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Electric Air Conditioning for Class 8 Tractors

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ABSTRACT

Air conditioning and heating of heavy-duty truck cabs is an important contributor to engine efficiency, fuel economy and driver comfort. The air conditioner condenser coil and engine radiator typically share a common cooling fan, making it necessary to run the large engine cooling fan to provide condenser cooling. Engagement of the radiator cooling fan consumes a large amount of energy, further contributing to engine exhaust and noise emissions. Even under moderate temperature conditions, when the conventional engine-driven air conditioning compressor is not in use, the belt drive system adds a small speed-dependent parasitic load to the engine.

Electrically driven air conditioning systems have the potential for lower energy consumption than their mechanical counterparts: Electrically driven air conditioning systems can reduce engine idle time by decoupling the air conditioner system from the engine cooling fan while offering near zero parasitic load when not in use.

This paper covers the design, integration, and testing of an electric air conditioning system for a Class 8 tractor for day cab cooling and is a continuation of the efforts initially published in SAE paper 2004-01-1478 [1]. A 42 VDC electric air conditioning system consisting of a variable speed compressor, remote condenser with a variable speed cooling fan, and a thermostatically controlled expansion valve was integrated into an existing Class 8 tractor. The OEM evaporator, in-vehicle ducting, and air speed control were unmodified. The electrical power for the electrified air conditioning system is supplied by a fuel cell auxiliary power unit. The Class 8 tractor has been in-service in the desert of Southern California.

Included in the paper is a detailed description of the different control schemes examined and the control scheme implemented. Energy consumption and driver comfort for each scheme is evaluated. Future system improvements and possible system enhancements are also identified.

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INTRODUCTION

Reduced emissions and increased fuel economy associated with fuel cells and the movement of the automotive industry towards higher voltage electrical systems create the opportunity for synergy between these two technologies. Hybrid vehicles using electrified accessories powered by small fuel cells and the resulting reduction in engine size can take advantage of this synergy and pave the way for full-fledged fuel cell vehicles in the future.

Air conditioning systems are excellent accessory systems for electrification for a number of reasons. The primary benefit of air conditioning system electrification is the decoupling of the compressor and condenser fan from the engine. This decoupling allows smaller sizing of the air conditioning components since they no longer require sizing for effective operation at the low speeds associated with engine idling. The electrification of the condenser fan also allows optimization of the fan speed, and thus heat rejection, for greatest efficiency.

The purpose of this paper is to analyze three different control schemes for an electric air conditioning system powered by a fuel cell and retrofitted to a 2002 Peterbilt 385 tractor trailer.

Although the electric air conditioning control schemes use a Class 8 tractor platform, they are generally applicable to all classes of vehicles due to similarities in air conditioning system design and operation.

Previous work in automotive air conditioning system efficiency improvement has focused on control of the expansion valve [3 and 4], air flow rate across the evaporator [3], and compressor-only control [5]. The focus of this paper is to analyze the effects on energy consumption and driver comfort for control schemes involving both compressor speed and condenser fan speed for the electric air conditioning system. An analysis on the impact of the condenser temperature and compressor speed on air conditioning system performance can be found in [2].

COMPONENTS

The OEM 134a air conditioning system consisted of a compressor, condenser, fixed orifice expansion valve, evaporator, and accumulator. To simplify the conversion into an electric air conditioning (A/C) system, the OEM evaporator and air handling blower were retained.

The 10 cubic inch Sanden SD7 7 cylinder mechanical compressor capable of 25000 Btu/hr was replaced with a 42 VDC Masterflux rolling piston compressor capable of 16500 Btu/hr. The more efficient design allows the max power consumption to be cut by more than one third and allows for full control of the compressor speed by the Masterflux controller due to engine-compressor decoupling.

The Masterflux variable-speed power electronics controller provides a 3-phase AC output to the compressor from the nominal 42 VDC input. The control signal to the controller was provided by a Rapid Prototyping Electronic Control System (RPECS™) processor and ranged from 0-5 VDC, corresponding to compressor speeds of 157-524 rad/s (1500-5000 rpm). The RPECS controller provides supervisory control for all vehicle electrified systems.

A new Modine 3X010632 condenser with comparable heat transfer capacity as the OEM condenser was mounted behind the cabin, replacing the condenser in the engine compartment. The new mounting location removed the condenser from the flow of air during vehicle movement, effectively eliminating any ram airflow effects typically present. A 280 mm variable speed electric fan from Engineered Machined Products (EMP) provided the required air flow across the condenser. The nominal operating speed range for the condenser fan was 0-576.0 rad/s (0-5500 rpm).

The OEM fixed orifice expansion valve was replaced with a 16000 Btu/hr Parker-Hannifin thermal expansion valve to maintain the superheat temperature of the fluid exiting the evaporator at a fixed temperature. The new valve takes advantage of the variable speed compressor by allowing a wider range of refrigerant flow rates.

System instrumentation included pressure and temperature sensors on the condenser and evaporator to calculate the subcooled and superheated values of the refrigerant. Other temperature sensors were also used for control: one measured the air stream of the evaporator outlet, and one measured the cabin temperature. The locations of the pressure and temperature transducers are show in Figure 1.

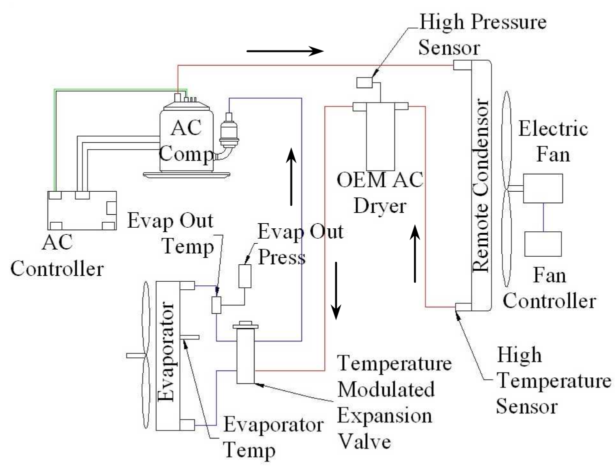

Figure 1: A/C System Instrumentation Schematic

CONTROL SCHEMES

For the purposes of this analysis, three cooling system control schemes were implemented:

EVAPORATOR REFRIGERANT OUTLET TEMPERATURE CONTROL

For the *Evaporator Refrigerant Outlet Temperature Control* scheme, the compressor speed command was proportional to the evaporator refrigerant outlet temperature. When the evaporator outlet temperature increased above an 8.5°C (47.3°F) set point, the A/C compressor command was increased,. When the low side temperature dropped below an 8.25°C (46.85°F) set point, the A/C compressor command was decreased. The two set points were separated by a 0.25°C (0.45°F) dead band to prevent compressor dithering.

The condenser fan speed was proportional to the condenser temperature differential defined as the ambient temperature subtracted from condenser outlet temperature. Hysteresis was added to the control such that a condenser temperature differential of 5°C was required to start increasing fan speed and a temperature differential of 4.85°C was required to start decreasing fan speed. The minimum condenser fan speed allowed was based on the compressor speed command. If the compressor speed command ever reached zero rad/s, the minimum fan speed allowed was 83.8 rad/s (800 rpm). If it never reached zero during operation, the lowest fan speed allowed was 366.5 rad/s (3500 rpm).

EVAPORATOR PRESSURE CONTROL

For the *Evaporator Pressure Control* scheme, the compressor operated according to a bang-bang, or full-on/off, control based on pressure at the evaporator. When the evaporator pressure differential increased above a 2.4 bar (35 psi) setpoint, the compressor speed command was set to 100%. When the evaporator pressure differential fell below a 1.4 bar (20 psi) setpoint, the compressor speed command was set to 0%. The condenser fan was run at a constant speed of 366.5 rad/s (3500 rpm).

CABIN TEMPERATURE CONTROL

Table 1: Air Conditioning System ControlScheme Summary

Control Scheme	**Compressor Logic**	**Condenser Fan Logic**
Evaporator Refrigerant Outlet Temperature Control	Proportional Control to Evaporator Outlet Temperature	Proportional Control to (Condenser Outlet Temperature - Ambient Temperature)
	Evaporator Outlet Temp Increase above 8.5°C to Increase Speed Command	> 5°C Temperature Differential to Ramp Up
	Evaporator Outlet Temp Decrease Below 8.25°C to Decrease Speed Command	< 4.85°C Temperature Differential to Ramp Down
		Minimum Speed 366.5 rad/s if Compressor Speed Command > 0
Evaporator Pressure Control	Bang-Bang Control to Evaporator Pressure	Constant 366.5 rad/s
	Evaporator Pressure > 2.4 bar = 100% Speed Command	
	Evaporator Pressure < 1.4 bar = 0% Speed Command	
Cabin Temperature Control	PI Control to Desired Cabin Temperature	Proportional Control to (Condenser Outlet Temp - Ambient Temp)
	Desired Cabin Temperature = 24°C	> 5°C Temperature Differential to Ramp Up
		< 4.85°C Temperature Differential to Ramp Down
		Minimum Speed 366.5 rad/s if Compressor Speed Command > 0

For the *Cabin Temperature Control* scheme, the compressor speed was controlled to a desired cabin temperature via a proportional/integral (PI) controller. This control scheme did not require hysteresis. For all tests conducted in this control scheme comparison, the desired cabin temperature was set to 24°C. The condenser fan speed control logic was the same as the condenser fan speed logic used in the *Evaporator Refrigerant Output Temperature Control scheme*.

The three control schemes summarized in Table 1 all share a certain performance characteristic. With a 0V compressor speed command provided by the RPECS controller, the compressor controller would maintain a minimum compressor speed of 157 rad/s (1500 rpm). This impacts the control schemes equally by always maintaining a minimum compressor speed.

RESULTS

The air conditioning system control schemes were each implemented on a Class 8 tractor and tested in the Southern California desert. For all tests, the vehicle was parked on a paved parking lot in full sunlight with the ignition off and the vehicle cabin doors and windows closed. Ambient conditions during testing for all three control schemes ranged from 44 to 46 degrees Celsius.

After soaking in the sun to heat the cabin, a particular air conditioning system control scheme was implemented until the cabin temperature reached the desired temperature. For the *Evaporator Refrigerant Output Temperature* and *Evaporator Pressure* control schemes, the evaporator fan speed was initially set to its highest setting. Once the desired temperature was reached, the vehicle operator reduced the speed to stabilize the cabin temperature. For the *Cabin Temperature* control scheme, the evaporator fan speed was set to its highest setting and was not adjusted by the vehicle operator. Comparable cabin temperatures were maintained to enable accurate energy comparisons among the three air conditioning system control schemes.

Upon reaching steady state, the air conditioning system operation continued while data was recorded for a period of fifteen minutes at five-second intervals. The procedure was repeated for each of the three air conditioning system control schemes.

EVAPORATOR REFRIGERANT OUTLET TEMPERATURE CONTROL

Figures 2 through 4 present the results of the *Evaporator Refrigerant Output Temperature Control* scheme.

During the air conditioning operation, the thermostatic expansion valve tended to hunt for a steady state operating point for the duration of the test, resulting in the evaporator outlet pressure and outlet temperature oscillations seen in Figures 2 and 3, respectively. Although valve hunting can result from an underdamped control scheme, the hunting in this application appeared to result from the expansion valves inherent proportional response to the evaporator outlet temperature. For constant compressor and fan speeds, the valve continued to hunt at a frequency independent of any system setting.

To buffer the compressor command response to the evaporator outlet temperature oscillations, an *average* evaporator outlet temperature was used to determine the compressor speed command. The average evaporator outlet temperature was calculated using the previous 25 seconds of evaporator outlet temperature as shown in Figure 3.

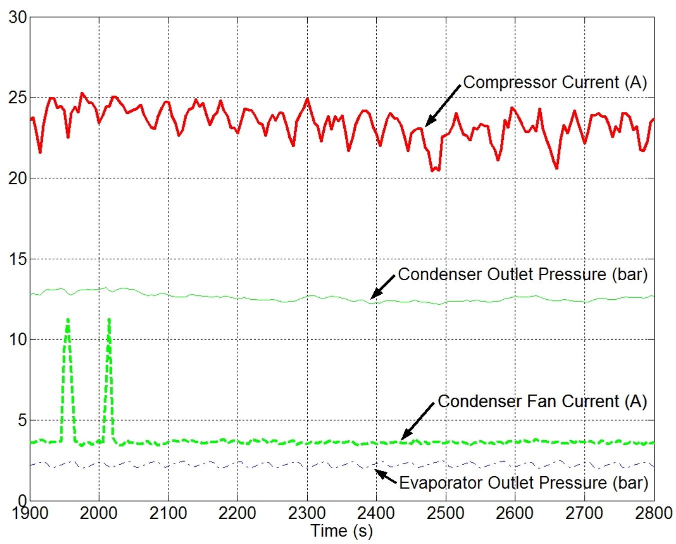

Figure 2: *Evaporator Refrigerant Output Temperature Control* Scheme A/C System Pressure Response to Compressor and Condenser Fan Current

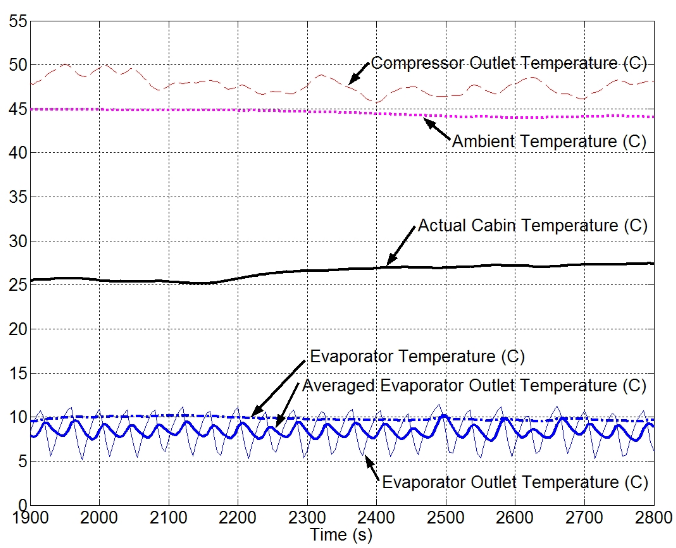

Figure 3: *Evaporator Refrigerant Output Temperature Control* Scheme Cabin Temperature Relationship to A/C System and Ambient Temperatures

As shown in the charts, the averaging period of 25 seconds was not sufficient to fully damp the evaporator outlet temperature oscillations. Longer periods necessary for increased damping, however, tended to disrupt the overall system controllability. The 25 second averaging period was chosen as a best case averaging period.

Figure 4 shows the upper and lower evaporator outlet temperature range of 8.25°C and 8.50°C, respectively. As expected, when the average evaporator temperature exceeded the 8.50°C upper limit, the compressor speed command increased proportionally to the average evaporator outlet temperature. Conversely, when the average evaporator outlet temperature fell below the 8.25°C lower limit, the compressor speed command decreased proportionally to the average evaporator outlet temperature. Due to the relatively narrow upper and lower evaporator outlet temperature limits selected, however, the compressor speed command tended to track the average evaporator outlet temperature oscillations (Figure 4). The resulting oscillatory current draw by the compressor is seen in Figure 2.

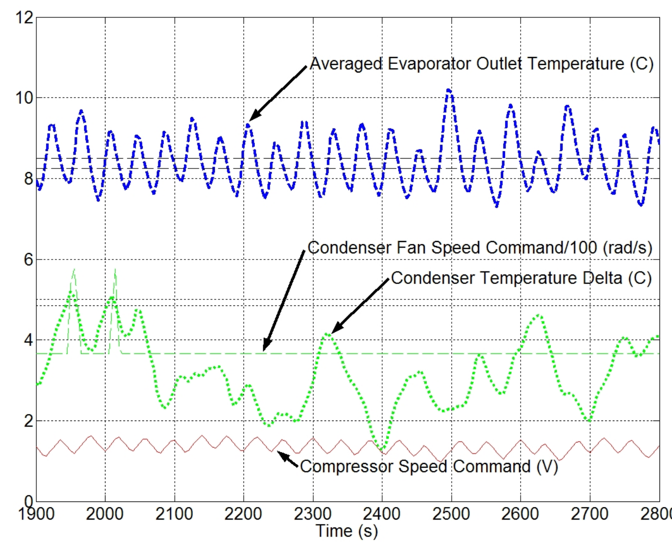

Figure 4: *Evaporator Refrigerant Output Temperature Control* Scheme Compressor and Condenser Fan Commands Resulting from Evaporator Outlet Temperature and Condenser Temperature Differential

Figure 4 shows the condenser fan speed command response to the condenser temperature differential (condenser outlet temperature – ambient temperature). As expected, when the condenser temperature differential exceeded the 5.00°C upper limit, the condenser fan speed command ramped up, and when the condenser temperature differential fell below the 4.85°C lower limit, the condenser fan speed command ramped down.

Since the compressor speed command never reached zero, the minimum condenser fan speed was 366.5 rad/s (Figure 4). As a result of the continuous condenser fan speed, the condenser temperature differential exceeded the upper temperature limit only twice during the test. For both of these occurrences, the condenser fan speed command ramped up as expected.

The resulting condenser fan current draw spikes can be seen clearly in Figure 2.

As seen in Figure 3, this control scheme was able to maintain the cabin temperature between 25°C and 28°C by means of evaporator fan speed control by the vehicle operator.

EVAPORATOR PRESSURE CONTROL

Figures 5 through 7 present the results of the *Evaporator Pressure Control* scheme.

For the *Evaporator Pressure Control* scheme, the compressor was controlled using a bang-bang, or full-on/off, control based on the pressure at the evaporator. As seen in Figure 7, the compressor command remained 100% throughout the test with the exception of 3 periods during which the evaporator pressure dropped below 1.4 bar (20 psi). During these three periods, the compressor speed command was set to 0%, and the compressor current dropped drastically as expected (Figure 5). Note that the compressor current shown in Figure 5 never reaches zero because the 0% speed command results in a minimum compressor speed of 157 rad/s (1500 rpm).

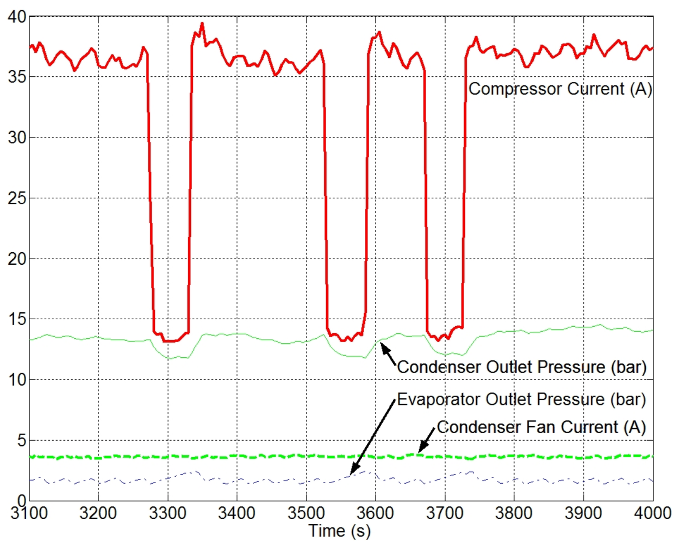

Figure 5: *Evaporator Pressure Control* Scheme A/C System Pressure Response to Compressor and Condenser Fan Current

The 0% compressor speed command resulted in drops in the condenser outlet pressure and compressor outlet temperature (Figures 5 and 6). The resulting increases in the evaporator temperatures (Figure 6) subsequently drove the evaporator outlet pressure up past the 2.4 bar (35 psi) threshold and reinstated the 100% compressor speed command as seen in Figure 7.

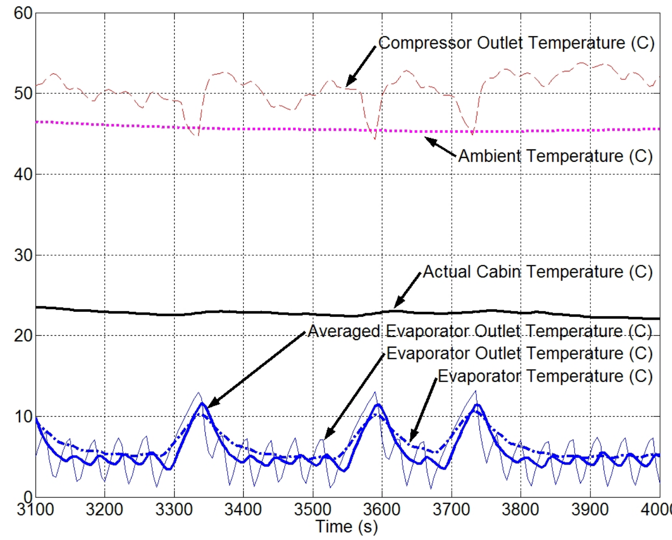

Figure 6: *Evaporator Pressure Control* Scheme Cabin Temperature Relationship to A/C System and Ambient Temperatures

Figure 7 verifies that the condenser fan speed command is fixed at 366.5 rad/s (3500 rpm) at all times in the *Evaporator Pressure Control* scheme. As expected, the condenser fan current also remains constant (Figure 5). The *Evaporator Pressure Control* scheme was able to maintain the cabin temperature between 22°C and 24°C (Figure 6) by means of evaporator fan speed control by the vehicle operator.

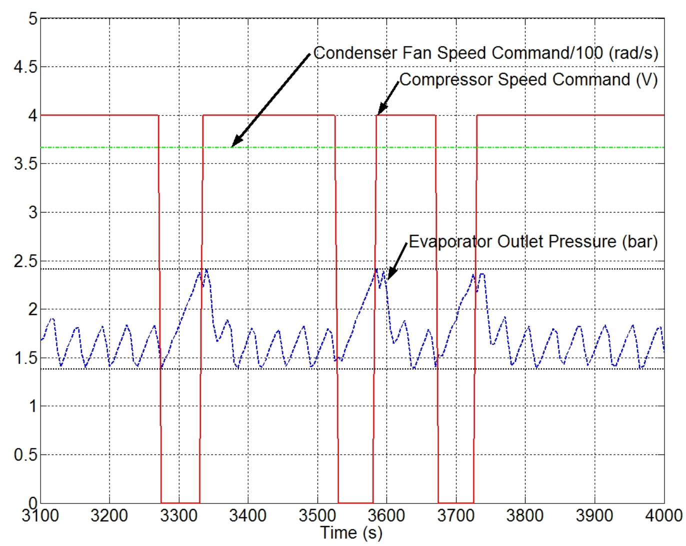

Figure 7: *Evaporator Pressure Control* Scheme Constant Condenser Fan Command and Compressor Command From Evaporator Pressure

CABIN TEMPERATURE CONTROL

The results of the *Cabin Temperature Control* scheme are shown in Figures 8 through 10:

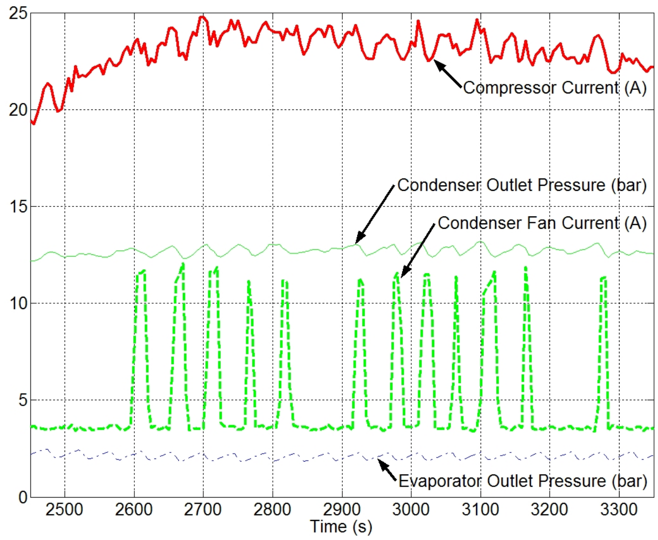

Figure 8: *Cabin Temperature Control* Scheme A/C System Pressure Response to Compressor and Condenser Fan Current

As with the *Evaporator Refrigerant Output Temperature Control* scheme, the thermostatic expansion valve hunted for a steady state operating point resulting in the evaporator outlet pressure and outlet temperature oscillations seen in Figures 8 and 9, respectively. Since the compressor speed command was controlled off the 24°C desired cabin temperature, however, the compressor speed command oscillations seen with the *Evaporator Refrigerant Output Temperature Control* scheme are no longer present (Figure 10).

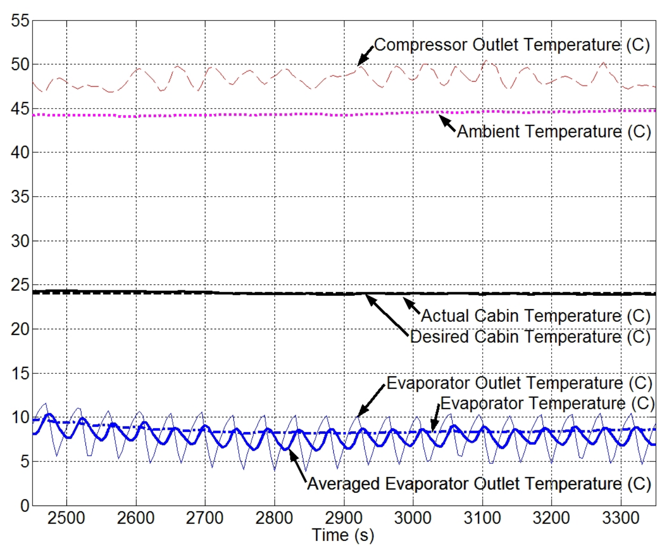

Figure 9: *Cabin Temperature Control* Scheme Cabin Temperature Relationship to A/C System and Ambient Temperatures

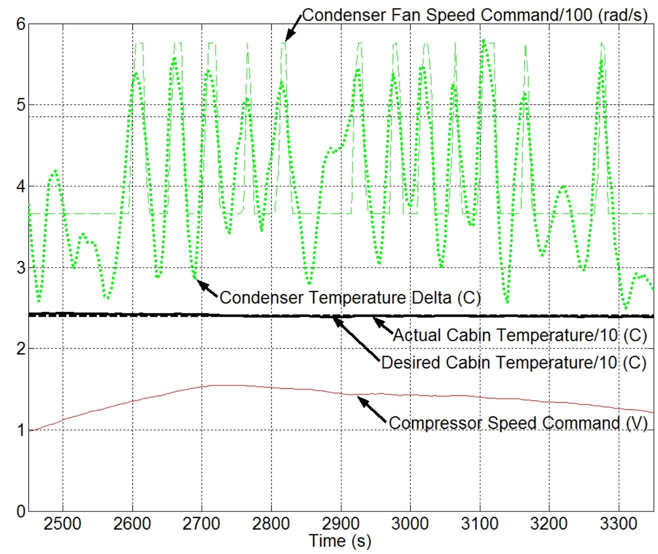

Figure 10: *Cabin Temperature Control* Scheme Compressor and Condenser Fan Commands Resulting from Cabin Temperature Control and Condenser Temperature Differential

Although the compressor speed command does not exhibit oscillatory behavior, the actual compressor current shown in Figure 8 shows signs of oscillations.

The condenser fan command logic used in *Cabin Temperature Control* scheme is identical to that of the *Evaporator Refrigerant Output Temperature Control* scheme. As with the *Evaporator Refrigerant Output Temperature Control* scheme, the compressor speed command never reached zero, so the condenser fan speed command exhibited a minimum of 366.5 rad/s (3500 rpm) (Figure 10). As shown in Figure 10, the condenser temperature differential crossed the 5.00°C upper and 4.85°C lower condenser differential temperature limits several times. Therefore, the condenser fan speed command ramped up and down proportionally. The expected corresponding spikes in the condenser fan current are apparent in Figure 8.

Since the Cabin Temperature Control scheme controlled the compressor speed command via proportional-integral control to the 24°C desired cabin temperature, it was expected that the cabin temperature would track 24°C closely. As shown in Figure 9 and 10, the cabin temperature was indeed controlled very close to 24°C without intervention by the vehicle operator.

RESULTS SUMMARY

Control Scheme Energy Comparison

Table 2 summarizes the energy consumption for the three control schemes. Current and voltage readings from transducers located throughout the system were used to calculate the compressor and condenser fan energy consumption. The total energy for each control scheme was calculated as the energy consumption sum for these two components.

Table 2: Air Conditioning System Energy Consumption for the Various Control Schemes

	Evaporator Refrigerant Outlet Temperature Control	**Evaporator Pressure Control**	**Cabin Temperature Control**
Compressor Logic	Proportional	Bang-bang	Proportional
Condenser Fan Logic	Proportional	Constant	Proportional
Total Energy (kJ)	932	1228	975
Compressor Energy (kJ)	796	1104	801
Condenser Fan Energy (kJ)	136	124	174
Average Ambient Temperature (°C)	44.5	45.6	44.4

Table 2 shows that the *Evaporator Refrigerant Output Temperature Control* scheme consumed the least energy, followed by the *Cabin Temperature Control*, then *Evaporator Pressure Control* scheme. The *Evaporator Pressure Control* scheme exhibited the highest energy consumption by the compressor due to its nearly constant 100% compressor command. For similar average compressor speeds, a control scheme resulting in larger or more frequent compressor oscillations will consume more energy than a control scheme resulting in smaller or less frequent compressor oscillations. The higher energy consumption results from compressor current oscillations. Although the compressor command in the *Evaporator Refrigerant Output Temperature Control* scheme oscillated throughout the test, the compressor command was generally lower than that of the *Cabin Temperature Control* scheme. As a result, the *Evaporator Refrigerant Output Temperature Control* scheme exhibited lower compressor energy consumption than the *Cabin Temperature Control* scheme.

The *Evaporator Pressure Control* scheme showed the lowest condenser fan energy consumption by using a steady 366.5 rad/s (3500 rpm) fan speed. Condenser fan energy consumption was proportional to use of the fan above the nominal 366.5 rad/s (3500 rpm) operating speed: the *Evaporator Refrigerant Output Temperature Control* scheme showed slightly higher condenser fan energy consumption, and the *Cabin Temperature Control* scheme showed the highest energy consumption. Due to the the cubic relationship between fan power and speed, the increase in fan energy consumption with speed is expected.

Control Scheme Driver Comfort Comparison:

In addition to energy consumption considerations, the driver comfort is an important factor in the control scheme evaluation. While all three control schemes adequately maintained the desired cabin temperature, several other factors influence the overall driver comfort.

The *Evaporator Refrigerant Ouput Temperature Control* and *Evaporator Pressure Control* schemes operation and therefore driver comfort are similar to that of typical OEM air conditioning systems. These control schemes are characterized by constantly cooling the air circulated through the vehicle cabin and using the evaporator fan speed as a basic means of controlling the cabin temperature. These control schemes also facilitate OEM defrost functions.

From a system durability standpoint, both schemes effectively monitor the refrigerant state in the evaporator, and are therefore capable of preventing compressor slugging, or refrigerant liquid return from the evaporator. Since the *Evaporator Pressure Control* scheme controls the compressor speed based on the evaporator pressure, there is inherent capability for compressor protection from loss of refrigerant. The *Evaporator Refrigerant Output Temperature Control* scheme lacks this inherent protection.

The *Cabin Temperature Control* scheme operates differently than typical OEM air conditioning systems because it varies the temperature of the evaporator air stream directly via the refrigerant temperature. As a result, this control scheme would require a specialized defrost mode. Although this control scheme requires only that the vehicle operator set the desired cabin temperature, the system response is highly dependent on the placement of the cabin temperature sensor.

Unlike the other control schemes, the *Cabin Temperature Control* scheme does not protect against compressor slugging. Additionally, the *Cabin Temperature Control* scheme does not protect the compressor against the loss of system refrigerant.

CONCLUSIONS

The design, integration, and testing of an electric air conditioning system for a Class 8 tractor was presented. Three control schemes were evaluated for energy consumption and driver comfort.

The *Evaporator Refrigerant Output Temperature Control* scheme was the most efficient control scheme of the three schemes tested with the specified hardware. As with most optimization work, however, substitution of different hardware components could show different results. The energy consumption of these control schemes relied heavily upon the compressor energy consumption at its minimum operating speed. Therefore, the compressor energy consumption at its minimum operating speed plays a major role in the overall energy consumption of the control scheme.

For all three of the control schemes, it was apparent that the thermostatically controlled expansion valve hunted arount it's mechanical operating point. An electronically controlled thermal expansion valve would reduce compressor energy consumption and allow better optimization of the control schemes.

While all three control schemes adequately provided driver comfort, the *Cabin Temperature Control* scheme had a slight edge over the other control schemes for maintaining the desired cabin temperature with nominal additional energy consumption. Other factors such as system fail-safe operation favor the *Evaporator Pressure Control* scheme.

FUTURE WORK

Possible future work includes;

- Optimization of the *Evaporator Refrigerant Output Temperature Control* scheme using wider low side temperature tolerances for compressor speed control.
- Revising condenser fan speed tolerances and proportional-integral control to reduce energy consumption in the *Cabin Temperature Control* scheme. The potential result would be lower energy consumption than the *Evaporator Refrigerant Output Temperature Control* scheme and more precise cabin temperature control.
- Optimizing control of the compressor speed and replacing the mechanical expansion valve with an electrical expansion valve to reduce hunting and associated compressor energy losses in the *Evaporator Refrigerant Output Temperature Control* scheme.
- Replacing the mechanical expansion valve with a fixed orifice tube and low-side accumulator. Although the accumulator could introduce a capacity impact during transients, the fixed orifice tube would remove the transients experienced with the mechanical expansion valve.
- Optimization through operation at maximum compressor speed and variable condenser fan speed or maximum condenser fan speed and variable compressor speed.
- Redesign the mechanical positioning and/or mounting of the remote condenser to include ram air from vehicle motion. The potential result would be a reduction of condenser fan energy consumption.

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