

Second Year Progress Report

Experimental and Theoretical Investigations of New Power Cycles and Advanced Falling Film Heat Exchangers

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ABSTRACT

The annual progress report for the period of October 1, 1999 to September 30, 2000 on DOE/UNM grant number DE-FG26-98FT40148 discusses the progress on both the theoretical analysis of advanced power cycles and the experimental investigation of advanced falling film heat exchangers.

The previously developed computer program for the triple cycle, based on the air standard cycle assumption, was modified to include actual air composition (%77.48 N₂, %20.59 O₂, %1.9 H₂O, and %0.03 CO₂). The actual combustion products were used in exergy analysis of the triple cycle. The effect of steam injection into the combustion chamber on its irreversibility, and the irreversibility of the entire cycle, was evaluated. A more practical fuel inlet condition and a better position of the feedwater heater in the steam cycle were used in the modified cycle. The effect of pinch point and the temperature difference between the combustion products, as well as the steam in the heat recovery steam generator on irreversibility of the cycle were evaluated.

Design, construction, and testing of the multitube horizontal falling film condenser facility were completed. Two effective heat transfer additives (2-ethyl-1-hexanol and alkyl amine) were identified and tested for steam condensation. The test results are included. The condenser was designed with twelve tubes in an array of three horizontals and four verticals, with a 2-inch horizontal and 1.5-inch vertical in-line pitch. By using effective additives, the condensation heat transfer rate can be augmented as much as 30%, as compared to a heat transfer that operated without additives under the same operating condition. When heat transfer additives function effectively, the condensate-droplets become more dispersed and have a smaller shape than those produced without additives. These droplets, unlike traditional turbulence, start at the top portion of the condenser tubes and cover most of the tubes. Such a flow behavior can be explained by the Marangoni effect (in terms of thermodynamic equilibrium) in connection with obtained surface tension data. In our experiments, we noted that the use of heat transfer additives such as 2-ethyl-1-hexanol for steam condensation was highly effective.

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EXECUTIVE SUMMARY

Excellent progress was made during the second year of the DOE HBCU/OMI (grant number DE-FG26-98FT40148), which was the period from October 1, 1999 to September 30, 2000. The major research tasks performed during this period were:

- 1: Second Law Analysis and Optimization of Combined Triple Power Cycle
- 2: Construction of the Multitube Condenser Facility and Additive Testing

The triple cycle developed in the first year effort, (based on the air standard cycle assumption), was improved to accommodate actual air composition along with a more practical fuel inlet condition, in addition to the position of the feedwater heater in the steam cycle. The triple cycle shows an excellent potential to achieve high plant efficiency.

Construction of the multitube falling film condenser facility is complete, with a data acquisition system capable of monitoring 10 temperatures, 3 pressures, 1 flow rate, and 5 auxiliary data inputs. Two effective heat transfer additives (2-ethyl-1-hexanol and alkyl amine) were identified and tested for steam condensation. We are interested in applying patents for regarding effective additives.

Interaction with the Public Service Company of New Mexico (PNM) has progressed. Our energy group has toured the San Juan coal power plant, which constitutes a primary power source for State of New Mexico. In addition, our annual meeting with a PNM officer proved fruitful.

Dr. Razani (PI) has participated in the Seventh Annual HBCU/OMI symposium, in Pittsburgh, Pennsylvania on June 8-9, 2000. The summary of activities in the symposium was presented to our energy group at a regularly scheduled bi-weekly group meeting. In the second year, a number of minority students participated in research activities under this grant. Mr. Irvin Marrero has finished his M.S. degree, studying computer modeling and optimization of new power cycles. Mr. Aaron Stone was graduated with B.S. degree and is now continuing his work to obtain M.S. degree. Research participants are:

Mr. Aaron Stone, minority graduate student, M.S. candidate in ME
Mr. Irvin Marrero, minority graduate student, graduated with M.S. degree in ME
Ms. Blanca Montoya, minority undergraduate ME student
Mr. Arne Lefsaker, graduated with B.S. degree in ME
Dr. Kwang J. Kim, Co-PI, and
Dr. Arsalan Razani, PI.

During this period, two technical papers have been submitted for review in major journals. Those papers are: "Second Law Analysis and Optimization of a Combined Triple Power Cycle," in *Energy Conversion and Management* (by Marrero, Lefsaker, Razani, and Kim) and "The Effective Use of Heat Transfer Additives for Steam Condensation," in *Applied Thermal Engineering* (by Kim, Lefsaker, Razani, and Stone).

THEORETICAL/EXPERIMENTAL RESULTS AND DISCUSSION

1.0 Second Law Analysis and Optimization of Combined Triple Power Cycle

Thermodynamic analysis of the combined triple power cycle composed of a gas Brayton topping cycle, an intermediate steam Rankine cycle, and an ammonia bottoming Rankine cycle, (using air-standard cycle assumption for the gas cycle and an exhaust steam temperature of 538 °C and pressure of 160 bar) was reported in the last progress report. In order to analyze a more practical cycle and to improve the model used for the triple cycle, the following modifications were made to the triple cycle:

- a) Since the analysis of power cycles and their sources of inefficiency are best performed by the exergy analysis, a more accurate model for the composition of the air in environment should be considered. A major modification undertaken in last year for the triple cycle was to use environmental air composition and actual composition of combustion products in exergy analysis of the cycle. The composition of air is taken as %77.48 N₂, %20.54 O₂, %1.9 H₂O, and %0.03 CO₂. However, the actual composition of combustion products influences the gas topping cycle analysis and the Heat Recovery Steam Generation (HRSG) performance. Due to this observation, the air-standard cycle assumption of previous investigation was removed. As a result, the exergy input and exit to the components and their irreversibilities are now computed more accurately.
- b) Another major modification to the previous model was to investigate the effect of steam injection into the combustion chamber on its irreversibility, the irreversibility distribution, and efficiency of the triple cycle. Steam is injected to the combustion chamber to mitigate the formation of NO_x at high inlet turbine temperatures. The steam injected into the combustion chamber is assumed to be saturated vapor at the operating pressure of the combustion chamber, which depends on the gas topping cycle pressure ratio. The effect of steam injection on combustion irreversibility and efficiency of entire cycle was evaluated. The effect of steam injection on cycle performance was first analyzed using a model for the

gas topping cycle shown in Figure 1. The model was then added to the triple cycle given in Figure 2.

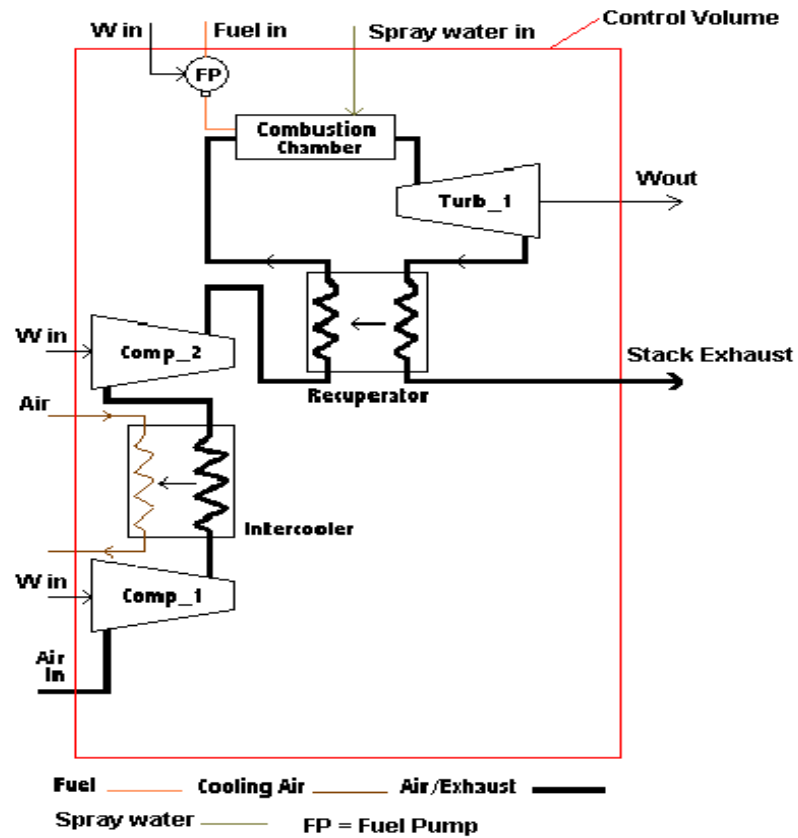


Figure 1. The topping cycle configuration

- c) The model for the triple cycle was modified to include the variation in the pressure of the steam cycle and the exhaust temperature of steam from HRSG. These two parameters were fixed in the previous model ($T=538^{\circ}\text{C}$ and $P=160$ bar). In the modified model the temperature of two feedwater heaters are calculated based on the steam pressure in HRSG. The feed water heaters are placed at equal temperature intervals in the steam cycle. In this manner an optimization can be performed with respect to steam pressure and temperature to find the optimum condition for the entire cycle. The position of the steam/ammonia condenser/evaporator is still fixed at a steam condensation temperature of 100°C , corresponding to steam pressure of 1 atm. The effect of the pinch point in HRSG and the temperature difference between the combustion

products and steam temperature in HRSG on the irreversibility of the cycle was evaluated.

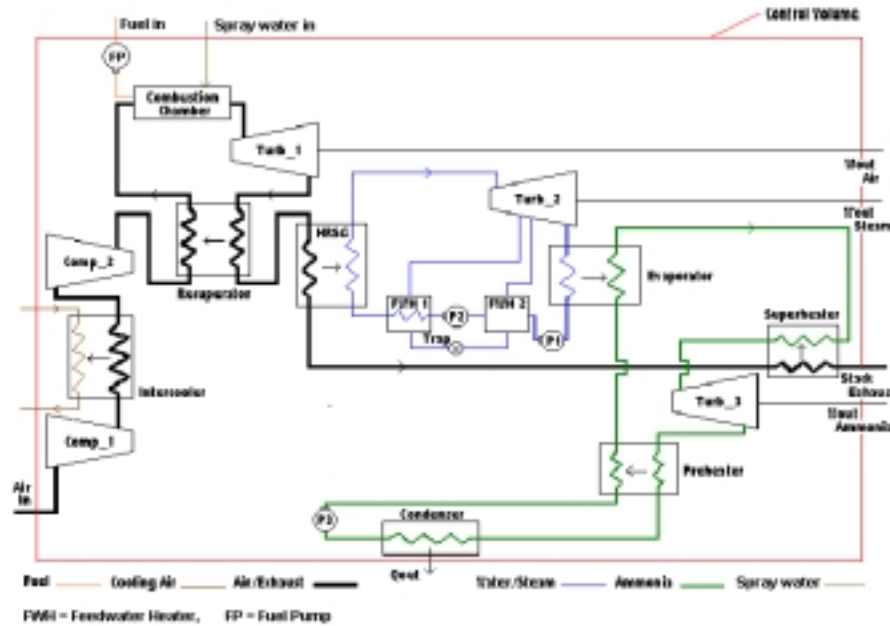


Figure 2. Flow diagram of the triple cycle

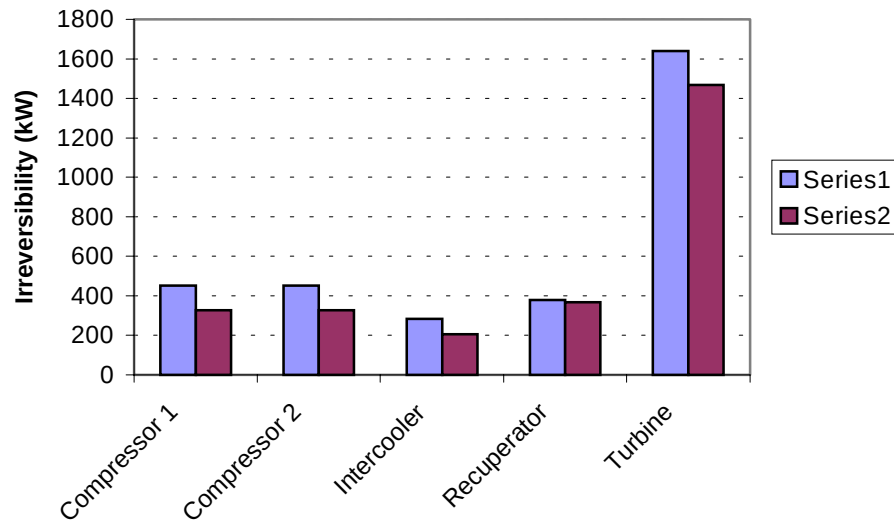


Figure 3. The Irreversibility of the different components for TIT =1600 K and pressure-ratio =35. Series 1 is for 0 kg/s spray water added. Series 2 is for 4 kg/s spray water added.

- d) The fuel is assumed to be natural gas, with an equivalent chemical formula of $\text{CH}_{3.88}$. Using the fuel, (and assuming complete combustion), the chemical composition of combustion products is determined. The composition changes depending on the amount of steam injected into combustion chamber. In addition, the turbine inlet temperature that is an important parameter in performance analysis. A fuel compressor was added to the cycle to increase the pressure of the fuel to the operating pressure of the combustion chamber. The exhaust steam is not used to heat the fuel in this model. Heating of fuel using the exhaust stream will increase the efficiency.

The nominal values of the parameters used in the analysis of gas topping cycle and the triple cycle are given in Tables 1 and 2, respectively. The irreversibility distribution for the gas topping cycle and the triple cycle are given in Figures 3 and 4, respectively. The second law efficiency of the triple cycle as a function of the pressure ratio is given in Figure 5. More results and discussion of analysis of the gas topping and triple cycles are given in Appendix A. This appendix is a copy of the paper submitted to Journal of Energy Conversion and Management for publication. Appendix B-1 and B-2 give the EES programs and sample results of gas topping and triple cycles, respectively.

Table 1. Nominal settings for the topping cycle

Parameter	Value	Units
Gas turbine inlet temperature	1600	K
Pressure-ratio	25	Dimensionless
Mass flow rate of fuel	1	Kg/s
Environmental temperature	298	K
Mass flow rate of spray water	0	Kg/s
Input energy	4200	KW
Input exergy	43680	KW
Heat exchanger effectiveness	0.9	Dimensionless
Turbine efficiency	0.9	Dimensionless
Pump efficiency	0.9	Dimensionless
Compressor efficiency	0.9	Dimensionless

Table 2. Nominal values of different parameters in the triple cycle

Parameter	Value	Units
Gas turbine inlet temperature	1600	K
Pressure Ratio	25	Dimensionless
Mass flow rate of fuel ($\text{CH}_{3.884}$)	1	Kg/s
Environmental temperature	298	K
MH2O spray	0	Kg/s
DELTA T	30	K
T _{pinch}	10	K
DELNH ₃	20	K
DELFWH	5	K
DELTA T _{evap}	4	K
DELTA T _{ph}	2	K
DELTA T _c	5	K
Input exergy	43680	KW
Input energy	42000	KW
Heat exchanger effectiveness	0.9	Dimensionless
Pump efficiency	0.7	Dimensionless
Turbine efficiency	0.9	Dimensionless
Compressor efficiency	0.9	Dimensionless

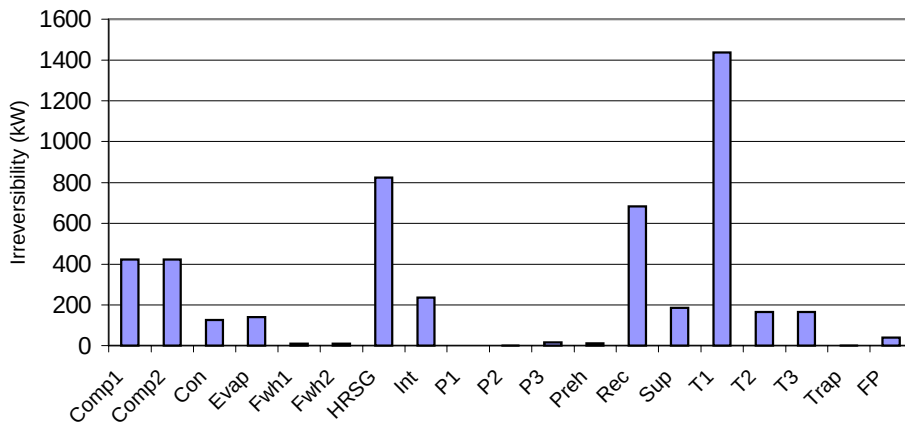


Figure 4. Component irreversibility for the triple cycle for T_{pinch} =20 K and DELTAT=90 K

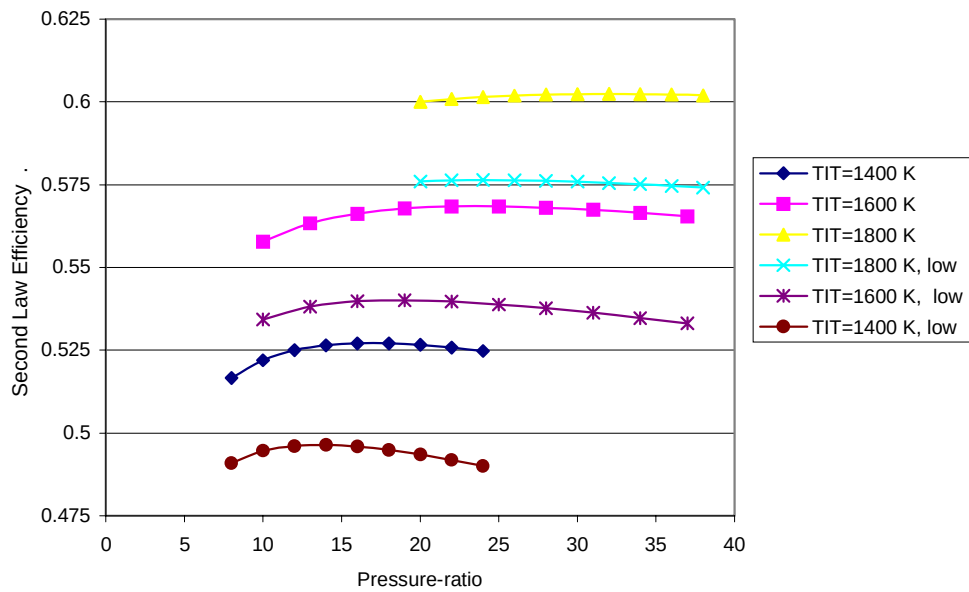


Figure 5. The effect of gas turbine inlet temperature (TIT) and pressure-ratio on the second law efficiency for the triple cycle with compressors and turbines efficiencies 0.9 and 0.85. The values of TIT marked “low” are the ones with turbine and compressor efficiencies of 0.85. The others represent nominal efficiency of 0.9.

2.0 Construction of the Multitube Condenser Facility and Additive Testing

2.1 Horizontal Tube Condensation – In General

The power industry commonly uses a bank of horizontal smooth (or vertical) tubes for a condenser design. In most horizontal tube condensers, the condensate flows as a thin film down the outside of each tube. A cold stream is circulated through the tubes while the condensation takes place at the interface between the condensate and vapor. The flow characteristics primarily depend on the condensate flow rate. When the condensate flow rate is small, the condensate falls in the form of droplets or long tails. Therefore, key engineering objectives are to obtain the complete coverage of the condensate over the heat transfer surface and to provide adequate mixing of the condensate so as to minimize the overall heat and mass transfer resistance at little cost. Although advanced surfaces are usually available, they are rather expensive. For a bank of horizontal tubes, the heat transfer coefficient for each tube differs in nature since the condensate from one tube will drop on the tube below it in the line. Although analytical approaches are possible, this situation is somewhat complicated to model and therefore, experimental approaches are convenient.

2.2 Construction of the Multitube Condenser Facility

Figure 6 shows a schematic drawing of the constructed multi-tube condenser facility. In Figure 7, the data acquisition system (LabView™) is shown. The condenser is connected with the cold reservoir, boiler, and water chiller. The condenser design is a closed system, made of carbon-steel. Steam is generated at the boiler by two immersion heaters with a total heating capacity of 1,650 W with a controller. The water evaporates and fills the rest of the condenser with steam. Copper tubes run through the top of the condenser. Cooling water flows through the copper tubes. When the steam hits the cold copper tubes, it condenses and eventually falls back to the bottom water reservoir. The condenser was designed with twelve tubes in an array of three horizontals and four verticals. A 2-inch horizontal and 1.5-inch vertical in-line pitch was selected. These condenser tubes were connected as shown; “in” designates where the cooling water enters

the condenser and “out” designates where the cooling water leaves the condenser. The rest of the tubes are connected starting from 1 and to the next increasing number (1 and 2, 3 and 4, 5 and 6, 7 and 8, 9 and 10). At the top surface of the condenser, a circular hole with a diameter of 6.7 inches was made to place a plastic dome that allows one to see through it, monitoring the condensation process during the experiments. This dome was also connected to a vacuum pump. A NeopreneTM rubber vacuum seal was placed between the condenser and the dome so as to obtain a complete vacuum. A number of shake-up tests were performed to meet a favorable leak rate less than 10^{-3} Torr/hour so as to minimize the detrimental effect of non-condensable to the condensation process. In order for the operator to oversee the condensation process from the side, a sight-glass with a diameter of 1.4 inches was placed in one side of the condenser, parallel to the condensation tubes. The other sight-glass with the same diameter was placed near the bottom of the condenser at the same side as the other window. The condenser tubes had a 16 mm outside diameter, with a wall thickness of 1 mm. The tubes extended 3 inches outside the condenser on each side. Holes were drilled through the steel plates for the tubes, and the tubes were welded to the condenser wall. The entire system was properly insulated.

2.3 Condensation Testing and Results

It appears that in the presence of effective additives, the steam condensation process observed in our experiment was mostly a dropwise-like condensation process. When effective heat transfer additives were present, the condensate water droplets became a smaller shape than those without additives and became more dispersed, exhibiting much higher frequencies. The observed size of the droplets with heat transfer additives was approximately 1-2 mm in diameter in the additive concentration range of 100-1,000 ppm (note that the additive concentration is defined by the mass of the added heat transfer additive divided by the total mass of the water). A further increase in additive concentration (particularly up to 10,000 ppm, =1%, at which an effective heat transfer additive, 2-ethyl-1-hexanol, was added) decreased the condensate droplet size to approximately 0.1-0.5 mm in diameter. These droplets are smaller with heat transfer additives present and, also, there was more condensation activity at the condenser surface. As soon as condensate appeared on the surface of the condenser tubes, the film

contracted into small droplets. As illustrated in Figure 8, effective additives displayed considerably vigorous disturbances. We also observed that film substrates appear to be thinner compared to those without additives. Such a flow condition was quite attractive for condensation in the absence of smooth, side-drainage, and splashing types. Overall, added surfactants play a pivotal role in improving dropwise-like condensate activity that leads to a significant benefit for condensation heat transfer.

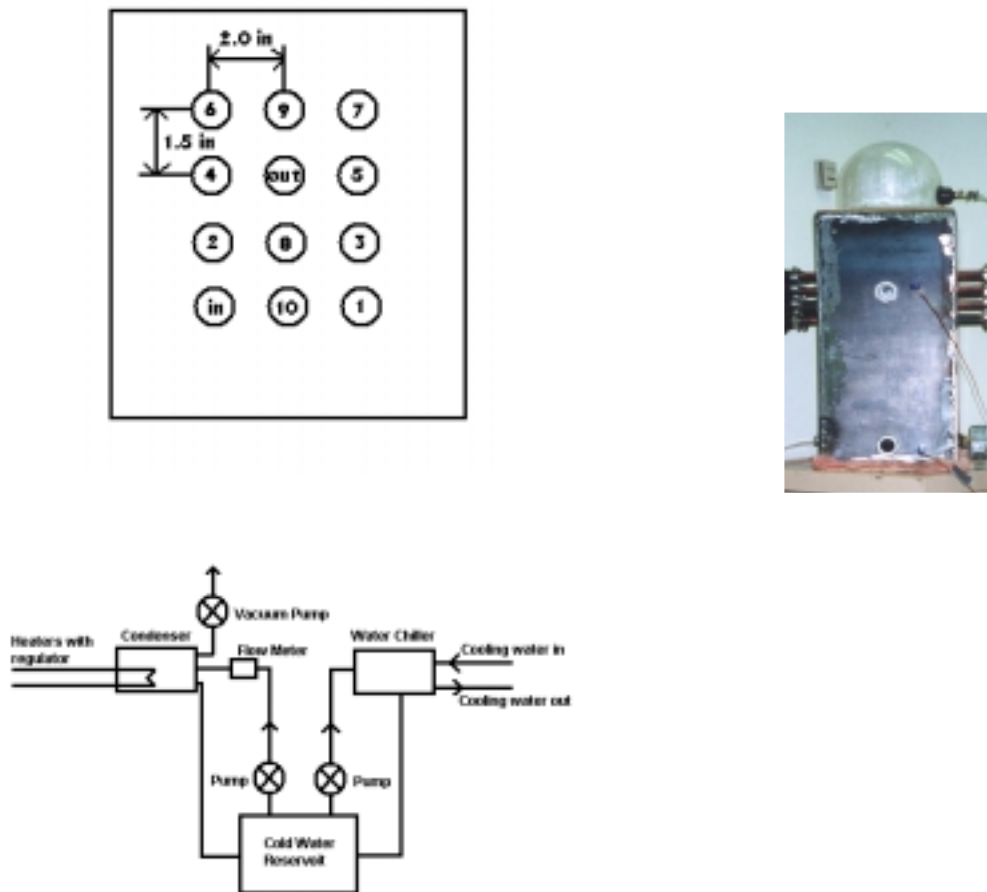


Figure 6. A schematic drawing of the apparatus along with tube spacing (left) and its photograph (right). Two thermocouples were placed inside the condenser: one to measure the liquid temperature at the boiler, and the other to measure the steam temperature. Five others were also placed on the outside surface of the copper tubes right outside the condenser: one for the inlet cooling water, one for the outlet cooling water, and three others in-between. The thermocouples were of the type T.



Figure 7. A photograph of the data acquisition system.
A virtual instrument was setup to carry out data acquisition so as to monitor and save the appropriate data set simultaneously.

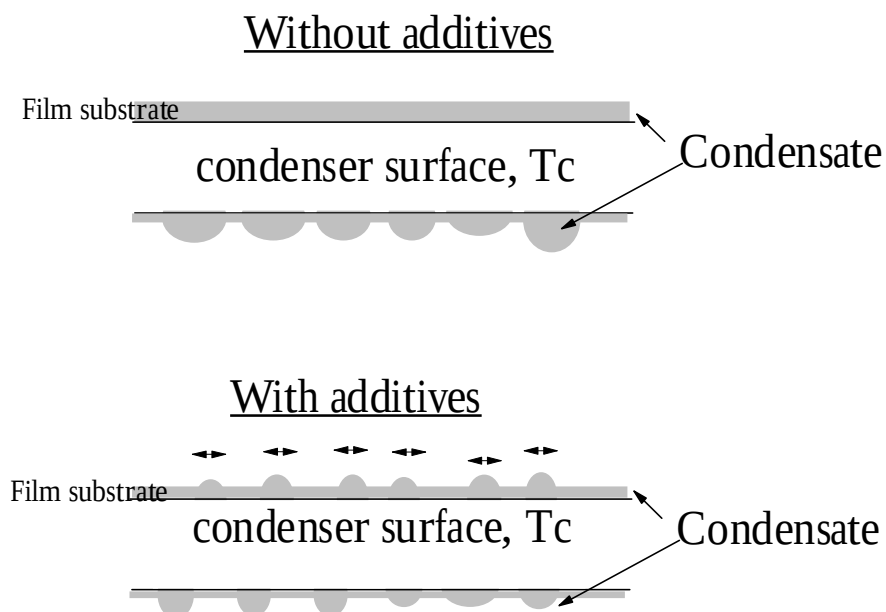


Figure 8. Schematic illustrations of observed condensate behavior.
When effective additives are present, usually film substrates get thinner and the condensate becomes a *dropwise-like* form with a much smaller droplet size than that of the usual filmwise condensate.

Small amounts of certain surfactant additives can change the condensation phenomena of steam. Since the amounts of surfactants used in our study is small and no micelles are formed at this small concentration range (no changes in bulk properties), the surface tension was the sole property that we considered. Therefore, surface tension can be subjected to study for its relationship to the condensation heat transfer coefficient, h_c ,

$$h_c = f(\sigma) \quad (1).$$

The effect of heat transfer additives on the condensation heat transfer was evaluated by introducing the enhancement factor, F_e , which is defined by,

$$F_e = \frac{\bar{h}_a}{\bar{h}_o} \quad (2),$$

where $\bar{h}_a$ and $\bar{h}_o$ denote the average condensation heat transfer coefficients with additives and without additives, respectively. Note that the overall condensation heat transfer coefficients, $\bar{h}$, can be found from,

$$\bar{h} = \frac{\dot{q}}{A_c (T_{sat} - T_w)} \quad (3),$$

where A_c , T_{sat} , T_w , and $\dot{q}$ are the condenser heat transfer area, saturation temperature, average wall temperature, and condenser load, respectively. The condenser load can be simply obtained by,

$$\dot{q} = \dot{m} C_p \Delta T \quad (4),$$

where $\dot{m}$ is the cooling water mass flow rate, C_p is the cooling water heat capacity and ΔT is the temperature gain in the cooling water. The saturation temperature, T_{sat} , was considered as the one inside the condenser with a thermocouple about two inches at the side of row number 2 from the top (middle row, Figure 6). The T_w was used as the mean temperature of the inlet and outlet wall temperature (also measured with thermocouples).

In order to evaluate the effectiveness of the additives, the heat input to the boiler was controlled to maintain the same temperature gain, ΔT , in the cooling water relative to one without additives. This approach is convenient to evaluate the performance of the additives. It is anticipated that effective additives exhibit higher values of the condensation heat transfer coefficient that lead to lower the corresponding saturation temperatures, T_{sat} . It is due to the less required boiler duty. The experiments were

carried out for three different controlled values of ΔT of 3.0, 4.5, and 6.2 °C, and results are provided in Figure 9-(a), (b), and (c), respectively.

In general we conceive that heat transfer additive-A can enhance the heat transfer coefficient by as much as 17 percent, which is believed to be a significant benefit for condenser design. As noted, the effect of adding heat transfer additives is progressively larger for increasing ΔT 's. For an obtained ΔT , enhanced condensation activity gained by effective additives leads to a higher heat transfer rate, which lowers the corresponding saturation temperature because of less power input to the boiler. It is obvious, from Figure 9, that heat transfer additive-A gives much higher average condensation heat transfer coefficients than heat transfer additive-B. It can be clearly seen that heat transfer additive-A is more useful for the purpose of this study. One extra run was done, adding 1 percent (10,000 ppm) of heat transfer additive-A (not plotted in Figure 9). A 30% increase in the average heat transfer coefficient was obtained.

There are two Marangoni numbers that can arise from the surface tension gradients at the condensation interface,

$$Ma_{add} = -\left(\frac{\partial\sigma}{\partial C_{add}}\right)\left(\frac{\Delta C_{add} L_{ch}}{\mu D_{add}}\right) \text{ and } Ma_T = -\left(\frac{\partial\sigma}{\partial T}\right)\left(\frac{\Delta T L_{ch}}{\mu\alpha}\right) \quad (6).$$

Where μ , α , D_{add} , and L_{ch} represent the dynamic viscosity and thermal diffusivity of the condensate, additive diffusivity, and length scale, respectively. Also, realizing that,

$$\frac{Ma_{add}}{Ma_T} \gg 1 \quad (7),$$

an important parameter is Ma_{add} . In order to determine the effect of Marangoni number on the results provided in Figure 9, we use a linear correlation with the Marangoni number, based on the turbulent scale. The effect of ΔT is addressed then:

$$F_e = \frac{\bar{h}_a}{h_o} \cong k_1 + k_2 \left(\frac{\Delta T_i}{\Delta T_o} \right) Ma_{add} \text{Pr}^{-1/2} \quad (8),$$

$$\begin{aligned}
& k_1 = 1 \text{ and } k_2 = 0; \text{ when } \left(\frac{\Delta T_i}{\Delta T_o} \right) Ma_{add} Pr^{-1/2} < 10^5 \\
& k_1 = 1 \text{ and } k_2 = 4.337 \times 10^{-8}; \text{ when } \left(\frac{\Delta T_i}{\Delta T_o} \right) Ma_{add} Pr^{-1/2} > 10^5
\end{aligned} \tag{9}.$$

Where k_1 and k_2 are experimentally determined parameters. ΔT_o is an arbitrary reference temperature gain ($\Delta T = 6.2^\circ\text{C}$ was chosen in this analysis). The parameters k_1 and k_2 include the effect of condenser geometry and dynamic surface tension effect, and Pr is the Prandtl number. Figure 10 shows such a correlation.

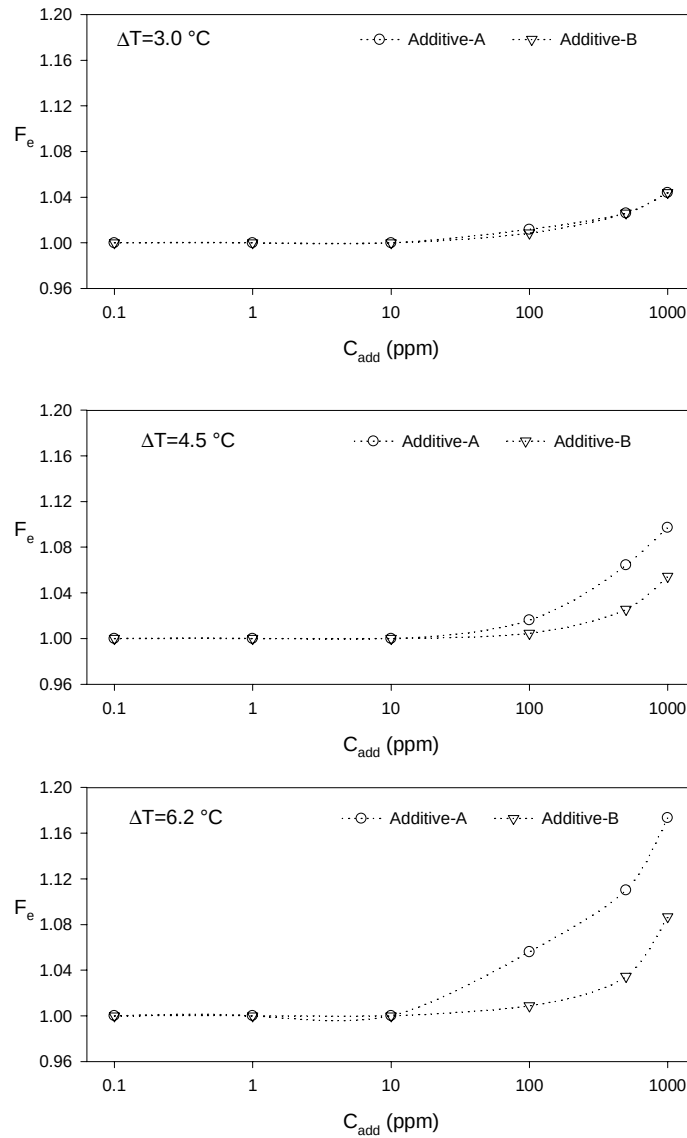


Figure 9. The effect on the heat transfer coefficient by adding heat transfer additives in terms of enhancement factor, F_e . The data were taken for $\Delta T=3.0, 4.5, 6.2$ °C, respectively. ΔT is the temperature gain in the cooling water.

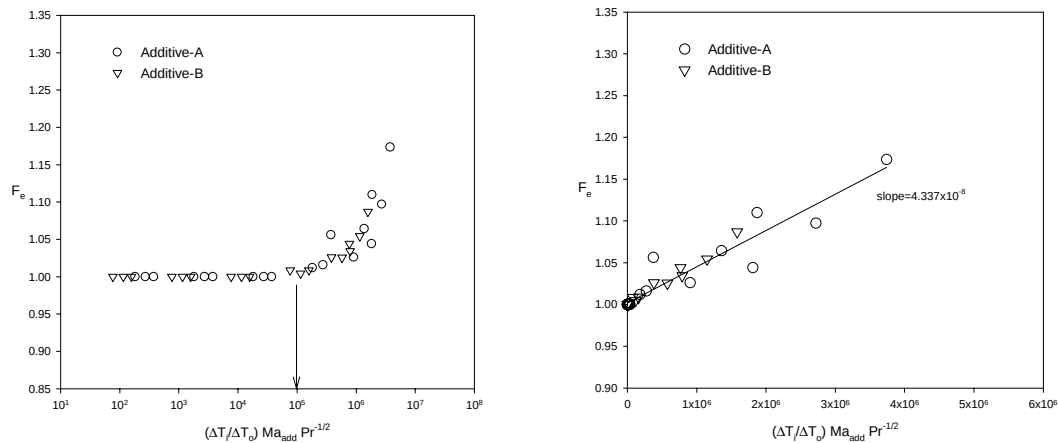
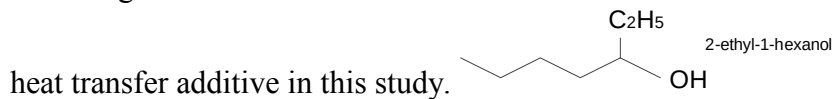


Figure 10. The correlation shows Equation (8). The x-axis was plotted based upon the log (left) and linear scale (right). Note that $R^2=0.991$. Note that the following chemical shows its structure of non-ionic surfactants tested as a



CONCLUSION

The irreversibility distribution of all components in the proposed combined triple cycle (Brayton/Rankine/Rankine)/(Gas/Steam/Ammonia) was evaluated and analyzed using both the environmental air composition and the actual combustion reaction. As expected, the irreversibility of combustion is, by far, the largest and decreases with increasing turbine inlet temperature. The irreversibility also increases with the amount of steam injected into the combustion chamber. The ammonia bottoming cycle provides a more efficient thermal matching for the triple cycle Heat Recovery Steam Generator (HRSG) than the HRSG of a conventional combined cycle. Feedwater heaters can be conveniently added to the steam cycle to increase its efficiency. The effects of the pinch point and the gas/steam approach temperature difference on the irreversibility of HRSG and the cycle were evaluated and their combined effect on the cycle efficiency was found to be small. ***The triple cycle is able to achieve 60 percent efficiency with modern gas turbines.***

Experimental studies of steam condensation to a bundle of horizontal tubes with small amounts of effective additives were successfully performed to evaluate their effect on condensation heat transfer and to identify the onset of heat transfer enhancement. ***An increase in condensation heat transfer of up to 30% was achieved by adding only 1% of 2-ethyl-1-hexanol.*** This additive was found to be more effective than alkyl primary amine. Due to interfacial turbulence, the true condensation heat transfer enhancement mechanism is caused by local variation in the surface tension. This means that it is primarily related to the additive concentration dependence of surface tension. In the presence of effective additives, the steam condensation process observed in our experiment produced a mostly *dropwise-like* condensation. The condensate water droplets were of a smaller shape than those produced without additives and became more dispersed, exhibiting much higher frequencies. These droplets, unlike traditional turbulence, start at the top portion of the condenser tubes and cover most of the tubes, hanging at the bottom in small droplet sizes. We also observed that film substrates appear to be thinner compared to those produced in the steam condensation process without additives. Overall, we believe that added surfactants play a pivotal role in improving

dropwise-like condensate activity and provide a significant benefit for condensation heat transfer.

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APPENDIXES

Appendix A: A Copy of the Paper, “Second Law Analysis and Optimization of a Combined Triple Power Cycle,” by Marrero, Lefsaker, Razani, and Kim

Appendix B-1: EES Program – Gas Turbine Cycle

Appendix B-2: EES Program - Triple Cycle

Appendix C: A Copy of the Paper, “The Effective Use of Heat Transfer Additives for Steam Condensation,” by Kim, Lefsaker, Razani, and Stone

Appendix A: A Copy of the Paper, “Second Law Analysis and Optimization of a Combined Triple Power Cycle,” by Marrero, Lefsaker, Razani, and Kim

SECOND LAW ANALYSIS AND OPTIMIZATION OF A COMBINED TRIPLE POWER CYCLE

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SECOND LAW ANALYSIS AND OPTIMIZATION OF A COMBINED TRIPLE POWER CYCLE

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Abstract

In this investigation, a combined triple (Brayton/Rankine/Rankine)/(Gas/Steam/Ammonia) power cycle is analyzed. In the triple cycle, the exhaust of the Brayton gas topping cycle is used in a Heat Recovery Steam Generator (HRSG) to produce steam for a Rankine steam middle cycle followed by an ammonia Rankine Bottoming cycle. The ammonia bottoming cycle provides a practical and more efficient hot and cold streams thermal matching for the triple cycle HRSG as compared to the HRSG of a conventional combined power cycle. Through exergy analysis of the cycle, the exergy of the exhaust streams and the irreversibility of each component in the cycle are determined using reasonably practical constraints for the system components. These constraints are mainly due to the size of components and are conveniently parameterized and analyzed. The triple cycle was analyzed and optimized with respect to important system parameters such as gas topping cycle pressure ratio, gas turbine inlet temperature, HRSG pinch point, gas/steam approach temperature difference, rate of steam injection into combustion chamber, and the effectiveness of heat exchangers. One goal of the study is to find what configuration will achieve a thermal efficiency of 60 percent when reasonably practical constraints for system parameters are used.

Keywords: Combined power cycle, Exergy analysis, Second law analysis, High efficiency power cycles.

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1. Introduction

The most efficient cycle between two thermal reservoirs is the Carnot cycle. Common cycles used in power plants are the Rankine and Brayton cycles. Given that water is widely used as the working fluid in power plants, many power plants work on a Rankine cycle. A typical power plant working on a simple Rankine cycle could be expected to have a thermal efficiency around 39 percent. Any plans to increase the efficiency of power plants beyond 50 percent would result in binary and combined cycles [1-3]. A combined cycle can achieve higher efficiencies by coupling a Brayton cycle with a bottoming Rankine cycle. This takes advantage of the fact that a Brayton cycle works at a high temperature regime while the Rankine cycle works at a low temperature one. Combined-cycle power plants have their special characteristics. For example, it has been shown that addition of feedwater heaters; in general, results in reduction of plant efficiency of combined cycles [4]. This is contrary to its effect in simple Rankine cycles. This is mainly due to an increase in stack exhaust exergy loss when feedwater heater is added to combined power cycles. Recently, primarily because of environmental considerations, high efficiency combined cycle power plants have been under intensive investigations [5-10].

In this study, we investigate a triple cycle utilizing gas (air and combustion products), water, and ammonia, as the working fluids in a Brayton/Rankine/Rankine configuration. The ammonia (Rankine) bottoming cycle is utilized to efficiently recover exhaust exergy of gases from the HRSG. In this manner a more effective heat exchange process between the ammonia bottoming cycle and exhaust gases is provided. In the triple cycle, a multi-pressure steam cycle is not used to reduce the irreversibility of HRSG [11]. Conventional feedwater heaters can be added to the steam cycle and ammonia bottoming cycle uses the exhaust exergy from HRSG. We study the combined triple cycle power plant using second-law analysis and optimize with constraints on the size of components in the system. The irreversibility of each component in the power plant will be determined to see which component contributes the most to the total irreversibility of the plant. The goal is to see how one can achieve a thermal efficiency close to 60 percent for the combined power cycles with a reasonable constraint on the size of each component [10]. Thermodynamic optimization, based on a criterion of

minimum total irreversibility of the system, is performed on selected parameters. The problem is complex because a reduction in the irreversibility of one component in the system may increase the irreversibility of other components. In addition, this optimization must be performed considering the constraints on the size of the components in the system [12,13]. The economic issues are not directly considered in this study. The emphasis is on the reduction of the irreversibility of heat transfer processes.

2. Analyses and Simulation

Exergy analysis and second law analysis and optimization of thermal systems are a well-established method in the field of thermal science and engineering and design of power systems [12-15]. In general, conservation of mass and energy and balance of exergy is written for each system component, the irreversibility of each component is determined, and the exergy entering and leaving the system is evaluated. For the triple cycle under investigation, these equations were written for each system components using EES software [16].

In the proposed triple (Brayton/Rankine/Rankine)/(Gas/Steam/Ammonia) power cycle, shown in Figure1, the exhaust gases of a Brayton gas topping cycle is used to produce superheated steam in the HRSG. This superheated steam is used in a Rankine intermediate steam cycle to produce power. Two Feedwater heaters are used in the intermediate steam cycle to increase its efficiency. Since the pressure ratio of gas topping cycle is an optimization parameter and the exhaust temperature of gases from the gas turbine is a variable, the boiler pressure in the steam cycle is evaluated using equal temperature intervals for the feedwater heaters. One feedwater heater is closed and the other is open also acting as a deaerator. Multi-pressure steam generation in HRSG or reheat is not used in the steam cycle. A condenser/evaporator (steam/ammonia) connects the Rankine steam intermediate cycle to a Rankine bottoming ammonia cycle completing the triple cycle. In this study, the steam condenser pressure is fixed at one atmosphere. An ammonia superheater uses the exhaust gases from the HRSG to superheat ammonia before expansion in an ammonia turbine. The ammonia bottoming cycle provides a practical and more efficient hot and cold stream thermal matching for the triple cycle HRSG as compared to the HRSG of a conventional Brayton/Rankine combined power

cycle. An ammonia preheater uses the exhaust exergy of ammonia turbine to increase the exergy of ammonia before entering the ammonia evaporator. The bottoming cycle is connected to the environment by heat rejection through an ammonia condenser. The composition of the inlet air to the gas topping cycle is assumed to consist of %77.48 N₂, %20.59 O₂, %1.90 H₂O, %0.03 CO₂. The mass flow rate of the fuel is set at 1 kg/sec and is assumed to be natural gas with an equivalent chemical formula of CH_{3.88}. The chemical exergy and physical exergy for inlet and exit streams for each component are determined based on the reference environment mentioned above [12,15]. The fuel is assumed to enter the system at the pressure and the temperature of the environment. Therefore, a fuel pump is used to increase the pressure of the fuel to the operational pressure of the combustion chamber.

In general, overall thermal efficiency is defined by,

$$\eta_a = \frac{\text{Net Energy Rate of the Product}}{\text{Rate of input energy}} \quad (1)$$

The product in the above cycle is the net power produced divided by the product of the Lower Heating Value (LHV) of the fuel and its mass flow rate (fixed at 1 Kg/s). The second law efficiency in this study is defined by,

$$\eta_b = \frac{\text{Net Exergy Rate of the Product}}{\text{Net Exergy Rate of Input}} \quad (2)$$

The net exergy rate of the product is the net power produced, and the net exergy rate of input is calculated by subtracting the exergy rate of all exit streams from the exergy rate of all input streams. In this investigation, the input streams are taken to be the fuel and steam spray into the combustion chamber, and the exit streams are the combustion products from the stack exhaust gases and air from the intercooler. The Brayton gas topping cycle was analyzed first. In this analysis, the concentration is on the combustion irreversibility and the effect of steam injection into combustion chamber on its irreversibility. When steam is injected into combustion chamber, the partial pressure of water vapor in the combustion products increases. Care must be taken in the calculation of physical and chemical exergies of combustion products [12,17]. The topping cycle is then connected to the steam intermediate and ammonia bottoming cycle to construct the triple cycle.

3. Results and Discussion

The results are presented in two sections. First the Brayton gas topping cycle is discussed and then the triple cycle is presented.

3.1 Brayton Gas Topping Cycle

The topping cycle consists of two compressors with an intercooler between them. The cycle also includes a combustion chamber, a turbine, and a recuperator. The topping cycle configuration is given in Figure 2. The cycle has the turbine inlet temperature (TIT) and the mass flow rate of steam spray into the combustion chamber as the important parameters for analysis. Table 1 gives the nominal values for parameters used in this study.

3.1.1 The effect of adding spray water vapor on combustion chamber irreversibility.

One goal of the analysis of topping cycle is to evaluate the effect of steam injection into combustion chamber on its irreversibility and the irreversibility of other components in the cycle. Figure 3 to 5 show the effect of spray water on combustion irreversibility for three values of TIT, three values of pressure ratio, and three different amounts of spray water vapor added (series 1 to 3). Figure 3 shows that the irreversibility of the combustion chamber increases with increasing amounts of spray water added for fixed values of TIT. As expected, the figure also shows that the irreversibility of combustion chamber decreases for higher values of TIT. Figure 4 shows the same result as Figure 3 for a different value of pressure ratio. The irreversibility of the combustion chamber increases with increasing amount of steam injection for all values of TIT. Comparing Figures 3 and 4, the corresponding irreversibilities are almost the same. Figure 5 shows the same results as Figures 3 and 4 for a different value of pressure ratio. For a pressure-ratio of 30 and TIT = 1400 K, the temperature of combustion gases at the turbine exit is so low that the recuperator has a negative irreversibility. In such cases the recuperator should not be used. Figures 3, 4 and 5 show that the irreversibility of the

combustion chamber increases with increasing amounts of spray water added, and that the irreversibility of the combustion chamber is almost independent of the pressure-ratio.

The rate of fuel entering the system is fixed at 1 Kg/s and the turbine inlet temperature is an important parameter of the study. The steam spray entering the combustion chamber is assumed to be saturated steam corresponding to the operating pressure of the combustion chamber. Therefore, for a given TIT, the conservation of energy on the combustion chamber gives the mass flow rate of air entering the combustion chamber. Since in these calculations the TIT and the mass flow rate of fuel are fixed, the mass flow rate of air is a floating variable. Therefore, in each calculation, the amount of oxidant in air must be checked to make sure enough excess air is available for complete combustion. The exergy associated with the spray water is an important quantity that influences the irreversibility of combustion chamber and the second law efficiency of the cycle is given by Equation 2.

3.1.2 Irreversibility distribution of the cycle components

A major goal of this study is to investigate the irreversibility distribution of cycle components. The irreversibility distribution is only represented for three values of TIT and pressure ratios selected close to the point where the cycle efficiencies are maximum. The points selected are: TIT=1400 K and pressure ratio=20, TIT=1600 K and pressure ratio=35, TIT=1800 K and pressure ratio=40. Figure 6 to 8 show the irreversibility distribution for two different values of spray water vapor added to combustion chamber, zero Kg/s (series 1) and 4 kg/s (series 2). Figure 6 shows that for increasing amounts of spray water added, the irreversibility of each component decreases except for the combustion chamber. This occurs when spray water is added and the mass flow rate of the working fluid decreases to maintain the energy balance in the combustion chamber. When the mass flow rate through the components decreases, there is an almost proportional decrease in irreversibility. For the combustion chamber, the exergy added by spray water vapor is larger than net exergy change of air going through the combustion chamber. The combustion irreversibility together with the exhaust exergies from recuperator and intercooler are also given in Figure 6, for zero and four kg/s spray

water vapor, for comparison. The exergy exhaust from the recuperator is large and is partially recovered by the bottoming cycle that will be studied in the next section. The irreversibility distribution for TIT of 1600 K and 1800 K is given in Figure 7 and 8, respectively. The pressure-ratios of 35 and 40 were selected to show the results for these two cases, respectively. As expected, a comparison of Figures 6,7, and 8, shows that combustion irreversibility decreases with an increase in TIT. The exit exergy from exhaust of the recuperator also increases in most cases when TIT increases. This shows that not only does the efficiency of the topping cycle increase with TIT, but there is also in general more opportunity to recover the exhaust exergy in the bottoming cycle. It should be pointed out that different pressure-ratios are used in these calculations and TIT is not the only significant parameter in the analysis of this cycle. As expected, the inlet exergy of water vapor added increases when the combustion chamber operates at higher pressure (higher pressure-ratio). The variation of the irreversibility of the recuperator is interesting. When water is added, the mass flow rates of hot and cold streams are different and the recuperator will be more unbalanced. There is an irreversibility associated with the imbalance in the recuperator heat exchanger [11]. In addition, because of changes in the pressure ratio and TIT, the inlet and exit temperatures and pressure also vary. Therefore, despite equal heat exchanger effectiveness used in all the calculations represented in Figures 6, 7 and 8, a specific trend in the irreversibility of the recuperator cannot be deduced.

3.2 Analysis of the Triple Cycle.

The flow diagram of the combined triple cycle is given in Figure 1. Nominal values of different parameters used in the analysis of the triple cycle are listed in the Table 2. In order to analyze the triple cycle under a practical condition, the temperature differences for the most important heat exchangers are given in Table 2 and are defined in nomenclature.

3.2.1 The effect of TIT, pressure-ratio, compressor and turbine efficiencies on the triple cycle performance.

The nominal values of efficiencies of the compressors and turbines were set to 0.9. The effect of pressure ratio on the second law efficiency for three values of TIT and two values of compressor and turbine efficiency of 0.85 and 0.9 are given in figure 9. As expected, Figure 9 shows that the second law efficiency goes down by approximately 2.5 percent when the turbines and compressor efficiency is set down from 0.9 to 0.85. All the six graphs have an optimum for the efficiency with respect to the pressure-ratio. It is interesting to note that the efficiency versus pressure ratio lines are flat and do not have sharp optimum. This is important when other constraints, such as cost and environmental issues, must be considered in the design of the system. It is noticeable that when the compressor and turbine efficiencies are set down to 0.85, the optimum occurs at lower pressure-ratios than for the value 0.9. As expected, when TIT increases, the second law efficiency increases. Figure 9 is for 0 kg/s spray water added. When spray water is added, second law efficiency continuously increases and an optimum is not observed in the range of the pressure ratio studied. A typical second law efficiency as a function of pressure ratio for TIT of 1800 K, turbine and compressor efficiencies of 0.9, and 4 Kg/s of steam spray is given in Figure 10. Comparison of Figures 9 and 10 shows that second law efficiency has a lower value throughout the interval of pressure ratio when steam is injected into the combustion chamber. Besides the effect of pressure-ratio there are also several other important design parameters that should be studied. Some of these parameters are:

- a- The superheat temperature for the water cycle.
- b- The boiling pressure for the steam cycle.
- c- The superheat temperature for the ammonia cycle.
- d- The pressure (or saturation temperature) for the two feedwater heaters.

In this study, the boiling pressure in steam cycle is not fixed and the highest temperature of the cycle (exit of superheater) is related to the exhaust temperature of gas turbine by a given approach with the temperature difference parameter DELTAT. One open and one closed feedwater heater are selected at equal temperature intervals in the

cycle between the boiler and the fixed pressure steam condenser (1 atm). A pinch temperature difference in HRSG also constrains the boiler pressure. Parametric studies have shown that the second law efficiency is not very sensitive to the position of feedwater heaters in the cycle.

3.2.2 The effect of pinch and approach temperature of HRSG and gas recuperator effectiveness on system performance.

The effect of recuperator effectiveness on second law efficiency was found to be small. Reducing the recuperator effectiveness from 0.95 to 0.85 resulted in almost no change in second law efficiency over a wide range of pressure ratio. In this case the increase in irreversibility of the recuperator is balanced with a reduction in the irreversibility of other components or the exhaust exergies. This is important because a reduction in the recuperator effectiveness corresponds to a reduction in its size and cost. For different values of approach temperature difference in steam cycle superheater (DELTA_T), the effect of pressure ratio on second law efficiency is shown in Figure 11. When DELTA_T is increased from 10 K to 90 K, the boiling pressure in the steam cycle drops from 47.88 bars to 24.23 bars. The heat transfer area needed in HRSG depends on DELTA_T and its capital cost decreases as DELTA_T increases. It is shown in Figure 11 that the difference in the second law efficiency for DELTA_T = 10 K and DELTA_T = 90 K is small. In the calculations, the pinch temperature difference is fixed at 10 K. Figure 12 shows the irreversibility of HRSG and the second law efficiency as a function of DELTA_T for the nominal setting.

Figure 12 also shows that the irreversibility of the Heat Recovery Steam Generator (HRSG) is doubled from 400 to 800 KW when DELTA_T increases from 10 K to 90 K. The second law efficiency decreases by less than one percent for the same DELTA_T interval. When DELTA_T increases, the power output for the middle cycle is reduced. On the other hand, the mass flow rate of ammonia in the bottoming cycle will increase resulting in an increase in the power output of the ammonia bottoming cycle. When the power output from the middle cycle and the bottoming cycle is added up,

variation in DELTAT has a small effect on the total power output and the second law efficiency as shown in Figure 11. It should be pointed out that the exergy of exhaust from the system also changes. For the nominal setting, most of the power is produced by the gas turbine of the topping cycle (approximately 80 percent). The remaining power is produced by the steam turbine (approximately 12 percent) and the ammonia turbine (approximately 8 percent). The effect of pinch temperature difference on HRSG irreversibility and second law efficiency is given on Figures 13 and 14, respectively. These figures are presented for three values of TIT and the pressure ratios corresponding to nearly optimum second law efficiency of Figure 9. These values correspond to TIT=1400 K and pressure ratio=16, TIT=1600 K and pressure-ratio =25, and TIT=1800 K and pressure-ratio=30.

Figure 13 shows that the effect on the HRSG irreversibility when T_{pinch} value is increased from 10 K to 20 K is not significant. The difference in HRSG irreversibility for T_{pinch}=10 and T_{pinch}=20 is increasing with increasing TIT. Since the variation in the HRSG irreversibility is very small, the variation in the second law efficiency for different T_{pinch} should also be small as shown in Figure 14. To see the effect of varying DELTAT and T_{pinch} on the whole cycle when it comes to the component irreversibility, two studies were performed where the irreversibility of each component is calculated for nominal settings for two different values of DELTAT = 10K and T_{pinch} = 10 K and DELTAT = 90 and T_{pinch} = 20. The results are shown in Figures 15 and 16, respectively. The irreversibility of combustion, intercooler and the exergy of exhaust gases are also given in the figures for comparison. Except for the combustion chamber, it is the turbine in the gas cycle that has the highest irreversibility. Figure 15 shows that the HRSG irreversibility is increased from 393 KW to 822 KW because of the increasing values for T_{pinch} and DELTAT. It is interesting to note that the exergy out by the exhaust (stack) is lower for DELTAT=90 and T_{pinch}=20 than for DELTAT=10 K and T_{pinch}=10 K. This should mean higher efficiency, but the total irreversibility of the triple cycle is higher for DELTAT=90 K and T_{pinch}=20 K than for the triple cycle with DELTAT=10 K and T_{pinch}=10 K. The difference in the second law efficiency for these two cases is small. The second law efficiency for the values in Figure 15 is 56.9 % and the second law efficiency for the values in Figure 16 is 56.3%. It should be also pointed

out that the ammonia bottoming cycle uses the exhaust exergy from HRSG. Therefore, higher exhaust exergy from HRSG is not necessarily lost.

3.2.3 The effect of environment on efficiency.

The effect of environmental temperature (T_o) on efficiency is also interesting. This parameter is more likely to vary from plant to plant, or more precisely from location to location. Figure 17 shows the effect of varying T_o on the second law efficiency of the triple cycle. The figure shows that the second law efficiency is increasing with decreasing T_o . The effect of going from 298 °K (25°C) down to 288 °K (15°C) is almost 1 percent. It is interesting that the graph in Figure 17 is very close to linear. The stack gas exhaust temperature influences the efficiency of the plant. For lower stack exhaust temperature, less thermal exergy is lost to the environment and power plant is producing more power. Recently developed fossil-fuel power plants (in particular the combined cycle gas turbine plant, CCGT) employ quite low gas exhaust temperature (less than 100 °C) thus reducing the loss of thermal exergy in the stack exhaust. In such cases, the loss of exergy associated with a failure to expand the product to atmospheric partial pressure (virtually impossible in practice) becomes comparable with the loss of thermal exergy [17]. For the triple cycle used in this study, the stack exhaust gas temperature is approximately 170 °C for the nominal setting. When the cycle is optimized, the exhaust gas temperature goes down and the loss of exergy is reduced. Even for the optimized cycle, the exhaust temperature of gases can be reduced by lowering the pressure (or temperature) of steam condenser/ammonia evaporator. As mentioned before, the steam condenser pressure is fixed in this study. The exhaust gas temperature can also be used to preheat the fuel or supply part of the exergy of steam injected into the combustion chamber. Finally, we should point out that the ammonia bottoming cycle may have an advantage over the conventional steam cycle when cooling water is scarce and an air-cooled condenser must be used.

4. Conclusions

The irreversibility distribution of all components in the proposed combined triple (Brayton/Rankine/Rankine)/(Gas/Steam/Ammonia) cycle was evaluated and analyzed. As expected, the irreversibility of combustion is by far the largest and decreases with increasing Turbine Inlet Temperature (TIT). The irreversibility also increases with the amount of steam injected into the chamber. Ammonia bottoming cycle provides a more efficient thermal matching for the triple cycle Heat Recover Steam Generator (HRSG) than the HRSG of a conventional combined cycle. Feedwater heaters can be conveniently added to the steam cycle to increase its efficiency. The effects of the pinch point and the gas/steam approach temperature difference on the irreversibility of HRSG and the cycle were evaluated and their effect on the cycle efficiency was found to be small. The triple cycle is able to achieve 60 percent efficiency with modern gas turbines. Further increases in efficiency are possible by further optimization, but it would be marginal. The steam condenser used in the triple cycle is more compact and the ammonia condenser can be advantageous when an air-cooled condenser is used.

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Nomenclature

CCGT	Combined-Cycle Gas Turbine
CCPP	Combined-Cycle Power Plant
Comp1	First compressor
Comp2	Second compressor
Con	Ammonia condenser
DELTA _T	Gas/steam temperature difference
DEL _{NH3}	Gas/ammonia temperature difference
DEL _{FWH}	Feedwater temperature difference
DELTA _T _evap	Evaporator steam-ammonia temperature difference
DELTA _{Tc}	Temperature difference between outlet ammonia condenser and To
Evap	Steam condenser/ammonia evaporator
FP	Fuel pump
Fwh1	First feedwater heater
Fwh2	Second feedwater heater
Ex_cao	Exhaust exergy from intercooler
Ex_exhaust	Exergy of stack exhaust gases
Ex_H2O	Exergy of spray water
Ex_rec	Exergy of recuperator exhaust
HRSG	Heat recovery steam generator
I _{comb}	Irreversibility of combustion
Int	Intercooler
I _{tot}	Total irreversibility of a given power cycle
P1	Pump1
P2	Pump2
P3	Pump3
Preh	Ammonia preheater
Rec	Recuperator
Sup	Superheater
T1	Turbine 1
T2	Turbine 2
T3	Turbine 3
T ₀	Ambient temperature
TIT	Gas Turbine Inlet Temperature
T _{pinch}	Pinch point temperature difference
W _{in}	Input power
W _{out}	Output power
η _a	Thermal efficiency, Eq. (1)
η _b	Second law efficiency, Eq. (2)

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Table 1: Nominal settings for the topping cycle.

Parameter	Value	Units
Gas turbine inlet temperature	1600	K
Pressure-ratio	25	Dimensionless
Mass flow rate of fuel	1	Kg/s
Environmental temperature	298	K
Mass flow rate of spray water	0	Kg/s
Input energy	4200	KW
Input exergy	43680	KW
Heat exchanger effectiveness	0.9	Dimensionless
Turbine efficiency	0.9	Dimensionless
Pump efficiency	0.9	Dimensionless
Compressor efficiency	0.9	Dimensionless

Table 2: The nominal values of different parameters in the triple cycle.

Parameter	Value	Units
Gas turbine inlet temperature	1600	K
Pressure Ratio	25	Dimensionless
Mass flow rate of fuel ($\text{CH}_{3.884}$)	1	Kg/s
Environmental temperature	298	K
MH2Ospray	0	Kg/s
DELTAT	30	K
Tpinch	10	K
DELNH3	20	K
DELFWH	5	K
DELTAT_evap	4	K
DELTAT_ph	2	K
DELTATc	5	K
Input exergy	43680	KW
Input energy	42000	KW
Heat exchanger effectiveness	0.9	Dimensionless
Pump efficiency	0.7	Dimensionless
Turbine efficiency	0.9	Dimensionless
Compressor efficiency	0.9	Dimensionless

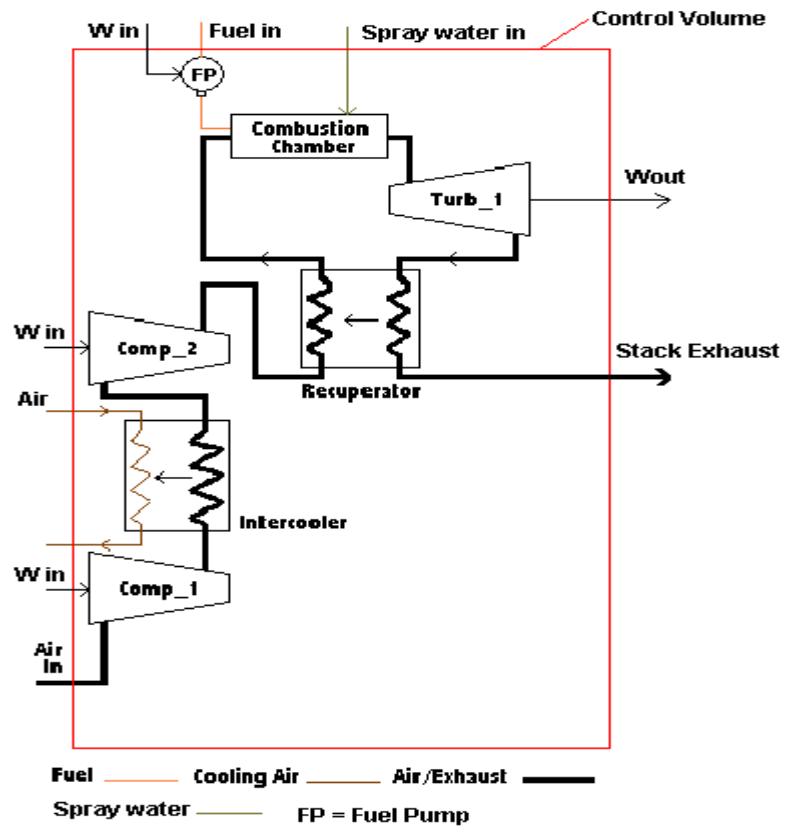


Fig. 2: The topping cycle configuration.

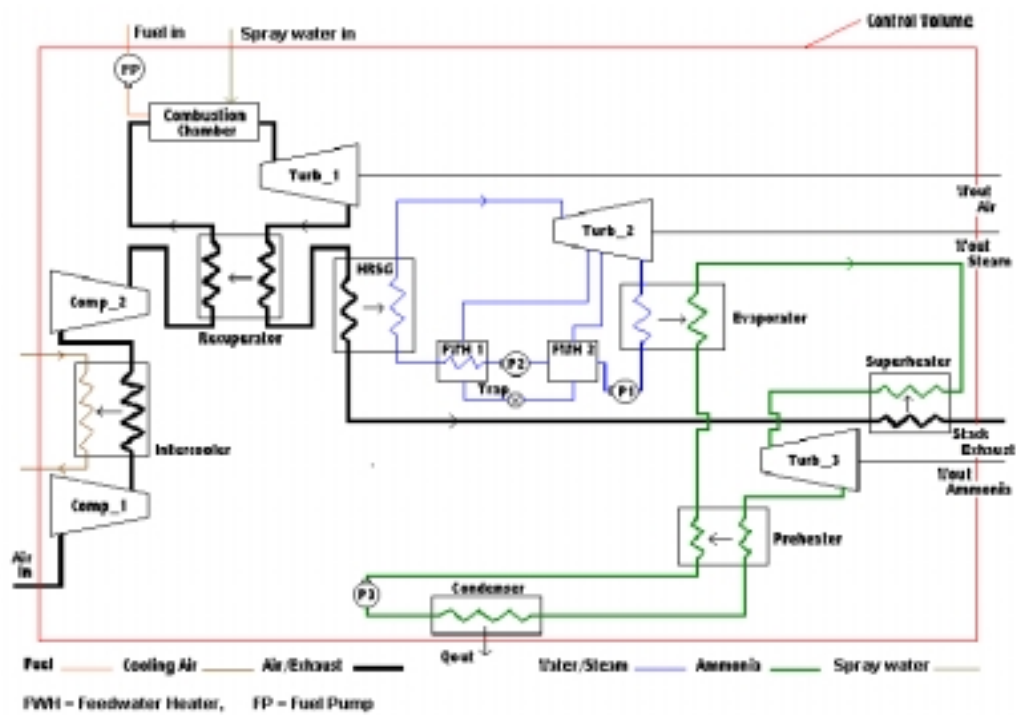


Fig. 1. Flow diagram of the triple cycle.

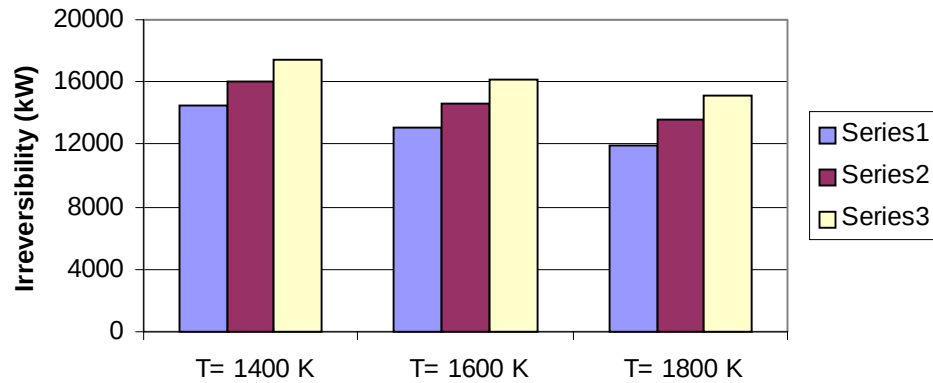


Fig. 3. The irreversibility of the combustion chamber for different values of TIT with pressure-ratio =10. Series 1 is for 0 Kg/s spray water added. Series 2 is for 2 kg/s spray water added. Series 3 is for 4 kg/s spray water added.

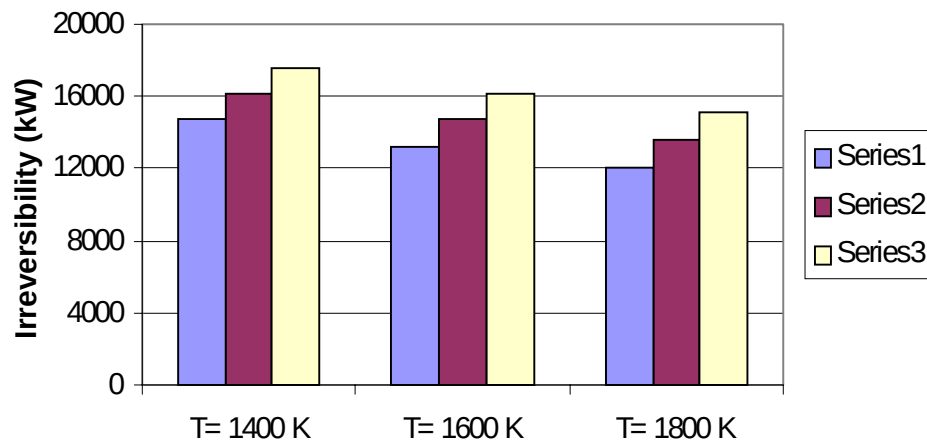


Fig. 4. The irreversibility of the combustion chamber for different values of TIT with pressure-ratio =20. Series 1 is for 0 kg/s spray water added. Series 2 is for 2 kg/s spray water added. Series 3 is for 4 kg/s spray water added.

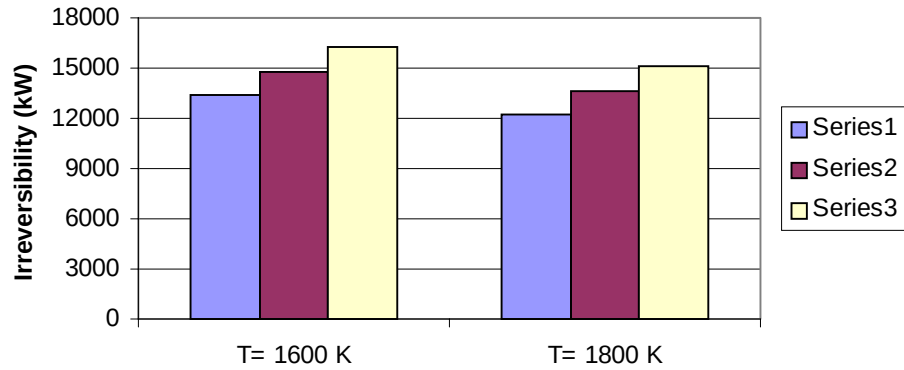


Fig.5. The irreversibility of the combustion chamber for different values of TIT with pressure- ratio =30. Series 1 is for 0 kg/s spray water added. Series 2 is for 2 kg/s spray water added. Series 3 is for 4 kg/s spray water added.

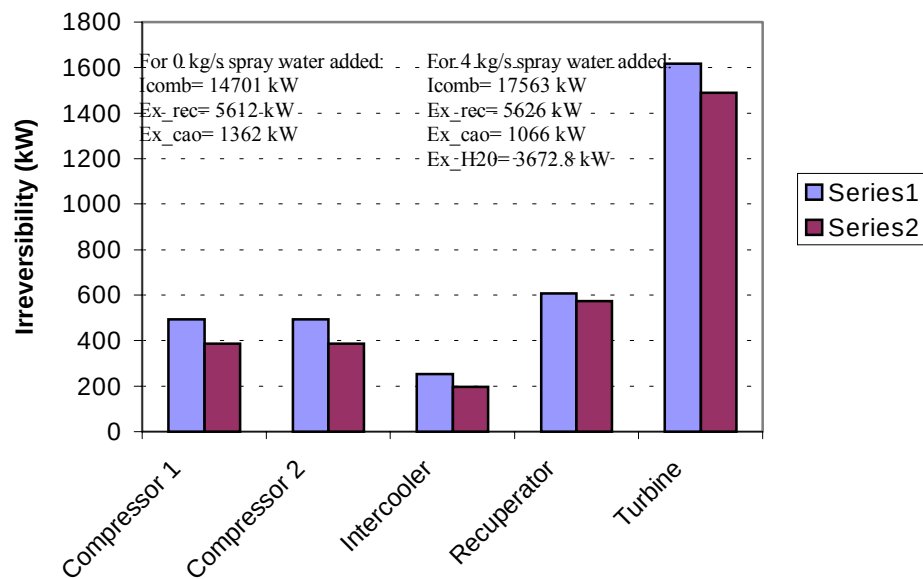
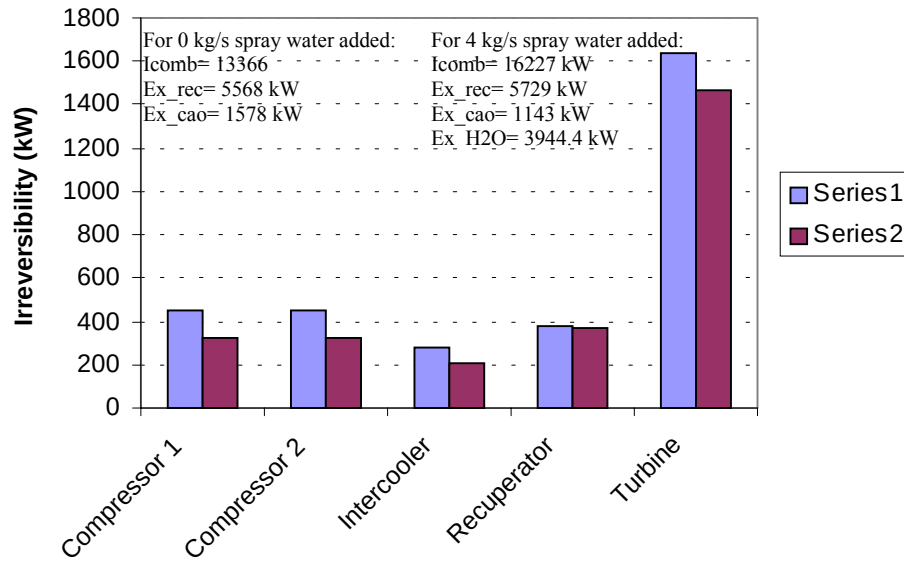


Fig. 6. The Irreversibility of the different components for TIT =1400 K and pressure-ratio =20. Series 1 is for 0 kg/s spray water added. Series 2 is for 4 kg/s spray water added.

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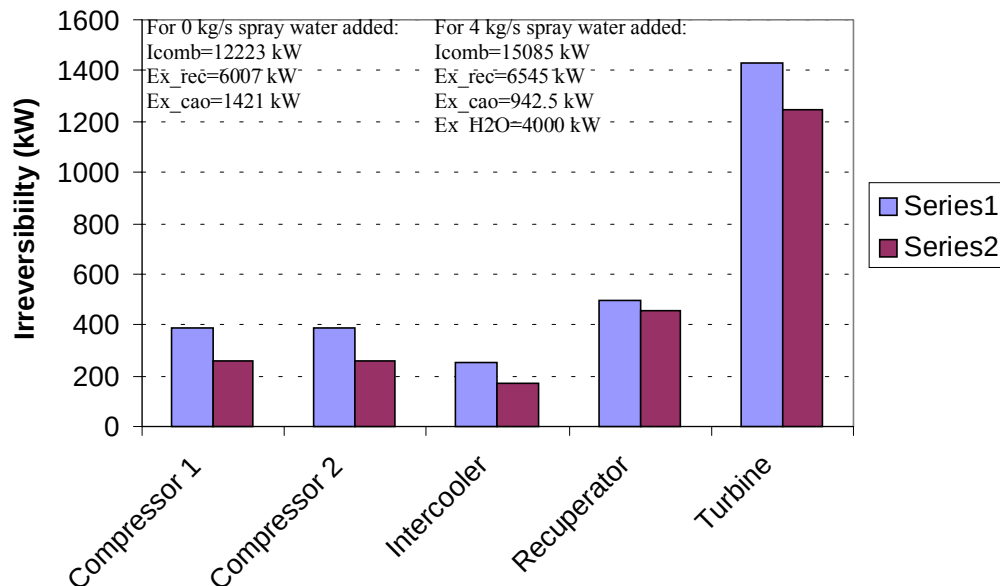


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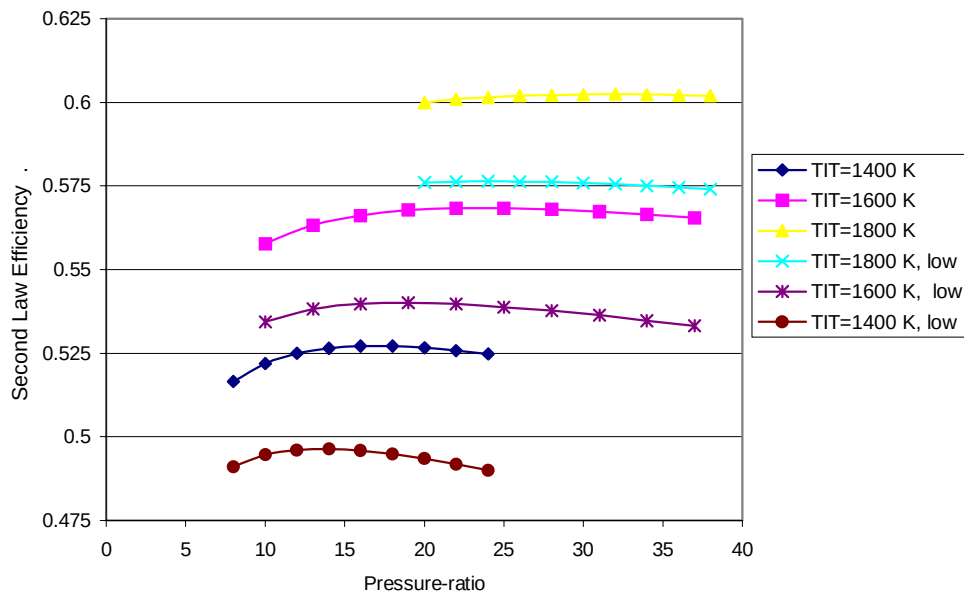


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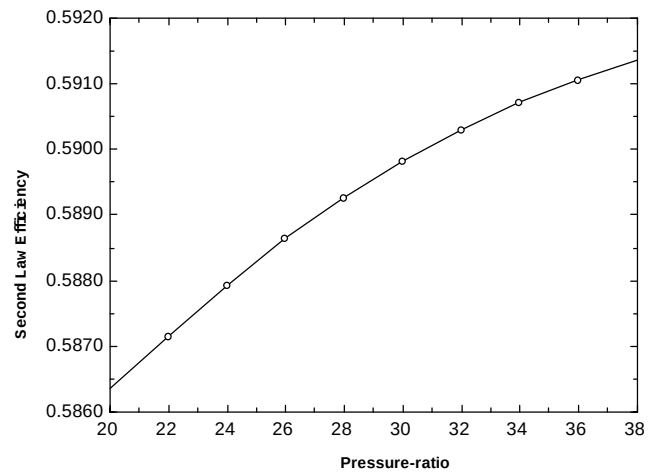


Fig. 10. Second law efficiency versus pressure-ratio for TIT=1800 K and 4 kg/s spray water added.

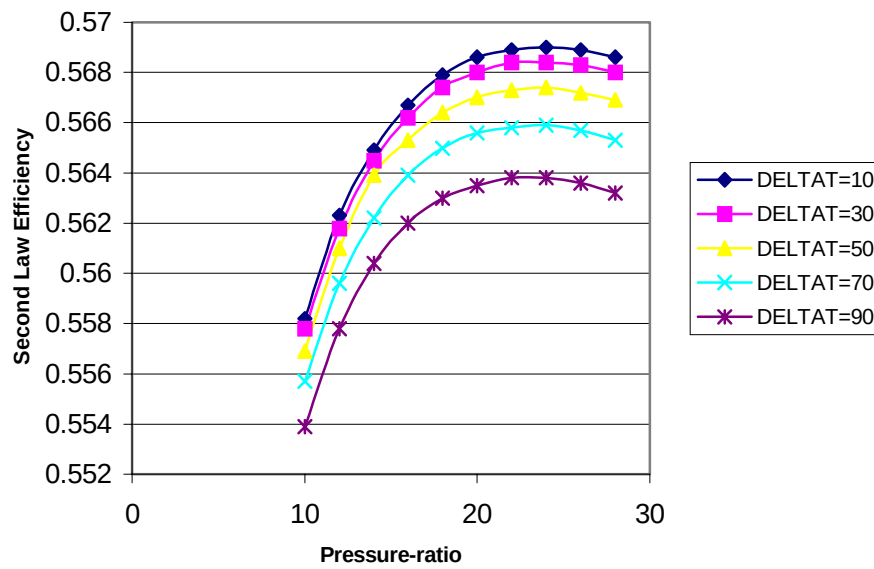


Fig. 11. Second law efficiency versus pressure-ratio for different DELTAT (Nominal setting).

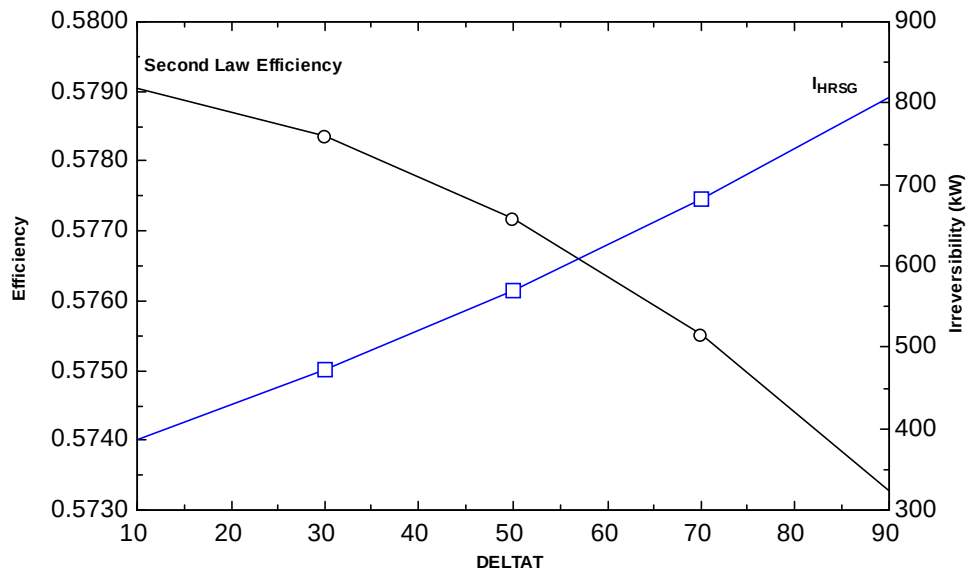


Fig. 12. Second law efficiency and the irreversibility of the Heat Recovery Steam Generator (HRSG) versus DELTAT (nominal settings).

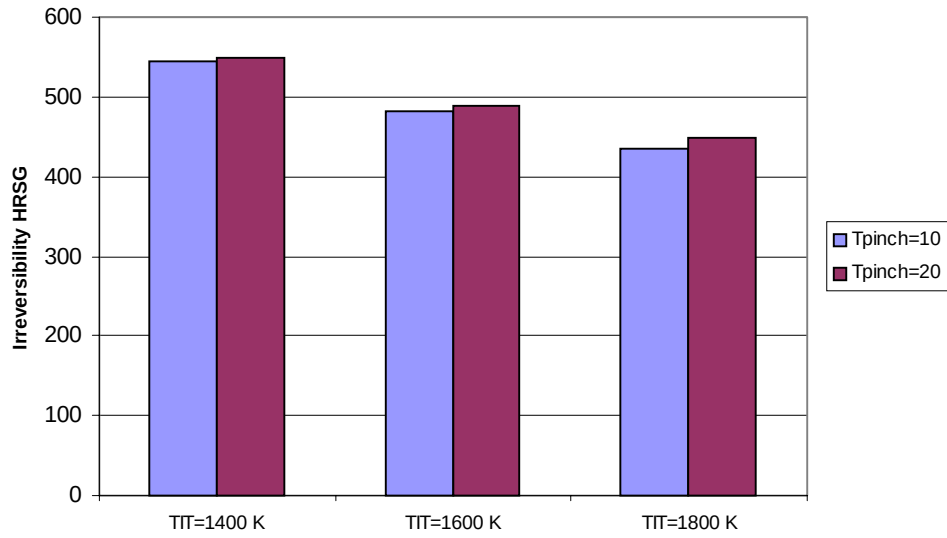


Fig. 13. The HRSG irreversibility for different values of Tpinch and TIT.

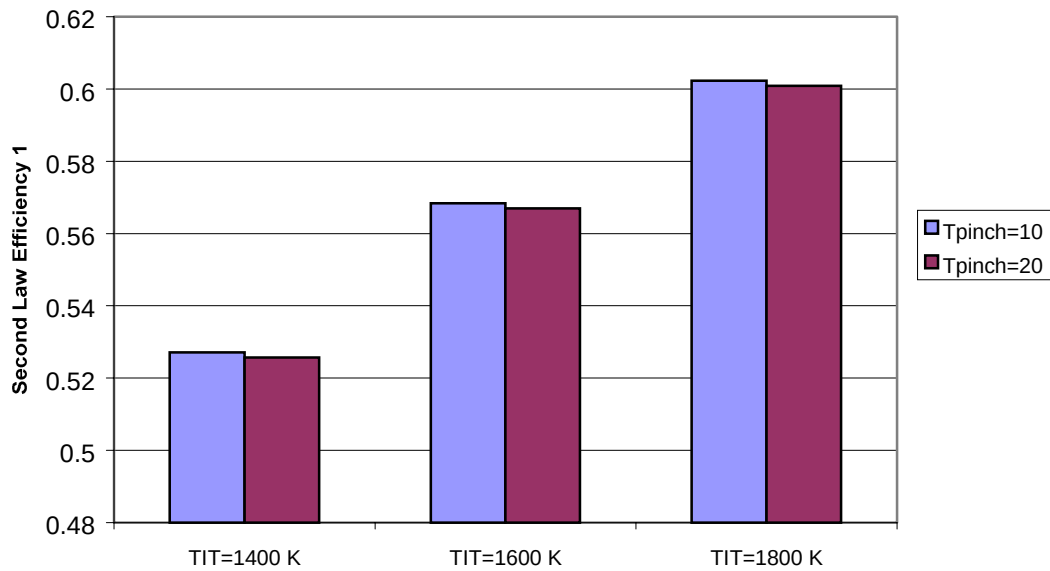


Fig. 14. The second law efficiency for different values of TIT and Tpinch.

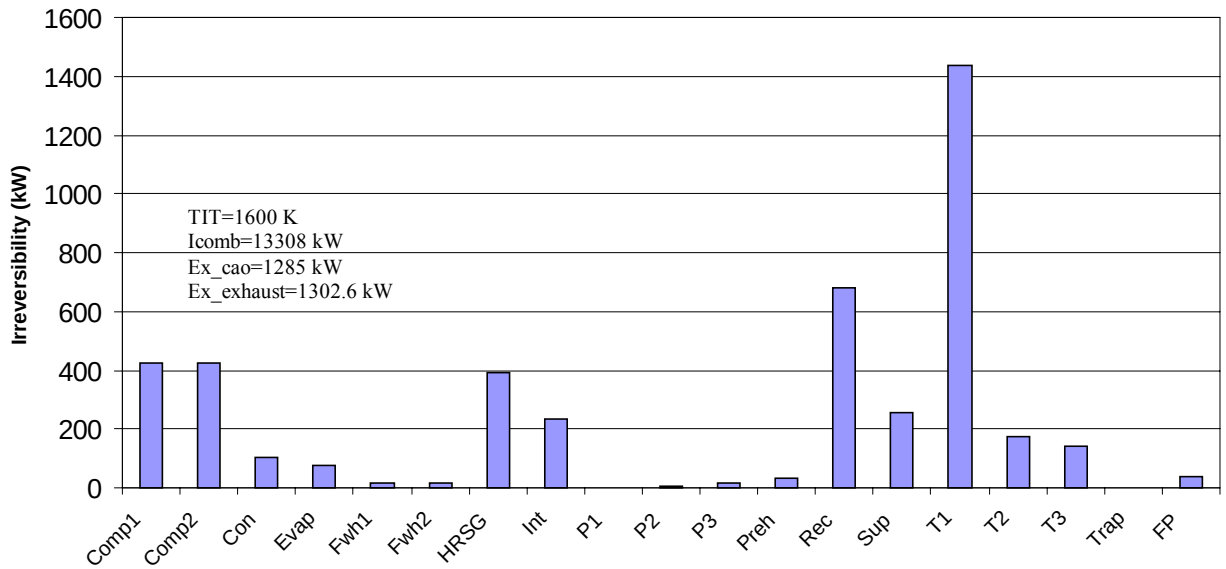
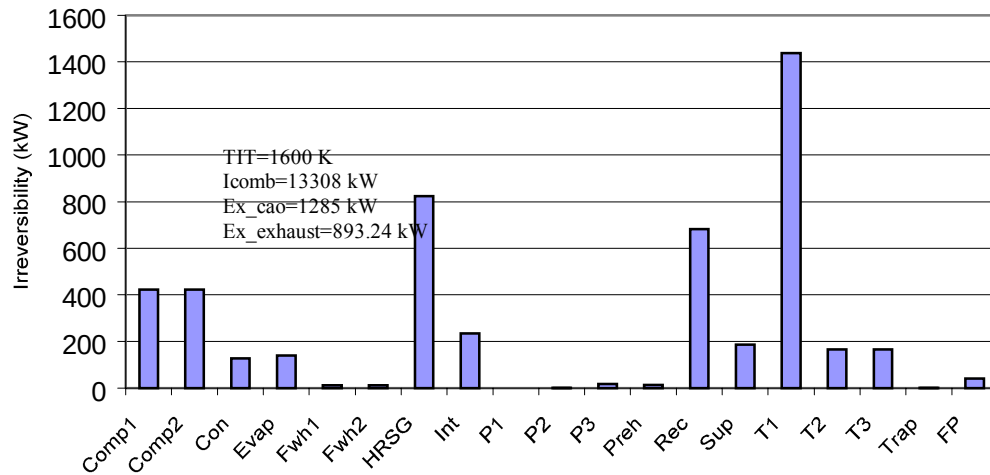


Fig. 15. Component irreversibility for the triple cycle with $T_{pinch}=10\text{ }^{\circ}\text{C}$ and $\Delta T_{AT}=10\text{ }^{\circ}\text{C}$. Nominal settings, where; Comp1= compressor 1, Comp2= compressor 2, Con= condenser, Evap= evaporator, Fwh1= feed water heater 1, Fwh2= feed water heater 2, HRSG= heat recovery steam generator, Int= intercooler, P1= pump 1, P2= pump 2, P3= pump 3, Preh= preheater, Rec= recuperator, Sup= superheater,



T1= turbine 1, T2= turbine 2, T3= turbine 3, Trap= trap/valve, FP= fuel pump

Fig. 16. Component irreversibility for the triple cycle for $T_{pinch}=20\text{ K}$ and $\Delta T_{AT}=90\text{ K}$.

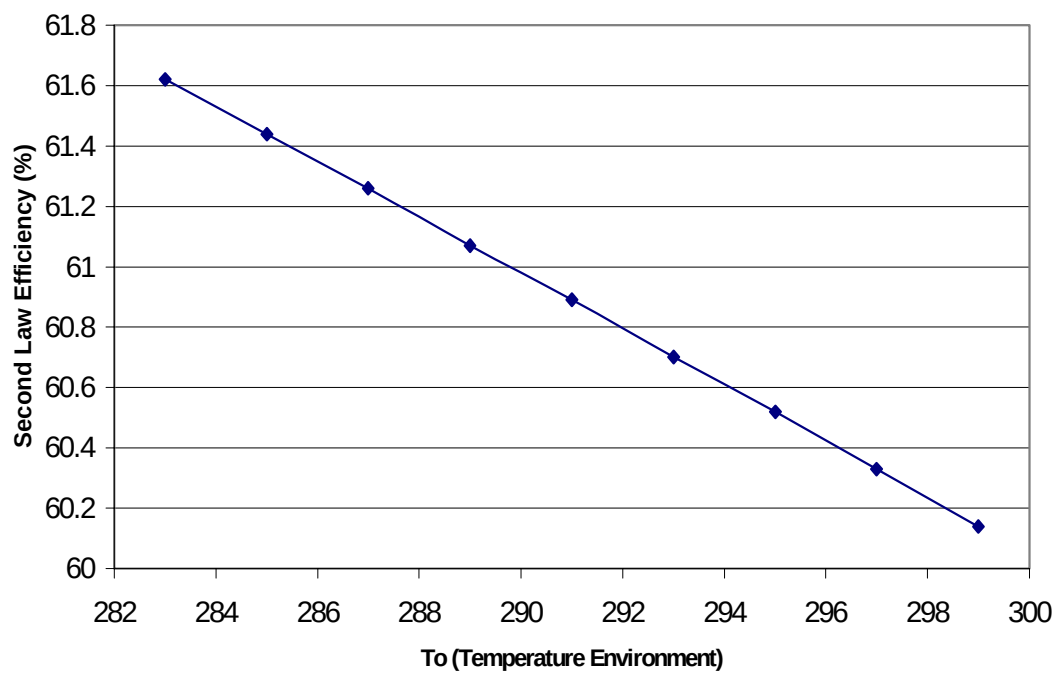


Fig. 17. The effect of varying the environment temperature (T_o) on the second law efficiency.

Appendix B-1: EES Program – Gas Turbine Cycle

"EES model of Gas-Cycle (Air) Power Plant"
 "Gas Cycle "

" _____ Input Parameters _____ "

epsilon_IC = .9
 hcf = 42000 "Natural Gas"
 ex_f = 43680
 ma2 = ma1 + mf1+mH2Ospray
 mca = ma1
 mf1 = 1
 Rf = .2870
 r_p = 30
 RIT=1800
 To = 298
 mH2Ospray=0

" _____ Adiabatic Efficiencies _____ "

eta_c1 = .9
 eta_c2 = .9
 eta_t1 = .9
 eta_CH4= .9
 eta_rec = 0.9

" _____ Calculating the AF _____ "

"62.93 CH₃.884+ a(2 O₂+7.526 N₂+0.1846 H₂O) + k H₂O = b CO₂ + c H₂O + d N₂ + e O₂"

b=62.93
 2*c=244.42 +2*k+2*a*0.1846
 2*d=a*15.052
 2*b+c+2*e=4*a + k+a*0.1846

R=hcf*15.884*mf1-Wm*mf1*15.884

z=b*44.01*ENTHALPY(CO₂,T=298)+c*18.02*ENTHALPY(H₂O,T=298)

airH₂O=a*3.326*(ENTHALPY(H₂O,T=T2ro)-ENTHALPY(H₂O,T=To))

62.93*R+a*64*ENTHALPY(O₂,T=T2ro)+210.64*a*ENTHALPY(N₂,T=T2ro)+airH₂O+18.02*k*(ENTHALPY(Steam_NBS,x=1,P=P2ro)-
 ENTHALPY(Steam_NBS,x=1,P=P1))=b*44.01*ENTHALPY(CO₂,T=T3)+c*18.02*ENTHALPY(H₂O,T=T3)+d*28.01*
 ENTHALPY(N₂,T=T3)+e*32*ENTHALPY(O₂,T=T3)-z

"b = numbre of CO₂ mol, c = number of H₂O mol, d = number of N₂ mol out, e = number of O₂ out mol"
 "f = number of O₂ mol inn, g = number of N₂ mol inn, k = number of spray H₂O, h=number of H₂O mol inn with air "

h=a*0.1846
 k=(mH2Ospray*1000)/18.02
 f = 2*a
 g = 7.52*a
 molinn = f+g+k+h
 molout = b+c+d+e

" Total mass of air =ma1"

ma1=a*(64+210.8+3.326)/1000

" Mass of the different components leaving combustion chamber"

```
mCO2 = (b*44.01)/1000
mO2 = (e*32)/1000
mH2O = (c*18.02)/1000
mN2 = (d*28.01)/1000
mO2inn = (a*64)/1000
mN2inn = (210.64*a)/1000
```

" Total mass leaving combustion chamber"

```
m2check=mCO2+mO2+mH2O+mN2
```

" ___Molfraction of gasses leaving combustion chamber___ "

```
yN2 = d/molout
yO2 = e/molout
yCO2 = b/molout
yH2O = c/molout
```

" _____Environment_____ "

```
xN2 = 0.7748
xO2 = 0.2059
xCO2 = 0.0003
xH2O = 0.019
```

"Chemical exergy "

```
Exch_N2 = 639
Exch_O2 = 3951
Exch_CO2 = 14176
Exch_H2Og = 8636
Exch_H2Ol = 45
```

"Molecular weight of air"

```
w_a=xN2*28.01+xO2*32+xCO2*44.01+xH2O*18.02
```

" _____Topping Cycle (Brayton) Air States_____ "

"State 1"

```
T1 = To
P1 = 1.013
h1 = (xCO2*44.01*(ENTHALPY(CO2,T=To)-ENTHALPY(CO2,T=To))) + xN2*28.01*(ENTHALPY(N2,T=To)-
ENTHALPY(N2,T=To)) + xH2O*18.02*(ENTHALPY(H2O,T=To)-ENTHALPY(H2O,T=To)) + xO2*32*(
ENTHALPY(O2,T=To)-ENTHALPY(O2,T=To))/w_a
s1 =
(xCO2*44.01*ENTROPY(CO2,T=To,P=(P1*xCO2))+xH2O*18.02*ENTROPY(H2O,T=To,P=(P1*xH2O))+xN2*28.01*
ENTROPY(N2,T=To,P=(P1*xN2)) + xO2*32*ENTROPY(O2,T=To,P=(P1*xO2)))/w_a
```

"State 2i"

```
P2is = ((r_p)^.5)*P1
s2is=(xCO2*44.01*ENTROPY(CO2,T=T2is,P=(P2is*xCO2))+xH2O*18.02*ENTROPY(H2O,T=T2is,P=(P2is*xH2O))
+xN2*28.01*ENTROPY(N2,T=T2is,P=(P2is*xN2)) + xO2*32*ENTROPY(O2,T=T2is,P=(P2is*xO2)))/w_a
s2is = s1
h2is=(xCO2*44.01*(ENTHALPY(CO2,T=T2is)-ENTHALPY(CO2,T=To))) + xN2*28.01*(ENTHALPY(N2,T=T2is)-
ENTHALPY(N2,T=To)) + xH2O*18.02*(ENTHALPY(H2O,T=T2is)-ENTHALPY(H2O,T=To)) + xO2*32*(
ENTHALPY(O2,T=T2is)-ENTHALPY(O2,T=To))/w_a
```

"State 2i Actual"

$$P_{2i} = ((r_p)^{.5}) * P_1$$

$$h_{2i} = (x_{CO_2} * 44.01 * (ENTHALPY(CO_2, T=T_{2i}) - ENTHALPY(CO_2, T=T_o)) + x_{N_2} * 28.01 * (ENTHALPY(N_2, T=T_{2i}) - ENTHALPY(N_2, T=T_o)) + x_{H_2O} * 18.02 * (ENTHALPY(H_2O, T=T_{2i}) - ENTHALPY(H_2O, T=T_o)) + x_{O_2} * 32 * (ENTHALPY(O_2, T=T_{2i}) - ENTHALPY(O_2, T=T_o))) / w_a$$

$$s_{2i} = (x_{CO_2} * 44.01 * ENTROPY(CO_2, T=T_{2i}, P=(P_{2i} * x_{CO_2})) + x_{H_2O} * 18.02 * ENTROPY(H_2O, T=T_{2i}, P=(P_{2i} * x_{H_2O})) + x_{N_2} * 28.01 * ENTROPY(N_2, T=T_{2i}, P=(P_{2i} * x_{N_2})) + x_{O_2} * 32 * ENTROPY(O_2, T=T_{2i}, P=(P_{2i} * x_{O_2}))) / w_a$$

$$\eta_{a_c1} = (h_1 - h_{2is}) / (h_1 - h_{2i})$$

"State 2o"

$$P_{2o} = P_{2i}$$

$$h_{2o} = h_{2ob}$$

$$h_{2o} = (x_{CO_2} * 44.01 * (ENTHALPY(CO_2, T=T_{2o}) - ENTHALPY(CO_2, T=T_o)) + x_{N_2} * 28.01 * (ENTHALPY(N_2, T=T_{2o}) - ENTHALPY(N_2, T=T_o)) + x_{H_2O} * 18.02 * (ENTHALPY(H_2O, T=T_{2o}) - ENTHALPY(H_2O, T=T_o)) + x_{O_2} * 32 * (ENTHALPY(O_2, T=T_{2o}) - ENTHALPY(O_2, T=T_o))) / w_a$$

$$s_{2o} = (x_{CO_2} * 44.01 * ENTROPY(CO_2, T=T_{2o}, P=(P_{2o} * x_{CO_2})) + x_{H_2O} * 18.02 * ENTROPY(H_2O, T=T_{2o}, P=(P_{2o} * x_{H_2O})) + x_{N_2} * 28.01 * ENTROPY(N_2, T=T_{2o}, P=(P_{2o} * x_{N_2})) + x_{O_2} * 32 * ENTROPY(O_2, T=T_{2o}, P=(P_{2o} * x_{O_2}))) / w_a$$

"State 2ri"

$$P_{2ris} = ((r_p)^{.5}) * P_{2o}$$

$$s_{2ris} = s_{2o}$$

$$s_{2ris} = (x_{CO_2} * 44.01 * ENTROPY(CO_2, T=T_{2ris}, P=(P_{2ris} * x_{CO_2})) + x_{H_2O} * 18.02 * ENTROPY(H_2O, T=T_{2ris}, P=(P_{2ris} * x_{H_2O})) + x_{N_2} * 28.01 * ENTROPY(N_2, T=T_{2ris}, P=(P_{2ris} * x_{N_2})) + x_{O_2} * 32 * ENTROPY(O_2, T=T_{2ris}, P=(P_{2ris} * x_{O_2}))) / w_a$$

$$h_{2ris} = (x_{CO_2} * 44.01 * (ENTHALPY(CO_2, T=T_{2ris}) - ENTHALPY(CO_2, T=T_o)) + x_{N_2} * 28.01 * (ENTHALPY(N_2, T=T_{2ris}) - ENTHALPY(N_2, T=T_o)) + x_{H_2O} * 18.02 * (ENTHALPY(H_2O, T=T_{2ris}) - ENTHALPY(H_2O, T=T_o)) + x_{O_2} * 32 * (ENTHALPY(O_2, T=T_{2ris}) - ENTHALPY(O_2, T=T_o))) / w_a$$

"State 2ri Actual"

$$P_{2ri} = ((r_p)^{.5}) * P_{2o}$$

$$h_{2ri} = (x_{CO_2} * 44.01 * (ENTHALPY(CO_2, T=T_{2ri}) - ENTHALPY(CO_2, T=T_o)) + x_{N_2} * 28.01 * (ENTHALPY(N_2, T=T_{2ri}) - ENTHALPY(N_2, T=T_o)) + x_{H_2O} * 18.02 * (ENTHALPY(H_2O, T=T_{2ri}) - ENTHALPY(H_2O, T=T_o)) + x_{O_2} * 32 * (ENTHALPY(O_2, T=T_{2ri}) - ENTHALPY(O_2, T=T_o))) / w_a$$

$$s_{2ri} = (x_{CO_2} * 44.01 * ENTROPY(CO_2, T=T_{2ri}, P=(P_{2ri} * x_{CO_2})) + x_{H_2O} * 18.02 * ENTROPY(H_2O, T=T_{2ri}, P=(P_{2ri} * x_{H_2O})) + x_{N_2} * 28.01 * ENTROPY(N_2, T=T_{2ri}, P=(P_{2ri} * x_{N_2})) + x_{O_2} * 32 * ENTROPY(O_2, T=T_{2ri}, P=(P_{2ri} * x_{O_2}))) / w_a$$

$$\eta_{a_c2} = (h_{2o} - h_{2ris}) / (h_{2o} - h_{2ri})$$

"State 2ro"

$$P_{2ro} = P_{2ri}$$

$$h_{2ro} = h_{2rob}$$

$$h_{2ro} = (x_{CO_2} * 44.01 * (ENTHALPY(CO_2, T=T_{2ro}) - ENTHALPY(CO_2, T=T_o)) + x_{N_2} * 28.01 * (ENTHALPY(N_2, T=T_{2ro}) - ENTHALPY(N_2, T=T_o)) + x_{H_2O} * 18.02 * (ENTHALPY(H_2O, T=T_{2ro}) - ENTHALPY(H_2O, T=T_o)) + x_{O_2} * 32 * (ENTHALPY(O_2, T=T_{2ro}) - ENTHALPY(O_2, T=T_o))) / w_a$$

$$s_{2ro} = (x_{CO_2} * 44.01 * ENTROPY(CO_2, T=T_{2ro}, P=(P_{2ro} * x_{CO_2})) + x_{H_2O} * 18.02 * ENTROPY(H_2O, T=T_{2ro}, P=(P_{2ro} * x_{H_2O})) + x_{N_2} * 28.01 * ENTROPY(N_2, T=T_{2ro}, P=(P_{2ro} * x_{N_2})) + x_{O_2} * 32 * ENTROPY(O_2, T=T_{2ro}, P=(P_{2ro} * x_{O_2}))) / w_a$$

"State 3"

$$T_3 = RIT$$

$$P_3 = P_{2ro}$$

$$s_3 = m_{CO_2} * ENTROPY(CO_2, T=T_3, P=((b/molout) * P_3)) / ma_2 + m_{O_2} * ENTROPY(O_2, T=T_3, P=((e/molout) * P_3)) / ma_2 + m_{H_2O} * ENTROPY(H_2O, T=T_3, P=((c/molout) * P_3)) / ma_2 + m_{N_2} * ENTROPY(N_2, T=T_3, P=((d/molout) * P_3)) / ma_2$$

$$h_3 = ((m_{CO_2} * (ENTHALPY(CO_2, T=T_3) - ENTHALPY(CO_2, T=298))) / ma_2) + ((m_{O_2} * (ENTHALPY(O_2, T=T_3) - ENTHALPY(O_2, T=298))) / ma_2) + ((m_{H_2O} * (ENTHALPY(H_2O, T=T_3) - ENTHALPY(H_2O, T=298))) / ma_2) + ((m_{N_2} * (ENTHALPY(N_2, T=T_3) - ENTHALPY(N_2, T=298))) / ma_2)$$

"State 4i"

$$P_{4is} = P_{4i}$$

$s_{4is} = s_3$
 $s_{4is} = m_{CO2} * ENTROPY(CO2, T=T_{4is}, P=((b/molout)*P_{4is}))/ma2 + m_{O2} * ENTROPY(O2, T=T_{4is}, P=((e/molout)*P_{4is}))/ma2 + m_{H2O} * ENTROPY(H2O, T=T_{4is}, P=((c/molout)*P_{4is}))/ma2 + m_{N2} * ENTROPY(N2, T=T_{4is}, P=((d/molout)*P_{4is}))/ma2$
 $h_{4is} = ((m_{CO2} * (ENTHALPY(CO2, T=T_{4is}) - ENTHALPY(CO2, T=298)))/ma2) + ((m_{O2} * ENTHALPY(O2, T=T_{4is}))/ma2) + ((m_{H2O} * (ENTHALPY(H2O, T=T_{4is}) - ENTHALPY(H2O, T=298)))/ma2) + ((m_{N2} * ENTHALPY(N2, T=T_{4is}))/ma2)$

"State 4i Actual"

$P_{4i} = P_1$
 $h_{4i} = h_3 - (\eta_{t1} * (h_3 - h_{4is}))$
 $h_{4i} = ((m_{CO2} * (ENTHALPY(CO2, T=T_{4i}) - ENTHALPY(CO2, T=298)))/ma2) + ((m_{O2} * ENTHALPY(O2, T=T_{4i}))/ma2) + ((m_{H2O} * (ENTHALPY(H2O, T=T_{4i}) - ENTHALPY(H2O, T=298)))/ma2) + ((m_{N2} * ENTHALPY(N2, T=T_{4i}))/ma2)$
 $s_{4i} = m_{CO2} * ENTROPY(CO2, T=T_{4i}, P=((b/molout)*P_{4i}))/ma2 + m_{O2} * ENTROPY(O2, T=T_{4i}, P=((e/molout)*P_{4i}))/ma2 + m_{H2O} * ENTROPY(H2O, T=T_{4i}, P=((c/molout)*P_{4i}))/ma2 + m_{N2} * ENTROPY(N2, T=T_{4i}, P=((d/molout)*P_{4i}))/ma2$

"State 4o"

$P_{4o} = P_{4i}$
 $h_{4o} = ((m_{CO2} * (ENTHALPY(CO2, T=T_{4o}) - ENTHALPY(CO2, T=298)))/ma2) + ((m_{O2} * ENTHALPY(O2, T=T_{4o}))/ma2) + ((m_{H2O} * (ENTHALPY(H2O, T=T_{4o}) - ENTHALPY(H2O, T=298)))/ma2) + ((m_{N2} * ENTHALPY(N2, T=T_{4o}))/ma2)$
 $s_{4o} = m_{CO2} * ENTROPY(CO2, T=T_{4o}, P=((b/molout)*P_{4o}))/ma2 + m_{O2} * ENTROPY(O2, T=T_{4o}, P=((e/molout)*P_{4o}))/ma2 + m_{H2O} * ENTROPY(H2O, T=T_{4o}, P=((c/molout)*P_{4o}))/ma2 + m_{N2} * ENTROPY(N2, T=T_{4o}, P=((d/molout)*P_{4o}))/ma2$
 $\eta_{rec} = (T_{2ro} - T_{2ri}) / (T_{4o} - T_{2ri})$

"Inlet intercooler"

$P_{cai} = P_1$
 $T_{cai} = T_o$
 $h_{cai} = (x_{CO2} * 44.01 * (ENTHALPY(CO2, T=T_{cai}) - ENTHALPY(CO2, T=T_o)) + x_{N2} * 28.01 * (ENTHALPY(N2, T=T_{cai}) - ENTHALPY(N2, T=T_o)) + x_{H2O} * 18.02 * (ENTHALPY(H2O, T=T_{cai}) - ENTHALPY(H2O, T=T_o)) + x_{O2} * 32 * (ENTHALPY(O2, T=T_{cai}) - ENTHALPY(O2, T=T_o))) / w_a$
 $s_{cai} = (x_{CO2} * 44.01 * ENTROPY(CO2, T=T_{cai}, P=(P_{cai} * x_{CO2})) + x_{H2O} * 18.02 * ENTROPY(H2O, T=T_{cai}, P=(P_{cai} * x_{H2O})) + x_{N2} * 28.01 * ENTROPY(N2, T=T_{cai}, P=(P_{cai} * x_{N2})) + x_{O2} * 32 * ENTROPY(O2, T=T_{cai}, P=(P_{cai} * x_{O2}))) / w_a$

"Outlet intercooler"

$P_{cao} = P_1$
 $h_{cao} = h_{caob}$
 $h_{cao} = (x_{CO2} * 44.01 * (ENTHALPY(CO2, T=T_{cao}) - ENTHALPY(CO2, T=T_o)) + x_{N2} * 28.01 * (ENTHALPY(N2, T=T_{cao}) - ENTHALPY(N2, T=T_o)) + x_{H2O} * 18.02 * (ENTHALPY(H2O, T=T_{cao}) - ENTHALPY(H2O, T=T_o)) + x_{O2} * 32 * (ENTHALPY(O2, T=T_{cao}) - ENTHALPY(O2, T=T_o))) / w_a$
 $s_{cao} = (x_{CO2} * 44.01 * ENTROPY(CO2, T=T_{cao}, P=(P_{cao} * x_{CO2})) + x_{H2O} * 18.02 * ENTROPY(H2O, T=T_{cao}, P=(P_{cao} * x_{H2O})) + x_{N2} * 28.01 * ENTROPY(N2, T=T_{cao}, P=(P_{cao} * x_{N2})) + x_{O2} * 32 * ENTROPY(O2, T=T_{cao}, P=(P_{cao} * x_{O2}))) / w_a$

" _____ Heat Exchange Components _____ "

"Intercooler"

$(h_{caob} - h_{cai}) = (h_{2i} - h_{2ob})$
 $h_{2ob} = h_{2i} - \epsilon_{IC} * (h_{2i} - h_{cai})$

"Recuperator"

$ma_1 * (h_{2rob} - h_{2ri}) = ma_2 * (h_{4i} - h_{4o})$

" _____ Exergy on the fuel entering the combustion chamber _____ "

$hm_1 = ENTHALPY(CH_4, T=T_o)$
 $hm_2 = ENTHALPY(CH_4, T=T_2)$

```

sm1=ENTROPY(CH4,T=To,P=P1)
sm2s=ENTROPY(CH4,T=T2s,P=P2ro)
sm1=sm2s
hm2s=ENTHALPY(CH4,T=T2s)
eta_CH4=(hm1-hm2s)/(hm1-hm2)
sm2=ENTROPY(CH4,T=T2,P=P2ro)
Wm=(hm1-hm2)
lm=To*(sm2-sm1)
ex_m2=ex_f-Wm+lm

```

"Spray H2O inlet"

```

Exph_5 = ENTHALPY(Steam_NBS,x=1,P=P2ro)-ENTHALPY(Steam_NBS,T=To,P=P1) -
To*(ENTROPY(Steam_NBS,x=1,P=P2ro) - ENTROPY(Steam_NBS,T=To,P=P1))
Exch_5 = Exch_H2O/18.02

Ex5 = Exph_5 + Exch_5

```

"enthaply of spray H2O entering the combustion chamber"

```

h5=ENTHALPY(Steam_NBS,x=1,P=P2ro)-ENTHALPY(Steam_NBS,T=To,P=P1)

```

" _____ Exergy on combustion chamber _____ "

"inlet"

```

Exph_21 = xCO2 *44.01* (ENTHALPY(CO2,T=T2ro)-ENTHALPY(CO2,T=To)) + xN2 *28.01*
(ENTHALPY(N2,T=T2ro)-ENTHALPY(N2,T=To)) + xH2O *18.02 * (ENTHALPY(H2O,T=T2ro)-
ENTHALPY(H2O,T=To)) + xO2*32 * (ENTHALPY(O2,T=T2ro)-ENTHALPY(O2,T=To))

Exph_221 = xCO2*44.01*To*(ENTROPY(CO2,T=T2ro,P=(P2ro*xCO2))-ENTROPY(CO2,T=To,P=(P1*xCO2))) +
xH2O*18.02*To*(ENTROPY(H2O,T=T2ro,P=(P2ro*xH2O))-ENTROPY(H2O,T=To,P=(P1*xH2O)))

Exph_222 = xN2*28.01*To*(ENTROPY(N2,T=T2ro,P=(P2ro*xN2))-ENTROPY(N2,T=To,P=(P1*xN2))) +
xO2*32*To*(ENTROPY(O2,T=T2ro,P=(P2ro*xO2))-ENTROPY(O2,T=To,P=(P1*xO2)))

Exph_22 = Exph_221+Exph_222

Exph_2 = Exph_21 - Exph_22

Exch_2 = xH2O*Exch_H2Og + xCO2*Exch_CO2 + xN2*Exch_N2 + xO2*Exch_O2 +
8.314*To*(xCO2*ln(xCO2)+xH2O*ln(xH2O)+xO2*ln(xO2)+xN2*ln(xN2))

Ex2m = Exph_2 + Exch_2

Ex2 = Ex2m/w_a

```

"outlet"

"number og mol liquid H2O in the dead state of the mixture"

```

nv=(0.0323)*(1-yH2O)

Exph_31 = yCO2*44.01 * (ENTHALPY(CO2,T=T3)-ENTHALPY(CO2,T=To)) + yN2*28.01 * (ENTHALPY(N2,T=T3)-
ENTHALPY(N2,T=To)) + yH2O *18.02* (ENTHALPY(H2O,T=T3)-ENTHALPY(H2O,T=To)) + yO2*32 *
(ENTHALPY(O2,T=T3)-ENTHALPY(O2,T=To))

Exph_321 = yCO2*44.01*To*(ENTROPY(CO2,T=T3,P=(P3*yCO2))-ENTROPY(CO2,T=To,P=(P1*yCO2))) +
yH2O*18.02*To*(ENTROPY(H2O,T=T3,P=(P3*yH2O))-ENTROPY(H2O,T=To,P=(P1*yH2O)))

```

$$\text{Exph_322} = y_{N2} * 28.01 * T_o * (\text{ENTROPY}(N2, T=T3, P=(P3 * y_{N2})) - \text{ENTROPY}(N2, T=T_o, P=(P1 * y_{N2}))) + y_{O2} * 32 * T_o * (\text{ENTROPY}(O2, T=T3, P=(P3 * y_{O2})) - \text{ENTROPY}(O2, T=T_o, P=(P1 * y_{O2})))$$

$$\text{Exph_32} = \text{Exph_321} + \text{Exph_322}$$

$$\text{Exph_3} = \text{Exph_31} - \text{Exph_32}$$

$$\text{Exch_3} = y_{H2O} * \text{Exch_H2Og} + y_{CO2} * \text{Exch_CO2} + y_{N2} * \text{Exch_N2} + y_{O2} * \text{Exch_O2} - (y_{H2O} - nv) * (\text{Exch_H2Og} - \text{Exch_H2Ol}) + 8.314 * T_o * (y_{CO2} * \ln(y_{CO2}) + y_{O2} * \ln(y_{O2}) + y_{N2} * \ln(y_{N2}) + nv * \ln(nv))$$

$$\text{Ex3m} = \text{Exph_3} + \text{Exch_3}$$

$$\text{Ex3} = \text{Ex3m} / w_c$$

"Molecular weight of the combustion gas"

$$w_c = y_{N2} * 28.01 + y_{O2} * 32 + y_{H2O} * 18.02 + y_{CO2} * 44.01$$

"Irreversibility of the Combustion Chamber"

$$I_Comb = \text{Ex2} * ma1 + \text{Ex5} * m_{H2O\text{spray}} + \text{ex_m2} * mf1 - \text{Ex3} * ma2$$

$$\text{EPSILON} = I_Comb / (\text{ex_m2} * mf1)$$

"Exergy out at state 4o (after recuperator)"

$$\text{Exph_41} = y_{CO2} * 44.01 * (\text{ENTHALPY}(CO2, T=T4o) - \text{ENTHALPY}(CO2, T=T_o)) + y_{N2} * 28.01 * (\text{ENTHALPY}(N2, T=T4o) - \text{ENTHALPY}(N2, T=T_o)) + y_{H2O} * 18.02 * (\text{ENTHALPY}(H2O, T=T4o) - \text{ENTHALPY}(H2O, T=T_o)) + y_{O2} * 32 * (\text{ENTHALPY}(O2, T=T4o) - \text{ENTHALPY}(O2, T=T_o))$$

$$\text{Exph_421} = y_{CO2} * 44.01 * T_o * (\text{ENTROPY}(CO2, T=T4o, P=(P4o * y_{CO2})) - \text{ENTROPY}(CO2, T=T_o, P=(P1 * y_{CO2}))) + y_{H2O} * 18.02 * T_o * (\text{ENTROPY}(H2O, T=T4o, P=(P4o * y_{H2O})) - \text{ENTROPY}(H2O, T=T_o, P=(P1 * y_{H2O})))$$

$$\text{Exph_422} = y_{N2} * 28.01 * T_o * (\text{ENTROPY}(N2, T=T4o, P=(P4o * y_{N2})) - \text{ENTROPY}(N2, T=T_o, P=(P1 * y_{N2}))) + y_{O2} * 32 * T_o * (\text{ENTROPY}(O2, T=T4o, P=(P4o * y_{O2})) - \text{ENTROPY}(O2, T=T_o, P=(P1 * y_{O2})))$$

$$\text{Exph_42} = \text{Exph_421} + \text{Exph_422}$$

$$\text{Exph_4} = \text{Exph_41} - \text{Exph_42}$$

$$\text{Exch_4} = y_{H2O} * \text{Exch_H2Og} + y_{CO2} * \text{Exch_CO2} + y_{N2} * \text{Exch_N2} + y_{O2} * \text{Exch_O2} - (y_{H2O} - nv) * (\text{Exch_H2Og} - \text{Exch_H2Ol}) + 8.314 * T_o * (y_{CO2} * \ln(y_{CO2}) + y_{O2} * \ln(y_{O2}) + y_{N2} * \ln(y_{N2}) + nv * \ln(nv))$$

$$\text{Ex4m} = \text{Exph_4} + \text{Exch_4}$$

$$\text{Ex4} = \text{Ex4m} / w_c$$

$$\text{Ex4out} = \text{Ex4} * ma2$$

"Second Law"

"Sink Temperature (K)"

$$Tl = 298$$

"Entropy Generation"

$$Sg_IC = mca * (scao - scai) - ma1 * (s2i - s2o)$$

$$Sg_ca1 = ma1 * (s2i - s1)$$

$$Sg_ca2 = ma1 * (s2ri - s2o)$$

$$Sg_rec = ma1 * (s2ro - s2ri) - ma2 * (s4i - s4o)$$

$$Sg_ta = ma2 * (s4i - s3)$$

```

Sg_cc=ma2*s3-ma1*s2ro

Sg_tot = Sg_IC+Sg_ca1+Sg_ca2+Sg_cc+Sg_ta+Sg_rec

"Irreversibility"
I_IC = Tl*Sg_IC
I_ca1 = Tl*Sg_ca1
I_ca2 = Tl*Sg_ca2
I_rec = Tl*( ma1*(s2ro-s2ri) - ma2*(s4i-s4o) )
I_ta = Tl*Sg_ta
Imact=Im*mf1

I_tot = I_IC+I_ca1+I_ca2+I_rec+I_ta+I_Comb+Imact

I_tot1=I_tot+Ex4out+E_cao

" _____ Exergy out at state cao _____ "

E_cao=mca*(hcao-h1-To*(scao-s1))

" _____ Total exergy out by stream _____ "

E_ex=Ex4out+E_cao

" _____ Net exergy out _____ "

E.net.out = netW+E_ex

" _____ Work out of the turbine _____ "

W_turb=ma2*(h3-h4i)

" _____ Work first compressor _____ "

W_comp1=ma1*(h1-h2i)

" _____ Work second compressor _____ "

W_comp2=ma1*(h2o-h2ri)

" _____ efficiencies of cycle _____ "

Wmact=Wm*mf1

netW=W_turb+W_comp1+W_comp2+Wmact

Thermal_efficiency=netW/(mf1*hcf)

Second_law_efficiency_1=netW/(ex_f*mf1+Ex5*mH2Ospray-E_ex)

Second_law_efficiency_2= netW/(ex_f*mf1 - E_ex)

Balance=E.net.out+ I_tot-(ex_f*mf1+Ex5*mH2Ospray)

```

Appendix B-2: EES Program - Triple Cycle

"EES model of Combined-Cycle (Air/Water/Ammonia) Power Plant"
 "Triple Cycle "

" _____ Input Parameters _____ "

Tpinch=10
 DELTAT=30
 DELFWH = 5
 DELNH3 = 20
 DELTAT_evap = 4
 DELTAT_ph = 2
 DELTATc=5
 epsilon_IC = .9
 hcf = 42000 "Natural Gas"
 ex_f = 43680
 ma2 = ma1 + mf1+mH2Ospray
 mca = ma1
 mf1=1
 Rf = .2870
 r_p=25
 RIT=1600
 To=298
 Tac = To+DELTATc
 Tsc = 373
 mH2Ospray=4

" _____ Adiabatic Efficiencies _____ "

eta_c1 = .9
 eta_c2 = .9
 eta_p1 = .9
 eta_p2 = .75
 eta_p3 = .75
 eta_t1 = .9
 eta_t2 = .9
 eta_t3 = .9
 eta_CH4= .9
 eta_rec=.9

" _____ Calculating the AF _____ "

"62.93 CH3.884+ a(2 O2+7.526 N2+0.1846 H2O) + k H2O = b CO2 + c H2O + d N2 + e O2"

b=62.93
 $2*c=244.42 + 2*k + 2*a*0.1846$
 $2*d=a*15.052$
 $2*b+c+2*e=4*a + k + a*0.1846$

$R=hcf*15.884*mf1-Wm*mf1*15.884$

$z=b*44.01*ENTHALPY(CO2,T=298)+c*18.02*ENTHALPY(H2O,T=298)$

$airH2O=a*3.326*(ENTHALPY(H2O,T=T2ro)-ENTHALPY(H2O,T=To))$

$62.93*R+a*64*ENTHALPY(O2,T=T2ro)+210.64*a*ENTHALPY(N2,T=T2ro)+airH2O+18.02*k*(ENTHALPY(Steam_NBS,x=1,P=P2ro)-ENTHALPY(Steam_NBS,x=1,P=P1))=b*44.01*ENTHALPY(CO2,T=T3)+c*18.02*ENTHALPY(H2O,T=T3)+d*28.01*ENTHALPY(N2,T=T3)+e*32*ENTHALPY(O2,T=T3)-z$

"b = nombre of CO2 mol, c = number of H2O mol, d = number of N2 mol out, e = number of O2 out mol"

"f = number of O2 mol inn, g = number of N2 mol inn, k = number of spray H2O, h=number of H2O mol inn with air "

```

h=a*0.1846
k=(mH2Ospray*1000)/18.02
f = 2*a
g = 7.52*a
molinn = f+g+k+h
molout = b+c+d+e

```

" Total mass of air =ma1"

```
ma1=a*(64+210.8+3.326)/1000
```

" Mass of the different components leaving combustion chamber"

```

mCO2   = (b*44.01)/1000
mO2    = (e*32)/1000
mH2O   = (c*18.02)/1000
mN2    = (d*28.01)/1000
mO2inn = (a*64)/1000
mN2inn = (210.64*a)/1000

```

" Total mass leaving combustion chamber"

```
m2check=mCO2+mO2+mH2O+mN2
```

" ____Molfraction of gasses leaving combustion chamber____"

```

yN2 = d/molout
yO2 = e/molout
yCO2 = b/molout
yH2O = c/molout

```

" _____Environment_____ "

```

xN2 = 0.7748
xO2 = 0.2059
xCO2 = 0.0003
xH2O = 0.019

```

```

Exch_N2 = 639
Exch_O2 = 3951
Exch_CO2 = 14176
Exch_H2Og = 8636
Exch_H2Ol = 45

```

"Molecular weight of air"

```
w_a=xN2*28.01+xO2*32+xCO2*44.01+xH2O*18.02
```

" _____Topping Cycle (Brayton) Air States_____ "

"State 1"

```

T1 = To
P1 = 1.013
h1 = (xCO2*44.01*(ENTHALPY(CO2,T=To)-ENTHALPY(CO2,T=To))) + xN2*28.01*(ENTHALPY(N2,T=To)-
ENTHALPY(N2,T=To)) + xH2O*18.02*(ENTHALPY(H2O,T=To)-ENTHALPY(H2O,T=To)) + xO2*32*
(ENTHALPY(O2,T=To)-ENTHALPY(O2,T=To))/w_a
s1 =
(xCO2*44.01*ENTROPY(CO2,T=To,P=(P1*xCO2))+xH2O*18.02*ENTROPY(H2O,T=To,P=(P1*xH2O))+xN2*28.01*
ENTROPY(N2,T=To,P=(P1*xN2)) + xO2*32*ENTROPY(O2,T=To,P=(P1*xO2)))/w_a

```

"State 2"

```
P2is = ((r_p)^.5)*P1
```

$s_{2is} = (x_{CO2} * 44.01 * \text{ENTROPY}(CO_2, T=T_{2is}, P=(P_{2is} * x_{CO2})) + x_{H2O} * 18.02 * \text{ENTROPY}(H_2O, T=T_{2is}, P=(P_{2is} * x_{H2O})) + x_{N2} * 28.01 * \text{ENTROPY}(N_2, T=T_{2is}, P=(P_{2is} * x_{N2})) + x_{O2} * 32 * \text{ENTROPY}(O_2, T=T_{2is}, P=(P_{2is} * x_{O2}))) / w_a$
 $s_{2is} = s_1$
 $h_{2is} = (x_{CO2} * 44.01 * (\text{ENTHALPY}(CO_2, T=T_{2is}) - \text{ENTHALPY}(CO_2, T=T_o)) + x_{N2} * 28.01 * (\text{ENTHALPY}(N_2, T=T_{2is}) - \text{ENTHALPY}(N_2, T=T_o)) + x_{H2O} * 18.02 * (\text{ENTHALPY}(H_2O, T=T_{2is}) - \text{ENTHALPY}(H_2O, T=T_o)) + x_{O2} * 32 * (\text{ENTHALPY}(O_2, T=T_{2is}) - \text{ENTHALPY}(O_2, T=T_o))) / w_a$

"State 2i Actual"

$P_{2i} = ((r_p)^{.5}) * P_1$
 $h_{2i} = (x_{CO2} * 44.01 * (\text{ENTHALPY}(CO_2, T=T_{2i}) - \text{ENTHALPY}(CO_2, T=T_o)) + x_{N2} * 28.01 * (\text{ENTHALPY}(N_2, T=T_{2i}) - \text{ENTHALPY}(N_2, T=T_o)) + x_{H2O} * 18.02 * (\text{ENTHALPY}(H_2O, T=T_{2i}) - \text{ENTHALPY}(H_2O, T=T_o)) + x_{O2} * 32 * (\text{ENTHALPY}(O_2, T=T_{2i}) - \text{ENTHALPY}(O_2, T=T_o))) / w_a$
 $s_{2i} =$
 $(x_{CO2} * 44.01 * \text{ENTROPY}(CO_2, T=T_{2i}, P=(P_{2i} * x_{CO2})) + x_{H2O} * 18.02 * \text{ENTROPY}(H_2O, T=T_{2i}, P=(P_{2i} * x_{H2O})) + x_{N2} * 28.01 * \text{ENTROPY}(N_2, T=T_{2i}, P=(P_{2i} * x_{N2})) + x_{O2} * 32 * \text{ENTROPY}(O_2, T=T_{2i}, P=(P_{2i} * x_{O2}))) / w_a$
 $\eta_{a_c1} = (h_1 - h_{2is}) / (h_1 - h_{2i})$

"State 2o"

$P_{2o} = P_{2i}$
 $h_{2o} = h_{2ob}$
 $h_{2o} = (x_{CO2} * 44.01 * (\text{ENTHALPY}(CO_2, T=T_{2o}) - \text{ENTHALPY}(CO_2, T=T_o)) + x_{N2} * 28.01 * (\text{ENTHALPY}(N_2, T=T_{2o}) - \text{ENTHALPY}(N_2, T=T_o)) + x_{H2O} * 18.02 * (\text{ENTHALPY}(H_2O, T=T_{2o}) - \text{ENTHALPY}(H_2O, T=T_o)) + x_{O2} * 32 * (\text{ENTHALPY}(O_2, T=T_{2o}) - \text{ENTHALPY}(O_2, T=T_o))) / w_a$
 $s_{2o} = (x_{CO2} * 44.01 * \text{ENTROPY}(CO_2, T=T_{2o}, P=(P_{2o} * x_{CO2})) + x_{H2O} * 18.02 * \text{ENTROPY}(H_2O, T=T_{2o}, P=(P_{2o} * x_{H2O})) + x_{N2} * 28.01 * \text{ENTROPY}(N_2, T=T_{2o}, P=(P_{2o} * x_{N2})) + x_{O2} * 32 * \text{ENTROPY}(O_2, T=T_{2o}, P=(P_{2o} * x_{O2}))) / w_a$

"State 2ri"

$P_{2ris} = ((r_p)^{.5}) * P_{2o}$
 $s_{2ris} = s_{2o}$
 $s_{2ris} =$
 $(x_{CO2} * 44.01 * \text{ENTROPY}(CO_2, T=T_{2ris}, P=(P_{2ris} * x_{CO2})) + x_{H2O} * 18.02 * \text{ENTROPY}(H_2O, T=T_{2ris}, P=(P_{2ris} * x_{H2O})) + x_{N2} * 28.01 * \text{ENTROPY}(N_2, T=T_{2ris}, P=(P_{2ris} * x_{N2})) + x_{O2} * 32 * \text{ENTROPY}(O_2, T=T_{2ris}, P=(P_{2ris} * x_{O2}))) / w_a$
 $h_{2ris} = (x_{CO2} * 44.01 * (\text{ENTHALPY}(CO_2, T=T_{2ris}) - \text{ENTHALPY}(CO_2, T=T_o)) + x_{N2} * 28.01 * (\text{ENTHALPY}(N_2, T=T_{2ris}) - \text{ENTHALPY}(N_2, T=T_o)) + x_{H2O} * 18.02 * (\text{ENTHALPY}(H_2O, T=T_{2ris}) - \text{ENTHALPY}(H_2O, T=T_o)) + x_{O2} * 32 * (\text{ENTHALPY}(O_2, T=T_{2ris}) - \text{ENTHALPY}(O_2, T=T_o))) / w_a$

"State 2ri Actual"

$P_{2ri} = ((r_p)^{.5}) * P_{2o}$
 $h_{2ri} = (x_{CO2} * 44.01 * (\text{ENTHALPY}(CO_2, T=T_{2ri}) - \text{ENTHALPY}(CO_2, T=T_o)) + x_{N2} * 28.01 * (\text{ENTHALPY}(N_2, T=T_{2ri}) - \text{ENTHALPY}(N_2, T=T_o)) + x_{H2O} * 18.02 * (\text{ENTHALPY}(H_2O, T=T_{2ri}) - \text{ENTHALPY}(H_2O, T=T_o)) + x_{O2} * 32 * (\text{ENTHALPY}(O_2, T=T_{2ri}) - \text{ENTHALPY}(O_2, T=T_o))) / w_a$
 $s_{2ri} = (x_{CO2} * 44.01 * \text{ENTROPY}(CO_2, T=T_{2ri}, P=(P_{2ri} * x_{CO2})) + x_{H2O} * 18.02 * \text{ENTROPY}(H_2O, T=T_{2ri}, P=(P_{2ri} * x_{H2O})) + x_{N2} * 28.01 * \text{ENTROPY}(N_2, T=T_{2ri}, P=(P_{2ri} * x_{N2})) + x_{O2} * 32 * \text{ENTROPY}(O_2, T=T_{2ri}, P=(P_{2ri} * x_{O2}))) / w_a$
 $\eta_{a_c2} = (h_{2o} - h_{2ris}) / (h_{2o} - h_{2ri})$

"State 2ro"

$P_{2ro} = P_{2ri}$
 $h_{2ro} = h_{2rob}$
 $h_{2ro} = (x_{CO2} * 44.01 * (\text{ENTHALPY}(CO_2, T=T_{2ro}) - \text{ENTHALPY}(CO_2, T=T_o)) + x_{N2} * 28.01 * (\text{ENTHALPY}(N_2, T=T_{2ro}) - \text{ENTHALPY}(N_2, T=T_o)) + x_{H2O} * 18.02 * (\text{ENTHALPY}(H_2O, T=T_{2ro}) - \text{ENTHALPY}(H_2O, T=T_o)) + x_{O2} * 32 * (\text{ENTHALPY}(O_2, T=T_{2ro}) - \text{ENTHALPY}(O_2, T=T_o))) / w_a$
 $s_{2ro} = (x_{CO2} * 44.01 * \text{ENTROPY}(CO_2, T=T_{2ro}, P=(P_{2ro} * x_{CO2})) + x_{H2O} * 18.02 * \text{ENTROPY}(H_2O, T=T_{2ro}, P=(P_{2ro} * x_{H2O})) + x_{N2} * 28.01 * \text{ENTROPY}(N_2, T=T_{2ro}, P=(P_{2ro} * x_{N2})) + x_{O2} * 32 * \text{ENTROPY}(O_2, T=T_{2ro}, P=(P_{2ro} * x_{O2}))) / w_a$

"State 3"

$T_3 = RIT$
 $P_3 = P_{2ro}$
 $s_3 = m_{CO2} * \text{ENTROPY}(CO_2, T=T_3, P=((b/\text{molout}) * P_3)) / m_a + m_{O2} * \text{ENTROPY}(O_2, T=T_3, P=((e/\text{molout}) * P_3)) / m_a + m_{H2O} * \text{ENTROPY}(H_2O, T=T_3, P=((c/\text{molout}) * P_3)) / m_a + m_{N2} * \text{ENTROPY}(N_2, T=T_3, P=((d/\text{molout}) * P_3)) / m_a$
 $h_3 = ((m_{CO2} * (\text{ENTHALPY}(CO_2, T=T_3) - \text{ENTHALPY}(CO_2, T=298))) / m_a) + ((m_{O2} * (\text{ENTHALPY}(O_2, T=T_3) - \text{ENTHALPY}(O_2, T=298))) / m_a) + ((m_{H2O} * (\text{ENTHALPY}(H_2O, T=T_3) - \text{ENTHALPY}(H_2O, T=298))) / m_a) + ((m_{N2} * (\text{ENTHALPY}(N_2, T=T_3) - \text{ENTHALPY}(N_2, T=298))) / m_a)$

"State 4i"

$P_{4is} = P_{4i}$
 $s_{4is} = s_3$
 $s_{4is} = mCO_2 * ENTROPY(CO_2, T=T_{4is}, P=((b/molout)*P_{4is}))/ma_2 + mO_2 * ENTROPY(O_2, T=T_{4is}, P=((e/molout)*P_{4is}))/ma_2 + mH_2O * ENTROPY(H_2O, T=T_{4is}, P=((c/molout)*P_{4is}))/ma_2 + mN_2 * ENTROPY(N_2, T=T_{4is}, P=((d/molout)*P_{4is}))/ma_2$
 $h_{4is} = ((mCO_2 * (ENTHALPY(CO_2, T=T_{4is}) - ENTHALPY(CO_2, T=298)))/ma_2 + ((mO_2 * (ENTHALPY(O_2, T=T_{4is}) - ENTHALPY(O_2, T=298)))/ma_2 + ((mH_2O * (ENTHALPY(H_2O, T=T_{4is}) - ENTHALPY(H_2O, T=298)))/ma_2 + ((mN_2 * (ENTHALPY(N_2, T=T_{4is}) - ENTHALPY(N_2, T=298)))/ma_2$

"State 4i Actual"

$P_{4i} = P_1$
 $h_{4i} = h_3 - (\eta_{t1} * (h_3 - h_{4is}))$
 $h_{4i} = ((mCO_2 * (ENTHALPY(CO_2, T=T_{4i}) - ENTHALPY(CO_2, T=298)))/ma_2 + ((mO_2 * (ENTHALPY(O_2, T=T_{4i}) - ENTHALPY(O_2, T=298)))/ma_2 + ((mH_2O * (ENTHALPY(H_2O, T=T_{4i}) - ENTHALPY(H_2O, T=298)))/ma_2 + ((mN_2 * (ENTHALPY(N_2, T=T_{4i}) - ENTHALPY(N_2, T=298)))/ma_2$
 $s_{4i} = mCO_2 * ENTROPY(CO_2, T=T_{4i}, P=((b/molout)*P_{4i}))/ma_2 + mO_2 * ENTROPY(O_2, T=T_{4i}, P=((e/molout)*P_{4i}))/ma_2 + mH_2O * ENTROPY(H_2O, T=T_{4i}, P=((c/molout)*P_{4i}))/ma_2 + mN_2 * ENTROPY(N_2, T=T_{4i}, P=((d/molout)*P_{4i}))/ma_2$

"State 4o"

$P_{4o} = P_{4i}$
 $h_{4o} = ((mCO_2 * (ENTHALPY(CO_2, T=T_{4o}) - ENTHALPY(CO_2, T=298)))/ma_2 + ((mO_2 * (ENTHALPY(O_2, T=T_{4o}) - ENTHALPY(O_2, T=298)))/ma_2 + ((mH_2O * (ENTHALPY(H_2O, T=T_{4o}) - ENTHALPY(H_2O, T=298)))/ma_2 + ((mN_2 * (ENTHALPY(N_2, T=T_{4o}) - ENTHALPY(N_2, T=298)))/ma_2$
 $s_{4o} = mCO_2 * ENTROPY(CO_2, T=T_{4o}, P=((b/molout)*P_{4o}))/ma_2 + mO_2 * ENTROPY(O_2, T=T_{4o}, P=((e/molout)*P_{4o}))/ma_2 + mH_2O * ENTROPY(H_2O, T=T_{4o}, P=((c/molout)*P_{4o}))/ma_2 + mN_2 * ENTROPY(N_2, T=T_{4o}, P=((d/molout)*P_{4o}))/ma_2$
 $\eta_{rec} = (T_{2ro} - T_{2ri}) / (T_{4o} - T_{2ri})$

"State 4p"

$P_{4p} = P_1$
 $T_{4p} = T_{11p} + T_{pinch}$
 $h_{4p} = ((mCO_2 * (ENTHALPY(CO_2, T=T_{4p}) - ENTHALPY(CO_2, T=298)))/ma_2 + ((mO_2 * (ENTHALPY(O_2, T=T_{4p}) - ENTHALPY(O_2, T=298)))/ma_2 + ((mH_2O * (ENTHALPY(H_2O, T=T_{4p}) - ENTHALPY(H_2O, T=298)))/ma_2 + ((mN_2 * (ENTHALPY(N_2, T=T_{4p}) - ENTHALPY(N_2, T=298)))/ma_2$
 $s_{4p} = mCO_2 * ENTROPY(CO_2, T=T_{4p}, P=((b/molout)*P_{4p}))/ma_2 + mO_2 * ENTROPY(O_2, T=T_{4p}, P=((e/molout)*P_{4p}))/ma_2 + mH_2O * ENTROPY(H_2O, T=T_{4p}, P=((c/molout)*P_{4p}))/ma_2 + mN_2 * ENTROPY(N_2, T=T_{4p}, P=((d/molout)*P_{4p}))/ma_2$

"State 5"

$h_5 = h_{5b}$
 $P_5 = P_1$
 $h_5 = ((mCO_2 * (ENTHALPY(CO_2, T=T_5) - ENTHALPY(CO_2, T=298)))/ma_2 + ((mO_2 * (ENTHALPY(O_2, T=T_5) - ENTHALPY(O_2, T=298)))/ma_2 + ((mH_2O * (ENTHALPY(H_2O, T=T_5) - ENTHALPY(H_2O, T=298)))/ma_2 + ((mN_2 * (ENTHALPY(N_2, T=T_5) - ENTHALPY(N_2, T=298)))/ma_2$
 $s_5 = mCO_2 * ENTROPY(CO_2, T=T_5, P=((b/molout)*P_5))/ma_2 + mO_2 * ENTROPY(O_2, T=T_5, P=((e/molout)*P_5))/ma_2 + mH_2O * ENTROPY(H_2O, T=T_5, P=((c/molout)*P_5))/ma_2 + mN_2 * ENTROPY(N_2, T=T_5, P=((d/molout)*P_5))/ma_2$

"State 6"

$h_6 = h_{6b}$
 $P_6 = P_1$
 $h_6 = ((mCO_2 * (ENTHALPY(CO_2, T=T_6) - ENTHALPY(CO_2, T=298)))/ma_2 + ((mO_2 * (ENTHALPY(O_2, T=T_6) - ENTHALPY(O_2, T=298)))/ma_2 + ((mH_2O * (ENTHALPY(H_2O, T=T_6) - ENTHALPY(H_2O, T=298)))/ma_2 + ((mN_2 * (ENTHALPY(N_2, T=T_6) - ENTHALPY(N_2, T=298)))/ma_2$
 $s_6 = mCO_2 * ENTROPY(CO_2, T=T_6, P=((b/molout)*P_6))/ma_2 + mO_2 * ENTROPY(O_2, T=T_6, P=((e/molout)*P_6))/ma_2 + mH_2O * ENTROPY(H_2O, T=T_6, P=((c/molout)*P_6))/ma_2 + mN_2 * ENTROPY(N_2, T=T_6, P=((d/molout)*P_6))/ma_2$

" _____ Mid-Cycle (Rankine) Water States _____ "

"State 7"

T7 = Tsc
x7 = 0
P7 = pressure(Steam_NBS,T=T7,x=x7)
h7 = enthalpy(Steam_NBS,T=T7,x=x7)
s7 = entropy(Steam_NBS,T=T7,x=x7)

"State 8"
P8s = PRESSURE(Steam_NBS,T=TFWH2,x=0)
s8s = s7
T8s =TFWH2
TFWH2=(TFWH1 + Tsc)/2
h8s = enthalpy(Steam_NBS,P=P8s,s=s8s)

"State 8 Actual"
P8 = P8s
h8 = ((h8s-h7)/eta_p1) + h7
T8 = temperature(Steam_NBS,P=P8,h=h8)
s8 = entropy(Steam_NBS,P=P8,h=h8)

"State 9"
x9 = 0
P9 = P8
T9 = temperature(Steam_NBS,x=x9,P=P9)
h9 = enthalpy(Steam_NBS,x=x9,P=P9)
s9 = entropy(Steam_NBS,x=x9,P=P9)

"State 10"
P10s = PRESSURE(Steam_NBS,T=TX,x=0)
s10s = s9
T10s = TX
TX=273+((T12-273)/1.54)
h10s = enthalpy(Steam_NBS,P=P10s,s=s10s)

"State 10 Actual"
P10 = P10s
h10 = ((h10s-h9)/eta_p2) + h9
T10 = temperature(Steam_NBS,P=P10,h=h10)
s10 = entropy(Steam_NBS,P=P10,h=h10)

"State 11"
T11 = TFWH1 + DELFWH
P11 = P10
h11 = enthalpy(Steam_NBS,T=T11,P=P11)
s11 = entropy(Steam_NBS,T=T11,P=P11)

"State 11p"
x11p = 0
P11p = P10
T11p = temperature(Steam_NBS,x=x11p,P=P11p)
h11p = enthalpy(Steam_NBS,x=x11p,P=P11p)
s11p = entropy(Steam_NBS,x=x11p,P=P11p)

"State 12"
T12 = T4o - DELTAT
P12 = P10
h12 = enthalpy(Steam_NBS,T=T12,P=P12)
s12 = entropy(Steam_NBS,T=T12,P=P12)

"State 13"
P13s = PRESSURE(Steam_NBS,T=TFWH1,x=0)
s13s = s12

h13s = enthalpy(Steam_NBS,P=P13s,s=s13s)
T13s = TFWH1
TFWH1 = (Tsc +TX)/2

"State 13 Actual"

P13 = P13s
h13 = h12 - (eta_t2*(h12-h13s))
T13 = temperature(Steam_NBS,P=P13,h=h13)
s13 = entropy(Steam_NBS,P=P13,h=h13)

"State 14"

x14 = 0
P14 = P13
T14 = temperature(Steam_NBS,x=x14,P=P14)
h14 = enthalpy(Steam_NBS,x=x14,P=P14)
s14 = entropy(Steam_NBS,x=x14,P=P14)

"State 14t"

h14t = h14
P14t = P9
T14t = temperature(Steam_NBS,h=h14t,P=P14t)
s14t = entropy(Steam_NBS,h=h14t,P=P14t)

"State 15"

P15s = P8
s15s = s12
T15s = temperature(Steam_NBS,P=P15s,s=s15s)
h15s = enthalpy(Steam_NBS,P=P15s,s=s15s)

"State 15 Actual"

P15 = P8
h15 = h12 - (eta_t2*(h12-h15s))
T15 = temperature(Steam_NBS,P=P15,h=h15)
s15 = entropy(Steam_NBS,P=P15,h=h15)
x15 = quality(Steam_NBS,P=P15,h=h15)

"State 16"

P16s = P7
s16s = s12
T16s = temperature(Steam_NBS,P=P16s,s=s16s)
h16s = enthalpy(Steam_NBS,P=P16s,s=s16s)
x16s = quality(Steam_NBS,P=P16s,s=s16s)

"State 16 Actual"

P16 = P7
h16 = h12 - (eta_t2*(h12-h16s))
T16 = temperature(Steam_NBS,P=P16,h=h16)
s16 = entropy(Steam_NBS,P=P16,h=h16)
x16 = quality(Steam_NBS,P=P16,h=h16)

" _____ Bottoming Cycle (Rankine) Ammonia States _____ "

"State 17"

x17 = 0
T17 = Tac
P17 = pressure(Ammonia_ha,x=x17,T=T17)
h17 = enthalpy(Ammonia_ha,x=x17,T=T17)
s17 = entropy(Ammonia_ha,x=x17,T=T17)

"State 18"

```

s18s = s17
P18s = P20
T18s = temperature(Ammonia_ha,s=s18s,P=P18s)
h18s = enthalpy(Ammonia_ha,s=s18s,P=P18s)

```

"State 18 Actual"

```

P18 = P18s
h18 = ( (h18s-h17)/eta_p3 ) + h17
T18 = temperature(Ammonia_ha,P=P18,h=h18)
s18 = entropy(Ammonia_ha,P=P18,h=h18)

```

"State 19"

```

h19 = hb19
P19 = P20
T19 = temperature(Ammonia_ha,h=h19,P=P19)
s19 = entropy(Ammonia_ha,h=h19,P=P19)

```

"State 20"

```

T20 = T7 - DELTAT_evap
x20 = 1
h20 = enthalpy(Ammonia_ha,T=T20,x=x20)
P20 = pressure(Ammonia_ha,T=T20,x=x20)
s20 = entropy(Ammonia_ha,T=T20,x=x20)

```

"State 21"

```

T21 = T5-DELNH3
P21 = P20
h21 = enthalpy(Ammonia_ha,T=T21,P=P21)
s21 = entropy(Ammonia_ha,T=T21,P=P21)

```

"State 22"

```

P22s = P17
s22s = s21
T22s = temperature(Ammonia_ha,P=P22s,s=s22s)
h22s = enthalpy(Ammonia_ha,P=P22s,s=s22s)

```

"State 22 Actual"

```

P22 = P17
h22 = h21- (eta_t3*(h21-h22s))
T22 = temperature(Ammonia_ha,P=P22,h=h22)
s22 = entropy(Ammonia_ha,P=P22,h=h22)

```

"State 23"

```

P23 = P17
T23 = T17 + DELTAT_ph
h23 = enthalpy(Ammonia_ha,P=P23,T=T23)
s23 = entropy(Ammonia_ha,P=P23,T=T23)

```

"Inlet intercooler"

```

Pcai = P1
Tcai = To
hcai = (xCO2 *44.01* (ENTHALPY(CO2,T=Tcai)-ENTHALPY(CO2,T=To)) + xN2 *28.01* (ENTHALPY(N2,T=Tcai)-
ENTHALPY(N2,T=To)) + xH2O *18.02 * (ENTHALPY(H2O,T=Tcai)-ENTHALPY(H2O,T=To)) + xO2*32 *
(ENTHALPY(O2,T=Tcai)-ENTHALPY(O2,T=To)))/w_a
scai=(xCO2*44.01*ENTROPY(CO2,T=Tcai,P=(Pcai*xCO2))+xH2O*18.02*ENTROPY(H2O,T=Tcai,P=(Pcai*xH2O))
+xN2*28.01*ENTROPY(N2,T=Tcai,P=(Pcai*xN2)) + xO2*32*ENTROPY(O2,T=Tcai,P=(Pcai*xO2)))/w_a

```

"Outlet intercooler"

```

Pcao = P1
hcao = hcaob

```

$h_{cao} = (x_{CO2} * 44.01 * (ENTHALPY(CO2, T=T_{cao}) - ENTHALPY(CO2, T=T_o)) + x_{N2} * 28.01 * (ENTHALPY(N2, T=T_{cao}) - ENTHALPY(N2, T=T_o)) + x_{H2O} * 18.02 * (ENTHALPY(H2O, T=T_{cao}) - ENTHALPY(H2O, T=T_o)) + x_{O2} * 32 * (ENTHALPY(O2, T=T_{cao}) - ENTHALPY(O2, T=T_o))) / w_a$
 $s_{cao} = (x_{CO2} * 44.01 * ENTROPY(CO2, T=T_{cao}, P=(P_{cao} * x_{CO2})) + x_{H2O} * 18.02 * ENTROPY(H2O, T=T_{cao}, P=(P_{cao} * x_{H2O})) + x_{N2} * 28.01 * ENTROPY(N2, T=T_{cao}, P=(P_{cao} * x_{N2})) + x_{O2} * 32 * ENTROPY(O2, T=T_{cao}, P=(P_{cao} * x_{O2}))) / w_a$

" _____ Heat Exchange Components _____ "

"Intercooler"

$(h_{caob} - h_{cai}) = (h_{2i} - h_{2ob})$
 $h_{2ob} = h_{2i} - \epsilon_{IC} * (h_{2i} - h_{cai})$

"Recuperator"

$ma_1 * (h_{2rob} - h_{2ri}) = ma_2 * (h_{4i} - h_{4o})$

"Heat Recovery Steam Generator (HRSG)"

$ma_2 * (h_{4o} - h_{4p}) = mw * (h_{12} - h_{11p})$
 $ma_2 * (h_{4p} - h_{5b}) = mw * (h_{11p} - h_{11})$

"Ammonia Evaporator"

$(mw - m_2 - m_3) * (h_{16} - h_7) = m_4 * (h_{20} - (h_{b19}))$

"Ammonia Superheater"

$ma_2 * (h_5 - h_{6b}) = m_4 * (h_{21} - h_{20})$

"Ammonia Preheater"

$m_4 * ((h_{b19}) - h_{18}) = m_4 * (h_{22} - h_{23})$

"Feed Water Heater I"

$mw * (h_{11} - h_{10}) = m_2 * (h_{13} - h_{14})$

"Feed Water Heater II"

$m_2 * (h_{14}) + m_3 * (h_{15}) + (mw - m_2 - m_3) * (h_8) = mw * (h_9)$

$s_{3o} = m_{CO2} * ENTROPY(CO2, T=T_o, P=((b/molout) * P_1)) / ma_2 + m_{O2} * ENTROPY(O2, T=T_o, P=((e/molout) * P_1)) / ma_2 + m_{H2O} * ENTROPY(H2O, T=T_o, P=((c/molout) * P_1)) / ma_2 + m_{N2} * ENTROPY(N2, T=T_o, P=((d/molout) * P_1)) / ma_2$

" _____ Exergy on the fuel entering the combustion chamber _____ "

$hm_1 = ENTHALPY(CH_4, T=T_o)$
 $hm_2 = ENTHALPY(CH_4, T=T_2)$
 $sm_1 = ENTROPY(CH_4, T=T_o, P=P_1)$
 $sm_{2s} = ENTROPY(CH_4, T=T_{2s}, P=P_{2ro})$
 $sm_1 = sm_{2s}$
 $hm_{2s} = ENTHALPY(CH_4, T=T_{2s})$
 $\eta_{CH_4} = (hm_1 - hm_{2s}) / (hm_1 - hm_2)$
 $sm_2 = ENTROPY(CH_4, T=T_2, P=P_{2ro})$
 $W_m = (hm_1 - hm_2)$
 $Im = T_o * (sm_2 - sm_1)$
 $ex_{m2} = ex_f - W_m + Im$
 $T_2 = T_{f1}$

" _____ Exergy on combustion chamber _____ "

"inlet"

Exph_21 = xCO2*44.01*(ENTHALPY(CO2,T=T2ro)-ENTHALPY(CO2,T=To)) + xN2*28.01*(ENTHALPY(N2,T=T2ro)-ENTHALPY(N2,T=To)) + xH2O*18.02*(ENTHALPY(H2O,T=T2ro)-ENTHALPY(H2O,T=To)) + xO2*32*(ENTHALPY(O2,T=T2ro)-ENTHALPY(O2,T=To))

Exph_221 = xCO2*44.01*To*(ENTROPY(CO2,T=T2ro,P=(P2ro*xCO2))-ENTROPY(CO2,T=To,P=(P1*xCO2))) + xH2O*18.02*To*(ENTROPY(H2O,T=T2ro,P=(P2ro*xH2O))-ENTROPY(H2O,T=To,P=(P1*xH2O)))

Exph_222 = xN2*28.01*To*(ENTROPY(N2,T=T2ro,P=(P2ro*xN2))-ENTROPY(N2,T=To,P=(P1*xN2))) + xO2*32*To*(ENTROPY(O2,T=T2ro,P=(P2ro*xO2))-ENTROPY(O2,T=To,P=(P1*xO2)))

Exph_22 = Exph_221+Exph_222

Exph_2 = Exph_21 - Exph_22

Exch_2 = xH2O*Exch_H2Og + xCO2*Exch_CO2 + xN2*Exch_N2 + xO2*Exch_O2 + 8.314*To*(xCO2*ln(xCO2)+xH2O*ln(xH2O)+xO2*ln(xO2)+xN2*ln(xN2))

Ex2m = Exph_2 + Exch_2

Ex2 = Ex2m/w_a

"outlet"

"number of mol liquid H2O in the dead state of the mixture"

nv=(0.0323)*(1-yH2O)

Exph_31 = yCO2*44.01*(ENTHALPY(CO2,T=T3)-ENTHALPY(CO2,T=To)) + yN2*28.01*(ENTHALPY(N2,T=T3)-ENTHALPY(N2,T=To)) + yH2O*18.02*(ENTHALPY(H2O,T=T3)-ENTHALPY(H2O,T=To)) + yO2*32*(ENTHALPY(O2,T=T3)-ENTHALPY(O2,T=To))

Exph_321 = yCO2*44.01*To*(ENTROPY(CO2,T=T3,P=(P3*yCO2))-ENTROPY(CO2,T=To,P=(P1*yCO2))) + yH2O*18.02*To*(ENTROPY(H2O,T=T3,P=(P3*yH2O))-ENTROPY(H2O,T=To,P=(P1*yH2O)))

Exph_322 = yN2*28.01*To*(ENTROPY(N2,T=T3,P=(P3*yN2))-ENTROPY(N2,T=To,P=(P1*yN2))) + yO2*32*To*(ENTROPY(O2,T=T3,P=(P3*yO2))-ENTROPY(O2,T=To,P=(P1*yO2)))

Exph_32 = Exph_321+Exph_322

Exph_3 = Exph_31 - Exph_32

Exch_3 = yH2O*Exch_H2Og + yCO2*Exch_CO2 + yN2*Exch_N2 + yO2*Exch_O2 - (yH2O-nv)*(Exch_H2Og-Exch_H2Ol) + 8.314*To*(yCO2*ln(yCO2)+yO2*ln(yO2)+yN2*ln(yN2)+nv*ln(nv))

Ex3m = Exph_3 + Exch_3

Ex3=Ex3m/w_c

"Molecular weight of the combustion gas"

w_c = yN2*28.01+yO2*32+yH2O*18.02+yCO2*44.01

"Spray H2O inlet"

Exph_5 = ENTHALPY(Steam_NBS,x=1,P=P2ro)-ENTHALPY(Steam_NBS,T=To,P=P1) - To*(ENTROPY(Steam_NBS,x=1,P=P2ro) - ENTROPY(Steam_NBS,T=To,P=P1))
Exch_5 = Exch_H2Ol/18.02

$$\text{Ex5} = \text{Exph}_5 + \text{Exch}_5$$

" Enthalpy of spray H2O entering the combustion chamber "

$$h5\text{spray} = \text{ENTHALPY}(\text{Steam_NBS}, x=1, P=P2ro) - \text{ENTHALPY}(\text{Steam_NBS}, T=To, P=P1)$$

" Irreversibility of the Combustion Chamber "

$$I_{\text{Comb}} = \text{Ex2} * \text{ma1} + \text{Ex5} * \text{mH2O}\text{spray} + \text{ex}_m * \text{mf1} - \text{Ex3} * \text{ma2}$$

$$\text{EPSILON} = I_{\text{Comb}} / (\text{ex}_m * \text{mf1})$$

" Exergy out at state 6 "

$$\begin{aligned} \text{Exph}_{61} = & y\text{CO2} * 44.01 * (\text{ENTHALPY}(\text{CO2}, T=T6) - \text{ENTHALPY}(\text{CO2}, T=To)) + y\text{N2} * 28.01 * (\text{ENTHALPY}(\text{N2}, T=T6) - \\ & \text{ENTHALPY}(\text{N2}, T=To)) + y\text{H2O} * 18.02 * (\text{ENTHALPY}(\text{H2O}, T=T6) - \text{ENTHALPY}(\text{H2O}, T=To)) + y\text{O2} * 32 * \\ & (\text{ENTHALPY}(\text{O2}, T=T6) - \text{ENTHALPY}(\text{O2}, T=To)) \end{aligned}$$

$$\begin{aligned} \text{Exph}_{621} = & y\text{CO2} * 44.01 * To * (\text{ENTROPY}(\text{CO2}, T=T6, P=(P6 * y\text{CO2})) - \text{ENTROPY}(\text{CO2}, T=To, P=(P1 * y\text{CO2}))) + \\ & y\text{H2O} * 18.02 * To * (\text{ENTROPY}(\text{H2O}, T=T6, P=(P6 * y\text{H2O})) - \text{ENTROPY}(\text{H2O}, T=To, P=(P1 * y\text{H2O}))) \end{aligned}$$

$$\begin{aligned} \text{Exph}_{622} = & y\text{N2} * 28.01 * To * (\text{ENTROPY}(\text{N2}, T=T6, P=(P6 * y\text{N2})) - \text{ENTROPY}(\text{N2}, T=To, P=(P1 * y\text{N2}))) + \\ & y\text{O2} * 32 * To * (\text{ENTROPY}(\text{O2}, T=T6, P=(P6 * y\text{O2})) - \text{ENTROPY}(\text{O2}, T=To, P=(P1 * y\text{O2}))) \end{aligned}$$

$$\text{Exph}_{62} = \text{Exph}_{621} + \text{Exph}_{622}$$

$$\text{Exph}_6 = \text{Exph}_{61} - \text{Exph}_{62}$$

$$\begin{aligned} \text{Exch}_6 = & y\text{H2O} * \text{Exch}_{\text{H2Og}} + y\text{CO2} * \text{Exch}_{\text{CO2}} + y\text{N2} * \text{Exch}_{\text{N2}} + y\text{O2} * \text{Exch}_{\text{O2}} - (y\text{H2O} - \text{nv}) * (\text{Exch}_{\text{H2Og}} - \\ & \text{Exch}_{\text{H2Ol}}) + 8.314 * To * (y\text{CO2} * \ln(y\text{CO2}) + y\text{O2} * \ln(y\text{O2}) + y\text{N2} * \ln(y\text{N2}) + \text{nv} * \ln(\text{nv})) \end{aligned}$$

$$\text{Ex6m} = \text{Exph}_6 + \text{Exch}_6$$

$$\text{Ex6} = \text{Ex6m} / w_c$$

" Exergy out at state 4o (after recuperator) "

$$\begin{aligned} \text{Exph}_{41} = & y\text{CO2} * 44.01 * (\text{ENTHALPY}(\text{CO2}, T=T4o) - \text{ENTHALPY}(\text{CO2}, T=To)) + y\text{N2} * 28.01 * \\ & (\text{ENTHALPY}(\text{N2}, T=T4o) - \text{ENTHALPY}(\text{N2}, T=To)) + y\text{H2O} * 18.02 * (\text{ENTHALPY}(\text{H2O}, T=T4o) - \\ & \text{ENTHALPY}(\text{H2O}, T=To)) + y\text{O2} * 32 * (\text{ENTHALPY}(\text{O2}, T=T4o) - \text{ENTHALPY}(\text{O2}, T=To)) \end{aligned}$$

$$\begin{aligned} \text{Exph}_{421} = & y\text{CO2} * 44.01 * To * (\text{ENTROPY}(\text{CO2}, T=T4o, P=(P4o * y\text{CO2})) - \text{ENTROPY}(\text{CO2}, T=To, P=(P1 * y\text{CO2}))) + \\ & y\text{H2O} * 18.02 * To * (\text{ENTROPY}(\text{H2O}, T=T4o, P=(P4o * y\text{H2O})) - \text{ENTROPY}(\text{H2O}, T=To, P=(P1 * y\text{H2O}))) \end{aligned}$$

$$\begin{aligned} \text{Exph}_{422} = & y\text{N2} * 28.01 * To * (\text{ENTROPY}(\text{N2}, T=T4o, P=(P4o * y\text{N2})) - \text{ENTROPY}(\text{N2}, T=To, P=(P1 * y\text{N2}))) + \\ & y\text{O2} * 32 * To * (\text{ENTROPY}(\text{O2}, T=T4o, P=(P4o * y\text{O2})) - \text{ENTROPY}(\text{O2}, T=To, P=(P1 * y\text{O2}))) \end{aligned}$$

$$\text{Exph}_{42} = \text{Exph}_{421} + \text{Exph}_{422}$$

$$\text{Exph}_4 = \text{Exph}_{41} - \text{Exph}_{42}$$

$$\begin{aligned} \text{Exch}_4 = & y\text{H2O} * \text{Exch}_{\text{H2Og}} + y\text{CO2} * \text{Exch}_{\text{CO2}} + y\text{N2} * \text{Exch}_{\text{N2}} + y\text{O2} * \text{Exch}_{\text{O2}} - (y\text{H2O} - \text{nv}) * (\text{Exch}_{\text{H2Og}} - \\ & \text{Exch}_{\text{H2Ol}}) + 8.314 * To * (y\text{CO2} * \ln(y\text{CO2}) + y\text{O2} * \ln(y\text{O2}) + y\text{N2} * \ln(y\text{N2}) + \text{nv} * \ln(\text{nv})) \end{aligned}$$

$$\text{Ex4m} = \text{Exph}_4 + \text{Exch}_4$$

$$\text{Ex4o} = \text{Ex4m} / w_c$$

" _____ Exergy out at state cao _____ "

$$E_{cao} = m_{ca} \cdot (h_{cao} - h_1 - T_o \cdot (s_{cao} - s_1))$$

" _____ Total exergy out by stream _____ "

$$E_{ex} = E_{x6} \cdot m_2 + E_{cao}$$

" _____ First Law _____ "

"Energy Inputs"

$$E_{in_air} = m_{a1} \cdot (h_1)$$

$$E_{in_chm1} = m_{f1} \cdot (h_{cf})$$

$$E_{in_ca} = m_{ca} \cdot (h_{cai})$$

$$E_{in_H2O\text{spray}} = m_{H2O\text{spray}} \cdot h_{5\text{spray}}$$

$$E_{in} = E_{in_air} + E_{in_chm1} + E_{in_ca} + E_{in_H2O\text{spray}}$$

"Energy Outputs"

$$E_{out_exh} = m_{a2} \cdot (h_6)$$

$$Q_{out} = m_4 \cdot (h_{23} - h_{17})$$

$$E_{out_ca} = m_{ca} \cdot (h_{cao})$$

$$W_{out_air} = m_{a2} \cdot (h_3 - h_{4i}) - m_{a1} \cdot (h_{2i} - h_1) - m_{a1} \cdot (h_{2ri} - h_{2o}) - m_{f1} \cdot (h_{m2} - h_{m1})$$

$$W_{out_w} = m_2 \cdot (h_{12} - h_{13}) + m_3 \cdot (h_{12} - h_{15}) + (m_w - m_2 - m_3) \cdot (h_{12} - h_{16}) - m_w \cdot (h_{10} - h_9) - (m_w - m_2 - m_3) \cdot (h_8 - h_7)$$

$$W_{out_am} = m_4 \cdot (h_{21} - h_{22}) - m_4 \cdot (h_{18} - h_{17})$$

$$W_{out} = W_{out_air} + W_{out_w} + W_{out_am}$$

$$E_{out} = Q_{out} + E_{out_exh} + W_{out} + E_{out_ca}$$

"Energy Balance"

$$EB = E_{in} - E_{out}$$

$$Balance = E_{net.out} + I_{tot} - (ex_f \cdot m_{f1} + Ex_5 \cdot m_{H2O\text{spray}})$$

$$Q_{out_IC} = m_{a1} \cdot (h_{2i} - h_{2o})$$

" _____ Second Law _____ "

"Sink Temperature (K)"

$$T_I = 298$$

"Entropy Generation"

$$S_{g_rec} = m_{a1} \cdot (s_{2ro} - s_{2ri}) - m_{a2} \cdot (s_{4i} - s_{4o})$$

$$S_{g_IC} = m_{ca} \cdot (s_{cao} - s_{cai}) - m_{a1} \cdot (s_{2i} - s_{2o})$$

$$S_{g_ca1} = m_{a1} \cdot (s_{2i} - s_1)$$

$$S_{g_ca2} = m_{a1} \cdot (s_{2ri} - s_{2o})$$

$$S_{g_p1} = (m_w - m_2 - m_3) \cdot (s_8 - s_7)$$

$$S_{g_p2} = m_w \cdot (s_{10} - s_9)$$

$$S_{g_p3} = m_4 \cdot (s_{18} - s_{17})$$

$$S_{g_trap} = m_2 \cdot (s_{14t} - s_{14})$$

$$S_{g_ta} = m_{a2} \cdot (s_{4i} - s_3)$$

$$S_{g_cc} = m_{a2} \cdot s_3 - m_{a1} \cdot s_{2ro}$$

$$S_{g_hrsg} = (m_w \cdot (s_{12} - s_{11})) - (m_{a2} \cdot (s_{4o} - s_5))$$

$$S_{g_sh} = (m_4 \cdot (s_{21} - s_{20})) - (m_{a2} \cdot (s_5 - s_6))$$

$$S_{g_tw} = (m_2 \cdot (s_{13} - s_{12})) + (m_3 \cdot (s_{15} - s_{12})) + ((m_w - m_2 - m_3) \cdot (s_{16} - s_{12}))$$

$$S_{g_evp} = m_4 \cdot (s_{20} - s_{19}) - (m_w - m_2 - m_3) \cdot (s_{16} - s_7)$$

$$S_{g_fwh2} = m_w \cdot (s_9) - (m_3 \cdot (s_{15}) + (m_w - m_2 - m_3) \cdot (s_8) + m_2 \cdot (s_{14t}))$$

$$S_{g_fwh1} = m_w \cdot (s_{11} - s_{10}) - m_2 \cdot (s_{13} - s_{14})$$

$$S_{g_tam} = (m_4 \cdot (s_{22} - s_{21}))$$

$$S_{g_ph} = (m_4 \cdot (s_{19} - s_{18})) - (m_4 \cdot (s_{22} - s_{23}))$$

$$S_{g_cond} = (Q_{out}/T_I) - (m_4 \cdot (s_{23} - s_{17}))$$

$$S_{g_tot} =$$

$$S_{g_IC} + S_{g_ca1} + S_{g_ca2} + S_{g_cc} + S_{g_ta} + S_{g_hrsg} + S_{g_sh} + S_{g_tw} + S_{g_evp} + S_{g_fwh2} + S_{g_fwh1} + S_{g_tam} + S_{g_ph} + S_{g_cond} + S_{g_rec}$$

"Irreversibility"

```

I_IC = Tl*Sg_IC
I_ca1 = Tl*Sg_ca1
I_ca2 = Tl*Sg_ca2
I_rec = Tl*( ma1*(s2ro-s2ri) - ma2*(s4i-s4o) )
I_p1 = Tl*Sg_p1
I_p2 = Tl*Sg_p2
I_p3 = Tl*Sg_p3
I_trap = Tl*Sg_trap
I_cc1 = I_Comb
I_ta = Tl*Sg_ta
I_hrs = Tl*Sg_hrs
I_sh = Tl*Sg_sh
I_tw = Tl*Sg_tw
I_evp = Tl*Sg_evp
I_fwh2 = Tl*Sg_fwh2
I_fwh1 = Tl*Sg_fwh1
I_tam = Tl*Sg_tam
I_ph = Tl*Sg_ph
I_cond = Tl*Sg_cond
I_mact = Im*mf1

I_tot =
I_IC+I_ca1+I_ca2+I_rec+I_p1+I_p2+I_p3+I_trap+I_cc1+I_ta+I_hrs+I_sh+I_tw+I_evp+I_fwh2+I_fwh1+I_tam+I_ph+I
_cond+I_mact

I_tot1=I_tot+E_ex

```

" _____ Net Exergy out _____ "

E.net.out = Wout + E_ex

" _____ efficiencies of cycle _____ "

eff1=Wout/(mf1*hcf)

eff2=Wout/(ex_f*mf1- E_ex)

eff2_1=Wout/(ex_f*mf1+Ex5*mH2Ospray-E_ex)

APPENDIX - C

The Effective Use of Heat Transfer Additives for Steam Condensation

by

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ABSTRACT

Reported here is the results of the experimental investigations on the effect of small amounts of heat transfer additives ($C_{\text{add}} < 1000$ ppm) on steam condensation in a bundle of horizontal tubes that are internally cooled. The condenser was designed with twelve tubes in an array of three horizontals and four verticals with a 2-inch horizontal and 1.5-inch vertical in-line pitch. By using effective additives, the condensation heat transfer rate can be augmented as much as 30% as compared with heat transfer that operated without additives under the same operating condition. The steam condensation which occurred in our experiments while using effective additives was mostly *dropwise-like*. When heat transfer additives function effectively, the condensate-droplets become more dispersed and have a smaller shape than those produced without additives. These droplets, unlike traditional turbulence, start at the top portion of the condenser tubes and cover most of the tubes. Such a flow behavior can be explained by the Marangoni effect (in terms of thermodynamic equilibrium) in connection with obtained surface tension data. Furthermore, we extend our discussion to the dynamic effect of surface tension in the steam condensation process and propose a general criterion for selecting effective additives. In our experiments, we noted that the use of heat transfer additives such as 2-ethyl-1-hexanol for steam condensation was highly effective.

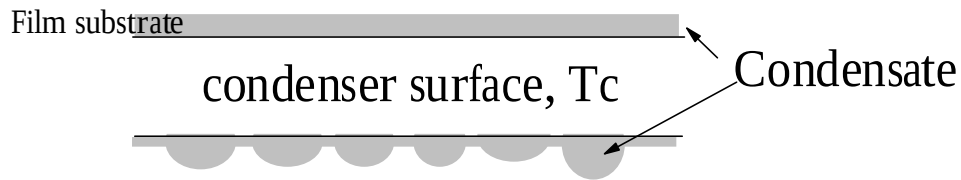
1. INTRODUCTION

It is well known that the use of *dropwise* condensation is an effective means of heat transfer for condenser design. In dropwise condensation most of the heat transfer is through small drops and results in transfer rates dramatically larger than those associated with *filmwise* condensation [1-2]. However, long-term dropwise condensation conditions are usually difficult to maintain. Most surface condensers, therefore, are designed to operate in the filmwise mode. Effectively and simply, *dropwise-like* condensation (discussed later in this section) can be achieved by introducing small amounts of heat transfer additives (surfactants) into the condenser [3-6]. The effective additives trigger *interfacial turbulence* that leads to the loss of hydrodynamic stability at the vapor/liquid interface and produce a spontaneous surface convection [7-11]. As a result, condensation heat transfer is significantly increased.

Interfacial turbulence (often called the surface convection or *Marangoni* effect) is an important mechanism in various transport processes such as condensation, extraction, crystal growth, distillation and absorption, and many other interfacial transport processes. It is widely accepted that interfacial turbulence is caused by local variations in the surface tension, meaning that it is principally related to the additive concentration dependence of surface tension. Usually, a surface tension gradient causes the convection flows and leads to a substantial increase in heat (and/or mass) transfer rate. For example, intensifying heat transfer in absorption cooling systems, interfacial turbulence created by heat transfer additives, was proven to be an effective way of overcoming the limitation that diffusion has without applying externally-driven, forced convection [7-8, 10-11].

It appears that in the presence of effective additives, the steam condensation process observed in our experiment was mostly a dropwise-like condensation process. When effective heat transfer additives were present, the condensate water droplets became a smaller shape than those without additives and became more dispersed, exhibiting much higher frequencies. The observed size of the droplets with heat transfer additives was approximately 1-2 mm in diameter in the additive concentration range of 100-1,000 ppm (note that the additive concentration is defined by the mass of the added heat transfer additive divided by the total mass of the water). A further increase in additive concentration (particularly up to 10,000 ppm, =1%, an effective heat transfer additive, 2-ethyl-1-hexanol, was added) decreased the condensate droplet size to approximately 0.1-0.5 mm in diameter. These droplets are smaller with heat transfer additives present and, also, there was more condensation activity at the condenser surface. As soon as condensate appeared on the surface of the condenser tubes, the film contracted into small droplets. As illustrated in Figure 1, effective additives displayed considerably vigorous disturbances. The dropwise-like condensation, unlike traditional turbulence, starts at the top portion of the condenser tubes and covers most of the tubes, hanging at the bottom with small droplet sizes. We also observed that film substrates appear to be thinner compared to those without additives. Such a flow condition was quite attractive for condensation in the absence of smooth, side-drainage, and splashing types. Overall, we believe that added surfactants play a pivotal role in improving dropwise-like condensate activity that leads to a significant benefit for condensation heat transfer.

Without additives



With additives

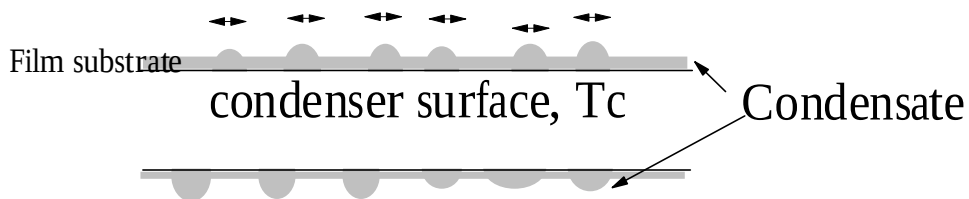


Figure 1. Schematic illustrations of observed condensate behavior. When effective additives are present, usually film substrates get thinner and the condensate becomes a *dropwise-like* form with a much smaller droplet size than that of the usual filmwise condensate.

2. EXPERIMENTAL

Figure 2 shows a schematic drawing of the constructed test apparatus, which is connected with the cold reservoir, condenser, and water chiller along with details of tube spacing. The condenser design was a closed system, made of carbon-steel. Steam is being generated at the boiler by two immersion heaters with a total heating capacity of 1,650 W with a controller. The water evaporates and fills the rest of the condenser with steam. At the top of the condenser, copper tubes run through the condenser. Cooling

water flows through the copper tubes. When the steam hits the cold copper tubes, it condenses and eventually falls back to the bottom water reservoir. The condenser was designed with twelve tubes in an array of three horizontals and four verticals. A 2-inch horizontal and 1.5-inch vertical in-line pitch was selected. These condenser tubes were connected as shown; “in” designates where the cooling water enters the condenser and “out” designates where the cooling water leaves the condenser. The rest of the tubes are connected starting from 1 and to the next increasing number (1 and 2, 3 and 4, 5 and 6, 7 and 8, 9 and 10). At the top surface of the condenser, a circular hole with a diameter of 6.7 inch was made to place a plastic dome that allows one to see through it, monitoring the condensation process during the experiments. This dome was also connected to a vacuum pump. A NeopreneTM rubber vacuum seal was placed between the condenser and the dome so as to obtain a complete vacuum. A number of shake-up tests were performed to meet a favorable leak rate less than 10^{-3} Torr/hour so as to minimize the detrimental effect of non-condensable to the condensation process [12-14]. In order for the operator to oversee the condensation process from the side, a sight-glass with a diameter of 1.4 inch was placed in one of the sides of the condenser parallel to the condensation tubes. The other sight-glass with the same diameter was placed near the bottom of the condenser at the same side as the other window. The condenser tubes were 16 mm outside diameter with wall thickness of 1 mm. The tubes went 3 inches outside the condenser on each side. Holes were drilled through the steel plates for the tubes, and the tubes were welded to the condenser wall. The entire system was properly insulated.

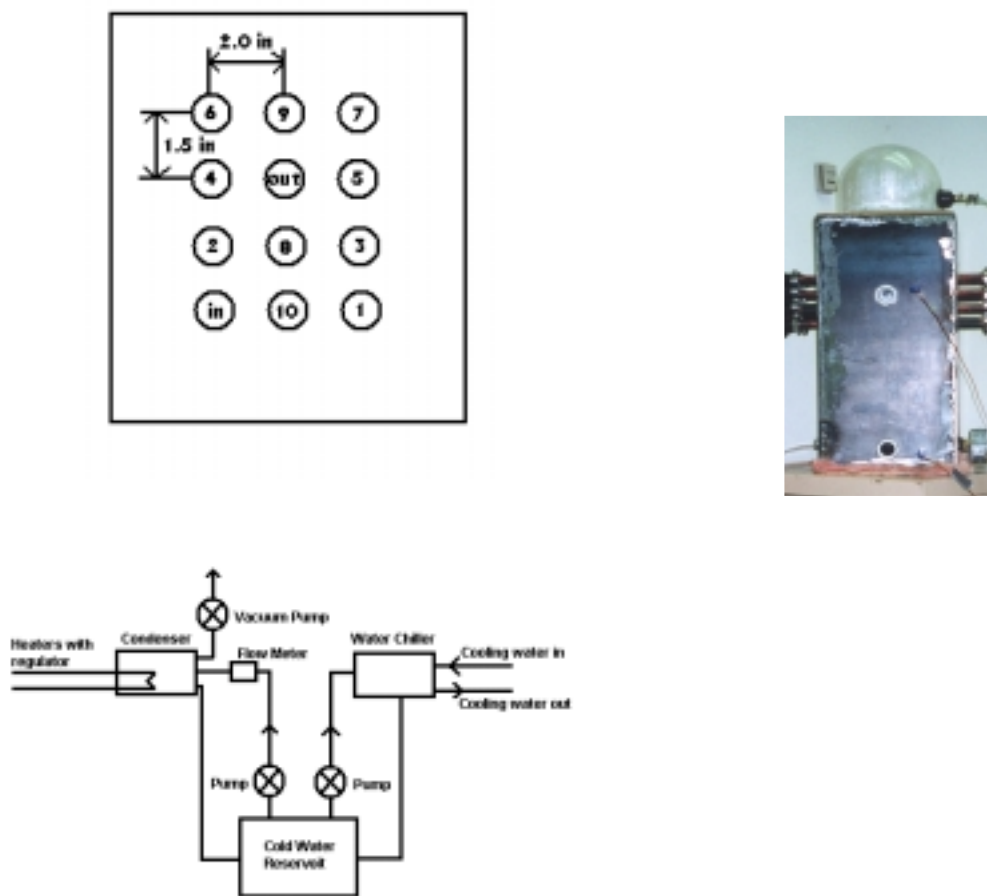


Figure 2. A Schematic drawing of the apparatus along with tube spacing (left) and its photograph (right). **Two thermocouples were placed inside the condenser: one to measure the liquid temperature at the boiler, and the other to measure the steam temperature. Five others were also placed on the outside surface of the copper tubes right outside the condenser: one for the inlet cooling water, one for the outlet cooling water, and three others in-between. The thermocouples were of the type T. The thermocouples were connected to a computerized data**

acquisition system (LabViewTM) where the temperatures were monitored and stored.

3. RESULTS AND DISCUSSION

As we discussed previously, small amounts of certain surfactant additives can change the condensation phenomena of steam. Since the amounts of surfactants used in our study is small and no micelles are formed at this small concentration range (no changes in bulk properties), the surface tension was the sole property that we considered. Therefore, surface tension can be subjected to study for its relationship to the condensation heat transfer coefficient, h_c ,

$$h_c = f(\sigma)$$

(1).

Figure 3 shows the measured surface tension of water with Additive-A (2-ethyl-1-hexanol) and Additive-B (alkyl primary amine), respectively. As expected, an increase in surfactant additives' concentration significantly reduces the surface tension of water and finally reaching their own surface tension values. It is of interest to briefly review the subject of the surface tension of a binary system. The Gibbs equation describes the surface tension, σ , as,

$$d\sigma = -\Gamma_1^s d\mu_1 - \Gamma_2^s d\mu_2 \quad (2),$$

where Γ_1^s and Γ_2^s are the arbitrary chosen dividing surface excesses per unit area for phases 1 and 2, respectively. Other notations μ_1 and μ_2 are the surface chemical potentials of each phase defined as,

$$\mu = RT \ln a \quad (3),$$

where a is the activity of the solute, R is the gas constant, and T is the absolute temperature. If the dividing surface is appropriately chosen so that Γ_1^s becomes zero, then Γ_2^s can be written as,

$$\Gamma_2^s = -\left(\frac{a}{RT}\right)\left(\frac{d\sigma}{da}\right) \quad (4).$$

If there is a surface excess of solute ($\Gamma_2^s > 0$), $d\sigma/da$ is negative. So, if there are surfactants in water, the surface tension of water drops as the surfactants' concentration increases. This effect is posed in Figure 3. However, note that the equilibrium surface is not established in a dynamic system (such as a steam

condensation process with additives) as fast as in pure liquid due to the time required for the solution to reach its equilibrium condition.

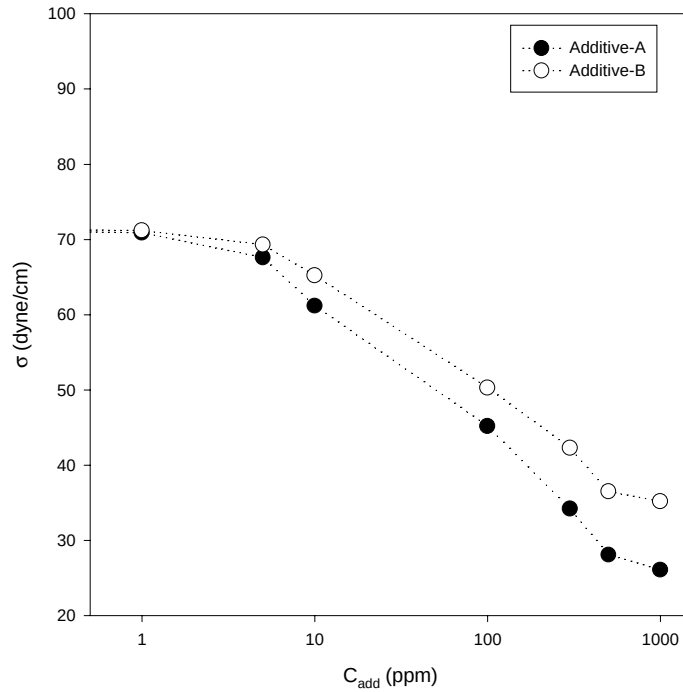


Figure 3. Measured surface tension data of water/additives.
The Du Nouy ring method was employed to determine the surface tension. Additive-A and Additive-B are 2-ethyl-1-hexanol and alkyl primary amine, respectively. The surfactant concentration dependence of surface tension is negative ($\partial\sigma / \partial C_{add} < 0$).

The effect of heat transfer additives on the condensation heat transfer was evaluated by introducing the enhancement factor, F_e , which is defined by,

$$F_e = \frac{\bar{h}_a}{\bar{h}_o} \quad (5),$$

where $\bar{h}_a$ and $\bar{h}_o$ denote the average condensation heat transfer coefficients with additives and without additives, respectively. Note that the overall condensation heat transfer coefficients, $\bar{h}$, can be found from,

$$\bar{h} = \frac{\dot{q}}{A_c (T_{sat} - T_w)} \quad (6),$$

where A_c , T_{sat} , T_w , and $\dot{q}$ are the condenser heat transfer area, saturation temperature, and the average wall temperature, and condenser load, respectively. The condenser load can be simply obtained by,

$$\dot{q} = \dot{m} C_p \Delta T \quad (7),$$

where $\dot{m}$ is the cooling water mass flow rate, C_p is the cooling water heat capacity and ΔT is the temperature gain in the cooling water. The saturation temperature, T_{sat} , was considered as the one inside the condenser with a thermocouple about two inches at the side of row number 2 from the top (middle row, Figure 2). The T_w was used as the mean temperature of the inlet and outlet wall temperature (also measured with thermocouples).

In order to evaluate the effectiveness of the additives, the heat input to the boiler was controlled to maintain the same temperature gain, ΔT , in the cooling water relative to one without additives. This approach is convenient to evaluate the performance of the additives. It is anticipated that effective additives exhibit higher values of the condensation heat transfer coefficient that lead to lower the corresponding saturation temperatures, T_{sat} . It is due to the less required boiler duty. The experiments were carried out for three different controlled values of ΔT of 3.0, 4.5, and 6.2 °C, and results are provided in Figure 4-(a), (b), and (c), respectively. In general we conceive that heat transfer additive-A can enhance the heat transfer coefficient by as much as 17 percent,

which is believed to be a significant benefit for condenser design. As noted, the effect of adding heat transfer additives is progressively substantial for larger ΔT 's. For an obtained ΔT , enhanced condensation activity gained by effective additives leads to higher heat transfer rate, which lowers the corresponding saturation temperature because of less power input to the boiler. It is obvious from Figure 4 that heat transfer additive-A gives much higher average condensation heat transfer coefficients than heat transfer additive-B. It can be clearly seen that heat transfer additive-A is more useful for the purpose of this study. Such a finding could be due to the larger value of $|\partial\sigma / \partial C_{add}|$ for additive-A as can be noticed in Figure 3 but much more careful study may be necessary to understand the detailed physics in the future study.

Although minute but significant changes in surface tension of water/additives starts at approximately 10 ppm (see Figure 3), a noticeable increase in condensation heat transfer truly starts at about 100 ppm (see Figure 4). So, there exists an onset in the range from 10-100 ppm for both additives. One extra run was done with adding 1 percent (10,000 ppm) of heat transfer additive-A (not plotted in Figure 4). An increase of the average heat transfer coefficient by 30 percent was obtained.

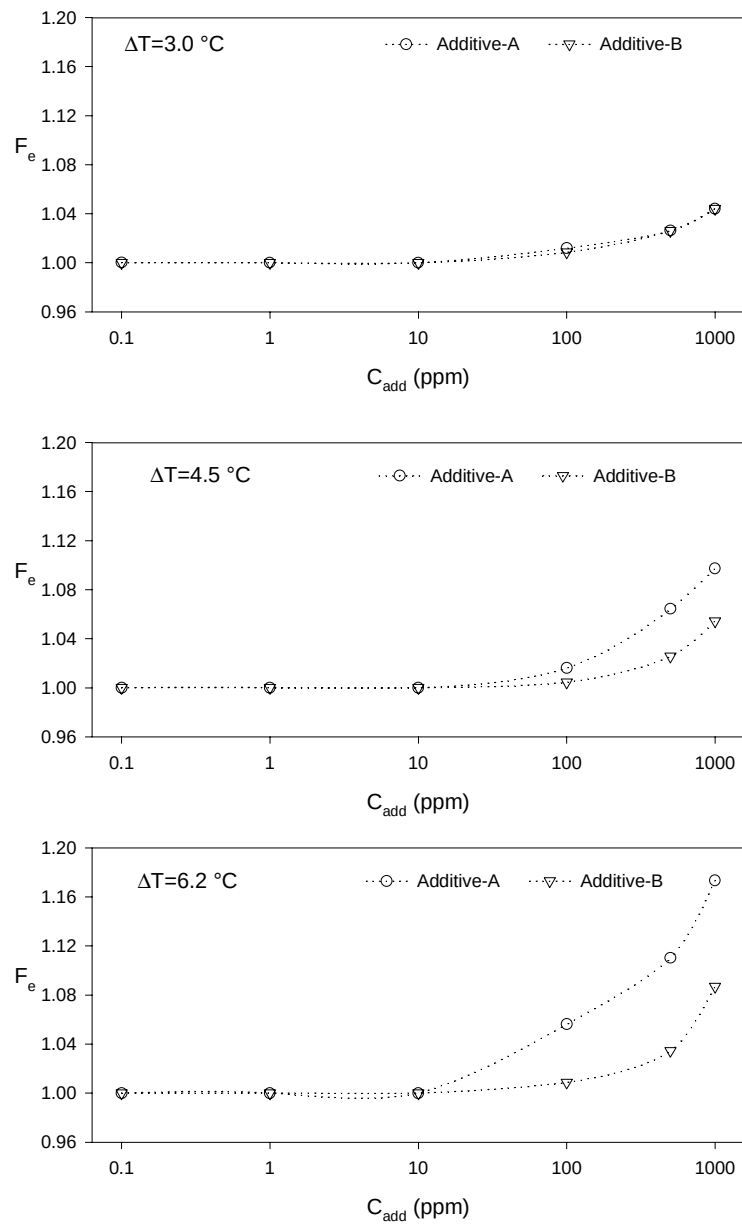


Figure 4. The effect on the heat transfer coefficient by adding heat transfer additives in terms of enhancement factor, F_e . The data were taken for $\Delta T=3.0, 4.5, 6.2\text{ }^{\circ}\text{C}$, respectively. ΔT is the temperature gain in the cooling water.

Another interesting way of revisiting the data presented in Figure 4 is to plot them in terms of enhancement factor, F_e , versus the temperature gain, ΔT , while maintaining the additive concentration the constant value. This method may be indirect but can provide more insights of the effect of heat transfer additives on the steam condensation process. As can be seen in Figure 5, as the cooling water temperature gain, ΔT , increases, the heat transfer coefficients substantially increases only in the presence of additives. Such an interpretation is possible that, as the condensation process accelerates, it could provide more additives at the interface, improving the condensation activity due to interfacial turbulence occurred.

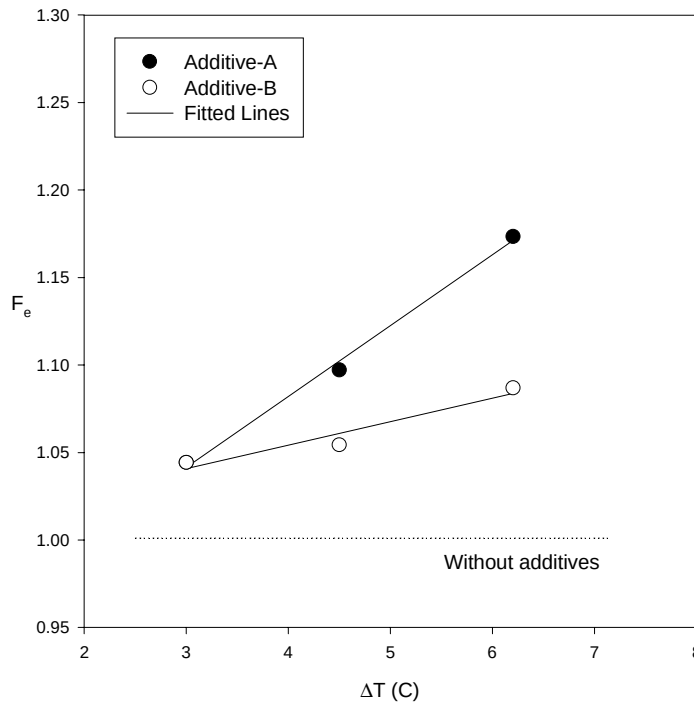


Figure 5. Dependency of heat transfer coefficient on the cooling water temperature gain. These data are plotted in terms of F_e vs. ΔT . Note that the additive concentration is set at 1,000 ppm.

vapor (steam + additive)

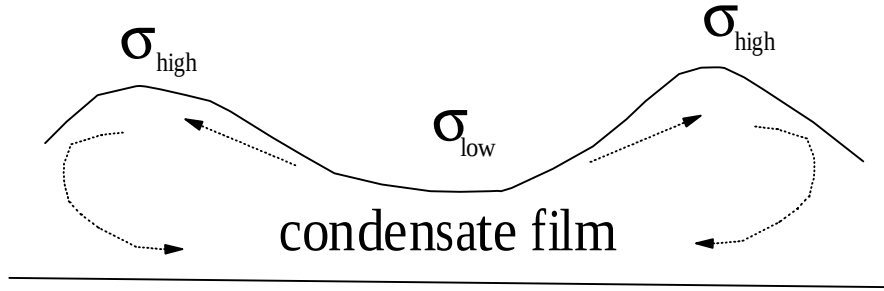


Figure 6. Marangoni instability mechanism in condensate film with additives.

In literature, there are some limited studies on binary-component vapor condensation [16-18]. Overall data is scattered. Nevertheless, the general consensus of these studies is that if local surface tension tends to increase with an increase in film thickness, then the film will be unstable. In other words, if there is a local region of thick film with higher interfacial tension, it will tend to draw liquid from adjacent thin film regions with lower surface tension (see Figure 6, [3-4, 7-8, 21]). In this sense, a general criterion for the Marangoni effect is,

$$\frac{\partial \sigma}{\partial \delta} > 0$$

$$\text{where } \frac{\partial \sigma}{\partial \delta} = \left(\frac{\partial \sigma}{\partial T} \right) \left(\frac{\partial T}{\partial \delta} \right) \quad \text{and/or} \quad \frac{\partial \sigma}{\partial \delta} = \left(\frac{\partial \sigma}{\partial C_{add}} \right) \left(\frac{\partial C_{add}}{\partial \delta} \right) \quad (8).$$

Notations δ , T and C_{add} are the condensate film thickness, its temperature and additive concentration, respectively. Equation (8) indicates that the stability of the condensate film depends on the properties of the mixture ($\partial \sigma / \partial T$ and/or $\partial \sigma / \partial C_{add}$). The fulfillment of Equation (8) can make the condensate unstable so as to enhance the heat transfer process

significantly. Knowing that temperature dependency of surface tension in this case ($\partial\sigma/\partial T$) is secondary, probably the additive concentration dependency ($\partial\sigma/\partial C_{add}$) is believed to be the primary parameter.

For liquids in equilibrium in contact with vapors, the distribution of molecules at the interfacial region is quite different from that in the bulk phase. This indicates that, for some cases, even the compositions differ. When a newly born surface is created for a finite time, it is required to reach equilibrium. Then, surface tension varies during this period [15]. When surfactants are present, the aging of the surface results in decreasing surface tension with time. Considering the surfactant diffusion process into the newly born surface of a condensate, the following simple diffusion equation is valid:

$$D_{add} \frac{\partial^2 C_s}{\partial x^2} = \frac{\partial C_s}{\partial t} \text{ with a boundary condition } \frac{dn(t)}{dt} = D_{add} \frac{\partial C_s(x=0, t)}{\partial x} \quad (9),$$

where D_{add} , C_s , x , t , and n are the mass diffusivity of surfactant, the surface excess concentration, the axis perpendicular to the interface, diffusion time, and the number of molecules at the interface. An analytical solution of Equation (9) is possible resulting in the dynamic surface tension, γ , in a simplified form of [19],

$$\gamma = \frac{A}{t^{1/2}} + C_1 \quad (10),$$

containing experimentally determined value of the lumped parameter A and constant C_1 , respectively. The previous study [9] has revealed that the value of A for 2-ethyl-1-hexanol is approximately ~ 4 dyne/cm-sec^{1/2} resulting mass diffusivity, D_{add} , of around 4×10^{-6} cm²/sec that is quite reasonable. The importance of Equation (10) is that the larger value of mass diffusivity of additives can reach the equilibrium value of the surface tension faster so that they can create better mixing or higher instabilities. By characterizing the values of A in Equation (10) for potential additives, we can have a better view of selecting effective ones. A further study on this issue is currently underway.

There are two Marangoni numbers that can arise from the surface tension gradients at the condensation interface,

$$Ma_{add} = -\left(\frac{\partial\sigma}{\partial C_{add}}\right)\left(\frac{\Delta C_{add} L_{ch}}{\mu D_{add}}\right) \text{ and } Ma_T = -\left(\frac{\partial\sigma}{\partial T}\right)\left(\frac{\Delta T L_{ch}}{\mu\alpha}\right) \quad (11).$$

Where μ , α , D_{add} , and L_{ch} represent the dynamic viscosity and thermal diffusivity of condensate, additive diffusivity, and length scale, respectively. Also, realizing that,

$$\frac{Ma_{add}}{Ma_T} \gg 1 \quad (12),$$

an important parameter is Ma_{add} . In order to determine the effect of Marangoni number on the results provided in Figure 4, we use a linear correlation with the Marangoni number based on turbulent scale and address the effect of ΔT , then;

$$F_e = \frac{\bar{h}_a}{\bar{h}_o} \cong k_1 + k_2 \left(\frac{\Delta T_i}{\Delta T_o} \right) Ma_{add} Pr^{-1/2} \quad (13),$$

$$\begin{aligned} k_1 = 1 \text{ and } k_2 = 0; \text{ when } \left(\frac{\Delta T_i}{\Delta T_o} \right) Ma_{add} Pr^{-1/2} < 10^5 \\ k_1 = 1 \text{ and } k_2 = 4.337 \times 10^{-8}; \text{ when } \left(\frac{\Delta T_i}{\Delta T_o} \right) Ma_{add} Pr^{-1/2} > 10^5 \end{aligned} \quad (14).$$

Where k_1 and k_2 are experimentally determined parameters. ΔT_o is an arbitrary reference temperature gain ($\Delta T = 6.2^\circ\text{C}$ was chosen in this analysis). The parameters, k_1 and k_2 include the effect of condenser geometry and dynamic surface tension effect and Pr is the Prandtl number. Figure 6 shows such a correlation.

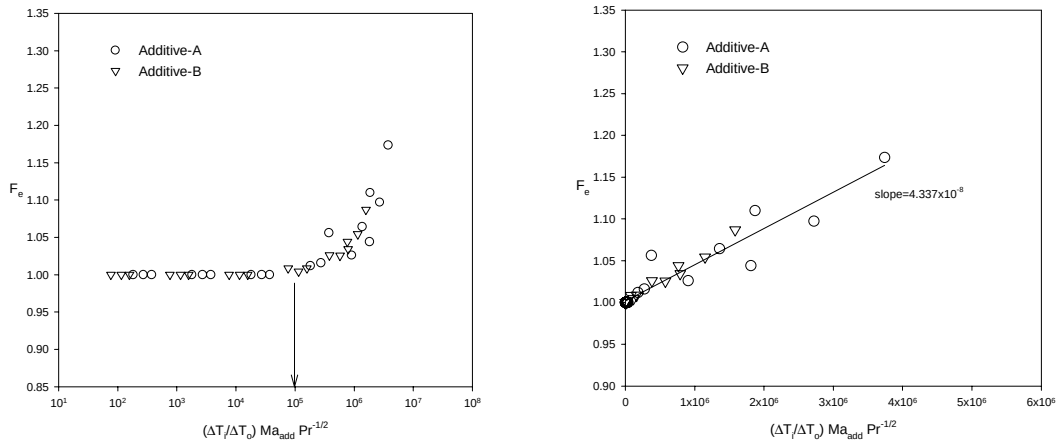


Figure 6. The correlation shows Equation (13). **The x-axis was plotted based upon the log (left) and linear scale (right). Note that $R^2=0.991$.**

In order to validate the data scatter, the generalized uncertainty analysis [20] was performed on Equation (6). Based upon measurements and instrumentation sensitivities, the average condensation heat transfer coefficient uncertainty was found to be approximate 3 percent.

4. CONCLUDING REMARKS

Experimental studies of steam condensation to a bundle of horizontal tubes with small amounts of effective additives were successfully performed to evaluate their effect on condensation heat transfer and to identify the onset of heat transfer enhancement. An appropriate heat transfer correlation was proposed. An increase in condensation heat

transfer of up to 30% was achieved by adding only 1% of 2-ethyl-1-hexanol. This additive was found to be more effective than alkyl primary amine.

Due to interfacial turbulence, the true condensation heat transfer enhancement mechanism is caused by local variation in the surface tension. This means that it is primarily related to the additive concentration dependence of surface tension. In addition, the importance of dynamic surface tension of the water/additives system was appropriately addressed. In the presence of effective additives, the steam condensation process observed in our experiment produced a mostly *dropwise-like* condensation. The condensate water droplets were of a smaller shape than those produced without additives and became more dispersed, exhibiting much higher frequencies. These droplets, unlike traditional turbulence, start at the top portion of the condenser tubes and cover most of the tubes, hanging at the bottom in small droplet sizes. We also observed that film substrates appear to be thinner compared to those produced in the steam condensation process without additives. Such a flow condition was quite attractive for condensation in the absence of smooth, side-drainage, and splashing types. Overall, we believe that added surfactants play a pivotal role in improving dropwise-like condensate activity and provide a significant benefit for condensation heat transfer.

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