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Energy and Exergy Analysis of a combined Diesel-Engine Gas-Turbine System for Distributed Power Generation

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Abstract

This paper presents a combined diesel-engine gas-turbine system that enables distributed power generation plants to attain high thermal efficiencies while enjoying the operational advantages of both diesel engines and gas turbines. The configuration of the system presented here improves upon previously proposed configurations through the appropriate use of regeneration and inter-cooling. Energy and exergy analyses of the system's performance were performed using an Excel-based thermodynamic model. The energy analysis studied the effect of the diesel-engine's compression ratio, the gas-turbine's inlet temperature, and the compressors' pressure ratios on the system's performance. This analysis determined the pressure ratios that permit the system to be designed with fewer compressors and turbines. The second-law analysis determined the main exergy losses and destructions of the system. Finally, the paper compares the thermal efficiency and net work of the proposed system at different modes of operation.

Keywords: *Combined Diesel-Brayton cycle; Small-scale power generation; Energy and exergy analysis*

1. Introduction

Due to their excellent thermal efficiency, reliability and high availability, diesel engines have become the most preferred prime movers in distributed generation as well as other medium and medium-large applications [1]. Diesel engine power plants can be set up quickly, normally in less than twelve months, to generate hundreds of megawatts of energy [2]. Another advantage of diesel power plants is that they can run on heavy diesel fuel, a low-grade relatively inexpensive product of oil refineries. Diesel engines are also suited to burn natural gas and bio-fuels either in their pure form or mixed with petroleum fuels [3]. Despite of the high thermal efficiency of diesel engines, their exhaust gases, cooling water and lubrication coil carry out with them a significant portion of the energy input in the combustion chamber [4]. Therefore, combined systems, such as the Organic Rankine cycle (ORC) [5] and the Diesel-Kalina cycle [6], have been developed for using the waste energy from the engine to produce additional power. However, both the Organic Rankine cycle that employs organic fluids and the Diesel-Kalina cycle that uses a water-ammonia solution require boilers, condensers, etc. Therefore, these combined systems need substantial initial costs and also limit the plant's flexibility. Moreover, these systems do not utilise all the work potential in the exhaust gas.

While the exergy losses in the cooling water and lubricant oil are purely thermal, the exergy loss in the exhaust gas is both thermal and mechanical because of its higher-than-atmospheric temperature and pressure. A concept that makes use of both the mechanical and thermal exergies in the exhaust gas is the compound cycle engine (CCE) [7]. The CCE utilises the energy of the diesel-engine's exhaust for producing extra expansion work, part of which is used in compressing the air prior to the engine's intake manifolds. The CCE received interest in aviation applications because it combines the lightweight pressure rise capability of a gas turbine with the high efficiency of a diesel engine. An extension of the CCE concept which is more relevant to stationary power generation is the combined Diesel-Brayton (CDB) cycle [8,9]. The CDB cycle also takes advantage of the energy content of the diesel-engine's exhaust gas but adds a complete gas-turbine with its compressor and combustion chamber. This arrangement, which enables the diesel engine and gas turbine to be run independently, gives the system great flexibility during operation. The CDB cycle enables small-scale power generation and industrial cogeneration systems to attain high thermal efficiencies, while the fuel flexibilities of diesel engines and gas turbine enables them to use heavy-diesel fuel, natural gas, or renewable bio-fuels [10]. Unlike the combined Brayton-Rankine cycle, which received considerable attention because of its suitability for large-scale power generation, published studies on CDB cycles are rather scarce.

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The present paper describes a combined diesel-engine gas-turbine (CDG) system that improves upon previous proposals through the appropriate use of regeneration and intercooling. For the analysis of the system's performance, an Excel-based computer model has been developed. The model enables optimum values of the compressors' pressure ratios to be determined for given values of the diesel-engine's compression ratio (*CR*) and gas-turbine's inlet temperature (*TIT*). Analysis of the system's performance with different values of *CR* and *TIT* determined the combination of compressor pressure ratios that permits the system to be designed with fewer compressors and turbines while maintaining a near-optimum performance. A second-law analysis that weighs the different exergy losses and destructions in the system is also presented. The paper also compares the performance of the proposed system in four modes of operation; (i) as a compound cycle diesel engine, (ii) as an intercooled regenerative gas turbine, (iii) as a combined Diesel-Brayton cycle, and (iv) as an air-bottoming cycle.

2. The Combined Diesel-Engine Gas-Turbine System

Figure 1 shows a schematic diagram of the combined system that consists of a compound diesel engine (compressor C1, intercooler IC1, diesel engine DE, and turbine T1) and an intercooled regenerative gas turbine (compressor C2, intercooler IC2, compressor C3, regenerator RG, combustion chamber CC, and turbine T2). Before discharged to the atmosphere, the exhaust gases from the two turbines are mixed and passed through the regenerator to reheat the high-pressure air going to the combustion chamber of the gas turbine.

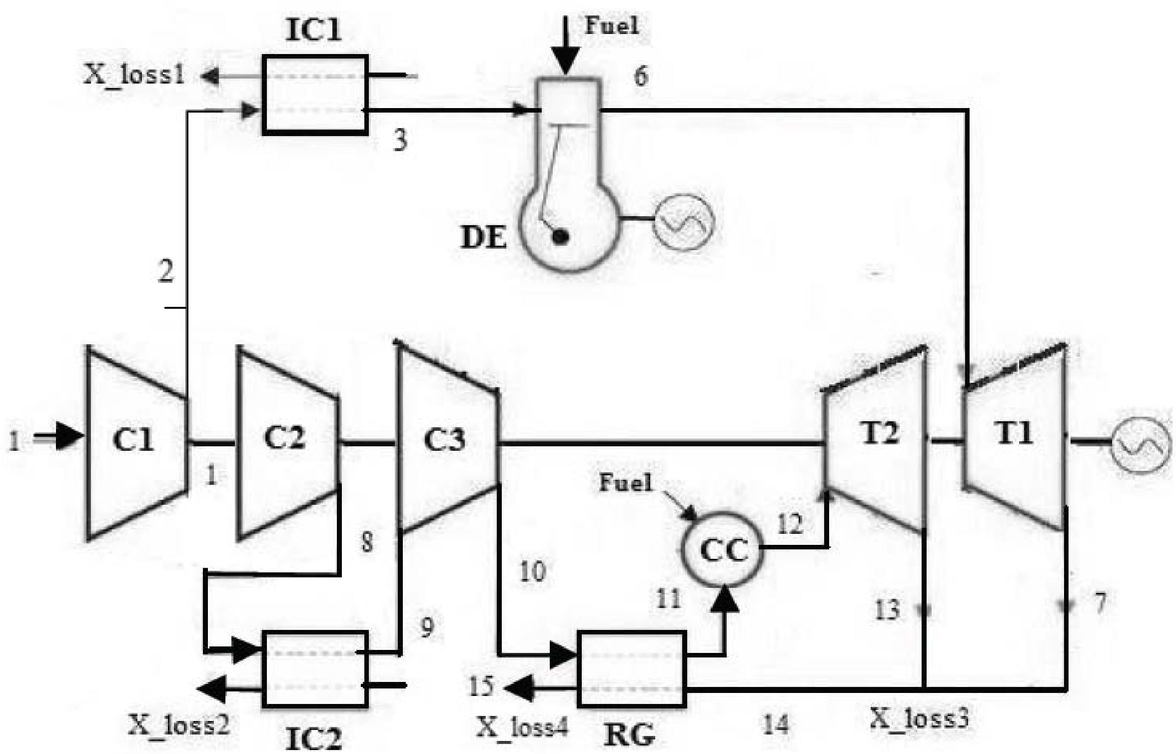


Fig. 1. The combined diesel-engine gas-turbine system with intercooling and regeneration

Compared to the configuration of the combined system proposed by Mukul and Agarwal [8], the present configuration offers greater design and operational flexibilities by introducing separate compressors and turbines for the Diesel and Brayton cycles. This arrangement also allows the two components of the combined system to be optimised almost independently since the only common part is the regenerator. Although intercooling causes an exergy loss, it has certain advantages in the implementation of the system. For the gas turbine, intercooling enhances the effect of regeneration and improves the cycle's thermal efficiency. For the diesel engine, cooling the compressed air before the engine's intake improves its volumetric efficiency and, therefore, reduces its size and installation cost. In the cycle proposed by Mukul and Agarwal [8], the diesel-engine's intake air is heated after compression, rather than cooled, which reduces the engine's volumetric efficiency. With two separate turbines, the present configuration also avoids the loss of exergy that is caused by mixing the hot gases coming from the gas-turbine's combustion chamber with the cooler diesel-engine's exhaust.

Compared to the configuration of proposed by Krishna and Renald [9], the present configuration improves the thermal efficiency by precompressing the air of the diesel engine and by introducing intercooling and regeneration in the gas turbine cycle. However, compared to the two systems proposed previously the present system consists of more components, which is bound to increase its initial cost as well as its overall losses. Therefore, the main objective of the following analysis is to explore the possibility of replacing the two low-pressure compressors by a single compressor and/or replacing the two turbines by a single turbine.

3. The Thermodynamic Model

In the following analysis, the system shown in Figure 1, which has two different low-pressure compressors and two different turbines, is referred to as system A. In system B, the two low-pressure compressors C1 and C2 are required to have the same pressure ratio (i.e. PR_{C1}=PR_{C2}) so that they can be combined in a single large compressor. Similarly, in system C the two turbines T1 and T2 are required to have the same pressure ratios (i.e. P_{10}=P_6) so that they can be replaced by a single large turbine. System D has a single low-pressure compressor as well as a single turbine. Figure 2 shows the *T-s* diagrams for the four configurations of the combined system. As the figure shows, systems A and B, respectively, are different from systems C and D in that the two turbines T1 and T2 have different compression ratios. Systems A and C, respectively, are different from systems B and D in that the low pressure compressors C1 and C2 have different pressure ratios.

In the four cycles shown in Figure 2, process 1-2 is an isentropic compression process in compressor C1, where ambient air taken at T_1, P_1 is delivered at T_2, P_2. The compressed air passes through an ideal constant pressure intercooler (process 2-3) after which its temperature is brought down to T_3=T_1. The cooled air goes to the intake of the diesel engine where it is compressed isentropically to state 4. Heat is then added to it in process 4-5 followed by the expansion stoke (process 5-6). In systems A and B, the gas discharged from the diesel engine is expanded in turbine T1 (Process 6-7) before discharged at ambient pressure.

In systems C and D, the diesel-engine's exhaust is mixed with the gas-turbine's products of combustion before the mixture is expanded in the common turbine in process 7-13. In all cycles, the gas-turbine's intake air that is compressed in compressor C2 to point 8 also goes through an ideal inter-cooler that brings its temperature T_9 = T_1. The compressed air then passes through the high-pressure compressor C3 (process 9-10), the regenerator (process 10-11), and the combustion chamber (process 11-12). In systems A and B, the heated gas is then expanded in the second turbine T2 (Process 12-13) before discharged at ambient pressure and mixed with the exhaust gas of turbine 1. The mixed gas is sent at T_{14} to the regenerator to preheat the air going the combustion chamber and leaves at state 15. In systems C and D, the exhaust gas from the single turbine at T_{13} is sent to the regenerator before discharged at state 14 to the atmosphere.

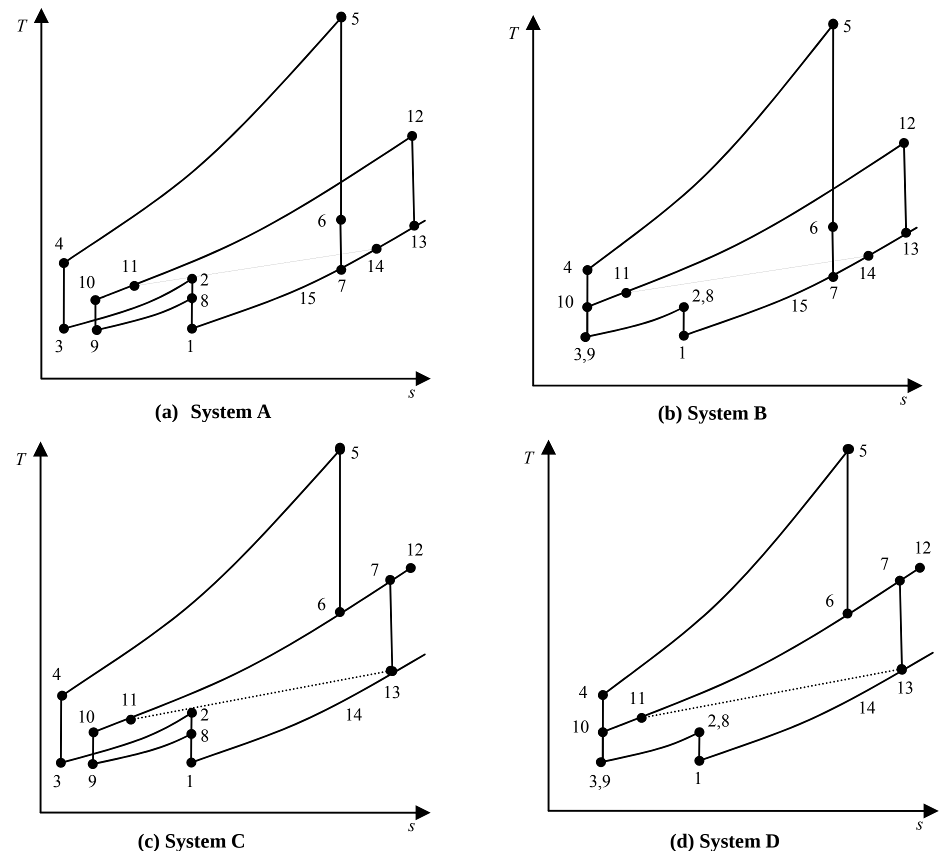


Fig. 2. The *T-s* diagrams of four arrangements of the combined system

The performance of the diesel engine depends on its compression ratio (*CR*) and cut-off ratio (*COR*) where,

$$CR = v_3 / v_4 \tag{1}$$

$$COR = v_5 / v_4 \tag{2}$$

Referring to systems A and B, the relations that determine the system's net work(w_{NET}) and thermal efficiency (η) are:

$$W_{C1} = m_1(h_2 - h_1) \quad (3)$$

$$W_{C2} = m_2(h_8 - h_1) \quad (4)$$

$$W_{C3} = m_2(h_{10} - h_9) \quad (5)$$

$$W_{T1} = m_1(h_6 - h_7) \quad (6)$$

$$W_{T2} = m_2(h_{12} - h_{13}) \quad (7)$$

$$W_{DE} = m_1[(h_5 - h_6) - (u_6 - u_3)] \quad (8)$$

$$Q_{DE} = m_1(h_5 - h_4) \quad (9)$$

$$Q_{GT} = m_2(h_{12} - h_{11}) \quad (10)$$

$$W_{NET} = (W_{DE} + W_{T1} + W_{T2}) - (W_{C1} + W_{C2} + W_{C3}) \quad (11)$$

$$\eta = \frac{W_{NET}}{(Q_{DE} + Q_{GT})} \quad (12)$$

A slightly different set of relations apply to the cycles of systems C and D. With the assumed idealisations, viz., ideal compression and expansion, perfect intercooling, and negligible pressure losses, the performance of the combined system, in terms of its thermal efficiency and net work-output, is affected by the following factors:

1. The compression ratio and cut-off ratio of the diesel engine (*CR* and *COR*)
2. The pressure ratios of the three compressors (PR_{C1}, PR_{C2} and PR_{C3}) and two turbines (PR_{T1}, PR_{T2})
3. The effectiveness of the regenerator (ε)
4. The mass flow rates of the air supplied to the diesel engine and the to the gas turbine (m_1 and m_2).

Depending on the flow rate of the air supplied to the diesel engine (m_1) and that supplied to the gas turbine (m_2), the system can be operated in one of the following modes:

(i) As a compound cycle engine only: $m_1>0$, $m_2=0$,

(ii) As an intercooled regenerative gas turbine only: $m_1=0$, $m_2>0$,

(iii) As a combined Diesel-Brayton cycle: $m_1>0$, $m_2>0$.
As an air-bottoming cycle; $m_1>0$, $m_2>0$, $T_{12}=T_{11}$

4. Analysis of the System's Performance

The following analysis of the combined system studies the effects of the key parameters on the system's performance so as to determine the values of these parameters that optimise its thermal efficiency and net work-output. This was achieved by varying the diesel-engine's compression ratio (*CR*) and the gas turbine inlet temperature (*TIT*). The pressure ratios of the three compressors were not treated as independent parameters. Instead, Excel's Solver was used to determine the pressure ratios that maximise the cycle's thermal efficiency at each combination of *TIT* and *CR*. A second-law analysis weighs the different exergy losses and destructions in the system. Finally, the study compares the performance of the selected system configuration in different mode of operation. The analyses were carried out based on the following assumptions:

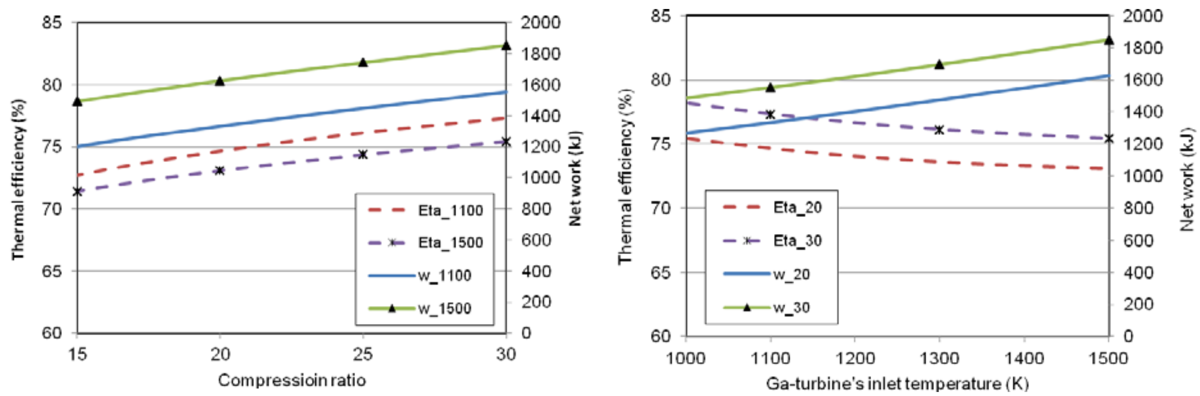
- Ambient temperature and the pressure are 300 K and 0.1 MPa, respectively.
- The diesel engine's cut-off ratio is 2.
- The regenerator effectiveness is 80%.
- Post-combustion products of the diesel engine and gas turbine are treated as air.

In the present model, the thermodynamic properties were determined by linking REFPROP [11] to Excel.

4.1. Performance of system A with optimised pressure ratios

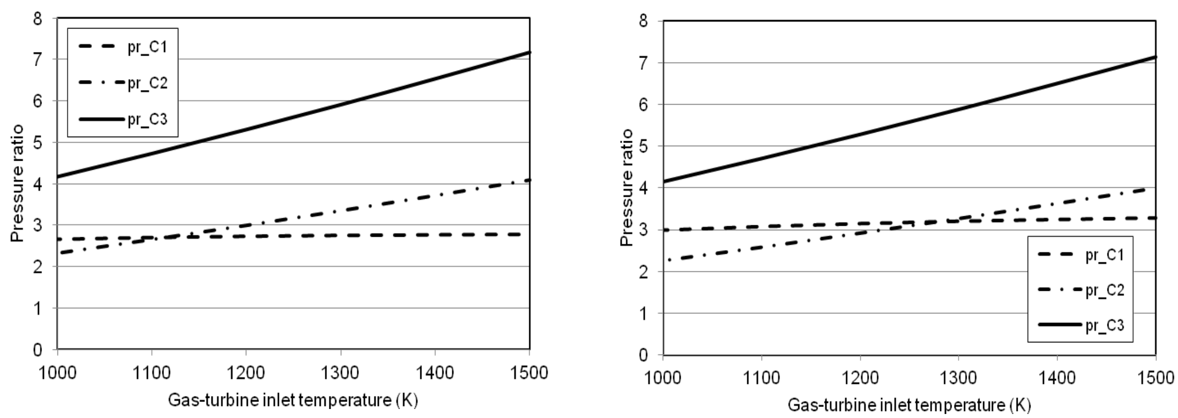
The mass flow rates in both the diesel engine and the gas turbine were taken as 1 kg, i.e. $m_2= m_1=1$. The thermal efficiency and net work were calculated at values of *TIT* in the range 1000-1500 K and values of *CR* in the range 15-30. Excel Solver was used to determine the pressure ratios of the three compressors that maximise the thermal efficiency at each set of *TIT* and *CR*. Figure 3.a shows the variation of cycle's thermal efficiency and net work with two values of *TIT* (1100K and 1500K) and *CR* varying from 15 to 30. The figure shows that both the net work and thermal efficiency increase with *CR*. Figure 3.b, which shows the results with two values of CR (20 and 30) and *TIT* varying from 1000K to 1500K, shows that increasing *TIT* increases the net work but reduces the thermal efficiency of the system. Therefore, in order to maintain the same level of thermal efficiency at a lower *TIT*, the diesel-engine's compression ratio must also be reduced.

It was mentioned earlier that the net work and thermal efficiency at each combination of *TIT* and *CR* were calculated at the compressors' pressure ratios that maximise the cycle's thermal efficiency. Figure 4 shows the variation of the optimum pressure ratios of the three compressors with *TIT* at *CR*=20 and *CR*=30. From the figure it can be seen that the pressure ratio for compressor C1 does not change significantly with both *TIT* and *CR*. The optimum values of the pressure ratio of compressor C2 and that of compressor C3 are also affected slightly with *CR*, but they both increase steadily as *TIT* increases. The higher compression ratio is always that of C3. Figure 4 also indicates that the two compressors C1 and C2 have the same optimum value at a certain value of *TIT*. At *CR*=20, the optimum values of PR_{C1} and PR_{C2} are both equal to 2.7 at *TIT* approximately equal to 1100K. For *CR*=30, the pressure ratios PR_{C1} and PR_{C2} become identical at about *TIT*=1300K.



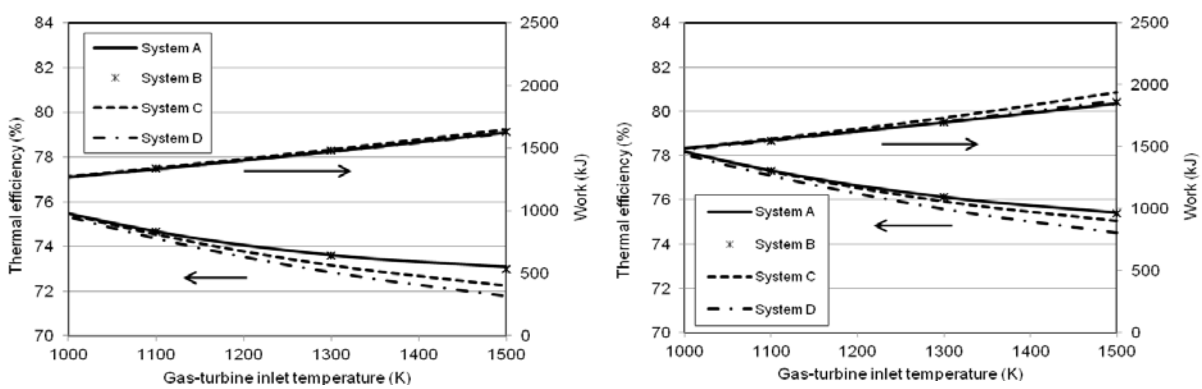
(a) (b)

Fig. 3. Variation of efficiency (Eta) and net work (w): (a) with *CR*, (b) with *TIT*



(a) (b)

Fig. 4. Variation of optimum compressor pressure ratios with *TIT*: (a) *CR*=20, (b) *CR*=30



(a) (b)

Fig. 5. Effect of constraining pressure ratios on the cycle's efficiency and net work: (a) *CR*=20 (b) *CR*=30

These results indicate that the two compressors C1 and C2 can be combined in a single large compressor that supplies compressed air to both the diesel engine and gas turbine. Similarly, the two turbines T1 and T2 can be replaced by a single large turbine if the pressure outlet of compressor C3 can be the same as that of the diesel-engine exhaust. This is the case of systems C and D shown in Figure (2) for which the pressure at point 6 is made equal to that of the points 10-12. Apart from the economical and operational advantages of having to deal with fewer compressors and turbines in the system, this arrangement might also reduce the losses and improve the overall efficiency of the system.

4.2. Performance of the modified systems B, C and D with optimised pressure ratios

Further simulations studied the performance of the systems with the constraint PR_{C1}= PR_{C2} and/or P_6 = P_{10}. Figure 5 compares the thermal efficiency and net work with these restrictions (systems B, C, D) to those of the cycle without such restrictions (system A). Figures 5.a and 5.b, respectively, show the results at *CR*=20 and *CR*=30 with *TIT* in the range 1000-1500K. The net work and thermal efficiency of system B remained close to those of system A at both values of *CR*. Although systems C and D produced more work at both values of CR, their thermal efficiencies were lower than that of system A particularly at high values of *TIT*. System C gives more work at higher thermal efficiency than system D for both values of *CR*.

Table 1 and Table 2 compare the key cycle parameters for system A, which was optimised without any conditions imposed on the pressure ratios of the compressors and turbines, with those for systems B, C and D, which were optimised with the restrictions PR_{C2}= PR_{C1} and/or P_6=P_{10}. Table 1 compares these parameters at *TIT* = 1100K and *CR*=20, while Table 2 compares them at *TIT* = 1100K and *CR* = 30. The figures on the tables show that system B maintains the thermal efficiency of system A but slightly reduces the cycle's net work. Although system C leads to different values of the compressors' pressure ratios, the figures show that it produces more work than systems A, B and D with minor reduction in the thermal efficiency. System D gives the lowest thermal efficiency for both values of *CR*. These results favour systems B and C over system D.

Table 1: Key performance parameters of the CDG systems at *TIT*=1100K, *CR*=20

	System A	*System B*	*System C*	*System D*
PR_{C1}	2.7	2.7	3.4	3.1
PR_{C2}	2.7	2.7	2.4	3.1
PR_{C3}	4.7	4.7	4.3	3.0
Net work (kJ/kg)	1333.9	1333.6	1340.5	1331.7
η (%)	74.7	74.7	74.5	74.4

Table 2: Key performance parameters of the CDG systems at *TIT*=1100K, *CR*=30

	System A	*System B*	*System C*	*System D*
PR_{C1}	3.1	2.8	3.5	3.2
PR_{C2}	2.6	2.8	2.5	3.2
PR_{C3}	4.7	4.5	4.5	3.1
Net work (kJ/kg)	1553.9	1547.0	1564.6	1550.5
η (%)	77.3	77.3	77.3	77.1

4.3. Exergy analysis

Exergy analysis determines the quality of wastes and losses of availability in the cycle. Normally, 30% of the fuel exergy is lost in the combustion and heat addition in the combustion chamber [12]. However, the present analysis does not take into consideration the exergy destruction and losses associated with the combustion process. Accordingly, the total exergy input (X_{Total}) to the cycle is taken as the net exergy added to the working fluid delivered to the Diesel cycle (X_{Diesel}) and that delivered to the Brayton cycle ($X_{Brayton}$). These are obtained from:

$$X_{Diesel} = X_5 - X_4 \tag{13}$$

$$X_{Brayton} = X_{12} - X_{11} \tag{14}$$

$$X_{Total} = X_{Diesel} + X_{Brayton} \tag{15}$$

where X is the flow exergy at the specified point in the cycle. Neglecting kinetic and potential exergies, X is given by:

$$X = (h - h_0) - T_0(s - s_0) \tag{16}$$

Since all compression and expansion processes are assumed to be ideal, exergy waste and exergy destruction occur at only certain points in the cycle: (1) exergy waste in the intercoolers, (2) exergy waste in the exhaust gases, (3) exergy destruction in mixing the two gas streams, and (4) exergy destruction in the regenerator. Referring to Figure 2, the exergy wastes in systems A and B are obtained from:

$$X_{los1}=X_2-X_3 \tag{17}$$

$$X_{los2}= X_8-X_9 \tag{18}$$

$$X_{los3}=X_{15} \tag{19}$$

The exergy destructions in the mixing process (X_{des1}) and the regenerator (X_{des2}) are obtained from:

$$X_{des1}= m_1X_7+ m_2X_{13}) - (m_1+m_2)X_{14} \tag{20}$$

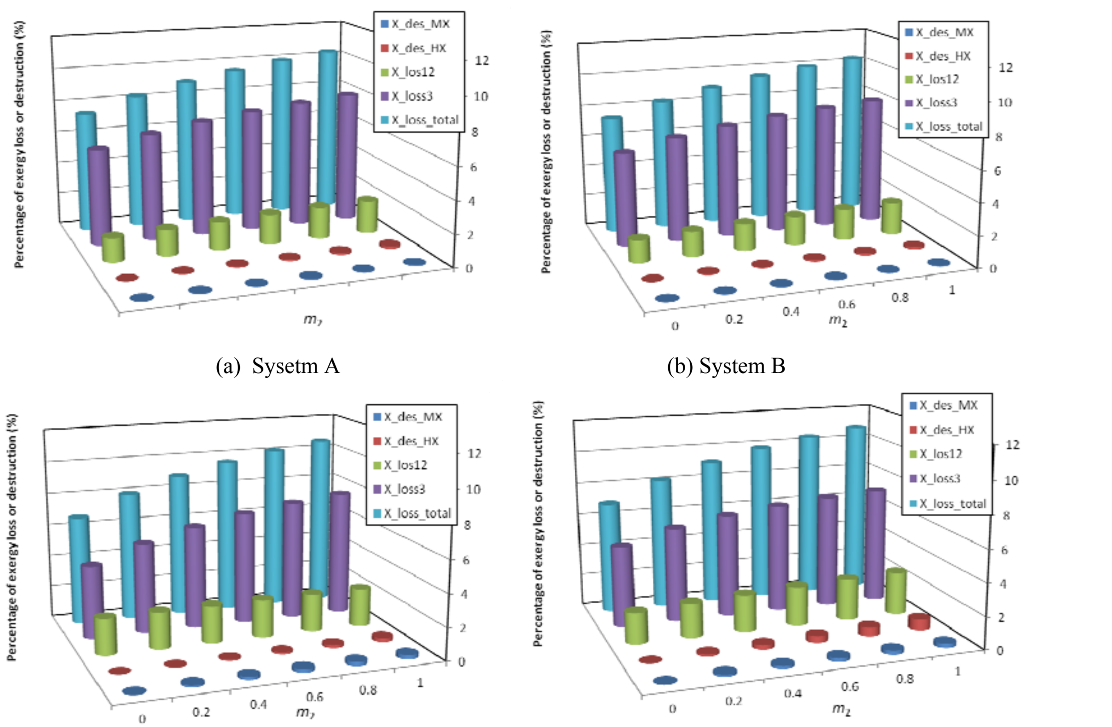
$$X_{des2}= (m_1+m_2)(X_{14}-X_{15})-m_2(X_{11}-X_{10}) \tag{21}$$

In the system configurations with a single turbine (systems C and D) mixing of the gases takes place before the turbine and not after it as expressed by Eqs. (19) -(21). Therefore, these three equations have to be suitably modified for these cases.

Figure 6 shows the percentages of the exergy losses and destructions for systems A, B, C and D in the total exergy input (X_{Total}). The losses were calculated at m_1 = 1 and different values of m_2 with *TIT*=1100K, *CR*=20. The figure shows that about 10% of the total exergy input is wasted or destroyed in all four systems. The major part of the exergy losses is that associated with exhaust gas (*X*_loss3). This loss increases with m_2 and at m_2= m_1=1 it amounts to about 8.1 in system A and system B, 7.7% in system C, and 7.2 in system D. Exergy waste in the intercoolers (X_loss12) amounts to about 2.0% in system A and system B, 2.4% in system C, and about 2.7% in system D. Although the exergy destructions due to gas mixing (X_des1) and due to heat transfer in the regenerator (X_des2) increase with m_2, their magnitudes remain considerably less than the two losses even at m_2=1. The figures show that more exergy is destroyed in systems C and D than in systems A and B. The total exergy loss and destruction in systems C and D is also more than that in systems A and B. These results and the fact that the exergy loss in the exhaust gas can be minimised by installing a more effective regenerator make system B more favourable than systems C and D.

4.4. Performance of system B at different modes of operation

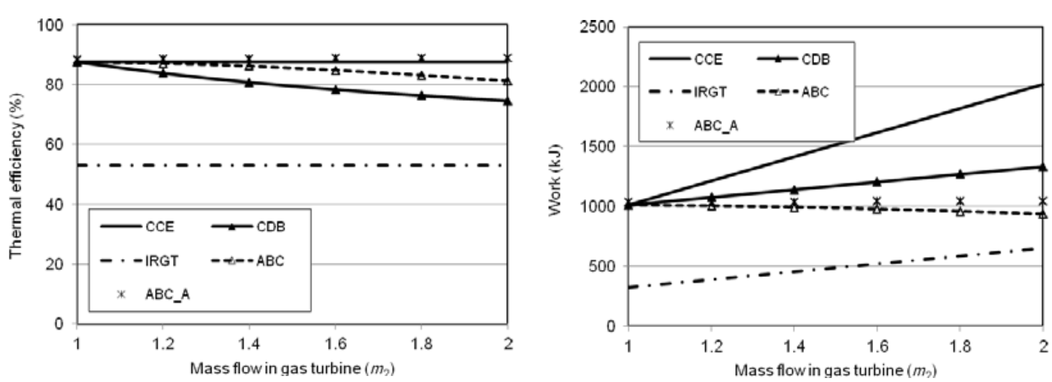
A CDG power plant that consists of a number of diesel engines and gas turbines can be operated in different modes in response to varying demand throughout the day and throughout the year. Further simulations of system B were performed in which the two performance indicators of the system were calculated at different values of m_1 and m_2, resembling the loading of more diesel engines or more gas turbines. In the compound cycle engine (CCE) mode, the value of m_2 was kept equal to 0.0 while m_1 was increased from 1.0 to 2.0. In the intercooled regenerative gas-turbine (IRGT) mode, the value of m_1 was kept equal to 0.0 while m_2 was increased from 1.0 to 2.0. In the combined Diesel-Brayton cycle (CDB) mode, the value of m_1 was kept equal to 1.0 and m_2 was increased from 0.0 to 1.0. The air-bottoming cycle (ABC) mode was similar to the CDB mode, but without heat addition in the gas-turbine (i.e. T_{12}=T_{11}).



(a) Sysetm A (b) System B

(c) System C (d) System D

Fig. 6. Variation of exergy losses and destruction with m_2 at *CR*=20, *TIT*=1100K



(a) (b)

Fig. 7. Performance of the system in different modes of operation at *CR*=20, *TIT*=1100K, (a) efficiency, (b) net work

Figure 7 shows the thermal efficiency and net work of the system with *TIT*=1100 and *CR* = 20 in the four modes of operation. Figure 7.a shows that the highest efficiency was obtained when the system was run as a compound engine. Figure 7.b, which shows the variation of the net work with m_2, shows that the compound engine mode also produced more work than the other arrangements for the same mass flow rate. Although the efficiency of the CDB cycle was lower than that of the CCE, it remained higher than that of the IRGT mode over the full range of m_2.

5. Conclusion

A combined diesel-engine gas-turbine system has been presented and optimised using an Excel-based thermodynamic model. The model enabled the pressure ratios of the system's compressors that maximise thermal efficiency to be determined at each combination of the diesel-engine's compression ratio and gas-turbine's inlet temperature. First-law analysis of four configurations of the system showed that it can be designed with a single large compressor that supplies air to both the diesel engine and gas turbine without undermining its performance. Alternatively, the system can be designed with a single turbine that expands the hot gasses from the gas-turbine combustion chamber and the diesel-engine's exhaust. However, in this case the study shows that the system's performance with two low-pressure compressors was better than with one compressor.

A second-law analysis of the system showed that the total exergy loss amounts to about 10.0 % . The major part of exergy losses was that associated with exhaust gas (≈7%) and secondly to it in magnitude was the exergy loss in intercooler IC1 (≈ 2%). The exergy destruction due to gas-mixing and that due to heat transfer in the regenerator both increased with m_2 but their contributions in the total exergy loss were minor. Therefore, further efforts to improve the systems use of fuel's energy should primarily concentrate on recovering the waste thermal energy in the exhaust gas followed by that in the cooling water. The study also analysed the performance of the combined system that has been optimised with a single low-pressure compressor at four modes of operation. By varying the mass-flow rate of the air through the gas-turbine (m_2), the system was simulated in a combined Diesel-Brayton (CDB) cycle, a compound cycle engine (CCE), a regenerative intercooled gas turbine (RIGT), or in an air bottoming cycle (ABC). For the same total flow rate of (m_1+ m_2), the maximum possible thermal efficiency and net work were obtained by running the system in the CCE mode. Although the thermal efficiency of the CDB cycle declined with m_2, it remained higher than that of the RIGT for all values m_2. These results indicate that gas turbines are to be run as peak-load machines. The analysis also showed that running the system as ABC reduced its thermal efficiency without producing any additional work.

Nomenclature

COR	[-]	Cut-off ratio
CR	[-]	Compression ratio
h	[kJ/kg]	Specific enthalpy
m	[kg]	Mass
P	[kPa]	Pressure
PR	[-]	Pressure ratio
Q	[kJ]	Heat transfer
s	[kJ/kg.K]	Specific entropy
T	[K]	Temperature
u	[kJ/kg]	Specific internal energy
v	[m^3/kg]	Specific volume
W	[kJ]	Work
X	[kJ/kg]	Specific flow exergy

Greek characters

ε	[-]	Heat-exchanger effectiveness
η	[-]	Thermal efficiency

Subscripts

C	Compressor
DE	Diesel engine
GT	Gas turbine
T	Turbine

Abbreviations

CCE	Compound cycle engine
CDB	Combined Diesel-Brayton cycle
CDG	Combined Diesel-engine gas-turbine system
IRGT	Intercooled regenerative gas-turbine
ORC	Organic Rankine cycle

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