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Thermal Modeling and Optimization of Irreversible Cogeneration Power Plant

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Abstract

This paper aims at studying the cogeneration system coupled with outer heat reservoir of finite heat capacity and comprises of both external as well as internal irreversibility. External irreversibility is due to the finite-rate heat transfer between the working fluid and heat reservoirs while internal irreversibility is due to heat dissipation of working fluid. Both types of irreversibility are involved in the study on introducing an internal irreversible parameter which is based on second law of thermodynamics and the variation of temperature of heat reservoir. The experimental analysis presented here is more useful when compared to those given by earlier researchers. The effect of the heat consumer temperature parameter (Ψ) and extreme temperature ratio (Φ) on the maximum total exergy rate and the corresponding exergy efficiency are investigated. The results are drawn in the form of figures by considering various parameters.

Keywords: cogeneration, exergy, endoreversible

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INTRODUCTION

Cogeneration is referred to the generation of energy for the process, from the excess energy supplied to another process. It is the production of electric power and other forms of useful energy such as heat or process steam from the same facility, in other words, cogeneration is said to be a concurrent production of two forms of energy e.g. steam and electricity from a single power plant. Various terms are applied in the name of Cogeneration. They are- “In plant generation (IPG)”, “By product power”, “Total energy”, “Combined heat and power (CHP)”. Most machines, either automobiles or steam turbines are built to perform a particular function. It takes so much energy to get the unit to a level pursuance capable of attaining the function, but it takes less energy indeed to do the task. This is the main reason behind development of cogeneration plant. In practical, the absolute changes in enthalpy and free energy in a process rarely arrives to the values of ideal thermodynamic enthalpy and free energy changes for that process.

Typically the actual consumption of enthalpy and free energy as fuel and other inputs are likely to be more than the ideal thermodynamic limit. It is important to use the differences of actual and ideal requirements of energy, enthalpy, free energy or availability as an index of how a process could be improved. This discussion is intended to meet the challenge by providing an expansion of standard thermodynamics that will forms a borderline on process variables taken in limited or finite time intervals. This discipline is known as finite time thermodynamics or finite temperature difference thermodynamics.

The goal of finite time thermodynamics is to estimate the ideal limits on heat and work of processes performing at finite rates. One approach is to make processes occur in an arbitrary but fix time interval. We then effectuate the analysis further, to determine the most favorable interval in which a process should be directed for optimizing power or any index of optimality. Classical thermodynamics describes about real processes as those which always produce less work and more entropy than the equivalent reversible processes.

Reversible processes are anyway, possible in the limit of infinite range of time. But one do not want to yield finite work in infinite time and/or to run the machine infinitely slow. Thus the concept of finite time thermodynamics transpired, which not only answers the above questions but also solve the problems of heat transfer and energy conversion systems.

Finite time thermodynamics is concerned with the finite temperature difference during external heat transfer between system and source/sink thermal reservoirs. Heat is a kind of energy, which can be transferred from one body to another because of temperature difference between them, but finite temperature difference makes the process irreversible.

Therefore, the heat transfers process, becomes a reversible one as the temperature difference between the two bodies tends to zero.

Although, finite amount of heat transfer through infinitesimal temperature difference may take infinite time or infinite heat transfer area. From the heat transfer theory, the measure of heat (Q) transfer between two bodies is proportional to the temperature difference, the contact area and the time taken i.e.

$$Q \propto A\, \Delta T\, t \quad \text{Or} \quad Q = U\, A\, \Delta T\, t$$

where U is the overall heat transfer coefficient, A is the heat transfer area, ΔT is the temperature difference between the bodies and t is the time taken in the process. For infinitesimal temperature difference [i.e. $\Delta T \rightarrow 0$], either $A \rightarrow \infty$ or $t \rightarrow \infty$ or $U \rightarrow \infty$. Since the materials have finite conductivity so, U will be finite; thus the only possibility is either $A \rightarrow \infty$ or $t \rightarrow \infty$.

If $A \rightarrow \infty$ then it means the heat exchanger area is infinite, the heat transfer unit becomes economically unviable, since as the heat exchanger units are quite expansive. If $t \rightarrow \infty$, that means it takes infinite time to obtain the finite amount of work, then the power {which is work per unit time i.e. $P = W/t \rightarrow 0$ as $t \rightarrow 0$) will be zero.

This means we need finite heat transfer in finite time, which leads to irreversibility. In practical, every machine has some power, which suggests finite work in finite time that means, there should be a finite temperature difference between the working fluid and the external reservoirs.

Thus, irreversible heat transfer due to finite temperature difference is known as “Finite Time Thermodynamics”.

From time to time over last three decades, we have heard about energy crisis. The term has widely used in technical as well as non-technical circle. Yet the term energy crisis is itself quite misleading and perplexing. In April 1977 energy message, President Carter coined a new word- “Cogeneration”. “Cogeneration is the simultaneous production of two forms of energy e.g. steam and electricity from a single power plant”

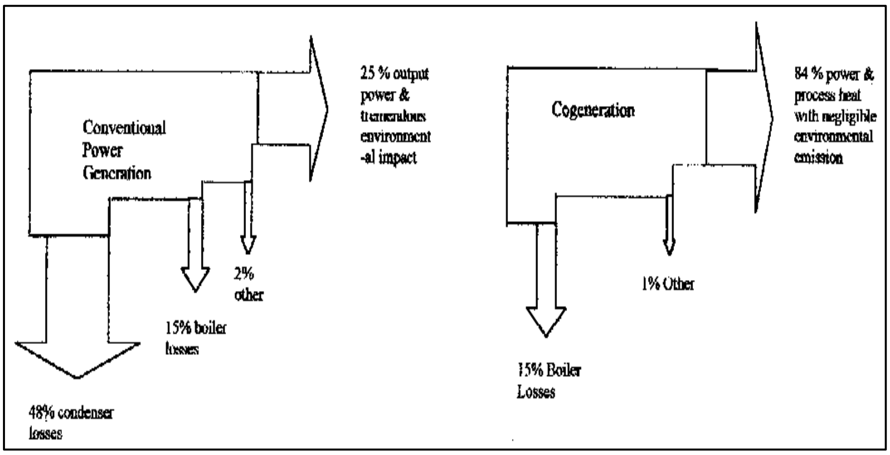

Fig. 1. *Conventional Versus Cogeneration Power Plant.*

Or "the coincident generation of necessary heat and power –electrical or mechanical- or the recovery of low level heat for power generation" investigated by Ingersoll et al. [1] Figure 1 shows the comparison between Conventional and Cogeneration power plant. Cogeneration is also known as, "In plant generation (IPG)", "By product power", "Total energy", "Combined heat and power (CHP)" by Hall.[2] Cogeneration systems often secure wasted thermal energy, usually from an electricity producing device like a heat engine (e.g., steam-turbine, gas-turbine, diesel-engine), and use it for space heating, water heating, industrial process heating, or as a thermal energy source for another system component. The principle technical advantage of cogeneration systems is their ability to improve the efficiency of fuel use in the production of electrical and thermal energy. Less fuel is required to obtain a given value of electrical and thermal energy in a single cogeneration unit than is needed to create the same quantities of both types of energy with separate, conventional technologies (e.g., turbine-generator sets and steam boilers). The technical advantages of cogeneration lead to important environmental advantages. That is, the increase in effectiveness and related reduction in fuel use by a cogeneration system, compared to other customary processes for thermal and electrical energy production, normally produce large reductions in greenhouse gas emissions. These reductions can be as large as 50% in some situations, while the same thermal and electrical services are presented by Mehmet Kanoglu et al. [3]

The optimal design of heat engines and refrigerator systems is a major objective of engineering thermodynamics. In classical thermodynamics, it is well known that the most efficient cycles are reversible cycles. The Carnot heat engine cycle, which is composed of four reversible processes, is the best known reversible cycle observed by Chambadal P. et al. [4] But in reality reversible processes require an infinite process time and/or an infinite system area which is not the case in practice. Chambadal P. et al. [4–6] used the concept of finite-time thermodynamics to improve the power output of a Carnot engine. However, reversible cycles provide the upper bounds for the performance criteria and therefore are considered as models for the actual systems by Bejan A. et al. [7] Nowadays cogeneration power plants (in which heat and power are produced together) are widely used to minimize the transmission and distribution losses by Chandra H. et al. [8]

According to Chandra, Hall and Wilson,[4,8,9] cogeneration power plants are also known as `in-plant generation (IPG)', `combined heat and power (CHP)', and `by-product power'. These cogeneration power plants are more beneficial in terms of energy and exergy efficiencies than plants, which produce heat and power separately by Sahin et al. [10] Sahin and Kodal [10] worked on the exergy optimization for an endoreversible cogeneration cycle which was based on finite-time thermodynamic having infinite heat capacity and evaluated the optimum values for design parameters of the cogeneration cycle at maximum exergy output.

The exergy optimization of irreversible cogeneration cycle based on finite time thermodynamics which deals with the internal irreversibility caused due to entropy generation during the internal process for more accurate result of optimum channel of design parameter Chandra H. et al.[11] However, the only advantage of their model was that they considered cogeneration plant to be coupled with thermal reservoir of infinite heat capacity and also, the design parameter are very much influenced by irreversibility of finite-rate heat transfer.

THERMODYNAMIC FORMULATION

Figure 2 shows the schematic model of irreversible cogeneration cycle coupled with finite heat capacity heat exchanger. The model of the irreversible cogeneration cycle (in which both external and internal irreversibility's are considered) and its T-S diagram is shown in Figure 3 and 4respectively. There are three heat sources, Q_H, Q_L and Q_K between which the cogeneration cycle operates. The temperatures of the working fluids exchanging heat with this heat sources at Q_H, Q_K and Q_L are T_H, T_K and T_L respectively. Cogeneration power plant model is shown in Figures 1 and 2, in which steam is extracted from some stages of steam turbine at intermediate pressure and feed to consumers for process applications such as heating. The condensed steam is then pumped back to boiler of steam plant.

(a) Q_H is the rate of heat transfer from heat source at T_{mH} to the warm working fluid at the constant temperature T_H in process 2'-3

(b) Q_K is the rate of heat transfer from the working fluid at constant T_K to the heat consuming device at T_{mK} in process 4'-5

(c) Q_L is the rate of heat transfer from the cold working fluid at constant T_L to the heat source at T_{mL} in process 6'-1

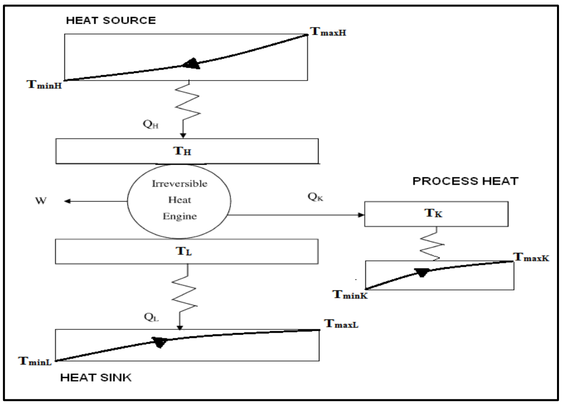

Fig. 2. *Schematic Model of Irreversible Cogeneration Cycle Coupled With Finite Heat Capacity Heat Exchanger.*

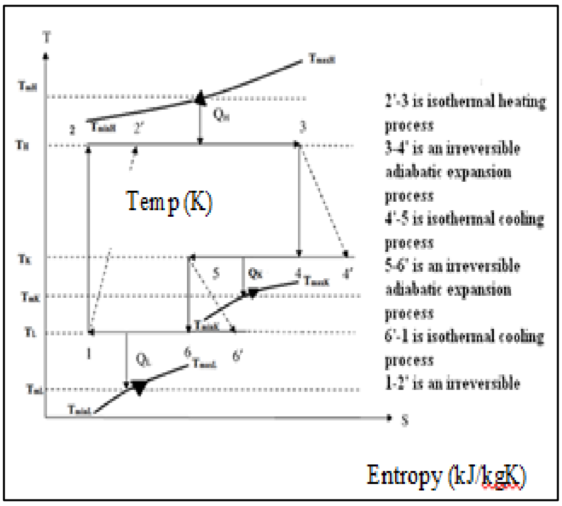

Fig. 3. *T–S Diagram of Irreversible Cogeneration Cycle Coupled With Finite Capacity Heat Exchanger.*

The rate of heat input from high temperature heat source to the cogeneration cycle is given by

$$Q^{\bullet}_{H}=\frac{Q_H}{t_H}=U_HA_H\left[\frac{(T_{maxH}-T_H)-(T_{minH}-T_H)}{\ln\left(\frac{T_{maxH}-T_H}{T_{minH}-T_H}\right)}\right]=m_HC_H(T_{maxH}-T_{minH})$$

(1)

We have

$$T_{minH}=T_H+(T_{maxH}-T_H)e^{-\left(\frac{U_HA_H}{m_HA_H}\right)} \quad (2)$$

Substituting the value of T_{minH} in Equation (1)

$$Q^{\bullet}_{H}=\frac{Q_H}{t_H}=m_HC_H(T_{maxH}-T_H)\left(1-e^{-\left(\frac{U_HA_H}{m_HA_H}\right)}\right) \quad (3)$$

$$T_{maxK}=T_K+(T_{minK}-T_K)e^{-\left(\frac{U_KA_K}{m_KA_K}\right)} \quad (4)$$

Exergy due to the work done is given by E_W

$$E_W=\alpha(T_{maxH}-T_H)-\beta(T_L-T_{minL})-\gamma(T_K-T_{minK}) \quad (5)$$

Exergy due to the process heat is given by:

$$E_Q=Q^{\bullet}_{K}\left(1-\frac{T_{mKL}}{T_K}\right) \quad (6)$$

Hence total exergy is given by,

$$E_T=\alpha(T_{maxH}-T_H)-\beta(T_L-T_{minL})-\gamma(T_K-T_{minL})\frac{T_{mKL}}{T_K} \quad (7)$$

Power to process heat ratio is given by

$$R=(Q_H-Q_L-Q_K)/Q_K \quad (8)$$

$$R=\frac{\alpha(T_{maxH}-T_H)-\beta(T_L-T_{minL})-\gamma(T_K-T_{minK})}{\gamma(T_K-T_{minK})} \quad (9)$$

Thus the Lagrangian function is given by,

$$F=\alpha(T_{maxH}-T_H)-\beta(T_L-T_{minL})-\gamma(T_K-T_{minK})\frac{T_{mL}}{T_K}+\mu_1\left\{\frac{\alpha I(T_{maxH}-T_H)}{T_H}-\frac{\beta(T_L-T_{minL})}{T_L}-\frac{\gamma(T_K-T_{minK})}{T_K}\right\}$$

$$+\mu_2\left\{\frac{\alpha(T_{maxH}-T_H)-\beta(T_L-T_{minL})-\gamma(TK-T_{minK})}{\gamma(TK-T_{minK})}-R\right\} \quad (10)$$

We have

$$T^*_L = T^*_H \sqrt{\frac{T_{minL}}{I \times T_{maxH}}} \tag{11}$$

Now putting value of T^*_L in equation (6), we get

$$T^*_H = \frac{\alpha T_{maxH} + \beta T_{minL} - \gamma (T^*_K - T_{minK})(1+R)}{\alpha + \beta \sqrt{\frac{T_{minL}}{I \times T_{maxH}}}} \tag{12}$$

Finally by solving the above equation 1-8 and putting the value T^*_H, we get

$$-(\alpha I + \beta + \gamma) T^*_K [\alpha T_{maxH} + \beta T_{minL} - \gamma (T^*_K - T_{minK})(1+R)] + (\alpha I T_{maxH} + \alpha \sqrt{I \times T_{maxH} \times T_{minL}}) T^*_K \times \left(\alpha + \beta \sqrt{\frac{T_{minL}}{I \times T_{maxH}}} \right) + \gamma T_{minK} [\alpha T_{maxH} + \beta T_{minL} - \gamma (T^*_K - T_{minK})(1+R)] = 0 \tag{13}$$

After simplifying the above equation, we get

$$T^{*2}_K - (T_{minK} + B + C) T^*_K + T_{minK} \times B = 0 \tag{14}$$

And further solving, we get

$$T^*_K = \frac{(T_{minK} + B + C) \pm \sqrt{(T_{minK} + B + C)^2 - 4 \times B \times T_{minK}}}{2} \tag{15}$$

Where

$$B = \frac{\alpha T_{maxH} + \beta T_{minL} + \gamma (1+R) T_{minL}}{(1+R)(\alpha I + \beta + \gamma)}$$

$$C = \frac{\alpha \beta \left(\sqrt{T\max_H} - \sqrt{T\min_L} \right)^2}{(1+R)(\alpha I + \beta + \gamma)}$$

Now from equation (9),

$$\alpha (T_{maxH} - T^*_H) = R \gamma (T^*_K - T_{minK}) + \beta (T^*_L - T_{minL}) + \gamma (T^*_K - T_{minK}) \tag{16}$$

Put the value of equation (16) in equation (7), we get

$$E^*_T = \gamma (T^*_K - T_{minK}) \left\{ 1 + R - \frac{T_{mKL}}{T^*_K} \right\} \tag{17}$$

RESULT AND DISCUSSION

In this section, the effect of the heat-consumer temperature parameter (Ψ) and extreme temperature ratio (Φ) on the maximum total exergy rate and the corresponding exergy efficiency are discussed.

Effect of Heat-Consumer Temperature Parameter

The effect of the heat-consumer temperature parameter on the maximum total exergy rate and the corresponding exergy efficiency, with the ratio of power output to process heat R is shown in Figures 4 and 5, respectively.

The value of Φ and I is taken as 4 and 1.12 and the value of (Ψ) is varied from 1.2 to 1.8 and the value of R is varied from 0 to 20. It is observed in this curve that for a particular value of heat-consumer temperature parameter (Ψ), exergy rate and corresponding efficiency varies inversely with respect to power to heat ratio.

With increasing value of (Ψ), slope of the curve increases. Initially up to R=6, the value of exergy rate and corresponding efficiency decrease and then with higher values of R, exergy rate and corresponding efficiency become constant.

The reason for these decreases is that the temperature T^*_K at maximum total exergy increases for increasing T_{minK}, which is the heat-consumer temperature.

The increase in T^*_K causes the isentropic expansion work of the heat engine and thus causes the exergetic performance of the irreversible cogeneration system to decrease.

The heat-consumer temperature must be kept low to decrease T^*_K in order to obtain a high exergetic performance from the cogeneration system.

The value of the maximum total exergy rate and exergy efficiency at maximum total exergy for the irreversible cogeneration cycle is lower than that for the endoreversible cogeneration cycle due to the presence of cycle irreversibility parameter in the irreversible cogeneration cycle.

The effect of external irreversibility on the exergy & exergy efficiency is on the lower side as compared to the previous papers i.e. exergy optimization for endoreversible cogeneration cycle by Sahin and Kodal &

Thermal exergy optimization for an irreversible cogeneration power plant by Chandra and Kaushik due to in cooperation of external irreversibility.

Effects of Extreme Temperature Ratio (Φ)

The effect of the extreme temperature ratio (Φ) for various values of R on E_{Tmax} and $\varepsilon*$ are shown in Figures 5–7, respectively. For this purpose, values of Ψ and I are taken as 1.33 and 1.11, while the value of Φ is varied from 2 to 7 and the value of R is varied from 0 to infinity.

Although both E_{Tmax} and $\varepsilon*$ increase for increasing Φ and decreasing R, the increase in E_{Tmax} is more rapid than $\varepsilon*$. Here, also the values of E_{Tmax} and $\varepsilon*$, for the irreversible cogeneration cycle is less than that for the endoreversible cogeneration cycle due to the presence of the cycle irreversibilities.

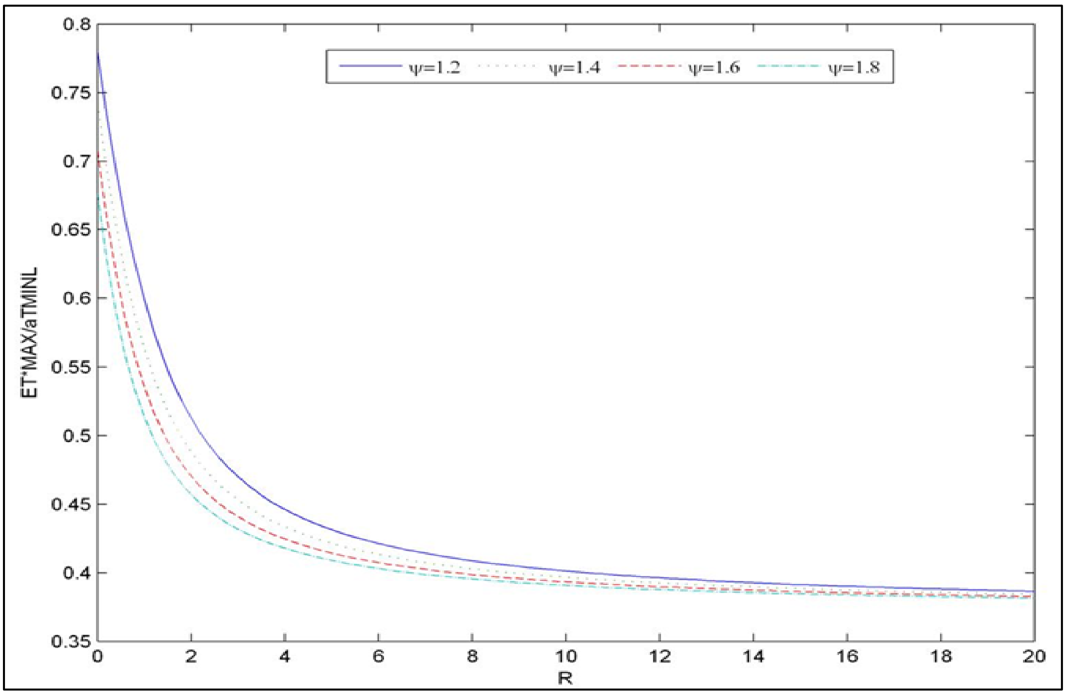

Fig. 4. *Variation of Maximum Total Energy Rate With Respect to R for Various Ψ, Φ=4,* $\propto = \beta = \gamma$.

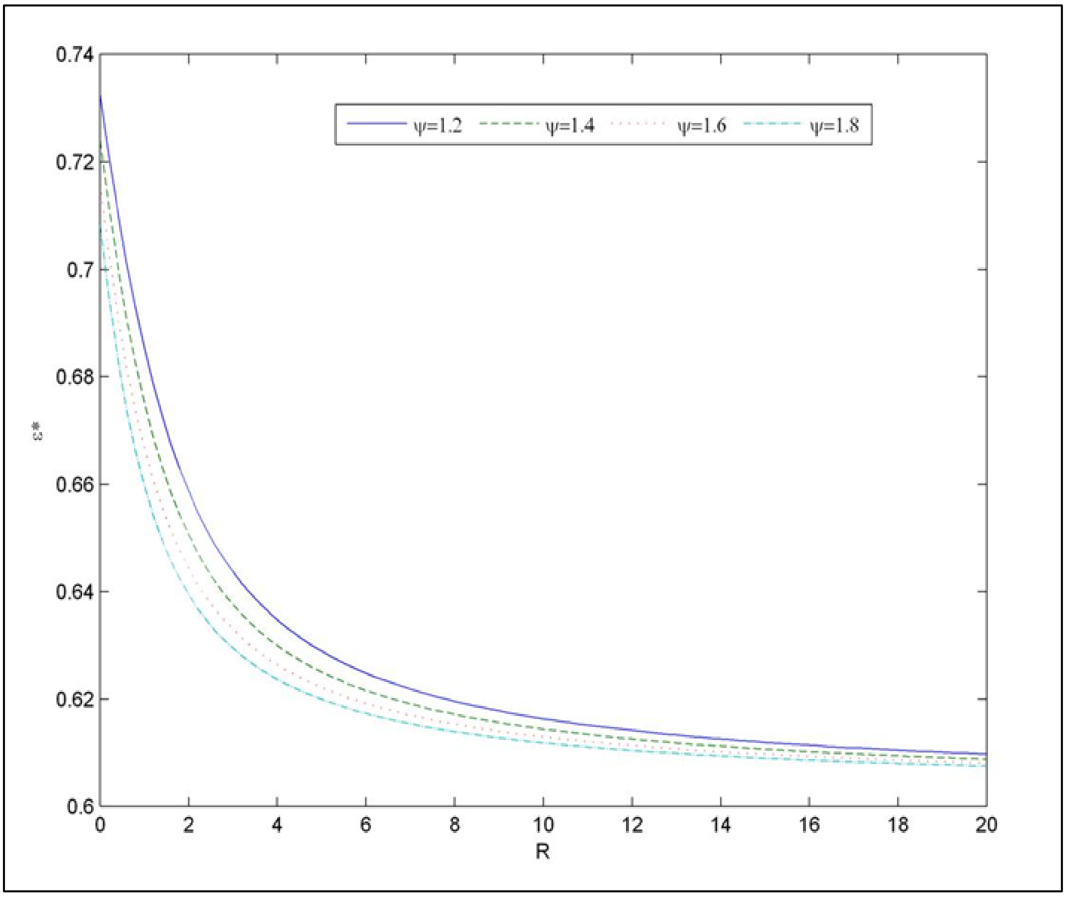

Fig. 5. *Variation of Exergy Efficiency at Maximum Total Energy With Respect to R forVarious Ψ , Φ=4,* $\propto = \beta = \gamma$.

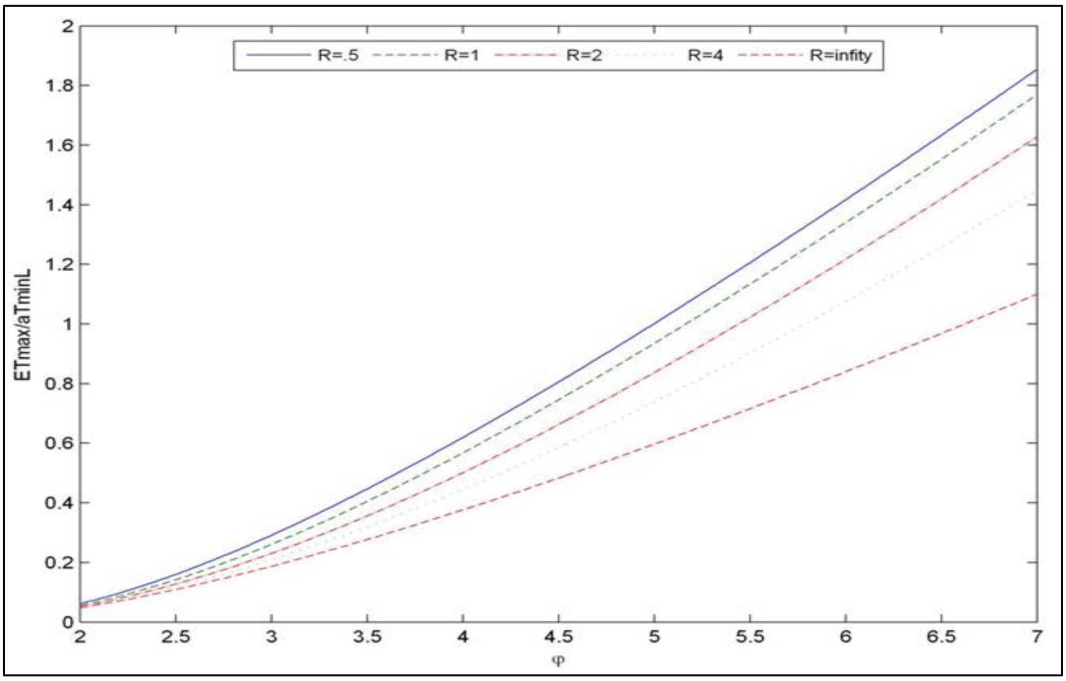

Fig.6. *Effects of Φ and R on total exergy rate for Ψ=1.33* $\propto = \beta = \gamma$.

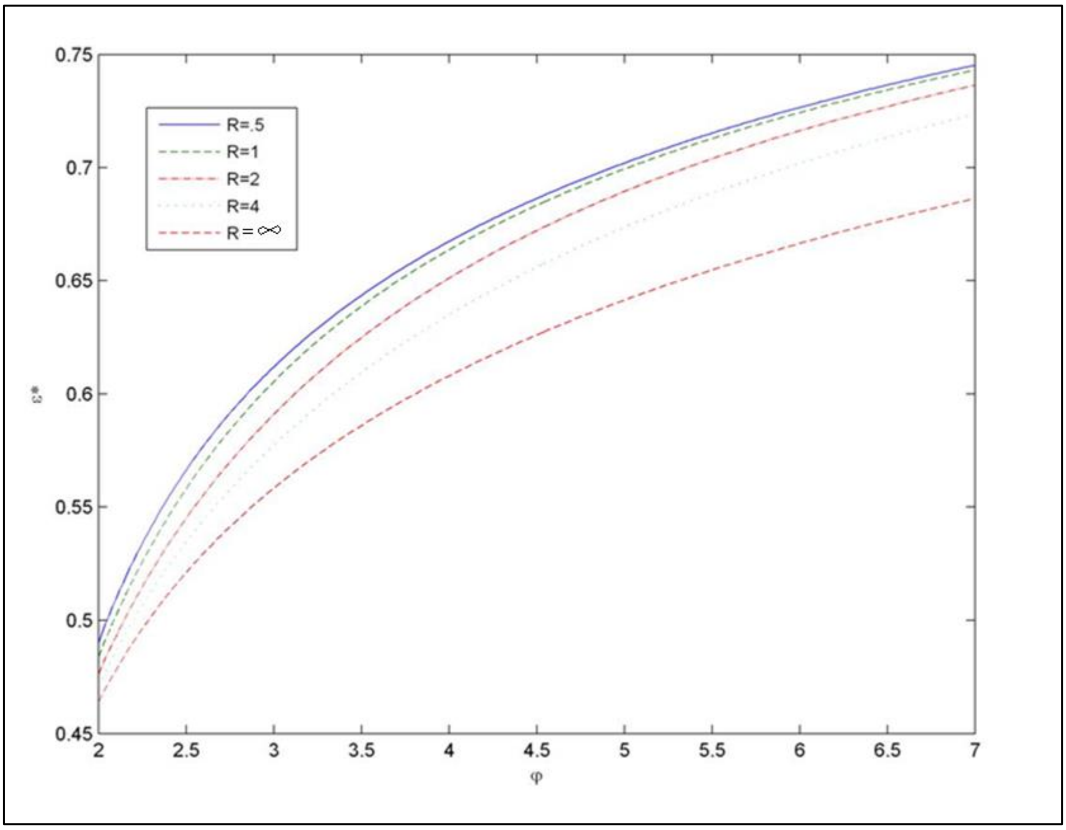

Fig. 7. *Effects of Φ and R on exergy efficiency at maximum total exergy for Ψ=1.33* $\propto = \beta = \gamma$.

CONCLUSION

The thermal exergy optimization for an irreversible cogeneration cycle has been carried out by introducing the cycle irreversibility parameter as a modified criterion. From this study it is clear that the endoreversible cogeneration cycle is a special case of the irreversible cogeneration cycle. The exergetic performance of the irreversible cogeneration cycle for different design parameters has been analyzed and the performance is always less than that for the endoreversible cogeneration cycle case, due to the presence of the cycle irreversibility parameter. In order to get the high exergetic performance of the irreversible cogeneration system the heat-consumer temperature should be as low as possible.

The value of the optimum conductance-allocation ratio on heat-source side is 0.5, and for the heat-sink and heat-consumer sides, depend R,Ψ & Φ In our continuing search for a theoretical upper limit for the exergetic performance, the new equations with cycle irreversibility shows a further insight over the exergetic performance of the endoreversible cogeneration cycle. The results obtained are quite useful for investigating the scope of cogeneration power plants.

Future work can be finite time exergy based ecological optimization of cogeneration cycle considering irreversibility between the heat source/sink and the irreversibility within the cycle. This consists of maximizing a function representing the compromise between the power output and entropy production rate of the cogeneration cycle.

NOMENCLATURE

T Temperature (K)
E Exergy rate (kJ/s)
I Irreversibility
Q Rate of heat transfer (kJ/s)
R power to process heat ratio
$R_{\Delta S}$ Cycle internal irreversibility parameter
S Entropy generation rate (kJ\kgK\s)
W Power generated from cogeneration plant (kJ/s)
c Thermal conductance(kJ/kgK)
C Thermal conductance (kJ/K)
m Mass flow rate of working fluid(kg/s)
1, 2, 3 State points
t Time(s)
U Over all heat transfer coefficient (kJ/m^2K)
A Area (m^2)
x Thermal conductance allocation ratio for the heat-source
y Thermal conductance allocation ratio for the heat-sink
z Thermal conductance allocation ratio for the heat-consumer side

Greek Letters

ε Effectiveness of heat exchanger
α, β, γ Product of thermal conductance and effectiveness of heat exchangers of heat source, heat sink and consumer heat, respectively
μ_1, μ_2 Lagrangian multipliers
λ Overall thermal conductance
Φ Ratio of heat-source to heat-sink temperature
Ψ Ratio of heat-consumer to heat-sink temperature
ξ Ratio of max. sink temp. to min. sink temp.

Subscript

max Maximum
min Minimum
H Heat source
L Heat sink
K Process heat
m Mean
T Total
sink Lower temperature reservoir
source Higher Temperature reservoir

Super Script

* Maximum total exergy condition
• Rate
' Irreversible process

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