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Electric Aircraft Thermal Management Using a Two-Phase Heat Transport System with Solid-State Thermal Switching Capability

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Advanced Cooling Technologies, Inc. (ACT) is collaborating with NASA Glenn Research Center (GRC) to develop a heat pipe-based thermal delivery system to efficiently manage the waste heat generated onboard an electric aircraft. The heat pipe system will interface with NASA GRC's thermoacoustic heat pump in order to recycle waste heat by transporting thermal energy to various end users onboard the aircraft. This paper discusses the layout of theoretical heat pipe networks for a MW-class commercial electric aircraft. This is followed by a discussion of progress made on the development of a novel two-phase heat transport system with solid-state thermal switching and control capabilities.

I. Introduction

Electrified Aircraft Propulsion (EAP) systems utilize electrical motors to provide some or all the thrust for an aircraft. Potential benefits of EAP include (1) low carbon emission (2) low noise (3) reduced fuel consumption (4) low operating cost and (5) high performance. NASA is currently leading the development of EAP technology in U.S. The near-term goal is to realize a MW-class, commercial electric aircraft by 2035. To fulfill this vision, an advanced thermal management technology that can handle (i.e., recycle, deliver and reject) all low-grade waste heat (~kW) generated within an aircraft is needed. Dyson[1] at NASA Glenn Research Center (GRC) is developing a novel system called "Thermal Recovery Energy Efficient System" (TREES), which uses thermo-acoustic heat engines to generate acoustic mechanical energy, distributes the acoustic wave via multiple acoustic tubes to acoustic heat pumps, where the low-grade waste heat is recovered and elevated to a high temperature.

As part of a Phase III NASA SBIR, Advanced Cooling Technologies, Inc. (ACT) is developing a heat pipe-based heat delivery system to integrate with the acoustic heat pump and deliver the high-grade waste heat from the acoustic heat pump back to various locations on the airplane requiring thermal energy (e.g., heating combustion air, wing de-icing etc.). These heat sinks will be referred to as "end users". Heat pipes represent a solution that fits the challenges encountered by the overall thermal management system. Heat pipes through their passive and two-phase nature of operation can provide thermal transport at large rates over large distances with minimal temperature drop. The passive solid-state nature of heat pipes also makes them a light-weight and reliable system ideal of aircraft applications.

This paper will discuss the following aspects of the work conducted under this SBIR program to develop a solid-state heat pipe-based heat delivery system for electric aircraft thermal management.

1. Selection of a representative hybrid electric aircraft and development of heat transfer networks linking various heat sources to the end users requiring thermal energy. The different heat transfer networks are grouped based on the required temperatures and locations of the end users.

2. Progress in the development and experimental demonstration of a two-phase heat transport system with solid-state thermal switching capabilities.

II. Heat Pipe Thermal Delivery System

A. Electric Aircraft Selection

The aircraft chosen as surrogate for this program was the Single-aisle Turboelectric AiRCraft with Aft Boundary Layer propulsion (STARC-ABL) developed by NASA in 2017.[2] The STARC-ABL is a 150-passenger class commercial transport with a traditional "tube-and-wing" shape. The plane relies on turboelectric propulsion, meaning that it uses electric motors powered by onboard gas turbines. As Figure 1 shows, the aircraft has two traditional jet engines that are mounted under the wings, which are coupled to electric generators for power generation. Electrical power is sent to the tail of the aircraft, where an all-electric propulsion takes advantage of an aerodynamic benefit known as boundary layer ingestion (BLI).

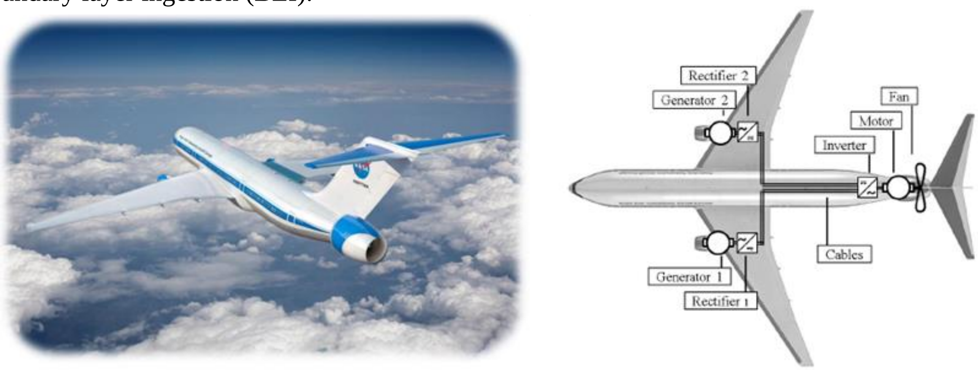

Figure 1. STARC-ABL Conceptual Design and the Notional Power System

B. Heat Sources and End Users

Based on an extensive literature survey,[2,3,4,5,6] ACT identified the heat sources and heat end users of this type of aircraft. Dominant heat sources on the airplane include power system, circuit breakers, cabin cooling and others (avionics, actuation, gearboxes etc.). Their power, temperature and locations are summarized in Table 1. ACT also identified potential heat end users (systems onboard the aircraft that benefit from thermal energy). The temperature and location information of the end users are listed in Table 2.

Table 1. Non-Exhaustive List of Heat Sources of the Electric Aircraft

Components	Qty	Power	Max. Allowable Temp.	Locations
Generator	2	50kW each	<100°C	Under wings
Rectifiers	2	15kW each	<100°C	Under wings
Circuit Breakers	4	7kW each	<100°C	Along cables (two behind rectifiers and two in front of inverter)
Inverter	1	25 kW	<100°C	Near tailcone
Motor	1	100 kW	<100°C	Near tailcone
Cabin Cooling	1	10 kW	< 30°C	Bottom
Others	n/a	2 kW	<100°C	Bottom

Table 2. Non-Exhaustive List of End Users on Electric Aircraft

Components	Temperature	Locations
Preheat Combustion Air	>600°C	Near Turbofans
Preheat Air into Auxiliary Power Unit (APU)	> 400°C	Near Tailcone
Anti-icing System	> 20°C	Wing, engine cowlings and tail flight control surfaces
Warm by-pass Air for Thrust Enhancement	> 50°C	Near Turbofans
Warm the Ram Air	> 30°C	Near Turbofans

C. Heat Pipe Networks

According to the temperature and relative locations of the heat sources and end users, three groups of heat pipe heat delivery systems are shown in Figure 2. The first group is located at the tailcone. Waste heat generated from motor, inverter and circuit breakers can be transferred by low temperature heat pipes to the cold end of the acoustic heat pump. Another set of low temperature heat pipes with thermal switching capability (i.e., multi-condenser VCHP described in Section III) will be used to distribute heat from the hot end of the acoustic heat pump to the outer mold line of the tail for de-icing purposes. The remaining waste heat can be dumped into exhaust stream. A high temperature heat pipe will be used to recycle the high-grade heat at the hot side of heat pump to the APU or combustion air. Similar configurations are shown in the other two groups of heat pipe systems to handle underwing heat sources and bottom heat sources.

The working fluid of the heat pipes will be based on the required temperature range, power and considering the envelope material. Potential working fluids for the low temperature heat pipes are toluene, methanol, ammonia or water. Water-based heat pipes can transfer more power but freezing of working fluid and start-up could be a challenge. Ammonia heat pipes have a maximum temperature limit of approximately 100°C. Toluene heat pipes have a much wider operating temperature range but less power can be carried. Ammonia and toluene are both compatible with aluminum envelopes. Methanol can carry more power than toluene and has a similar temperature range but is only compatible with stainless steel and copper envelopes, resulting in mass penalties. The high temperature heat pipes will use an alkali metal such cesium, potassium or sodium. These high temperature fluids are compatible with stainless steel, Inconel and Haynes 230.

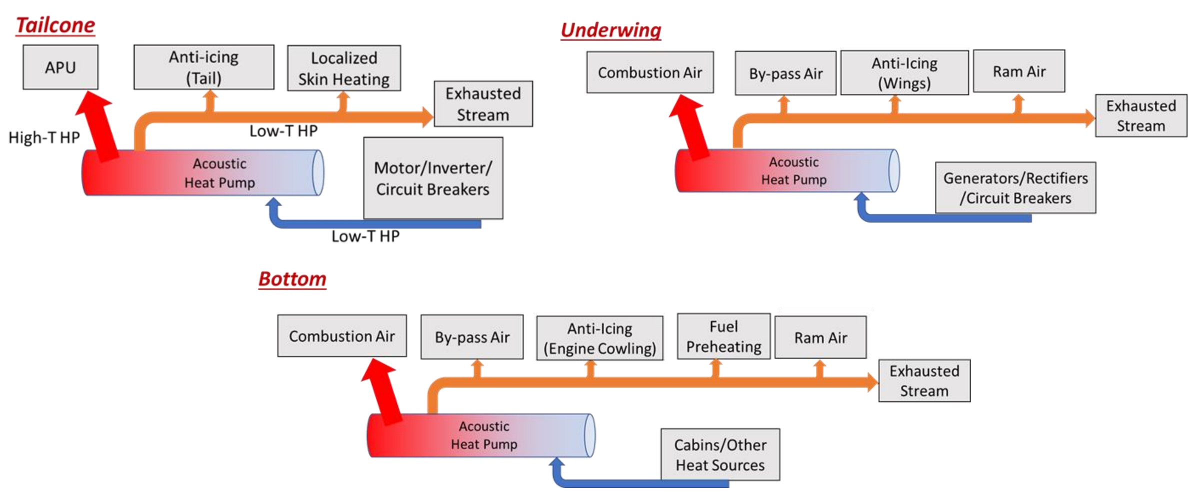

Figure 2. Three groups of heat pipe heat delivery systems for the electric aircraft.

The elevation of the end users is higher than the end sources, which is favorable for heat pipe design because the heat pipes will operate as thermosyphons (gravity-assisted heat pipe). In a typical heat pipe, liquid is passively returned from the condenser to the evaporator through capillary pumping of a wick structure. When the heat source is below the heat sink gravity can be used to return the liquid to the evaporator. Thermosyphons are capable of transporting large amounts of thermal energy over long distances.

Figure 3 shows a detailed layout of the high-temperature and low-temperature heat pipe delivery system for the tailcone. A high-temperature heat pipe is used to preheat air entering the APU while a system of low-temperature heat pipes distributes waste heat along the tail for deicing or dump the waste heat into the exhaust stream.

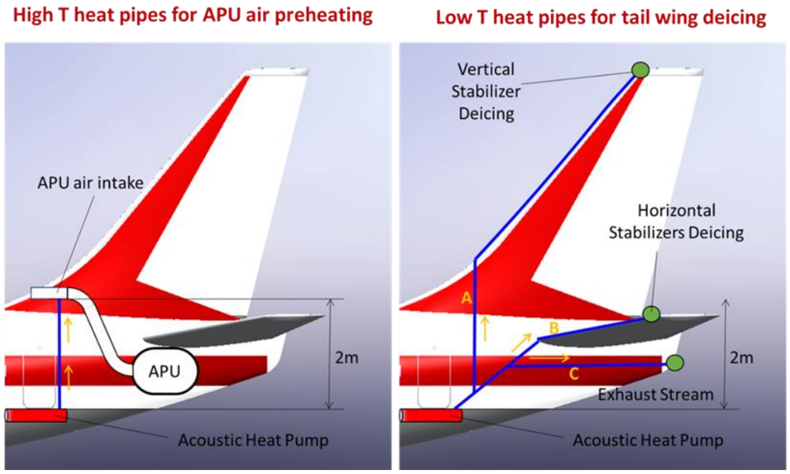

Figure 3. Tailcone heat pipe layout (a) high temperature heat pipe for APU preheating (b) low temperature with multiple condensers for tail wing deicing

Figure 4 illustrates the high temperature heat pipe for combustion air pre-heating operating under the wing. As the figure shows, the heat pipe is gravity-aided with 5.5° inclination. The operating temperature for this heat pipe should be higher than 600C. For a better view on pipe sizing, the power capability of alkali metal (sodium and potassium) heat pipes with different diameters are plotted in Figure 5. A 2-inch outer diameter (OD) heat pipe charged with sodium can carry around 10kW of heat. Note that the power capability calculation shown in Figure 5 are the flooding limits[7] of regular thermosyphon with 5.5° inclined angle. As liquid flows down the walls of the thermosyphon it experiences a shear force from the rising vapor. As the heat flux increases the shear force eventually becomes large enough that it begins impeding the return flow of the falling liquid. This can result in a flooding of the condenser and dry-out of the evaporator. Flooding limits of the thermosyphon is governed by ID, working fluid and operating temperature.

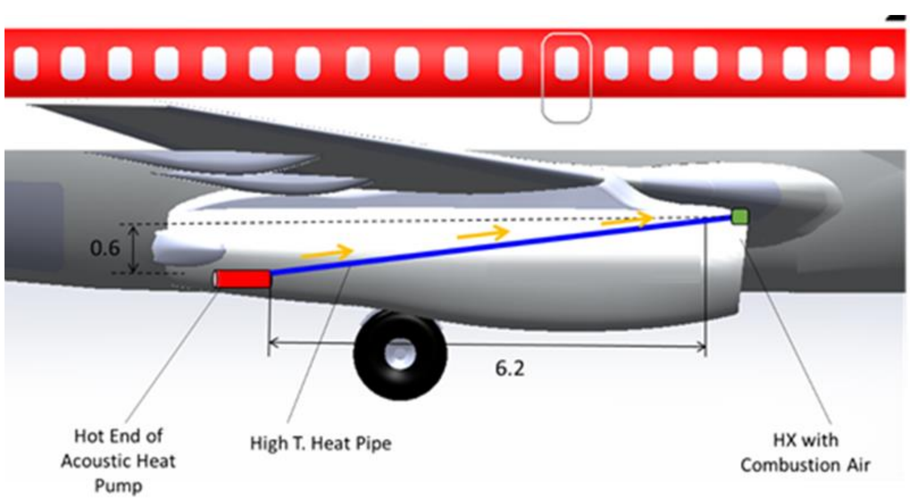

Figure 4. High temperature heat pipe to preheat the combustion Air (blue line represents the heat pipe, yellow arrows show heat transfer direction)

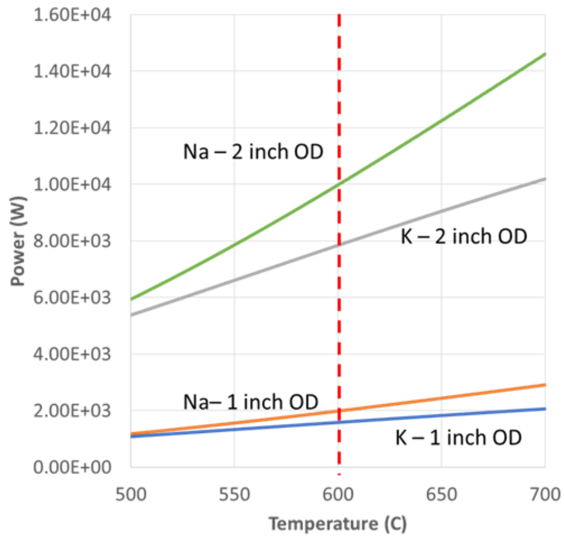

Figure 5. Power capability of the high temperature heat pipes for combustion preheating

III. Two-Phase Heat Transport System with Solid-State Thermal Switching Capability

A. Multi-Condenser Variable-Conductance Heat Pipe Concept

The proposed heat pipe heat delivery networks shown in Figure 2, shows thermal energy being transported from a single source to multiple heat sinks or “end users”. For example, waste heat under the wing can be carried from the acoustic heat pump and used to warm by-pass air, for anti-icing and warming ram air. Depending on the particular end user and phase of flight it may be desirable to block or reduce heat transfer to a particular end user. In order to minimize the number of heat pipes required to transport this power and provide sufficient control, ACT developed a two-phase heat transport system with solid-state thermal switching/control capabilities based on a *variable conductance heat pipe* (VCHP). A VCHP contains a reservoir of non-condensable gas (NCG). The NCG volume expands and contracts due to changes in the thermal conditions (vapor temperature and sink temperature). As the NCG expands into the condenser, the area available for heat transfer is reduced. VCHPs are typically used when it is desirable for the system to passively adjust to minimize the influence of the external thermal conditions on the vapor temperature. It is also possible to actively control VCHPs by heating/cooling the reservoir of NCG. Applying heat to the reservoir will cause the NCG temperature to rise. The NCG will then expand into the condenser reducing area available for heat transfer shutting down the heat pipe.

Figure 6 illustrates the *multi-condenser VCHP* (MCVCHP) developed by ACT to deliver heat from a single source to multiple end users and provide solid-state switching and control capability. The system shown in Figure 6, has a single evaporator connected to four separate condensers (vertical branches) that each have their own reservoir of NCG. Note that the MCVCHP concept can be applied to an arbitrary number of condensers with arbitrary geometry. Such a system reduces the total number of pipes necessary to transport heat to the various end users and provides thermal control capabilities. Heat is supplied to the system from the Acoustic Heat Pump via an annular evaporator that interfaces with the Acoustic Heat Pump. Vapor formed in the evaporator is free to travel to any of the 4 condensers. It is assumed that power input from the Acoustic Heat Pump will be constant as power must be continuously dissipated from the electronics onboard. The end users connected to Condensers 1-3 may require varying amounts of heat depending on the phase of flight so Reservoirs 1-3 can be heated in order to shut down these condensers on demand. Condenser 4 represents a heat sink that can always accept any heat load that may be available such as pre-heating engine air or the ambient environment.

The system illustrated in Figure 6 operates as follows:

- In Figure 6a, Condensers 1-3 are fully open and these end users are receiving the maximum amount of heat they require. Areas of heat transfer are highlighted by the red vapor. Non-condensable gas is highlighted in green. Condenser 4 is partially open to accept any remaining heat load. The majority of Condenser 4 is occupied by NCG. Recall that Condenser 4 represents a heat sink capable of handling an arbitrary amount of heat such pre-heating engine air or rejecting heat to the atmosphere.

- In Figure 6b, heat has been applied to Reservoir 1 causing the NCG to expand into Condenser 1. This blocks heat transfer to this end user. Because the amount of power supplied by the Acoustic Heat Pump is constant, limiting the area for heat transfer will cause a rise in vapor temperature. The vapor is a saturated fluid so an increase in temperature causes an increase in pressure. This pushes the boundary of Vapor and NCG in Condenser 4 increasing the surface area available for heat transfer within Condenser 4. As a result, Condenser 4 passively accepts the heat load that was rejected at Condenser 1 with only a slight raise in vapor temperature. The NCG/Vapor boundary in Condensers 2 and 3 will also move however by properly sizing the reservoirs the boundary motion in Condenser 4 will be much larger than Condensers 2 and 3.
- In Figure 6c, heat has been applied to Reservoirs 1-3 fully blocking Condensers 1-3. Condenser 4 has passively adjusted to reject all the heat load supplied to the system with minimum vapor temperature rise.
- With this system Condensers 1-3 can be controlled independently and continuously from Fully Open to Fully Closed.

Note that Diebold et al.[8] presented a preliminary proof-of-concept demonstration of the MCVCHP, but the system illustrated in Figure 6, and demonstrated experimentally below, is more sophisticated than the initial prototype. The initial prototype demonstration by Diebold et al.[8] required manual adjustments to the power in order to maintain a constant vapor temperature as heat transfer to the various end users was switched on and off. The system demonstrated in this paper utilized an additional condenser/reservoir pair (Condenser/Reservoir 4 in Figure 6) to allow the system to passively maintain a nearly constant vapor temperature while rejecting a constant overall heat load but varying the heat loads to individual condensers.

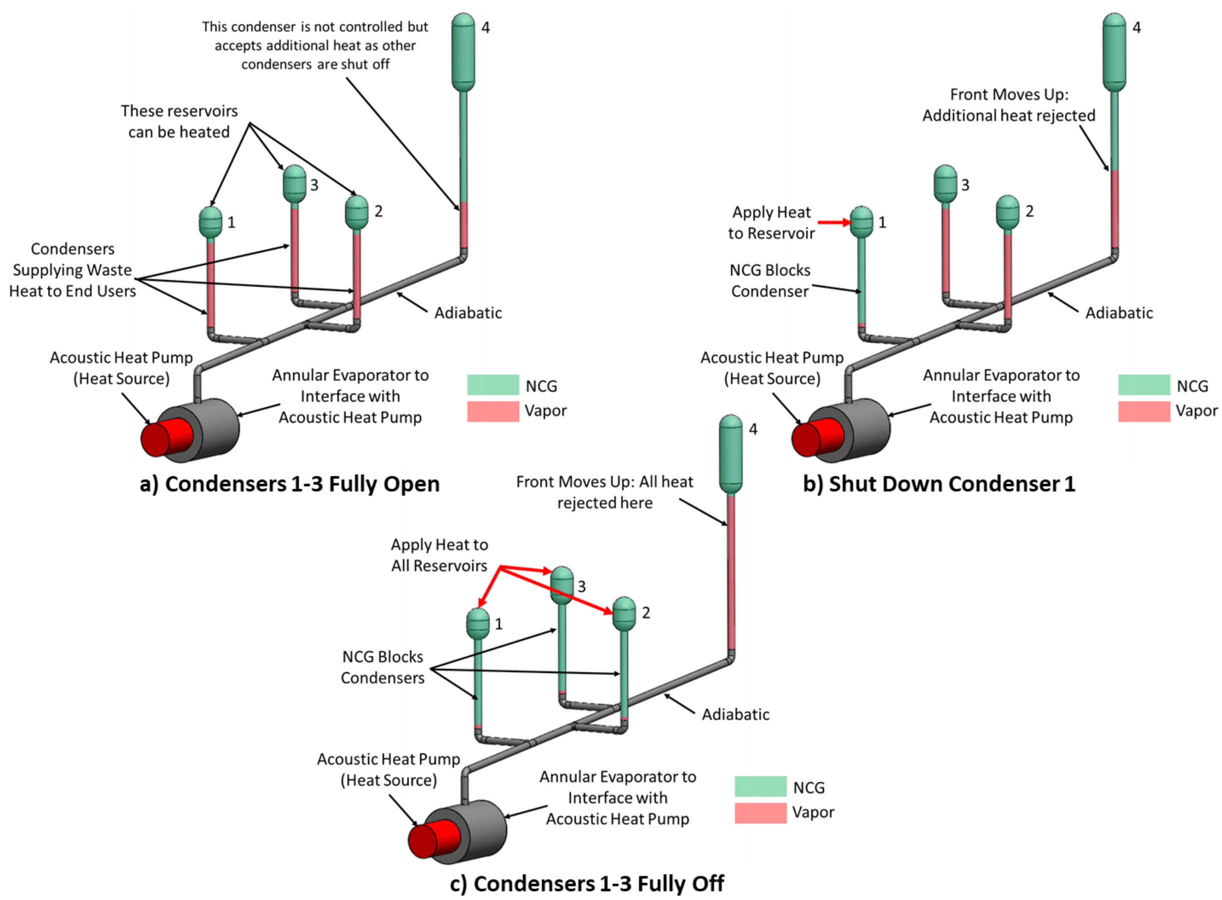

Figure 6. Illustrations of the Multi-Condenser VCHP prototype under construction at ACT. Heat is supplied to the system by the Acoustic Heat Pump at the Annular Evaporator. Vapor is free to travel to Condensers 1-4. a) Condensers 1-3 are fully open, condenser 4 accepts additional heat load. b) Reservoir 1 is heated to shutdown Condenser 1. The additional heat load is passively taken by Condenser 4. c) Reservoirs 1-3 are heated fully shutting down Condensers 1-3. The entire heat load is now rejected by Condenser 4.

The MCVCHP concept has the follow advantages:

- Passive two-phase heat transfer.
- Active thermal control with no moving parts.
- Heat transfer to a given sink can be controlled continuously from fully off to fully on.
- A system can be designed for an arbitrary number of end users (condensers).
- The required heater power applied to a reservoir in order to shut down a given branch only depends on the system geometry, the vapor temperature, the ambient temperature and heat leaks from the reservoir. A relatively small amount of heater power applied to the reservoir (accounting for heat leaks from the reservoir) can control an arbitrarily large amount of power being transported through the system.

The key to designing a MCVCHP is properly sizing the reservoirs to ensure the correct amount of NCG is captured in each branch during the startup process. Prior to startup, the entire system is at the ambient temperature and no power is being applied to the evaporator. In this state, the working fluid will be nearly all condensed liquid or may even be frozen in the evaporator region, and as a result most of the system will be occupied by NCG at a relatively low pressure. As power is applied to the evaporator the working fluid will begin to evaporate and the pressure will begin to increase pushing the NCG front towards the condensers. After the NCG front passes the intersection of the main pipe and the first branch, the amount of NCG in the first branch becomes fixed. The main NCG front will continue to advance in the main pipe while a secondary front in the first branch will move towards Reservoir 1. Fronts will eventually be formed in in the remaining branches. The reservoirs must be sized such that at the design vapor temperature, the NCG front in the end user's condensers (Condensers 1-3 in Figure 6) are located at the respective reservoirs. In other words, at the design vapor temperature all end user condensers are fully open. The displacement of the NCG front for a given increase in vapor pressure is larger for condensers with larger reservoirs. ACT modeled the motion of the NCG fronts using Flat-Front Theory.[8]

B. Multi-Condenser Variable-Conductance Heat Pipe Prototype Fabrication

The fabricated prototype is shown in Figure 7. The prototype was built with stainless steel and used acetone as the working fluid. Stainless steel was selected for this prototype because the relatively low thermal conductivity makes the NCG front location easier to detect which is useful for assessing the behavior of the system. The prototype was designed to operate as a thermosyphon, the liquid was returned to the evaporator by gravity. Figure 7 shows heater tape wrapped around the reservoirs, each reservoir heater was controlled by an independent power source. Note that while Reservoir 4 had heater tape it was never used, Reservoir 4 remained at the ambient temperature. Condensers 1-3 were each 6 in. long while Condenser 4 was 14 in. long. The four condensers were cooled with fans. When Reservoirs 1-3 were heated, shutting down the respective condensers, all heat was rejected at Condenser 4. Because the surface area of Condenser 4 was smaller than the total surface area of Condensers 1-3 it was necessary to use additional fans to improve the cooling over Condenser 4, and this can be seen in Figure 7.

The sizes of the reservoirs were calculated based on flat front theory. The design operating point was defined to be: Condensers 1-3 fully open and only 10% of Condenser being utilized at a vapor temperature of 90°C. The four reservoir volumes are listed in Table 3. Note that the volume of Reservoir 4 was selected and then Reservoirs 1-3 resulted from the geometry of the system and the design temperature. Based on the design temperature and system geometry, the total number of moles of NCG required to reach the correct operating condition was 0.119.

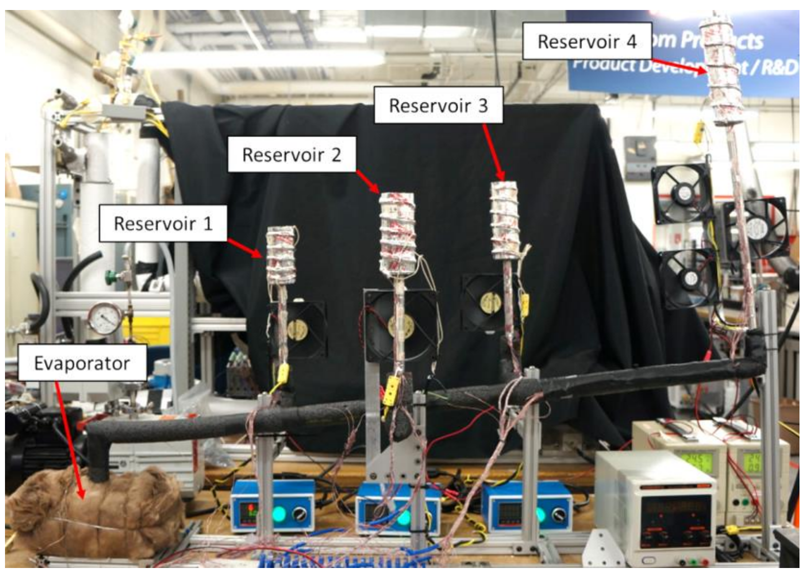

Figure 7. MCVCHP experimental prototype.

Table 3. Reservoir volumes for MCVCHP prototype.

Reservoir 1 (in.^{3})	Reservoir 2 (in.^{3})	Reservoir 3 (in.^{3})	Reservoir 4 (in.^{3})
10.9	12.6	15.5	20.0

The adiabatic sections of the heat pipe were covered in insulation. The prototype was outfitted with thermocouples along the entire length to monitor the location of the NCG/vapor front.

C. Multi-Condenser Variable-Conductance Heat Pipe Prototype Experimental Results

Prior to testing, the system was charged with acetone and after adding the working fluid, the system was charged with 0.119 moles of nitrogen gas. Power was then applied to the system and increased in steps until the design operating condition, vapor temperature of 90°C at a power of 160W, was reached.

Figure 8 shows the temperature distribution of the system near its nominal design operating condition, or the "Fully-On" condition, Condensers 1-3 were fully open. Note that the different sections of the pipe are color coded, and the size of the plot is not to scale. The power to the system was 160W. Applying more power to the system would require more effective cooling at the condensers. At this power the average temperature of the adiabatic TCs was approximately 92°C, slightly above the design temperature of 90°C. The temperatures of Reservoirs 1-3 were slightly above ambient due to their proximity to the hot vapor. Most of the length of Condenser 4 and all of Reservoir 4 were at the ambient temperature. By design, approximately 10% of Condenser 4 was utilized in this condition.

Figure 9 shows the temperature distribution after applying heater power to Reservoir 1 in order to block heat transfer to Condenser 1. The temperature of the outer surface of Reservoir 1 was approximately 90°C and the NCG front can clearly be seen extending into the adiabatic section of Branch 1. The MCVCHP prototype was operated at a constant power applied to the evaporator, representing a constant output from the acoustic heat pump. As Condenser 1 was shut down, the area available for heat transfer was reduced. This resulted in an increase in vapor temperature which pushed the NCG fronts in Condensers 2-4 towards the reservoirs. Reservoir 4 had the largest volume and therefore the NCG front in Condenser 4 was most sensitive to vapor temperature and the front rapidly moved towards Reservoir 4. This increased the surface area available for heat transfer thus minimizing the vapor temperature rise. With Condenser 1 blocked, the average temperature of the adiabatic section increased to approximately 93.2°C, an increase of only 1.2° above the "Fully-On" condition (Figure 8). If the NCG front in Condenser 4 did not move, based on the reduction in surface area resulting from shutting down Condenser 1, the vapor temperature would need to be approximately 121°C in order to reject 160W to the ambient. In other words, if the system did not passively adjust then shutting down Condenser 1 while maintaining a constant power would have required a vapor temperature rise of approximately 29°C.

Figure 10 shows the temperature distribution when heater power was applied to Reservoirs 1 and 2. Blocking Condenser 2 resulted in the vapor temperature rising to 93.9°C and the NCG front moving further up Condenser 4 in order to continue rejecting the total load. Note that due to the increased vapor temperature it was also necessary to increase the temperature of Reservoir 1. The surface of Reservoir 1 was at approximately 110°C while the surface of Reservoir 2 was at approximately 52°C.

Finally, Figure 11 shows the temperature distribution when heater power was applied to Reservoirs 1-3. The average adiabatic section temperature increased to 94.8°C and nearly all of Condenser 4 was fully utilized. The surface temperatures of Reservoirs 1-3 were approximately 124°C, 70°C and 53°C, respectively. In the "Fully-On" condition (Figure 8) approximately only 10% of the total heat load was being rejected by Condenser 4, i.e., Condensers 1-3 represented approximately 90% of the surface area available for heat transfer. Due to the passive adjustment of the NCG in Condenser 4, shutting down Condensers 1-3 resulted in an overall vapor temperature increase of only 2.8°C. If the NCG front in Condenser 4 did not move from its position in the "Fully-On" condition (Figure 8), the vapor temperature would have increased by approximately an order of magnitude to reject 160W to ambient with Condensers 1-3 shutdown. By including the passive condenser/reservoir the system was able to maintain excellent temperature stability regardless of the control input.

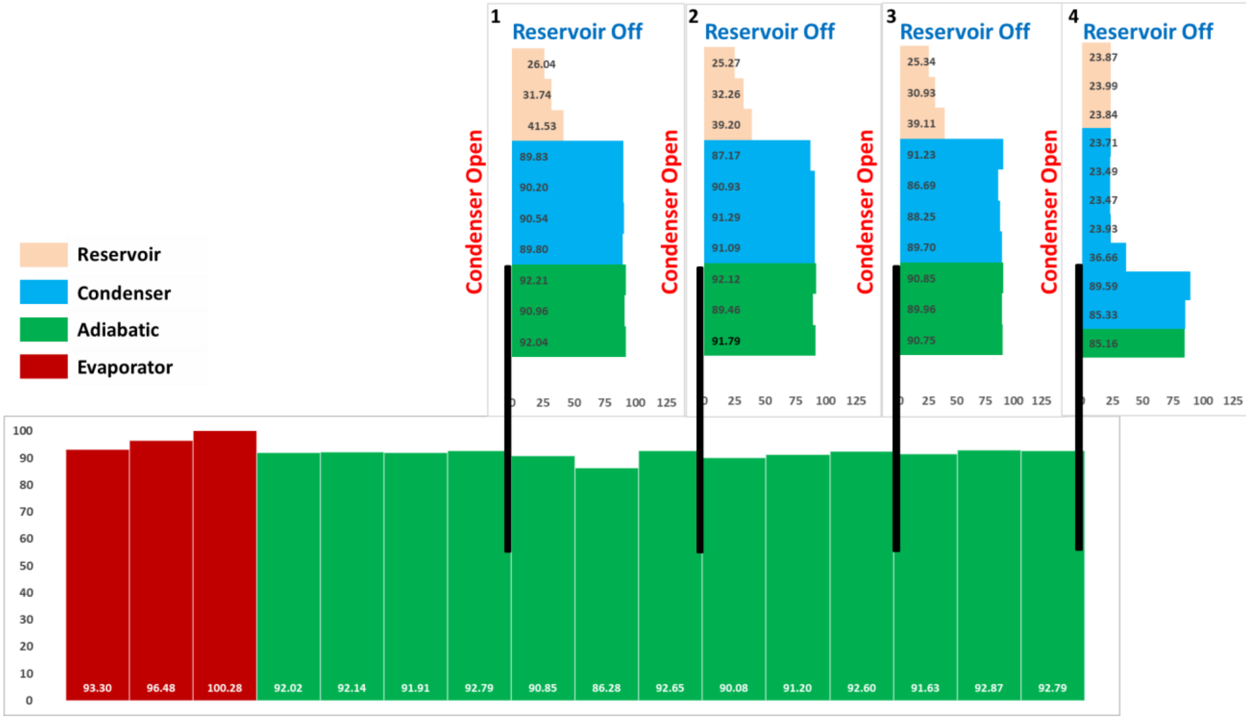

Figure 8. Temperature distribution of MCVCHP Prototype at the design condition. No heater power applied to reservoirs. Condensers 1-3 fully open. Power = 160W.

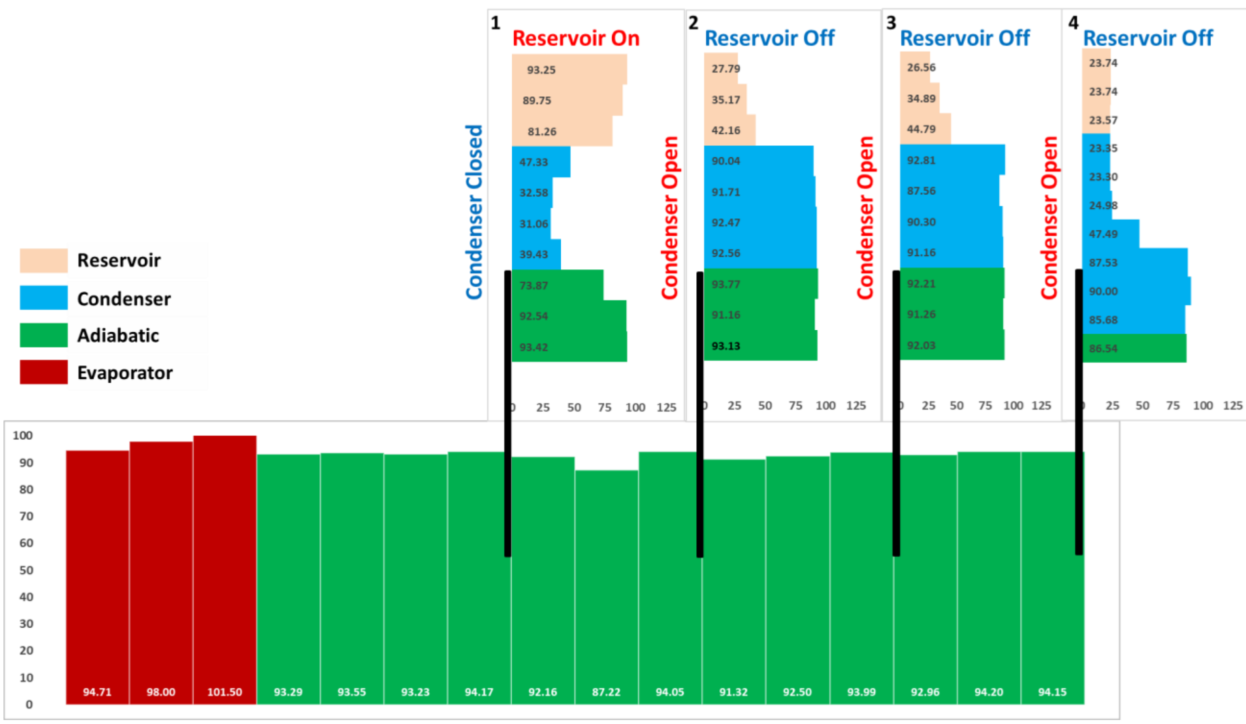

Figure 9. Temperature distribution of MCVCHP Prototype with heater power applied to Reservoir 1. Condenser 1 Closed, Condensers 2-3 Open. Power = 160W.

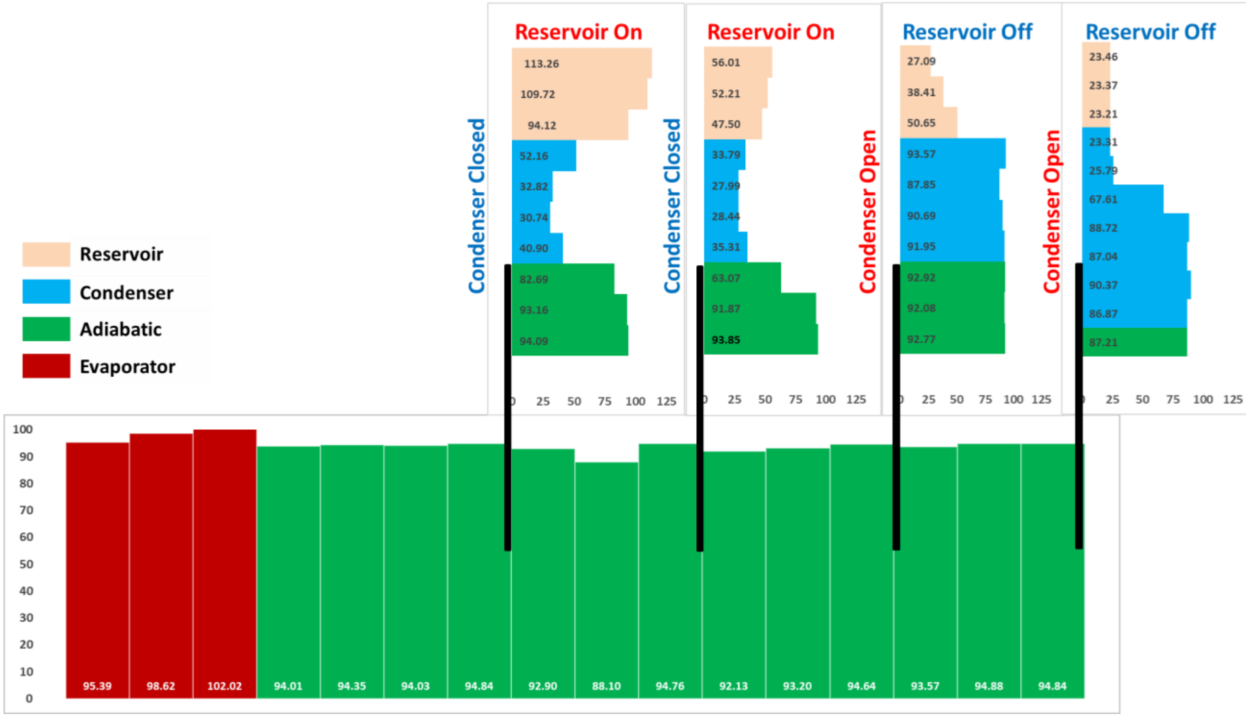

Figure 10. Temperature distribution of MCVCHP Prototype with heater power applied to Reservoirs 1-2. Condensers 1-2 Closed, Condenser 3 Open. Power = 160W.

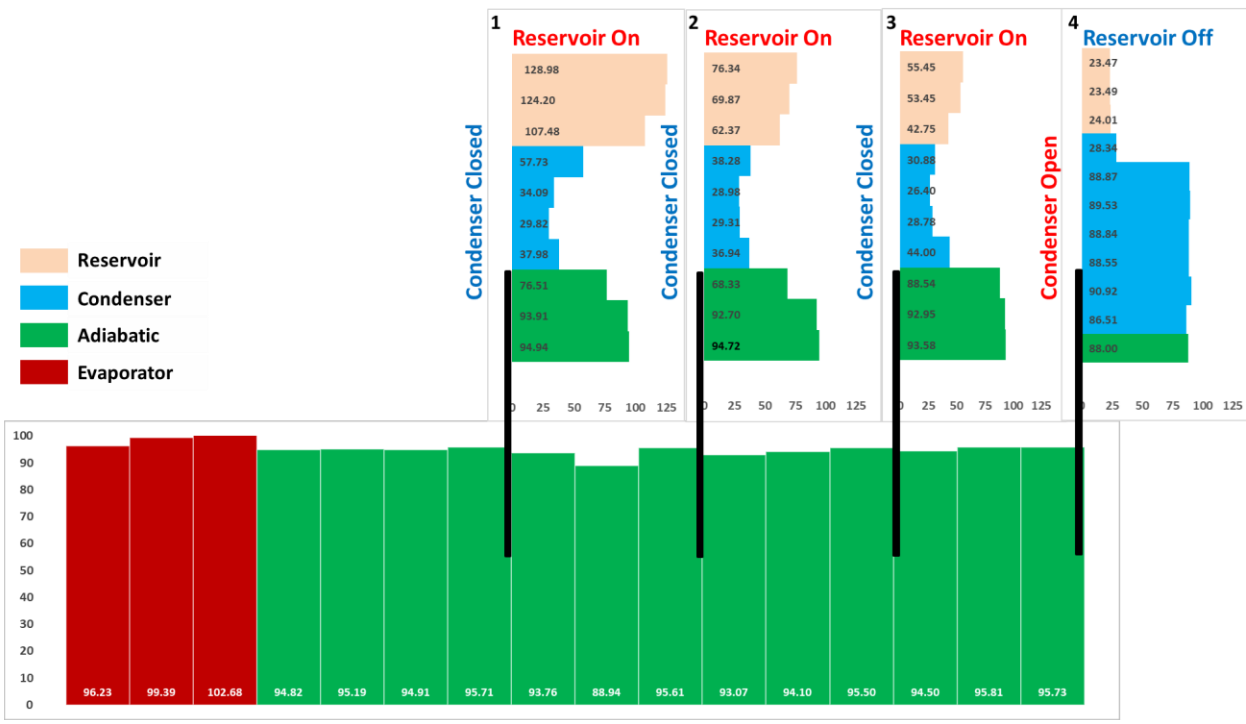

Figure 11. Temperature distribution of MCVCHP Prototype with heater power applied to Reservoirs 1-3. Condensers 1-3 Closed. Power = 160W.

IV. Summary and Conclusions

This paper discusses the development of a two-phase solid-state waste heat delivery system for electric aircraft. An acoustic heat pump developed by Dyson[1] will be used to elevate low-grade waste heat throughout the aircraft to a higher temperature. ACT is designing heat pipe-based thermal delivery networks to transport this high-grade waste heat to various end users throughout the aircraft. Using NASA's STARC-ABL as a representative aircraft, ACT identified the temperature and relative locations of the heat sources and end users and designed three groups of heat pipe heat delivery systems based on the temperature and power requirements.

A prototype demonstration of a multi-condenser VCHP system with thermal switching capability that improved upon the demonstration of Diebold et al.[8] was presented. The system is capable of transporting heat from a single heat source to multiple end users via a single interconnected network of pipes. The heat transferred to each individual end user can be independently controlled and the system passively adjusts to minimize the change in operating temperature regardless of the control input. The MCVCHP system reduces the number of individual heat pipes required to distributed thermal energy and provides solid-state thermal control. The system is temperature driven and therefore a relatively small amount of power delivered to the NCG reservoirs can be used to control an arbitrary amount of power transferred through the heat pipe system.

Acknowledgments

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