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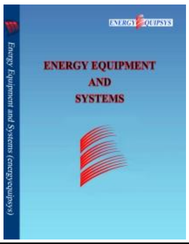

Analysis of micro-recuperators in small-sized gas turbines –manufacturing potential of Iran

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ABSTRACT

Heat recuperation is often used to improve the overall cycle efficiency of gas turbines. However, generally in small- scale gas turbines, it has a negative effect on turbine inlet temperature, pressure ratio and pressure drops, and thus decreases the overall cycle efficiency. In this paper, a thermodynamic model is performed to evaluate recovered heat as a function of heat exchanger effectiveness, pressure drops, and defines the overall cycle energy and exergy efficiency. A high heat exchanger effectiveness, and low pressure drops are favorable to achieve maximal cycle energy efficiency. The main challenge in recuperator designis to find compromise between these conflicting requirements. Hence, a thermodynamic model is developed to determine recuperator design with an aim to maximize overall cycle energy and exergy efficiency. Further, to analyze Iran's manufacturing and its technological capabilities,a pre-requisite is considered for the proposed model. Hence, Iran's capability level could be determined. Finally, a case study of selecting recuperator of a 200kW gas turbine is conducted. Industrial gas turbines show performance characteristics that distinctly depend on ambient and operating conditions. They are influenced by site elevation, ambient temperature, and relative humidity. Proper application of gas turbines requires consideration of these factors. Thus, in this study, the effect of ambient conditions on overall cycle energy and exergy efficiency is performed by the proposed thermodynamic model.

Keywords: Recuperator, Gas Turbine, Efficiency.

1. Introduction

The gas turbine technology has progressed over the past five decades. Micro- and mini-gas turbines (to produce power range of 5–500 kW) are new class of global power generation technology. They play an important role in generating power for standalone cases, and interconnections.

A micro-gas turbine (MGT) consists of a generator, compressor, combustion chamber, turbine, and a frequency converter that function together to generate power for small-scale use; they have good fuel flexibility. The small turbines enable small energy users to generate their own electricity to secure power supply, even at peak load periods, or power shortage. In many geographical places, the micro turbines could be employed to produce power at a competitive cost.

Microturbines with energy efficiency of about 30% are well-suited to meet energy requirements of small users, such as schools, apartment buildings, restaurants, offices, and small business centers [1]. Micro- and mini-

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gas turbines are designed and developed (power range of —500kW) in many developed countries for distributed power generation [1]. Figure 1 shows the general schematic of microturbine cycle.

Once gone through filter, the air enters the compressor, and after that the compressed air goes through recuperator. In recuperator, turbine exhaust gas causes increase in air temperature. After being heated, the air passes into combustion chamber, mixes with the fuel and burns. The combusted air/fuel mixture expands through the turbine, and the rotation of the turbine drives, and the shaft which powers both the compressor and generator. The exhaust from the turbine is fed back through the recuperator to preheat the compressed air. This cycle repeats and causes the generator to rotate and generate power [2].

Heat recuperation is used to improve the overall cycle efficiency of standard gas turbines. In small size (micro and mini) gas turbines, this advancement is question able to some extent. The reason is that both achievable pressure ratios and turbine inlet temperatures are significantly lower, and pressure drops are much larger compared to conventionally-sized gas turbines. Hence, a thermodynamic model should be performed to evaluate recovered heat as a function of heat exchanger effectiveness and pressure drops, and defines overall cycle efficiency. Several works have been published on MGT recuperators focusing on theoretical performance [3,4].Galanti et al., [5] present thermoeconomic analysis of micro-gas turbines up to 500 kWe. This analysis emphasizes the need to minimize specific capital cost, and optimize the MGT size to match market needs. This paper is focused on the basis of thermodynamic, geometric, and capital cost parameters of the main MGT devices, and on economic scenario, such as,fuel cost and cost of electricity for different micro-gas turbines size in the range of 25–500 kWe.

Clay et al., [6] used computational fluid dynamics to investigate a recuperator design, based on industry standard stock materials, and manufacturing techniques. Arecuperator for a 1-kW MGT CHP unit is established. Dimensional reasoning confirmed the existence of optimum pipe diameter combinations for minimum pressure drop. Moreover, maximum heat exchanger effectiveness is achieved using minimum pressure drop pipe combination with pipe length.

Shih et al., [7] simulated numerically Swiss-roll recuperator for the future higher efficiency microturbines. The proposed Swiss-roll recuperator is basically primary surface type. It is composed of two flat-plates that are wrapped around each other, creating two concentric channels of rectangular cross-section. In this study, from a theoretical analysis, the number of turns, the corresponding channel widths, and the required number of transfer unit of recuperator is defined. The consequent pressure loss through the recuperator is predicted.

McDonald [8] discussed recuperator concept that meets the demand requirements for microturbines. The role of the recuperator has on turbo generator performance is discussed, together with a summary of the early prototype heat exchanger development. The major requirements, features, and cost goals for a compact primary surface recuperator for microturbine service are covered.

Further, some research has been done on

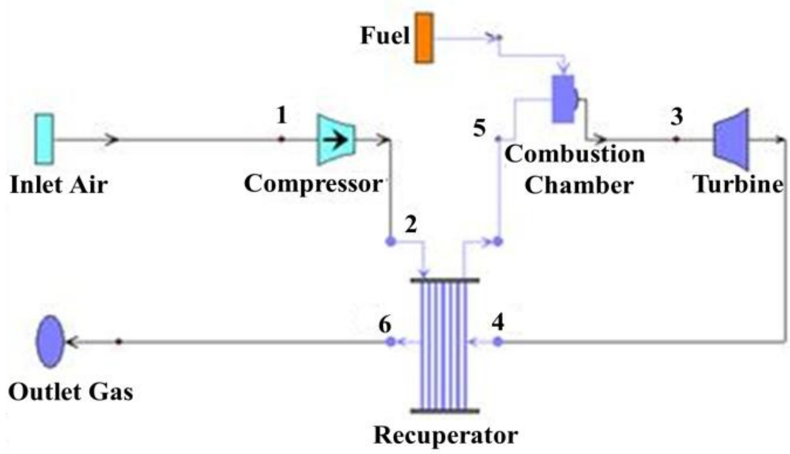

Fig.1. Schematic of recuperated microturbine cycle

technology development [9,10,11]) and system optimization. For example,Stevens et al., [12]showed that the optimal cold and hot side pressure drops of a micro-recuperator are uniquely correlated, when integrated in a gas turbine. A simplified optimization procedure is applied to design a micro-recuperator in a 1.5 kW gas turbine.

Liu et al., [13] showed an optimization method for the recuperator surface, with objectives of heat-transfer performance, exchanger weight, and pressure loss. The influencing factors of optimization are to determine the independent design variables. A multi-objective optimization design model is established, and the specific expressions of each objective are deduced. The result shows that the subjective factors can be avoided in choosing corrugated foil geometrical sizes, and the best design of optimum performance can be obtained.

The objective of system optimization, and choice of decision variables are not independent; they are to be matched each other to establish a suitable optimization strategy [14]. Because, the laws of thermodynamics allow to see individual part of energy conversion systems, and the complete system that makes the selection of thermodynamics derived functions as the most suitable for a global analysis of a complex energy system, or its various sub-systems [15]. The analytical model of complete system is introduced to determine the effect of pressure drops and heat exchanger effectiveness on the energy efficiency of the cycle. The results of this model are verified with the results of *GasTurb10* software. Further, the technological potential of Iran in the case of recuperated microturbines is determined, and the model justified to Iran's capability. Then, the proposed model was used to study the optimal energy efficiency of designed microturbine in Turbine Machine m.e. Co.(MAPTA). Finally, the effects of ambient temperature, elevation, and relative humidity on the performance of the designed micro-gas turbine are shown.

Nomenclature

Alphabet

E^{CH}	Chemical Exergy
C_{Pi}	Cpesific heat capacity of stream in ith component (C=compressor, T=turbine)
$E_{D,i}$	Exergy destruction of ith component
E_i	Exergy of ith stream
E^{KN}	Kinetic Exergy
E^{PH}	Physical Exergy
E^{PT}	Potential Exergy
h_i	Enthalpy of ith stream
k, k'	The ratio of specific heat
LHV	Low heating value of natural gas
$\dot{m}_i$	Mass flowrate of ith stream
Q_{in}	Input heat
s_i	Entropy of ith stream
T_i	Temperature of ith stream
u_i	Internal energy of ith stream
v_i	Specific volume of ith stream
W_C	Compressor work
x_j	Concentration of species j in the stream i

Greek

β	Fuel to air ratio
ε	Turbine to compressor pressure ratio
η	Efficiency of ith component (C=compressor, T=urbine, R=recuperator)
η_{energy}	Energy efficiency
η'_i	Exergy efficiency of ith component
π_i	Pressure ratio in ith component (C=compressor, T=turbine)

2.Thermodynamic Analysis

To study the effect of recuperator pressure drop, and its effectiveness on the gas turbine cycle efficiency, an analytical model is established, according to basic thermodynamic relations. The operating cycle is depicted in Fig.2.

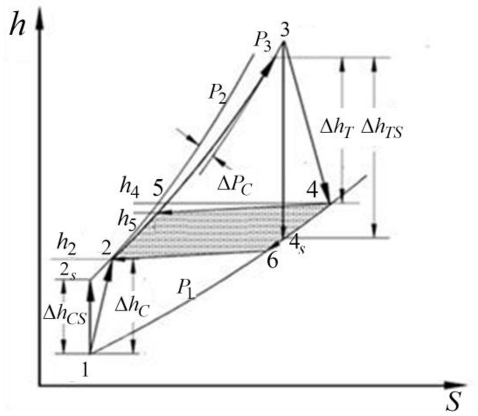

Fig.2. Recuperated gas turbine h-s diagram

$$\eta_{energy} = \frac{W_T - W_C}{Q_{in}} \quad (1)$$

where:

$$W_T = (\dot{m}_1 + \dot{m}_f)(h_3 - h_4) = (\dot{m}_1 + \dot{m}_f)\overline{C_{PT}}(T_3 - T_4) \quad (2)$$

$$W_C = \dot{m}_1(h_2 - h_1) = \dot{m}_1\overline{C_{PC}}(T_2 - T_1) \quad (3)$$

$$Q_{in} = (\dot{m}_1 + \dot{m}_f)\overline{C_{PCC}}(T_3 - T_5) \quad (4)$$

From the definition of compressor efficiency, the following could be dervied:

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{T_{2s}-T_1}{T_2-T_1},$$

and

$$\frac{T_{2s}}{T_1} = \pi_C^{m_C}$$

where

$$\pi_C = \frac{P_2}{P_1},$$

and

$$m_c = \frac{k-1}{k} \quad (5)$$

Hence, it can be written as:

$$\frac{T_2}{T_1} = (1 + \frac{\pi_C^{m_C} - 1}{\eta_C}) \quad (6)$$

Further, from the definition of turbine efficiency, the following could be derived:

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

and

$$\frac{T_{4s}}{T_3} = \pi_T^{-m_T}$$

where

$$\pi_T = \frac{P_3}{P_4}$$

and

$$m_T = \frac{k' - 1}{k'} \quad (7)$$

Hence, it can be written as:

$$\frac{T_4}{T_3} = 1 - \eta_T(1 - (\epsilon\pi_C)^{-m_T})$$

where

$$\pi_T = \epsilon\pi_C \quad (8)$$

The effectiveness of the recuperator defines as:

$$\eta_R = \frac{h_5 - h_2}{h_4 - h_2} = \frac{T_5 - T_2}{T_4 - T_2}$$

$$\xrightarrow{Therefore} T_5 = T_2 + (T_4 - T_2)\eta_R \quad (9)$$

By using Eq.(9), in Eq.(4):

$$Q_{in} = \dot{m}_1(1 + \beta)\overline{C_{PCC}}(T_3 - (1 - \eta_R)T_2 - T_4\eta_R) \quad (10)$$

By inserting Eqs. (6) and (8), in Eqs.(2) and (3); and finally inserting Eqs.(2) and (3), in Eq.(1), objective function derives.

$$\eta_{energy} = \frac{\overline{C_{PT}}\eta_T\theta[1 - (\epsilon\pi_C)^{-m_T}](1 + \beta) - \frac{1}{\eta_C}\overline{C_{PC}}(\pi_C^{m_C} - 1)}{\overline{C_{Pcc}}(1 + \beta)\left\{\theta(1 - \eta_R + \eta_R\eta_T - \frac{\eta_R\eta_T}{(\epsilon\pi_C)^{m_T}}) - (1 - \eta_R)(1 + \frac{\pi_C^{m_C} - 1}{\eta_C})\right\}}$$
$$= \frac{\overline{C_{PT}}\eta_T\theta[1 - (\epsilon\pi_C)^{-m_T}](1 + \beta) - \frac{1}{\eta_C}\overline{C_{PC}}(\pi_C^{m_C} - 1)}{\overline{C_{Pcc}}(1 + \beta)\left\{\theta(1 - \eta_R + \vartheta) - (1 - \eta_R)(1 + \frac{\pi_C^{m_C} - 1}{\eta_C})\right\}}$$

$$\vartheta = \eta_R\eta_T[1 - (\epsilon\pi_C)^{-m_T}] \quad (11)$$

where

$\theta = \frac{T_3}{T_1}, = \frac{1-(\zeta_{RA})_{air}-\zeta_{CC}}{1+(\zeta_{RA})_{gas}}$,

$(\zeta_{RA})_{air} = \frac{P_2-P_5}{P_2}$, $(\zeta_{RA})_{gas} = \frac{P_4-P_6}{P_6}$

and $\zeta_{CC} = \frac{P_5-P_3}{P_2}$.

Exergy analysis, which is the combination of first and second laws of thermodynamics, helps to highlight the thermodynamic inefficiencies of a system. By analyzing the exergy destroyed by each component in a process, one could see where it should focus the efforts to improve system efficiency. It can also be used to compare components, or systems to help make informed design decisions.

Exergy and energy have been differentiated to avoid confusion. Energy flows into and out of a system through mass flow, heat transfer, and work (e.g., shafts, piston rods). Energy is conserved, not destroyed. Exergy is an entirely different concept. It represents quantitatively the useful energy, or the ability to do work of the great variety of streams (mass, heat, and work) that flow through the system. The first attribute of the property exergy is that it makes it possible to compare on a common basis interactions (inputs, outputs) that are quite different in a physical sense. Another benefit is that by accounting for all the exergy streams of the system, it is possible to determine the extent to which the system destroys exergy. The destroyed exergy is proportional to the generated entropy. In systems, exergy is always destroyed, partially, or totally. The destroyed exergy, or the generated entropy is responsible for less than the theoretical efficiency of the system.

The total exergy of a stream consist of four components: physical exergy E^{PH}, chemical exergy E^{CH}, kinetic exergy E^{KN} ,and potential exergy E^{PT} [16].

$$E_{stream} = E^{PH} + E^{CH} + E^{KN} + E^{PT} \qquad (12)$$

The physical exergy of stream i associated with a thermodynamic system is given by:

$$E_i^{PH} = (u_i - u_0) + p_0(v_i - v_0) - T_0(s_i - s_0) \qquad (13)$$

where u_i , v_i, and s_i represent the internal energy, volume, and entropy of the stream i, respectively. The subscript 0 denotes the state of the same stream at ambient conditions [16].

The chemical exergy of streams depends on the change of the components concentration compared to the environment.

$$E_i^{CH} = T_0 R \sum x_j \ln \frac{x_j}{x_{j,0}} \qquad (14)$$

where, x_j is the concentration of species j in the stream i and $x_{j,0}$ denotes the concentration of given species in the environment [17].

Also, the chemical exergy of natural gas uses as fuel is corresponding to LHV of the natural gas as follows [18].

$$E_{Fuel}^{CH} = 1.04 \times LHV \qquad (15)$$

Neglecting changes in potential and kinetic energy, exergy efficiency and destruction of system components are as follows:

For compressor exergy destruction and efficiency are as follows:

$$E_{D,compressor} = E_1 - E_2 + W_{compressor} \qquad (16)$$

$$\eta'_{compressor} = \frac{E_2 - E_1}{W_{compressor}} \qquad (17)$$

Combustion chamber exergy destruction and efficiency are presented in Eqs.(18) and (19), respectively.

$$E_{D,c.ch} = E_{Fuel} + E_5 - E_3 \qquad (18)$$

$$\eta'_{c.ch} = \frac{E_3}{E_{Fuel} + E_5} \qquad (19)$$

Gas turbine exergy destruction and efficiency can be calculated as follows:

$$E_{D,Turbine} = E_3 - E_4 - W_{turbine} \qquad (20)$$

$$\eta'_{turbine} = \frac{W}{E_3 - E_4} \qquad (21)$$

For recuperator exergy destruction and efficiency is calculated as Eqs.(22) and (23), respectively.

$$E_{D,recup} = (E_4 - E_6) - (E_5 - E_2) \qquad (22)$$

$$\eta'_{recup} = 1 - \frac{E_{D,recup}}{\sum_{i,recup} E_i} \qquad (23)$$

In these equations E_i is the exergy of the stream *i*, and $\dot{E}_d$ is the system exergy destruction.

Finally, the overall system efficiency is calculated as follows:

$$\eta'_{overall} = 1 - \frac{E_{D,overall}}{E_{Fuel} + E_1 - E_6} \qquad (24)$$

where $E_{D,overall}$ is the cycle exergy destruction and is calculated as Eq.(25).

$$E_{D,overall} = \sum_{i=Compressor,c.ch,Turbine,recup} E_{D,i} \qquad (25)$$

3.Technological Potential

Gas turbine designers today can use a wide variety of design alternatives to raise both engine, and overall system efficiencies. However, microturbine designers face relatively tough system complexity and cost constraints. System improvements available to larger engines, such as, combined cycles are very expensive to include in a microturbine.

Microturbines cannot afford to use the latest generation of expensive materials capable of handling very high operating temperatures, and some advanced design techniques, such as, integral cooling of turbine blades are not practical for microturbines. Microturbine designers must minimize both prime cost and maintenance costs over the full life cycle of the machine. Further, material availability is a great concernin design.

Many microturbines have evolved from hybrid vehicle to aerospace applications. They were designed to be compact, and relatively lightweight, taking advantage of the potential power/weight superiority of a gas turbine engine. Recuperators are relatively large and heavy components, where more effectiveness means more surface area, and thus more weight and volume. A designer, thus, must compromise recuperator effectiveness, if engine weight/volume limits are to be met. However, commercial building, and industrial market opportunities for today's Iran market are stationary applications. How heavy the system is not a concern, if it is within reasonable limits. Hence, a microturbine designed specifically for stationary applications is free to incorporate a very effective recuperator.

In Iran, technology of manufacturing has been engineered with the design latitude of a conventional plate-fin heat exchanger, and the ability to vary the size to meet performance requirements. Moreover, mostly the stainless steel 347 is used as a base material for producing the recuperators in large quantities. The characteristics of stainless steel 347, which is used in recuperators are provided by Mc Donald [9] as follows:

Table 1. Characteristics of recuperator material

Title	Value
Maximum metal temperature	675°C
Nickel content	11.2%
Elative thermal expansion coefficient at 675°C	1
Approximately energy efficiency	29.5
Recuperator effectiveness	0.87

A heat exchanger thermal density of a candidate recuperator, with energy efficiency of 30%, specific power of 100 kW/kg/s,and effectiveness of 85% is about 8 MW/m3 [9].

4.Case Study

The proposed thermodynamic model is used to study the effects of pressure drops in recuperator manifolds, and the recuperator effectiveness on the efficiency of designed 200kW capacity microturbine engine, according to Iran's manufacturing and technological level. Hence, all of parameters in the equation (11), except η,R, and ε, kept constant, and equal to their practical values; these numerical values are shown in Table 2.

Table 2. Case study characteristics parameters

$\overline{C_{Pc}} = \overline{C_{Pcc}}$	$\eta_T = 0.85$
$\theta = 3.97$	$\pi_C = 4.5$
$\zeta_{CC} = 0.01$	$\eta_C = 0.80$
$\beta = 0.0093$	$m_C = 0.2857$
$m_T = 0.2537$	$(\zeta_{RA})_{gas} \cong 2(\zeta_{RA})_{air}$

5. Results

By inserting the values of Table 2 in the Eq.(11), the curve of energy efficiency *vs.*effectiveness and $(\zeta_{RA})_{air}$can is illustrated in Fig. 3. Zeta is pressure drop ratio in air side of recuperator. In the figure, three different pressure drops ratio, energy efficiency *vs.*effectiveness are illustrated.

Figure 3 shows the energy efficiency increases with increasing the effectiveness and decreased with increasing air pressure drop.

The inlet gas temperature of the recuperator in the designed microturbine is about580°C; so, the temperature of the metal does not reache to 675°C.

The minimum value of $(\zeta_{RA})_{air}$, and the maximum value of η_R in the designed microturbine are 2%, and 85%.Hence, it can be determined that the maximum attainable efficiency of the microturbine with the Eq.(11) as $\eta_{th} = 29.3\%$.

Sensitivity Analysis

Industrial gas turbines show performance characteristics that distinctly depend on ambient and operating conditions. They are influenced by site elevation, ambient temperature, and relative humidity. Proper application of gas turbines requires consideration of these factors. Effects of ambient temperature, site elevation, and relative humidity on the energy, and exergy efficiency of the cycle are illustrated in Figs.4–9.

The effect of increasing ambient temperature from 5 to 35 °C on cycle energy efficiency is depicted in Fig. 4. When ambient temperature is increased, the required compressor work will be enhanced. Increase in compressor work decreases net power generation, and hencecycle energy efficiency falls.

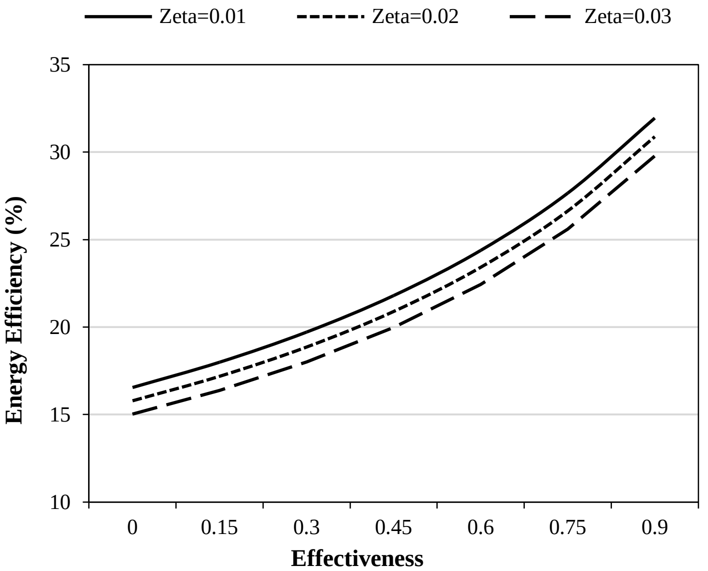

Fig.3. Energy efficiency of the cycle as a function of effectiveness

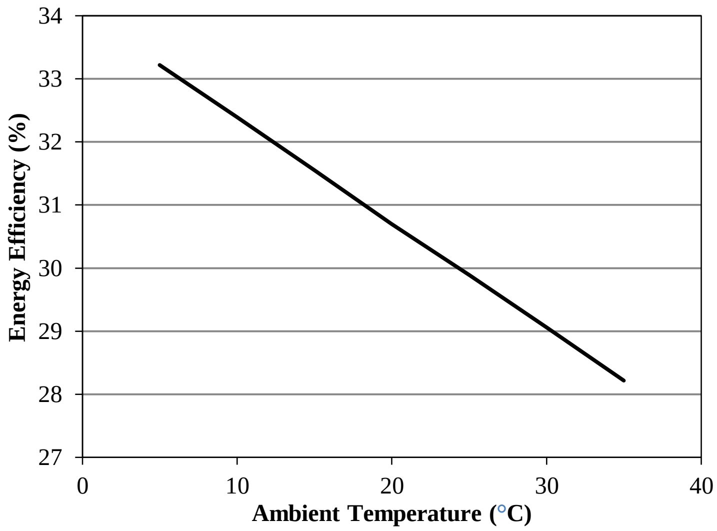

Fig.4. Energy efficiency vs.ambient temperature

The relationship between ambient temperature and cycle exergy efficiency is shown in Fig. 5. As temperature increases, cycle exergy efficiency increases. Further, it can be seen that at higher temperatures, exergy efficiency increases more smoothly. It is because of ambient condition definition in exergy equations.

Figure 6 illustrates the cycle energy efficiency *vs.*site altitude. It can be seen by varying site elevation from 0 to 1800, cycle energy efficiency decreases <1%. Hence, it can be concluded that cycle overall energy efficiency is more affected by temperature than site elevation change, because of the sensitivity of density to temperature and altitude. Density is much more sensitive to temperature than elevation change.

It can be seen in Fig.7 that the cycle overall exergy efficiency is a linear function of site altitude. As the site altitude increases, cycle overall exergy efficiency falls. Site altitude affects stream inlet pressure. Exergy efficiency depends on pressure, and hence affected by changing site altitude.

Figure 8shows the calculated cycle energy efficiency at different relative humidity. The results show that system energy efficiency is independent of relative humidity. This is expected that an increase in the ambient humidity, does not extremely affect inlet stream density as temperature, and altitude do.

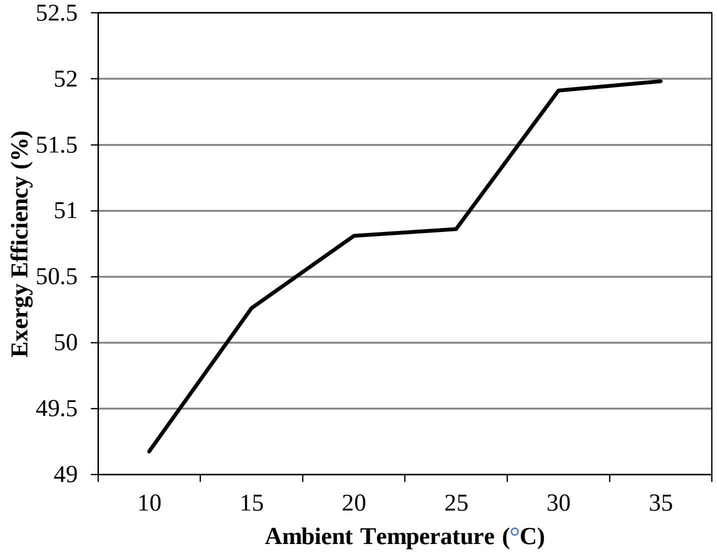

Fig.5. Exergy efficiency vs.ambient temperature

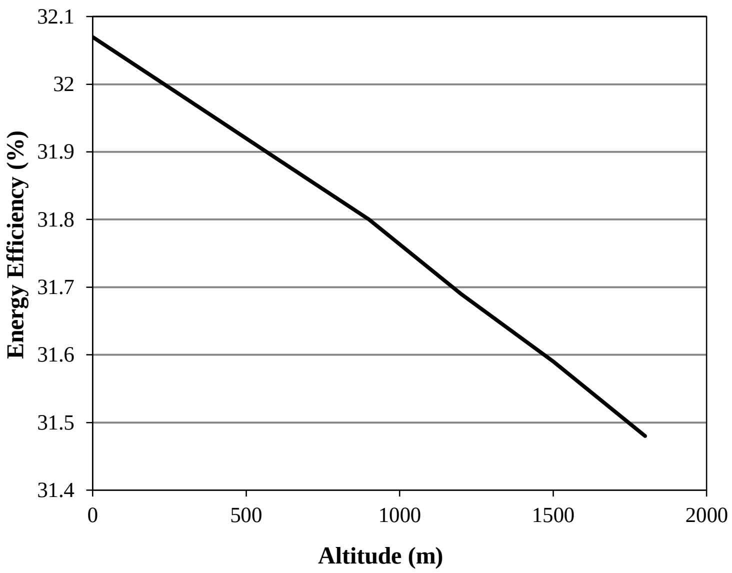

Fig.6. Energy efficiency as a function of altitude

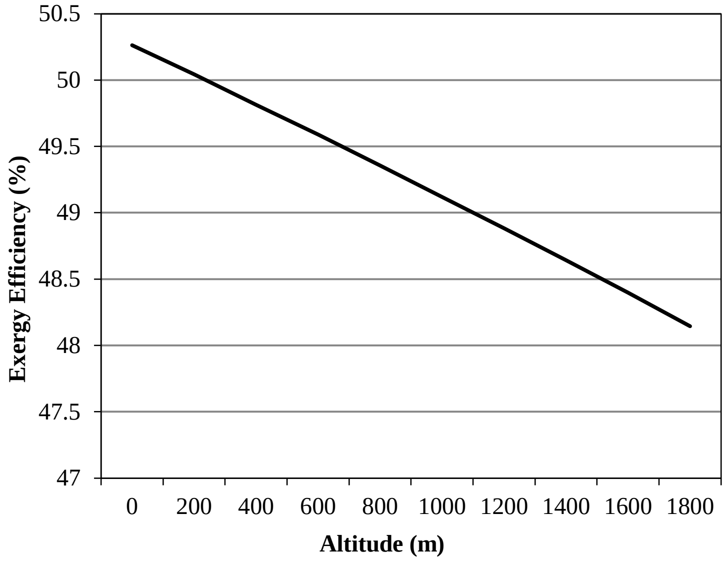

Fig.7. Exergy efficiency as a function of altitude

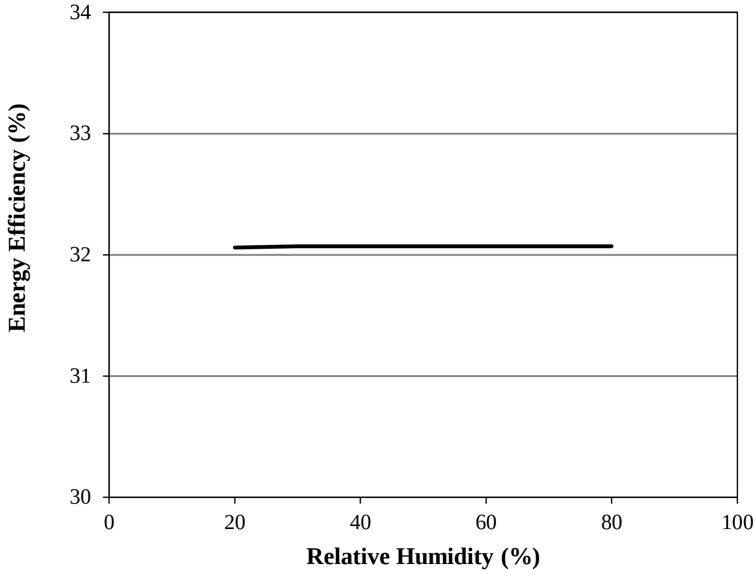

Fig.8. Energy efficiency vs.ambient relative humidity

Unlike energy efficiency, exergy efficiency changes with varying ambient humidity slightly. This is shown in Fig. 9. It can be seen exergy efficiency changes slightly with changing in ambient humidity. By varying ambient relative humidity from 20 to 80%, exergy efficiency increases only 1%.The independent behavior of exergy efficiency at higher ambient relative humidity of 70% is because of the ambient condition definition in exergy equations.

6.Conclusion

In this study, a thermodynamic model has been performed to evaluate energy efficiency as a function of effectiveness, and air pressure drop in MGT. The results show that the energy efficiency increases with increasing the effectiveness and decreased with increasing air pressure drop. Further, by using proposed model, the influence of ambient operating conditions (temperature, altitude, and relative humidity) on energy and exergy efficiency is studied. It can be concluded that ambient temperature has an extreme effect on efficiencies. Efficiencies are independent of ambient relative humidity.

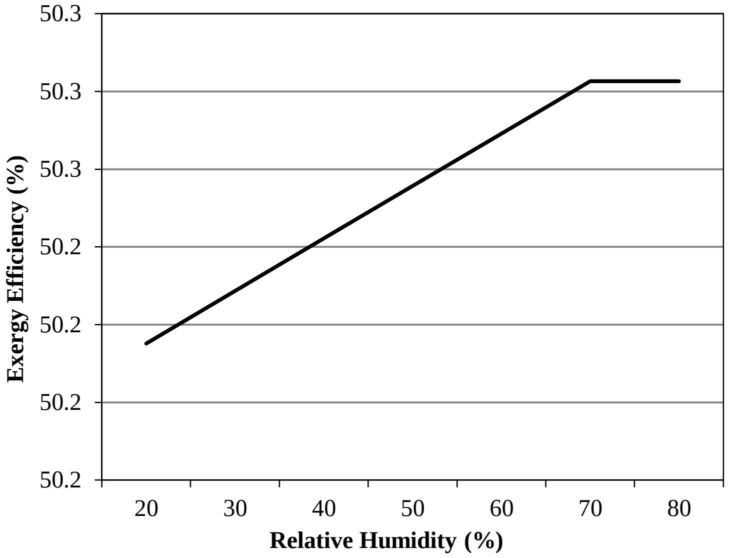

Fig.9. Exergy efficiency vs.ambient relative humidity

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