Analysis of Refrigerator / Freezer Appliances Having Dual Refrigeration Cycles

André I. Gan

Sanford A. Klein, Ph.D.
Fellow ASHRAE

Douglas T. Reindl, Ph.D., P.E.
Member ASHRAE

ABSTRACT

This paper investigates the benefits of using two separate refrigeration cycles to meet demands for both the freezer and fresh food compartments in domestic refrigerators. The energy savings that can be obtained by delivering refrigeration at the higher temperature as required by the fresh food compartment is found to be a function of the cabinet load ratio (defined as the ratio of the fresh food to the freezer cabinet loads) and the ratio of the freezer and refrigerator cycle COPs. Depending on the values of these two parameters, the energy requirement for a dual-cycle system can be up to 30% lower than that for a comparable single-cycle† system meeting the same cabinet loads. This energy-saving approach can help manufacturers meet the Department of Energy's year 2001 energy use standards for domestic refrigerators. The dual-cycle system also offers the advantages of reduced defrost (not included in the previously mentioned energy savings estimate) and the ability to maintain higher humidity conditions in the fresh food compartment. The feasibility of using the fresh food compartment as a sink to subcool liquid refrigerant prior to its entry into the freezer capillary tube was also investigated. The advantage of the subcooler was found to be most significant at low cabinet load ratios in dual cycles for which suction-line heat exchangers were not present. When high-effectiveness suction-line heat exchangers in the fresh food and the freezer cycles were employed, a maximum reduction of 3% in total electrical power requirements at a cabinet load ratio of 1.0 resulted from the indirect mechanical subcooling modification.*

INTRODUCTION

In most domestic refrigerators, refrigeration for both the fresh food and freezer compartments is provided by a single vapor-compression cycle that operates at the freezer evaporating saturation temperature. Although there are capital cost and space advantages to using a single refrigeration cycle, the overall performance of the refrigerator is reduced because the coefficient of performance (COP) for production of refrigeration at the freezer temperature is lower than that for production of refrigeration at the fresh food evaporator conditions. For this reason, some energy savings can be expected if two separate cycles are used to meet the respective cooling loads for the freezer and fresh food cabinets. The extent of the energy savings depends on the relative cabinet loads. A larger increase in system performance is expected if the majority of the cooling is provided to meet the fresh food compartment load.

With a single refrigeration cycle, refrigeration demands for the fresh food compartment are met by exchanging air between the freezer and fresh food compartments. However, the dew-point temperature of the air in the freezer is approximately equal to the freezer temperature so that the humidity of the freezer air supplied to the fresh food compartment is quite low. Such low-humidity air is undesirable in the fresh food compartment because it rapidly desiccates stored foods. In addition, the water extracted from the fresh food compartment must be removed from the freezer evaporator by a

* Stand-alone independent refrigeration cycles for both the fresh food compartment and the frozen food compartment.

† A single refrigeration cycle that provides refrigeration to both the freezer and the fresh food compartments while operating at the freezer temperature.

André I. Gan is a design engineer for the Amana Company, Amana, Iowa. **Sanford A. Klein** is a professor in the Solar Energy Laboratory at the University of Wisconsin, Madison, and **Douglas T. Reindl** is an assistant professor and director of the HVAC&R Center at the University of Wisconsin.

defrost cycle, which further reduces the overall energy efficiency of single-cycle systems.

Peak refrigeration demands in a domestic refrigerator are quite small, approximately 150 W. When two separate refrigeration cycles are used, the required steady-state capacity of each compressor is necessarily reduced. Smaller capacity compressors tend to operate less efficiently due to increased frictional losses attributable to increased compressor surface area to volume ratio. Using oversized compressors also leads to a performance penalty as a result of short cycle run times, which increases the number of cycles per day and cycling losses. In addition to the extra cost and machine compartment space required to locate an additional compressor, the use of two smaller, less efficient compressors may erode the inherent thermodynamic advantage of providing refrigeration at a higher temperature for the fresh food compartment. This paper explores these performance trade-offs.

The use of two cycles to separately provide refrigeration to the fresh food and freezer compartments also presents the opportunity to utilize a form of mechanical subcooling to improve the performance of the low-temperature freezer compartment cycle. Mechanical subcooling relies on the operation of a second, higher temperature refrigeration cycle to subcool the high-pressure liquid refrigerant before it is throttled to the freezer temperature. Thornton et al. (1994) found a 10% improvement in overall COP in their study of mechanical subcooling for supermarket refrigeration systems. In a domestic refrigerator, the refrigeration cycle for the fresh food compartment could be used to directly subcool the condensate for the freezer cycle, thereby shifting some of the cooling load from the freezer to the fresh food cycle. Strictly speaking, both cycles have to operate simultaneously to allow mechanical subcooling by this approach. An alternative that accomplishes the same objective without the constraint of simultaneous cycle operation is to place the subcooling heat exchanger within the fresh food compartment. This arrangement allows the thermal capacity of the fresh food contents to provide short-term cycle-to-cycle energy storage for subcooling.

This paper begins by comparing the COP of a dual-cycle system to a corresponding single-cycle system. Parameters in the analysis include the distribution of cabinet loads and the compressor efficiency. In a related study, Bare (1992) found a 23% improvement in overall COP for a dual-cycle system using refrigerants R-142b and R-152a when the total cabinet loads are evenly distributed between the freezer and the fresh food compartments. The effects of suction-line heat exchangers for domestic refrigerators using refrigerant R-134a is studied. Finally, this paper investigates the potential performance benefits of indirect mechanical subcooling for a refrigerator that currently uses dual cycles with and without suction-line heat exchangers. Additional details are provided by Gan (1999).

COMPARISON OF SINGLE AND DUAL-CYCLE SYSTEMS

In a single-cycle system, both the fresh food and freezer cabinet loads are met by a single refrigeration cycle operating at the freezer temperatures. The steady-state power required to meet the loads (not considering defrost) can be represented as indicated in Equation 1.

$$Power_{1-cycle} = \frac{Load_{fz} + Load_{ff}}{\mathrm{COP}_{fz}} \qquad (1)$$

where

$Load_{ff}$ = fresh food cabinet load (W or Btu/h)

$Load_{fz}$ = freezer cabinet load (W or Btu/h)

COP_{fz} = COP of the freezer cycle

The primary motivation for considering a dual-cycle system is the thermodynamic advantage of providing refrigeration capacity to the fresh food compartment at a higher evaporating temperature. The ability of a dual-cycle system to enhance the performance depends on the operating characteristics of the two compressors and the ratio of the fresh food to the freezer cabinet loads. The steady-state power required for a dual-cycle system is shown in Equation 2.

$$Power_{2-cycle} = \frac{Load_{fz}}{\mathrm{COP}_{fz}}\left(1 + \frac{LR}{CR}\right) \qquad (2)$$

where

$$LR = \frac{Load_{ff}}{Load_{fz}} \qquad (3)$$

$$CR = \frac{\mathrm{COP}_{ff}}{\mathrm{COP}_{fz}} \qquad (4)$$

and COP_{ff} is the COP of the fresh food refrigeration cycle.

Equation 2 indicates that the steady-state power consumption of a dual-cycle system is a function of the cabinet loads and the COPs of the cycles that provide refrigeration to meet these loads. Since COP_{fz} is ordinarily lower than COP_{ff}, it is evident that, for the same loads, the total power consumption of the dual-cycle system should be lower than that for a single-cycle system. A figure of merit that can be used to compare the steady-state performance of single- and dual-cycle systems for the same cabinet loads is the power difference ratio DR, defined in Equation 5. The power difference ratio is seen to depend only on LR and CR. A plot showing contours for different values of DR as a function of LR and CR appears in Figure 1.

$$DR = \frac{Power_{1-cycle} - Power_{2-cycle}}{Power_{1-cycle}} = \frac{LR(1 - 1/\mathrm{CR})}{(1 + LR)} \qquad (5)$$

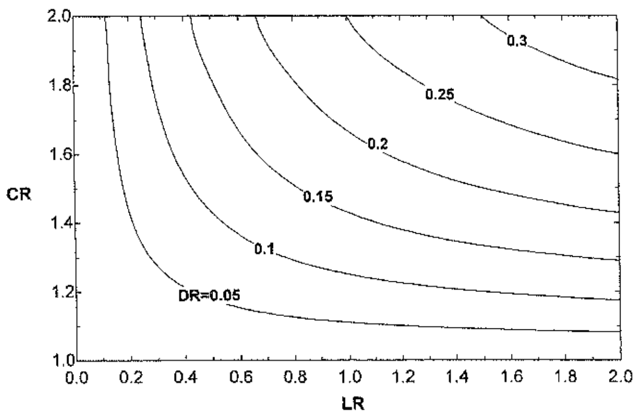

Figure 1 Power difference ratio (DR) as a function of COP ratio (CR) and load ratio (LR).

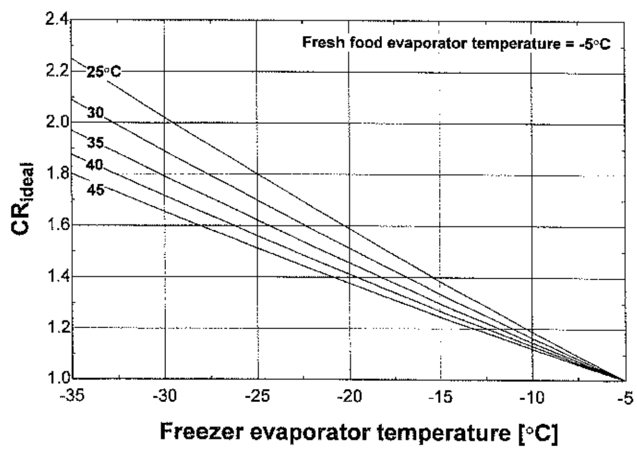

Figure 2 COP ratio based on Carnot COP for a range of condensing and evaporator freezer temperatures at a fresh food evaporator temperature of 265 K.

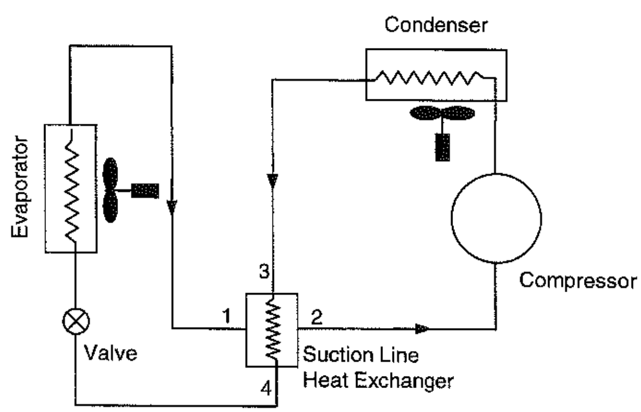

Figure 3 Schematic of a vapor compression cycle employing suction line heat exchanger.

Figure 1 indicates that there may be a significant performance advantage for dual-cycle systems depending upon the values of the load ratio and COP ratio. The load ratio for residential refrigerators depends on the relative cabinet size and use pattern, but it is independent of the system configuration. A simple estimate of the COP ratio is provided as the ratio of the Carnot COPs at the fresh food and freezer conditions given in Equation 6.

$$CR_{ideal} = \frac{T_{ff}/(T_{cond} - T_{ff})}{T_{fz}/(T_{cond} - T_{fz})} \tag{6}$$

A plot of CR_{ideal} for a fresh food evaporator temperature of –5°C (23°F) is shown in Figure 2 as a function of the condensing and freezer evaporator temperatures. During typical operation, the condensing temperature is approximately 35°C (95°F) while the freezer and refrigeration evaporator temperatures may be –20°C (–4°F) and –5°C (23°F), respectively. For these conditions, CR_{ideal} is 1.456. Assuming that the actual COP ratio is equal to the ideal value with a load ratio of unity, the dual-cycle system would require about 16% less power than the single-cycle system. The value of CR_{ideal} given in Equation 6 and Figure 2 applies to non-ideal compressors provided that the isentropic efficiencies of the freezer and refrigerator compressors are equal.

Actual values of the load ratio and COP ratio will vary. The best way to estimate the COP ratio would be to use compressor performance maps provided by a compressor manufacturer. As noted earlier, the smaller compressors used in a dual-cycle system may be expected to have a somewhat lower isentropic efficiency than the larger capacity compressor used in a single-cycle system, and this effect should be considered; however, there clearly is a strong incentive to use two cycles from a performance standpoint.

SUCTION-LINE HEAT EXCHANGER

Suction-line heat exchangers are provided in many refrigerators to exchange energy between the cool gaseous refrigerant leaving the evaporator and warm liquid refrigerant exiting the condenser. Suction-line heat exchangers can serve several purposes. Their primary function is to help ensure that refrigerant entering the compressor is fully vaporized and, in some cases, they may improve system performance. Figure 3 illustrates a simple vapor compression refrigeration cycle utilizing a suction-line heat exchanger.

The performance of the suction-line heat exchanger is quantified in terms of the heat exchanger effectiveness, defined in Equation 7.

$$\varepsilon = \frac{(T_2 - T_1)}{(T_3 - T_1)} = \frac{(T_{vapor,\,out} - T_{vapor,\,in})}{(T_{liquid,\,in} - T_{vapor,\,in})} \tag{7}$$

The effect of suction-line heat exchangers was recently studied by Klein et al. (1999). They show that the effect of a suction-line heat exchanger on the steady-state performance of a refrigeration cycle (neglecting the effects of pressure losses) can be quantified in terms of the relative capacity index, which is a function of the heat exchanger effectiveness and two additional parameters, as indicated in Equation 8.

$$RCI/\varepsilon = -3.0468 + 19.3484\,D - 19.091\,D^2 + 1.2094\,L + 0.02101\,L^2 - 5.9980\,DL - 0.002797\,DL^2 + 5.52865\,D^2L \quad (8)$$

where

$$RCI = \left(\frac{Capacity - Capacity_{no\ hx}}{Capacity_{no\ hx}}\right) 100\% \quad (9)$$

$$D = \Delta h_{vap}/(c_{p,L}\ T_c) \quad (10)$$

where

Capacity = the refrigeration capacity with a liquid-suction heat exchanger (W or Btu/h)

$Capacity_{no\ hx}$ = the refrigeration capacity for a system operating at the same condensing and evaporating temperatures without a liquid-suction heat exchanger (W or Btu/h)

Δh_{vap} = the refrigerant enthalpy of vaporization at the evaporator pressure (kJ/kg or Btu/lb_m)

$c_{p,L}$ = the specific heat of saturated liquid refrigerant at the evaporator temperature (kJ/kg·K or $Btu/lb_m \cdot R$)

T_c = the critical temperature of the refrigerant (K or R)

L = the temperature lift, i.e., the difference in saturated condensing and evaporating temperatures (K or R)

The parameter D is a dimensionless indicator of the ratio of the latent to sensible energy storage capacities for the refrigerants. For domestic refrigerators employing R-134a as the refrigerant, the dimensionless parameter D ranges between 0.40 at 0°C (32°F) to 0.46 at –30°C (–22°F). A plot of the relative capacity index as a function of evaporator and condenser saturation temperatures for refrigerant R-134a appears in Figure 4. In typical domestic refrigerator applications, the condensing temperature is 35°C (95°F) while the fresh food and freezer saturation temperatures may be –5°C (23°F) and –20°C (–4°F), respectively. At these conditions, the maximum increase in capacity resulting from the use of suction-line heat exchangers (effectiveness of unity and no pressure drop) is about 4% for the fresh food cycle and by about 8% for the freezer cycle, as shown by the symbols in Figure 4.

"MECHANICAL" SUBCOOLING

Mechanical subcooling uses a small separate refrigeration cycle to subcool, i.e., reduce the temperature of the liquid refrigerant exiting the condenser of the primary refrigeration cycle. Subcooling the refrigerant liquid before it is throttled increases its specific capacity (refrigeration capacity per unit refrigerant mass), resulting in a reduction in the required

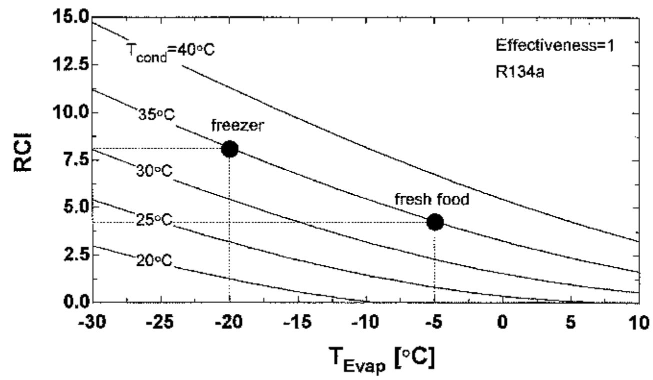

Figure 4 *Relative capacity index for a suction-line heat exchanger as a function of evaporator and condenser saturation temperatures for refrigerant R-134a.*

refrigerant flow rate and compressor power to meet a given refrigeration load. The smaller subcooling cycle requires additional power to sensibly cool the refrigerant in the primary cycle, but the subcooling cycle operates over a smaller temperature lift compared to the primary cycle and, consequently, at a higher COP. Ideally, the increase in capacity and COP resulting from subcooling the primary cycle more than compensates for the additional power needed to operate the smaller subcooling refrigeration cycle, resulting in a net increase in overall system COP.

A refrigerator that uses separate refrigeration cycles for the fresh food and freezer compartments already possesses one of the characteristics required for mechanical subcooling, i.e., two refrigeration cycles. It should be possible to use the fresh food cycle compartment as a means of providing subcooling for the freezer cycle. One advantageous effect of subcooling is that some of the work required by the freezer compressor is shifted to the fresh food refrigeration cycle. If the fresh food cycle operates a COP higher than the freezer cycle, this type of indirect mechanical subcooling will reduce the total power requirement assuming parasitic losses (additional pressure drops and fan power) are small.

In a strict sense, the fresh food and freezer refrigeration cycles have to operate simultaneously in order to implement a direct mechanical subcooler (since there must be direct heat exchange between the refrigerants in the two cycles). In practice, the fresh food and freezer refrigeration cycles must be allowed to operate independently since the fresh food and freezer cabinet refrigeration demands are usually not coincident. Since the run times with fixed-speed compressors differ for the two cycles, controlled simultaneous operation is not an option. Simultaneous operation of the two cycles can be avoided if energy storage is available. The thermal capacitance of the contents of the fresh food compartment provides energy storage, and this storage can be exploited to implement indirect mechanical subcooling as indicated in Figure 5.

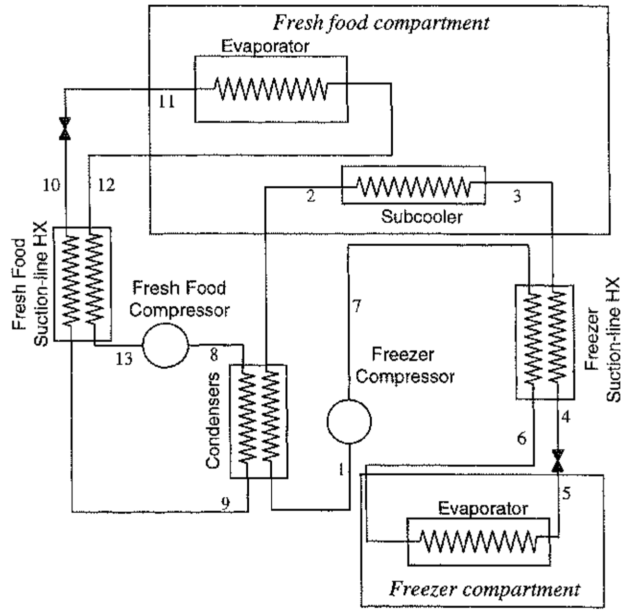

Figure 5 *A possible arrangement of freezer and fresh food cycles to provide mechanical subcooling utilizing energy storage in the fresh food compartment.*

In the configuration shown in Figure 5, the subcooler heat exchanger would be placed inside the fresh food compartment, physically piped in series with the evaporator. Air inside the fresh food compartment would then be circulated past the subcooler when either the fresh food or freezer cycle is operating. The energy transfer from the subcooler to the air would occur, thereby increasing the fresh food cabinet load. Thermostatic controls would activate the fresh food compressor when the temperature in the fresh food compartment exceeds a specified setpoint. Arrangement of the subcooler and evaporator in this manner will necessarily increase the required fan power since the run time and the pressure loss increase. The additional fan power is not considered in the following analysis.

Mechanical Subcooling Modeling

A simulation model was developed using a commercial equation-solving program (Klein and Alvarado 1998) to study the performance of refrigerators employing suction-line and indirect mechanical subcooling heat exchangers. The compressor mass flow rate and power are characterized using the semi-empirical model in Equations 11 and 12, as described by Jaehnig (1999).

$$\dot{m} = \left[1 + C - C\left(\frac{p_{dis}}{p_{suc}}\right)^{\frac{1}{n}}\right] \cdot \frac{V \cdot Speed}{v_{suc}} \tag{11}$$

$$Power \cdot \eta_{comb} = \dot{m} \cdot \frac{n}{n-1} \cdot p_{suc} \cdot v_{suc}\left[\left(\frac{p_{dis}}{p_{suc}}\right)^{\frac{n-1}{n}} - 1\right] \tag{12}$$

where

$\dot{m}$ = refrigerant flow rate (kg/s or lb_m/s)

C = a clearance volume ratio that is used as a fitting parameter to represent compressor data

p_{dis} = compressor discharge pressure that is approximately equal to the saturated condensing pressure (kPa or psia)

p_{suc} = compressor suction pressure that is represented as the product of the evaporator saturation pressure and the factor $(1\text{-}f_p)$ where f_p is determined by regressing compressor calorimeter data (kPa or psia)

V = compressor cylinder displacement volume (m^3 or ft^3)

Speed = compressor rotation speed such that the product of V and *Speed* provides the displacement rate

n = isentropic index for the refrigerant at the suction conditions

v_{suc} = specific volume of refrigerant at pressure p_{suc} and the compressor inlet temperature (m^3/kg or ft^3/lb_m)

η_{comb} = a combined efficiency factor for the electric motor and compressor that has been found (Jaehnig 1999) to be correlated to the evaporator pressure for small compressors as shown in Equation 13

$$\eta_{comb} = a + b/P_{evap} \tag{13}$$

In this feasibility study, the refrigerant was assumed to exit each evaporator as a saturated vapor (states 6 and 12) and saturated liquid at the condenser outlets (states 2 and 9). These assumptions eliminate the need to explicitly model the capillary tube and fluid inventory. The evaporators in both the fresh food and freezer compartments were modeled with quasi-steady mechanistic relations in Equations 14a and 14b.

$$\dot{Q}_{evap,ff} = (UA)_{ff}(T_{ff} - T_{11})$$
$$\dot{Q}_{evap,fz} = (UA)_{fz}(T_{fz} - T_5) \tag{14a, b}$$

where

$\dot{Q}_{evap,ff}$ and $\dot{Q}_{evap,fz}$ = respective evaporator heat transfer rates for the fresh food and freezer compartments, assuming quasi steady-state operation (W or Btu/h)

$(UA)_{ff}$ and $(UA)_{fz}$ = overall heat transfer coefficient-area products for the fresh food and freezer compartments, respectively (W/K or Btu/h·R)

T_{ff} and T_{fz} = average air temperatures in the fresh food and freezer compartments (K or R)

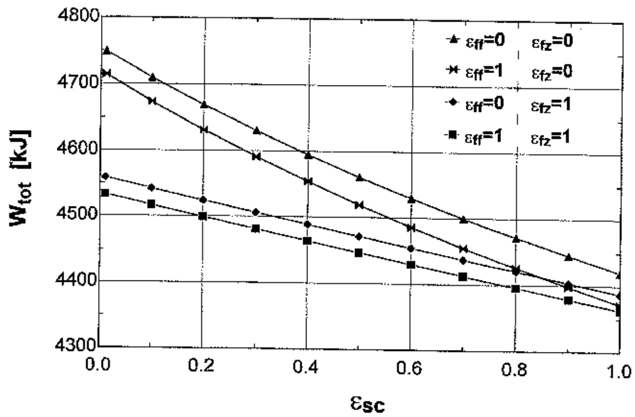

Figure 6 *Calculated work for a 24-hour period as a function of subcooler effectiveness with different combinations of suction-line heat exchangers.*

T_5 = saturated evaporator temperature for the frozen food compartment (K or R)

T_{11} = saturated evaporator temperature for the fresh food compartment (K or R)

The thermal performance of the subcooler heat exchanger was represented in terms of a heat exchanger effectiveness, defined as

$$\varepsilon_{SC} = \frac{(T_2 - T_3)}{(T_2 - T_{ff})}. \tag{15}$$

Fractional compressor operating times were calculated by determining the ratio of the refrigeration load to available capacity of the refrigeration cycle operating to meet the specified loads. System performance losses due to cycling were not considered. A direct figure of merit is the total compressor work (considering the fractional operating times) required to meet the fresh food and freezer loads over 24 hours of operation.

Mechanical Subcooling Modeling Results

Figure 6 shows the calculated 24-hour work (for both compressors) as a function of the subcooler effectiveness for four different combinations of suction-line heat exchanger effectivenesses. The assumed values of system parameters used in these calculations are provided in Table 1. The compressor parameters in Table 1 represent the operating characteristics of commercially available compressors used in domestic refrigerators. Figure 6 shows that, for all cases, the total compressor work is reduced as the effectiveness of the subcooler is increased. This behavior occurs because for the assumed compressor parameters, the COP of the fresh food cycle is higher than that of the freezer cycle. The physical size of the subcooler heat exchanger increases with increasing effectiveness. In addition, the necessary fan power and refrigerant pressure drop would also likely increase. These effects were not considered in the results shown in Figure 6.

TABLE 1
Assumed Values of System Parameters

Parameter	Fresh Food Cycle	Freezer Cycle
V	5.7 cm^3	7.0 cm^3
Speed	3500 rpm	3500 rpm
C	0.033	0.029
n	1.10	1.10
f_p	0.014	0.017
a	0.738	0.652
b	-17.1 kPa^{-1}	-13.3 kPa^{-1}
Cabinet air temperature	3.3°C	-15°C
Cabinet load	4320 kJ/day	6910 kJ/day
Evaporator heat transfer coefficient	0.090 kW/K	0.090 kW/K
Condensing temperature	35°C	35°C

The results in Figure 6 also show that for a refrigerator that does not have suction-line heat exchangers for either cycle, implementation of mechanical subcooling can decrease the total work by up to 7%. The extent of the improvement depends on the operating characteristics of the fresh food and freezer compressors and on the effectiveness of the subcooler heat exchange process. However, when suction-line heat exchangers are employed, the performance improvement resulting from mechanical subcooling is reduced. The presence of a suction-line heat exchanger in the fresh food cycle has little effect. When a suction-line heat exchanger with an effectiveness of unity is provided in the freezer cycle, the maximum effect of mechanical subcooling (with no penalty for additional pressure drop) is reduced from 7% to about 3.5% for these compressors.

It should be noted that Figure 6 provides estimates for the maximum effect of subcooling, assuming perfect heat transfer in the subcooler and no consideration of additional fan power and refrigerant pressure drops. The actual improvement would certainly be lower due to these effects. The reduction in the benefit of the subcooler occurs because the function provided by indirect mechanical subcooling is somewhat redundant to the function provided by the suction-line heat exchanger for the freezer refrigeration system. In both cases, the temperature of the liquid refrigerant exiting the condenser is reduced.

The results in Figure 6 were determined for fresh food and freezer cabinet loads of 4320 and 6910 kJ/day, respectively. These loads were chosen to represent a large modern appliance with side-by-side freezer and fresh food compartments. The effect of indirect mechanical subcooling was found to depend on the load ratio, LR, as defined in Equation 3. The effect of the load ratio was investigated by calculating the ratio of the daily total compressor work with a subcooler having an

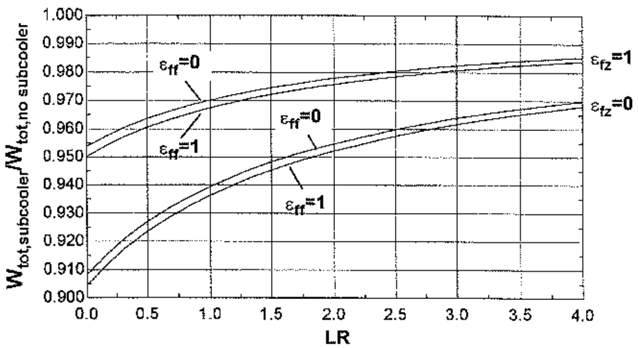

Figure 7 *Influence of indirect mechanical subcooling for a range of load ratios.*

effectiveness of unity to the work required if a subcooler is not utilized. Figure 7 displays calculated values of this work ratio (representing the maximum benefit of indirect mechanical subcooling) for LR ranging between 0 and 4. There are four curves in Figure 7 corresponding to the four combinations of suction-line heat exchanger effectiveness values shown in Figure 6.

The effect of the suction-line heat exchanger for the fresh food cycle was found to be very small. The effectiveness of the freezer suction-line heat exchanger is significant, as seen in Figure 6. When LR is 0, there is no load for the fresh food compartment and the refrigerator cycle operates solely to provide subcooling for the freezer cycle. With no fresh food compartment load, mechanical subcooling can result in a maximum reduction of 9% in compressor power if a liquid suction heat exchanger in the freezer cycle is not employed, but this performance improvement is reduced to about 5% when a high-effectiveness liquid suction heat exchanger is employed. The results in Figure 7 were found to be insensitive to the choice of compressor parameters.

CONCLUSION

A significant improvement in energy was demonstrated in a system that uses two cycles to independently provide refrigeration to the fresh food and freezer compartments. The extent of these savings depends on the ratio of the fresh food to the freezer cabinet loads, *LR*, and the ratio of the COPs of the fresh food and freezer refrigeration cycles, *CR*. High load and COP ratios increase the benefits of the dual-cycle design. Performance advantages of up to 30% are possible. In addition, the dual-cycle system should provide better humidity control in the fresh food cabinet and reduced defrosting. The disadvantages of the dual-cycle design are the additional cost and space required for two refrigeration cycles. These considerations have to be factored into the decision-making process for implementing a dual-cycle system.

The effect of suction-line heat exchangers was investigated for a dual-cycle system using refrigerant R-134a in both cycles. The suction-line heat exchanger on the freezer cycle was found to reduce the required compressor work by as much as 8%, whereas the suction-line heat exchanger for the fresh food cycle only provided a 4% reduction. The suction-line heat exchangers are inexpensive and easy to implement. They also provide a some protection from liquid entering the compressor. Implementation of suction-line heat exchangers can be recommended, especially for the freezer cycle.

Indirect mechanical subcooling was found to enhance the performance of the dual-cycle system by as much as 9%. However, this figure corresponds to a situation with no fresh food cabinet load and no suction-line heat exchanger in the freezer cycle. The performance improvement corresponding to more typical conditions is about 3%. It is important to note that the results presented here provide an upper bound on the benefit of indirect mechanical subcooling in that the additional pressure drops and fan power that would be needed were not considered. The number of ways to cost-effectively reduce the energy use of a modern refrigerator is limited. However, this analysis indicates that indirect mechanical subcooling would likely not be a viable alternative.

ACKNOWLEDGEMENTS

This project was jointly funded by grants from the Sub-Zero Freezer Company and University Industry Relations at the University of Wisconsin. The authors would like to gratefully acknowledge the guidance and input from the Sub-Zero staff, including John Jaschinski, Paul Sikir, and Chris Rieger.

REFERENCES

Bare, J.C. 1992. Simulation results of single refrigerants for use in a dual-circuit refrigerator/freezer. *J. Air Waste Management Assoc.*, vol. 42, no. 2.

DOE. 1997. Refrigerator energy efficiency standards (R-97-027). U.S. Dept. of Energy. http://www.osti.gov/html/doe/whatsnew/factsheet/fs97027.html

Gan, A.I. 1999. The design of better household refrigerators with improved energy consumption. M.S. thesis, Solar Energy Laboratory, Department of Mechanical Engineering, University of Wisconsin–Madison.

Jaehnig, D. 1999. A semi-empirical method for modeling reciprocating compressors in residential refrigerators and freezers. M.S. thesis, Solar Energy Laboratory, Department of Mechanical Engineering, University of Wisconsin–Madison.

Klein, S.A., and F.L. Alvarado. 1998. EES: Engineering Equation Solver, Version 4.900. University of Wisconsin–Madison.

Klein, S.A., D.T. Reindl, and K.A. Brownell. 1999. Refrigeration system performance using liquid-suction heat exchangers. Submitted to *International Journal of Refrigeration.*

Thornton, J.W., S.A. Klein, and J.W. Mitchell. 1994. Dedicated mechanical subcooling design strategies for supermarket applications. *Int. J. Refrig.*, vol. 17, no. 8.