Enhanced Condensation for Organic Rankine Cycle

2nd Quarterly Progress Report

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1. Back Ground

Generating electricity from low grade heat sources has attracted attention due to rising fuel price and increasing energy demand. The organic Rankine cycle (ORC) system is the most practical solution among technologies developed for low grade heat recovery. However, the efficiency of a typical small scale ORC is 10% or less. Most energy loss in the ORC is attributed to thermodynamically irreversible heat transfer processes occurring in its heat exchangers: the evaporator and condenser. In particular for waste heat recovery ORCs, economical success is mainly determined by effectiveness of the condenser because, while their heat source is provided at no cost, heat rejection accounts for most of operation cost. Almost half of total cost for operation and maintenance of an ORC system can stem from its condenser. We investigate and demonstrate heterogeneous condensing surfaces that potentially reduce the irreversibility during the condensation of organic fluids.

2. Progress Report

We have made progress during the reporting period (May 1 ~ July 31) and progress activities are described below.

Task 1: Model Development

In this project we use a heterogeneous condensing surface with two different wettabilities, which are hydrophilic and hydrophobic, in order to enhance condensation heat transfer in organic Rankine cycles. Figure 1 illustrates the heterogeneous condensing surface. Condensation occurs in different manners depending on the wettability of condensing surface. Most metals used for condensing surface in industrial power plants possess high surface energy, which attracts liquids with high bonding energy. As a result liquids wet the surface, leading to conventional filmwise condensation. The area with low surface energy coating has the opposite characteristics. Due to low wettability the hydrophobic surface may promote dropwise condensation. Modeling has been carried out for two cases: coexistence of dropwsie condensation and filmwise condensation and filmwise dominant condensation.

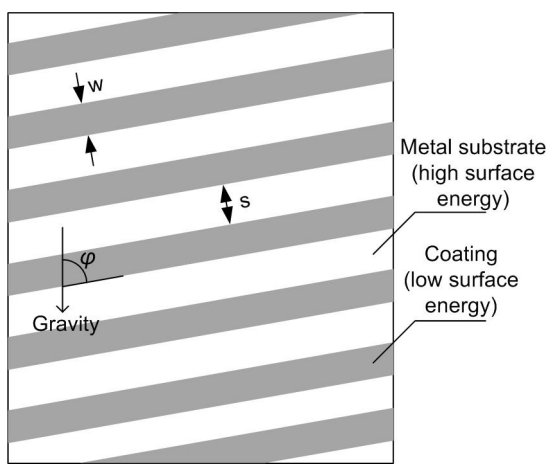

Figure 1: Heterogeneous condensing surface

Coexistence of dropwise and filmwise condensations: the two modes of condensation may happen in one condenser when the proposed condensing surface with the coated area having high

hydrophobicity is used under a moderate condensation rate condition. For this case, both filmwise and dropwise condensation heat transfers determine the average heat transfer coefficient of the condenser. The heat and mass transfer where dropwise condensation occurs can be predicted by calculating population of the droplets and heat transfer rate of individual drops.

Population of droplets is determined by the population balance theory [1, 2]. The growth rates of drops of radius r_1 and r_2 are called G_1, and G_2, respectively. The population density of drops $n(r)$ in an arbitrary area A is defined as the number of drops of radius r per unit area. In order for the number of drops to be conserved in a certain size range, the number of drops entering by growth must equal the sum of the number of drops leaving by growth and the number of drops swept away by larger drops; i.e.,

$$An_1G_1dt = An_2G_2dt + Sn_{1-2}\Delta rdt \tag{1}$$

where S and t are the sweeping rate and the time. As Δr approaches zero, $n_{1\text{-}2}$ becomes a point value, and the equation can now be written by

$$\frac{d}{dr}(Gn) + \frac{n}{\tau} = 0 \tag{2}$$

where the sweeping period $\tau = A/S$.

The heat transfer rate through a drop of radius r with the contact angle θ equals to the rate at which the enthalpy H_{fg} of newly condensing vapor changes, and therefore one can write the single drop heat transfer q_d as a function of the drop growth rate G:

$$q_d = \rho H_{fg} 2\pi r^2 (1 - \cos\theta) G \tag{3}$$

Meanwhile, heat transfer rate through a single drop can be obtained by evaluating the network of thermal resistances residing on the vapor-liquid interface and inside of the drop. From the vapor side to the liquid and solid surface, there are four thermal resistances affecting the condensation heat transfer. They are the vapor-liquid interfacial resistance, thermal resistance due to the drop curvature, conduction resistance of the liquid, and conduction resistance of the coating material. Each component of thermal resistance contributes to temperature drop between the thermal boundaries. Temperature drop by the vapor-liquid interfacial resistance ΔT_{if} varies with the drop radius, drop contact angle, and interfacial heat transfer coefficient h_{if}

$$\Delta T_{if} = \frac{q_d}{h_{if} 2\pi r^2 (1 - \cos\theta)} \tag{4}$$

Drop curvature also creates temperature drop ΔT_c as a function of the surface tension σ and the average temperature of the drop T_{avg} [3]

$$\Delta T_c = \frac{2T_{avg}\sigma}{H_{fg} r \rho} \tag{5}$$

Since, the wall subcooling determines the minimum viable drop radius [4],

$$r_{\min} = \frac{2T_{avg}\sigma}{H_{fg}\rho\Delta T} \tag{6}$$

Equation (6) is reduced to

$$\Delta T_c = \frac{r_{\min}}{r}\Delta T \tag{7}$$

The conduction effect inside of the drop ΔT_{drop} depends strongly on the shape of drop and the thermal conductivity k_d.

$$\Delta T_{drop} = \frac{q_d \theta}{4\pi r k_d \sin\theta} \tag{8}$$

Temperature decrease due to the resistance of the coating material on the contact surface is proportional to the thickness of coating layer δ and the inverse of thermal conductivity k_{coat} of the coating material [5]

$$\Delta T_{coat} = \frac{q_d \delta}{k_{coat}\pi r^2 \sin^2\theta} \tag{9}$$

From Eqs. (4), (7), (8), and (9), the total temperature drop between the vapor and the condensing surface can be computed

$$\begin{aligned}\Delta T &= \Delta T_{if} + \Delta T_{drop} + \Delta T_{coat} + \Delta T_c \\ &= q_d \frac{1}{\pi r^2}\frac{1}{1 - r_{\min}/r}\left(\frac{\delta}{k_{coat}\sin^2\theta} + \frac{r\theta}{4k_d \sin\theta} + \frac{1}{2h_{if}(1-\cos\theta)}\right)\end{aligned} \tag{10}$$

Solving Eq. (2) after substituting Eq. (10) into Eq. (3) we have the drop population equation

$$\begin{aligned}n(r) = \frac{(Gn)_{\min}}{G}\exp\Bigg(&\frac{A_2}{\tau A_1}\left[\frac{(r - r_{\min}^2)}{2} + 2r_{\min}(r - r_{\min}) + r_{\min}^2 \ln(r - r_{\min})\right] \\ &+ \frac{A_3}{\tau A_1}\left[r - r_{\min} + r_{\min}\ln(r - r_{\min})\right]\Bigg)\end{aligned} \tag{11}$$

where $A_1 = \dfrac{\Delta T}{2\rho H_{fg}}$, $A_2 = \dfrac{\theta(1-\cos\theta)}{4k_d \sin\theta}$, and $A_3 = \dfrac{1}{2h_{if}} + \dfrac{\delta(1-\cos\theta)}{k_{coat}\sin^2\theta}$

The steady state dropwise condensation heat transfer rate per unit area of the hydrophobic condensing surface can be obtained by multiplying the heat transfer rate through a single drop with the population density,

$$q''_{dropwise} = \int_{r_{\min}}^{r_{\max}} q_d(r)n(r)dr \tag{12}$$

Condensation on the non-coated condensing area must be filmwise. For this area we can take the Nusselt condensation model [6]

$$q''_{filmwise} = 0.943\left[\frac{k_c^2 H_{fg} g(\rho_c - \rho_v)\Delta T^{3/4}}{L\nu}\right]^{1/4} \tag{13}$$

where g, ρ_c, ρ_v, L, and ν denote the gravitational acceleration, condensate density, vapor density, height of the condensing surface, and kinematic viscosity. When dropwise condensation on the coated area and filmewise condensation on the non-treated area coexist in the proposed condensing surface shown in Figure 1, the average heat transfer coefficient is determined

$$q'' = \frac{wq''_{dropwise} + sq''_{filmwise}}{w+s} \tag{14}$$

Filmwise dominant condensation: the entire condensing surface is curved by a wavy film of condensate. The variation of surface tension is the driving force of the wavy profile of condensate film. It also produces convection flows inside the film and thus enhances the heat transfer. The wavy profile and convection flows are affected by the relative effect of surface tension variation against viscous force of flow. Not only these, many conditions such as thermal properties of fluid, geometry of solid surface, and degree of subcooling also affect the heat transfer process. Thus, experiment is a better way to study the filmwise dominant condensation on the proposed surface. Further investigation will be carried out with experiments.

Task 2: Design and Construction of Testing Apparatus

Design for a condensation experiment apparatus in a SolidWorks environment was completed during the previous reporting period as shown in Figure 2. The test setup consists of the condensing chamber, evaporator, condensing surface and heat conduction rod, and cooling chamber.

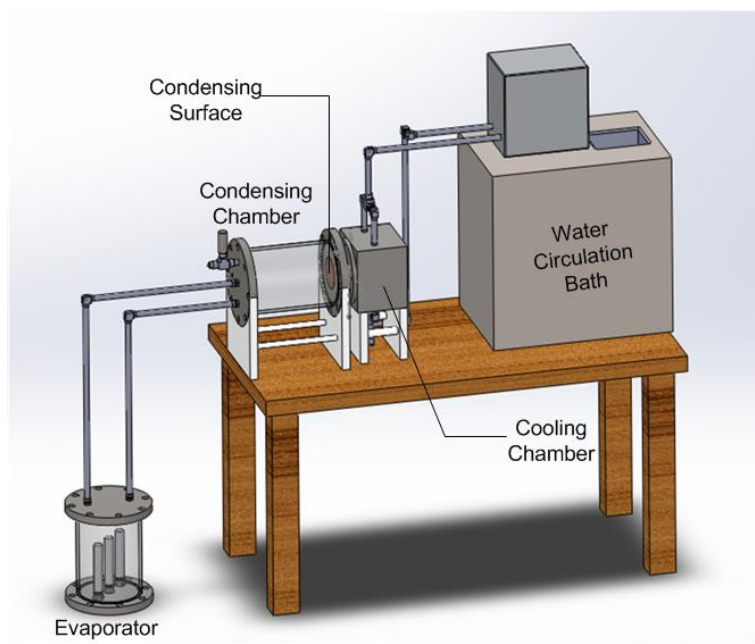

Figure 2: Design of condensation experiment apparatus

Based on the design, we constructed a complete vertical flat plate condensation apparatus as photographed in Figure 3. The apparatus requires a constant vapor source and a heat sink, to dump the heat absorbed by the condensing block from the vapor. The heating chamber constantly heats

and vaporizes the organic fluid. The vapors are then transferred to the condensing chamber though the upper pipe line. The cooling chamber, which removes the latent heat from the organic vapors, is fed with cooling water. Cooling water is circulated by the chiller pump (also called chilled water circulation bath), which maintains the set cooling water temperature. The condensate (condensed vapor) then flows back to the boiling chamber through the lower pipe line. This configuration allows the condensation process to be continuous and steady.

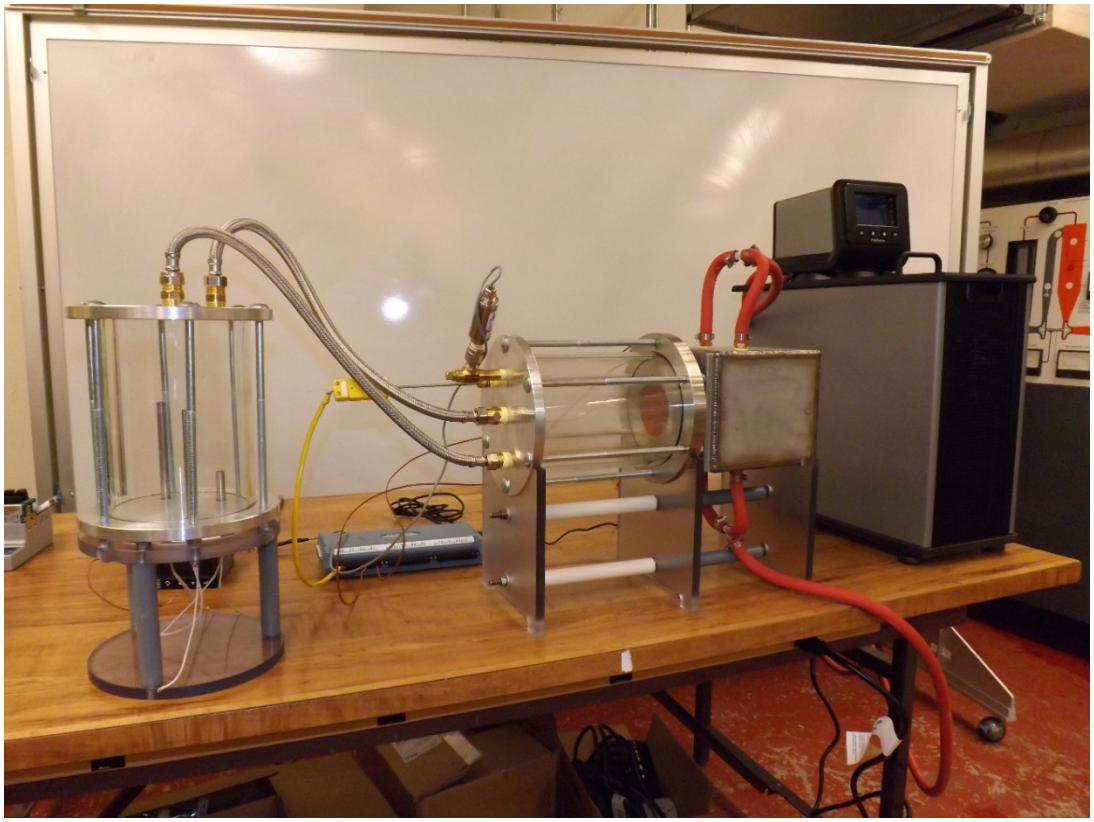

Figure 3: Condensation apparatus that consists of evaporator, condensing chamber, condensing surface, cooling chamber, and water circulation bath

1. Evaporator (Boiling Chamber)

The chamber in which the organic liquid is heated to its boiling point and evaporates is known as the evaporator or boiling chamber. The evaporator must withstand high pressures as well as vacuum pressures, for example, if we run the condenser with a high pressure fluid such as water as working fluid, the fluid tends to get superheated, which also raises the pressure. On the other hand, if we run a low pressure fluid such as most organic fluids, we would observe vacuum pressures. (Normal operating conditions of industrial power plant condensers are vacuum conditions.) In both cases, we need a boiling chamber which would withstand the aforementioned conditions. Moreover, the chamber should be air tight under positive and negative pressures.

A borosilicate glass cylinder is chosen as the wall for the boiling chamber because of its low thermal conductivity, and clear optical properties. Thanks to its low thermal conductivity it is resistant to thermal shocks. The length of the glass is eight inches. The outer diameter of the glass cylinder is six

inches. The thickness of the cylinder is one third of an inch. Two aluminum disks are used to cover the two open ends of the glass cylinder. The aluminum disks are each eight inches in diameter and half inch thick. These aluminum disks have a groove so that the glass cylinder would sit tight in to the aluminum disks. Figure 4 shows the disk with a gasket in the groove. One eighth inch thick neoprene gasket sits into the groove which provides an air tight seal. The two aluminum disks are tightened with nuts and bolts. On one of the aluminum disk there are three holes of diameter of three eighths of an inch, which are tapped so that three cartridge heaters can be screwed in to the tapped holes. Figure 5 shows the tapped holes for the heaters. On the other aluminum disk there are two holes which are tapped with one half of an inch pipe fitting threads, in which a brass nipple is fitted so that the vapor can flow through one of the pipe lines connected to the condensing chamber,, and in other pipe line, the condensed fluid will flow back to the heating chamber from the condensing chamber. Figure 6 shows the borosilicate glass sitting in the groove on the aluminum disk.

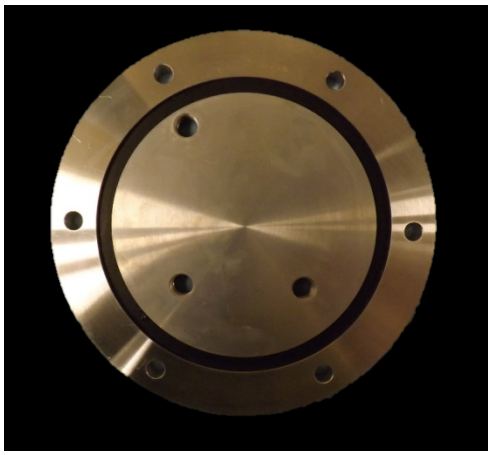

Figure 4: Aluminum disk with a groove for gasket

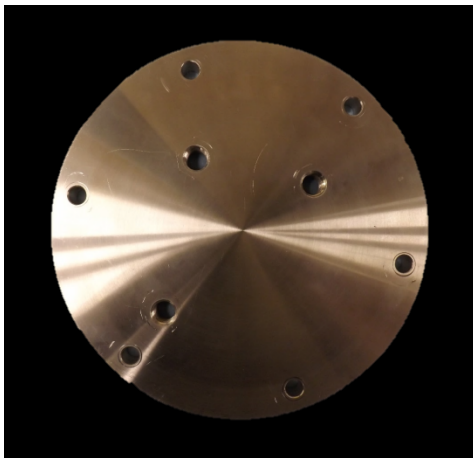

Figure 5: Aluminum disk with tapped holes for heaters

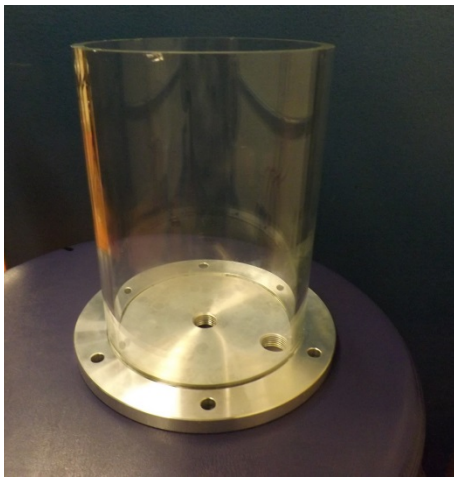

Figure 6: Borosilicate glass, sitting in the aluminum disk's groove

2. Condensing chamber

The condensing chamber is similar to the boiling chamber in size, dimensions, and materials. Here borosilicate glass is used for the same reasons as mentioned above for the boiling chamber, and the clear optical properties of borosilicate glass allow us to visually observe the condensation type that the vapor is going through. This is the main reason for choosing borosilicate glass cylinder. In the condensing chamber, instead of having two aluminum disks we have one aluminum disk and one stainless steel disk. The reason for choosing aluminum disk is that, aluminum has a high thermal conductivity; so a system with aluminum would come to equilibrium faster when compared to stainless steel and plastics. This is the reason why we chose aluminum for the boiling chamber as well. The aluminum disk in the condensing chamber also has two tapped holes of half inch diameter in which a nipple is screwed in, and to which pipes are connected from the boiling chamber. The other end of the borosilicate glass is closed with a stainless steel disk instead of aluminum. The stainless disk has a three inch diameter hole in the center which houses the condensing block. Figure 7 shows the stainless steel disk with the 3 inch hole in the center. The condensing block is made out of copper. Because the condensing block and the plate housing it are made of dissimilar metals, the thermal expansion coefficient of copper and the thermal expansion coefficient of stainless steel are to be considered. The thermal expansion coefficients of both copper and stainless steel are close enough that no leaking problems should occur at the copper and stainless steel interface. Aluminum has a higher thermal expansion coefficient than copper, so it is not advisable to use an aluminum disk. The thickness of the stainless steel disk is three quarters of an inch, it is thicker than the aluminum disks because in addition to the grooves for gaskets it also has a groove on the other side to incorporate an O-ring and it has tapped holes for the socket head screws. Figure 8 shows the O-ring on the other side of the stainless steel disk. The purpose of the O-ring on the other side of the stainless steel disk is to ensure that there is no leak between the copper condensing block and the stainless steel disk. This is explained in detail in the cooling chamber section below. The tapped holes are to join the condensing chamber with the cooling chamber. The condensing block bridges the gap, allowing heat to transfer from condensing chamber to the cooling chamber.

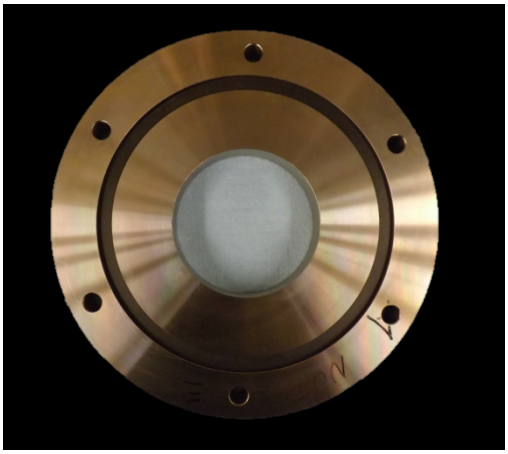

Figure 7: Stainless steel disk with three inches diameter hole in the center

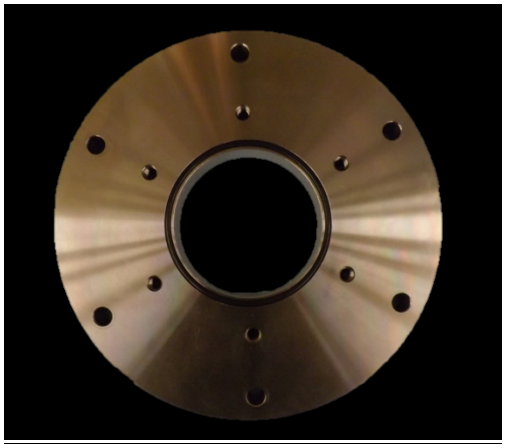

Figure 8: Other side of stainless steel disk with o-ring and tapped holes

3. Condensing block

The copper rod of width 3.5 inches diameter and length of 4.5 inches was machined in lathe in such a way so that the diameter of the copper rod is 3 inches except for a 1 inch section where there is a step of 0.5 inches. The step begins at 0.75 inches from right end. The step is machined so that the vacuum in the condensing chamber cannot force the copper condensing block through the housing hole in the stainless steel plate. The step rests against the O-ring of the stainless steel disk. 0.75 inches of the copper block fits into stainless steel disk. The other part of the copper goes into the cooling chamber. There are three holes on the step surface of the copper, to allow thermocouples to measure the temperatures of the copper at known spacing intervals. Figure 9 shows the copper block with holes for the thermocouples.

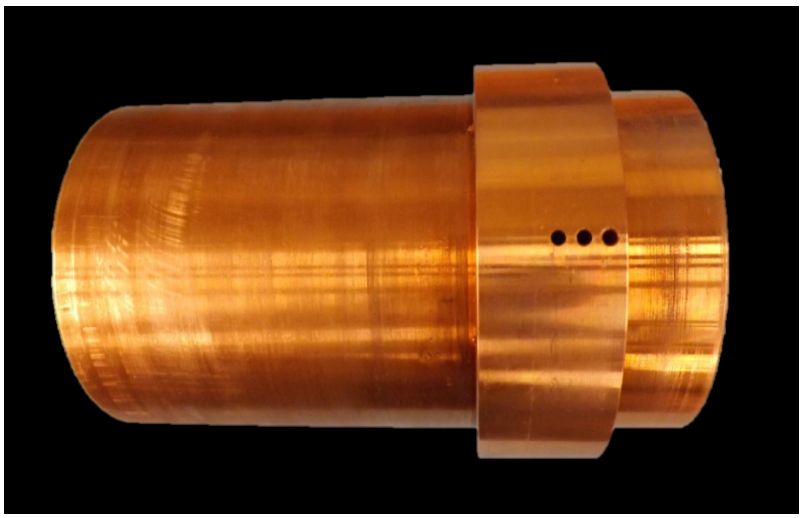

Figure 9: Copper block with holes on step surface

4. Cooling chamber

The cooling chamber is made out of stainless steel type 304, because it is easy to machine, does not rust, has a high thermal resistance for a metal, and has a thermal expansion coefficient lower than that of copper. The cooling chamber is a 6”×6”×4” cube. All sides are welded to avoid any possible leaks of cooling water which is supplied by the chiller pump. The front face of the cooling chamber has a three inch diameter hole, in which the copper block is held. The 3.5 inch diameter step on the copper block rests against the O-ring, which sits in the groove. The copper block rests against the O-ring so as to prevent any possible leaks. The front face of the cooling chamber has six holes in a circular pattern, so that the specially manufactured socket head bolts sit in the holes, and a plate with holes identical to the circular pattern would sit on the front face in such a way that the socket head bolts can freely rotate, but would not come out of the holes. The plate is bolted to the cooling chamber using screws. The condensing chamber stainless steel disk has threaded holes on its outer surface, which incorporates these bolts. By tightening the bolts, the heating chamber and the cooling chamber tighten against the copper block, which gives a leak proof configuration in both the condensing chamber and the cooling chamber. Figure 10 shows the front face of cooling chamber with the holes for the socket head bolts, and the tapped holes for the bolts to fix to the plate.

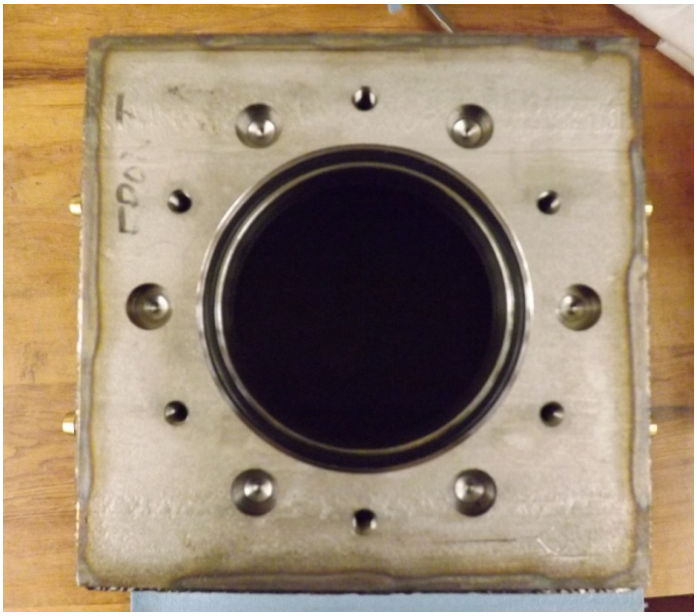

Figure 10: Cooling chamber. Front face showing holes for socket head bolts and tapped holes for screws and a 3" diameter hole for copper block

5. Heaters

We use three immersion type heaters (cartridge heaters) for heating the liquid in the boiling chamber as shown in Figure 11. The heaters have NPT threads for the connection into the boiling chamber. Of three heaters, two heaters are 250 watts, and the third one is 75 watts. These heaters give enough heat to boil the organic fluid.

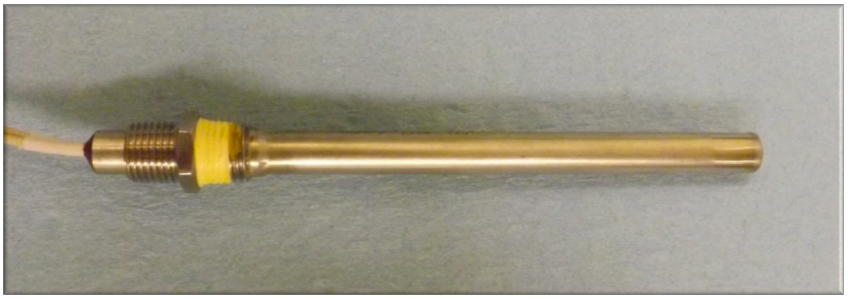

Figure 11: Cartridge heater

6. Variac

It is important to control the heat input and the heat that is being transferred through the copper block. A variac which is also known as a variable autotransformer is used to control the output voltage, for a steady input AC voltage. The variac has a dial on the top which is used to control the voltage output. Three heaters are plugged in to the variac, and by adjusting the dial on the top, we can control the voltage to the heaters. By doing this we can control the heat input from the heaters.

7. Measurement

For the measurement of the temperatures, we use Omega's hermitically sealed tip "K" type thermocouples. These thermocouples were chosen because the tips for these thermocouples are

sealed with an insulating material, which does not expose metal tip of the thermocouple. Because of this we can make sure that the metal tip is not in contact with the copper rod since thermocouples are inserted in the copper rod. We use an Omega pressure transducer to measure the pressure and an NI DAQ to read the measured temperatures and pressures.

8. Chiller pump

By using a chiller pump we can control the heat transfer through copper tube. We use chiller pump for fine adjustment of the heat transfer, since we can adjust the chiller fluid temperature to 0.01 degree Celsius. We will be using a Polyscience chiller pump, which has a temperature range from 5 degree Celsius to 100 degree Celsius. The chiller pump will be using DI water as the working fluid, which will be pumped by the chiller pump into the cooling chamber. This is a closed cycle in which the DI water will take away heat from the copper rod and then the water rejects that heat to the chiller pump and gets recirculated in to the cooling chamber. The figure 12 below shows the chiller pump connected to the cooling chamber

Figure 12: Chiller pump connected to the cooling chamber of the condenser

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