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## Modelling the Regenerator for Multi-Objective Optimisation of the Air-Bottoming Cycle with Excel

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## Keywords

Thermal-fluid systems, optimisation analysis, property add-ins

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# Modelling the Regenerator for Multi-Variable Optimisation of the Air-Bottoming Cycle with Excel

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## Abstract

Spreadsheet software can be utilised in energy-related courses to enhance the students' problem-solving skills and to train them for future research efforts. In this respect, the present paper demonstrates the capabilities of Microsoft Excel for conducting thermodynamics and heat-transfer optimisation analyses of energy systems by considering the air-bottoming power-generation cycle. The paper presents a model that uses the effectiveness-NTU method to explicitly take into consideration the design particulars of the regenerator, such as its size and overall heat-transfer coefficient. The study, which uses the exact method of analysis, compares the effectiveness-NTU model with the usual effectiveness ( $\epsilon$ ) model for multi-variable optimisation of the air-bottoming cycle.

Keywords: Thermal-fluid systems, optimisation analysis, Excel, Solver, property add-ins

## 1. Introduction

With the availability of computers and computer software, the desire to meet industrial needs has made computational tools important elements of modern mechanical engineering curricula. Thus, computer-aided design software has replaced traditional methods in structural design analysis and computational fluid dynamics (CFD) software has become indispensable tools for designing fluid systems. By appealing to students more than traditional "chalk and talk" methods, the use of computers and computer software as teaching tools helps to engage the students more actively in the learning process and motivates them to undertake private studies beyond taught classes [1,2]. In this respect, success stories of integrating software packages, such as Matlab, Mathcad, and HYSYS, in various engineering courses have been reported [3-5]. Using computational tools in undergraduate research projects also encourages the students to exercise critical thinking and to appreciate the importance of life-long learning [6].

While even an iterative solution can be an obstacle for students in a traditional computer-less environment, computer-aided methods allow them to conduct more realistic analyses of energy systems by removing the tedium of manual calculation methods [7]. In this respect, the present paper demonstrates the use of Excel for conducting multi-variable optimisation analyses of the air-bottoming (AB) power systems by using a combined thermodynamics and heat-transfer model. Thermodynamic analyses of the AB cycle usually model the regenerator by using a simple effectiveness ( $\epsilon$ ) model that does not allow for the design particulars of the heat-exchanger to be specified. Using the  $\epsilon$ -NTU method to determine the exit temperatures of the hot and cold fluids from the regenerator allows the optimisation process to account for the size and other particulars of the regenerator in an explicit

manner. The present Excel-based model, which applies the exact variable specific-heat method in its formulation, determines the properties of the working fluid by using the multi-substance Thermax add-in [8].

## 2. The air-bottoming cycle

Figure 1 shows a schematic diagram of a power generation system that works on the air-bottoming cycle. In the AB cycle, the energy of the exhaust gases in the top Brayton cycle is transferred to the compressed air of the bottom cycle. According to Korobitsyn [9], the bottom cycle adds 20–30% to the power output, which boosts the combined-cycle efficiency by up to 45%. Unlike the combined Brayton-Rankine cycle, the AB cycle doesn't require abundant supply of water or expensive water treatment facilities and can be run unmanned. Figure 2 shows the  $T$ - $s$  diagram for the AB cycle in which the two compressors and two turbines are assumed to be adiabatic but not isentropic.

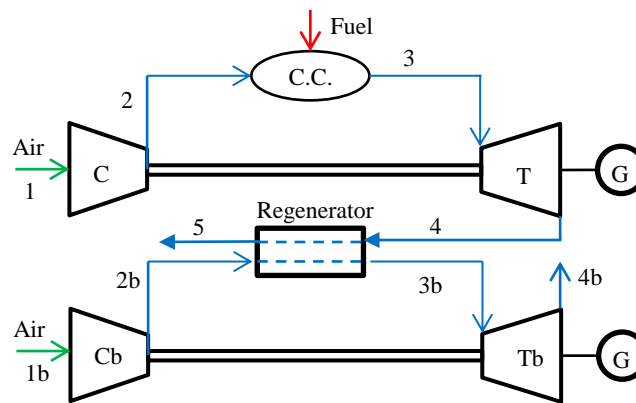


Figure 1: Schematic diagram of the air-bottoming system

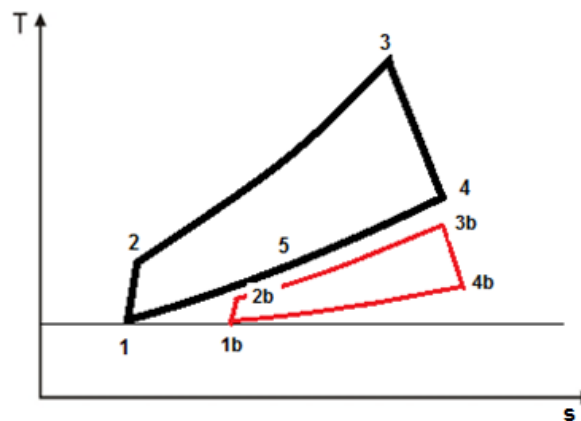


Figure 2:  $T$ - $s$  diagram for the air-bottoming cycle

Taking the working fluid as pure air, which is treated as an ideal gas, and given the intake air temperature ( $T_1$ ) and the turbine inlet temperature ( $T_3$ ), the compressor work ( $w_c$ ) and turbine work ( $w_t$ ) of the top cycle per each kg of air are given by:

$$w_c = (h_1 - h_2) \quad (1)$$

$$w_t = (h_3 - h_4) \quad (2)$$

Enthalpy values at states 1 and 3 are determined by the temperatures of inlet-air and the combustion chamber, respectively. Enthalpy values at states 2 and 4 are determined from the respective temperatures  $T_2$  and  $T_4$ . The temperatures  $T_2$  and  $T_4$  themselves are determined by using the variable specific-heat method. According to this method, the relative pressures  $P_{r2}$  and  $P_{r4}$  can be calculated from [10]:

$$P_{r2} = P_{r1} \times r_p \quad (3)$$

$$P_{r4} = P_{r3} / r_p \quad (4)$$

Where  $P_{r1}$  and  $P_{r3}$  are the relative pressures at  $T_1$  and  $T_3$  and  $r_p$  is the pressure ratio ( $P_2/P_1$ ) in the top cycle. The heat input in the combustion chamber per kg of inlet air ( $q_{in}$ ) is obtainable from:

$$q_{in} = (h_3 - h_2) \quad (5)$$

Similarly, the compressor work ( $w_{cb}$ ) and turbine work ( $w_{tb}$ ) per unit mass flow in the bottom cycle can be calculated from:

$$w_{cb} = (h_{1b} - h_{2b}) \quad (6)$$

$$w_{tb} = (h_{3b} - h_{4b}) \quad (7)$$

The relative pressures  $P_{r2b}$  and  $P_{r4b}$  can be calculated from:

$$P_{r2b} = P_{r1b} \times r_{pb} \quad (8)$$

$$P_{r4b} = P_{r3b} / r_{pb} \quad (9)$$

Where  $P_{r1b}$  and  $P_{r3b}$  are the relative pressures at  $T_{1b}$  and  $T_{3b}$ , respectively, and  $r_{pb}$  is the pressure ratio ( $P_{2b}/P_{1b}$ ) in the boom cycle. Assuming that  $m_b$  kilograms of air go through the bottom cycle for each kilogram of air in the top cycle, the overall net work ( $w_{net}$ ) is given by:

$$w_{net} = (w_t + w_c) + m_b (w_{tb} + w_{cb}) \quad (10)$$

The overall thermal efficiency of the combined AB cycle ( $\eta$ ) is then obtained from:

$$\eta = \frac{w_{net}}{q_{in}} \quad (11)$$

### 3. Modelling the regenerator

The regenerator is a key component in the air-bottoming system that strongly affects its performance and economic feasibility. Therefore, appropriate modelling of the regenerator is particularly important in the analyses of the air-bottoming cycle. Thermodynamic analyses usually adopt a simple effectiveness ( $\epsilon$ ) model for the regenerator, but computer-based methods enable more realistic models that allow multi-variable optimisation of the energy system with respect to its efficiency, cost, size, etc.

### 3.1. The effectiveness model

Figure 1 shows the regenerator as a counter-flow heat exchanger in which the hot exhaust gas from the top cycle enters at  $T_4$  and the cold air from the compressor of the bottom cycle enters at  $T_{2b}$ . The effectiveness model determines the exit temperature of the bottom air ( $T_{3b}$ ) from [10]:

$$T_{3b} = T_{2b} + \varepsilon(T_4 - T_{2b}) \quad (12)$$

Where  $\varepsilon$  is the heat-exchanger's effectiveness, which is below 0.85 for most heat exchangers used in practice [10]. The simple heat-exchanger effectiveness method expressed by Equation (12) is not suitable for modelling the regenerator in the air-bottoming cycle which is complicated by the fact that the hot and cold air mass flow rates and their specific heats may differ by large margins. If Equation (12) were used without modification, the variation of the net specific work and thermal efficiency with the mass flow rate in the bottom cycle ( $m_b$ ) will be as shown in Figure 3. The figure indicates that both the thermal efficiency and net work of the bottom cycle continuously decreasing with increasing the ratio of mass flow ratio ( $m_b$ ) and, therefore, the optimum value for  $m_b$  is zero.

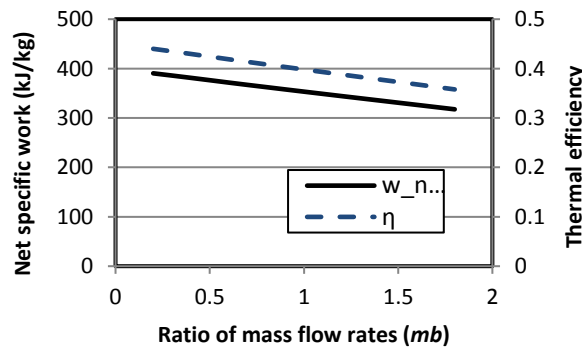


Figure 3: Variation of the net specific work and thermal efficiency with the ratio of mass flow rates

An improved effectiveness model that takes into consideration the heat capacities in the top and bottom cycles ( $C_h$  and  $C_c$ , where  $C=mc_p$ ) was used by Czaja et al [11]. According to this model, the temperatures of the two streams leaving the heat-exchanger ( $T_{3b}$  and  $T_5$ ) are determined from those of the two streams entering the heat-exchanger ( $T_{2b}$  and  $T_4$ ) by the following relationships:

$$T_{3b} = T_{2b} + \varepsilon(T_4 - T_{2b}) \quad \text{If } C_c < C_h \quad (13)$$

$$T_5 = T_4 - \varepsilon(T_4 - T_{2b}) \quad \text{If } C_c \geq C_h \quad (14)$$

Energy balance over the heat exchanger determines the enthalpy of the exhaust gas in the top cycle as:

$$h_5 = h_4 - m_b(h_{3b} - h_{2b}) \quad (15)$$

Assuming a constant specific heat, Equation (15) can be approximated by:

$$T_5 = T_4 - m_b(T_{3b} - T_{2b}) \quad (16)$$

Rearranging:

$$T_{3b} = T_{2b} + \frac{1}{m_b}(T_4 - T_5) \quad (17)$$

Substituting for  $T_5$  from Equation (14) into Equation (17), yields:

$$T_{3b} = T_{2b} + \frac{\varepsilon}{m_b}(T_4 - T_{2b}) \quad (18)$$

Therefore, with this modified  $\varepsilon$ -model, the temperature  $T_{3b}$  is obtained as follows:

$$T_{3b} = T_{2b} + \varepsilon(T_4 - T_{2b}) \quad C_c < C_h \quad (19)$$

$$T_{3b} = T_{2b} + \frac{\varepsilon}{m_b}(T_4 - T_{2b}) \quad C_c \geq C_h \quad (20)$$

### 3.2. The effectiveness-NTU model

The effectiveness model described in the previous section doesn't explicitly take into consideration the heat exchanger's type and size. A more realistic model for the regenerator can be developed by applying the heat-transfer effectiveness-NTU method to determine the exit temperatures from the heat-exchanger [12]. According to this method, the rate of heat-transfer between the two streams ( $Q$ ) is obtained by the following formula [13]:

$$Q = \varepsilon Q_{\max} \quad (21)$$

Where,  $\varepsilon$  is the heat-exchanger effectiveness and  $Q_{\max}$  is the maximum possible rate of heat-transfer as given by:

$$Q_{\max} = C_{\min}(T_{hi} - T_{ci}) \quad (22)$$

Where  $C_{\min} = \text{MIN}(C_h, C_c)$ . Figure 4 shows two arrangements of a single pass, cross-flow heat exchanger that can be used as the regenerator of the AB system.

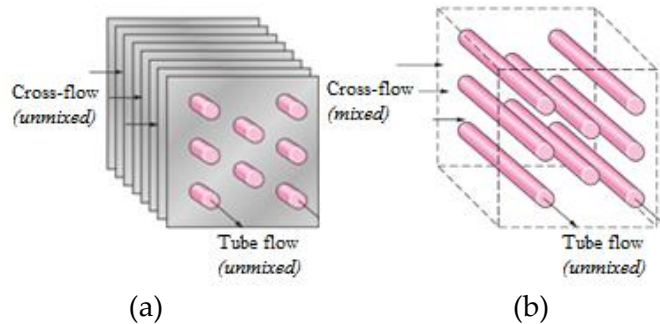


Figure 4: Two arrangements of a cross-flow heat exchanger (adopted from [13])

The two arrangements shown in Figure 4 have different formulae for calculating the effectiveness. For example, the effectiveness of the heat exchanger shown in Figure 4.a (both streams unmixed) is determined by [13]:

$$\varepsilon = 1 - \exp\left\{\frac{NTU^{0.22}}{c} \left[\exp(-cNTU^{0.78}) - 1\right]\right\} \quad (23)$$

Where,  $NTU$  stands for the number of transfer units:

$$NTU = AU / C_{\min} \quad (24)$$

$$c = C_{\min} / C_{\max} \quad (25)$$

In Equation (24),  $A$  is the surface area of the heat exchanger and  $U$  is its overall heat transfer coefficient. Since the values of  $\varepsilon$  for different types of heat exchangers are obtained by using relevant formulae, this model takes into consideration the type of the heat exchanger and the flow patterns of the two fluids [13].

Once  $Q$  is determined from Equation (21), the exit temperatures of the two streams can be obtained from:

$$T_{co} = T_{ci} + Q / C_c \quad (26)$$

$$T_{ho} = T_{hi} - Q / C_h \quad (27)$$

The exit temperatures obtained by the  $\varepsilon$ -NTU method can now be used in the analytical model of the AB cycle. Accordingly, the temperature  $T_{3b}$  in Figure 1 is assigned the value of  $T_{co}$  calculated from Equation (26) and  $T_5$  is assigned the value of  $T_{ho}$  calculated from Equation (27).

#### 4. The computer model

Determination of the pressure ratio ( $r_{pb}$ ) and mass flow rate ( $m_b$ ) in the bottom cycle that maximise the overall thermal efficiency of the AB cycle requires an optimisation process that suits a computer-based solution method. The computer model developed for the present study uses Excel as a modelling platform. Two Excel sheets were developed for the two alternative models of the regenerator. Fluid properties ( $h$ ,  $P_r$ ,  $c_p$ ) at various temperatures are determined by using the Thermax add-in [8]. To illustrate the optimisation process, consider a gas-turbine power plant that operates on a simple Brayton cycle which is to be retrofitted with a bottoming gas turbine. The pressure ratio of the original plant is 10 and the turbine inlet temperature is 1400 K. Intake air to the compressors of both top and bottom cycle is at ambient temperature and pressure of 298.15 K and 103.1 kPa, respectively. The isentropic efficiency for both compressors is 0.85 while that for both turbines is 0.87.

Figure 5 shows the sheet developed for the  $\varepsilon$ -model in which the upper part includes the basic data and calculations for the top cycle while the lower part stores the data of the bottom cycle and its calculations. A value of  $\varepsilon = 0.85$  is used. The formula tab in the sheet reveals the formula used to determine the regenerator exit temperature  $T_{3b}$ . Figure 5 shows that per 1 kg of air the top cycle requires 891.2479 kJ of heat input ( $Q_{in}$ ) to produce 288.1597 kJ of net work ( $w_{nett}$ ). Therefore, the thermal efficiency of the top cycle standing alone ( $\eta_{top}$ ) is 0.323322. For the bottom cycle, the sheet shows the calculations for a pressure ratio ( $P_{ratio_b}$ ) of 5 and mass ratio ( $m_b$ ) of 1. The exhaust gas leaves the top cycle ( $T_4$ ) at 870.182 K and passes part of its energy to the compressed air of the bottom cycle that enters the heat exchanger ( $T_{2b}$ ) at 499.5133 K and leaves ( $T_{3b}$ ) at 814.5817 K. The heated air then expands in the bottom cycle turbine to produce a net work ( $w_{netb}$ ) of 60.93141 kJ per kg of air flow in the bottom cycle. The overall thermal efficiency ( $\eta_{ABC}$ ) at the specified pressure ratio is 0.391688, which is higher than that of the top cycle alone by about 21%.

T <sub>3b</sub>		=F(mcp>mcpb,T <sub>2b</sub> +ε*(T <sub>4-T<sub>2b</sub></sub> ),T <sub>2b</sub> +(mcp/mcpb)*(T <sub>4</sub> -(T <sub>4</sub> -ε*(T <sub>4-T<sub>2b</sub></sub> ))))													
	B	C	D	E	F	G	H	I	J	K	L	M			
1															
2	T <sub>1</sub>	298.15	K	h <sub>1</sub>	297.9887		h <sub>3</sub>	1515.758		w <sub>c</sub>	326.5215				
3	P <sub>1</sub>	101.3	kPa	Pr <sub>1</sub>	1.35393		Pr <sub>3</sub>	454.3223		w <sub>t</sub>	614.6812				
4	T <sub>3</sub>	1400	K	Pr <sub>2s</sub>	13.5393		Pr <sub>4s</sub>	45.43223		w <sub>nett</sub>	288.1597				
5	P <sub>ratio</sub>	10		T <sub>2s</sub>	568.705		T <sub>4s</sub>	786.8156							
6	eff <sub>c</sub>	0.85		h <sub>2s</sub>	575.532		h <sub>4s</sub>	809.2281		Q <sub>in</sub>	891.2479				
7	eff <sub>t</sub>	0.87		h <sub>2</sub>	624.5102		h <sub>4</sub>	901.077		η <sub>top</sub>	0.323322				
8				T <sub>2</sub>	615.1512		T <sub>4</sub>	870.182							
9															
10	T <sub>1b</sub>	298.15	K	h <sub>1b</sub>	297.9887		mcp	1.110196		h <sub>4b</sub>	573.393				
11	P <sub>1b</sub>	101.3	kPa	Pr <sub>1b</sub>	1.35393		mcpb	1.037131		T <sub>4b</sub>	566.6676				
12	eff <sub>cb</sub>	0.85		Pr <sub>2sb</sub>	6.76965		T <sub>3b</sub>	814.5817							
13	eff <sub>tb</sub>	0.87		T <sub>2sb</sub>	469.7377		h <sub>3b</sub>	839.6628		w <sub>cb</sub>	205.3384				
14	ε	0.85		h <sub>2sb</sub>	472.5264		Pr <sub>3b</sub>	51.8665		w <sub>tb</sub>	266.2698				
15				h <sub>2b</sub>	503.3271		Pr <sub>4sb</sub>	10.3733		w <sub>netb</sub>	60.93141				
16	P <sub>ratiob</sub>	5		T <sub>2b</sub>	499.5133		T <sub>4sb</sub>	528.6325		w <sub>net</sub>	349.0911				
17	mb	1					h <sub>4sb</sub>	533.6055		η <sub>ABC</sub>	0.391688			21.14502	
18															

Figure 5: Excel sheet for energy analysis of the AB cycle with the ε-model

T <sub>3b</sub>		=T <sub>co</sub> +273																	
	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R		
1																			
2	T <sub>1</sub>	298.15	K	h <sub>1</sub>	297.9887		h <sub>3</sub>	1515.758		w <sub>c</sub>	326.5215		Input data				NTU-method		
3	P <sub>1</sub>	101.3	kPa	Pr <sub>1</sub>	1.35393		Pr <sub>3</sub>	454.3223		w <sub>t</sub>	614.6812		U	0.03	kW/m <sup>2</sup> .K	C <sub>h</sub>	1.110195826		
4	T <sub>3</sub>	1400	K	Pr <sub>2s</sub>	13.5393		Pr <sub>4s</sub>	45.43223		w <sub>nett</sub>	288.1597		A	467.6712	m <sup>2</sup>	C <sub>c</sub>	1.03713061		
5	P <sub>ratio</sub>	10		T <sub>2s</sub>	568.705		T <sub>4s</sub>	786.8156					T <sub>ci</sub>	226.5133	oC	C <sub>min</sub>	1.03713061		
6	eff <sub>c</sub>	0.85		h <sub>2s</sub>	575.532		h <sub>4s</sub>	809.2281		Q <sub>in</sub>	891.2479		T <sub>hi</sub>	597.182	oC	C <sub>max</sub>	1.110195826		
7	eff <sub>t</sub>	0.87		h <sub>2</sub>	624.5102		h <sub>4</sub>	901.077		η <sub>top</sub>	0.323322		Flow <sub>h</sub>	1	kg/s	C	0.934187092		
8				T <sub>2</sub>	615.1512		T <sub>4</sub>	870.182					Flow <sub>c</sub>	1	kg/s	Q <sub>max</sub>	384.4318199		
9													cp <sub>h</sub>	1.110196	kJ/kg.K	NTU	13.52784007		
10	T <sub>1b</sub>	298.15	K	h <sub>1b</sub>	297.9887		T <sub>3b</sub>	814.5798		w <sub>cb</sub>	205.3384		cp <sub>c</sub>	1.037131	kJ/kg.K	eff	0.849994764		
11	P <sub>1b</sub>	101.3	kPa	Pr <sub>1b</sub>	1.35393		h <sub>3b</sub>	839.6607		w <sub>tb</sub>	266.2698					T <sub>co</sub>	326.7650342		
12	eff <sub>cb</sub>	0.85		Pr <sub>2sb</sub>	6.76965		Pr <sub>3b</sub>	51.86603		w <sub>nwtb</sub>	60.93072					T <sub>ho</sub>	302.8510093		
13	eff <sub>tb</sub>	0.87		T <sub>2sb</sub>	469.7377		Pr <sub>4sb</sub>	10.37321		w <sub>net</sub>	349.0904								
14	ε	0.85		h <sub>2sb</sub>	472.5264		T <sub>4sb</sub>	528.6313		η <sub>ABC</sub>	0.391687								
15				h <sub>2b</sub>	503.3271		h <sub>4sb</sub>	533.6042											
16	P <sub>ratiob</sub>	5		T <sub>2b</sub>	499.5133		h <sub>4b</sub>	573.3916											
17	mb	1					T <sub>4b</sub>	566.6662											
18																			

Figure 6: Excel sheet for energy analysis of the AB cycle with the ε-NTU model

Figure 6 shows the sheet developed for the ε-NTU model which adds the data and calculations involved in the ε-NTU method as shown on the right side of the sheet. In this model, the area (A) and overall heat-transfer coefficient (U) of the heat exchanger have to be specified. For a gas-to-gas heat exchanger, the range of U is 10-40 W/m<sup>2</sup>.°C [13]. In the present analysis, U is taken as 30 W/m<sup>2</sup>.°C. The area A is specified such that the exit temperature of the hot-air in the bottom cycle (T<sub>3b</sub>) is the same as that obtained by the effectiveness model, i.e. 814.58K. The corresponding surface area (obtained by Goal-Seek) is 467.7 m<sup>2</sup>. Note that the values of the net work and overall thermal efficiency calculated by this model are identical to those obtained by the ε-model. The formula ribbon here reveals how the temperature T<sub>3b</sub> is assigned the absolute value of T<sub>c,o</sub> as calculated by the ε-NTU method:

$$T_{3b} = T_{co} + 273 \tag{28}$$

The conversion of T<sub>3b</sub> from Celsius to absolute degree is required by the ideal-gas analysis of the AB cycle.

## 5. Cycle optimisation analyses

The two Excel sheet described in the previous section that apply the two regenerator models will be now be used to determine the optimum pressure ratio and mass flow rate for the bottom cycle by studying the effects of these parameters on the net work from the AB cycle and its thermal efficiency.

### 5.1. Effect of the pressure ratio

Keeping the mass flow ratio of air in the bottom cycle ( $m_b$ ) equal to 1, the two sheets were used to determine the variation of the combined net specific work ( $w_{net}$ ) and thermal efficiency ( $\eta_{ABC}$ ) with the pressure ratio of the bottom cycle ( $r_{pb}$ ). Figures 7 and 8 show the results obtained by the two regenerator models. Both figures indicate that the maximum net work and the maximum thermal efficiency occur at the same pressure ratio which is  $\approx 4$ .

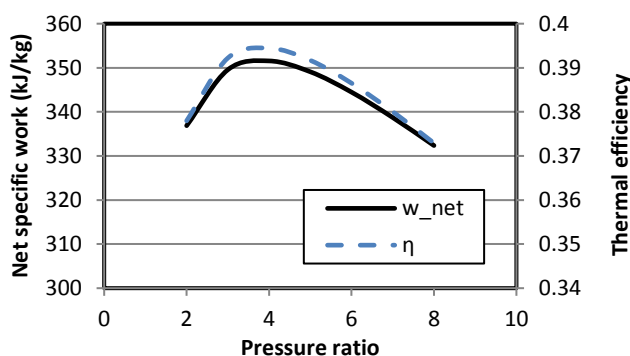


Figure 7: Effect of the bottom-cycle pressure ratio ( $r_p$ ) on the combined net specific work and thermal efficiency with the  $\epsilon$ -model

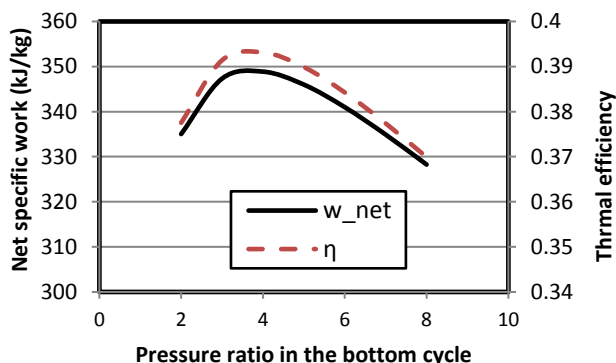


Figure 8: Effect of the bottom-cycle pressure ratio ( $r_p$ ) on the combined net specific work and thermal efficiency with the  $\epsilon$ -NTU model

### 5.2. Effect of the mass flow-rate

With the pressure ratio in the bottom cycle fixed at 4, the two sheets were then used to determine the variation of the combined net specific work ( $w_{net}$ ) and thermal efficiency ( $\eta_{ABC}$ ) with the mass flow ratio in the bottom cycle ( $m_b$ ). Figures 9 and 10 show the results obtained at selected values of the air flow ratio in the cycle  $0.2 \leq m_b \leq 1.8$ . Unlike Figure 3, the two figures indicate that there is an optimum value for  $m_b$  at which the AB cycle yields its maximum net specific work and thermal efficiency. Both figures indicate that the ratio of air in the bottom cycle per each kg of

air in the top cycle is about 1.2 kg, but the  $\epsilon$ -NTU method shows a smoother variation with  $m_b$  than the  $\epsilon$ -model and yields significantly higher values of  $w_{net}$  and  $\eta_{ABC}$  at values of  $m_b$  greater than the optimum value. Estimates of the optimum values of  $r_{pb}$  and  $m_b$  can be improved by repeating the calculations starting with the newly determined value of  $m_b = 1.2$ .

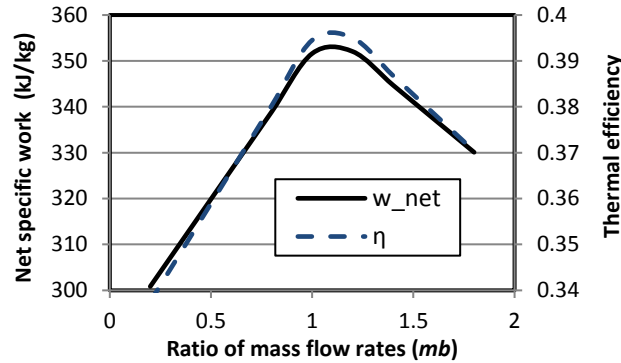


Figure 9: Effect of mass flow ratio on the net specific work and thermal efficiency with the  $\epsilon$ -model

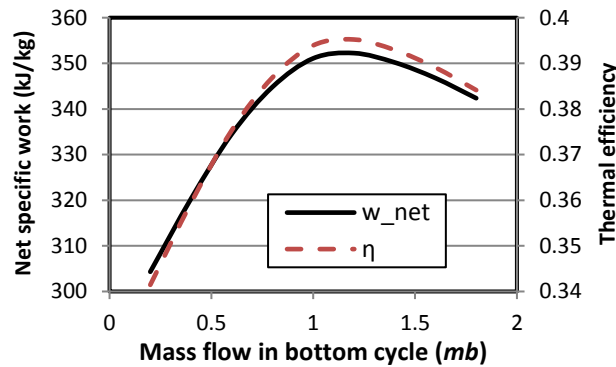


Figure 10: Effect of mass flow ratio on the net specific work and thermal efficiency with the  $\epsilon$ -NTU model

### 5.3. Cycle optimisation with Solver

The combination of  $r_p$  and  $m_b$  that optimises either the overall thermal efficiency or the net specific work of the air-bottoming cycle can be determined easily and accurately by using the Solver add-in. Solver allows Excel to find the maximum or minimum value of the function in a “target cell” by modifying the values stored in the “changing cells”. Figure 11 shows the set-up for Solver for optimising the overall thermal efficiency ( $\eta_{ABC}$ ).

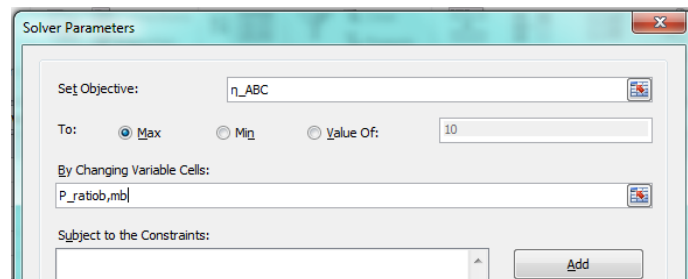


Figure: 11 Solver set-up to maximise the overall efficiency

As calculated by the sheet using the  $\varepsilon$ -model, the values determined by Solver for the combination of pressure ratio and mass flow ratio that maximise the overall thermal efficiency, are 2.88 and 1.086, respectively. At these values of  $r_{pb}$  and  $m_b$ , the net specific work and overall thermal efficiency are 353.96 kJ/kg and 0.397, respectively. The corresponding values calculated by the sheet using the  $\varepsilon$ -NTU model are 3.80 and 1.097, at which the net specific work and overall thermal efficiency are 353.44 kJ/kg and 0.397, respectively. Although the maximum values of  $w_{net}$  and  $\eta_{ABC}$  obtained by the two models are almost identical, the  $\varepsilon$ -model obtains these values at a much lower value of the pressure ratio (2.88) compared to the  $\varepsilon$ -NTU model (3.8).

## 6. Conclusions

With its wide distribution, ease of use, and powerful graphics tools, Excel can be utilised to introduce the basic concepts in introductory energy-related courses. The Thermax property add-in enables the spreadsheet to be used as an open modelling platform for thermal-fluid systems that helps students in advanced energy-related courses to conduct design projects of energy systems. In this respect, the paper presents a case study that demonstrates the advantages of using Excel in the optimisation analysis of the air-bottoming gas-turbine cycle. The example shows that the integration of thermodynamic and heat-transfer methods enables us to develop more useful models of thermal-fluid systems.

By using the  $\varepsilon$ -NTU method to model the generator in the air-bottoming cycle, the computer model allows for different types of heat exchangers. Therefore, the optimisation process can take into consideration the economics as well as the compactness of the system. Such studies help students to practice self-learning and to prepare for future energy-related research activities.

The present optimisation analysis of the air-bottoming cycle, which used Thermax to apply the variable specific-heat method, can also be performed by using the IdealGas add-in developed at the University of Alabama [14]. In the lack of such property add-ins, the analysis can be conducted by using the approximate constant specific-heat method of analysis [10].

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