Condensate Retention Effects on the Air-Side Heat Transfer Performance of Automotive Evaporator Coils

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Abstract

The effect of condensate accumulation and shedding on the air-side thermal performance of automotive evaporator coils has been studied. Experiments under wet and dry conditions were conducted to expose the impact of condensate on five different coils. Condensate retention data were collected in both real-time and at steady-state to quantitatively determine how condensate load up on a coil surface. Sensible Colburn $j$ factors and friction factors were calculated from the experimental data, and the relative performance of different coils was discussed. A dynamic drainage test was developed to study the nature of water draining out of a heat exchanger. It was found the simple drainage test qualitatively predicted how much condensate would be retained by different coils under operating conditions. Current retention modeling techniques were adapted to include automotive evaporators.
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Nomenclature

$A$  area ($m^2$)

$A_{drop}$  area of a droplet ($cm^2$)

$A_{fr}$  frontal area ($m^2$)

$b$  intercept of least squares fit line

$C$  constant of integration

$C_d$  drag coefficient

$C_p$  specific heat at constant pressure (kJ/kg-K)

$D$  diameter ($m$)

$D_{AB}$  binary mass diffusion coefficient ($m^2/s$)

$D_{drop}$  diameter of droplet ($m$)

$D_h$  hydraulic diameter ($m$)

$\Delta P_{HX}$  heat exchanger differential pressure (kPa)

$f$  friction factor

$f_s$  fin spacing (mm)

$F_d$  air drag force ($N$)

$F_g$  gravitational force ($N$)

$F_s$  surface tension force ($N$)

$g$  gravitational acceleration ($9.81 \, m/s^2$)

$G$  mass velocity based on minimum free flow area ($kg/m^2-s$)

$h$  enthalpy (kJ/kg)

$h_o$  air-side heat transfer coefficient ($W/m^2-K$)

$h_i$  coolant-side heat transfer coefficient ($W/m^2-K$)

$h_m$  mass transfer coefficient ($kg/m^2-s$)

$j$  sensible $j$ factor

$k$  thermal conductivity ($W/m-K$)

$L_f$  length or depth of fin ($m$)

$Le$  Lewis number

$\cdot$  mass flux ($kg/s$)

$m_o$  fin efficiency parameter
\( Nu \)  Nusselt number
\( Pr \)  Prandtl number
\( q \)  heat transfer rate (KW)
\( R \)  thermal resistance (K/W)
\( R_c \)  coolant flow meter output (pulse/s)
\( Re \)  Reynolds number
\( T \)  temperature (°C)
\( t \)  time (s)
\( V \)  velocity (m/s)

**Greek symbols**

\( \alpha \)  angle of inclination (radians)
\( \delta \)  fin thickness (m)
\( \phi \)  relative humidity
\( \gamma \)  surface tension (mN/m)
\( \eta \)  fin efficiency, surface effectiveness
\( \mu \)  dynamic viscosity (N-s/m\(^2\))
\( \theta \)  dimensionless temperature
\( \theta_A \)  advancing contact angle (radians)
\( \theta_M \)  mean contact angle (radians)
\( \theta_R \)  receding contact angle (radians)
\( \rho \)  density (kg/m\(^3\))
\( \sigma \)  contraction ratio (\( A_{\text{min}}/A_{\text{fr}} \))

**Subscripts**

\( \text{air} \)  air
\( \text{atm} \)  atmospheric pressure (atm)
\( \text{ave} \)  average
\( \text{c} \)  coolant
\( \text{cali} \)  calibrated
\( \text{dp} \)  dewpoint
dry  dry condition
f    fin
fr   frontal
i    tube-side
in   inlet
l    liquid
mair mean air
min  minimum
max  maximum
o    air-side
out  outlet
sens sensible
t    total
wet  wet condition
Chapter 1 Introduction and Literature Review

1.1 Introduction

In automotive air-conditioning systems, the air-side surface temperature of the vapor-compression evaporator is usually below the dew point of the conditioned air, and it is common for water to condense onto the air-side heat-transfer surface. Condensate accumulates on the surface and is retained by surface tension until it is removed by either gravitational or flow forces. Retained condensate plays an important role in the overall performance of the air-conditioning system; it can profoundly affect the heat transfer and pressure drop performance, but there is disagreement on the overall impact in some heat exchanger geometries. Condensate retention also has important implications on air quality. Condensate that blows off with the conditioned air stream can affect comfort, and water provides a medium for biological activity on air-handling surfaces. The post-operation (off-cycle) condensate draining behavior is extremely important in this respect, because the warm, moist conditions prevailing after system shut down are conducive to biological growth.

The focus of this project was on the effect of condensate on air-side thermal-hydraulic performance, with an overall goal to develop new fin-design guidelines that maximize performance under dehumidifying conditions. A wind tunnel was designed and constructed for testing heat exchangers under dry and condensing conditions. Experiments were conducted to obtain steady state and real-time measurements of condensate retention. Furthermore, heat transfer and pressure drop data for heat exchangers under dry and condensing conditions were recorded. Extensive drainage tests
were conducted using a dynamic drainage test apparatus. The data from the experiments were used to aid in development and validation of a retention model.

1.2 Literature Review

The majority of the work on air-side thermal performance under dehumidifying conditions has concentrated on heat exchangers with round tubes, as opposed to the flat-tube, brazed-plate geometry of this study. Nevertheless, the prior work is helpful in developing an understanding of the impact of condensate retention and shedding. This section includes a discussion of past work on flat plates, finned-tube coils with plain and enhanced fins, and the limited work in the open literature on automotive-style evaporators. Finally, a discussion of condensate retention, drainage, and modeling is presented.

1.2.1 Early Studies

Bettanini (1970) performed numerous experiments in heat and mass transfer for a vertical plate and reported an enhancement in sensible performance under condensing conditions. Bettanini postulated the effect was caused (in part) by an increase in surface roughness caused by condensation on the surface. Two types of experiments were conducted to verify this effect. A soap and water solution was sprayed on the surface to temporarily cause filmwise condensation, and heat transfer measurements were taken until the condensation turned to dropwise, thus increasing surface roughness. The sensible heat transfer coefficient increased by approximately 20% when the condensation turned to dropwise, supporting the roughness effect idea. Additional experiments were done using gypsum chips to simulate water droplets on the surface. The results from these tests showed approximately a 10% increase in performance. While these tests do show an
impact from surface roughness, the experimental apparatus and procedure used was simple and not well controlled (as noted by Bettanini), and the extension of these results to more complicated heat exchangers and flow regimes is unclear.

Yoshii et al. (1973) examined the effects of dropwise condensation on the pressure drop and heat transfer performance of wavy-fin heat exchangers. It was found that pressure drop for wet heat exchangers was 50 to 100% higher than for dry exchangers. Under wet conditions, a 20 to 40% enhancement in heat transfer coefficient was found for the heat exchangers. To investigate the effect of condensate on the flow dynamics, scale-up models of the heat exchangers with simulated condensate were made and tested in a water channel at similar Reynolds numbers. Yoshii and coworkers reported that drops on the flat fin surface promoted turbulence when the droplet adhered on the ridge or valley in the wavy fin, but caused separation when adhered to the area between bends. Additionally, water bridges between the tube and fins significantly increased the wake region downstream of the tube, but downstream droplets can direct flow into the wake region. From these observations Yoshii and coworkers concluded the overall impact on condensate depends on both location and shape of the droplets.

Guillory and McQuiston (1973) and McQuiston (1976) studied developing flow between horizontal flat plates and found a heat transfer enhancement of about 30% for wet surface conditions. In agreement with Bettanini, Guillory and McQuiston explained that the condensate that formed on the heat exchanger increased the surface roughness of the exchanger walls and this increased roughness explained the increase in heat transfer and pressure drop found under wet conditions. Tree and Helmer (1976) also studied a parallel plate heat exchanger under condensing conditions. Unlike Guillory and
McQuiston, they found that condensation did not affect the sensible heat transfer and pressure drop during laminar flow. However, agreement was found in the transitional and turbulent regime, where condensate was found to increase heat transfer and pressure drop. For plain-fin-and-tube geometries, Myers (1967), Elmahdy (1975), and Ekels and Rabas (1987) have reported a sensible enhancement under wet conditions.

Inconsistent with a simple roughness effect, McQuiston (1978a,b) found the enhancement in plain finned-tubes to be strongly dependent on fin spacing. For circular-finned tubes, Jacobi and Goldschmidt (1990) found the enhancement to be Reynolds number dependent. A degradation was observed at low Reynolds numbers, and an enhancement was found at high Reynolds numbers. Jacobi and Goldschmidt suggested that their results, and those of McQuiston, were due to condensate retention. At low Reynolds numbers, retained condensate would occupy heat exchanger area with a deleterious effect, but at high Reynolds numbers, vapor shear would remove retained condensate and roughness effects of the remaining condensate would dominate. This explanation has since been supported by the work of Uv and Sonju (1992).

Further complicating the issue, spatial variations of the dry-surface local heat transfer coefficient may or may not have a significant impact on the fin efficiency (Huang and Shah, 1991; and Kearney and Jacobi, 1996), and the same could be true for a wet fin. Hu et al. (1994) conducted detailed local heat transfer experiments with simulated condensate. Their results indicate that for circular fins the effect on fin efficiency is small, as did Kearney and Jacobi, but the overall impact of condensate retention on average heat transfer can be significant. For circular fins, the average sensible heat transfer coefficient
can be increased by as much as 30% due to condensate effects at high Reynolds numbers. The impact in other geometries (e.g., louvered fins) remains unclear.

Hong (1996) examined the use of hydrophilic coatings to improve wettability and thereby decrease the pressure drop associated with wet-fin operation. Wavy, lanced, and louvered fins were studied, and at fixed face velocity of 2.5 m/s, the ratio of wet-to-dry pressure drop was 1.2 for each geometry tested. A model to predict the carry-over velocity was developed and compared to experimental data. Carry-over velocity is dependent on surface tension forces that depend on contact angle. Hong presented contact angle data obtained from a sessile drop goniometer test; however, because a static test procedure was adopted, no measure of contact angle hysteresis was obtained—a result between the advancing and receding is all that can be achieved through such an approach. Hong found that after approximately 1,000 wetting cycles the coated and uncoated test surfaces all exhibited contact angles of approximately 60 degrees.

Korte and Jacobi (1997) studied the effects of condensate retention on the air-side performance of plain-fin-and-tube heat exchangers. Experiments were conducted under dry conditions and then repeated under condensing conditions. It was found that the heat transfer performance under condensing conditions was dependent on the fin spacing. An enhancement in heat transfer for wet conditions was seen for a 6.35 mm fin pitch heat exchanger but not for a 3.18 mm fin pitch heat exchanger. The results for the heat exchanger with a 3.18 mm fin pitch showed the heat transfer performance under wet conditions to sometimes be better and sometimes worse than for dry conditions. It was also found that the effect of condensation on friction factor was dependent on fin spacing. Similar friction factors were observed for a 6.35 mm fin pitch heat exchanger under wet
and dry conditions. At 3.18 mm fin pitch, there was a significant increase in friction factor under wet conditions. However, with increasing air-flow rates the quantity of retained condensate and the increase in friction factor decreased.

Wang et al. (1997) studied the performance of plain finned-tube heat exchangers under dehumidifying conditions. The effects of fin spacing, number of tube rows, and inlet conditions were investigated. Nine plain-fin-and tube heat exchangers were tested with fin spacing ranging from 1.82 mm to 3.2 mm and 2, 4, and 6 tube rows. Heat transfer performance and friction factors were observed for the exchangers at a relative humidity of 50% and 90%. The friction factors for wet coils were found to be much larger than those of dry coils. For fully wet conditions, the friction factors were found to be 60 to 120% higher than for dry conditions and insensitive to change in inlet air relative humidity, fin spacing, and the number of tube rows. Sensible factors under dehumidifying conditions were not found to be dependent on the inlet air conditions. Under wet conditions, a degradation in sensible heat transfer was seen at low Reynolds numbers. At high Reynolds numbers, a small enhancement in heat transfer performance was observed under wet conditions but the enhancement disappeared as the number of tube rows increased.

Ha et al. (1999) studied the hydraulic performance of wet fin-and-tube heat exchangers with various wettability coatings. Contact angle measurements obtained were used to characterize each of the different surfaces. For all surfaces, an increase in pressure drop was found for heat exchangers under wet conditions. The increase in pressure drop was greater with increasing contact angles. It was also found that surfaces with smaller contact angles retained less condensate and required less time to reach a steady value of
retained condensate. Furthermore, pressure drop models for dry and wet heat exchangers with dropwise condensation were developed.

Yin and Jacobi (1999) studied the effect of condensate retention on thermal performance for plain-fin and wavy-louvered fin heat exchangers exposed to air frontal velocities from 0.8 m/s to 2.0 m/s. They reported the amount of condensate was independent of face velocity for these geometries and air-flow rates, but dependent on fin geometry and contact angles. A greater amount of condensate was retained in the wavy-louvered coils. Under wet conditions, Colburn $j$ factor decreased, and this degradation was greater at greater fin densities—consistent with a greater amount of water being retained. Additionally, under dry conditions, the wavy-louvered had a higher $j$ factor relative to the plain fin coil, but the enhancement disappeared under wet conditions. The work of Kim and Jacobi (1999) similarly showed a decrease in thermal performance for both plain and enhanced fins (slit-fin) round tube evaporators. The increased pressure drop was concluded to be caused by the blockage effect of retained condensate and the decrease in wet heat transfer due to fouling of the air-side heat transfer surface in plain fins, and additional fouling of the louvers or slits by condensate bridging.

1.2.2 Automotive Evaporator Condensate Drainage and Thermal Performance

Little work has been reported in the open literature to address condensate drainage on automotive-style evaporator coils; however, a few studies addressing the thermal performance of such coils have been reported. Wang et al. (1994) found an increased air-side heat transfer coefficient under wet conditions and surmised condensate on the fins acted as an enhancement by increasing surface roughness. Their study included only two
different heat exchangers, giving useful but limited results. Osada et al. (1999) performed heat transfer and visualization experiments on single fin columns of flat tube evaporators. They investigated the effects of surface wettability, louver geometry, and inclination angle on condensate drainage. Osada and co-workers concluded that fin surface characteristics near the air-flow-exit face of the heat exchanger were an important factor in condensate drainage. This region of the fin is important because airflow forces push condensate toward that part of the fin—it accumulates there until it is drained by gravity. Osada and co-workers found that decreasing the non-louvered length on the fin promotes better drainage; they suggest that increasing the louver cut length decreases the amount of louver blockage caused by condensate bridging within the inter-louver space.

Very recently, McLaughlin and Webb (2000a) studied the impact of fin geometry on drainage characteristics and retention using a table-top test cell for experiments with a single-fin column. They undertook several methods to validate their test methods, and they argued that this approach with a single fin specimen provides results representative of a full-scale heat exchanger. Single-fin tests were conducted with an air flow of 2.5m/s and an entering relative humidity greater than 95%. During these experiments, tube-side cooling was provided by cold water circulated through a tube brazed to one side of the fin. This simplification allowed optical access through glass on the other side of the fin, but it certainly compromised the thermal boundary conditions, and the fin-glass interface modified the geometry and surface tension boundary conditions on retained condensate—as recognized by McLaughlin and Webb. Their results suggest louver pitch is the single most important parameter determining drainage characteristics of louver fin evaporators. They state that a critical louver pitch between 1.1 mm and 1.3 mm exists for a louver
angle of 30°, where condensate retention increases by 26%. In a related study, they found up to a 40% decrease in air-side heat transfer coefficient under wet conditions for the 1.1-mm-louver-pitch evaporator, no change under wet conditions for the 1.33-louver-pitch coil (McLaughlin and Webb, 2000b). They conclude that condensate bridging within the louvers is responsible for the increase in retention and degradation of performance.

1.2.3 Modeling Condensate Retention

Several models of condensate retention have been proposed. Rudy and Webb (1981) reported a method for measuring condensate retention for the condensation of pure fluids on integral low-finned tubes. They found that condensate retention was intensified for a close fin spacing and suggested that the gravity-drained model of Beatty and Katz (1948) was inadequate because it neglected surface tension effects; they later developed their own models (Webb, et al., 1985a,b). Unfortunately, there is little hope of generalizing these geometrically specific results. Jacobi and Goldschmidt (1990) presented a simplified model of condensate retention in the “bridges” formed between neighboring fins. Their model was qualitatively successful in predicting heat transfer effects; unfortunately, it is not possible to trivially generalize this model for other condensate geometries or more complex fins.

Korte and Jacobi (1997) developed a model to predict the quantity of retained condensate for uncoated aluminum plain-fin-and-tube heat exchangers with a fin spacing of 6.35 mm. The quantity of retained condensate was determined by calculating the volume of retained condensate and multiplying this volume by the density of the water. Unlike previous studies, the model incorporated advancing and receding contact angles
that were used to determine surface tension forces. Modeling techniques were relatively successful in predicting the quantity of retained condensate for the 6.35 mm fin pitch heat exchanger, but many higher order effects were not included. The model of Korte and Jacobi incorporated only droplets adhering to the surface, other features such as bridges occurring between fins, fillets and bridges at the fin-tube junction, or other condensate geometries were not included. However, they developed initial force balances to assist in determining the sizes of some simple bridges. The droplet size distributions used in the model were based on the work of Graham (1969). Air-flow forces were included by assuming a constant drag coefficient and approximating the local velocity using laminar flow approximations. Gross surface coverage values were estimated from experimental observations and, as noted by Korte and Jacobi, there may be variations in area covered through the length of the fin due to sweeping effects.

The retention model of Korte and Jacobi was adapted to include condensate bridging at the fin-tube junction by Yin and Jacobi (1999). Furthermore, the modeling technique was applied to a heat exchanger with over twice the fin density. The decreased fin spacing made it difficult to determine the droplet size distributions on the heat exchanger. Therefore, a stock fin sample was studied in the controlled environment of a glove box. Using image analysis software, Yin and Jacobi determined not only the droplet size distribution (still based on the concepts developed by Graham), but also investigated the vertical variations of droplet size density due to sweeping along the fin. The model successfully predicted the amount of condensate retention in a 2.13 mm fin pitch coil, but over predicted the mass in heat exchangers with 1.59 mm and 1.27 fin pitches. This discrepancy was attributed to the assumption in the model of zero
interaction between droplets on adjacent fins. It is likely that in a heat exchanger with tighter fin spacing for two droplets on adjacent fins would coalesce, creating a fin bridge, and cause a greater sweeping effect as the larger fin bridge is shed. A similar approach in modeling was studied by Kim and Jacobi (1999), and a corresponding over-prediction of mass as heat exchanger geometry became more complex was reported.

1.3 Objectives

The objectives of this project were to determine the effect of condensate retention on air-side heat transfer performance of automotive evaporator coils, to investigate the drainage characteristics during operation and after system shutdown, and finally to develop a condensate retention model to predict the amount of condensate retention based on prior modeling efforts for plain-fin-and-tube heat exchangers. Air-side heat transfer performance under and condensing conditions were measured, and condensate retention measurements were taken in real-time and at steady state. A dynamic drainage test rig was developed and the results were used to aid in the understanding of retention and shedding of condensate. A model was developed to predict total condensate retention under normal operating conditions.
Chapter 2 Experimental Apparatus and Methods

A closed-loop wind tunnel was designed and constructed for testing heat exchangers under condensing conditions. Heat exchanger performance and condensate retention measurements were obtained using the apparatus. A condensate visualization chamber was built to study retention on small pieces of fin stock. A water drainage test apparatus was constructed to test drainage behavior of heat exchangers. This chapter describes the experimental apparatus, instruments, experimental procedures, and heat exchangers tested for this research.

2.1 Experimental Apparatus

2.1.1 Wind Tunnel

The thermal performance and condensate retention apparatus consisted of a closed-loop wind tunnel, a test section for testing heat exchangers exposed to horizontal air-flow, and a coolant loop that circulates a single-phase coolant. The wind tunnel is shown schematically in Figure 2.1. It was used to obtain measurements of retained condensate and heat transfer performance for various types of heat exchanger geometries. Experiments were conducted with a horizontal flow of air, and specimens were tested at various air flow rates typical to mobile air-conditioning applications. The closed-loop wind tunnel allowed control of temperature, humidity, and air flow rate. Air temperature was controlled by varying the power supplied to ten electrical resistance heaters using a PID controller; the total capacity of these heaters was 7.5 kW. A feedback Type-K thermocouple was located just downstream of the main tunnel contraction. The heaters were located in two banks, one upstream of the blower and the other downstream of the blower. The second bank was used only during testing at the highest heat duties. Evenly
spaced Type-T thermocouples were used both upstream and downstream of the test section to measure the inlet and outlet air temperature. A six-thermocouple grid was used upstream and a twelve-thermocouple grid was used downstream to measure the average inlet and outlet air temperatures. The upstream temperature readings for all thermocouples varied from the mean less than 0.8°C at the lowest air velocity and less than 0.3°C at the highest velocity, with the upper row always reading higher than the lower row. The relatively large variation at the low flow rates was caused by improper mixing in the thermal-mixing chamber. The upper duct discharged directly into the top of the chamber and caused excessive stratification. Each thermocouple was individually referenced to a thermocouple located in an ice bath, and calibrated to a NIST traceable mercury-in-glass thermometer using a thermostatic bath. Calibration data were fit with fifth order polynomials for each thermocouple. The dewpoints of the air were measured by chilled mirror hygrometers with a measurement uncertainty of ±0.2°C. Air was supplied to the chilled mirrors through sampling tubes located 30-cm upstream and downstream of the test section. A small, medical diaphragm air pump drew air through the sampling tubes. The dewpoint of the incoming air was maintained using a steam injection system. The humidifier was a boiler capable of providing 11.5 kg/hr of steam. The output of the humidifier was controlled by varying the input heater power using a PID controller. The upstream dewpoint monitor provided the control signal for the steam injector. The steam was injected into the tunnel through a perforated pipe 50-cm downstream of the first bank of heaters. An axial fan, belt driven by a DC motor, mixed the airstream and provided volumetric flow rates up to 8.5 m³/min. Upstream of the test section, air was drawn from a thermal mixing chamber and passed through a set of
screens, honeycomb flow straighteners, and a 9:1 contraction to obtain steady laminar flow before passing through the test section. Additional, smaller contractions were required just upstream of the test coil to match the flow to each geometry. These elliptical contractions were cut from Styrofoam, covered with aluminum tape, and attached to the wind tunnel walls with adhesive.

The test section, shown in Figure 2.2, was designed for testing wet heat exchangers. The design allowed for both real-time and steady-state measurements of the mass of retained condensate. The test section was constructed using clear acrylic to allow for optical access. In order to limit conduction losses when observations were not being made, the test section was insulated with 1.27 cm thick polyethylene foam insulation with an insulation factor of $0.08 \text{ W/m}^2\text{K}$. Upstream and downstream pressure taps were located on the upper and lower walls of the rectangular test section for measuring the pressure drop across the heat exchanger. The pressure taps were located approximately 7.5 cm upstream and downstream of the heat exchanger, with two taps at each location spaced 7.5 cm apart centered on each side of the test section. An electric manometer with an uncertainty of $\pm0.124 \text{ Pa}$ was used to measure the air-side pressure drop across the heat exchanger. Face velocities were measured at the test section using a constant temperature thermal anemometer. The face velocity was determined by taking twelve equally spaced measurements traversing the height of the heat exchanger in three places at each of the four velocity measurement locations shown in Figure 2.3. The twelve measurements were recorded and an average face velocity was determined. The velocity measurements were within 11% of the average at the lowest velocity and 8% at the highest velocity. Turbulence intensity measurements were taken using a hot wire.
anemometer at the velocity measurement locations, with a plain-fin-and-tube installed in
the test section. Measurements were made at three locations and except for the small
wake region behind the thermocouples, the turbulence intensity was less than 2.5%.

A single-phase ethylene glycol (DOWTHERM 4000) and water mixture was
circulated on the tube side of the heat exchanger. Over the course of this project, two
different concentrations of ethylene glycol were used, 32.6% and 40.0%. The
concentrations were mixed and maintained by measuring the specific gravity of the
mixture using a NIST traceable hydrometer. The required specific gravity was obtained
by interpolating on manufacturer provided tables. Coolant-side temperatures were
measured using type-T immersion thermocouples located approximately two meters
upstream and downstream of the heat exchanger. Each thermocouple was individually
referenced to a thermocouple located in an ice bath, and calibrated in the same manner as
the wind tunnel grid thermocouples. An R-502 liquid-to-liquid, variable-speed chiller was
used to control the coolant temperature. Temperature of the solution was controlled using
an immersion temperature probe on the supply line as the control signal for a proportional
controller driving the compressor. The mixture was circulated through a copper tubing
loop by two pumps. An integral, centrifugal, recirculation pump provided a 200-kPa head
to a positive displacement rotary gear pump. The gear pump was belt driven to minimize
vibrations by a two horsepower motor. The coolant flow was controlled by using an
inverter to vary the drive motor speed. The test heat exchanger was connected to the
copper tubing with flexible, reinforced, PVC tubing that terminated with quick
disconnect couplings to facilitate removal of the coil. All tubing was insulated with 9.5
mm polyethylene foam insulation with an insulation factor of 0.05 W/m²K. Coolant flow
rate was measured on the return line using a positive displacement disc flow meter with a measurement uncertainty of ±1.0%. A transmitter attached to the flow meter provided a 1-5V pulse with a frequency proportional to the volumetric flow rate. A Philips programmable timer/counter was used to count the number of pulses over a timed cycle with an uncertainty of ±2 pulses. Mixing cups were not used in the coolant lines prior to the thermocouples to help minimize line pressure losses, but the flow is well mixed. The supply line was insulated and typical flow Reynolds numbers were from 4000 to 7000 which should maintain a flat temperature profile. The return line was also well insulated and the highly interrupted internal geometry of the tubes of the tested coils is such that it mixes the coolant.

The data acquisition system consisted of a control unit, programmable timer/counter, and personal computer. The control unit contained an 20 bit analog-to-digital converter and samples 23 channels. The channel outputs are read twice a second, averaged over 11 measurements, and recorded every 45 seconds. The personal computer receives the recorded outputs and stores them in a data text file for subsequent analysis. The temperatures, dewpoints, and coolant flow meter readings are recorded by the data acquisition system. The air velocity, barometric pressure, and core pressure drop are manually recorded during a test.

2.1.2 Condensate Visualization

A condensate retention visualization apparatus was built to quantify the nature of retained condensate on small pieces of fin stock. The apparatus is an acrylic box with volume of approximately 0.07 m³ containing a humidity source and test section. The clear acrylic provides optical access for taking photographs from a variety of orientations.
A plastic glove is mounted around an access hole in one side to allow non-intrusive manipulation of the test section. The visualization apparatus is shown in Figure 2.4. The test section consists of a sample fin (or fin stock) mounted to a Peltier thermoelectric device. The Peltier device is water-cooled and capable of removing 50 Watts using a DC power supply. The sample is bonded to a piece of aluminum stock using thermal epoxy and then clamped to the Peltier device. A submersible pump with a 0.2 L/s capacity circulates water from an icebath through a heat exchanger connected to the hot side of the Peltier device. A beaker of hot water provides water vapor and a small fan mixes the air to provide a uniform distribution of water vapor.

2.1.3 Dynamic Drainage

A schematic diagram of the drainage apparatus is shown in Figure 2.5. The apparatus consists of a moving water reservoir and mechanism to suspend and weigh the heat exchanger. The moving reservoir has a volume of 68 liters and is positioned using a hydraulic jack. This simple arrangement allows a smooth, consistent lowering of the water reservoir. The heat exchanger is suspended from a balance using an acrylic frame and attaching mechanism. The frame is large enough to accommodate both the balance and the width of any coil (see Fig 2.5). The attaching mechanism depends on the particular heat exchanger being tested. In this study, nine coils were tested. Four coils were attached to the weighing mechanism by permanently fixing aluminum “wings” to the outside as detailed in Figure 2.6a. The wings are 58mm x 90mm rectangles attached to a two-piece narrow aluminum strap. The lower strap is connected by a pin and clamp to the wing, allowing angular adjustment, and the upper strap is connected to the frame by a bolt. The two pieces of the strap are connected by a clamp to allow height
adjustments. In this way a test coil that can be tilted in two directions during a test, allowing experiments on orientation effects on water drainage. The other coils did not have an external attachment site for the same wings, so heavy gauge wire was looped under the upper tube row or manifold as shown in Figure 2.6b. The wire was then bent into a rectangle, interlocked with the frame and heat exchanger providing a stable support. Adjustments were done by lengthening or shortening the wire loop until proper orientation was obtained.

A precision balance was used for the mass measurements. It has a readability of 0.1 grams and a reported uncertainty of <0.1 grams. An adjustable base that allows the balance to be leveled or moved vertically if required supports the balance. The balance has an RS-232 port and a personal computer could be used to record mass readings. Based on initial repeatability of experimental data, it was decided for this study to manually read and record all data.

The dynamic drainage apparatus was modified to obtain real-time condensate retention measurements as shown in Figure 2.7. A frame was built to support the balance and suspension mechanism over the test section. The aforementioned adjustable attaching technique allowed the minute adjustments required to align the test coil with the incoming airstream. The metal hanging straps acted as springs to offset the moment created by the coolant lines and airflow forces. The coolant lines were securely clamped to the wind tunnel frame as far as possible from the heat exchanger. This long, horizontal run after the sturdy support helped eliminate measurement errors created by fluid momentum and vibrations. The apparatus was tested by operating under dry conditions to assess the measurement errors and the observed mass varied less than three grams at a
face velocity of approximately 3.0 m/s. It is impossible in this test set-up to operate in a closed system and record real-time measurements since the evaporator has to be free-floating. Side plates of thin acrylic and an aluminum bottom tray were constructed to minimize air losses. The gap between the test section and the coil was also made as small as possible.

2.1.4 Contact Angle Measurements

Most studies use a goniometer to measure contact angles, whether advancing, receding, or solely static angles are measured. A new method using digital photography and image analysis software was developed for this study. Initial results were compared to values recorded with a goniometer and found to be in close agreement. The test set-up is outlined in Figure 2.8. A syringe of distilled water and a test specimen platform are mounted on a ring stand. A CCD camera with a 1-6.5 zoom lens is oriented horizontally, focused on the test specimen. Scion Image Acquisition and Analysis software is used to acquire a video clip at 20 frames per second while water is added and removed from the sample with the syringe. Individual frames are extracted from the video and the contact angles are measured with built-in image analysis tools. The advancing contact angle is the angle between the substrate and water the moment before the droplet contact line moves as water is added. Alternatively, the receding contact angle is the angle just before the droplet contact line moves as water is removed.

2.1.5 Heat Exchanger Specifications

The specifications for the tested heat exchangers are displayed in Table 2.1. The evaporators are numbered 1 through 7 and are referred to by number throughout this document. Coils 1 through 5 were tested for thermal performance and steady-state
retention. Real-time condensate retention tests were performed on Coils 4 and 5. Dynamic drainage test data were collected on Coils 2 through 7, with additional drainage tests completed on Coils 3 through 5 with the coil face tilted.

2.2 Experimental Conditions and Procedures

2.2.1 Thermal Performance

Experiments were conducted under both dry and wet conditions. Dry experiments were conducted by setting the inlet coolant temperature so that the temperature at the tube wall is above the dewpoint of the air throughout the heat exchanger. Dry conditions were verified by comparing the inlet and outlet dewpoints. The heat duty of the some of the evaporators in this study is high enough to require the inlet air temperature to be elevated above 45° C to avoid "pinching-off" part of the coil. During the dry experiments the inlet temperatures were used to determine when the system had reached steady-state and data are recorded and averaged for at least three minutes in the steady-state condition while pressure drop and airflow are measured and recorded manually.

Wet experiment procedures are very similar to the dry experiments. The operating conditions are set so that the entire heat exchanger is wet by ensuring the outlet coolant temperature is below the outlet dewpoint. Steady-state in wet experiments is determined by the air inlet temperature and inlet air dewpoint, and to ensure the condensate retention has reached steady-state the test is allowed to run for at least one hour. Once the system has reached steady-state data are recorded and averaged over at least a three-minute interval while pressure drop and airflow are measured and recorded manually.

2.2.2 Steady-state Condensate Retention

Steady-state condensate retention measurements were recorded either after thermal performance data is recorded or during test runs solely for retention
measurements. The procedure is identical in either case. The wind tunnel is operated at steady-state for a minimum of one hour to ensure condensate retention is also at steady-state. The wind tunnel is shut down and a small tray is inserted under the test heat exchanger to catch any condensate that is knocked off during the removal process. The heat exchanger is removed from the wind tunnel and disconnected from the coolant lines. The inlet and outlet tubes are sealed off by the quick release couplings with automatic valves that are used to connect the coolant lines to the heat exchanger. The heat exchanger and tray are weighed and the total mass is recorded. The heat exchanger and tray are then allowed to thoroughly dry and are weighed again. The difference between these two weights is the mass of the retained condensate. To hasten the drying process (normally at least 24 hours) compressed air was used to blow off a large amount of the water from the coil, and a heat gun was used to further dry the heat exchanger. In instances when these rapid drying techniques were employed, the evaporator was often weighed again after several hours to ensure the recorded dry weight was accurate.

2.2.3 Real-time Condensate Retention

Real-time condensate retention measurements required using two heat exchangers—the test coil, and a ‘dummy’ coil. The dummy coil was necessary to minimize the transient response of the inlet conditions during a test. The wind tunnel system requires about one to two hours to reach steady state. However, a test coil only requires approximately fifteen minutes to reach a maximum retention mass.

Before system start up, the test coil is placed in the test section on the weighing system and adjusted for proper orientation. By removing the lower adjustment clamp the coil could be pivoted and removed from the tunnel without changing any other
adjustments. The dummy coil is then inserted and the wind tunnel is turned on and allowed to reach steady state. Once the tunnel reaches steady state it is shut down and the coils are rapidly changed. Timing and experience are the essential elements to how the tunnel is turned back on and the resulting quality of data recorded. The two most important considerations are when to turn the steam on and zero the balance. The steam injection is the slowest responding element of the wind tunnel system (heaters, chiller, blower, etc.). The steam could not be left on during changeover because it likely would saturate both the tunnel and the dewpoint monitors. If the steam were turned on too late, the coil would dehumidify the air quicker than the humidifier could respond, resulting in a lag in the retention measurements and a large overshoot of the dewpoint set point and subsequent oscillations. Through repetition, the timing was refined so the transient response of inlet dewpoint was less than three minutes.

The balance needed to be zeroed prior to deposition of condensate but after the coolant pump and blower were operating. Once the entire system is running and the data acquisition program is started, mass measurements are recorded every ten seconds for 900 seconds then every 60 seconds for an additional 360 seconds. To assist in synchronizing the mass measurements with the inlet air conditions, the inlet dewpoint is recorded manually approximately once a minute. This also served as a check of the steam injection system response. The time at which the first condensate drips from the bottom of the coil is also recorded. After the amount of condensate retained on the coil reaches a steady value, the system is shut down and the steady-state value is recorded as previously described. Checking the steady-state value helped verify the test procedures and results.
2.2.4 Dynamic Drainage

For each experimental drainage test run, a dry test coil was suspended over the water reservoir, the orientation was checked using a standard bubble level, and adjustments were made to ensure proper alignment on two axes. Test runs were also conducted with the test specimen tilted 10 degrees. Alignment in these tests was achieved using a plumb bob and adjusting the heat exchanger until the plumb string intersected marks scribed on the external surface, producing the proper degree of tilt.

The balance was turned on and zeroed, and a final alignment check was performed. Zeroing the balance at this stage allowed direct reading of the mass of water retained on the coil. The water reservoir was raised until the entire test coil was submerged. Due to the large amount of horizontal fin surface in the coils, there was concern that a significant amount of trapped air might remain in the submerged coil. This possibility was investigated by conducting tests during which the water was vigorously agitated and the coil turned through 180° while submerged. It was found that a dry coil when submerged contains a negligible amount of trapped air and simply agitating the water and causing flow through the coil could remove this small amount of air. However, if a coil is partially removed from the water after a "false start" and needs to be resubmerged (say if an alignment problem is noticed as the coil starts to drain), a large amount of air will be trapped upon re-submerging the wet coil. This increased amount of air is the result of isolated areas within the coil that are surrounded by water bridges, and care must be taken to avoid re-submerging a wet coil.

Additional verification of this potential test problem was attained by performing desktop tests on single fins (similar to those of McLaughlin and Webb). The fin sample
was placed between two pieces of clear acrylic and clamped in place. An initially dry fin was placed into a clear glass container and observed. As expected very few air bubbles remained on the fin. Then the fin was removed and replaced after it had begun to drain. Large air bubbles, bridging five to ten louvers, were easily observed. As a result of these tests it is recommended that dip testing should not include the re-submerging of any coil as part of the test procedure.

With an initially dry test specimen fully submerged in the reservoir, testing was initiated by lowering the reservoir and starting a stopwatch at the instant the bottom of the coil cleared the water. Weight readings were recorded at five-second intervals for ninety seconds, then at 30-second intervals for an additional 240 seconds. Other data points of longer duration were also recorded during various tests to help fully characterize the nature of water drainage. Repeated observations of the test coil were obtained, with special attention to the early drainage behavior (in the first thirty seconds), to ensure orientation remains correct and the heat exchanger remains stable.
Figure 2.1 Horizontal flow wind tunnel. (A) 36-cm diameter round sheet metal duct. (B) Thermal mixing chamber. (C) Screens and honeycomb flow straighteners. (D) 9:1 contraction. (E) Test heat exchanger. (F) Inlet/outlet measurement sections. (G) Strip resistance heaters. (H) Steam injection tube. (I) Axial blower.

Figure 2.2 Test Section for Wet and Dry Runs. (A) Pressure taps (top and bottom). (B) Chilled mirror hygrometer sensors. (C) Insulated clear acrylic. (D) Drainage tray. (E) Thermocouple grid (inlet and outlet).
Figure 2.3 Air velocity measurement locations.

Figure 2.4 Closed environment glove box apparatus for examining condensing fin samples. (A) Beaker with water. (B) Fin stock. (C) Peltier device and liquid heat exchanger. (D) Glove. (E) Fan.
Figure 2.5 Dynamic drainage apparatus.

Figure 2.6 Attaching mechanism for drainage test coils.
Figure 2.7 Real time retention apparatus. (A) Wind tunnel. (B) Suspension mechanism components. (C) Balance. (D) Inlet/outlet coolant lines. (E) Test heat exchanger. (F) Drain.

Figure 2.8 Contact angle measurement apparatus.
### Table 2.1 Tested coil descriptions.

<table>
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<tr>
<th>Coil</th>
<th>External Dim (HxWxD)</th>
<th>Fin type</th>
<th>Louver pitch</th>
<th>Louver angle</th>
<th>Louver width</th>
<th>Fin width</th>
<th>Fin thickness</th>
<th>Fin pitch</th>
<th>OF Strip Height</th>
<th>Contact Angle</th>
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<td>1</td>
<td>213 219 92</td>
<td>Louver</td>
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<td>N/A</td>
<td>6.35</td>
<td>9.14</td>
<td>0.13</td>
<td>2.12</td>
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<td>64 35</td>
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<td>9.53</td>
<td>0.13</td>
<td>2.12</td>
<td></td>
<td>60 30</td>
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<td>Louver</td>
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<td>8.00</td>
<td>0.10</td>
<td>1.81</td>
<td></td>
<td>68 44</td>
</tr>
<tr>
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<td>6.35</td>
<td>8.00</td>
<td>0.10</td>
<td>1.81</td>
<td></td>
<td>68 44</td>
</tr>
<tr>
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<td>6.35</td>
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All dimensions in mm except contact angles in degrees.
Chapter 3 Results and Discussion

Condensate retention affects air-side heat transfer and pressure drop characteristics of heat exchangers; however, the direction of the effect is heavily dependent on geometry and operating conditions. The main objective of this work is to quantify these effects on the performance of automotive evaporators. The experimental approach was to establish the impact of retained condensate by performing thermal performance experiments under dry and wet conditions, and then record both steady-state and real-time condensate retention measurements to begin understanding how condensate is retained on the surface. Additionally, extensive drainage tests were completed that assist in developing how the condensate affects performance. Furthermore, these drainage tests show the geometrical dependence of the post operation drainage that affects long-term air quality, another important parameter in measuring overall system performance.

3.1 Thermal Performance

Thermal performance data were collected for five different heat exchangers under dry and condensing conditions. The results are presented in both dimensional and non-dimensional forms. Data reduction and interpretation follows the methods detailed by the ARI Standard for condensing heat exchangers, with special attention to related recent work (e.g., Jacobi and Goldschmidt, 1990; Korte, 1997; Hong and Webb, 1996). A combination of FORTRAN and Engineering Equation Solver (EES) routines were used to reduce the data. Dimensional plots of the sensible air-side heat transfer coefficient versus frontal velocity are given in Figures 3.1 through 3.5 and air-side core pressure drop in millimeters of water versus frontal velocity are shown in Figures 3.6 through 3.10. The data reduction procedure used to calculate the heat transfer coefficients is
discussed in Appendix A, Data Reduction. The pressure drop is recorded manually during the experimental test using an electronic manometer. The experimental results are non-dimensionalized by calculating the sensible Colburn \( j \) factors and the Fanning friction factors for the data sets. These parameters are determined from temperature, mass flow, pressure drop, and geometrical data using the following equations:

\[
\begin{align*}
  j &= \frac{St}{Pr^{3/5}} = \frac{Nu}{Re_{Dk} Pr^{1/3}} = \frac{h Pr^{3/5}}{G C_p} \\
  f &= \frac{2 \Delta P_{HX} \rho_a}{G^2} \left( \frac{A_{\text{min}}}{A_{\text{tot}}} \right) - (1 + \sigma^2) \left( \frac{\rho_{a,1}}{\rho_{a,2}} - 1 \right) \left( \frac{A_{\text{min}}}{A_{\text{tot}}} \right) \left( \frac{\rho_a}{\rho_{a,1}} \right)
\end{align*}
\]

Plots of \( j \) and \( f \) factors versus air-side Reynolds number are given in Figures 3.11 through 3.15. The sensible heat transfer coefficient for all coils decreased under condensing conditions, an effect that has been contributed in recent literature (Osada et al. (1999), and McLaughlin and Webb(2000)) to condensate bridges in the inter-louver space. While the relative performance of a coil under dry and wet conditions is of interest in understanding the impact of condensate retention on performance, the absolute performance is of paramount importance in application. The experimental results will be presented and discussed giving exposure to both the relative performance of wet versus dry conditions for a single coil and comparative performances of different coils.

The heat transfer results for Coils 1 and 2 shown in Figures 3.1 and 3.2, respectively, are lower under both wet and dry conditions than the performance of Coils 3
through 5 shown in Figures 3.3-3.5, but they clearly demonstrate the effect of geometry on wet performance. The sensible heat transfer coefficient for Coil 2 decreased 30% when the coil was fully wet, compared to a 12% decrease for Coil 1. The main differences between Coil 1 and Coil 2 are louver design and louver pitch. Coil 1 has a complex 'scoop' louver design while Coil 2 is more of a standard louver design. Coil 2 has a much smaller louver pitch, 1.59 mm, than the 5.08 mm louver pitch of Coil 1. The impact of these geometrical differences under dehumidifying conditions is to give a larger relative amount of surface area where retained condensate acts not as performance enhancements, but as flow interruptions. The condensate sticking on the Coil 2 surface interacts with the flow such that the flow is redirected into duct or channel flow, probably through the formation of louver bridges. When condensate ceases to exist as droplets and totally blocks a flow passage, the enhancement due to boundary layer tripping and vortex shedding disappears. The highly compact design of these coils can also cause the pressure drop across the coil to increase even in duct flow as the condensate on the fin surface decreases the minimum flow area, especially in the instances where fin bridges are formed. This effect is consistent with the findings of McLaughlin and Webb (1999) where it was observed that a coil with a smaller louver pitch (a critical louver pitch between 1.1 and 1.3 mm for their study) had a larger propensity for inter-louver bridging by condensate. The pressure drop performance shown in Figures 3.6 and 3.7 for Coils 1 and 2 are consistent with the condensate interrupting the flow in Coil 2 more then Coil 1. Coil 1 had a much larger (>30%) pressure drop across the coil at all air velocities, likely caused by the greater core depth, thicker tubes, and higher effective louver angle of the scoop louver. However, the pressure drop increase under wet conditions of Coil 1 is
considerably smaller than the increase in Coil 2. Coil 2 had a >25% increase over the entire velocity test range while Coil 1 had <10% increase at lower velocities and <20% at higher velocities. Both the heat transfer and pressure drop performance trends are also presented for Coils 1 and 2 in non-dimensional $j$ and $f$ factor plots in Figures 3.11 and 3.12, where the same trends are seen over the entire Reynolds number range.

As previously stated, the heat transfer performance for Coils 3 through 5 is higher than the corresponding performance of Coils 1 and 2. This increase in heat transfer coefficient does result in the expected higher pressure drop across the coils from the higher louver angles, smaller louver pitch, and decreased fin spacing. The flow over each louver (or offset strip) can be viewed as flow over a flat plate. The Reynolds analogy for flat-plate flows from White (1991) states,

\[
\frac{C_f}{C_h} = 2 \Pr^{\frac{1}{3}}
\]  

(3.3)

or alternately, friction is proportional to heat transfer. It should be stressed Equation 3.3 is reliable only for certain flow conditions such as low pressure gradients. Thus, by increasing the number of louvers a streamline flows over (by decreasing louver pitch or by increasing louver angle) in an array to achieve increased heat transfer will also increase pressure drop.

Both Coils 4 and 5 had higher sensible heat transfer coefficients than Coil 3 under both wet and dry conditions; in fact, the wet performance of Coils 4 and 5 is the same as the dry performance of Coil 3 within the experimental uncertainty. The main geometrical differences between the coils are louver pitch, louver angle, and number of louver banks. Coil 3 has a 1.2 mm louver pitch compared to the 1.0 mm louver pitch in Coils 4 and 5. Coil 3 also had a louver angle of 30° versus the 36° and 42° louver angles for Coils 4 and
5 respectively. Coil 3 also has four louver banks (three turnaround louvers) while Coils 4 and 5 each have two relatively long louver banks with only a single turnaround section.

Under dry operating conditions (and to a lesser degree wet) the net result of the geometry differences is a higher heat transfer coefficient and larger pressure drop. The heat transfer results are shown in Figures 3.3-3.5 and the pressure drop in Figures 3.8-3.10. The main contributor to both effects is the smaller louver pitch, which causes a greater number of louvers for the same coil depth, and thus a greater number of boundary layer restarts, resulting in a higher average Nusselt number and friction factor. This geometrical effect partially explains the greater pressure drop across Coils 4 and 5 relative to Coil 3. The greater louver angles also cause an increase in pressure drop, as evidenced in the distinctly greater pressure drop in Coil 5 with a 42° louver angle versus Coil 4 with a 36° louver angle. Increasing the louver angle increases the effective flow depth of the coil. Under wet operating conditions Coils 3 and 5 performed similarly to Coil 2 with a 25-30% increase from dry to wet for Coil 3 and a 20% increase in pressure drop for Coil 5. Coil 4 responded to wet conditions with only a 10% increase in core pressure drop, similar to Coil 1, though for a different reason. Coil 1 had a lower increase in pressure drop because the geometry is much more open with wider fin spacing and larger louver pitch. The reason for the wet pressure drop performance in Coil 4 will be seen in the next section—at steady-state it retained considerably less condensate than the other coils.
3.2 Condensate Retention

Condensate retention data were recorded under transient and steady-state conditions. The transient, real-time retention experiments were conducted for several reasons. Yin and Jacobi (1999) and Kim and Jacobi (1999) reported an overshoot in condensate retention for some geometries, where the quantity of condensate retained reached a maximum, then decreased to a steady-state value. This behavior existed at all tested face velocities, and at some velocities there was a 15% difference between the maximum and steady-state values. Since there were no other discernible oscillations present after the initial overshoot, the dynamics that caused the behavior were not addressed. Korte and Jacobi (1997) discussed two possible scenarios for condensate retention. (1) Condensate may accumulate until deposition is balanced with shedding, resulting in a steady value under a given operating condition. (2) Condensate retention could be cyclic and oscillate between a maximum and minimum value due to contact angle hysterisis and shedding characteristics.

Condensate accumulates on a surface by condensation and coalescence, and is retained by surface tension forces. The water droplets are held until they are shed either when flow and gravitational forces overcome the surface tension forces, or the droplet is swept by another droplet being shed. This sweeping is likely an important shedding mechanism, especially under the fully wetted operating conditions of interest. Potentially, for certain geometries, sweeping could remove a considerable amount of water, essentially resetting the heat exchanger surface to a partially loaded condition. The surface would then start the loading process again, resulting in condensate retention oscillations in time. A caveat to this idea is the latent load the coil is operating under that
could offset the sweeping effect by causing the heat transfer surfaces to effectively be constantly shedding water. If the water mass flux onto the surface is high enough, then as soon as a droplet is swept another droplet condenses on the now clean surface and little oscillation would be observed. Yin and Jacobi (1999) noticed a difference in droplet size distribution vertically on a flat plate, but it was observed in a very low latent load environment in a glove box that caused only a relatively small frequency of sweeping.

For the heat exchangers in this study, a droplet drains from the coil through one (or a combination) of three main drainage routes. (1) Flow forces push the water toward the downstream edge of the fin array and the droplet drains down the edge of the fins. (2) The droplet drains through the fins until it reaches the bottom. (3) The droplet moves down the channels along the tube wall to the bottom of the coil. Both geometry and flow conditions will affect the relative quantity of condensate draining through each of the different modes. Osada et al. (1999) studied the effects of surface condition, louver cut length, and a dividing section on drainage. Osada and coworkers found a 2.0 mm center dividing section promoted drainage along the tube wall in the center of the array versus the water moving along the fin length to the trailing edge before draining. This finding is an important point regarding the performance of Coils 4 and 5, which have a relatively large open turnaround section in the center of the array that may behave similar to an actual division.

3.2.1 Real-time Retention

Condensate retention has been shown to have a large impact on heat transfer, and therefore the actual amount of water on the surface at a given time affects interpretation of performance data. The existence of oscillations in condensate retention would affect
the validity of the thermal performance and steady-state retention data, dependent on the frequency of the oscillations. Additionally, the overall time required for the condensate on the heat exchanger to reach a steady-state value was initially unknown, and the actual test duration required could affect experimental procedures. In summary, real-time retention experiments were conducted to investigate condensate loading characteristics, to validate the thermal performance and steady-state retention data, and to determine any required changes to experimental procedures.

A plot showing condensate retention and humidity ratio versus time for Coil 5 is shown in Figure 3.16. The retention data were recorded over a 45 minute interval, longer than any other test run. The humidity ratio is shown to verify the operating conditions and validate the experimental apparatus and procedure. Inlet air temperatures were also recorded and showed less than 1.0°C variation over the course of a test due to the relatively large thermal mass of the tunnel. The approximately three minute transient period in the humidity ratio is caused by the humidifier controller adjusting the steam injection rate after restarting the wind tunnel as discussed in Chapter 2. The quantity of condensate retained increased asymptotically to a steady value. There were virtually no oscillations in the mass after twenty minutes, and what small fluctuations that were noted are likely caused by system vibrations and not an unsteadiness in the quantity of retained condensate.

The repeatability of the experimental technique is shown in the plot of two test runs with Coil 4 in Figure 3.17. The two data sets were collected at face velocities that were within the experimental uncertainty of each other. The large uncertainty in the velocity measurement is caused by the necessity to leave the coil free floating and not
seal the wind tunnel while recording the data. As discussed in the experimental procedure, the real time retention measurement apparatus is set-up as consistently as possible to mitigate the velocity measurement error.

The results in Figure 3.17 show a similar trend to that of Coil 5 (see Figure 3.16), but there is an abrupt flattening of the curves for several minutes, then a gradual climbing to a steady-state value. This flat region is in the same area where previous studies (Yin and Jacobi (1999), Kim and Jacobi (1999)) observed an overshoot in condensate retention, and the effect is more pronounced in the $V_f = 1.4 \text{ m/s}$ test. Actual overshoot was observed for the same coil (Coil 3) at a face velocity of 1.5 m/s as displayed in Figure 3.18. Also shown in Figure 3.18 is a test run with $V_f = 0.8 \text{ m/s}$ where neither an overshoot nor flattening trend is observed, only a gradual climb to a steady-state value. The steady-state values measured by removing the heat exchanger from the wind tunnel and weighing as described for steady-state tests are also plotted for the two runs in Figure 3.18. Unlike the plain-fin-round-tube heat exchangers studied by Yin and Jacobi (1999) where the overshoot was dependent only on geometry, operating conditions appear to affect the transient retention behavior of automotive style evaporators.

The real-time condensate retention behaviors of the tested evaporators were well behaved in the sense that there were no substantial oscillations in time that would effect data acquisition or interpretation. Under some operating conditions, there was a single oscillation where the mass of condensate overshoot the final steady-state value by 10% and then smoothly decreased to the steady value. This overshoot was only observed at face velocities of approximately 1.4 – 1.5 m/s. The likely cause of the overshoot existing only at certain velocities is the drainage dynamics of the coil. As stated, a surface will load up
with condensate until the water is shed through gravitational or flow forces. The rate of deposition depends on the prevailing operating conditions of both the incoming moist air stream and the fin surface. The difference in this deposition rate is clearly seen in Figure 3.18. The slopes of the graphs between 100 and 300 seconds is the rate condensate is being deposited on the surface (there was no observed drainage during this phase). This may also be calculated from the following,

\[
\dot{m}_{\text{Cond}} = \dot{m}_{\text{Air}} (\omega_{\text{in}} - \omega_{\text{out}}),
\]

and at steady-state this quantity is also equal to the drainage rate of condensate, since the quantity of water removed from the airstream also must leave the heat exchanger at steady-state.

An overshoot in condensate retention would be expected in only the following special case. For a given coil at different operating conditions, the relative amount of drainage occurring through the aforementioned drainage routes will vary. For instance, at higher velocities, one would expect more drainage along the downstream edge because of the increased shear forces. Additionally, the order in which the drainage modes occur will also depend on operating conditions, at lower velocities, drainage along the tube walls will likely will occur before any condensate is pushed to the back of the coil by the air flow. However, each of these drainage modes, especially the pure gravity driven ones, necessarily has a maximum rate constrained by the geometrical parameters that create the routes through which condensate can drain. Thus, the circumstances where an overshoot in condensate retention could occur can now be identified. If the maximum drainage rate for drainage along the tube wall or through the fin is reached before airflow forces have pushed enough condensate to the back of the fin for drainage, then an overshoot could
occur. This overshoot would be temporary because as the condensate effectively ‘backs-up’ on the fin surface, higher shear forces will occur on the larger droplets, and they will be forced to the downstream edge and a balance will be eventually reached. At higher flow rates, shear forces would dominate from the beginning, and at lower velocities the gravity drainage modes would be dominant. It should be stressed here that this explanation is only a hypothesis, and would be difficult to directly verify with the existing test set-up. Also, since the resulting effect of this overshoot is minimal with no long-term effect on heat transfer, it isn’t likely that the transient nature of condensate retention is an important design consideration. However, these drainage modes will effect steady-state retention, a very important design consideration.

3.2.2 Steady-state Retention

Steady-state retention measurements on all five coils are shown in Figures 3.19 and 3.20. Total condensate retained is plotted in Figure 3.19 versus frontal velocity and quantity of condensate retained per unit of total heat transfer area versus frontal velocity is plotted in Figure 3.20. Condensate retention for all five coils displays a heavy dependence on frontal velocity, though the magnitude of the impact varies with geometry. Coil 2 had a 50% increase in retention when the frontal velocity changed from 2.0 m/s to 1.0 m/s. By comparison, Coil 4 retention only increased 25% with the same velocity change. Similar data trends were reported by Korte and Jacobi (1997) for uncoated, plain-fin-and-tube heat exchangers in a downward flow wind tunnel with velocities in the range of 1.5 m/s to 8.0 m/s based on minimum free flow area. Yin and Jacobi (1999) observed condensate retention to be independent of air velocity over the
range of face velocities from 0.8 m/s to 2.0 m/s for plain-fin-and-tube coils in a horizontal flow wind tunnel.

The large influence of air-flow rate on the quantity of retained condensate for the studied automotive evaporators studied is likely caused mainly by geometry and not surface condition, though surface condition is a contributing factor in general for condensate retention. Kim and Jacobi (1999) observed almost a 50% decrease in condensate retention for a heat exchanger treated with a hydrophilic coating over an uncoated coil. Korte and Jacobi (1997) reported condensate retention on a hydrophilic surface coil had little dependence on air velocity. Three of the five coils (coils 3-5) tested in this study had virtually identical surfaces. They were manufactured at roughly the same time by the same laboratory and had been exposed to testing conditions the same number of times. The other two coils were older and tested more frequently, and as a result, though they initially had moderately hydrophilic surfaces, by the time steady-state retention tests were conducted the advancing and receding contact angle were similar to the other three coils. Therefore, the differences in the condensate retention between the coils are caused by geometric differences.

3.3 Dynamic Drainage

Dynamic drainage tests were completed for coils 2 through 5 and two additional coils that were not tested in the wind tunnel for this study. Most of the research on condensate retention has focused on its effects on thermal performance. Korte and Jacobi (1997, 2000), Yin and Jacobi (1999), and Kim and Jacobi (1999) state that the effects on thermal performance depend directly on the nature and quantity of retained condensate, and that such retention can be modeled if the flow forces, gravitational force, and surface
tension forces (and surface condition) are considered. They have developed retention models to predict the quantity of condensate retained on a coil during operation. Unfortunately, none of this work addresses the important issue of post-operation drainage. Condensate drainage must be understood if designers are to develop fast-draining coils to mitigate biological activity on the air-handling surfaces and improve air quality. The automotive industry typically relies on the "dip test" to provide drainage information. In this test, the coil is submerged in a reservoir of water, withdrawn and weighed. There is little consistency in how this test is conducted, namely in the time delay between dipping and weighing and in heat exchanger handling during the dip test. The dynamic drainage test developed during this study alleviates some of the potential misinterpretation due to differences in dip test techniques. This new variation to the standard dip test is used to develop a deeper understanding of post-operation condensate drainage behavior, which is linked to the actual condensate retention during operation.

Multiple test runs were conducted for most coils, initially to ascertain the repeatability of the test procedures, and finally for periodic verification of results. Figure 3.21 shows the results from three different tests on one heat exchanger demonstrating the repeatability of the experimental results. As expected, the maximum error occurs during the rapid draining phase in the first 60 seconds as illustrated in the partial chart of the results in Figure 3.22. Overall, the differences between tests are within the experimental uncertainty, and this repeatability served as an additional check of test conditions.

Drainage test results for the six heat exchangers are depicted in Figures 3.23 and 3.24. Unless otherwise noted, all mass values are on per unit heat transfer area basis. The basic behavior shown in the figures resembles an exponential-like decrease in retained
water, with one group of coils (in Fig. 3.23) showing a short time constant (low-$\tau$), and
one group exhibiting a long time constant (high-$\tau$). The low-$\tau$ coils are the three coils that
reach within 20% of their final value (based on a 300 second total test time) in the first 20
seconds, and high-$\tau$ coils are the remaining three coils that show distinctly different
drainage patterns.

Test runs over an extended time interval were completed on a selection of coils
and are presented in Figure 3.25. Data were recorded at one-hour intervals after initial
data points were collected to verify proper orientation by comparison to other test runs.
Lab conditions were not controlled in regards to temperature and humidity so there was
initially concern that evaporation could lead to a misinterpretation of the results. A tray
was placed between the test heat exchanger and the water reservoir after fifteen minutes
and by observing the amount of water that drained into the tray it was concluded that
evaporation accounted for a negligible amount of the mass change during a four-hour
test. Coils 6 and 7 continued to drain a significant amount of water, especially during the
first hour of the test, although they appeared to have very slow drainage rates in the
shorter tests. In comparison, Coils 3 through 5 drained rapidly in the first five minutes,
then lost only 13% more, whereas Coil 6 lost an additional 26% over the extended time
period.

Data were also recorded for Coils 3 through 5 with the coil face tilted 10° from
vertical. A representative graph in Figure 3.26 displays the results for a single coil and
Table 3.1 outlines the effects on each coil. The tilt tests were conducted using exactly the
same procedure as the vertical tests. Multiple experiments were again performed to
ensure the repeatability of the test results. The results typically varied less then 4% over
the entire test. Coils 3 and 4 each drained approximately an additional 25% when tilted, but Coil 5 was distinctly different, draining only 15% more. This difference is likely due to the way water is being retained on each coil rather than just the quantity.

Examination of the tilt test results in conjunction with the extended time results provides information about where water is being held in the heat exchangers. Germane to the discussion is a comment on the different areas water can be held in a fin array. Essentially, water can stick on the fin surface, on the tube surface, or as a bridge between louvers, fins, or fin-tube junctions. Water contained on the fin surface is held in equilibrium by surface tension and gravitation forces as shown in Figure 3.27a,b. In Figure 3.27a, the droplet is on a horizontal section and in Figure 3.27b, the droplet is on an inclined surface, such as a louver. In both of these cases, tilting the heat exchanger potentially can cause the droplet to move, depending on the actual surface tension forces and degree of tilt. Likewise, water held along the fin-tube junction as a fillet, also will be affected by an orientation change. Furthermore, water in the channels formed by the fin-tube junction and the fold of the fin itself will not exist as droplets, but as a film that may extend the entire fin depth, and may be easily displaced from a tilted coil. However, a water bridge in a louver is a very stable entity as pictured in Figure 3.27c. The water bridge is held in the louver space predominantly by surface tension forces, and has a maximum size constrained by geometry. Louver angle likely also contributes to louver-bridge stability by changing the effective height of the bridge. The higher the angle, subject to limit of probably around 60° where gravity can begin to dominate, the more stable a bridge will be because of the ability of the solid-liquid-gas interface to be closer to the proper contact angle, and thus have the surface tension holding force as large as
possible. It is unlikely that any degree of orientation change will affect a louver bridge. This was verified by desk-top experiments on a piece of fin stock, and indeed, a fin could be rotated through an entire $360^\circ$ and a louver bridge remained. Bridges between offset strips, as shown in Figure 3.27d, appear to be less stable because of the significantly larger space, relative to a louver, that the water occupies in an offset strip, thus gravity is a large factor in displacing the bridge. The susceptibility of larger bridges to be shed by gravity agrees with the results of McLaughlin and Webb (2000) where a larger louver pitch, and thus larger bridges, had substantially less louver bridging.

The tilt results presented in Table 3.1 show that Coil 5 significantly differed in response from the other two coils tested. All three coils are identical except for fin design, so it may be stated with confidence that the fin geometry of Coil 5 accounted for the difference. From the preceding argument, louver bridging is the likely reason Coil 5 retains more water when tilted relatively to the other three coils. The louver angle in Coil 5 is $42^\circ$, higher than Coil 4, which has only $36^\circ$ louvers. Coil 5 likely has a greater amount of water retained as louver bridges than Coil 4, up to 11% more, or at least louver bridges that are more stable and less apt to swept out during drainage. The extended time drainage results support this stable louver bridge idea. Between the one and four hour data points Coil 6 drained an additional 26% while Coil 5 only drained 13%, consistent with the idea that louver bridges in Coil 5 are locked in and have less tendency to drain.

The tilt test can help determine how condensate is retained by showing how much water is ‘easily’ removed from the coil by gravity forces. Furthermore, under actual operating conditions, flow forces coupled with gravitational forces may create a resultant force on retained condensate that is similar to the tilt test, and thus both vertical and tilt
test results give insights into how much condensate is retained on in a coil under operation. Indeed, the steady-state retention results in Figure 3.19 show qualitatively that Coil 4 drained better and also held less condensate over the entire range of tested velocities. Osada and coworkers (1999) found that coil inclination greatly influenced the thermal performance of an evaporator, and further investigation of this effect could be a valuable tool in future fin development.
Figure 3.1 Coil 1 air-side sensible heat transfer coefficient versus face velocity.

Figure 3.2 Coil 2 air-side sensible heat transfer coefficient versus face velocity.
Figure 3.3 Coil 3 air-side sensible heat transfer coefficient versus face velocity.

Figure 3.4 Coil 4 air-side sensible heat transfer coefficient versus face velocity.
Figure 3.5 Coil 5 air-side sensible heat transfer coefficient versus face velocity.

Figure 3.6 Coil 1 air-side pressure drop in mm of water versus face velocity.
Figure 3.7 Coil 2 air-side pressure drop in mm of water versus face velocity.

Figure 3.8 Coil 3 air-side pressure drop in mm of water versus face velocity.
Figure 3.9 Coil 4 air-side pressure drop in mm of water versus face velocity.

Figure 3.10 Coil 5 air-side pressure drop in mm of water versus face velocity.
Figure 3.11 Coil 1 $j$ and $f$ factors versus air-side Reynolds number.

Figure 3.12 Coil 2 $j$ and $f$ factors versus air-side Reynolds number.
Figure 3.13 Coil 3 j and f factors versus air-side Reynolds number.

Figure 3.14 Coil 4 j and f factors versus air-side Reynolds number.
Figure 3.15 Coil 5 j and f factors versus air-side Reynolds number.

Figure 3.16 Real time condensate retention and humidity ratio for Coil 5 versus time.
Figure 3.17 Real time condensate retention repeatability for Coil 4.

Figure 3.18 Real time plot for Coil 4 showing overshoot at higher velocity.
Figure 3.19 Steady state retention for all coils versus face velocity.

Figure 3.20 Steady state retention per unit of heat transfer area versus face velocity.
Figure 3.21 Drainage test plot for Coil 3 illustrating the repeatability of the experimental technique.

Figure 3.22 Partial plot of the repeatability test results showing the area of maximum error.
Figure 3.23 Dynamic test results for the fast draining coils.

Figure 3.24 Dynamic test results for the sustained draining coils.
Figure 3.25 Extended time drainage test results.

Figure 3.26 Comparison plot between a vertical coil and a coil tilted 10°.
Table 3.1. Tilt test and vertical test comparison.

<table>
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<th>Orientation</th>
<th>Mass</th>
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<th>Mass</th>
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<td>390</td>
<td>25</td>
<td>374</td>
<td>27</td>
</tr>
<tr>
<td></td>
<td>Tilted</td>
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<td></td>
<td>304</td>
<td></td>
<td>284</td>
<td></td>
</tr>
<tr>
<td>4</td>
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<td>23</td>
<td>342</td>
<td>26</td>
<td>335</td>
<td>27</td>
</tr>
<tr>
<td></td>
<td>Tilted</td>
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</tr>
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Figure 3.27 Droplet forces and bridges.
Chapter 4 Conclusions and Recommendations

Heat transfer, pressure drop, and retention data have been collected and presented for five automotive evaporators. The thermal performance tests were conducted under dry and wet conditions and both steady-state and transient retention data were recorded. Additionally, a dynamic drainage test was developed and a variety of coils tested. This chapter contains general conclusions about the effects of condensate on the overall performance of an automotive air-conditioning system, and a recommendation for the focus of future work in this area.

4.1 Conclusions

4.1.1 Thermal-Hydraulic Performance

The heat transfer coefficient decreased and the pressure drop increased for all coils tested. Condensate retention causes the decrease in heat transfer coefficient by forming bridges in the inter-louver space and redirecting the flow from the desired louver-directed flow to duct-directed flow. Additionally, condensate may also form bridges between fins effectively creating a dead region on the fin. Larger louver pitches helped prevent louver bridging, but decreased absolute performance of the coil due to reduced boundary layer restarting (and resulting thicker boundary layers).

The pressure drop increases under wet conditions even though the flow is redirected into duct flow and that would seem to decrease pressure drop. The presence of condensate on the fin surface decreases the effective minimum flow area and causes increased pressure drop. This effect will be even greater with the presence of fin bridges. Coil 4 was shown to have only a slight increase in pressure drop from dry to wet conditions, and a corresponding lower quantity of retained condensate.
4.1.2 Condensate Retention

A highly repeatable method of measuring condensate load-up on evaporators was developed. The quantity of retained condensate reached a steady value in 15-25 minutes depending on operating conditions. The greater the water mass-flux the shorter the time to reach steady-state, as expected. At some specific air-flow rates, an initial 10% overshoot in condensate retention was observed. This effect may be important in determining the drainage characteristics of a coil. It was hypothesized the relative quantity of condensate draining through different modes causes the overshoot, though further experiments are required. Extended test runs were done to investigate the possibility of oscillations in retention caused by shedding mechanisms that may impact the performance characteristics and data interpretation methods. However, the quantity of condensate remained within 1.5% over a 45 minute test, once a steady value was attained.

Steady-state retention results showed a large influence of frontal velocity on the quantity of retained condensate for all the tested coils. The impact of the airflow shear forces varied with geometry over the tested range of face velocities. At the highest velocities the quantity of retained condensate for similar geometries (same heat transfer area, different louver designs) showed a slight tendency to converge, indicating that shear forces start to dominate gravity forces and geometrical influences on retention. A large turnaround section, similar to the centerline divide investigated by Osada and coworkers (1999), resulted in lower condensate retention in Coil 4 compared to Coil 5, though the impact of the higher louver angle is unclear. However, heat transfer was observed to decrease by a greater amount in Coil 4 under wet conditions, likely due to a lower absolute number of louvers without condensate bridging.
4.1.3 Dynamic Drainage

A dynamic drainage test apparatus was built to test the drainage of a variety of coils. Initial drainage response (in a 300 second interval), long-term drainage, and orientation effects were investigated. Two distinct drainage patterns were observed; fast draining coils that reached a steady-state value within about 120 seconds, and sustained drainage coils that continued to substantially drain water for up to four hours. The long-term results, and the results of the tests with the coil face tilted 10° from vertical show the influence of louver bridging on drainage. When a louver bridge exists, gravitational forces are not sufficient to remove the bridge unless the bridge is very large, such as one that may exist between offset strips. Thus, in long-term testing, louvered fins drain much slower than offset strip fins. This drainage behavior is potentially important to biological growth on the fin surface.

The drainage tests also qualitatively predicted the condensate retention in the wind tunnel for Coils 2 through 5; that is, the rank-order for condensate retention in wind-tunnel testing was congruent with the drainage test results. The drainage test over-predicted the absolute quantity of retention, but with the large influence of shear forces on retention this result would be expected. The drainage test reflects retention at the zero-velocity operating condition. Furthermore, the test procedure developed is more accurate than traditional ‘dip-test’ methods, because the relative rates of drainage are important.

4.2 Design Guidelines

In order to compare two fin designs and one design declared ‘better’, it is necessary to define the objective function as it relates to system performance. The optimized design attributes can be any combination of the following parameters.
1) Higher heat transfer coefficient, under either dry, wet, or wet compared to dry.

2) Lower pressure drop across the coil, again under either dry, wet, or wet compared to dry.

3) Lower quantity of condensate retention, under operation or post-operation.

A higher heat transfer coefficient is obviously desirable to either increase the heat duty of a heat exchanger or alternatively and perhaps more importantly, decrease the size of the coil while maintaining the same capacity. A lower pressure drop is beneficial because of pumping power and noise. Overall system efficiency will be improved if less pumping power (the blower fan in automotive applications) is required. Increased blower speed also generally causes increased noise levels. As discussed in the introduction, water remaining on the air-side surface of an evaporator coil after system shutdown is a prime environment for biological activity, and the result is a degradation of air quality through odor and thermal performance through fouling. In summary, care must be taken when rating heat exchangers and developing design guidelines, as it is critical to define the proper objective function based on the focus of the optimization. From the results of this study, the following are recommended design guidelines for improved fin design:

- Keep louver angle at or below 36°. The comparative results of Coils 4 and 5 showed only a slight increase in heat transfer coefficient from using a 42° louver and large increase in core pressure drop.

- Increase the effect of the turnaround section on condensate drainage by increasing the length or openness of the turnaround section, effectively dividing the fin. One interesting possibility is to actually remove part of the heat transfer surface in the center. This may allow a smaller ‘drainage’ section
to obtain the same benefits, thus leaving more area for the high heat transfer louver.

- Offset-strip fins appear to drain condensate better than louver fins over the long-term, such as after system shutdown (a tentative conclusion based on limited test results), and as discussed should form less condensate bridges. Thermal-hydraulic performance data for offset strip arrays would be required to further quantify potential benefits of that design.

4.3 Recommendations for Future Studies

There are several areas where additional research can either enhance the ideas generated in this study, further the condensate retention modeling efforts, or refine the design guidelines for developing improved fins. Three specific research directions are discussed in this section.

Drainage and sweeping effects for automotive evaporators are substantially different from plain-fin coils, there are important interactions between fins and fins and fins and tubes in the automotive evaporator. As a result, it is expected the wet performance of the fins vary will vertically in the fin column due to condensate distribution. The variance in heat transfer coefficient from top to bottom in the fin array would be important to determine both the impact of condensate on performance and refine the retention model by indirectly predicting how much louver and fin bridging exists. McLaughlin and Webb (2000) suggested that lower header design is key to determining how much condensate will 'back-up' vertically in a coil, and vertically resolved performance data would also develop this idea. A heat exchanger could be partitioned into thirds or fourths vertically by embedding thermocouples in the tube at
specific points depending on circuiting, and matching the downstream air temperature thermocouples to the tube locations.

Related to vertical partitioning, is more sophisticated single fin column testing with optical access. One of the keys to condensate modeling is predicting the droplet distribution on the surface, and the highly compact geometry of automotive evaporators make it virtually impossible to observe the fins during operation. Previous efforts at single column testing (Osada et al., 1999, and McLaughlin and Webb, 2000) used only single-sided cooling of the fin, which could affect the distribution on the fin, and definitely affects the distribution on what would be the outer tube surface. The tube-side geometry also needs to be closely matched; however, this will be difficult to accomplish while maintaining optical access, as will cooling both sides of the fin. However, one solution may be to treat the inside of the glass to make it hydrophilic, and then make observations through the water film. This would at least allow a better approximation of drainage routes. Cooling the tube surface would be easier, as dry, refrigerated air could be blown across the outer surface. Results of single column tests would yield important observations required to model droplet, and more importantly, bridge distributions along the fin depth and height.

The final recommended research focus is in asymmetric fin design. All the fins tested in this study (and in the open literature), are symmetric along the fin depth centerline. The large influence of shear forces on condensate retention necessarily means there exists a condensate distribution along the fin depth. Fin designs that have good heat transfer characteristics do not necessarily manage condensate effectively, and by designing the fin to promote high heat transfer in the front of the fin and better condensate drainage in the back, overall performance can be increased. The larger gaps
within an offset strip array may have a tendency to form less condensate bridges. McLaughlin and Webb (2000) noted the louver cut end provides an area where a condensate meniscus can easily grow into a louver bridge. A true offset strip would have no narrowing at the cut end. Offset strips though, do not have the heat transfer performance for the same fin depth as louvers. So a potentially superior fin design is to use small pitch high angle louvers for the highest possible heat transfer in the front of the array and offset strips or larger pitched louvers in the rear of the fin to decrease bridging and promote drainage. The actual configuration of the design would be a primary objective. Initial experiments on scale-up models would be an efficient technique to determine a starting point for an optimal design. Single fin column testing would also be useful, and a modular type test specimen could be built with partial fins that are different types of louvers and offset strips could be built and different configurations could be easily tested.
Appendix A Data Reduction

This appendix describes the data reduction techniques used for this study. Many of the parameters are straightforward calculations. Further discussion of the calculations is presented as needed. The output file from the data acquisition system and the manually recorded data described in Chapter 2 are used as the input variables for the main data reduction routine maintained on a commercial equation solver, Engineering Equation Solver (EES). The advantages of using EES are the thermophysical property functions for moist air are built in and a parametric table can be used to manage multiple test runs. The EES code used for the data reduction is shown in Tables A.1 and A.2

A.1 Mass Fluxes

The coolant-side mass flow rate was calculated from the density and volumetric flow rate; measured with a flow meter that provided a 5-volt dc pulse with 1.849x10⁶ pulses per cubic meter of liquid. Equation A.1 was used to calculate the coolant mass flow rate where $R_e$ is the number of pulses per second. The outlet coolant temperature was used to calculate the coolant density since the meter was located on the return line.

\[ \dot{m}_c = R_e \rho_c \left( \frac{1}{1.849 \times 10^6} \right) \]  

(A.1)

The air-side mass flow rate was calculated from the frontal velocity, flow area, and density. The velocity of the air at the heat exchanger face was measured using a constant temperature thermal anemometer. The anemometer was calibrated by the manufacturer at standard temperature and pressure so the measured velocity needed to be corrected based on the temperature and pressure at the heat exchanger face. The actual velocity was calculated using Equation A.2.
\[ V_{p,c} = V_f \left( \frac{273 + T_f}{294.1} \right) \left( \frac{101.325}{P_{atm}} \right) \]  

(A.2)

The atmospheric pressure is measured with a NOVA laboratory barometer and the air temperature is from the upstream thermocouples. All other air properties were computed using thermophysical property functions that were built into EES.

A.2 Heat Transfer Rates

Equations A.3 through A.5 were used to calculate the heat transfer rates. Calculations were based on measurements made at the test section inlet and outlet. The data used for this study required that air-side and coolant-side heat transfer rates were within 10% including the bias error discussed in Appendix B.

\[ q_{sens} = \dot{m}_{air} C_{p,air} (T_{in,air} - T_{out,air}) \]  

(A.3)

\[ q_{s} = \dot{m}_{air} (h_{in,air} - h_{out,air}) \]  

(A.4)

\[ q_{c} = \dot{m}_{c} C_{p,c} (T_{out,c} - T_{in,c}) \]  

(A.5)

A.3 Fin Efficiency

The fin geometry in the tested automotive evaporator coils can be modeled for fin efficiency calculations as a straight fin with an adiabatic tip. Furthermore, the adiabat for each fin is taken as the centerline due to circuiting and coolant flow rates. Most of the heat exchangers tested use a parallel flow manifold with three coolant passes so each side of the fin is at the same temperature with the exception of the two fins that are along the turnarounds. Additionally, the coolant flow rate is such that the temperature change is between 1.5 and 5.0 degrees Celsius, so any deviation of the adiabat from the centerline for any fin will be much smaller than the fin width.
Fin efficiency for the dry condition tests was computed using a standard equation commonly found in heat transfer texts (for example Incoprera and Dewitt, 1990). The expression for a straight fin with adiabatic tip is,

\[ \eta_f = \frac{\tanh(m_0L)}{m_0L} \]  \hfill (A.6)

where,

\[ m_0 = \sqrt{\frac{h_oP}{k_fA_o}} \]  \hfill (A.7)

For fins where the fin width is much greater than fin thickness \( m_0 \) reduces to

\[ m_0 = \sqrt{\frac{2h_o}{k_f t_f}} \]  \hfill (A.8)

For calculating the fin efficiency under condensing conditions when the fin surface is full wetted, the method presented by Wu and Bong (1994) is used. They consider the driving forces for heat and mass transfer separately. Wu and Bong assumed a linear relationship between \( \omega_s \), the humidity ratio of the saturated air at the wet surface, and \( T_s \), the surface temperature. This assumption allowed them to analytically solve the governing fin surface temperature differential equation when the Colburn-Chilton heat and mass analogy holds. Namely, the heat transfer coefficient and the mass transfer coefficient are related by

\[ \frac{h_o}{h_m} = CpLe^{2/3} \]  \hfill (A.9)

With the fin surface temperature distribution known expressions for the heat transfer from the fin and the maximum heat transfer (occurring if the entire fin was at the fin base temperature) are derived.
\[ q_{\text{fin}} = k_f A_o L m^2 (\theta_b + \theta_p) \tanh(ml) \]  
(A.10)

\[ q_{\text{max}} = k_f A_o L m^2 (\theta_b + \theta_p) \]  
(A.11)

Where \( m \) is related to \( m_o \) in the dry fin equation by,

\[ m^2 = m_o^2 (1 + b \xi) \]  
(A.12)

\[ \xi = \frac{i_{ls}}{C_p L e^{\sqrt{b}}} \]  
(A.13)

and \( b \) is the average slope of the saturation line on the psychometric chart from \( T_b < T_i < T_f \),

\[ b = \frac{\omega_{s,i} - \omega_{s,b}}{T_i - T_b} \]  
(A.14)

The fin efficiency is defined as the ratio of heat transfer to maximum possible heat transfer. Hence,

\[ \eta_f = \frac{\tanh(ml)}{ml} \]  
(A.15)

which is same as the expression for fin efficiency in the dry case with \( m_o \) modified by Eq A.12. The fully wet fin efficiency of Wu and Bong is relatively independent of the humidity ratio of the incoming air.

With the fin efficiency known, the overall surface effectiveness for both wet and dry conditions can be calculated from,

\[ \eta_i = 1 - \frac{A_f}{A_i} (1 - \eta_f) \]  
(A.16)
A.4 Heat Transfer Coefficients

A modified Wilson-plot methodology was used to calculate the air-side heat transfer coefficient. The technique used was an adaptation of the ideas discussed by Briggs and Young (1969), and presented by Rohsenow et al. (1985). Wilson (1915) devised a technique whereby individual thermal resistances could be extracted from the overall system thermal resistance. The general thermal circuit for the studied heat exchangers neglecting air-side and tube-side fouling and no tube-side enhancements is:

\[
\frac{1}{UA} = \frac{1}{(UA)_c} = \frac{1}{(UA)_o} = \frac{1}{(\eta hA)_o} + \frac{1}{(hA)_c} + R_w
\]  

(A.17)

The basic idea behind building a data set for the Wilson-plot method is to hold one side of the heat exchanger (the air, or hot side in this study) constant and systematically vary the flow on the other side. Solving Equation A.17 for \( \frac{1}{U_c} \) and grouping terms yields Equation A.18.

\[
\frac{1}{U_c} = \left( \frac{A_c}{(\eta hA)_o} + R_w A_c \right) + \frac{1}{h_c}
\]  

(A.18)

For turbulent flow through constant cross-sectional ducts, the Nusselt number correlation has the form of

\[
Nu = C_o \, Re^{0.8} \, Pr^{0.4}
\]  

(A.19)

and an equivalent form of Equation A.18 is

\[
\frac{1}{U_c} = \left( \frac{A_c}{(\eta hA)_o} + R_w A_c \right) + \frac{1}{C_i V^{0.8}}
\]  

(A.20)
A plot of \( \frac{1}{U_c} \) versus \( \frac{1}{V^{0.8}} \) with the air-side temperature and mass flux held constant and Equation A.19 being valid is linear and has the form of,

\[
y = mx + b
\]  

(A.20a)

where,

\[
y = \frac{1}{U_c}, m = \frac{1}{C_1}, b = \left( \frac{A_r}{(\eta h)_o} + R_w A_t \right)
\]  

(A.20b)

The slope and intercept of the resulting data set are calculated by a least squares fit to the data points. Equation A.18 can be solved for \( \eta h_o \) directly or iteratively for \( h_o \) with a fin efficiency equation and \( j \) factors can be computed.

Briggs and Young modified the Wilson-plot routine by incorporating the real possibility that the fluid temperature on either side of the heat exchanger could vary from test run to test run. Instead of plotting a velocity function they used the tube-side Nusselt number directly in the form of

\[
Nu_c = C_2 \text{Re}^{n} \text{Pr}^{0.4} \left( \frac{\mu_w}{\mu_m} \right)^n
\]  

(A.21)

and proceeded in a similar manner. A linear regression analysis is carried out to determine the appropriate Reynolds number exponent that minimizes the least squares fit to the data points.

For this study, further modification of the Briggs and Young technique is used to construct the Wilson-plot. The tube-side Nusselt number is calculated using the Gnielinski correlation for transitional flow in tubes:

\[
Nu_D = \frac{(f/8)(\text{Re}_D-1000)\text{Pr}}{1+12.7(f/8)^{1/2}(\text{Pr}^{2/3}-1)}
\]  

(A.22)
where,

\[ f = (0.79 \ln \text{Re}_D - 1.64)^{-2} \]  

(A.23)

The Reynolds number is based on hydraulic diameter, \( D_h = \frac{4A_c}{P} \). The Wilson-plot was constructed by plotting \( \frac{1}{UA} \) versus \( \frac{1}{Nu_D} \). Thus the linear relationship is,

\[
\left( \frac{1}{UA} \right) = \left( \frac{1}{Nu_D} \right) + \left( R_w + \frac{1}{(\eta h A)_o} \right)
\]  

(A.24)

Where the first term on the right hand side is dependent on tube-side Reynolds number and the second group is independent and held constant by maintaining the air-side conditions. Furthermore, the wall resistance for the range of Reynolds numbers in this study accounted for less than 5% of the total resistance and was neglected. The intercept of the least squares fit line to a single set of Wilson-plot data is the air-side resistance,

\[ b = \frac{1}{(\eta h A)_o} \]  

(A.25)

This Wilson-plot technique has effectively allowed the tube-side resistance to be forced to zero by extrapolating to an infinite tube-side Nusselt number. Equation A.25 and the fin efficiency equations are then solved iteratively to obtain the air-side heat transfer coefficient.

The methodology for building a complete set of data is described next. Since the focus of this study is air-side heat transfer an effort was made to maximize the amount of air-side data while maintaining the lowest uncertainty. The slope of the Wilson-plot is independent of the air-side velocity, so the adopted technique was to collect a baseline data set at an air-side Reynolds number at approximately the middle of the range of
interest (~700 based on hydraulic diameter). This baseline data set consisted of Wilson-plot points generated from tube-side Reynolds numbers from approximately 4,000 to 9,000. Subsequent data sets were recorded for other air-side Reynolds numbers (from ~300-1100) at only three or four tube-side flow rates where the Reynolds number varied between 5,000 and 8,000. The slope of the linear fit to the baseline data set was induced on all other data sets and the intercept was calculated from a modified, least squares fit. This technique was really beneficial in condensing experimental test runs because of the difficulty in maintaining constant air-side conditions for the extended duration required to record data for the different tube-side Reynolds numbers. An example modified Wilson-plot is shown in Figure A.1.
Figure A.1 Example of Modified Wilson-plot.
"EES code for computing friction factors and data for Wilson plots.

10 input parameters:
Air inlet/outlet temperatures
Coolant inlet/outlet read voltages
Inlet/outlet depoints in F
Air velocity
Coolant flow meter reading
Atmospheric pressure
Pressure drop across heat exchanger"

"Heat Exchanger Geometry"
A_min=.0252
A_tot=2.52
A_fr=.03923
sigma=A_min/A_fr

"Tube-side Calculations"
Tin_cF=Tin_cC*1.8+32
Tout_cF=Tout_cC*1.8+32

"Tube-side Thermocouple calibration curves"
Tin_cC=1.208200E-01+2.565600E+01*Volt_in_ref+4.646300E-01*Volt_in_ref^2-1.170900E+00*Volt_in_ref^3
Tout_cC=1.045000E-01+2.567500E+01*Volt_out_ref+4.694600E-01*Volt_out_ref^2-1.176600E+00*Volt_out_ref^3

"Coolant flowrate"
Q...c=R_c/l0/700*0.003785
m_c=Q...c*Rho_c
Re_tube=4/(50/1000)*m_c/Vis_c

"Coolant Properties, 40% Concentration"
Rho_c=(3.703704E-7*Tout_cF/3-8.214286E-5*Tout_cF^2-1.150132E-2*Tout_cF+6.789952E+1)/0.06243 {Density Kg/m^3}
Vis_c=(-1.759259E-5*Tin_cF/3+3.764286E-3*Tin_cF^2-3.183122E-1*Tin_cF+12.73762)*(1E-03) {Viscosity Ns/m^2}
k_c=(-1.851852E-8*Tin_cF^3+1.785714E-6*Tin_cF^2+2.839947E-4*Tin_cF+12.73762)/.5778/1000{Conductivity W/mK}
Cp_c=(9.259259E-9*Tin_cF^3-3.571429E-6*Tin_cF^2+4.312169E-4*Tin_cF+.7923810)/2.389E-4/1000{Specific Heat KJ/KgK}

"Tube side heat rate"
Q_ref=m_c*Cp_c*(Tout_cC-Tin_cC)

"Tube-side Nu Calculation using Gniel. correlation"
Nu_D=(fTube/8*(Re_tube-1000)*Pr_tube)/(1+12.7*(sqrt(fTube/8)*(Pr_tube^(2/3)-1)))
fTube=(.79*ln(Re_tube)-1.64)^(-2)
Pr_tube=Cp_c*Vis_c/k_c

"Air-side Calculations"
Table A.1 Friction factor and Wilson plot data EES code listing.
\[
T_{dp\_inC} = \frac{(T_{dp\_inF} - 32)}{1.8} \quad \text{(degrees F to degrees C)}
\]
\[
T_{dp\_outC} = \frac{(T_{dp\_outF} - 32)}{1.8} \quad \text{(degrees F to degrees C)}
\]
\[
T_{dp\_inC} = \text{DewPoint}(\text{AirH}_2\text{O}, T=T_{in\_air}, P=P_{\_atm}, w=w_1) \quad \{\text{determine absolute humidity}\}
\]
\[
T_{dp\_outC} = \text{DewPoint}(\text{AirH}_2\text{O}, T=T_{out\_air}, P=P_{\_atm}, w=w_2) \quad \{\text{determine absolute humidity}\}
\]
\[
P_{\_atm} = (P_{\_hg}) \ast \text{convert(inHg,kPa)} \quad \{\text{Pressure conversion}\}
\]
\[
\text{RH\_in} = \text{RelHum}(\text{AirH}_2\text{O}, T=T_{in\_air}, P=P_{\_atm}, w=w_1) \quad \{\text{Relative Humidity}\}
\]
\[
\text{RH\_out} = \text{RelHum}(\text{AirH}_2\text{O}, T=T_{out\_air}, P=P_{\_atm}, w=w_2) \quad \{\text{Relative Humidity}\}
\]
\[
T_{\_mair} = \frac{(T_{in\_air} + T_{out\_air})}{2}
\]
\[
w_{\_mair} = \frac{(w_1 + w_2)}{2}
\]
\[
\text{Rho\_air1} = \text{Density}(\text{AirH}_2\text{O}, T=T_{in\_air}, P=P_{\_atm}, w=w_1) \quad \{\text{Kg/m}^3\}
\]
\[
\text{Rho\_air2} = \text{Density}(\text{AirH}_2\text{O}, T=T_{out\_air}, P=P_{\_atm}, w=w_2) \quad \{\text{Kg/m}^3\}
\]
\[
\text{Vis\_air} = \text{Viscosity}(\text{AirH}_2\text{O}, T=T_{\_mair}, P=P_{\_atm}, w=w_{\_mair}) \quad \{\text{Ns/m}^2\}
\]
\[
k_{\_air} = \text{Conductivity}(\text{AirH}_2\text{O}, T=T_{\_mair}, P=P_{\_atm}, w=w_{\_mair}) \quad \{\text{W/mK}\}
\]
\[
\text{Cp\_in\_air} = \text{SpecHeat}(\text{AirH}_2\text{O}, T=T_{in\_air}, P=P_{\_atm}, R=\text{RH\_in}) \quad \{\text{KJ/KgK}\}
\]
\[
\text{Cp\_out\_air} = \text{SpecHeat}(\text{AirH}_2\text{O}, T=T_{out\_air}, P=P_{\_atm}, w=w_2) \quad \{\text{KJ/KgK}\}
\]
\[
\text{Cp\_mair} = \frac{(\text{Cp\_in\_air} + \text{Cp\_out\_air})}{2} \quad \{\text{KJ/KgK}\}
\]
\[
\text{hin\_air} = \text{Enthalpy}(\text{AirH}_2\text{O}, T=T_{in\_air}, P=P_{\_atm}, w=w_1) \quad \{\text{KJ/Kg}\}
\]
\[
\text{hout\_air} = \text{Enthalpy}(\text{AirH}_2\text{O}, T=T_{out\_air}, P=P_{\_atm}, w=w_2) \quad \{\text{KJ/Kg}\}
\]
\[
h_{\_mair} = \frac{(\text{hin\_air} + \text{hout\_air})}{2} \quad \{\text{KJ/Kg}\}
\]

"Air flowrate"
\[
\text{m\_dot\_air} = \text{Vol\_air} \ast \text{Rho\_air} \quad \{\text{Kg/s}\}
\]
\[
\text{Vol\_air} = \text{Vel\_air} \ast \text{Flow\_area} \ast \frac{(273 + \text{Tin\_air})}{294.1} \ast \frac{101.4}{P_{\_atm}} \quad \{\text{Velocity probe correction}\}
\]
\[
\text{Flow\_area} = 8 \ast 0.0254 \ast 12 \ast 0.0254 \quad \{\text{Wind Tunnel geometry}\}
\]
\[
\text{V\_max} = \frac{V\_air}{(A_{\_fr}/A_{\_min})}
\]
\[
G\_air = V\_max \ast \text{Rho\_air}
\]
\[
V\_air = \frac{\text{Vol\_air}}{A_{\_fr}}
\]
\[
Pr\_a = \frac{\text{Cp\_mair} \ast \text{Vis\_air}}{k_{\_air} \ast 1000}
\]
\[
\text{Re}_{\_air} = \frac{\text{G\_air} \ast 2.30}{1000 \ast \text{Vis\_air}} \quad \{2.30=\text{hydraulic diameter}\}
\]

"Heat Rates"
\[
\text{q\_sens} = \text{m\_dot\_air} \ast \text{Cp\_mair} \ast (T_{in\_air} - T_{out\_air})
\]
\[
\text{q\_tot} = \text{m\_dot\_air} \ast (\text{hin\_air} - \text{hout\_air})
\]
\[
\text{q\_ave} = \frac{(\text{q\_ref} + \text{q\_tot})}{2}
\]
\[
\text{q\_err} = \frac{(\text{q\_ref} - \text{q\_ave})}{\text{q\_ave}}
\]
\[
\text{LMTD} = (\text{Large} - \text{Small})/\ln(\text{Large}/\text{Small})
\]
\[
\text{Large} = T_{in\_air} - T_{out\_C^C}
\]
\[
\text{Small} = T_{out\_air} - T_{in\_C^C}
\]

"Wilson Plot Data"
\[
\text{Wilsy} = (\text{LMTD} / \text{q\_sens})
\]
\[
\text{Wilsx} = 1 / \text{Nu\_D}
\]

"f factor"
\[
\text{f\_air} = 2 \ast \text{dp} \ast \text{Rho\_air} \ast G\_air \ast 2 \ast A_{\_min} / A_{\_tot} \ast (1 - \sigma^2) \ast (\text{Rho\_air1} / \text{Rho\_air2} - 1) \ast (A_{\_min} / A_{\_tot}) \ast (\text{Rho\_air} / \text{Rho\_air1})
\]
\[
\text{dp} = \text{deltaP} \ast \text{convert(inH2O,pa)}
\]

Table A.1 (cont.) Friction factor and Wilson plot data EES code listing.
"EES code for computing j factors
11 input parameters:
Atmospheric pressure
Inlet/outlet coolant temperatures
air mass flux
Wilson plot intercept
air Prandtl number
air specific heat
air density
air conductivity
inlet/outlet humidity ratios"

\[ \text{st}=\frac{h}{(G \cdot C_p_a)} \]
\[ j=st \cdot \text{Pr} \cdot A \left( \frac{2}{3} \right) \]
\[ \frac{1}{(\eta_o \cdot h \cdot A_t)} = \text{intercept} \]

\[ \eta_{\text{fin}} = \frac{\tanh(m \cdot L)}{m \cdot L} \]
\[ m_0 = \sqrt{2 \cdot h / (k \cdot \text{thickness})} \]
\[ m = m_0 \cdot \sqrt{1 + b \cdot \xi} \]
\[ \text{thickness} = 0.004 \cdot \text{convert(in,m)} \]
\[ k = 0.154 \text{ (kW/m K)} \]
\[ L = 0.15625 \cdot \text{convert(in,m)} \]
\[ \eta_o = 1 - A_{\text{fin}} / A_t \cdot (1 - \eta_{\text{fin}}) \]
\[ A_t = 2.542 \]
\[ 2.296 = A_{\text{fin}} \]

"Wu and Bong"
\[ \frac{(\theta + \theta_p)}{(\theta_b + \theta_p)} = \cosh(M \cdot (L - \text{finwidth}) / \cosh(m \cdot L)) \] (fin temp. distribution)
\[ \text{finwidth} = 0.15625 \cdot \text{convert(in,m)} \]
\[ \theta = T_a - T_t \]
\[ \theta_b = T_a - T_b \]
\[ \theta_p = \xi \cdot C_O / (1 + b \cdot \xi) \]

\[ \xi = \frac{h \cdot f_g}{(C_p_a \cdot \text{Le} \left( \frac{2}{3} \right))} \]
\[ b = \left( w_s_t - w_s_b \right) / (T_t - T_b) \]
\[ a = w_s_b - \left( w_s_t - w_s_b \right) / (T_t - T_b) \cdot T_b \]
\[ C_0 = w_a - a - b \cdot T_a \]
\[ w_s_t = \text{humrat(AirH2O,T=T_t,P=P_atm,D=T_t)} \] (Hum ratio at fin tip)
\[ w_s_b = \text{humrat(AirH2O,T=T_b,P=P_atm,D=T_b)} \] (Hum ratio at fin base)

\[ \text{Le} = \frac{k_a}{(\rho_a \cdot C_p_a \cdot D_{AB}) / 1000} \]
\[ D_{AB} = 0.00143 \cdot T_a ^{1.75} / (P_{\text{atm}} \cdot M_{AB}^{0.5} \cdot (\Sigma_{\text{nu,A}}^{1/3} + \Sigma_{\text{nu,B}}^{1/3})^{2}) \]
\[ \Sigma_{\text{nu,A}} = 19.7 \]
\[ \Sigma_{\text{nu,B}} = 13.1 \]
\[ M_{AB} = 2 \cdot (1 / M_A + 1 / M_B)^{-1} \]
\[ M_A = \text{MOLARMASS(Air)} \]
\[ M_B = \text{MOLARMASS(Steam)} \]

Table A.2 j factor EES code listing.
\[ \rho_a = \text{density(airH}_2\text{O,P=P}_\text{atm},T=T_a,W=w_a) \]
\[ w_a = (w_1 + w_2)/2 \]
\[ h_{fg} = h_3 - h_2 \]
\[ h_2 = \text{ENTHALPY(Steam,x}=0,P=P_1) \]
\[ h_3 = \text{ENTHALPY(Steam,x}=1,P=P_1) \]
\[ P_1 = \text{PRESSURE(Steam,T}=T_a,x=0) \]
\[ 2*T_b = \text{Tin}_C_c + \text{Tout}_C_c \]

Table A.2 (cont.) j factor EES code listing.
Appendix B Uncertainty Analysis

Uncertainties in the experimentally measured and reduced data are presented in this appendix. The errors in the measured parameters are discussed and propagated to estimate the uncertainties in the calculated parameters. Most of the uncertainty calculations are straightforward, but several warrant discussion and clarification.

B.1 Uncertainty in Measured Parameters

The errors associated with the various experimental measurements are shown in Table B.1. The dewpoint of the air was measured by chilled mirror hygrometers at the inlet and outlet and had a measurement uncertainty of ±0.2°C. Coolant flow rate was measured using an oscillating disc type flow meter with a measurement uncertainty of ±1.0%. Air-flow velocities were measured using a constant temperature thermal anemometer with a calibrated uncertainty of ±2.0% of the measured reading. Finally, an electric manometer with an uncertainty of ±0.124 Pa was used to measure the air-side pressure drop across the heat exchanger. Type-T thermocouples were used to measure the air temperature and the coolant temperature. Each thermocouple was individually referenced to a thermocouple located in an ice bath and calibrated to a NIST traceable mercury-in-glass thermometer. Calibration data were fit with fourth order polynomials for each thermocouple. The uncertainties associated with the thermocouples were ±0.2°C. A precision electronic balance was used to measure condensate quantities and had an uncertainty of ±0.1 grams, which is less than 0.05% over the entire range of measurements and will normally be neglected.
B.2 Uncertainty in Calculated Values

The uncertainties in calculated experimental values were determined using techniques by Kline and McClintock (1953). The propagation of error through the data reduction equations introduces an uncertainty in calculated parameters. Equation B.1 was used to determine the uncertainties in the calculated values.

\[ W_y = \left[ \sum_{m=1}^{n} \left( \frac{\partial Y}{\partial X_m} \right) W_m \right]^2 \]

(B.1)

Where \( W_m = \) uncertainty of variable \( m=1,2,3,\ldots,n \)

\( W_y = \) propagating uncertainty in result

\( \frac{\partial Y}{\partial X_m} = \) partial derivative of result with respect to variable \( m \).

When \( Y \) is simply related to \( X_m \) by the following form,

\[ Y = f(X_1, X_2, \ldots, X_n) \]

(B.2)

then Equation B.1 may be rewritten as,

\[ \frac{W_y}{Y} = \left[ \sum_{m=1}^{n} \left( \frac{W_{X_m}}{X_m} \right)^2 \right]^{1/2} \]

(B.3)

B.2.1 Tube-side

A. Heat Transfer Rate

The uncertainty in coolant heat transfer rate is calculated using Equation B.4, where the first three terms on the right hand side is uncertainty from the mass flow rate. As published by the manufacturer, the volumetric mass flow rate meter has an uncertainty of 1.0%. Using a conservative 2.0% uncertainty in the ethylene glycol mixture properties
the uncertainty in tube-side heat transfer rate is estimated to be 10%. However, in the beginning of this study, it was determined a bias error of approximately 5% existed where the computed heat transfer rate on the tube-side was lower than the computed heat transfer rate on the air-side. Different sources of error on both sides of the heat exchanger were investigated and it was concluded the immersion thermocouples on the tube-side were the cause. The coolant flow-rate was required to be relatively large to maintain turbulent flow through the heat exchangers and this caused tube-side temperature differences of less than 2.0°C giving an uncertainty of 10% from the thermocouples. The actual bias error is probably the result of errors in the calibration that are compounded by the high uncertainty in temperature reading. The errors in the average heat transfer rate were very consistent and varied less than 4% from the 5% bias, giving an overall uncertainty within 10%.

\[
\frac{W_{q_{c}}}{q_{c}} = \left[ \left( \frac{W_{\rho_{c,\text{out}}}}{\rho_{c,\text{out}}} \right)^2 + \left( \frac{W_{Rc}}{Rc} \right)^2 + (1.0\%)^2 + \left( \frac{W_{Cp}}{Cp} \right)^2 + \left( \frac{W_{\Delta T}}{\Delta T} \right)^2 \right]^{1/2}
\]  

(B.4)

**B.2.2 Air-side**

**A. \(V_{\text{max}}\)**

Equation B.5 was used to determine the propagated uncertainty for \(V_{\text{max}}\). The frontal velocity was measured directly using a constant temperature thermal anemometer with an uncertainty of 2.0% in the measured reading. Heat exchanger dimensions were measured using a caliper with an uncertainty of 0.025 mm., and the uncertainty for each parameter is based on number of measurements required. \(A_{\text{min}}\) has an additional source of
error due to orientation when the coil is placed in the wind tunnel and is approximately 4%. The uncertainty in $V_{\text{max}}$ was 6% with an uncertainty in $A_r$ of 2.0%.

$$\frac{W_{V_{\text{max}}}}{V_{\text{max}}} = \left[ \left( \frac{W_{V_{\text{air}}}}{V_{\text{air}}} \right)^2 + \left( \frac{W_{A_r}}{A_r} \right)^2 + \left( \frac{W_{A_{\text{min}}}}{A_{\text{min}}} \right)^2 + \left( \frac{W_{\rho_{\text{air}}}}{\rho_{\text{air}}} \right)^2 \right]^\frac{1}{2} \quad (B.5)$$

**B. Reynolds Number**

The uncertainty in air-side Reynolds number based on hydraulic diameter is calculated using Equation B.6. The uncertainty in hydraulic diameter is approximately 1.5%, and the uncertainty in air mass flux is the same as for $V_{\text{max}}$, 6%. Therefore, the uncertainty in Reynolds number is approximately 6.5%.

$$\frac{W_{Re_h}}{Re_{h_Dh}} = \left[ \left( \frac{W_{G_{\text{air}}}}{G_{\text{air}}} \right)^2 + \left( \frac{W_{D_h}}{D_h} \right)^2 + \left( \frac{W_{\mu_{\text{air}}}}{\mu_{\text{air}}} \right)^2 \right]^\frac{1}{2} \quad (B.6)$$

**C. Friction Factor**

The uncertainty in air-side friction factor is determined by Equation B.7. With an uncertainty in $A_{\text{tot}}$ of 4.5% the uncertainty in air-side friction factor is calculated to be 12%.

$$\frac{W_{f}}{f} = \left[ \left( \frac{W_{\Delta P_{\text{ext}}}}{\Delta P_{\text{HX}}} \right)^2 + \left( \frac{2 W_{G_{\text{air}}}}{G_{\text{air}}} \right)^2 + \left( \frac{W_{\rho_{\text{air}}}}{\rho_{\text{air}}} \right)^2 \right]^\frac{1}{2} + \left( \frac{W_{A_{\text{min}}}}{A_{\text{min}}} \right)^2 + \left( \frac{W_{A_{\text{tot}}}}{A_{\text{tot}}} \right)^2 \quad (B.7)$$
D. Heat Transfer Coefficient

The difficult factor in determining the uncertainty in heat transfer coefficient is estimating the error in the Wilson plot intercept. As detailed in Appendix A, the Wilson plot is constructed by plotting $1/UA$ versus $1/\text{Nu}_{\text{tube}}$ and fitting a least squares line to the data for each air-side condition. The variance in the intercept for a single line can be estimated using an equation from Beers (1957),

$$S_b = \left( \frac{\sum (\delta y_n)^2}{k - 2} \right) \times \left( \frac{1}{\sqrt{k \sum x_n^2 - (\sum x_n)^2}} \right)$$  \hspace{1cm} (B.8)

Where $k$ is the number of samples and the first term is the estimated variance in $y$. The 95% interval can now be used to estimate the uncertainty in the $y$-intercept of a single Wilson plot line. Several values were computed and the maximum uncertainty for a data set was 8%. It should be noted that additional statistical analysis would be required to estimate the added uncertainty of fitting subsequent data sets to the same slope. The uncertainty in air-side sensible heat transfer coefficient is calculated using Equation B.9 when the coil is wet and is 11%. For dry coil calculations, only the last three terms in Equation B.9 are used and the uncertainty is 9%.

$$\frac{W_h}{h} = \left[ \frac{W_{r_{\text{air,in}}}}{T_{\text{air,in}}} \right]^2 + \left[ \frac{W_{r_{\text{air,out}}}}{T_{\text{air,out}}} \right]^2 + \left[ \frac{W_{r_{\text{dp,in}}}}{T_{\text{dp,in}}} \right]^2 + \left[ \frac{W_{r_{\text{dp,out}}}}{T_{\text{dp,out}}} \right]^2 \left( \frac{W_{G_{\text{air}}}}{G_{\text{air}}} \right)^2 + \left( \frac{W_{A_{\text{fr}}}}{A_{\text{fr}}} \right)^2 + \left( \frac{2S_b}{b} \right)^2 \right]^{1/2}$$  \hspace{1cm} (B.9)
E. Sensible $j$ factor

The only significant contributions to the $j$ factor uncertainty are from the mass velocity and the air-side heat transfer coefficient $h$. The uncertainty in sensible $j$ factor is calculated using Equation B.10 and determined to be 15%.

\[
\frac{W}{j} = \left[ \left( \frac{W_h}{h} \right)^2 + \left( \frac{W_{G_{air}}}{G_{air}} \right)^2 + \left( \frac{W_{C_{P,air}}}{C_{P,air}} \right)^2 + \left( \frac{W_{Pr}}{Pr} \right)^2 \right]^{1/2}
\]  
(B.10)

B.3 Uncertainty in Measured Condensate Retention

The major sources of error contributing to the uncertainty in condensate retention measurements are from the experimental apparatus and procedures. The error associated with the electronic balance is negligible compared to the other errors. Steady-state retention measurements were collected after the coil was exposed to condensing conditions in the closed wind tunnel. When the tunnel is shut off, the coil continues to drain water and until the catch tray is inserted, the drained amount of condensate is lost. To ascertain the possible quantity of lost condensate a test run using the real-time retention apparatus was performed. After the coil had reached steady-state the tunnel was shut down and mass readings were recorded every 10 seconds for four minutes. It was found the coil drained 5% of the retained condensate in the first 50 seconds. It normally took less than 30 seconds to open the test section and insert the catch tray, so 5% is likely a conservative estimate of the uncertainty in the mass of retained condensate for the steady-state retention tests. The uncertainty for the mass of condensate in the real-time retention tests is also less than 5%. The final value recorded in the real-time tests was
compared to the value obtained by removing the heat exchanger after the final reading
and proceeding to measure the retained condensate using the same procedure as for the
steady-state tests.

B.4 Uncertainty in Dynamic Drainage Tests

Evaluation of the experimental uncertainty for the drainage experiments requires
further consideration of the techniques used during a test run. The maximum source of
error is synchronizing the mass reading with the stopwatch. The first 18 readings are
recorded at five-second intervals, when the maximum drainage occurs, at rates up to 30
g/s. A conservative estimate of the timing error is 0.5 seconds and this translates to a
maximum uncertainty of 5% from data recording. Additional sources of error are coil
orientation and the balance. Tests were conducted to determine the effect of small angular
orientation errors. If the coil is vertical to within two degrees the results varied less than
3% in the initial 60 seconds and less than 2% in the extended-time mass measurements,
and this uncertainty is within the synchronization uncertainty. Equation B.11 combines
the three and yields an overall test uncertainty of 6%.

\[
\frac{W_{m,\text{Drain}}}{m_{\text{Drain}}} = \left[ \left( \frac{W}{m_{\text{balance}}} \right)^2 + \left( \frac{W_{m,\text{min,g}}}{m} \right)^2 + \left( \frac{W_{m,\text{mass}}}{m} \right)^2 \right]^{1/2} \tag{B.11}
\]
Table B.1 Uncertainties in measured parameters.

<table>
<thead>
<tr>
<th>Measured Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta P_{HX}$</td>
<td>$\pm 0.001$ inches of water</td>
</tr>
<tr>
<td>$T_{air, in}$</td>
<td>$\pm 0.2^\circ C$</td>
</tr>
<tr>
<td>$T_{air, out}$</td>
<td>$\pm 0.2^\circ C$</td>
</tr>
<tr>
<td>$T_{c, in}$</td>
<td>$\pm 0.2^\circ C$</td>
</tr>
<tr>
<td>$T_{c, out}$</td>
<td>$\pm 0.2^\circ C$</td>
</tr>
<tr>
<td>$T_{dp, in}$</td>
<td>$\pm 0.2^\circ C$</td>
</tr>
<tr>
<td>$T_{dp, out}$</td>
<td>$\pm 0.2^\circ C$</td>
</tr>
<tr>
<td>Pulses</td>
<td>$\pm 0.5%$</td>
</tr>
<tr>
<td>$V_{air}$</td>
<td>$\pm 2%$</td>
</tr>
</tbody>
</table>
Appendix C Retention Modeling

This section describes condensate modeling work, presents the pertinent equations, and discusses adaptations of the model to the complex fin designs of this study. The two essential elements for predicting the volume of water retained on a coil, and thus the mass, are the distribution of water on the coil surface, and the distribution of droplet sizes in those wetted areas. Additionally, non-standard accumulation (i.e. condensate bridges) will be particularly important as the modeling techniques are applied to more compact geometries.

C.1 Prior Work

Much of the foundation for the retention model discussed herein was set forth by Korte and Jacobi (1997, 1999), and a brief synopsis of their methodology will be given. They started with a droplet with a circular contact line as shown in Figure C.1, and derived the maximum droplet size by performing a force balance between gravitational, air-flow, and surface tension forces.

\[ F_{g,x} + F_{d,x} + F_{s,x} = 0 \]  

(C.1)

The gravitational force is simply

\[ F_{gx} = -\rho_{s}g\forall \sin \alpha \]  

(C.2)

Where the droplet volume is computed from approximating the droplet as a hemispherical cap interfacing the surface at an average contact angle \( \theta_M \).

\[ \forall_{\text{droplet}} = \frac{\pi D^3}{24} \left( \frac{2 - 3 \cos \theta_M + \cos^3 \theta_M \sin \theta_M}{\sin^3 \theta_M} \right) \]  

(C.3)
The drag force is computed using the results of Al-Hayes and Winterton (1981) where a constant drag coefficient of 1.22 was found to apply to bubbles on a submerged surface for $Re_d = \rho_d ud / \mu_a$ from 20 to 400. Since the flow regime would not be expected to change for the Reynolds numbers of this study (and that of Korte and Jacobi) the same $C_d$ is used. The drag force on a droplet is calculated by,

\[ F_{d,x} = -C_d \rho_d u^2 A_p / 2 \]  \hspace{1cm} (C.4)

where $A_p$ is the projected area and $u$ is the local velocity at the droplet mid-height, $h_d / 2$.

\[ A_p = d^2 (\theta_M - \cos \theta_M \sin \theta_M) / \left(4 \sin^2 \theta_M \right) \]  \hspace{1cm} (C.5)

\[ h_d = d \left(1 - \cos \theta_M \right) / \left(2 \sin \theta_M \right) \]  \hspace{1cm} (C.6)

Korte and Jacobi used a laminar (Blasius) velocity profile midway through the heat exchanger using a freestream velocity based on the maximum velocity in the coil. A simplification for the current model is the assumption of a linear velocity profile still computed at the same height using the maximum velocity in the heat exchanger. The justification for this simplification is the relatively low droplet heights and comparison of the results to the more tedious boundary layer results.

The surface tension force was determined by integrating the surface tension force along the contact line assuming a linear variation in contact angle (for full derivation see Korte and Jacobi, 1997.)

\[ F_{ss} = \frac{\gamma a d}{2 \left[ \frac{\sin \theta_R - \sin(\theta_A - \pi)}{\theta_R - \theta_A + \pi} + \frac{\sin \theta_R - \sin(\theta_A + \pi)}{\theta_R - \theta_A - \pi} \right]} \]  \hspace{1cm} (C.7)

A maximum retained droplet size can now be estimated by combining Eqs. C.1, C.2, C.4, and C.7 into:

\[ \Xi_2 d_{max}^2 + \Xi_1 u^2 d_{max} + \Xi_0 = 0 \]  \hspace{1cm} (C.7)
where,

\[ \Xi_2 = -\frac{\rho_w g \pi \sin \alpha}{24} \left( \frac{2 - 3 \cos \theta_M + \cos^3 \theta_M}{\sin^3 \theta_M} \right) \]  \hspace{1cm} (C.8)

\[ \Xi_1 = -\frac{\rho_s C_d (\theta_M - \cos \theta_M \sin \theta_M)}{8 \sin^2 \theta_M} \]  \hspace{1cm} (C.9)

\[ \Xi_0 = \frac{\gamma \pi}{2} \left[ \frac{\sin \theta_R - \sin(\theta_A - \pi)}{\theta_R - \theta_A + \pi} + \frac{\sin \theta_R - \sin(\theta_A + \pi)}{\theta_R - \theta_A - \pi} \right] \]  \hspace{1cm} (C.10)

Korte and Jacobi used the ideas developed by Graham (1969) to calculate the droplet size distribution based on the maximum droplet diameter. Graham conducted detailed photographic studies of condensation and divided the droplets into two different groups and used the following power-law fits.

\[ \Delta N = B_1 d^{1.73} \]  \hspace{1cm} for \( 10 \mu m < d < 0.2d_{\text{max}} \) \hspace{1cm} (C.11a)

\[ \Delta N = B_2 d^{2.8} \]  \hspace{1cm} for \( 0.2 d_{\text{max}} < d < d_{\text{max}} \) \hspace{1cm} (C.11b)

where \( \Delta N \) is the number of drops of diameter \( d \pm 0.2d \) per cm\(^2\). Through digital image analysis the constants \( B_1 \) and \( B_2 \) were determined to be \( 2.042 \times 10^6 \) \( \mu m^{1.73} / cm^2 \) and \( 4.467 \times 10^9 \) \( \mu m^{2.8} / cm^2 \) respectively. Equation C.11 is then recast into a form for the number of droplets a certain diameter per unit area by using \( \Delta d = 0.4\Delta N \).

\[ n(d) = 5.104 \times 10^6 d^{2.73} \]  \hspace{1cm} for \( 10 \mu m < d < 0.2d_{\text{max}} \) \hspace{1cm} (C.12a)

\[ n(d) = 1.117 \times 10^{10} d^{3.8} \]  \hspace{1cm} for \( 0.2 d_{\text{max}} < d < d_{\text{max}} \) \hspace{1cm} (C.12b)

The total mass of condensate on a heat exchanger surface can now be calculated by integrating over all droplet diameters over the entire surface.

\[ M = \rho_w \int_{A} \int_{d} n(\xi) v(\xi) d\xi dA \]  \hspace{1cm} (C.13)
Korte and Jacobi (1999) used the model to successfully predict the quantity of retained condensate on a plain-fin-and-tube heat exchanger with a fin spacing of 6.35 mm, but did not predict the retention for a heat exchanger with a fin spacing of 3.18 mm. Korte and Jacobi concluded interactions between the fins become at smaller fin spacings, and their model does not account for any fin interactions. Yin and Jacobi (2000) added to the model by including condensate bridges between fins at the fin-tube junction. The model of Yin and Jacobi overpredicted the retention for a heat exchanger with a fin spacing of 2.18 mm, and again it was concluded that interactions between droplets was changing the distribution of water on the surface.

C.2 Adaptations

The focus of the modeling efforts for the studied automotive evaporator coils is to attempt to use the model in its current form by separating the heat transfer surface into four different areas with different orientations and restrictions. While this will still fall short of accounting for interactions between droplets on adjacent fins, it will show the applicability of the modeling ideas and give some additional insight into condensate retention.

The studied coils have vertical, inclined, and horizontal surfaces. The tubes and the fin-tube interface are vertical. The louver surfaces are inclined at the louver angle. The turnaround section and the fin area between the louver-end and the tube wall is horizontal. Each of these areas are treated seperately and slightly different. The mass of retained condensate is predicted at face velocities of 1.0 to 2.25 m/s, encompassing the tested range of the wind tunnel retention tests. The force balance was used to compute
maximum droplet size was altered to investigate the possibility of droplet detachment from gravity on the underside of the fin surface. It was found that a droplet is removed from the surface by flow forces prior to gravity for all cases—a result that is especially dependent on contact angles. The actual size of the droplets on the different surface sections are constrained by geometry (the smallest computed $d_{\text{max}}$ was 3.3 mm), and two different ideas were pursued to account for this. The first is to simply set $d_{\text{max}}$ equal to largest possible droplet size such as louver pitch for the louver area and fin pitch for the vertical area. However, this artificially modifies the distribution functions—the number of drops calculated for example at 0.2 mm is dependent on the maximum droplet size which should be derived from the surface conditions not size. Furthermore, by setting $d_{\text{max}}$ all dependency on air velocity is lost, which is not accurate for the tested coils. A more logical approach (and the adopted one) is to compute the maximum droplet size in the normal fashion and adjust Equation C.13 to integrate only up to maximum allowable droplet size. Any droplets that are bigger are assumed to be retained in a different manner or shed. The constraints on the sizes in the different sections are based on observations of fin stock in the condensate visualization glove-box. It was observed that a droplet larger than the louver pitch can remain on a louver by being non-axisymmetric.

Table C.1 displays the relative area percentages of each of the different sections and the constraints on $d_{\text{max}}$. Table C.2 is a summary of the results for applying the model to Coil 4 showing the mass per unit area for each of the different sections and the total predicted mass. The model prediction is compared to the experimentally measured value graphically in Figure C.2, and the EES code is shown in Table C.3.
Figure C.2 shows the model underpredicted the retained mass at all frontal velocities, an opposite trend than observed by Yin and Jacobi (2000) and Kim and Jacobi (2000). In both those studies the authors concluded interactions between droplets on adjacent fins caused the discrepancy. A droplet that is shed on one fin and sweeps along the fin surface could coalesce with a large droplet on a neighboring fin and cause premature sweeping on that fin, thus decreasing the actual mass by a mechanism not captured by the model. In this study, the likely cause for the underprediction is bridging between fins and louvers. The droplets that were larger than the constraint were simply discarded. It is logical that these large drops would be the first to form bridges. The trend in Figure C.2 that at higher velocities the prediction and measured quantities are closer also supports this idea. At higher velocites there should be less bridging as flow forces clear out the bridges (though this phenomenon has not been experimentally observed in this study it is logical).

The model developed by Korte and Jacobi has been adapted and applied to a heat exchanger tested in this study. The model correctly predicts the trends and is within 12% of the measured value throughout the tested velocity range. The model was also applied to Coil 5 and underpredicted the retention by 30-40%, a result that does not signify failure of the model, but rather emphasizes the importance of condensate bridging to the problem. Coil 5 was concluded to have a larger relative amount of condensate bridging than Coil 4. The model does not yet include the physics of bridging or the influence of the vertical variation in condensate retention discussed in Chapter 4, but the limited succes in applying it to a vastly different goemetry than it was originally developed for definitely displays its robustness.
Figure C.1 Forces acting on a droplet on an inclined surface due to gravity, air-flow, and surface tension.

Table C.1 Relative surface areas and maximum droplet size constraints for Coil 4.

<table>
<thead>
<tr>
<th></th>
<th>Percent of Area</th>
<th>Restriction on Dmax (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Louver</td>
<td>40.9</td>
<td>1.5</td>
</tr>
<tr>
<td>Vertical</td>
<td>9.7</td>
<td>1.8</td>
</tr>
<tr>
<td>Turnaround</td>
<td>26.8</td>
<td>2.5</td>
</tr>
<tr>
<td>Flat</td>
<td>22.6</td>
<td>4</td>
</tr>
</tbody>
</table>
Table C.2 Summary of model results for Coil 4.

<table>
<thead>
<tr>
<th>Velocity (m/s)</th>
<th>Mass per unit area (g/m²)</th>
<th>Total Area 2.54 (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Vertical</td>
<td>Louver</td>
</tr>
<tr>
<td>1</td>
<td>65.1</td>
<td>50.5</td>
</tr>
<tr>
<td>1.25</td>
<td>65.3</td>
<td>50.8</td>
</tr>
<tr>
<td>1.5</td>
<td>65.6</td>
<td>51.1</td>
</tr>
<tr>
<td>1.75</td>
<td>65.8</td>
<td>51.5</td>
</tr>
<tr>
<td>2</td>
<td>66.1</td>
<td>52.0</td>
</tr>
<tr>
<td>2.25</td>
<td>66.5</td>
<td>52.5</td>
</tr>
</tbody>
</table>

Figure C.2 Measured and predicted values of steady-state condensate retention for Coil 4.
**EES program to calculate maximum droplet diameter and mass of water per unit area**

\[ \theta_a = 64\times(\pi/180) \quad \text{(advancing contact angle in radians)} \]

\[ \theta_r = 44.0\times(\pi/180) \quad \text{(receding contact angle in radians)} \]

\[ \alpha = \alpha_{d}\times(\pi/180) \quad \text{(angle between fin surface normal and gravity in radians)} \]

\[ u_{\max} = u_a\times1.642 \quad \text{(air-side face velocity, m/s)} \]

\[ f_s = 0.00181 \quad \text{(interfin spacing in m)} \]

\[ \rho = \text{DENSITY(Water, T=20, X=0.0)} \quad \text{(density of liquid water, kg/m}^3) \]

\[ \rho_a = \text{DENSITY(Air, T=25, P=101)} \quad \text{(density of air, kg/m}^3) \]

\[ \sigma = 0.072 \quad \text{(surface tension, N/m)} \]

\[ g = 9.8 \quad \text{(gravitational acceleration, m/s}^2) \]

\[ C_d = 1.22 \quad \text{(drag coefficient per Al-Hays and Winterton)} \]

\[ \theta = (\theta_a + \theta_r)/2 \quad \text{(Average contact angle)} \]

\[ \theta = (\theta_a + \theta_r)/2 \]

\[ \frac{X_{CO}}{(x_{LOa} + x_{LOb}) \times \sigma \times \pi/2} \]

\[ x_{LOa} = \frac{(\sin(\theta_r) - \sin(\theta_a - \pi)}{(\theta_r - \theta_a + \pi)} \]

\[ x_{LOb} = \frac{(\sin(\theta_r) - \sin(\theta_3 + \pi)}{(\theta_r - \theta_a - \pi)} \]

\[ \frac{X_C1}{-\rho_a \times C_d \times (\theta - \cos(\theta) \times \sin(\theta))}{2} \]

\[ X_C2 = \frac{-((2 - 3 \times \cos(\theta) + \cos(\theta) - 3)/3) \times \rho \times g \times \pi \times \sin(\alpha)}{24} \]

\[ y = d_m \times (1 - \cos(\theta)) / 4 \]

\[ u = u_{\max} \times (2 \times y / f_s) \quad \text{(simplification to the model of Korte and Jacobij)} \]

\[ X_{I_0} = (x_{I_0a} + x_{I_0b}) \times \sigma \times \pi / 2 \]

\[ x_{I_0a} = (\sin(\theta_r) - \sin(\theta_a - \pi) / (\theta_r - \theta_a + \pi) \]

\[ x_{I_0b} = (\sin(\theta_r) - \sin(\theta_a + \pi) / (\theta_r - \theta_a - \pi) \]

\[ X_{I_1} = -\rho_a \times C_d \times (\theta - \cos(\theta) \times \sin(\theta)) / 2 \]

\[ X_{I_2} = \frac{-((2 - 3 \times \cos(\theta) + \cos(\theta) - 3)/3) \times \rho \times g \times \pi \times \sin(\alpha)}{24} \]

\[ y = d_m \times (1 - \cos(\theta)) / 4 \]

\[ u = u_{\max} \times (2 \times y / f_s) \]

\[ X_{I_2} \times (d_m)^2 + X_{I_1} \times (d_m) \times u^2 + X_{I_0} = 0 \quad \text{[d_m is maximum diameter in m]} \]

\[ d_{\max} = d_m \times 1e6 \quad \text{[d_max is maximum diameter in microns]} \]

**Integration of droplet size distribution functions**

\[ n_s = (5.104e06) \times (d_s)^{-2.73} \quad \text{(number of droplets with} \quad d < 0.2d_{\max} \text{, per cm}^2) \]

\[ n_b = (1.117e10) \times (d_b)^{-3.80} \quad \text{(number of droplets with} \quad d > 0.2d_{\max} \text{, per cm}^2) \]

\[ V_s = (\pi \times d_s^3) \times ((2 - 3 \times \cos(\theta) + \cos(\theta) - 3)/(\sin(\theta)^3))/24 \quad \text{(small droplet volume, microns}^3) \]

\[ V_b = (\pi \times d_b^3) \times ((2 - 3 \times \cos(\theta) + \cos(\theta) - 3)/(\sin(\theta)^3))/24 \quad \text{(big droplet volume, microns}^3) \]

\[ VT_s = \text{INTEGRAL}(n_s \times V_s, d_s, 10, d_{\max}/5) \quad \text{(total} \quad d_s \text{ volume in microns}^3 \text{ per cm}^2) \]

\[ VT_b = \text{INTEGRAL}(n_b \times V_b, d_b, d_{\max}/5, d_{\max} \_\text{constrained}) \quad \text{(total} \quad d_b \text{ volume in microns}^3 \text{ per cm}^2 \text{ with max drop size constrained)} \]

\[ MpA = \rho \times (VT_s + VT_b) \times (1e-11) \quad \text{(total retention in g/m}^2) \]

**Table C.3 EES code for computing mass per unit area for different surfaces.**
References


