ChemCon'04 Mumbai

Thermodynamic cycle analysis of combined cycle heat engines

Amitabh Babar[#], Pratik Donde[#], Bhanu Samyal[#] and Santanu Bandyopadhyay Name[*, $]

[#] *Department of Mechanical Engineering, Bharati Vidyapeeth Engineering College, CBD Belpada, Navi Mumbai 400 703, India.*
[*] *Energy Systems Engineering, Department of Mechanical Engineering, Indian Institute of Technology, Bombay, Powai, Mumbai 400 076, India.*
[$] *Corresponding author. Tel.: +91-22-257167894 extension: 7894. Fax: +91-22-25723480.*
E-mail address: santanu@me.iitb.ac.in

Abstract

Thermodynamic analysis of combined cycle power plants from the point of view of obtaining maximum power has been carried out in this paper. Combinations of reciprocating and continuous heat engines in combined cycle power plants have been analytically examined for this purpose. It has been observed that efficiency, which corresponds to the operating cost per unit of power produced, while producing maximum power, is the same for all combinations of combined cycle engines considered in this paper and is equal to that of an internally power optimized heat engine. The corresponding power developed, however, varies; the highest power being produced by the Continuous-Continuous combined cycle heat engine and the least by the Continuous-Reciprocating combined cycle heat engine [Ref. Table 1]. For simplicity and consistency, all configurations of heat engines are considered to be internally reversible.

1 Introduction

An internally reversible heat engine is a simplification of a real engine wherein thermal irreversibility is concentrated at the heat exchangers. This allows the remainder of the engine to be considered as reversible. A combined cycle engine is used where the available temperature drop can not be efficiently harnessed by a conventional single cycle heat engine. A simple combined cycle heat engine is formed when two or more heat engines are connected in series such that the heat rejected by one is fed to the other in part or whole. The cycle connected to the high temperature heat sink is called the topping cycle whereas the one connected to the low temperature heat reservoir is called the bottoming cycle. From the practical point of view, the operating temperature of steam turbines commonly used for power generation (working close to the Rankine cycle), is significantly lower than the maximum available temperature (that can be practically produced) due to severe stress and creep problems associated with boilers and turbines. This has lead to the commercial exploitation of combined cycle power plants wherein the topping cycle is generally a gas turbine cycle or mercury cycle whereas the bottoming cycle is the usual Rankine cycle.

Single cycle internally reversible heat engines were first analyzed by Chambadal [1] and Novikov [2] and then by Curzon and Ahlborn [3]. The efficiency of such an engine at the condition of maximum power production is

$$\eta_{CA} = 1 - \sqrt{\frac{T_C}{T_H}}$$

Similar power optimizations involving combined cycle heat engines consisting of two continuous cycles have been carried out by some researchers [4, 5, 6]. The aim of this literature is to extend this analysis to combined cycle heat engines involving combinations of continuous and reciprocating engines and establish a basis for the comparison of the same.

The important assumptions made in the analysis are as follows:

1. All engines are internally reversible. This means that irreversibility is assumed to be concentrated at the heat exchangers only.
2. The thermodynamic cycles constituting the engines are assumed to be reversible Carnot cycles.
3. All combined cycle engines are assumed to work between similar thermal reservoirs.
4. The rate of heat exchange at the heat exchangers depends upon the temperature difference and the conduction coefficient K such that K=UA. The value of the overall coefficient of heat transfer U covers up the thermal resistance offered during heat transfer by conduction, convection as well as radiation.
5. The heat exchangers are perfect, i.e., no heat is lost to the surroundings.
6. The total thermoeconomic input which is dictated by the arithmetic sum of the conduction coefficient for individual heat exchangers, for all cases, is the same.
7. All heat engines are analyzed at the condition of maximum power production.

2 Internally reversible heat engine

The schematic diagram of an internally reversible heat engine is shown in Figure 1. The engine is assumed to deliver power continuously. A practical example of such an engine is the gas or steam turbine, which while consuming heat continuously, deliver uninterrupted power. The heat engine, which works between a heat source at temperature $T_1=T_{max}$ and a heat sink at a temperature $T_4= T_{min}$, is bounded on either side by heat exchangers having overall heat transfer coefficients U_i, effective area for heat exchange A_i and hence, the total thermal conductivity K_i; ($K_i=U_iA_i$). The value of K depends on thermoeconomic considerations which dictate the size of the heat exchangers and the material used for their construction. The rates of heat transfer through the heat exchangers are assumed to be linear (Newtonian) as may be generalized by the following equation for the ith heat exchanger:

$$\dot{Q}_i = K_i\left(T_n - T_{n+1}\right) \qquad (1)$$

The power output of the engine can be given by the first law of thermodynamics as

$$\dot{W} = \dot{Q}_1 - \dot{Q}_2 \qquad (2)$$

By Clausius' inequality, for a thermodynamic system, $\oint \frac{Q}{T} \le 0$. In the current case, the above equation can be generalized as $\frac{\dot{Q}_1}{T_1} = \phi \frac{\dot{Q}_2}{T_2}$ where $\phi \le 1$ the irreversibility factor such that $\phi \le 1$. The value of ϕ depends on the presence of irreversibilities such as friction, heat loss to the surroundings, etc. For an internally reversible heat engine, the value of ϕ is taken as unity. Hence, the equation may be modified as

$$\frac{\dot{Q}_1}{T_2} = \frac{\dot{Q}_2}{T_3} \qquad (3)$$

Combining equations (1), (2) and (3), the net power output from the engine is obtained as

$$\dot{W} = \frac{1}{\frac{1}{K_1} + \frac{1}{K_2}}\left(T_1 - \frac{T_4}{\tau}\right)\left(1 - \tau\right) \qquad (4)$$

where $\tau = \frac{T_3}{T_2}$. From the above equation, it is seen that the total power output would be zero when τ=1 (thermal short circuit) or τ=0 (thermal open circuit). The maximum power is produced at an intermediate value of τ. At this optimum temperature ratio, the power output is found by partially differentiating the above equation with respect to τ and equating the result to zero.

$$\frac{\partial \dot{W}}{\partial \tau} = 0$$

The optimum temperature ratio thus calculated is

$$\tau_{opt} = \sqrt{\frac{T_4}{T_1}} = \sqrt{\frac{T_{\min}}{T_{\max}}} \qquad (5)$$

The corresponding maximum power output delivered is

$$\dot{\mathrm{W}} = \frac{1}{\frac{1}{K_1} + \frac{1}{K_2}} \left(\sqrt{T_{\max}} - \sqrt{T_{\min}}\right)^2 \qquad (6)$$

This equation shows that the power output of a heat engine is a function of the heat source temperature $T_{\max}$, heat sink temperature $T_{\min}$ and the coefficient of heat transfer of the heat exchangers involved. The value of the heat transfer coefficient would depend on the capital available and the general thermoeconomic considerations. Let us assume that from thermoeconomic considerations, the total available heat transfer coefficient is K. Hence, K=K_1+K_2. The optimum value of K_1 and K_2 for maximum power output is found to be

$$K_1 = K_2 = \frac{K}{2} \qquad (7)$$

This result signifies that the thermal conductances need to be distributed equally amongst the heat exchanger, to obtain maximum power. The corresponding maximum power output of the engine is

$$\dot{\mathrm{W}} = \frac{K}{4}\left(\sqrt{T_{\max}} - \sqrt{T_{\min}}\right)^2 \qquad (8)$$

The efficiency of a heat engine is defined as the ratio of work produced to the heat input. Hence the efficiency of this internally reversible heat engine at the condition of maximum power production can be stated, by correlating equations (4) and (5), as

$$\eta_{\max \dot{W}} = 1 - \sqrt{\frac{T_{\min}}{T_{\max}}} \qquad (9)$$

The above equation shows that the efficiency an internally reversible heat engine at the condition of maximum power production is independent of the heat transfer coefficient and is only a function of the maximum and minimum operating temperatures.

A reciprocating engine is the one that accepts and rejects heat discontinuously so as to develop power through fixed intervals of time. An example of such an engine is an internal combustion engine working on the Otto or Diesel cycle. In the schematic of the engine shown in Figure 1, the heat is accepted for a time t_1 and rejected for a time t_2, such that the ith heat exchanger works for a time t_i.

Hence the heat flow through the heat exchangers may be generalized as

$$Q_i = K_i t_i \left(T_n - T_{n+1}\right) \qquad (10)$$

A Carnot cycle for power generation necessarily consists of two isothermal heat exchange processes and two isentropic processes. The time consumed by the isentropic processes can be neglected. In FTT, due to the presence of external irreversibilities, heat exchange occurs over a finite interval of time, which is the time t_i for which the ith heat exchanger works. Hence the total cycle time to produce one power stroke is the algebraic sum of the time for which the heat exchangers work; ie. t_1+t_2. The effective power developed is

$$\dot{W} = \frac{W}{t_1 + t_2} = \frac{Q_1 + Q_2}{t_1 + t_2} \qquad (11)$$

The power produced by such a heat engine at optimum temperature ratio and optimum time for heat rejection and acceptance is calculated by correlating equations (10) for heat transfer, (2) for energy conservation, (3) governing Clausius' inequality and (4) as

$$\dot{W} = \frac{1}{\left(\frac{1}{\sqrt{K_1}} + \frac{1}{\sqrt{K_2}}\right)^2}\left(\sqrt{T_{\max}} - \sqrt{T_{\min}}\right)^2 \tag{12}$$

If we define a parameter $K=K_1+K_2$, denoting total heat transfer coefficient as in the previous section, the optimum values of K_1 and K_2 for maximum power generation are found to be the same as the ones calculated in equation (7). The corresponding maximum power is

$$\dot{W} = \frac{K}{8}\left(\sqrt{T_{\max}} - \sqrt{T_{\min}}\right)^2 \tag{13}$$

Note the difference in the values of total power produced in equations (8) and (13). From this result, we can conclusively state that a continuous heat engine can produce twice as much power as that produced by a reciprocating heat engine, when working at the maximum power condition.

The efficiency of the internally reversible reciprocating engine is found to be the same as that of the continuous engine as stated in equation (9). Hence, we can conclude that the efficiency of an internally reversible engine working at the maximum power condition is independent of the type of the engine.

3 Combined cycle engines

An internally reversible combined cycle heat engine is equivalent to the actual combined cycle heat engine with the practical thermodynamic cycles replaced by theoretical internally reversible cycles. The following types of these engines have been analyzed in this paper:

- Continuous-Continuous combined cycle heat engine: In this set-up, the topping as well as bottoming thermodynamic cycles comprise of continuous heat engines. A practical example sporting such a set-up is a Brayton-Rankine combined cycle heat engine.
- Reciprocating-Reciprocating combined cycle heat engine: In this set-up, the topping as well as the bottoming thermodynamic cycles comprise of reciprocating heat engines. A combined cycle heat engine consisting of two Sterling engines is an example of this type.
- Reciprocating-Continuous combined cycle heat engine: The topping engine in this set-up is reciprocating whereas the bottoming engine is continuous. The Dicold-Organic Rankine cycle combine cycle heat engine is an example of this type.
- Continuous-Reciprocating combined cycle heat engine: The topping engine in this set-up is continuous and the bottoming engine is reciprocating. An example of this set-up is the Brayton-Sterling combined cycle heat engine.

In the following sub-sections, such combined cycles are analyzed thermodynamically.

3.1 Continuous-Continuous combined cycle heat engine

An example of such a set-up in practice is a gas turbine feeding heat to a steam turbine. To minimize losses, we assume that the heat rejected by the first cycle is completely accepted by the second cycle without any losses. This arrangement is shown in figure 2. As in section 2, the ith heat exchanger is supposed to handle heat $\dot{Q}_i$ as defined in equation (1). The corresponding power developed is generalized as

$$\dot{W}_i = \dot{Q}_i - \dot{Q}_{i+1} \tag{14}$$

As per Clausius' inequality, for each engine, the following equations are true:

$$\frac{\dot{Q}_1}{T_2} = \frac{\dot{Q}_2}{T_3} \tag{15a}$$

$$\frac{\dot{Q}_2}{T_4} = \frac{\dot{Q}_3}{T_5} \tag{15b}$$

Temperatures T_2, T_3, T_4 and T_5 are variables and may be replaced by temperature ratios τ_1 and τ_1, which are defined as

$$\tau_1 = \frac{T_3}{T_2} = \frac{\dot{Q}_2}{\dot{Q}_1} \qquad (16a)$$

$$\tau_2 = \frac{T_5}{T_4} = \frac{\dot{Q}_3}{\dot{Q}_2} \qquad (16b)$$

The net power output of the engine is arithmetic sum of the power produced by individual cycles.

$$\dot{W} = \dot{W}_1 + \dot{W}_2 = \dot{Q}_1\left(1 - \tau_1\tau_2\right) \qquad (17)$$

The value of $\dot{W}$ independent of the intermediate temperatures, entirely in terms of the temperature ratios τ_1 and τ_2, is

$$\dot{W} = \frac{1}{\dfrac{1}{K_1} + \dfrac{1}{K_2} + \dfrac{1}{K_3}}\left(T_1 - \frac{T_6}{\tau_1\tau_2}\right)\left(1 - \tau_1\tau_2\right) \qquad (18)$$

From the above equation, we see that the engine would deliver no power at all if τ_1=0 or τ_2=0 (thermal open circuit), or when τ_1=1 and τ_2=1 (thermal short circuit). The maximum power is produced at an intermediate value of τ_1 and τ_2, which is calculated by partially differentiating the power in equation (18) with respect to τ_1 and τ_2 and equating the result to zero. The corresponding calculations lead to the following result:

$$\dot{W} = \frac{1}{\dfrac{1}{K_1} + \dfrac{1}{K_2} + \dfrac{1}{K_3}}\left(\sqrt{T_{\max}} - \sqrt{T_{\min}}\right)^2 \qquad (19)$$

The above equation shows that the power output of a continuous combined cycle engine is a function of the heat source temperature T_{max}, heat sink temperature T_{min} and heat transfer coefficients of various heat exchangers. The maximum power can thus be increased by increasing the temperature range or by increasing the capital cost of the heat exchanger, thereby improving *K*. Hence, let us introduce the total heat transfer coefficient K=K_1+K_2+K_3 which will signify the total cost involved from the thermoeconomic point of view. The optimum value of K_1, K_2 and K_3 for maximum power production is found to be

$$K_1 = K_2 = K_3 = \frac{K}{3} \qquad (20)$$

The corresponding maximum power is

$$\dot{W}_{\max} = \frac{K}{9}\left(\sqrt{T_{\max}} - \sqrt{T_{\min}}\right)^2 = 0.111K\left(\sqrt{T_{\max}} - \sqrt{T_{\min}}\right)^2 \qquad (21)$$

On comparing equations (8) and (21), we find that an ordinary internally reversible heat engine produces 2.25 times more power than an internally reversible continuous combined cycle heat engine working in the same temperature range and having similar thermoeconomic constraints. The reason for this may be attributed to the fact that there are 3 heat exchangers in this the combined cycle engine compared to 2 in the single cycle engine. However, in practice, a single cycle heat engine cannot be used efficiently over a large temperature drop due to problems associated with working fluid properties. Hence, in reality, combined cycle heat engines are more desirable.

The efficiency calculated by correlating equations (17) and (19) is

$$\eta_{\max \dot{W}} = 1 - \sqrt{\frac{T_{\min}}{T_{\max}}} \qquad (22)$$

This value is found to be the same as that in equation (9) for a single cycle heat engine. We can thereby conclude that the cost of generating power in a continuous combined cycle engine is the same as that for a simple engine working under similar conditions.

3.2 Reciprocating-Reciprocating combined cycle heat engine

A reciprocating heat engine, in this case, feeds heat to another reciprocating heat engine. In order to minimize direct heat losses, we assume that the heat rejected by the first engine is completely accepted by the second engine. The i^{th} heat exchanger is assumed to work for time t_i, so that the heat exchanged would be

$Q_i = \dot{Q}_i t_i = K_i t_i (T_n - T_{n+1})$. The basic set-up for this arrangement is shown in figure 2. The net power developed would be the algebraic sum of the power developed by individual cycles.

$$\dot{W} = \frac{Q_1 - Q_2}{t_1 + t_2} + \frac{Q_2 - Q_3}{t_2 + t_3} \tag{23}$$

The power developed in terms of the temperature ratios τ_1 and τ_2 is

$$\dot{W} = \frac{1}{\frac{1}{K_1 t_1} + \frac{1}{K_2 t_2} + \frac{1}{K_3 t_3}} \left(T_1 - \frac{T_6}{\tau_1 \tau_2} \right) \left[\frac{(1-\tau_1)}{t_1 + t_2} + \frac{\tau_1 (1-\tau_2)}{t_2 + t_3} \right] \tag{24}$$

At optimum temperature ratios and optimum operating time, maximum power developed is

$$\dot{W} = \frac{1}{\left(\sqrt{\frac{1}{K_1} + \frac{1}{K_3}} + \sqrt{\frac{1}{K_2}} \right)^2} \left(\sqrt{T_{\max}} - \sqrt{T_{\min}} \right)^2 \tag{25}$$

The dependence of this power on the heat transfer coefficient can be analyzed by defining constant $K=K_1+K_2+K_3$ as in the previous section. The optimum heat transfer coefficient for individual engines for maximum power production is

$$K_1 = K_3 = \frac{K}{2^{\frac{1}{3}} + 2} = 0.307K \tag{26}$$

$$K_2 = \frac{K}{2^{\frac{2}{3}} + 1} = 0.386K \tag{27}$$

The corresponding maximum power developed is

$$\dot{W}_{\max} = \frac{K}{\left(2^{2/3} + 1\right)^3} \left(\sqrt{T_{\max}} - \sqrt{T_{\min}} \right)^2$$

$$= 0.058K \left(\sqrt{T_{\max}} - \sqrt{T_{\min}} \right)^2 \tag{28}$$

On comparing equations (21) and (28), we find that the maximum power developed by a continuous combined cycle heat engine is 1.92 times that developed by a reciprocating combined cycle engine working under identical conditions. The reason for this may be attributed to the fact that a continuous cycle produces twice as much power as a reciprocating cycle, as shown in section 2. Similarly, on comparing equations (13) and (28), we find that the simple reciprocating heat engine produces 2.17 times more power than a combined cycle reciprocating engine working under similar conditions. This is mainly because of the additional heat exchanger and the associated losses involved with the later.

The efficiency at maximum power production is found to be the same as that in previous set-up, as in equations (22).

3.3 Reciprocating-Continuous combined cycle heat engine

A reciprocating cycle feeds heat to a continuous cycle in this case. A practical example of this set-up is the internal combustion engine - turbocharger assembly in automobiles. The set-up is as shown in figure 2. The first cycle accepts heat $Q_1 = K_1 t_1 (T_1 - T_2)$ while rejecting $Q_2 = K_2 t_2 (T_3 - T_4)$. However, since the second cycle accepts heat continuously, the effective heat added per unit time to it would be $\dot{Q}_2 = \frac{K_2 t_2}{t_1 + t_2}(T_3 - T_4)$. The heat rejected by the second cycle would be $\dot{Q}_3 = K_3 (T_5 - T_6)$. This would ensure continuous working of the second cycle without any heat losses.

The power developed by the combined cycle is the arithmetic sum of the power developed by the individual cycles.

$$\dot{W} = \left(\frac{Q_1 - Q_2}{t_1 + t_2} \right) + \left(\dot{Q}_2 - \dot{Q}_3 \right) \tag{29}$$

The above value in terms of the temperature ratios τ_1 and τ_2 is

$$\dot{W} = \frac{1}{(t_1 + t_2)\left(\frac{1}{K_1 t_1} + \frac{1}{K_2 t_2} \right) + \frac{1}{K_3}} \left(T_{\max} - \frac{T_{\min}}{\tau_1 \tau_2} \right) (1 - \tau_1 \tau_2) \tag{30}$$

At optimum time for heat rejection and acceptance and optimum temperature ratios, the maximum power developed by the combined cycle is

$$\dot{W} = \frac{1}{\frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} + \frac{2}{\sqrt{K_2 K_3}}} \left(\sqrt{T_{max}} - \sqrt{T_{min}} \right)^2 \tag{31}$$

On solving mathematically, the contribution of individual heat transfer coefficients for maximum power production is found to be

$$K_1 = 0.26K$$

$$K_2 = K_3 = 0.37K$$

The corresponding maximum power developed is

$$\dot{W}_{max} = 0.0682K \left(\sqrt{T_{max}} - \sqrt{T_{min}} \right)^2 \tag{32}$$

The above results show that the coefficient of heat transfer of the second and third heat exchangers contribute more towards the total coefficient. The reason for this should be the fact that the second and third exchangers exchange heat continuously, whereas the first one exchanges heat discontinuously, hence their additional influence.

On comparing the equation (32) with previous results in equations (21) and (28), we find that the maximum power developed by a Reciprocating-Continuous heat engine is between that developed by a Continuous combined cycle engine and a Reciprocating combined cycle heat engine. This is due to the presence of one, two and zero continuous cycles respectively in the above heat engines. The efficiency for this set-up though, remains the same as the others, as in equation (22).

3.4 Continuous-Reciprocating combined cycle heat engine

In this case, the first cycle is continuous in nature whereas the second one is discontinuous. Hence, for the second cycle to receive the entire heat rejected by the first cycle, a thermal reservoir needs to be incorporated, which would accept heat continuously from the first cycle and reject it discontinuously to the second cycle. Such a set-up is shown in figure 3.

Energy balance at the heat reservoir gives the following equation:

$$\dot{Q}_2 = \frac{Q_3}{(t_3 + t_4)} \tag{33}$$

The total power, which is the sum of the powers developed by the individual cycles is

$$\dot{W} = \left(\dot{Q}_1 - \dot{Q}_2 \right) + \left(\frac{Q_3 - Q_4}{t_3 + t_4} \right) \tag{34}$$

The maximum power produced at optimum temperature ratios τ_1 and τ_2 and optimum operating times τ_3 and τ_4 is

$$\dot{W} = \frac{\left(\sqrt{T_{max}} - \sqrt{T_{min}} \right)^2}{\frac{1}{K_1} + \frac{1}{K_2} + \left(\frac{1}{\sqrt{K_3}} + \frac{1}{\sqrt{K_4}} \right)^2} \tag{35}$$

This result is in general agreement with previous results, and in a broader sense, is a combination of the maximum power developed by simple continuous and reciprocating engines, as in equations (6) and (12). The optimum heat transfer coefficients of individual heat exchangers at maximum power production are

$$K_1 = K_2 = 0.2071K$$

$$K_3 = K_4 = 0.2929K$$

The corresponding maximum power is

$$\dot{W}_{max} = \frac{K}{\left(\sqrt{4\sqrt{2} + 9} + 1 \right)^2} \left(\sqrt{T_{max}} - \sqrt{T_{min}} \right)^2 = 0.0429K \left(\sqrt{T_{max}} - \sqrt{T_{min}} \right)^2 \tag{36}$$

This value is lower than that calculated for Continuous, Reciprocating and Reciprocating-Continuous combined cycle heat engines as in equations (21), (28) and (32) respectively. The reason for this drop in power production in this case is the presence of the additional heat exchanger provided due to the thermal reservoir.

The efficiency of the Continuous-Reciprocating combined cycle heat engine at maximum power production is found to be the same as for other set-ups, as in equation (22).

4. Discussion

The results obtained in equations (21), (28), (32) and (36) are tabulated in table 1.

As is evident, the Continuous- Continuous combined cycle heat engine develops the most power. The reason for this is the fact that a simple continuous heat engine produces twice as much power as a similar reciprocating engine as shown in section 2. Due to the same reason, the Reciprocating-Continuous engine produces the next highest power, followed by the Reciprocating-Reciprocating combined cycle heat engine. The Continuous-Reciprocating heat engine produces the least power due to the provision of an extra heat exchanger.

The efficiency in all the above set-ups is the same. This means that the cost of power production is the same in all the cases. Hence, when exposed to a certain heat source, the Reciprocating-Continuous engine would consume the least amount of heat and deliver the least power amongst all, while working at the maximum power condition, and so on.

Conclusion

The power developed by an internally reversible heat engine depends upon the external system conditions, the engine type and the thermoeconomic constraints. The external system conditions include the heat source and heat sink temperatures and the technological inputs associated with the set-up. The engine type mainly refers to whether the engine is of reciprocating type or continuous type; i.e. the power delivery pattern and frequency of the engine. The thermoeconomic constraints are the financial implications imposed on the design and construction of the set-up. An engine with the greatest difference between the temperatures of the heat sink and source, a continuous power delivery pattern and the greatest thermoeconomic consideration will produce the highest power. The efficiency at the maximum power production condition, however, is an entity independent of individual engine characteristics (type of engine, etc.) and is linked more with external system conditions and thermoeconomics.

While high temperatures can be now produced chemically, actual thermodynamic cycles work satisfactorily only in a certain range of temperatures which is dictated mainly by the physical and thermodynamic properties of the fluids involved therein. This deficiency can be corrected by using combined cycle engines made by cascading one or more heat engines.

The theoretical power production limit has been analytically derived in this paper which should give a designer some idea as regards combined cycle to be selected keeping in mind the constraints imposed by actual physical requirements.

References

1. Chambadal P, Les Centrales, Paris, Armand Colin, 1957.
2. Novikov H, Atomnaya Energiya, 1957, 3:409.
3. Curzon FL and Ahlborn B, "Efficiency of Carnot engine at maximum power output", Am. J. Phys. 43, 22-24 (1975).65.
4. Bandyopadhayay S, Bera N C, Bhattacharyya S, ‘Thermoeconomic optimization of Combined Cycle Power plants’, Energy Conversion and Management, 42 (2001), 359-371.
5. C Wu, G Karpouzian, Kiang R L, ‘The optimal power performance of an Endoreversible Combined Cycle’, Journal of Institute of Energy, 1992, 65, pp 41-45.
6. Ozkaynak S, ‘The theoretical efficiency limits for a combined cycle under the condition of maximum power output’, Journal of Applied Physics, 1995, 28:2024.