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A Low Emission, Electrified Solution For Refrigerated Trucking

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ABSTRACT

Multiple factors are driving commercial trucking toward electrified solutions to lower harmful greenhouse gas emissions, improve efficiency and reduce fuel consumption. As part of this, the refrigerated trucking industry is searching for methods to meet these new requirements. Currently, most refrigerated truck solutions use a diesel engine to provide power for the refrigeration system. This diesel engine is a source of pollutants which are federally regulated by the Environmental Protection Agency (EPA) and state regulated by groups such as the California Air Resource Board (CARB). In addition, this diesel engine is a source of noise and excessive fuel consumption. Today electric systems do exist in the form of eutectic plate evaporator systems. These systems utilize electric compressors which are plugged in at night to freeze a bank of eutectic plates. During the delivery route these units use the stored cooling in the plates to keep product cold. This operation leads to low emissions, but route duration and box temperature control can pose a challenge preventing these units from being used in some applications.

A refrigeration system has been developed which significantly decreases carbon dioxide impact and eliminates fuel consumption while at the same time allowing for precise temperature control for an entire 10-hour route. The system utilizes a fully electric condensing unit featuring a high efficiency variable speed scroll compressor with economized vapor injection and 48-volt input variable speed drive. Inside the refrigerated body is a dual evaporator system consisting of eutectic plates and direct expansion evaporator. Powering the system is a 48-volt battery pack which is charged by utility power at night and via solar panels during the day. The entire system is controlled with a single system controller which is programmed to optimize both efficiency and reliability. This paper will present the system design, estimates of anticipated performance and field test data to validate these estimates. Results will be compared to traditional refrigeration technologies.

1. INTRODUCTION

Multiple factors are driving refrigerated trucking away from diesel engine based systems and toward electric solutions. Government regulation of emissions is occurring in this industry with federal greenhouse gas (GHG) regulations from the EPA in effect in 2018 and state regulations such as the California Air Resource Board (CARB) actively controlling emissions based on model year. Financial justification is also present to switch from diesel engine based systems to electric systems. Fuel consumption for powering the refrigeration system is eliminated in favor of utility power which significantly lowers the cost per energy delivered. Maintenance costs are reduced by eliminating the engine, belts and seals required in a traditional system in favor of a totally hermetic system that is free from refrigerants leaks associated with shaft seals. Finally, electronics technology has advanced in the fields of power electronics, solar energy and batteries to make it possible to provide adequate power to refrigerate a class 7 vehicle for a full day of operation.

This paper discusses the design of a fully electric refrigerated truck system, presents predictions on system performance versus thermal loading and results from the field as compared to a traditional engine driven refrigeration system.

2. TRADITIONAL TECHNOLOGIES IN REFRIGERATED TRUCKING

Truck refrigeration systems traditionally fall into 3 basic designs: self-powered, direct-drive, and eutectic. In self-powered systems a dedicated diesel engine is integrated into the system and powers all refrigeration loads. This typically includes a belt driven compressor and electric fans driven by an on-board alternator. Short refrigerant line runs to and from the compressor enable higher efficiency in this type of system. The presence of an on-board diesel engine however adds to overall size, cost and required maintenance of the system.

A direct-drive system derives its power by utilizing the internal combustion engine already present on the truck. In this design a compressor is mounted to the truck engine and refrigerant lines are routed to remote heat exchangers located elsewhere on the truck. Electrical power is supplied by the truck electrical system. Removal of the dedicated diesel engine is an improvement in size and cost over the self-powered system, however long line runs to and from the compressor negatively impact the efficiency of the system.

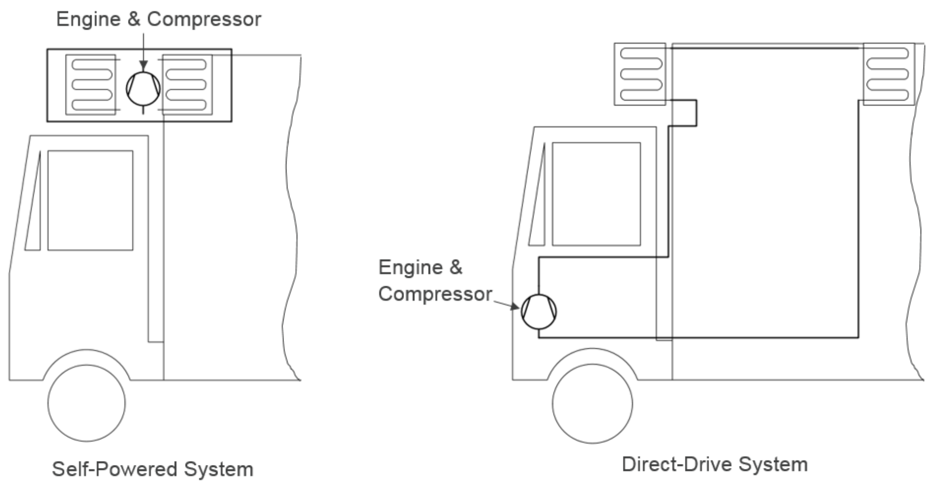

Figure 1: Self-Powered & Direct-Drive Refrigeration Systems

Eutectic systems, sometimes called "cold plate systems", are quite different and in fact are all electric. This design uses stored energy in the form of frozen fluid to cool the refrigerated box. The eutectic plates contain a brine solution designed to freeze at an appropriate temperature corresponding to the goods it is carrying. The plates act as an evaporator in the refrigeration cycle with an electric condensing unit mounted elsewhere on the truck. Typical operation is to power the condensing unit only overnight to freeze plates. During the delivery route the box is cooled by using the stored energy in the plates. This method of refrigeration has the notable benefit of eliminating the diesel engine and associated fuel and maintenance costs. The main drawback of this method is the inability of the refrigeration system to run while on route. This reduces the achievable route duration and temperature control. As a result, these systems are typically employed for frozen goods and dairy products.

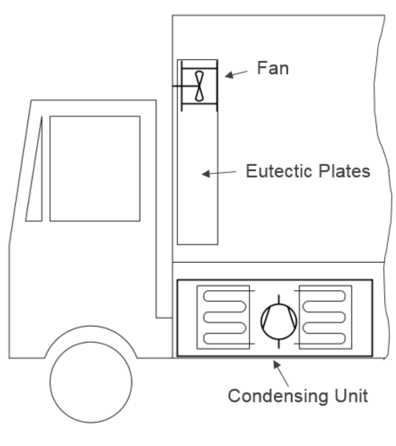

Figure 2: Eutectic Plate Refrigeration System

3. HYBRID EUTECTIC SYSTEM

The subject system of this paper is referred to as a hybrid eutectic system. The term hybrid refers to the nature of how the box is cooled by utilizing both a direct expansion evaporator and eutectic plate storage. The truck refrigeration system is fully electric and contains a 48-volt condensing unit, eutectic plate evaporator (containing 4 plates), and direct expansion evaporator. Power to run the system is supplied by an on board 48-volt battery pack. 48-volt power was selected to stay below the 60-volt level at which point additional safety precautions must be used. 48-volt power is also knowns to be an emerging technology in the automotive market, Truett (2016). Batteries are charged via utility power and on-board charging system at night while the truck is docked and via roof mounted solar panels during the day. The solar panels measure 139.3 ft^2 (12.9 m^2) and have the capability of producing 12.9 Watts/ ft^2 (139 Watts/m^2).

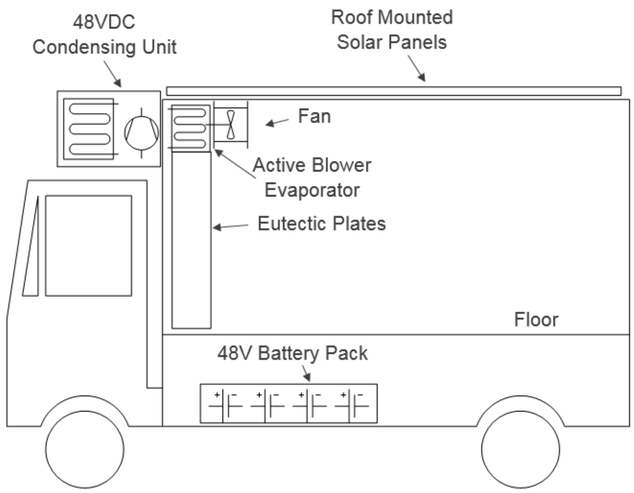

Figure 3: Truck System Diagram

The condensing unit contains a 48-volt variable speed drive powering a variable speed scroll compressor, micro-channel type condenser and economized vapor injection (EVI) circuit. The EVI circuit allows further capacity and efficiency improvements, particularly at high ambient operating conditions. Benefits have been shown by Bahman, A., Groll, E (2014). Solenoid valves are placed in the system to allow adjustment of evaporator operation. This gives the ability to cool the box via eutectic only, direct expansion evaporator only, or both combined.

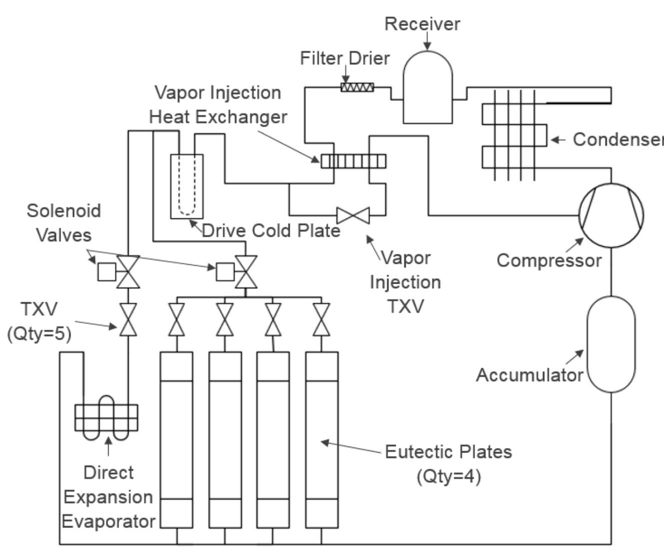

Figure 4: Refrigeration System Diagram

Of note is the 48-volt condensing unit design which can run at a low 48-volt level but at the same time produce the required refrigeration capacity to maintain a 20 ft. (6.1 m) long refrigerated truck at 36°F (2.2°C) under all ambient requirements. Two new components were developed specifically for this application, the 48-volt variable speed drive and variable speed compressor.

3.1 48-Volt Variable Frequency Drive

The variable frequency drive (VFD) was developed specifically for this application by optimizing for the low voltage DC input. A traditional VFD takes in AC line voltage, rectifies it to DC and stores voltage in a DC Bus before utilizing pulse width modulation (PWM) to run a motor at varying frequencies. The VFD discussed in this paper eliminates the rectification circuit and instead takes in DC voltage in the range of 39-59 volts and boosts the voltage to a sufficient DC bus to power the compressor. The power output of the VFD is 2.6 kW at a nominal 48-volt input. Due to the low voltage input high amperage will exist on the input side of the drive. To enable peak power output and long-term reliability the drive is mounted to a plate heat exchanger and cooled by subcooled liquid refrigerant from the economized vapor injection circuit. Utilizing this cooling method provides the drive with a stable cooling medium below 100°F (38°C) in all ambient environments.

3.2 Variable Speed Compressor

The compressor developed for this application is a variable speed scroll compressor. Displacement was selected to be 23 cubic centimeters (cc) with a speed range of 1200 to 5400 RPM. These values were selected to match the required cooling load as well as the available power output from the variable speed drive. A vapor injection port has been added allowing injection into a mid-compression pocket of the scroll. The combination of compressor and VFD allow the system to produce 21,000 Btu/hr (6149 Watts) assuming R404A refrigerant, a 20°F (-6.7°C) evaporating temperature and 100°F (37.8°C) ambient temperature. The ability of the compressor to run efficiently at low speed is a key contributor to conservation of battery power while the truck is on a delivery route.

4. PERFORMANCE SIMULATION

Prior to assembly of the truck performance simulations were run to verify the system would meet real world requirements. The refrigeration system capacity was modeled and compared to the predicted thermal load on the refrigerated truck.

4.1 Refrigeration System Performance Estimate

Refrigeration system capacity and efficiency was modeled over the known operating pressure range of a medium temperature, 36°F (2.2°C) refrigeration application using the below equations.

Capacity may be calculated as a function of mass flow through the evaporator and known inlet and outlet pressure and temperatures at evaporator outlet and exit. Additionally, losses from drive cooling must be accounted for, this loss is directly related to drive efficiency and was measured in laboratory testing to be from 5% to 15% of the total drive power. For the present analysis a conservative estimate of 15% was used. Analysis assumes all heat from drive inefficiency manifests itself in capacity reduction.

$$\dot{Q}_{EVAP} = (\dot{m}_{EVAP}) * (h_1 - h_4) \tag{1}$$

$$\dot{Q}_{EVAP}: Evaporating\ Capacity, W$$
$$\dot{m}_{EVAP}: Evaporator\ Mass\ Flow\ Rate, kg/s$$
$$h_1: Enthalpy\ Exiting\ Evaporator, J/kg$$
$$h_4: Enthalpy\ Entering\ Evaporator, J/kg$$

Mass flow may be calculated as a function of the compressor displacement, speed, refrigerant density at compressor inlet and volumetric efficiency. For the scroll compressor in this study representative test conditions were run and volumetric efficiency is measured to be 92% on average.

$$\dot{m}_{EVAP} = n_V * \omega * D * \rho \tag{2}$$

$$n_V = Volumetric\ Efficiency, \%$$
$$\omega = Rotational\ Speed\ Of\ Scroll, Hz$$
$$D = Displacement\ Of\ Scroll, m^3$$
$$\rho = Density, kg/m^3$$

Combining the above equations yields an equation for evaporator capacity as a function of known parameters.

$$\dot{Q}_{EVAP} = (n_V * \omega * D * \rho) * (h_1 - h_4) \tag{3}$$

Drive power required may be calculated by applying an assumed isentropic efficiency. For the scroll compressor in this study representative test conditions were run and actual isentropic efficiency is measured to be 65% on average.

$$P = \frac{\dot{Q}_{EVAP}}{(COP_{THEO}) * n_I} \tag{4}$$

$$COP_{THEO} = \frac{h_1 - h_4}{h_1 - h_{2s}} = Theoretical\ C.O.P., Dimensionless$$
$$h_{2s} = Enthalpy\ at\ compressor\ discharge\ J/kg$$
$$assuming\ constant\ entropy\ compression, J/kg$$
n_I=Isentropic efficiency, %

The final system capacity can now be written.

$$\dot{Q}_{EVAP} = (n_V * \omega * D * \rho) * (h_1 - h_4) - 0.15 * P \tag{5}$$

4.2 Thermal Model

Thermal loading on the refrigerated truck body can be modeled as an energy balance as shown below in Figure 5.

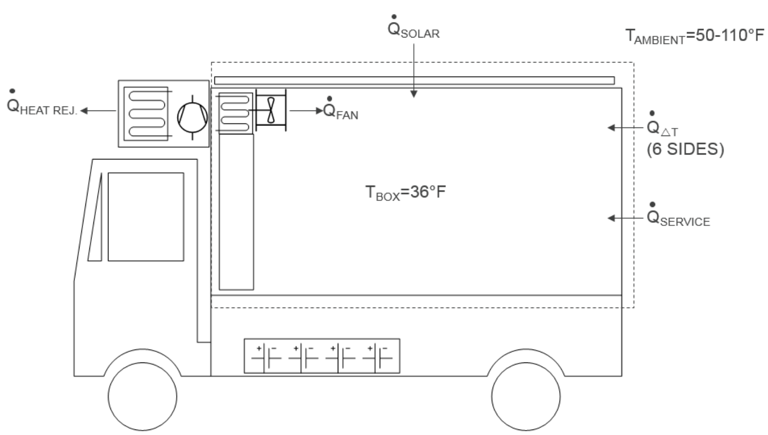

Figure 5. Truck Thermal Model

The total required cooling load during a route can be calculated using the energy balance and assumptions below. Note the service load is an empirically determined relationship developed by truck manufacturers to model the impact of door openings, convective heat transfer and solar loading.

$$\dot{Q}_{LOAD} = \dot{Q}_{FAN} + \dot{Q}_{SOLAR+}\dot{Q}_{\Delta T} + \dot{Q}_{SERVICE} \tag{6}$$

$$\dot{Q}_{FAN} = Fan\ Power, W$$
$$\dot{Q}_{SOLAR} = 1000, \frac{W}{m^2}$$
$$\dot{Q}_{\Delta T} = U * A * \Delta T, W$$
U=Heat Transfer Coefficient, W/m^2*K
A= Surface Area of Box, m^2
$$\Delta T = T_{AMBIENT} - T_{BOX}$$
$$\dot{Q}_{SERVICE} = Service\ Load\ (Emperical\ Equation), W$$

Table 1. Thermal Model Assumptions

Metric	Assumption	Source
Fan Power	100% of power goes to heating	Customer requirement
Solar Load	1000 W/m^2, Included in service load	Customer requirement
Wall Thickness	3” (7.6 cm) to 6” (15.2 cm)	Supplier
Service Factor	2	Supplier

Results of the system performance model are compared to the thermal model to verify adequate sizing of the refrigeration system. Modeling shows the refrigeration system capacity exceeds the expected thermal loading by approximately 1.5 kW at worst case loading.

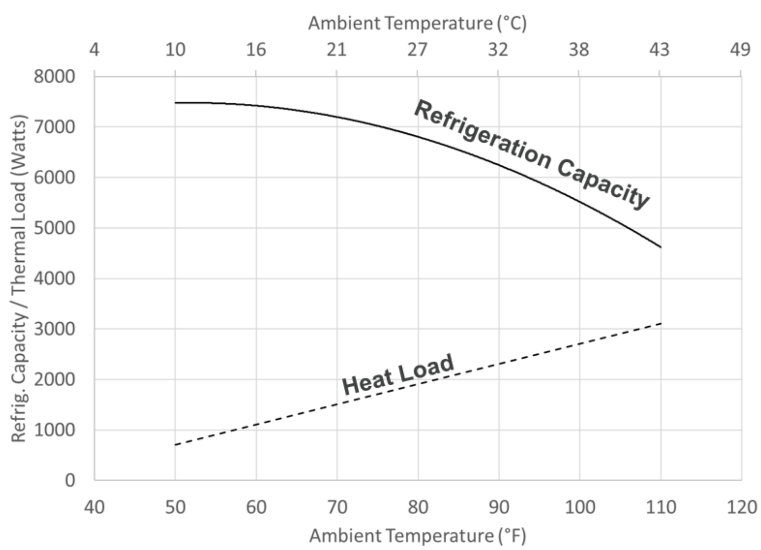

Figure 6. Simulation Results of System Capacity versus Thermal Load

4.3 Battery Life Simulation

Customers require a vehicle capable of completing a 10-hour route duration. During this 10-hour route the refrigeration system must have the ability to maintain the box temperature in the range of 34 to 38°F (1.1 to 3.3°C) while running entirely from energy stored in the on-board battery system with supplemental energy from the roof mounted solar panels. Bin analysis was performed with the following assumptions.

Table 2. Battery Life Assumptions

Metric	Assumption	Source
Route Duration	10 hours	Customer requirement
Ambient Temperature	70°F (21°C) to 110°F (43°C)	Customer requirement
Thermal Load	5,100 to 10,600 Btu/hr (1,493 to 3,104 W)	Section 4.2
Solar Energy Input	138 W/m^2 At Peak Sun	Supplier
Refrigeration Capacity	11,500 to 13,500 Btu/hr (3,513 to 4,100 W) Compressor: 3000 RPM	Section 4.1
Run Time	Ratio of thermal load to refrigeration capacity	Calculation
Power Draw From Refrigeration System	Sum of compressor, evaporator, condensing fans	Calculation
Battery Capacity	255 Amp*hr – Hourly amps draw.	Calculation

Results of the bin analysis are shown below in Figure 7 for a 10-hour route which is assumed to begin 2 hours before sunrise with an ambient temperature starting at 70°F (21°C) and ending at 110°F (43.3°C).

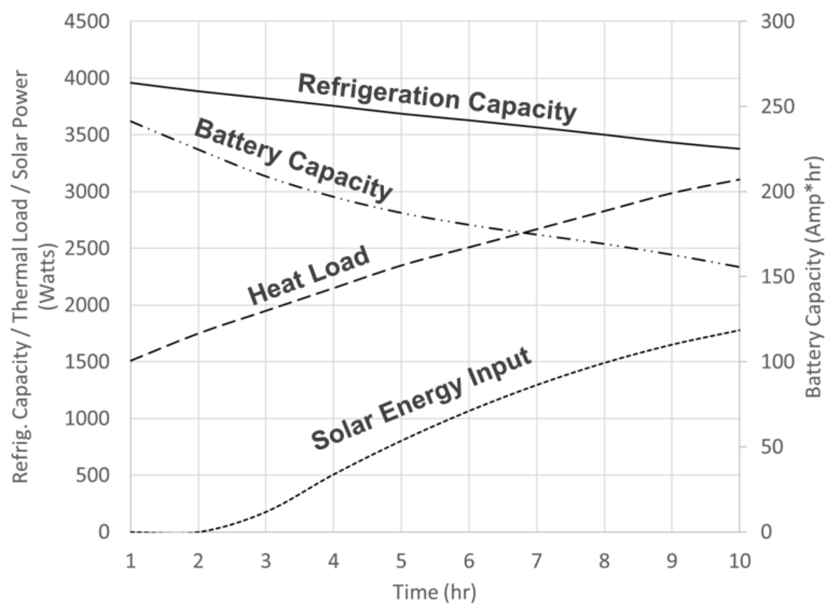

Figure 7. Battery Life Bin Analysis Results

Results of the simulation show an expected total energy consumption of 13.5 kW*hr over the course of a high ambient 10-hour delivery route. Since power is derived from a 48-volt battery pack the estimated Amp-hr requirement may be found by dividing the Watt*hr consumption by 48 VDC. While this analysis does not consider the fluctuations in voltage as batteries are charged and depleted, it can be used as a first pass estimate to size the battery pack. Using this analysis, with an assumed 255 Amp*hr battery pack shows the batteries depleted to roughly 156 Amp*hr or 61% at the end of the 10-hour route. This result suggests the battery pack is sized adequately to maintain box temperature. Running this same extreme ambient analysis with zero solar energy input, such as on a rainy day, suggests the batteries may be fully depleted. In this scenario the presence of the eutectic plates allows the system to continue to provide capacity and maintain box temperature until the end of the route at which time the truck is plugged into utility power and batteries can be re-charged.

5. FIELD RESULTS

Field testing began in April 2017 and is planned to last for 2 years. The field test is sponsored by the San Joaquin Valley Air Pollution Control District and includes power measurement and carbon footprint estimates of two trucks running similar delivery routes in Fresno, CA. The first truck contains the fully electric system described in this paper. In this truck total power measurement of the refrigeration system is measured and recorded when the vehicle is plugged in overnight to recharge batteries and eutectic plates. Power consumption is also measured during over the road operation using controllers for both the solar panels and refrigeration system. The second truck is outfitted with a traditional self-powered refrigeration system containing diesel engine. For this system the fuel consumption of the engine is measured directly. The cost to operate the vehicles is then calculated using the current cost of diesel fuel and electricity combined with maintenance costs. Results may be found below in Table 3 for a 3-month time in Summer 2017.

Table 3. Field Results of Electric vs. Self-Powered System

	Self-Powered	**All Electric**
Route Hour Operation	113.7	113.7
Total Fuel Consumption	152 gallons (575 liters)	0
Fuel Cost/unit	$3.05/gallon	$3.05/gallon
Total Fuel Cost	$464	$0
Dock Hours Operation	401	401
Total Electric Usage	6,015 kW	2,667 kW
Electric Cost	$0.14	$0.14
Total Electric Cost	$842	$373
Maintenance Cost per Hour	$1.00	$0.10
Total Cost Over 3 Months	**$1.420**	**$385**

In addition to power consumption and operating cost the environmental impact is quantified by calculating the carbon footprint savings associated with eliminating all fuel usage of the refrigeration system. EPA (2005) showed 1 gallon (3.8 liters) of diesel fuel to contain approximately 22 pounds (10 kg) of carbon dioxide content on average. Multiplying the total fuel consumed by the self-powered unit by the above-mentioned carbon dioxide content in diesel yields a savings of roughly 2500 lb. (1134 kg) of carbon dioxide over the 3-month period.

To gauge the benefit of the electric hybrid eutectic system versus a traditional eutectic, the truck was operated as a eutectic only system for a select number of days. That is the refrigeration system was operated only when the truck was plugged in to utility power and the box temperature was maintained during the route by only pulling air over the frozen eutectic plates. This experiment provided two relevant results. First, improved temperature control and route duration was documented on the hybrid truck. Figure 8 below shows recorded box temperatures on similar temperature days with high temperature of 102°F (39°C) and low temperature of 72°F (22°C). Note the hybrid truck can maintain box temperature to the desired setpoint range for the full 10-hour route whereas the eutectic only system is unable to pull the box back to setpoint after approximately 3.5 hours due to repeated door openings.

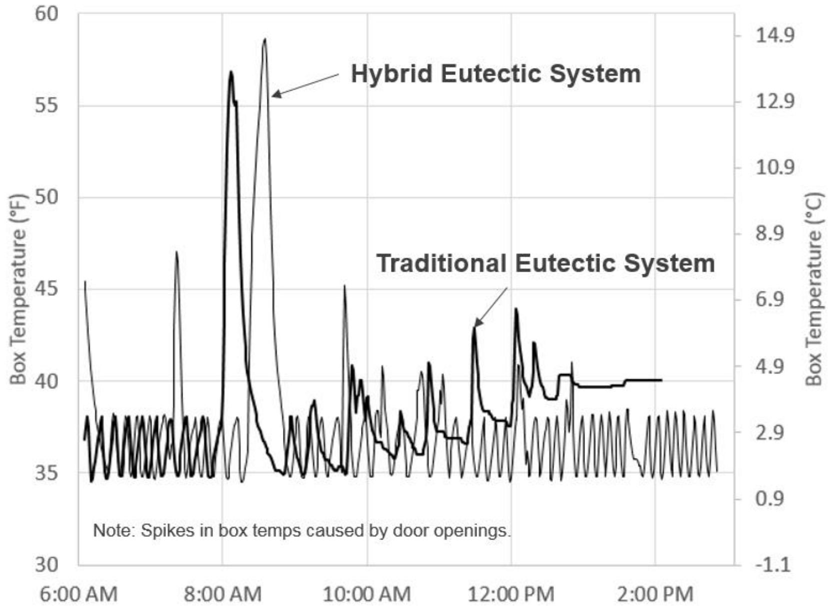

Figure 8. Box Temperature Maintenance of Hybrid vs. Eutectic Only System

The second result is a measurable reduction in box pulldown time after a door opening. Figure 10 below compares pulldown time to set point after door closing following a 15-minute door opening at an ambient temperature of 80°F (27°C). A 12-minute reduction in the time needed to pull the box temperature back to setpoint is recorded.

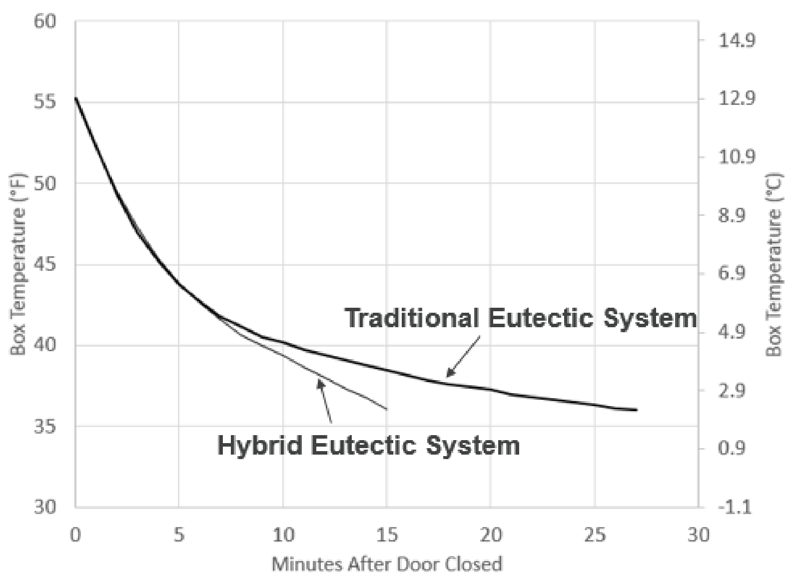

Figure 9. Pulldown Time Comparison After Door Opening

6. CONCLUSIONS

An all-electric refrigeration solution for urban delivery routes has been developed and presented. The system utilizes a 48-volt battery pack and solar panels to power a refrigeration system combining eutectic plate refrigeration with a direct expansion evaporator. Models of refrigeration system performance and thermal loading are presented and confirm the viability of the solution. Finally, field results are presented which confirm the viability of the system and quantify cost savings of $1,038 per truck with a carbon footprint reduction 2,500 lb. (1134 kg) of carbon dioxide over a 3-month period versus a self-powered diesel system.

NOMENCLATURE

DC	Direct Current	(volts)
AC	Alternating Current	(volts)
RPM	Revolutions per Minute	(RPM)
COP	Coefficient of Performance	(dimensionless)

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