Final Report

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

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ABSTRACT

The final report for the DOE/UNM grant number DE-FG26-98FT40148 discusses the accomplishments of both the theoretical analysis of advanced power cycles and experimental investigation of advanced falling film heat exchangers. This final report also includes the progress report for the third year (period of October 1, 2000 to September 30, 2001).

Four new cycles were studied and two cycles were analyzed in detail based on the second law of thermodynamics. The first cycle uses a triple combined cycle, which consists of a topping cycle (Brayton/gas), an intermediate cycle (Rankine/steam), and a bottoming cycle (Rankine/ammonia). This cycle can produce high efficiency and reduces the irreversibility of the Heat Recovery Steam Generator (HRSG) of conventional combined power cycles. The effect of important system parameters on the irreversibility distribution of all components in the cycle under reasonable practical constraints was evaluated. The second cycle is a combined cycle, which consists of a topping cycle (Brayton/gas) and a bottoming cycle (Rankine/ammonia) with integrated compressor inlet air cooling. This innovative cycle can produce high power and efficiency. This cycle is also analyzed and optimized based on the second the second law to obtain the irreversibility distribution of all components in the cycle. The results of the studies have been published in peer reviewed journals and ASME conference proceeding.

Experimental investigation of advanced falling film heat exchangers was conducted to find effective additives for steam condensation. Four additives have been selected and tested in a horizontal tube steam condensation facility. It has been observed that heat transfer additives have been shown to be an effective way to increase the efficiency of conventional tube bundle condenser heat exchangers. This increased condensation rate is due to the creation of a disturbance in the liquid condensate surround the film. The heat transfer through such a film has increased due to the onset of Marangoni convection as well as the population of “dropwise-like” condensation increased. The results have been published in peer reviewed journals.

TABLE OF CONTENTS

	Page
EXECUTIVE SUMMARY.....	5
 PART I: Thermodynamic Analysis and Optimization of New Combined Power Cycles	
1. Second-Law Analysis and Optimization of New Combined Power Cycles ...	7
1.1 Second Law Analysis and Optimization of Combined Triple Power Cycle...	7
1.2 Second Law Analysis and Optimization of a Combined Cycle with Integrated Compressor Inlet Air Cooling.	8
1.2.1 Cycle Description and Analysis	8
1.2.2 Typical Results and Discussion	9
1.3 Conclusions	10
1.4 Literature Cited.....	11
 PART II: Experimental Investigation of Advanced Falling Film Heat Exchangers....	
2. Introduction	17
2.1 Background	17
2.1.1 Dropwise Condensation.....	18
2.1.2 Marangoni Effect.....	21
2.2 Experimental Setup	24
2.2.1 Introduction	24
2.2.2 Single Tube Setup	24
2.2.3 Multiple Tube Setup	25
2.2.4 Surface Tension Setup	26
2.2.5 Additive Selection	27
2.3 Results and Discussion	28
2.3.1 General Description	28
2.3.2 Heat Transfer Coefficient Calculation	29
2.3.3 Single Tube Results	30
2.3.4 Multi-tube Results	30
2.3.5 Surface Tension Calculation	31
2.3.6 Surface Tension Results	31
2.4 Conclusions	32
2.5 References	32
 APPENDIX I	
A copy of the paper presented at ASME conf. "SECOND LAW ANALYSIS OF A COMBINED CYCLE WITH INTEGRATED COMPRESSOR INLET AIR COOLING"	56

EXECUTIVE SUMMARY

Excellent progress was made during the third year of the DOE HBCU/OMI grant number DE-FG26-98FT40148, which was the period from October 1, 2000 to September 30, 2001. The final report includes the results of research for three years including the third year. The research and training of students are concerned with two main areas:

- 1: Thermodynamic Analysis and Optimization of New Combined Power Cycles**
- 2: Experimental Investigation of Advanced Falling Film Heat Exchangers**

Two new thermodynamic cycles were proposed and investigated based on the second law of thermodynamics. Two computer programs were developed to find effect of important system parameters on the irreversibility distribution of all components in the cycle: 1) The first cycle is based on a combined triple (Brayton/Rankine/Rankine)/(Gas/steam/ammonia) cycle capable of producing high efficiencies. 2) The second cycle is a combined (Brayton/Rankine)/(gas/ammonia) cycle with integrated compressor inlet air-cooling capable of producing high power and efficiency. The proposed cycles and the results obtained from the second law analyses of the cycles have been published in a journal (Energy Conversion and Management) and ASME proceedings (IMEC&E 2001).

Experimental investigation of advanced falling film heat exchangers was conducted via three different pieces of experimental apparatus that were constructed in order to obtain the empirical data necessary for analyzing and describing heat transfer characteristics inside a condenser chamber. First, a single vertical tube experiment was constructed and utilized in order to make preliminary measurements of heat transfer coefficients using additives vs. steam alone. Second, with the construction of a horizontal multi-tube condenser (horizontal tube bundles are predominantly used in industry), four chemical additives were tested. It has been observed that inducing the Marangoni convective effect is highly dependent on the surface tension behavior of additives. Third, in order to show the relationship between the increased heat transfer rate and the surface tension due to additive/water solution, an apparatus was constructed to measure the surface tension effects on water due to the additives. Collecting both heat transfer and surface tension data provides more insight for a correlation between surface tension and heat transfer coefficients. Based upon these work, two journal publications have been produced (Applied Thermal Engineering and International Journal of Environmentally Conscious Design and Manufacturing).

Interaction with the Public service company of New Mexico (PNM) was helpful for participants to see the practical and economical aspects of power production. Our energy group has participated in Reeves power plant tour in Albuquerque and San Juan coal power plant in New Mexico.

Dr. Razani (PI) has participated in the three Annual (1998, 1999, 2000) HBCU/OMI symposia and presented the results of research under this grant. The summary of activities in the symposia was presented the energy group at our group meetings. A total of eleven undergraduate and graduate students participated in the research under this grant and most have graduated. The participants are:

Mr. Aaron Stone, BS Mechanical Engineering, 1999
Mr. Antonio Gonzales, BS Mechanical Engineering, 1999
Ms. Blanca Montoya, BS Mechanical Engineering, 1999
Mr. Irvin Marrero, MS Mechanical Engineering, 1999
Mr. Arne Lefsaker, BS/MS Mechanical Engineering, 2000
Mr. Jason Paquette, BS, Mechanical Engineering, 2001
Ms. Terresa Martinez, BS Mechanical Engineering, 2001
Mr. Robert Smith, Mechanical Engineering undergraduate student
Mr. Mike Patterson, Mechanical Engineering undergraduate student
Mr. Jonah Dunham, Mechanical Engineering undergraduate student
Mr. Aaron Stone, Mechanical Engineering graduate student
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PART I:

Thermodynamic Analysis and Optimization of New Combined Power Cycles

1. Second-Law Analysis and Optimization of New Combined Power Cycles

Two new combined power cycles were proposed and analyzed based on the second law of thermodynamics. To find the components and processes responsible for the inefficiency of the cycles, the effect of different system parameters of the irreversibility distribution of all components were obtained. The cycles and the summary of important results and accomplishments are given in this section. The detail of the cycles and results were reported previously (Marrero, 1999; Lefsaker, 2000; Marrero, et al., 2002; Razani, et al., 2001; Razani and Kim, 2000).

1.1 Second Law Analysis and Optimization of Combined Triple Power Cycle

The schematic of the proposed triple cycle is given in Figure 1.1. In the proposed triple (Brayton/Rankine/Rankine)/(Gas/Steam/Ammonia) power cycle the exhaust gases of a Brayton gas topping cycle is used to produce superheated steam in a Heat Recovery Steam Generator (HRSG). This superheated steam is used in a Rankine steam intermediate cycle to produce power. A condenser/evaporator (steam/ammonia) connects the Rankine steam intermediate cycle to a Rankine ammonia bottoming cycle completing the triple cycle. 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 (gas/steam) combined power cycle. In order to optimize and analyze the triple combined power cycle based on the second law, a computer program using EES® software (Flow Chart, 1995) was developed. Through exergy analysis of the triple cycle, the exergy of exhaust streams and the irreversibility of each component in the cycle are determined. 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 fuel mass flow rate is set at 1 Kg/S and assumed to be natural gas with equivalent chemical formula of CH_{3.88}. **The goal of the study is to find what configuration will achieve a thermal efficiency of 60 percent when reasonably practical constraints for system components are used.** These constraints are mainly related to the size of the system components and are conveniently parameterized as input to the computer program. The triple cycle was analyzed and optimized using the important system parameters such as gas topping cycle pressure ratio, gas topping cycle turbine inlet temperature, HRSG pinch point temperature, gas/steam approach temperature difference, gas/ammonia approach temperature difference, steam/ammonia (condenser/evaporator) temperature difference, mass flow rate of steam injected into combustion chamber, effectiveness of heat exchangers, and the environmental temperature. The irreversibility of each component in the cycle was evaluated and the irreversibility distribution of the cycle components was determined. The effect of steam injection into the combustion chamber of the gas topping cycle and gas turbine inlet temperature on the irreversibility of combustion process and the efficiency of the triple cycle was also evaluated (Lefsaker, 1999 and Marrero, et al., 2002). Most of the results were obtained under the fixed steam condition at the exit of HRSG ($T = 538\text{ }^{\circ}\text{C}$ and $P = 160\text{ bar}$). The effect of steam pressure in HRSG and the position of the feed water heaters on system performance were evaluated (Lefsaker, 1999). The position of steam/ammonia

condenser/evaporator is fixed at steam condensation temperature of 100 °C corresponding to 1 atm. The effect of pinch point and gas/steam approach temperature difference on the irreversibility of HRSG and the cycle efficiency was investigated. As expected, the irreversibility of the combustion chamber is by far the largest followed by the magnitudes of the irreversibility of gas turbine and HRSG. It is shown that the efficiency of 60 percent for the triple cycle is possible but the increase over this 60 percent efficiency will be marginal (considering practical constraints) unless innovative methods are used to reduce the combustion irreversibility.

1.2 Second Law Analysis and Optimization of a Combined Cycle with Integrated Compressor Inlet Air Cooling

1.2.1 Cycle Description and Analysis

Since this cycle has not been discussed in previous progress reports and was investigated in the third year of the grant, a more detail of the analysis and results will be reported in this section. A flow diagram of the gas/ammonia combined cycle with an integrated ammonia vapor compression ammonia cycle is given in Figure 1.2. The Brayton topping cycle includes two compressors with an air intercooler in-between and a recuperator to preheat the air using turbine exhaust gases. The exhaust gases from the recuperator produce superheated ammonia. In addition to the ammonia turbine and condenser, the ammonia Rankine bottoming cycle includes ammonia preheater to recover the energy from the exhaust of the ammonia turbine. The ammonia enthalpy from the turbine exit may be high enough to boil the ammonia liquid entering the preheater. The ammonia is superheated in the Heat Recovery Ammonia Generator (HRAG). HRAG is the component connecting the Brayton gas topping cycle and Rankine ammonia bottoming cycle. The ammonia exit from the preheater is condensed in an ammonia air condenser. Ammonia exit from the condenser is assumed to be saturated liquid, a fraction of which is pumped to high pressure and the remaining is expanded in a Joule Thompson expansion valve before entering the ammonia evaporator. The ammonia evaporator is used to cool the inlet air to the compressor for the purpose of increasing the power output of the gas turbine cycle. The ammonia exit from the evaporator is assumed to be saturated vapor and is compressed to the condenser pressure. It is then combined with the exhaust of the ammonia preheater before entering the ammonia condenser. Exergy analysis and second law analysis and optimization of thermal systems are well-established methods of power system design in the fields of thermal science and engineering and are not discussed here. In general, conservation of mass and energy in addition to the balance of exergy are written for each system component. The irreversibility of each component is determined and the exergy entering and leaving the system is evaluated. For the cycle under investigation in this paper, these equations were written for each system components using EES software (F-chart, 1995).

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 78.10% N₂, 20.76% O₂, 1.11% H₂O, and 0.03% CO₂. The volume flow rate of air is set at 100 m³/s and the fuel is assumed to be natural gas with an equivalent chemical formula of CH_{3.88}. The chemical and physical exergy for the inlet and exit streams of each component are determined based on the reference environment mentioned above (Bejan, et al., 1996). The fuel is assumed to enter the system at a pressure of three atmospheres and the temperature of the environment. Therefore, a fuel compressor must be used to increase the pressure of the fuel to

the operational pressure of the combustion chamber. In general, overall thermal efficiency is defined by:

$$\eta_1 = (\text{Net Rate of Energy of Product}) / (\text{Rate of Input Energy of Fuel}) \quad (1.1)$$

The product in the above cycle is the net power produced and the rate of input energy is the product of the Lower Heating Value (LHV) of the fuel and its mass flow rate. The second law efficiency in this study is defined by:

$$\eta_2 = (\text{Net Rate of Exergy of Product}) / (\text{Net Rate of Input Exergy}) \quad (1.2)$$

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 air. The exit streams are the combustion products from the stack exhaust gases and the cooling air discharges from the intercooler and ammonia condenser. It should be pointed out that second law efficiency given by equation (1.2) assumes that the exergy in the exhaust stream is partly recoverable. In practice, the system should be designed assuming the exergy of the exhaust streams is lost. Since the product is net power produced and the input exergy and energy of the fuel are almost the same, in this study, the efficiencies given by both equations would be approximately the same if the exergy of the exhaust streams were assumed to be lost. Therefore, the second law efficiency given by equation (1.2) indicates the contribution of the exergy of the exhaust streams. A high value of the efficiency may point to potentially recoverable lost exergy.

The chemical exergy of inlet and exit streams for each component is calculated using the Table of Chemical Energy of Compounds (Bejan, et al., 1996). The difference in composition of air used in this study and the reference on which the chemical exergy of compounds is used, results in a very small error when energy balance is calculated for the system. This error is found to be smaller than 0.03 percent of the inlet fuel exergy. The exergy balance for the system is evaluated to check the calculation of total irreversibility for the system. It should be pointed out that the chemical exergy of the ammonia inlet and exit streams of each component in the ammonia cycle cancel out when the irreversibility of each component is evaluated. The chemical exergy of the fuel is corrected for the equivalent chemical formula used in this study.

Gas turbines are essentially constant volume machines; any change in the mass flow of air through the turbine affects their output. Mass flow of air is proportional to its density, which is affected by the compressor inlet temperature. In this analysis a fixed volume flow rate to the inlet of the compressor is assumed. Important parameters in the cycle are the pressure ratio, environmental air temperature, turbine inlet temperature, and compressor inlet air temperature. To analyze the effect of the compressor inlet air cooling, its temperature is varied while the volume flow rate of air into the compressor is held constant. The flow rate of input fuel is a variable. It is calculated from the energy and mass balance on the combustion chamber such that a given Turbine Inlet Temperature (TIT) is produced.

1.2.2 Typical Results and Discussion

Table 1.1 gives the nominal values for the system parameters used in this study. For simplicity of analysis and preliminary parametric studies, a nominal value of 0.9 for the turbine and compressor efficiencies is assumed. The important system parameters under investigation are the environmental air temperature to the evaporator inlet (Toa), compressor inlet air

temperature (T_1), gas turbine inlet temperature (TIT), and the gas cycle pressure ratio (Pr). Figures 1.3 to 1.5 show the normalized power as a function of compressor inlet air temperature (T_1) when gas turbine inlet air temperature is fixed (TIT = 1600K), and the environmental air temperature is constant ($T_{oa} = 45^\circ\text{C}$) for three different values of gas cycle pressure ratios of 16, 25 and 36 respectively. The air pressure exit of the first compressor is the square root of the pressure ratio. The first law efficiency (η_1) and the second law efficiency (η_2), as defined by equations (1) and (2), are also given in the same figures. Since the exergy of the exhaust stream is small, the values of the first and second law efficiencies are very close. Even though the optimum conditions are different, the values of the optima are very close. Therefore, other system constraints and economics will dictate the actual system design. Since the air temperature into the ammonia evaporator is cooled progressively from 45°C to 5°C , the net power of the combined cycle increases as air is cooled to lower temperatures before entering the air compressor. The specific power (power per unit volume flow rate) is normalized to a compressor inlet air temperature of 15°C . To find the effect of compressor inlet air cooling, the volume flow rate of inlet air to the compressor is fixed at $100\text{m}^3/\text{s}$. Therefore, as the inlet air is cooled further, the mass flow rate into the gas cycle increases resulting in higher net power output. This, of course, necessitates an increase in the mass flow rate of the fuel for a fixed TIT = 1600K. It should be pointed out that the fuel mass flow rate is obtained from the energy balance on the combustion chamber. The effect of the pressure ratio on the net power produced by the combined cycle is more complex because it influences several aspects of the cycle. As the pressure ratio increases, the exergy out of the intercooler wasted to the environment increases, while the bottoming cycle becomes more efficient.

The role of the recuperator effectiveness in the gas cycle is also important in this regard. The effect on efficiency is clear. As the pressure ratio increases the first and second law efficiencies are reduced. It is interesting to note that there is a shallow optimum for the second law efficiency as compressor inlet air temperature is reduced. The optimum values shift toward lower values of T_1 as Pr is increased. The first law efficiency also has an optimum but it occurs at lower values of T_1 . In fact, the optimum does not occur for $Pr = 36$ even at the lowest value of $T_1 = 5^\circ\text{C}$, used in this study. Compressor inlet air temperatures lower than 5°C requires separation of moisture even for the typical dry air used in this study.

1.3 Conclusions

Two new cycles were proposed and analyzed. It was shown using the exergy analysis that the triple combined cycle made of a topping gas Brayton cycle, an intermediate steam Rankine cycle and an ammonia Rankine bottoming cycle can produce a high efficiency in excess of 60 percent with reasonable practical constraints. The effect of important cycle parameters on the irreversibility distribution of all components in the cycle was determined. The second law analysis and optimization of the cycle show that efficiency improvement beyond 60 percent will be marginal. A new combined cycle with integrated compressor air cooling capable of producing high efficiency and power was proposed. This cycle is made of a topping gas Brayton cycle and a bottoming ammonia Rankine cycle. First and second law analyses of the cycle show that a power boosting of more than 15 percent is possible by compressor air cooling during summer. The irreversibility distribution of all component of the cycle was determined. Both proposed cycles are convenient for air cooling of condenser because ammonia is used in the bottoming cycles. Further studies of both cycles including economic analysis must be performed.

1.4 Literature Cited

1. Bejan, A., 1996, *Entropy Generation Optimization*, CRC Press, Inc., Boca Raton.
2. Bejan A., Tsatsaronis G., and Moran M., 1996, *Thermal Design and Optimization*, John Wiley and Sons, Inc.
3. DeLucia M., Bronconi R., and Carnevale E., 1994, "Performance and Economic Enhancement of Cogeneration Gas Turbine Through Compressor Inlet Air Cooling", ASME Journal of Gas Turbine and Power, Vol. 116, pp. 360-365.
4. Ebeling J.E., Halil R.O., Bantam D., Bakenhaus B., Schreiber J., and Wendland R., 1992, "Peaking Gas Turbine Capacity Enhancement Using Ice Storage for Compressor Inlet Air Cooling", ASME Paper No. 92-GT-265.
5. F-Chart Software, 1995, EES, Engineering Equation Solver, F-Chart Software, Middlton, WI.
6. Lefsaker, A.M., 1999, Second Law Analysis of Combined Power Cycles and Advanced Falling Film Heat Exchangers, BS/MS project in Mechanical Engineering, The University of New Mexico.
7. Kraft J.E., Everlein D., 1999, "Inlet Air Cooling Systems Boost Summer Peaking Capacity at WE", Power, pp. 74-79.
8. Marrero, I.O., 1999, Second Law Analysis and Optimization of Combined Power Plants, MS thesis in Mechanical Engineering, The University of New Mexico.
9. Marrero, I.O., Lafsaker, A.M., Razani, A., and Kim, K.J., 2002, "Second Law Analysis and Optimization of a Combined Triple Power Cycle" Energy Conversion and Management," pp. 1-17, April, 2002, to appear.
10. Razani, A. and Kim, K.J., October 1999, Annual Progress Report, DOE Grant # DE-FG26-98FT40148.
11. Razani, A. and Kim, K.J., October 2000, Annual Progress Report, DOE Grant # DE-FG26-98FT40148.
12. Razani, A., M. Patterson and Kim, K.J., "Second Law Analysis of a Combined Cycle with Integrated Compressor Air Cooling," Proceedings of 2001 ASME International Mechanical Engineering Congress and Exposition, IMECE 2001/AES-23670, PP. 1-9, 2001.

Table 1.1: Nominal Settings for Gas/Ammonia Combined Cycle with Integrated Compressor Inlet Air Cooling

Parameter	Value	Units
Ammonia pump efficiency	0.7	Dimensionless
Ammonia preheater approach temperature difference	4	°C
Gas cycle pressure ratio	16	Dimensionless
Enthalpy of combustion for fuel ($\text{CH}_{3.884}$)	42	MW/kg
Chemical exergy of natural gas	43.68	MW/kg
Volume flow rate of air into gas cycle compressor	100	m ³ /s
TIT	1600	K
Temperature of inlet air to compressor	15	°C
Pinch temperature of ammonia evaporator	10	°C
Ammonia condenser reservoir temperature	Toa	°C
Toa	45	°C
Ammonia turbine inlet pressure	62	Bars
Lower bound on stack exhaust temperature	70	°C
Gas turbine efficiency	0.9	Dimensionless
Ammonia turbine efficiency	0.9	Dimensionless
All compressor efficiencies	0.9	Dimensionless
Recuperator effectiveness	0.9	Dimensionless
Intercooler effectiveness	0.9	Dimensionless
Ammonia evaporator effectiveness	0.8	Dimensionless
Gas/ammonia approach temperature	80	°C
Ammonia boiler pressure	62	bars
Inlet fuel pressure	3	atmospheres

Figure 1.1: Flow diagram of the triple cycle.

Figure 1.2: Flow diagram of the Gas/Ammonia combined cycle with integrated compressor inlet aircooling

Figure 1.3: Normalized power, first and second law efficiencies as a function of compressor inlet temperature, $Pr=16$.

Figure 1.4: Normalized power, first and second law efficiencies as a function of compressor inlet temperature, $Pr=26$.

Figure 1.5: Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, Pr=38.

PART II:

Experimental Investigation of Advanced Falling Film Heat Exchangers

2. Introduction

In this experimental study, heat transfer additives are introduced into the working fluid of a vertical single-tube condenser and a horizontal multi-tube condenser. The additives act to decrease the surface tension of the working fluid thus decreasing the liquid film thickness (substrate) on the condensing surface and also enhancing condensate mixing and ultimately increasing the heat transfer rate of the exchanger. A total of four additives were tested in each apparatus. The dynamic surface tension of each additive was also measured and used to correlate with heat transfer enhancement.

2.1. Background

The most common type of condensation involved in heat exchangers is surface condensation where a cooled surface, at a temperature less than the local saturation temperature of the vapor, is placed in contact with that vapor. In this situation, the vapor molecules that come near the cold surface may stick to it and condense into liquid. The resulting liquid (i.e., condensate) will accumulate in one of two ways. If the liquid “wets” the cold surface, the condensate will form a continuous film, and this mode of condensations is called “filmwise condensation” (Figure 2.1). If the liquid does not “wet” the cold surface, it will form into numerous microscopic droplets. This mode of condensation is called “dropwise condensation.” Dropwise condensation results in much larger heat transfer coefficients than “filmwise condensation” due to the type of heat transfer evoked. Dropwise condensation also promotes higher convective heat transfer rates than filmwise condensation (Figure 2.2). Therefore, numerous studies have been conducted concerning this issue, examples and references to follow.

Typically, condenser heat exchangers are set up as airtight chambers with a bank of horizontal tubes running through the chamber. Cold fluid is pumped through the tubes, while vapor condenses and falls, by force of gravity, on the outside of the tubes. Condenser tubes can be positioned in several ways. Traditionally, tubes are either positioned as an in-line array, or a staggered array (Figure 2.2). The flow of steam over these tubes, also affects the condensation process (Figure 2.3). The tubes in the first few rows act as a turbulence grid, which increases the heat transfer coefficient for tubes in the following rows.

It has been well established that designing for filmwise condensation is a safe design practice because of the difficulty in maintaining dropwise condensation for a sustained period of time (Jakob, 1936). Dropwise condensation is difficult to maintain for three main reasons. First, non-condensable gases (i.e. air with 78% nitrogen, 21% oxygen, 1% other) inside the condenser can significantly hinder condensation rates. Non-condensable gases are carried with the condensable vapor toward the surface of the tubes. Here the vapor changes phase where it becomes a liquid. The partial pressure of these non-condensable gases rise near the vicinity of the condenser tubes as the condensable vapor condenses and falls away. Because the total pressure in the condenser is kept constant, the increase in partial pressure of non-condensable gas tends to decrease the partial pressure of the condensable gas, thus lowering the overall condensation rate (Zhang, 1996). Condenser chambers are evacuated to pressures lower than atmospheric pressure to remove non-condensable gases from the chamber.

A second factor in determining the efficiency of a condenser is the mass flow rate of the cooling water through the tubes. Increasing the mass flow rate through the tubes will increase the heat transfer rate for the cold side of the condenser. The heat transfer for the system will increase until, at some point, an increase in mass flow rate will be inversely proportional to overall heat transfer. This inverse relationship is due to an increased liquid film thickness around the tubes. As the condensation rate increases, so does the film thickness on the tube. As this film thickness increases the condenser efficiency falls drastically. This decrease in heat transfer is due to the high thermal capacity of water as opposed to that of the condensing surface. At some limiting point, an increase in mass flow will have an adverse effect in the condensation rate of the condenser (Zhang, 1996). Also, increasing pumping power would increase the energy input to the system.

The third problem associated with the overall performance of a condenser is called condenser inundation. As condensate film thickness increases, the heat transfer coefficient decreases, and the tube surfaces become inundated with water. As previously mentioned, a condenser tube covered with a thick film of water will have drastically lower heat transfer rates than one without. Numerous researchers have attacked the problem of condenser inundation.

2.1.1 Dropwise Condensation

Once the heat transfer rate is successfully computed an effort is made for its enhancement. Due to factors such as geometry; differing vapor and condensate flows over each tube; differing temperature and pressure gradients throughout the tube bundle, the flow of vapor and condensate along the surface of condenser tubes is complicated. In general, condensers rely on gravity to pull the film loose from the tubes so continued condensation can occur. Taking a force balance around the tubes of a condenser, it can be seen that gravity is at a constant battle with the surface tension of the water surrounding the tubes. Decreasing this surface tension will decrease the film thickness, thus increasing the condensation rate, and ultimately increasing the overall performance of the condenser. Also, inducing turbulent flow around the condenser tubes would increase the overall heat transfer rate inside the condenser by combining convection with conduction within the liquid film. This would allow warmer liquid further from the cooling surface to come in closer contact with that surface.

Dropwise condensation has been proven to be superior to filmwise for several reasons. As steam begins to condense on a cold surface, the surface will become 'wet' (see Figure 2.1). Once the surface is entirely wet the interaction between the vapor and the liquid film need only be considered. That is, the vapor must condense on to a film of liquid, and then heat must be transferred through this film to the cold surface by conduction. Typically, these liquid films have large thermal resistances associated with them and can severely hinder heat transfer rates. For the case of steam and water, heat must be conducted through a film of water with a thermal conductivity value of 0.613 W/mK at a reference temperature of 300 K , where a copper surface has a thermal conductivity value of 400 W/mK at the same temperature. Obviously, a liquid water film acts to insulate the cold surface. In dropwise condensation, small droplets form and coalesce thus continuously leaving a very thin film exposed for further condensation. The smaller the droplet, and the thinner the film, the smaller the thermal resistance associated with conduction (Figure 2.4) Although dropwise condensation produces much higher heat transfer rates than filmwise, it requires promotion. It has been accepted that clean steam, whether or not it contains non-condensable gas, eventually condenses in a filmwise manner on clean surfaces, rough or polished (Jakob, 1936).

In dropwise condensation, there are two different viewpoints regarding the mechanism of the formation of droplets. The purpose of a condenser heat exchanger is to condense a working fluid from a gas or vapor to a liquid. Energy is transferred to the cooling surface during this phase change. The change in enthalpy that the substance experiences during the phase change process is called the latent heat of the substance. The latent heat is defined such that it is equal to the amount of energy required to change the phase of a unit mass of the substance at a given pressure and temperature (Black and Hartley, 1996). During the condensation process, the energy transferred is called the latent heat of liquefaction. After the condensation process, heat is transferred to the cooling surface by thermal conduction through a liquid condensate. It is widely accepted that dropwise condensation yields heat transfer rates significantly higher than filmwise condensation. However, the mechanism for the formation of droplets has been investigated and disputed. Two theories describing this process have been proposed. One theory suggests that a thin unstable film develops on the surface, and then coalesces into droplets (Jakob, 1936; McCormick and Baer, 1963). Another theory states that the drops appear to have the same arrangement on the surface and the existence of particular nucleation sites occurs on the surface (Song et al., 1991). Recent work by Song et. al (1991) supports the theory that a thin film of condensate still exists at the spot of droplet temperature.

Song et al. conducted experiments on the mechanism of dropwise condensation. It was concluded that “a thin film of condensate exists on the area among the droplets and thin film of condensate still exists at the spots of droplet departure” (Song et al., 1991), (Figure 2.3). Testing for the condensate film utilized the reflectance spectrum method. According to the optic principle, when a ray of light projects on a transparent film adhered to the metal, the reflectance is: $R=f(n_1, n_2, n_3, k_3, \theta_1, \lambda, h)$. Where n_1 is the index of refraction gas, n_2 is the index of refraction of film, n_3 is the index of refraction of metal (substrate), k_3 is the extinction coefficient of substrate, θ_1 is the angle of incident light, λ is the wavelength of incident light, and h the film thickness. Using this method, experimenters could detect the existence of a film during the transient dropwise process (Song et al., 1991). Although the mechanisms for dropwise condensation are generally recognized, detailed description of heat transfer process is complicated due to the many variables affecting the process. Also, dropwise condensation is difficult to maintain. Dropwise condensation is attractive, however, because it can often produce heat transfer rates twenty times larger than filmwise. Efforts to induce and maintain dropwise condensation are discussed in succeeding sections. One such way is to decrease the film thickness and size of droplets to allow for greater heat transfer rates. Also, increasing the frequency with which drops leave the surface would allow for more activity or greater amount of steam to condense over a given period of time.

2.1.2 Marangoni Effect

In order to decrease the surface tension of the bulk fluid and induce interfacial turbulence, the Marangoni effect is employed at the surface of the tubes. Employing this effect is an inexpensive, and environmentally conscious method for the reduction in film thickness, and improvement of surface wetting along the condenser tubes. This method would increase condensation rates and allow for the other possible improvements, such as, increasing the flow rate of cold water to a new upper bound.

The Marangoni effect, often called interfacial turbulence or surface convection, is an important transport mechanism in various interfacial heat and mass transfer processes. It is understood that this film turbulence is caused by local variations in the interfacial tension. Usually, an interfacial tension gradient causes the intensified convection flows and leads to a

substantial increase in heat and mass transfer rate. An investigation of the Marangoni effect and how to induce it will follow.

The Marangoni instability in a two-phase flow leads to the loss of hydrodynamic stability at the interface and a spontaneous periodic convection pattern or chaotic behavior develops at the interface. These two different types of surface flows have been reported in numerous studies. As mentioned earlier, one is the spontaneous development of periodic convection patterns that show a high degree of periodicity and a coherent structure, in most cases called “roll cells,” or “structured convective cells.” The other is the chaotic pulsations, which manifest marked random motions in a turbulent fluid. Both structured and chaotic convection motions are able to improve interfacial transport significantly (typically 2-3 times higher transfer coefficients) (Kim et al., 1996). Ford and Missen’s work on the binary-component vapor condensation was concentrated on investigation of a general criterion for the Marangoni effect (1968). Other studies include the condensate films for pentane-methanol and pentane-methylene dichloride (Mirkovich and Missen, 1963) and ammonia/water (Morrison and Deans, 1997). The general conclusion of these studies is that if a local surface tension tends to increase with an increase in film thickness, then the film will be unstable.

To begin, Mirkovich and Missen (1963) offer some motivation for this approach. Comparisons for condensation among several binary vapors and pure substances were made in their paper. It was reported that several binary vapors displayed properties of non-filmwise condensation. In fact the binary vapors were described as a “pure” substance exhibiting dropwise condensation. However, the condensing surface required no “promotion” for binary vapors to condense in a dropwise fashion as it would for pure substances (Ford and Missen, 1968). This shows dropwise condensation as a direct result of the Marangoni effect.

The Marangoni effect is characterized by movement of fluid at the surface caused by local variations in the surface tension. If a region of the condensate film is at relatively high surface tension and is in contact with a region of lower surface tension, the region of high tension spontaneously tends to contract in order to lower the surface free energy. This movement of fluid is often visible and can be described by turbulent flow of condensate around the tubes (Ford and Missen, 1968). Ford and Missen (1968) considered a falling film to be inherently unstable and set out to prove how the Marangoni effect provides a criterion for stability with respect to surface tension fluctuations. That is, in an inherently unstable fluid, these instabilities could grow or decay with film movement caused by surface tension. First, if thin regions of the film are of higher surface tension, the film is stabilized by contraction of the region and spreading of the thick regions so as to create a more uniform thickness. On the other hand, if the thick regions are of higher surface tension, their contraction will accentuate the non-uniformity and the film will give the appearance of instability (Figure 2.16). Ford and Missen show the latter to be the case. As this non-uniformity increases the condensate flow will approach drop-wise condensation.

The next issue would be to investigate how to artificially induce the Marangoni effect. Obviously several methods of triggering surface tension gradients along an interface exist, such as changes in thermal physical properties of the bulk fluid (i.e. temperature, pressure, viscosity, etc), or chemical instabilities in the bulk fluid. More importantly than just triggering this effect is to describe this flow in terms of heat and mass transfer inside a condenser. The available literature provides some insight into this problem. Some of it is discussed here with appropriate references.

From an engineering design point of view, it is most desirable that the results of mass transfer can be analyzed, generalized, and formulated in a convenient form for calculations (Lu

et al., 1997; 1996). Several different researchers have extensively studied this issue of triggering the Marangoni effect. The problem of describing mass transfer is dependent on several factors, and is thus difficult to determine. Sternling and Scriven made initial studies in 1959. They set out to determine what is the mechanism of interfacial turbulence and in what way can it be recognized. Their purpose was to develop a model to predict the effect of interfacial turbulence on the rate of mass transfer between phases. It was already well known that longitudinal variations of surface tension not only cause movement in the liquid surface but also bring forces to bear on the underlying liquid, setting it in motion (Sternling and Scriven, 1959). Sternling and Scriven proposed a simplified model that lays the groundwork for handling more complete models. This model is dependent on diffusivity and concentration of the surfactant, and the viscosity of the bulk fluid. These parameters are synonymous with concepts of hydrodynamics because interfacial turbulence involves gross fluid motion. It was found that not only was interfacial tension dependent on the fore mentioned parameters, but also contingent on the direction in which interfacial tension changes with solute concentration. Several complications arose in the development of the model (a linear stability analysis) and it was concluded that the analysis is based on a model too simplified to be reproduced in the laboratory.

The paper by Wolf and Stichlmair (1996) investigate mass transfer as it relates to the Marangoni effect. The objective of the papers was to quantify the influence of the Marangoni-convection on mass transfer rate and to include the shear stress at the interface in the development of criteria for the prediction of the onset of the interfacial convection. According to Wolf et al. Sternling and Scriven developed the best-known model based on linear stability analysis in 1959. This model was investigated and refined by Wolf and Stichlmair. Their results of the experiments reveal that not only the direction of mass transfer but also the concentration difference and the velocity at the interface are decisive for the onset of Marangoni-convections. The criteria of Sternling and Scriven developed for quiescent bulk phases are not able to reliably predict the occurrence of interfacial instabilities for flowing bulk phases.

Other methods of determining mass transfer due to Marangoni effect were proposed by Ruckenstein et al in 1970. Ruckenstein recognized that the induced flow was unsteady. For this reason they modified the physical situation by continuously feeding the drop into the supporting liquid (water). In this manner, a quasisteady velocity field was sustained outside the drop and it may be treated theoretically and experimentally in a simpler manner. Ruckenstein described his simplified flow by simultaneously solving the convection-diffusion equation along with the Navier-Stokes equations. Ruckenstein concluded that his treatment was valid for not only partially miscible liquids, but also for completely miscible ones (Ruckenstein et al., 1970).

Lu, Yang, and Maa (1996) studied what effect the artificially provoked Marangoni convection had at a gas/liquid interface on absorption of CO₂. Marangoni convection was provoked by continuously dissolving aqueous binary mixtures of n-alcohols at the water surface. The alcohols used were methanol, ethanol, and n-propanol. Overall mass transfer coefficients were then evaluated as a function of the concentration of each alcohol. The spreading liquids were delivered into the supporting liquid by a small capillary tube at the surface of the liquid water, in the exact way that Ruckenstein had previously done. The spreading of the liquid film is due to the surface tension gradient. This surface tension produces a shear stress in the fluid that leads to fluid motion in the vicinity of the spreading liquid. In order to describe the flow induced by the delivery of an alcohol the convective-diffusion equation must be solved together with the Navier-Stokes equations (not shown)

subject to the boundary condition that the radial surface tension gradient is balanced by the shear stress in the fluid (Lu et al., 1996).

Lu also showed that the change in surface tension with respect to the radius of the tube, and thusly the film thickness, depended not only the concentration but also the species of the spreading liquid. Furthermore, in addition to Ruckenstein, a Reynolds number of interfacial turbulence was proposed in order that the induced Marangoni convection by different spreading liquids could be quantitatively evaluated.

$$Re = \frac{\rho u_{i,ave} R}{\mu}, \quad (2.1)$$

where ρ is the fluid density, $u_{i,ave}$ is the average velocity, i denotes species, and μ is the fluid viscosity. In general the value of the Reynolds number increases as the concentration of the spreading liquid is increased. Finally, it was concluded that significant enhancement of the absorption rate can be achieved by this approach. The effectiveness of absorption enhancement increases with concentration of spreading liquid (Lu et al., 1996).

Lu, Yang, and Maa closely studied inhibitions to the induced Marangoni effect in which aqueous solutions of three different types of surfactants (additives) over relatively wide ranges of concentration were used as supporting (absorbing) liquids. Their goal was to identify inhibition mechanisms of surfactants on artificially induced Marangoni convection. Lu found that mass transfer rates with artificially induced Marangoni convection could be as much as 3 to 4 times larger than those without it. They found the overall resistance to mass transfer due to surfactants could be described by resistance to absorption of surfactants in the liquid phase. The absorption of surfactants tends to spread the interface thus increasing surface tension in that area and producing a restorative force to the induced Marangoni convection (Lu et al., 1996).

Vazquez, Navaza, and Santos (1995) studied the absorption of CO_2 by water and surfactant solutions in the presence of induced Marangoni effect. The Marangoni effect was induced by the deposition of several low surface tension liquids on the liquid-gas interface. These convection-inducing liquids were 20-100% aqueous solutions of methanol, ethanol, 1-propanol, 2-propanol or acetone. Vazquez showed that the absorption rates for CO_2 in the presence of induced convection were between 3 and 4 times higher than the rates measured in its absence (Vazquez et al., 1995).

Another method of inducing turbulent flow is also presented. Mendes-Tatsis and De Ortiz (1996) presented mechanical instabilities in liquid-liquid systems caused by chemical reaction instabilities in the liquid. They investigated claims that a necessary but not sufficient condition for interfacial instability is the instability of the reaction itself. The stability of a reaction between a surfactant and a metal ion was studied. It was concluded that chemical systems could cause film instabilities in either direction of the reactions (Mendes-Tatsis and De Ortiz, 1996).

Wasan, Nikolov, and Brenner (2001) investigate interfacial turbulence driven by temperature gradients of a liquid condensing on a surface. Three different mechanisms of droplet flows are discussed, and an expression for the bulk fluid velocity due to surface tension is given. Maximum fluid velocity can be obtained by forcing capillary flow in the same direction of the induced Marangoni flow. Modifying the solid surface so that a droplet grown on a hotter hydrophobic portion of the surface moves to the cooler hydrophilic portion of the surface does this. Due to non-uniform surface tension throughout the drop the curvature of the advancing meniscus becomes greater than that of the receding meniscus, and capillary pressure moves the droplet further. The expression for the velocity due to the Marangoni effect is:

$$V_{\max} = \frac{d}{2\mu} \frac{d\sigma}{dL}. \quad (2.2)$$

Where d is the thickness of the spreading droplet, μ is the viscosity of the droplet liquid, σ is the surface tension, and L is the drop length. This expression ignores the gravitational and capillary contribution and is only a balance between the shear forces at the gas-liquid surface and the tractive forces due to the surface tension gradient (Wasan, et al., 2001).

Overall data regarding the Marangoni-induced heat and mass transfer phenomena in the literature 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 (Figure 2.16). In this sense, a general criterion for the Marangoni effect is,

$$\frac{\partial\sigma}{\partial\delta} > 0 \quad (2.3)$$

$$\text{where } \frac{\partial\sigma}{\partial\delta} = \left(\frac{\partial\sigma}{\partial T} \right) \left(\frac{\partial T}{\partial\delta} \right) \quad \text{and/or} \quad \frac{\partial\sigma}{\partial\delta} = \left(\frac{\partial\sigma}{\partial C_{add}} \right) \left(\frac{\partial C_{add}}{\partial\delta} \right). \quad (2.4)$$

Notations δ , T and C_{add} are the condensate film thickness, its temperature and additive concentration, respectively. The above equation 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 that equation 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; 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 (Adamson, 1983). 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:

$$D_{add} \frac{\partial^2 C_s}{\partial x^2} = \frac{\partial C_s}{\partial t} \quad \text{with a boundary condition} \quad \frac{dn(t)}{dt} = D_{add} \frac{\partial C_s(x=0, t)}{\partial x}, \quad (2.5)$$

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 the diffusion equation is possible resulting in the dynamic surface tension, γ , in a simplified form of (Adamson, 1983),

$$\sigma = \frac{A}{t^{1/2}} + C_1, \quad (2.6)$$

containing experimentally determined value of the lumped parameter A and constant C_1 , respectively. The previous study (Adamson, 1983) 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 the surface tension equation is that the

larger the value of mass diffusivity of additives, the faster the equilibrium value of surface tension can be reached so that they can create better mixing or higher instabilities. By characterizing the values of A in surface tension equation for potential additives, we can have a better view of selecting effective ones.

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

$$Ma_{add} = -\left(\frac{\partial\sigma}{\partial C_{add}}\right)\left(\frac{\Delta C_{add} L_{ch}}{\mu D_{add}}\right) \text{ and } Ma_T = -\left(\frac{\partial\sigma}{\partial T}\right)\left(\frac{\Delta T L_{ch}}{\mu\alpha}\right). \quad (2.7)$$

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,

$$\frac{Ma_{add}}{Ma_T} \gg 1, \quad (2.8)$$

as an important parameter is Ma_{add} .

2.2 Experimental Setup

2.2.1 Introduction

Three different pieces of experimental apparatus were constructed in order to obtain the empirical data necessary for analyzing and describing heat transfer characteristics inside a condenser chamber. First, a single vertical tube experiment, (Figure 2.5) was constructed and utilized in order to make preliminary measurements of heat transfer coefficients using additives vs. water alone. The single tube was constructed for two reasons. The single tube can be considered a simplified or scaled down version of a multi-tube condenser typically seen in industry. Also, the single tube condenser was relatively inexpensive to construct. By making experimental measurements using the single tube, enough data was generated to justify the need for a more realistic version of a multi-tube condenser. Therefore, the experimental results from the single tube provided the necessary criteria for the construction and implementation of a multi-tube condenser in the energy lab (Figure 2.6).

With the construction of a horizontal multi-tube condenser (horizontal tube bundles are predominantly used in industry), four chemical additives were selected to test the Marangoni phenomena. Comparative measurements were made among all four additives and the case where no additive was present inside the condenser. Finally, as mentioned previously, inducing the Marangoni effect is highly dependent on the surface tension behavior of additives. Also, in order to show the relationship between the increased heat transfer rate and the surface tension due to additive/water solution, an apparatus was constructed to measure the surface tension effects on water due to the aforementioned additives. Collecting both heat transfer and surface tension data provides more insight for a correlation between surface tension and heat transfer coefficients.

2.2.2 Single Tube Setup

Figure 2.5 shows the setup of the vertical, single tube condenser. The condenser design was a closed system made from clear Lexan™ plastic to allow for the visualization of the condensation process along the entire length of the tube. Steam is being generated at the boiler with a voltage controller by an immersion heater with a total heating capacity of 7,200 W. The copper tube (a minor portion zinc) is filled with turbulators in order to insure turbulent flow through the condensing tube. The turbulators are approximately 1/8 inch square thin

perforated copper pieces with a half circle shape. When the steam hits the cold copper tube, it condenses and falls back to the boiler. The entire condenser chamber was evacuated to a vacuum pressure of $15(\pm 0.5)$ in Hg. 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 gases inside the condenser chamber. The condenser tube was 16 mm outside diameter with wall thickness of 2 mm. The entire system was properly insulated. The apparatus was fitted with five type-T thermal couples in order to take temperature measurements. A thermal couple was placed at the inlet and outlet of both the hot (steam) side, and the cold side. A thermal couple was placed on the boiler. The thermal couples were connected to a computerized data acquisition system (LabView™) where the temperatures were monitored and stored. LabView™ hardware setup includes a MAC-MIO-16 board and a temperature terminal connected to a SCXI-1000. A pressure gage was positioned inside the chamber to monitor the pressure conditions. Initial experiments were conducted with deionized water to allow researchers to familiarize themselves with the apparatus, and to gain a reference frame in which to measure the additives against. In addition to the original apparatus, a chiller was built and connected to the cooling water of the condenser. A twenty-gallon thermal reservoir was also added for cooling water circulation.

The working fluid was propelled by gravity and pressure gradients. Once the system had been evacuated, the boiler was started along with the chiller. The boiler would convert the working fluid to a gaseous phase where, by pressure driven forces in the boiler, it would travel up from the boiler, to the top of the condenser chamber. Steam would condense on the vertical tube and fall, by force of gravity, back to the boiler beneath. When the steam generated enough pressure to enter the chamber, the cooling water pump was started. The cooling water was run in two separate loops into and out of the 25-gallon reservoir. One loop went through the chiller. The chiller was manufactured by Elkay® model number ERW32 1A. The water was pumped through the chiller with a submersible utility pump, Tempest 2, 1/6 HP. This maintained the reservoir temperature at $9 (\pm 0.5) ^\circ\text{C}$. The chiller was supplied with cooling water from the tap, which was disposed of at the sink drain. The other loop went to the condenser. This water was also pumped by a submersible utility pump, Tempest 2, 1/6 HP. The water flowed through PVC pipes, 12.7 mm inner diameter, to the condenser, and then back to the reservoir. Two valves on the PVC tubing controlled the flow rate. Primer and PCV cement was used to ensure tight and no leakage fittings between the tubes and valves. A rotameter from Omega® model number FL4205 with a range of 0 to 60 GPH was added to this loop to control and measure the volumetric flow rate.

2.2.3 Multiple Tube Setup

The most common condenser design in the power industry is to use a bank of horizontal smooth tubes. In most horizontal tube condensers, the condensate usually flows as a thin film down the outside of each tube (Figure 2.7). The condensation takes place at the interface between condensate and vapor while cold fluid is circulating through the tubes. The characteristics of the condensate flow primarily depend on the condensate flow rate. A small condensate rate results in droplets or long tails for the condensate flow. Therefore, key engineering objectives are:

- i) To obtain the complete coverage of the condensate over the heat transfer surface
- ii) 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 available, they are rather expensive or difficult to manufacture. For a bank of tubes, the heat transfer coefficient differs for each tube since the condensate from one tube will drop on to the tube beneath. Since it may be very difficult to do an analytical approach, an experimental approach is definite.

Figure 2.6 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. Copper tubes (Cu/Zn) with <5% Zn run horizontally through the top of the condenser chamber. 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 Figure 6; “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 consecutively starting from 1 and to the next increasing. Nalgene® ½ inch diameter tubing covered with proper insulation was used for connecting the individual horizontal tubes. 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 for visualization of the condensation process. This dome was also connected to a vacuum pump. A Neoprene™ rubber vacuum seal was placed between the condenser and the dome. 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 gas 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 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 chamber at the same side as the other window. The condenser tubes were 16 mm outside diameter with wall thickness of 1 mm. The tubes protruded 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.

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 chamber: 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 (LabView™) where the temperatures were monitored and stored.

2.2.4 Surface Tension Setup

In order to better quantify the effectiveness of each additive, its effect on the surface tension of water was measured and compared to the surface tension of water alone. **The experiment focused on measuring the dynamic surface tension of water in the presence of additive vapor.** The drop volume method is applied in making the surface tension measurements (Harkins and Brown, 1919). An apparatus was constructed, (Figure 2.8) for this experiment. Experimental apparatus consists of a test chamber, drop creator, collection flask, scale, additive flask, heat mantle, reservoir, cold trap, and pump. The test chamber is constructed from transparent lexan™ tubing with an inside diameter of 0.09 m, and is 0.3 m in length. The drop creator is constructed from copper tubing 6.35 mm in outside diameter, a

valve, and surgical hose connecting it to a large reservoir. The valve controls the mass flow rate and thus the drop frequency. The reservoir is open to atmospheric pressure and can be considered a constant head tank. This constant head assures that the drop frequency will be constant. The drop delivery system is entirely gravity fed. Inside the test chamber the drop falls from the tip of the drop creator and into a collection flask which is placed on the scale. The scale is a Fisher Scientific model number XD-400D.

Inside the test chamber a constant flow rate of additive vapor and air are flow across the chamber from left to right. This flow is set up by a vacuum pump attached to the right side of the chamber. The pump is a two-stage J/B Fast Vac pump with serial number 598. Additive vapor is supplied to the chamber by boiling the additive in a flask equipped with a heating mantle. A cold trap is added in between the test chamber and the pump in order to condense the additive vapor into a liquid for proper disposal.

The initial set of data obtained using the surface tension apparatus was inconclusive. This was due to lack of vapor reaching the water drops. A modification was made to the apparatus to insure additive vapor would diffuse into the water drop. Initially, the system was set up with a pump that would draw air and vapor through the chamber. However, the chamber was set up such that atmospheric pressure was maintained due to several orifices open to the outside environment. These orifices would intake air due to the pump running. A non-uniform flow field would obstruct the vapor from reaching the drop. To address this problem the vapor delivery section was extended to about 1 inch from the dropper, as opposed to 6 inches. This modification placed the droplet directly in the path of the additive vapor. Additives were brought to a boil in order to create enough vapor to inundate the drop. At the onset of boiling, vapor did not immediately flow into the test chamber. Vapor had to be given ample time in order to climb through the delivery section and into the test chamber. The deliver section was approximately 6 inches from the boiling additive and was connected by $\frac{1}{4}$ inch tube. The vapor was realized to be present in the test section.

2.2.5 Additive Selection

Additive selection was based on three different criterions:

- i) Diffusivity.
- ii) Vapor Pressure.
- iii) Static Surface Tension.

These three parameters were crucial for inducing the Marangoni effect. The lower the surface tension the greater the possible disturbance created in the film. Just as important, however, are the vapor pressure and the diffusivity of the additive in water. When the water/additive solution is boiler together, it is necessary that the original working fluid does not separate from the additive vapor. If the vapor pressure of the two fluids is very different, no communication (or at least very little communication) between the additive and the working fluid is possible. Should the additive not condense together with the working fluid, there can be no enhancement regardless of the surface tension of the additive. The other key factor is the diffusivity of the additive in the working fluid. It is the gradient in the surface tension in the liquid and the driving potential for the additive to diffuse from a region of high concentration to regions of low concentration which cause bulk fluid motion in the condensate and promote enhanced mixing.

2.3 Results and Discussion

2.3.1 General Discussion

Experimental procedures discussed in the previous section produce heat transfer experiments which Marangoni convection is the primary mode of heat transfer. The literature review has shown the flow around the condenser tube bundles to be complicated. These particular experiments were conducted in order to provide insight into these types of Marangoni convective flows. Enhanced heat transfer coefficients were found for four additives introduced into the working fluid of the condenser. Several experiments were run to obtain data on the single vertical tube condenser. With steam introduction to the chamber, drops began to form on the surface of the vertical tube. Eventually, the entire tube was covered with water that would flow down the tube in a filmwise manner, that is, the tube was entirely 'wet', (Figure 2.7). When an effective additive was introduced into the steam, the initial condensation formed small droplets, the tube appeared to be sweating as opposed to just 'wet'. As the drops began to flow down the surface of the tube under the influence of gravity, they would coalesce and flow in sheets. With additives present, the flow along the condensing surface was dropwise-like. That is, the surface appeared to be less wetting than when compared to the situation when no additive was present. In the vertical tube apparatus, flow near the bottom 1/3 of the tube sheeted downward, while small drops continued to form near the top of the tube. Along with this visualization, a significant increase in heat transfer coefficient was noticed. With 500 ppm of octanol present, the effective heat transfer coefficient was more than twice the effective heat transfer coefficient without additive, see Figure 2.9. Octanol was the only additive tested in the vertical single tube experiment.

Many experiments were run to obtain data on the multi-tube horizontal condenser. The multi-tube condenser was run with a substantial amount more than the single vertical tube because multi-tube condenser performance is our primary focus. Also, four additives were tested using this apparatus. Additives tested were: octanol, 2-ethyl-1-hexanol, 2-methyl-1-pentanol, and hexyl alcohol. A sufficient amount of research has been conducted on the effect of heat transfer additives used to induce the Marangoni effect. The mechanism to induce the Marangoni effect is not well understood; this reduces the search for new additives to a trial and error experimental approach (Perez-Blanco and Sheehan, 1995).

As previously mentioned the single tube was run in order to provide insight into this type of condensation, and to justify the construction of a multi-tube condenser. The multi-tube was also run at more than one value of temperature difference between the inlet and outlet cooling water. By increasing the power to the heaters, the heat transfer to the cooling water could be increased, therefore, the outlet cooling water temperature was increased. Increasing the power to the heaters increased the amount of steam inside the chambers but did not force the steam to enter the super heated region. As noted before, the multi-tube also displayed increased heat transfer with a transition from large droplets condensing on the surface to smaller droplets condensing and falling from the surface of the condenser tubes. Here again, significant increases in heat transfer coefficient were found.

Again, several experiments were run on the surface tension apparatus in order to confirm the initial hypothesis as to why the additives were an effective method in increasing condenser performance. The drop-volume method was employed to take obtain this data. The experimental work of Harkins and Brown established this method on a sound basis (Harkins and Brown, 1919; Yao et al, 1991). A simple formula for the surface tension of a drop given the

radius of the tube and the weight of the drop does not exist. However, the correction factor, f , allows for this method to be utilized with sufficient accuracy. The drop-volume method is well accepted surface tension measurement technique due to the ease of experimental operation. In this case, the drop volume method allowed for easy introduction of additive vapor into the drop. Drop frequency was a variable within the experiments. Data was obtained for three different drop frequencies. Drop frequencies of 0.1 Hz, 0.5 Hz, and 0.05 Hz were tested due to the dynamic nature of the surface tension of binary fluids. The drop frequency was set before the experiment was run. For example, 0.1 Hz was obtained for the drop frequency before the vapor was introduced to the test section. Then, as vapor was introduced, the drop frequency would increase due to the diffusion of vapor additive into the water droplet. This increase in frequency signified smaller droplets falling due to less surface tension, that is, the diffusion control. Setting the initial drop frequencies to as low as 0.05 Hz gave the additive vapor a considerable enough time to diffuse. As more additive diffused into the drop, and as more time was allowed for the surface tension of the solution to reach a pseudo-steady state value, more interfacial tension was created and the surface tension of the drop decreased (Equation 2.6).

2.3.2 Heat Transfer Coefficient Calculation

In order to calculate the heat transfer coefficient, two well-known equations were used. The first one calculated the heat flux from the steam to the cooling water.

$$\dot{q} = \dot{m}C_p\Delta T \quad (2.9)$$

Where $\dot{q}$ is the heat transfer rate, $\dot{m}$ is the mass flow rate, C_p is the heat capacity and ΔT is the temperature difference from outlet and inlet cooling water (=temperature gain). To calculate the heat transfer coefficient, Eq. (2.10) was used.

$$\bar{h} = \frac{\dot{q}}{A(T_{sat} - T_{sur})} \quad (2.10)$$

Where $\bar{h}$ is heat transfer coefficient, A is the surface area of the condenser tubes, T_{sat} is the saturation temperature, and T_{sur} is the outside temperature of the cooling tubes. For the single tube apparatus, T_{sat} was measured inside the condenser with a thermocouple approximately 4 inches from the top of the steam inlet. T_{sur} was determined by taking the average between the cooling water inlet and cooling water outlet. For the multi-tube apparatus, T_{sat} was measured inside the condenser with a thermocouple about 2 inches from the top at the side of row number 2 (middle row). The T_{sur} was considered to be the surface temperature of the condenser tubes, however, this temperature was not measured directly. The surface temperature is approximated as the average temperature of the inlet and outlet cooling water temperatures, also measured with thermocouples, see Figures 2.5 and 2.6. This is a reasonable approximation because the temperature of the copper tube is very close to that of the cooling water. Note that the tube resistance is small and the thermal capacity of water is significantly larger than that of copper; therefore, this assumption is valid.

To present the enhancement in the heat transfer coefficient in a convenient way, an enhancement factor h_{eff} was defined. This heat transfer coefficient was normalized by finding the ratio between the heat transfer coefficient with additive to heat transfer coefficient without additive. This normalized value is called the effective heat transfer coefficient.

$$h_{\text{eff}} = \frac{\bar{h}_w}{\bar{h}_{w/o}} \quad (2.11)$$

Where $\bar{h}_w$ is the heat transfer coefficient with heat transfer additive added, and $\bar{h}_{w/o}$ is the heat transfer coefficient without heat transfer additive. Note that this value, h_{eff} , is a direct measure of the heat transfer enhancement.

2.3.3 Single Tube Results

Initial experiments with a vertical single tube produced favorable results. Experimental run time was approximately 1.5 hours. At this time the steady state temperature values were tabulated. Five thermal couple readings yielded temperatures for the boiler, inlet and outlet steam at the top and bottom of the condenser chamber, and inlet and outlet water through the condenser chamber. At approximately 1.5 hours, temporal changes in temperature readings were negligible. The heat transfer coefficient for the system was found using the method described above for each run and then averaged. Figures 2.9 and 2.10 show the dependence of the effective heat transfer coefficient on the amount of n-octanol present in the system. As can be seen in the Figure, the additive's effectiveness increases with an increase in amount of additive; however, it will eventually reach an asymptotic value. One additional run of 1% additive (10,000 ppm) was taken (not shown). The increase in effective heat transfer coefficient from 500 parts per million to 1% of additive was small. Note that additive concentration is the charge concentration, and local concentration is not known.

2.3.4 Multi-tube Results

The system was evacuated and the heaters were powered for the desired temperature difference in cooling water. Again, the system reached steady state values in approximately 1.5 hours. At steady state, the temperature values for five thermal couples were tabulated. The five thermal couples yielded temperature values for the chiller, the cooling water reservoir, the inlet and outlet cooling water temperatures, and the saturation temperature for the steam inside the chamber. The heat transfer coefficient for the multi-tube condenser was calculated and normalized in exactly the same way as the single vertical tube. Figures 2.11-2.14 show the dependence of the effective heat transfer coefficient vs. the amount of additive present in the condenser chamber for each additive. The figures also compare this dependence between a temperature difference, ΔT , of 4.5 and 6.2 °C over the inlet and outlet cooling water. As can be seen, the effectiveness of the additive increases with an increase in additive amount and as expected, an increase in temperature difference between the inlet and outlet temperatures. One additional run of 1% (10,000 ppm) additive was taken (not shown). The increase in effective heat transfer coefficient from 500 to 10000 parts per million of additive was small. This shows that the effectiveness of the additive reaches an asymptotic value as more additive is introduced into the condenser. When a liquid contains more than one molecular species, equilibrium will be reached only when the surface layers comprise those molecules that make the free energy of the surface a minimum. There will thus be a formed surface phase differing in composition from the bulk of the liquid. This difference may be any degree, ranging from the case where a single layer of closely-packed molecules is adsorbed at the boundary, down to the case where there is near the surface merely a slight excess of one kind of molecule, determined by the balance between the forces tending to promote adsorption and those causing diffusion (Burdon,

1949; Joos, 1999). It appears that the condensation that occurred in these experiments were mostly dropwise-like condensation. When heat transfer additives were added, the droplets became more dispersed and had a smaller shape than the case for without additive (Figure 2.15).

The Marangoni effect could cause the droplets to fall off the condensation tube at an earlier stage, which means smaller droplets, also the size of the droplets without heat transfer additive was around 1-2 mm in diameter. When 1,000 ppm of heat transfer additive was added, the droplet size was around 0.5-1 mm in diameter. With heat transfer additives added, the drops are smaller and have a larger population along the condensing surface. The rate of falling drops is higher; therefore, there is more “activity” in the condenser. Thus, more surface area that is not covered by a condensate drop will allow for “new” steam to get in contact with the condenser tube and be condensed. This increased activity will enhance the heat transfer coefficient. When 10,000 ppm (1 percent) of heat transfer additive was added, the droplets were smaller than the ones for 500 ppm of the same heat transfer additive. When 10,000 ppm of heat transfer additive hexyl alcohol was added, the droplet size reduced to approximately 0.1-0.5 mm diameter.

2.3.5 Surface Tension Calculation

In order to calculate the surface tension of the water/additive solution, Newton’s second law is applied to the drop, (Figure 2.17). The force balance is as follows,

$$mg = 2\pi r\sigma \quad (2.12).$$

Where m is the mass of one drop, g is the gravitational acceleration, r is the outside radius of the tip and σ is the surface tension of the liquid. However, it has been found that a small portion of the drop remains on the tip, and causes an error in the calculated surface tension. Therefore, a correction factor, f , introduced by Harkins (1952), is used to correct for this effect and to achieve better accuracy (Harkins and Brown, 1919). This correction is a dimensionless parameter, $f(r/V^{1/3})$. Note that f was determined by measuring the surface tension of pure water and comparing the obtained value with published values. Now, solving for the surface tension and adding the correction factor, the equation becomes,

$$\sigma = \frac{mg}{2\pi f} \quad (2.13).$$

2.3.6 Surface Tension Results

In order to run the surface tension apparatus approximately 15 ml of additive were brought to a boil. Using a stopwatch the desired drop frequency was achieved. The experiment was run at different frequencies to test for the dynamic nature of surface tension. The additive vapor was connected to the test chamber, and the pump was started. The vapor ran through a cold trap in order to condense it back to liquid. Next, the number of drops were counted and weighed over a two-minute period. Five runs for each additive at each frequency were tabulated. The surface tension for each run was calculated and averaged over the number of runs.

Figure 2.18-2.21 shows the dependence of surface tension on frequency of the drops. As the figure shows, surface tension is a definite dynamic process. Once diffusion has occurred a finite time exist for the solution to reach its steady state surface tension value.

Some difficulties in obtaining surface tension data were present. Maintaining steady flow of vapor inside the test chamber was a challenge. Due to boiling action inside the flask, vapor would pulse in to the test chamber; therefore the vapor concentration was not quantified in the experiment. However, the intention of the experiment was to make relative comparisons among the different additives tested. For each additive tested in the same way, comparative results could be obtained, and their standard deviations are noted as error bars. In order to combat these difficulties multiple tests runs were taken for each additive and averaged.

Two immediate indications provided insight into the diffusive nature of the additives. First, an increase in drop frequency and a reduction in drop size were noticed in the presence of additives. Second, liquid additives could be seen in the collection flask after each run therefore additive had definitely diffused into the drop. Multiple runs of each additive were taken in order to develop an accurate average of the surface tension of the water in presence of the four additives.

The surface tension of n-octanol and 2-ethyl-1-hexanol had the highest surface tension values. Their values were very close together. Hexyl alcohol and 2-methyl-1-pentanol had the lowest surface tension values, and were also very close in value. Most interestingly, a nearly linear relationship was observed between σ and $t^{1/2}$ (Figure 2.22).

2.4 Conclusions

During this experimental investigation two significant results have been shown. First, heat transfer additives have been shown to be an effective way to increase the efficiency of conventional tube bundle condenser heat exchangers. This increased condensation rate is due to the creation of a disturbance in the liquid film surrounding the tubes. Because of this disturbance, it appears the liquid film thickness has been decreased. It is postulated that the heat transfer through this film has increased due to the onset of convection as well as conduction through the liquid film, also the population of dropwise-like condensation increased. Second, a direct correlation may be made between the surface tension and the enhanced heat transfer coefficient of each additive. The addition of both hexyl alcohol and 2-methyl-1-pentanol bring the surface tension of water/additive system down to 35 dyne/cm. It is the Marangoni effect that causes the disturbance in the liquid film, and the mechanism for this disturbance is the surface tension. The dynamic nature of diffusion of effective additives causes interfacial turbulence and bulk convective flows in the liquid film along the condenser tubes.

The additive n-octanol produced the lowest enhancement factor of the four tested, and also had the highest surface tension and lowest vapor pressure. For n-octanol at 500 ppm; h_{eff} was 1.025 and 1.034 at temperature differences of 4.5 and 6.2 °C respectively (Figures 2.11-2.12, 2.18-2.19). The additive 2-ethyl-1-hexanol produced the next lowest enhancement factor. The additives hexyl alcohol and 2-methyl-1-pentanol produced the largest enhancement factor, out of the four additives tested. 2-methyl-1-pentanol, also, had the lowest surface tension out of the four additives tested.

2.5 Reference

1. Adamson, A. W., Physical Chemistry of Surface, John Wiley & Sons, New York 1983.
2. Beckwith, Thomas G., Marangoni, Roy D., Lienhard V, John H., Mechanical Measurements, 5th edition, Addison-Wesley 1993.

3. Black, William Z., Hartley, James G., Thermodynamics, 3rd edition, HarperCollins publishing, 1996.
4. Burdon, R. S., Surface Tension and the Spreading of Liquids, 2nd edition, Cambridge Press, 1949.
5. Ford, J. D., Missen R. W., On the Conditions for Stability of Falling Films Subject to Surface Tension Disturbances; The Condensation of Binary Vapors, The Canadian Journal of Chem. Engr., Vol. 46, pp 309-312, Oct. 1968.
6. Fuks, S. N. , Heat Transfer with Condensation Flowing in a Horizontal Tube Bundle, Teploenergetika, Vol. 4, pp 35-39, 1957.
7. Grant, I. D. R., and Osment, B. D. J., The Effect of Condensate Drainage on Condenser Performance, NEL Rep 350, April 1968.
8. Harkins, William D., Brown, F. E., The Determination of Surface Tension (Free Surface Energy), And The Weight of Falling Drops: The Surface Tension of Water and Benzene By The Capillary Height Method, Jour. Am. Chem. Soc., Vol. 41, pp 499-524, 1919.
9. Hatamiya, Shigeo, Tanaka, Hiroaki, Dropwise Condensation of Steam at Low Pressures, International Journal of Heat and Mass Transfer, Vol. 30, No. 3, pp 497-507, 1987.
10. Hoke, Bryan C. Jr., Chen, John C., Binary Aqueous-Organic Surface Tension Temperature Dependence, Journal of Chem. Eng. Data, Vol. 36, No. 3, pp 322-326, 1991.
11. Incropera, Frank, and Dewitt, David, Introduction to Heat Transfer, Third Edition, Wiley & Sons 1985.
12. Jakob, Max, Heat Transfer in Evaporation and Condensation – II, Mechanical Engineering, pp 729-739, 1936.
13. Joos, Paul, Dynamic Surface Tension, VSP BV 1999.
14. Kakac, S. and Lin, H. Heat Exchangers; Selection, Rating and Thermal Design. pp 201-214, 1998.
15. Kern, D. Q., Mathematical Development of Loading in Horizontal Condensers, AIChE J., Vol. 4 pp 157-160, 1958.
16. Kim, K. J., Berman N. S., Wood, B. D., “Surface tension of aqueous lithium bromide + 2-ethyl-1-hexanol”, *J. Chem. Engr. Data*, No. 39, pp 122-124, 1994.
17. Kim, K. J., Berman N. S., Wood, Byard D., Absorption of Water Vapor into LiBr Solutions with 2-Ethyl-1-Hexanol, AIChE Journal, Vol. 42, No. 3, pp 884-888, 1996.
18. Kim, K. J., Lefsaker, A. M., Razani, A., Stone, A., “The effective use of heat transfer additives for steam condensation,” *Applied Thermal Engineering*, Vol. 21, No. 18, pp 1863-1874, 2001.
19. Kim, K. J., Janule, V. P., “Dynamic surface tension of aqueous lithium bromide with 2-ethyl-1-hexanol”, *Int. Comm. Heat Mass Transfer*, Vol. 21, No. 6, pp 839-848, 1994.
20. Lu, H., and Yang, Y., and Maa, J., Surfactant Effects on Absorption in the Presence of Induced Interfacial Turbulence, AIChE Journal, Vol. 43, No. 7, pp 1909-1913, 1997.
21. Lu Hsin-Hsen, and Yang, Yu-Min, and Maa, Jer-Ru, Effect of Artificially Provoked Marangoni Convection at a Gas/Liquid Interface on Absorption, *Ind. Eng. Chem. Res.* Vol. 35, pp 1921-1928, 1996.
22. McBain, James W., Sharp, Evert R., A Comparative Study of Surfaces of Solutions with the Film Balance and Surface Tension Equipment, *Journal of American Chemical Society*, Vol. 63, no. 34, pp 1422-1426, 1941.

23. McCormick, James, Baer, Eric, On the Mechanism of Heat Transfer in Dropwise Condensation, *Journal of Colloid Science*, Vol. 18, pp 208-216, 1963.
24. Mendes-Tatsis, M. A., and De Ortiz, E.S. Perez, Marangoni Instabilities In Systems with an Interfacial Chemical Reaction, *Chemical Engr. Science*, vol. 51, No. 15, pp 3755-3761, 1996.
25. Mirkovich, V. V., and Missen R. W., A Study of the Condensation of Binary Vapors of Miscible Liquids, *The Canadian Journal of Chem. Engr.*, pp73-78, April 1963.
26. Moffat, R.J., Using Uncertainty Analysis in the Planning of an Experiment, *Journal of Fluids Engineering*, Vol. 107, pp 173-178, June 1985.
27. Omega, *The Flow and Level Handbook*, Omega Engineering Inc. 1992.
28. Oosthuizen, Patrick, and Naylor, David, *Introduction to Convective Heat Transfer Analysis*, McGraw-Hill 1999.
29. Perez-Blanco, H, Sheehan, D. S., Effect of additive concentration on falling film absorption, *HVAC&R Research*, Vol. 1, No. 4, pp 273-281, 1995.
30. Putman, Richard E., *Steam Surface Condensers: Basic Principles, Performance Monitoring, and Maintenance*, ASME Press 2001.
31. Ruckenstein, E, Smigelschi, O, Suciu, D. G., A steady dissolving drop method for studying the pure Marangoni effect, *Chemical Engineering Science*, Vol. 25, pp 1249-1254, 1970.
32. Savitch, Walter, *Problem Solving with C++ The Object of Programming*, 2nd edition, Addison-Wesley 1999.
33. Song, Yongji, Xu, Dunqi, Lin, Jifang, A study on the mechanism of dropwise condensation, *-International Journal of Heat and Mass Transfer*, Vol 34, No. 11, pp 2827-2831, 1991.
34. Sternling, C.V., Scriven, L.E., Interfacial Turbulence: Hydrodynamic Instability and the Marangoni Effect, *AIChE Journal*, pp 514-523, 1959.
35. Stone, Aaron, Kim, Kwang J., Razani, A., "Enhanced Steam Condensation as a result of Heat Transfer Additives", Accepted for publication in *International Journal of Environmentally Conscious Design and Manufacturing*
36. Vazquez, G., Antorrena, G., Navaza, J.M., Santos, V., Absorption of CO₂ by Water and Surfactant Solutions in the Presence of Induced Marangoni Effect, *Chemical Engineering Science*, Vol. 51, no. 12, pp 3317-3324, 1996.
37. Vazquez, Gonzalo, and Alvarez, Estrella, and Navaza, Jose M., Surface Tension of Alcohol + Water from 20 to 50 °C, *Journal of Chemical Engineering Data*, Vol. 40, pp 611-614, 1995.
38. Wasan, Darsh T., and Nikolov, Alex D. and, Brenner, Howard, Droplets Speeding on Surfaces, *Science*, vol. 291, pp 605-606, 2001.
39. Webb, Ralph L., *Principles of Enhanced Heat Transfer*, John Wiley & Sons, Inc., 1994.
40. Wilson, J. L., The Design of Condensers by Digital Computers, *IchE Sym. Ser.*, no. 35, pp. 132-151, 1972.
41. Wolf, Stefan, Stichlmair, Johann, Interfacial Instabilities in Liquid-Liquid Systems, *Veroffentlichung aus dem Tagungsband ISEC*, Essay 2, 1996.
42. Yao, Wen, Bjurstrom, Henrik, Setterwall, Fredrik, Surface Tension of Lithium Bromide Solutions with Heat-Transfer Additives, *J. Chem. Eng. Data*, Vol. 36, pp 96-98, 1991.

43. Zhang Chao, Local and Overall Condensation Heat Transfer Behavior in horizontal Tubes Bundles, Heat Transfer Engineering, Vol. 17, no.1 pp19-30, 1996.

Figure 2.1: Wettability of a Surface. For a contact angle, ϕ , greater than 90° , the surface is said to be non-wetting. For a contact angle, ϕ , less than 90° the surface is said to be wetting.

Figure 2.2: Cross section of horizontal tube bundle. a) in-line array of horizontal tubes, b) staggered array of horizontal tubes.

Figure 2.3: Turbulent flow of steam over in-line horizontal condenser tubes.

Arrows denote direction of vapor flow. First column in tube bundle may act as a turbulence grid.

Figure 2.4: Dropwise condensation showing a thin film on the cooling surface as drops coalesce.

Figure 2.5: Experimental setup – vertical falling film facility

Figure 2.6: 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. The thermocouples were connected to a computerized data acquisition system (LabView™) where the temperatures were monitored and stored (Kim et al., 2001).

Figure 2.7: Dropwise condensation (a) vs. filmwise condensation (b) on horizontal condenser tubes.

Figure 2.8: Schematic of surface tension experiment.

Figure 2.9: The effect on the heat transfer rate by adding heat transfer additives in terms of enhancement factor, h_e . Defined in Eq. (2.11)

Figure 2.10: Single tube results for n-octanol additive.

As amount of additive is increased the enhanced heat transfer coefficient increases. Error bars represent standard deviation of experimentally measured values. Points fitted with a linear curve.

Figure 2.11: Enhanced heat transfer coefficient for n-octanol.

As amount of additive is increased, the enhanced heat transfer coefficient increases. Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.12: Enhanced heat transfer coefficient of 2-ethyl-1-hexanol.

As amount of heat transfer additive is increased, the enhanced heat transfer coefficient increases. Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.13: Enhanced heat transfer coefficient of 2-methyl-1-pentanol.

As amount of heat transfer additive is increased, the enhanced heat transfer coefficient increases. Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.14: Enhanced heat transfer coefficient of hexyl alcohol.

As amount of heat transfer additive is increased, the enhanced heat transfer coefficient increases. . Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.15: 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 (Kim et al., 2001).

Figure 2.16: Marangoni instability mechanism in condensate film with additives (Kim et al, 2001).

Figure 2.17: Drop force balance on the tip of the tube (Beckwith, et al. 1993; Black, 1996).
 Note that the size of the drop is a function of frequency due to the additive's diffusional time in contact with water.

Figure 2.18: Surface tension vs. drop frequency of n-octanol.

As drop frequency is manually decreased, additives have a greater time to diffuse into the droplet, and surface tension decreases. Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.19: Surface tension vs. drop frequency of 2-ethyl-1-hexanol.

As drop frequency is manually decreased, additives have a greater time to diffuse into the droplet, and surface tension decreases. Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.20: Surface tension vs. drop frequency of 2-methyl-1-pentanol.

As drop frequency is manually decreased, additives have a greater time to diffuse into the droplet, and surface tension decreases. Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.21: Surface tension vs. drop frequency of hexyl alcohol.

As drop frequency is manually decreased, additives have a greater time to diffuse into the droplet, and surface tension decreases. Error bars represent standard deviation of the mean of experimentally measured values. Points fitted with a linear curve.

Figure 2.22: Surface tension, σ , as a function of $t^{1/2}$.

Note that additive #1 and additive #2 are hexyl and 2-methyl-1-pentanol respectively.

Figure 2.23: Picture of filmwise condensate on one horizontal copper tube.
There is no additive present in the working fluid.

Figure 2.24: Picture of dropwise-like condensate on one horizontal copper tube.
Approximately 250 ppm of hexyl alcohol present in working fluid.

SECOND LAW ANALYSIS OF A COMBINED CYCLE WITH INTEGRATED COMPRESSOR INLET AIR COOLING

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ABSTRACT

A gas/ammonia combined cycle is proposed in which exhaust gases from a Brayton gas topping cycle are used to produce superheated ammonia in a Heat Recovery Ammonia Generator (HRAG). To increase the power of the gas turbine in the combined cycle, when the environmental air temperature is high, inlet air to the compressor is cooled in the evaporator of an ammonia refrigeration cycle added to the combined air/ammonia power cycle. In this integrated combined power cycle a small fraction of high-pressure ammonia liquid, from the exit of the ammonia pump, is used in the ammonia refrigeration cycle to cool the air. The second law analysis and optimization of the above combined cycle is presented. The effect of important system parameters on the irreversibility of components in the cycle and the exergy of exhaust streams are evaluated. Reasonable constraints for system components are assumed. The power and efficiency of the cycle are evaluated and their dependences on system parameters are presented.

Keywords: combined power cycles, power augmentation, second law analysis

NOMENCLATURE

°C	Degree Centigrade
Comp1	First air compressor
Comp2	Second air compressor
BWR	Back Work Ratio
ΔT_{app}	Gas/Ammonia temperature difference
ΔT_{ATC}	Temperature difference between outlet of ammonia condenser and To
FP	Fuel Compressor
Excao	Exhaust exergy from intercooler
Exstack	Exergy of stack exhaust gasses
G	Generator

GT	Gas Turbine
HRAG	Heat Recovery Ammonia Generator
I_{comb}	Irreversibility of combustion
IC	Intercooler
K	Degree Kelvin
I_{tot}	Total irreversibility of a given power cycle
Pr	Pressure Ratio
Rec	Recuperator
Toa	Ambient Air temperature
TIT	Gas Turbine Inlet Temperature
T_{pinch}	Pinch point temperature difference
W_{in}	Input power
W_{out}	Output power
η_1	Thermal efficiency, Eq. (1)
η_2	Second law efficiency, Eq. (2)

INTRODUCTION

One problem with gas turbine power plants is that power is lowest during the hot summer months. Ironically, in most cases, demand is highest during the summer months as well. One option for utilities to remedy this situation is the cooling of the compressor inlet air (Kraft, et al., 1999). Adding an air cooling system to gas turbine power plants designed to generate peak power can provide up to a 25 percent increase in output power (Ebeling, 1992). Gas turbine power augmentation, in a cogeneration power plant, considering different options for inlet air cooling, has been investigated (Ondryas, et al., 1991). Ondryas, et al. analyzed the availability of power augmentation anytime when the chilled air temperature is below the ambient temperature, but especially during summer on-peak hours. They considered different cooling options including absorption chillers, mechanical chillers, and thermal energy storage. The most

suitable cooling systems, both absorption and compression refrigeration with and without thermal storage, for power augmentation of a cogeneration power plant, have been investigated (DeLucia, et al., 1994). In the study by DeLucia et al. both technical and economical analysis have been performed. They conclude the power output may be increased by almost 20 percent using an absorption unit to cool the compressor inlet air to 10°C. They also find that the cost of an absorption unit is a small fraction of the value of potential energy savings in the warmest months of the year. More recently, the optimum power boosting of gas turbine cycles with inlet air refrigeration has been studied analytically (Ait-Ait, 1994). Unlike the previous studies, this investigation considers cooling of air well below practical temperatures where potential ice formation in the compressor suction line becomes a problem. Ait-Ali argues that there are relatively inexpensive industrial solutions for

removal of water vapor from inlet air that are outweighed by the potential power gain that can be achieved from its refrigeration. He then proceeds to optimize the system using the pressure ratio and compressor inlet air temperature as parameters.

A typical power plant working on a simple Rankine or Brayton cycle has efficiencies on order of 35 percent, much lower than the efficiency of a combined cycle that couples a Brayton gas topping cycle with a bottoming Rankine cycle (Harlock, 1995, Kehlhofer, 1997 and Khartchenko, 1998). The combined power cycles achieve efficiencies well above 50 percent and are approaching an efficiency of 60 percent (Briesch, et al., 1995, Dechamps, 1998, Razani and Kim, 2000). In this study a gas ammonia (Brayton/Rankine) combined cycle is proposed in which an ammonia compression refrigeration cycle is added to the bottoming cycle. The ammonia refrigeration cycle is used to cool the compressor

Fig. 1. Flow diagram of the Gas/Ammonia combined cycle with integrated compressor inlet air cooling.

inlet air. This integrated combined cycle produces high power and efficiency.

CYCLE DESCRIPTION AND ANALYSIS

A flow diagram of the gas / ammonia combined cycle with an integrated ammonia vapor compression ammonia cycle is given in Fig. 1. The Brayton topping cycle includes two compressors with an air intercooler in-between and a recuperator to preheat the air using turbine exhaust gases. The exhaust gases from the recuperator produce superheated ammonia. In addition to the ammonia turbine and condenser, the ammonia Rankine bottoming cycle includes an ammonia preheater to recover the energy from the exhaust of the ammonia turbine. The ammonia enthalpy from the turbine exit may be high enough to boil the ammonia liquid entering the preheater. The ammonia is superheated in the Heat Recovery Ammonia Generator (HRAG). HRAG is the component connecting the Brayton gas topping cycle and Rankine ammonia bottoming cycle. The ammonia exit from the preheater is condensed in an ammonia/air condenser. Ammonia exit from the condenser is assumed to be saturated liquid, a fraction of which is pumped to high pressure and the remaining is expanded in a Joule Thompson expansion valve before entering the ammonia evaporator. The ammonia evaporator is used to cool the inlet air to the compressor for the purpose of increasing the power output of the gas turbine cycle. The ammonia exit from the evaporator is assumed to be saturated vapor and is compressed to the condenser pressure. It is then combined with the exhaust of the ammonia preheater before entering the ammonia condenser. Exergy analysis and second law analysis and optimization of thermal systems are well-established methods of power system design in the fields of thermal science and engineering. (Kotas, 1995, Bejan, et al., 1996, Horlock, et al. 2000).

In general, conservation of mass and energy in addition to the balance of exergy are written for each system component. The irreversibility of each component is determined and the exergy entering and leaving the system is evaluated. For the cycle under investigation in this paper, these equations were written for each system components using EES software (F-chart, 1995).

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 78.10% N₂, 20.76% O₂, 1.11% H₂O, and 0.03% CO₂. The volume flow rate of air is set at 100 m³/s and the fuel is assumed to be natural gas with an equivalent chemical formula of CH_{3.88}. The chemical and physical exergy for the inlet and exit streams of each component are determined based on the reference environment mentioned above (Bejan, et. al., 1996). The fuel is assumed to enter the system at a pressure of three atmospheres and the temperature of the environment. Therefore, a fuel compressor must be used to increase the pressure of the fuel to the operational pressure of the

combustion chamber. In general, overall thermal efficiency is defined by:

$$\eta_1 = \frac{\text{Net Energy Rate of the Product}}{\text{Rate of Input Energy}} \quad (1)$$

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. The second law efficiency in this study is defined by:

$$\eta_2 = \frac{\text{Net Exergy Rate of Production}}{\text{Net Exergy Rate of Input}} \quad (2)$$

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 air. The exit streams are the combustion products from the stack exhaust gases and the cooling air discharges from the intercooler and ammonia condenser. It should be pointed out that second law efficiency given by equation (2) assumes that the exergy in the exhaust stream is partly recoverable. In practice, the system should be designed assuming the exergy of the exhaust streams is lost. Since the product is net power produced and the input exergy and energy of the fuel are almost the same, in this study, the efficiencies given by both equations would be approximately the same if the exergy of the exhaust streams are assumed to be lost. Therefore, the second law efficiency given by equation (2) indicates the contribution of the exergy of the exhaust streams. A high value of the efficiency may point to potentially recoverable lost exergy.

The chemical exergy of inlet and exit streams for each component is calculated using the Table of Chemical Energy of Compounds (Bejan, et al., 1996). The difference in composition of air used in this study and the reference on which the chemical exergy of compounds are based, results in a very small error when energy balance is calculated for the system. This error is found to be smaller than 0.03 percent of the inlet fuel exergy. The exergy balance for the system is evaluated to check the calculation of total irreversibility for the system. It should be pointed out that the chemical exergy of the ammonia inlet and exit streams of each component in the ammonia cycle cancel out when the irreversibility of each component is evaluated. The chemical exergy of the fuel is corrected for the equivalent chemical formula used in this study. Gas turbines are essentially constant volume machines; any change in the mass flow of air through the turbine affects their output. Mass flow of air is proportional to its density, which is affected by the compressor inlet temperature. In this analysis a fixed volume flow rate to the inlet of the compressor

is assumed. Important parameters in the cycle are the pressure ratio, environmental air temperature, turbine inlet temperature, and compressor inlet air temperature. To analyze the effect of the compressor inlet air cooling, its temperature is varied while the volume flow rate of air into the compressor is held constant. The flow rate of input fuel is a variable. It is calculated from the energy and mass balance on the combustion chamber such that a given Turbine Inlet Temperature (TIT) is produced.

RESULTS AND DISCUSSION

Table 1 gives the nominal values for the system parameters used in this study. For simplicity of analysis and preliminary parametric studies, a nominal value of 0.9 for the turbine and compressor efficiencies is assumed. The important system parameters under investigation are the environmental air temperature to the evaporator inlet (T_{oa}), compressor inlet air temperature (T_1), gas turbine inlet temperature (TIT), and the gas cycle pressure ratio (Pr). Figures 2 to 4 show the normalized power as a function of compressor inlet air temperature (T_1) when gas turbine inlet air temperature is fixed (TIT = 1600K), and the environmental air temperature is constant ($T_{oa} = 45^\circ\text{C}$) for three different values of gas cycle pressure ratios of 16, 25 and 36 respectively. The air pressure exit of the first compressor is the square root of the pressure ratio. The first law efficiency (η_1) and the second law efficiency (η_2), as defined by equations (1) and (2), are also given in the same figures. Since the exergy of the exhaust

Fig. 2 Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, $Pr=16$.

stream is small, the values of the first and second law efficiencies are very close. Even though the optimum conditions are different, the values of the optima are very close. Therefore, other system constraints and economics will dictate the actual system design. Since the air temperature into the ammonia evaporator is cooled progressively from 45°C to 5°C , the net power of the combined cycle increases as air is cooled to lower temperatures before entering the air compressor. The specific power (power per unit volume flow rate) is normalized to a compressor inlet air temperature of 15°C . To find the effect of compressor inlet air cooling, the volume flow rate of inlet air to the compressor is

Table 1. Nominal Settings for Gas/Ammonia Combined Cycle with Integrated Compressor Inlet Air Cooling

Parameter	Value	Unit
Ammonia pump efficiency	0.7	Dimensionless
Ammonia preheater approach temperature difference	4	$^\circ\text{C}$
Gas cycle pressure ratio	16	Dimensionless
Enthalpy of combustion for fuel ($\text{CH}_{3.884}$)	42	MW/kg
Chemical exergy of natural gas	43.68	MW/kg
Volume flow rate of air into gas cycle compressor	100	m^3/s
TIT	1600	K
Temperature of inlet air to compressor	15	$^\circ\text{C}$
Pinch temperature of ammonia evaporator	10	$^\circ\text{C}$
Ammonia condenser reservoir temperature	T_{oa}	$^\circ\text{C}$
T_{oa}	45	$^\circ\text{C}$
Ammonia turbine inlet pressure	62	bars
Lower bound on stack exhaust temperature	70	$^\circ\text{C}$
Gas turbine efficiency	0.9	Dimensionless
Ammonia turbine efficiency	0.9	Dimensionless
All compressor efficiencies	0.9	Dimensionless
Recuperator effectiveness	0.9	Dimensionless
Intercooler effectiveness	0.9	Dimensionless
Ammonia evaporator effectiveness	0.8	Dimensionless
Gas/ammonia approach temperature	80	$^\circ\text{C}$
Ammonia boiler pressure	62	bars
Inlet fuel pressure	3	atmospheres

Fig. 3 Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, $Pr=25$.

Fig. 4 Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, $Pr=36$.

fixed at $100\text{m}^3/\text{s}$. Therefore, as the inlet air is cooled further, the mass flow rate into the gas cycle increases resulting in higher net power output. This, of course, necessitates an increase in the mass flow rate of the fuel for a fixed $TIT = 1600\text{K}$. It should be pointed out that the fuel mass flow rate is obtained from the energy balance on the combustion chamber. The effect of the pressure ratio on the net power produced by the combined cycle is more complex because it influences several aspects of the cycle. As the pressure ratio increases, the exergy out of the intercooler wasted to the environment increases, while the bottoming cycle becomes more efficient. The role of the recuperator effectiveness in the gas cycle is also important in this regard. The effect on efficiency is clear. As the pressure ratio increases the first and second law efficiencies are reduced.

Fig. 5 Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, $TIT=1400$.

It is interesting to note that there is a shallow optimum for the second law efficiency as compressor inlet air temperature is reduced. The optimum values shift toward lower values of T_1 as Pr is increased. The first law efficiency also has an optimum but it occurs at lower values of T_1 . In fact, the optimum does not occur for $Pr = 36$ even at the lowest value of $T_1 = 5^\circ\text{C}$, used in this study. Compressor inlet air temperatures lower than 5°C requires separation of moisture even for the typical dry air used in this study. Figures 5 and 6 show the same quantities as figures 2 except the gas turbine inlet temperature has changed to 1400K and 1800K , respectively.

Fig. 6 Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, $TIT=1800$.

Comparison of figures 2, 5 and 6 show that the effect of the change in TIT on normalized power at a fixed compressor inlet temperature is not significant. As expected, the effect of TIT on both the first and second law efficiencies is significant. In fact, the first law efficiency of the combined cycle surpasses the 60 percent level and has an optimum close to $T_1=15^\circ\text{C}$. It should be pointed out that turbine inlet temperatures higher than 1100K require turbine blade cooling not included in this analysis.

To assess the effect of environmental air temperature on power and efficiency, normalized power and efficiency are shown in figures 7 and 8 for $T_{oa}=30^{\circ}\text{C}$ and $T_{oa}=15^{\circ}\text{C}$, respectively. Naturally, as environmental air temperature goes down the effect of inlet air cooling on power augmentation is reduced. Figure 8 shows the normalized power and efficiency

Fig. 7 Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, $T_{oa}=30^{\circ}\text{C}$.

Fig. 8 Normalized power, first and second law efficiencies as a function of compressor inlet air temperature, $T_{oa}=15^{\circ}\text{C}$.

for the standard air temperature of 15°C (ISO Condition). The power and efficiency increase very slowly with compressor inlet air cooling. Obviously, the increase does not justify the capital cost of the addition of the ammonia bottoming cycle when the environmental temperature is low.

Figures 9 to 11 show the gas topping cycle back work ratio (BWR) and the ammonia bottoming cycle back work ratio (BWR), as a function of the compressor inlet air temperature (T_1) for three different pressure ratios. Figure 9 corresponds to the high environmental air temperature of 45°C . Figure 9 shows that as the compressor inlet temperature is reduced the ammonia BWR increases dramatically and is not a strong function of gas cycle pressure ratio. This is due to the fact that the pressure of the ammonia cycle is fixed at 62 bars in these calculations. When the environmental air temperature is 45°C the ammonia compressor power is zero but the ammonia pump

is still needed in the cycle and the ammonia BWR is reduced to about 5 percent. As expected, the BWR of the gas topping cycle is not a strong function of compressor inlet air temperature and increases with the gas cycle pressure ratio.

Fig. 9 Back work ratio for gas topping and ammonia bottoming cycle as a function of compressor inlet air temperature for three values of pressure ratios ($T_{oa} = 45^{\circ}\text{C}$).

Figures 10 and 11 show the same results for the environmental air temperature of $T_{oa}=30^{\circ}\text{C}$ and $T_{oa}=15^{\circ}\text{C}$ respectively.

Fig. 10 Back work ratio for gas topping and ammonia bottoming cycle as a function of compressor inlet air temperature for three values of pressure ratios ($T_{oa} = 30^{\circ}\text{C}$).

The BWR for the ammonia cycle is reduced substantially as T_{oa} goes down. This is due to the fact that ammonia compressor power is reduced while the ammonia turbine power is increased. The increase in ammonia cycle power is the result of lower ammonia condenser temperature as T_{oa} is reduced. It should be pointed out the nominal values for the air-cooled condenser temperature difference are fixed at 20°C in this study. Therefore, the ammonia pressure at the exit of the ammonia turbine goes down as T_{oa} is reduced producing higher power.

Fig. 11 Back work ratio for gas topping and ammonia bottoming cycle as a function of compressor inlet air temperature for three values of pressure ratios ($T_{oa} = 15^\circ\text{C}$).

The comparison of figures 9,10 and 11 also show that the BWR of the gas topping cycle is also reduced for a given pressure ratio (P_r) as T_{oa} is reduced. As expected this effect is not as dramatic as the BWR for the ammonia cycle.

The recuperator plays an important role in performance of the gas/ammonia combined cycle. For a fixed pressure ratio the gas and ammonia approach temperature difference (ΔT_{app}) and recuperator effectiveness control the ammonia cycle turbine inlet temperature. It is important to analyze the impact of recuperator effectiveness and ΔT_{app} on irreversibilities of the recuperator and ammonia boiler.

Fig. 12 provides this for the nominal settings. As expected, the irreversibility of the ammonia boiler is a strong function of ΔT_{app} .

Fig. 12 The effect of recuperator effectiveness and Gas/Ammonia approach temperature difference on ammonia boiler and recuperator irreversibilities.

It should be pointed out that in these calculations the pinch point temperature of the ammonia boiler is set at 10°C . For a fixed ΔT_{app} , the irreversibility of the ammonia boiler is reduced almost linearly as the recuperator effectiveness increases. Naturally, higher values of ΔT_{app} and lower values of recuperator effectiveness correspond to reduced heat exchange surface area and lower the capital cost of that component in the system. As expected, for a fixed pressure

ratio, the irreversibility of the recuperator is only affected by the recuperator effectiveness and is not a function of ΔT_{app} . It should be pointed out that the recuperator is not a balanced heat exchanger and a portion of its irreversibility is due to this imbalance (Bejan et al., 1996). Figures 13 and 14 give the irreversibility distribution of all components in this cycle from 45°C to 15°C and the ISO condition with no cooling, respectively. The calculations are provided for the nominal settings. The irreversibility for each component is reported as a percent of the sum of the total irreversibility of all components in the system and the total exergy losses in all system exit streams. The exergy losses of the exit streams are also given in the same figures. The figures show the irreversibility of the combustion chamber is by far the largest, followed by the gas turbine and ammonia boiler. Since no cooling is used for the ISO condition ($T_{oa} = 15^\circ\text{C}$, $T_1 = 15^\circ\text{C}$) of Figure 14, the irreversibilities of all components associated with cooling are zero. Figure 15 shows the specific work, defined as the net work produced divided by the mass flow rate of fuel, as a function of compressor inlet air temperature for three pressure ratio values.

Fig. 15 Specific work and total specific irreversibility for different pressure ratios.

The specific total irreversibility is also given on the same figure for comparison. From the figure it can be seen that the maximum of the specific work corresponds to the minimum of the specific irreversibility. It should be pointed out that the total irreversibility in this calculation includes the exergy of the exit streams and a balance of the specific irreversibility due to the variation of specific input exergy into the system. In order to calculate the balance specific irreversibility, a predetermined bound for the specific input exergy must be defined. Therefore, the absolute values given in Fig. 15 for the specific total irreversibility do not have particular physical significance and are only used for the second law optimization. The balance irreversibility defined in this study, called the Room-to-Move irreversibility, has been discussed in second law optimization previously (Bejan, 1996).

CONCLUSIONS

A gas / ammonia combined cycle with integrated air cooling system is proposed and analyzed. The power and efficiency of the system are evaluated and preliminary analysis shows that the cycle can produce high power and efficiency. The effect of the compressor inlet air cooling on the system performance is evaluated and it is shown that the reduction of power during hot summer months can be substantially reduced. The irreversibility of each component under reasonable system constraints is determined to see how the combined cycle can be improved. The effects, of varying important system parameters, on both power and first and second law efficiency are presented. Optimization of the system with respect to the important system parameters is discussed. More elaborate analysis including pressure drops in all components and thermoeconomic studies must be performed to determine the final design condition. The proposed cycle, capable of producing high power and efficiency along with its potential for air cooling of the condenser, merits further investigation.

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REFERENCES

- Ait-Ali, M.A., 1997, "Optimum Power Boosting of Gas Turbine Cycles with Compressor Inlet Air Cooling", ASME Journal of Engineering for Gas Turbines and Power, Vol. 119, pp 124-30.
- Bejan, A., 1996, *Entropy Generation Optimization*, CRC Press, Inc., Boca Raton.
- Bejan A., Tsatsaronis G., and Moran M., 1996, *Thermal Design and Optimization*, John Wiley and Sons, Inc.
- Briesch M.S., Bannister R.C., Diakunchak I.S., and Huber, D.J., "A Combined Cycle Designed to Achieve Greater than 60 Percent Efficiency", ASME Journal of Engineering for Gas Turbines and Power, Vol. 117, pp. 734-741.
- Dechamps P.J., 1998, "Advanced Combined Cycle Alternatives with the Latest Gas Turbines", ASME Journal of Engineering for Gas Turbines and Power, Vol. 120, pp. 351-357.
- DeLucia M., Bronconi R., and Carnevale E., 1994, "Performance and Economic Enhancement of Cogeneration Gas Turbine Through Compressor Inlet Air Cooling", ASME Journal of Engineering for Gas Turbines and Power, Vol. 116, pp. 360-365.
- Ebeling J.E., Halil R.O., Bantam D., Bakenhaus B., Schreiber J., and Wendland R., 1992, "Peaking Gas Turbine Capacity Enhancement Using Ice Storage for Compressor Inlet Air Cooling", ASME Paper No. 92-GT-265.
- F-Chart Software, 1995, EES, Engineering Equation Solver, F-Chart Software, Middleton, WI.
- Horlock J.H., 1995, Combined Power Plants-Past, Present, and Future, ASME Journal of Engineering for Gas Turbines and Power, Vol. 117, pp. 608-616.
- Horlock J.H., Young J.B., and Mantrida G., 2000, "Exergy Analysis of Modern Fossil Fuel Power Plants", ASME Journal of Engineering for Gas Turbines and Power, Vol. 122, pp. 1-7.
- Kehlhofer R., 1997, *Combined-Cycle Gas and Steam Turbine Power Plants*, Pennwell Pub.
- Khartchenko N.A., 1998, *Advanced Energy Systems*, Taylor and Francis.
- Kotas T.J., 1995, *The Exergy Method of Thermal Plant Analysis*, Krieger Publishing Company, Malabar, Florida.
- Kraft J.E., Everlein D., 1999, "Inlet Air Cooling Systems Boost Summer Peaking Capacity at WE", Power, pp. 74-79.
- Ondryas I.S., Wilson D.A., Kwamoto M., Haub G.L., 1991, "Options in Gas Turbine Power Augmentation Using Inlet Air Chilling", ASME Journal of Engineering for Gas Turbines and Power, Vol. 113, pp. 203-211.
- Razani A. and Kim K.J., 2000 "Second Law Analysis and Optimization of Combined Triple Cycle", 7th DOE/HBCU/OMI Symposium, Pittsburgh, PA.

Fig. 13 Irreversibility distribution for different components in the gas / ammonia combined cycle with cooling ($T_{oa} = 45^{\circ}\text{C}$, $T_1 = 15^{\circ}\text{C}$), nominal values.

Fig. 14 Irreversibility distribution for different components in the gas / ammonia combined cycle with no cooling ($T_{oa} = 15^{\circ}\text{C}$ and $T_1 = 15^{\circ}\text{C}$), nominal values.