# Modes of Heat Transfer

## Progress Report

**Transport Phenomena of Chemical Engineering Lab 301**
**Thursday A-5**

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**Nomenclature**

| Symbol | Term | Units |
|---|---|---|
| A | Surface Area of the Heater | $m^2$ |
| $A_c$ | Surface area of the Coil | $m^2$ |
| $A_{cy}$ | Surface Area of the Cylinder | $m^2$ |
| D | Diameter of Heated Cylinder | $m$ |
| F | Area of geometric factor | - |
| $h_r$ | Heat Transfer Coefficient: Radiation | $\frac{W}{m^2 * k}$ |
| $h_c$ | Heat Transfer Coefficient: Natural Convection | $\frac{W}{m^2 * k}$ |
| $h_f$ | Heat Transfer Coefficient: Forced Convection | $\frac{W}{m^2 * k}$ |
| $m_w$ | Mass Flow Rate of Cooling water | $\frac{kg}{s}$ |
| $C_{pw}$ | Specific Heat of Water | $\frac{J}{kg * K}$ |
| k | Thermal Conductivity | $\frac{W}{m * k}$ |
| Nu | Nusselt Number | - |
| Pr | Prandtl Number | - |
| $Q_r$ | Heat Transfer: Radiation | $W$ |
| $Q_c$ | Heat Transfer: Natural Convection | $W$ |
| $Q_f$ | Heat Transfer: Forced Convection | $W$ |
| $Q_w$ | Energy transferred to the Condenser | $W$ |
| $Q_{total}$ | Total Heat Loss | $W$ |
| Re | Reynolds Number | - |
| $T_s$ | Temperature of Heater Surface | ℃ |
| $T_A$ | Temperature of Surrounding Air | ℃ |
| $T_f$ | Average Film Temperature of the Air | ℃ |
| $\Delta T_{LM}$ | Log mean temperature difference between the cooling water and the condensing vapor | ℃ |
| $U_a$ | Air Velocity | $\frac{m}{s}$ |
| $\sigma$ | Stefan Boltzmann Constant for Radiation | $\frac{W}{m^2 * K^4}$ |
| $\varepsilon$ | Emissivity of the cylinder | - |
| $\upsilon$ | Kinematic Viscosity of Air or of the Water | $\frac{m^2}{s}$ |
| $\beta$ | Coefficient of thermal expansion of air | - |
| g | Gravitational acceleration | $\frac{m}{s^2}$ |
| C | | - |
| n | | - |

| | | |
|---|---|---|
| Ra | Rayleigh Number | - |
| Gr | Grashof Number | - |
| $\mu$ | Linear velocity of the Water Flow | $\frac{m}{s}$ |
| U | Overall Heat Transfer Coefficient at the Condenser Coil | $\frac{W}{m^2 * k}$ |
| Ø | Heat flux from the cylinder | $\frac{W}{m^2}$ |
| h | Heat transfer coefficient between the heater surface and the surrounding liquid | $\frac{W}{m^2 * k}$ |

## 1.0 Introduction and Background

Generally, heat is defined as energy transfer due to temperature gradients. There are three basic ways by which heat is transferred: radiation, conduction, and convection. These modes of heat transfer occur constantly throughout the universe and are applicable any time a process, object, or space requires heat. To begin, radiation is characterized as the transfer of heat by electromagnetic waves, where radiated photons may either be reflected, absorbed, or transmitted. The energy propagated through such electromagnetic waves does not require a medium. In fact, the heat received on Earth from the Sun is the result of electromagnetic waves traveling through the void of space between the Earth and the Sun [1]. Conduction, on the other hand, is the transfer of heat from a hot body to a cold body, usually a solid, in direct contact with one another. Lastly, convection requires the transfer of heat from one position to another by the movement of fluids, such as gases or liquids.

In this convective and radiant heat transfer experiment, the group could calculate the heat loss for radiation ($Q_r$) and natural and forced convection ($Q_c$) from the heating element. Additionally, the fraction of heat loss by radiation and natural and forced convection to the air at steady-state was calculated by using experimental data found by testing the equipment. Lastly, the group recorded the total heat transfer ($Q_{total}$) from the cylinder. In summary, this experiment involved the testing of the radiant and natural convection heat transfer by varying the temperature and finding the amount of heat lost through convection and radiation. The second part of the experiment involved varying the air velocity into the cylinder as the air blower was on and analyzing the relationship between air velocity and forced convection heat transfer.

Depending on how the flow originates, the form of convection present can be either natural or forced. Natural convection is considered a heat transport process of any fluid movement between two locations that happens on its own, in which no external sources are needed. For example, when a hot object is exposed to cold air, the temperature of the exterior of the object will drop, while the temperature of the nearby air rises. Therefore, the driving force for natural convection is a result of differences in fluid densities due to the temperature gradient. For instance, when a hot object is exposed to the cold atmosphere, the temperature of the air adjacent to the hot object is higher, thus making it not as dense as the cool air. Therefore, the hot air surrounding the hot

object will experience an upward buoyant force, which allows the hot air to rise while the cooler, denser, air of the atmosphere sinks. Natural convection can be found throughout nature, such as the natural updraft birds and gliders, which seek to extend flight time, or any wind or air movement in hurricanes and tornadoes [2].

The other form of convection is forced convection, which is the force of fluid movement by induced external forces such as a fan or a pump. These external forces, in general, intensify the motion of the fluid, which in turn increases the overall heat transfer; thus, a direct relationship can be observed between velocity and heat transfer [3]. Forced convection applications are found commonly in everyday life. As an illustration, the most common and practical example is central heating in most homes. The most common method used in central heating involves the burning of fossil fuels, which heats the equipment. The heat is fed to multiple locations by forced-air through duct work or by water circulating through pipes. In this example, the fan provides a driving force for the fluid to move and transfer the heat gained into the room.

Although radiation and convection can be observed separately, in reality these modes of heat transfer are extremely difficult to isolate. Thus, the Combined Convection and Radiation Heat Transfer Unit provides a method of observing a combination of natural or forced convection with radiation by minimizing heat loss due to conduction. First, radiation and heat loss by natural convection can be measured. For instance, by increasing the surface temperature of the unit, the heat will be transferred from the hotter surface to the surrounding air. The air that meets the hot surface heats up because of differences in fluid densities due to temperature gradients, which causes it to rise and therefore a convection flow occurs. Additionally, some heat is also transferred to the surroundings due to radiation from the heated surface.

Second, the combined heat transfer unit can measure radiation and heat loss by forced convection when the blower is on. Because forced convection greatly increases the movement of air past the heated cylinder, the heat transfer rate from the surface improves greatly. A blower transfers the heated air throughout the equipment by increasing the air velocity flowing through the cylinder. Because increasing velocity by an external force directly affects forced convection,

the result of increasing air velocity flowing through the cylinder can be used to measure the effects on radiant heat transfer.

In the previous mode of heat transfer experiment with the Combined Convection and Radiation Heat Transfer Unit, the experiment purely dealt with heat transfer by convection to single phase fluids or simply pure liquids or pure gases. During the past few decades, great interest has developed in convection heat transfer processes associated with boiling and condensation, which differ from other forms of convection because they depend on the latent heat of vaporization of the fluid and the surface tension at the liquid-vapor interface. In addition, the heat transfer coefficients associated with boiling and condensation are typically much higher than those encountered in other forms of single phase convection processes.

To begin, boiling is associated with a change in phase from liquid to vapor. This process occurs at the solid-liquid interface when a liquid is brought into contact with a surface maintained at a temperature sufficiently above the saturation temperature of the liquid. When a liquid at saturation temperature is in contact with the surface of a solid, usually a metal, at a higher temperature, heat is transferred to the liquid, and a phase change of some of the liquid begins to occur.

There are two basic types of boiling: pool boiling and flow boiling [3]. Pool boiling is the process in which a heated surface is submerged in a stagnant, non- agitated, liquid pool, where no bulk fluid flow is present. On the other hand, flow boiling, or forced convection boiling, has the presence of bulk fluid flow in the flowing stream.

The most important invention in the field of boiling was done in 1934, by S. Nukiyama, who observed that boiling takes different forms depending on the value of the excess temperature. As the value of the excess temperature increases, the curve in Figure 1 traverses four different regimes: natural convection, nucleate boiling, transition boiling, and film boiling.

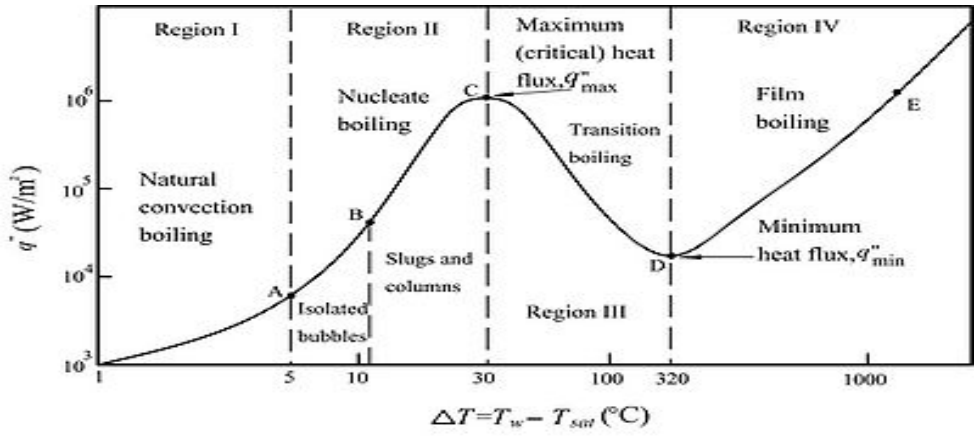



**Fig 1**. Pool Boiling in Water on a Horizontal Wire at Atmospheric Pressure

Before the first bubble is created at point A, all heat transfer is done by natural convection currents. As the liquid begins to heat, the heat of the liquid at higher temperature transfers from the liquid to the bubble, causing the bubble to multiply and rise to the top under the influence of buoyancy. The temperature difference between the vapor in the forming bubbles and the surrounding liquid provides the driving force for heat transfer between the two phases. This region becomes known as nucleate boiling. At point B, the heat transfer coefficient begins to gradually decrease due to an increase in bubbles, which lessen the total surface to liquid contact. Although the heat transfer coefficient decreases, the heat rate continues to rise until Point C due to an increase in excess temperature. Beyond the peak of the boiling curve at point C, the transition boiling regime is entered. In this regime, a vapor film forms around the wire, and portions of this film break off and rise, briefly exposing a portion of the wire surface [3]. Between points C and D, there is a combination of both types of boiling. Because the surface is now primarily in contact with a combination of bubbles and a vapor film, the heat transfer coefficient begins to reduce drastically; therefore, causing the overall heat transfer rate to decrease until point D. Finally, when the transition boiling regime reaches the upper temperature limit, the heating surface and the bulk liquid are completely separated by a stable vapor film. Thus, this is the stable film-boiling region. In this region, there is no longer any liquid-surface contact; however, the heat transfer rate continues to increase as excess temperature increases because heat transfer by radiation becomes significant.

Like boiling, there are two major modes of condensation: film condensation and dropwise condensation. Film condensation is the process where a liquid dampens a surface and spreads out, forming a film upon a surface. In dropwise, the surface is not initially wetted causing droplets to form and trickle down the surface, fusing with another condensate drops as they encounter one another. Thus, heat transfer at very high rates begins to occur on these exposed areas. In turn, the overall heat transfer coefficient for dropwise condensation is five to ten times larger than that of the film-wise coefficient [3].

In the boiling heat transfer portion of the experiment, a chamber equipped with a cylindrical copper heating element inside was used to observe the different aspects of convective and boiling heat transfer. First, the group calculated the effect on the overall heat transfer coefficient (U) and heat duty ($Q_w$) when the condenser flow rate was varied. With the changing of condenser flow rate, the Reynolds Number for the water flow was also able to be determined and correlated with a corresponding flow regime for the water flow. For technical objective two, the effect of heat flux (Ø) and heat transfer coefficient (h) on heat transfer from the heater surface to the surrounding liquid was calculated. Then, a boiling curve of log ($\frac{Q}{A}$) versus ($T_1 - T_2$) was plotted and the respective boiling regimes for each trial condition was identified. In summary, through a detailed analysis of experimental flow rates for technical objective one, the group could determine the effects of the condenser cooling water flow rate on the heat duty and overall heat transfer coefficient. Additionally, for technical objective two, the group could determine the effects of increased power input on the overall heat transfer coefficient of the apparatus.

## 2.0 Experimental Methodology

*2.1 Equipment and Apparatus (Boiling Heat Transfer)* – A boiling heat transfer unit was used to complete the technical objectives by providing data to determine the effect on the heat transfer coefficient by varying the condenser water flow rate and boiling regimes. The actual boiling heat transfer unit used for this experiment is pictured in Figure 2.

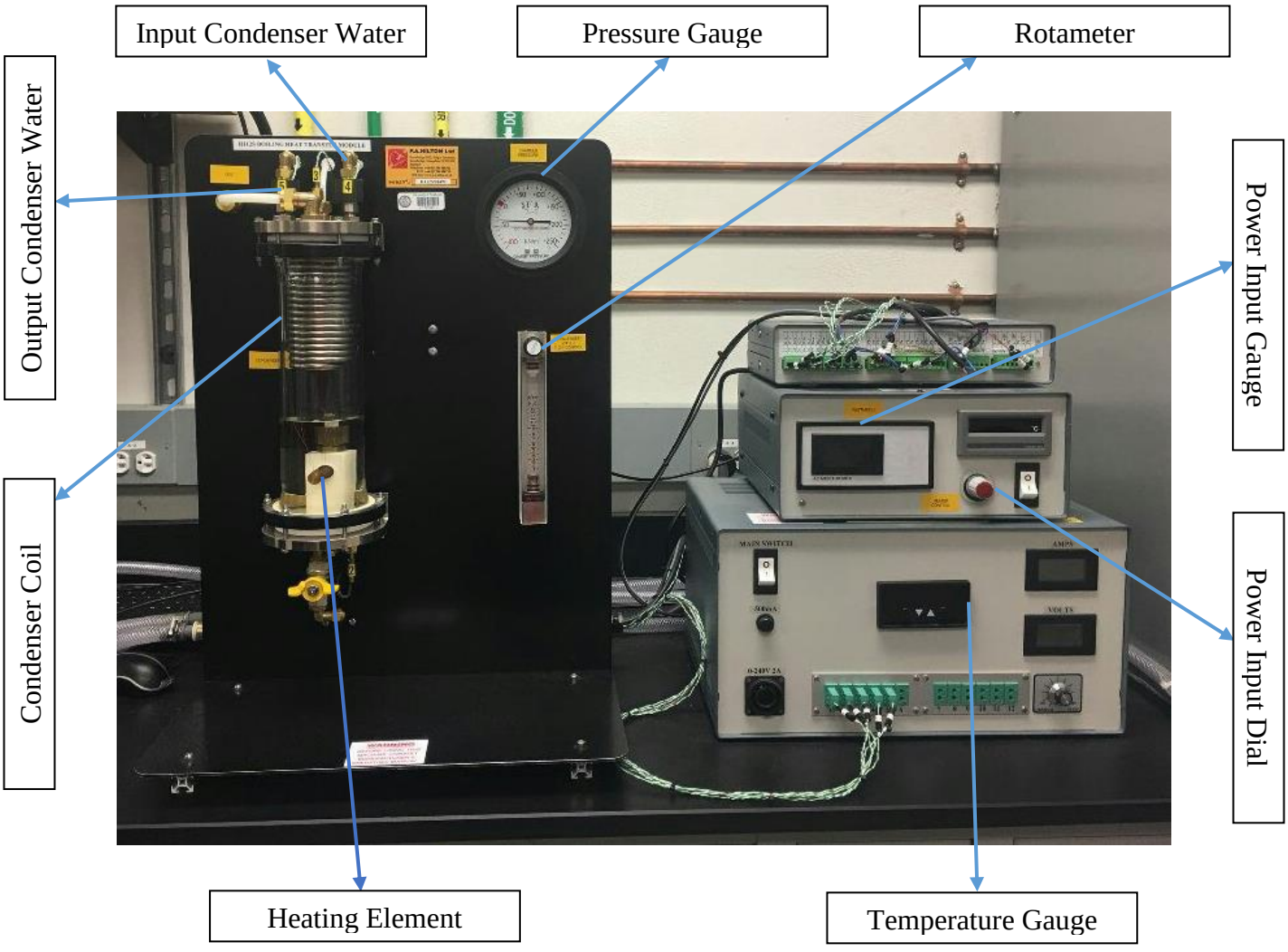



**Fig. 2.** Boiling heat transfer unit used in the experiment.

The boiling heat transfer unit consists of a chamber which houses the SES36, condenser coil, and heating element. Power input to the heating element is controlled by a dial, and the wattage to the unit is displayed on a digital gauge. Condenser water flow rate is controlled by a rotameter that can adjust the flow in units of g/s. The gauge pressure within the chamber is displayed on the

pressure gauge in units of $kN/m^2$. Throughout the unit, five temperatures are also measured ($T1_b$: heating element surface temperature $T2_b$: liquid SES36 temperature $T3_b$: vapor SES36 temperature $T4_b$: condenser water inlet temperature $T5_b$: condenser water outlet temperature) and displayed on a digital gauge.

During the experiment, power is input to the heating element which heats the liquid SES36. Once heated, the SES36 begins to boil. The vapor then rises and hits the condenser coil which condenses the SES36 vapor. This cycle happens constantly in the chamber throughout the experiment, and once the cycle reaches steady-state, data is recorded.

*2.2 Experimental Procedures (Boiling Heat Transfer)* – The first technical objective was to calculate the effect of varying the condenser cooling water flow rate on the heat duty of the condenser cooling water and the overall heat transfer coefficient at the condenser coil. The effect on the heat duty and heat transfer coefficient were calculated by varying the cooling water flow rate while holding all other experimental variables constant.

At the start of the experiment, the chamber pressure was confirmed that it was at the saturation pressure of the SES36. The rotameter needle valve was adjusted to set the condenser water flow rate to about 12 g/s. At the same time the red heater control dial on the small control console box was adjusted to set the power input to the heater to 200 W. Once the water flow rate and input power to the heater were set, the system was allowed to reach steady-state. Steady-state was defined when the chamber pressure, along with the temperatures of the liquid SES36, $T2_b$, and of the vapor SES36, $T3_b$, in the chamber, stabilized. When steady-state was declared, the chamber pressure, condenser water flow rate, power input, and the five temperatures throughout the system were recorded.

After recording the steady-state data for 12 g/s, the condenser water flow rate was adjusted to 10 g/s while keeping all other variables constant. Again, the system was allowed to reach steady-state and the data was recorded. This procedure was performed four more times at 8 g/s, 6 g/s, 4 g/s, and 2 g/s. During each trial the system reached steady-state and during steady-state the data was recorded for each flow rate. With each trial's data, analysis was carried out to determine the effects of the condenser water flow rate in technical objective one.

The goal of the second technical objective was to observe and compare the behavior of nucleate boiling and film boiling while also comparing the heat transfer coefficient between the heating element and liquid SES36. Before starting the second technical objective, the chamber pressure was confirmed to be at the saturation pressure of SES36. After the chamber pressure was confirmed, the control dial of the power input for the heater was set to 50 W and the rotameter valve was adjusted to set the condenser water flow rate at 1 g/s. The chamber pressure was continuously monitored throughout the technical objective to maintain a gauge pressure between -10 to 10 $kN/m^2$. If the chamber pressure stabilized outside this range, the condenser water flow rate was adjusted accordingly to get the pressure back within the range. Data for the 50 W trial was taken after it was affirmed that the temperature of the liquid SES36 and chamber pressure were stable. The data taken included all five temperatures, condenser water flow rate, chamber pressure, and input power to the heater. A picture was also taken in order to analyze the boiling regime at 50 W.

The power input was increased to 100 W and the condenser water flow rate was adjusted accordingly to maintain a chamber gauge pressure around 10 $kN/m^2$. Once the chamber pressure and $T2_b$ were stable, data was recorded as it was during the 50 W trial. The same procedure was carried out, data was recorded, and a photo was taken at the following power inputs: 150 W, 200W, and 250 W. The data and images were analyzed to study properties of nucleate boiling as well as the heat transfer coefficient of the heating element for this mode of boiling.

Film boiling was analyzed in the second part of technical objective two. The dial for the power input to the heater was increased to 300-325 W. As film boiling was observed, a picture was taken of the heating element to aide with observations of the boiling regime. At this point, the power input to the heater was gradually lowered while trying to sustain film boiling. Once the heating element surface temperature, $T1_b$, was stable below 160 °C with film boiling still occurring, the chamber pressure, all five temperatures, condenser water flow rate, and the heater power input were all recorded. Like the nucleate boiling data, the film boiling data was used to analyze the heat transfer coefficient of the heating element during film boiling.

*2.3 Equipment and Apparatus (Convective and Radiant Heat Transfer)* – The technical objectives in this experiment were completed using a convective and radiant heat transfer unit. This apparatus enabled data to be taken in order to measure the fraction of total heat loss from the heating element by natural convection and radiation in technical objective one. In technical objective two, the effect of forced convection on the heating element surface temperature and rate of heat transfer was evaluated. The equipment used in the experiment is pictured in Figure 3.

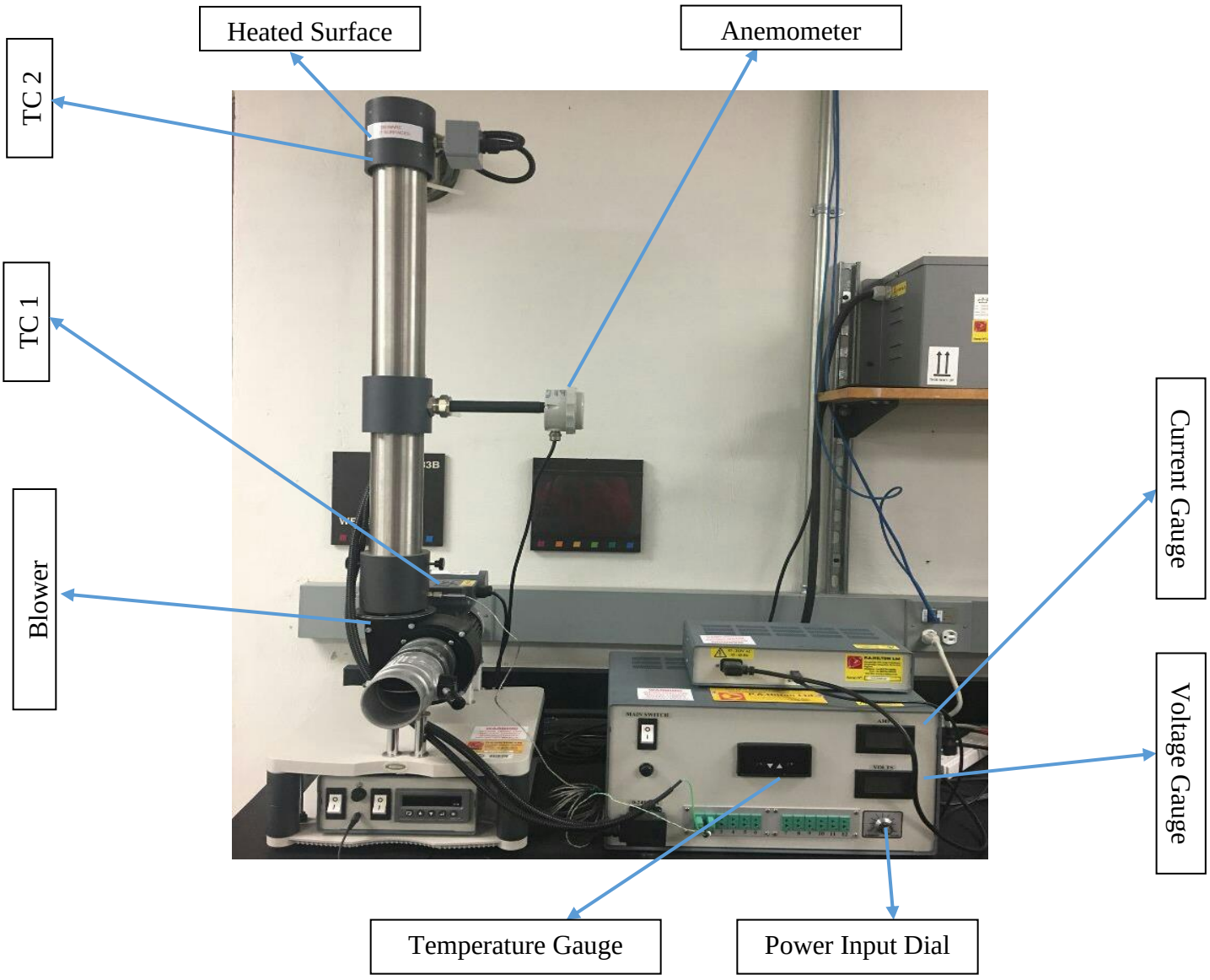



**Fig. 3.** Convective and radiant heat transfer unit used in the experiment.

Key components of the convective and radiant heat transfer unit include: a heating element that heats a surface which is the source of the convective and radiant heat transfer that takes place in

the column, and a blower in order to create forced convection. In the experiment, power is sent to the heating element which heats the surface. As the surface is heated, its temperature is measured as well as the air in the column that heat is being transferred to. The temperature gauge enables the temperatures to be measured, and the voltage gauge allows the power input to be read. The unit includes a blower in order to create forced convection by blowing the air by the heated surface. With these components, data can be taken to study the effects of natural and forced convection when it comes to heat loss and heat transfer.

*2.4 Experimental Procedures (Convective and Radiant Heat Transfer)* – In the first technical objective, the fraction of total heat loss from the heating element was determined. The heat loss determined was due to radiation and natural convection to the air measured at steady-state between heating element surface temperatures of 100 $^{o}$C and 400 $^{o}$C.

Before the start of the objective, the air flow damper was fully opened and the blower was confirmed to be off. The power input to the heating element was then adjusted to achieve a surface temperature, $T2_c$, of 100 $^{o}$C. When the temperature of the air, $T1_c$, and $T2_c$ were stable, both temperatures were recorded as well as the electric current and voltage supply to the heating element. The power input to the heating element was further adjusted to obtain temperatures of 200 $^{o}$C, 300 $^{o}$C, and 400 $^{o}$C. At each temperature, $T1_c$ and $T2_c$ were allowed to stabilize and once steady-state was reached, they were recorded with the electric current and voltage supply. With this data, the fraction of total heat loss from the heating element was analyzed and determined.

Technical objective two included evaluating the effect of air velocity on the heating element surface temperature and on the rate of convective and radiant heat transfer from the heating element at steady-state.

To start the second objective, the air flow damper was fully opened and the power to the fan and air velocity meter were turned on. The voltage to the heating element was turned on to 200 volts while the air flow damper was adjusted to maintain an air velocity of about 0.5 m/s. Once $T1_c$ and $T2_c$ were stabilized, the two temperatures, air velocity, electric current, and voltage supply were recorded. After the data was collected, the air flow damper was adjusted in order to increase air velocity by approximately 0.5 m/s. As $T1_c$ and $T2_c$ were stabilized, both temperatures, heating

element, air velocity, voltage supply, and electric current were recorded. The air flow damper was adjusted to increase the air velocity by another 0.5 m/s and the system was allowed to stabilize and data was recorded again. The air flow continued to be increased by 0.5 m/s with data collected at each new air velocity until the maximum air velocity was reached. The data collected at each air flow was used to evaluate how the effect of air velocity impacts the rate of convection and rate of radiation from the heating element at steady-state.

**3.0 Results**

Technical objective 1 of the Boiling Heat Transfer experiment was to evaluate how adjusting the condenser water flow rate affects the heat duty and heat transfer coefficient of the boiling heat transfer apparatus. Data was collected every 5 minutes to identify when steady state was reached. Calculations and data analysis were based on steady state values when the temperatures stabilized. The flow rate was adjusted from 12 g/s to 2 g/s by intervals of 2 g/s. Based on the temperatures recorded, charts were constructed to show the overall heat transfer coefficient and the heat transferred from the boiling refrigerant to the condenser water coil as represented by Figures 4 and 5.

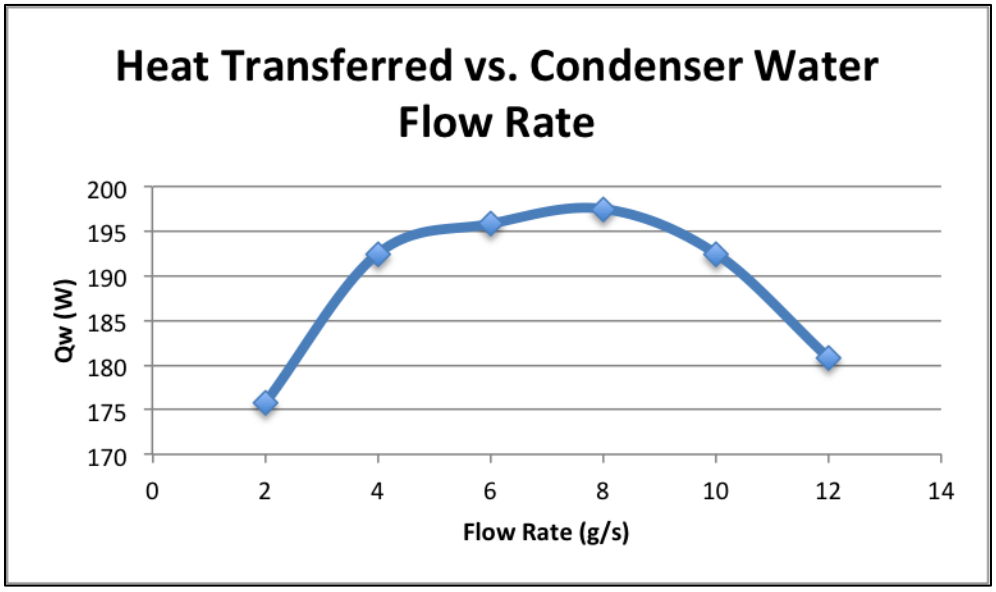



**Fig. 4.** Plot of heat transfer versus condenser water flow rate

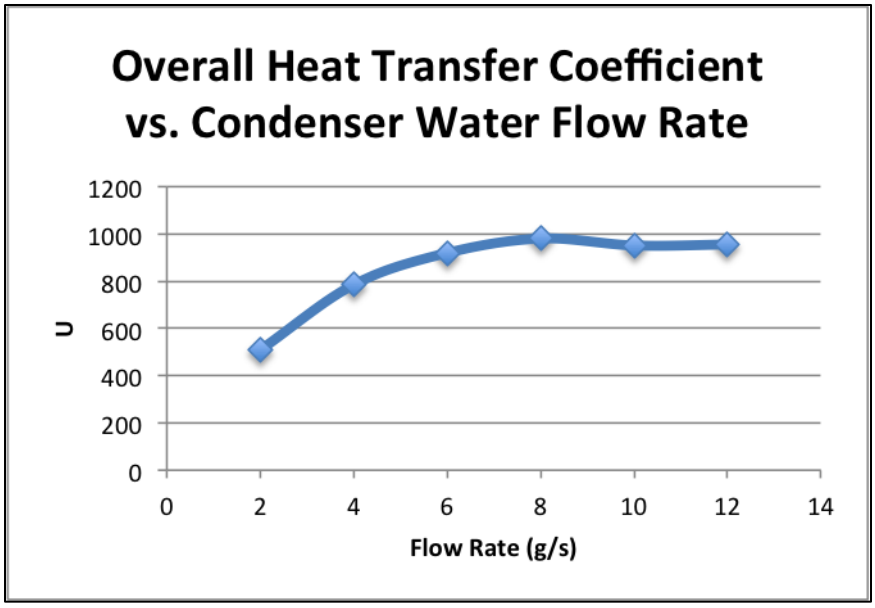



**Fig. 5.** Plot of the heat transfer coefficient versus condenser water flow rate

To better understand the flow regimes of the condenser water coil in technical objective 1, Reynolds numbers were calculated for each flow rate as shown in Figure 6 below.

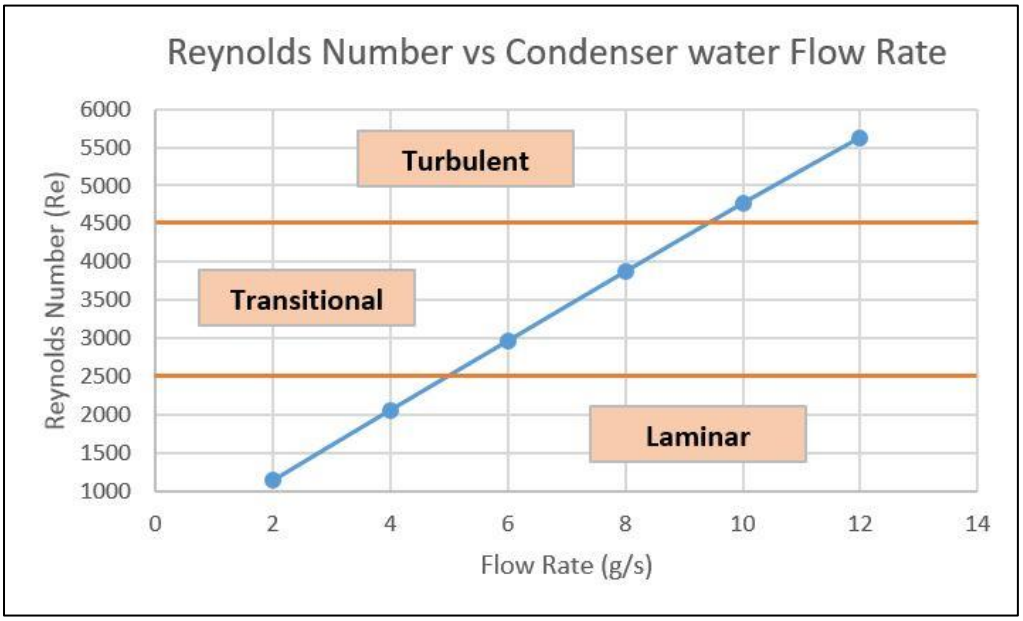



**Fig. 6.** Plot of the Reynolds number versus condenser water flow rate

The flow regimes are also depicted on the graph. The different types of flow are Laminar, Transitional, and Turbulent. Laminar flow has a Reynolds number less than 2500 while Turbulent

flow has a Reynolds number greater than 4500. Transitional flow falls between 2500 and 4500 for its Reynolds number.

The second technical objective for Boiling Heat Transfer was to calculate the heat flux and boiling regime based on heat transfer. The heat input was recorded for each trial and the area was known as 0.0018 $m^2$.

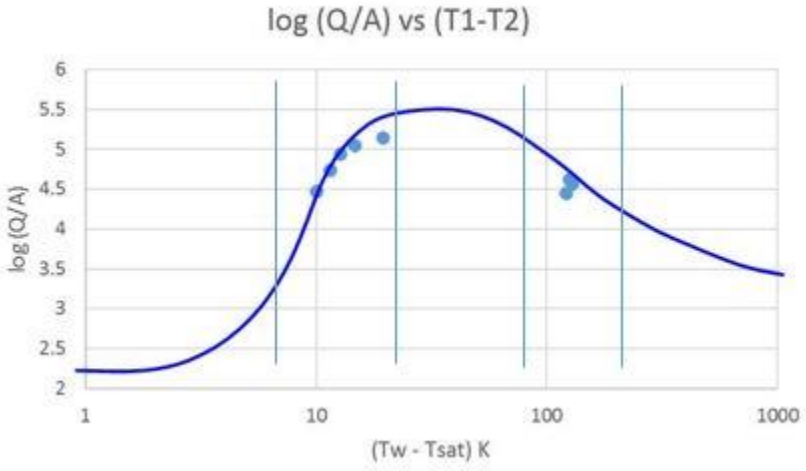



**Fig. 7.** Plot of log(Q/A) vs. (T1-T2)

The first technical objective for the Convective and Radiant Heat Transfer was to compare the heat loss from both natural convection and radiation. The power input, the surface temperature, air temperature, electric current and voltage were all recorded.

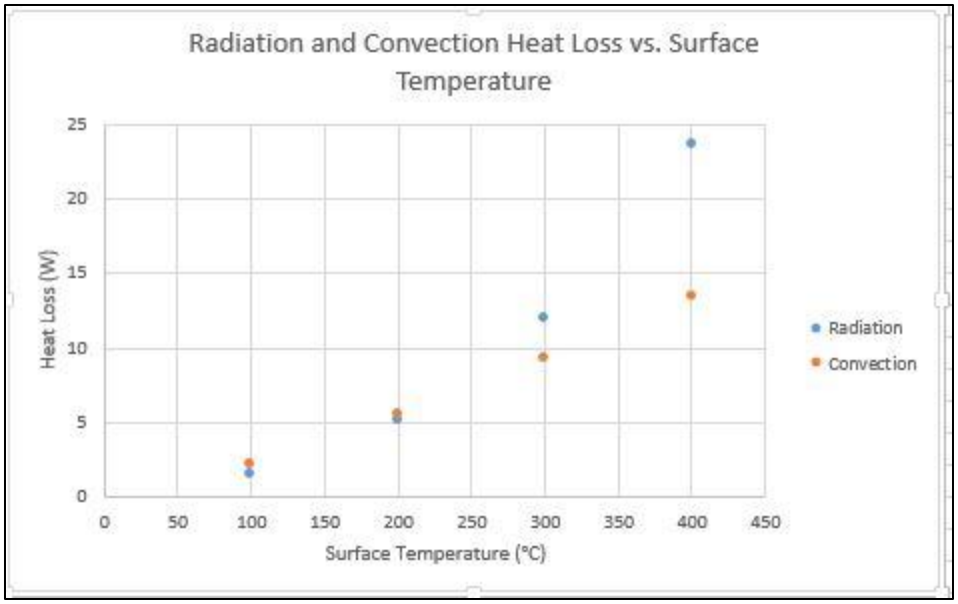



**Fig. 8.** Heat Loss of Radiation and Convection

The surface temperature was recorded for each trial and plotted versus the calculated the heat loss. Figure 8 shows both the heat loss from the radiation and convection.

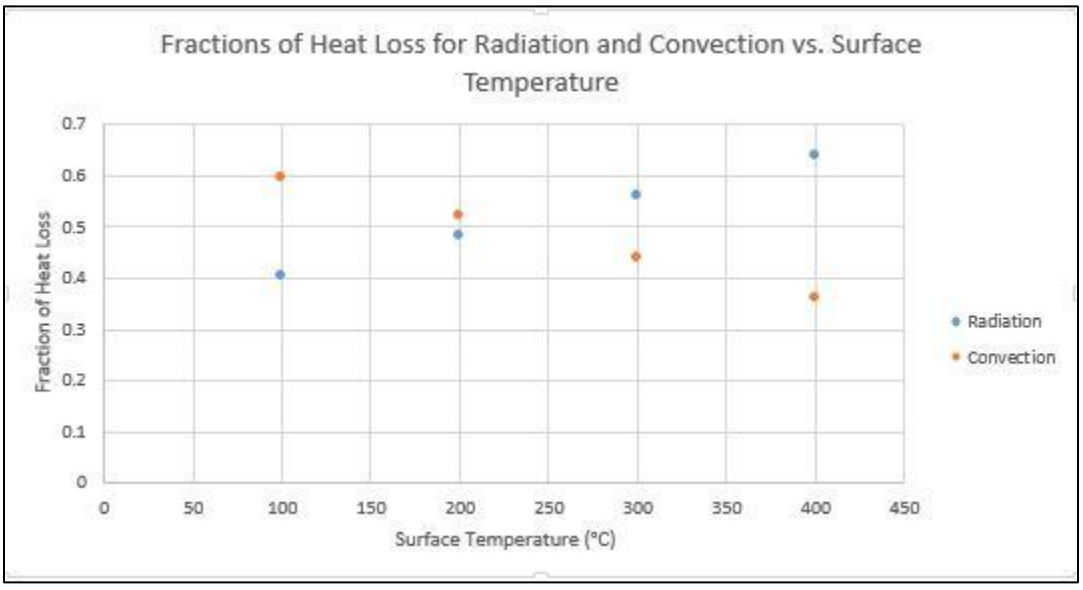



**Fig. 9.** Fractions of Heat Loss for Radiation and Convection

The second calculation of technical objective 1 was to calculate the fraction of heat loss for both radiation and convection. Figure 9 plots both fractions. Since there is only radiation and convection heat losses, as one increases, the other will decrease.

The second technical objective was to compare the heat loss from forced convection to the heat loss from radiation. This technical objective was done twice, and what is shown is the second trial; the first trial contained a human error and can be found in Appendix II. To find the heat loss due to radiation, the radiant heat transfer coefficient, the surface area of the heater, and the difference between the heated surface temperature and the air temperature were calculated. This was done for each effective air velocity that was measured. Figure 10 contains the plot of the radiant heat loss against the effective air velocity.

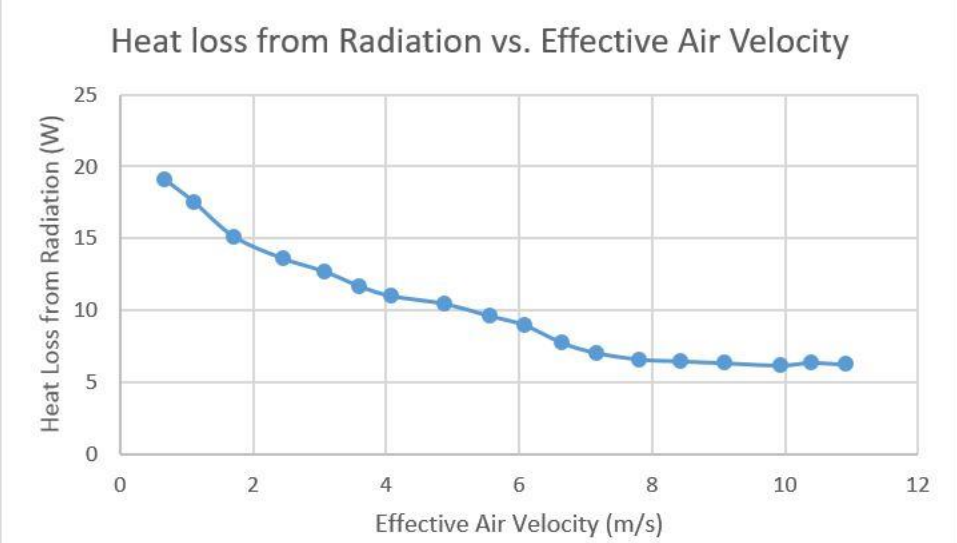



**Fig. 10.** Heat Loss from Radiation vs. Effective Air Velocity

The heat loss due to forced convection was found using the convective heat transfer coefficient ($h_c$), the surface area of the heater, and the temperature difference between the heated surface and the flowing air. To find $h_c$, the Nusselt Number needed to be found using the Prandtl Number and Reynolds number. The thermal conductivity, Prandtl Number, and kinematic viscosity were found using the Engineering Toolbox and interpolating values of each respective measurement for air at different temperatures [4]. Figure 11 displays the results of this analysis.

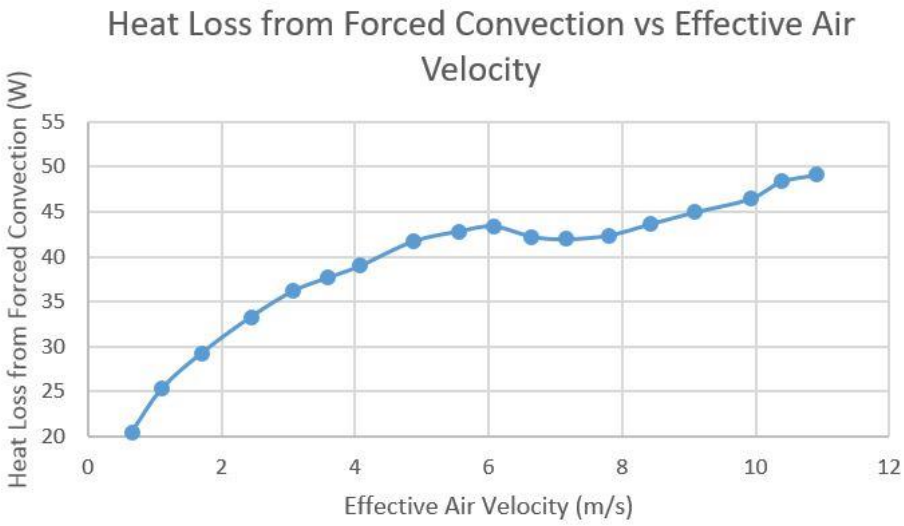



**Fig. 11.** Heat Loss from Forced Convection vs. Effective Air Velocity

Figure 12 shows both types of heat loss on the same graph to make for an easier analysis of the relationship between the two heat losses.

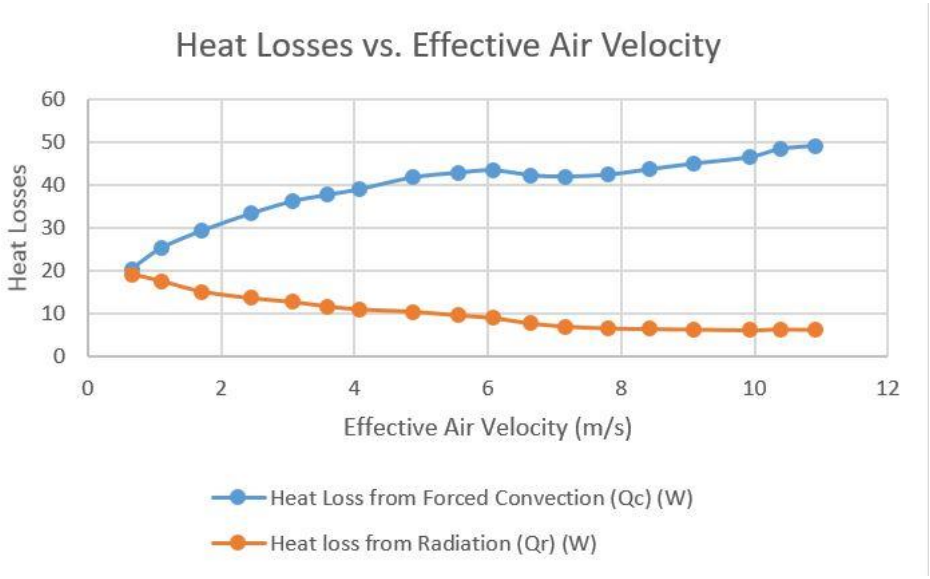



**Fig. 12.** Heat Losses vs. Effective Air Velocity

In addition, the surface temperature of the top of the column was plotted against the effective air velocity for each reading. Figure 13 shows the trend.

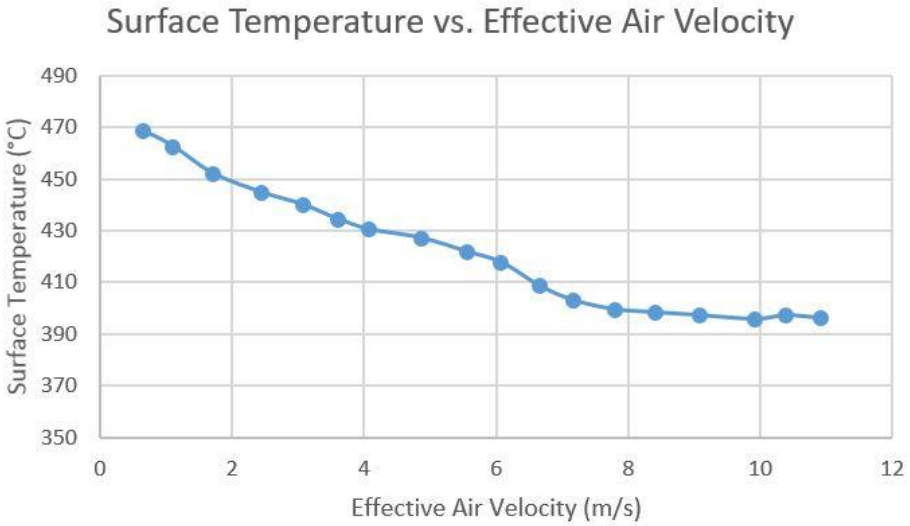



**Fig. 13.** $T2_c$ vs. Effective Air Velocity

The last part of the second technical objective was to analyze the flow regimes of the air for each trial. Figure 14 shows each Reynolds number plotted against each respective air velocity as well as the cutoffs for each flow regime.

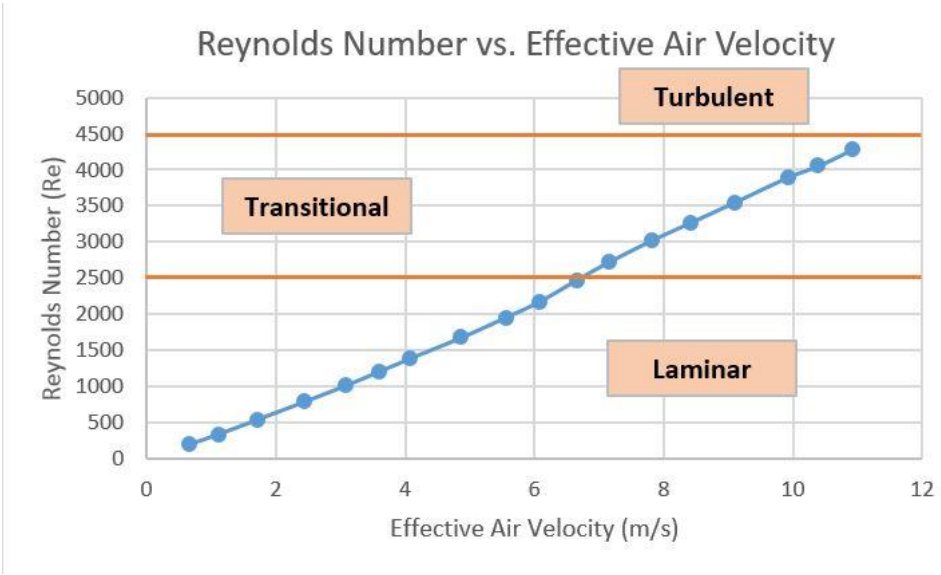



**Fig. 14.** Reynolds Number vs. Effective Air Velocity

## 4.0 Analysis and Discussion

### *4.1 Boiling Heat Transfer*

For the first technical objective, adjusting the condenser water flow greatly affected the overall heat transfer coefficient and slightly affected the heat transferred between the boiling refrigerant and the condenser coil. The $Q_w$ values ranged from approximately 175 to 195 W, which is a fairly tight group, while the overall heat transfer coefficient ranged from approximately 500 to 980 $W/m^2K$. The $Q_w$ values do not change substantially since the heat moving into the system remains constant. As the condenser water flow rate increased to 8 g/s, the heat transfer, $Q_{w,}$ also increased. The peak for $Q_w$ occurred when the flow rate was set to 8 g/s, which maximized the heat transfer between the boiling refrigerant and the condenser coil. The water flowing through the coil was traveling at the perfect speed to transfer as much heat as possible. This is because as the flow rate increased, more water was able to transfer heat with the hot boiling refrigerant. When the condenser water flow rate was fastest (10 g/s and 12 g/s), the water was moving too quickly to efficiently transfer heat with the boiling refrigerant.

As flow rate increased, the Reynolds number for the condenser water flow also increased. At the slower flow rates of 2 and 4 g/s, the flow was Laminar since the Reynolds numbers were less than 2500. The medium flow rates of 6 and 8 g/s were Transitional flow regimes since the Reynolds numbers were between 2500 and 4500. At the fastest flow rates of 10 and 12 g/s, the flow was Turbulent since the Reynolds numbers were greater than 4500. The heat transfer $Q_w$ was greatest in the transitional flow regimes. This could be because the water was moving at the perfect velocity to transfer heat with the boiling refrigerant most effectively.

The heat transfer coefficient increased as the condenser water flow rate increased. However, it did start to level off after the flow rate reached 8 – 12 g/s. In the laminar flow regimes, the overall heat transfer coefficient experienced the greatest loss and was significantly smaller than it was in the other flow regimes. The largest heat transfer coefficient was found at 8 g/s for the condenser water flow rate and leveled off at higher flow rates. From this, one can conclude that there is a limitation on the amount of heat that can be transferred in this system.

The second technical objective for boiling heat transfer involved finding the heat flux from the cylinder and calculating the heat transfer coefficient at each of the following power inputs: 50W, 100W, 150W, 200W and 250W as well as 50W when film boiling occurred. Similar parameters from technical objective 1 were recorded in Excel including $T_1$-$T_5$, the power input, the condenser water flow rate, and the chamber pressure. In order to calculate the heat flux, the power input to the cylinder was divided by the surface area (0.0018 $m^2$). This was done for the power inputs of 50W through 250W and each of the film boiling trials. The heat flux increased for each of the first 5 trials which was expected since the voltage was increasing each time. It then decreased with the 3 trials of film boiling since the power was reduced.

The next part was to find the heat transfer coefficient by using Equation 5 in the appendix. The difference in temperatures between $T1_b$ (surface temperature) and $T2_b$ (liquid temperature) was calculated for each trial. The difference in temperatures for film boiling trials was much larger than the difference for the first five trials. To then find the heat transfer coefficient, the heat flux was divided by the change in temperature. The calculations were all done in Excel and can be observed in Appendix I. The heat transfer coefficient increased for the power inputs from 50 W to 200 W but then slightly decreased at 250 W. The heat transfer coefficient of each of the trials for the film boiling was much less than that of regular boiling by an order of magnitude. This means that less heat was passed through at this regime. This makes sense because the difference in temperatures for the film boiling, as stated above, was very large. Since the temperature was so high for these trials, there was a vapor layer between the heating wall and the liquid preventing heat to be transferred. This is why the heat transfer coefficient was so much less during film boiling.

The log of heat flux was plotted against the difference in temperatures to show the different regimes of boiling. While the bubbles were first starting to form, free convection was occurring. As the data was collected, the difference of temperatures put the first trials at the nucleate-boiling stage (known as regime II). This region on the graph has a difference in temperatures of 10 K. The temperature continued to rise until the heating surface turned off due to the temperature being too high. However, by turning it back on and dropping the power source to just below 50 W, stable film boiling occurred in regime V. This is where the graph dips dramatically around a difference

of temperature of 100 °C. In between the nucleate boiling and film boiling is the transitional boiling where the temperature continued to rise as the power had to be turned off and then back on again.

*4.2 Convective and Radiant Heat Transfer*

The first technical objective for the convective and radiant heat transfer experiment required the following data parameters to be collected: the surface temperature, the air temperature, the electric current, and voltage supply. Each was measured by adjusting the power input until a surface temperature of the following temperatures for each trial was reached: 100 °C, 200 °C, 300 °C, and 400 °C. These were all recorded and used in the calculations to solve for the radiant and convective heat losses. In order to calculate the heat losses of each, an Excel sheet was created and can be observed in Appendix I. Equation 7 in Appendix I was used to solve for the heat loss of radiation. In order to solve for the heat loss, the heat transfer coefficient was solved for. The emissivity of the cylinder was given as 0.95, the shape factor as 1.0 and the Stefan-Boltzman constant as 5.67E-8 $W/m^2*K^4$. These values were used in Equation 6. Equation 12 was used to solve for the heat loss from natural convection. Multiple steps were taken in order to calculate the heat transfer coefficient. First, the Grashof Number was calculated by Equation 8. Beta (the coefficient of thermal expansion for air) and the kinematic viscosity were looked up on Engineering Toolbox by using the average film temperature and the area was known as 0.01 $m^2$ [4]. The Prandtl number and thermal conductivity were also looked up using this procedure. The Rayleigh number was calculated using Equation 9. C and n were looked up in Table 20.2 based on the Rayleigh and used to calculate the Nusselt Number in Equation 10 [3]. From the Nusselt number, the heat transfer coefficient of natural convection was calculated by Equation 11. As this experiment was run, these two types of heat loss accounted for the total heat loss that was observed.

The first part of technical objective 1 was to analyze the results of heat loss from radiation and convection versus surface temperature. Figure 8 shows the heat loss from radiation and convection versus the surface temperature. As one can observe, the heat loss from natural convection increased at almost a linear rate and was greater than the heat loss from radiation at a lower temperature. However, the heat loss from radiation grew at an exponential rate and became much greater at a

higher surface temperature. The difference between the two increased from the surface temperature of 100 °C to 400 °C which was expected since the radiation increase was exponential. The observed heat loss was expected to be much higher for natural convection at a lower temperature because of the fluid being in contact with the pipe. However, the amount of heat loss from radiation increased as the temperature increases because the radiative heat transfer coefficient focused on the difference of each of the surface and air temperature to the fourth power. This means that the transfer coefficient grew significantly with temperature.

The second part of technical objective 1 was to analyze the fractions of heat loss from radiation and convection. Since each of these values were a fraction of the total heat loss, it was expected that as one decreased, the other heat loss would increase. This was true and can be observed in Figure 9. As stated above, the radiative heat transfer coefficient depended greatly on the temperature. So, as the temperature increased, the amount of heat loss from radiation should increase as well. At a temperature of 100 °C, the fraction of heat loss from convection was 0.6 and the radiation heat loss is almost 0.4. But as temperature increased to 400 °C, the fraction of convective heat loss decreased to 0.36 and the radiative heat loss increased to 0.64. Overall, the increase in volts to the system to heat the fluid caused an increase in radiative heat transfer and a decrease in convective heat transfer.

For the second technical objective of the convective and radiant heat transfer experiment, the surface temperature at the top of the column decreased as the air velocity increased. The heat loss from radiation decreased as the effective air velocity increased, and the heat loss from forced convection increased as the effective air velocity increased. This makes sense because forced convection is usually caused by a velocity flow of a substance. In this case, air was flowing through the column at greater speeds, resulting in a greater contribution from forced convection heat loss.

The decrease in radiation heat loss is a result of the decrease in $T2_c$ as effective air velocity increased. Surfaces with lower temperatures do not radiate as much heat as surfaces with higher temperatures, so as $T2_c$ decreases, the radiative heat loss should as well, as Figure 10 shows.

The air flow regimes for each trial were either in laminar or transitional flow, according to each calculated Reynolds Number. The same boundary conditions for flow were used for this experiment as the boundary conditions for the Boiling Heat Transfer experiment. A Reynolds Number below 2,500 was considered to be laminar flow, above 4,500 was considered to be turbulent flow, and between those two values was considered to be transitional flow. About half of the trials for this technical objective resulted in laminar flow, while the other half resulted in transitional flow. This is most likely due to the relatively high kinematic viscosity of air, which results in a lower Reynolds number.

During transitional flow, each of the graphs started to level off towards an asymptote. From this information, one can assume that the surface temperature during high transitional/turbulent flow would be more resistant to change as effective air velocity increases. Therefore, increasing the Reynolds number decreases the change in radiant and forced convective heat loss as effective air velocity increases. These asymptotes suggest that there is a minimum radiant heat loss (from a minimum $T2_c$) and a maximum forced convective heat loss as effective air velocity approaches infinity.

**5.0 Summary and Conclusions**

Two modes of heat transfer, convection and radiation, were observed in the two transfer methods in this experiment. For technical objective one, the goal for the natural convection and radiation was to compare the heat loss from radiation and natural convection to the air at steady-state. Because natural convection is used, it does not utilize external forces such as an air blower. Instead, natural convection uses the forces from density differences caused by temperature variations in the air. In order to measure the variation in radiation and natural convection, the surface temperature was increased. From Figure 8, the heat loss from radiation and natural convection was observed. The graph illustrated that as the temperatures increased from (100-400) ℃ both the heat loss calculated from radiation and natural convection increased from 1.5 W to 23.5 W and 2.5 W to 14.0 W, respectively. Additionally, the results indicated the fractional heat loss from natural convection ($\frac{Q_C}{Q_{total}}$) decreased from 0.60 to 0.36 as the temperature increased from (100-400) ℃. Additionally, the fractional heat loss from radiation ($\frac{Q_r}{Q_{total}}$) increased from 0.4 to 0.64 as the

temperature increased from (100-400) ℃. These results provided a logical conclusion that the portion of heat loss at high cylinder temperatures is lost from the surface due to the radiation and the portion of heat loss at relatively low cylinder temperatures are mainly due to the natural convection.

On the contrary, technical objective two for forced convection and radiation corresponds with the evaluation of the effect of air velocity on the heating element surface temperature. Unlike natural convection, forced convection uses an external force, which required the air blower to remain on. Therefore, the velocity in technical objective 1 was increased at intervals of 0.5 $\frac{m}{s}$ until 9.0 $\frac{m}{s}$ was reached. From Figures 10 and 11, the heat loss versus effective air velocity from radiation and forced convection was observed. For Figure 10, the heat loss from radiation versus effective air velocity displayed a decreasing trend line, but for Figure 11 the heat loss from forced convection versus effective air velocity displayed an increasing trend line. Using the Reynolds Number versus the effective air velocity, the flow regimes for each air trial were able to be determined. These results provided a logical conclusion that as the air velocity increases the top of the column surface temperature would decrease.

Convection was additionally observed in the boiling heat transfer method. For the first technical objective, the effect of adjusting the condenser water flow rate on the heat duty and overall heat transfer coefficient was observed. The calculations recorded were based on steady state values when each of the temperatures stabilized, while the flow rate was decreased in intervals of 2 $\frac{g}{s}$ from 12 to 2 $\frac{g}{s}$. From Figure 4, the heat transfer versus the condenser water flow rate was observed. The graph illustrated a negative parabola with a peak at 8 $\frac{g}{s}$, which provided a valid conclusion that as the flow rate increased more water was present in the coil. This enabled more transfer of heat, as illustrated in Figure 5. Furthermore, the Reynolds numbers were calculated for each flow rate and corresponded to the correct flow regime: laminar, turbulent, and transitional. In the laminar flow regime, the Reynolds number was significantly smaller than in the turbulent and transitional regimes, which created a drop off in the overall heat transfer coefficient of the system. From the remaining two flow regimes, the heat transfer $Q_w$ was found to be the greatest in the

transitional flow regime, which provided a logical conclusion that as the flow rate considerably increased, the Reynolds number also increased creating the greatest heat transfer.

The second technical objective for the boiling heat transfer involved finding the heat flux from the cylinder and calculating the heat transfer coefficient as the power inputs were changed from 50-250 W as well as 50 W when film boiling occurred. The heat input over the known surface area was recorded for each trial in Figure 7. The graph demonstrated an increase in heat flux for each of the five continuous trials, which was reasonable since the voltage to the apparatus was increased. On the other hand, when the power was reduced to 50 W for the next 3 trials of film boiling, a decrease in heat flux was seen. Next, the temperature difference between the surface and liquid was calculated. By measuring the difference in temperatures, the heat transfer coefficient was then able to be determined by dividing the heat flux by the change in temperature. The results demonstrated that the heat transfer coefficient increased for the power inputs from 50 W to 200 W but then decreased at 250 W. In contrast, the heat transfer coefficient of the each of the trials for the film boiling was recorded to be much less than that of regular boiling. Therefore, less heat was being transferred through this regime. These results provided an accurate conclusion that as the temperature difference in the surfaces increases for film boiling a vapor layer forms between the heating wall and the liquid preventing heat to be transferred easily. Therefore, the heat transfer coefficient is so much less during film boiling.

## 6.0 References

[1] T. P. Classroom, "Methods of heat transfer," 1996. [Online]. Available: http://www.physicsclassroom.com/class/thermalP/Lesson-1/Methods-of-Heat-Transfer. Accessed: Oct. 17, 2016.

[2] "Heat transfer,". [Online]. Available: http://physics.bu.edu/~duffy/py105/Heattransfer.html. Accessed: Oct. 17, 2016.

[3]J. R. Welty, C. E. Wicks, G. L. Rorrer, and R. E. Wilson, *Fundamentals of momentum, heat and mass transfer 5E*, 6th ed. Chichester, United Kingdom: John Wiley [distributor], 2008.

[4] (2016). *Fluid Properties; Air*. The Engineering ToolBox. http://www.engineeringtoolbox.com/dry-air-properties-d_973.html

**Appendix I**

*Sample Calculations*

*Boiling Heat Transfer Technical Objective 1*

Sample Calculations for 8 g/s flow rate:

Calculating the heat transfer for water

$$Q_W = m_W * C_{pW} * \Delta T \quad \text{Equation 1}$$

$Q_w$ = (8 g/s)(4.184 J/g˚C)(29.7-23.8 ˚C) = 197.4848 W

u = (8 g/s) / (0.9965125 g/cm$^3$) / ( 0.072965 cm$^2$)*(1 m / 100 cm) = 1.10024 m/s

Reynolds Number

$$R_e = \frac{u * D}{v}$$

$R_e$ = (1.10024 m/s)(.003048 m)/(8.66975 x $10^{-7}$) = 3868.08 (Transitional)

Calculating the log mean temperature

$$\Delta T_{lm} = \frac{(T_{Hin} - T_{Cout}) - (T_{Hout} - T_{Cin})}{\ln \frac{(T_{Hin} - T_{Cout})}{(T_{Hout} - T_{Cin})}} \quad \text{Equation 2}$$

$\Delta T_{lm}$ = [(33.5 – 29.7˚C)–(33.5 – 23.8˚C)] / [ln((33.5 – 29.7˚C)/(33.5 – 23.8˚C))] = 6.29585 ˚C and K (when adding 273 to each value, it cancels out)

Solving for U

$$Q_W = U * A * \Delta T_{LM} \quad \text{Equation 3}$$

U = (197.4848 W) / [(.032 m$^2$)(6.29585 K) = 980.23256 W/m$^2$K

*Boiling Heat Transfer Technical Objective 2*

Heat flux, where area = 0.0018 m$^2$

$$\emptyset = \frac{Q}{A} \quad \text{Equation 4}$$

$$\emptyset = \frac{53\ W}{0.0018\ m^2} = 29444.44\ \frac{W}{m^2}$$

Heat transfer coefficient based on Q = h * A * ΔT

$$h = \frac{\emptyset}{(T_1 - T_2)}$$ Equation 5

$$h = \frac{29444.44\frac{W}{m^2}}{(44 - 34)K} = 2944.44\frac{W}{m^2 * K}$$

To plot the log(Q/A) vs ΔT:

$$\log\frac{Q}{A} = \log\left(\frac{53}{0.0018}\right) = 4.469$$

$$(T_1 - T_2) = (44 - 34)K = 10K$$

*Convective and Radiant Heat Transfer Technical Objective 1*

<u>Heat loss from Radiation</u>

Radiant heat transfer coefficient

$$h_R = \beta * F * \Omega * (T_s^4 - T_a^4)/(T_s - T_a)$$ Equation 6

$$h_R = 0.95 * 1 * 5.67 * 10^{-8}\frac{W}{m^2 * K^4} * \left(\frac{372.9^4 - 291.7^4}{372.9 - 291.7}\right)K^3 = 8.024\frac{W}{m^2 * K}$$

Heat loss from radiation

$$Q_R = h_R * A * \Delta T$$ Equation 7

$$Q_R = 8.024\frac{W}{m^2 * K} * 0.0022m^2 * (372.9 - 291.7)K = 1.433\ W$$

<u>Heat loss from Convection</u>

Grashof Number

$$Gr = \frac{g * \beta * (T_S - T_A) * D^3}{v^2}$$ Equation 8

$$Gr = \frac{\frac{m}{s^2} * 0.003\frac{1}{K} * (99.9 - 18.7)K * (0.01m)^3}{(1.89 * 10^{-5}\frac{m^2}{s})^2} = 6.69 * 10^3$$

Rayleigh Number

$$Ra = Gr * Pr \quad \text{Equation 9}$$

$$Ra = 6.69 * 10^3 * 0.709 = 4743.17$$

Nusselt Number

$$Nu = C * Ra^n \quad \text{Equation 10}$$

$$Nu = 0.85 * 4743.17^{.188} = 4.17369$$

Convective Heat Transfer Coefficient

$$h_c = \frac{Nu * k}{D} \quad \text{Equation 11}$$

$$h_c = \frac{4.17369 * 0.0285 \frac{W}{m * K}}{0.01\ m} = 11.895\ \frac{W}{m^2 K}$$

Heat Loss from Convection

$$Q_C = h_C * A *\ \Delta T \quad \text{Equation 12}$$

$$Q_C = 11.895 \frac{W}{m^2 * K}\ * 0.0022 m^2 * (99.9 - 18.7) K = 2.12492\ W$$

<u>Fraction of heat loss</u>

Fraction of Heat Loss from Radiation

$$\frac{Q_R}{Q_R + Q_C} = \frac{1.4334}{1.4334 + 2.12492} = 0.403 \quad \text{Equation 13}$$

Fraction of Heat Loss from Convection

$$\frac{Q_R}{Q_R + Q_C} = \frac{2.12492}{1.4334 + 2.12492} = 0.597 \quad \text{Equation 14}$$

*Convective and Radiant Heat Transfer Technical Objective 2*

For the second technical objective of the forced convection experiment, the heat loss from radiation was found using,

$$Q_R = h_R * A * \Delta T.$$

With the surface area given, $h_R$ and $\Delta T$ had to be found. The temperature difference was found using,

$$\Delta T = T2_C - T1_C,$$

$$\Delta T = 368.3℃ - 22.9℃ = 345.4℃ \; or \; K.$$

To find $h_R$, the following equation was used:

$$h_R = \beta * F * \Omega * (T2_C{}^4 - T1_C{}^4)/(T2_C - T1_C),$$

$$h_R = (0.95) * (1.0) * \left(5.67 * 10^{-8} \frac{W}{m^2 K^4}\right) * \frac{(641.3\ K)^4 - (295.9\ K)^4}{641.3\ K - 295.9\ K} = 25.182 \frac{W}{m^2 K}.$$

Once these values were calculated, the radiant heat loss could then be calculated by,

$$Q_R = (25.182\ \frac{W}{m^2 K}) * (0.0022\ m^2) * (345.4\ K) = 19.135\ W.$$

The heat loss due to forced convection was found using,

$$Q_C = h_C * A * \Delta T.$$

The temperature difference and area were the same as in the analysis for the radiant heat loss. However, finding $h_C$ required several steps. First, the thermal conductivity, Prandtl Number, and kinematic viscosity of air were all found by interpolating values from the engineering toolbox [4]. They were interpolated at the film temperature, where,

$$T_f = (T1_C + T2_C)/2, \qquad \text{Equation 15}$$

$$T_f = \frac{641.3\ K + 295.9\ K}{2} = 468.6\ K.$$

Then, the Reynolds Number was calculated using,

$$Re = \frac{u * D}{v}, \qquad \text{Equation 16}$$

$$Re = \frac{(1.22*0.54\frac{m}{s})*(0.01\ m)}{(3.376146*10^{-5}\frac{m^2}{s})} = 193.970.$$

This number was also used to describe the flow regime. Below 2,500 was considered to be laminar flow, and above 4,500 was considered to be turbulent flow. Any number in between was considered to be transitional flow. In this case, the flow was laminar. Then, the Reynolds Number and the Prandtl Number were used to find the Nusselt Number using,

$$Nu_D = 0.3 + \frac{0.62*Re_D^{\frac{1}{2}}*Pr^{1/3}}{[1+(\frac{0.4}{\mathrm{Pr}})^{\frac{2}{3}}]^{1/4}} * [1 + (\frac{Re_D}{282{,}000})^{5/8}]^{4/5}, \qquad \text{Equation 17}$$

$$Nu_D = 0.3 + \frac{0.62*(193.970)^{\frac{1}{2}}*(0.683)^{1/3}}{[1+(\frac{0.4}{0.683})^{\frac{2}{3}}]^{1/4}} * \left[1 + \left(\frac{193.970}{282{,}000}\right)^{\frac{5}{8}}\right]^{\frac{4}{5}} = 7.01393.$$

The Nusselt Number was then used to find $h_C$ using,

$$h_C = \frac{Nu*k}{D},$$

$$h_C = \frac{(7.01393)*\left(0.3822\frac{W}{m*K}\right)}{(0.01\ m)} = 26.885\frac{W}{m^2*K}.$$

Finally, the heat loss due to forced convection could be found using,

$$Q_C = \left(26.885\ \frac{W}{m^2K}\right) * (0.0022\ m^2) * (345.4\ K) = 20.430\ W.$$

*Excel Plots for Calculations*

Boiling Heat Transfer Technical Objective 1

Table 1. Data Collected for Boiling Heat Transfer TO. 1

| Time (min) | Flow (g/s) | Voltage (watt: | Pressure (bar) | T1 (°C) | T2 (°C) | T3 (°C) | T4 (°C) | T5 (°C) |
|---|---|---|---|---|---|---|---|---|
| 0 | 12 | 197 | -15 | 46.3 | 30.2 | 29.8 | 22.5 | 25.9 |
| 5 | 12 | 190 | -12 | 46.2 | 30.8 | 30.3 | 22.9 | 26.3 |
| 10 | 12 | 209 | -11 | 48 | 31.6 | 31.2 | 23.2 | 27 |
| **15** | **12** | **200** | **-10** | **47.6** | **31.8** | **31.5** | **23.6** | **27.2** |
| 20 | 10 | 193 | -8 | 47.8 | 32.5 | 32.2 | 23.6 | 28.2 |
| 25 | 10 | 199 | -8 | 48.3 | 32.6 | 32.4 | 23.7 | 28.3 |
| **30** | **10** | **204** | **-8** | **48.8** | **32.8** | **32.7** | **23.8** | **28.4** |
| 35 | 8 | 205 | -2 | 49.1 | 33.6 | 33.4 | 23.8 | 29.6 |
| 40 | 8 | 194 | -2 | 48.6 | 33.5 | 33.2 | 23.8 | 29.5 |
| **45** | **8** | **205** | **-2** | **50** | **33.7** | **33.5** | **23.8** | **29.7** |
| 50 | 6 | 212 | 0 | 51.2 | 35.2 | 35.1 | 23.7 | 31.7 |
| 55 | 6 | 203 | 0 | 50.5 | 35 | 34.9 | 23.7 | 31.5 |
| **60** | **6** | **203** | **0** | **50.5** | **35.1** | **35** | **23.7** | **31.5** |
| 65 | 4 | 201 | 4 | 52.8 | 38 | 37.9 | 23.5 | 34.6 |
| **70** | **4** | **198** | **4** | **53.2** | **38.3** | **38.3** | **23.5** | **35** |
| 75 | 2 | 203 | 40 | 59.7 | 45.4 | 45.4 | 23.5 | 41.2 |
| 80 | 2 | 200 | 50 | 61.5 | 47.5 | 47.5 | 23.7 | 43.7 |
| **85** | **2** | **201** | **51** | **62.3** | **48.3** | **48.3** | **23.8** | **44.8** |

Boiling Heat Transfer Technical Objective 2

Table 2. Data Collected for Boiling Heat Transfer TO. 2

| | | | | T1 | T2 | T3 | T4 | T5 | Chamber pressure | Flow rate | heater power |
|---|---|---|---|---|---|---|---|---|---|---|---|
| T1 | surface | | | 44 | 34 | 32.6 | 20.5 | 29.8 | -5 | 1 | 53 |
| T2 | liquid | | | 50.1 | 38.6 | 38.3 | 20.8 | 34 | 10 | 1.5 | 98 |
| T3 | vapor | | | 50.7 | 38 | 37.6 | 20.6 | 34.2 | 10 | 3 | 157 |
| T4 | H2O in | | | 53.1 | 38.4 | 38.3 | 20.7 | 34.7 | 10 | 3.5 | 201 |
| T5 | H2O out | | | 57.8 | 38.3 | 38.2 | 20.6 | 34.4 | 10 | 4.5 | 248 |
| | | | film | 154.2 | 29.2 | | | | -20 | 4.5 | 74 |
| Pressure | kN/m^2 | | | 157.6 | 28.9 | | | | -20 | 4.5 | 67 |
| | | | | 150.3 | 28.9 | | | | -20 | 4.5 | 50 |

Table 3. Calculation for Heat Flux and Heat Transfer Coefficient

| 1 Heat flux =Q/A | | 2 heat transfer coefficient | | 3 Plot log Q/A vs T1-T2 |
|---|---|---|---|---|
| | | T1-T2 | h | log(Q/A) |
| 29444.4 | | 10 | 2944.444444 | 4.469 |
| 54444.4 | | 11.5 | 4734.299517 | 4.73595 |
| 87222.2 | | 12.7 | 6867.891514 | 4.94063 |
| 111667 | | 14.7 | 7596.371882 | 5.04792 |
| 137778 | | 19.5 | 7065.527066 | 5.13918 |
| 41111.1 | | 125 | 328.8888889 | 4.61396 |
| 37222.2 | | 128.7 | 289.2169559 | 4.5708 |
| 27777.8 | | 121.4 | 228.8120081 | 4.4437 |

Convective and Radiant Heat Transfer Technical Objective 1

Table 4. Heat Loss from Radiation

| T1 (air) | T2 (surface | Amps | volts | T2-T1 | hr | Qr |
|---|---|---|---|---|---|---|
| 291.7 | 372.9 | 0.073 | 50 | 81.2 | 8.024034047 | 1.433413442 |
| 292.1 | 473.6 | 0.127 | 85 | 181.5 | 12.77007459 | 5.099090783 |
| 292.5 | 572.8 | 0.18 | 116 | 280.3 | 19.28025575 | 11.88936251 |
| 292.9 | 673.7 | 0.238 | 148 | 380.8 | 28.09799779 | 23.53937863 |

| Heat loss from Radiation Q=hr*A*(Ts-Ta) | |
|---|---|
| area | 0.0022 |
| β | 0.95 |
| F | 1 |
| Ω | 5.67E-08 |
| hr = β*F*Ω*(Ts^4-Ta^4)/(Ts-Ta) | |

Table 5. Heat Loss from Convection

| T1(air) | T2 (surface | Amps | volts | T2-T1 | Tf = (Ta+Ts)/ | k (T=Tf) (W/m K | v (T=Tf) m^2/s | B (T=Tf) 1/K | Pr (T=Tf) | Gr | Ra | Nu | hc | Qc |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 18.7 | 99.9 | 0.073 | 50 | 81.2 | 59.3 | 0.0285 | 1.89E-05 | 3.00E-03 | 0.709 | 6.69E+03 | 4743.172487 | 4.1737 | 11.895 | 2.12492 |
| 19.1 | 200.6 | 0.127 | 85 | 181.5 | 109.85 | 0.0321 | 0.000024145 | 0.00262 | 0.7015 | 7986.62 | 5602.612499 | 4.3064 | 13.824 | 5.51977 |
| 19.5 | 299.8 | 0.18 | 116 | 280.3 | 159.65 | 0.0358 | 2.39E-05 | 2.32E-03 | 0.69 | 7159.64 | 4940.154255 | 4.2057 | 15.057 | 9.28477 |
| 19.9 | 400.7 | 0.238 | 148 | 380.8 | 210.3 | 0.0414 | 0.000039862 | 0.00195 | 0.681 | 4584.4 | 3121.975573 | 3.8581 | 15.972 | 13.3811 |

| Heat loss from Convection Qc=hc*A*(Ts-Ta) | |
|---|---|
| area | 0.0022 |
| diameter | 0.01 |
| C | 0.85 |
| n | 1.88E-01 |
| g | 9.81 |

Table 6. Fraction of Heat Loss

| Fraction of Heat Loss due to radiation and natural convection | | | |
|---|---|---|---|
| Volts | Qr+Qc | Qr/(Qr+Qc) | Qc/(Qr+Qc) |
| 50 | 3.55833769 | 0.4028323 | 0.5971677 |
| 85 | 10.6188572 | 0.480192 | 0.519808 |
| 116 | 21.1741282 | 0.5615042 | 0.4384958 |
| 148 | 36.920473 | 0.6375698 | 0.3624302 |

Convective and Radiant Heat Transfer Technical Objective 2

Table 7. Data Collected for Convective and Radiant Heat Transfer TO. 2

| Air Velocity (m/s) | T1 (°C) | T2 (°C) | Electric Current (A) | Voltage Supply (V) |
|---|---|---|---|---|
| 0.54 | 22.9 | 368.3 | 0.33 | 201 |
| 0.91 | 23.8 | 355.2 | 0.328 | 200 |
| 1.4 | 24.3 | 333.8 | 0.328 | 200 |
| 2 | 24.6 | 319 | 0.325 | 197 |
| 2.52 | 24.5 | 309.4 | 0.327 | 199 |
| 2.95 | 24.5 | 298.3 | 0.324 | 197 |
| 3.34 | 24.5 | 290.4 | 0.331 | 200 |
| 3.99 | 24.2 | 283.9 | 0.334 | 201 |
| 4.55 | 24.2 | 273.3 | 0.331 | 201 |
| 4.98 | 24.2 | 265 | 0.333 | 201 |
| 5.45 | 24 | 247.2 | 0.328 | 199 |
| 5.87 | 23.5 | 236.4 | 0.332 | 200 |
| 6.4 | 23.6 | 229.1 | 0.329 | 199 |
| 6.9 | 23.6 | 226.9 | 0.332 | 200 |
| 7.45 | 23.5 | 224.7 | 0.333 | 202 |
| 8.13 | 23.4 | 221.9 | 0.331 | 200 |
| 8.51 | 23.2 | 225 | 0.334 | 202 |
| 8.95 | 23.2 | 222.8 | 0.333 | 201 |

Boiling Heat Transfer Technical Objective 2 Trial 2

Table 8. Data Collected for Boiling Heat Transfer TO. 2 Trial 2

| | | | | T1 | T2 | T3 | T4 | T5 | Chamber pressure | Flow rate | heater power |
|---|---|---|---|---|---|---|---|---|---|---|---|
| T1 | surface | | 50 | 42.6 | 32.3 | 29.2 | 19.3 | 26.4 | -10 | 1 | 53 |
| T2 | liquid | | 100 | 49.6 | 38 | 37.3 | 19.6 | 32.7 | 9.5 | 1.5 | 101 |
| T3 | vapor | | 150 | 51.3 | 38.4 | 37.7 | 19.7 | 33.7 | 10.5 | 2.5 | 151 |
| T4 | H2O in | | 200 | 52.8 | 38.3 | 37.9 | 19.8 | 33.8 | 10.5 | 3.5 | 195 |
| T5 | H2O out | | 250 | 56.8 | 38.3 | 38 | 19.8 | 33.7 | 10.5 | 4.5 | 245 |
| | | | film | 117.3 | 29 | 25 | 18.7 | 23.3 | -20 | 4.5 | 50 |
| Pressure | kN/m^2 | | | | | | | | | | |

Pictures of Boiling Heat Transfer, TO. 2

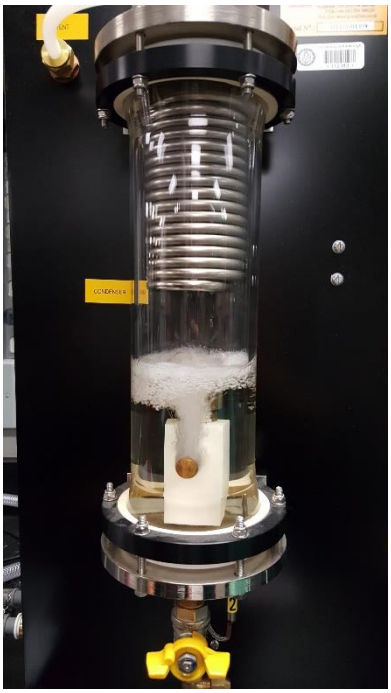


**Fig. 15**. 50 Watts

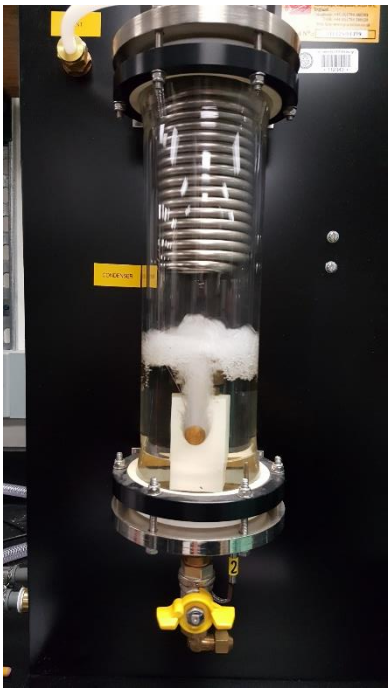


**Fig. 16.** 100 Watts

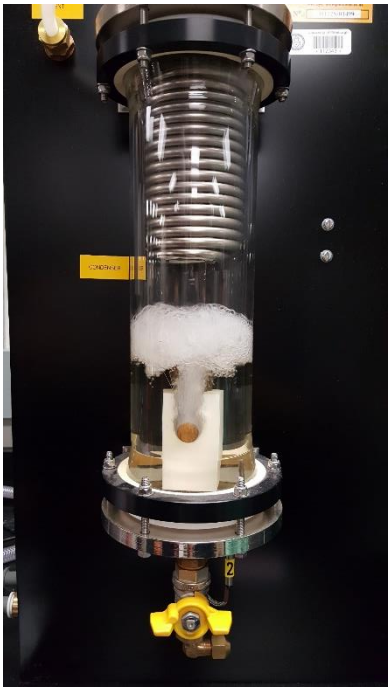


**Fig. 17.** 150 Watts

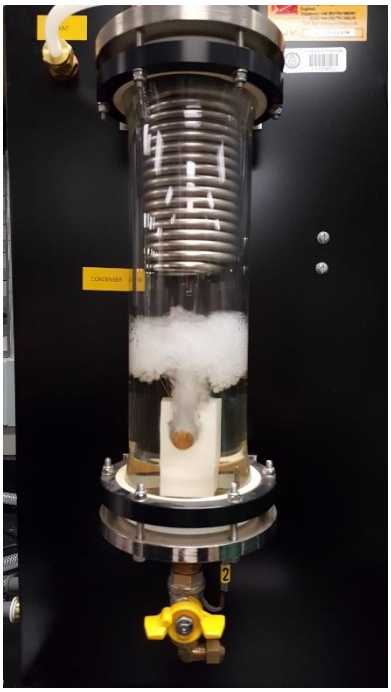


**Fig. 18.** 200 Watts

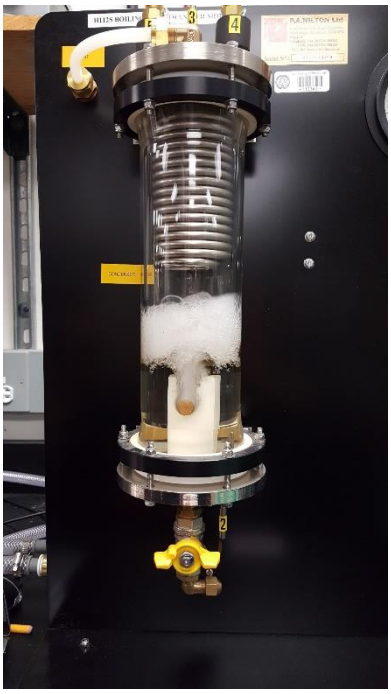


**Fig. 19.** 250 Watts



**Fig. 20.** Film Boiling

**Appendix II**

The results and analysis below contains information gathered from the first trial of the second technical objective of the convective and radiant heat transfer experiment as well as the second trial of the second technical objective for the boiling heat transfer experiment.

Results

The second technical objective for boiling heat transfer was repeated. The following data points were once again recorded: power, chamber pressure, $T_1$-$T_5$, condenser water flow rate, and heater power input. Similar results were found when plotting the log of the heat flux versus the difference in temperature and can be observed in Figure 21.

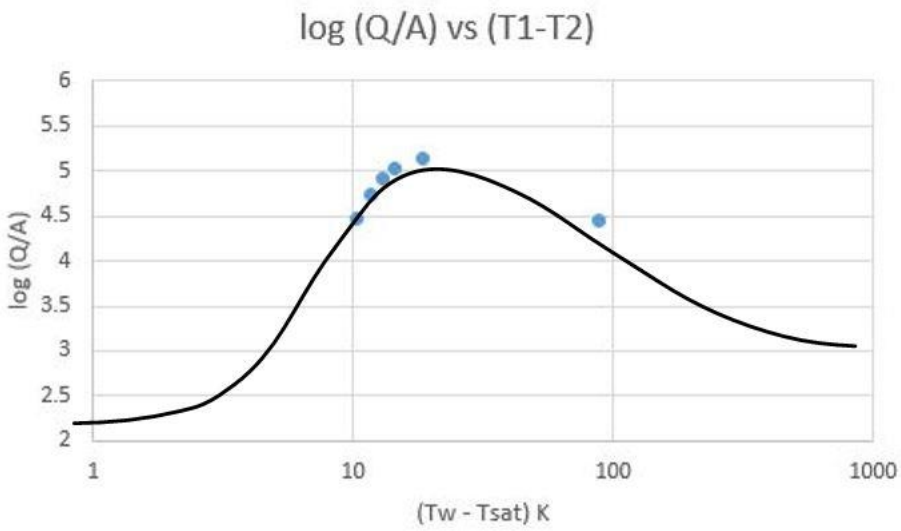



**Fig. 21.** Log of the Heat Flux vs. Difference in Temperature of the Surface and the Liquid

The second technical objective for the Convective and Radiant Heat Transfer lab was to compare the heat loss from forced convection to the heat loss from radiation.

To find the heat loss due to radiation, the radiant heat transfer coefficient, the surface area of the heater, and the difference between the heated surface temperature and the air temperature were

calculated. This was done for each effective air velocity that was measured. Figure 22 contains the plot of the radiant heat loss against the effective air velocity.

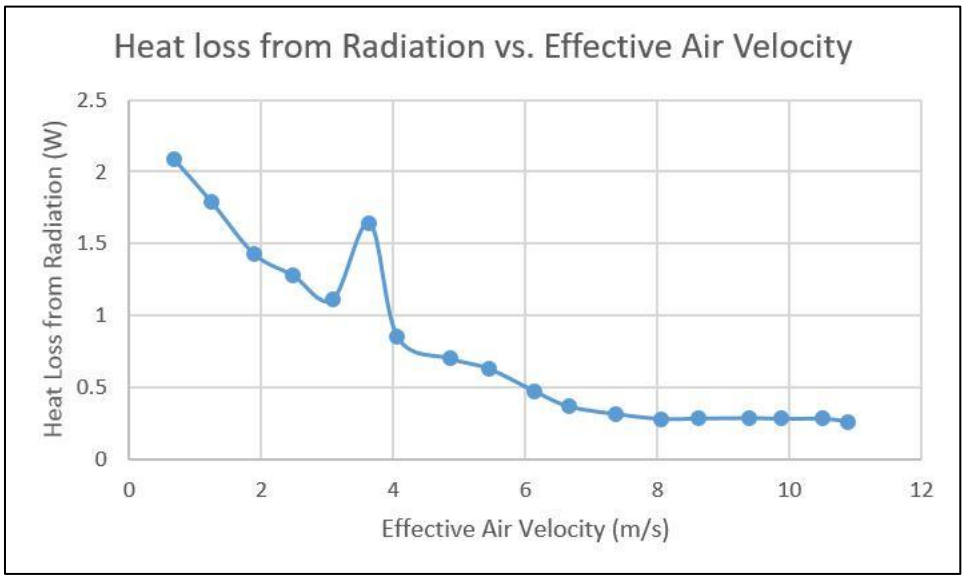



**Fig. 22.** Heat Loss from Radiation vs. Effective Air Velocity

The heat loss due to forced convection was found using the convective heat transfer coefficient ($h_c$), the surface area of the heater, and the temperature difference between the heated surface and the flowing air. To find $h_c$, the Nusselt Number needed to be found using the Prandtl Number and Reynolds Number. The thermal conductivity, Prandtl Number, and kinematic viscosity were found using the engineering toolbox and interpolating values of each respective measurement for air at different temperatures [4]. Figure 23 displays the results of this analysis.

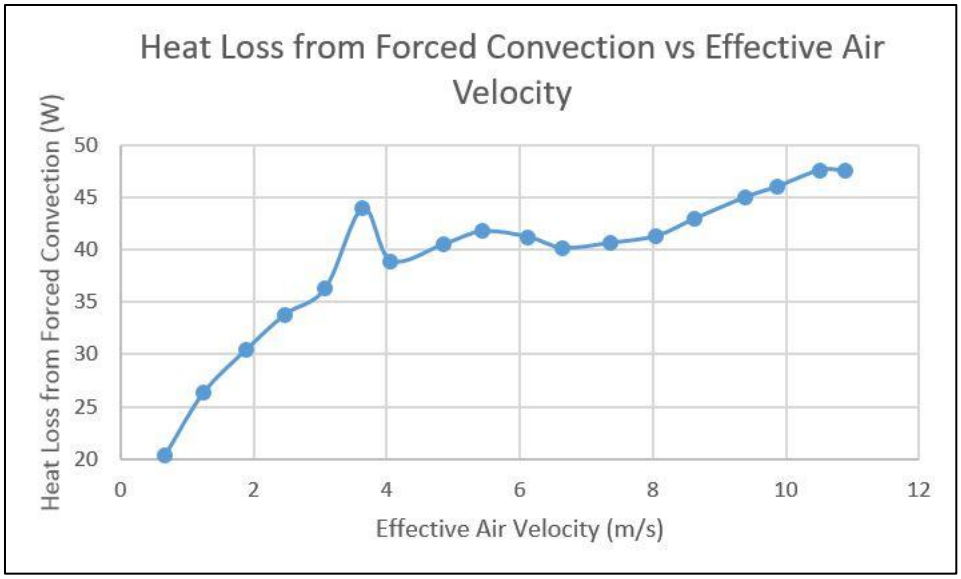



**Fig. 23.** Heat Loss from Forced Convection vs. Effective Air Velocity

In addition, the surface temperature of the top of the column was plotted against the effective air velocity for each reading. Figure 24 shows the trend.

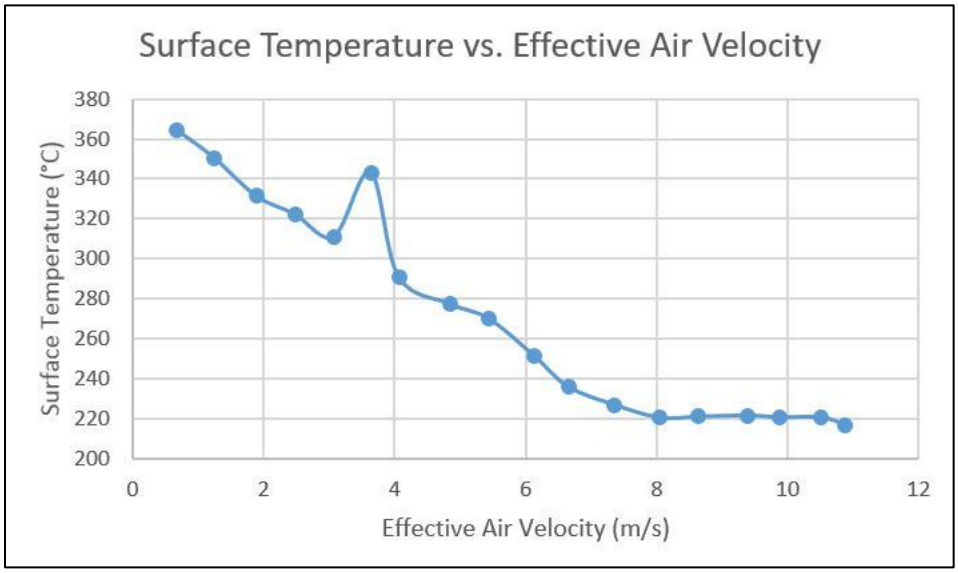



**Fig. 24.** $T2_c$ vs. Effective Air Velocity

The last part of the second technical objective was to analyze the flow regimes of the air for each trial. Figure 25 shows each Reynolds number plotted against each respective air velocity as well as the cutoffs for each flow regime.

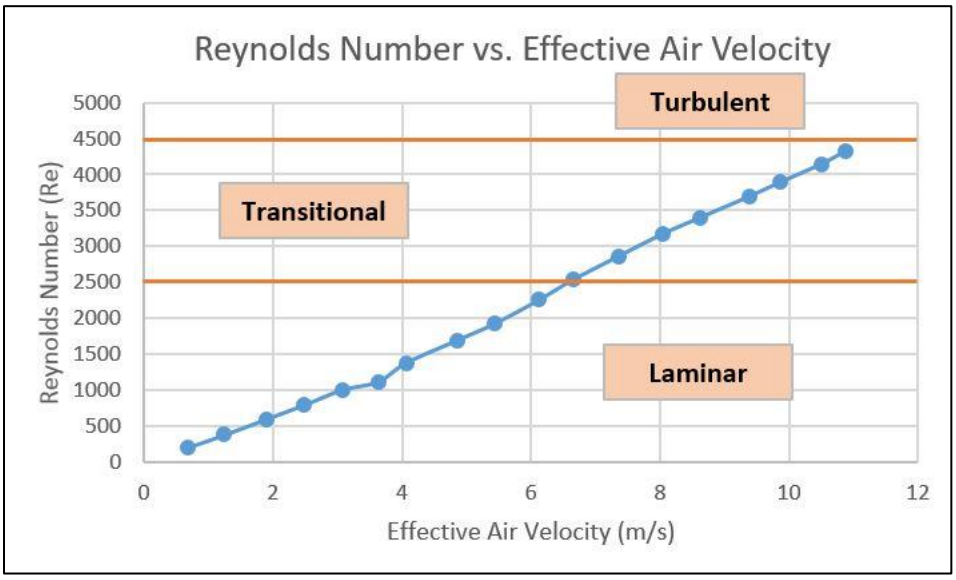



**Fig. 25.** Reynolds Number vs. Effective Air Velocity

## 4.0 Analysis and Discussion

For the second technical objective of the Convective and Radiant Heat Transfer experiment, there was a spike in the surface temperature when the effective air velocity reached about 3.6 m/s. For every other air velocity, the surface temperature at the top of the column would decrease as the air velocity increased. This deviation could be the result of several things. The two most likely suspects of this result are human error or a potential critical point in the experiment. Maybe there is a point in which the air velocity creates a maximum top surface temperature, resulting in a heightened loss of heat from both forced convection and radiation. However, it is probably more likely that there was an error in data collection.

Ignoring the spike in the data, the heat loss from radiation decreased as the effective air velocity increased, and the heat loss from forced convection increased as the effective air velocity increased. This makes sense because forced convection is usually caused by a velocity flow of a substance. In this case, air was flowing through the column at greater speeds, resulting in a greater contribution from forced convection heat loss.

The decrease in radiation heat loss is a result of the decrease in $T2_c$ as effective air velocity increased. Surfaces with lower temperatures do not radiate as much heat as surfaces with higher temperatures, so as $T2_c$ decreases, the radiative heat loss should as well, as Figures 22 and 24 show.

The air flow regimes for each trial were either in laminar or transitional flow, according to each calculated Reynolds Number. The same boundary conditions for flow were used for this experiment as the boundary conditions for the Boiling Heat Transfer experiment. A Reynolds Number below 2,500 was considered to be laminar flow, above 4,500 was considered to be turbulent flow, and between those two values was considered to be transitional flow. About half of the trials for this technical objective resulted in laminar flow, while the other half resulted in transitional flow. This is most likely due to the relatively high kinematic viscosity of air, which results in a lower Reynolds Number.

During transitional flow, each of the graphs started to level off towards an asymptote. From this information, one can assume that the surface temperature during high transitional/turbulent flow would be more resistant to change as effective air velocity increases. Therefore, increasing the Reynolds Number decreases the change in radiant and forced convective heat loss as effective air velocity increases. These asymptotes suggest that there is a minimum radiant heat loss (from a minimum $T2_c$) and a maximum forced convective heat loss as effective air velocity approaches infinity.