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DEMONSTRATION OF A PASSIVE CONDENSER LOOP

CHIEN-HUA CHEN
ADVANCED COOLING TECHNOLOGIES, INC.

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DEMONSTRATION OF A PASSIVE CONDENSER LOOP

Ojas Govardhan[2], Chien-Hua Chen[1], Josh Charles[1], Sean Hoeing[1], Mike Ellis[1], and Richard Bonner[1]

[1]Advanced Cooling Technologies, Inc. Lancaster, PA

[2]CBRE, Seattle, WA

ABSTRACT

Building HVAC consumes significant amount of energy. It is estimated that about 7% of the total electricity consumed by water cooled chiller is used to drive the condenser water pump. A passive condenser loop is developed to curb this energy consumption by replacing the open pumped loop to a closed loop thermosyphon system. The loop thermosyphon uses waste heat to circulate the condenser fluid, eliminating the electrical pumping power requirement and the large circulation pump. Since it is a closed loop, it also reduces the maintenance cost and enables both wet and dry cooling modes. A demonstration unit with a riser about 40 feet (12.2 meters) tall is fabricated and tested at powers up to 25 kW. The result has successfully showed that the loop thermosyphon is able to transfer the heat near isothermally (ΔT < 0.4^{o}C) over the long distance without the need of any pump and consuming any electricity.

Keywords: Loop Thermosyphon, Evaporative Condenser, Passive Heat Transfer, Building HVAC

NOMENCLATURE

HVAC	Heating, Ventilation, and Air Conditioning
DOE	Department of Energy
BTO	Building Technologies Office
$\dot{m}_L$	Liquid mass flow rate of refrigerant
$\dot{m}_V$	Vapor mass flow rate of refrigerant
D_L	Liquid line diameter
D_V	Vapor line diameter
ρ_L	Refrigerant liquid density
ρ_V	Refrigerant vapor density
EMI	Electromagnetic Interference
WBT	Wet Bulb Temperature
DBT	Dry Bulb Temperature
$R_{overall}$	Overall thermal resistance
$\dot{m}_{measured}$	Measured mass flow rate
$\dot{m}_{fg}$	Calculated mass flow rate (Heat of evaporation / input power)
Q_{in}	Input power
h_{fg}	Heat of evaporation

1. INTRODUCTION

Open-loop evaporative cooling tower is commonly used in large building for heat rejection. It requires a cooling water pump to circulate the cooling water from the chiller to the cooling tower. This cooling water pump accounts for roughly 7% of the total cooling energy consumption [1]. The total electricity consumption from the water-cooled chiller (commercial building) is estimated roughly 98B kWh in 2023 [2]. Based on these numbers, the cooling water pump accounts for 6.86B kWh/year with cost of $686M/year (assuming $0.1/kWh). This electricity consumption and the associated cost can potentially be saved by using a passive condenser loop, which consists of a loop thermosyphon and an evaporative condenser.

A loop thermosyphon uses heat to circulate working fluid between the heat source and the heat sink. Waste heat from the chiller condenser vaporizes the working fluid, and the resultant density and pressure difference between the vapor line and liquid line provide a driving force to circulate the flow. The vapor is condensed by an evaporative condenser, rejecting to the ambient air through the force air convection (dry mode) or evaporative cooling (wet mode). The condensate backs to the heat source by gravity, and then the cycle repeats. This mechanism enables a passive heat transfer over a very long distance (both vertically and horizontally). Compared to the conventional thermosyphon, the loop thermosyphon separates the liquid return path from the vapor transport line, which eliminates the flooding limit and is able to use more compact pipe size for large power transport.

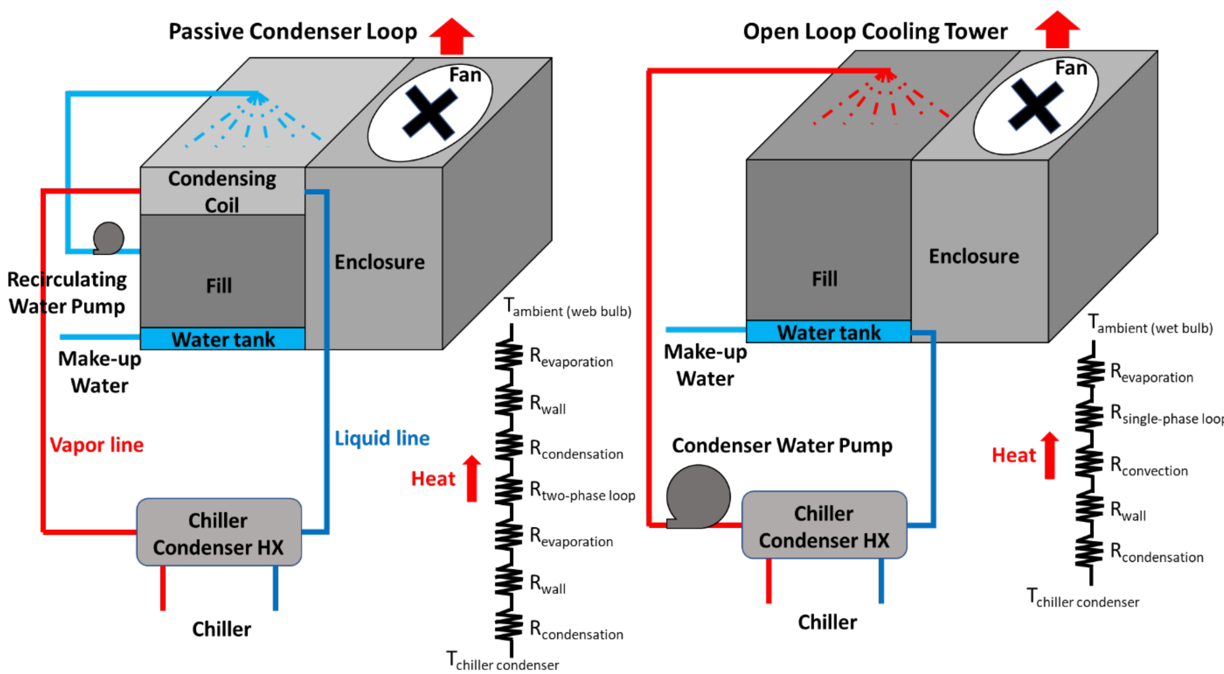


Figure 1. Schematics of the passive condenser loop and the open loop cooling tower. Note in the passive condenser loop, a small recirculating pump is still needed for spraying the cooling water to the condensing coil, but the electrical energy consumption is much smaller compared to the condenser water pump in the open loop cooling tower.

Figure 1 shows the schematics of a passive condenser loop and a conventional open loop cooling tower. The passive condenser loop uses a loop thermosyphon to transfer the heat via the latent heat of the refrigerant, while the open loop cooling tower uses an active pump to transfer the heat via the sensible heat of the cooling water. The single-phase thermal resistance of the open loop cooling tower in Figure 1 ($R_{single\text{-}phase\ loop}$) can be calculated via $1 / \dot{m}Cp$. On the other hand, the two-phase thermal resistance of the passive condenser loop ($R_{two\text{-}phase\ loop}$) can be calculated by the saturation temperature difference between the evaporator and condenser divided by the input power. For refrigerants with steep saturation curves under the normal operational temperature, the saturation temperature difference of the loop thermosyphon can be very small ($\Delta T < 1^oC$), compared to a typical pumped condensing water loop that has the ΔT around 5^oC. The reduced two-phase loop ΔT helps compensate the less efficient in-direct evaporative cooling of the evaporative condenser.

The evaporative cooling consumes a large portion of water supply (e.g. 28% of total water consumption in office buildings [3]). The open-loop cooling tower cannot operate without water, while the closed passive condenser loop can operate in both wet and dry modes. This feature reduces water consumption when the ambient temperature is low, and increases building resilience during the water scarcity scenario.

In addition to energy and water saving, the closed loop reduces the amount of regular maintenance as well as the chemical treatment compared to the conventional open loop cooling tower, which provides additional cost saving. The use of refrigerant as the working fluid also eliminates the freezing issue during the winter.

2. EXPERIMENTAL SETUP

A passive condenser loop was set up to demonstrate the feasibility of using a loop thermosyphon for the building HVAC application. The system consists of four major components:

1. Condenser – Evaporative Condenser
2. Evaporator – Heat Exchanger
3. Vapor Line
4. Liquid Line

Figure 2 shows the schematic of the test loop. Details of each component are described in the following subsections.

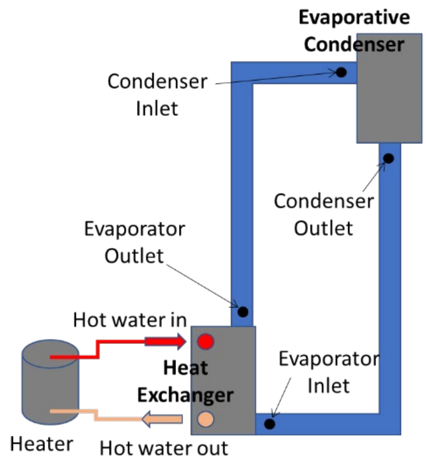


Figure 2. Schematic of the passive condenser loop test setup.

2.1 Condenser (Evaporative Condenser)

Since the loop thermosyphon is a closed system, a closed loop condenser is needed. A closed evaporative condenser is commercially available. While the evaporative condenser is commonly integrated with a vapor compression system, it fits well to serve as the condenser for the loop thermosyphon. The evaporative condenser includes several parallel serpentine coils where the vapor phase refrigerant is condensed to the liquid phase. Due to the closed system, the heat rejection can be accomplished by either wet cooling (with water spray) or dry cooling (without water spray). For wet cooling, a small water pump is needed to circulate the cooling water within the cooling tower. The evaporative condenser was placed on the rooftop of the building where ACT is located (Figure 3). The rooftop is about 40 feet from the floor, where the evaporator heat exchanger is located.

Figure 3. An evaporative condenser integrated with the loop thermosyphon. The condenser is located on the rooftop of Burle business park where ACT is located.

The pressure drop inside the condenser is estimated via a two-phase, Friedel's correlations. The calculation shows that pressure drop is minimal (less than 1 Pa) for the operational range due to the very small mass flow rate and the large effective cross section area. The pressure drop caused by the condenser is insignificant compared to the pressure drop in the vapor and liquid lines.

2.2 Evaporator

A flat plate heat exchanger with four ports is used to serve as the evaporator. Two ports are connected to the loop thermosyphon with working fluid (R134a) condensate entering from the bottom port and leaving from the top port. The other two ports are connected to a circulation hot water heater to serve as the heat load (Figure 2). The heat exchange is orientated in a countercurrent fashion, in which the hot water enters from the top port and leaves from the bottom port. The heater uses resistance coils to increase the temperature of the water and this hot water, connected by pipes to the evaporator inlet and outlet exchange heat with the cool refrigerant in the loop. The input power (Q_{in}) to the hot water can be adjusted by increasing the heater resistance and a maximum of 25 kW heat can be inputted to the system. The water mass flow rate to the heat exchanger can be adjusted to control the outlet temperatures of the hot water ($Q = \dot{m} \times C_p \times \Delta T$). The working fluid pressure drop inside the evaporator is also calculated using Friedel's correlations. Due to the large size of the evaporator heat exchanger used in the setup, the calculated frictional pressure drop is also very small (2.3 Pa).

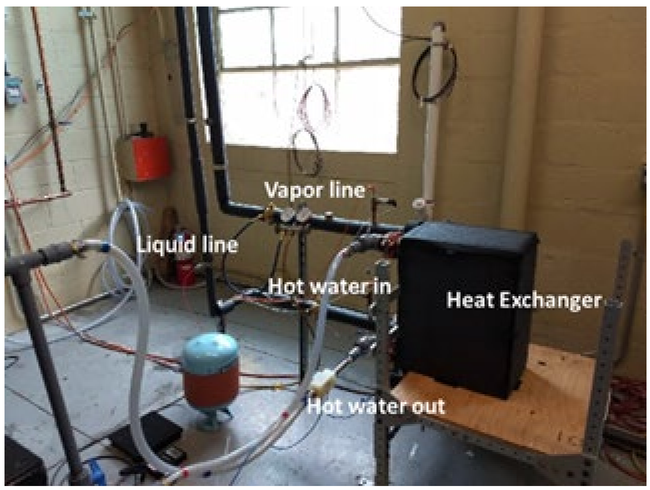


Figure 4. A flat plate heat exchanger connected to the loop thermosyphon and a recirculation water heater.

2.3 Vapor Line

The vapor line carries the refrigerant in either single-phase vapor or two-phase mixture depending on the charge and input power. To ensure the circulation motion of the flow, and therefore the continuous heat rejection, the following equation (Equation 1) has to be satisfied:

$$\Delta P_{\text{liquid head}} = \Delta P_{\text{vapor line}} + \Delta P_{\text{liquid line}} + \Delta P_{\text{evaporator}} + \Delta P_{\text{condenser}} \qquad (1)$$

Due to high velocity in gas phase, the $\Delta P_{\text{vapor line}}$ is likely dominant in the right hand side of the Equation 1. For the single phase (vapor) flow, the friction part of the $\Delta P_{\text{vapor line}}$ can be estimated via Darcy–Weisbach equation. In the current test setup design, it is assumed that the refrigerant exits the evaporator at the saturation conditions of 30°C and 770.1 kPa. The length of the vapor line is the combination of the vertical tubing (the height between the evaporator and the condenser, i.e. 40 feet) and the horizontal tubing (10 feet). There are also number of elbows that can cause minor losses. The design of the vapor line pipe sizing is to balance the performance (pressure drop) and the cost. The larger the pipe size, the lower the pressure drop, but also the higher the plumbing cost. A 1.5” nominal diameter copper pipe was selected for the vapor line and the calculated frictional pressure drop is 2,332 Pa.

2.4 Liquid Line

The liquid line is the piped connection between the exit of the condenser to the evaporator. This downcomer line carries the refrigerant in liquid state after the liquid changes its phase in the condenser. The liquid line is expected to carry the same mass flow rate of refrigerant as that of the vapor line. To evaluate the dimensions of the liquid line, the mass flow rate in the vapor line is equal to the mass flow rate in the liquid line.

For constant mass flow rate,

$$\dot{m}_L = \dot{m}_V \tag{2}$$

Therefore,

$$D_L = ((U_V / U_L) \times (D_V^2 \times \rho_V) / \rho_L)^{0.5} \tag{3}$$

Since the vapor velocity is higher than the liquid velocity ($U_V / U_L > 1$), $D_L > ((D_V^2 \times \rho_V) / \rho_V)^{0.5}$.

Considering

$D_V = 1.5$” and densities at 30°C are $\rho_L = 1387.46$ kg/m^3 and $\rho_V = 37.54$ kg/m^3

Minimum Diameter for liquid line has to be > 0.25”.

Hence, from these calculations and considering the market availability, we choose the liquid line pipes to be 0.5” in diameter connecting the outlet of the condenser to the inlet of the evaporator.

Figure 5 Left shows the vapor and liquid lines. Figure 5 Right shows the 3D drawing of the test setup with aforementioned design inputs.

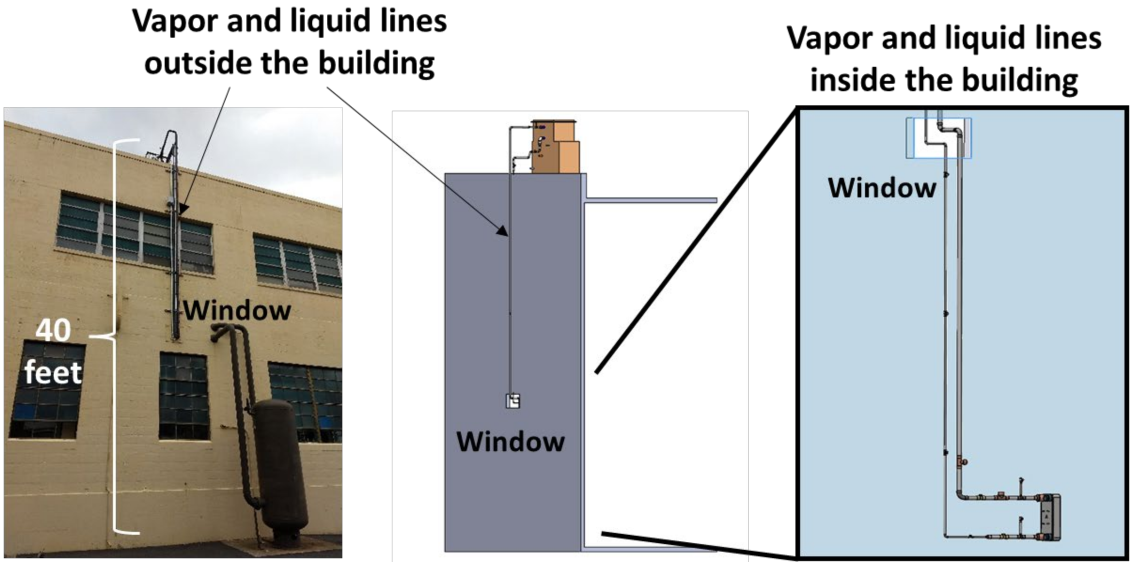


Figure 5. Left: Picture of the vapor and liquid lines outside the building. Middle: 3D drawing of the test setup. Right: Vapor and liquid lines inside the building.

2.5 Instrumentation and Total Volume

The entire testing loop needs to be monitored for pressures, temperatures, and flow rates to predict the performance of the loop. Four Resistance Temperature Detectors (RTDs) and Pressure Transducers (PTs) are used to monitor the temperature and the pressure at the evaporator outlet, condenser inlet, condenser outlet, and evaporator inlet. A flow meter is installed on the liquid line section before the evaporator inlet to measure the liquid refrigerant flow rate in the system. Three sight glasses on the liquid line at the height of 3 feet, 8 feet and 13 feet front the ground are installed in order to check the height of the liquid column. Another sight glass is installed 3 feet below the condenser on the liquid line in order to check the quality of refrigerant at the exit of the condenser. Two sight glasses are installed in the vapor line to visualize the quality of refrigerant leaving the evaporator.

The total volume of the loop was calculated to be 88.46 liters. Figure 6 shows the system volume distribution of each component.

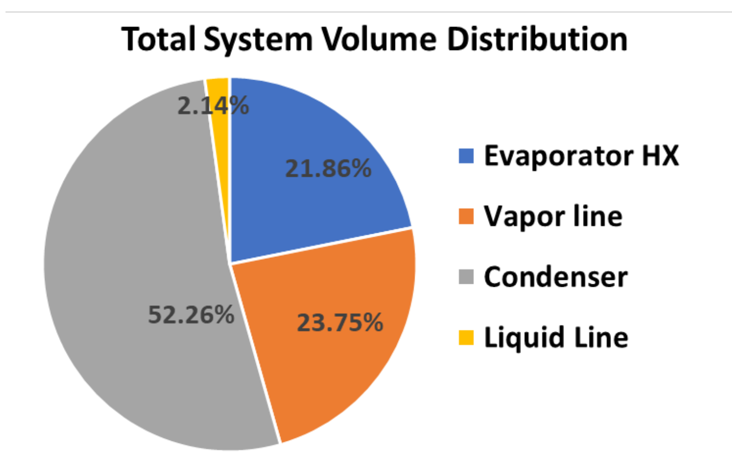


Figure 6. System volume distribution.

3. EXPERIMENTAL RESULTS

The passive condenser loop prototype was tested at multiple powers under different charges to evaluate the thermal performance. The system was charged initially to 60 lb (27.2 kg) of R-134a refrigerant. Then 10 lb (4.54 kg) of charge was added in the system and the same tests were conducted at 70, 80, 90, 100 and 110 lb. charges. All the parameters including temperature, pressure, and refrigerant mass flow rate were measured for each charge. During the testing, the heater power was increased from 15 kW to 25 kW with a step of 2.5 kW. Note that since the hot water circulation pump also generates heat, the measured powers (by circulation water calorimetry) were slightly higher than the heater input power as shown in the Figure 8 to Figure 12. A time interval of approximately 40 minutes between each test was allotted to ensure the system reaches to the steady state.

3.1 60 LB CHARGE

At the charge of 60 lb (Figure 7), the refrigerant occupied around 26% of the total loop volume. Figure 7 shows the temperature data of 60 lb charge under different input powers. The condenser inlet temperature showed an early spike but the evaporator outlet maintained the same temperature with the rest of the loop suggesting the superheated vapor was caused by the vapor line wall conduction. As the heater power increases, the evaporator outlet temperature also showed a spike (at 20 kW) indicating the evaporator heat exchanger is drying out. The system performance was hampered as the hot water out temperature increased significantly with the increased power. The results indicated that the system was under charge.

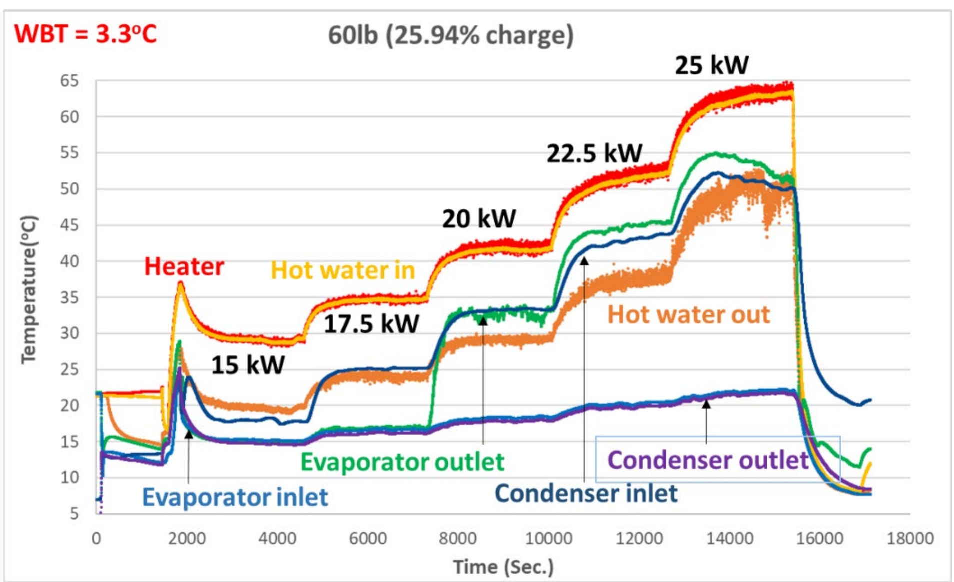


Figure 7. 60 lb charge test data.

3.2 70 LB CHARGE

By adding 10 lb charge (70 lb, Figure 8), the system yields much better results than the previous test. The heater temperature is around 45ºC at 25 kW input power compared to > 60 ºC for the 60 lb charge, even with higher wet bulb temperature (WBT, 12.8ºC vs. 3.3 ºC). The hot water out temperatures maintain close to the loop liquid line temperatures (condenser outlet and evaporator inlet) throughout the tests. While at the heater powers of 20-25 kW, the vapor line shows superheated vapor, it does not impact too much on the hot water out temperature as 60 lb charge does. This suggests that no dry out happens in the evaporator heat exchanger and the superheated vapor in the vapor line is through vapor line wall conduction.

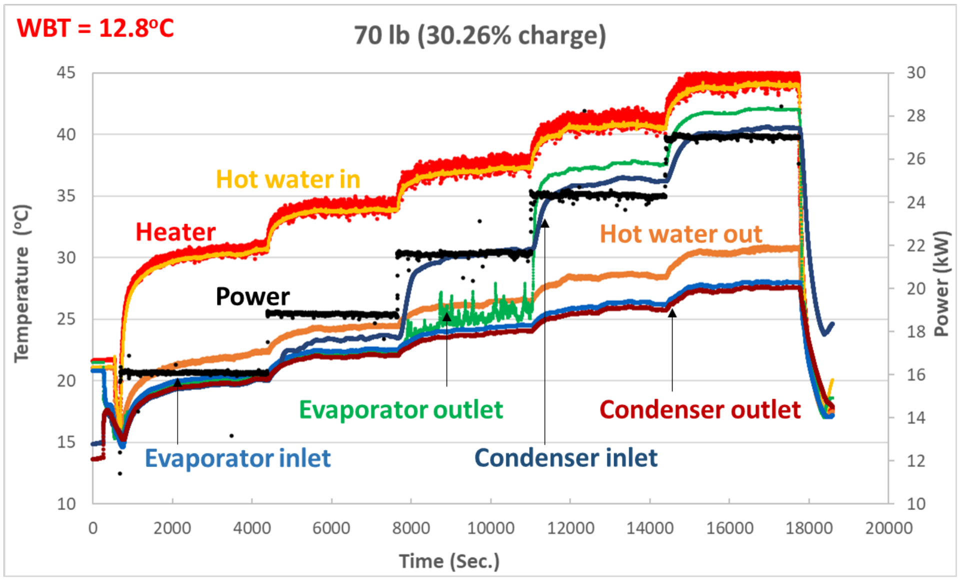


Figure 8. 70 lb charge test data.

3.3 80 LB CHARGE

At the charge of 80 lb (Figure 9), the volume occupied by the refrigerant equals 30.6 liters which accounts for 35% of the total loop thermosyphon volume. The heater water temperature is about 27°C initially and rises to 40°C at 25 kW power. This lower heater temperature is contributed by the lower WBT. At this charge, the vapor line and the liquid line temperatures are almost equal (isothermal loop behavior) throughout the test. The condenser inlet temperature is slightly lower than the evaporator outlet temperature indicates two phase flow in the vapor line. The whole loop system maintains isothermal two-phase flow as no superheated vapor or subcooled liquid is observed.

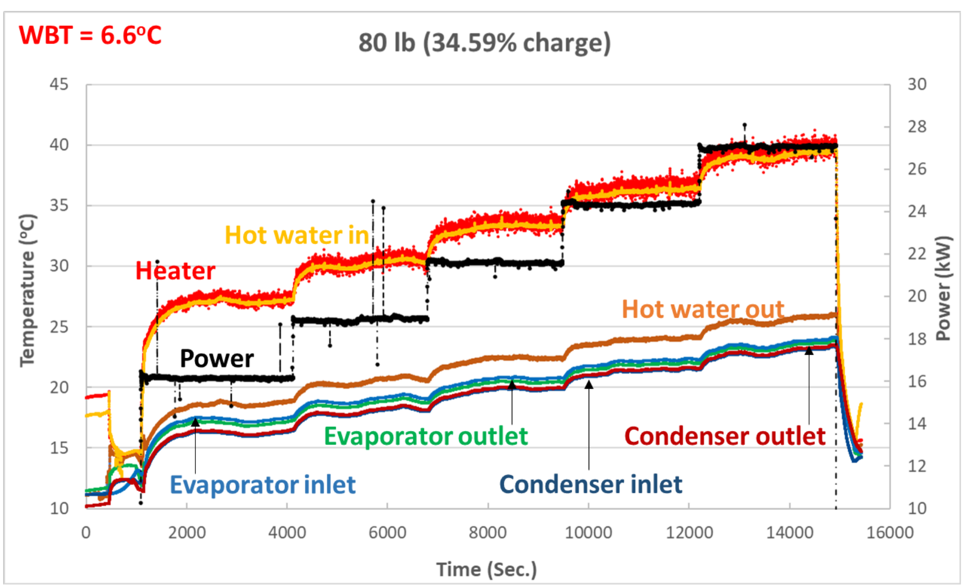


Figure 9. 80 lb charge test data.

3.4 90 LB CHARGE

At the charge to 90 lb (Figure 10), the loop behaves similar to 80 lb, except the slight subcooled liquid is observed at higher powers as the condenser outlet temperature falls below the condenser inlet temperature. This is because under this charge and the input powers, the liquid line is completely filled with liquid and the excess liquid refrigerant reaches inside the condenser leading to subcooling in the system.

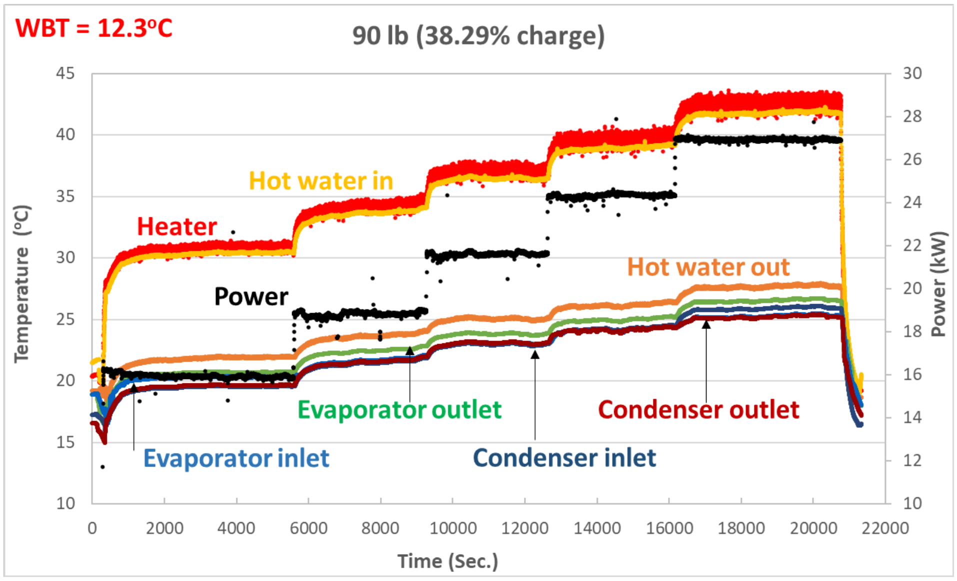


Figure 10. 90 lb charge test data.

3.5 100 LB CHARGE

At a charge of 100 lb (Figure 11), prominent subcooling of 1~2°C is observed in the loop throughout the test. Some fluctuations in the heater power are observed due to electromagnetic interference (EMI). Under this charge, the flow in the vapor line is two-phase and the liquid line is completely filled with subcooled liquid.

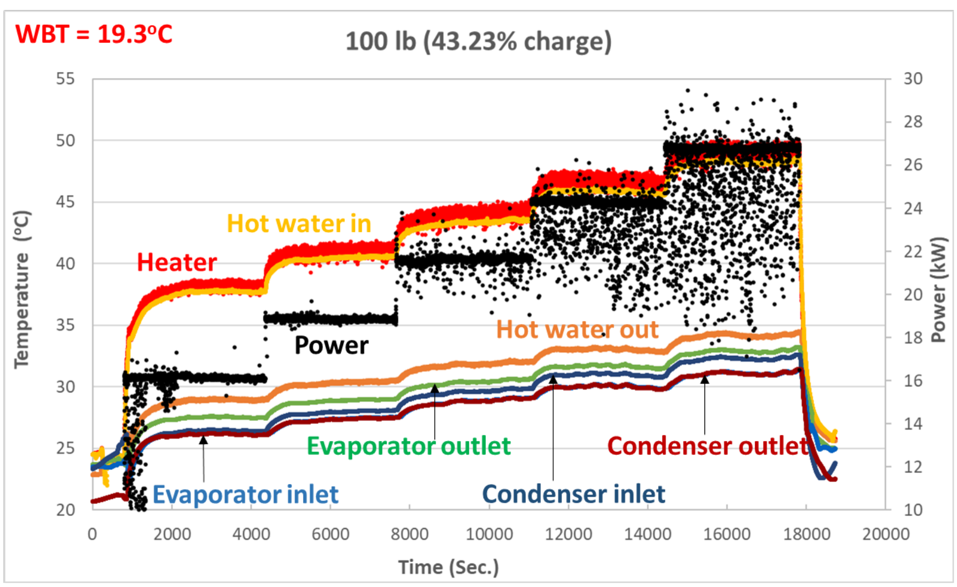


Figure 11. 100 lb charge test data.

3.6 110 LB CHARGE

At the charge of 110 lb (Figure 12), the system is almost half (48%) volumetrically charged and a more significant subcooling of 3~4°C is seen in the liquid line.

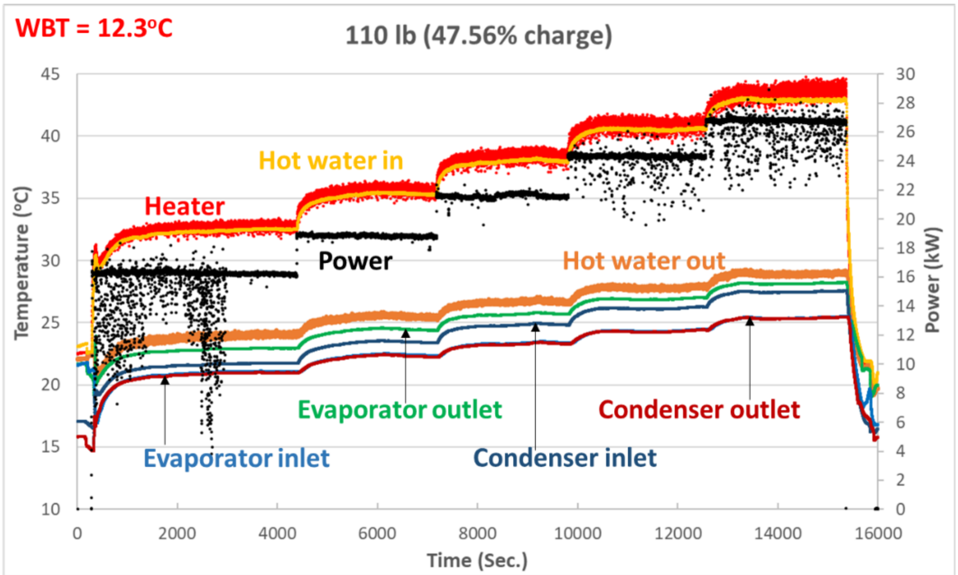


Figure 12. 110 lb charge test data.

4. DISCUSSION

4.1 Loop Thermosyphon ΔT vs. Different Charges

The loop thermosyphon performance can be evaluated by the ΔT between the evaporator outlet and the condenser outlet (locations were shown in Figure 2). Figure 13 shows the ΔT under 80 lb to 110 lb charges. As can be seen, the increased charge increases the ΔT. In addition, when the liquid head in the liquid line reaches to the condenser (90 lb, 22.5kW), an increase in ΔT is observed due to subcooled condensate in the condenser outlet.

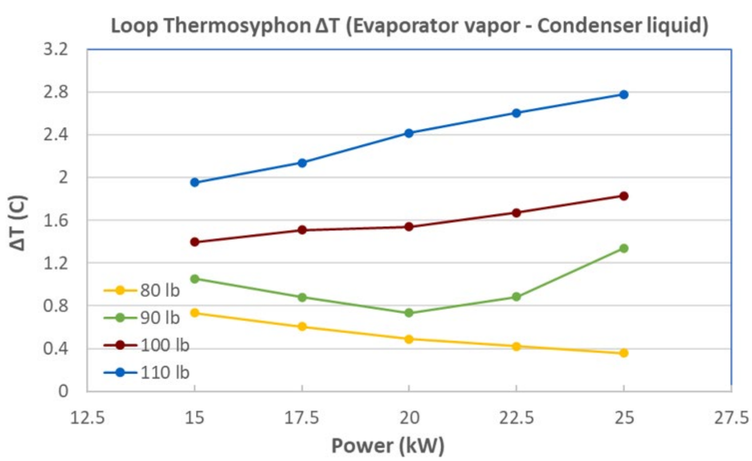


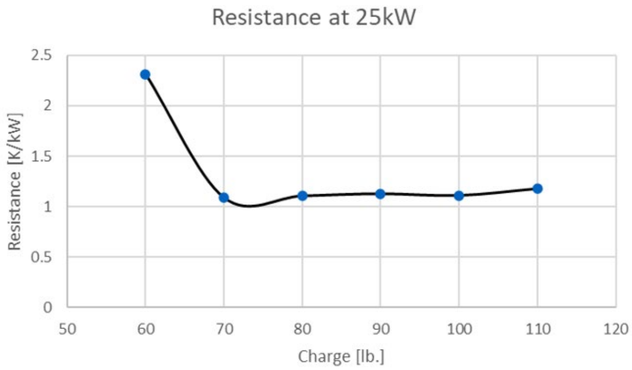
Figure 13. Loop thermosyphon ΔT vs. input power for different fluid charges.

4.2 Overall Thermal Resistance

The overall thermal performance is evaluated via the overall thermal resistance calculated via the following equation:

$$R_{overall} = (\text{Heater Temperature} - \text{Wet Bulb Temperature}) / \text{Input Power} \quad (4)$$

The input power is calculated from the calorimetry of the hot water loop. Figure 15 shows the overall thermal resistance for different charges at 25 kW heater input. At 60 lb charge, the evaporator has a significant dry out region at 25 kW power input, leading to high overall thermal resistance. At 70 lb, while the evaporator outlet temperature shows superheated vapor in the vapor line, the thermal resistance is lower compared to other charges. After 70 lb, further increasing charge shows a slight increase of the overall thermal resistance (Figure 14).



Charge (lb)	Dry (°C)	Humidity (%)	Wet (°C)
60	7	60	3.3
70	19	50	12.8
80	12	44	6.6
90	20	40	12.3
100	22.8	73	19.3
110	22.7	31.1	12.3

Figure 14. Left: Overall thermal resistance vs. fluid charge. Right: Wet bulb temperature at different charge tests.

5. CONCLUSION

The feasibility of a passive condenser loop has been demonstrated for refrigerants with steep saturation curves under the normal operational temperature via a large-scale loop thermosyphon experiment. The system was tested at six different fluid charges (R134a, from 60 lb to 110 lb at a 10 lb increment) and at 5 different powers (15 kW to 25 kW at a 2.5 kW increment). The results validate the loop is able to deliver a large power (25 kW) over a long distance (40 feet vertical + 10 feet horizontal) with a minimum ΔT (< 0.4°C).

At 60 lb charge, dry out in the evaporator was observed resulting in a large overall thermal resistance. When the charge increases to 70 lb, no dry out in the evaporator as the overall resistance becomes minimal (Figure 14). However, an increased vapor line temperature was observed at 22.5 kW due to superheated vapor (Figure 8). When the charge increases to 80 lb, the superheated vapor is eliminated, but the overall resistance slightly higher than the 70 lb. When the charge increases to 90 lb, the ΔT of the loop thermosyphon increases at higher power (22.5 kW, Figure 13) due to the liquid head reaching to the condenser and therefore subcooled liquid. The ΔT increases with higher charges (100 lb and 110 lb) and higher input powers since more liquid is pushed into the condenser and therefore more subcooling. The overall thermal performance of the current setup however was observed less dependent from different charges due to much larger thermal resistances of the evaporator (between the recirculating hot water and the loop thermosyphon) and the condenser (between the evaporative condenser and the ambient).

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REFERENCES

[1] Frank Morrison, "Saving Energy with Cooling Towers," ASHRAE Journal, February 2014

[2] https://scout.energy.gov/baseline-energy-calculator.html

[3] https://www.epa.gov/sites/production/files/2017-01/documents/ws-commercial-factsheet-offices.pdf