Experimental Investigation of the Heat Transfer Performance of a Hybrid Cooling Fin Thermosyphon

Christina A. Pappas[1]
Department of Mechanical and Aerospace Engineering,
University of Virginia,
122 Engineer's Way,
Charlottesville, VA 22904
e-mail: caj5p@virginia.edu

Donald A. Jordan
Senior Research Scientist
Department of Mechanical and Aerospace Engineering,
University of Virginia,
122 Engineer's Way,
Charlottesville, VA 22904
e-mail: dj8n@virginia.edu

Pamela M. Norris
Professor
Fellow ASME
Department of Mechanical and Aerospace Engineering,
University of Virginia,
122 Engineer's Way,
Charlottesville, VA 22904
e-mail: pamela@virginia.edu

The effect of fill volume on the heat transfer performance of a hybrid cooling fin thermosyphon, characterized by an airfoil cross-sectional shape and a slot-shaped cavity, is investigated. The performance was examined at three fill volumes, expressed as a percentage of the evaporator section: 0%, 60%, and 240%. These were chosen to represent three distinct regimes: unfilled, filled, and overfilled evaporator sections, respectively. The cross section of this copper–water thermosyphon has a NACA0010 shape with a chord length of 63.5 mm and an aspect ratio (ratio of the length of the evaporator section to the cavity width) of 1.109. The evaporator length comprises 8.3% of the total thermosyphon length. The air-cooled condenser section was placed in a uniform air flow in the test section of an open return wind tunnel. The rate of heat transfer, or performance, was measured as a function of fill volume and evaporator temperature. The heat transfer performance increased by 100–170% by adding 0.86 ml of working fluid (de-ionized water), i.e., when the fill volume increased from 0% to 60%, which illustrates the improvement of a cooling fin's heat transfer rate by converting it to a hybrid cooling fin thermosyphon. Of the fill volumes investigated, the thermosyphon achieves a maximum heat transfer rate and highest average surface temperature at the 60% fill volume. Overfilling the evaporator section at 240% fill results in a slight decrease in performance from the 60% fill volume. The results of this study demonstrate the feasibility of hybridizing a cooling fin to act both as a cooling fin and a thermosyphon. [DOI: 10.1115/1.4028000]

Keywords: thermosyphon, cooling fin, airfoil, fill volume, small aspect ratio

1 Introduction

In many aerospace applications, such as commercial aircraft, there exists a need for the enhancement of air-cooled thermal management techniques. Cooling fins are often employed to enhance the heat transfer from a device to its surroundings. A potential way to improve the heat transfer rate of a cooling fin would be to transform it from a simple extended surface into a closed two-phase thermosyphon. This type of heat pipe is characterized by its very simple structure; it is a closed, evacuated vessel that is filled with a small amount of a working fluid. The section of the thermosyphon where heat input takes place is called the evaporator section, and the section where heat output takes place is called the condenser section. The flow of vaporized working fluid from the evaporator to the condenser and subsequent flow of the condensed fluid back to the evaporator creates the heat flow cycle. The effective thermal conductivity of a heat pipe made of copper can be 90 times greater than the thermal conductivity of a plain copper rod of the same size [1]. This implies that a properly functioning thermosyphon fabricated in the form of a cooling fin, assuming favorable orientation in the gravitational field, could enhance the thermal performance of that cooling fin.

Cylindrical copper–water thermosyphons (with cylindrical cavities) are frequently used in studies of thermosyphon performance, and the evaporator and condenser sections typically have similar lengths. The effect of fill volume (the amount of fluid in the thermosyphon cavity) on performance has also been investigated extensively, though in recent years studies have focused primarily on the effect of nanofluids [2–4]. Hung et al. [5], Lee and Mital [6], Negishi and Sawada [7], and Noie et al. [8] tested this effect for cylindrical copper–water thermosyphons, and all focus on fill volumes less than 100% of the evaporator section. One of the primary conclusions that may be drawn from these studies is that using small fill volumes risks dry out of the evaporator section. Also, all found that the rate of heat transfer increases as fill volume increases up to a certain point, which varies based on the study. The thermosyphons tested in these studies have evaporator lengths that range from 19% of the total thermosyphon length (Noie [8]) to 67% of the total length (Lee and Mital [6]). A general consensus is difficult to declare as each investigation examines the effect of evaporator length in a slightly different way. One conclusion that may be made, however, is that very short evaporator sections are not typically investigated and of the evaporator lengths investigated in these studies, none are smaller than 19% of the total thermosyphon length. In contrast to this, the length of the evaporator section here is 8.3% of the total length. The fundamental differences between the thermosyphons mentioned in the studies above and the one investigated here are the cross-sectional shape, the cavity shape, and the aspect ratio, and so these results may not inform the present study.

The use of thermosyphon technology in an air-cooled waste heat management scenario would require an aerodynamic shape and a very long condenser section, and an airfoil-shaped hybrid cooling fin thermosyphon would satisfy both these requirements. These two characteristics (the airfoil shape and short evaporator section/long condenser section) would make it unlike the vast majority of thermosyphons investigated in the literature. The plausibility and effectiveness of a cooling fin thermosyphon have been preliminarily investigated by Randolph [9], De Cecchis [10], and Pappas et al. [11]. In their cylindrical thermosyphon experiments with small evaporator sections, both Randolph and De Cecchis found that the thermosyphons exhibited much higher rates of heat transfer than an empty shell when in the presence of cooling air flow. However, the cylindrical thermosyphon does not lend itself to air-cooled applications due to the fact that nonstreamlined bodies in a flow produce losses that either require a larger fan or more power input for the same flow rate. Therefore, an airfoil-shaped thermosyphon is better suited to the application of an air-cooled cooling fin than the standard cylindrical thermosyphon.

[1]Corresponding author.

Contributed by the Heat Transfer Division of ASME for publication in the JOURNAL OF HEAT TRANSFER. Manuscript received August 5, 2013; final manuscript received July 2, 2014; published online July 29, 2014. Assoc. Editor: Patrick E. Phelan.

The primary goal of this work is to investigate the effect of fill volume and evaporator temperature on the rate of heat transfer (performance) of a low aspect ratio airfoil cooling fin thermosyphon. The scope of the work in this paper is as follows. An airfoil-shaped cooling fin having a slot-shaped cavity was designed and fabricated out of copper using standard Computer Numeric Control (CNC) machining techniques and wire electrical discharge machining (EDM). Next, the condenser section of the thermosyphon was installed in the test section of an open return wind tunnel while the evaporator section, which remained outside the test section, was heated with electrical cartridge heaters. Experiments were conducted to examine the heat transfer rate (as a function of evaporator temperature and fill volume) by measuring the output heat transfer rate, the temperature distribution over the surface of the condenser section, and the cavity pressure.

2 Experimental Details

Figure 1(*a*) is a diagram of the experimental setup for this study and is very slightly adapted from Pappas et al. [11]. The thermosyphon is placed in the test section of an open return wind tunnel to simulate an air-cooling heat sink. Its evaporator section is heated using a heater block containing electrical cartridge heaters (heat input) and its condenser section is convectively cooled (heat output) by the airflow in the test section. The measurement of the output rate of heat transfer is achieved by conducting every experiment at steady state, where the input rate of heat transfer is equivalent to the output rate of heat transfer (once any heat transfer from the heater block to the surroundings is subtracted out). It is difficult to measure the output heat transfer rate directly because the heat is transferred to the airstream whose temperature increases only marginally. Therefore, the input heat transfer rate is what is physically measured in order to determine the thermosyphon output heat transfer rate.

2.1 Design, Fabrication, and Setup. The cooling fin thermosyphon assembly (shown in Fig. 1(*b*)) has three components fabricated separately out of solid pieces of ultraconductive copper (alloy 101) and will henceforth be referred to as the top end cap, the main body or shell, and the bottom end cap. The main body has a NACA0010 airfoil-shaped cross section with a slot-shaped cavity. An illustration of the cross section, including dimensions, is displayed in Fig. 1(*c*). The bottom end cap is simply a block designed to act as both an end cap and the means by which the evaporator section is heated. The block had two 6.35 mm holes drilled through the sides where two cylindrical 200 W electrical cartridge heaters are positioned to deliver thermal energy to the evaporator section of the thermosyphon. The block was heavily insulated to minimize heat transfer from the block to its surroundings. The top end cap serves to connect the thermosyphon to instrumentation and the rest of the experimental setup. These end caps were fabricated using standard CNC machining techniques, and the shell was fabricated using wire EDM. The end caps were attached to the main body of the thermosyphon using a metal-filled repair epoxy. After the thermosyphon components were assembled, the setup was hydrostatically pressure-tested up to an internal pressure of 827 kPa (120 psig) with no sign of structural failure.

The top end cap was fitted with two stainless steel tubes: one connects to a pressure transducer to allow for monitoring of the cavity pressure and the second connects to a tee that leads to the fluid reservoir and to the vacuum pump. The model of the vacuum pump is a Pfeiffer TSH-064D pumping station with an IKR-251 cold cathode gauge. In order to streamline the experimental process, the thermosyphon was evacuated and filled while it remained positioned in the test section of the wind tunnel. In light of the very small evaporator volume (1.43 ml), a 10 ml burette with a resolution of 0.05 ml was used as a fluid reservoir, and the working fluid investigated is de-ionized water.

2.2 Experiment. As mentioned previously, the primary quantity of interest in this study is the thermosyphon's output heat transfer rate. Prior to the assembly of the individual thermosyphon components, the input rate of heat transfer required to keep the bottom end cap alone at a constant temperature was measured. This heat transfer rate was defined as an offset and does not contribute to the total output rate of heat transfer, i.e., from the condenser to the air flow, and is therefore subtracted from the measured power supplied to the cartridge heaters. This was done for each evaporator temperature 121, 135, 149, 157, and 163 °C,

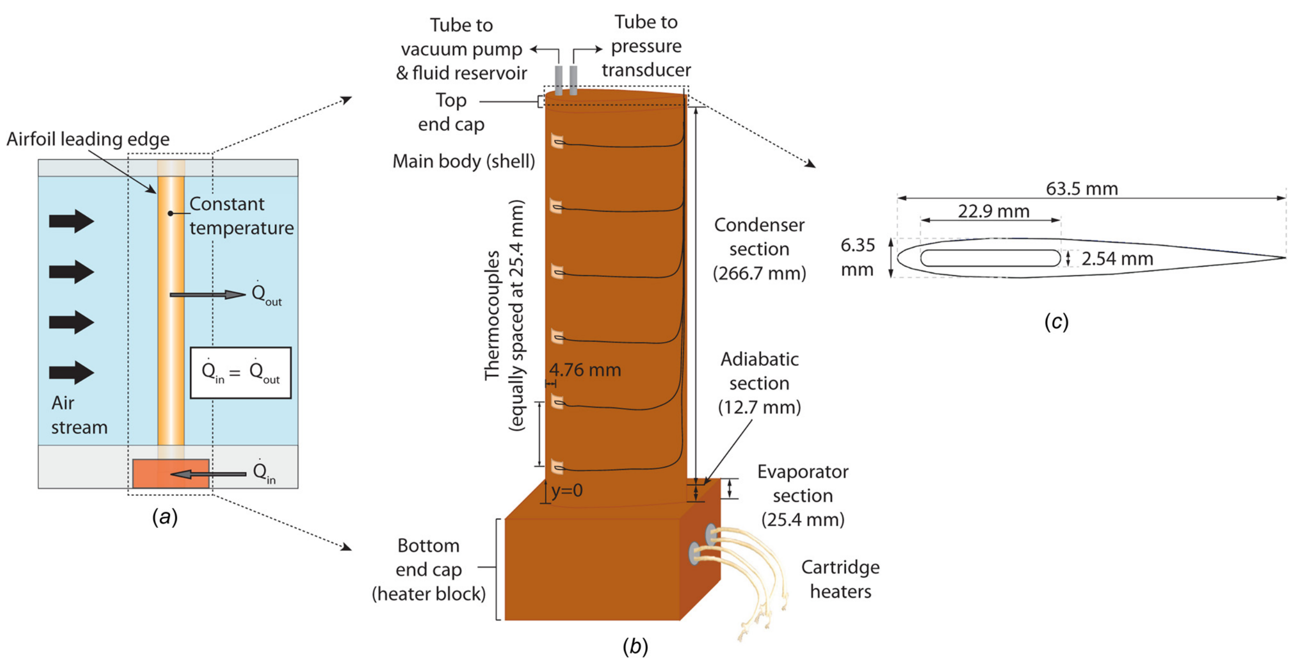

Fig. 1 (*a*) Schematic of the steady state experimental setup in the wind tunnel test section, (*b*) side view of the thermosyphon assembly with thermocouples attached, and (*c*) thermosyphon cross section with labeled dimensions

yielding offset values of 26.0, 27.4, 28.4, 30.0, and 31.6 W, respectively. The adjusted rate of heat transfer is equivalent to the output heat transfer rate from the condenser to the air stream at steady state. During each experiment, three quantities were measured and recorded: the input rate of heat transfer, the surface temperature along the length of the condenser section at several locations, and the cavity pressure.

Six OMEGA self-adhesive precision fine wire type-K thermocouples were placed along the length of the thermosyphon's condenser section and were spaced equal distances from each other. The locations of the thermocouples are illustrated in Fig. 1(*b*). The surface was covered in foil tape to ensure that the thermocouples did not get pulled off by the air flow and also to preserve the aerodynamic shape of the thermosyphon. For data acquisition, the thermocouples were connected to an OMEGA TempScan/1100 high speed temperature measurement system with an OMB-TEMPTC-32B thermocouple scanning module installed. Data acquisition was performed with the use of the program Chartview, which was installed on a computer used throughout the experiments. The same type of thermocouple was also used to measure the temperature of the heater block, which was attached to the top of the block. In order to estimate the possible discrepancy between the evaporator temperature and the temperature measured by the thermocouple at the surface of the heater block, the evaporator temperature was calculated using the standard expression for cylindrically symmetric steady state conduction from the cartridge heaters to the evaporator section as outlined in Incropera and De Witt [12]. The inputs for this calculation included the measured power supplied to the cartridge heaters (offset removed), the temperature of the top of the heater block, the thermal conductivity of copper and the high thermal conductivity paste between the cartridge heater and the copper block. In every case, the approximated evaporator temperature was within the error of the temperature as measured by the thermocouple placed on top of the heater block.

The power delivered to the cartridge heaters (equivalent to the sum of the output heat transfer rate and the heat transfer rate from the insulated block to the atmosphere), or the input rate of heat transfer, was manually controlled using a model 3PN1210B Staco Energy variable autotransformer (commonly referred to as a Variac) and was measured using a Power Monitors, Inc., Eagle 120 recorder. The Eagle 120 was connected to the data acquisition computer using a USB port, and data acquisition was performed using the program ProVision.

For a given experiment, the thermosyphon heat transfer rate was measured at five evaporator temperatures (121, 135, 149, 157, and 163 °C) and three fill volumes (0%, 60%, and 240% of the evaporator volume or pool height of 0 mm, 15.2 mm, and 61.0 mm). These particular fill volumes were chosen to represent three regimes: an unfilled evaporator section (control case), an underfilled evaporator section, and an overfilled evaporator section. These fill volumes were measured at room temperature prior to the beginning of each set of experiments. The wind tunnel frequency was set to a value such that the wind speed in the test section, as measured by Pitot-static tube, yielded a reading of approximately 44.7 m/s (100 mph). This air speed was chosen as it is within the range of jet engine bypass air flow speeds in a commercial aircraft, and it was kept constant throughout the experiments. The ambient temperature of the airflow through the test section remained approximately constant at 23 °C. It took approximately 2 h for the evaporator temperature to reach a steady temperature of 121 °C (the lowest evaporator temperature tested). When the heater block thermocouple read a temperature that did not vary by more than 1 deg Fahrenheit for several minutes, it was concluded that steady state was achieved. At this point, the surface temperatures and the input power were measured for 10 min at a sampling rate of 1 Hz and were averaged over that time. The time required to reach steady state at each subsequent evaporator temperature was approximately 20 min.

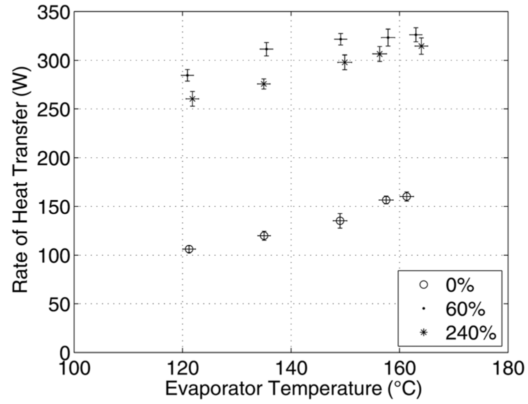

Fig. 2 The rate of heat transfer as a function of evaporator temperature for the 0%, 60%, and 240% fill volumes

3 Experimental Results

The rate of heat transfer from the thermosyphon to the cooling airstream as a function of fill volume for each of the five evaporator temperatures tested is displayed in Fig. 2. The total error (including repeatability and propagation of error) associated with the rate of heat transfer is approximately ±5 W. Similarly, the total error associated with each temperature measurement for this experiment (evaporator temperature, surface temperatures, and air temperature) is approximately ±2 °C. The rate of heat transfer increases approximately linearly with evaporator temperature for all fill volumes. At 0% fill volume, the thermosyphon is effectively a cooling fin. With no working fluid, the heat transfer rate of the thermosyphon is the lowest of all fill volumes tested and is considered the baseline level of performance. A marked increase in heat transfer rate is observed upon the addition of 0.86 ml of fluid into the cavity, i.e., the transition of fill volume from 0% to 60% of the evaporator section. Depending on the evaporator temperature, this increase in heat transfer rate ranges from 100% to 170%. The rate of heat transfer decreases slightly at all evaporator temperatures when the fill volume transitions from 60% to 240% of the evaporator volume. The evaporator section is overfilled at the 240% fill volume so that the working fluid is heated mostly from the bottom, as a pot of water is heated on a stove, which may explain the decreased performance when compared to the 60% fill

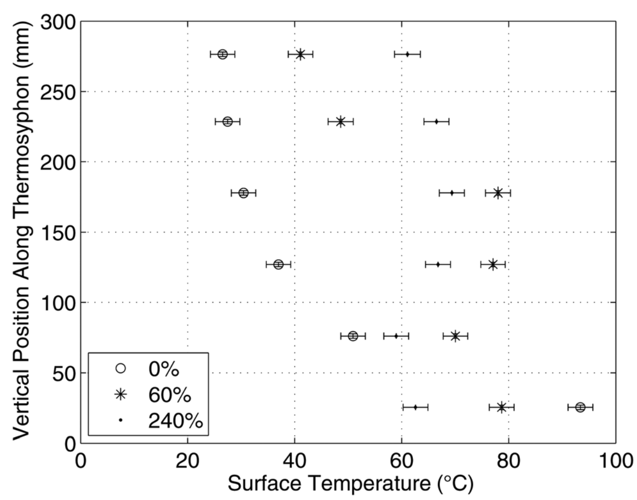

Fig. 3 The surface temperature as measured by thermocouples placed along the length of the condenser section of the thermosyphon for the 0%, 60%, and 240% fill volumes when the evaporator temperature is 157 °C

Table 1 Average surface temperature, measured vapor pressure, and estimated saturated temperature for 60% and 240% fill volumes

	60% Fill volume			240% Fill volume		
Evaporator temperature (°C)	Average surface temperature (°C)	Measured pressure (kPa)	Estimated saturated temperature (°C)	Average surface temperature (°C)	Measured pressure (kPa)	Estimated saturated temperature (°C)
121	60.56	45.16	78.89	59.77	36.06	73.47
135	63.71	63.57	87.48	62.13	42.06	77.15
149	65.12	79.57	93.39	64.15	48.68	80.73
157	65.58	89.15	96.46	64.21	49.78	81.28
163	65.79	95.29	98.29	64.29	50.54	81.66

volume. The overfilled configuration has less direct contact between the working fluid and the heated walls of the evaporator section than the 60% case. This leads to a lesser degree of vaporization due to the fact that a smaller fraction of the working fluid is in direct contact with the evaporator walls.

Figure 3 displays the surface temperature distribution along the length of the thermosyphon's condenser section for each fill volume at a representative evaporator temperature of 157 °C. The total error associated with the measurement of the vertical position of the thermocouple along the length of the condenser section is approximately ±1.6 mm. The surface temperature distribution for the 0% fill volume is distinctly nonisothermal. This may be because the primary mode of heat transfer in this case is conduction up the walls from the evaporator section. This temperature distribution is characterized by very high temperatures near the evaporator section and very low temperatures near the top of the condenser section. The rapid decrease of temperature along the length of the thermosyphon results in a lower average temperature which yields lower rates of heat transfer into the cooling air stream. In contrast to this, the surface temperature distribution for the thermosyphon at 60% fill volume is much more isothermal. The exhibition of the more isothermal surface temperatures implies that typical thermosyphon behavior is occurring. It should be noted, however, that there is a significant drop in temperature near the top of the condenser. This may be due to all the vaporized working fluid condensing prematurely, i.e., prior to reaching the top of the condenser section, on the thermosyphon walls as a result of the long condenser section and very small amount of working fluid. Finally, at the 240% fill volume, the surface temperature distribution is more isothermal than the 60% case. The average temperature is lower for the 240% fill volume than it is for the 60% fill volume, which is consistent with the lower rate of heat transfer observed for this case in Fig. 2.

Table 1 displays the internal pressures experienced by the thermosyphon during each experiment for the 60% and 240% fill volumes and also the corresponding average surface temperature. The reported pressures have a total error of approximately ±10 kPa. The highest pressures were experienced at the 60% fill volume, though the internal pressure never reached atmospheric pressure at any fill volume. The internal pressure increases with evaporator temperature for both the 60% and 240% fill volumes. Given the lack of instrumentation of the thermosyphon cavity, the internal temperature is estimated using the Antoine equation [13] and the measured internal pressure. This equation is a derived form of the Clausius–Clapeyron equation and relates the vapor pressure of a pure substance to its temperature. The resulting estimated internal temperatures are also displayed in Table 1. For every evaporator temperature, the estimated internal temperature is greater than the corresponding average surface temperature. This is expected due to the presence of two resistances to heat transfer to the environment: the thin layer of condensed water on the walls and the copper walls themselves.

4 Conclusions

The effect of fill volume on the heat transfer rate of a copper–water cooling fin thermosyphon with a slot-shaped cavity is investigated. The performance of this air-cooled thermosyphon was measured at three fill volumes representing three distinct regimes expressed as a percentage of the evaporator volume: 0% (unfilled), 60% (filled), and 240% (overfilled). The cross section of the thermosyphon has an airfoil (NACA0010) shape and an aspect ratio, defined as the ratio of the evaporator section length to the total cavity width, of 1.109. The rate of heat transfer was also measured as a function of evaporator temperature in the test section of an open return wind tunnel, where the airflow had a constant speed of 44.7 m/s and temperature of 23 °C. The five evaporator temperatures at which the rate of heat transfer was measured were 121, 135, 149, 157, and 163 °C. With no working fluid, the cooling fin's heat transfer rate is the baseline case in terms of performance and reflects the lack of thermosyphon action. The rate of heat transfer increased by 100–170% (depending on the particular evaporator temperature) when the fill volume increased from 0% to 60%, which is equivalent to adding 0.86 ml of working fluid. For the fill volumes investigated, the cooling fin thermosyphon achieves the highest rate of heat transfer and also the highest average surface temperature along the condenser section at 60% fill volume. This occurs because the fraction of working fluid in direct contact with the evaporator section walls is largest at the 60% fill volume. At 240% fill volume, the evaporator section is overfilled and results in a slight performance decrease when compared to the 60% fill volume. These results demonstrate the feasibility of the use of an airfoil-shaped cooling fin thermosyphon for air-cooled thermal management applications.

Acknowledgment

Christina Pappas was supported by the U.S. Department of Defense (DoD) through the National Defense Science & Engineering Graduate Fellowship (NDSEG) Program and the Virginia Space Grant Consortium.

References

[1] Faghri, A., 1995, *Heat Pipe Science and Technology*, Taylor & Francis, Washington, DC.

[2] Huminic, G., Huminic, A., Morjan, I., and Dumitrache, F., 2011, "Experimental Study of the Thermal Performance of Thermosyphon Heat Pipe Using Iron Oxide Nanoparticles," Int. J. Heat Mass Transfer, **54**(1), pp. 656–661.

[3] Shanbedi, M., Heris, S. Z., Baniadam, M., and Amiri, A., 2013, "The Effect of Multi-Walled Carbon Nanotube/Water Nanofluid on Thermal Performance of a Two-Phase Closed Thermosyphon," Exp. Heat Transfer, **26**(1), pp. 26–40.

[4] Sureshkumar, R., Mohideen, S., and Nethaji, N., 2013, "Heat Transfer Characteristics of Nanofluids in Heat Pipes: A Review," Renewable Sustainable Energy Rev., **20**, pp. 397–410.

[5] Hung, N., Groll, M., and Thong, D., 1982, "Experimental Investigation of Closed Two-Phase Thermosyphons," J. Energy, **6**(4), pp. 283–284.

[6] Lee, Y., and Mital, U., 1972, "A Two-Phase Closed Thermosyphon," Int. J. Heat Mass Transfer, **15**(9), pp. 1695–1707.

[7] Negishi, K., and Sawada, T., 1983, "Heat Transfer Performance of an Inclined Two-Phase Closed Thermosyphon," Int. J. Heat Mass Transfer, **26**, pp. 1207–1213.

[8] Noie, S. H., Emami, M. R. S., and Khoshnoodi, M., 2007, "Effect of Inclination Angle and Filling Ratio on Thermal Performance of a Two-Phase Closed Thermosyphon Under Normal Operating Conditions," Heat Transfer Eng., **28**, pp. 365–371.

[9] Randolph, T. L., 2008, "Thermosyphon Technology for Heat Management in High-Bypass Jet Engines Aboard Next Generation Boeing 737 Aircraft," M.S. thesis, University of Virginia, Charlottesville, VA.

[10] De Cecchis, P. M., 2010, "Investigation of the Effect of Fill Volume on Heat Transfer From Air-Cooled Low Aspect Ratio Thermosyphons," M.S. thesis, University of Virginia, Charlottesville, VA.

[11] Pappas, C., De Cecchis, P., Jordan, D., and Norris, P., 2013, "The Effect of Fill Volume on Heat Transfer From Air-Cooled Thermosyphons," ASME J. Heat Transfer, **135**(4), p. 044504.

[12] Incropera, F., and De Witt, D., 2002, *Fundamentals of Heat and Mass Transfer*, 5th ed., Wiley, New York.

[13] Poling, B. E., Prausnitz, J. M., and O'Connell, J. P., 2001, *The Properties of Gases and Liquids*, 5th ed., McGraw-Hill, New York.