Wasit Journal for Science & Medicine **2008 1 (2) : (116 - 132)**

Exergy Analysis of a Single-Shaft Gas Turbine Power Plant with Steam Injection.

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تحليل المتاحية لمحطة قدرة غازية أحادية المحور عند حقن البخار

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الخلاصة:

غرفة عند مدخل تحليل المتاحية اجري على محرك توربيني غازي احادي المحور مع حقن البخار
الضاغط يتضمن البحث دراسة تأثير الزيادة في قيم درجة حرارة الدخول الى الاحتراق.
على الخسارة T_{O3} ، ودرجة حرارة الدخول الى التوربين PR_{cd} الضاغط نسبة انضغاط ، T_{O1}
ليس له اي T_{O1}). لقيمة معينة من كتلة جريان البخار: الزيادة في STIGبالمتاحية لمكونات دورة (
تاثير على الخسارة بالمتاحية للضاغط والتوربين بينما يؤدي الى زيادة الخسارة بالمتاحية في غرفة
يقود الى زيادة الحسارة بالمتاحية لكل اجزاء محطة القدرة. الزيادة في PR_{cd} الاحتراق. الزيادة في
يؤدي الى تقليل الخسارة بالمتاحية لغرفة الاحتراق، الزيادة الخسارة بالمتاحية للتوربين بينما ليس T_{O3}
لقيم معينة له اي تأثير يذكر على الضاغط. وجد ان اكبر خسار بالمتاحية تحدث في غرفة الاحتراق.
حقن البخار يؤدي الى تقليل الخسارة بالمتاحية لغرفة الاحتراق بينما يحث T_{O3}، و PR_{cd} ، T_{O1}للـ
عكس ذلك في التوربين ، بينما ليس له أي تأثير على الضاغط.

Abstract:

Exergy analysis has been performed on a single-shaft gas turbine with steam injection upstream (STIG) of the combustion chamber. The analysis will be performed to study the effect of increasing of compressor inlet temperature T_{O1} design compressor pressure ratio PR_{cd} and turbine inlet temperature T_{O3} on the exergy destruction of STIG cycle components. For a specific value of steam mass flow rate: the increasing in T_{O1} has a worthless effect on the exergy destruction of compressor and turbine while leads to increase in the exergy destruction of the combustion chamber. The increasing in PR_{cd} leads to increase in the exergy destruction for all gas turbine power plant components. The increasing in T_{O3} leads to decrease in the exergy destruction of the

combustion chamber, increase in the exergy destruction of the turbine, while has a worthless effect on the compressor. The biggest exergy destruction occurs in the combustion chamber

For a specific values of T_{O1}, PR_{cd}, and T_{O3} steam injection will lead to improve the exergy destruction of the the combustion chamber, while has a reverse effect on the turbine, and has no effect on the compressor.

Nomenclature

Symbols	Definition
	Ratio of turbine mass flow rate to that of compressor
AF	Air to fuel ratio.
Cp	Specific heat capacity at constant pressure [*kJ/kg K]*.
C_x	Axial velocity. [m/s].
DPs	Steam pressure losses in HRSG [kPa].
DTA	HRSG temperature approach [K]
HRSG	Heat Recovery Steam Generator
h	Enthalpy [kJ/kg]
$\dot{m}$	Mass flow rate [kg/s].
M	Mach number
P	Pressure [kPa].
PR	Pressure ratio
R	Gas constant [*kJ/ kg K*]
T	Temperature [K]
U	Rotational speed at off-design conditions [rpm].
W	Specific work [kJ/kg].
ΔP	Pressure loses
ΔT	e difference [K].
ϕ	Flow coefficient
φ	Stage loading factor
γ	Heat capacity ratio
η	Efficiency
ρ	Density [kg/m^3]

Subscript	Definition
a	air
ac	Compressor air mass flow rate at off-design conditions
acd	Compressor air mass flow rate at design conditions
af	Mixture of air and fuel
at	Turbine air mass flow rate at off-design conditions
atd	nass flow rate at itions
as	Mixture of air and steam
asf	Mixture of air, steam, and fuel
c	Compressor
cc	Combustion chamber
cd	Compressor design
d	Design
e	Turbine exit
ec	Economizer
f	Fuel
fg	Mixture of liquid and steam
p	Inlet gases to economizer
pp	Pinch point
$ppmin$	Minimum pinch point
s	Steam
t	Turbine
td	Turbine at design
1	Inlet to compressor stage at off-conditions.
$1d$	Inlet to compressor stage at design conditions
O	Dead state conditions
$O1$	Inlet to compressor
$O2$	Outlet from compressor
$O3$	Inlet to turbine
$O5$	Exit of gases from economizer
PC	Polytropic compressor
PT	Polytropic turbine
W	Water

1 Introduction

Exergy is defined as the maximum(or minimum) theoretical work obtainable as the system interacts with its surroundings and comes to equilibrium. Once a system is in equilibrium with its surroundings, its is not possible to use the energy within the system to produce work. At this point, the exergy of the system has been completely destroyed. The state in which the system is in equilibrium with its surroundings known as the dead state. In order to quantify the exergy of a system, we must specify both the system and the surroundings. The *exergy reference environment* is used to standardize the quantification of exergy. The exergy reference environment or simply the environment is assumed to be large, simple compressible system. The temperature and pressure of the reference environment are assumed to be uniform at T_O and P_O respectively.

Exergy is generally not conserved as energy but destroyed in the system. Exergy destruction is the measure of irreversibility that is the source of performance loss and it can be measured by quantifying the entropy-generation of the components . Therefore, an exergy analysis assessing the magnitude of exergy destruction, identifies the location, the magnitude and the source of thermodynamics inefficiencies in a thermal system. Exergy analysis is based on the first law of thermodynamics (FLT), and Second law of thermodynamics (SLT), and the analysis is not complete if not includes SLT analysis [1, 2, 3, 4, 5].

Quantitative exergy balance for each component of gas turbine power plant and for whole system was considered by [6, 7, 8, 9]. They verified that the biggest exergy destruction occurs through combustion chamber.

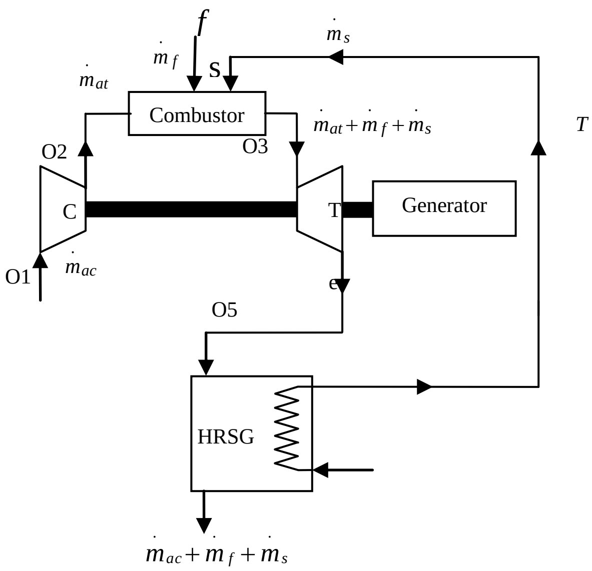
As a test case, FLT analysis of steam injection gas turbine (STIG) is performed on a typical gas turbine engine with steam injection upstream of the combustion chamber [10]. Tornado which is manufactured by Ruston company is used as a basis for this purpose [11, 12].

In our study, *exergy analysis* will be performed on the same gas turbine engine. Our objective is to predict the amount of exergy destruction by quantifying the entropy generation for each component in STIG cycle. This analysis will be performed for different values of T_{O1}, PR_{cd} and T_{O3} with steam injection.

2 Steam injection gas turbine plant (STIG):

STIG cycle analysis has been investigated by several authors [10, 13, 14, 15, 16, 17]. In STIG cycle the superheated steam generated by *heat recovery steam generator* (HRSG) will be injected directly at upstream of the combustion chamber, see Fig.(1). Therefore, gas turbine engine will operate at off-design conditions. Because the increase in the turbine mass flow rate will lead to increase the pressure ratio across both the turbine and compressor[10]. For the analysis of gas turbine cycle the following assumption are made:

1- Effect of bleed flow rate on the turbine performance is neglected.
2- The following data is considered to be constant with the steam injection at design and off-design conditions: PR_{cd} =12 (or 16,20) , $\varphi_d = 0.4$, $\Delta P_{CC} = 3\%$, $\Delta P_t = 4\ kPa$, T_{O3} = 1273 K (or 1400 K, 1500 K) [10] according to the tested case , $AF = 50$, $A = \frac{\dot{m}_{atd}}{\dot{m}_{acd}} = \frac{\dot{m}_{at}}{\dot{m}_{ac}} = 0.95$, η_{PC} = 0.89, η_{PT} =88 [12, 18].



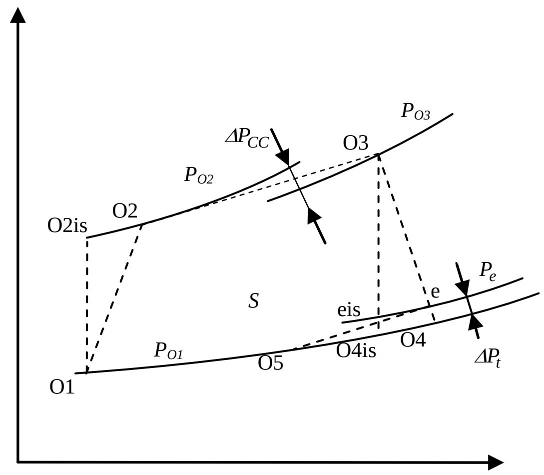


Fig.(1) Single-shaft engine with HRSG.

Fig.(2) T-S diagram of a single shaft engine.

3 Exergy Analysis STIG Cycle:

Generally the exergy balance for steady flow processes equal to [1] :

$$\dot{E}_w = \sum_{i=1}^{n} (\dot{E}_Q)_i + \sum_{in} \dot{m}\, e_x - \sum_{out} \dot{m}\, e_x - T_o\, \dot{S}_{gen} \qquad (1)$$

Where $\dot{E}_W$ represents the available work (exergy), $\sum_{i=1}^{n}(\dot{E}_Q)_i = \sum_{i=1}^{n}\left(1-\frac{T_o}{T_i}\right)\dot{Q}_i$ exergy via heat transfer, $\sum_{in}\dot{m}\,e_x, \sum_{out}\dot{m}\,e_x$ intake and release of flow exergy via mass flow rate, and $T_o\,\dot{S}_{gen}$ lost of available work due to exergy destruction. The specific exergy e_x are expressible in terms of four components: physical exergy e_{xPH}, kinetic exergy e_{xKN}, potential exergy e_{xPT}, and chemical exergy e_{xCHE} [12]:

$$e_x = e_{xPH} + e_{xKN} + e_{xPT} + e_{xCHE}$$

e_{xKN} and e_{xPT} are not considered because the analysis is performed on a power plant which has no change its potential and kinetic exergy. e_{xPH} represent specific exergy of the substance (air, fuel, and product of combustion) as a function of pressure and temperature. e_{xCHE} represent the chemical exergy of the fuel.

Air is assumed to be ideal gas, therefore specific exergy is calculated from [1]:

$$e_x = Cp\,T_O\left(\frac{T}{T_O}-1-\ln\frac{T}{T_O}\right)+R\,T_O\ln\frac{P}{P_O} \qquad (2)$$

Where T is in K.

Second law efficiency η_{II} is equal to

$$\eta_{II} = \frac{Exergy\,\text{recovered}}{Exergy\,\text{supplied}} = \frac{Exergy\,\text{supplied - Exergy destruction}}{Exergy\,\text{supplied}}$$

$$\eta_{II} = 1-\frac{\text{Exergy destruction}}{Exergy\,\text{supplied}} \qquad (3)$$

According to Gouy-Stodola theorem [1]

$$Exergy\,destruction = \dot{W}_{lost} = T_O\,\dot{S}_{gen}$$

(4)

Therefore, second law efficiency becomes

$$\eta_{II} = 1-\frac{T_O\,\dot{S}_{gen}}{Exergy\,\text{supplied}} \qquad (5)$$

4 Gas turbine components exergy balance:

We should refer to Fig.(1) for specifying input and output points for each component.

Compressor exergy balance: Applying Eq.(1), with assuming that the compression process is adiabatic (i.e, $\dot{E}_Q = 0$) and $\dot{E}_w = -\dot{W}_c$, the minus sign indicates that the exergy is consumed through compressor.

$$-\dot{W}_c = \dot{m}_{ac}\, e_{x1} - \dot{m}_{ac}\, e_{x2} - T_O\, \dot{S}_{gen}$$

Divided by $\dot{m}_{acd}$ and solve for compressor exergy destruction χ_C

$$\chi_C = T_O\, s_{gen} = W_c - \frac{\dot{m}_{ac}}{\dot{m}_{acd}}(e_{x2} - e_{x1}) \qquad (6)$$

Where e_{x1}, e_{x2} inlet and outlet exergy of the compressor respectively. They are calculated by using Eq.(2). Compression process is assumed to be a polytropic process, therefore, with reference to Fig.(2) T_{O2} and W_C are calculated from [18]:

$$T_{O2} = T_{O1}\,(PR_c)^{\frac{R_a}{\eta_{PC}\,(Cp_a)_{12}}} \qquad (7)$$

Compression process is assumed to adiabatic process

$$W_C = (\dot{m}_{ac}/\dot{m}_{acd})\,(Cp_a)_{12}\,(T_{O2} - T_{O1}) \qquad (8)$$

Where $(Cp_a)_{12}$ is calculated at average temperature from Eq.(15).

Inlet conditions to the compressor are according to ISO conditions are $T_{O1} = 15^o C, P_{O1} = 101.3\, kPa$ [6]. For *reference environment* the values of $T_O = 25^o C,\ P_O = 101.3\, kPa$ [1].

Applying Eq.(5) for compressor second law efficiency η_{IIC}

$$\eta_{IIc} = 1 - \frac{\chi_C}{W_c} \qquad (9)$$

Combustion chamber exergy balance: Applying Eq.(1) with adiabatic Combustion process. Therefore, the exergy balance equation for combustion chamber becomes:

$$A\frac{\dot{m}_{ac}}{\dot{m}_{acd}}e_{x2} + \frac{\dot{m}_f}{\dot{m}_{acd}}(e_{xf} + e_{xCHE}) + \frac{\dot{m}_s}{\dot{m}_{acd}}e_{xs} - (A\frac{\dot{m}_{ac}}{\dot{m}_{acd}} + \frac{\dot{m}_f}{\dot{m}_{acd}} + \frac{\dot{m}_s}{\dot{m}_{acd}})\,e_{x3} - T_O\, s_{gen} = 0$$

Where e_{xf}, e_{xs}, e_{x3} are fuel specific exergy, steam specific exergy, and outlet specific exergy of the combustion chamber, they are calculated by using Eq.(2). Solve for combustion chamber exergy destruction χ_{CC} (where $\chi_{CC} = T_O s_{gen}$)

$$\chi_{CC} = \left(A\frac{\dot{m}_{ac}}{\dot{m}_{acd}} e_{x2} + \frac{1}{AF}(e_{xf} + e_{xCHE}) + \frac{\dot{m}_{s}}{\dot{m}_{acd}} e_{xs} \right) - \left(A\frac{\dot{m}_{ac}}{\dot{m}_{acd}} + \frac{1}{AF} + \frac{\dot{m}_{s}}{\dot{m}_{acd}} \right) e_{x3} \tag{10}$$

Power plant normally used natural gas (methane) as the fuel. Chemical exergy for methane is equal to $e_{xCHE} = 51748.43\ kJ/kg$ [1]. Fuel injection conditions are P_f =30 bar T_f = 25^o C and is constant at design and off-design process [6].

Combustion chamber second law efficiency η_{IICC} can be calculated from Eq.(5)

$$\eta_{IICC} = 1 - \frac{\chi_{CC}}{A\frac{\dot{m}_{ac}}{\dot{m}_{acd}} e_{x2} + \frac{1}{AF} e_{xf} + \frac{\dot{m}_{s}}{\dot{m}_{acd}} e_{xs}} \tag{11}$$

*Exergy balance for turbine***:** Applying Eq.(1)

$$W_t = \left(\frac{1}{AF} + A\frac{\dot{m}_{ac}}{\dot{m}_{acd}} + \frac{\dot{m}_{s}}{\dot{m}_{acd}} \right) e_{x3} - \left(\frac{1}{AF} + A\frac{\dot{m}_{ac}}{\dot{m}_{acd}} + \frac{\dot{m}_{s}}{\dot{m}_{acd}} \right) e_{xe} - T_O\, s_{gen}$$

Solve for turbine exergy destruction χ_t

$$\chi_t = T_O\, s_{gen} = \left(\frac{1}{AF} + A\frac{\dot{m}_{ac}}{\dot{m}_{acd}} + \frac{\dot{m}_{s}}{\dot{m}_{acd}} \right) (e_{x3} - e_{xe}) - W_t \tag{12}$$

Where e_{xe} outlet exergy from turbine, is calculated by using Eq.(2).

Expansion process is assumed to be a polytropic process, therefore, with reference to Fig.(2), T_e and W_t are calculated from [18]:

$$T_e = T_{O3}\,(1/PR_t)^{\frac{\eta_{PT} R_{asf}}{(Cp_{asf})_{3e}}} \tag{13}$$

Expansion process is assumed to adiabatic process

$$W_t = [(1/AF) + (\dot{m}_s/\dot{m}_{acd}) + A(\dot{m}_{ac}/\dot{m}_{acd})](Cp_{asf})_{3e}\,(T_{O3} - T_e) \tag{14}$$

Where $(Cp_{asf})_{3e}$ is calculated at average temperature for air, steam, and fuel.

Air, steam, and products of combustion (assuming complete combustion) are assumed to behave as an ideal gas. Heat capacity for either air and superheated steam will be calculated at average temperature from [19]

For air $$Cp_a = 1.003 + 1.816\times10^{-4} T \tag{15}$$

For steam $$Cp_s = 4.6 - 103\times T^{-0.5} + \frac{967.2}{T} \tag{16}$$

Where T in K.
Turbine second law efficiency η_{IIt}, applying Eq.(5);

$$\eta_{IIt} = 1 - \frac{\chi_t}{\left((1/AF) + (\dot{m}_s/\dot{m}_{acd}) + A\,(\dot{m}_{ac}/\dot{m}_{acd})\right)\left(e_{x3} - e_{xe}\right)} \quad (17)$$

Total exergy destruction $\chi_{TOT} = \chi_C + \chi_{CC} + \chi_t$.
Total cycle second law efficiency:

$$\eta_{IITOT} = 1 - \frac{\chi_{TOT}}{(1/AF)\,e_{xCHE} - \left((1/AF) + (\dot{m}_s/\dot{m}_{acd}) + A\,(\dot{m}_{ac}/\dot{m}_{acd})\right)e_{xe}} \quad (18)$$

Off-design performance:
Due to steam injection, turbine will operate at off-design conditions. Therefore, turbine will be assumed to be chocked at design and off-design conditions in order to calculate pressure ratio of turbine and compressor. Therefore, for chocked turbine the term ($\dot{m}_t \sqrt{R} / P_{O3}\sqrt{(\gamma)_{3e}}$) will remain constant, i.e. PR_t equal to [10]:.

$$PR_t = \frac{\left[(1/AF) + (\dot{m}_s/\dot{m}_{acd}) + A(\dot{m}_{ac}/\dot{m}_{acd})\right]\sqrt{R_{asf}}}{K1\sqrt{(\gamma_{asf})_{3e}}} \quad (19)$$

Where $$K1 = \frac{(1/AF + A)\sqrt{R_{af}}}{PR_{td}\sqrt{(\gamma_{af})_{3ed}}} = \text{constant} \quad (20)$$

Compressor (or turbine) pressure ratio is calculated at design and off-design conditions from [18]

$$PR_c = \frac{(1 + \Delta P_t / P_{O1})}{(1 - \Delta P_{CC}/100)}\,PR_t \quad (21)$$

Off- design compressor mass flow rate:
Evaluation the variation in compressor mass flow rate at off-design conditions will start with [10]:

$$\dot{m}_{ac} = \rho_1\, C_{X1}\, A_1 \quad (22)$$

Where A_1 inlet stage area of the compressor [m^2].

From perfect gas law $$\rho_1 = \frac{P_1}{R_a\, T_1} \quad (23)$$

For compressible flow

$$\frac{T_{O1}}{T_1} = (1 + \frac{\gamma - 1}{2} M_1^2) \qquad (24)$$

$$\frac{P_{O1}}{P_1} = (1 + \frac{\gamma - 1}{2} M_1^2)^{\frac{\gamma}{\gamma - 1}} \qquad (25)$$

Substitute for P_1 and T_1 from Eqs.(24) and (25) into Eq.(22) and use Eq.(22) to calculate mass flow rate at design and off-design conditions. Assume that A_1, R_a, P_{O1}, T_{O1}, U, M_1, are constant at design and off-design conditions. Therefore,

$$\frac{\dot{m}_{ac}}{\dot{m}_{acd}} = \frac{\phi}{\phi_d} \qquad (26)$$

Where, $\phi = C_{x1} / U$, $\phi_d = C_{x1d} / U_d$. Where U_d represent compressor rotational speed at design conditions.

To evaluate the variation ϕ / ϕ_d, the compressor stage loading factor φ is used [21]

$$\varphi = \frac{\Delta W_C}{\mathrm{U}^2} = 1 - \phi (\tan\alpha_1 + \tan\beta_2) \qquad (27)$$

Where, α_1, β_2 are the flow outlet angles from compressor stator and rotor repectively. Cascade data suggest that $\alpha_1 + \beta_2$ are constant at design and off-design conditions, hence

$$\frac{\phi}{\phi_d} = \frac{\varphi / \varphi_d - 1/\varphi_d}{1 - 1/\varphi_d} \qquad (28)$$

Assume that the compressor has very large number of small stages with equal ΔW_C (now any turbomachine may be regard as being composed of very large number of small stages irrespective of the actual number of stages in the machine. If each small stage has the same efficiency of the whole machine , this efficiency can be used to compare between different machines have different pressure ratio [18, 22]). From definition of stage loading factor Eq.(27), and substitute for W_C from Eq.(8) , compressor discharge temperature from Eq.(7) , and for design and off-design conditions with noting $U = U_d$ .Therefore,

$$\frac{\varphi}{\varphi_d} = \frac{PR_c^{(R_a/\eta_{PC}(Cp_a)_{12})} - 1}{PR_{cd}^{(R_a/\eta_{PC}(Cp_a)_{12})} - 1} \qquad (29)$$

5 HRSG Energy balance

Fig.(3) shows a schematic diagram of HRSG components. HRSG is fitted to gas turbine engine exhaust. Typical temperature profile of HRSG is shown in Fig.(4) . HRSG will be considered as a cross-flow exchanger. The main effective parameters of HRSG are $\Delta T_{pp\,min}$, DTA [14, 15, 16]

Energy balance for boiler and superheater:

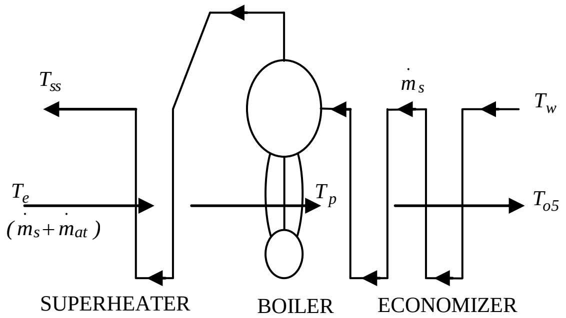
If DTA known then

$$T_s = T_e - \mathrm{DTA} \tag{30}$$

$$T_p = T_e - \frac{(\dot m_s/\dot m_{acd})\left[Cp_w\,\Delta T_{ec} + h_{fg} + (Cp_s)_{ssfg}\,(T_s - T_{fg})\right]}{\left[(1/AF) + (\dot m_s/\dot m_{acd}) + A(\dot m_{ac}/\dot m_{acd})\right](Cp_{asf})_{ep}} \tag{31}$$

If ΔT_{pp} is known then

$$T_p = T_{fg} + \Delta T_{PP} \tag{32}$$



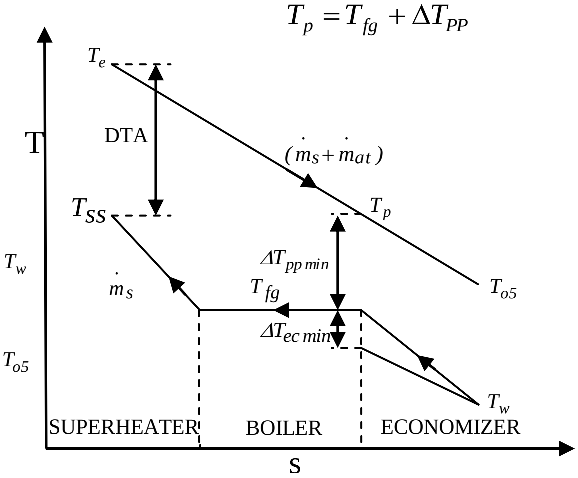


Fig.(3) Schematic diagram of HRSG components. Fig.(4) HRSG temperature profile.

Then

$$T_s = T_{fg} + \frac{1}{(Cp_s)_{sfg}}\left[\left(1 + \frac{(1/AF) + A(\dot m_{ac}/\dot m_{acd})}{\dot m_s/\dot m_{acd}}\right)\times(Cp_{as})_{ep}\,(T_e - T_p) - Cp_w\,\Delta T_{ec} - h_{fg}\right] \tag{33}$$

Energy balance for economizer gives

$$T_{O5} = T_p - \frac{(\dot{m}_s / \dot{m}_{acd})\, Cp_w\,(T_{fg} - \Delta T_{ec} - T_w)}{\left[(1/AF) + (\dot{m}_s / \dot{m}_{acd}) + A(\dot{m}_{ac} / \dot{m}_{acd})\right](Cp_{as})_{p5}} \qquad (34)$$

T_{fg} and h_{fg} are calculated at P_s from steam table by fitting a polynomial for the required pressure and temperature [21]. $P_s = P_{O2}$ + DPs. The numerical values of $\Delta T_{PP\,min} = 15^o C$, $\Delta T_{ec\,min} = 30^o C$, $DTA_{min} = 30^o C$, $\Delta T_{ec\,min} = 30^o C$, $T_w = 27^o C$, $T_{stack} = 121^o C$, DPs=100 kPa [10]. $Cp_w = 4.2\, kJ/kg\,K$, $R_s = 0.461\, kJ/kg\,K$ [22].

6 STIG Cycle Calculation Procedure

The basic concept of the computational procedure is that the unknown variable is assumed first, and through a series of calculation a new value of the unknown variable is calculated. After that the new value is compared with assumed value and the procedure is repeated until the convergence is obtained.

For each steam mass flow rate, $(\dot{m}_{ac} / \dot{m}_{acd})$ first and then PR_t, through a series of calculations is performed until the calculation procedure is converged.

7 Results and discussion:

Compressor: we can see from Figs.(5)-(10) that the changing of the parameters T_{O1}, PR_{cd}, T_{O3}, and also steam injection have a worthless effect on the compressor exergy destruction and exergy efficiency. We should refer to Fig.(7) and (8), the increasing in PR_{cd} leads to increasing in exergy destruction Fig.(7) at the same time increasing in exergy efficiency Fig.(8). Increase in PR_{cd} means increase in work but the relation is not linear, because the pressure line in T-S diagram diverge. That means the increasing in compressor work greater than increasing in exergy destruction, therefore exergy efficiency will increase with increasing PR_{cd}. The same effect can be shown in turbine Figs.(19) and (20).

Combustion chamber: For a specific value of $(\dot{m}_s / \dot{m}_{acd})$ (or for $\dot{m}_s / \dot{m}_{acd}$=0), the increase in T_{O1}, PR_{cd} will lead to increase in exergy destruction. That is due to the increase in T_{O2} (Eq.(7)) which will cause to increase in input exergy to the combustion chamber and that means less exergy extracted from the fuel and more exergy will be destructed. Therefore, exergy efficiency will decrease Figs.(11) - (14). Increase in T_{O3} leads to more exergy extracted from fuel combustion and that will reduce exergy destruction, Figs.(15) and (16).

For a specific values of T_{O1}, PR_{cd}, T_{O3}, the increasing in the steam mass flow rate will lead to decrease in exergy destruction Figs.(11), (13), and (15). More steam injection into combustion means more exergy extracted from fuel combustion. Therefore, exergy efficiency will increase, Figs.(12), (14), and (16).

Turbine: For a specific value of $(\dot{m}_s / \dot{m}_{acd})$ (or for $\dot{m}_s / \dot{m}_{acd}$=0), T_{O1} has a worthless effect on exergy destruction and exergy efficiency, see Figs.(17) and (18). While, the increase in PR_{cd}, T_{O3} will lead to increase in exergy destruction see Figs.(19) and (21). Exergy efficiency will decrease with the increasing T_{O3} Fig.(22) but with increasing PR_{cd}, exergy efficiency will increase also Fig.(20), that is for same reason mentioned in the compressor discussion.

The most attractive results is that for a specific value of T_{O1}, PR_{cd}, T_{O3} the increase in steam mass flow rate will lead to increase in exergy destruction and decrease in exergy efficiency.

Steam injection will lead to increase the turbine inlet pressure as mentioned in. At the same time temperature difference through turbine approximately remains constant [10]. Therefore, input exergy to the turbine will increase due to increase in the inlet pressure according to Eq.(2), and exergy destruction will increase [see Eq.(12)].

Whole cycle: Figs.(23) and (24) shows a comparison between gas turbine components, we see that the best improvement occurs in the combustion chamber because the injection of the steam will extract more exergy from the fuel combustion, therefore combustion exergy efficiency will increase. Compressor exergy destruction and exergy efficiency still constant with steam mass flow rate. While for turbine there is a little percent increase in exergy destruction and decrease in exergy efficiency.

8 Conclusions

1- There is a worthless effect of changing of T_{O1}, PR_{cd}, T_{O3} and steam injection on compressor exergy destruction and exergy efficiency.
2- The main improvement of steam injection can be shown on the combustion chamber. Exergy destruction will decrease and exergy efficiency will increase. The improvement reaches to 25%. The biggest improvement can be shown by increasing T_{O3}.
3- The effect of steam injection will lead to increase exergy destruction of turbine.

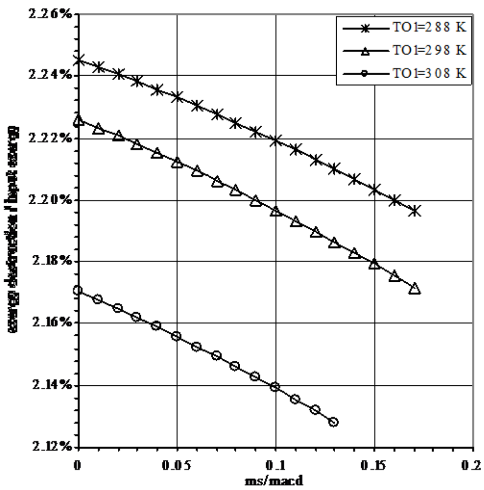


Fig.(5) Change in compressor exergy destruction due to change in compressor inlet temperature

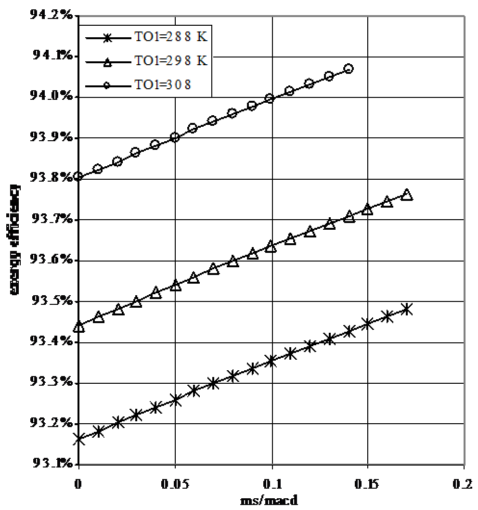


Fig.(6) Change in compressor exergy efficiency due to change in compressor inlet temperature

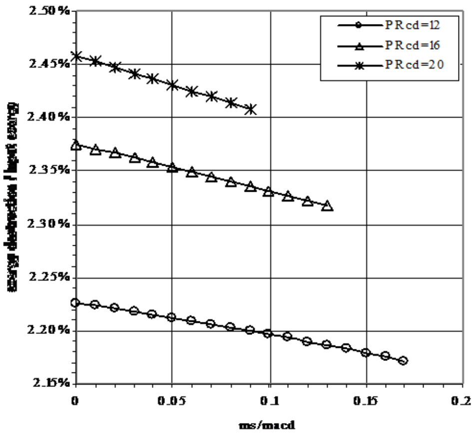


Fig.(7) Change in compressor exergy destruction due to change in design compressor pressure ratio

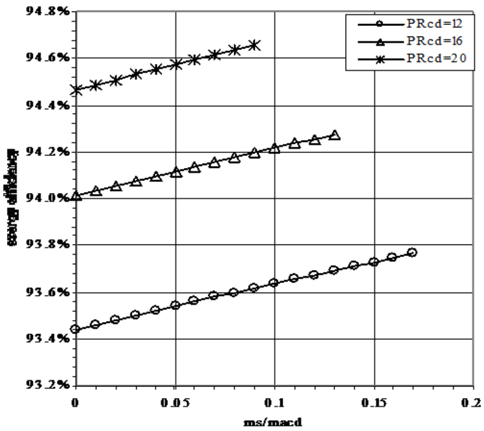


Fig.(8) Change in compressor exergy efficiency due to change in design compressor pressure ratio

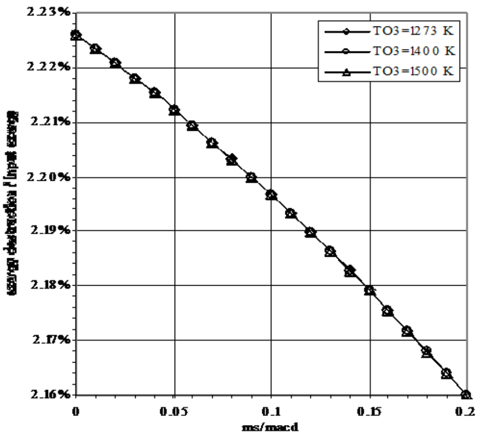


Fig. (9) Change in copmressor exergy destruction due change in turbine inlet temperature

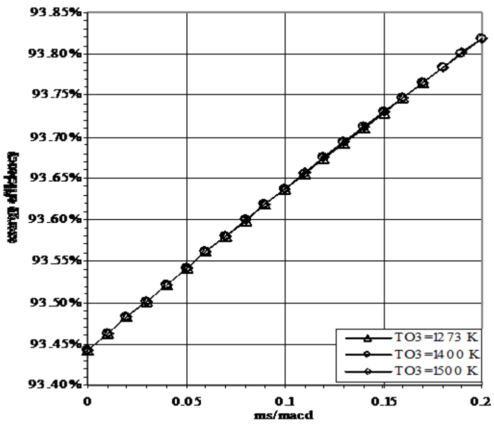


Fig.(10) Change in compressor exergy efficiency due to change in turbine inlet temperature

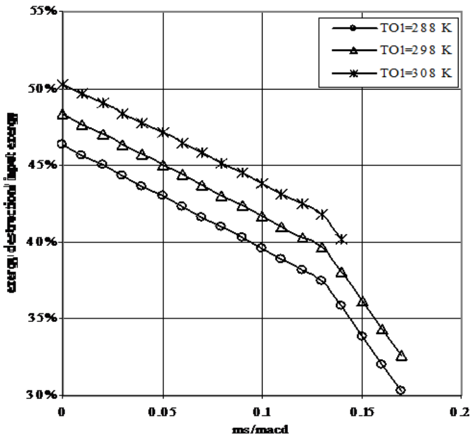


Fig.(11) Chnage in combustion chamber exergy destruction due to change in compressor inlet temperature

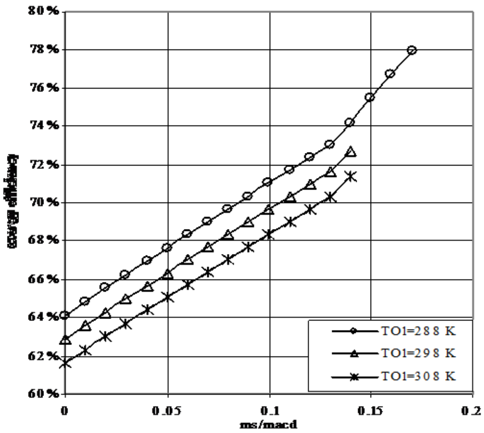


Fig.(12) Change in combustion chamber exergy efficiency due to chancge in compressor inlet temperature

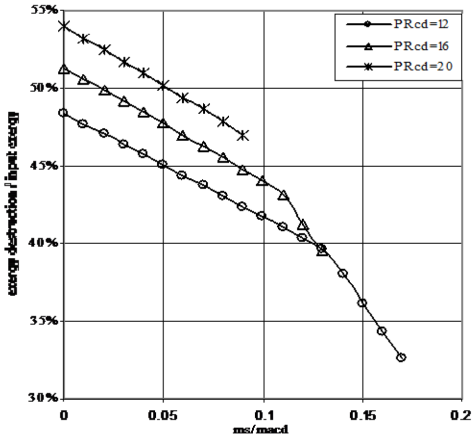


Fig.(13) Change in combustion exergy destruction due to change in compressor pressure ratio

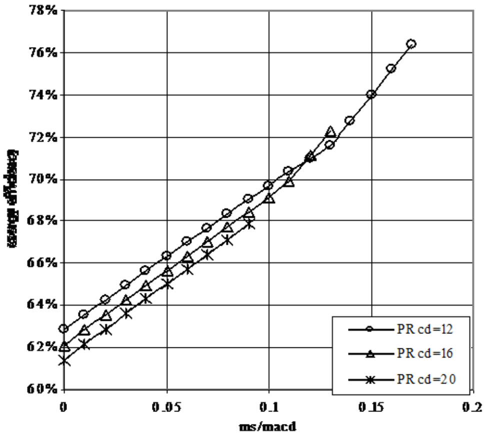


Fig.(14) Change in combustion chamber exergy efficiency due to change in compressor pressure ratio

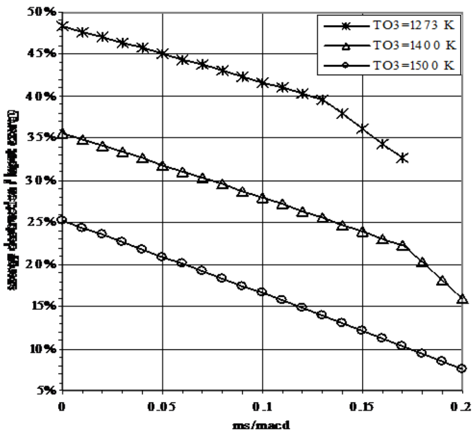


Fig.(15) Change in combustion chamber exergy destruction due to change in turbine inlet temperature

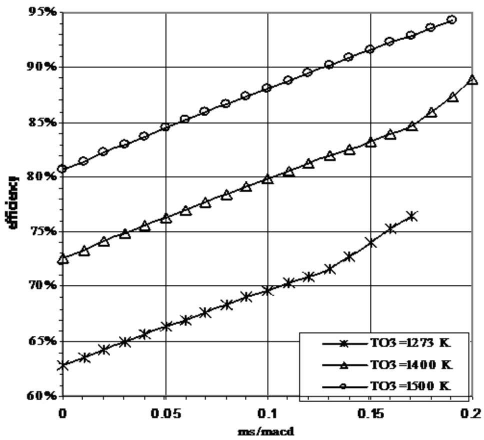


Fig.(16) Change in combustion chamber exergy efficiency due to change in turbine inlet temperature

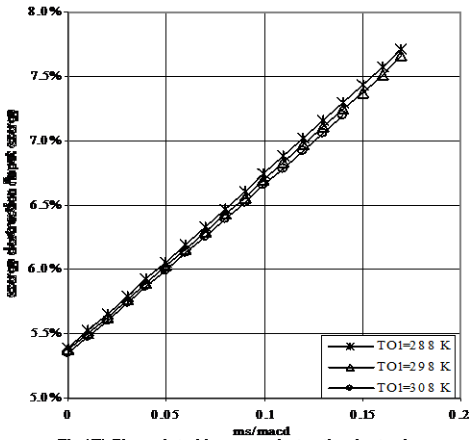


Fig.(17) Change in turbine exergy destruction due to change in compressor inlet temperature

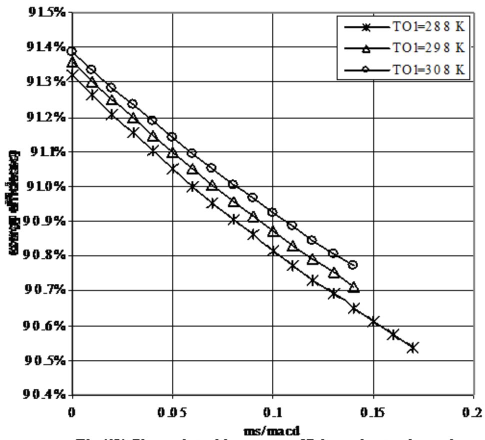


Fig.(18) Change in turbine exergy efficiency due to change in compressor inlet temperature .

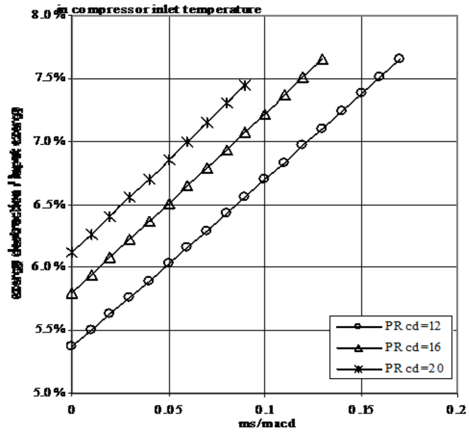


Fig.(19) Change in turbine exergy destruction due to change in compressor design pressure ratio

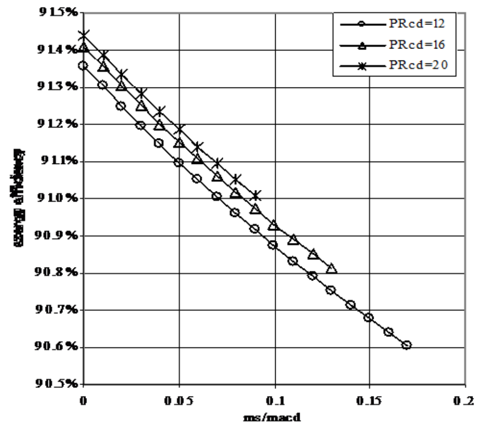


Fig.(20) Change in turbine exergy efficiency due to change in design compressor pressure ratio

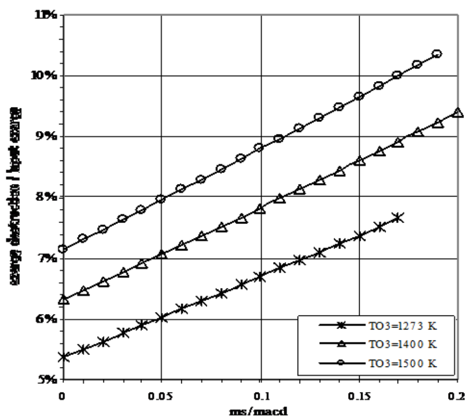


Fig.(21) Change in turbine exergy destruction due to change in turbine inlet temperature

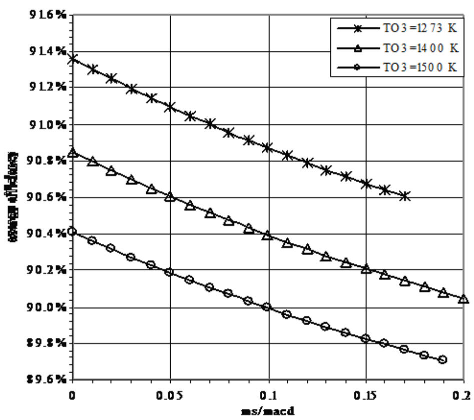


Fig.(22) Change in turbine exergy efficiency due to change in turbine inlet temperature

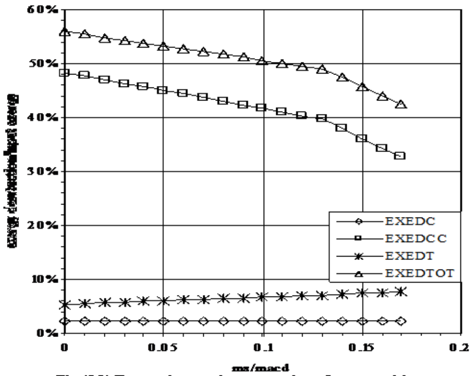


Fig.(23) Exergy destruction comparison for gas turbine components

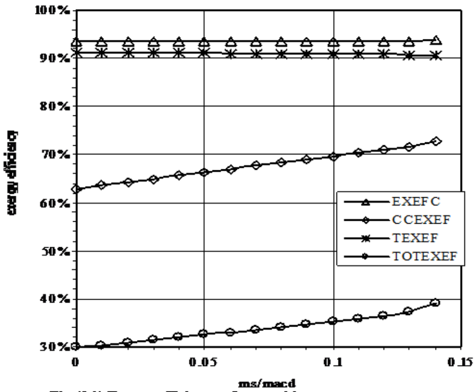


Fig.(24) Exergy efficiency of gas turbine components

Note for Figs.(23) and (24): EXEDC: compressor exergy destruction, EXEDCC: combustion chamber exergy destruction, EXEDT: turbine exergy destruction, EXEDTOT: total exergy destruction, EXEFC: compressor exergy efficiency, CCEXEF: Combustion chamber exergy efficiency, TEXEF: turbine exergy efficiency, TOTEXEF: total exergy efficiency.

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