Secondary Coolant Systems for Supermarkets

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The recent developments in supermarket refrigeration systems reflect the factors shaping the supermarket industry: severe competition, small profit margin, high energy cost, high refrigerant price, regulatory pressures, and public perception/image. Secondary coolant technology has evolved in the last decade as the most reliable solution to these factors mainly through refrigerant charge reduction, refrigerant leak elimination, maintenance simplification, and product quality improvement.

New advanced designs and operational features are applied for energy parity with the traditional centralized direct-expansion system. These features include lower floating condensing pressure, deeper liquid subcooling, lower vapor superheat and pressure drop in the refrigerant return lines, simpler oil management, and reduced or eliminated oil accumulation in the heat exchangers/coils.

Additional benefits of secondary coolant systems are improved product quality and reduced shrink in fresh foods, opportunities to use more efficient and environment-friendly refrigerants, and to reduce demand and dependence on qualified technicians during installation and operation.

The environmental impacts of refrigerants leaked into the atmosphere, such as ozone depletion, resulted in global and local environmental regulations that were unprecedented two decades ago in the supermarket industry. Additionally, unknown or potentially negative impacts have raised public awareness to the extent that new regulations are constantly introduced while existing regulations are becoming more restrictive.

The no venting rule under the U.S. Clean Air Act currently applies to CFC and HCFC refrigerants, *and their substitutes*, including HFCs. The maximum annual refrigerant leak rate of 35% for adding refrigerants into the supermarket refrigeration systems containing CFCs and/or HCFCs without triggering leak repair or system retrofit requirements appears to be more obsolete and unacceptably high even in the views of the supermarket industry. A strong belief exists that HFC refrigerants should be included as well. The conditions for stricter regulations on the leak rate in U.S. supermarkets are in place, and it is a matter of time for more stringent limitations on refrigerant leak rate to be imposed. Outside of the U.S. the awareness of a potentially detrimental impact of the halogenated hydrocarbons leaked into the atmosphere has resulted in regulations discouraging or even banning their use.

The higher production cost of the HFC chemicals led to refrigerant prices that are an order of magnitude higher than prices paid for CFCs and HCFCs

Secondary coolant systems help improve product quality and reduce shrink in fresh foods.

in the not too distant past. The impact on the installed cost of a new refrigeration system with a charge of 3,000 lb (1400 kg) can be estimated at around $30,000 per store including labor. An annual refrigerant leak of 33% in the same store will add about $10,000 to the yearly operating cost. For an industry operating with a profit margin of 3% to 4%, neither the incremental installed cost nor the operating cost increase is acceptable.

These two circumstances contributed largely to the revival of the secondary coolant systems (SCS) more than a decade ago. However, it took nearly 10 years for design engineers to perfect these systems to the extent that they can compete with the well-established centralized direct-expansion (DX) refrigeration systems. Past experiences with secondary systems have been that the indirect refrigeration systems are 30% more expensive and consume 30% more energy.[1] These numbers reflected the poor

thermophysical properties of the limited selection of secondary coolants (brines) and poor initial design practices applied to the first installations. Broadening the selection with better secondary coolants based on water solutions of organic salts[2] combined with the advanced engineering practice developed in the last decade, positioned the secondary coolant technology to successfully compete with the traditional DX systems in terms of both installed cost and energy consumption.[3] From an environmental point of view, however, the SCS are superior to the DX systems and are the only known technology that has a potential to provide zero-leak supermarket refrigeration systems.

SCS: Low-Charge and Zero-Leak

The statistical data from the supermarket industry indicates that the major occurrence of refrigerant leaks is in the thousands of feet of pipe and hundreds of joints in the field-installed distribution piping, including liquid refrigerant supply lines, refrigerant vapor return lines, and hot-gas supply lines in the instances of hot-gas defrost.[4] Analysis of the causes leads to the following conclusions. The field installations are performed by contractors who cannot guarantee consistent qualification of all of their employees. In addition, few of them can afford a regular training program and periodic skill testing. In many instances, the piping is done under difficult conditions with limited availability of sophisticated tools to facilitate the work and improve the quality. A third adverse circumstance is that most of the piping is in locations that are not easily accessible for identifying/locating and repairing leaks. The unavoidable disruption in store operation is one of the most undesirable factors in the search for, and repair of, leaks.

On the other hand, refrigerant leaks can be eliminated completely in the rest of the refrigeration system, which is much more complicated, contains most of the major components, and has an order of magnitude more connections. And, if and when a leak occurs, it is much easier to be located and repaired without disruption of the store operation because of the better accessibility of the equipment.

Figure 1 illustrates the reduction in the number of leaky systems over a period of five years in a U.S. factory specializing in manufacturing supermarket refrigeration systems and experienced with more than 400 secondary systems in North

About the Authors

Georgi S. Kazachki, Ph.D., is director of research and development, refrigeration systems division, and **David K. Hinde**, is project manager, research and development at Hill Phoenix in Covington, Ga.
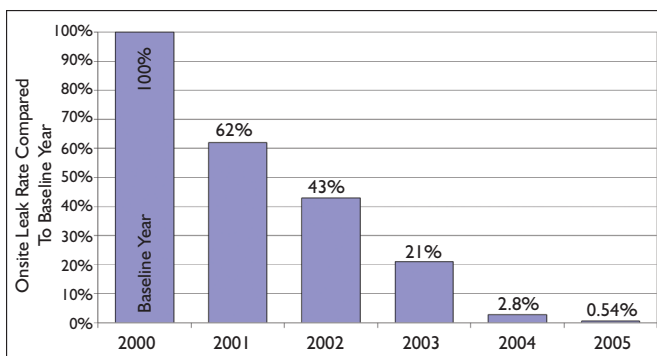
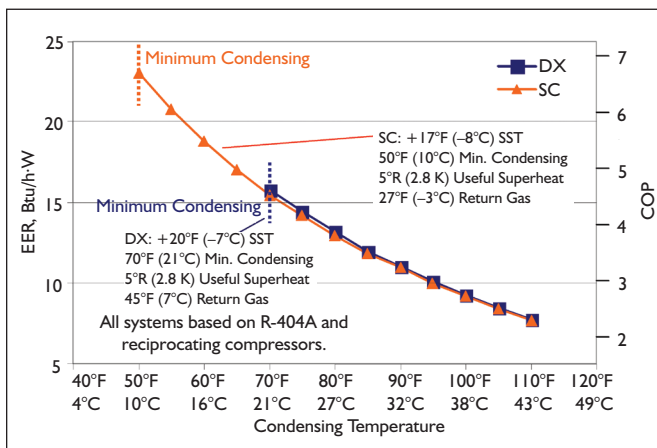Figure 1: Effects of leak reduction techniques applied in factory.

Figure 2: Direct expansion vs. secondary coolant piping arrangement.

Figure 3: Medium-temperature compressor energy.

Figure 4: Low-temperature compressor energy.

America. The phenomenal reduction of leaks was due to measures implemented in the factory: stringent quality control of purchased components and the manufacturing process; worker training and periodic recertification; provision of specialized tools for better and more ergonomic approach to the work piece; development of a better and more technological manufacturing process; and use of more sensitive methods for leak detection. Although in theory, these leak reduction methods could be similarly applied to the distribution piping in the field, this would be cost-prohibitive in practice and there is no indication that contractors who perform these installations are moving in such a direction.

The situation with the leaks in the distribution piping and in the rest of the system leads in a logical way to the concept of the secondary coolant system (*Figure 2*). The refrigerant is contained in the portion of the system that can be manufactured leak-free in a controlled factory environment and is easily accessible for monitoring and maintenance since the refrigerant-containing piping is entirely located in a machine room or mechanical center. A potential leak can be quickly and easily located and repaired. The leak occurrence identification and signaling can be automated and converted into a service call.

The leak-prone, field-installed portion of the system is transformed into a low-pressure hydronic system, which is much easier to install and service and much less likely to leak. This portion of the system can be installed using plastic piping, which is a trend that has quickly gained gr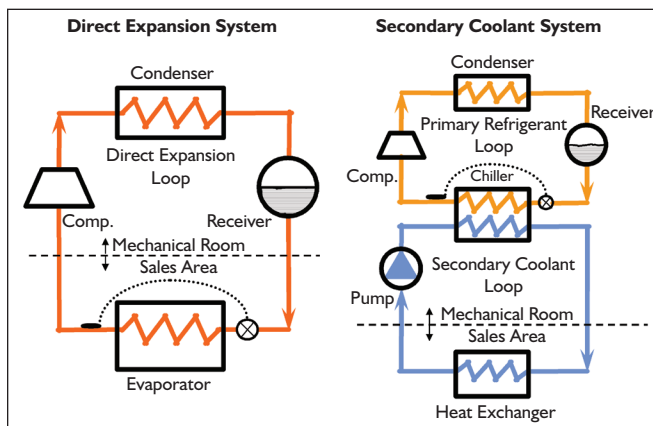ound in the last few years because of the simple, easy, and faster installation at reduced installed cost compared to the same piping made out of copper.

An additional benefit of the secondary coolant arrangement is the reduction in refrigerant charge. In a 60,000 ft² (5500 m²) supermarket, the refrigerant charge in a centralized DX refrigeration system with loop piping averages 1,700 lb (770 kg) while a secondary coolant system has only 700 lb (320 kg) or less of refrigerant. A major portion of this refrigerant is located in the condenser, which is typically air-cooled or sometimes evaporative for this type of system. If water-cooled condensers are implemented, further significant reductions in refrigerant charge can be accomplished.

Factors Affecting System Energy Efficiency

Secondary coolant systems have two unique features that differentiate them from centralized and distributed DX refrigeration systems: short liquid refrigerant supply lines and short vapor refrigerant return lines. The associated advantages of these features are:

- Reduced heat exchange between the refrigerant and ambient;
- Negligible pressure drop in the liquid supply and vapor return lines;
- Controlled single-digit vapor superheat;
- Excellent oil return;

System Type	LT System Energy	MT System Energy	Combined Total	
			System Energy	Compared to DX
Atlanta Results				
DX With 50°F (10°C) Liquid	334,808 kWh/year	566,798 kWh/year	901,606 kWh/year	—
SC With 50°F (10°C) Liquid	331,818 kWh/year	586,625 kWh/year	897,469 kWh/year	−0.5%
SC With 40°F (4°C) Liquid	316,133 kWh/year	594,293 kWh/year	889,453 kWh/year	−1.3%
SC With 30°F (−1°C) Liquid	301,117 kWh/year	601,573 kWh/year	881,717 kWh/year	−2.2%
Boston Results				
DX With 50°F (10°C) Liquid	325,988 kWh/year	517,603 kWh/year	843,591 kWh/year	—
SC With 50°F (10°C) Liquid	308,631 kWh/year	500,268 kWh/year	787,926 kWh/year	−6.6%
SC With 40°F (4°C) Liquid	296,150 kWh/year	506,175 kWh/year	781,352 kWh/year	−7.4%
SC With 30°F (−1°C) Liquid	282,192 kWh/year	512,814 kWh/year	774,033 kWh/year	−8.2%
Los Angeles Results				
DX With 50°F (10°C) Liquid	324,922 kWh/year	517,387 kWh/year	842,309 kWh/year	—
SC With 50°F (10°C) Liquid	331,384 kWh/year	555,404 kWh/year	865,815 kWh/year	2.8%
SC With 40°F (4°C) Liquid	314,379 kWh/year	563,460 kWh/year	856,866 kWh/year	1.7%
SC With 30°F (−1°C) Liquid	299,872 kWh/year	570,583 kWh/year	849,482 kWh/year	0.9%

DX System: MT with R-404A at 20°F (−7°C) SST, 70°F (21°C) minimum condensing, 5°R (2.8 K) useful superheat, 45°F (7°C) return gas, LT with R-404A at −20°F (−29°C) SST, 70°F (21°C) minimum condensing, 15°R (8 K) useful superheat (w/SLHE), 45°F (7°C) return gas, 50°F (10°C) liquid from mechanical subcooling.
SC System: MT with R-404A at 17°F (−8°C) SST, 50°F (10°C) minimum condensing, 5°R (2.8 K) useful superheat, 27°F (−3°C) return gas, secondary system using 30% propylene glycol with 5°R (2.8 K) chiller approach, 7°R (3.9 K) fluid temperature change, 70 ft (23 kPa) design pump head. LT with R-404A at −23°F (−31°C) SST, 40°F (4°C) minimum condensing, 5°R (2.8 K) useful superheat, −13°F (−25°C) return gas, varying liquid temperature from mechanical subcooling as indicated. Secondary system using Dynalene HC-30 with 5°R (2.8 K) chiller approach, 7°R (3.9 K) fluid temperature change, 70 ft (23 kPa) design pump head.

Table 1: Total annual system energy consumption for DX vs. SCS at selected locations.

- Elimination of oil trapping in the evaporators; and
- Improved heat transfer in the coils.

On the other hand, secondary coolant systems have two characteristics associated with energy penalties: the presence of circulation pumps for the secondary fluid and the presence of intermediate heat exchangers. The circulation pumps draw energy in addition to the energy used in the DX systems, and place a portion of this energy into the system in the form of increased refrigeration load. The intermediate heat exchangers introduce an additional temperature step between the refrigerant and the secondary coolant.[5] The balance between the advantages and penalties associated with the SCS is key to their success. Good design practice is required to take full advantage of the benefits of the SCS and to reduce and offset the energy penalties. The practical aspects of the advantages will be discussed in the order they were listed.

The heat exchange between the liquid refrigerant supply lines and ambient can have either a positive or a negative role on the performance of the DX system. When the temperature around the pipes is lower than the refrigerant temperature, for instance, when the outdoor temperature and the related condensation temperatures are high and the refrigerant supply lines are passing through air-conditioned space, the resulting subcooling increases the refrigeration capacity and the energy efficiency ratio (EER) of the system.

An additional benefit is the stable and reliable operation of the thermostatic expansion valves (TXV) when subcooled

liquid feeds into them. The opposite phenomena occur when the temperature around the liquid refrigerant piping (even when insulated) is higher than the refrigerant temperature. This occurs during hot days and intensive sun when large portions of the liquid lines are run in unconditioned space or above/on the roof. The liquid refrigerant can reach its boiling point and a mixture of liquid and vapor enters the TXVs, resulting in an unstable operation of the expansion valves, starvation of the coils, and performance and efficiency degradation of the whole system. The heat transfer between the ambient and the long liquid lines in DX systems places a limitation on how low the condensing temperature can float when low outdoor ambient conditions are present. The pressure drops in the long supply lines further exacerbate the situation.

While the liquid refrigerant supply distribution lines in a DX system may consist of thousands of feet of pipe, the same lines in a SCS are only a few feet to a few tens of feet.[5] The reduced heat transfer through the short insulated pipe lines and the reduced pressure drop allow taking full advantage of the floating condensing pressure. The limitations in this case come from the compressor operational envelopes. Within a specific geographic area, using floating condensing pressure as low as the ambient allows significantly improves EER of the compressors and reduces the yearly energy consumption of SCS compared to DX systems. In addition, SCS are well-suited for use of electronic expansion valves (EEVs) that can operate over a wider range of condensing

pressures without need for seasonal readjustment. Although DX systems can also implement EEVs, their use in these systems is generally cost-prohibitive due to the large number of required valves and associated electronics.

Figure 3 illustrates the EER of a medium-temperature (MT) reciprocating compressor in a secondary and a DX system as a function of the condensing temperature. The current practice in the supermarket industry is to float the condensing pressure in DX refrigeration systems down to 70°F (21°C) to secure adequate pressure and subcooling at the inlet of the expansion valves. Because of the short liquid lines, the condensing pressure in the MT secondary systems can be floated down to 50°F (10°C) without concern about improper feed of the TXVs.

This results in an increase of the EER from approximately 16 Btu/h·W (COP of 4.7) to 23 Btu/h·W (COP of 6.7), which amounts to an increase of more than 40%. This increase will be realized only at ambient temperatures providing for condensing temperatures below 70°F (21°C), which

limits this advantage to certain climatic conditions and for a certain portion of the year. It is also worth noting that lower condensing temperatures are possible in DX systems if thicker insulation is applied on the liquid supply line. However, this increases the installed cost for materials and labor although the potential requirement for seasonal readjustment of the TXVs, and the potential for warming of the liquid refrigerant up to its boiling point still remains.

The advantages of the SCS are even more apparent in the low-temperature (LT) systems where the liquid refrigerant is subcooled by the MT system (referred to as mechanical subcooling) with resulting increase in the cooling capacity and EER. Common design practice in state-of-the-art DX systems is to use the MT system to mechanically subcool the refrigerant liquid in the LT system to 50°F (10°C). In addition to allowing lower floating condensing pressures, the SCS can achieve a deeper subcooling of the LT liquid refrigerant to levels lower than 50°F (10°C), which are considered impractical in the DX systems due to the extensive distribution piping network.
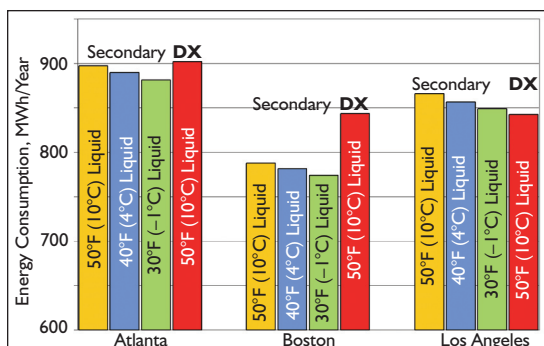

Figure 5: Total annual system energy consumption for DX vs. SCS at selected locations.

The subcooling in the SCS can take advantage of the lowest suction saturation temperature (SST) available in the MT system. Thus, if the lowest MT SST is 20°F (–7°C), the liquid refrigerant in the LT system can be subcooled down to 25°F (–3.9°C). *Figure 4* demonstrates the impact of the floating condensing pressure and the deeper subcooling of the liquid refrigerant on EER, which can be improved by up to 50% compared to the LT DX system. The lower condensing temperature improves the compressor EER while the lower subcooling improves the EER of the system. The same qualifier as with the MT system applies for the LT system. The floating condensing pressure follows the ambient temperature and, therefore, is dependent on the geographic/climatic conditions and the time of the year. One also can argue that the same floating condensing and the same subcooling can be applied in DX systems. The limitations are determined again by the heat gains into the liquid refrigerant and by the low pressure in the liquid lines leading to boiling of the refrigerant, unstable operation of the TXVs, and inefficient performance of the system.

The short return lines in SCS with negligible pressure drops result in higher capacity and EER of the MT and LT compressors, compared with DX systems, which are commonly designed for pressure drop equating to a 2°R (1.1 K) change in equivalent saturation temperature.[6] In addition, the limited heat exchange between the vapor and ambient in the short return lines leads to low refrigerant vapor superheat at the compressor inlet, causing further improvements to compressor capacity and EER.[5] It can be argued that the secondary coolant supply and return heat gains can have higher heat transfer rates to ambient than in DX systems. These can be mitigated, if not eliminated, by using plastic piping materials, which have lower thermal conductivity than copper piping, increased insulation thickness, and better insulating materials. The cost can be covered from the savings in installation time associated with plastic piping.
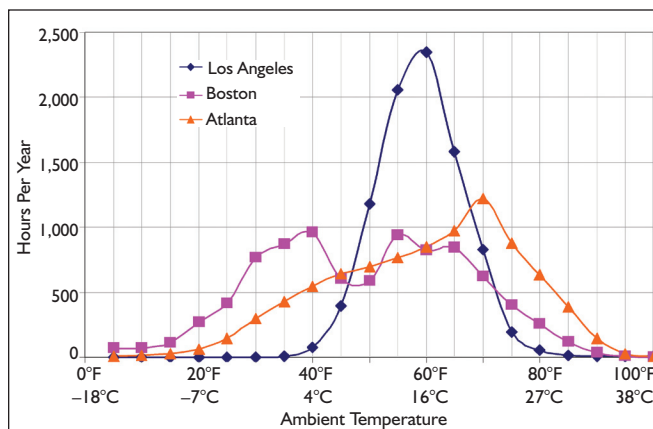
Figure 6: Ambient temperature bin data for selected locations.

An important benefit of the low refrigerant vapor superheat is that since the process starts from a lower inlet temperature, the isentropic compression ends at a lower outlet temperature. This allows application of more efficient refrigerants without or with a limited use of liquid injection, which provides for a primary cycle with a higher EER. (The thermodynamic analysis of refrigerants' performance in a vapor compression cycle shows that refrigerants providing for high cycle and system EER have inherently low throttling losses and high discharge temperatures, e.g., R-22, ammonia, R-410A.) The separation of the primary refrigeration system from the sales area also allows the potential use of refrigerants that were previously unsuitable for DX systems (ammonia or hydrocarbons).

An important feature of the SCS is the limited length of the oil circulation loop and the simplified oil return at a negligible pressure drop. Not only oil presence but also oil piling in the coils that often occurs in DX systems is eliminated, and the heat transfer coefficient is correspondingly improved. Oil traps located in front of risers also are eliminated in the field piping.

Because the coils are flooded with the SC, the whole heat transfer area is used and the desired discharge air temperature can be achieved with a higher SC supply temperature than the evaporating temperature in the DX systems. However, in common design practice and for the analysis done in this study, the secondary coolant supply temperature is assumed to be the same as the DX evaporation temperature. The automatic selection of lower SC supply temperatures compared to DX evaporating temperatures is a common design misconception disproved by more than 10 years of laboratory testing and hundreds of field installations.

Energy Comparison of DX vs. SC

A comparison was performed to better understand the effects of the DX and SC system characteristics on annual system energy efficiency. The compound effects of floating condensing pressure and the deeper mechanical subcooling on energy consumption have a significant impact on this comparison and the results are illustrated in *Figure 5* and *Table 1* for three different climatic conditions: Atlanta, Boston, and Los Angeles. *Figure 6* represents the number of hours per year for each temperature in each location.

The benefits of the SCS are optimal at the conditions in the Northeast U.S. where the number of hours with low ambient conditions is the largest. As a result, the annual energy consumption of the SCS is lower than the annual energy consumption of the DX system by 6.6% to 8.2% depending on the subcooling level.

We need to point out that the deeper subcooling is associated with additional expenses for a larger subcooler, and additional MT rack capacity partially offset by the reduced LT rack capacity. The advantages of the SCS at the Atlanta climatic conditions are

balanced to the extent that its annual energy consumption is only 0.5% to 2.2% lower than the annual energy consumption of the DX system. The energy comparison in the Los Angeles area has exactly the opposite results: the energy consumption of the SCS is 0.9% to 2.8% higher than the energy consumption of the DX system. Since the annual energy difference between SCS and DX system in both Atlanta and Los Angeles conditions are within ±3%, for practical purposes we can consider the two systems to be at parity.

This analysis demonstrates that the benefits of the short liquid lines with the reduced heat exchange between the refrigerant and ambient and with the negligible pressure drop are enough to offset the circulation pumps and the additional temperature difference in the intermediate heat exchangers to the extent that the SCS can be in parity or even more energy efficient than DX systems on an annual basis. In fact, this situation has been observed in the field.

Secondary Coolant Loop Design Considerations

An error with the most adverse consequences on SCS energy efficiency is the selection of the secondary coolant temperature change in the heat exchangers. A number of secondary coolant systems developed in the early periods of design experience attempted to approximate the refrigerant temperature profile in DX coils, i.e., to run the coils with a minimum SC temperature change. The associated SC flow rates led to disastrous results with dual negative impacts. First, large circulation pumps consumed an excessive amount of electricity and second, most of the energy was input into the system requiring more compressors, condensers, energy, etc., to compensate. The magnitude of the error is best illustrated with the correlation between the pump power and the SC temperature change shown in Equation 1.

$$\frac{P_A}{P_B} = \left(\frac{\Delta T_B}{\Delta T_A}\right)^3$$

<div align="right">1</div>

where
P = pumping power (at design condition A or B)
ΔT = fluid temperature change in heat exchangers (at design condition A or B)

Equation 1 implies that if the pump size in a certain SCS is 5 hp (3.7 kW) at 6°R (3.3 K) SC temperature change, the required pump size in the same system at 2°R (1.1 K) SC temperature change will be $5 \times 3^3 = 135$ hp (101 kW). Considering the temperature profile on the airside of the coil with a temperature difference of 10°R to 12°R (5.6 K to 6.7 K), it becomes clear that a SC temperature difference of 6°R or even 8°F vs. 2°R (3.3 K or even 4.4 K vs. 1.1 K) is acceptable and desirable. The reduced mean logarithmic temperature difference in the coils can be offset to a large extent, if not completely by, the larger effective internal heat transfer surface with the SC vs. DX and by the higher overall heat transfer coefficient.

The role of the proper selection of the secondary coolant temperature difference is illustrated in *Figure 7*. The pump energy as a percent of the total system energy in a SCS is shown as a function of the SC temperature change at three pump heads determined by the pressure drops in the system. A SC temperature difference of 7°R (3.9 K) appears to be a good choice resulting in pump energy accounting for 5% of the system energy. A temperature difference of 8°R (4.4 K) reduces the pump energy to 2.5% and may be well justified if the desired discharge air temperature doesn't require a substantial decrease in the SC supply temperature. One degree lower temperature difference, i.e., 6°R (3.3 K) may be justified for certain display cases but not for the entire system since it increases the pumping power to between 5% and 10%. It becomes clear then that temperature differences of 3°R (1.7 K) or even 4°R (2.2 K) are a poor design practice. In the past, an SC temperature difference of 5°R (2.8 K) was proposed as optimal, however, the percentage of the pump energy at this design condition clearly indicates how detrimental such temperature difference can be for the system energy efficiency. Nonetheless, these temperature differences were common, leading to the misleading conclusions about the efficiency of secondary systems.

Closely related to the energy efficiency of the SCS is the choice of proper secondary coolant from the standpoint of the material and concentration. Proper coolant selection is critical to pumping energy of the system, and design and selection of the heat exchangers. The following Equation 2 developed in 1996 provides for a comparative analysis of two secondary coolants (Fluids A and B) without a phase change based on the characteristic fluid properties.[7]

$$\frac{P_A}{P_B} = \left(\frac{\rho_A}{\rho_B}\right)^{1.8} \left(\frac{\upsilon_A}{\upsilon_B}\right)^{0.2} \left(\frac{c_{pA}}{c_{pB}}\right)^{-2.8}$$

2

where
- P = pumping power
- ρ = density
- υ = kinematic viscosity
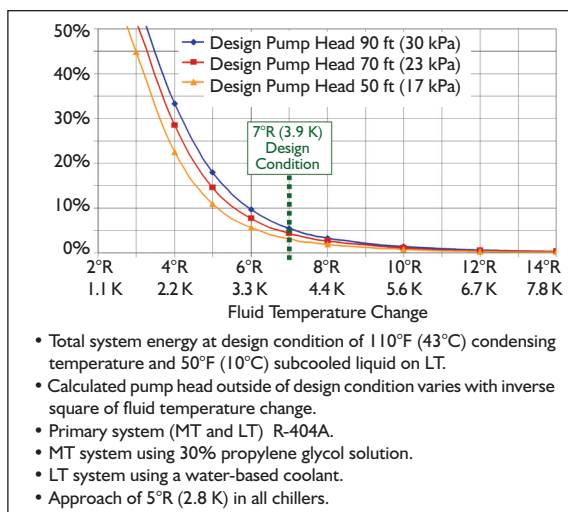- c_p = specific heat

This equation provides a useful screening tool for eliminating fluids that are not appropriate for secondary systems. Combining the most suitable fluids with the recent and more sophisticated heat exchanger modeling programs led to heat exchanger designs with the desired high fluid temperature differences and resulting benefits to system energy consumption.

- Total system energy at design condition of 110°F (43°C) condensing temperature and 50°F (10°C) subcooled liquid on LT.
- Calculated pump head outside of design condition varies with inverse square of fluid temperature change.
- Primary system (MT and LT) R-404A.
- MT system using 30% propylene glycol solution.
- LT system using a water-based coolant.
- Approach of 5°R (2.8 K) in all chillers.

Figure 7: Pump energy consumption vs. fluid temperature change at varying design pump head.

Conclusions

Operational and design characteristics of secondary coolant system have been detailed:
- An analysis of the most critical aspects of secondary coolant systems was performed in comparison to direct-expansion systems;
- Commonly known, nonproprietary, manufacturing practices have been presented that allow significant reduction of leaks in manufactured equipment;
- An energy analysis was performed showing that secondary coolant systems achieve energy parity or better compared with direct expansion systems and depending on climate conditions; and
- A method of evaluating the efficiency of secondary fluids was presented.

Proper application of the presented criteria will lead to secondary coolant systems that are equally or more efficient than traditional direct expansion systems with the benefits of superior environmental performance long associated with indirect systems. Secondary coolant systems are the only technology to substantially reduce refrigerant charge and to achieve the potential of a zero-leak supermarket refrigeration system. The analysis of the environmental and energy-efficiency aspects of the secondary systems shows that these systems are the best available technology to meet the broad supermarket requirements and to successfully replace direct-expansion systems.

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