International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2010

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ABSTRACT

Physics-based dynamic model of a transcritical CO2 supermarket refrigeration system providing cooling capacity for medium- and low-temperature food cabinets, racks, and cold rooms has been developed using equation-based software Dymola. The main motivation for developing such a model is to develop control algorithms and verify their performance and robustness in the absence of physical prototypes. In this paper, the effects of 1) one- and two-dimensional gas cooler models, 2) discretization of the gas cooler outlet line, and 3) gas cooler natural convection are investigated using the developed system model. It has been observed that two-dimensional gas cooler model gives more accurate system performance prediction when the gas cooler fans were running at moderate speeds compared to one dimensional model. It has also been noted that by increasing the pipe discretization, the modeled transport delay time was increased, and finally that the natural convection caused by the gas cooler frame height has a very large impact on the system performance when the gas cooler fans are shut down.

1. INTRODUCTION

Since the synthetic refrigerants, such as CFC, HCFC and HFC, can cause global warming and deplete the ozone, natural refrigerants, which are friendly to the environment, are attracting more and more research interest and being considered as the long term replacement for synthetic refrigerants in the past two decades. Kim *et al.* (2004) pointed out that CO2 was the only non-flammable and non-toxic fluid that could also operate in a vapor compression cycle below $0\,^{\circ}C$ among natural refrigerants, i.e. fluids like water, air, noble gases, hydrocarbons, ammonia and CO2. Since supermarket refrigeration systems have a significant amount of piping and a number of pipe joints, the refrigerant leakage is often considerable and can vary from +30% for the older and around 15% or somewhat lower for the new systems (Baxter, 2003). Baxter also pointed out (2003) that the large refrigerant charge and high loss rate for the multiplex direct expansion refrigeration systems resulted in high values of TEWI (total equivalent warming impact) with direct refrigerant loss impact accounting for about half of the total. Therefore, it would be very important to develop and optimize the supermarket system employing the use of natural refrigerants, such as CO2 (Ge and Tassou, 2009).

There has been a significant amount of work done to investigate the vapor compression refrigeration cycle and components performance in supermarket application system, automobile air conditioning system, airliner cooling system and heat pump water heater system with CO2 as refrigerant, e.g., Ge and Tassou (2009), Sawalha (2008a and 2008b), Karim and Tummescheit (2008), Sienel and Finckh (2007), Kim *et al.* (2004), Pfafferott and Schmitz (2004), Girotto *et al.* (2004), Boewe *et al.* (1999) and Lorentzen (1993). It was observed that the measured annual energy

consumption of MT (medium temperature) CO2 system was about 10% lower in central and northern Europe compared with properly installed R404A system (Sienel and Finckh, 2007).

Due to different configuration, size and location of supermarket refrigeration systems, it is very difficult and expensive to build a general laboratory to test the whole system performance, robustness, control, and reliability before launching the refrigeration system to the market. Computer simulation is one of the valuable means to make the design process of refrigeration systems more efficient and the product performance better if the developed simulation model could provide stable, rapid and accurate results (Ding, 2007). Ge and Tassou (2009) developed an integrated system model for a medium temperature retail food refrigeration system with detailed condenser/gas cooler model, a simplified compressor model, an isenthalpic expansion process and constant evaporating temperature and superheating. Based on this model, the authors optimized the controlled discharge pressure for two transition temperatures of $16\,°C$ and $21\,°C$. With the developed high pressure control strategy in seasonal simulations, it was demonstrated that the system could save approximately 18% energy. Sawalha (2008a) developed two kinds of simplified steady state system models to simulate the performance of parallel and centralized supermarket refrigeration system for low and medium temperature levels with CO2 transcritical cycle in EES (Engineering Equations Solver). He found that the two-stage centralized system solution gave the highest COP for the ambient temperature range 10-40 $°C$ compared with the one-stage centralized system solution and parallel system solution. Sawalha (2008b) also compared the simulated annual energy consumption for centralized trans-critical CO2 system, NH3-CO2 cascade system and conventional R404A DX system and found that centralized trans-critical CO2 system is good solution for cold climates, which would be good alternatives to R404A DX system for supermarket refrigeration.

In order to have a good understanding of CO2 supermarket system behaviors under different boundary conditions and understand the control robustness, a physics-based dynamic model was developed using equation-based software Dymola™ and Air Conditioning Library™ for a "CO2 Booster supermarket system", which was thought as a cost-effective, reliable and promising approach to integrate the CO2 MT and LT systems for supermarket refrigeration system by Sienel and Finckh (2007). In Section 2, the architecture of the modeled CO2 Booster refrigeration system and description of the model physics captured are presented. Section 3 illustrates some of the system level model validation results. In Section 4, the effect of one- and two- dimensional gas cooler models, the effect of discretization number for the liquid pipe between gas cooler and receiver, and the effect of natural convection on the gas cooler when the fan was shut down were investigated using validated system model. Finally, conclusions are given in Section 5.

2. CO2 BOOSTER SYSTEM AND MODEL DESCRIPTION

Figure 1 illustrates the CO2 Booster supermarket system diagram presented in Sienel and Finckh (2007). Generally, the main components of the system include MT compressors, gas cooler, HP valve, receiver, MP valve, MT cabinets/cold rooms (including MT EXV and evaporator), LT cabinets/cold rooms (including LT EXV and evaporator) and LT compressors. The components and system model were developed in Dymola™ using the Air Conditioning Library™ developed by Modelon using Modelica language, an object-oriented equation-based modeling language used to model large, complex and heterogeneous physical systems (Pfafferott and Schmitz, 2004). Model library assumptions include: one-dimensional, one- and two-phase refrigerant flow are considered, and bi-directional flows are supported. Finite volume method is used to discretize the refrigerant and air flow passages by applying mass, momentum and energy balance equations used in refrigerant pipes and heat exchanger models. The general mass and energy balance equation forms for the *i*th finite volume are listed below:

$$\frac{dM_i}{dt} = \dot{m}_{i,in} - \dot{m}_{i,out} \tag{1}$$

$$\frac{dU_i}{dt} = \dot{m}_{i,in} \times h_{i,in} - \dot{m}_{i,out} \times h_{i,out} + \dot{Q}_i + W_i \tag{2}$$

Zhang *et al.* (2009) pointed out that implementation of either dynamic or steady-state momentum equations gives similar results both at the component and system level. Therefore, the steady state momentum equation is used in the developed component models in the following form:

$$0 = p_i - p_{i+1} + \Delta p_{acc,i} - \Delta p_{fri,i} - \Delta p_{gra,i} \qquad (3)$$

The equation used for calculating the friction loss term, Δp_{fri}, is the common quadratic equations related with mass flow rate, but with density profile modification from the ThermoFluid library inserted in Dymola and Air Conditioning library. The state variables of $\{M, U\}$ were usually transformed into a form with $\{p, h\}$ as state variables in our calculation.

In the following subsections, the component models developed within this work and based on the above general equations, such as heat exchanger (gas cooler and evaporator) and compressor models are discussed. The other component models, directly used from the Air Conditioning Library, are not discussed in this paper.
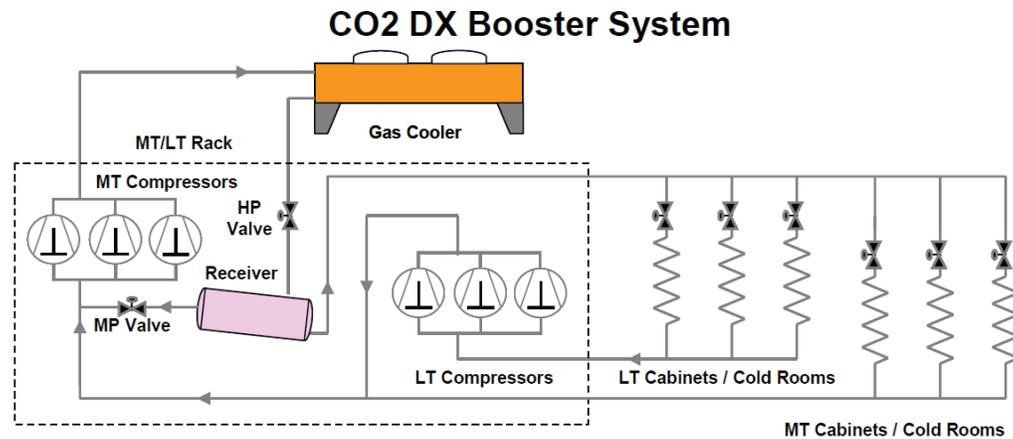

Figure 1: CO2 Booster supermarket system configuration

2.1 Gas Cooler/ Evaporator Model

Air Conditioning library has a 3-dimensional heat exchanger model. However, in order to speed up the system simulation while maintaining required accuracy required for controls design, one- and two-dimensional models for heat exchangers with round tube and plate fin have been developed. Both, the one- and two-dimensional models assume equal refrigerant flow circuit length and equal refrigerant distribution among different circuits. Therefore, only one refrigerant circuit is needed for discretization and calculation. The tube wall model applies the same discretization methodology as the one used in refrigerant circuit. On the air side, counter or parallel flow are assumed by setting equal air flow passage area for every refrigerant circuit in one-dimensional model. However, in the two-dimensional model, cross flow is assumed between the air and the refrigerant. It is also assumed that there is no air flow exchange between the adjacent air flow passages as indicated in Figure 2 because the fans are physically separated. By connecting corresponded finite volumes between refrigerant flow cell and air flow cell matrixes and ignoring the axial thermal conduction within the tube wall, the heat transfer rate matrix could be obtained for each control volume.
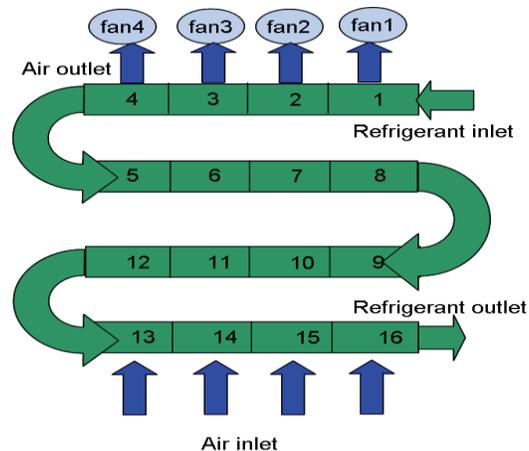

Figure 2: Two dimensional heat exchanger model diagram

2.2 Compressor/Valve Models

The compressor model was developed using a manufacturer provided map that correlates the volume flow rate and power with pressure ratio, speed and inlet conditions. The valve characteristic curves are also based on the manufacturer data, but the mass flow rate equation is based on the generic model from Air Conditioning library.

3. MODEL VALIDATION

The dynamic system model in Dymola is shown in Figure 3. In the model calibration/validation procedure, the following variables were used as model inputs: compressor speed, valve position, gas cooler fans status, cabinet side air inlet and outlet temperature and flow rate, and outside ambient temperature. The system model was validated using field data from one discounter store on June 09, 2009 in Germany by comparing model outputs with actual field data for the variables of interest in control design. A subset of the validation results are shown in Figure 4. It can be seen in Figure 4 that the general trend and steady state value for high pressure (HP) and gas cooler outlet temperature predicted by the system model correspond well with the field test data.
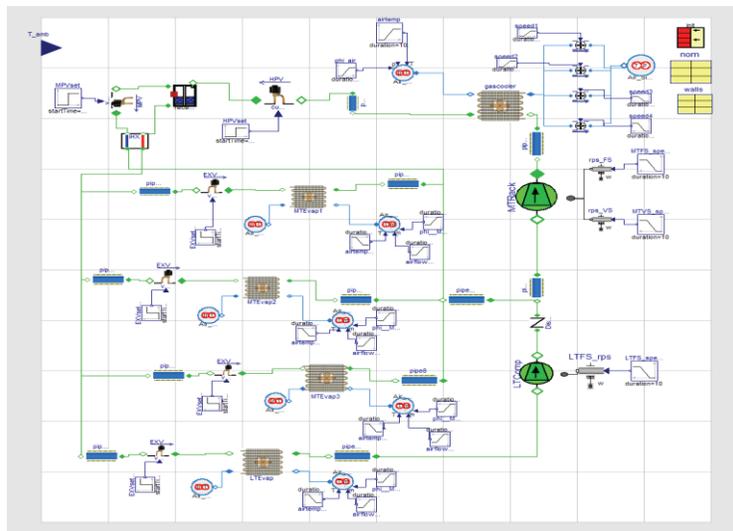

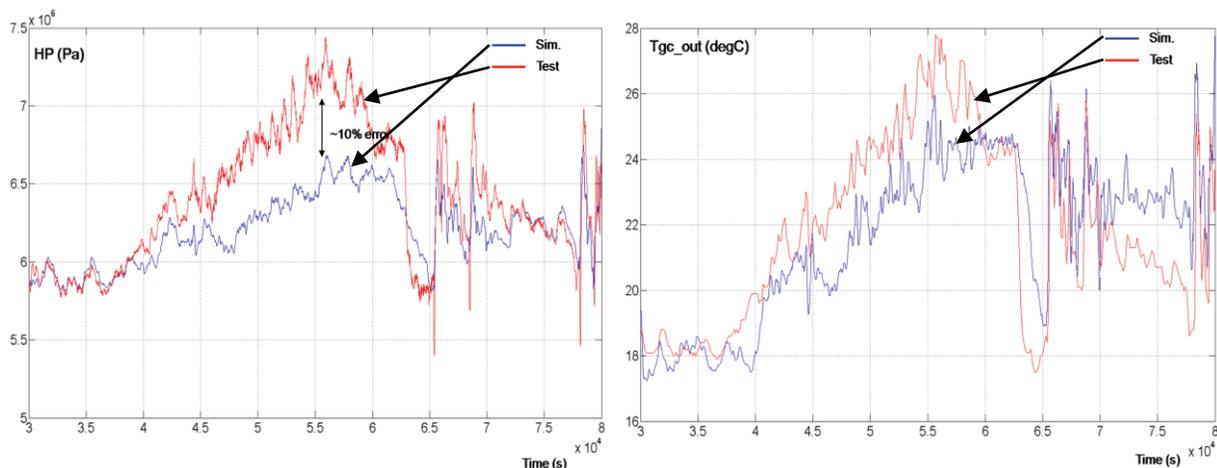Figure 3: CO2 discounter booster system model diagram

Figure 4: CO2 discounter booster system model validation

4. RESULTS AND ANALYSIS

4.1 1-D and 2-D Gas Cooler Model Comparison

In winter conditions and lower ambient temperatures, some or all gas cooler fans are required to run at lower than maximum speeds to maintain head pressure. In order to design a good and robust control system using system level models, gas cooler model accuracy is very important. System simulation results with 1-D and 2-D gas cooler models are shown in Figure 5. It can be seen that when all the fans are running at full speed or shut down, the difference between the two different models is very small. That means the 1-D model could be used in the simulation if the air flow through the coil didn't vary significantly over different part of the coil. However, when some of the fans were shut down, the difference would increase with the number of fans shut-down. Obviously, using a two-dimensional heat exchanger model is better as it is a more accurate representation of the actual heat transfer process in a round-tube-plate-fin heat exchanger. When some of the fans are shut down, in 2-D model both the air flow rate and the heat transfer area can be reduced as indicated in Figure 2, whereas, in one-dimensional model, only the air flow rate can be reduced.
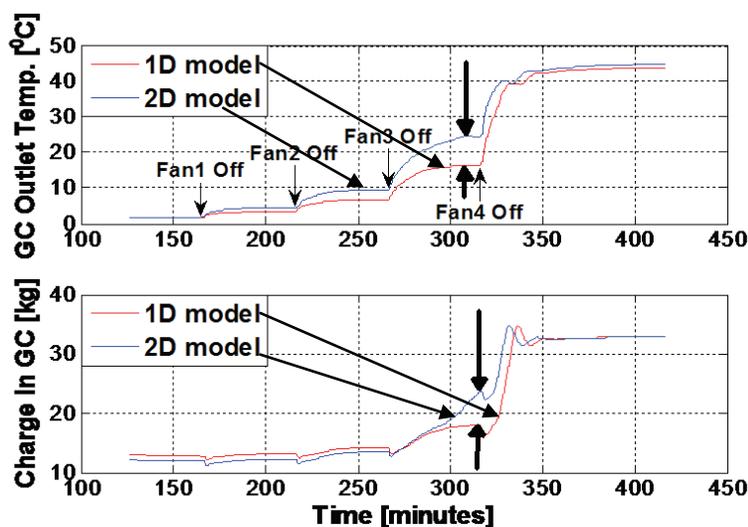

Figure 5: 1-D and 2-D gas cooler model comparison results

4.2 Effect of Discretization Number for Liquid Pipe

Accurate prediction of transport delays is another very important parameter for control design. Sensor location decisions are often made considering transport delay as it has significant effect on overall system controllability and control system stability and performance. In field installations, the gas cooler is usually placed outside of the supermarket store. However, the compressor rack may be put in the store's mechanical room. Given different sizes and configurations of supermarket stores, the pipe length between the gas cooler and the compressor rack may vary significantly. Considering that refrigerant speed in the liquid pipe can be very low, the transport delay in the liquid pipe is even more important when the pipe length is very long.

As it can be seen from Figure 6, the discretization number has a significant impact on the liquid pipe outlet temperature transient profile when the inlet temperature suddenly changes. With the constant inlet mass flow rate assumption, refrigerant velocity also changes by ~2.5% as a result of inlet temperature change as shown in Figure 6. The difference in outlet temperature is mainly caused by the increase of transport lag with increased discretization number. However, in the system level simulation of a step change in outside ambient temperature (OAT) illustrated in Figure 7, the normalized value of high-side pressure and gas cooler outlet temperature are a lot less sensitive to discretization number. Such ambient temperature change does not significantly affect the liquid pipe inlet temperature due to system thermal inertia.
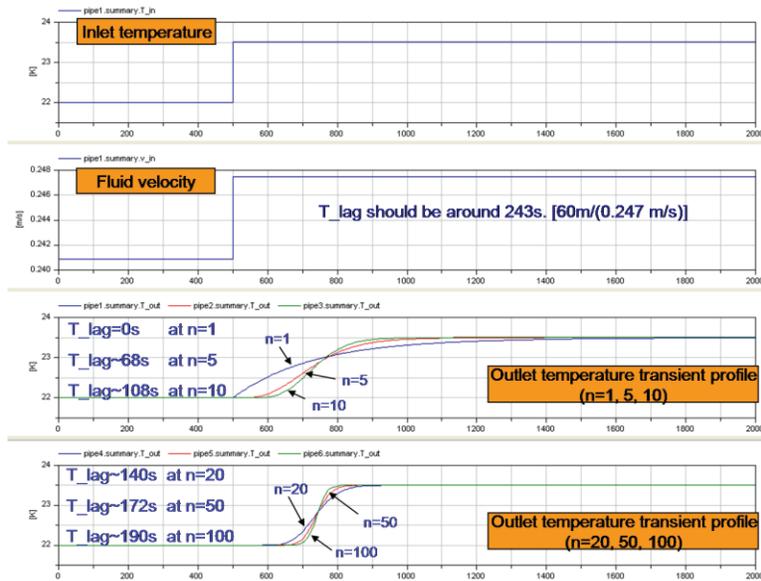
Figure 6: Component level results with different discretization number for liquid pipe
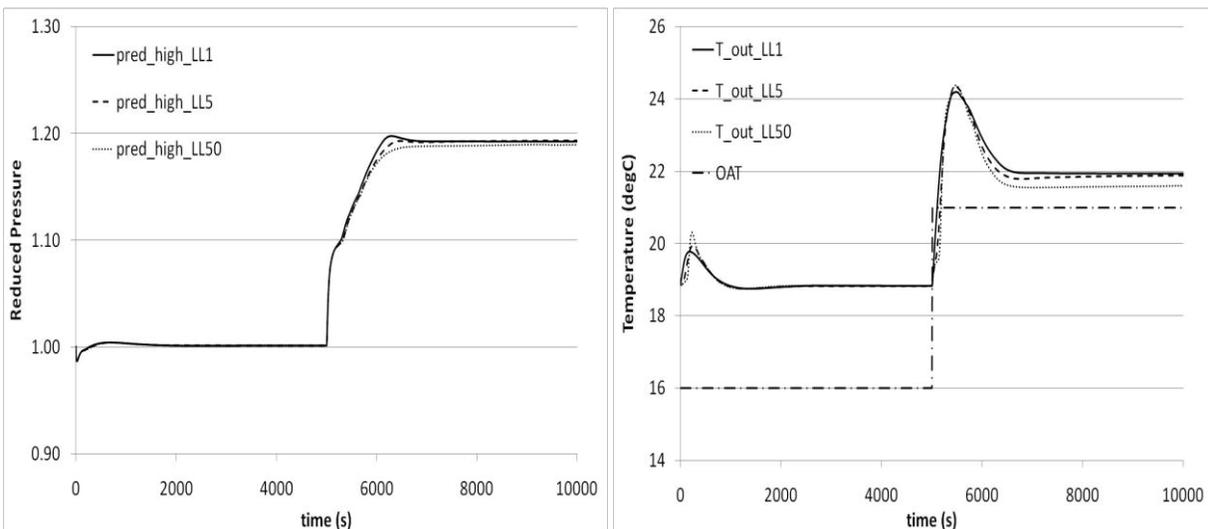

Figure 7: System level results with different discretization number for liquid pipe

4.3 Natural Convection Impact on Gas Cooler Model

In order to design good pressure control and predict the performance of the system when the gas cooler fans are shut down and ambient temperature is low, the effect of the natural convection on the gas cooler heat exchange needs to be taken into account. As a result, the density-driven buoyancy effect is included in the system model.

For each fan compartment :

$$
\begin{aligned}
\Delta p_{buoyancy} &= \Delta p_{coil} + \Delta p_{frame} \\
&= \rho_{amb} \cdot g \cdot H_{total} - \int_0^{H_{total}} \rho_{air} \cdot g \cdot dH \\
&= \rho_{amb} \cdot g \cdot H_{total} - \int_0^{H_{coil}} \rho_{air} \cdot g \cdot dH - \int_{H_{coil}}^{H_{total}} \rho_{air} \cdot g \cdot dH \\
&= \rho_{amb} \cdot g \cdot H_{total} - \left(\frac{\rho_{amb}}{2} + \rho_{out,Node1} + \rho_{out,Node2} + \rho_{out,Node3} + \frac{\rho_{out,coil}}{2} \right) \cdot g \cdot \frac{H_{coil}}{4} - \frac{\rho_{out,coil} + \rho_{out,fan}}{2} \cdot g \cdot \left(H_{total} - H_{coil} \right)
\end{aligned}
\tag{4}
$$

where :

$\rho_{out,coil} = \rho_{out,fan}$, if frame walls are assumed to be adiabatic.

Based on the gas cooler and coil frame diagram as shown in Figure 8, the density-driven pressure difference caused by buoyancy was obtained using equation (4).
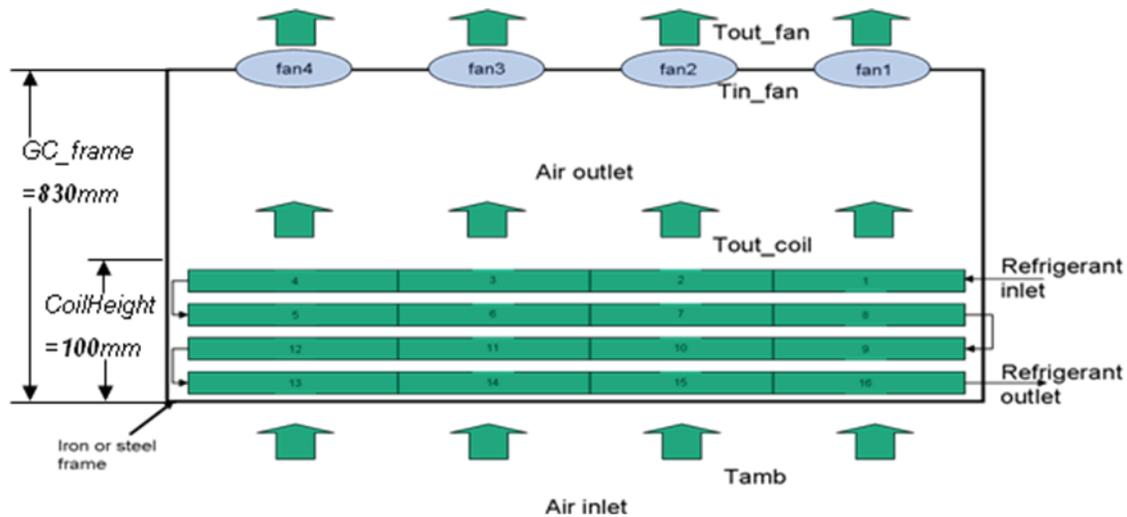

Figure 8: Gas cooler and coil frame diagram

Figure 9 shows the system simulation results with and without natural convection when fan 4 was off and the other 3 fans were on. It can be seen that the implementation of the natural convection effect results in ~3% of normalized high pressure difference, and $\sim 0.8\,^{\circ}C$ gas cooler outlet temperature difference with natural convection being about 4% of the total air flow rate for one fan. Therefore, the natural convection term is very important to include in the system model as it has a significant effect when the fan(s) are shut down.
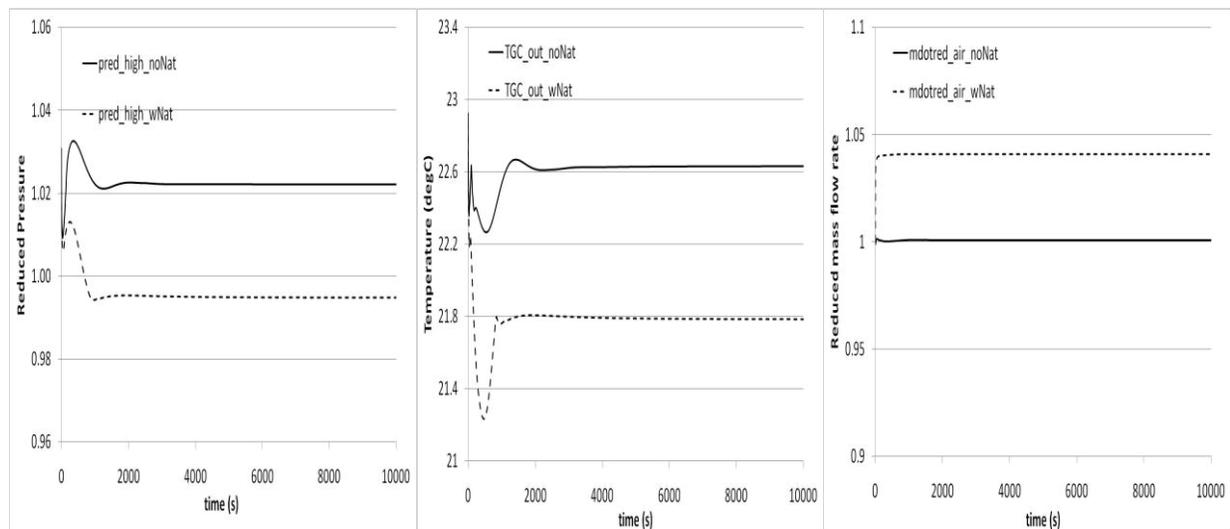

Figure 9: comparison of system results with and without natural convection included

5. CONCLUSIONS

A physics-based dynamic model of a CO2 booster supermarket refrigeration system was built to develop and verify performance and robustness of system controls in the absence of physical prototypes. This model was successfully validated using field data. Because of the particular importance for verifying control performance and robustness at low ambient temperatures, the effects of 1) one- and two-dimensional gas cooler models, 2) discretization number for the liquid pipe, and 3) natural convection on the gas cooler were investigated. The results showed that there

were large differences between the 1-D and 2-D gas cooler models on the system performance when the gas cooler fans were running at moderate speed. By increasing the pipe discretization, the modeled transport delay time was increased in the component level test. However, no significant change was observed in the system level simulations. Finally, the simulation results clearly showed that the natural convection caused by the gas cooler frame height has a very large impact on the system performance when the gas cooler fan(s) are shut down.

NOMENCLATURE

			Subscripts	
g	gravity acceleration	(m/s)		
H	height	(m)	acc	acceleration
h	specific enthalpy	(J/kg)	amb	ambient
M	Mass	(kg)	fri	friction
$\dot{m}$	mass flow rate	(kg/s)	gra	gravity
p	pressure	(Pa)	in	inlet
$\dot{Q}$	heat flow rate	(W)	out	outlet
t	time	(s)		
U	Internal energy	(J)		
W	work	(W)		
ρ	density	(kg/m^3)		

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ACKNOWLEDGEMENT