

DEVELOPMENT OF ENERGY EFFICIENT TRANSPORTATION REFRIGERATION SYSTEM WITH HEAT STORAGE

A. Ardiyansyah¹, Kwang-II Choi¹, Jong-Taek Oh^{2*}, Hoo-Kyu Oh³

¹ Graduate School, Chonnam National University, San 96-1, Dunduk-Dong, Yeosu, Chonnam 550-749, Korea

²Professor, Department of Refrigeration and Air Conditioning Engineering, Chonnam National University, San 96-1, Dunduk-Dong, Yeosu, Chonnam 550-749, S. Korea

³Professor, Department of Refrigeration and Air Conditioning Engineering, Pukyung National University, Yongdang-Dong, Nam-Ku, Busan 608-749, S. Korea

Abstract: This paper presents the development of an energy efficient refrigeration system for transportation purposes. The system stores the heat wasted during refrigeration cycles and utilizes it for hot gas defrost process. A computer model is developed to design an evaporator and a condenser and to determine heat storage size based on pre-determined cooling condition. Simulations with the selected components were carried out to investigate refrigeration system performance under various operating conditions during refrigeration and defrost cycles. An experimental investigation on cooling capacity tests is performed on a 3,225 x 1,740 x 1,640 mm full-scale refrigerated box to validate the model. The evaporator coil defrost process analysis provides quantitative information on the defrost energy efficiency. The simulation values show a good agreement with the experimental results with the deviation for cooling capacity and defrost time less than 7.5% and 13.8% respectively. These results are important in order to avoid excessive parasitic load and temperature rise inside the refrigerated container.

Key Words: *Refrigeration system, performance, simulation, experiment, reverse cycle, hot gas, defrost*

1 INTRODUCTION

Transportation of sliced-raw fish from the processing plant to the market and consumer forms an important parts of the product's cold chain which is mainly distributed through road vehicle equipped with refrigeration system. Worldwide, more than 1 million units of refrigerated road vehicles are in operation for various products transportation (IIR, 2006). The performance and efficiency of such refrigerated vehicle is essential to maintain good quality of the transported products, save energy and minimize environmental impacts.

Many studies have been carried out in order to increase the performance and energy efficiency of the system. One of the considerations is the frost formation which reduces the refrigeration capacity and the performance of the system. To overcome these problems, the defrosting process should be performed periodically. One of the methods is the reverse cycle-hot gas defrost which is used widely in industrial refrigeration applications (Hoffenbecker et al. 2005).

In the present study, a refrigerated vehicle system for sliced-raw fish transportation has been developed. During the refrigeration cycle operation, the equipment acts as a pre-condenser and stores a portion of refrigerant heat rejection as depicted in Figure 1. During the defrost cycle, the system uses superheat of refrigerant gas discharged from the compressor, to defrost the evaporator coils.

Various computer or numerical models have been developed based on physical equations, empirical correlations using the curve fitting method, or combination of both. This model uses physical equations combined with the curve fitting empirical correlations to develop a model with simple and minimum input requirements. Based on the system operating conditions, refrigeration system models can be divided into steady-state (Hoffenbecker et al. 2005), (Jolly et al. 2000) and (Ardiyansyah et al. 2007) and transient models. The steady-state model is used because in the transportation refrigeration systems, the equipment is mainly intended to keep the refrigerated room in steady-state conditions within the set temperature range. Also, the cargo being carried is already under low temperature conditions when it is loaded (Jolly et al. 2000).

The objective of this study is to develop a computer model which can be used to analyze the effects of various operating variables on the performance of refrigeration system for sliced-raw fish transportation, equipped with heat storage during the steady-state refrigeration cycles. Further experimental investigation on cooling capacity tests is also performed on a full-scale refrigerated box to validate the model.

2 REFRIGERATION SYSTEM

2.1 Model Development and Simulation

Figure 1 shows the schematic of a refrigeration system equipped with heat storage. The heat storage equipment uses water or brine which stores the heat. During the defrost process, the system directs hot refrigerant from compressor through the heat storage (bypassing the condenser) to the evaporator for defrosting the evaporator coils. The refrigerant ejects its heat and condenses into liquid. It then flows back into the heat storage equipment. The refrigerant is re-evaporated by the heat stored in the equipment before entering the compressor and circulates again.

A steady state energy balance equation over the entire system is described in equation (1).

$$P + q_e = q_c + q_{th} \quad (1)$$

The compressor is modeled based on published manufacturers' specification data and empirical correlations from previous study (Ardiyansyah et al. 2007). The direct-expansion evaporator is modeled based on the single zone heat transfer process with the two-phase refrigerant heat transfer process, between the refrigerant inside the evaporator coils and the air flow. It is derived from the e-NTU relation for two-phase heat exchanger (Incropera and DeWitt 1990).

A quasi-steady state approach is applied to the heat transfer process in the heat storage area. For a determined period of time, the heat storage temperature increases by (Boeis et al. 2005):

$$\Delta T_w = (T_2 - T_w) \frac{\dot{m}.c_p}{M.c_v} \left(1 - \exp \left[- \frac{UA}{\dot{m}.c_p} \right] \right) \Delta t \quad (2)$$

The air-cooled condenser used in this study is modeled based on two-zone analysis, namely supercooling and condensation zones. The refrigerant leaving the condenser is assumed to be in a saturated liquid phase. The heat transfer within the supercooling is obtained from equation (3-4) where ε_c is the heat transfer effectiveness.

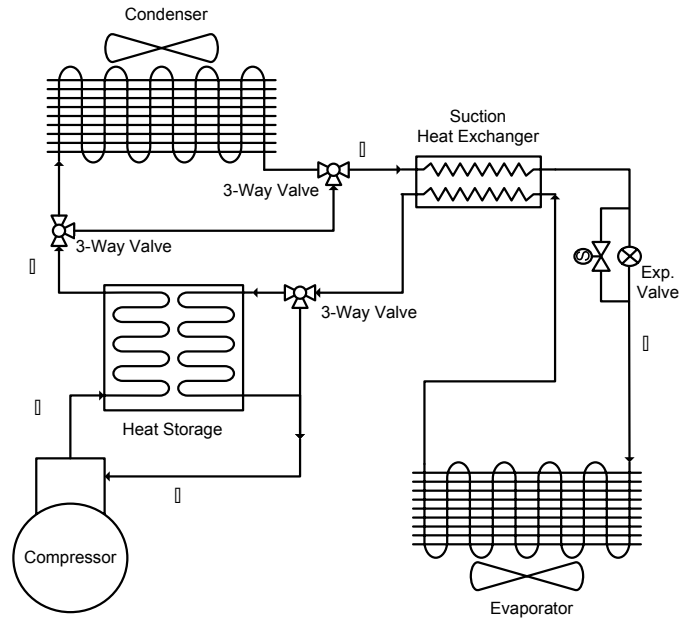


Figure 1: Schematic of a refrigeration system equipped with heat storage

$$q_{c,sp} = \dot{m}_r c_r (T_3 - T_{b,in}) \cdot \varepsilon_c \quad (3)$$

$$\varepsilon_c = 1 - \exp \left(- \frac{\dot{m}_a \cdot c_a}{\dot{m}_r \cdot c_r} \left(1 - \exp \left(\frac{U_{c,sp} A_{c,sp}}{\dot{m}_a \cdot c_a} \right) \right) \right) \quad (4)$$

The two-phase heat transfer for the condensation zone is given in (5).

$$q_{c,tp} = \dot{m}_b c_b (T_c - T_{b2}) \left[1 - \exp \left(- \frac{U_{c,tp} A_{c,tp}}{\dot{m}_b \cdot c_b} \right) \right] \quad (5)$$

The corresponding overall heat transfer coefficient for each zone is obtained from heat exchanger equation similar to equation (3) for the condenser. The refrigerant R-404A properties used in the analysis are obtained from data generated by REFPROP 8.0 software (Lemmon et al. 2007). The air side heat transfer coefficients for both evaporator and condenser are calculated using the correlation from (Wang et al. 2000). The Gungor-Winterton correlation (Gungor and Winterton 1997) is used for the two-phase heat transfer coefficient.

The simulation algorithm of the refrigeration system is depicted in Figure 2. The analysis is reduced into the solution of simultaneous equations at the balance point of the system and solved using Newton-Raphson approach. At each determined time step, the program solves the equations and generates operating variable values for the corresponding condition.

Then, the increase in the heat storage temperature as given by equation (2) is used to update the water temperature inside the heat storage area. The water temperature affects the operating parameters at the balance point; therefore, the program is repeated again to find the new balance point. The program runs until the determined operation time is reached.

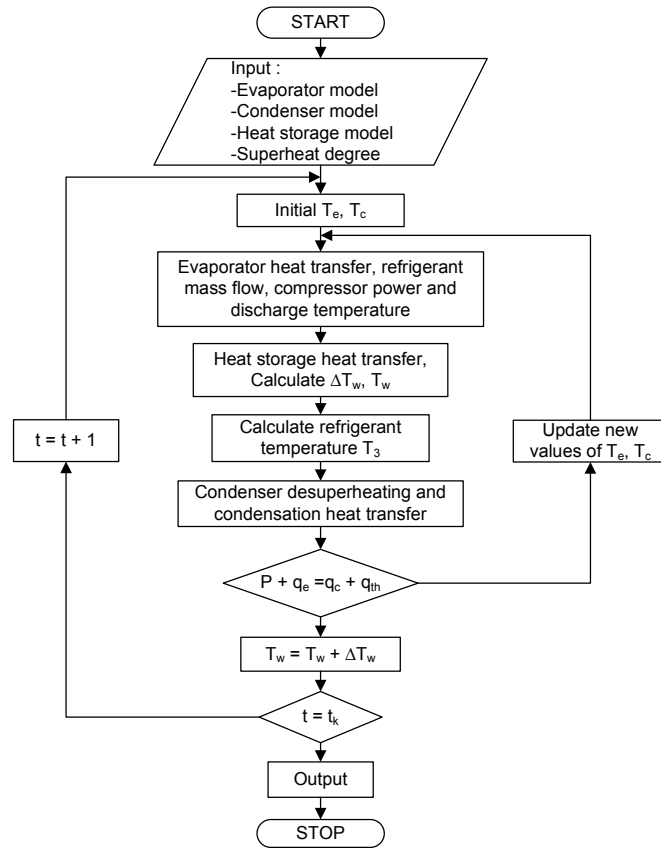


Figure 2: Flowchart of simulation algorithm

The program results include evaporating and condensing temperatures, refrigeration capacity, compressor power consumption, refrigerant mass flow rate, condenser heat ejection, system coefficient of performance, heat storage temperature and the heat energy available in the heat storage equipment.

2.2 Evaporator Defrosting Analysis

During the defrost process, the evaporator fan is shut-down hence the heat and mass transfer process between frost and air inside the refrigerated room is in a natural convection mode. In this study, the following simplifications are considered: the refrigerant phases are in thermodynamic equilibrium, the frost mass per heat exchanger unit area and defrosting rate are uniform in all evaporator sections, and the heat conduction in the axial direction of the pipe is negligible.

During the defrost process, part of the heat is stored in the coil, frosts and surface water and the remainder is transferred from the refrigerant to the coil, frost, surface water and the ambient air. Krakow (Krakow et al. 1992) divided the process into preheating, melting, vaporizing and dry heating stages. The energy equation for each heat exchanger element is described as:

$$Q_r = Q_s + Q_c + Q_f + Q_v \quad (6)$$

Each energy components are described in equations (9)-(12). Q_s is the stored heat energy in tube and fin, frost or surface water, which depends on the corresponding stages during the process. Q_c is the convective heat released into the ambient air and is a function of air film conductance h_a , surface area and temperature of the surface and ambient air. Q_f is the heat

energy conducted from the coils to the frost, where h_{aw} and T_f are air/water film conductance and frost temperature respectively. The heat from condensing refrigerant is also used to vaporize surface water that resulted from melted frost, described as Q_v and depends on the rate of mass vaporization (R_v) and the enthalpy of vaporization of water (h_{fg}).

$$Q_s = C \frac{dT}{dt} \quad (7)$$

$$Q_c = h_a A_{eff} (T_{cw} - T_a) \quad (8)$$

$$Q_f = h_{cw} A_{eff} (T_{cw} - T_f) \quad (9)$$

$$Q_v = R_v h_{fg} \quad (10)$$

The rate of vaporization is assumed to be proportional to the difference between water vapor density at the wall temperature and water vapor density at ambient temperature, effective wetted area of the coil and surface water vaporization coefficient (Krakow et al. 1992). The value of air film conductance, air/water film conductance and rate of vaporization obtained experimentally by Krakow are used in this study.

The simulation is conducted by dividing the process into small time steps and applying the energy equations for each time step. The preheating stage is analyzed from the beginning of the defrost process and terminated when the coil wall temperature T_{cw} approaches 0°C. The melting stage ends and the vaporizing stage begin when the frost mass, m_f reaches 0. The dry heating stage is not considered, the defrost process ends when the surface water that resulted from the melted frost is completely vaporized.

3 EXPERIMENTAL INVESTIGATION

A prototype refrigeration system is developed and installed. The cooling capacity tests are performed on a 3,225 x 1,740 x 1,640 mm full-scale refrigerated box. The operation, including electrical valves, time and box temperature set, is controlled by microprocessor-based controller designed specifically for the system. The temperature sensor setting, position and measurement during heat leak and cooling capacity tests are based on the calibrated-box method of transport refrigeration unit described in ARI 1110-2006 (ARI 2006). Along with temperature sensor at the compressor, condenser, evaporator and heat storage, the temperature measurement are recorded simultaneously by Darwin DA 100 data acquisition unit. The air flow from evaporator and condenser and box humidity are measured using Solomat MPM 100 environmental meter.

The system specifications and test conditions are described in Table 1.

Table 1: System specification and test conditions

Component	Specification
Compressor	Seltec TM-15HD 6 cylinder, Nominal 2,000 rpm, Displacement 147 cm ³
Refrigerant	R 404A
Evaporator	Aluminum finned, 6 rows, 9.53 mm diameter aluminum tube, total area 8.52 m ² , frontal area 0.172 m ² ; Fan: 2x250 mm diameter, 80 W, 12/24 V, 1,400 m ³ /hr
Condenser	Aluminum finned, 4 rows, 9.53 mm diameter Cooper tube, total area 13.635 m ² , frontal area 0.2 m ² ; Fan: 2x250 mm diameter, 80 W, 12/24 V, 1,400 m ³ /hr

Expansion valve	FSE-1/2-C
Heater	5 kW heater element with single fan
Heat storage	2 rows x 4 circuit Cooper tube immersed in 50 L of water
Heater	5 kW heater element with single fan
Test condition	Heat leak and cooling capacity for -10, -5 and 0 °C box set temperature based on ARI 1110 at ambient temperature of 15 °C

The defrost time measurement are performed by using surface coil temperature and visual observations. The defrost process is considered end when the evaporator coil temperature reaches 0 °C and there are no surface water observed at the coils.

4 RESULTS AND DISCUSSION

The inputs for the computer simulation include evaporator and condenser model, air flow and temperature, ambient and box set point temperature, and heat storage model. The comparison of simulation and experimental investigation are performed for cooling capacity and heat storage temperature for three different operating conditions. The experimental investigation also measures the system power input and system COP.

The average refrigerated box heat transfer constant of 14.39 W/°C at 15 °C ambient temperature is obtained from the heat leak tests prior to the cooling capacity tests.

The simulated and experimental values of cooling capacity are shown in Figure 3. The simulated cooling capacity over predicts the experimental values with higher deviation occurs at lower refrigerated box set temperature. The deviation may be attributed mainly to the pressure losses at the suction line which are neglected during the simulation. The lower suction pressure during the experiment leads to the lower mass flow rate, hence the simulation results over predict the cooling capacity.

The system power input and system COP is depicted in Figure 4. The measured power input includes compressor's motor, evaporator and condenser fans and control operations and other losses. The simulation and experimental comparison of compressor power draw values, which is different from the total power input, is not presented here. From Figure 4, the compressor power is more required at higher refrigerated box set point temperature, to achieve the balance point of the system; hence the system power input increases with the increase of the refrigerated box set point temperature. Given the COP is directly proportional to cooling capacity, the COP increases as the refrigerated box temperature increases.

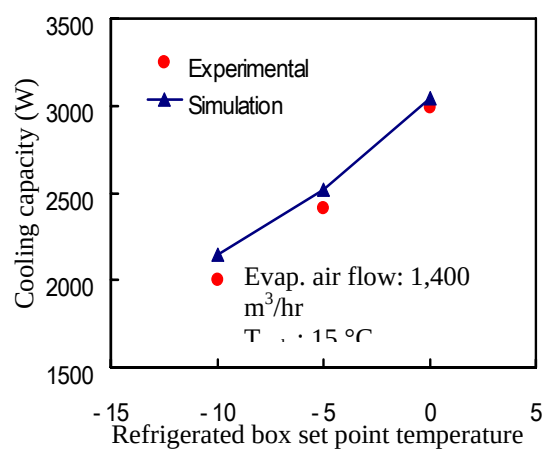


Figure 3: Simulation and experimental results of cooling capacity with box set temperature

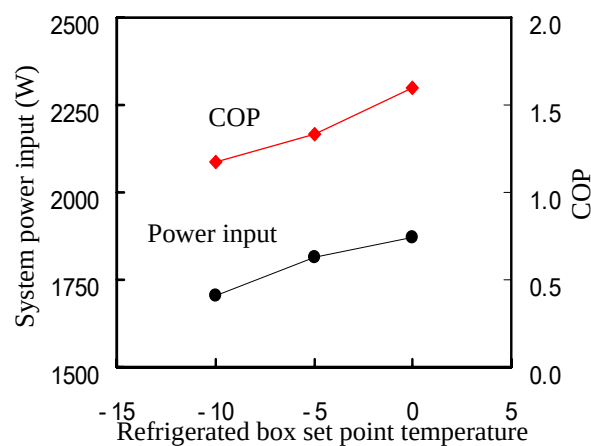


Figure 4: Experimental results of power input and COP with box set temperature

The simulation and experimental investigation of heat storage temperature is shown in Figure 5. Using water, both simulation and experimental results are in a good agreement. The equation (2) shows good prediction of the heat storage temperature increase. The experimental results become lower than the simulation values at higher heat storage temperature with water temperature and longer time. The deviation may be caused by heat losses in the heat storage equipment.

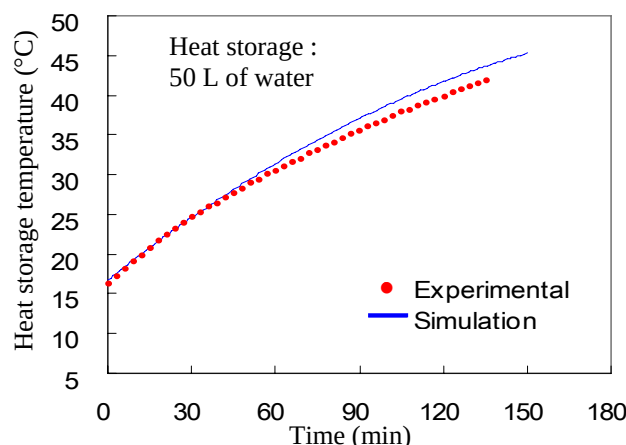


Figure 5: Heat storage temperature with time

The heat transferred from the hot refrigerant during the defrost process is not only used to melt the frost, part of the energy is stored in the coils, frost, and surface water, and is absorbed during the vaporization of the surface water and convected to the ambient air. This unavioded excess energy become part of cooling load during the next refrigeration cycle. The simulation results for refrigerant gas temperatures of 60 °C are used to analyze the energy distribution during the defrost process and are listed in Table 2. The defrost efficiency (Cole 1989) is defined as the ratio of the energy used to melt the frost and the total energy provided during the defrost process.

The experimental result of defrost time indicates a good agreement with the simulation values with 13.8% deviation. The lower values from the simulation may be attributed to the fact that in the actual defrost process, there are transient processes at the beginning of the defrost cycle. Some liquid refrigerant still exist at the evaporator coils when the hot gas from compressor is directed to evaporator at the beginning, which is not considered in the defrost model analysis.

Table 2: Defrost process energy distribution

Hot gas refrigerant temperature (°C)	Defrost time (s)*		Energy stored, Q_s (%)	Energy to vaporize surface water, Q_v (%)	Energy convected, Q_c (%)	Defrost deficiency Q_f (%)
	S	E				
60	320	364	24.3	26.6	25.1	24.0

* S: simulation value; E: Experimental result

5 CONCLUDING REMARKS

A refrigerated vehicle for sliced-raw fish transportation is developed. The refrigeration system is equipped with heat storage for reverse cycle-hot gas defrost (the stored heat is used during defrost cycle of the system). Simulation of the system at ambient temperature of 15 °C and three different box operating temperatures are performed and validated experimentally. The simulation values show good agreement with the experimental results with the deviation for cooling capacity and defrost time less than 7.5% and 13.8% respectively. These results

are important in order to avoid excessive parasitic load and temperature rise inside the refrigerated container. Further investigation for different system configurations and operating conditions are required to improve the reliability of the model.

6 NOMENCLATURE

A	: area [m^2]
C_v	: specific heat [J/kg K]
h	: enthalpy [J/kg]
m	: mass flow rate [kg/s]
M	: mass of water [kg]
P	: compressor power [W]
q	: heat transfer rate [W]
Q	: heat energy [J]
t	: operation time [s]
T	: temperature [K]
U	: overall heat transfer coef. [$\text{kW/m}^2 \text{K}$]

Subscript

c	: condenser
e	: evaporator
g	: gas
l	: liquid
r	: refrigerant
s	: stored energy
th	: heat storage
v	: vaporization
w	: water
cw	: evaporator coil wall
tp	: two phase

7 ACKNOWLEDGEMENTS

The authors wish to acknowledge that this work was supported by the Ministry of Maritime Affairs and Fisheries of Korea (KIMST Grant No. 20060090) and Jeil SPV Co.

8 REFERENCES

- IIR. 2006. "Refrigeration Drives Sustainable Development," Report Card of IIR
- Hoffenbecker N., S.A. Klein, D.T. Reindl. 2005. "Hot Gas Defrost Model Development and Validation," International Journal of Refrigeration 28: 605-615.
- Jolly P.G., C.P. Tso, Y.W. Wong and A.M. Ng. 2000. "Simulation and Measurement on the Full Load Performance of Refrigeration System in a Shipping Container," International Journal of Refrigeration 23: 112-126.
- Ardiyansyah, K.I. Choi, J.T. Oh, H.K. Oh, 2007, "Modelling of a High Efficiency Refrigeration System with Heat Storage for Reverse Cycle Hot Gas Defrost," Int. J. of AC and Ref. 15(4), 175-179

Incropera F.P. and David P. DeWitt 1990. *Fundamental of Heat and Mass Transfer 3rd edition*, John Wiley and Son, Singapore.

Boeis A.M., K.O. Homan. J.H. Davidson. Wei Liu 2005. A Variable Effectiveness Model for Indirect Thermal Storage Devices. Proceedings of ASME Summer Heat Transfer Conference HT2005-72711 Research 25:859-880.

Lemon, E.W., M.L. Huber, and M.O. Mc Linden. 2007. REFPROP: Reference Fluid Thermodynamic and Transport Properties -NIST Standard Reference Database 23-Version 8.0. USA.

Wang Chi-Chuan. kuan-Yu Chi. Chun-Jung Chang 2000. Heat Transfer and Friction Characteristics of Plain Fin-and-tube Heat Exchangers. part II: Correlation. International Journal of Heat and Mass Transfer 43: 2693-2700

Gungor K.E.. & Winterton. R.H.S. 1987. Simplified General Correlation for Saturated Flow Boiling and Comparisons of Correlations with Data. Chemical Engineering Research and Design 65: 148-156.

ARI, 2006, "Performance Rating of Mechanical Transport Refrigeration Units-Standard 1110," ARI, USA

Krakow K.I.. S.Lin. L. Yan 1992. A Model of Hot-Gas Defrosting of Evaporators. ASHRAE Trans. 98(1): 451-474

Cole R.A. 1989. Refrigeration Loads in a Freezer Due to Hot Gas Defrost and Their Associated Costs ASHRAE Trans. 95(2): 1149-1154