International Journal on Emerging Technologies 5(2): 7-13(2014)

ISSN No. (Print) : 0975-8364
ISSN No. (Online) : 2249-3255

Performance Improvement Cycle effect Supermarket High Cooling & Low power Refrigeration systems

Bhavna Dubey* and Dr Manoj Chopra**
*Research Scholar, Department of Mechanical Engineering,
**Asst. Professor, Department of Mechanical Engineering,
RKDF, Bhopal, (MP) India

(Corresponding author: Bhavna Dubey)
(Received 05 June, 2014 Accepted 11 July, 2014)

ABSTRACT: Today phase-out of ozone-depleting refrigerants and the improvement of waste heat recovery are the main issues of the supermarket refrigeration industry. The negative impact of refrigerant leakages on global warming and ozone depletion has stimulated the development of new supermarket refrigeration systems requiring less quantities of refrigerant. The advanced system presented in this paper involves secondary fluid loops on both refrigerating and condensing sides, and heat recovery with brine-to-air heat pumps and passive heat exchangers. This integrated concept has a considerable potential to reduce combined refrigeration and HVAC energy use in supermarkets multiplex refrigeration systems with more conventional heat recovery approaches. It also may reduce up to 70% of the quantity of required. energy consumption of 1,000 kWh/m2/yr. Conventional multiplex refrigeration systems account for about 50% of this energy consumption and require large refrigerant charges: 1000 to 2500 kg (2,200 to 5,500 lb) of HCFC or HFC per store. Hundreds of meters of piping, and many valves and brazed joints provide 15% to 30% of refrigerant annual losses. Most of these systems recover by desuper heating between 30% and 40% of compressors' total heat rejection. In the Canadian cold climate, this amount of energy is not sufficient to completely eliminate the use of fossil fuels (natural gas, propane) for space and hot water heating. Environmental issues have stimulated the development of new supermarket refrigeration systems that require less quantities of refrigerant.1 among these systems, decentralized compressors and completely.

Keywords- supermarket refrigeration, energy, compressor racks.

I. INTRODUCTION

Supermarkets are one of the [3] most energy-intensive types of commercial buildings. Significant electrical energy is used to maintain chilled and frozen food in both product display cases and walk-in storage coolers. Supermarkets [11] have a wide range of sizes. In North America, store sizes vary from roughly 2,000 to 11,000 square meters. A typical supermarket consumes roughly 2 million kWh annually, and roughly half is for refrigeration [1]. Thus, improvement in energy efficiency of supermarket refrigeration will affect the store's bottom line of profit margin. The most commonly used refrigeration system for supermarkets today is the parallel rack direct expansion system using a HFC refrigerant such as R404A. a typical parallel rack system. Multiple compressors operating at the same saturated suction temperature (SST) are mounted on a skid, or rack, and are piped with common suction and discharge refrigeration lines. Using multiple compressors in parallel provides a means [17] of capacity control, since compressors can be turned on and off to meet refrigeration load. All display cases and cold store rooms use direct expansion (DX) air-refrigerant evaporator [18] coils that are connected to compressor racks in a remote machine room typically located in the back or on the roof of the store. Heat rejection is usually done with air-cooled condensers because these are least costly to install and maintain. A typical supermarket requires 1400 to 2300 kg of refrigerant. In response to the environmental concern of global warming, efforts have been made in supermarket refrigeration industry to design or develop refrigeration systems that operate with less refrigerant charge and energy consumption. The "advanced" systems that have been used or developed and have much less refrigerant charge than the parallel rack system include the distributed, [16] self-contained, glycol secondary loop, and CO_2 [15] secondary loop/cascade system, etc.

The distributed refrigeration system is similar to the parallel rack, and the difference is that several small compressor racks are located in cabinets that are distributed throughout the [14] store and close-coupled to the display case lineups or storage rooms they serve. With this approach, both the machine [2] room and the long lengths of piping needed to connect the cases with large remote compressor racks are eliminated. The advanced self-contained system [19] consists of display cases or storage coolers each having their own compressor and water-cooled condenser with warm water pumped to the rooftop fluid cooler for heat rejection. The self-contained system has advantages in extremely low refrigerant change (one tenth of rack systems), [12] easy and low-cost installation, and flexibility in time to order and remodeling. However, the self-contained system has some inherent disadvantages including high Equipment cost and[10] low efficiency due to heat transfer penalty of water-cooled condensing. A single-phase[8] secondary loop system employs one or more chillers to refrigerate a secondary fluid (generally glycol/water

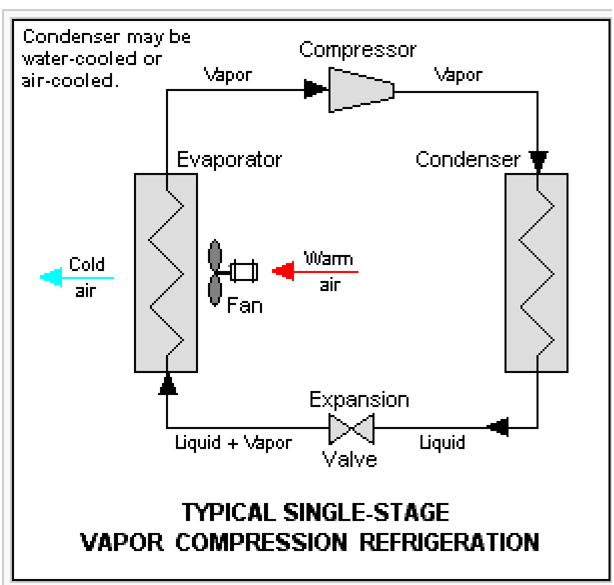

Fig. 1. Refrigeration system.

Mechanical [7] refrigeration is accomplished by continuously circulating, evaporating, and condensing a fixed supply of refrigerant in a closed system. Evaporation occurs at a low temperature and low pressure while condensation occurs at a high temperature and high pressure. Thus, it is possible to transfer heat from an area of low temperature (i.e., refrigerator cabinet) to an area of high temperature (i.e., kitchen).Referring to the illustration below, [4] beginning the cycle at the evaporator inlet the low-pressure liquid expands, absorbs heat, and evaporates, changing to a low-pressure gas at the evaporator outlet

The compressor pumps this gas from the evaporator through the accumulator increases its pressure, and discharges the high-pressure gas to the condenser [23] The accumulator is designed to protect the compressor by preventing slugs of liquid refrigerant from passing directly into the compressor. An accumulator should be included on all systems subjected to varying load conditions or frequent compressor cycling. In the condenser, heat is removed from the gas, which then condenses and becomes a high-pressure liquid. In some systems, this high-pressure liquid drains from the condenser into a liquid storage or receiver tank On other systems, both the receiver and the liquid line valve are omitted. A [24] heat exchanger between the liquid line and the suction line is also an optional item, which may or may not be included in a given system design. Between the condenser and the evaporator an expansion device is located. Immediately preceding this device is a liquid line strainer/drier which prevents plugging of the valve or tube by retaining scale, dirt, and moisture. The flow of refrigerant into the evaporator is controlled by the pressure differential across the expansion device or, in the case of a thermal expansion valve, by the degree of superheat of the suction gas. [9] Thus, the thermal expansion valve shown requires a sensor bulb

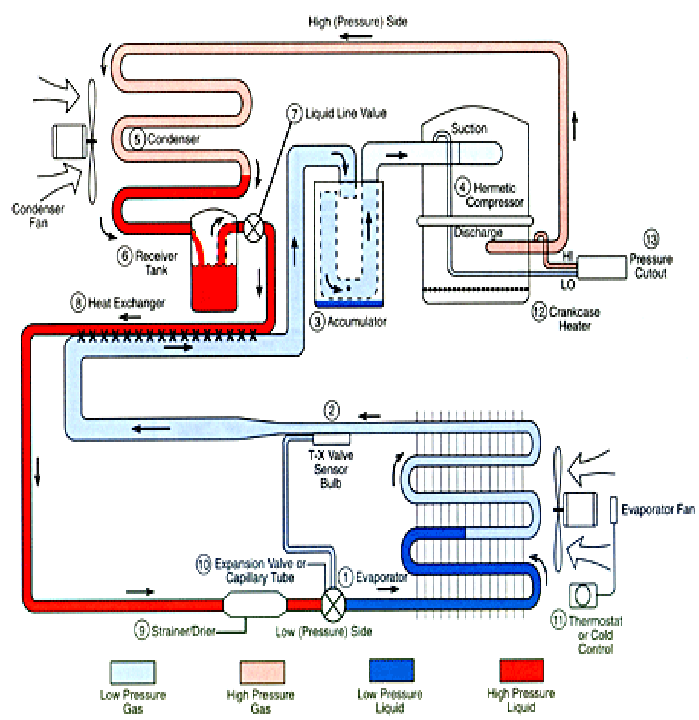

Fig. 2. Basic refrigeration cycle.

Located at the evaporator outlet. In any case, the flow of refrigerant into the evaporator normally increases as the evaporator load increases.

As the high-pressure [6] liquid refrigerant enters the evaporator, it is subjected to a much lower pressure due to the suction of the compressor and the pressure drop across the expansion device. Thus, the refrigerant tends to expand and evaporate. In order to evaporate, the liquid must absorb [5] heat [3] from the air passing over the evaporator. Eventually, [22] the desired air temperature is reached and the thermostat or cold control will break the electrical circuit to the compressor motor and stop the compressor. As the temperature of the air through the evaporator rises, the thermostat or cold control remakes the electrical circuit. The compressor starts, and the cycle continues. In addition to the accumulator, a compressor crankcase heater is included on many systems [21]. This heater prevents accumulation of refrigerant in the compressor crankcase during the non-operating periods and prevents liquid slugging or oil pump out on startup. Additional protection to the compressor and system is afforded by a high- and low-pressure cutout [20] This control is set to stop the compressor in the event that the system pressures rise above or fall below the design operating range.

II. METHODS AND APPLICATIONS OF HEAT RECOVERY

Supermarket refrigeration system direct expansion system type, in which direct expansion evaporator coils are located within the refrigerated display cases and walk-in coolers and freezers, and the compressors and condensers are located in a machine room at the back of the store or on the roof of the store. Supply and return piping delivers refrigerant to and from the evaporators in the display cases and walk-in coolers/freezers. As shown in Figure 2, in a basic direct expansion (DX) refrigeration [25] system, a heat recovery heat exchanger (HRC) can be used to transfer heat from the compressor discharge gas to either the retail space or to process water. The heat to be extracted from the high pressure, high temperature discharge gas consists of both superheat and condensing heat. Any excess heat that is not recovered is rejected to the outdoor ambient via an air-cooled condenser. The escalating cost of energy has drawn much more attention on improving the energy efficiency of supermarket operations. In a supermarket, refrigeration systems consume a large amount of energy in maintaining chilled and frozen food, meanwhile [29] a HVAC (heating, ventilating, and air-conditioning) system is used to assure thermal comfort for occupants and suitable climatic conditions for refrigerated cases.

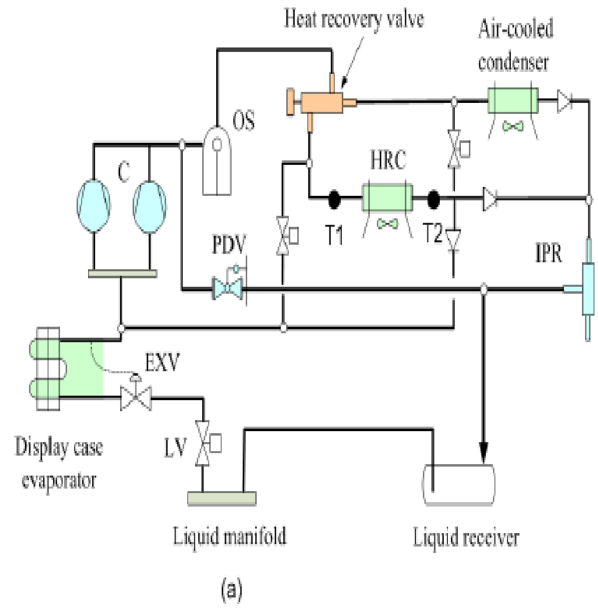

Fig. 3. Supermarket refrigeration system with heat recovery.

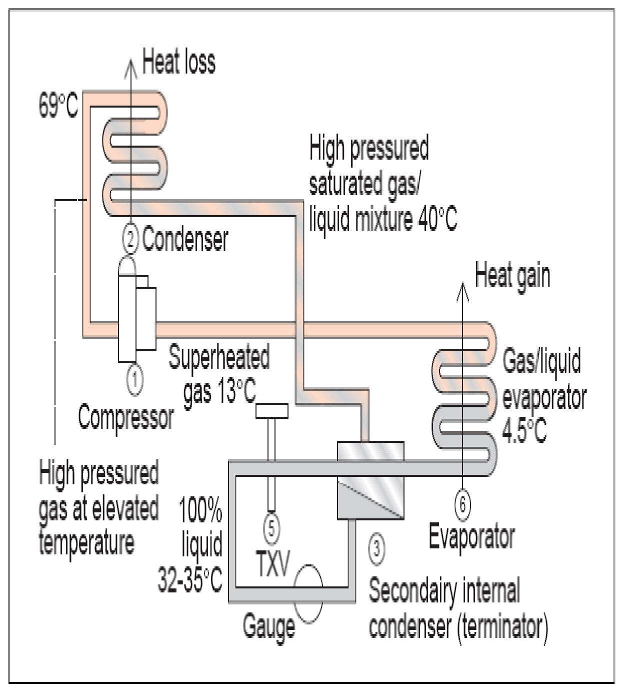

Fig. 4. Liquid line system diagram.

More specifically, the supermarket refrigeration system has normally two temperature levels medium temperature for preservation of fresh food and low temperature for frozen products.

Fresh food is maintained between 1 8C and 14 8C, while frozen food is kept at _12 8C and _18 8C. The evaporating temperature of medium-temperature system varies between _15 8C and 58C, while for a low-temperature system it ranges from _30 8C to _40 8C [1]. Due to such a low evaporating temperature, the COPs of low- and medium-temperature systems are quite low. The typical values are 1 and 2, respectively. The supermarket HVAC system includes an air-cooled or water-cooled chiller/heat hump, cooling tower, pumps and air handling units. The chilled water supply temperature is around 6.7 8C in cooling season and the heated water supply temperature is 45 8C in heating season. The evaporating temperature of chiller system is between 0 8C and 6 8C. The typical COP of the entire HVAC system is around 3–4.5. For saving energy in supermarkets, energy efficient design and optimal control of HVAC and refrigeration systems are two major means. Energy efficient design includes using evaporative condensing to minimize system power requirements [2], application of heat recovery and floating condensing [3], mechanical sub cooling [4–6] for low-temperature system and so on. Mechanical sub coolers are heat exchangers that use refrigeration effect from the medium-temperature system to provide sub cooling for the low-temperature system, which is more efficiency [26] since the medium-temperature system has a higher COP than the low temperature system. Optimal control of HVAC and refrigeration systems can be implemented by using variable speed drives on compressors, pumps, fans with an optimized control system instead of on/off operation [7–9] or developing an efficient control strategy for refrigeration system itself [10]. Another increasing interest [27] of researches is the integration of supermarket HVAC and refrigeration systems. For instance, the refrigerated cases are better operating with low ambient relative humidity (40–45%). The dehumidification is traditionally carried out with HVAC system by over-sizing the coil and re-heating the treated air, which might not be energy efficient. To offset the disadvantages, hybrid HVAC system with chemical dehumidification was studied recently [11,12]. Moreover, the heat rejection from refrigeration systems can be recovered to warm the store’s ventilation air and heat the domestic water supply by employing the water-source heat pump and the integrated system can take advantage of climatic conditions by modulating temperature and condensation pressure to match the building’s Frost forms on evaporator coils by the water vapour in the air condensing and freezing when the surface temperature of the coil falls below 0°C. A small amount [28] of frost may improve the heat transfer performance of the coil by increasing the surface area and surface roughness which induces increased turbulence [29] However, significant frost accumulation deteriorates the coil performance by reducing the air flow and thereby the refrigerating capacity of the evaporator. Maintaining the store humidity at low levels, and using air curtains to prevent penetration of humid store air into the display cases, reduces the rate of frost formation on display case evaporators to some extent, but does not eliminate it completely due to the disturbance of the air curtain by shoppers and staff loading up the display case. Consequently, frosting is a major problem in retail refrigeration systems and evaporators need to be actively defrosted periodically to maintain system performance and temperature control in the display cases. The most commonly used defrost methods in display cases are hot or cool gas defrost and electric defrost. In electric defrost, the thermal energy to melt the ice is provided by an electric strip heater which is situated across the face of the coil. During defrost the refrigerant supply to the display case is switched off, the electric heater is switched on, and the evaporator fans blow air which is heated by the strip heater through the coil, [32] melting the ice from the coil surface. This method of defrost can be implemented on both conventional, single compressor refrigeration systems, and multiplex refrigeration systems which are now widely used in large retail stores. These consist of three or four compressors connected in parallel, providing flexibility [31]in system capacity control and maintenance. Accurate determination of the energy consumption during the defrost cycle would require comprehensive instrumentation to measure the power consumption of the refrigeration packs during defrost. Although it will be useful to carry out such an exercise, this is beyond the scope of the present paper. [33] It is interesting, however, to make an estimate of the energy consumed during defrost and this can be based on the following assumptions The refrigerant (R22) flow rate was measured using a Coriol is mass flow meter. Temperatures and pressures of the refrigerant were measured at five points in the system using thermocouples and pressure transducers. The power consumption of the compressors was recorded using a power transducer. The display cabinet and the coil were fitted with thermocouples on the surface. The operation of the system was controlled by a standard Supermarket controller board. For example, an individual compressor can be rated by comparing the energy needed to run the compressor versus the expected refrigeration capacity based on inlet volume flow rate. [30] It is important to note that both CoP and PF for a refrigeration system are only defined at specific operating conditions, including temperatures and thermal loads. Moving away from the specified operating conditions can dramatically change a system's performance.

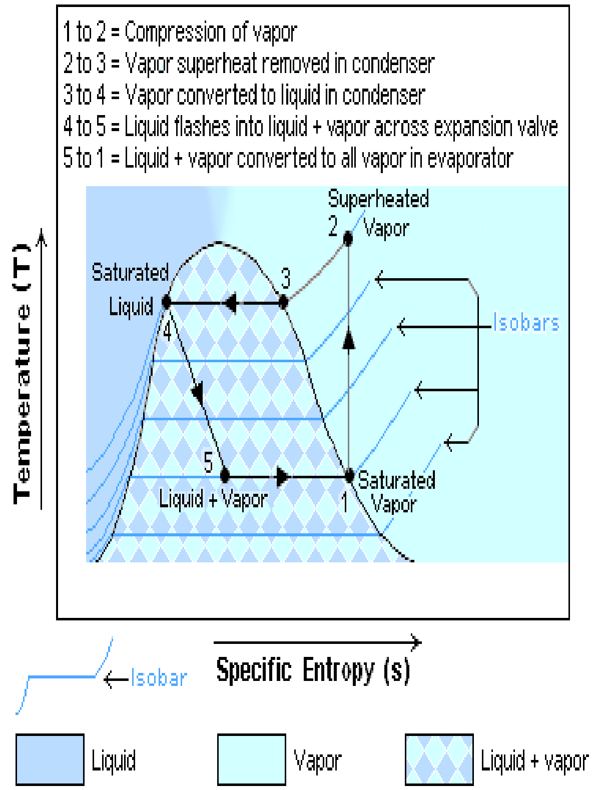

Fig. 5. Temperature–Entropy diagram.

A refrigeration system's coefficient of performance (CoP) is very important in determining a system's overall efficiency. It is defined as refrigeration capacity in kW divided by the energy input in kW. While CoP is a very simple measure of performance, it is typically not used for industrial refrigeration in North America. Owners and manufacturers of these systems typically use performance factor (PF). A system's PF is defined as a system's energy input in horsepower divided by its refrigeration capacity in TR. Both CoP and PF can be applied to either the entire system or to system components.

III. RESULTS

Simulation results are presented here for the performance of the cooling system. The effect of the variation of the inlet generator heating water temperature is shown in The cooling capacity varies approximately linearly starting from a low value of 0.47 kW up to 16 kW. The COP rises from a low value of 0.82 to reach a constant value of 0.94. The cooling capacity increases as the inlet generator temperature increases. The COP of the system increases slightly when the heat source temperature increases.

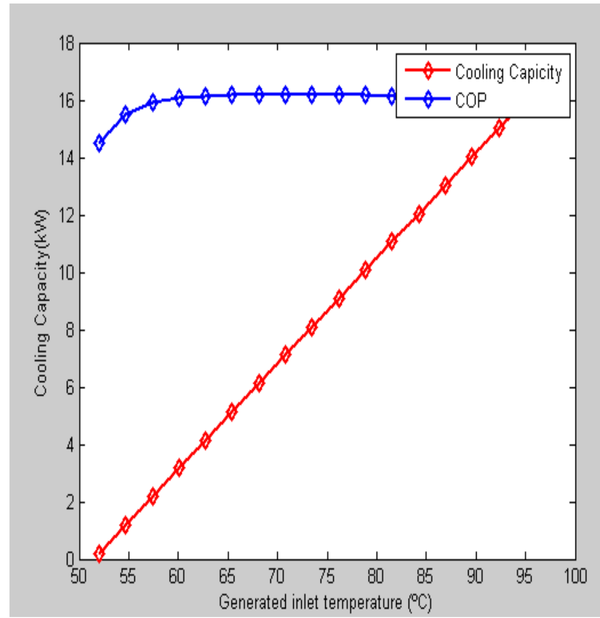

Fig. 6. Effect of generator inlet temperature on cooling capacity and COP.

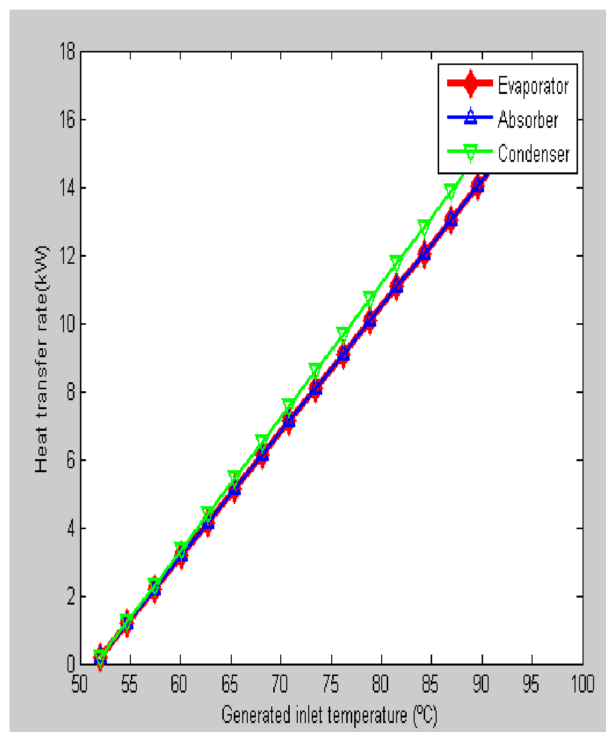

Fig. 7. Effect of generator inlet temperature on evaporator, absorber, condenser and generator heat transfer rates.

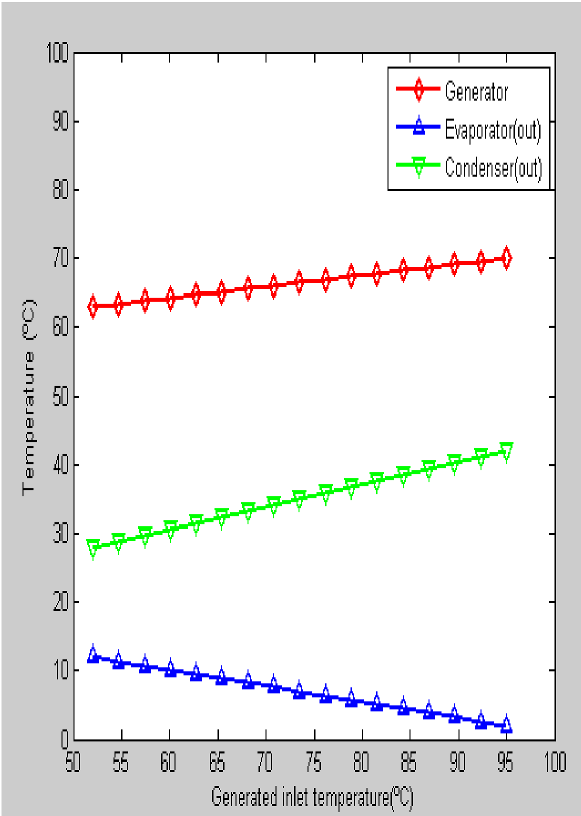

Fig. 8. Effect of generator inlet temperature on generator, evaporator and condensertemperatures.

IV. CONCLUSIONS

A supermarket refrigeration system which is performed on a timed basis consumes excess electrical energy. Defrosting of display cases also disturbs the temperature control of the case resulting in temperatures which exceed the design temperature over a significant time period. The energy consumption during the defrost process can be reduced using more advanced defrost initiation and termination techniques based on demand rather than timed defrost. Although a number of different demand defrost strategies have been proposed in the past, none has found wide acceptance in the food retail refrigeration industry due to poor reliability and high capital cost.

REFERENCES

[1]. Stoecker, W.F., 1957, "How Frost Formation on Coils affects Refrigeration Systems", *Refrigerating Engineering,* **65**(2), 55.

[2]. Heinzen, R.A., 1988, "How adaptive Defrost maintains Refrigeration System efficiency", Australian Refrigeration, Air conditioning and Heating, 12.

[3]. Ciricillo, S.F., 1985, "Heat pump de-icing/controlling for energy conservation and costs", Clima 2000, *Congress on heating, ventilation and air conditioning, Copenhagen, 87.*

[4]. Muller, E.D., 1975, "A new concept for defrosting refrigeration plants", *Kalte*, **28**(2), 52.

[5]. Llewelyn, 1984, "A significant advance in defrost control", *Int. J Refrigeration*, **7**(5), 334.

[6]. Woodley, C.B.C, Aug 1989, "Saving on the defrost", Air conditioning and Refrigeration News, 62.

[7]. Paone, N., and Rossi, G., 1991, "Fiber-optic ice sensors for refrigerators", *Proc of SPIE*, 1511, 129.

[8]. Datta D., Tassou S. A. and Marriott D., "Demand Defrost of Supermarket Refrigeration System", Proc, CIBSE National Conference, Eastbourne, 1995.

[9]. O'Neal, D.L., and Tree, D. R., 1985, "A review of Frost Formation in Simple Geometries", *ASHRAE Trans,* **91**(2B), 267.

[10] ASHRAE Handbook of Fundamentals, ASHRAE, Atlanta, USA, 1997.

[11] Wilbur, P.J., and Mitchell, C.E., "Solar absorption air conditioning alternatives", *Solar Energy*, vol. **17**, no. 3, pp. 193-199, 1975.

[12] Li, Z.F., and Sumathy, K., "Simulation of a solar absorption air conditioning system", *Energy Conversion and Management*, vol. **42**, no. 3, pp. 313-327, 2001.

[13] Li, Z.F., and Sumathy, K., "Experimental studies on a solar powered air
conditioning system with partitioned hot water storage tank", Solar Energy, vol.**71**, no. 5, pp. 285-297, 2001.

[14] Florides, G.A., Kalogirou, S.A., Tassou, S.A., and Wrobel, L.C., "Modelling and
simulation of an absorption solar cooling system for Cyprus", Solar Energy, vol.**72**, no. 1, pp. 43-51, 2002

[15] Atmaca, I., and Yigit, A., "Simulation of solar-powered absorption coolingsystem", *Renewable Energy*, vol. **28**, no. 8, pp.1277-1293, 2003.

[16] Florides, G.A., Kalogirou, S.A., Tassou, S.A., and Wrobel, L.C., "Design and construction of Li Br-water absorption machine", *Energy Conversion & Management*, vol. 44, no. 15, pp. 2483-2508, 2003.

[17] Assilzadeh, F., Kalogirou, S.A., Ali, Y., and Sopian, K., "Simulation and optimization of a LiBr solar absorption cooling system with evacuated tube collectors", *Renewable Energy*, vol. **30**, no. 8, pp. 1143-1159, 2005.

[18] Mittal, V., Kasana, K.S., and Thakur, N.S., "Performance evaluation of solarabsorption cooling system of Bahal (Hatyana)", *Journal of the Indian Institute of Science*, vol. **85**, pp. 295-305, 2005.

[19] Sayegh, M.A., "The solar contribution to air conditioning systems for residential buildings", Desalination, vol. **209**, vol. 1-3, pp. 171-176, 2007.

[20] Balghouthi, M., Chahbani, M.H., and Guizani, A., "Feasibility of solar absorptionair condition in Tunisia", Building and Environment, vol. **43**, no. 9, pp. 1459-1470, 2008.72

[21] Howell, R.H., Van, N.Q., and Smith, C.E., "Heat and moisture transfer through turbulent recirculated plane air curtains", *ASHRAE Transactions*, vol. **82,** no. 2, pp. 191-205, 1976.

[22]. Mateus, T., and Oliveira, A.C., “Energy and economic analysis of an integrated solar absorption cooling and heating system in different building types andclimates”, *Applied Energy*, vol. **86**, no. 6, pp. 949-957, 2009.

[23]. Howell, R.H., and Adams, P.A., “Effects of indoor space conditions on refrigerated display case performance”, Final report, ASHRAE-596 RP, 1991.

[24]. Howell, R.H., “Effects of store relative humidity on refrigerated display case performance”, ASHRAE Transactions, vol. **99**, no. 1, pp. 667-678, 1993.

[25]. Howell, R.H., “Calculation of humidity effects on energy requirements ofrefrigerated display cases”, *ASHRAE Transactions*, vol. **99**, no. 1, pp. 679-693,1993.

[26]. Tassou, S.A., and Datta, D., “Influence of supermarket environmental parameterson the frosting and defrosting of vertical multideck display cabinets”, *ASHRAE Transactions,* vol. **105**, no. 1, pp. 491-496, 1999.

[27]. Orphelin, M., Marchio, D., and D’Alanzo, S.L., “Are there optimum temperatureand humidity set points for supermarkets?”, ASHRAE Transactions, vol. 105, no.1, pp. 497-507, 1999.

[28] Rosario, L., and Howell, R.H., “Relative humidity and temperature measurement and predictions in supermarket”, *ASHRAE Transactions*, vol. **107**, no. 2, pp. 415-423, 2001.

[29] Kosar, D., and Dumitrescu, O., “Humidity effects on supermarket refrigerated case energy performance: a database review”, *ASHRAE Transactions*, vol. **111**, no. 1, pp. 1051-1060, 2005.

[30] Chen, Y.-G., and Yuan, X.-L., “Experimental study of the performance of singlebandair curtains for a multi-deck refrigerated display cabinet”, *Journal of Food Engineering,* vol. **69**, no. 3, pp. 261-267, 2005.

[31] Smale, N.J., Moureh, J., and Cortella, G., “A review of numerical models of airflow in refrigerated food applications”, *International Journal of Refrigeration,* vol. **29**, no. 6, pp. 911-930, 2006.

[32] Getu, H.M., and Bansal, P.K., “Modeling and performance analyses of evaporators in frozen-food supermarket display cabinets at low temperatures”, *International Journal of Refrigeration*, vol. **30**, no. 7, pp. 1227-1243, 2007.

[33] Ge, Y.T., Tassou, S.A., and Hadawey, A., “Simulation of multi-deck medium temperature display cabinets with the integration of CFD and cooling coilmodels”, *Applied Energy*, vol. **87**, no. 10, pp. 3178-3188, 2010.