

# Energy and Exergy Analyses of the Evaporative-Regenerative Gas-Turbine Cycle Using Excel-Thermax

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**ABSTRACT:**This paper deals with the analyses of advanced power generation cycles by using Microsoft Excel and the Thermax add-in that determines the thermo-physical properties for a wide range of substances used in thermal fluid systems. Thermax enables Excel to be used as a modelling platform for conducting parametric and optimisation analyses of power generation and refrigeration cycles, air-conditioning systems, and combustion processes. The present paper focusses on using the Excel-Thermax platform for conduction energy and exergy analyses of the evaporative regenerative gas-turbine (ERGT) cycle. The relevant Thermax functions are validated by comparing their results with those provided by Bathie and shown to be in good agreement with his data. Exergy analysis of the cycle shows that the combustion and humidification processes cause the most exergy destruction in the system and, therefore, any efforts to improve the system's performance should focus on these two processes.

**KEYWORDS:** Evaporative regenerative gas turbine; thermal-fluid analyses, exergy analysis; Excel-Thermax.

## I. INTRODUCTION

Microsoft Excel is a general-purpose spreadsheet application that comes as part of the widely-distributed Microsoft Office software. Being an extremely versatile and easy to use application, the potential of Excel as an effective educational tool for introducing the concepts of engineering sciences has been realised by many [1-6]. Unfortunately, the lack of fluid-property built-in functions in Excel undermines its use for thermo-dynamic analyses that involve significant property variations. Various educational and industrial institutions attempted in the past to overcome this deficiency by developing their own Excel add-ins for thermo-fluid property determination. In their website, the University of Virginia provides Excel-based models for thermodynamics analyses of Rankine, Brayton, Otto, and Diesel Cycles [3]. Goodwin [4] developed an Excel add-in for educational purposes, called *TPX*, which determines the thermodynamic properties of selected gases ( $H_2O$ ,  $H_2$ ,  $O_2$ ,  $N_2$ ,  $CH_4$ ) and the HFC refrigerant R-134a. Rivas et al [5] used *TPX* to demonstrate the capability of Excel to define, analyse and optimise models of thermal-fluid systems and processes. They used Excel's Solver to analyse and optimise a regenerative Rankine cycle with three closed feed-water heaters. Similar add-ins were developed by other academic and research groups, but most of these could only be used for a limited number of substances [6]. El-Awad [7] developed an Excel add-in, called *Thermax*, that provides a library of user-defined functions for determining the properties of most substances usually met in thermal-fluid analyses. Thermax enables parametric studies and design optimisation of most types of thermal-fluid systems to be performed via Excel's user-interface and iterative tools. This paper aims to demonstrate the capabilities of the Excel-Thermax platform for analysing an advanced power generation cycle, which is the evaporative-regenerative gas turbine (ERGT) cycle. Analysis of the ERGT cycle requires properties of a mixture of two fluids (air and water vapour) to be determined, which makes it more challenging to model than a conventional regenerative gas-turbine cycle. The paper presents energy and exergy analyses of the ERGT cycle and compares the results with those given by Bathie [8] to confirm the adequacy of the Excel-Thermax platform for such analyses.

## II. THE EVAPORATIVE-REGENERATIVE GAS-TURBINE CYCLE

Fig. 1 shows a schematic diagram of the ERGT system in which pressurised water is injected at the compressor discharge. Apart from increasing the mass flow through the turbine, the injected water cools the compressed air and makes regeneration more effective. Like compressor intercooling, water injection reduces the gas turbine thermal efficiency if done without regeneration. The water must be delivered at a pressure which is higher than that of the compressor. Theoretically, as much water as that needed to saturate the compressed air can be injected, but practically the injected amount of water should not adversely affect the systems' performance. The ERGT cycle is a type of advanced gas-turbine cycles commonly known as humidified air turbine (HAT) cycles [8, 9].

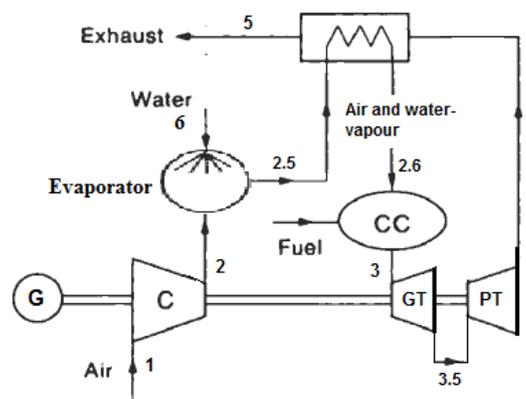


Fig. 1. Schematic diagram of the evaporative-regenerative gas turbine (adapted from Horlock [9])

In what follows, the suffixes *a* and *w* refer to pure air and water, respectively. Per unit mass flow rate of atmospheric air to the compressor, the compressor work ( $w_c$ ) is given by:

$$w_c = 1 \times (h_{2a} - h_{1a}) \quad (1)$$

Where  $h_{1a}$  and  $h_{2a}$  are enthalpy of pure air at states 1 and 2, respectively. The amount of water ( $m_w$ ) injected per unit mass flow rate of air is determined from an energy balance across the evaporator. Given the temperature and pressure of the injected water ( $P_6, T_6$ ), the temperature of air before water injection ( $T_2$ ), and the temperature of the air-water vapour mixture after injection ( $T_{2.5}$ ), energy balance gives [8]:

$$1 \times h_{2a} + m_w h_{6w} = 1 \times h_{2.5a} + m_w h_{2.5w} \quad (2)$$

Where  $h_{2a}$  and  $h_{2.5a}$  are enthalpy of pure air and  $h_{6w}$  and  $h_{2.5w}$  are enthalpy of water at these points. Rearranging (2):

$$m_w = \frac{(h_{2.5a} - h_{2a})}{(h_{6w} - h_{2.5w})} \quad (3)$$

In (3), the two enthalpy values for water ( $h_{6w}$  and  $h_{2.5w}$ ) are approximated by  $h_f$  at  $T_6$  and  $h_g$  at  $P_{2.5}$ , respectively. Given the effectiveness of the regenerator ( $\varepsilon$ ), the temperature of the humidified air after the regenerator ( $T_{2.6}$ ) is determined from:

$$T_{2.6} = T_{2.5} + \varepsilon(T_4 - T_{2.5}) \quad (4)$$

Where  $T_4$  is the temperature of the exhaust gas at state 4 that will be evaluated later. The heat-input ( $Q_{in}$ ) is obtained from:

$$Q_{in} = (1 + m_w) \times (h_3 - h_{2.6}) \quad (5)$$

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Where the enthalpy of the humidified air at a given temperature is calculated from those of pure air and water vapour according to the following relationship:

$$h = \sum f_i h_i = (1 \times h_a + m_w \times h_w) / (1 + m_w) \quad (6)$$

Where  $f_i$  is the mass fractions of the two mixture components, air and water-vapour. Since the work produced by the gas-generator turbine ( $w_{gt}$ ) equals the compressor work ( $w_c$ ), the enthalpy at point 3.5 ( $h_{3.5}$ ) is given by:

$$h_{3.5} = h_3 - w_{gt} / (1 + m_w) = h_3 - w_c / (1 + m_w) \quad (7)$$

The temperature of the humidified air exiting the gas generator turbine ( $T_{3.5}$ ) is then determined from the requirement that:

$$h_{3.5} = (1 \times h_{3.5a} + m_w \times h_{3.5w}) / (1 + m_w) \quad (8)$$

However, since the working fluid consists of two components, (8) has to be solved by iteration to find  $T_{3.5}$ . The ideal temperature at the exit from the gas generator turbine ( $T_{3.5i}$ ) is determined from the enthalpy after an ideal expansion in the generator turbine ( $h_{3.5i}$ ), which can be calculated from:

$$h_{3.5i} = h_3 - w_{gti} / (1 + m_w) \quad (9)$$

Where  $w_{gti}$  is the ideal work from the gas-generator turbine which is related to the actual work from the gas-generator turbine ( $w_{gt}$ ) by:

$$w_{gti} = w_{gt} / \eta_{gt} = w_c / \eta_{gt} \quad (10)$$

The temperature  $T_{3.5i}$  is then determined from  $h_{3.5i}$ . The pressure  $P_{3.5i}$ , that follows an ideal isentropic expansion to  $T_{3.5i}$ , can now be calculated by using the following relationship:

$$s_3 - s_{3.5i} = s_3^0 - s_{3.5i}^0 - R \ln \frac{P_3}{P_{3.5i}} = 0 \quad (11)$$

Where  $s_3^0$  is the temperature-dependent part of entropy change and  $R$  is the gas constant for humid air, which is calculated by a relationships similar to (6). In (11),  $s_3^0$  and  $s_{3.5i}^0$  are evaluated at the temperatures  $T_3$  and  $T_{3.5i}$ , respectively. Rearranging (11) to give the unknown pressure  $P_{3.5} = P_{3.5i}$ :

$$P_{3.5} = \frac{P_3}{e^{(s_3^0 - s_{3.5i}^0) / R}} \quad (12)$$

The ideal temperature after an isentropic expansion in the power turbine ( $T_{4i}$ ) is determined from the following relationship that applies for an ideal isentropic expansion between points 3.5 and 4i:

$$s_{3.5}^0 - s_{4i}^0 - R \ln \frac{P_{3.5}}{P_{4i}} = 0 \quad (13)$$

Rearranging (13):

$$s_{4i}^0 = s_{3.5}^0 - R \ln \frac{P_{3.5}}{P_{4i}} \quad (14)$$

The temperature  $T_{4i}$  can now be determined from the value obtained for  $s_{4i}^0$ , but this also requires an iterative solution. The power-turbine work can be calculated from:

$$w_{pt} = \eta_{pt} \times (1 + m_w) (h_{3.5} - h_{4i}) \quad (15)$$

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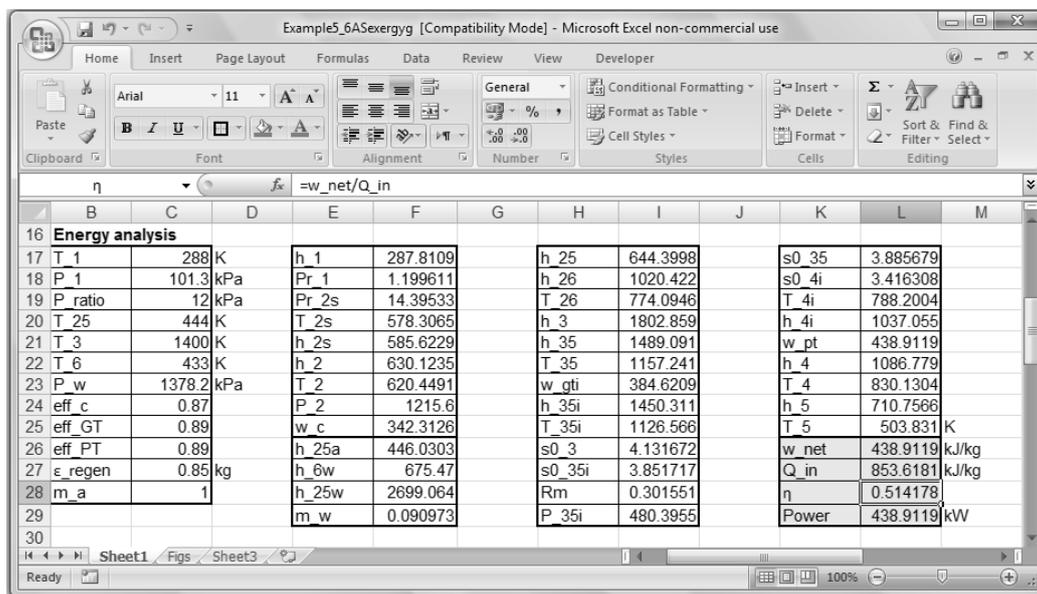
Where  $\eta_{pt}$  is the isentropic efficiency for the power turbine and  $h_{4i}$  is the enthalpy of the mixture after an isentropic expansion in the power turbine to the exhaust pressure  $P_4=P_1$ . Having determined the power-turbine's work, the actual enthalpy of the gas at the turbine's discharge ( $h_4$ ) can be calculated from:

$$h_4 = h_{3.5} - w_{pt} / (1 + m_w) \quad (16)$$

The temperature  $T_4$  is then determined from  $h_4$ .

### III. ENERGY ANALYSIS OF THE ERGT CYCLE

Bathie [8] analysed the ERGT cycle with the following data;  $T_1=288$  K,  $P_1=101.3$  kPa, Pressure ratio = 12,  $T_3=1400$  K,  $P_6=1447$  kPa,  $T_6= 433$  K (160°C),  $T_{2.5}=444$  K, compressor isentropic efficiency ( $\eta_c$ ) = 87%, gas generator and power turbine isentropic efficiency ( $\eta_{tg}$ ,  $\eta_{pg}$ )= 89%, and regenerator effectiveness ( $\epsilon$ )= 85%. Taking the compressor air intake as 1 kg/s, he calculated the thermal efficiency and net power [kW] developed by the gas turbine. The results obtained by Bathie [8] will be used for validating the Thermax functions used in the analysis. Fig. 2 shows the Excel sheet prepared for the analysis of the ERGT cycle. The figure shows the input data on the left side of the sheet and the key performance parameters,  $w_{net}$ ,  $Q_{in}$ ,  $\eta$ , and  $Power$ , on the right side of the sheet. The calculation part shows the temperature of the compressed air ( $T_2=620.45^\circ\text{C}$ ), the compressor work ( $w_c = 342.31$  kJ/kg), the pressure after the gas-generator turbine ( $P_{3.5i}=480.46$  kPa), the power-turbine work ( $w_{pt}=438.91$  kJ/kg), the amount of injected water ( $m_w=0.091$  kJ/kg), and the temperature of the exhaust gas at the discharge point ( $T_5=503.83$ K).



	B	C	D	E	F	G	H	I	J	K	L	M
16	<b>Energy analysis</b>											
17	T 1	288	K	h 1	287.8109		h 25	644.3998		s0 35	3.885679	
18	P 1	101.3	kPa	Pr 1	1.199611		h 26	1020.422		s0 4i	3.416308	
19	P_ratio	12	kPa	Pr 2s	14.39533		T 26	774.0946		T 4i	788.2004	
20	T 25	444	K	T 2s	578.3065		h 3	1802.859		h 4i	1037.055	
21	T 3	1400	K	h 2s	585.6229		h 35	1489.091		w_pt	438.9119	
22	T 6	433	K	h 2	630.1235		T 35	1157.241		h 4	1086.779	
23	P_w	1378.2	kPa	T 2	620.4491		w_gti	384.6209		T 4	830.1304	
24	eff_c	0.87		P 2	1215.6		h 35i	1450.311		h 5	710.7566	
25	eff_GT	0.89		w_c	342.3126		T 35i	1126.566		T 5	503.831	K
26	eff_PT	0.89		h 25a	446.0303		s0 3	4.131672		w_net	438.9119	kJ/kg
27	epsilon_regen	0.85	kg	h 6w	675.47		s0 35i	3.851717		Q_in	853.6181	kJ/kg
28	m_a	1		h_25w	2699.064		Rm	0.301551		eta	0.514178	
29				m_w	0.090973		P 35i	480.3955		Power	438.9119	kW
30												

Fig. 2. Excel sheet for the ERGT cycle – energy analysis

Table I lists Thermax functions used in the Excel sheet. The working fluid properties at various states were determined by using functions from three Thermax groups: (i) ideal-gas, (ii) water, and (iii) pressurised humid air. The first four of these function, which belong to the ideal-gas group, were used to determine properties of dry air before the humidifier. The following two functions that come from the water-group were used to determine enthalpy of liquid-water prior to the humidifier and enthalpy of water-vapour after the humidifier. The last three functions come from the pressurised humid-air group and were used to determine the properties of the working fluid after the humidifier. Enthalpy of humid air at a given temperature is determined from (6) by using the add-in function PhaTKHs\_h. The temperature  $T_{3.5}$  has to be determined from (8) by an iterative process by using the function Phahs\_TK. The same function is used to determine the temperature  $T_{2.6}$  from  $h_{2.6}$ , the temperature  $T_{3.5i}$  from  $h_{3.5i}$ , the temperature  $T_4$  from  $h_4$ , and the temperature

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$T_5$  form  $h_5$ . Equation (15), which also requires an iterative solution, could be solved with the help of the add-in function Phas0Hs\_TK.

TABLE I. FUNCTIONS USED IN THE ERGT CYCLE ANALYSIS

Function	Input	Output	Usage
gasTK_h	"Air", $T$	$h$	$h_1, h_{2s}, h_{2.5a}$
gasTK_Pr	"Air", $T$	$P_{rel}$	$P_{rel1}$
gasPr_TK	"Air", $P_r$	$T$	$T_{2s}$
gash_TK	"Air", $h$	$T$	$T_2$
WatTx_h	$T, x$	$h$	$h_{6w}$
WatPx_h	$P, x$	$h$	$h_{2.5w}$
PhaTKHs_h	$T, \omega$	$h$	$h_{2.5}, h_3, h_{4i}$
PhaTKHs_s0	$T, \omega$	$s^0$	$s_3^0, s_{3.5i}^0, s_{3.5}^0, s_{4i}^0$
PhahHs_TK	$h, \omega$	$T$	$T_{2.6}, T_{3.5i}, T_{3.5}, T_4, T_5$
Phas0Hs_TK	$s^0, \omega$	$T$	$T_{4i}$
PhaHs_Rm	$\omega$	$R$	$R$

Table II compares the solution obtained by using Thermax functions with that given by Bathie [8]. It can be seen from the table that the present model slightly overestimated the amount of injected water per kg of dry air ( $m_w$ ). However, in spite of this error, the figures on the table show that the present model accurately estimated the key cycle performance parameters. The deviations from those of Bathie [8] are less than 1%.

TABLE II. TABLE I: ENERGY ANALYSIS OF THE ERGT CYCLE

Quantity	Bathie [8]	Thermax	Error (%)
$m_w$ [kg]	0.089	0.091	2.217
$T_2$ [K]	623.0	620.449	-0.409
$T_{2.6}$ [K]	769.0	774.095	-0.6625
$T_4$ [K]	831	830.130	0.104645
$w_c$ [kJ/kg]	343.0	342.313	0.200292
$w_{pr}$ [kJ/kg]	436.6	438.912	-0.52952
$Q_{in}$ [kJ/kg]	851.0	853.618	-0.30765
$\eta$	0.513	0.514	-0.22963
Power [kW]	436.6	438.911	-0.52952

The performance improvements due to water injection can be assessed by comparing the thermal efficiency and net power output at different rates of water injection. Fig. 3 shows the thermal efficiency and new power of the gas turbine at water injection rates that vary from 0 to 0.09. As the figure shows, water injection increases both the efficiency and power of the turbine. At the maximum rate of injection, the ERGT cycle increased the power and thermal efficiency of the gas turbine by 31.6% and 9%, respectively. These improvements are close to those achievable by combined Brayton-Rankine systems without the significant increase in the required initial and running costs incurred by the additional hardware.

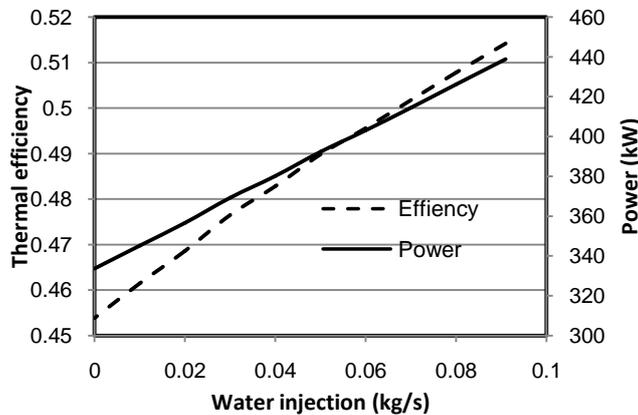


Fig. 3. Effect of water-injection rate on the ERGT power and thermal efficiency

#### IV. EXERGY ANALYSIS OF THE ERGT CYCLE

Exergy analysis, which is based on the second-law of thermodynamics, enables the locations and true magnitudes of waste and loss to be determined. For open systems, flow exergy ( $E_f$ ) is defined as follows [10, 11]:

$$E_f = (h - h_0) - T_0(s - s_0) + \frac{V^2}{2} + gz \quad (17)$$

Where  $T_0$  is the dead-state temperature and  $h_0$  and  $s_0$  are the working fluid enthalpy and entropy values at the dead-state ( $T_0, P_0$ ). The last two terms that represent contributions from kinetic and potential energy are usually small compared to the other terms. Neglecting these terms, the definition of flow exergy becomes:

$$E_f = (h - h_0) - T_0(s - s_0) \quad (18)$$

The rate of exergy increase or decrease across each component (control volume) of the system is governed by [10]:

$$\frac{dE_{cv}}{dt} = \sum_j \left( 1 - \frac{T_0}{T_j} \right) \dot{Q}_j - \left( \dot{W}_{cv} - P_0 \frac{dV_{cv}}{dt} \right) + \sum_i \dot{m}_i e_{fi} - \sum_e \dot{m}_e e_{fe} - \dot{E}_d \quad (19)$$

Where  $\dot{E}_d$  refers to the rate of exergy destruction and the sign convention of heat-in (+) and work-out (+) applies. For steady, single-pass flow devices, the equation reduces to:

$$\dot{E}_d = \left( 1 - \frac{T_0}{T} \right) \dot{Q} - \dot{W}_{cv} + \dot{m}(e_{fi} - e_{fe}) \quad (20)$$

For devices that involve only adiabatic process (e.g. the air-compressor and turbine), (20) can be reduced to the following form:

$$\dot{E}_d = \dot{m}T_0(s_e - s_i) \quad (21)$$

For the combustion chamber that does not involve work-transfer, (20) reduces to:

$$\dot{E}_d = \left( 1 - \frac{T_0}{T} \right) \dot{Q} + \dot{m}(e_{fi} - e_{fe}) \quad (22)$$

The humidifier involves no heat or work transfer and, therefore, the equation reduces to:

$$\dot{E}_d = \sum_i \dot{m}_i e_{fi} - \dot{m}_e e_{fe} \quad (23)$$

Similarly, for the regenerator the equation becomes:

$$\dot{E}_d = \sum_i \dot{m}_i e_{fi} - \sum_e \dot{m}_e e_{fe} \quad (24)$$

Exergy analysis of the ERGT cycle could be performed by making suitable extensions to the Excel sheet developed for the energy analysis. Fig. 4 shows the rate of exergy change across each device in the ERGT system. Neglecting the relatively small exergy input in the hot pressurised water injected in the humidifier, the only external exergy input in the cycle is the heat addition in the combustion chamber, which amounts to about 622 kJ. By cooling the compressed air, the humidifier results in a loss of nearly 54 kJ, but the heat-transfer in the regenerator provides about 194 kJ. The gas-generator turbine converts about 350 kJ into work to drive the compressor and the power turbine produces about 460 kJ of useful work. Therefore, the second-law efficiency of the plant approaches 71%. Fig. 5 shows the percentage of exergy destruction in the different components of the system. The figure shows that the combustion process causes the most exergy destruction in the system closely followed by the process in the humidifier. Therefore, any efforts to improve the system's performance should focus on these two processes.

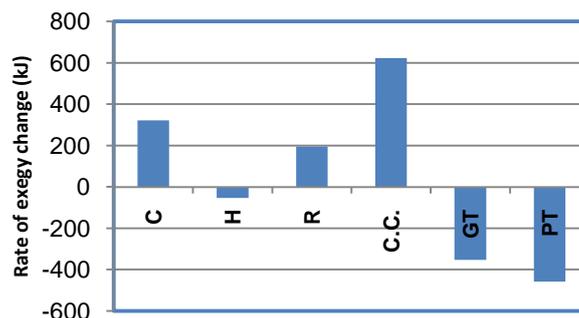


Fig. 4. Exergy change in the ERGT system

C=compressor, H=humidifier, R=regenerator, C.C.=combustion chamber, GT=generator turbine, PT=power turbine

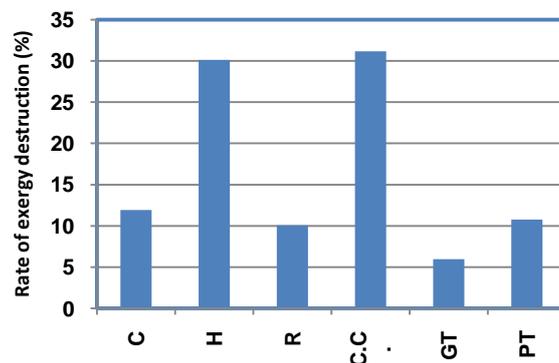


Fig. 5. Percent exergy destruction in the ERGT system

C=compressor, H=humidifier, R=regenerator, C.C.=combustion chamber, GT=generator turbine, PT=power turbine

## V. CONCLUSIONS

Energy and exergy analyses of the ERGT cycle demonstrate the suitability of the Excel-Thermax combination for the analyses of such advanced power generation cycles. Energy analysis of the ERGT cycle shows that, at the maximum rate of water injection, the cycle increased the power and thermal efficiency of the gas turbine by 31.6% and 9%,

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respectively. Exergy analysis of the cycle shows that the combustion chamber and the humidifier are the two processes that mostly cause the destruction of exergy in the system and, therefore, any efforts to improve the system's performance should focus on these two processes.

With its numerous property functions, Thermax enables many other power generation and refrigeration cycles to be analysed by using Excel. For example, the functions provided for gases, water, and superheated steam can be used to analyse various configurations of Brayton and Rankine cycles. The functions provided for vapour-compression refrigerants enable the students to easily analyse multi-stage and cascade refrigeration cycles and cycles that involve suction-line heat exchangers. Both ammonia-water and water-lithium bromide absorption refrigeration cycles can be analysed by using the relevant add-in functions. Finally, it should be mentioned that Thermax makes Excel particularly useful for conducting optimisation analyses of thermal-fluid systems by means of the Solver add-in.

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## BIOGRAPHY



Dr El-Awad currently works as an associate professor at Sohar College of Applied Sciences (Oman). He is on secondment from the University of Khartoum (Sudan). Dr El-Awad, who previously taught at two Malaysian universities, UPM and UNITEN, obtained his M.Sc. at the University of Khartoum in 1984 and Ph.D. at the University of Manchester Institute of Science and Technology (UMIST) in 1993. For his M.Sc. project, he worked on the problem of steam-turbine blade erosion for which he developed a thermodynamic model for the expansion line through a multi-stage steam turbine. For his Ph.D., he developed a vectorised finite-element CFD model for the dispersion of heavier-than-air gases in the atmosphere. His current research interests focus on Excel-aided modelling of thermal-fluid systems and the development of efficient and renewable energy systems.