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Energy Analysis of Various Supermarket Refrigeration Systems

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ABSTRACT

Modeling and analysis work was done on various supermarket refrigeration systems for their energy efficiency, TEWI (Total Equivalent Warming Impact), and annual operating cost. The systems which were modeled in detail include parallel racks, distributed, self-contained, glycol secondary loop, and CO_2 secondary loop (medium temperature) and cascade (low temperature). Based on R404A modeling results, distributed systems with scroll compressors have energy usage 6 to 9% lower than the baseline parallel rack system. On the other hand, self-contained units with horizontal scroll compressors and water-cooled condensers have energy consumption 11% higher than parallel racks, and glycol fluid secondary loop systems have energy consumption 15% higher than parallel racks. CO_2 secondary loop/cascade systems with propane as primary refrigerant have energy consumption comparable to parallel racks. The CO_2 systems also have low TEWI.

1. INTRODUCTION

Supermarkets are one of the 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 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. 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. Figure 1 shows a diagram of 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 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 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, self-contained, glycol secondary loop, and CO_2 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 store and close-coupled to the display case lineups or storage rooms they serve. With this approach, both the machine room and the long lengths of piping needed to connect the cases with large remote compressor racks are eliminated. The advanced self-contained system 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), easy and low-cost installation, flexibility in time to order and remodeling. However, the self-contained system has some inherent disadvantages including high equipment cost and low efficiency due to heat transfer penalty of water-cooled condensing. A single-phase secondary loop system employs one or more chillers to refrigerate a secondary fluid (generally glycol/water

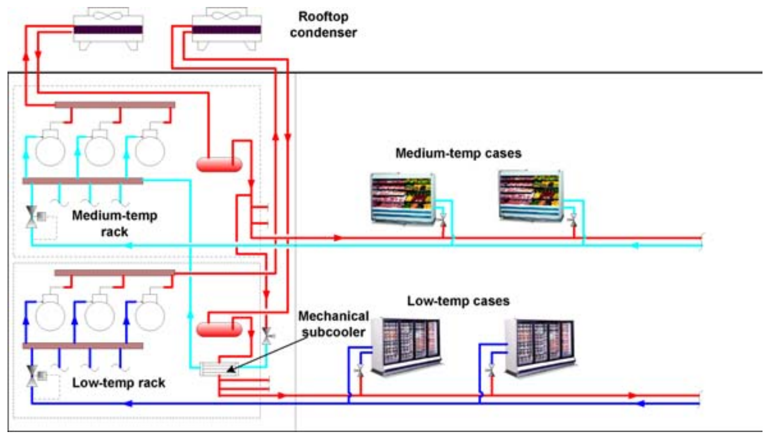

Figure 1: Diagram of parallel rack refrigeration system

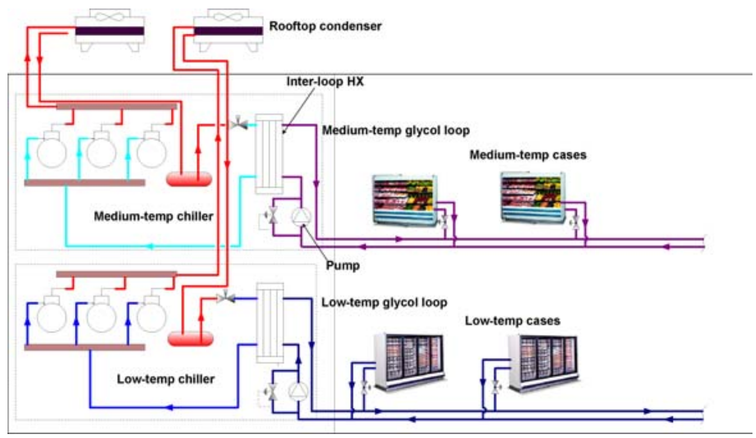

Figure 2: Diagram of single-phase secondary loop system

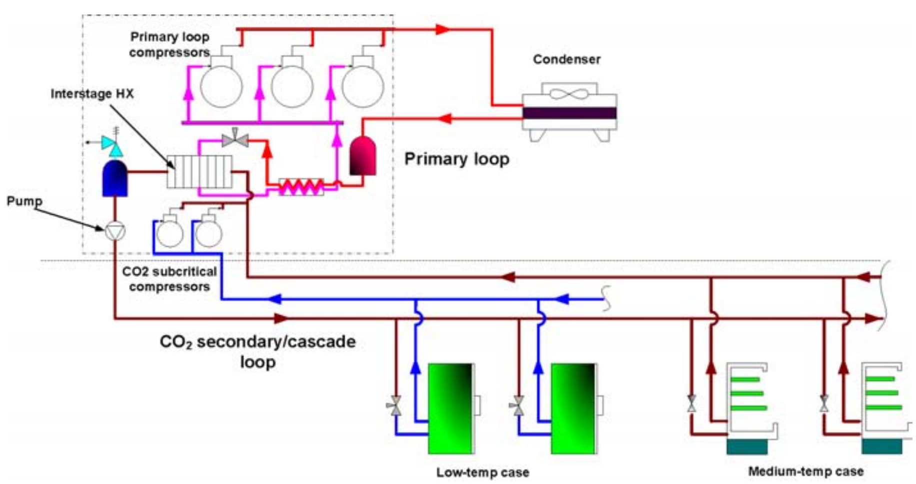

Figure 3: Diagram of CO2 secondary/cascade system

mixture) that is then pumped to the display cases and storage rooms. Figure 2 shows the elements of a secondary loop approach. In this example, the chillers, similar in configuration to multiplex compressor racks, provide chilled secondary fluid through an inter-loop heat exchanger. Generally the secondary loop glycol system is expected to have higher energy consumption due to the additional heat transfer process needed to cool the secondary fluid and the power needed to operate the secondary refrigerant circulation pump(s). To solve the problems with high pumping power, low energy efficiency, and large pipe diameters with single-phase secondary loops, systems of secondary fluid with phase change such as liquid CO_2 have been developed and drawn much attention. For medium temperature applications, liquid CO2 is pumped to display case evaporators where it is evaporated partially, and then leaves evaporators as two-phase mixture back to the inter-loop heat exchanger. For low temperature application, the same design (liquid CO_2 with pump circulation) or CO_2 cascade can be used. For the cascade design, the low stage is a CO_2 DX system with subcritical CO2 compressors. A flow diagram of a CO_2 secondary loop (medium temperature)/cascade (low temperature) system is shown in Figure 3. The primary loop refrigerant can be HFC, ammonia, or propane.

It is valuable to do modeling of these supermarket refrigeration systems to compare their annual energy consumption, TEWI, and life cycle cost. In fact some work has been done in this filed. Walker and Baxter (2003) reported their modeling of annual energy consumption and lifecycle cost for several types of supermarket refrigeration systems using only HFC refrigerants. Arias and Lundqvist (2005) simulated supermarket energy usage covering both HVAC and refrigeration systems, but it seemed that their program is a general tool without focusing on specific characteristics of the refrigeration systems. The present work intended to model and analyze all the refrigeration systems described above. The goal was to get our insight on various refrigeration systems and various refrigerants based on the input data or assumptions that we thought are realistic. The analysis results can help choose the most suitable refrigeration systems, and can be a basis for development of new alternative refrigeration technologies.

2. MODELING WORK

2.1 Supermarkets Modeled

Two typical supermarkets, one with 2800 square meters and the other with 5100 square meters were chosen for the analysis work. The refrigeration schedule of the stores defined the connected refrigerated fixtures, made up of display cases and walk-in storage coolers, and gave rated discharge air temperatures, evaporating temperatures, and refrigeration loads at design indoor conditions (24^{o}C, 55% RH). The refrigeration loads were then assigned to the rack, distributed, and secondary loop systems, respectively so a refrigeration system description was formulated for each type of refrigeration. The resulting system configuration information for the 2800 square meter store is shown in Tables 1 and 2 for the rack and distributed systems, respectively. The refrigeration loads were assigned to the compressor rack or sub-system based on their location and evaporator temperature. The SST (Suction Saturation Temperature) of each compressor or sub-system was close to, but lower than, the lowest evaporating temperature in the group of display cases.

Table 1: Rack refrigeration system configuration for 2800 square meter store

	SST ^oC (^oF)	Design Refrigerated Load KW (Btu/h)
(Medium-temp) Rack A	-7.2 (19)	149.9 (511,390)
(Low-temp) Rack B	-31.7 (-25)	54.3 (185,260)

Table 2: Distributed refrigeration system configuration for 2800 square meter store

	SST ^oC (^oF)	Design Refrigerated Load KW (Btu/h)
(Medium-temp) Unit A	-8.9 (16)	65.6 (223,680)
Unit B	-7.2 (19)	89.9 (307,060)
(Low-temp) Unit D	-31.7 (-25)	24.0 (82,040)
Unit E	-24.4 (-12)	21.9 (74,820)

The secondary loop system was assumed to have the same number of units and refrigeration load as the rack system (two units for the 2800 square meter store - one for medium and the other for low temperature).

Three representative geographical locations, St. Louis, MO, Boston, MA, and Dallas, TX, were compared by using the corresponding ASHRAE hourly temperature distribution data for annual energy calculation.

2.2 Modeling Method

A computer program, CoolPack developed by the Technical University of Denmark, was used to calculate energy efficiency under a specific operation condition. The CoolPack is especially suitable for supermarket refrigeration analysis for which we have well-defined component performance and need to catch the main system characteristics while neglecting some details. The energy efficiency under different operating conditions from the CoolPack was exported to an Excel spreadsheet where weather bin data, fan/pump data, and add-in functions were used to calculate the entire system annual energy consumption and TEWI. In addition, several EES (Engineering Equation Solver) programs were developed to evaluate the effect of heat transfer and pressure drop on system performance.

2.3 System or Component Characteristics Related to Modeling Work

2.3.1 Refrigeration load factor: When ambient temperature deviates from the design point, the indoor air temperature and relative humidity will change and this will cause refrigeration loads to deviate from the design loads defined by the store refrigeration schedule. Walker and Baxter (2003) proposed the following load factor to address this issue:

$$Load\_factor = \left(1-(1-\min)\frac{(85-T_{amb})}{(85-40)}\right) \tag{1}$$

Where min is the minimum fraction of design load (0.66 for medium temperature and 0.8 for low temperature), and T_{amb} is ambient dry-bulb temperature (^oF).

2.3.2 Compressor performance: Isentropic efficiency was used to define compressor performance. The data of isentropic efficiency for typical compressor types was obtained from both compressor manufacturers and internal testing, and were used to develop a correlation of isentropic efficiency vs. compression ratio.

2.3.3 Condenser and condensing temperature: There are two types of rooftop condensers in supermarket refrigeration: air-cooled, and evaporative. The air-cooled condenser is most common because it requires the least maintenance and operates reliably. Evaporative condensers are used in some supermarkets, primarily in drier climates where a substantial difference in dry-bulb and wet-bulb temperature exists. Evaporative condensers can operate at a lower condensing temperature. However, water treatment and consumption and related cost are major issues that prevent widespread use of evaporative condensers in supermarkets. In the present work, air-cooled condensers were used for all systems except the self-contained system to get consistent results and fair comparison of various systems. As for self-contained units (one compressor per display case), water-cooled condensers were used, and warm water/glycol was pumped to air-cooled fluid coolers on rooftop to reject heat.

Air-cooled condensers are sized based on 5.6^{o}C (10^{o}F) TD for low temperature systems, and 8.3^{o}C (15^{o}F) TD for medium temperature systems, where TD is the difference between condensing temperature and ambient dry-bulb temperature.

When ambient temperature is very low in winter, condenser fans are cycled on/off to maintain condensing temperature above 21^{o}C (70^{o}F), except for the condensing temperature of the low temperature distributed system (with scroll compressors) which is maintained to a minimum 10^{o}C (50^{o}F).

2.3.4 Evaporating temperature and saturated suction temperature: The refrigeration schedule of the stores defines evaporating temperature for each display case and walk-in cooler. When a group of display cases or walk-in coolers are assigned to a multi-compressor refrigeration system, the saturated suction temperature (SST) of the system is lower than the lowest evaporating temperature in the group.

2.3.5 Parasitic losses: For the rack system using R404A, the suction line is sized to have pressure drop equivalent to 1.7°C (3°F) for low temperature units and 1.1°C (2°F) for medium temperature units. The distributed system has lower suction line pressure drop because of shorter pipe length.

2.3.6 Secondary loop related issues: *Air cooler*: For the glycol secondary loop system (propylene glycol/water for medium temperature and potassium formate/water for low temperature), airflow in each display case is cooled by a fluid cooler, rather than an evaporator. The temperature rise of the glycol in the air cooler was chosen to be 7°F and fluid flow rate was based on this.

Inter-loop heat exchanger: For secondary loop systems, an intermediate heat exchanger is needed to provide cooling for the secondary fluid. This intermediate heat exchanger is the evaporator for the primary loop and condenser (for CO_2) or fluid cooler (for single phase fluid) for the secondary loop fluid. A 5°F temperature difference between the primary loop evaporating temperature and the secondary fluid leaving temperature was used for single phase fluid while a 7°F difference between the primary loop evaporating temperature and CO_2 saturation temperature was used for the CO_2 secondary loop. These values result in a good balance between energy efficiency and intermediate heat exchange size.

3. RESULTS AND DISCUSSION

3.1 Energy Consumption and Operating Cost

All the results presented here are for the 2800 square meter store. The results and conclusions for the 5100 square meters store are similar, except the larger store's rack units are larger and their suction manifolds are grouped based on SST and thus the rack system has relatively higher energy efficiency and lower cost. Figure 4 and Table 3 show comparison of annual energy consumption for the refrigeration systems analyzed. One can see from Figure 4 and Table 3 that the trend of relative energy consumption level is the same for the three locations (St. Louis, Boston, and Dallas). Thus only the data for St. Louis area are given in the next several tables. The rack system is considered the baseline, since it is the most commonly configuration.

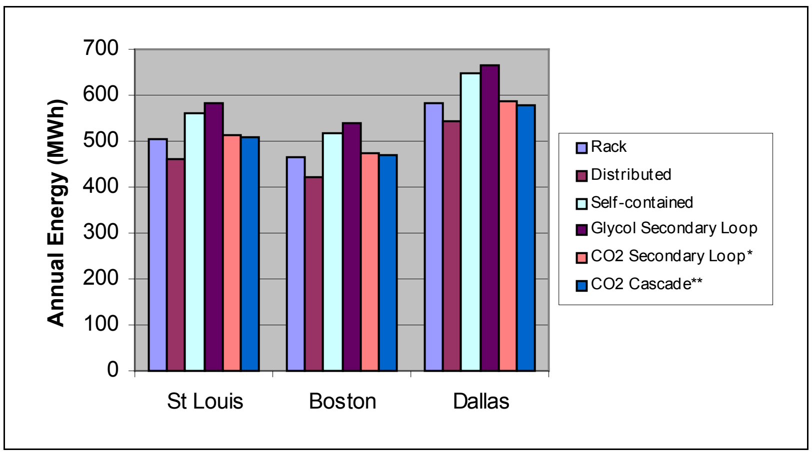

Figure 4: Annual energy consumption of refrigeration systems analyzed

Table 3: Annual energy consumption (MWh) of refrigeration systems analyzed

	Rack	Distributed	Self-contained	Glycol Secondary Loop	CO_2 Secondary Loop*	CO_2 Cascade**
St Louis	506	462	561	584	514	511
Boston	465	420	516	540	476	474
Dallas	582	543	646	665	587	582

Notes:

1. Energy consumption includes that of compressors, condenser fans, and pumps

(evaporator fans and case lighting not included).

2. Except for the CO_2 secondary loop and cascade system using propane as primary refrigerant, all other systems, including the primary loop of the glycol secondary system, use R-404A refrigerant.
3. *: "CO_2 secondary loop" means CO_2 secondary loop for both medium temperature and low temperature
4. **: "CO_2 cascade" means CO_2 secondary loop for medium temperature and CO_2 cascade for low temperature.

As shown in Figure 4 and Table 3, the distributed system with scroll compressors has energy use about 6 to 9% lower than the baseline rack system. This results from the factors that the compressors of the distributed system are closer to the display cases causing lower parasitic losses, and saturated suction temperature (SST) employed for each distributed unit can closely match the evaporator temperature of the display cases.

The calculated energy consumption of the self-contained system with scroll compressors and the glycol secondary loop system is significantly higher than the baseline rack system (the self-contained system is about 11% higher and the glycol system is about 15% higher). The high energy consumption of the self-contained system can be attributed to the additional temperature difference of the water-cooling condensers and the low efficiency of horizontal scroll compressors. The low efficiency of the glycol secondary loop system can be attributed to the temperature difference of the inter-loop heat exchangers and energy consumption by the glycol pumps. Table 4 shows the breakdown of energy consumption for various systems. The energy of the glycol pumps is approximately 7% of the compressor energy. It is worthy to note that some other reports stated brine secondary loop systems could be more efficient than rack systems. For example, work by Faramarzi and Walker (2004) concluded a state-of-the-art glycol secondary loop system was about 4.9% more efficient than a baseline rack system. They attributed the energy saving to three features:

- The use of multiple, parallel brine pumps and low viscosity Dynalene organic salt-water significantly reduced pump energy consumption;
- Subcooling from warm brine defrost provided energy saving;
- The display case heat exchangers were re-designed, which were larger than DX refrigerant evaporators, and were better suited for use with secondary fluid

In addition, the parasitic loss of the baseline rack system in their work was 2.3 to 3.3^oC (4 to 6^oF), compared to 1.1 to 1.7^oC (2 to 3^oF) in the present work.

Table 4: Breakdown of annual energy consumption (MWh)

Systems	Compressors	Secondary Loop Pumps	Condenser/Fluid Cooler Fans	Total
Rack	448	N/A	58	506
Distributed	408	N/A	54	462
Self-contained	503	N/A	58	561
Glycol Secondary Loop	489	36.4	58	584
CO2 Secondary Loop	453	3.5	58	514
CO2 Cascade	448	1.5	58	507

Both CO_2 secondary loop and cascade systems have energy consumption comparable to rack systems. Since the primary loop or high stage of the two CO_2 systems used propane as refrigerant, the results show we can develop a totally "green" system which will have energy efficiency comparable to a R404A rack system. The good energy efficiency of the CO_2 systems compared to the glycol system is due to the very low energy consumption of the secondary loop pump (see Table 4) and the higher cycle efficiency of propane compared to R404A.

In this study, we assumed an equal evaporating temperature of CO_2 (in CO2 secondary loop systems) and R404A (in rack or other DX systems) for the same product temperature of the display cases. However, the CO_2 evaporating temperature may be 1.5 to 2^oC (2.7 to 3.6^oF) higher due to its better heat transfer/pressure drop performance and improved air temperature and frost distribution. Thus, CO_2 secondary loop and cascade systems may be more efficient than R404A direct expansion. If the use of propane or ammonia as the primary loop refrigerant is not practical in North America, one can consider using R404A as primary loop refrigerant, and the combination of R404A/CO2 can be as efficient as R404A DX rack system because the higher CO2 evaporating temperature and elimination of parasitic loss can overcome the loss at the inter-loop heat exchanger.

For low temperature applications (frozen food or ice cream), the energy efficiency of the CO_2 secondary loop system is close to the CO_2 cascade system unless the evaporating temperature is extremely low (-35^{o}F or lower). The CO_2 secondary loop system is simpler than CO_2 cascade because the former does not have CO_2 compressors and is lubricant free. These factors make the CO_2 secondary loop system attractive even for low temperature applications. The disadvantages of the CO_2 secondary loop compared to CO_2 cascade for low temperature include higher CO_2 mass flow rate and cooling loss on the CO_2 return line, and larger primary loop compressors.

It is worthy to note that the size of CO_2 pipes for CO_2 secondary loop and cascade systems is only about half of R404A for DX systems. This is good for containing high pressure of CO_2, and offers potential for cost reduction.

Table 5 compares annual operating cost for all the systems analyzed. The operating cost includes energy and refrigerant cost. As expected, distributed units had about 12.7% lower annual operating cost than racks. Even though their refrigerant cost is much lower, the glycol system and self-contained system have respectively 8% and 3% higher total annual operating costs than the baseline due to significantly higher energy costs. The CO_2 secondary loop or cascade system had approximately 6% lower operating cost than the rack baseline. This can be completely attributed to lower refrigerant cost.

Table 5: Annual operating cost of various refrigeration systems

System	Annual Energy (MWh)	Refrigerant	Charge (lb)	Leak rate	Cost ($)			Cost saving over Rack ($)	% Saving over rack
					Energy	Refrigerant	Total		
Rack	506	R404A	2,500	0.15	30,363	2,329	32,691	0	0
Distributed	462	R404A	1,300	0.10	27,719	807	28,527	4,165	12.7
Self-contained	561	R404A	300	0.01	33,672	19	33,691	-999	-3.1
Glycol secondary loop	584	R404A	500	0.04	35,016	124	35,391	-2,699	-8.3
		Propylene /Potassium	5,000	0.10		250			
CO2 secondary loop	514	Propane	500	0.04	30,837	10	30,947	1,744	5.3
		CO2	2,000	0.10		100			
CO2 cascade	507	Propane	500	0.04	30,423	10	30,533	2,159	6.6
		CO2	2,000	0.10		100			

Note: Cost calculation is based on energy rate of $0.06/kWh, R404A refrigerant cost $6.21/lb, and other fluids $0.50/lb.

3.2 TEWI

Figure 5 shows the annual TEWI (Total Equivalent Warming Impact) for various systems. Table 6 explains the details of TEWI calculation.

Table 6: TEWI (Total Equivalent Warming Impact) of various refrigeration systems

System	Refrigerant	Charge (lb)	Leak	GWP/kg	Annual Energy (kWh)	TEWI (kg of CO_2)		
						Direct	Indirect*	Total
Rack	R404A	2,500	0.15	3,874	506,044	658,967	328,929	987,896
Distributed	R404A	1,300	0.10	3,874	461,987	228,442	300,292	528,734
Self-contained	R404A	300	0.01	3,874	561,200	5,272	364,780	370,052
Glycol secondary loop	R404A	500	0.04	3,874	583,608	35,145	379,345	414,490
	Propylene /Potassium formate		0.1	0		0		
CO2 secondary loop	Propane	500	0.04	20	513,954	181	334,070	334,320
	CO2	1,500	0.1	1		68		
CO2 cascade	Propane	500	0.04	20	507,043	181	329,578	329,827
	CO2	1,500	0.1	1		68		

*: Based on conversion factor of 0.65 kg CO_2/kWh.

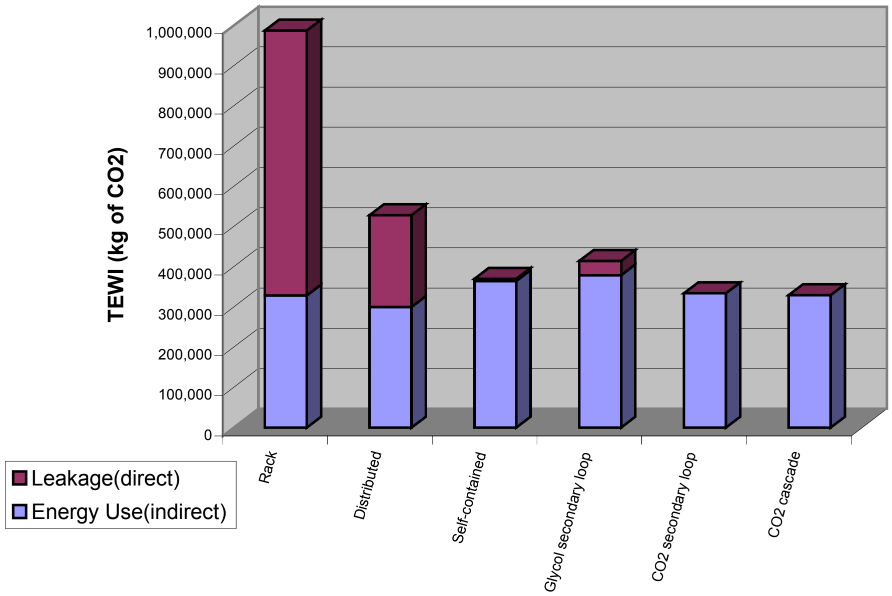

Figure 5: TEWI (Total Equivalent Warming Impact) of various refrigeration systems

The conventional rack system has the highest TEWI due to high direct contribution by refrigerant leakage. With the leak rate of 15%, almost 70% of the TEWI for the rack system is from direct contribution by refrigerant leakage. Distributed systems have significantly lower TEWI than the rack system because of smaller system size, less refrigerant charge, and shorter pipes causing less refrigerant leakage. As expected, secondary loop systems, including glycol and CO_2, have low TEWI.

4. CONCLUSIONS

Existing and new supermarket refrigeration systems were modeled and analyzed for their energy efficiency, TEWI, and cost. The modeling work was done using CoolPack, Excel spreadsheet, and EES programs. Based on modeling for representative supermarkets, distributed systems have energy usage 6 to 9% lower than the baseline rack system. CO_2 secondary loop/cascade systems with propane as primary refrigerant have energy consumption comparable to R-404A parallel racks. The CO_2 systems have the lowest TEWI.

REFERENCES

Arias, J., Lundqvist, P., 2005, Modeling Supermarkets Energy Usage, *Proc. Vicenza Conf.*, IIR: p. 109-116.

Faramarzi, R.T., Walker, D.H., 2004, Investigation of Secondary Loop Supermarket Refrigeration Systems, *report prepared for California Energy Commission.*

Walker, D. H., Baxter, V.D, 2003, Analysis of advanced, low-charge refrigeration for supermarkets, *ASHRAE Transactions*, v 109 PART 1: p 285-292.