Thermodynamic Investigation of a Typical Commercial Refrigeration System

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Abstract

In this study, the thermodynamic performance of a typical commercial refrigeration system is investigated and the degradation from design condition is evaluated. A mathematical model is developed to simulate thermal behavior of the refrigeration system and to obtain the coefficient of performance (COP). A number of measuring equipment including thermocouples, pressure transducers, flow meters, and power meters are installed on the refrigeration system to obtain the real-time data for a period of three months. The acquired data is utilized to investigate the thermodynamic behavior of refrigeration system in addition to model validation. A good agreement between the simulation results and the real-time data, with a maximum discrepancy of 9%, is achieved. The results represent a range of 1.1-1.4 for current COP of the refrigeration system under different operating conditions. It is shown that due to degradation, the current COP is lower than design COP by 29%-39% in different duty cycles.

Keywords

Performance, Commercial Refrigeration, Thermodynamic Modelling, Duty Cycle

Nomenclature

W	Power (W)
c_0 to c_8	Compressor polynomial constants
d_0 to d_8	Compressor polynomial constants
cf_0 to cf_3	Fan polynomial constants
C	Thermal capacity $(\mathrm{W/°C})$
C^*	Thermal capacity ratio, $C_{\min}/C_{\max}$
h	Enthalpy $(\mathrm{kJ/kg°C})$
$\dot{m}$	Mass flow rate $(\mathrm{kg/s})$
NTU	Number of transfer units
$\dot{Q}$	Heat transfer rate (W)
T	Temperature $(\mathrm{°C})$
UA	Overall heat transfer coefficient $(\mathrm{W/°C})$
ε	Heat exchanger effectiveness
η	Efficiency
Subscripts	
c	Condenser
e	Evaporator
E	Electrical
M	Mechanical
in	Inlet
out	Outlet
max	Maximum
min	Minimum
ref	Refrigerant
a	Air
$comp$	Compressor
$cond$	Condenser
$evap$	Evaporator
f	Fan
nom	Nominal
TEV	Thermostatic expansion valve

1. Introduction

Refrigeration units consume a tremendous amount of electrical power leading to large quantities of greenhouse gas emissions (GHG) and contain refrigerants which significantly contribute to the ozone depletion and global warming process through their ozone depleting and global warming potentials. The world's demand for commercial refrigeration equipment including reach-in and walk-in coolers, freezers and display cabinets, has increased 4.6% per year during 2008-2011 [1]. Keeping the temperature and humidity of food products in the safe range during the storage step, is the main reason of using refrigeration systems in food provision industry. Studies have shown that refrigeration is the source for 35- 50% of the total energy consumption in the abovementioned industry [2–5]. Walk-in freezers are widely used to control a space temperature in the range of -12 to -18 °C for safekeeping of frozen foods. Due to such a low cooled space temperature, coefficient of performance (COP) for walk-in freezers, typically 1, is quite low in comparison with the other refrigeration systems [2,3]. As a result of aging, continuous performance degradation is inevitable for all refrigeration systems including the walk-in freezers. To develop strategies for improving the performance of existing walk-in freezers, current COP of the refrigeration system should be obtained first. The obtained magnitude can be then compared with the design condition for finding the aging-related deviation. Evaluating the current COP deviation from the design condition assists in finding potential energy saving for the existing walk-in freezers [6].

Mathematical simulation of vapor compression refrigeration (VCR) systems is an effective methodology to evaluate thermodynamic and thermal parameters of these systems to find the cooling power and COP [7,8]. A number of models have been published for different types of refrigeration systems, including domestic refrigerators and refrigerating cassettes [9,10], automotive air conditioning and shipping containers [11–13], supermarket refrigeration systems [1,2,14], etc. Although there are numerous studies in the literature focused on refrigeration system in different applications, commercial walk-in freezers have not been well covered. Additionally, performance degradation, as the main source of higher energy consumption for old refrigeration systems in comparison to brand new systems, has not been studied for walk-in freezers.

Accordingly, present study focuses on evaluating the current performance and its aging-related degradation from design condition for a typical walk-in freezer room. A walk-in freezer room equipped with a 10-year old condensing unit located in a restaurant in Surrey, BC, Canada, is selected for the investigation. The major goals of the study are to: i) investigate the current performance of the typical walk-in freezer, and ii) find the performance deviation from the original design condition (brand-new). The study includes mathematical modeling, real-time data collection, and performance analysis of the freezer.

2. Typical commercial walk-in freezer

The Surrey Central Pub and Brewery in Surrey, British Columbia, Canada, which has five walk-in freezers and refrigerator rooms, is chosen as a pilot commercial unit. Figure 1 shows the selected freezer room for this study and its condensing unit (10 years old) installed on the ceiling of room. The compressor installed for the selected system is model “CS14K6E-PFV” with a nominal cooling capacity of 2.3 kW. HFC 507 is used as the refrigerant in this system.

Figure 1: The selected freezer room (left) and its condensing unit on the roof (right)

3. Mathematical model

A mathematical model is developed for evaluating the performance of VCR system and to simulate thermal parameters under various working conditions. The major output parameter from the modeling of a VCR system is the coefficient of performance (COP = cooling power/Input power). In addition to the COP, other parameters of a VCR system including cooling power, input power, and refrigerant mass flow rate are obtained from the modeling results.

3.1. Compressor Sub-Model

A map-based model is used in this study for thermodynamic simulation of the compressor as shown through Eqs. (1) to (3) in Table 1. It was experimentally shown that this approach had a good agreement with experimental data [12], which is less than 2.5% maximum error. Based on these equations, with known inlet conditions, the state point of the refrigerant gas at outlet can be calculated.

3.2. Condenser and Evaporator Models

Energy balance correlations between the refrigerant and air flows in condenser and evaporator are used to model heat transfer in these components [15]. In the present study, following [16], an $\varepsilon - NTU$ model is employed to derive the mathematical model for the condenser and evaporator. The correlations are presented in Table 1, see Eqs. (4) to (10). In addition to the heat transfer model for the condenser and evaporator, map-based correlations are employed to obtain the fans power consumption using manufacturer’s datasheets [17], see Eq. (11).

3.3. Thermostatic Expansion Valve Model

Following [12], an isenthalpic model is selected for thermodynamic simulation of the thermostatic expansion valve in the present study. In this model, as a result of adiabatic assumption, the inlet and outlet refrigerant enthalpies are considered the same, see Eq. (12).

3.4. Numerical Solver

Developing the mathematical model for VCR system, including correlations of Table 1 as well as the condenser and evaporator overall heat transfer relationships in addition to thermodynamic properties correlations of HFC-134a, leads to a set of 20 coupled nonlinear equations that has to be solved simultaneously. The known and unknown parameters of the VCR model are listed in Table 2. A Newton-Raphson method is employed to develop an iterative numerical code in C for solving the set of nonlinear correlations [18].

4. Data Collection

To validate the present VCR model, real-time data collection is required. A number of measuring equipment is installed on the system components (condensing unit and evaporator) to collect the required parameters including refrigerant pressure, refrigerant and air temperature and mass flow rates, and input power to the compressor and fans. Figure 2 shows some of the installed equipment on the existing condensing unit for data collection.

5. Results and Discussions

After installing the measuring devices, all the refrigerant-side and the air-side required parameters are collected for three months and used to investigate performance of the selected VCR system. After detailed investigation into the collected data, four ‘duty cycles’ have been defined for different working hours of the selected site during weekdays and weekends (see Table 3).

Compressor sub-model	$W_{comp} = c_0 + c_1 T_e + c_2 T_c + c_3 T_e^2 + c_4 T_e T_c + c_5 T_c^2 + c_6 T_e^2 T_c + c_7 T_e T_c^2 + c_8 T_e^2 T_c^2$	(1)
	$\dot{m}_{ref} = d_0 + d_1 T_e + d_2 T_c + d_3 T_e^2 + d_4 T_e T_c + d_5 T_c^2 + d_6 T_e^2 T_c + d_7 T_e T_c^2 + d_8 T_e^2 T_c^2$	(2)
	$\dot{m}_{ref} (h_{ref,comp,out} - h_{ref,comp,in}) = W_{comp} \times \eta_E \times \eta_M$	(3)
Condenser and evaporator sub-model	$\varepsilon = \dot{Q} / \dot{Q}_{\max}$	(4)
	$\dot{Q}_{\max} = C_{\min} (T_{hot,in} - T_{cold,in})$	(5)
	$\dot{Q} = \varepsilon C_{\min} (T_{hot,in} - T_{cold,in})$	(6)
	$\varepsilon = \dfrac{1 - \exp[-NTU(1 - C^*)]}{1 - C^* \exp[-NTU(1 - C^*)]}$	(7)
	$C^* = C_{\min} / C_{\max}$; $NTU = UA / C_{\min}$	(8)
	$\dot{Q} = \left\| \dot{m}_a \left(h_{a,in} - h_{a,out} \right) \right\|$	(9)
	$\dot{Q} = \left\| \dot{m}_{ref} \left(h_{ref,out} - h_{ref,in} \right) \right\|$	(10)
	$W_f = W_{f,nom} \left(cf_0 + cf_1 \left(\dfrac{\dot{m}_a}{\dot{m}_{a,nom}} \right) + cf_2 \left(\dfrac{\dot{m}_a}{\dot{m}_{a,nom}} \right)^2 + cf_3 \left(\dfrac{\dot{m}_a}{\dot{m}_{a,nom}} \right)^3 \right)$	(11)
Expansion valve sub-model	$h_{ref,TEV,in} = h_{ref,TEV,out}$	(12)

Table 1: Mathematical model correlations

Input parameters		
η_E	UA	$d_0 - d_8$
η_M	$\dot{m}_a$	$cf_0 - cf_3$
$C_{\min}$	$h_{a,in}$	$\dot{m}_{a,nom}$
$C_{\max}$	$c_0 - c_8$	$W_{f,nom}$
Output parameters		
Cooling power	W_{comp}	$W_{f,cond}$
Heat rejection	NTU_{evap}	NTU_{cond}
COP	$h_{ref,evap,in}$	$\dot{m}_{ref}$
$h_{ref,comp,in}$	$h_{ref,comp,out}$	$h_{ref,TEV,in}$
$W_{f,evap}$	$h_g @ T_e$	T_e
ε_{evap}	ε_{cond}	$h_g @ T_c$
T_c	$h_f @ T_c$	

Table 2: Input and output parameters of the mathematical model

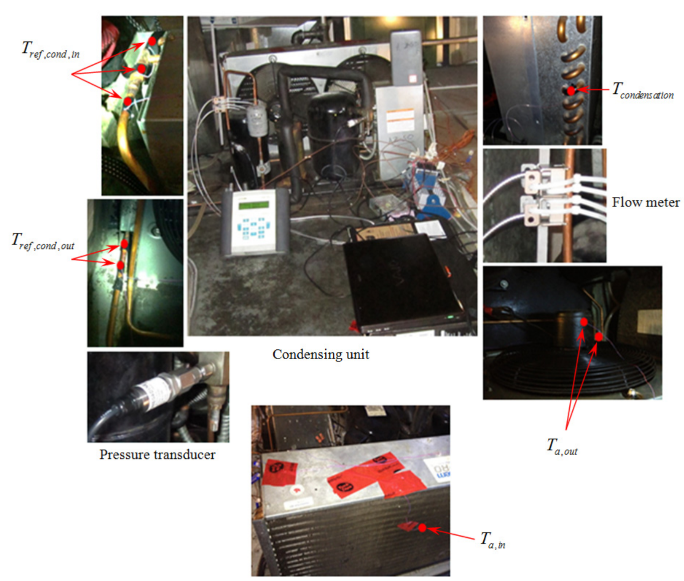

Figure 2: Installation of thermocouples, pressure transducers, and flow meters on the condensing unit. (Red dots point at the locations of thermocouple installation.)

Duty cycle	Description	Days	Start hour	End hour
DT1	Weekends work-hours	Saturday-Sunday	10 AM	12 AM
DT2	Weekends off-hours	Saturday-Sunday	12 AM	10 AM
DT3	Weekdays work-hours	Monday-Friday	11 AM	11 PM
DT4	Weekdays off-hours	Monday-Friday	11 PM	11 AM

Table 3: Duty cycle definition

In Fig. 3, samples of the collected temperature data for the evaporating and condensing units during weekdays are illustrated. Figure 4 also represents a sample of the measured input electrical power to the condensing unit. These plots reflect different types of fluctuations in the temperature and power as a result of variant operating conditions, cooling demand, and activities. The maximum uncertainty of the presented data in these figures is less than 3%. The following can be concluded from the acquired results:

- A frequent defrost cycle is employed every 6 hours to melt down the formed ice from the evaporator coil. During the defrost cycle, the compressor is shut off and an electrical heater (1,600 Watt) is switched on. It is noticed that the defroster does not function properly, thus it is recommended to install a humidity-based controlled defroster instead of the current timer-based one to improve the overall energy efficiency.
- The compressor frequently cycles on-off during the day hours due to door openings and loading/unloading of food products in the freezer room. It also cycles during night hours due to malfunction of the degraded insulation over time. In addition, it is noticed that an incandescent light (100 Watt) and a door anti-sweat heater (200 Watt) are always kept on. These unnecessarily imposed loads can be eliminated by using a motion sensor along with a high efficiency lamp and a humidity-based anti-sweat heater for the door.

To find the average performance of the refrigeration system and evaluate its deviation from the first-day or 'design' condition, the average values of the measured parameters are obtained from the field data. Also, all the required input parameters to the mathematical model of VCR cycle are calculated from the measured data during three months. To evaluate the performance of a VCR unit, the average delivered cooling effect and consumed power should be obtained over each duty cycle. Employing this approach also enables us to assess the deviation of current performance from the design performance that can be calculated from condensing unit's manufacturer data.

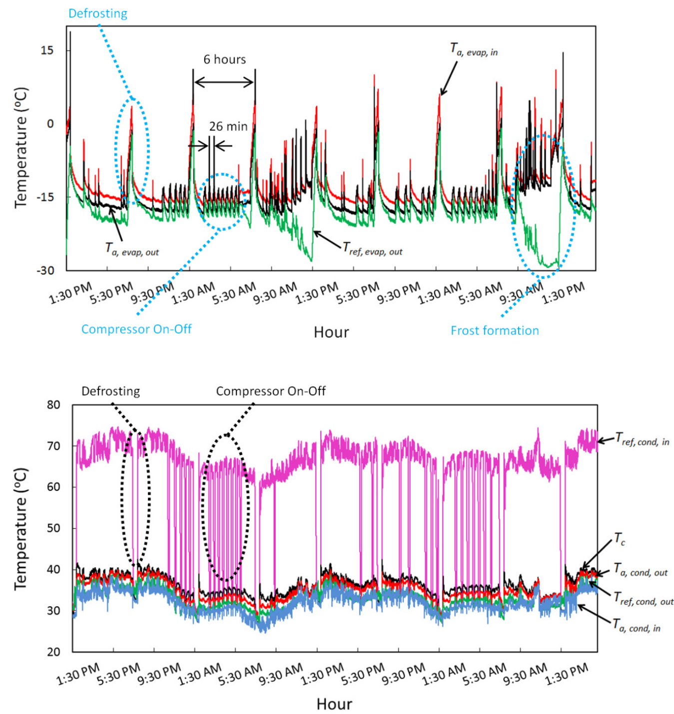

Figure 3: Sample measured temperature during weekdays

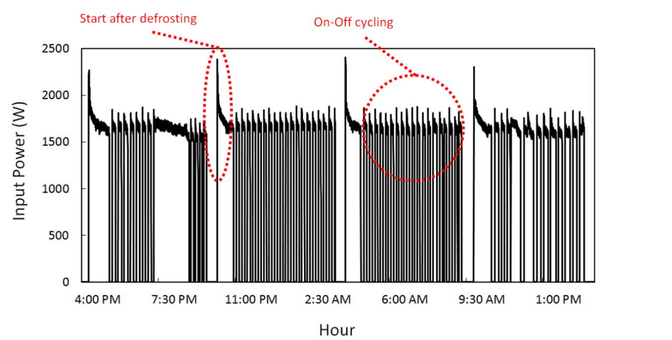

Figure 4: Sample measured input electric power to condensing unit

Figure 5 represents three important parameters obtained from modeling and measurements over the duty cycles. The average cooling power, input power, and evaporator outlet air temperature are presented and compared, which shows a good agreement between the simulation and measured values with less than 10% relative difference. The maximum uncertainty of the obtained results in this figure is less than 5%. Based on the validated simulation results, the provided cooling power by the refrigeration system varies in a range of 1.75-2.05 kW for the different duty cycles. In addition, the total input power to the refrigeration system varies in a range of 1.56-1.68 kW.

Figure 6 shows the COP of VCR system based on modeling and measurements for different duty cycles. The design COP in this figure is calculated based on manufacturer's data to find the system's current performance deviation from the first day condition. The results show a good agreement between simulated and measured values with maximum relative difference of 12%. The results indicate that the system currently functions with a 29-39% lower COP than the first day. The reduction in COP can be a result of various parameters, mainly: i) wear and tear; ii) refrigerant leakage; iii) fouling on the condenser and evaporator coils. Based on the simulation and measurements, the VCR system (including its defroster) currently consumes 16,300 kWh/year that is 4,600 kWh/year more than a new system (brand new of the same condensing unit). This significantly higher energy consumption indicates the necessity of either replacing the current system with efficient refrigeration system or resolving the performance-related issues of it. Our observations indicate that the system is relatively new (~only 10 years old); fairly well designed; and well maintained. From the calculated COP values, one can conclude that even for such a relatively new refrigeration system a significant degradation takes place over years. There can be found many refrigeration units running around the world that are older than the studied system. Accordingly, a significant amount of energy can be saved worldwide by replacing or improving the efficiency of old commercial refrigeration units.

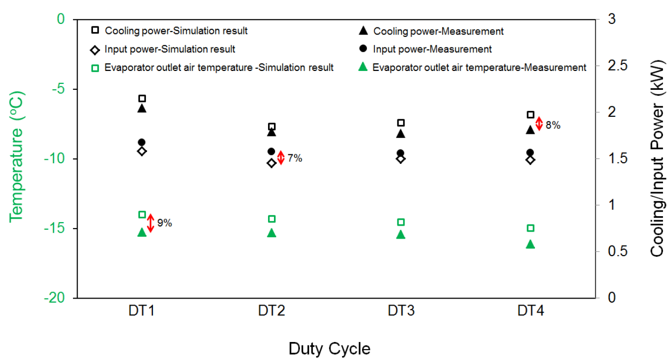

Figure 5: Validation of simulation results with measured values

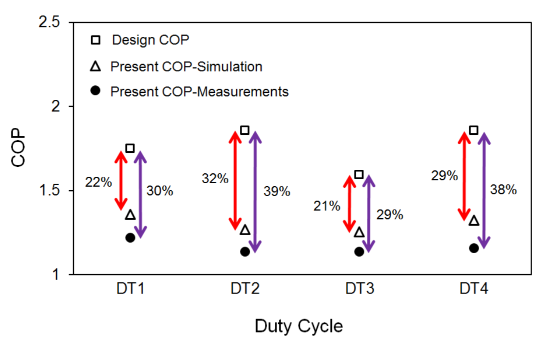

Figure 6: Present and design COP of VCR system for different duty cycles

6. Conclusions

This paper presented an investigation into the current performance and its degradation from the design condition for a typical VCR system installed in a walk-in freezer room. A mathematical model was developed for simulating the cooling capacity, power consumption, and COP of the system. A variety of measuring devices were also employed to collect data for the mathematical model validation and detailed study. Four duty cycles were defined and the performance was assessed for each of these cycles separately. Using the collected real-time data, the mathematical model was validated and employed for the COP simulations. The results showed that the current COP of system has dropped by 29-39% compared to the design condition. Energy consumption calculations indicated that the studied VCR system consumes 16,300 kWh/year. Additionally, the calculations showed that as a result of COP degradation, the VCR system is currently consuming 4,600 kWh/year more electrical energy in comparison with a new system for providing a same cooling power.

Acknowledgments

This work was supported by Automotive Partnership Canada (APC), Grant No. APCPJ 401826-10, and Metro Vancouver Office, Burnaby, BC. The authors would like to thank the kind support from Central City Brew Pub, 13450 102nd avenue, Surrey, BC.

References

[1] Gaspar P. D., Carrilho Gonçalves L. C., and Pitarma R. a., 2011, "Experimental analysis of the thermal entrainment factor of air curtains in vertical open display cabinets for different ambient air conditions," Appl. Therm. Eng., 31(5), pp. 961–969.

[2] Yang L., and Zhang C.-L., 2010, "Analysis on energy saving potential of integrated supermarket HVAC and refrigeration systems using multiple subcoolers," Energy Build., 42(2), pp. 251–258.

[3] Pasific Gas and Electric Company, 2013, Supermarket refrigeration; 2013 California building energy efficiency standards.

[4] Tassou S. a., Ge Y., Hadawey a., and Marriott D., 2011, "Energy consumption and conservation in food retailing," Appl. Therm. Eng., 31(2-3), pp. 147–156.

[5] Kullheim J., 2011, "Field measurement and evaluation of CO2 refrigeration systems for supermarket," KTH, School of Industrial Engineering and Management.

[6] Zhao L., Cai W.-J., Ding X., and Chang W., 2013, "Decentralized optimization for vapor compression refrigeration cycle," Appl. Therm. Eng., 51(1-2), pp. 753–763.

[7] Anand S., Gupta a., and Tyagi S. K., 2013, "Simulation studies of refrigeration cycles: A review," Renew. Sustain. Energy Rev., 17, pp. 260–277.

[8] Ding G., 2007, "Recent developments in simulation techniques for vapour-compression refrigeration systems," Int. J. Refrig., 30(7), pp. 1119–1133.

[9] Hermes C. J. L., Melo C., Knabben F. T., and Gonçalves J. M., 2009, "Prediction of the energy consumption of household refrigerators and freezers via steady-state simulation," Appl. Energy, 86(7-8), pp. 1311–1319.

[10] Waltrich M., Hermes C. J. L., and Melo C., 2011, "Simulation-based design and optimization of

refrigeration cassettes," Appl. Energy, 88(12), pp. 4756–4765.

[11] Tian C., and Li X., 2005, "Numerical simulation on performance band of automotive air conditioning system with a variable displacement compressor," Energy Convers. Manag., 46(17), pp. 2718–2738.

[12] Jolly P. G., Tso C. P., Wong Y. W., and Ng S. M., 2000, "Simulation and measurement on the full-load performance of a refrigeration system in a shipping container," Int. J. Refrig., 23, pp. 112–126.

[13] Bagheri F., Fayazbakhsh M. a., Cheppudira Thimmaiah P., and Bahrami M., 2015, "Theoretical and experimental investigation into anti-idling A/C system for trucks," Energy Convers. Manag., 98, pp. 173–183.

[14] Arias J., and Lundqvist P., 2006, "Heat recovery and floating condensing in supermarkets," Energy Build., 38(2), pp. 73–81.

[15] Bagheri F., Fakoor-Pakdaman M., and Bahrami M., 2014, "Utilization of orthotropic graphite plates in plate heat exchangers, analytical modeling," Int. J. Heat Mass Transf., 77, pp. 301–310.

[16] Shah R. K., and Sekulic D. P., 2003, Fundamentals of heat exchanger design, John Wiley & Sons.

[17] Zhao L., Cai W., Ding X., and Chang W., 2013, "Model-based optimization for vapor compression refrigeration cycle," Energy, 55, pp. 392–402.

[18] Abbasbandy S., 2003, "Improving Newton–Raphson method for nonlinear equations by modified Adomian decomposition method," Appl. Math. Comput., 145(2-3), pp. 887–893.