DESIGN AND DEVELOPMENT OF A LOW PRESSURE EVAPORATOR/CONDENSER UNIT (ECU) FOR WATER-BASED ADSORPTION TYPE CLIMATE CONTROL SYSTEMS

A Thesis presented

By

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Electric vehicles (EVs) are the future of clean transportation and driving range is one of the most important parameters which dictates its marketability. In order to increase driving range, electrical battery energy consumption should be minimized. Vapor-compression refrigeration systems currently employed in EVs for climate control consume a significant fraction of the battery charge. Thus, by replacing this traditional heating ventilation and air-conditioning system (HVAC) with an adsorption based climate control system one can have the capability of increasing the drive range of EVs.

The Advanced Thermo-adsorptive Battery (ATB) for climate control is a water-based adsorption type refrigeration cycle. An essential component of the ATB is a low pressure evaporator/condenser unit (ECU) which facilitates both the evaporation and condensation processes required for operation of the ATB. The thermal design of the ECU relies predominantly on the accurate prediction of evaporation/boiling heat transfer coefficients since the standard correlations for predicting boiling heat transfer coefficients have large uncertainty at the low operating pressures of the ATB. This work describes the design and development of the low pressure ECU as well as the thermal performance characterization of the actual ECU prototype.
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Chapter 1: Introduction

The transportation sector in the United States is responsible for 26% of total greenhouse gas emissions (GHG) [1]. Electric vehicles (EVs) can reduce GHG emissions by 32% [2]. Average driving range of current generation EVs is about 100 Miles [3] and 77% of customers are concerned about the range limitations of EVs [4].

Coefficient of performance (COP) of standard R-12 vapor compression refrigeration system currently employed in EVs for climate control is 2.35 [5] and it consumes a significant fraction (35%) of battery charge [6]. Adsorption heat pumps are one of the energy efficient alternatives to the traditional vapor compression system for heating and cooling applications [7]. The advanced thermo-adsorptive battery (ATB) is a water-based adsorption type climate control system that has the potential to reduce the battery energy consumption and increase the drive range of EVs significantly. This section describes the basic working principle of ATB system and also the different modes of operation when it is deployed in an actual EV.

1.1 Background

The ATB system has two main components 1) adsorption bed- which uses high density adsorbent materials (MgY Zeolite) with very high hydrophilicity, water uptake capacity and heat of adsorption [8] and 2) evaporator/condenser unit (ECU) - which employs micro finned tubes to generate and condense vapor at low pressures. As the vapor is generated in the ECU by the evaporator, the adsorption bed draws all the vapor and generates heat. No compression or any other external means of vapor transport is required for transporting refrigerant vapor between the two components [9]. This process is known as discharging.
which is shown in Figure 1(a). The coolant can then be circulated across the ECU and the adsorption beds to accomplish both cooling and heating [10]. The entire process can be reversed by applying heat energy into the adsorption beds which regenerates the beds by releasing vapor molecules back to the ECU. Finally, the condenser coils condense the hot vapor back to the liquid state. This process is known as recharging which is shown in Figure 1(b).

![Figure 1 Schematic of ATB operation during a) discharging mode (adsorption) and b) recharging mode (desorption).](image)

During the previous phase of this project, efforts were made to integrate both the ECU and the adsorption bed as a single unit [11]. Although this monolithic integration was advantageous both in terms of space and weight, there were some difficulties in manufacturing the embedded evaporator prototype. The motivation for change in system architecture arose after some successful experimental results obtained at MIT using an external coiled evaporator [12]. This is the starting point of this thesis and the main objective of this work involves the design and development of a low pressure ECU and
evaluating its thermal performance. In addition to this, several control strategies are explored to tackle vehicle preconditioning requirements and also to reduce the cost and complexity of the ATB system.

1.2 ATB modes of operation

ATB interacts with two additional heat exchangers when it is deployed in an EV, they are cabin climate core and radiator. These are liquid and air based heat exchangers with minimal electrical power requirements for coolant circulation and air flow that are used to exchange heat between different components and environment [10]. There are three different modes of operation for ATB they are cooling, heating and regeneration. Cooling and heating modes are accomplished during adsorption phase where the ECU operates as an evaporator and regeneration is accomplished during desorption phase where the ECU works as a condenser. The internal pressure inside the ECU varies between 700 Pa and 60kPa during adsorption and desorption process respectively.

1.2.1 Cooling

In this mode of operation, the primary objective of the ECU is to provide chilled coolant to the climate core heat exchanger so that it can supply cooled air ($\leq 15^\circ C$) inside the passenger cabin. Hence in this mode of operation the ECU is connected to the climate core heat exchanger and the heat generated inside the adsorption bed is dumped into the environment using a radiator [9]. The schematic of the cooling mode is shown is Figure 2. In addition to achieving this target cabin air temperature, the ECU should also be able to deliver a target thermal power of 1600 watts for a period of 4 Hours. During peak summer conditions the inlet air temperature entering the climate core heat exchanger can be as high as 43°C.
1.2.2 Heating

In this case the primary objective is to supply hot air at 28°C inside the passenger cabin and therefore in this mode of operation, heat from the adsorption bed is utilized by coupling it to the climate core heat exchanger and the ECU is coupled in a closed loop with Radiator [9]. In severe winters, the coolant entering the ECU can be below the freezing temperature of refrigerant (water). So to achieve evaporation at such low (below freezing) temperature, auxiliary electric heaters are installed on the bottom plate of the ECU enclosure.
1.2.3 Regeneration

During adsorption, the stacks inside the beds start to become saturated with water until they cannot adsorb any more water molecules. Hence the adsorption beds need to be recharged after each adsorption cycle before it can start operating again. This recharging process also known as desorption requires heating up the beds to a very high temperature (150 to 350°C) [13]. In order to achieve this, the side walls of the adsorption bed enclosures are equipped with external heaters which are electrically powered from the grid. Hence the regeneration mode occurs when the electric vehicle is in charging mode. The ECU is coupled with the radiator during this mode of operation. Vapor from the adsorption bed enters the ECU at a much higher temperature (60°C). The ECU now operates as a condenser and condenses the hot vapor entering the enclosure and the refrigerant gets stored back into the ECU. This mode completes the ATB cycle of operation. The air temperature across the radiator (T_{air}) during severe winters can be as low as -18°C and in summer it can be as high as 43°C. Figure 4 depicts the schematic of regeneration mode and the ECU should be able to deliver a power of about 2700 Watts during this mode of operation.

![Figure 4 Schematic of ATB regeneration mode where the ECU is coupled to the radiator and adsorption bed is coupled to external electric grid.](image-url)
Chapter 2: Structural design of the evaporator/condenser unit

Principal functions of the ECU include 1) Evaporating refrigerant to create vapor 2) Condensing the hot vapor from the adsorption beds during regeneration mode 3) Store enough refrigerant for ATB operation and 4) Maintain structural stability while operating under vacuum conditions. These functions provided the design constraints that guided the initial shape of the ECU. The schematic of the ECU is shown in Figure 5.

![Figure 5 Schematic of initial structural design of Evaporator/Condenser Unit.](image_url)

Figure 5 shows the ECU schematic with ripple like structures on top and bottom plates of the enclosure. These structural perturbations were primarily created to retain structural stability while maintaining the enclosure weight below 20 kg. However, welding thin stainless steel sheets (<2 mm) posed a major manufacturing difficulty.

Apart from the principal functions, the ECU should also ensure smooth vapor transport to the adsorption beds without any major mass transfer resistance and pressure drop. The final tapered design of the ECU shown in Figure 6 takes care of this objective by facilitating the accumulation of vapor on top the chamber before it exits the ECU. Another important factor was to ensure that the entire unit fits inside the hood of an EV. This was
accomplished by having several fitting exercises. The above mentioned factors transformed the shape and size of the ECU enclosure from the initial design to a tapered design as shown in Figure 6. The vapor port (3.38 CF flange) is placed in the top part of the front plate to favor vapor extraction from the highest section (158 mm). The final dimensions of the ECU enclosure after the fitting exercise were 750×450 mm. The final ECU enclosure weight and volume were 41 kg and 39 liters respectively.

Figure 6 Isometric view of final ECU enclosure design. The fabrication of the unit was outsourced to Super Radiator Coils in Virginia.

Inside the ECU, evaporator heat exchanger tubes are placed in 2 rows with 10 and 8 passes respectively. Whereas the condenser heat exchanger tubes are placed in a single row with 8 passes. The tube bending radius which depends on the outside diameter (O.D) dictated the spacing between each pass of the tube in both evaporator and condenser heat exchanger. This in turn finalized the width of the ECU enclosure (450 mm). Some clearance is provided between the walls of the ECU enclosure and the heat exchanger tubes for manufacturing feasibility.
Figure 7 Exploded view of ECU showing the heat exchanger arrangements in both evaporator and condenser respectively.

Furthermore, a set of two coolant ports each for the evaporator and condenser heat exchangers are welded on to the back plate of the enclosure. These ports are spaced in a specific manner for connecting the inlets / outlets of the evaporator and condenser tubes inside the enclosure to the external heat exchangers. The size of the coolant ports is determined by the size of the evaporator and condenser tubes respectively.

Figure 8 Side view of the final ECU enclosure design showing the coolant ports for evaporator/condenser and ports for pressure transducer/drain on top and bottom plate of enclosure.
Inside the ECU apart from evaporator and condenser heat exchangers there are structural members known as *Baffles* that were incorporated to 1) provide support for evaporator and condenser heat exchanger assemblies and 2) serve as structural reinforcement to the ECU enclosure under vacuum conditions. Apart from these, the baffles were also designed in a way to facilitate vapor and water transport inside the ECU enclosure. The dimensions of the baffle design are shown in Figure 9. The slots in the middle section are for supporting the evaporator and condenser heat exchanger tubes while the remaining slots are for water and vapor transport inside the enclosure. The two heat exchangers spaced in such a way that it can accommodate 10.7 liters of refrigerant (water) between them to have the energy capacity 6.4 Kw-hr. Baffle thickness, number of baffles and spacing between them were finalized after conducting several iterations of FEA analysis for the entire ECU enclosure which is discussed in the next section.

Figure 9 Baffle blue print with detailed geometry. Slots in the middle are for supporting both evaporator and condenser heat exchanger while the remaining slots are for water and vapor transport across the enclosure.
2.1 Static structural analysis

As discussed in the previous section, there were several factors and constraints that dictated the geometry and shape of the enclosure. To ensure structural stability several iterations of finite element analysis (FEA) were performed by utilizing ANSYS Workbench 14.0. Static structural analysis was performed not only to ensure that the ECU enclosure will withstand the maximum load under vacuum conditions during its operation but also to eliminate any structural errors that would result in severe stress concentration regions or points inside the ECU enclosure.

The enclosure material selected from the library of ANSYS Workbench 14.0 is stainless steel alloy with a 7850 kg/m$^3$ density. The yield strength and ultimate strength of stainless steel are 208 MPa and 460 MPa respectively. Computational time and memory required for finite element modelling can be made more efficient by exploiting the planes of symmetry[14]. Symmetry assumption is valid only when the restraints (boundary conditions) and the geometry are symmetric about a plane which is present in the ECU enclosure design [14]. Hence only one half of the enclosure geometry is utilized for structural analysis. The boundary condition (type of contact) selected between the enclosure and baffles was bonded (type). This type of contact restricts any type of separation and relative motion between surfaces [15]. FEA analysis was performed by assuming vacuum conditions and a maximum pressure load of 1 Atmosphere or 0.1 MPa acting in the normal direction on all sides of the enclosure. After several trials the optimal thickness of different parts of the ECU enclosure to achieve a static safety factor of 1.43 was determined. These dimensions are listed in Table 1.
Table 1 Thickness details for different parts of the ECU

<table>
<thead>
<tr>
<th>Part</th>
<th>Top and bottom plates</th>
<th>Baffles</th>
<th>Side plates</th>
<th>Larger half-cylinder (front end)</th>
<th>Smaller half-cylinder (back end)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thickness</td>
<td>4.572 mm</td>
<td>4.572 mm</td>
<td>4.572 mm</td>
<td>6” pipe SCH40</td>
<td>4.5” pipe SCH40</td>
</tr>
</tbody>
</table>

The values of maximum stress, maximum deformation and static safety factor from the static structural analysis are listed in Table 2. This design ensures that there are no structural irregularities that could result in major stress concentration.

Table 2 FEA results for the final ECU enclosure design under vacuum conditions

<table>
<thead>
<tr>
<th>X-Axis Max. Deformation (mm)</th>
<th>Y-Axis Max. deformation (mm)</th>
<th>Z-Axis Max. Deformation (mm)</th>
<th>Von-Mises Equivalent Stress (Mpa)</th>
<th>Static Factor of safety</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.014529</td>
<td>0.2146</td>
<td>0.022477</td>
<td>145.39</td>
<td>1.43</td>
</tr>
</tbody>
</table>

Figure 10 shows the von-mises stress distribution across the ECU enclosure for a pressure load of 0.1 MPa. Maximum von-mises stress is 145 MPa and it occurs in the slots provided on the baffles for water and vapor transport. Figure 11 shows the deformation profile of the ECU enclosure under vacuum conditions. The maximum deformation under a pressure load of 0.1 MPa is 0.21 mm and it is located on the top plate of the enclosure.
Figure 10 Distribution of von-mises equivalent stress for the ECU enclosure. Maximum von-mises stress under a uniformly distributed of 0.1 Mpa is 145 Mpa and is located in the slots provided in the baffles for water and vapor transport.

Figure 11 Location of maximum deformation for the ECU enclosure under a load of 0.1 MPa. Maximum deformation is 0.21 mm and is located on the top plate of the enclosure.
2.2 Fatigue analysis

One of the design requirements is that the ECU should be able to hold vacuum without any leak and maintain structural rigidity at least for a period of 5 years (8000 Cycles). Fluctuations in the pressure within the ECU and the fact that these loads are cyclic motivated the need to perform a fatigue failure analysis. The enclosure absolute pressure varies between 700 Pa during adsorption and 60 kPa during regeneration (desorption). This type of fluctuating loading has a non-zero mean stress. So to investigate the effect of this non-zero mean stress on fatigue life the Soderberg Line method (more conservative) has been utilized to ensure reliability of the design [16].

From the static structural analysis, the maximum von-misses stress in the ECU enclosure under an external pressure load of 0.1 MPa (adsorption) and 0.04 MPa (desorption) are 145.39 MPa ($S_{\text{max}}$) and 47.78 MPa ($S_{\text{min}}$) respectively. From these values, the alternating stress ($S_a$) and mean stress ($S_m$) values are computed using the following relations [16],

$$S_a = \frac{S_{\text{max}} - S_{\text{min}}}{2} \quad (1)$$

$$S_m = \frac{S_{\text{max}} + S_{\text{min}}}{2} \quad (2)$$

Typical endurance limit ($S_e$) of stainless steel (AISI type 302) is 234 Mpa [17]. Finally, the fatigue safety factor ($N$) is computed using the following relation [16],

$$\left(\frac{S_a}{S_e}\right) + \left(\frac{S_m}{S_y}\right) = \left(\frac{1}{N}\right) \quad (3)$$
Table 3 Fatigue analysis results of final ECU enclosure design using Soderberg line method.

<table>
<thead>
<tr>
<th>Alternating Stress (Sa)</th>
<th>Mean Stress (Sm)</th>
<th>Yield Strength of Stainless steel (Sy)</th>
<th>Endurance Strength of Stainless Steel (Su)</th>
<th>Fatigue safety factor (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>MPa</td>
<td>MPa</td>
<td>MPa</td>
<td>MPa</td>
<td>1.4</td>
</tr>
<tr>
<td>48.76</td>
<td>96.585</td>
<td>208</td>
<td>234</td>
<td></td>
</tr>
</tbody>
</table>

The values of alternating stress, mean stress and fatigue safety factor are listed in Table 3. A fatigue safety factor of 1.4 is obtained for the ECU enclosure final design and this suggests that the amplitude of alternating stress (Sa) and the effect of mean stress (Sm) for the loading conditions during ATB system operation is not high enough to cause fatigue failure.
Chapter 3: Evaporator performance characterization

This section deals with the methodology followed for estimating the average evaporation/boiling heat transfer coefficient at low pressures. Among the two heat transfer mechanisms inside the ECU (evaporation and condensation), the performance is mainly limited by evaporation/boiling process. Hence more importance is given to the design of evaporator heat exchangers than condenser to deliver the required power at desired coolant outlet temperature. Only standard heat exchanger tubes and fin arrangements are evaluated in order to reduce the complexity of the ATB system. Figure 12 shows the different thermal resistances encountered during the evaporation process inside the ECU.

![Thermal Resistance Model](image)

Figure 12 Thermal Resistance Model for flow inside a pipe with evaporation/boiling on the external tube surface.

From the thermal resistance model in Figure 12, the evaporation heat transfer ($Q_{evap}$) can be determined by using the following expression.

$$Q_{evap} = \frac{1}{1 + \frac{1}{h_{int} A_{int}} + \frac{\ln(r_2 / r_1)}{2\pi k L} + \frac{1}{h_{evap} A_{ext}^*}} \left(T_{in} - T_{sat}\right)$$

(4)
where $T_{in}$ is the coolant inlet temperature and $T_{sat}$ (K) is water (refrigerant) saturation temperatures (K), $h_{int}$ is the internal heat transfer coefficient and $h_{evap}$ is the evaporation/boiling heat transfer coefficient (W·m$^{-2}$·K$^{-1}$), $r_2$ and $r_1$ are the outer and inner tube radius (m), $k$ is the tube material (Copper) thermal conductivity (W·m$^{-1}$·K$^{-1}$), $L$ is the tube length (m), and $A_{int}$ and $A_{ext}^*$ are the internal and external surface areas of the tube (m$^2$), respectively. Here $A_{ext}^*$ is the effective surface area and $A_{ext}^* = A_{tot} \eta_{tot}$ where $A_{tot}$ is the total external surface area and $\eta_{tot}$ is the total fin efficiency.

From estimating the values of all the resistances in the system, it was evident that the evaporator/boiling heat transfer is the major resistance in the system. Thus, the accurate prediction of the external evaporation/boiling heat transfer coefficient is a key objective to properly design the heat exchanger.

Initially, Rohsenow’s correlation was adopted for estimating the external evaporation/boiling heat transfer coefficient with water as a refrigerant. But there was a large sensitivity in the evaporator length required to achieve a target thermal power of 1600 watts due to surface material and finishing, e.g. 5m and 18 m for a scored and lapped copper surface respectively.

During adsorption process, the ECU operates at very low chamber pressure (700-800 Pa) and at low heat flux and wall superheat temperatures (0.5-4 W/cm$^2$ and 4- 10°C) to generate vapor. At these operating conditions, only few nucleation sites are active for boiling and the bubble departure frequency is also very low. Heat flux levels of 60 w/cm$^2$ are required to increase the bubble departure frequency in order to maintain continuous boiling and steady wall temperature at low pressures [18]. Hence the standard Rohsenow’s nucleate
boiling correlation for predicting heat transfer coefficient cannot be utilized for designing the ECU for ATB operating conditions.

When the chamber pressure is very low (700-800 Pa) the weight of the liquid column becomes significant. Hence, the pressure and the saturation temperature changes along the liquid column (Figure 13) inside the ECU. These variations creates a non-homogeneity in the bulk fluid properties that results in a boiling process entirely different from those observed at atmospheric conditions [19].

![Figure 13 Variation of saturation temperature with liquid height (h) for a vapor saturation pressure of 750 Pa (T_{sat}=2.85°C).](image)

Sub-atmospheric pool boiling experiments were conducted over a plain horizontal copper tube surface at pressures ranging between 1.8 – 3.3 kPa [20]. However, the correlations for heat transfer coefficient obtained from these experiments cannot be utilized for sizing the evaporator as the ATB system operates at a much lower pressure (700-800 Pa) and also employs a micro finned tube for evaporation/boiling (S/T Trufin® 67-114025 tube from Wolverine Inc.). This enhanced tube surface with micro fins will also have a significant effect on boiling performance [21]. Thus, the lack of correlation for predicting external
boiling heat transfer coefficient at the desired pressure range (700-800 Pa) and the large uncertainty of Rohsenow’s model stimulated the need for experiments.

The external boiling heat transfer coefficient $h_{\text{evap}}$ strongly depends on the wall superheat temperature and this changes along the length of phase change heat exchanger which is shown in Figure 14. Hence several experiments are carried out at different superheat temperatures (4 – 10°C) that corresponds to ATB operational conditions and finally an average approximation for variation in $h_{\text{evap}}$ with respect to $\Delta T_e$ is obtained. The resulting average heat transfer coefficient $\bar{h}_e$ is then utilized to accurately size the heat exchanger.

![Figure 14](image.png)

*Figure 14* Variation of the superheat temperature difference along the heat exchanger. As the external surface temperature varies, so does the superheat temperature and the boiling heat transfer coefficient.

The refrigerant and coolant selected for this analysis are distilled water and 50/50 ethylene glycol-water mixture. Water has a high heat of vaporization and high thermal conductivity apart from being environmental friendly and cheap [21]. Furthermore, the heat of adsorption (3300-4200 kJ/kg) and desorption temperature (200°C) for zeolite-water pair
is also very high [22]. In severe winters the coolant that enters the ECU can be as low as -18°C. Hence to prevent freezing of coolant at such low temperature a 50/50 ethylene glycol-water composition (volume) was selected which has a freezing point of -40°C [23].

### 3.1 Selection and fabrication of enhanced surface tubes

Two different micro finned tubes are procured from Wolverine tubes Inc. and used for evaporator characterization experiments. Tube 1 (TURBO-Chill® 56-4050228) is a micro finned tube with external fins on the outside diameter and ridges on the inner diameter. These ridges on the interior surface aids heat transfer by promoting turbulence of the working fluid [24]. Tube 2 (S/T Trufin® 67-114025) has integral helical fins that are spaced very closely to one another while the inner surface is smooth with no internal protrusions. This tube has a larger surface area when compared with Tube 1 and it increases the heat transfer coefficient during evaporation and condensation [25].

![Figure 15](image-url) Images of Tube 1 (TURBO-Chill® 56-4050228) and Tube 2 (S/T Trufin® 67-114025) from Wolverine Inc. that were utilized for fabrication and testing inside the ECU for estimating the average boiling heat transfer coefficient.
The end faces of the micro finned tubes are machined to the required diameter and then soldered to a hollow copper rod (5/8” outside diameter). The soldered joints are leak tested by pressurizing the fabricated tube at 30 Psi. Once the joints are leak proof, they are filed to get a smooth surface finish. This is primarily done to eliminate additional nucleation sites for boiling during experiments.

After soldering, compression fittings are utilized to couple these tubes to the liquid feedthrough inside the vacuum chamber. Polypropylene compression tube fittings from *Mc-Master-Carr* were installed to ensure that the evaporation/boiling process takes place only on the micro finned tubes during the evaporator characterization experiments. These compression fittings can withstand very high pressure and temperature (220 Psi and 107°C). Polypropylene fittings are also inert which makes them ideal for high purity applications such as distilled water [26].

![Fabrication of Micro Finned Tube](image)

**Figure 16** Fabrication of Micro Finned Tube for estimating the average boiling heat transfer coefficient at Low Pressures. The compression fittings are utilized to couple the tubes to liquid feedthrough which in turn is connected to the coolant lines.
3.2 Calibration of temperature sensor and flow meter

The reliability of the evaporator experiments for estimating the external boiling heat transfer coefficient largely relies on the accurate measurement of the wall superheat. Hence the calibration of thermocouples and other temperature sensors was necessary.

Three known standard temperature points, boiling (100°C) and freezing point of water (0°C) and constant temperature bath of Dry Ice/ Iso-Propyl Alcohol (-77°C) were chosen to calibrate the K-type thermocouples and RTD probes from Omega Inc. The thermocouples and RTD probes are inserted in a glass beaker as shown in Figure 17. Care is taken to ensure that the temperature sensors are not in contact with the walls of the beaker. The sensors are then connected to a data acquisition system (Agilent 34980A) for recording the temperature data during the experiments. During each experiment the data is collected until the temperature becomes steady.

![Set up for calibrating the temperature sensors (K-type thermocouple and 4-wire RTD probes from Omega Inc.) at three standard temperature points. Calibration curve for the temperature sensor.](image)
After the temperature calibration experiments, the measured temperature is compared with the reference temperature (standard) and a straight line is obtained for each and every temperature sensor. The resulting straight line equation is then utilized for correcting the temperature measurement during the evaporator characterization experiments.

Coolant flow rate impacts the calculated heat flux during the evaporator characterization experiments. Flow meter (FPR 302) calibration is carried out by pumping 50/50 ethylene glycol-water mixture (volume) through the sensor and collecting it in a graduated 20 liters container for 20 seconds (measured using a chronometer). This procedure is repeated for different flow rates and the recorded readings by flow meter is compared with the calculated flow rates (Flow rate= collected volume/time). Figure 18 shows the experimental set up used for calibrating the flow meter and the linear relationship that is obtained between the calculated flow rate and recorded flow rate by the flow meter (FPR 302) from Omega Inc.

![Experimental set up for calibrating coolant flow meter for evaporator characterization experiments and calibration curve (measured vs recorded data).](image-url)

Figure 18  Experimental set up for calibrating coolant flow meter for evaporator characterization experiments and calibration curve (measured vs recorded data).
3.3 Experimental procedure for estimating average boiling heat transfer coefficient

The set up consists of a cylindrical vacuum chamber 15.2 cm (6 inches) inner diameter and about 35.6 cm (14 inches) long. The micro finned tube to be tested is placed inside the vacuum chamber by using a fabricated support structure shown in figure 19. Both the micro finned tubes to be tested have an outside diameter of 19.05 mm and both the tubes are 1’ long (304.8 mm). The ends of the tube are connected to a series of pumps and a chiller through a liquid feedthrough port attached to the vacuum chamber. The Polystat chiller/heater from Cole Parmer (12111-02) is used to control the coolant inlet temperature.

![Image of acrylic support structure](image_url)

Figure 19 Acrylic support structure utilized for supporting the micro finned tubes during evaporator characterization experiments
The vapor generated inside the chamber during experiments is captured using a set of dry ice cold traps from Laco Technologies (LIT-10025). The cold traps are connected in parallel with each other in order to lower the vapor flow rate which increases the residence time for freezing and ensure better vapor retention capacity. Pressure inside the chamber is reduced to the required saturation pressure (750 Pa) by using a RV-3 Edwards vacuum pump that is connected to the vacuum chamber through the cold traps. The vacuum chamber also has a view port on one end to visualize the boiling process. In addition to these components, there is a KF 40 butterfly valve from Kurt J. Lesker that is installed on top of the cylindrical vacuum chamber to control the system pressure. The entire vacuum chamber is covered with 12.5 mm thick nitrile rubber insulation ($k_{ins}= 0.25 \text{ Wm}^{-1}\text{K}^{-1}$) to prevent losses. Two PX409 series absolute pressure transducers from Omega Inc. were installed on both the vacuum chamber and also on the cold trap line to keep track of the chamber pressure. In order to measure the wall superheat temperature, two K-type thermocouples are attached to the surface of the tubes and one additional K-type thermocouple was installed near the tube surface to measure the water temperature ($T_{\text{sat}}$). K-type thermocouples have errors of ± 2.2°C [27]. In addition to this, there were two 4-wire RTD’s that were measuring coolant inlet and outlet temperature ($T_{C-IN}$, $T_{C-OUT}$). RTD’s are more accurate than standard K-type thermocouple (. Two more K-type thermocouples were attached to the top and bottom surface of insulation to measure the heat gain from the environment. A flow meter (FPR 302) from Omega Inc. and a needle valve were also attached to the coolant lines right after the pump to measure and control the coolant flow rate. These sensors are then connected to an Agilent data acquisition system (34980A) to record the data during experiments.
In order to securely attach the thermocouples to the tube surface a Dow-Corning silicone heat sink compound and a kapton tape was used. The heat sink compound ensured that there was minimal contact resistance between the thermocouple and the tube surface. A 50-50 ethylene glycol-water mixture (volume) was used as a coolant for these experiments because of its ability to operate at wider temperature ranges without any phase change.

Before starting the experiments, the water inside the chamber is degassed to remove any non-condensable gases. This is done by freezing the water and then melting it. The detailed procedure for degassing is described in the next section. Once degassing is complete, the vapor valve is closed and the cold traps are filled with dry ice before the vacuum pump is turned on. Coolant is then circulated at 8 lit/min and at a specific inlet temperature controlled by the chiller. As soon as the coolant inlet temperature reaches the target temperature set in the chiller, vapor control valve is opened. The pressure inside the
chamber starts to drop. This is accompanied by drop in chamber surface temperature and liquid water (Refrigerant) temperature. Once the pressure falls below 800 Pa, the vapor valve is throttled to maintain the chamber pressure around 750 Pa ($T_{sat}=2.85^\circ C$). The experiment is then continued until the steady conditions are reached. If the vapor valve is not throttled the pressure in the vacuum chamber drops below 700 Pa and the surface of water freezes stopping the evaporation/boiling process inside the chamber.

The experiments are repeated for different coolant inlet temperatures in order to obtain different wall superheats and corresponding heat fluxes. After a set of data points are obtained for Tube 1, it is then replaced by Tube 2 and the same experimental procedure is followed. The average boiling/evaporation heat transfer coefficients, $\overline{h_e}$ (W·K$^{-1}$·m$^{-2}$) for two different types of enhanced surface tubes from Wolverine Tube Inc. were estimated from an energy balance of the submerged heat exchanger which is shown below.

$$\dot{m}C_{p_{coolant}} (T_{C-IN} - T_{C-OUT}) = \overline{h_e} A_{evap} \Delta T_e$$ \hspace{1cm} (5)

Here, $\overline{T_e} (T_{surface} - T_{sat})$ is the average wall superheat temperature, where $T_{surface}$ is the average tube surface temperature and $T_{sat}$ is the liquid water saturation temperature. $A_{evap}$ is the evaporator nominal external surface area ($m^2$), $\dot{m}$ is the coolant mass flow rate (kg·s$^{-1}$), and $C_{p_{coolant}}$ are the coolant density (kg·m$^{-3}$) and specific heat capacity (J·kg$^{-1}$·K$^{-1}$) evaluated at the average coolant temperature, respectively.
Chapter 4: Full scale ECU prototype testing

After structural and thermal analysis, the geometries of both the ECU enclosure and the heat exchangers were finalized. Manufacturing drawings were prepared using SolidWorks 14.0 and the fabrication of the ECU enclosure along with the evaporator and condenser heat exchangers were outsourced to Super Radiator Coils (SRC) in Virginia. They were able to complete the entire manufacturing process along with some preliminary leak tests in 4 weeks.

Figure 21 Fabrication of evaporator and condenser heat exchangers for the ECU full scale prototype. The manufacturing of the ECU was outsourced to Super Radiator Coils in Virginia.

There are two primary reasons for conducting component level experiments before integrating the ECU with the rest of the ATB system. 1) To evaluate the Thermal performance of the ECU at different coolant flow rates and 2) To explore novel control strategies for ECU that would not only simplify the ATB system control but also provide means to tackle cabin preconditioning requirements inside an electric vehicle.
4.1 Vacuum leak test for the ECU enclosure

Super Radiator Coils were able to pressure test the heat exchanger coils at 150 Psi and they also tested for leaks inside the chamber at 3 Psi. However, in order to ensure that the ECU can hold vacuum for a long period of time (8000 Cycles) a series of vacuum leak tests were conducted at Northeastern University. The ECU was coupled to a RV-3 vacuum pump through a set of cold traps that were connected in parallel. There were two KF 40 butterfly valves in this set up and one was installed near the ECU vapor valve and the other one near RV-3 Vacuum Pump. Furthermore, PX409 series pressure transducer from Omega was installed on the pressure transducer port on top of ECU. Figure 22 shows the set up utilized for conducting vacuum leak tests for the ECU enclosure. Once the set up was complete both the valves were kept open and the vacuum pump was started. The pressure in the chamber is reduced below 100 Pa (absolute) and then the vapor control valve was closed. The data for pressure inside the chamber was collected by connecting the pressure transducer to an Agilent Data acquisition system(34980A). The chamber was left under vacuum for over 60 hours and then the data for pressure and temperature inside the chamber was collected for 24 Hours.

![Experimental set up for vacuum leak testing of the ECU (<100 Pa) using RV-3 Edwards vacuum pump and dry ice cold traps (LIT-10025) from Laco technologies.](image-url)
From Figure 23, it is clear that the ECU enclosure can hold vacuum for a considerable period of time. The small rise in pressure (0.011 Pa/Min) is because of the heat addition from the environment.

![Graph](image)

Figure 23 Vacuum leak test for the ECU prototype conducted at Northeastern University. Small variation of chamber pressure (<10 Pa pressure increase over 24 hours) with respect to time after 60 hours of vacuum suggests that the chamber is leak proof.

### 4.2 Degassing methodology

One of the factors that can affect the performance of ECU and the ATB system as a whole, is the presence of non-condensable gases (NCG). Evaporation/boiling process inside the ECU strongly depends on the wall superheat temperature. The presence of NCG in the ECU will increase the total pressure of the system as well as the saturation temperature of water. Thus reducing the wall superheat temperature considerably affecting the evaporator performance [28]. During the condensation process any NCG present in the ECU will accumulate over the condenser tube surface creating an additional resistance to heat transfer which in turn will reduce the rate of condensation significantly [29]. In order to
avoid this problem, a series of degassing cycles were carried out to remove all the non-condensable gases out of the ECU before conducting any experiments. Removal of NCG in the ECU is generally carried out by freezing the water to separate working fluid from NCG [30]. Furthermore, the solubility of NCG decreases at higher temperatures [28]. Hence the degassing procedure followed for removing NCG in the ECU includes a set of heating and cooling cycles. The experimental set up for degassing includes a RV-3 Edwards vacuum pump and dry ice cold traps from Laco technologies connected in parallel to each other. The procedure for degassing is outlined below.

1. Start vacuum pump and open the vapor control valve to reduce the pressure to 500 Pa. (Refrigerant Starts to Freeze below 600 Pa)
2. Once the chamber pressure reaches 500 Pa, close the vapor control valve and stop the vacuum pump.
3. Turn on electric heaters that are attached to the bottom plate of ECU.
4. When the chamber temperature reaches 60 °C and chamber pressure is approximately 20 kPa (saturation pressure at 60°C) turn off the heaters.
5. Start vacuum pump and open vapor control valve to reduce the chamber pressure back to 500 Pa.
6. Close the vapor control valve and allow the chamber to go back to ambient conditions (Chamber pressure 2400 Pa and Chamber Temperature 21°C).
7. Repeat step 6 and close the vapor control valve to complete degassing cycles for ECU.

Dry ice cold traps used in this experiment has a very limited capacity for capturing vapor. Once the cold traps are saturated with vapor the system pressure starts to increase. Thus
after each cooling and heating cycle, the cold traps are isolated from the set up by closing the vapor control valve and the condensed vapor is removed from the cold traps before re-installing. This process is indicated as “cold trap cleaning” in figure 24 and it ensures that the cold traps never become saturated with vapor.

![Diagram](image)

**Figure 24** Degassing procedure followed for removing non-condensable gases from liquid refrigerant (water) inside the ECU. Figure also shows the vapor control valve opening and vacuum pump operation at different pressures.

The amount of water that is lost during degassing cycles is about 2 to 2.5 Liters. Once the degassing experiments are completed the pressure and the temperature of refrigerant (water) inside the ECU very precisely matches the saturation conditions and this assures the elimination of non-condensable gases.
Figure 25  Figure shows the plot between chamber temperature and time during degassing cycles. The heaters installed at the bottom of the ECU are used for heating the ECU enclosure up to 60°C.

4.3 Thermal performance characterization

After vacuum leak test, the ECU enclosure was insulated with 1” thick nitrile rubber insulation ($k_{ins} = 0.25$ Wm$^{-1}$K$^{-1}$) to prevent heat addition into the chamber from the environment. The performance of dry ice cold traps along with vacuum pump used in this experimental set up mimics the adsorption bed characteristics. Hence the behavior of ECU with this experimental set up and the control strategies explored would be applicable when the unit is actually coupled with the adsorption bed. The proxy adsorption bed used for this experiments consists of an Evans 3 vacuum pump with a total discharge capacity of 5.7 CFM and two dry ice cold traps from Laco Technologies (LIT 10025) with a total vapor retention capacity of 2 Liters. The schematic of the experimental set up is shown in Figure 26.
Figure 26 Schematic of experimental set up utilized for the ECU thermal performance testing. Basic idea behind this experimental set up is that the dry ice cold traps along with vacuum pump mimics the performance of adsorption bed.

Furthermore, two electric heaters (1150 Watts each) submerged in a coolant bath (5 liters capacity) and powered by a varying AC power supply were used as thermal load for the experiments. Two Shurflo diaphragm pumps (5050-1310-D011) connected in parallel and controlled by a varying DC power supply (BK1900 Precision) is utilized for pumping coolant (50-50 ethylene glycol-water mixture) at different flow rates into and out of the chamber. A flow meter (FPD 3004) was used to measure the flow rate and two PX 409 series pressure transducers from Omega were installed for measuring pressure inside the ECU and also in the cold trap line. Furthermore, a set of two 4-Wire RTD’s (P-M-1/10-1/4-5-0-PS-4) were installed right next to coolant ports for measuring the coolant inlet and outlet temperatures for evaporator heat exchangers. In addition to this, one K-type thermocouple was introduced into the chamber for measuring vapor temperature and two more K-type thermocouples were installed on top and bottom of the insulation to measure the losses.
Figure 27 Actual experimental set up integrated at Northeastern University for the ECU thermal performance characterization. Coolant pumps are connected in parallel to deliver a maximum flow rate of 20 lit/min during experiments.

One of the target performance metrics for ATB system during cooling mode is to provide chilled coolant at 13° C to the climate core heat exchanger while maintaining a thermal power of 1600 watts across the evaporator for 4 hours.

Thus in this project, evaporator performance is evaluated by its ability to provide chilled coolant at various thermal powers during steady state conditions. For evaluating the ECU thermal performance, the inlet and outlet ports of evaporator heat exchanger were connected to the coolant lines. Initial set of experiments were conducted at a coolant flow rate of 20 lit/min and the input power to the electric heaters were controlled by an AC variable transformer. Before starting each experiments, care was taken to ensure that the level of refrigerant (distilled water) inside the ECU was always the same (30 mm from the top surface of evaporator coils). Hence the saturation pressure near the heat exchanger
surface between experiments is always constant. After this step, the degassing cycles are carried out to eliminate the effect of non-condensable gases and during this process, the amount of water lost inside the ECU is 2 to 2.5 Liters. Thus before the start of each experiment the height of water column from the top surface of evaporator coils is 24mm. The vapor valve is closed after the last degassing cycle and then the coolant is circulated inside the chamber at 20 lit/min without turning on the electric heaters. Once the system reaches ambient conditions (chamber pressure and temperature at 2400 Pa and 21° C) and the coolant inlet and outlet temperatures becomes steady, the electric heaters are turned on by applying a constant voltage.

The vapor control valve is then slowly opened to drop the pressure inside the chamber. As soon as the chamber pressure approaches 800 Pa, the vapor valve is adjusted accordingly to keep the pressure constant at 750 Pa. This in turn will regulate all the other parameters within the system and the experiment is continued until the system reaches steady conditions. The same experimental procedure is then repeated for different thermal powers and at different coolant flow rates.
Chapter 5: Results and discussion

5.1 Boiling performance of enhanced surface tubes

Figure 28 shows the plot between heat flux and average wall superheat for two types of enhanced surfaced tubes from Wolverine Inc. and also compares it with the Rohsenow’s model. It is evident from the graph that Rohsenow’s model was under predicting the performance of the Tube 2 (S/T Trufin®). Tube 2 with regular circular fins that are closely spaced on the outside diameter seems to perform much better than Tube 1 which has staggered fins on its outside diameter. One of the reasons for this may the larger surface area of Tube 2 that is available for convection at these low superheat temperatures.

Heat fluxes measured for Tube 2 were approximated using a third order polynomial function of the wall superheat ($\Delta T_e$), $q''_{evap/boiling} = 108.77\Delta T_e^3 - 1315.9\Delta T_e^2 + 7389.8\Delta T_e - 14394$. This approximation (shown as average model in Figure) was fed into the finite difference model (FD) developed at Northeastern University to determine a heat transfer rate of about 2800 W (a safety factor of 1.7 was used for the target thermal power) when the coolant enters the evaporator at 10.3°C and 20 lit/min and the chamber pressure is at 750 Pa.

At low superheat temperatures (4-7°C), the variations in heat flux for the same wall superheat temperature is significant. Hence a worst case scenario of thermal performance was evaluated using the lowest heat flux measured when the superheat temperature was between 4 and 7°C. This model is labelled as lower limit model in the figure 28. Here the heat flux and superheat has an exponential relationship, $q''_{evap/boiling} = 8.1717 exp (1.052\Delta T_e)$. This model was then fed into the FD model and the evaporator heat transfer
rate was 1610 Watts (no safety factor for the target thermal power) when coolant enters at 10.3°C and at 20 lit/min with chamber pressure at 750 Pa.

Figure 28 Variation of average wall superheat with heat flux for tube 1 and tube 2. Results show that Rohsenow’s model clearly under predicts the performance of Tube 2 at low pressure (750 Pa).

5.2 Thermal performance of full scale ECU prototype

ECU performance experiments were conducted for different thermal powers (450 – 1200 Watts) and at two different flow rates (10 Lit/Min and 20 Lit/Min). The thermal power was evaluated using the following equation,
\[ P_{th} = \dot{m}_{\text{coolant}} C_{p_{\text{coolant}}} (T_{\text{coolant-in}} - T_{\text{coolant-out}}) \]  

(6)

Where \( \dot{m}_{\text{coolant}} \) is the mass flow rate of coolant (Kg/sec), \( C_{p_{\text{coolant}}} \) is the specific heat capacity of coolant (J/kg K), \( T_{\text{coolant-in}} \) \& \( T_{\text{coolant-out}} \) (K) are the coolant inlet and outlet temperatures, respectively.

As soon as the experiment is started with the vapor control valve in the fully opened position, there is a high peak in the thermal power mainly because of the thermal mass of the coolant tank utilized in the experimental set up. This transient behavior may differ during ATB system testing. Figure 32 shows the variation of thermal power with time during the evaporator performance experiments for three different powers.

![Image](image_url)

**Figure 29** Thermal Power Vs Time shows the Transient Behavior of ECU during performance evaluation experiments. The initial peak in the thermal power is because of the thermal mass of the coolant tank utilized for this experiments.
Even though the transient behavior of ECU will vary when the proxy set up (vacuum pump and dry ice cold traps) is replaced by the adsorption bed, the steady coolant temperatures (inlet and outlet) will remain the same for every thermal power evaluated (450 to 1200 Watts) during these experiments. This knowledge of steady coolant outlet temperature leaving the ECU at different powers will be very useful for maintaining the passenger cabin at the desired temperatures for climate control.

Figure 30 Thermal Power Vs Coolant Outlet Temperature at Steady State for Two different flow rates (10Lit/Min and 20 Lit/Min). The maximum power tested (1270 Watts) was limited by the capacity of vacuum pump and cold traps.

Figure 33 shows the variation of coolant outlet temperature with thermal power at two different flow rates. The results obtained demonstrated a coolant outlet temperature <11°C and evaporation thermal powers up to 1270 Watts at steady state conditions. The maximum thermal power tested with this set up is limited by the pumping power of the vacuum pump.
and cold trap capacity. However, these results suggest that the ECU could provide coolant at 13°C while maintaining a thermal power of 1600 Watts across the evaporator which in turn is one of the performance metrics for ATB.

5.3 Optimization of vapor control valve for climate control in EVs

The ability to control the extent of cooling and heating supplied to the EV-cabin is one of the requirements for practical implementation of ATB system [10]. Experiments from thermal performance evaluation of evaporator revealed that at each steady power there was a corresponding angle for the vapor control butterfly valve at which the entire system reached steady state conditions. As the steady state thermal power increased, the valve angle also increased towards the fully open position. This formed the baseline for the series of control strategies that were explored. However, with the available vacuum pump exhaust rate and cold traps capacity the thermal power tested was limited to 1300 watts.

<table>
<thead>
<tr>
<th>Thermal Power (Watts)</th>
<th>Steady Vapor Valve Angle for 20 LPM (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>430</td>
<td>29</td>
</tr>
<tr>
<td>525</td>
<td>35</td>
</tr>
<tr>
<td>640</td>
<td>40</td>
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<tr>
<td>750</td>
<td>43</td>
</tr>
<tr>
<td>930</td>
<td>50</td>
</tr>
<tr>
<td>1276</td>
<td>60</td>
</tr>
</tbody>
</table>

Figure 31 Vapor control butterfly valve used for full scale ECU thermal performance characterization experiments on the left and variation of valve angle with thermal power on the right.
Experiments were also conducted at two different flow rates for a given steady state thermal power. The variation of vapor control valve angle with steady state thermal power and at two different flow rates is shown in figure 29. It is evident from the figure that the vapor control valve angle is independent of changes in coolant flow rate.

![Figure 32](image)

**Figure 32** Variation of Vapor Control Valve Angle with Thermal Power for Two Different Flow rates. The results suggest that the valve angle is independent of coolant flow rate.

In order to find the effect of refrigerant level on valve angle, experiments were conducted at three different refrigerant levels from the top surface of the evaporator coils. These experiments revealed two important results 1) the steady state coolant outlet temperature is not constant at different refrigerant level for a given thermal power and 2) the vapor control valve angle remained the same for a given thermal power at different refrigerant levels. Thus the valve angle is independent of both coolant flow rate and refrigerant level.
This dependability of valve angle on only one parameter reduces the ATB system complexity in terms of control. Figure 30 shows the variation of coolant outlet temperature at different refrigerant levels from the top surface of evaporator coils inside the ECU.

![Figure 30](image)

**Figure 30** Variation of coolant outlet temperature at different refrigerant levels from the top surface of evaporator coils inside the ECU.

5.4 Control Strategies for vehicle-preconditioning and cabin climate control

Vehicle preconditioning is one of the common features of modern EVs and it allows the passenger to start their journey with a desired cabin temperature [31]. So in order to achieve this using the ATB system, the ECU should deliver very low coolant temperature at a very high thermal power for a short period of time.
The initial transient behavior of the ATB system at the beginning of the adsorption process provides a window of opportunity for extracting high thermal power while supplying coolant at a fairly low temperature (13°C). Hence the main objective of the different control strategies explored was to modulate this high initial transient power to a lower value but for longer periods of time either by controlling the pressure (by throttling the vapor valve) or by controlling the coolant flow rate. A third hybrid control strategy of controlling both pressure and coolant flow rate in order to maintain peak initial power was also pursued during the component level testing of the ECU.

Besides the need to extract high initial transient power and acquiring the desired coolant outlet temperature, the fundamental reason to throttle the vapor valve and control the pressure drop inside the chamber is to avoid freezing of the refrigerant (water). Hence the basic control for the ECU involves manually throttling the vapor control valve when the system pressure approaches 800 Pa and then controlling the valve opening until the pressure reaches a steady value of 750 Pa. In this base-line control, the flow rate is kept constant at a design value of 20 lit/min and only the chamber pressure is controlled to reach the steady state conditions. The figure 34 depicts the pressure and time at which the vapor control valve was manually throttled in order to reach a steady state pressure of 750 Pa. This strategy along with the flow control can greatly reduce the complexity and cost of the ATB system control.
Figure 34 The base-line control strategy for the ATB system requires the control of the vapor valve until a steady evaporation pressure is achieved. This figure shows the variation of pressure (top left), coolant outlet temperature (to right), coolant flow rate (bottom left), and Thermal power (bottom right).

In the next control strategy (ALT 1) the knowledge of vapor control valve angle for achieving a given thermal power along with flow control were utilized to maintain the peak initial transient power for considerable period of time before reaching the steady state power. In this strategy (ALT 1), the experiment was started at a low coolant flow rate (15 lit/Min) and then it was gradually increased to maintain the peak initial transient power until it reached a target flow rate of 20 Lit/min. Once the flow rate reaches its maximum value, the vapor valve was immediately set at a particular angle to reach the corresponding steady state thermal power. Figure 35 depicts this first hybrid control strategy.
Figure 35 ALT 1 control strategy (maximization of thermal power) This hybrid control strategy for the ATB system targets to have a constant thermal power delivered during the beginning of the evaporation experiment. This figure compares ALT 1 strategy with the Original strategy (base-line control) on the temporal variation of pressure (top left), coolant outlet temperature (to right), coolant flow rate (bottom left), and Thermal power (bottom right).

This control strategy (ALT 1) provides an opportunity to extract high initial thermal power for a considerable period of time and save energy (reducing the amount of refrigerant adsorbed) by reaching the steady state power at a much faster rate than the original control strategy. One of the drawback of this control is that the coolant temperature reduction rate is lower than the original control strategy (>13°C for more than 30 minutes) which will affect the temperature of the air blown into the EV passenger cabin.
The second alternative control strategy (ALT 2) involves starting the experiment with no coolant flow and allowing the system pressure to drop to 1500 Pa. The system pressure is lowered to this target pressure (1500 Pa) within a few seconds from the start of the experiment as there is no evaporation/boiling inside the ECU. As soon as the system reaches 1500 Pa ($T_{\text{sat}}=13^\circ\text{C}$), the coolant is pumped into the system at a very low flow rate (10 Lit/min) and then it is gradually increased to 20 lit/min to maintain the peak initial transient power. Meanwhile, the system pressure keeps dropping and then the vapor control valve is set at a pre-determined angle when the system reaches 700 Pa. As soon as the valve is throttled from the fully opened position to the target angle, the pressure in the system spikes up initially and then equilibrates to the steady state pressure of 750 Pa.

This control strategy provides the possibility of maintaining a high initial thermal power for a considerable period of time while at the same time lowering the coolant outlet temperature at a much faster rate than the previous control strategies (< 10°C in 15 mins). Hence this strategy will have an important impact in the preconditioning process of an electric vehicle. In addition to this, reducing the ECU pressure from ambient conditions (2400 Pa) to lower pressure (1500 Pa) without any coolant flow has significantly reduced the amount of refrigerant adsorbed. Hence by adopting this control strategy, the ECU can provide coolant at lower temperatures (< 15°C) for a longer period of time when compared with the previous strategy. At peak summer conditions the ECU should be able to cool the cabin air temperature from 43°C to 21°C in a short period of time and the initial ECU pressure will be 6000 Pa ($T_{\text{sat}} = 36^\circ\text{C}$) before the start of adsorption. ALT 2 control will be the best strategy to use in order to ensure that the ECU can meet preconditioning requirements and also provide coolant at a lower temperature for a longer period of time.
Figure 36 ALT 2 Control Strategy (faster reduction of coolant temperature) This hybrid control strategy for the ATB system targets to have a higher temperature reduction rate of the coolant leaving the evaporator heat exchanger. This figure compares this ALT 2 strategy with the Original strategy on the temporal variation of pressure (top left), coolant outlet temperature (to right), coolant flow rate (bottom left), and Thermal power (bottom right).

Exploration of these control strategies can eliminate the need for expensive/complex control systems and feedback devices which in turn can make the system more robust and cost effective. Figure 37 compares all the three control strategies in terms pressure, flow rate, coolant outlet temperature and thermal power.
Figure 37  This figure compares ALT 1 and ALT 2 control strategy with Original strategy on the temporal variation of pressure (top left), coolant outlet temperature (to right), coolant flow rate (bottom left), and Thermal power (bottom right).
Chapter 6: Conclusions and future work

Experimental determination of average boiling heat transfer coefficient has refined the design of the ECU. However, because of the large fluctuations in heat flux and average wall superheat during the experiments, a very high safety factor (1.5) has been used to ensure that the ECU can meet the target performance metrics of the ATB system.

The final ECU enclosure design was modeled after several fitting exercises to ensure that the entire unit fits inside the hood of an electric vehicle and also facilitate vapor transport to the adsorption bed without any major mass transfer resistance. The fabrication of the ECU was outsourced to Super Radiator Coils in Virginia and they were able to manufacture the entire enclosure with preliminary leak tests in 4 weeks.

Even though the ECU design meets the size metrics of the project, the volume and weight of the enclosure is still on the higher end. One way to reduce the weight is to use a lightweight enclosure material like aluminum instead of stainless steel. However, in order to use a lightweight material like Aluminum, a special coating is required for the enclosure walls to prevent its interaction with refrigerant (water) so that there are no generation non-condensable gases.

The accuracy of evaporator characterization experiments for predicting the boiling heat transfer coefficient at low pressures can be improved by slightly altering the experimental set up. The major fluctuations in the heat flux for a given wall superheat can be reduced significantly by utilizing cartridge heaters to heat the tube surface. This also prevents the variation of surface temperature along the length of the heat exchanger.
From the ECU thermal performance evaluation, it was evident that for each and every thermal power there was a corresponding angle for the vapor control valve at which the entire system reached steady state conditions. This valve angle was independent of both coolant flow rate and refrigerant level inside the ECU and was dependent only on the thermal power. This dependability of valve angle on only one parameter reduces the ATB system complexity in terms of control.

ECU full scale prototype testing also showed that the ECU could provide coolant at a temperature $<11^\circ C$ and maintain a thermal power of 1270 watts during steady state conditions. The thermal power tested was limited to 1300 watts because of the vacuum pump and cold traps capacity. However, these results suggest that the ECU could deliver coolant at $13^\circ C$ to the climate core heat exchanger while maintaining a thermal power of 1600 watts across the evaporator thus meeting the performance metrics for the ATB system during cooling mode. The knowledge of coolant outlet temperatures at different thermal power can be useful for maintaining desired cabin temperatures.

The different control strategies explored during the component level testing of the ECU has their own advantages/ disadvantages and all these strategies can be implemented in the ATB system operation based on the power requirements and the desired cabin air temperature. These strategies have significantly reduced the complexity, weight and cost of the vapor control valve (initially estimated as $6500 by MKS Instruments).
REFERENCES


S. Narayanan, H. Kim, A. Umans, S. Yang, and X. Li, “A Thermophysical Battery for Storage-based Climate Control†,” pp. 1–14 unpublished.


