INVESTIGATION OF AMMONIA-SECONDARY FLUID SYSTEMS IN SUPERMARKET REFRIGERATION SYSTEMS

Timothy P. McDowell
Student Member ASHRAE

John W. Mitchell, Ph.D., P.E.
Member ASHRAE

Sanford A. Klein, Ph.D.
Member ASHRAE

ABSTRACT

International agreements have legislated the phaseout of many refrigerants, including R-502 and R-12, which are commonly used in supermarket refrigeration systems. R-22 and ammonia (R-717) are candidate replacement refrigerants having appropriate thermodynamic properties. The toxicity of ammonia at low concentrations requires that it be confined to the equipment room, so a secondary fluid is needed to distribute cooling to the refrigerated cases. This paper investigates ammonia-secondary fluid systems and compares their performance with equivalent R-22 systems. Both R-22 and ammonia have high compressor discharge temperatures, necessitating staged compression. Three methods of staging the compression were compared for both refrigerants. Six secondary fluids were evaluated for use with ammonia in the supermarket system. The overall system performance of the ammonia-secondary fluid refrigeration system, including both compressor and secondary fluid pump power, is governed by a large set of design parameters. The influence of these parameters on the overall system performance was studied in a systematic manner. From this parametric study, design rules leading to optimal ammonia-secondary fluid systems were developed. The performance of well-designed ammonia-secondary fluid systems was found to be 4% to 10% lower than that of R-22 systems operating under similar conditions.

INTRODUCTION

Environmental concerns have led to international agreements to eliminate substances that cause ozone depletion or global warming, including many of the refrigerants currently being used. The refrigerants of greatest concern are fully halogenated chlorofluorocarbons (CFCs) and chlorofluorocarbons that are not fully halogenated (HCFCs). Many supermarket refrigeration systems currently utilize R-12 or R-502, both of which are scheduled to be phased out. A near-term replacement refrigerant for R-12 and R-502 in supermarket applications is R-22. However, R-22 is an HCFC and its production is scheduled to be phased out by the year 2030. A possible replacement refrigerant is ammonia (R-717).[1]

The advantages and disadvantages of ammonia as a refrigerant are well known (Stoecker 1989). The major disadvantage of ammonia is that it is toxic at low concentrations. If ammonia is used in a supermarket, it must be confined to the equipment room, necessitating a secondary heat exchange loop to distribute cooling to the refrigerated cases, as shown in Figure 1.

This paper compares the thermal performance of an R-22 supermarket refrigeration system and an ammonia system with a secondary heat transfer fluid. Although there are other replacement fluids, R-22 was used for this comparison because its properties are well-established and it is in common use. Comparisons of the performance of an ammonia secondary loop system with other candidate refrigerants can be indirectly made by using the R-22 system results as a basis for comparison.

MODEL DEVELOPMENT

A computer model of the ammonia with a secondary fluid refrigeration system was written utilizing an equation-solving program (Klein and Alvarado 1993) to determine the performance and to develop design rules. The models of the refrigerant system components are outlined here. A more detailed presentation of these models is provided by McDowell (1993).

Condenser

The condenser model is representative of an air-cooled condenser and uses the effectiveness-NTU method (Kays and London 1964). The total heat transfer in an actual process includes the desuperheating, condensing, and possibly the subcooling of the refrigerant. The major heat flow is due to condensation, and the mechanism equation used in this

[1]Since this research was completed, the supermarket industry has moved rapidly to non-ozone-depleting HFC alternatives such as R-404A and R-507, thereby reducing some of the environmental advantages of ammonia as a primary refrigerant.

Timothy P. McDowell is a research intern at the Thermal Storage Applications Research Center, Madison, WI. **John W. Mitchell** and **Sanford A. Klein** are professors in the Department of Mechanical Engineering at the University of Wisconsin, Madison.

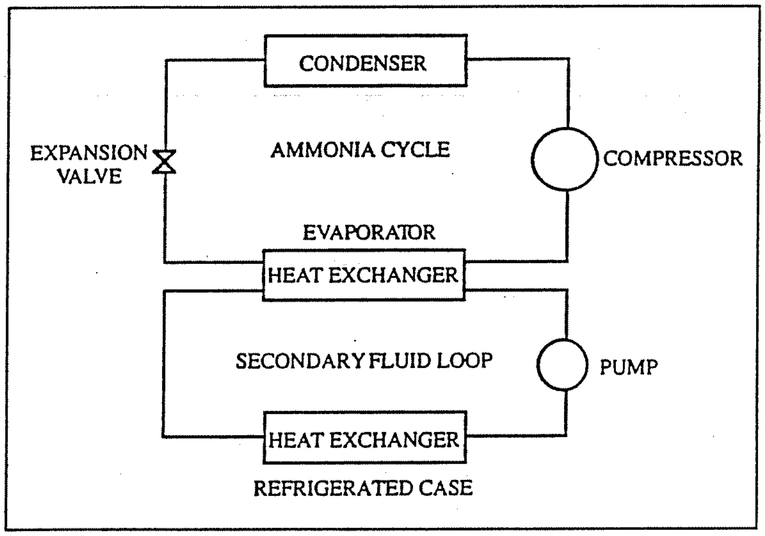

Figure 1 *Diagram of an ammonia-secondary fluid refrigeration system.*

condenser model assumes that the refrigerant is isothermal at the condensing pressure, as recommended by Stoecker and Jones (1982). The effectiveness-NTU equation for an isothermal phase change is used to calculate the heat transfer rate and the change in enthalpy of the ammonia, as indicated in Equations 1 and 2. The physical size and construction details of the heat exchanger can be related to the heat exchanger NTU.

$$\varepsilon = 1 - \exp(-NTU) \tag{1}$$

$$Q = \varepsilon \ \dot{m}_{air} Cp_{air} (T_{ref} - T_{amb}) \tag{2}$$

Compressor

A mechanistic model based on actual physical dimensions is used to describe the performance of the reciprocating compressor (Threlkeld 1970; Chlumsky 1965). A polytropic exponent (n) is used to relate the entering and exiting states of the refrigerant according to Equation 3:

$$P_{in} v_{in}^{n} = P_{out} v_{out}^{n} \,. \tag{3}$$

Compressor volumetric efficiency is defined based on the clearance ratio, r, of the compressor, as well as the compressor's pressure ratio and the polytropic exponent.

$$\eta_{vol} = 1 - r\left[\left(\frac{P_{out}}{P_{in}}\right)^{\frac{1}{n}} - 1\right] \tag{4}$$

The polytropic exponents were set to be 1.15 for R-22 and 1.30 for ammonia, which are essentially equal to the isentropic exponents. The clearance ratio was set to 0.053.

The volumetric efficiency is used to calculate the refrigerant flow rate as a function of the displacement volume, compressor speed, and specific volume of the inlet refrigerant.

$$\dot{m}_{ref} = \left[\frac{1}{v_{in}}\right] V_{disp} N \eta_{vol} \tag{5}$$

The compressor power, W_{comp}, is given by

$$W_{comp} = \frac{W_{isentropic}}{\eta_{isentropic}} \tag{6}$$

where $W_{isentropic}$ is the power required for isentropic compression and $\eta_{isentropic}$ is an isentropic efficiency that was set to 0.65.

Evaporator

The evaporator model is for a flooded shell-and-tube heat exchanger with the secondary fluid flowing through the tubes and ammonia in the shell. The secondary fluid enters the tube bundle of the heat exchanger, where it is cooled by the ammonia and then recirculated to the refrigerated case. The ammonia enters as a liquid-vapor mixture after leaving the expansion valve. All expansion valves are assumed to be isenthalpic. As the ammonia cools the secondary fluid, it evaporates and saturated ammonia vapor flows to the compressor. The effectiveness of the heat exchanger is calculated from the heat exchanger geometry and the heat transfer coefficients on the inside and outside of the pipes. On the inside of the pipes, the flow is considered to be developing hydrodynamically and developed thermally. For laminar flow, the Hausen correlation is used to calculate the Nusselt number (Chapman 1984):

$$Nu_D = 3.66 + \frac{0.0668\left(\frac{D}{L}\right)Re_D Pr}{1 + 0.4\left[\left(\frac{D}{L}\right)Re_D Pr\right]^{2/3}} \tag{7}$$

For turbulent flow, an equation developed by Nusselt that accounts for the entry length effects is used (Chapman 1984):

$$Nu_D = 0.036 Re_D^{0.8} Pr^{1/3}\left(\frac{D}{L}\right)^{1/18} \tag{8}$$

On the outside of the pipes, ammonia is evaporated in a pool boiling process. To calculate the heat transfer coefficient, a correlation developed by Rohsenow (1952) for pool boiling is used:

$$h = \mu_l h_{fg}\left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} \tag{9}$$

$$\left[\frac{Cp_l}{C_{sf} h_{fg}(Pr_l)^{1.7}}\right]^3 (T_{wall} - T_{saturated})^2$$

Once the heat transfer coefficients are calculated, the overall heat transfer coefficient and the *UA* product are determined. The heat transfer and temperature changes are calculated using the effectiveness-NTU method (Kays and London 1964).

Refrigerated Case

The refrigerated case is modeled as a heat exchanger that cools the air circulating in the refrigerated case with a secondary fluid flowing through a tube-bundle-oriented crossflow to the airstream. Standard heat exchanger modeling techniques are used to determine the overall heat transfer coefficient. The secondary fluid flow is assumed to be developing hydrodynamically and developed thermally, and the Hausen correlation for laminar flow and the Nusselt correlation for turbulent flow are used again. Air is circulated on the outside of the pipes, resulting in heat transfer by forced convection over horizontal pipes. It is assumed that the pipes do not have extended surfaces. Churchill and Bernstein developed a correlation equation for this geometry (Chapman 1984):

$$Nu_D = 0.3 + \frac{0.62 Re_D^{1/2} Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \tag{10}$$

$$\left[1 + \left(\frac{Re_D}{2.82 \times 10^5}\right)^{5/8}\right]^{4/5}$$

The effectiveness-NTU method is used to determine the heat transfer and temperature changes. The effectiveness of the case heat exchanger is calculated using the formula for crossflow heat exchangers with both fluids unmixed (Incropera and Dewitt 1985):

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right)(NTU)^{0.22}\{\exp[-C_r(NTU)^{0.78}] - 1\}\right] \tag{11}$$

The heat transfer rate in the case heat exchanger is the refrigerated case load.

Piping Thermal Losses

The secondary fluid piping system between the two heat exchangers involves both thermal losses and pumping requirements. In order to determine the thermal losses to the environment, it is necessary to calculate the overall heat transfer coefficient of the piping system. The flow is assumed to be fully developed both hydrodynamically and thermally and to be turbulent. The heat transfer coefficient on the inside of the pipes is calculated using the Dittus-Boelter correlation (Incropera and Dewitt 1985):

$$Nu_D = 0.023 Re_D^{0.8} Pr^{0.4} \tag{12}$$

The pipes are exposed to the air of the supermarket, and a constant heat transfer coefficient of 6 $W/m^2 \cdot °C$ (1.05 $Btu/h \cdot ft^2 \cdot R$) based on convection and radiation is used for the outside of the pipes (Chapman 1984). The thickness of the insulation on the pipes is a design parameter. Heat transfer resistance due to conduction through the pipe walls is neglected. Since the heat transfer properties are dependent on the bulk temperature of the secondary fluid and the temperatures depend on the heat transfer from the pipes to the environment, both an energy balance and a heat transfer rate equation are necessary to determine the inlet and outlet temperatures and the heat transfer rate. The heat transfer rate is based on the log-mean temperature difference of the secondary fluid and air temperatures.

Piping Head Losses

The first step in determining the pump power is to calculate the head losses in the pipes arising from the frictional losses and the minor losses due to bends and valves. The frictional losses are calculated using the friction factor (*f*) from the Moody diagram (White 1986):

$$head = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \tag{13}$$

The head losses from the minor losses due to bends and valves in the piping system depend on the number and type of bends and valves. Each bend and valve is assigned an equivalent loss coefficient (K_{eq}), and then the total equivalent loss coefficient is the sum of all the equivalent loss coefficients (White 1986):

$$head = K_{eq}\left(\frac{v^2}{2g}\right) \tag{14}$$

The total head loss is the sum of the losses due to friction and the minor losses. Once the total head losses are determined, the pressure drop and the pump power are calculated. The pump power is calculated, accounting for the pressure drop in the distribution lines, the refrigerated case heat exchanger, and the ammonia heat exchanger. Motor and mechanical inefficiencies are not included in the pump power.

The inherently higher cycle efficiencies of ammonia are offset by the additional pump power required to circulate the secondary fluid. The pump power is added to the compressor power to define a system coefficient of performance (COP):

$$COP = \frac{RefrigerationLoad}{W_{compressor} + W_{pump}} \tag{15}$$

Compressor Staging

The practical use of ammonia necessitates a means of controlling the compressor discharge temperatures. The temperature leaving the compressor can be reduced by staging the compression and intercooling between the stages. Three methods of staging the compression were compared for use with ammonia and R-22 (McDowell 1993). The first method is known as *basic staged compression* (Gosney 1982). This method involves extracting some of the refrigerant leaving the condenser at an intermediate pressure and mixing it with the refrigerant leaving the first compressor at the same intermediate pressure. The advantage of basic staging is that the refrigerant is desuperheated to saturated conditions between the two compressors, causing the compressed gas to exit the second stage of compression at a lower temperature. The lower temperature also leads to a higher volumetric efficiency in the second compressor. The gas entering the second compressor has a smaller specific volume than it would if no desuperheating took place, allowing a smaller compressor to be used in the second stage. However, a higher mass flow rate of refrigerant is needed to provide both the refrigeration load and the intercooling.

A second method, known as *staged compression and evaporation*, differs from basic staging in that all of the refrigerant leaving the condenser is expanded at an intermediate pressure (Gosney 1982). The liquid refrigerant separated out at the intermediate pressure is expanded again for use in the evaporator, while the vapor is used to mix with and cool the vapor leaving the first compressor, as shown in Figure 2. This type of staging produces the same desuperheating advantage as for the staged compression method but it is less effective than staged compression since the vapor is not cooled to such a low temperature. However, an advantage of the staged compression and evaporation method is that the refrigerant is expanded twice, so the enthalpy difference across the evaporator is greater, therefore less mass flow of refrigerant is needed to meet the refrigeration load, reducing the size of the first compressor stage.

A third method utilizes a flash tank between the condenser and the evaporator and between the two stages of compression (Stoecker and Jones 1982), as shown in Figure 3. This method is similar to the staged compression and evaporation method in that the refrigerant leaving the condenser is expanded at an intermediate pressure and some of the resulting liquid is used in the evaporator. However, it differs in that the vapor entering the second compressor is saturated. The liquid and vapor from the expansion of the refrigerant leaving the condenser enter a flash tank, where some of the liquid is extracted and sent to the evaporator. The refrigerant leaving the evaporator is compressed to the intermediate pressure in the low-pressure compressor and bubbled through the remaining liquid and vapor in the flash tank. The resulting saturated vapor in the flash tank is removed and compressed in the second compressor to the condensing pressure. A higher mass flow rate through the

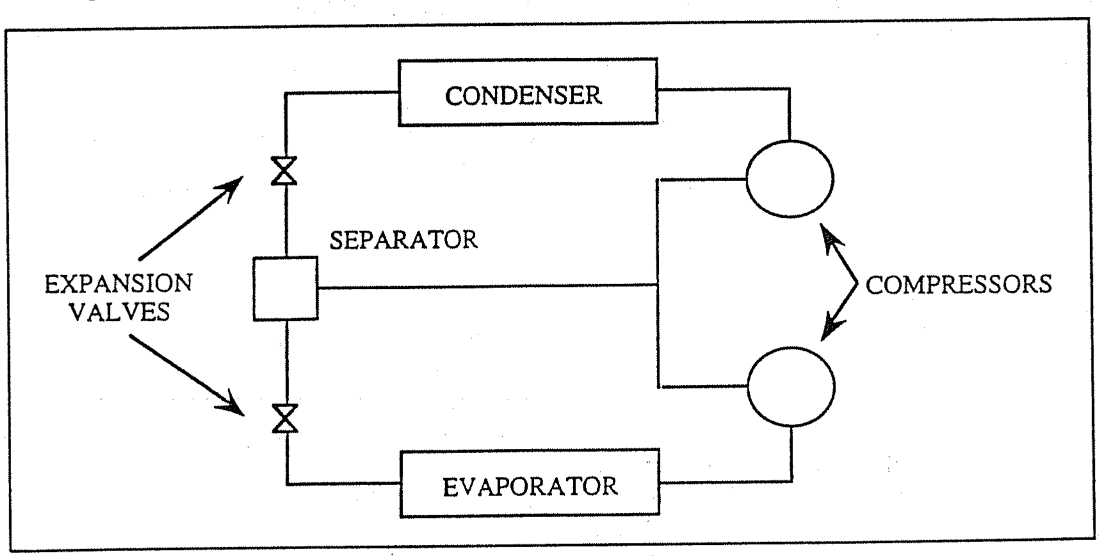

Figure 2 Refrigeration cycle with staged compression and evaporation.

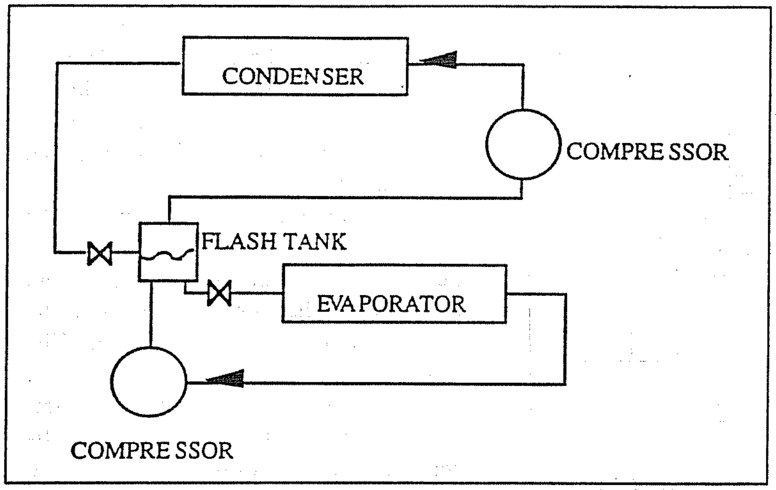

Figure 3 *Refrigeration cycle with staged compression and flash tank.*

condenser is needed than in the staged compression and evaporation method to provide the refrigerant for intercooling. However, the refrigerant is desuperheated to the saturation point, resulting in increased volumetric efficiency and decreased size for the second-stage compressor. The expansion is staged and has the same refrigeration capacity advantage as that in the staged compression and evaporation method. An optimal intermediate pressure, discussed in the "Design Rules" section, gives the highest performance.

The models of the different staging methods with R-22 and ammonia as the refrigerant were compared for evaporator temperatures of 244 K (–20°F) and 267 K (21°F) and a refrigeration load of 52.8 kW (15 tons). The COPs for the different systems (not including the pump power for a secondary loop) are shown in Table 1. With R-22 as the refrigerant, staged compression and evaporation has the highest performance. R-22 has a smaller superheating horn than ammonia and does not benefit as much from desuperheating and intercooling. Thus the higher flow rate needed in staged compression with a flash tank penalizes the performance of R-22 more than the advantage of intercooling. With ammonia as the refrigerant, staged compression with a flash tank yields the highest performance. Staged compression with a flash tank is used with ammonia, and staged compression and evaporation is used with R-22 in the rest of this study. At the same evaporator and condenser temperatures, ammonia performs better than R-22, but the pump power and heat transfer resistances for the secondary loop will penalize the performance of the ammonia system.

SECONDARY FLUID SELECTION

The secondary fluids evaluated in this study were propylene glycol, ethylene glycol, mineral oil, ethanol, propane, and a silicone-based heat transfer fluid. Propylene glycol-water solutions are used in applications in which oral toxicity is a concern, such as applications with drinking water or food processing. Ethylene glycol is less viscous than propylene glycol and it provides greater heat transfer and better low-temperature performance. It is, however, moderately orally toxic and should be used with caution in situations where accidental contact with food can occur. A low-temperature mineral oil fluid—polyalphaolefin—is a nontoxic substance that meets the Food and Drug Administration's (FDA) regulation for use as a synthetic white mineral oil for nonfood articles in contact with food. The low-temperature silicone-based heat transfer medium is a specially formulated silicone polymer—dimethyl polysiloxane. Ethanol (ethyl alcohol) is both flammable and explosive. Propane can also be used as a secondary fluid, although it is also highly flammable and explosive. It is necessary to ensure that the propane pressure is high enough that the propane remains in liquid form throughout the system.

Correlations were developed to relate the transport properties of the different fluids to temperature (and concentration for the glycols) (McDowell 1993). These correlations were then used in the refrigeration system model, and the overall system performance was calculated for each fluid. Figure 4 shows the overall system COP (defined in Equation 15 to include both compressor and pump power) versus the temperature difference across the heat exchanger of the refrigerated case. The temperature difference across the heat exchanger of the refrigerated case is an important design parameter; its effect on system performance is discussed in the following section. At a refrigerated case temperature of 267 K (21°F), propane has the highest performance. Propy-

TABLE 1
Compression Staging Performance Comparison

Method	Evaporating Temperature			
	244 K (-20°R)		267 K (21°R)	
	COP R22	COP NH_3	COP R22	COP NH_3
Single stage of compression	1.44	1.52	2.50	2.90
Staged compression	1.43	1.61	2.49	3.03
Staged compression and evaporation	1.69	1.69	2.76	3.09
Staged compression with flash tank	1.68	1.84	2.74	3.25

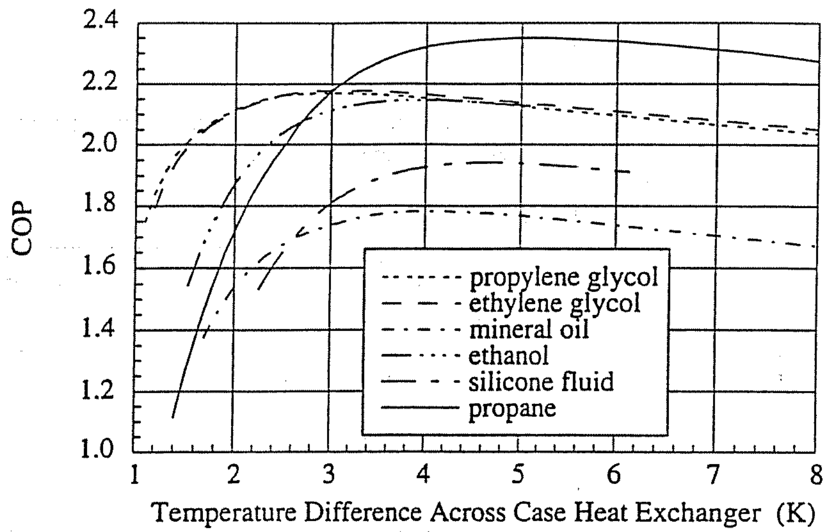

Figure 4 *Performance comparison of the secondary fluids for a case temperature of 267 K (21°F).*

lene glycol and ethylene glycol have the next highest performance. Ethanol has a performance almost as high as propylene glycol and ethylene glycol. The silicone-based fluid and the mineral oil have the lowest performance. The results for a case temperature of 244 K (–20°F) show the same performance rankings.

Figure 4 shows that the choice of propane as the secondary fluid will yield the highest performance by about 10%. However, propane is both flammable and explosive, while propylene glycol and ethylene glycol are both nonflammable. Propylene glycol is nontoxic, and ethylene glycol is orally toxic. In a supermarket, safety is a concern, and propylene glycol would likely be the best choice of the six secondary fluids examined here.

COMPARISON OF R-22 AND AMMONIA WITH SECONDARY FLUID SYSTEMS

Ammonia-secondary fluid refrigeration systems will be practical only if their performance is comparable to the performance of the R-22 systems. The model used to evaluate the performance of the ammonia-secondary fluid systems was written to include the pumping and thermal losses associated with the heat exchangers and the distribution of the secondary fluid throughout the supermarket. To compare the performance to that of an R-22 system, it was necessary to develop a model of an R-22 refrigeration system that included a direct-expansion refrigerated case model and pressure losses due to the liquid and vapor R-22 flows in the long pipe runs between the mechanical room and the display cases. These pressure losses result in increased compressor power rather than in separate pump power, as in the secondary loop system.

The equations used to calculate the thermal losses and pressure drops in the supply and return lines for the refrigerated case in the R-22 system model are identical to those used in the ammonia-secondary fluid system model. The refrigerated case is assumed to have R-22 circulated in a tube-bundle-oriented crossflow to the airstream. The heat transfer coefficients of the airflow on the outside of the pipes are calculated in the same manner as in the ammonia-secondary fluid model. The inside heat transfer coefficient was assumed to be at least an order of magnitude greater than the heat transfer coefficient on the outside of the pipes due to the phase change of the R-22 on the inside of the pipes. The effectiveness of the heat exchanger is determined using the effectiveness-NTU equation for heat exchangers with one stream-changing phase.

The performance of the R-22 system (using staged compression and evaporation) and the system using ammonia with propylene glycol (using staged compression with a flash tank) was compared at refrigerated case temperatures of 267 K (21°F) and 244 K (–20°F). The results are shown in Table 2. The system COP for the R-22 system is around 4% higher than the system COP for the ammonia with propylene glycol system at 267 K (21°F) and 10% higher at 244 K (–20°F). This difference between the two systems could possibly be slightly reduced with improvements in the heat exchanger design used in the ammonia-propylene glycol system.

The comparison of the R-22 and the ammonia-secondary loop systems is made solely on the basis of thermal performance. Other considerations, such as the local codes and regulations for use with ammonia systems and the cost associated with the heat exchange equipment and secondary fluids, have not been considered in this study.

DESIGN RULES

Intermediate Pressure

The selection of the operating pressure ratio of the staged compression in the ammonia refrigeration cycle is important in providing the greatest increase in intercooling and refrigeration capacity, resulting in the highest COP. An analysis of the influence of the exponent in the pressure ratio equation on the overall system performance shows that the highest performance occurs when the exponent (X) is between 0.5 and 0.6 (McDowell 1993):

$$\frac{P_{inter}}{P_{evap}} = \left[\frac{P_{cond}}{P_{evap}}\right]^X \tag{16}$$

TABLE 2
Performance Comparison of R-22 System and Ammonia with Propylene Glycol System

Refrigerant	COP @ 267 K (21°F)	COP @ 244 K (-20°F)
R22	2.84	1.40
Ammonia with propylene glycol	2.72	1.25

This result agrees with the estimate that the maximum performance will occur at the geometric mean of the condensing and evaporating pressures (Stoecker and Jones 1982).

Temperature Difference Across Refrigerated Case Heat Exchanger

The temperature difference across the refrigerated case heat exchanger is a design consideration. As the temperature change of the secondary fluid through the case heat exchanger increases, colder ammonia temperatures are required in the ammonia-secondary fluid heat exchanger. The colder the ammonia, the more compressor power is needed. If the temperature difference decreases, the ammonia temperature can be higher and the compressor power is reduced, but the pump power needed to circulate the secondary fluid increases. The highest system performance will occur at a temperature difference that balances the compressor and pump power. The relative influence of the ratio of the compressor power and the pump power was calculated for four heat exchanger-piping system combinations, as shown in Figure 5, where "num" stands for the number of pipes in the heat exchanger and "radius" is the inside radius of the pipes. The highest overall system performance occurs when the ratio of pump power to compressor power is between 0.01 and 0.03.

Relative Heat Exchanger Sizes

The supermarket refrigerated case system with ammonia and a secondary fluid utilizes two heat exchangers to provide the cooling. The first is between the ammonia and the secondary coolant, and the second is in the refrigerated case. Most refrigerated cases are designed to use direct expansion of refrigerant. The refrigerated case for an ammonia secondary loop system will necessarily require redesign to accommodate a different, larger heat exchanger.

As the mass flow rate-specific heat ratio of the secondary fluid stream increases, the effectiveness of the refrigerated case heat exchanger increases and the effectiveness of the ammonia-secondary fluid heat exchanger decreases, leading to an optimization problem involving the overall loss coefficients (*UA*) of the two heat exchangers.

The system performance was calculated for four total heat transfer areas and pipe diameter combinations. The total heat transfer area is the combined heat transfer area of the refrigerated case heat exchanger and the ammonia-secondary fluid heat exchanger. The results of the comparison as a function of the ratio of the *UA* value of the refrigerated case

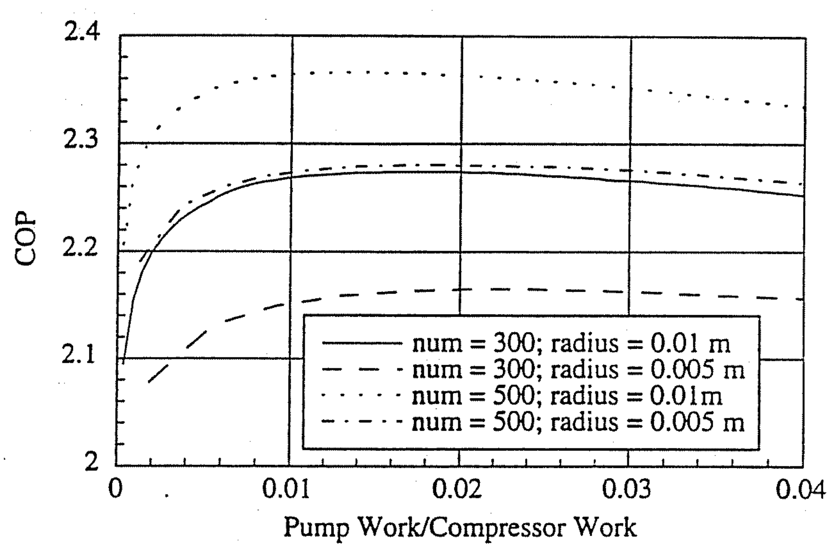

Figure 5 System performance as a function of compressor power-pump power ratio.

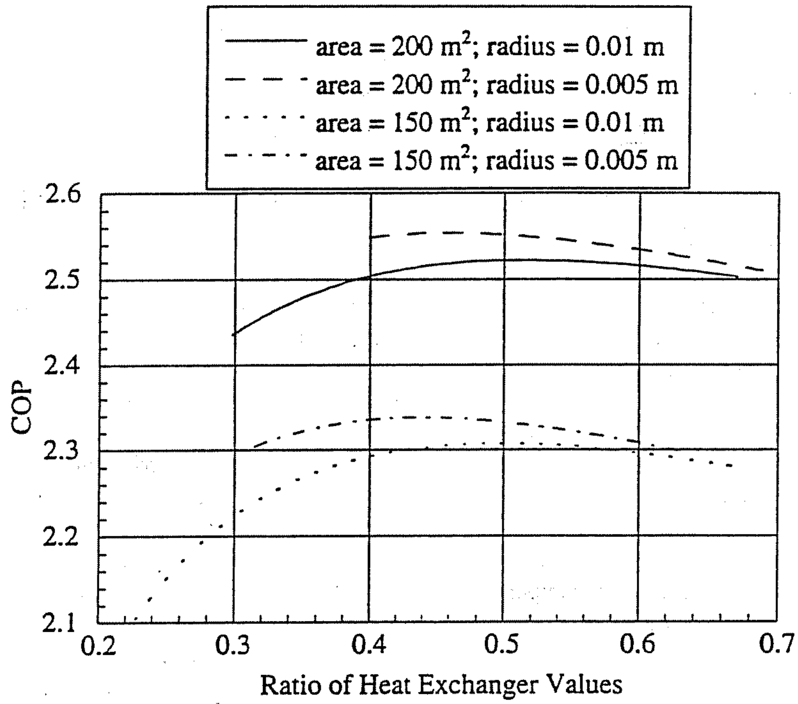

Figure 6 *System performance as a function of UA value ratio.*

heat exchanger to the *UA* value of the ammonia-secondary fluid heat exchanger are shown in Figure 6, where "area" is the total heat transfer area and "radius" is the inside radius of the pipes in the heat exchanger. All of the plots show a maximum performance at *UA* ratios between 0.4 and 0.5. Optimal performance occurs when the heat exchangers are sized such that the *UA* value for the ammonia-secondary fluid heat exchanger is 0.4 to 0.5 times the *UA* value of the refrigerated case heat exchanger.

Piping Diameters and Lengths

The length, diameter, and insulation thickness of the secondary fluid supply and return pipes are important in the design because they influence the pump power and thermal losses. It is assumed that the same length, diameter, and amount of insulation are used for both pipes. Maps of the performance of an ammonia-propylene glycol system at a refrigerated case temperature of 267 K (21°F) as a function of pipe length, diameter, and insulation thickness were developed. The ranges of the parameters were pipe length from 10 to 80 m (32.8 to 262.5 ft), pipe diameter from 0.05 to 0.30 m (2 to 12 in.), and insulation thickness from 0.01 to 0.03 m (0.40 to 1.2 in.). The influence of each individual parameter is different from the influence when all three parameters are taken together. The other parameters in the model were held constant at their base values, and the temperature difference across the case heat exchanger was set to 3 K (5.4°F), which is the optimal ratio for the base values. The maximum system COP in this comparison range was calculated at a pipe length of 10 m (32.8 ft), a pipe diameter of 0.10 m (4 in.), and an insulation thickness of 0.03 m (1.2 in.).

To develop the performance maps, the combinations of pipe length, pipe diameter, and amount of insulation that yielded system COPs that were 97.5%, 95.0%, 92.5%, 90%, and 80% of the maximum were determined and plotted. The performance maps are shown here in three parts: Figure 7 shows the map with 0.01 m (0.40 in.) of insulation, Figure 8 shows it with 0.02 m (0.8 in.) of insulation, and Figure 9 shows it with 0.03 m (1.2 in.) of insulation.

The maps provide an easy way to estimate an optimal design. Assume, for example, that the supermarket requires 40 m (131 ft) of pipe between the two heat exchangers. Using 0.01 m (0.40 in.) of insulation, the system can attain a COP between 95% and 92.5% of the maximum COP by using pipe diameters between 0.05 and 0.12 m (2 and 4.7 in.). With 0.02 m (0.8 in.) of insulation, performance between 95% and 92.5% of the maximum COP can be attained with pipe diameters between 0.05 and 0.21 m (2 and 8.3 in.). With 0.03 m (1.2 in.) of insulation, performance between 95% and 92.5% can be attained with pipe diameters between 0.05 and 0.29 m (2 and 11.4 in.).

The maps can also be used to determine the amount of insulation needed to attain a specified performance level with a specific pipe length and diameter. If a pipe length of

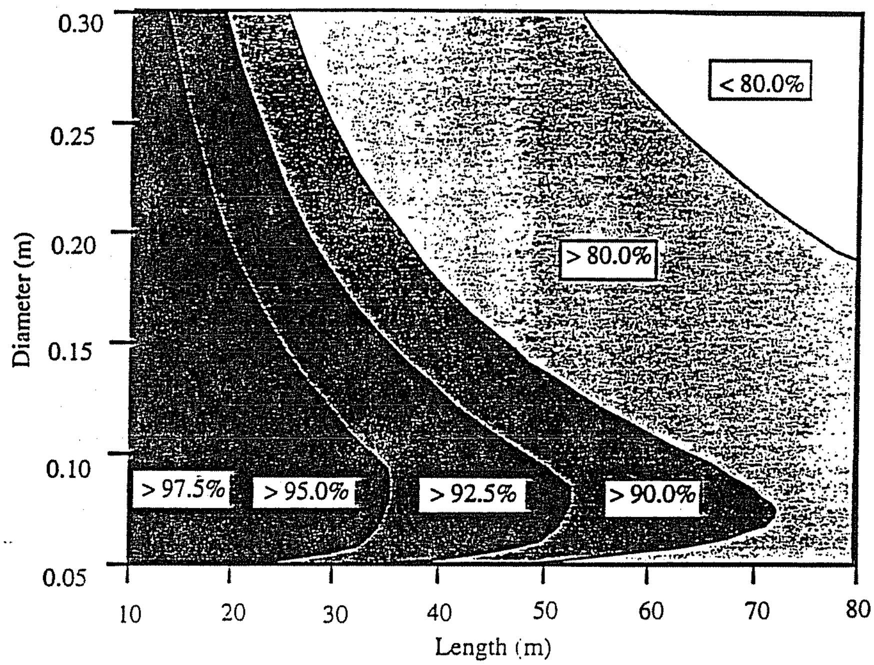

Figure 7 *Performance map with 0.01 m (0.40 in.) of insulation.*

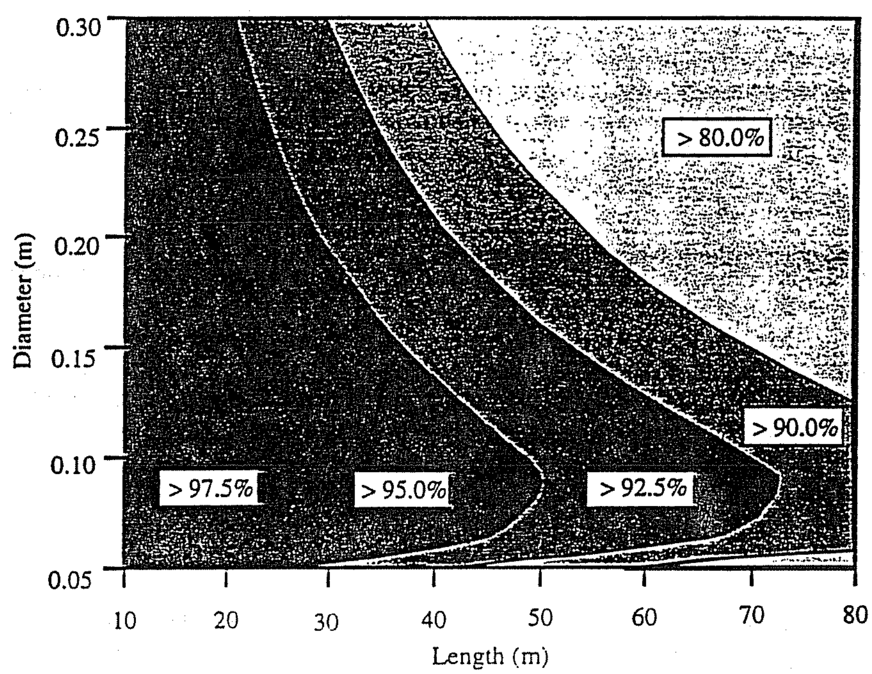

Figure 8 *Performance map with 0.02 m (0.8 in.) of insulation.*

25 m (82.0 ft) and a diameter of 0.17 m (6.7 in.) are to be used, 0.01 m (0.4 in.) of insulation will give a performance between 95% and 92.5% of the maximum, 0.02 m (0.8 in.) of insulation will give a performance between 97.5% and 95%, and 0.03 m (1.2 in.) of insulation will give a performance within 97.5% of maximum.

CONCLUSIONS

Ammonia is a proven refrigerant that has been in use for many years. This study shows that a well-designed supermarket system that uses ammonia with propylene glycol as a secondary fluid will have a performance that is 4% at 267 K

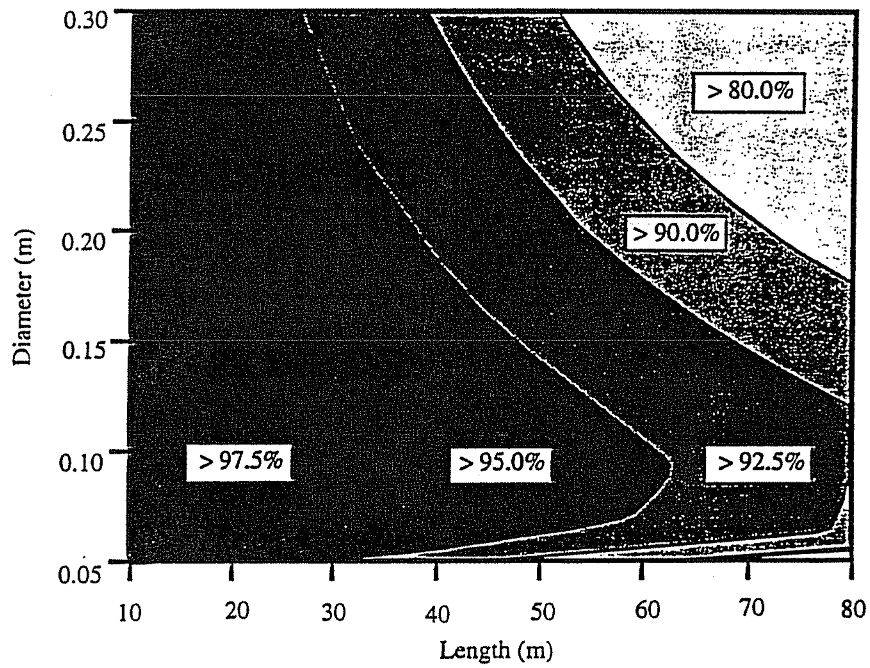

Figure 9 *Performance map with 0.03 m (1.2 in.) of insulation.*

(21°F) to 10% at 244 K (–20°F) lower than the performance of R-22 systems under the same operating conditions. The performance of the ammonia secondary loop systems is worse at low temperatures due to the high viscosity of the secondary fluid and the resulting pumping power. This design includes an ammonia system that utilizes staged compression with a flash tank to provide desuperheating and increased refrigeration capacity. The pressure ratio used for the staging provides an exponent for the pressure ratio equation (Equation 4) of between 0.5 and 0.6. The pump power should be around 0.02 times the compressor power. The ammonia-secondary fluid heat exchanger *UA* should be approximately 0.4 to 0.5 times the *UA* value for the refrigerated case. The secondary fluid piping system is selected using the performance maps (Figures 7 through 9) to achieve the highest possible performance level.

This research has quantified the thermal performance of ammonia secondary loop systems in supermarket applications. There are a number of practical issues that would need to be addressed if ammonia were used. Redesign of refrigerated cases to accommodate a large cooling coil has economic implications for the both the case manufacturer and the supermarket. Codes may prohibit the use of ammonia in supermarket applications, even if it were confined to the equipment room. Since the performance of ammonia secondary loop systems is at best comparable to that of alternatives, there appears to be no incentives to actively pursue the use of ammonia for supermarket applications.

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NOMENCLATURE

COP	=	coefficient of performance	
Cp	=	specific heat	
C_r	=	mass flow rate-specific heat product ratio for heat exchangers	
C_{sf}	=	empirical constant for pool boiling	
D	=	diameter	
f	=	friction factor	
g	=	acceleration of gravity	
h	=	heat transfer coefficient	
head	=	heat loss	
h_{fg}	=	heat of vaporization of refrigerant	
k	=	thermal conductivity	
K_{eq}	=	equivalent length for minor losses	
L	=	length	
$\dot{m}$	=	mass flow rate	(17)
n	=	polytropic exponent	
$\dot{n}$	=	compressor rotational speed	(18)
NTU	=	number of transfer units	
Nu	=	Nusselt number = $h{\cdot}D/k$	
P	=	pressure	

Pr	=	Prandtl number
Q	=	heat transfer
r	=	ratio of clearance to displacement volume
Re	=	Reynolds number
T	=	temperature
UA	=	heat transfer conductance-area product
v	=	specific volume; velocity
V_{disp}	=	compressor displacement volume
$\dot{W}$	=	power (19)
X	=	pressure ratio exponent
ε	=	heat exchanger effectiveness; pipe roughness
$\eta_{isentropic}$	=	isentropic efficiency of compressor
η_{vol}	=	compressor volumetric efficiency
σ	=	surface tension
ρ	=	density
μ	=	viscosity

Subscripts

air	=	air properties
amb	=	ambient conditions
comp	=	compressor
cond	=	condenser
D	=	diameter
evap	=	evaporator
in	=	into compressor
inter	=	intermediate pressure
isentropic	=	constant entropy process
l	=	liquid state
out	=	out of compressor
pump	=	to the pump
ref	=	refrigerant
saturated	=	saturated conditions
v	=	vapor state
wall	=	surface between ammonia and secondary fluid in heat exchanger

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