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THERMO-ECONOMIC OPTIMIZATION AND PARAMETRIC STUDY OF AN IRREVERSIBLE REGENERATIVE BRAYTON CYCLE

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ABSTRACT
Thermo-economic optimizations along with a detailed parametric analysis of an irreversible regenerative Brayton cycle with finite heat capacity of external reservoirs have been carried out. The external irreversibilities due to finite temperature difference and internal irreversibilities due to fluid friction losses in compressor/turbine, regenerative heat loss, and pressure loss are included in the analysis. The thermo-economic function is the ratio of power output to the sum of annual investment, energy consumption and maintenance cost. The optimization of the objective function is done for maximizing the power output and thermal efficiency of the model. A detailed analysis on the efficiency of turbine and compressor, effectiveness of various heat exchangers, heat source inlet temperature and pressure drop irreversibility on power, efficiency and economic function has been carried out. Parametric analysis shows that turbine efficiency has more effect on the performance of model than compressor efficiency for the same set of operating conditions. The model analyzed in this paper gives lower values of various performance parameters as expected and replicates the results of an irreversible regenerative Brayton cycle model discussed in the literature at pressure recovery coefficients of $\alpha_1=\alpha_2=1$.

INTRODUCTION
Brayton cycles have been extensively used in gas power plants, aircrafts, ship propulsion and various industrial usages. Salamon et al. [1] examined externally irreversible Carnot cycle and calculated optimum values of objective function for various performance parameters of endoreversible Carnot heat engine. Leff [2] analysed an endoreversible Brayton heat engine following Curzon and Ahlborn [3] and observed the change in Brayton cycle temperatures while altering maximum work in the cycle. Wu and Kiang [4] optimized power output of Brayton cycle using finite time thermodynamics. Wu [5] optimized the power of an endoreversible Brayton gas heat engine. Wu & Kiang [6] integrated real compression and expansion in Brayton heat engine and found that engine power and engine efficiency are strong functions of the compressor and turbine efficiencies. Cheng et al. [7] used finite time thermodynamics for ecologically optimizing the power output of a closed Brayton cycle and calculated optimal values of power, thermal efficiency and second law efficiency of Brayton cycle. Sahin and Kodal [8-11] performed several thermos-economic optimization studies for refrigerators/ heat pumps based on endoreversible and irreversible mode using FTT approach. Kaushik et al. [12] applied finite time thermodynamic approach to an irreversible regenerative closed Brayton cycle. Wang et al. [13] applied the hypothesis of finite time thermodynamics to analyze an irreversible closed intercooled regenerated Brayton cycle and optimized the intercooler pressure ratio for optimum power and corresponding efficiency. Kaushik et al. [14] performed a thermodynamic analysis of an irreversible regenerative Brayton cycle with isothermal heat addition and optimized the power output in context with working medium temperature. They observed an improvement of 15% in the thermal efficiency of Brayton cycle with heat addition at constant temperature. Tyagi et al. [15] investigated a complex Brayton cycle under maximum ecological and maximum economic conditions and found optimum values of various performance parameters at which the cycle attains maximum values of economic function,

ecological function, and power output and cycle efficiency. Till now, numerous researchers have done thermo-economic optimization studies for different thermal energy conversion systems based on endoreversible and irreversible configurations [16-27]. Finite time thermodynamics has touched various fronts since this technique had been used to analyze and optimize the performance of real thermodynamic processes, devices and cycles. On the basis of recent literature, a model of an irreversible regenerative Brayton cycle with pressure drop as supplementary irreversibility is considered in this paper and expressions for maximum thermo-economic function and corresponding power output/ thermal efficiency of the cycle are obtained. In addition, thermo-economic function chosen as the objective in this study includes annual investment cost, energy consumption cost and maintenance cost. The effect of effectiveness of various heat exchangers, efficiency of turbine and compressor, heat source inlet temperature, pressure drop and economic parameters have been studied in detail and the results are presented on the graphs. The model analyzed in this paper gives lower values of various performance parameters as expected.

THERMODYNAMIC ANALYSIS

An irreversible regenerative Brayton cycle coupled with a heat source and heat sink of finite heat capacity is shown in Fig. 1. In this model, state 1 is the entry point of working medium at compressor and compressed up to state 2. Then the working medium enters the regenerator where its partial heating up to state 2R is done by the turbine exhaust. The working medium next enters the hot side heat exchanger with a pressure drop which is reflected using pressure recovery coefficient, $\alpha_1 = p_3/p_2$ and heated up to state 3, while the heat source temperature decreases from T_{H1} to T_{H2}. The working medium now enters the turbine and expands up to state 4. After expansion, the working medium enters the regenerator to transfer heat partly and then enters the cold side heat exchanger with a pressure drop which is reflected using another pressure recovery coefficient, $\alpha_2 = p_1/p_4$. The working medium is cooled up to state 1, while the heat sink temperature increases from T_{L1} to T_{L2}. Therefore, we consider the closed Brayton cycle 1-2-2R-3-4R-1 with real compression / expansion processes and pressure drop irreversibilities for finite heat capacity of external reservoirs. Process (1-2s) and process (3-4s) are isentropic in nature as shown by dotted lines in Figure 1. The following assumptions are taken into account:

(a) The proposed model operates at steady rate.
(b) The variable heat capacity of heat source and heat sink is considered.
(c) The working fluid used in the proposed model behaves like an ideal gas.
(d) The constant specific heat of the working fluid is assumed.
(e) The pressure recovery coefficients (α_1 and α_2) are assumed to be equal.

The hot, cold and regenerative side heat transfer rates can be presented as [12, 19]:

$$Q_H = U_H A_H (LMTD)_H = C_H (T_{H1} - T_{H2}) \quad (1)$$

$$Q_L = U_L A_L (LMTD)_L = C_L (T_{L2} - T_{L1}) \quad (2)$$

$$Q_R = U_R A_R (LMTD)_R = C_W (T_4 - T_{4R}) \quad (3)$$

where,

$$(LMTD)_H = \frac{(T_{H1} - T_3) - (T_{H2} - T_{2R})}{\ln\{(T_{H1} - T_3)/(T_{H2} - T_{2R})\}} \quad (4)$$

$$(LMTD)_L = \frac{(T_{4R} - T_{L2}) - (T_1 - T_{L1})}{\ln\{(T_{4R} - T_{L2})/(T_1 - T_{L1})\}} \quad (5)$$

$$(LMTD)_R = \frac{(T_4 - T_{2R}) - (T_{4R} - T_2)}{\ln\{(T_4 - T_{2R})/(T_{4R} - T_2)\}} \quad (6)$$

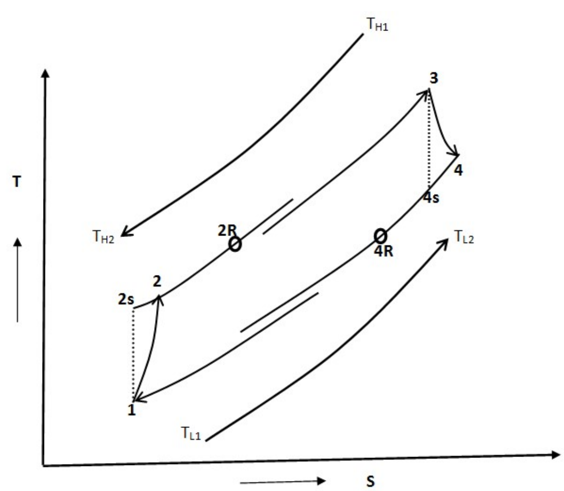

Fig. 1 T-S diagram for irreversible regenerative Brayton heat engine Cycle

From equations (1) to (6),

$$Q_H = \varepsilon_H C_{H,\mathrm{m}} (T_{H1} - T_{2R}) = C_W (T_3 - T_{2R}) \quad (7)$$

$$Q_L = \varepsilon_L C_{L,\mathrm{m}} (T_{4R} - T_{L1}) = C_W (T_{4R} - T_1) \quad (8)$$

$$Q_R = \varepsilon_R C_W (T_4 - T_2) = C_W (T_{2R} - T_2) \quad (9)$$

where ε_H, ε_L and ε_R are the effectiveness of the hot side, cold side and regenerative side heat exchangers respectively and presented as [19]:

$$\varepsilon_H = \frac{1-e^{-N_H(1-C_{H,\min}/C_{H,\max})}}{1-\frac{C_{H,\min}}{C_{H,\max}}e^{-N_H(1-C_{H,\min}/C_{H,\max})}} \tag{10}$$

$$\varepsilon_L = \frac{1-e^{-N_L(1-C_{L,\min}/C_{L,\max})}}{1-\frac{C_{L,\min}}{C_{L,\max}}e^{-N_L(1-C_{L,\min}/C_{L,\max})}} \tag{11}$$

$$\varepsilon_R = \frac{N_R}{1+N_R} \tag{12}$$

The heat capacitance rates and number of heat transfer units can be calculated as:

$$C_{H,\min} = \min(C_H, C_W)$$

$$C_{H,\max} = \max(C_H, C_W)$$

$$C_{L,\min} = \min(C_L, C_W)$$

$$C_{L,\max} = \max(C_L, C_W)$$

$$N_H = \frac{U_H A_H}{C_{H,\min}};\ N_L = \frac{U_L A_L}{C_{L,\min}};\ N_R = \frac{U_R A_R}{C_W}$$

The compressor and turbine efficiencies can be written as:

$$\eta_c = \frac{T_{2s}-T_1}{T_2-T_1} \tag{13}$$

$$\eta_t = \frac{T_3-T_4}{T_3-T_{4s}} \tag{14}$$

Now from equations (7) to (14),

$$T_{4R} = (1-\varepsilon_R)T_4 + \varepsilon_R T_2 \tag{15}$$

$$T_{2R} = (1-\varepsilon_R)T_2 + \varepsilon_R T_4 \tag{16}$$

$$T_1 = (1-b)T_{4R} + bT_{L1} \tag{17}$$

$$T_3 = (1-a)T_{2R} + aT_{H1} \tag{18}$$

$$T_{2s} = (1-\eta_c)T_1 + T_2\eta_c \tag{19}$$

$$T_{4s} = (1-\eta_t^{-1})T_3 + T_4\eta_t^{-1} \tag{20}$$

Parameters 'a' and 'b' are given in Appendix-I. For a given model,

$$T_1T_3 = \alpha T_{2s}T_{4s} \tag{21}$$

where $\alpha = (\alpha_1\alpha_2)^{\frac{k-1}{k}}$ and k is specific heat ratio of the working fluid.

Substituting the values of T_1, T_3, T_{2s} and T_{4s} from equations (17) - (20) into equation (21), we get the quadratic equation in T_2 as:

$$XT_2^2 + YT_2 + Z = 0 \tag{22}$$

Parameters X, Y and Z are given in Appendix- I. Solution of equation (22) gives,

$$T_2 = \frac{-Y+\sqrt{Y^2-4XZ}}{2X} \tag{23}$$

From the first law of thermodynamics,

$$\begin{aligned} P &= Q_H - Q_L \\ &= \varepsilon_H C_{H,\min}(T_{H1}-T_{2R}) - \varepsilon_L C_{L,\min}(T_{4R}-T_4) \end{aligned} \tag{24}$$

Substituting the value of T_{2R} and T_{4R} from equations (15-16) into equation (24) and (7), P and Q_H can be written as:

$$P = z_6 - x_7T_2 - y_7T_4 \tag{25}$$

$$Q_H = z_7 - x_8T_2 - y_8T_4 \tag{26}$$

Parameters x_7, x_8, y_7, y_8, z_6 and z_7 are given in the Appendix-I.

For optimal operation of gas power plant, cost should be as small as possible with maximum power output. Hence, for optimization power output per unit cost is considered as the objective function as proposed by earlier researchers [2, 8-9], and can be written as:

$$F = \frac{P}{C_i + C_e + C_m} \tag{27}$$

Here C_i, C_e and C_m indicate annual investment cost, energy consumption cost and maintenance cost of irreversible regenerative Brayton cycle model respectively. The annual investment cost covers cost of various heat exchangers, compressor and turbine used in Brayton cycle model. Investment cost of the heat exchanger can be assumed to be proportional to total heat transfer area [8, 9] and investment cost of compressor and turbine is assumed to be proportional to their compression / expansion capacities. Therefore, the total annual investment cost of the model can be presented as

$$C_i = a_a(A_H + A_L + A_R) + a_p P$$
$$= a_a(A_H + A_L + A_R) + a_p(Q_H - Q_L) \tag{28}$$

Here a_a is the proportionality constant for the investment cost of the heat exchanger which is the product of the capital recovery factor and investment cost per unit heat exchanger area and a_p is the proportionality constant for the investment cost for the compression and expansion devices which is the product of capital recovery factor and investment cost per unit power output. By using capital recovery factor, the initial investment cost is calculated in terms of equivalent annual payment [8, 9]. The annual energy consumption [8, 9] and the maintenance cost [2] are proportional to the energy input and power output respectively, which can be presented as:

$$C_e = a_q Q_H \tag{29}$$

$$C_m = b_p P = b_p(Q_H - Q_L) \tag{30}$$

a_q is defined as equivalent annual operation hours times price per unit energy input [8, 9] and b_p is defined as equivalent annual operation hours times price per unit power output [10]. Putting equations (28) - (30) into equation (27), the objective function can be written as:

$$F = \frac{P}{a_a(A_H + A_L + A_R) + \beta(Q_H - Q_L) + a_q Q_H} \tag{31}$$

Here $\beta = a_p + b_p$ Thus from equations (27) – (30), we have

$$\beta F = \frac{z_6 - x_7T_2 - y_7T_4}{k_1k_3 + (z_6 - x_7T_2 - y_7T_4) + k_2(z_7 - x_8T_2 - y_8T_4)}$$
$$= \frac{z_6 - x_7T_2 - y_7T_4}{z_9 - x_9T_2 - y_9T_4} \tag{32}$$

Parameters x_9, y_9, k_1, k_2 and k_3 are given in Appendix- I. Equation (32) shows that βF is a bi-variable function of T_2 and T_4 but T_2 is a function of T_4 and other parameters are constant for a typical set of operating conditions. Hence, optimizing equation (32) in context with T_4 i.e. $\frac{\partial \beta F}{\partial T_4} = 0$ and solution for this equation, gives:

$$(x_{11} + T_4)\frac{\partial T_2}{\partial T_4} = z_{11} + T_2 \tag{33}$$

Parameters x_{11} and z_{11} are given in Appendix- I. Putting the values of $\frac{\partial T_2}{\partial T_4}$ and T_2 from equation (23) into equation (33),

$$X_1T_4^2 + Y_1T_4 + Z_1 = 0 \tag{34}$$

Parameters X_1, Y_1 and Z_1 are given in Appendix-I. Solution of equation (34) gives optimum value of T_4 as

$$T_{4,opt} = \frac{-Y_1 + \sqrt{Y_1^2 - 4X_1Z_1}}{2X_1} \tag{35}$$

RESULTS AND DISCUSSION

In order to have mathematical approval of the results, the effects of various performance parameters viz. efficiency of turbine and compressor, effectiveness of various heat exchangers, heat source inlet temperature, pressure drop recovery coefficients and economic parameters on an irreversible regenerative Brayton heat engine model are investigated. Each one of above mentioned parameter is examined by keeping rest parameters constant as T_{H1}=1250 K, T_{L1}=300 K, $\eta_t = \eta_c$=0.8, C_W=1.05 kWK^{-1}, $C_H = C_L$=1 kWK^{-1}, $U_H = U_L = U_R$=2.0 kWK^{-1}m^{-2}, $\alpha_1 = \alpha_2$=0.95, k_1=0.5, k_2=0.1 [12, 18-19]. The obtained results are presented on graphs and discussed in detail as follows:

Effects of turbine and compressor efficiencies (η_t and η_c)

The variations of turbine and compressor efficiencies on maximum thermo-economic function and the corresponding power output/ thermal efficiency of an irreversible regenerative Brayton heat engine cycle with finite capacity heat reservoir are shown in figures 2(a) to 2(c). It is seen from these figures that maximum thermo-economic function and the corresponding power output/ thermal efficiency increases with the increase in component efficiencies (η_t and η_c) which indicates that larger the component efficiency is, better the performance of the cycle. It is also found that turbine efficiency (η_t) has more effect than the compressor efficiency (η_c) not only on the thermodynamic performance but also on the economic performance of an irreversible regenerative Brayton heat engine cycle. Hence, for practical Brayton heat engine, lots of research and investigation is still required on compressor efficiency.

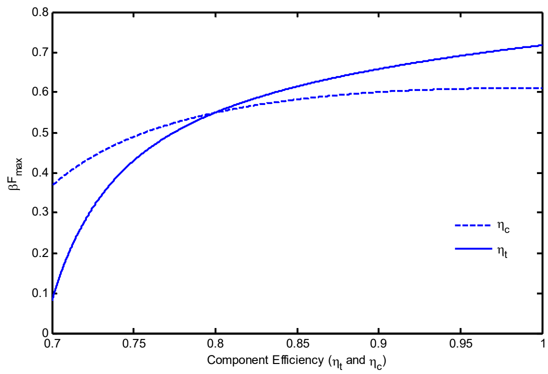

Fig. 2 (a) Variations of max economic function with respect to component efficiency (η_c and η_t)

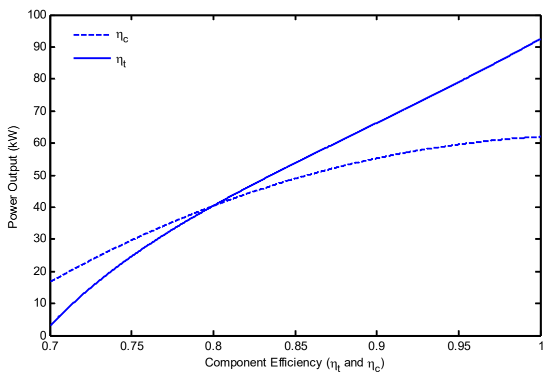

Fig. 2 (b) Variations of power output with respect to component efficiency (η_c and η_t)

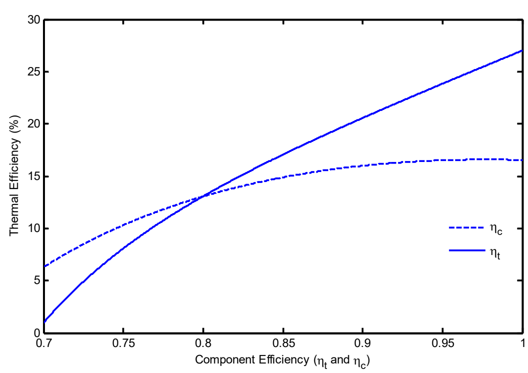

Fig. 2 (c) Variations of thermal efficiency with respect to component efficiency (η_c and η_t)

Effects of heat source inlet temperature (T_{H1})

The variation of heat source inlet temperature on various performance parameters of an irreversible regenerative Brayton heat engine cycle are shown in figure 3(a) to 3(c). It is seen from these figures that the maximum thermo-economic function and the corresponding power output/ thermal efficiency first increases, attains its maximum value and then starts decreasing at higher values of heat source inlet temperature. The power output and thermal efficiency starts decreasing at higher source inlet temperature because of inclusion of pressure drop in the analysis. Because at high source inlet temperature, pressure recovery coefficients decrease and therefore, power output and thermal efficiency starts decreasing. It is further observed that at unity pressure recovery coefficients, the performance parameters increases with increase in heat source inlet temperature. It is also seen that the effect of T_{H1} is more prominent for power output as compared to other performance parameters. Hence, efforts should be made to increase heat source inlet temperature of realistic Brayton heat engine cycle.

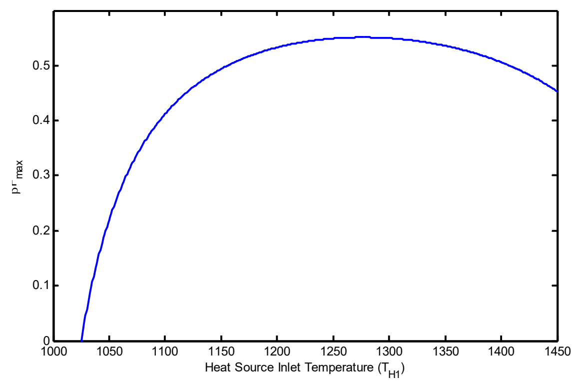

Fig. 3 (a) Variations of max economic function with respect to heat source inlet temperature (T_{H1})

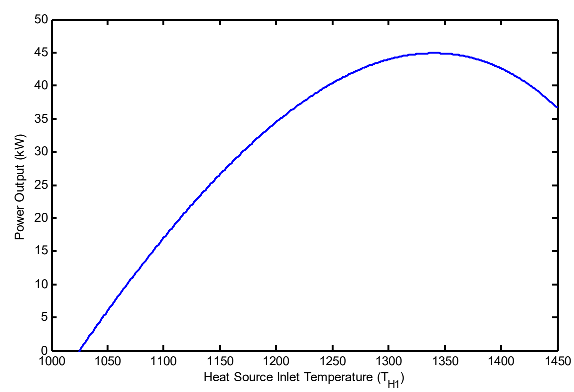

Fig. 3 (b) Variations of power output with respect to heat source inlet temperature (T_{H1})

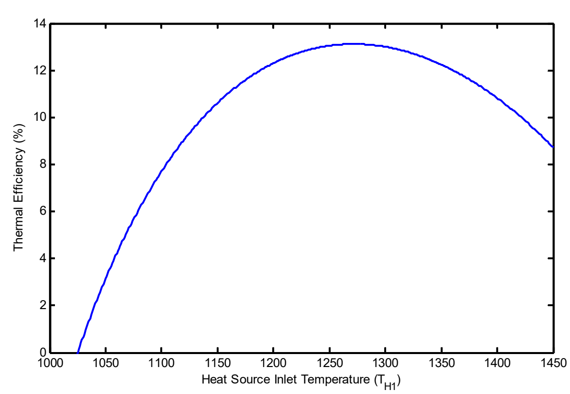

Fig. 3 (c) Variations of thermal efficiency with respect to heat source inlet temperature (T_{H1})

Effect of ε_H, ε_L and ε_R

The variations of hot side, cold side and regenerative side effectiveness on different performance parameters are shown in figures 4(a) to 4(c). It is seen from these figures that the maximum thermo-economic function and the corresponding power output/ thermal efficiency increases as the effectiveness on either side of heat exchanger (ε_H, ε_L and ε_R) is increased. It is also found that the effect of ε_L is more prominent than ε_H and ε_R for all the performance parameters of Brayton heat engine cycle. The results obtained can also be correlated with heat transfer area. It is required to increase the heat transfer area as the effectiveness is increased which results in increase of cost of the system. However, in general, the variations of the performance parameters with respect to various effectiveness of heat exchangers (ε_H, ε_L and ε_R) are not linear.

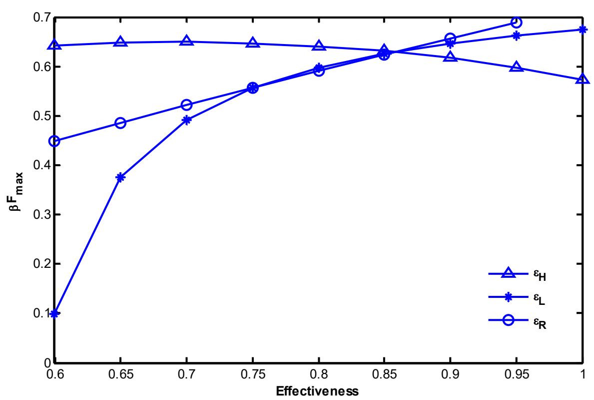

Fig. 4 (a) Variations of maximum economic function w.r.t. effectiveness of various heat exchangers

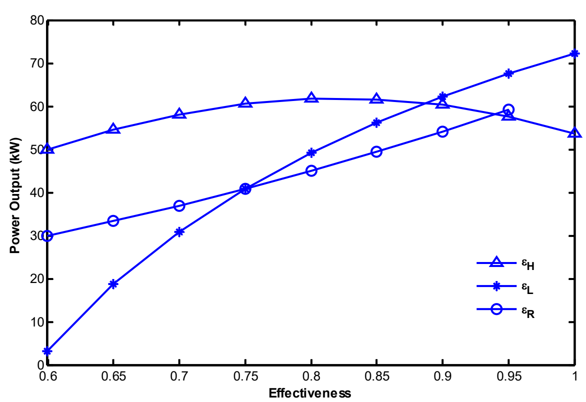

Fig. 4 (b) Variations of power output with respect to effectiveness of various heat exchangers

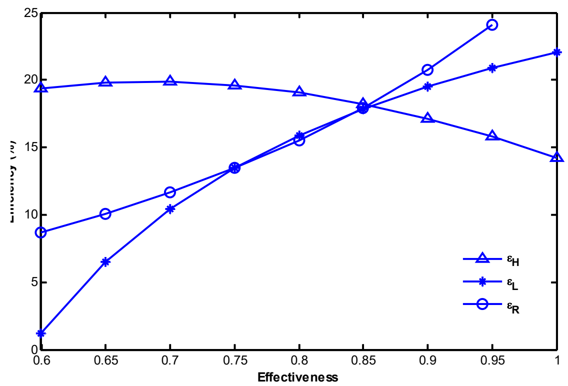

Fig. 4 (c) Variations of thermal efficiency with respect to effectiveness of various heat exchangers

Effects of pressure drop

Figure 5(a) to figure 5(c) shows the effect of pressure drop on various performance parameters of an irreversible regenerative Brayton heat engine cycle. It is seen from these figures that maximum thermo-economic function and the corresponding power output/ thermal efficiency increases as the pressure drop is decreased. It is also seen from these figures that various performance parameters attains their maximum value at zero pressure drop which cannot be achieved in realistic Brayton heat engine cycle. Further, negative value indicates that at lower values of pressure recovery co-efficient, performance of the system deteriorates in terms of power output and thermal efficiency. Keeping this view in mind, the pressure recovery co-efficient should be as high as possible.

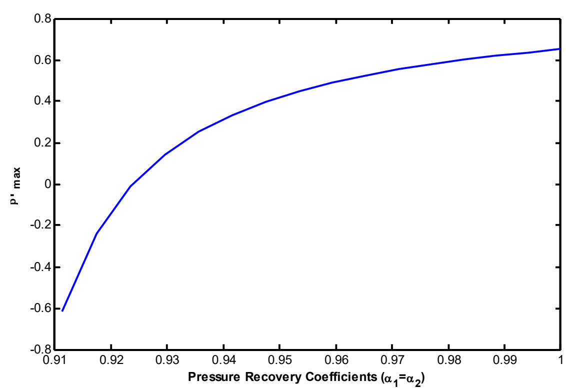

Fig. 5 (a) Variation of max economic function with respect to pressure recovery coefficients

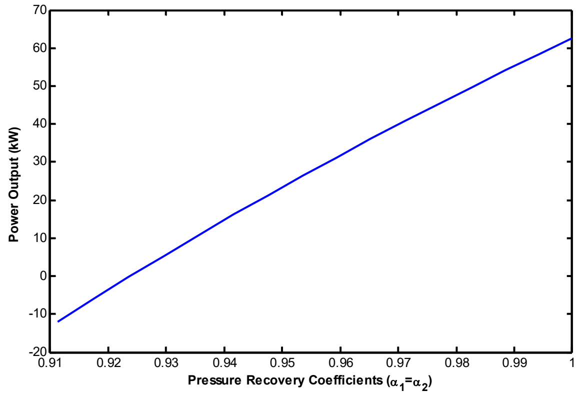

Fig. 5 (b) Variation of power output with respect to pressure recovery coefficients

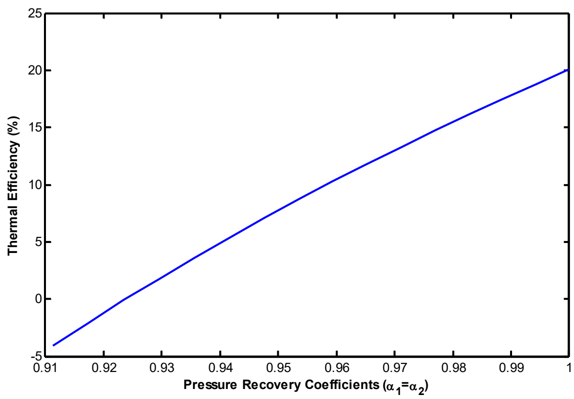

Fig. 5 (c) Variation of thermal efficiency with respect to pressure recovery coefficients

Further, maximum thermo-economic function reflects logarithmic while linear variations with corresponding power output and thermal efficiency.

Effects of economic parameters (k_1 and k_2)

Fig 6 shows the effect of various economic parameters (k_1 and k_2) on maximum thermo-economic function of an irreversible regenerative Brayton heat engine cycle. It is seen from figure 6 that the maximum thermo-economic function decreases with increase in the value of economic parameters (k_1 and k_2) but the effect of k_2 is more prominent than k_1. Since the economic parameter k_1 belongs only to investment cost while k_2 belongs to investment cost and running cost both. Hence, it is proved and verified from figure 6 that running cost is more effective than that of the investment cost.

Comparison of results with Ref [12]:

The results obtained for irreversible Brayton heat engine are compared with results obtained by Kaushik et al. [12] under same set of analytical conditions in Table 1. It is found that the values of P and η are 36.5% and 32.4% lower compared with Ref [12] because of inclusion of pressure drop irreversibility. Hence, it is proved and verified that with the inclusion of irreversibility in the system, the performance parameter decreases. Further, the proposed model replicates the results of an irreversible regenerative Brayton cycle model discussed in the literature at pressure recovery coefficients of $\alpha_1=\alpha_2=1$.

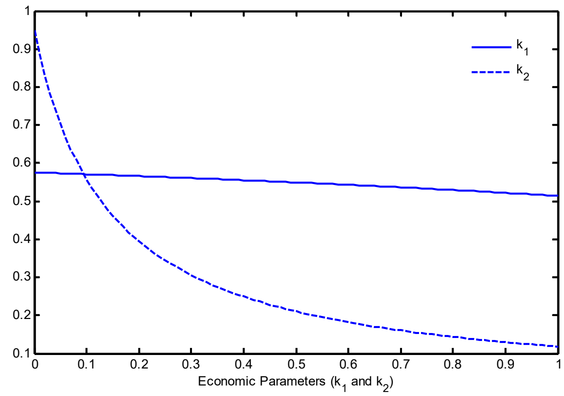

Fig. 6 Variations of maximum economic function with respect to economic parameters (k_1 and k_2)

CONCLUSION

A more practical irreversible regenerative Brayton heat engine cycle model with pressure drop as supplementary irreversibility is examined in this paper. The thermo-economic function is optimized in context with cycle temperature. The corresponding power output and thermal efficiency are calculated for analytical set of operating conditions. The major outcomes of current analysis are:

1. The thermo-economic function value increases with effectiveness of various heat exchangers, component efficiencies, heat source inlet temperature, pressure recovery factors and decreases for economic parameters.
2. It is proved and verified that running cost is more effective than that of the investment cost.
3. It is found that the cold side effectiveness is more prominent for the power output while regenerative effectiveness is dominant factor for thermal efficiency.
4. It is also found that the effect of turbine efficiency is more as compared with compressor efficiency on maximum objective function and the corresponding power output and thermal efficiency.
5. The model analyzed in this paper gives lower values of various performance parameters as expected and replicates the results of an irreversible regenerative Brayton cycle model discussed in the literature at pressure recovery coefficients of $\alpha_1=\alpha_2=1$.

For a typical set of operating parameters, the results obtained here are of large value to understand the divergence of actual performance from ideal performance while counting pressure drop as supplementary irreversibility. The work can further be extended by developing a correlation between component efficiencies and capital cost of compressor / turbine while doing thermo-economic analysis of proposed model.

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Table 1

Systems			Design Variables							Objectives		
	ε_H	ε_L	ε_R	α_1	α_2	T_{H1}	T_{L1}	η_t	η_c	P (kW)	η (%)	βF^*
Present Study	0.75	0.75	0.75	0.95	0.95	1250	300	0.8	0.8	39.80	12.5	0.53
Ref [12]	0.75	0.75	0.75	1	1	1250	300	0.8	0.8	63.00	18.58	-

NOMENCLATURE

A	Area (m^2)
C	Heat Capacitance Rate (WK^{-1})
F	Dimensionless thermo-economic function
k_1	Ratio of a_a to (a_p+b_p)
k_2	Ratio of a_q to (a_p+b_p)
P	Power output (W)
Q	Heat transfer rate (W)
R	Gas Constant ($Jmol^{-1}K^{-1}$)
T	Temperature (K)
U	Overall heat transfer Coefficient, $kWm^{-2}K^{-1}$

Greek letters

η	Thermal efficiency
ε_H	Heat source side effectiveness
ε_L	Heat sink side effectiveness
ε_R	Effectiveness of regenerator

Subscripts

1,2,3,4	State points
H	Heat source
L	Heat sink
R	Regenerator
s	Ideal state

APPENDIX-I

$$X = x_1x_3 - x_2x_4$$

$$X_1 = x_5^2 - 4Xy_5 - x_{13}^2 / x_{12}^2$$

$$x_1 = \varepsilon_R(1-b)(1-\eta_c) + \eta_c$$

$$x_2 = (1-a)(1-\varepsilon_R)$$

$$x_3 = \alpha(1-a)(1-\eta_t^{-1})(1-\varepsilon_R)$$

$$x_4 = (1-b)\varepsilon_R$$

$$x_5 = x_1y_3 + x_3y_1 - x_4y_2 - x_2y_4$$

$$x_6 = x_1z_3 + x_3z_1 - x_4z_2 - x_2z_4$$

$$x_7 = C_W\{a(1-\varepsilon_R) + b\varepsilon_R\}$$

$$x_8 = C_W a(1-\varepsilon_R)$$

$$x_9 = x_7 + k_2x_8$$

$$x_{14} = (x_5x_6 - 2Xy_6)x_{11} - (x_6^2 - 4Xz_5)$$

$$Y = x_5T_4 + x_6$$

$$Y_1 = x_5x_6 - 2Xy_6 - 2x_{13}x_{14} / x_{12}^2$$

$$y_1 = (1-\varepsilon_R)(1-b)(1-\eta_c)$$

$$y_2 = (1-a)\varepsilon_R$$

$$y_3 = \alpha\{\eta_t^{-1} + (1-a)(1-\eta_t^{-1})\varepsilon_R\}$$

$$y_4 = (1-\varepsilon_R)(1-b)$$

$$y_5 = y_1y_3 - y_2y_4$$

$$y_6 = y_1z_3 + z_1y_3 - z_2y_4 - y_2z_4$$

$$y_7 = C_W\{b(1-\varepsilon_R) + a\varepsilon_R\}$$

$$y_8 = aC_W\varepsilon_R$$

$$z_1 = (1-\eta_c)bT_{L1}$$

$$z_2 = aT_{H1}$$

$$z_3 = \alpha a(1-\eta_t^{-1})T_{H1}$$

$$z_4 = bT_{L1}$$

$$z_5 = z_1z_3 - z_2z_4$$

$$z_6 = C_W(aT_{H1} + bT_{L1})$$

$$z_7 = C_W aT_{H1}$$

$$z_8 = z_6 + k_2z_7$$

$$z_9 = k_1k_3 + z_8$$

$$z_{10} = y_9z_6 - y_7z_9$$

$$z_{11} = z_{10} / y_{10}$$

$$x_{10} = x_7 z_9 - x_9 z_6 \qquad y_9 = y_7 + k_2 y_8 \qquad a = \frac{C_H \varepsilon_H}{C_W}$$

$$x_{11} = x_{10} / y_{10} \qquad y_{10} = y_7 x_9 - y_9 x_7 \qquad b = \frac{C_L \varepsilon_L}{C_W}$$

$$x_{12} = 2Xz_{11} + x_5 x_{11} - x_6 \qquad Z = y_5 T_4^2 + y_6 T_4 + z_5$$

$$x_{13} = x_{11}(x_5^2 - 4Xy_5) - (x_5 x_6 - 2Xy_6) \qquad Z_1 = x_6^2 - 4Xz_5 - x_{14}^2 / x_{12}^2$$