Design and Fabrication of an Internal Condensation Loop for Effectiveness and Robustness Testing of Nanostructured Superhydrophobic Steam Condenser Tubes

By

Dhananjai Saranadhi

Submitted to the Department of Mechanical Engineering in partial fulfillment of the

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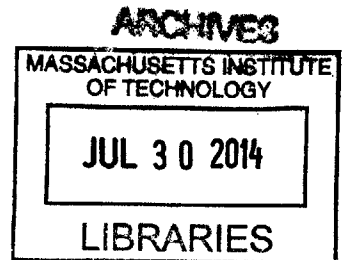
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Abstract

The Rankine cycle is at the heart of steam-electric power stations, which are responsible for generating about 90% of the world's electricity. Improving the efficiency of the cycle thus of great importance, and the greatest possible gain lies in improving the condensation process. Industrial condensers feature once-through water cooling, and the substantial amount of water they consume coupled with the increasing scarcity of freshwater supplies provides further motivation to focus on the condensation process. Condensation in these systems occurs predominantly *via* the filmwise mechanism, in which a thin film of water forms upon the condensing surface, adversely affecting its heat transfer abilities. However, forming a nanostructure and adding certain hydrophobic coatings on the heat exchanging surface of the condenser can render them superhydrophobic. This causes condensation to instead occur *via* the jumping droplet mechanism, which promises drastically improved heat exchanging performance. This thesis discusses the design and fabrication of an internal condensation loop which will allow us to test the heat transfer, fluid dynamic performance of the novel jumping droplet internal mode, and the durability and robustness of various hydrophobic coatings at the lab scale.

Acknowledgements

Most importantly, I must thank Jean Sack for bringing me on board this project, and for being my mentor on it. I am grateful to her for friendship, for sharing a great deal of knowledge with me, and for sparking my interest in superhydrophobic surfaces and ways in which they can be used to conserve energy in a number of applications. My Masters work will revolve around using such surfaces to reduce drag in water, and I have Jean to thank for introducing me to this fascinating research topic.

I also owe a debt of gratitude to Dion Antao and Dan Preston, who were my advisors on this project. Their wealth of experience allowed us to avert many a disaster, and ensure that the system was engineered to a high standard.

I would also like to thank Professor Evelyn Wang, the Principal Investigator on this project, for allowing me to use her lab and encouraging me in my work.

The last 4 years at MIT have been some of the best years of my life, and I am infinitely grateful for my family and friends for making them so.

Contents

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Chapter 1: Introduction

One of the defining topics of the 21st century has been the search for clean, efficient and renewable energy sources[1]. Natural gas, which is certainly better than coal by the above metrics[2] has become one of the top energy-producing fuels. Meanwhile, Europe continues to produce a large portion of its energy through nuclear and hydro power[3] and considerable money and resources are being investing in the development of solar-thermal power. Despite the apparently vast differences in the nature of these fuels, they share common ground in that their stored energy is converted to electricity. As about 85% of the world's electricity is currently generated by steam-electric power plants [4] it is unsurprising that a substantial amount of research and process engineering has been conducted in order to optimize the steam cycle taking place within these plants. One method of improving the condensor efficiency in a steam cycle is by inducing jumping droplet condensation, which has much higher heat transfer rates than observed in currently used filmwise or dropwise condensation.[5] This thesis proposes the design of a setup to test and optimize internal jumping condensation on superhydrophobic nanostructures, with the goal of implementation in industry.

1.1 The Rankine Cycle

The Rankine cycle[6] is a series of thermodynamic processes that convert heat into mechanical work. Using water as its working fluid, it is used in steam power plants, and is thus responsible for 90% of the world's electricity[7]. In its simplest form, the Rankine cycle consists of four primary processes:

(i) Compression: The working fluid begins in liquid form (water), and is pumped to high pressure. This process takes us from point 1 to point 2 on the diagram in Figure 1, below.

(ii) Heating: The water, now at high pressure, is heated at constant pressure in a boiler. Energy is required for this process, provided by an external source. The water boils to become saturated steam. This takes us from point 2 to 3 in Figure 1.

(iii) Expansion: The steam expands as it passes through a turbine. Energy is extracted from it as the turbine blades spin to produce mechanical work which is then converted into electricity. As the working fluid has lost energy, some of it condenses, and the fluid that exits the turbine is a vapor-liquid mixture (3-4)

(iv) Condensation: The working fluid is now fully condensed, and returns to being a saturated liquid, which is then returned to the compressor to continue the cycle (4-1).

Figure 1 on the next page is a Temperature-Entropy (T-S) diagram for the simple Rankine cycle.
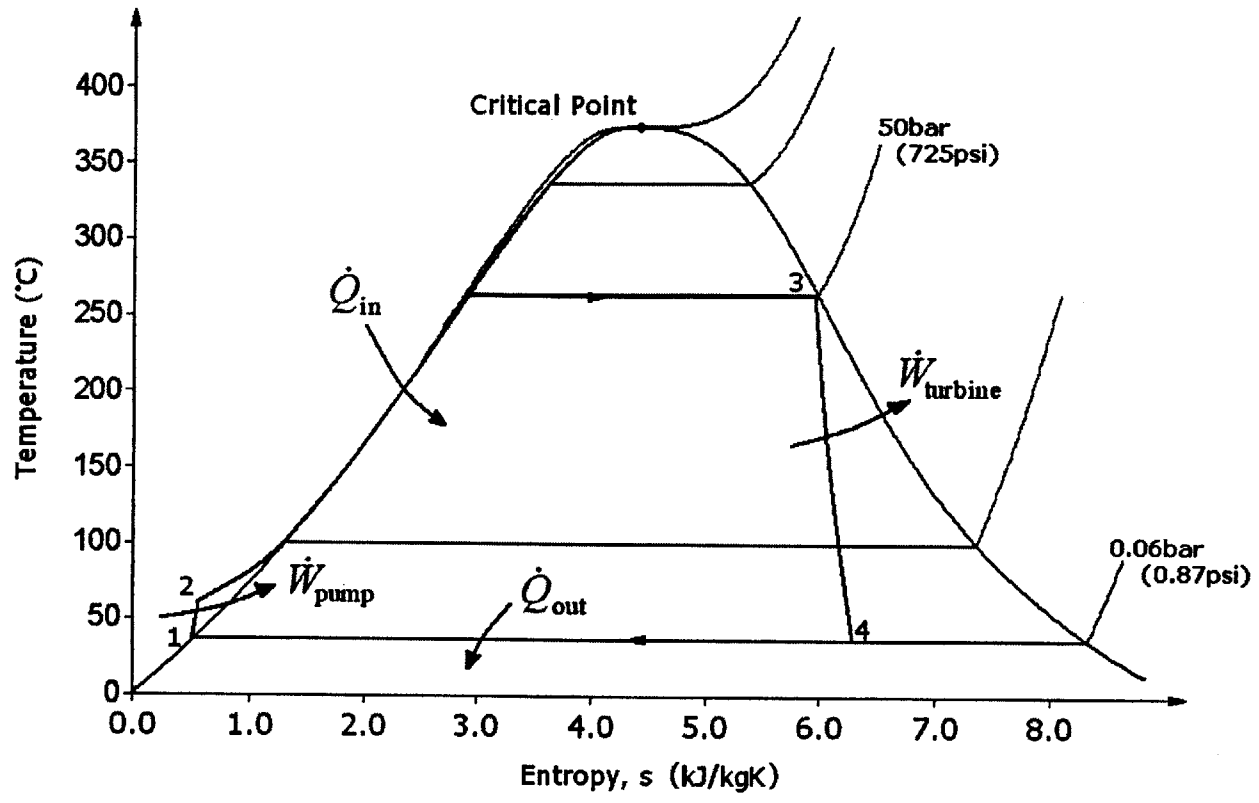
Figure 1: T-s Diagram of the Rankine Cycle[8]

Chapter 2: Condensation

While a great deal of time and money has been spent optimizing each step of the Rankine cycle since its inception in industry, this research effort focuses on the condensation process. The motivation for this is many-fold. Previous studies have indicated that the greatest net gain in efficiency within modern steam plants lies in focusing upon the condensation process[9] by decreasing the steam condensing temperatures and accordingly the turbine back-pressure, as well as the steam-side pressure drops [10,11]. Nearly every modern steam plant uses water cooling in its condensers, as it is substantially more effective than air-cooling. However, water-cooling uses a great deal of water, to the tune of up to 5000 gal/min[12]. With fresh water scarcity rapidly emerging as a vital issue in many regions, power plants employing this technology are proving to be unsustainable, as the recent series of plant shutdowns due to water shortages indicate.[13]

Condensation occurs when steam contacts the wall of the condenser, which acts as a heat exchanger. The condensation then occurs, and we will consider three possible condensation mechanisms: filmwise, dropwise and jumping droplet[14]. The former, as its name suggests, is characterized by the formation of a continuous film of liquid, in this case water, upon the heat-exchanging surface. This film then acts as a barrier to further heat transfer, causing the condensation process to become drastically more inefficient.[15] Dropwise condensation has displayed far superior heat transfer capabilities, but has proven difficult to achieve on industrial scales,[15] with the result that the majority of steam plants today rely on filmwise condensation.

Figure 2 below depicts filmwise condensation on the left, dropwise condensation in the center and jumping droplet condensation on the right.

Figure 2: Filmwise condensation (L) dropwise condensation (Center) and jumping droplet condensation (R) (courtesy of Nenad Miljkovic)

2.1 Jumping Droplet Condensation

Dropwise condensation occurs when the condensing surface is only partially wetted by the water. Where filmwise condensation sees the condensing surface get covered in a film of water, which drastically reduces its ability to foster further condensation, dropwise condensation instead features the formation of water droplets that gradually coalesce and then fall from the surface through the action of gravity.[15] For this to occur, the surface must by hydrophobic, which prevents the water from wetting to the surface.

If, instead, a superhydrophobic condensing surface is used, another mode of condensation may occurs, known as jumping droplet condensation. The minimization of surface energy (and subsequent conversation of excess energy to kinetic energy) leads to the condensing droplets being forcibly ejected from the condensing surface upon coalescence. This results in a smaller drop size distribution and corresponding lower conduction thermal resistance

through the drops at all times, drastically increasing the heat transfer properties, and hence promoting condensation. However, the condensing surfaces used in industry are almost inevitably made of metal, due to its prevalence as a large-scale construction material, as well as their good heat conductive properties. The issue with this is that water generally wets well to metals, resulting in filmwise (rather than dropwise or jumping droplet) condensation. As afore mentioned, jumping droplet condensation requires a superhydrophobic surface, and this required the condensing surface to have two characteristics: it must have 'inherent' superhydrophobicity, which is to say that its chemical makeup causes it to have very poor attraction to water molecules, and it must have a nano-scale roughness, which also prevents the wetting of water. This can be addressed by applying a nanostructures and superhydrophobic coatings which drastically reduces the surface energy, resulting in very high contact angles. Nanostructures considered in this thesis were copper oxide nanoblades and zinc oxide nanowires, with trichlorosilane, stearic acid, or polymer superhydrophobic monolayer coatings.

An inevitable problem with such coatings is that, over time, they may be removed from the metal. Large quantities of steam pass by them at high pressure on a yearly basis, and this can easily lead to the coating gradually wearing off. The ability of the coating to remain in place and maintain its effectiveness is referred to as its robustness, and is a major criterion in the selection of a coating for industrial applications.

Chapter 3: Internal Condensation Loop

Ultimately, the goal of this research initiative is to demonstrate the feasibility and superior performance of super-hydrophobic heat exchanging surfaces that promote jumping droplet condensation. In order to accomplish this, we must provide quantitative evidence in the form of experimental results that demonstrate improved heat transfer and durability. This requires a testing apparatus that enables us to compare the performance of various condensation surfaces and their corresponding mechanism.

The first step in this process is to define the performance metrics that we will use. As the purpose of this research initiative is to promote increased condensation through increased nucleation and droplet removal, the first and most obvious metric will be the measurement of the effectiveness of the condensation, and hence the overall effectiveness of the heat exchanger. This is most directly done by measuring the quality of the steam before and after condensation, since the greater the change in quality, the more successful the condensation process is.

In theory, if the steam is fully saturated, the condensation process is isothermal. However, since the steam is being condensed as it flows through a tube, as opposed as being allowed to coalesce in an infinitely large space, in reality, a pressure drop and a temperature change are inevitable. This pressure drop indicates how close we are to achieving the most efficient possible condensation, with 0 being the theoretical best case scenario. As mentioned in the introductory section, the greatest efficiency gains in the Rankine Cycle lie in decreasing the steam condensation temperatures and the steam-side pressure drop[9]. As such, the second

performance metric is the pressure drop across the condensing section (to be known hereafter as the test section), with lower values indicating superior performance. Initial estimates of the vapor flow inside the tubes during the proposed jumping droplet mechanism indicate to low pressure drops, and we will seek to verify this.

Next, we sought to define what variables we could modify in order to observe their effect upon the above performance metrics:

(i) The surface of the heat exchanger. This particular aspect of the project is targeted at demonstrating the superior performance of the jumping droplet mechanism *via* the use of super hydrophobic coatings, so it is imperative that the apparatus allow for the substitution of various surfaces, coatings and nanostructures to measure and compare their effect upon condensation.

(ii) The geometry and orientation of the heat exchanger. In order to simulate current industry standards, the heat exchanger will consist of a section of tube with a nanostructure and coating applied to the inner surface. The external surface will be cooled, resulting in condensation of the steam travelling through the tube. To characterize the performance of the heat exchanger, flexibility of the diameter, length and orientation of the test section is important.

(iii) The saturation pressure difference across the test section. This can be manipulated by controlling the temperature of the steam upstream of the test section, and by setting the

temperature of the cold reservoir into which the steal will eventually flow. This pressure

differential will ultimately be responsible for the motion of the steam.

It was determined that the ideal way to carry out the above measurements would be via an

internal condensation loop, a schematic of which is given below, in Figure 3.
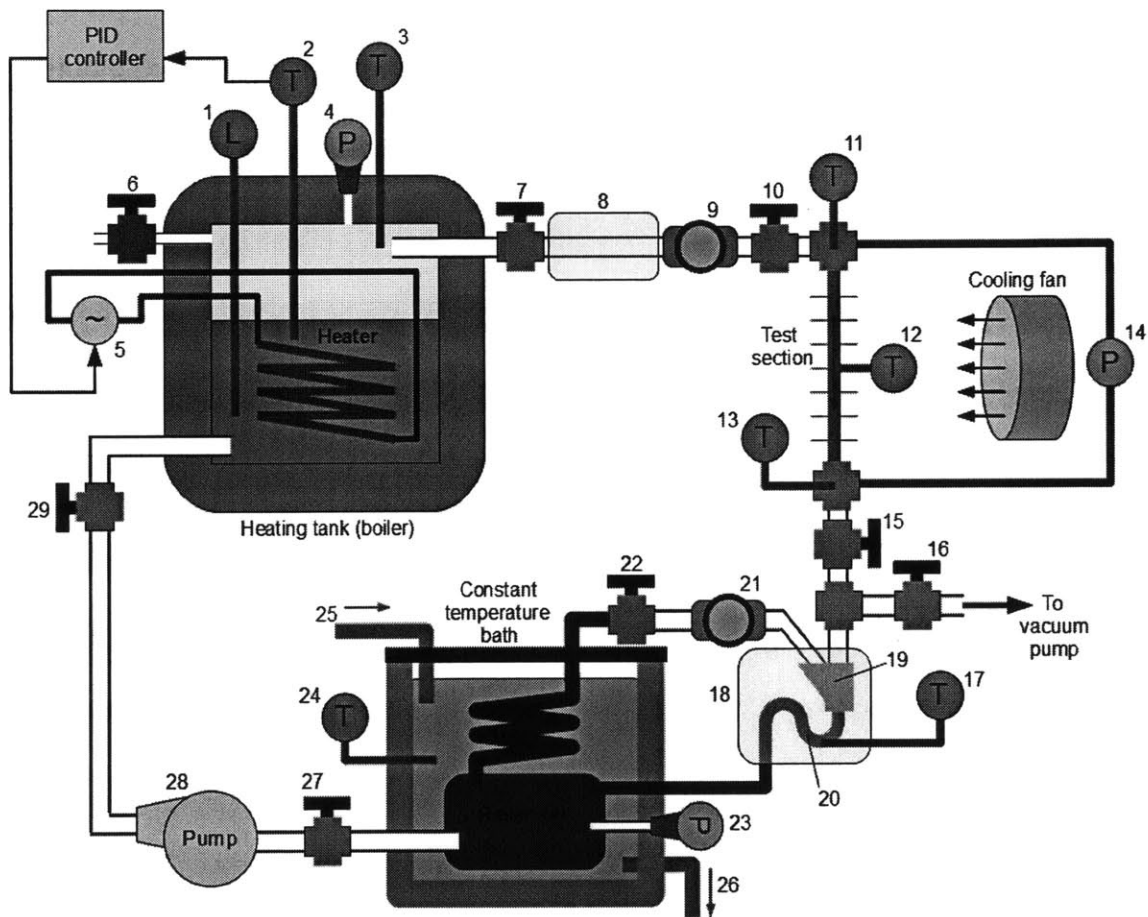
Figure 3: Schematic of internal condensation loop (Image courtesy of Dian Preston and Jiansheng Feng)

The water begins in liquid phase in the boiler, where it is converted to dry, saturated steam.

It proceeds through valve 7, and through a mass flow meter (9), and its temperature is

measured prior to entering the test section. It is cooled as it passes through the test section using an external condenser (depicted as a cooling fan), causing some of the steam to condense into water, using the walls of the test section as a heat exchanging surface. The mixture of steam and water then passes through a separator (19). The steam part of the mixture passes through a second mass flow meter, and rejoins the water part in the cooling reservoir, which is maintained at a fixed temperature using an external chiller. The steam condenses in the cooler, and water is then pumped back from the cooler to the boiler to restart the cycle. The extent of the condensation is measured in two different ways:

(i) The difference in mass flow is given by mass flow meter 9, which measures the amount of saturated steam entering the test section, and flow meter 21, which measures how much of that steam remains after condensation, tells us how much water was condensed out.

(ii) The cooling that promotes condensation in the test section is carried out using a water-cooled heat exchanger (described in detail ahead). Measuring the increase in temperature of this water will tell us the heat flux created by the condensation. Using measurements of pressure before and after the condensation will then allow us to use steam tables to calculate the quality of the steam-water mixture after condensation has occurred.

It is important to note that the pump only serves to move water from the cooler back to the boiler; it does not drive the flow of steam through the test section. This is achieved by setting and maintaining a temperature differential between the boiler and cooler, which

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creates a pressure gradient. The magnitude of this differential can be modified in order to obtain different steam flow rates.

The system is designed to run in vacuum to eliminate non-condensable gases before introducing pure water vapor, and also to reduce the boiling point of the water, thereby reducing the energy input. As mentioned earlier, the temperature difference between the boiler and cooler means that there is a pressure gradient between the two. We call this the saturation pressure, and it is this pressure differential that will drive the flow of the steam from the boiler to the cooler. This gradient must be able to overcome a head loss associated with the internal roughness of our tubing, a pressure drop across the condenser (inevitable in a practical system), as well as a two-phase pressure drop, caused by the conversion of pure steam to a saturated mixture, and a pressure drop due to gravity if the fluid must flow against gravity. The sum of these pressure drops will be equal to the difference in saturation pressures of the boiler and cooler[16].

This is displayed below in equation form:

$$P_{saturation} = P_{friction} + P_{two-phase} + P_{gravity}$$

Thus, we have but to select one parameter, for instance the mass flow rate or the velocity, and the remaining parameters, such as boiler and cooler temperature (and hence the driving pressure differential), test section diameter and length are set. It is also more than likely that we mirror the conditions used in industry, but in a scaled down form, where we maintain certain dimensionless groups. Space constraints prevent us from maintaining length scales. However, the key dimensionless group to match industry in is the Reynolds number, defined as

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$$Re = \frac{4\,\dot{m}}{\pi\,\mu\,D}$$

The Reynolds number is the ratio of inertial to viscous effects, and thus is a characteristic of the fluid flow. In terms of the heat-transfer, the Nusselt number, a ratio of convective heat transfer to conductive heat transfer, must be kept constant. Since the Nusselt number is a function of the Reynolds number and the Prandtl number, which is defined as the ratio of the viscous diffusion rate to the thermal diffusion rate in our system. This proves to be dependent on only the temperature of the boiler, and is therefore constant. Therefore, keeping the Reynolds number constant ensures that flow and heat transfer both scale. Finally, while the pressure drops above all have different magnitudes as compared to those in industry, we can achieve scaling in this regard by maintaining the value of the Euler number, which gives us a non-dimensional way of considering the pressure-drop:

$$Eu = \frac{\Delta P}{(\frac{\dot{m}}{A_c}\,\rho)^2}$$

The exact values of the pressure drops need to be determined experimentally, but ff the Reynolds number is kept constant, and the tube is horizontal so that the flow does not have to work against gravity, then they scale as follows:

$$P_{friction} \sim \frac{L}{D^3}$$

$$P_{2\phi} \sim \frac{L}{D^2}$$

Thus, setting the value of the Reynolds number allows us to scale the pressure drops, which in turn will affect the mass flow rate, and hence the Reynolds number itself. The system can

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thus be solved fully, allowing us to select attainable geometry to achieve the testing parameters that we seek.

Based on these calculations, let us now examine the design and fabrication of each part of the system.
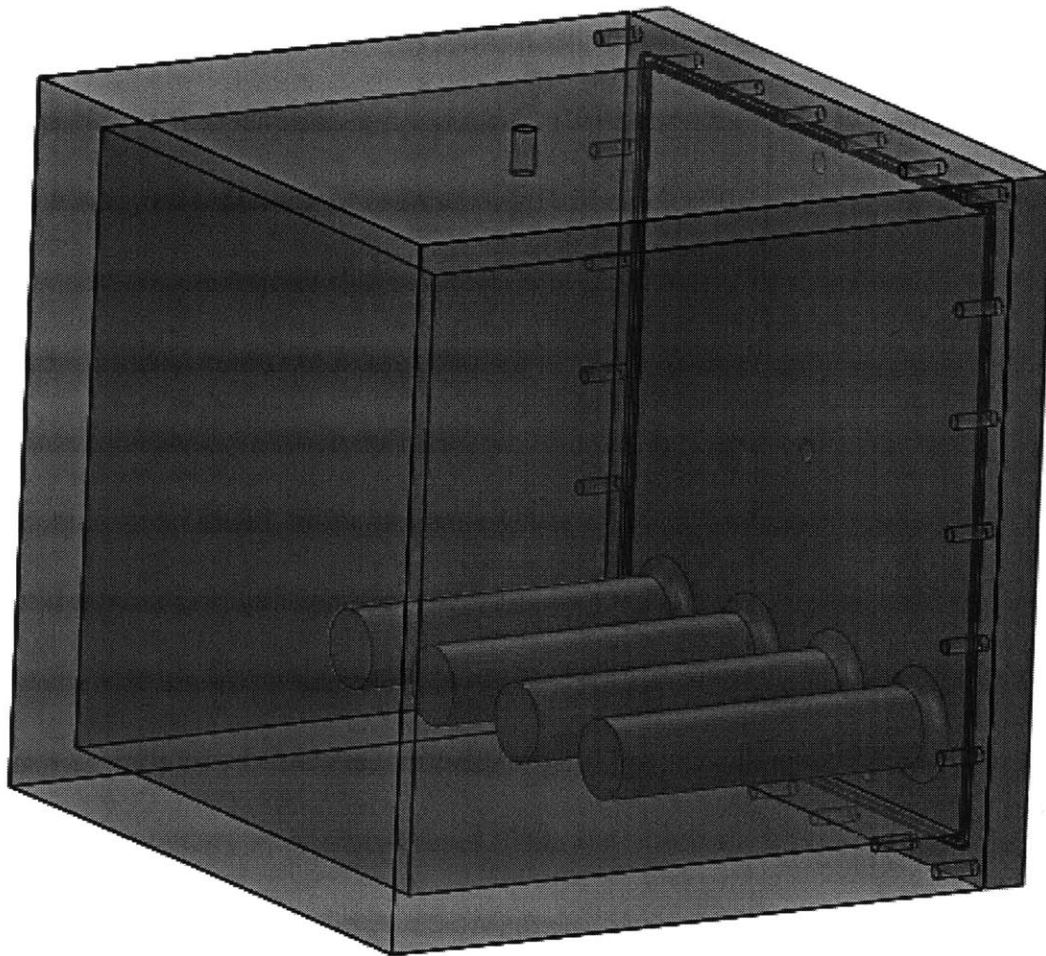
3.1 Boiler

The cycle begins in the boiler, where the water begins in liquid state. It is heated until it boils completely, producing a saturated vapor. The steam then leaves the boiler and moves towards the test section. In designing the boiler, there were several requirements had had to be met. Firstly, as with all parts of the system, it had to be completely resistant to corrosion leading to the selection of stainless steel. Since it would also have to be very carefully sealed and withstand vacuum, we selected 304 for its weldability and machinability. Using immersion heaters would allow for faster temperature control and response, but would create sealing issues, so one of the faces has holes with cylindrical insertions into which the cartridge heaters are inserted, as shown in figure 4, below.
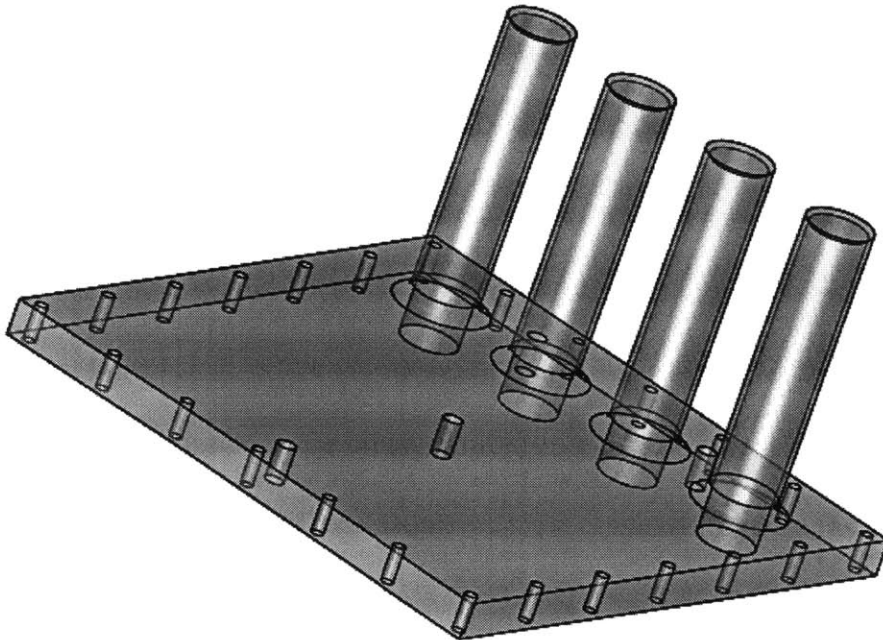
Figure 4: Removable face of boiler, showing inserts for cartridge heaters

Initially, we had decided to make the boiler a cube, with 6 of the faces vacuum welded together, and the 6th face, which also contained the inserts for the cartridge heaters (Figure 4) removable. The screw holes that we would use to bolt this 'heating face' to the rest of the boiler are shown in the figure above, along with holes for the insertions of the water-in line, and thermocouples to monitor the status of the water and the exiting steam. A Viton o-ring seal on the joining surface would ensure no leakage. The overall component was expected to look like in figure 5 below:

Figure 5: CAD rendering of early boiler design

The outer dimensions are an 8-inch side cube with 0.5" thick walls, to withstand the internal vacuum, and allow the screw holes to be tapped. The hole at the top provides an outlet for the steam. We decided to make the heating face on the side as it enabled the box to lie flat on a surface. This would not be possible if the heating face were on the bottom, as the leads from the heater would interfere. Furthermore, we feared the possibility of super heating the steam if the water level were to fall below the cartridge heaters, and locating the heaters in the lower part of a side face lowered the chances of this happening.

However, the design suffered from the fact the 4 walls had to be vacuum welded onto the base plate, leading to a long lead time and a high cost. Hollowing out an 8-inch side cube would require a tremendous amount of machining, and casting it out of stainless steel would be even more expensive. The solution we selected was to alter the geometry, making the boiler a cylinder with end caps as opposed to a cube. This involved purchasing a stainless tube with the required diameter, and have the end caps, which were square plates, welded on. This drastically reduced the amount of welding required. One of the end caps had a hole of the same diameter as the pipe in it, creating a continuous opening. We would bolt another plate on this, which allowed us to remove the top face of the chamber to clean and troubleshoot. The new geometry is shown in figure 6 below.
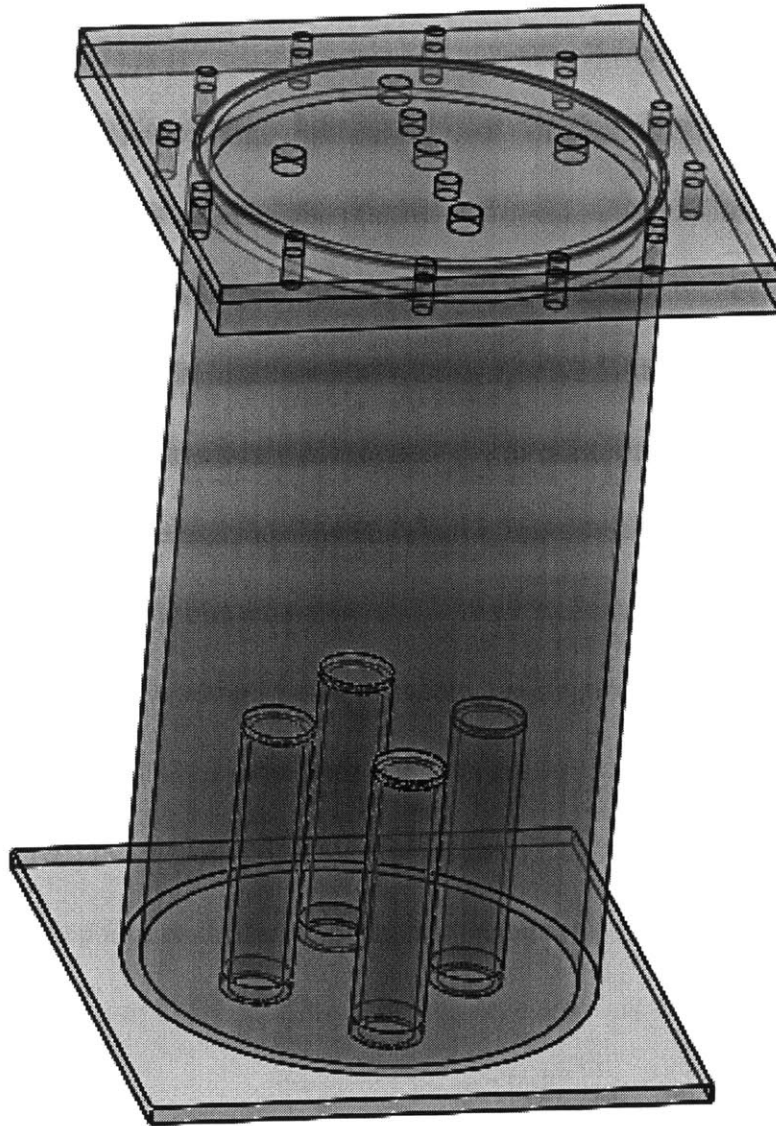
Figure 6: CAD rendering of final boiler design

The lack of corners allowed us to use thinner walls, which are 0.28" thick in the pipe section

that we selected. The pipe is a foot long with an internal diameter of 6.065", allowing for a

total internal volume of 347 in^3 or 5.68 L, slightly above the target of 5L. Once again, an o-

ring is inserted between the end-plate welded to the pipe and the lid piece that includes the holes for the gas outlet and for the thermocouples and pressure probes.

In order to ensure accurate results, the temperature and pressure of the steam that exits the boiler must be tightly regulated. It is important the steam should be dry, but not superheated. This is best achieved using a control loop. It takes as inputs the temperature and pressure of the steam measured directly outside the boiling chamber and adjust the power provide by the immersion heaters accordingly to result in steam of the pressure and temperature desired. The entire boiler is insulated using foam, so it is assumed that all the heat provided by the heaters goes towards heating and boiling the water. However, the heat must still travel through the stainless steam inserts and only then heat the water. We attempted to improve the heat transfer by using 4 cartridge heaters, thereby exposing a large heating surface area to the water, and reducing local heating, and by making the walls of the inserts very thing (0.01"). It would in fact have been idea to make the boiler out of Titanium to get better heat transfer properties, but that would have the material cost drastically higher, and made welding much more difficult.

The control loop that will be used to regulate the steam temperature is pictured schematically in figure 7 below:
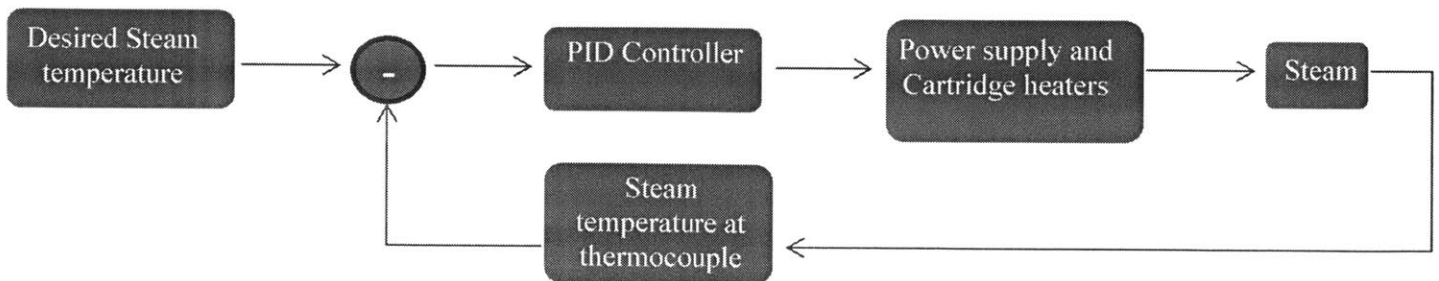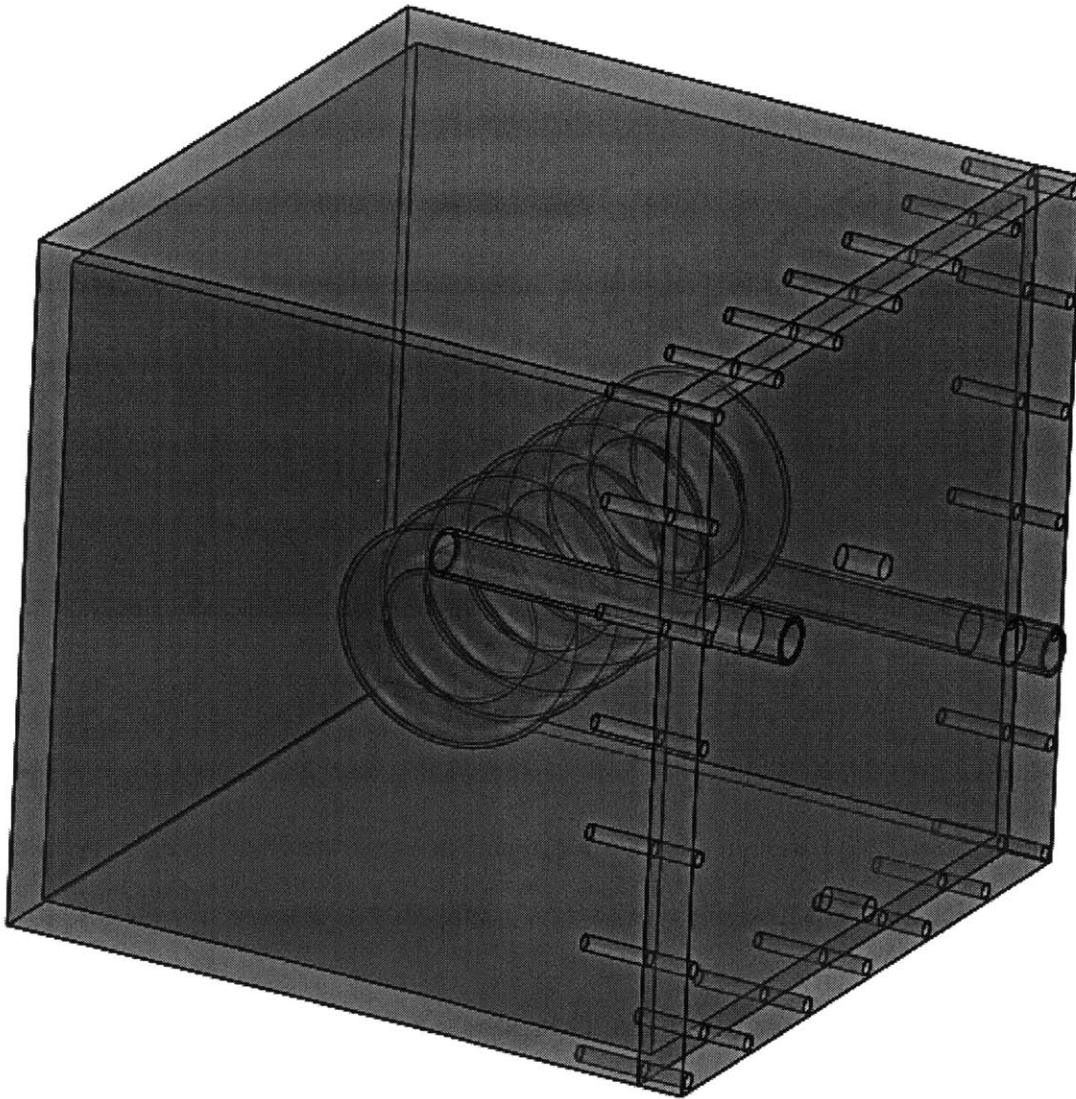

**Figure 7: Control loop for boiler**

3.2 Cooler

The function of the cooler is the mirror image of that of the boiler. As such, its dimensions are design closely mirror that of the boiler. The initial design for the cooler also featured a cube with 5 sides welded together, and the sixth one bolted on and removable.

Figure 8: CAD rendering of initial cooler design

The main differences between the cooler and boiler are mostly located on this removable sixth face. Instead of the slots for the cartridge heaters, the cooling face features entry points for a cooling loop. It is upon this surface that the steam will condense back into water after passing through the test section. Like the boiler, the cooler features an o-ring to seal the removable face. It is assumed that most of the steam will have condensed by the tie it reaches the cooler, and the remaining steam will follow suit once it contacts the cooling loop, so leakage is less of an issue. As we will still be dealing with vacuum conditions, the walls of the cooler as thick as those of the boiler, and it will also be made out of stainless steel in order to avoid corrosion. As with the boiler, it was found that the cost of manufacturing was prohibitively high, primarily due to the cost of vacuum welding. As such, the same revision as for the boiler was adopted, with the body of the coiler consisting of a stainless steel tube with flanges welded to its ends.

The cooling coil is made out of flexible copper tubing. The coil is looped into a helix in order to maximize the cooling surface area that can be exposed to the steam. The coolant used here is water, which comes from an external loop connected to a chiller. This allows us to control the temperature of the cooler, which is extremely important as the temperature differential between the boiler and the cooler is what determines the mass flow rate of the steam. As the cooling reservoir gradually fills, it has an outlet to the pump, which can either be run continuously or in bursts in regular intervals in order to move water back from the cooler to the boiler. Figure 9 on the next page depicts the final cooler design.
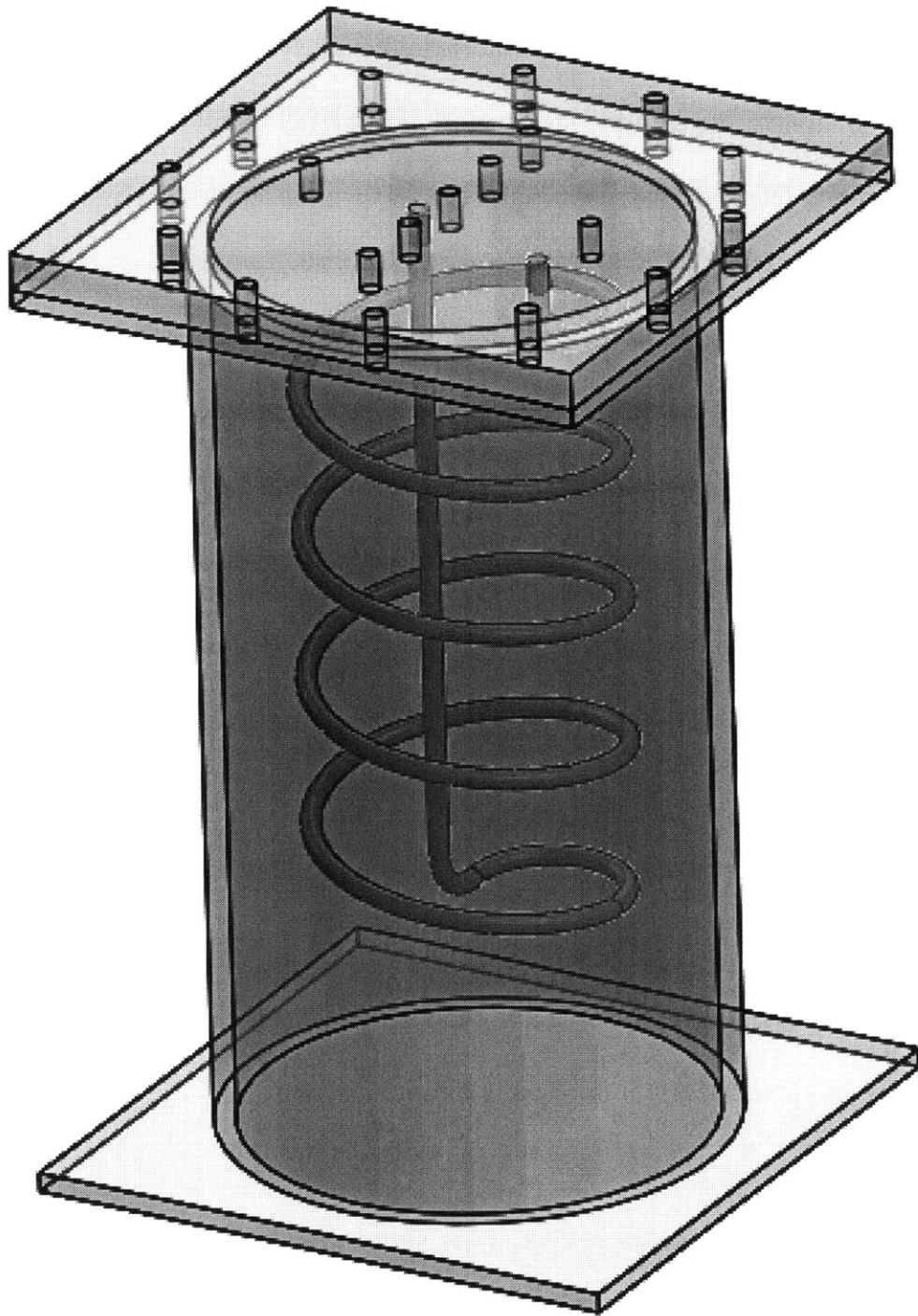
**Figure 9: CAD rendering of final cooler design**

3.3 Test Section Condenser

The condenser is the third part that was fabricated exclusively for use in this project. As with all bespoke parts, the design was carefully tailored to fulfill our requirements. Its purpose is simple: as steam passes through the test section, it must be induced to condense, which requires cooler. We considered several options to accomplish this. The schematic in Figure 3 illustrates the cooling mechanism as a simple fan. However, this would introduce variability and a dependence on environmental conditions into the system, so for initial testing we opted to select a method that allowed us a greater measure of control and repeatability over the cooling process. One possibility would have been to surround the test section using flexible tubing and pass cold water through the tubing. The drawback to this approach was the fact that the heat would have to pass through two surfaces (the thickness of the test section as well as the thickness of the cooling tubing), which made effective cooling difficult. This would make obtaining an accurate estimate of thermal resistance difficult, as we would require a model to account for the resistance due to the copper tube, cooper sleeve, solder, vacuum grease and the gap between the sleeve and test section. It would also require a large number of coils of cooling tube about the test section to ensure uniform cooling.

A solution was found that addresses all of the above drawbacks. The concept itself is a simple one: a removable tube that surrounds the test section. Water from external chiller flows in and out of this tube, in direct contact with the test section, carrying heat away from the steam. Measuring the temperature change in the cooling water also allows us to calculate the heat flux caused by the condensation. Using this information, along with

temperature and pressure readings of the steam before and after it passes through the test section, will allows us to calculate a value for the change in quality of the steam caused by the condensation process. The selection of the flow rate of cooling water is thus of vital importance. There is a tradeoff between a fast flow rate and better cooling versus a slower flow rate that allows for a clearly measurable change in temperature in the cooling water for heat flux and quality calculations.

A condenser can be treated as a special case of a heat exchanger. However, there are a few considerations that arise due to the fact that the steam condenses rather than changes heat. There are two methods of analyzing heat exchangers, and both of these will be considered. The first of them is known as the Log Mean Temperature Difference method, or LMTD. The temperatures of the cooling water and the condensing steam are shown in Figure 10:
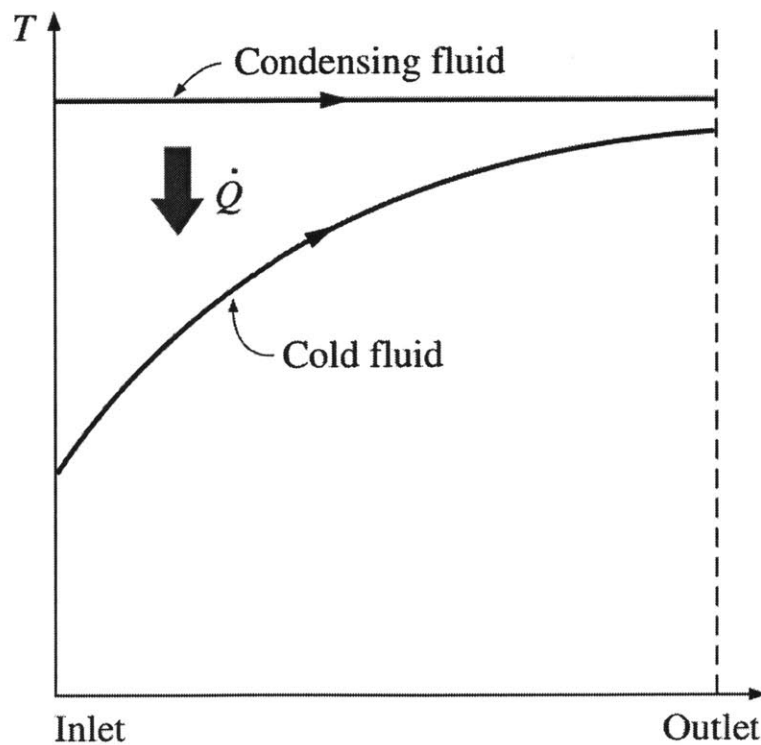

Figure 10: Temperature of cooling water and condensing steam[17]

Where the cooling water is the cold fluid, and the steam is the condensing fluid. This assumes that the condensation is isothermal, i.e. the temperature of the steam does not change. While this may be true in theory, in practice a change in temperature in such conditions are inevitable, and the effects of this will be discussed further ahead.

The log mean temperature is defined as

$$LMTD = \frac{\Delta T_a - \Delta T_b}{ln\left(\frac{\Delta T_a}{\Delta T_b}\right)}$$

The definitions of ΔT_a and ΔT_b depends on whether the cooling water and steam are in the parallel flow or counter-flow configurations, depicted in figure 11 below:

(a) Parallel flow   (b) Counter flow
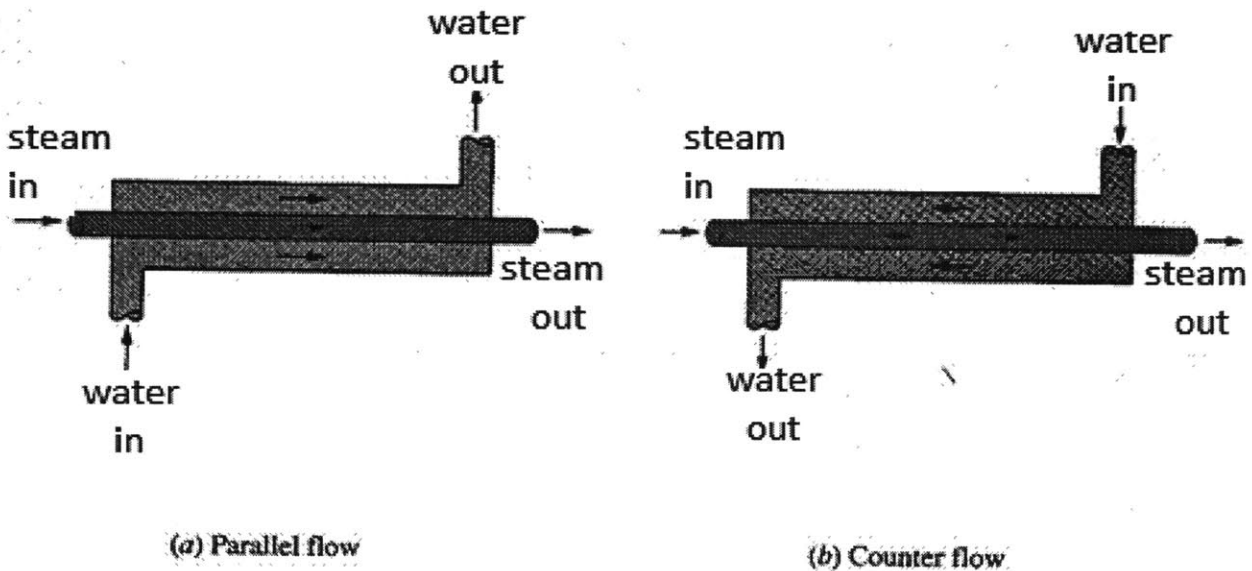
Figure 11: Parallel and counter flow in heat exchangers[18]

In similar operating conditions, the counter flow configuration offers superior heat exchange, so we will select that option. In this case, ΔT_a and ΔT_b are defined as follows:

$$\Delta T_a = \frac{T_{steam,in} - T_{water,out}}{T_{steam,out} - T_{water,in}}$$

Using this definition, the rate of heat transfer from the condensing steam to the cooling water is given by

$$\dot{Q}_1 = U * A * LMTD$$

Where U is the overall heat transfer coefficient. This has been estimated to be in the range of 1000-6000 W/m²K for condensers[19]. A is the surface area on which the heat transfer occurs. In our case, that is the outer surface area of the test section that is exposed to the cooling water, so that

$$A = \frac{\pi D^2 L}{4}$$

Where D is the diameter of the test section, and L is its length.

A second expression for the rate of heat transfer can be obtained we look at the cooling water alone:

$$\dot{Q}_2 = c_p \dot{m}_{water}(T_{water,out} - T_{water,in})$$

Similarly, a third expression of this kind can be obtained we focus exclusively upon the condensing steam:

$$\dot{Q}_3 = h_{fg} \dot{m}_{condensation}$$

All these three expressions for $\dot{Q}$ should be equal. Solving for $\dot{m}_{condensation}$ allows us to solve for the quality of the steam at the exit of the test section using the equation

$$x_{out} = \frac{\dot{m}_{steam} - \dot{m}_{condensation}}{\dot{m}_{steam}}$$

This assumes that the steam that enters the test section is completely dry (in our case, a valid assumption). These three expressions will now allow us to solve the system of equations by selecting values for some of the parameters. For instance, if the geometry of the test section is constrained by the space available for the setup, and we want to maintain similarity with industry standards, the mass flow rate of steam is selected for us. Similarly, we can select the value of the water as it enters the cooling loop. These equations will then allow us to solve for the temperature of the water as it leaves the cooling loop, and hence for the quality of the steam as it leaves the test section, as shown in the MATLAB code in Figure (12) on the following page.

```
c_latent = 2270*1000; %J/kg %latent heat of water

T_w_in = 15; %user defined
T_w_out = 16; %guess
T_s_in = 60; %user defined
T_s_out = 60; %assuming isothermal condensation
T1 = T_s_in - T_w_in;
T2 = T_s_out - T_w_out;
LMTD = (T1-T2)/(log(T1/T2));

d = 12/1000; l = 25/100; A = pi*d*l;
U = 1500; %Conservative estimate

Q1 = U*A*LMTD;
Q2 = C_water * (T_w_out-T_w_in);
Q3 = c_latent*m_dot_steam*(1-x_out);

n = 1;
while n<1000
    if abs(Q1-Q2)>10
        T_w_out = T_w_out +0.1;
        Q2 = C_water * (T_w_out-T_w_in);
        T1 = T_s_in - T_w_in;
        T2 = T_s_out - T_w_out;
        LMTD = (T1-T2)/(log(T1/T2));
        Q1 = U*A*LMTD;
        n = n+1;
    else
        break;
    end
end

disp(T_w_out);
m_dot_condensation = Q1/(c_latent);
x_out = (m_dot_steam-m_dot_condensed)/(m_dot_steam);
disp(x_out);
```

Figure 12: MATLAB code to calculate steam quality exiting the test section

As the code above shows, we first guess a value for $T_{water,out}$, and then use an iterative

process to find the value of $T_{water,out}$ that renders the two heat transfer rates equal. It should

be noted here that we assumed that the condensation process is isothermal. This is

theoretically true, but as mentioned before, in practice, a pressure drop and a temperature

change is inevitable. However, as long as this temperature change is small, then it can be

neglected, and we will assume that this is the case.

A second method of solving this condenser system is using the $\mathcal{E}$-NTU method, in which we

define[20]

$$NTU = \frac{UA}{c_{water} \dot{m}_{water}}$$

And

$$\mathcal{E} = 1 - e^{1-NTU}$$

We then find the heat transfer rate using the formula

$$\dot{Q} = \mathcal{E} \, c_{water} \, \dot{m}_{water} \, (T_{steam,in} - T_{water,out})$$

Quality is calculated as before, by dividing out the latent heat.

Both methods were found to yield the same answers for a range of inputs. Using the results,

we were able to select for each proposed mass rate a required water flow rate in order to

obtain both a measurable change in temperature in the water, as well as promote the

condensation of the steam.

Figure 13 below displays a CAD rendering of the condenser, showing the endcaps and the

pipe.

Figure 13: CAD rendering of the condeser tube

3.4 Test Section

The test section is arguably the heart of the setup, as it is where the actual condensation will take place. In many ways, the rest of the system had to be designed around the test section, in order to ensure that the desired pressure drops can be achieved during the condensation process by varying the mass flow rate and cooling rates. In order to achieve this, we had to select an appropriate length and diameter for the test section. It is worth mentioning that both of these can be changed, as most of the tubing and connectors are interchangeable Swagelok fittings. This allows us to easily swap out connectors to extend the length of a section of pipe, and use fitting to connect tubes of different diameters to each other. . In terms of pure mechanical design, however, very little has to be done for the test section itself. It consists of a tube of that can be coated on the inside with a nanostructure and superhydrophobic coating, or any other substance for comparison purposes. Designing and fabricating the condenser (described in the section above) was by far the greatest challenge in producing the test section.

3.5 Funnel Separator

Measuring the heat flux through the condenser, as described above, is one way of measuring the quality of the steam as it exits the test section. Another, more direct, way of doing this would be to separate the steam-water mixture after it emerges from the test section, and then either measure the volume of water obtained, or find the mass flow rate of the steam. The former is difficult to achieve, as the water may take time to drip off the surface of the separation surface. The difference between the initial mass flow rate, measured prior to the steam passing through the separator, and the mass flow rate as it exits, gives us the amount of water condensed.

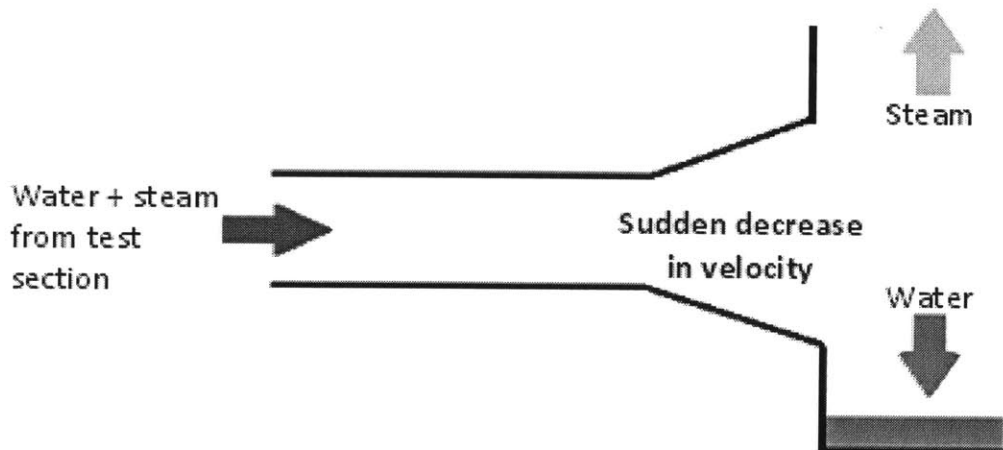The challenge lies in separating the water from the steam. We will do this using a diffuser:

Figure 14: Diffuser separation effect

As can be seen in figure 13 above, the process rests on the fact that an increase in the area of the diffuser will cause the steam to slow down. The water droplets that are being carried in the steam are being borne by the drag force, which varies as the square of the velocity of the steam. Therefore, when the velocity of the steam decreases, gravity dominates over the drag force, causing the water droplets to fall.

The key calculation involves finding the ratios of the inlet and outlet areas, where region 1 is the inlet, and region 2 is the outlet.
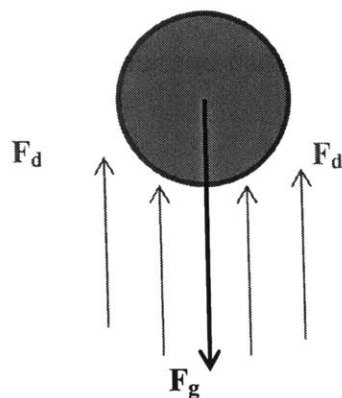
By conservation of mass:

$$v_1 A_1 = v_2 A_2$$

$$v_2 = v_1 \frac{A_1}{A_2}$$

V_1 can be found because we know the A_1 (simply by selecting an inlet area) and the mass flow rate.

V_2 can be found from a force balance on a water droplet, shown in figure 14 below:

Figure 15: Forces on a water droplet

$$F_d = \frac{1}{2} C_D \rho v_2{}^2 \pi R^2$$

$$F_g = \frac{4}{3} \pi R^3 \rho g$$

Where F_g is the force of gravity and F_d is the drag force, and R is the radius of the droplet.

For the water droplets to fall out, the gravity force must dominate the drag force.

$$F_g = F_D$$

$$V_2{}^2 = \frac{8 R g}{3 C_D}$$

Where C_d is the coefficient of drag, dependent upon the Reynold's number.

Using approximate values, V_1 is 5 to 10 times V_2, and hence A_2 is 5 to 10 times A_1, which is certainly feasible.

3.6 Sensors

A considerable fraction of the project cost is being directed towards sensors. These included thermocouples, pressure sensors, mass flowmeters and level sensors. Each of these will now be discussed in greater detail.

The control of pressure and temperature are all important in this system, and therefore must be carefully regulated. The temperature differential between the boiler and the cooler creates a pressure gradient that drives the steam flow, and we needed to be able to monitor the temperatures at various parts of the system to ensure consistency in our testing. This was done using thermocouple probes. Common criteria in selecting these include[21]:

(i) The temperature range that the probe can work with. Since we are working in a vacuum, the temperature range of the system is relatively small, spanning between 5°C (the lowest temperature we are likely to set the chiller at), and the 70°C, which is the highest temperature that the boiler will be placed at.

(ii) The accuracy of the measurements. Precise control of the system's temperature is required, and this calls for highly accurate temperature measurements.

(iii) Chemical resistance. The only fluid the thermocouple will come in contact with is water/steam at various stages of saturation. As such, chemical resistance and corrosion of the thermocouples was deemed to be a relatively minor concern.

(iv) Abrasion and vibration resistance. As the system will be stationary and mounted on a vibration-damping table, this should not be an issue.

(v) Installation requirements. This primarily involves the ability to provide holes in the equipment into which the thermocouples can be inserted. As we are designing and fabricating most of the components ourselves, this is not an issue, as we added holes of the correct diameter where needed.

On basis of the above, we opted for Omega type J thermocouples, with a measurement range of 0-750 °C, with a stainless steel sheaths. Their measurement range easily encompasses the values that they are likely to see in our system, they are highly accurate, and the stainless sheaths provide an adequate measure of corrosion resistance.

A similar thought process dictated the selection of the Omega PX2300 low pressure wet/wet transducer.

In order to measure quality, we required measurement of the mass flow rates of steam before and after condensation, which called for two mass flow meters. Selecting these proved to be a challenge. The majority of commercially available mass flow meters are designed to operate with air, water or oil, but not steam. Finding a flow meter that could work with saturated steam and high temperatures was even more difficult, especially given the low mass flow rates that we working with.

Meanwhile, in order to render the system truly automated, the pump would ideally activate itself when the water level in the boiler dropped below a certain point, or if the water level in the cooler got too high. This requires the use of level sensors that are submersible and functional in vacuum conditions, as well as meeting the criteria required by the temperature transducer. In this case, the accuracy of the instrument was somewhat less important, as the

only risk is either reservoir running out of water, but this is unlikely to happen if the

pumping process is slightly delayed or advanced. However, the ability to function under

water and corrosion resistance remains of importance. Based on this, we selected the Omega

LVU1501 sensor.

3.7 System

Figure 16 below shows a CAD rendering by Jean Sack of the full system. Please rotate sheet counter-clockwise.
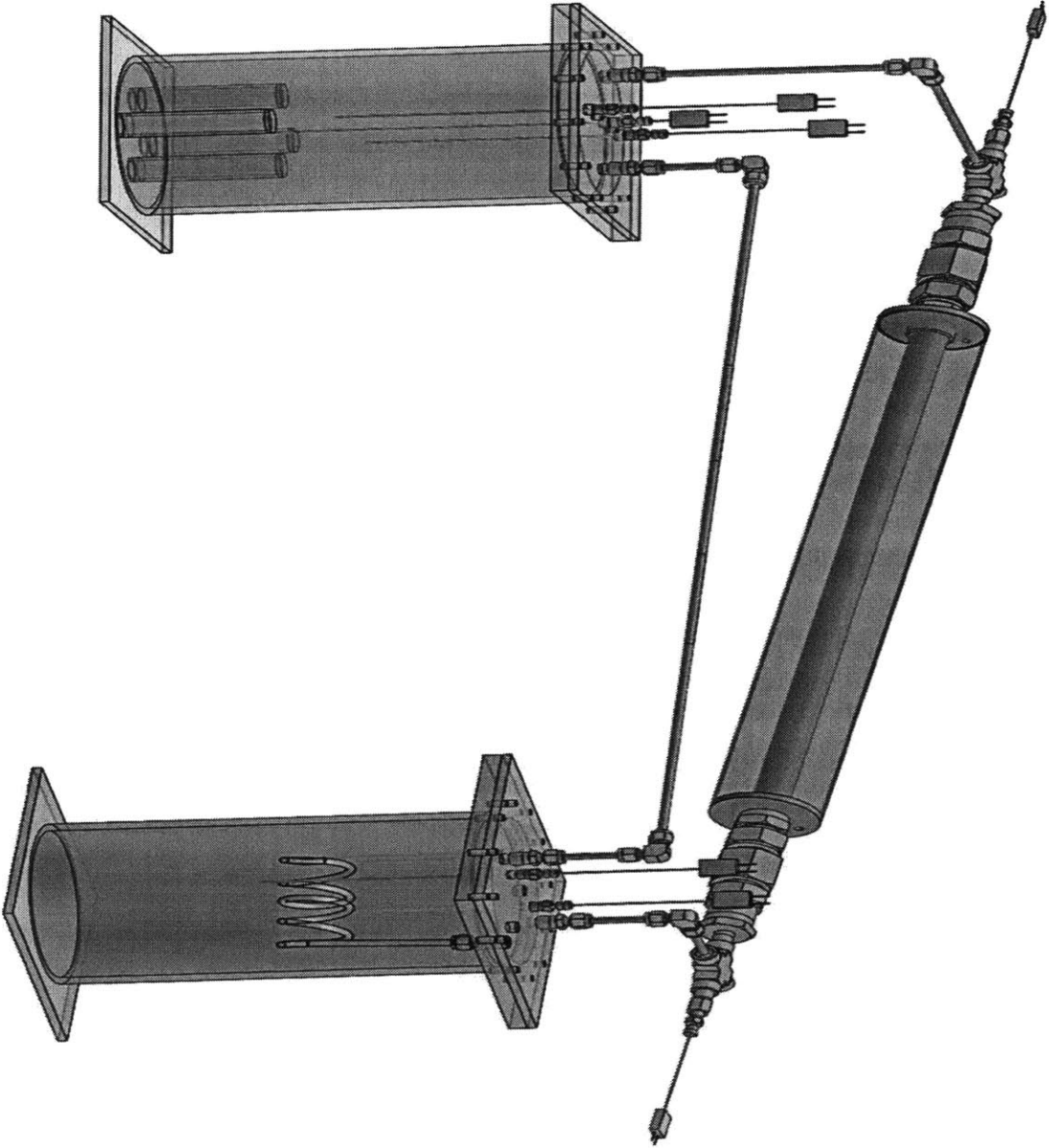
Figure 16: Full System Schematic (Image courtesy of Jean Sack)

Chapter 4: Future Steps

Fundamentally, the internal condensation loops was created to provide a proof of concept that superhydrophobic nanostructures can allow internal condensation via the jumping droplet mechanism, and hence drastically increase the effectiveness of condensation, at a lab scale. The next step will be to scale the model up further, and make additional improvements to more closely simulate industrial conditions.

The internal condensation loop was designed to allow careful control over all the system parameters. As intended, the setup will be used to test the effectiveness of various coatings in promoting condensation. We will be able to measure both the change in quality as well as the pressure drop in the condensing test section for various mass flow rates of air, mass flow rates of cooling water and orientations of the test section for each coating on the interior of the test section, which would give us a clear picture of the effect of each parameter upon the cooling. It could also enable us to come up with a holistic model that relates all of these parameters in a single equation. While the system was designed based upon a theoretical model, our results will enable to account for properties like the angle of the test section, and introduce any correction factors are we see fit.

Another important aspect of the testing that this setup will be used for is robustness testing of the superhydrophobic nanostructures. Particularly for industrial applications, where coatings could see tens of thousands of kilograms of steam pass through and condense upon them in a single day, it is important they should not wear off, and must retain their

superhydrophobicity over an extended period of time (on the order of years) without requiring replacement or major overhaul. While the internal condensation loop setup cannot subject coatings to conditions identical to those they will face in industry, it can assess their robustness on a much smaller scale, and help to eliminate coatings that noticeable suffer from robustness issues.

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