frac{1}{0.1714} = \textbf{5.83}$$

(*a*)
$$\eta_{(overall)} = \frac{200 \times 1000}{m_f \times CV} = \frac{200 \times 1000}{10.168 \times 42 \times 10^3} = \textbf{0.47} = \textbf{47\%}$$

(*b*)
$$W_g = 242.1 \times m_{a1} = 242.1 \times 415 = \textbf{100472 kW}$$

$$\therefore \quad \eta_g = \frac{W_g}{m_{f1} \times \text{C.V.}} = \frac{100472}{5.312 \times 42 \times 10^3} = 0.45 = \textbf{45\%}$$

(*c*)
$$W_s = 240 \times m_{a1} = 240 \times 415 = \textbf{99600 kW}$$

$$\eta_s = \frac{W_s}{m_{f2} \times CV} = \frac{99600}{4.856 \times 42 \times 10^3} = 0.49 = \textbf{49\%}$$

EXERCISES

25.1. What are the major advantages of a combined cycle system in the present power picture of the world ?

25.2. Draw the line diagrams of repowering system using steam turbine only and boiler only. Discuss their relative merits and demerits.

25.3. What is the importance of gasification of coal ? Draw a neat diagram of such integration system and explain its working.

25.4. What are the relative merits and demerits of using air or O_2 in a gasification plant when the gasification plant is integrated with closed cycle ?

25.5. What do you understand by PFBC ? What are its outstanding features over conventional FBC ?

25.6. Draw the line diagrams of two different PFBC systems which are commonly used and discuss their relative merits and demerits.

25.7. What are the main difficulties faced in developing the combined cycles with PFBC ? What is LASH ? What are its specific advantages ?

25.8. Why and when organic fluid is preferred over water in the bottoming cycle ? What are its advantages ? Draw a line diagram of such a system and represent both cycles on *T-s* diagram.

25.9. Discuss the part load behaviour of combined cycle plant and compare with conventional gas turbine plant of the same capacity.

25.10. What future developments are expected in combined cycle plants ?

26
Waste Heat Recovery Systems

26.1. INTRODUCTION

The present escalating costs of fossil fuels and electric power are forcing plant operators to examine the potential saving and to reduce fuel and power uses. The need for energy recovery from low and medium heat sources has gained importance due to increasing price trend of the fossil fuels.

In the industrial community ; millions of kJ of heat is lost for every day. Such waste heat is typically in the form of unburnt but combustible waste materials, sensible and latent heat exhausted from stacks. Waste heat recovery is of prime importance to the efficient use of energy in the industrial processes. Although, present waste heat recovery systems have been non-storage type, future trend may well see a significant increase in the use of thermal energy storage. This will be necessary in order to expand the rational use of large waste heat to match temporarily the cyclic variations and to permit the possibility of energy cascading and storage to overcome imbalance in plant systems.

Recovery of heat exhausted from industrial processes and combustion equipment can often reduce fuel consumption and improve thermal efficiency, markedly. It may be used to generate steam, to heat air or water or fuel depending on the temperature, pressure and flow rate of the exhaust. When the temperatue of the waste gas exceeds 300°C, steam geneation becomes most economical method of heat recovey.

Because of the growing quantities of waste heat discharged and increasing ecological concern with energy growth, energy utilization and thermal discharge problems have stimulated an examination of methods for using energy presently wasted to the environment. In this regard, ineffective utilization of energy in building and industrial processes constitutes a major component of the national energy problem.

All the available waste heat appears as low temperature heat. However the term *utilization of waste heat* refers to the performance of useful functions with heat before it is discharged to the environment. Even though waste heat recovery systems are higher in first cost than conventional steam turbines, gas turbines or diesel plants but becomes viable on life cycle cost basis because their operating energy costs are zero.

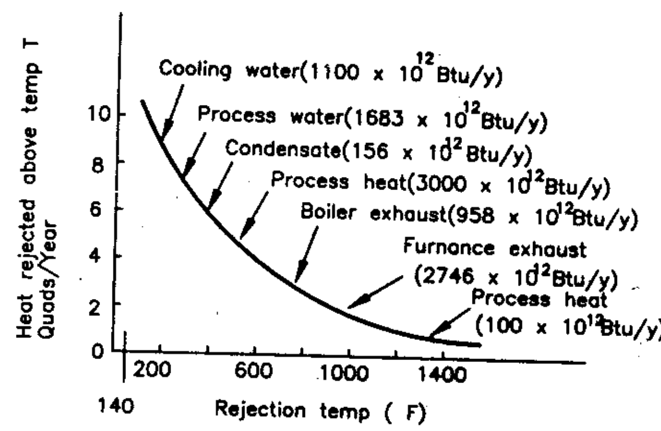
26.2. SOURCES OF WASTE HEAT AND THEIR GRADING

Waste heat is the heat which is not used and exhausted out as a waste product. Waste heat is generally available from the industry in the form of process steam and water which are discharged at higher temperature. The heat is also discharged with the exhaust gases in every type of industry. The heat which is discharged is either in the form of sensible heat (hot gases and water) or in the form of latent heat (process steam).

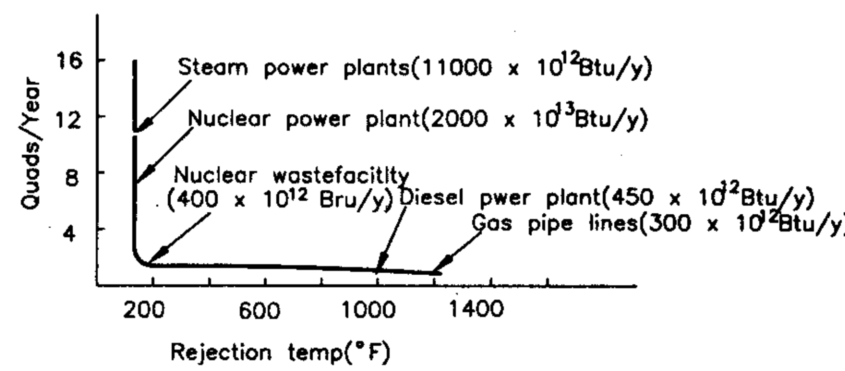
The waste heat is generally classified as low grade and high grade waste heat. The heat carried away higher than 300°C, particularly by gases, is considered high grade and heat carried away below 300°C by gas stream is classified as low grade heat. Sometimes, the recovery of low grade waste heat is not economically justifiable.

Presently USA, with only 6% of the world's population, consumes 30% of its energy. The utilization of energy that is currently discharged from U.S. industrial plants to the air and rivers of America can be a substantial energy source if it is recovered through appropriate technology. These thermal discharges amount to about 11 quads. Figure 26.1 (*a*) presents the distribution of industrial thermal discharges as a function of temperature. The current total energy consumption of the U.S. as a whole is 778 quads and of the industrial sector, 28 quads. Thus, the thermal wastes of industrial amount to 14% of the U.S. energy consumption and 39% of the industrial energy consumption. These industrial thermal wastes are attractive for conversion into useful thermal, mechanical or electric power due to the large volumes of heat involved at individual locations and temperature at which it is available. Other thermal discharges available from the utilities are shown in Fig. 26.1 (*b*). Although, the temperature potentials of these wastes are lower, they amount to about 13 quads. Thus, in the industrial and utility sectors, there are about 24 quads of thermal waste. This gives an idea of waste heat available in the world for positive utilization if proper technology is developed.

High grade waste heat is the heat carried by the flue gases with a temperature of 300°C and higher, can be readily recovered through properly designed heat transfer equipments. The use of high grade waste heat is the result of technical necessity rather than for economical reasons. The most economical method of utilization is by waste heat boiler because



Distribution of Industrial Waste Heat Flows
1 quad = 10^{15} ; Btu = 293×10^9 kWh.
Fig. 26.1. (*a*)



Distribution of Utility and Federal Waste Heat Flows.
Fig. 26.1. (*b*)

steam generated by cooling the gas can be used in the process as well as in turbines. Besides boiler, other types of heat recovery systems are used in industrial plants.

In the past, dirty gases were not considered as a source of heat energy, because related erosion, corrosion and cleaning problems could not be solved economically. Continuous increase in the price and scarcity of fossil fuel now justify dirty gases as a viable source of energy for industries as cement, iron, paper, glass and ceramic industries as potential sources for heat recovery.

Low grade waste heat is usually in the form of flue gases and drain waters. Previously, this type of heat was not considered economically feasible. But due to energy shortages and higher prices, a number of economically viable uses have been identified as new energy storage systems are developed. Low grade heat sources and recovery are commonly found in many industrial plants like aluminium, food processing and chemical industries.

Energy Cascading. The first step in the waste heat recovery utilization is to identify the waste heat sources and the heat energy users. The second step is to find out the proper technology to reuse the waste heat. The storage and transportation become important factors when source and user are located remotely. This is because, it is often not economical to transport heat long distances, especially if the temperature level of the waste heat is low. Another inherent problem in the waste heat utilization technology is the time span encompassing when waste heat is available and when heat is required. This time span presents a storage problem. Therefore, systematic approach and a careful review should be made in all areas where waste heat is available and heat resources are required.

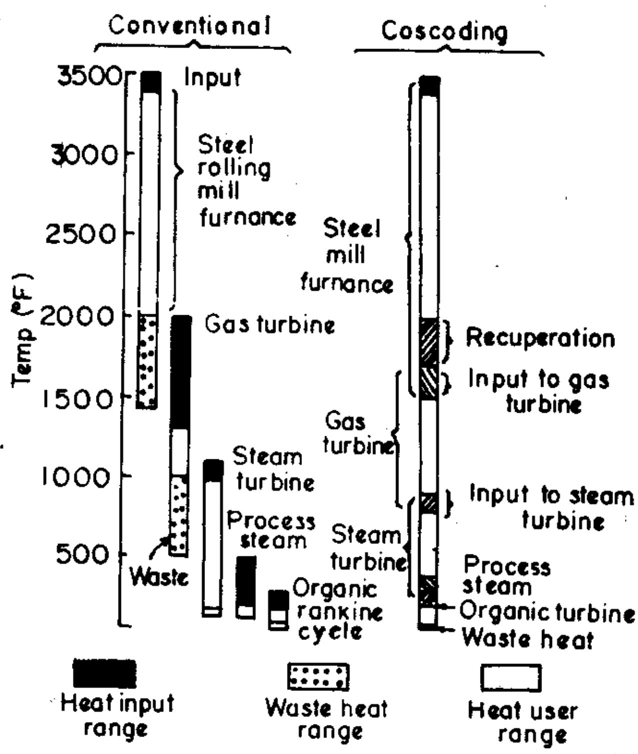


Fig. 26.2. Energy cascading concept.

To determine the feasibility of waste heat recovery, the energy mapping and cascading concept can be studied. Mapping means, a diagram of all the systems in a plant giving information about heat input and waste heat and cascading means, to pass energy from one system (exhaust) to another system (input).

Energy cascading, matching the equality (temperature) of available energy to the needs of the task, is simply a restatement of good engineering design practice based upon the Second Law of Thermodynamics. Refer to Fig. 26.2 for a clear illustration of the energy cascading concept. Five processes or cycles are listed on the left hand side. Each process uses heat at a higher temperature and rejects heat at lower temperature. The maximum temperature of heat that occurs is that of a typical flame temperature of about 2000°C. The steel rolling mill needs the full temperature available, the exhaust temperature is about 1000°C. An industrial gas turbine operates between the temperature range of 1000°C to 300°C. Steam turbine operates over a range of about 600°C to 60°C. The process steam is used in the range of 200°C to 100°C. Low temperature organic Rankine cycles can still extract reasonable amount of energy from 150°C down to the available cooling water temperature of 30°C. Each one of these five processes use only part of the temperature range available. By combining all the five processes so that the waste heat of one can be input to the next and in this way, the full temperature energy potential can be utilized.

26.3. THERMODYNAMIC CYCLES FOR WASTE HEAT RECOVERY

To recover waste heat optimally, it is necessary to bottom the heat energy of the waste heat source to the maximum extent and at the same time to convert the bottomed heat energy into power at the greatest efficiency. In addition to this, it is also becoming increasingly important to find ways to effectively use renewable thermal energy sources such as solar, sea thermal gradient and biomass. Since these sources are well as low temperature waste heat sources are at a low temperature (< 200°C), the heat engine cycle selected to use the heat must have a high efficiency at these temperatures.

Rankine cycles for waste heat recovery are attractive because they have relatively high thermal effficiency even at low temperatures compared to other dynamic energy conversion systems. This characteristic of the Rankine cycle provides a unique match for the low level waste streams. Thus, a rebirth of the Rankine cycle is taking place as energy costs escalate.

There is a thermodynamic limitation as to the amount of energy that can be extracted from the waste stream. This limitation is a function of the waste stream temperature and the temperature of the cooling medium such as air or river water. The higher the waste stream temperature and lower the cooling medium temperature, the greater is the amount of energy in both absolute and percentage terms that can be recovered. Unfortunately, in actual practice, thermodynamic limits cannot be achieved. Practical bottoming cycles can operate in the range indicated in Fig. 26.3.

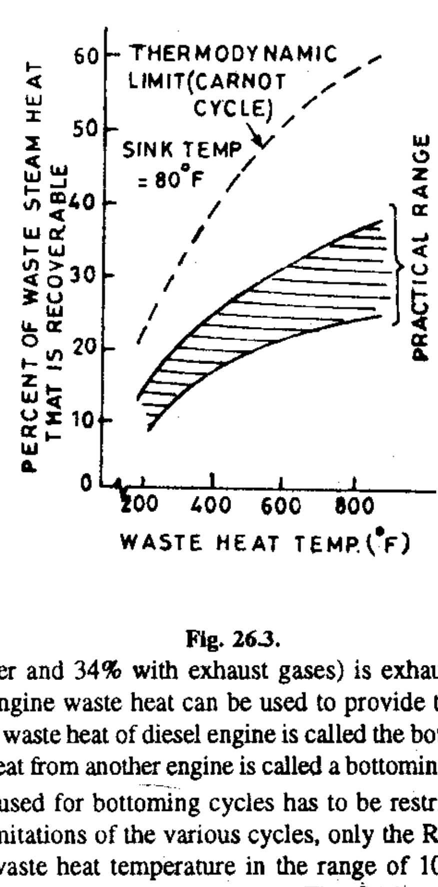


Fig. 26.3.

Any process that converts the chemical energy of the fuel to heat and work results in a loss. Some of the losses are not possible to recover due to laws of thermodynamics, others are not physically feasible to recover but some are in a form which can be used to produce further work. Considering a diesel engine, 38% of the fuel energy is converted into work and the remaining 62% in the form of heat (28% to cooling water and 34% with exhaust gases) is exhausted to atmosphere. This 62% of the fuel energy represented as engine waste heat can be used to provide the heat input to another engine. This second engine operating on the waste heat of diesel engine is called the bottoming cycle. Thus, any heat engine being powered by the rejected heat from another engine is called a bottoming cycle.

The selection of engine type and cycle that can be used for bottoming cycles has to be restricted to external combustion. Further based upon the efficiency limitations of the various cycles, only the Rankine, Stirling and Absorption cycles could be considered for waste heat temperature in the range of 100°C to 400°C. The absorption cycle can be used only for heating and cooling purposes. The Stirling engine developments are much further away than the Rankine cycle. Therefore, the only choice within the foreseeable future for bottoming cycles of high efficiency seems to be the Rankine cycle.

The Rankine cycle has a number of good features as operable over a wide range of source and sink temperatures depending upon selection of working fluid, operable with external heat addition, high ratio

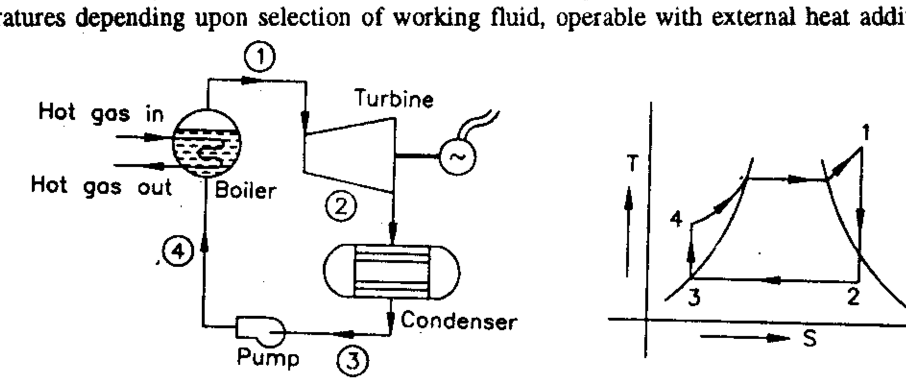


Fig. 26.4.

of cycle work output to pump work and operatable with highly reliable components such as axial expanders and centrifugal pumps.

The use of Rankine cycle using waste heat of hot gases is shown in Fig 26.4.

The working fluid used plays very important role in the operating efficiency of the Rankine cycle. Figure 26.5 shows the relationship between the temperature of the working fluid and heat source versus heat transferred. The line *ab* shows the temperature drop of heat source flowing in a counterflow heat exchanger and the line *cf* represents the temperature rise of a working fluid flowing in the opposite direction. If water is used as a working fluid with medium temperature heat source, due to high latent heat of vaporizaton of water, the temperature rise of the working fluid caused by the temperature drop *ab* is represented by the line *cdef*. Steam of a high pressure is hard to generate and this results in lower recovery from the available waste heat.

If the aim is to raise the steam pressure for higher power recovery, the temperature rise of the working fluid will be represented by the line *cd'e'f* which causes the temperature drop of the heating gas as shown by line *ab'*. This results in low heat recovery. Any attempt made to recover more power from the heat available by increasing the pressure of the working fluid in the Rankine cycle, the heat recovery from the waste heat is reduced and ultimately the purpose is lost.

The only possibility to achieve the above aim is to use a refrigerant with a low boiling point and small latent heat of vaporization as working fluid in Rankine cycle (Fl_2 is superior to NH_3 as its latent heat is considerably less). The temperature rise of a refrigerant against the temperature drop of the waste heat *ab* is shown by the line *cghf*. This means that the pressure and temperature of the working fluid can be high enough for efficient power recovery. The power recovery from waste gas by R-85 and steam are shown in Fig. 26.6. This indicates an importance of using refrigerant as working fluid in the Rankine cycle for converting waste heat into work in the lower and medium temperature range.

In case of low and medium temperature waste heat recovery plants, the temperature difference between the heat source and working fluid is always small and requires comparatively large heat exchangers or boilers which have big masses to accumulate heat.

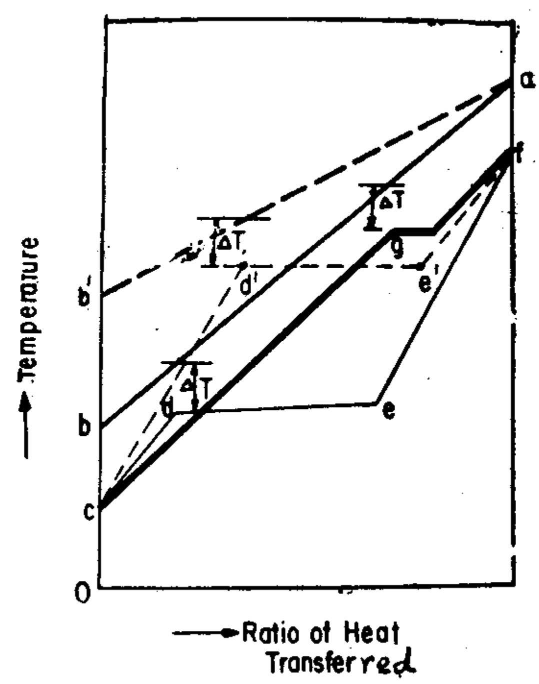


Fig. 26.5.

A 14 MW Rankine cycle waste heat recovery power plant was delivered to the Kimitsu Works of Nippon Steel Corporation, one of the largest mills in 1981. The plant recovers electric power from waste gases of the sintering process. This plant uses R-85 as working fluid and recovers 20 to 30% higher power from low and medium temperature heat source (100°C – 250°C) than did earlier techniques. The plant is equipped with a modern well developed micro-computer aided automatic control system which allows fully unattended operation and remote start-up and shutdown.

The capital cost in converting waste heat into useful work is one of the major considerations in selecting the equipments required. It has been shown that there exists a cycle temperature that produces a minimum equipment (capital) cost for Rankine system as shown in Fig. 26.7. It is interesting to point out that the minimum cost of Rankine cycle coincides with current steam power plants.

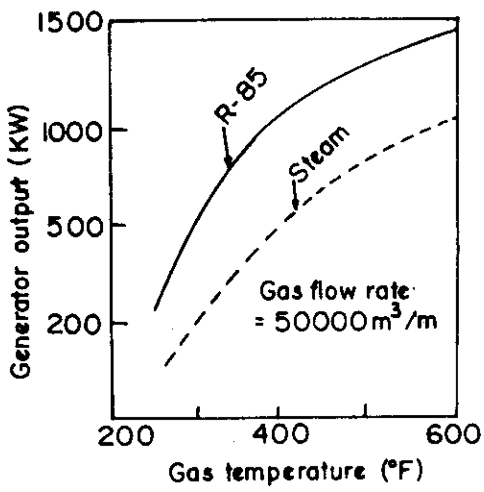


Fig. 26.6.

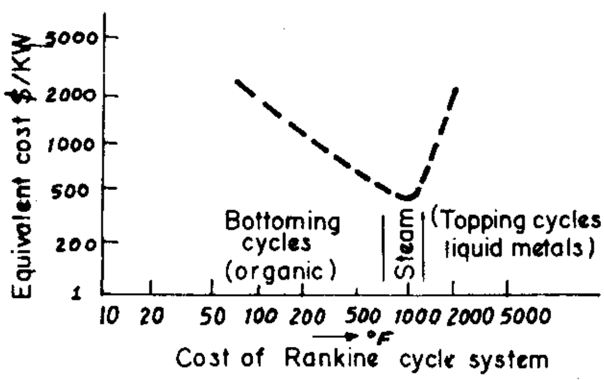


Fig. 26.7.

There are two factors which would tend to push the consideration of Rankine cycle away from the minimum cost point. In the direction of the higher temperatures, the factor is higher and overall efficient use of the thermodynamic potential of fuel. However, these higher temperatures imply the use of liquid metals which require the use of higher temperature materials which are more and more expensive to fabricate. Erosion and corrosion are also the problems in the use of liquid metals. This was the main reason for discarding mercury-steam system. In addition to this, there are other cycles (Brayton, Stirling) that offer higher efficiency at higher temperatures than does the Rankine cycle. Combining any of these cycles with a lower Rankine cycle in a cascaded system can result in a higher cycle efficiency.

In the direction of lower temperatures, the factor is the use of free waste heat. The exclusive use of low temperature Rankine cycles means low overall thermal efficiency and large heat exchangers. This is the cause of increasing the capital cost of the Rankine cycle but at lower temperatures, there are no other alternative dynamic systems.

There is economic regime for Rankine cycles which depends on the temperature of the stream and the size of the stream (amount of waste heat). The temperature affects the overall thermodynamic efficiency of the cycle and governs the amount of power that can be recovered for a given stream size. Stream size affects the total amount of power that can be recovered at a given temperature. More power per unit piece of machinery reduces capital cost per kW. This economic trade is shown in Fig. 26.8 for both a condensing vapour waste heat source and hot gas source.

Presently the steam plant cost is $ 1000/kW and nuclear plant costs $ 1200/kW. These costs are comparable to waste heat system as shown in Fig. 26.8 which lies between $ 800 to 1000/kW if the range of waste heat lies between 260°C to 100°C. In addition to this, the waste heat recovery system does not use any fuel.

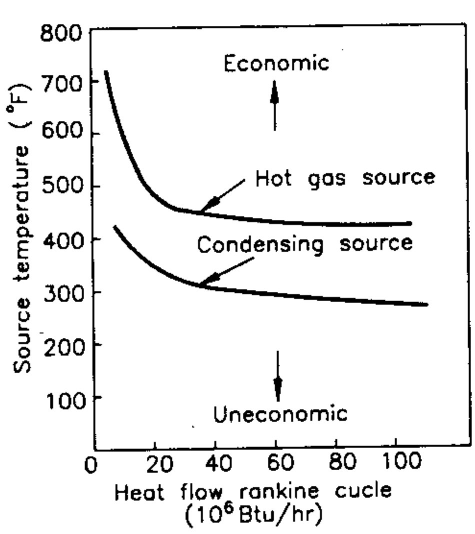


Fig. 26.8. Economic application of Bottoming cycle.

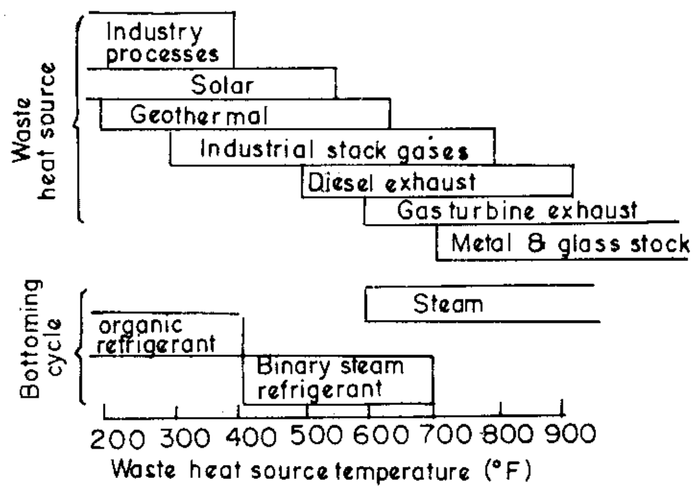
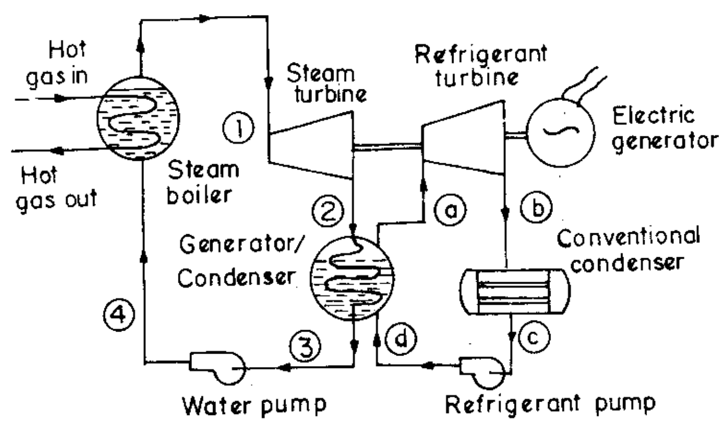


Fig. 26.9. Waste Heat Sources and Bottoming Cycle Systems.



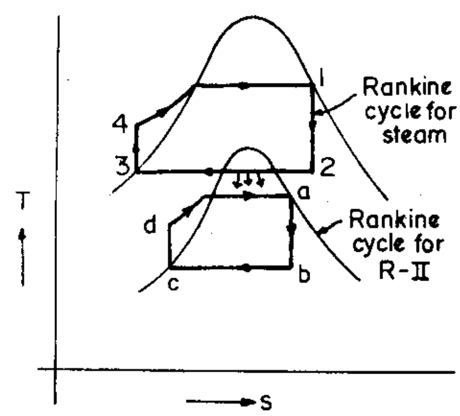


Fig. 26.10. (*a*) **Fig. 26.10.** (*b*)

There are number of fluids which can be effectively used in a Rankine cycle in variety of applications as shown in Fig. 26.9 using a variety of heat sources.

The principles of energy cascading are also receiving greater acceptance as energy costs escalate. The philosophy of energy cascading encourages the use of combined cycles as shown in Fig. 26.10 in order to increase overall system efficiency and to use more effectively the quality of energy sources.

The use of two fluids permits each fluid to operate in a range where it is most effective and maximises the use of the thermodynamically available work.

The combined cycles are more complex and their reliability plays a key role in their acceptance.

A binary Rankine bottoming cycle has been developed at MIT electrical utility at Rockville, New York. Using steam and R-11 as working fluids for using the waste heat from an exhaust of 5.5 MW diesel generator set, available at 300°C. The use of water avoids placing flammable or toxic fluid in the hot heat exchanger in the diesel exhaust and the selection of R-11 avoids the problem of a large condensing heat exchanger with high vacuum steam. The waste heat recovery system used at this station has increased the power output by 10% without any increase in fuel consumption.

MIT has presently developed hermetic power generator system, each of 2.5 MW capacity. The hermetic feature greatly simplifies the equipment and significantly improves reliability.

Working Fluids for Bottoming Cycle. One of the key design decisions is the selection of the working fluid or fluids to match the available source and sink temperature. In waste heat recovery, both source and sink temperatures are constrained by existing facilities and processes.

The working fluids which are used in the bottoming cycle or in cascade system must possess the following properties :

(1) Non-toxic (2) non-flammable (3) reasonable priced (4) readily available (5) no psychological barriers and (6) good thermodynamic properties.

The properties of few fluids which are commonly used are listed below.

Critical temp. and pressure						*145.7°C & 32.6 bar*
Properties ↓	R-12	R-11	R-85	Water	NH_3	R-114
Molecular weight			85.74			170.93
Specific gravity			1.37			
Boiling point at 1.013 bar			76.1°C			3.6°C
Freezing point			– 63.3°C			
Latent heat at 30°C						
Pressures at 100°C & 30°C						
Saturation temp.						
Latent heats at 100°C and 30°C						
Saturation temp.						
Thermal Stability			Stable (315-380°C)			
Flammability			Non-flammable			
Toxicity			Non-toxic			
Corrosiveness			Less than water			

26.4. HEAT RECOVERY FORMS AND METHODS

In most of the cases, the heat from the waste hot streams is recovered either in the form of sensible heat or latent heat. When the heat recovery takes place in the form of sensible heat, the fluid used for carrying heat is either air or water. In this case, the heat recovered from waste hot stream is used for heating the air which can be used either for combustion or for winter air-conditioning.

The hot gases coming out from the metal furnaces are used for heating the steel scrap. The waste heat can also be used for heating the air even upto 300°C which is further used for melting the metals by burning the fuels. The effect will be considerable saving of fuels in the combustion chambers of the furnaces. The possible saving in the fuel with increasing temperature of preheated air as parameter is shown in Fig. 26.11.

(1) Sensible Heat Recovery. The gas-air heaters which are commonly used are of counter-flow type. The temperature variations of the fluids are shown in Fig. 26.12. The transfer of the heat depends upon $(T_{ho} - T_{ci})$ and $(T_{hi} - T_{co})$ values as $T_{ho} > T_{ci}$ and $T_{hi} > T_{co}$ are the necessary conditions for the transfer of heat in counterflow heat exchanger. These conditions limit the percentage of heat recoverable from the exhaust gases.

The heat transfer Q is given by

$$Q = m_g \, C_{pg} \, (T_{hi} - T_{ho}) = UA \, (\Delta T)_m$$

$$\therefore \quad A = U \left[\frac{Q}{(\Delta T)_m} \right]$$

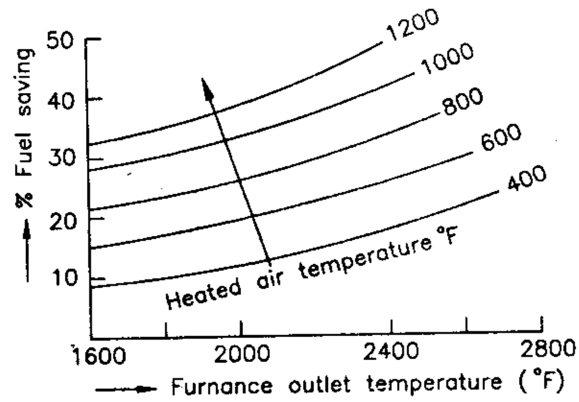


Fig. 26.11.

Fig. 26.12.

Now by selecting various values of T_{ho}, we can calculate the percentage increase in recovery and the relative area required for each percentage. The results are shown in Fig. 26.13. It can be seen from the figure that to increase the recovery from 40% to 60%, area increase is 71% but when recovery increases from 40% to 80%, increase in area is 216%. To double the heat recovery the area is to be increased by three times.

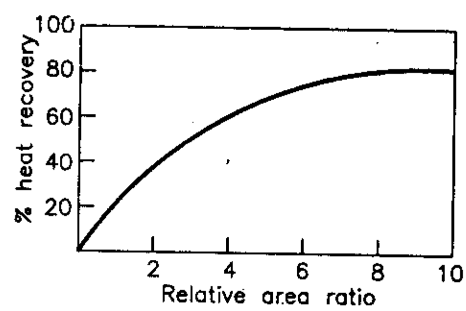


Fig. 26.13.

The recovery can be increased by increasing the value of U but at the expense of increased running cost because of higher pressure loss due to increased velocity. In some cases, like gas turbines, even above method is not permitted as there is a limitation on back pressure.

The optimum heat extraction should be done by increasing A as well as U in such a way that the extraction of heat costs minimum.

(2) Latent Heat Recovery. This is most common and versatile method of waste heat recovery when used for power generation. Because, waste heat boilers can be found virtually in any size and in the pressure range of 2 to 50 bar, the advantages of recovering the heat in latent form are listed below.

(1) This form of recovery is attractive when latent heat of working fluid is high.

(2) A reasonably low temperature difference (5°C) can be maintained even when percent heat recovery is high.

(3) The heat transfer coefficients of vaporizing sides are very high (50—100 times higher than air).

(4) The use of fluids with lower boiling point than water (refrigerants) is possible to recover the heat from low temperature streams.

As the fluids used in this mode of heat recovery change the phases during the cycle process, they offer special advantages as listed below.

(1) They require smaller pipe work as they operate in liquid phase which has high density.

(2) It gives higher values of U.

(3) The system pressure can be kept low (4-5 bar) as most fluids have low vapour pressures at operating temperatures.

26.5. OTHER USES OF WASTE HEAT

Another area, where large quantity of heat at lower thermal potential (70°C) is discharged to the atmosphere, is the heat dissipated by the thermal and nuclear plants. There will be very little waste heat at temperatures above river water which cannot be effectively and economically collected by cascading. Therefore, new applications should be developed to use large amounts of very low temperature waste heat. A few of them are listed below.

Many agricultural uses as spray irrigation, soil heating, environmental control of animal shelter and green houses offer a way to use the thermal discharges without affecting electrical energy production.

(1) Agriculture. Control of temperature for increasing agricultural yield is very recent. Spray irrigation and soil heating can be used to lengthen the growing season and prevent frost production in certain regions.

The temperature is one of the most important factors which govern the germination of seeds. Favourable temperatures at the seeding growth stage enhance the growth and increase the yield. It has been observed that soyabean production increased from 2.25 to 3.75 tons-acre when the soil was heated. Similar yield growth with varying percentages is also observed with tomatoes, bush-beans and many others. Heated water is used for soil sterilization which retards certain diseases in cotton and improves the fibre quality.

(2) Green Houses. It has been suggested that the green house might be constructed adjacent to nuclear plants in cold countries to use the waste heat and it may replace cooling towers which would be required otherwise. The use of waste heat for green houses climates is the use of fuel without affecting the efficiency of the power plant. Crop yield can be improved through the use of low temperature waste heat in a controlled glass or plastic houses providing an added incentive for the use of waste heat.

Figure 26.14 shows the recent green house arrangement where hot waste water is used for heating the air circulated in the green house. The present green houses are made of plastic which reduces the heat loss considerably. The air temperature and relative humidity required for the different vegetables are controlled by controlling the quantity of hot water passing through the heat exchanger and wetting pads.

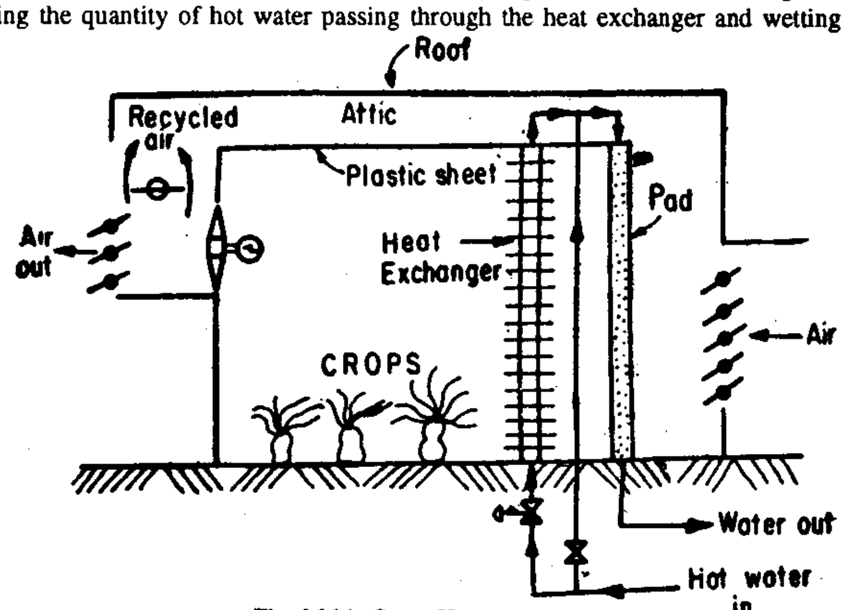


Fig. 26.14. Green House Arrangement.

Vegetables cultured at warm daytime temperatures of 30°C to 40°C and night time temperatures of 20 to 25°C include squash, water melon and cucumbers. In the temperature range of 20 – 75°C (daytime) and 15 – 17°C (night time) are tomatoes, peppers, egg-plants and onions.

The air and water quantities with their conditions for using evaporative pads for green houses and replacing cooling towers of the Fort St. Vrain nuclear plant of 330 MW capacitive are listed below.

Ambient condition		Air flow kg/hr	Water flow kg/hr	Green house condition		Range of water
DBT°C	RH %			DBT°C	RH%	Temp. °C
35	16	140×10^3	40090	24 – 30	80 – 67	19.5 – 32

The gains might be realised in green house operation if sizable fraction of the waste heat from power station is used otherwise interest on the investment will be too large. A glass house which uses 25% of the waste heat of 100 MWe plant requires $ 25 million and occupy about 250 acres of land. There is one such establishment with single operation in Hungary presently working.

This use of waste heat from the power plants will not solve thermal pollution problems as the use of heat through this mode is hardly 5% of the total thermal discharge of power plant in USA. But it can reduce the impact of thermal effluents on the local ecology, conserve energy resources and save money for both electric utility and agricultural operation.

In case of nuclear plants, the hazards of radioactivity on the crops must be considered and public acceptance of the products would have to be analysed before going for such systems.

(3) Animal shelters. The growth rate of some animals is strongly influenced on the environmental temperature. Proper control of temperature using waste heat can decrease feed consumption and increase productivity. This is particularly more effective for small animals like poultry and swine.

(4) Aquacultural Uses. The fish species are intensively cultivated in controlled environments. In a pond with supplemented feeding, the yield increased to 2000 lbs/acre-year compared with 100 to 600 lbs/acre-year in a pond with phosphorus fertilizer. Most impressive is running water with intensive feeding as used in Japan which gave 0.3 – 0.8 million lbs/acre-year. Dissolved O_2 and nutritional adequacy are some of the factors in addition to temperature for their growth rate.

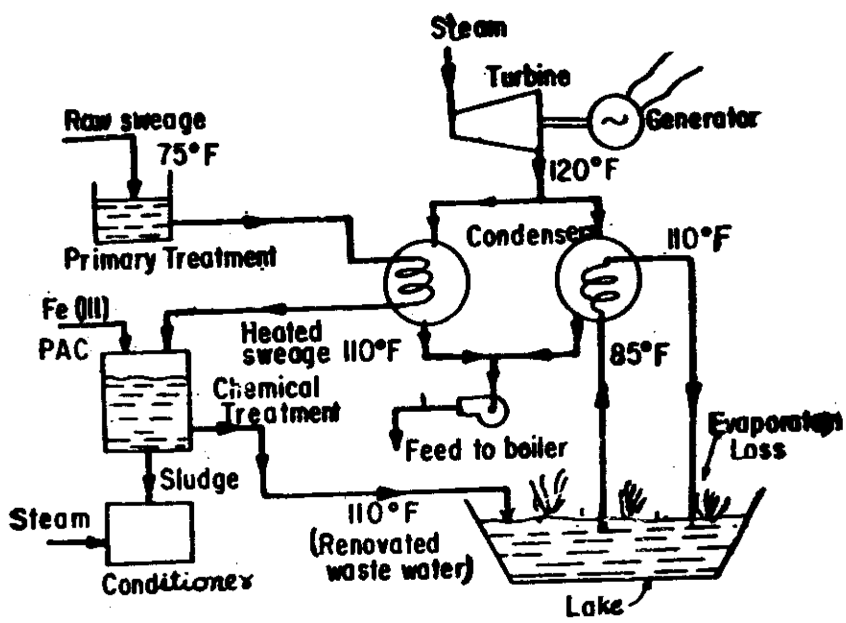


Fig. 26.15. Waste-water treatment cooling lake complex.

(5) Waste Heat Utilization for Waste Water Treatment. Municipal and industrial waste water effluents are growing rapidly. They are increasingly being looked upon, not as pollutants but as secondary water resource if treated properly using waste heat. This can be used as cooling water in power plant. A system used for treating waste water with waste heat at a power plant is shown in Fig. 26.15.

26.6. HEAT PUMP SYSTEMS

A conventional heat pump system is shown in Fig. 26.16. A heat pump takes heat from a low energy source (T_1) and pumps to the sink at high temperature (T_2). The sink is always a system which is to be heated in winter season. The coefficient of heat pump is given by

$$COP = \frac{Q_2}{W} = \frac{Q_1 + W}{W} = 1 + \frac{Q_1}{W}$$

It is obvious from the above expression that a heat pump always delivers more heat energy than the work energy used. This is why a heat pump is attractive.

Heat pumps can take heat from waste streams and deliver that heat to the process at required process temperature. Electric heat pumps for industrial use are not new. But engine driven heat pumps which use engine exhaust as a heat source are under development and are expected to come into the market as energy costs escalate. [The readers are requested to see the chapter on heat pumps from the book Refrigeration and Air-conditioning by the same authors for further details].

The effect of source temperature on the COP taking sink temperature as parameter is shown in Fig. 26.17.

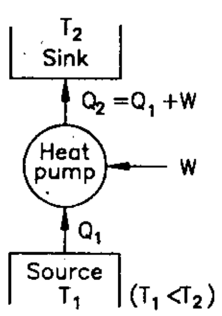


Fig. 26.16.

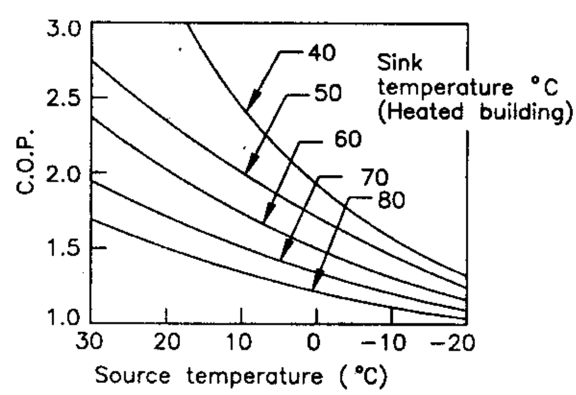


Fig. 26.17. Performance of heat pump.

26.7. DIFFERENT WASTES FOR POWER GENERATION

The waste material which can be used as fuel for power generation includes mainly municipal waste, industrial waste, paper waste, plastic and rubber waste and many others. These wastes created the dumping problems as well as pollution problems in past as they were never considered as substitute fuels for thermal electric power plants. The waste potential particularly of big cities like London, New York, Bombay is considerably large and therefore it can be used as fuel and its use as fuel will also solve the problem of dumping and pollution. With the escalating prices of conventional fuels, the use of waste was thought an economic proposal for power generation.

Figure 26.18 shows the heating values of some of the wastes. The energy content in most of the fuels is high and that great amount of energy content can be recovered if used as fuel. The mean calorific value of the wastes is approximately 12000 kJ/kg.

The wastes mentioned above can be used for any one of the uses listed below.

(1) Power generation.

(2) Steam generation which can be used for different industrial processes.

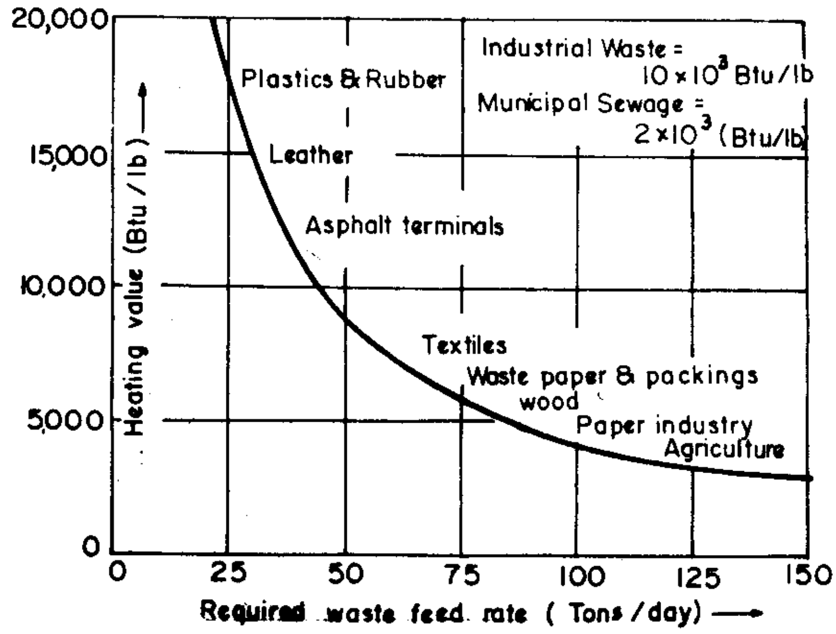


Fig. 26.18.

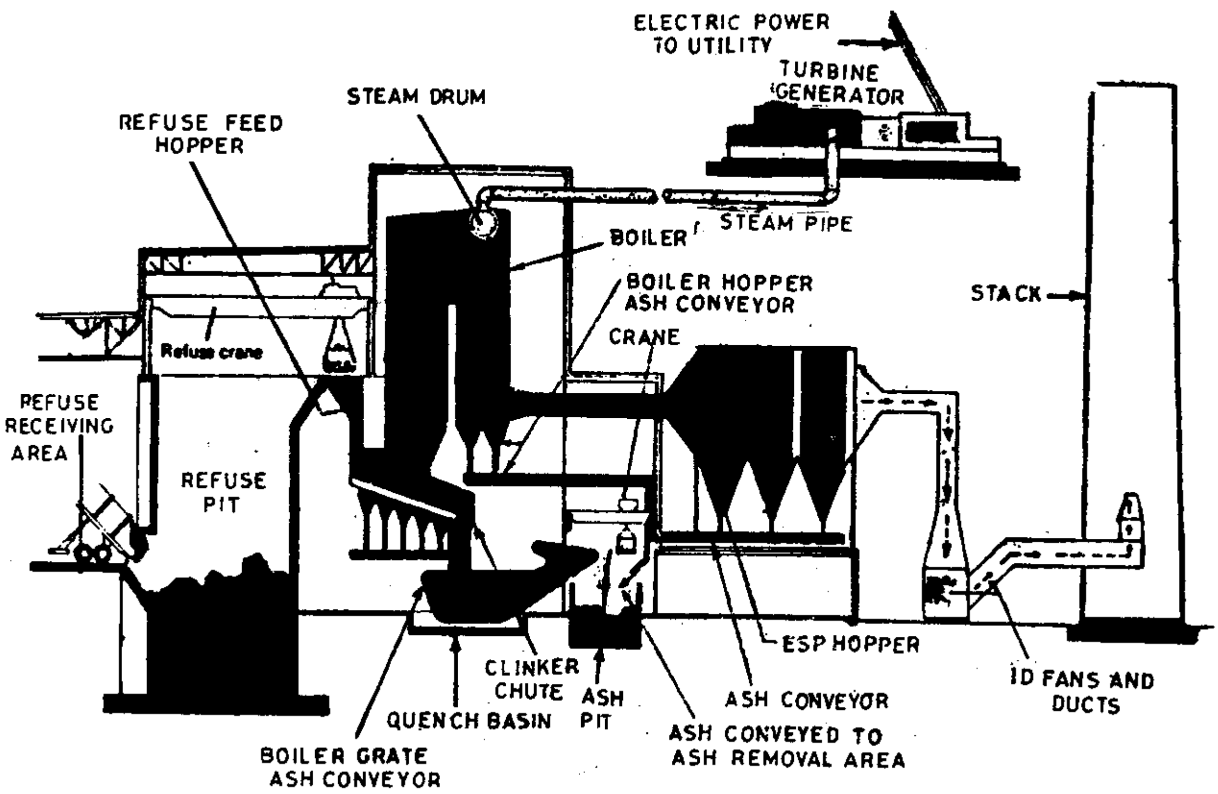


Fig. 26.19.

(3) It can be used as a supplementary fuel in boilers.

(4) Used for district heating and cooling processes.

(5) Gas production.

There are three main points to be considered in using waste as a fuel.

(1) It requires extensive preparation and cost before using it as a fuel like drying, separating and compacting.

(2) It creates unconventional air pollution as the types of the gases generated during combustion are very different than SO_2 and NO_x which create more problems of corrosion and health problems.

(3) The ash produced is highly sticky in nature particularly when rubber and plastic are used as fuels. It prevents power slag flow and clogging takes place frequently if proper temperature is not controlled.

The Peekskill Power Plant (New York) on the bank of the Hudson River using waste 1200 tons/day is considered the largest mass burning waste to energy plant in USA. A components arrangement of the system is shown in Fig. 26.19.

Another notable plant is working at Iiilich of 1.5 MWe capacity using rubber and plastic wastes at the rate of 400 kg/hr.

Another way to use the waste is to produce the gas out of waste burning in a controlled atmosphere and then use the same for power generation. Such a system developed at Andco-Torrax (England) is shown in Fig. 26.20.

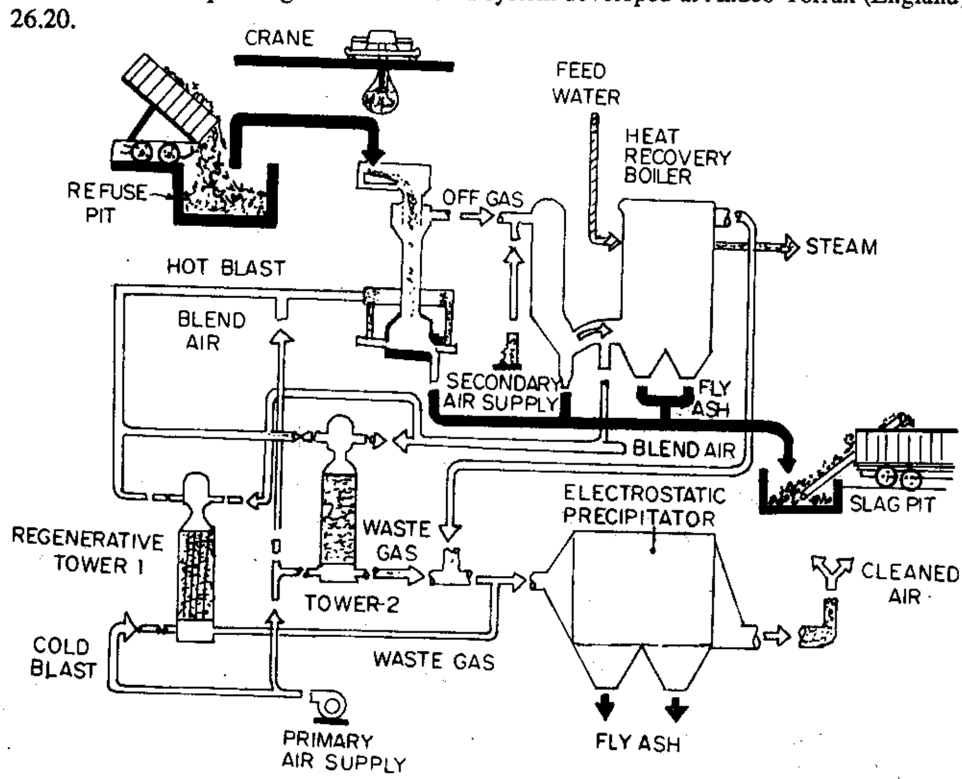


Fig. 26.20.

More recently in USA, the proposal has been to burn refuse milled to 50 mm in pulverised fuel-fired boilers. Much of the development work on this process was done on a 100 MW unit at Meramec Station using refuse from the city of St Louis, 20 miles away from the station. The rated refuse burning capacity is 325 ton/day.

26.8. HEAT RECOVERY AND WASTE HEAT BOILERS

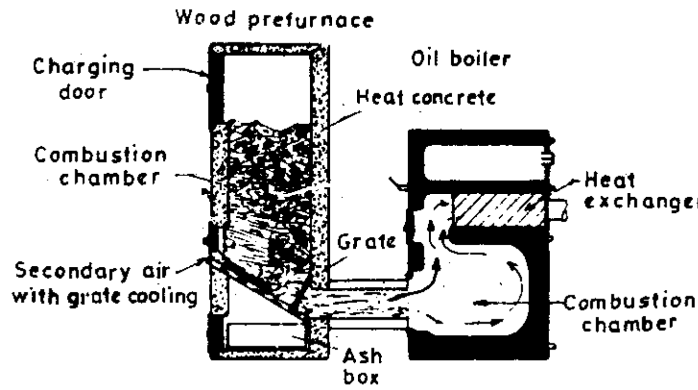
This article includes those boilers which either use the waste heat in gases coming out of diesel engines and gas turbines at high temperature or use the waste as a fuel in the incinerators. The boilers which use industrial dirty gases for power generation are not included as it is a subject of its own.

Heat recovery boiler is the most convenient and widely used method of recovering heat from high temperature streams as it gives lowest installation cost, compact heat recovery systems and no-operating problems.

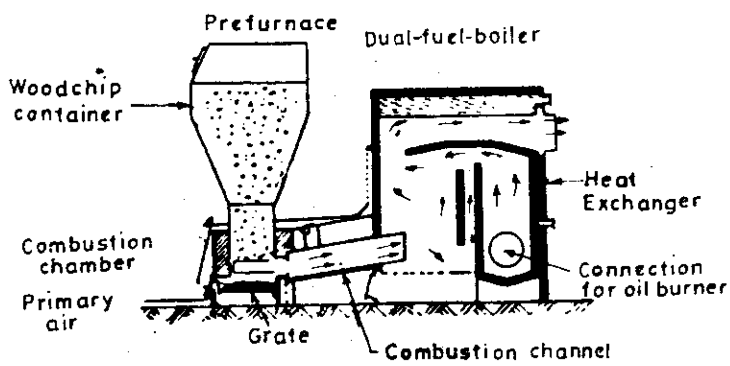
Fire tube boilers are used to recover heat from low pressure gases exhausted by diesel engines or gas turbines. As a general guide, shell and tube boilers can be used for dusty gases where the gas temperatures are well below the dust and ash softening point. The gas flows are below 2000 kg/hr and steam pressures below 40 bar. Outside these limits, water tube boilers are preferred.

Natural draught is generally used for shell and tube, bidrum or tridrum boilers and forced draught is preferred for coil and horizontal multi-loop boilers.

When hot gases from the high temperature furnaces are to be used, the basic problem associated is the removal of dust and dirt carried by the gases otherwise heavy fouling on the boiler surfaces hampers the heat transfer. A big steel work generates nearly 10^6 m^3 of gas per hour in the temperature range of 400 to 800°C. This itself gives an idea about the waste heat potential available for power generation. It has been estimated that waste heat recovery in the industry of U.K. has a potential for saving between 6 to 8 million tons of coal equivalent a year.



(*a*)



(*b*)

Fig. 26.21.

Addition of Mgo to the gases from the furnaces before entering into the boilers is essential as it modifies the nature of the deposits to make them easier to remove. Without addition of correct Mgo, the sulphate, pyrosulphate and bisulphate would combine to form a sticky eutectic which is very difficult to remove from the tube surfaces.

Arrangement of a boiler using waste heat from the furnace gases (278 m/min at 1260°C) is shown in the Fig. The boiler capable of producing 435 kW indicates the saving of \$ 100 × 10^3/year.

Dry biomass can also be used without thermal processing for heat and power generation. It has been observed that power generation from straw and wood is more economical in Germany in some instances. It is more important for developing countries with large resources of biomass or for countries that are able to grow biomass for energy. The only problem in using this as fuel is to modify the combustion chambers. Two typical biomass burning systems are shown in Fig. 26.21 (*a* & *b*).

EXERCISES

26.1. What are the different economic sources of waste heat ? How are they graded ?

26.2. What do you understand by cascading ? What is the importance of cascading in waste heat recovery system ?

26.3. Why Rankine cycle is preferred in waste heat recovery systems ? What is the importance of fluid used in waste heat recovery ? Illustrate your answer with neat graphical representation.

26.4. What do you understand by economical extraction of waste heat ? How the working fluid affects the economy of heat extraction ?

26.5. Draw a neat diagram of a cascade system for extraction of waste heat and to generate the power ? Why such cascading is preferred for waste heat from gas turbine plant ?

26.6. Draw a Cascade system used for extracting heat when steam turbine is used as basic prime mover ? What type of fluid will be used in the secondary circuit?

26.7. Represent the cascade systems on T-s diagrams when Gas turbine and steam turbine are used and steam-turbine and refrigerant turbine are used for waste heat recovery purpose.

26.8. What are the basic requirements of a fluid used for bottom cycle ? Discuss the merits and demerits of R-11, R-12, R-114 and R-85 when used in bottoming cycle for waste heat recovery

26.9. Discuss the other uses than power generation of waste heat recovery in details.

26.10. Explain the importance of heat pump for waste heat recovery purposes.

26.11. What are the basic features of a waste heat boiler compared with conventional ?
Draw a neat sketch of a boiler where waste wood is used as a fuel.

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