EXPERIMENTAL INVESTIGATION OF AIRFOIL THERMOSYPHONS FOR THERMAL MANAGEMENT OF NEXT GENERATION AIRCRAFT

Christina A. Pappas
Department of Mechanical and Aerospace Engineering
University of Virginia, Charlottesville, VA
Advisor: Pamela Norris

Abstract

The effect of fill volume and evaporator temperature on the heat transfer performance, i.e. rate of heat transfer, of airfoil-shaped thermosyphons for cooling applications is investigated. Two copper-water thermosyphons, one with a cylindrical cavity and the other with a slot-shaped cavity, were fabricated for testing. The rate of heat transfer for both thermosyphons was measured at three fill volumes (expressed as a percentage of the volume of the evaporator section): 0%, 60%, and 240% and for five evaporator temperatures: 250°F, 275°F, 300°F, 315°F, and 325°F. The condenser section was air-cooled in an open return wind tunnel with wind speeds of 100 mph in the test section and an ambient temperature of 73°F. The difference in heat transfer rate between the 0% and 60% fill volumes for the slot is approximately 200 W at all evaporator temperatures, while for the cylinder thermosyphon the difference is approximately 25 W. The rate of heat transfer was highest for both thermosyphons at 60% fill volume at the 325°F evaporator temperature. As fluid is added to the thermosyphons, the surface temperature rises and the temperature distribution becomes more isothermal, which is indicative of thermosyphon action taking place and explains the observed increase in performance.

Introduction

Boeing, the aerospace company and aircraft manufacturer, is continually exploring approaches to improve existing aircraft technology. One recent initiative is to upgrade aircraft auxiliary systems, such as cabin air conditioning, to run on electrical power. This will require replacing the current two-generator configuration with four more powerful electric generators. These added power sources will increase the amount of waste heat carried by the generator oil loop that must be expelled from the aircraft. At present, one small brick-shaped heat exchanger per jet engine is used for this heat transfer as shown in Fig. 1.

This heat exchanger is positioned inside the jet engine and down-wind from flow straighteners called outlet guide vanes (OGVs) that straighten the bypass airflow to transfer...
heat into the airstream via forced convection. The geometry and size of this heat exchanger make this technology inadequate for Boeing’s intended upgrades for two reasons: limited ability to dissipate excess heat and decreased engine thrust if scaled to a larger size.

Prior work has determined that the most viable solution to this thermal management problem involves heat pipe technology, specifically wickless heat pipes called thermosyphons. The internal operation of a closed two-phase thermosyphon is illustrated in Fig. 2. This type of thermosyphon is a closed, evacuated vessel containing a particular amount of fluid, and its primary function is to rapidly transfer heat from one location to another via latent heat of vaporization. At the heat source, the fluid vaporizes and moves rapidly toward the heat sink where it re-condenses and then is returned to the evaporator with the aid of gravity.

Therefore, the OGVs, whose current function is to straighten the flow of bypass air, will be hybridized to function as both flow straighteners and thermosyphons.

To simulate operating conditions, the condenser section of the thermosyphon is air-cooled in the test section of a wind tunnel. The conceptual idea for testing is shown in Fig. 3. The thermosyphon performance is defined as its output rate of heat transfer. When the experiment has settled to steady state, the rate of heat transfer to the air stream (RHT out) is equivalent to the electrical power put into the thermosyphon (RHT in). Therefore, the thermosyphon performance is ascertained by measuring the electrical power input under steady state conditions.

Methods

The ultimate goal of this work is to transition from previous work dealing with cylindrical-shaped thermosyphons to develop a design for an airfoil-shaped thermosyphon that may be utilized as a hybrid flow straightener and heat exchanger and to test its performance. Improvements to the design must be made until the rate of heat transfer is at a level such that an array of these thermosyphons will meet the heat dissipation
needs of Boeing’s new four-generator configuration.

**Construction**

Several iterations of finite element analysis (FEA) were performed using the COSMOS tool, which bundled with the solid modeling software SolidWorks, to predict the wall displacements for an airfoil-shaped shell, with varied cavity shapes and sizes, when the cavity is subjected to high internal pressures. Due to the size of the OGVs (10.5” span, 2.5” chord, and 0.25” maximum thickness), the cavity is necessarily small in size. Adequate space in the cavity is crucial to avoid entrainment of the condensate into the vapor flow upward so that the thermosyphon functions properly. Both the results of the FEA analysis and these entrainment considerations give opposing constraints on the design. With this in mind, two different cavity shapes, a cylinder (strong structure but small cross-sectional area) and a slot (weaker structure but larger cross-sectional area), were chosen to be appropriate for fabrication and subsequent testing. The cross-sections of the two airfoils are shown in Fig. 4 with their cavity shapes. The cross-sections are NACA0010 airfoils with a 2.5” chord length and 0.25” maximum thickness.

![Figure 4: Airfoil cross-section cavity shapes](image)

The airfoil bodies were constructed from solid pieces of copper and the internal cavities were created using a method called wire electrical discharge machining (EDM). The end caps were also made out of copper and were attached to the airfoil body using a metal-filled repair epoxy. The structural integrity of the complete thermosyphon assemblies was tested using a hydrostatic pressure test where the cavities were mostly filled with deionized water and then subjected to increasing internal pressure up 100 psig. This was done to ensure that the thermosyphons would not fail if the cavity pressurized at high temperatures. Any large leaks were sealed using more of the same epoxy, and smaller leaks were sealed using a high vacuum sealant spray.

**Experiment**

The evaporator section is electrically heated with cartridge heaters to simulate the generator oil loop heating the base of the thermosyphon, and the condenser section is exposed to the airflow in the test section of the wind tunnel to simulate the bypass airflow over the OGVs. By using a temperature controller to set a chosen heater temperature and waiting until the experiment settled to steady state, the energy dissipated into the air stream is measured from the electrical power input to the cartridge heaters. The surface temperature of the thermosyphon is measured by thermocouples placed along the length of thermosyphon at the location of the cavity. The internal pressure is measured using a pressure transducer attached to the top end cap of the thermosyphon.

An experimental setup was created to allow for both the evacuation of the cavity and filling the thermosyphon while it is positioned in the wind tunnel test section. An image of one of the thermosyphons placed in the test-section is shown in Fig. 5.
Both thermosyphons were tested with three different fill volumes: 0%, 60%, and 240%. These fill volumes are given as a percentage of the volume the evaporator section. For each fill volume, the thermosyphon’s rate of heat transfer was measured at five different evaporator (cartridge heater) temperatures: 250°F, 275°F, 300°F, 315°F, and 325°F, with the design point being 315°F.

**Results**

The rate of heat transfer for the thermosyphon with the cylinder cavity as a function of evaporator temperature is displayed in Fig. 6 for all three fill volumes. The 0% fill volume set of rate of heat transfer data ranges from approximately 75 to 125 W. The 60% fill volume data set barely outpaces the 0% fill volume data set, and the 240% fill volume data set performs or the same or worse than the 0% data set.

The surface temperatures were measured using thermocouples placed over the cavity and along the length of condenser section, beginning with one inch above the top of the evaporator section. A plot of the surface temperature as a function of this vertical position at the design point evaporator temperature of 315°F for the three fill volumes is displayed in Fig. 7. The 0% fill volume surface temperature data is characterized by very high temperatures near the evaporator section and much lower temperatures near the top of the condenser section. The 60% and 240% fill volume data sets exhibit a similar surface temperature pattern but do not achieve as a high temperature near the evaporator section.

The internal pressures (as measured by a pressure transducer) encountered during each experiment with the cylinder cavity thermosyphon are reported in Table 1. The
internal pressures remain approximately the same for all evaporator temperatures at the 0% fill volume. For the 60% and 240% fill volumes, the pressure increases with evaporator temperature but remains well below atmospheric pressure.

Table 1: Cylinder Thermosyphon Internal Pressures (psia)

<table>
<thead>
<tr>
<th>Evap. Temp.</th>
<th>Fill Volume</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0%</td>
</tr>
<tr>
<td>250°F</td>
<td>0.85±1.60</td>
</tr>
<tr>
<td>275°F</td>
<td>0.84±1.60</td>
</tr>
<tr>
<td>300°F</td>
<td>0.85±1.60</td>
</tr>
<tr>
<td>315°F</td>
<td>0.86±1.60</td>
</tr>
<tr>
<td>325°F</td>
<td>0.90±1.60</td>
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</tbody>
</table>

The rate of heat transfer for the thermosyphon with the slot-shaped cavity is plotted as a function of evaporator temperature in Fig. 7. The 0% fill volume set of rate of heat transfer data ranges from 100 to 150 W. The 60% and 240% fill volume data sets have rates of heat transfer that are over twice as large as those for when the thermosyphon contains no working fluid.

The surface temperature measured as a function of vertical position is plotted at the design point evaporator temperature of 315°F is displayed in Fig. 9. The surface temperature distribution for the 0% fill volume is distinctly non-isothermal. The 60% and 240% fill volume data sets are more isothermal than the 0% data set by a considerable amount. The 60% fill volume surface temperature exhibits a significant drop in temperature near the top of the condenser section. This phenomenon occurs at all evaporator temperatures, becoming more marked as the evaporator temperature increases.

The internal pressures encountered during the experiments with the slot-shaped cavity thermosyphon are reported in Table 2. The internal pressures remain approximately the same for all evaporator temperatures at the 0% fill volume. Both the 60% and 240% fill volumes have elevated pressures even for the lowest evaporator temperatures. For the 60% fill volume, the pressure increases with evaporator temperature and almost reaches atmospheric pressure.
Table 2: Slot Thermosyphon Internal Pressures (psia)

<table>
<thead>
<tr>
<th>Evap. Temp.</th>
<th>Fill Volume</th>
<th>0%</th>
<th>60%</th>
<th>240%</th>
</tr>
</thead>
<tbody>
<tr>
<td>250°F</td>
<td>0.53±1.60</td>
<td>6.55±1.67</td>
<td>5.23±1.65</td>
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<tr>
<td>275°F</td>
<td>0.59±1.60</td>
<td>9.22±1.71</td>
<td>6.10±1.67</td>
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<tr>
<td>300°F</td>
<td>0.63±1.60</td>
<td>11.54±1.74</td>
<td>7.06±1.68</td>
<td></td>
</tr>
<tr>
<td>315°F</td>
<td>0.70±1.60</td>
<td>12.93±1.76</td>
<td>7.22±1.68</td>
<td></td>
</tr>
<tr>
<td>325°F</td>
<td>0.38±1.60</td>
<td>13.82±1.78</td>
<td>7.33±1.68</td>
<td></td>
</tr>
</tbody>
</table>

**Discussion**

The rate of heat transfer for both thermosyphons when there is no water inside was by far the lowest and is considered the base level of performance. The slot thermosyphon, however, did exhibit a slightly higher heat transfer rate by approximately 25 W at every evaporator temperature for the 0% fill volume.

The surface temperature distribution for the 0% fill volume is distinctly non-isothermal for both thermosyphons, and this is due to the fact that the heat transfer is attributable to conduction up the walls from the evaporator section as opposed to heat transfer via the phase change of the working fluid when it is present. The rapid decrease of temperature along the length of the thermosyphons results in lower rates of heat transfer.

The 60% fill volume exhibited the best performance in terms of rate of heat transfer (and without regard to whether or not thermosyphon action is taking place) for all evaporator temperatures for both the cylinder and slot-shaped cavity thermosyphons. There is a particular characteristic of the slot thermosyphon that may explain the dramatically higher rate of heat transfer at this fill volume as compared to that of the cylinder thermosyphon; this is the larger surface area that is associated with the slot cavity geometry. The slot cavity provides much more surface area for the vaporized working fluid to condense on, which leads to a larger surface having a higher temperature and consequently more heat transfer. The smaller internal surface area associated with the cylindrical cavity thermosyphon may also explain why the rate of heat transfer at 60% fill volume barely outpaced the rate of heat transfer at 0% fill volume.

The surface temperature distribution for the slot thermosyphon at 60% fill volume is much more isothermal than that for the cylinder thermosyphon. The exhibition of approximately isothermal surface temperatures for the slot thermosyphon implies that thermosyphon action is occurring to some extent. There is a significant drop in temperature near the top of the condenser section, which may be attributed to all the vaporized working fluid condensing prior to reaching the top of the thermosyphon. The reason for this behavior may only be guessed at, as the instrumentation used in these experiments does not offer any insight into the internal thermosyphon behavior. In contrast to this, the cylinder thermosyphon’s surface temperature distribution at this fill volume has the same pattern as the 0% fill volume, and this similarity is indicative of the fact that the cylinder thermosyphon is behaving as a shell of copper material with no working fluid.

The 240% fill volume is the only fill volume where the evaporator section is overfilled, which may explain the decreased performance when compared to the 60% fill volume. When the evaporator section is overfilled, the working fluid is heated mostly from the bottom, as a pot of water is heated on a stove. The heat transfer rate at the 240% fill volume is slightly lower than that for the 60% fill volume for all evaporator temperatures for both thermosyphons. For the cylindrical cavity thermosyphon, the 240% fill volume data is more comparable to the 0% data, indicating that the primary heat transfer method is via conduction up through the thermosyphon walls. For the slot thermosyphon, the heat transfer rate for this fill volume is lower than that for the 60% data, but the difference is small.
As is the case for the 0% fill volume, the surface temperature distribution for the 240% fill volume is clearly non-isothermal for the cylinder thermosyphon. In contrast to this, the surface temperature distribution at this fill volume for the slot thermosyphon is approximately constant along the length of the thermosyphon. This exhibition of an isothermal surface temperature implies that typical thermosyphon behavior is taking place. The average temperature is lower for the 240% fill volume than it is for the 60% fill volume, which is also revealed in the lower rate of heat transfer for both thermosyphons.

Conclusions
Two airfoil-shaped thermosyphons were designed, fabricated, and tested to investigate their performance (rate of heat transfer) for cooling applications. Both thermosyphons, one with a cylindrical cavity and the other with a slot-shaped cavity, were made of copper, and the working fluid used was deionized water. The fill volumes (as a percentage of the volume of the evaporator section) tested were 0%, 60%, and 240%. The condenser section was air-cooled in the test section of a wind tunnel with wind speeds of 100 mph and subjected to a range of evaporator temperatures (250°F, 275°F, 300°F, 315°F, and 325°F). With no working fluid, the heat transfer rate for both thermosyphons is the lowest of all fill volumes tested and is considered the base level of performance. The rate of heat transfer was highest for both thermosyphons at 60% fill volume. For the slot thermosyphon, the rate of heat transfer increased significantly after the addition of some working fluid, which was not the case for the cylinder thermosyphon. The difference in heat transfer rate between the 0% and 60% fill volumes for the slot is approximately 200 W at all evaporator temperatures, while for the cylinder thermosyphon the difference is approximately 25 W. At 240% fill volume, the heat transfer rate for both thermosyphons was lower than that at 60% fill volume. The rate of heat transfer increased approximately linearly with evaporator temperature for both thermosyphons. Therefore, the highest heat transfer rates were achieved an evaporator temperature of 325°F for all fill volumes.

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