

Utilization of Ice Storage in Secondary Loop Automotive Air-Conditioning Systems

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ABSTRACT

Due to the influence of energy use on electric vehicle range, latent energy storage options could be used to increase thermal comfort and decrease energy consumption during driving. This study focuses on the implications of thermal storage on transient performance of a typical secondary loop system and a combined secondary loop with ice storage system. The use of ice storage in assisting the vapor compression cycle during cabin pull-down and continued cooling, as well as cooling during compressor off periods was experimentally investigated. It was found that the ice storage system was able to decrease energy consumption during pull-down by about 20% and decrease time to comfort by about 15% compared to a regular secondary loop system. Ice storage showed superior performance for preventing cabin warm-up and reducing cabin humidification during off-cycle periods of up to 15 minutes, while showing the potential to either replace or severely reduce the use of a vapor compression system during daily commutes when used in combination with cabin preconditioning. While the current ice storage box was found to be oversized and the heat exchanger could be optimized further, experimental results give an outlook on the possible use of ice storage as an effective method of reducing air conditioning energy and enhancing thermal comfort.

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INTRODUCTION

Thermal storage in automotive air-conditioning has received more attention in recent years for several reasons. Due to increased use of start/stop operation in internal combustion engine passenger vehicles, the lack of cooling capacity during off-cycle periods needs to be addressed. Another advantage of thermal storage may be the availability of a larger initial cooling or heating capacity when the storage medium is charged. Some air-conditioning systems, such as a secondary loop system have a certain amount of thermal storage "built in", due to their increased thermal mass compared to direct expansion systems. Secondary loop systems in automotive settings have been investigated by Ghodbane et al. [1]. The system was tested in several prototype vehicles. Ghodbane et al. [2] used 3.2 liters of a water-glycol mixture as secondary working fluid. The "built in" thermal storage capability of a secondary loop systems was shown. Lemke et al. [3] recently studied the implementation of paraffin-based phase change materials in a secondary loop system. Charging and discharging processes of a compact PCM unit during a virtual test drive were shown in terms of passenger compartment temperature.

The present research strives to quantify thermal storage benefits in terms of thermal comfort, power consumption and system efficiency during pull-down scenarios, as well as cabin warm-up during off-cycle periods. Combination of thermal storage concepts with energy efficiency concepts, such as cabin preconditioning and alternative air-conditioning system controls is experimentally investigated.

TEST FACILITY

Several test facilities were built for the present study. A direct expansion (DX) system featuring a variable speed fixed displacement compressor, a microchannel condenser, an electronic expansion valve, and a plate-and-fin type evaporator was built and tested with R134a as refrigerant as a baseline for comparison purposes. Subsequently, the facility was modified by replacing the evaporator with an intermediate heat exchanger which cools down a secondary working fluid, as well as adding a secondary loop (2LP), as described below. R290 was used as refrigerant in the vapor compression cycle (VCC) of the secondary loop system.

Experimental Test Bench

The secondary loop test facility is described in [Figure 1](#). The main components are marked with numbers. The vapor compression cycle included a compressor (1), a condenser (2), an electronic expansion valve (3) and an intermediate heat exchanger (4). The automotive compressor was belt driven by an electric motor (6), enabling a variable speed control of the compressor. A torque meter combined with an rpm meter allowed for measurement of compressor power consumption. The condenser was housed in a windtunnel (11) for air-side capacity measurement. The condenser wind tunnel, as well as the compressor and associated lines were set up within an environmental chamber to simulate ambient climate conditions. A variable speed pump (8) was used to circulate the chilled coolant from the intermediate heat exchanger through a liquid to air heat exchanger, the air cooler (5). The cooler was housed in a closed loop featuring an air handling unit (AHU) (11). In an automobile, the cooler would be located underneath the dashboard to provide cool air to the passenger cabin. A reservoir (9) was used as pressure expansion tank. Temperatures and coolant mass flow rate were measured to determine capacities of the chiller and the cooler in the secondary loop.

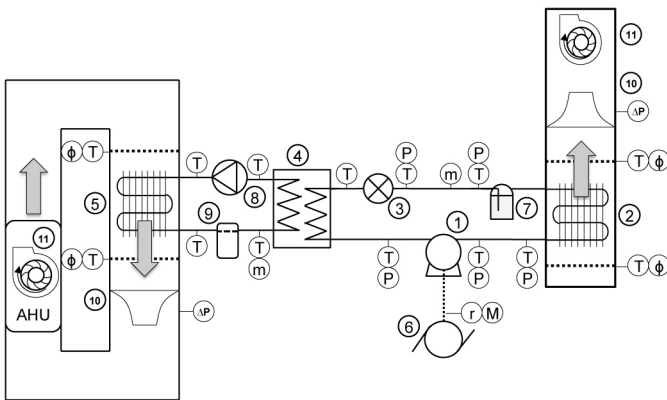


Figure 1. Secondary Loop Test Facility Schematic

The secondary loop test facility was subsequently modified to include an ice storage box. A copper tube-and-fin heat exchanger was built to fit a 15 liter box. The amount of water used was determined by experimental results of energy needed to cool down the passenger cabin of a midsize sedan driving at highway speed for a range of ambient temperature between 30°C and 41°C. At 41°C 19.8 kg of ice would be sufficient, while at 30°C 8.6 kg would be needed if the ice storage could be used as stand-alone without a VCC. The amount of 15 liters of water was chosen for a first generation test. The ice storage heat exchanger is shown in [Figure 2](#).

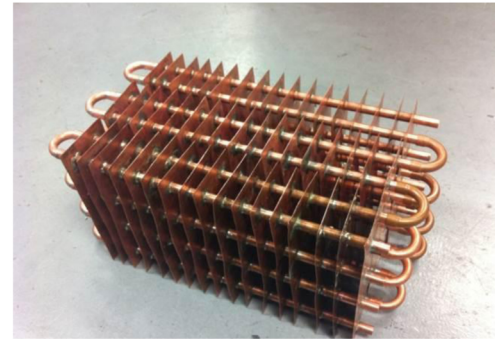


Figure 2. Ice Storage Heat Exchanger

The modified test facility is shown in [Figure 3](#). The figure omits the primary loop, i.e. the VCC, since the refrigerant-side was not modified. The main changes to the secondary loop facility are the introduction of an ice storage box (3) and the integration of three sets of ball valves (6), which allow for four operational models of the facility. Mode 1 bypasses the ice storage heat exchanger, so as to use only the basic secondary loop system for cooling. Mode 2 is a charging mode in which the cooler is bypassed to allow the cooling capacity of the VCC to be used for ice production. Mode 3 enables the combined capacity of the VCC and the ice storage to cool the passenger cabin. Mode 4 forces cooling of the passenger cabin to be performed by the ice storage box only.

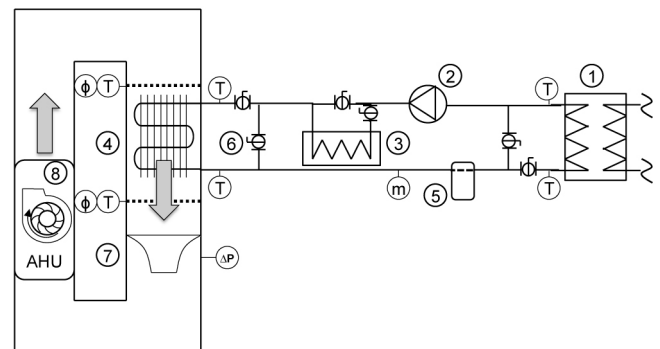


Figure 3. Combined Secondary Loop and Ice Storage Facility Schematic

Data Acquisition and Controls

Temperature, relative humidity, and air flow rate in the condenser wind tunnel and evaporator psychrometric loop, could be controlled manually for steady-state tests, or by PID control for transient tests. For drive cycle tests, the electric motor driving the compressor and the condenser fan were controlled by a PID controller, while the compressor clutch was actuated by a digital on/off control.

A lumped passenger cabin model was developed, based on the work of Gado [4]. By real-time communication with the controller of the air-handling unit upstream of the evaporator (DX) or cooler (2LP), the cabin model enables transient testing. As shown in [Figure 4](#), the core of the model

consists of a set of energy and mass balance equations. In every timestep, temperature, relative humidity, and air flow rate downstream of the evaporator (DX) or cooler (2LP) were measured. Based on physical and thermal characteristics of the cabin, as well as thermal loads, the cabin model determined temperature of the cabin air, temperature of interior mass, and relative humidity inside the cabin. The cabin air was then allowed to mix with ambient air, based on a recirculation ratio. The recirculation ratio of a MAC system is typically controlled automatically based on operating conditions. The return temperature and relative humidity determined by the cabin model were subsequently used as set points for the PID control of the air handling unit (AHU), which controlled the condition of the air upstream of the evaporator (DX) or cooler (2LP). This procedure was repeated every timestep and allowed for the simulation of a real passenger cabin and therefore transient testing in a laboratory setting.

A thermostat control, using a virtual heater core, was added to allow for a stable cabin temperature at the cabin temperature setpoint. Despite the possibility to control cabin temperature within limits by changing coolant flow rate, a low evaporation temperature is often needed for dehumidification, which necessitates a series reheating approach. The virtual heater core used a software side PID controller which added a certain magnitude to the actual, measured supply temperature downstream of the evaporator/cooler. The cabin model energy and mass balance equations therefore received an artificially higher supply temperature signal. This resulted in an increase of the calculated return temperature in the model, and therefore the upstream evaporator air temperature, controlled by the AHU. Therefore, within one timestep, the virtual heater core would lead to a real increase in process temperature, enabling the facility to work as if an actual heater core were present. The heater core load was subsequently determined by using the virtual temperature difference across the heater.

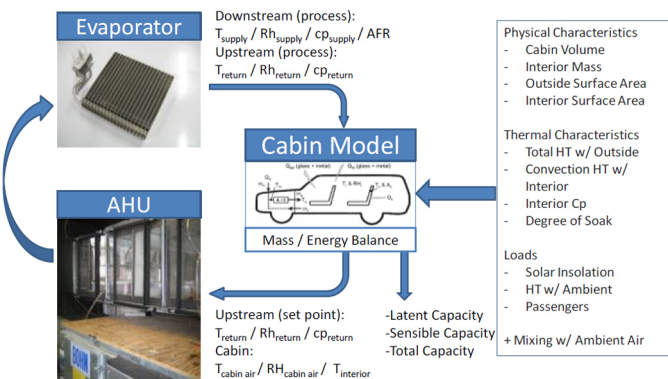


Figure 4. Cabin Model Operational Structure

EXPERIMENTAL RESULTS

Ice Storage Charging

To form ice, the cooler of the secondary loop was bypassed and the vapor compression cycle cooled part of the secondary loop to sub zero temperatures. The water-glycol mixture was routed through the ice storage heat exchanger to charge the ice storage box. Thermocouples were placed in the storage box to evaluate average temperature during charging and during cooling process. It was observed that the charging time for the ice storage box was about 60 minutes if the initial average temperature was above 10°C. This time is subject to the mass of ice. Sensibly cooling the ice stretches only a small portion of the overall time, so that charging time does not increase significantly at higher initial temperatures. Figure 5 shows energy used for charging the ice storage box with respect to initial average temperature. It was observed that about 1,500 Wh are needed to freeze the 15 L of water and subcool the ice to -6°C average storage box temperature. At average initial temperatures close to 0°C, parts of the box can already/still be frozen and the relation of charging energy to initial temperature becomes unreliable. The transient performance factor (TPF) is a Coefficient of Performance (COP), which relates the integrated air-side cooling capacity over the time of a test with the integrated power consumption, as defined by Equation (1).

$$TPF = \frac{\int_{t=0}^t \dot{Q}}{\int_{t=0}^t P_{total}} \quad (1)$$

TPF does not change significantly with initial average temperature. On average, ice charging TPF was measured to vary between 0.55 and 0.7. Ice charging TPF and charging time may be improved by optimizing the ice storage heat exchanger, optimizing the amount of ice needed for a specific application, and take charging performance into considerations when designing the VCC.

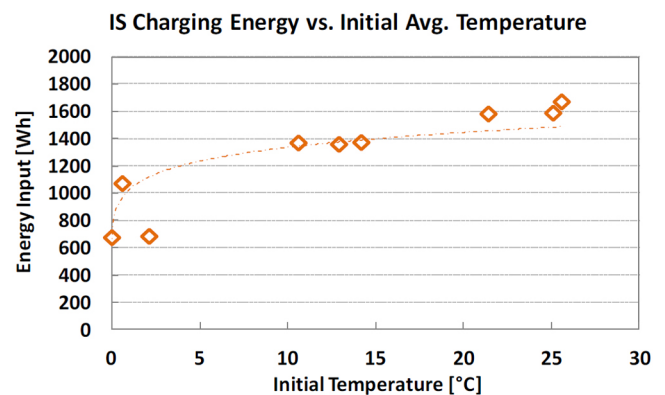


Figure 5. Ice Box Charging Capacity versus Initial Temperature