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Equipment Configurations for Use of Ammonia in Supermarket Applications

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ABSTRACT

International agreements have legislated the phase-out of many of the refrigerants currently being used in the world, including R502 and R12 which are commonly used to provide the cooling for the refrigerated cases in supermarkets. R22 and ammonia (R717) are candidate replacement refrigerants having appropriate thermodynamic properties and less environmental effects. This paper identifies the optimal design for ammonia - secondary fluid systems and compares their performance to that of R22 systems. Both R22 and ammonia have high discharge temperatures leaving the compressor, 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 is governed by a large set of design parameters. The influence that these parameters have on the overall system performance was studied in a systematic manner. From this parametric study, design rules leading to optimum ammonia - secondary fluid systems were developed. The performance of a well-designed ammonia - secondary fluid system was found to be only about 4% lower than that of an R22 system.

INTRODUCTION

The growing concern about the environment has 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 most concern are the fully halogenated chlorofluorocarbons (CFCs) and the non-fully halogenated chlorofluorocarbons (HCFCs). Most supermarket refrigeration systems currently utilize either refrigerant R12 or R502, both of which are scheduled to be phased out. For both R12 and R502, the near-term replacement refrigerant in supermarket applications is the refrigerant R22. However, R22 is an HCFC and according to current plans the production of HCFCs will be phased out before the year 2030. Another possible replacement refrigerant for supermarket applications is ammonia (refrigerant R717).

The advantages of ammonia as a refrigerant are discussed by Stoecker (Stoecker 1989). Ammonia is cheaper and has higher cycle efficiencies, higher heat transfer coefficients, and a higher critical temperature than most CFCs and HCFCs. Because ammonia has a pungent odor, it is easy to detect leaks. Water vapor remains in solution with ammonia and will not separate and freeze as it can with other refrigerants. However, water contamination can still cause chemical changes and should be avoided. Oils and ammonia are virtually insoluble in each other, while oils and hydrocarbon liquids are mutually soluble. Oil that collects in the ammonia system will have to be drained off at an inactive point in the system and returned to the compressor. A major advantage is that ammonia has zero ozone depletion. When released into the atmosphere, it reacts with water in the air to form ammonium hydroxide and is quickly removed from the atmosphere. Ammonia also has a zero Global Warming Potential (GWP).

There are drawbacks to the use of ammonia as a refrigerant (Stoecker 1989). The behavior of ammonia with oils can also be considered a disadvantage, because an oil separator is required. Ammonia is not compatible with copper and copper bearing alloys, so steel and aluminum must be used as the construction materials. Hermetically sealed compressors cannot be used with ammonia because the ammonia destroys the copper wiring in the motors. Open compressors must be used instead. The temperature of ammonia leaving the compressors of refrigeration systems is very high and steps, such as cooling the heads of the compressors or staged compression with intercooling, need to be taken to reduce the temperature (R22 has the same problem). Ammonia is weakly flammable at concentrations of 16 to 25% by volume in air. However, the major drawback is the low concentrations at which it is considered toxic. Special precautions must be taken to prevent the build-up of dangerous concentrations in occupied areas.

A refrigeration system utilizing R22 as the refrigerant consists of a condenser and compressor rack in the mechanical room and distribution pipes that transport the refrigerant to the refrigerated cases where evaporation occurs. A diagram of the system is shown in Figure 1.

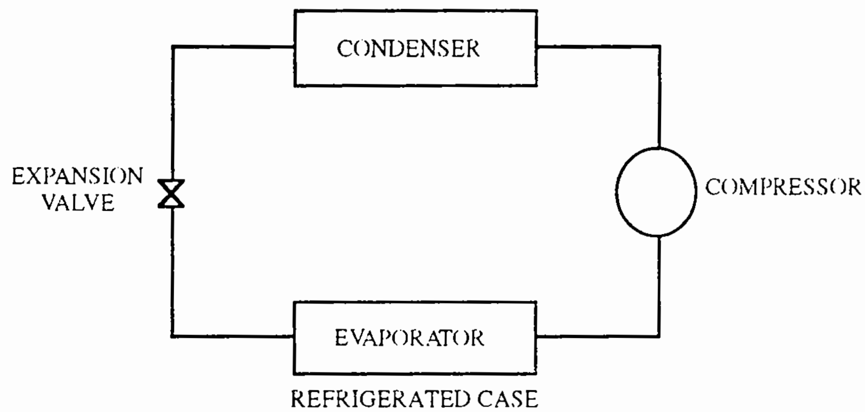

Figure 1: Diagram of R22 refrigeration system

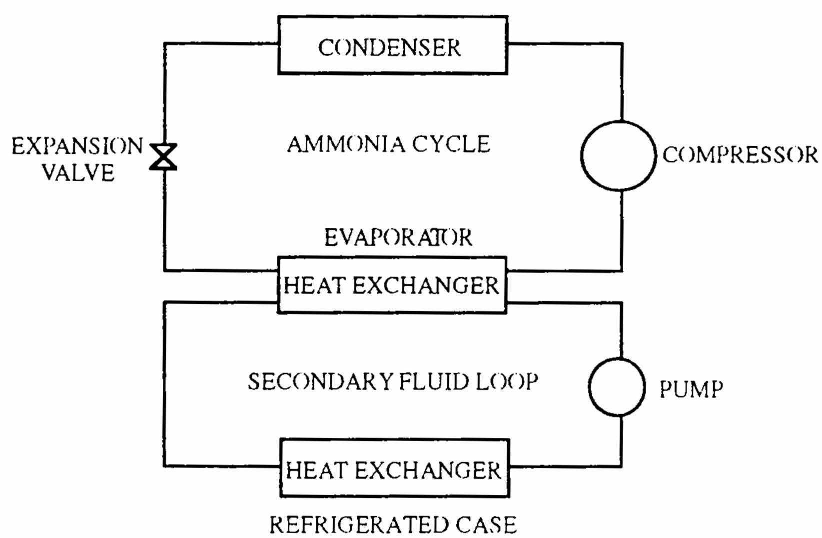

Figure 2: Diagram of an ammonia - secondary fluid refrigeration system

A system using ammonia would require a different configuration. Because of its toxicity, it would be a potential health risk to circulate ammonia throughout the supermarket. The ammonia would have to be confined to a well-ventilated mechanical room. To provide the cooling to the refrigerated cases, ammonia would be used to cool a secondary heat transfer fluid which would then be used in a heat exchanger to cool the air in the refrigerated case. The warm secondary fluid would return to the evaporator of the ammonia system where it would be cooled. A diagram of such a system is shown in Figure 2.

MODEL DEVELOPMENT

The performance of conventional refrigeration systems is well known. In contrast, the performance of ammonia systems with secondary loops has not been studied. 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 methods used to model the different system components are discussed here.

Condenser

The condenser model is representative of an air-cooled condenser and uses the effectiveness-NTU method to solve for the heat transfer (Chapman 1984). The total heat transfer in an actual process includes both the desuperheating, the condensing, and possibly the subcooling of the refrigerant. The major heat flow is due to condensation, and the mechanism equation used in this condenser model assumes that the refrigerant is isothermal at the condensing pressure as recommended by Stoecker and Jones (Stoecker and Jones 1982). The effectiveness-NTU equation for an isothermal phase change is used to calculate the heat transfer and the change in enthalpy of the ammonia:

$$\varepsilon = 1 - \exp(-NTU) \qquad (1)$$

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

Compressor

A polytropic model based on actual physical dimensions is used to model the reciprocating compressor (Threlkeld 1970 and Chlumsky 1965). The polytropic exponent (n) of the refrigerant determines the relationship between the entering and exiting states. The power used by the compressor is calculated from the flow rate of the refrigerant, the enthalpy change in the refrigerant, and the polytropic efficiency of the compressor. The polytropic process is defined by

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

The compressor work is given in terms of the polytropic efficiency by

$$W_{comp} = \frac{\dot{m}_{ref} \Delta h_{ref}}{\eta_{polytropic}} \tag{4}$$

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, and is cooled by the ammonia and then recirculated to the refrigerated case. The ammonia enters as a mixture of liquid and vapor after leaving the expansion valve. All expansion valves are assumed to be isenthalpic. As the ammonia cools the secondary fluid, it evaporates and the saturated ammonia vapor flows to the compressor where it is compressed. 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):

$$Nu_D = 3.66 + \frac{0.0668\left(\frac{D}{L}\right) Re_D Pr}{1 + 0.4\left[\left(\frac{D}{L}\right) Re_D Pr\right]^{2/3}} \tag{5}$$

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

$$Nu_D = 0.036 Re_D^{0.8} Pr^{1/3} \left(\frac{D}{L}\right)^{1/18} \tag{6}$$

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 for pool boiling is used (Rohsenow 1952):

$$h = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma}\right]^{1/2} \left[\frac{Cp_l}{C_{sf} h_{fg} (Pr_l)^{1.7}}\right]^3 (T_{wall} - T_{saturated})^2 \tag{7}$$

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 (Chapman 1984).

Refrigerated Case

The refrigerated case is modeled as a heat exchanger which cools the air circulating in the refrigerated case with a secondary fluid flowing through a tube bundle oriented cross-flow to the air stream. Standard heat exchanger modeling techniques are used to determine the overall heat transfer coefficient. The secondary fluid flow through the pipes in the heat exchanger is assumed to be developing hydrodynamically and developed thermally, and the Hausen correlation for laminar flow and the Nusselt correlation for turbulent flow are again used. 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):

$$Nu_D = 0.3 + \frac{0.62 Re_D^{1/2} Pr^{1/3}}{\left[1 + (0.4/Pr)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re_D}{2.82 \times 10^5}\right)^{5/8}\right]^{4/5} \tag{8}$$

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 cross-flow heat exchangers with both fluids unmixed (Incropera and Dewitt 1985):

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right)(NTU)^{0.22}\left\{\exp\left[-C_r (NTU)^{0.78}\right] - 1\right\}\right] \tag{9}$$

The heat transfer in the case heat exchanger is the load met by the refrigerated case. Either the temperature change of the secondary fluid can be provided to calculate the case load, or the required load can be specified to determine the necessary temperature change.

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):

$$Nu_D = 0.023 Re_D^{0.8} Pr^{0.4} \tag{10}$$

The pipes are exposed to the air of the supermarket, and a constant heat transfer coefficient of 6 W/m^2-C (1.05 BTU/hr-ft^2-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. 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 work is to calculate the head losses in the pipes. The head losses arise from the friction losses and the minor losses due to bends and valves. The friction losses are calculated using the friction factor (f) from the Moody diagram (White 1986):

$$\text{head} = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right) \tag{11}$$

The head losses from the minor losses due to bends and valves in the piping system depend on the number and type of the 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 of the equivalent loss coefficients (White 1986). For the calculations completed in this study it was assumed that each pipe had a sharp entrance and exit. This equates to a K_{eq} of 8 per pipe for each pipe in the heat exchangers and for the supply and return pipes:

$$\text{head} = K_{eq}\left(\frac{v^2}{2g}\right) \tag{12}$$

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 work are calculated. The pump work is calculated accounting for the pressure drop in the distribution lines, the heat exchanger in the refrigerated case, and the heat exchanger with ammonia. Motor and mechanical inefficiencies are not included in the pump work.

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

$$\text{COP} = \frac{\text{Refrigeration Load}}{W_{\text{compressor}} + W_{\text{pump}}} \tag{13}$$

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 making use of intercooling between the stages. Three methods of staging the compression were compared for use with ammonia and R22 (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 and the intercooling.

The second method, known as staged compression and evaporation, differs from basic staging because 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. 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 as low a 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 and less mass flow of refrigerant is needed to meet the refrigeration load, reducing the size of the first compressor stage.

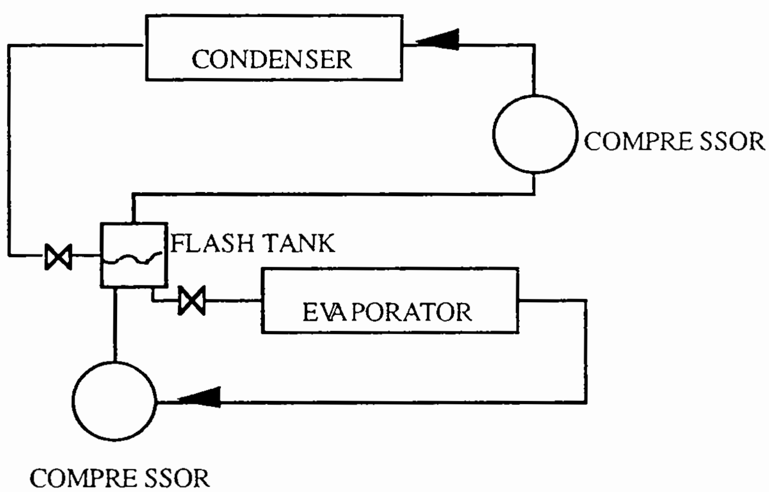

Figure 3: Refrigeration cycle with staged compression and flash tank

The 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 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 the staged compression and evaporation method. An optimal intermediate pressure, discussed in the section on Design Rules, gives the highest performance.

The models of the different staging methods with R22 and ammonia as the refrigerant were compared for evaporator temperatures of 244 and 267 K (434 and 481°R) and a refrigeration load of 52.8 kW (15 tons). The COPs for the different systems are shown in Table 1. With R22 as the refrigerant, staged compression and evaporation has the highest performance. R22 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 R22 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 R22 in the rest of this study.

Method	Evaporating Temperature			
	244 K (434 °R)		267 K (481 °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

Table 1: Compression staging performance comparison

SECONDARY FLUID SELECTION

The secondary fluids evaluated in this study are propylene glycol, ethylene glycol, mineral oil, ethanol, propane, and a silicone-based heat transfer fluid. Propylene glycol-water solutions are used in applications where oral toxicity is a concern, such as applications with drinking water or food processing. Ethylene glycol is less viscous than propylene glycol, and it generally provides greater heat transfer and better low temperature performance. It is, however, moderately orally toxic and should be used with caution where accidental contact with food can occur. A low temperature mineral oil fluid, polyalphaolefin, is a non-toxic substance that meets the Food and Drug Administration (FDA) regulation for use as a synthetic white mineral oil for non-food 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 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 properties of the different fluids to temperature and concentration (when applicable) (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 (Eq. 3) versus the temperature difference across the refrigerated case heat exchanger. At the smaller temperature differences, a larger mass flow rate of secondary fluid is needed to meet the refrigeration load and the pump work is higher. At higher temperature differences, the temperature of the ammonia in the ammonia - secondary fluid heat exchanger needs to be lower and the compressor work is higher. With a refrigerated case temperature of 267 K (481 °R), propane has the highest performance. Propylene 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.

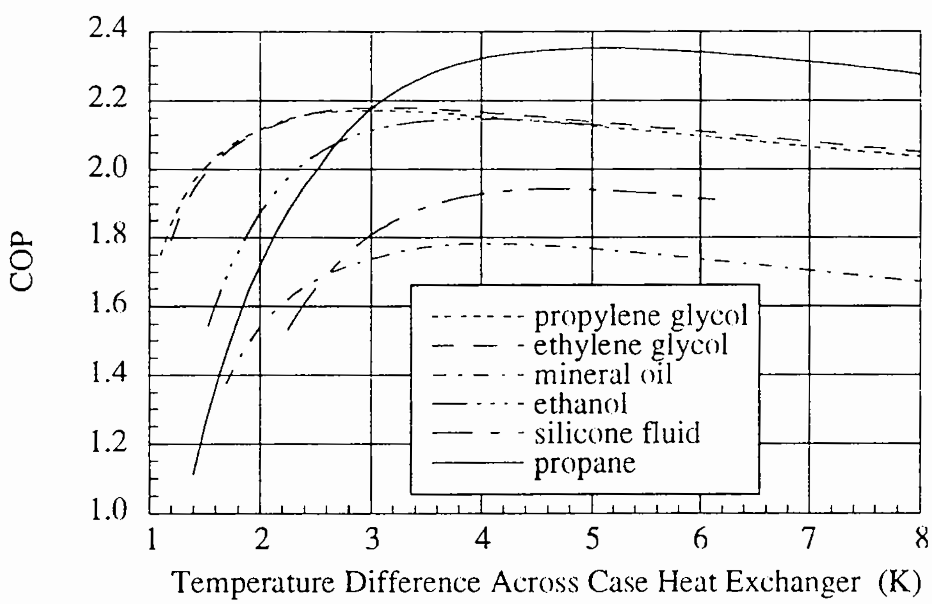

Figure 4: Performance comparison of the secondary fluids

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 non-flammable. Propylene glycol is non-toxic, 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 R22 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 R22 systems that they will replace. 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 a R22 system, it was necessary to develop a model of a R22 system that included a heat exchanger in the refrigerated case and pumping costs.

The equations used to calculate the thermal losses and the pressure drops in the supply and return lines for the refrigerated case in the R22 system model are identical to those used in the ammonia - secondary fluid system model. The refrigerated case is assumed to have R22 circulated in a tube bundle oriented cross-flow to the air stream. The heat transfer coefficients of the air flow on the outside of the pipes are calculated in the same manner as in the ammonia - secondary fluid model. The correlation equations become confounded by the forced convection during evaporation. An assumption was made that the inside heat transfer coefficient would 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 R22 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 R22 system (using staged compression and evaporation) and the ammonia with propylene glycol system (using staged compression with a flash tank) were compared at refrigerated case temperatures of 267 K (481 °R) and 244 K (434 °R). The results are shown in Table 2. The system COP for the R22 system is around 4% higher than the system COP for the ammonia with propylene glycol system at 267 K (481 °R) and 10% higher at 244 K (434 °R). 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.

Refrigerant	COP (267 K) (481 °R)	COP (244 K) (434 °R)
R22	2.84	1.40
Ammonia with propylene glycol	2.72	1.25

Table 2: Performance comparison of R22 system and ammonia with propylene glycol system

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 most intercooling and refrigeration capacity increase, 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).

$$\frac{P_{inter}}{P_{evap}} = \left[\frac{P_{cond}}{P_{evap}}\right]^X \qquad (14)$$

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 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 work is needed. If the temperature difference decreases, the ammonia temperature can be higher and the compressor work is reduced, but the pump work needed to circulate the secondary fluid increases. The highest system performance will occur at a temperature difference that balances the compressor and pump work. The relative influence of the ratio of the compressor work and the pump work was calculated for four heat exchanger and 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 work to compressor work is between 0.01 and 0.03.

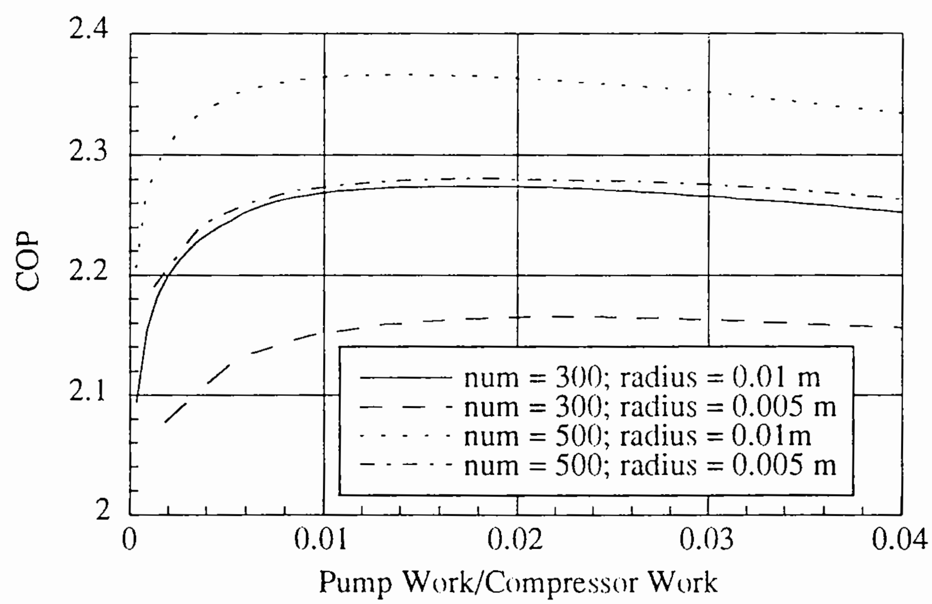

Figure 5: System performance as a function of compressor work - pump work ratio

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. 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.

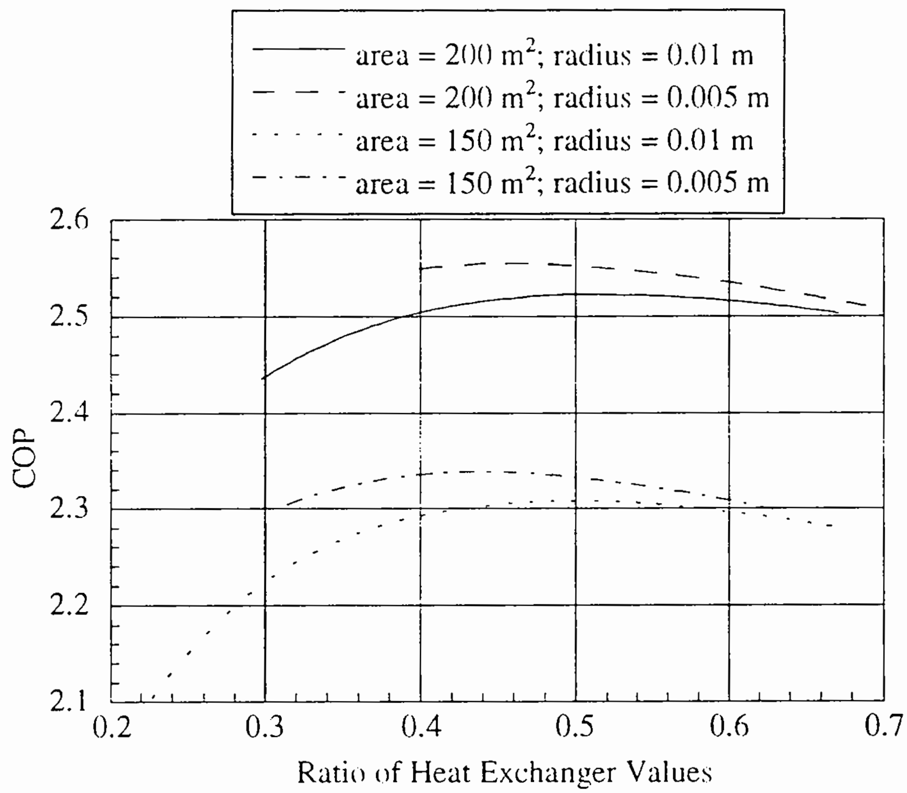

Figure 6: System performance as a function of UA value ratio

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 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, so for optimal performance the heat exchangers should be sized in such a manner 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 work 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 (481 °R) 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 (0.164 to 0.984 ft), and insulation thickness from 0.01 to 0.03 m (0.0328 to 0.0984 ft). The influence of each individual parameter is different than 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 °R), 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 (0.328 ft), and 0.03 m (0.0984 ft) of insulation.

To develop the performance maps, the combinations of the 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 (0.0328 ft) of insulation, Figure 8 with 0.02 m (0.0656 ft) of insulation, and Figure 9 with 0.03 m (0.0984 ft) of insulation.

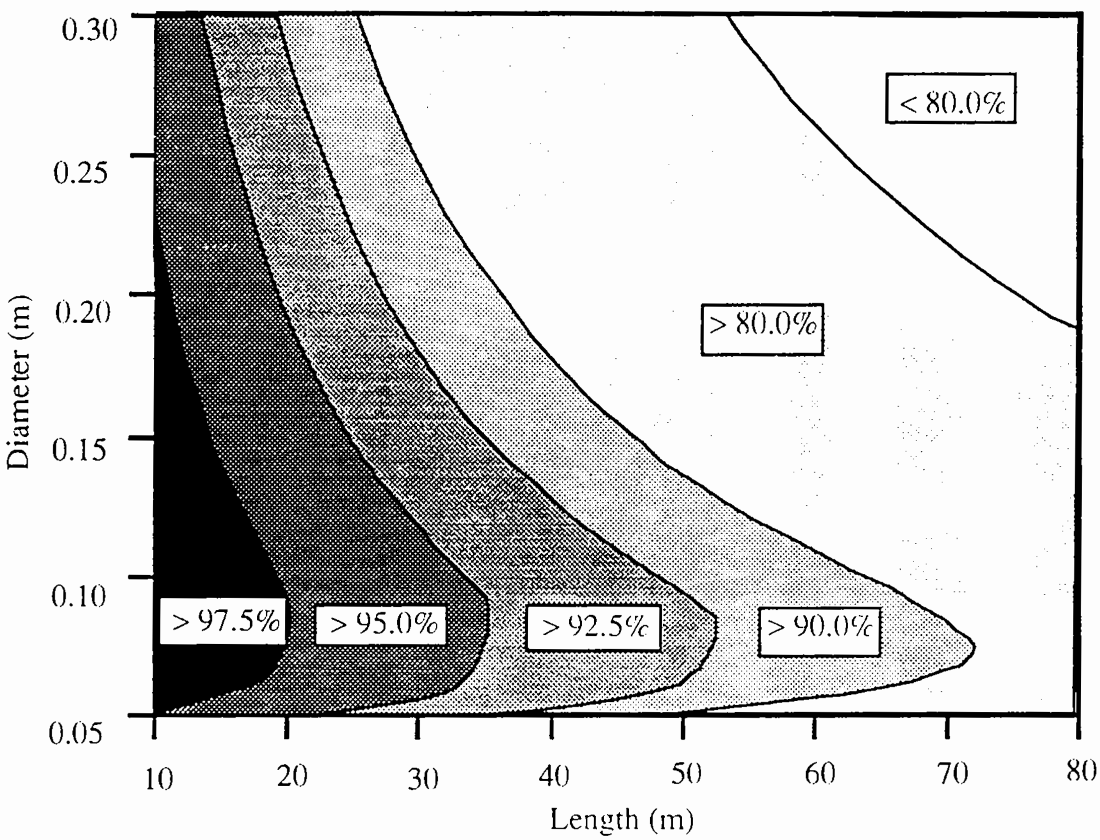

Figure 7: Performance map with 0.01 m (0.0328 ft) of insulation

The maps provide an easy way to estimate an optimal design. Assume, for example, that the supermarket requires 40 m (131.2 ft) of pipe between the two heat exchangers. Using 0.01 m (0.0328 ft) 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 (0.164 and 0.394 ft). With 0.02 m (0.656 ft) 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 (0.164 and 0.689 ft). With 0.03 m (0.984 ft) of insulation, performance between 95 and 92.5% can be attained with pipe diameters between 0.05 and 0.29 m (0.164 and 0.951 ft).

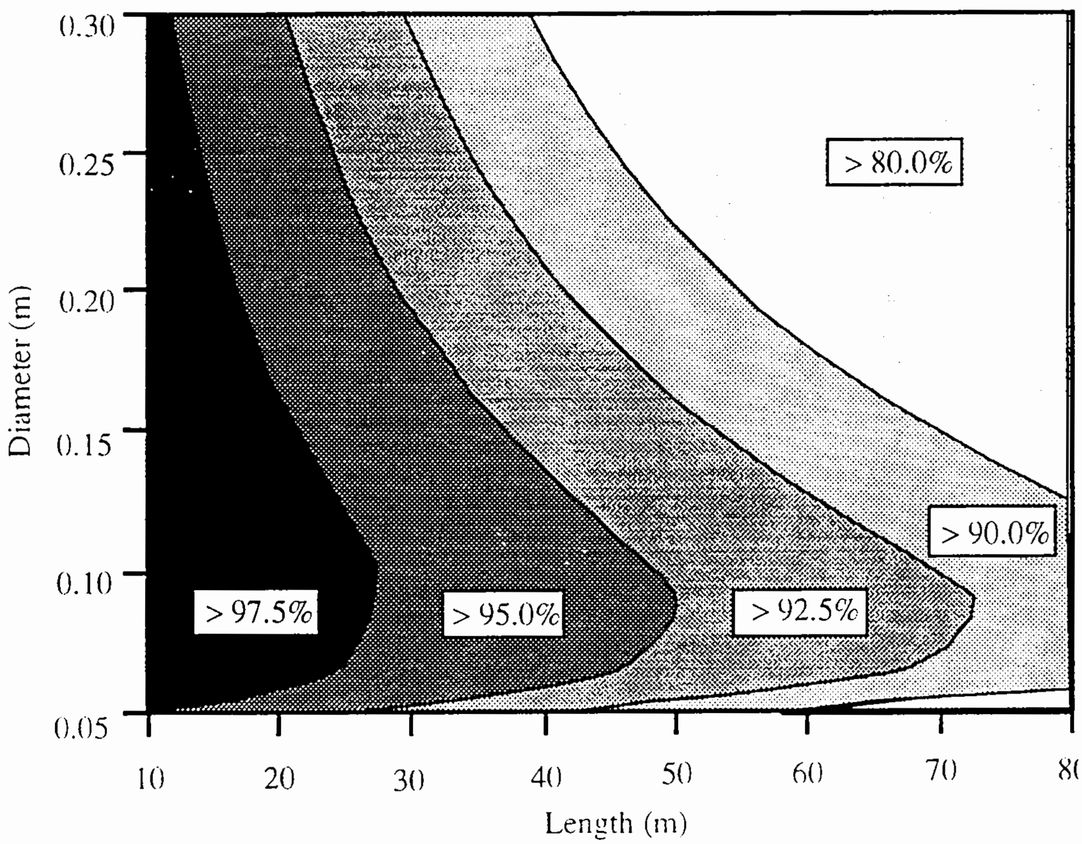

Figure 8: Performance map with 0.02 m (0.0656 ft) of insulation

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 25 m (82.0 ft) and a diameter of 0.17 m (0.558 ft) is to be used, 0.01 m (0.0328 ft) of insulation will give performance between 95 and 92.5% of the maximum, 0.02 m (0.0656 ft) of insulation between 97.5 and 95%, and 0.03 m (0.0984 ft) of insulation within 97.5% of maximum.

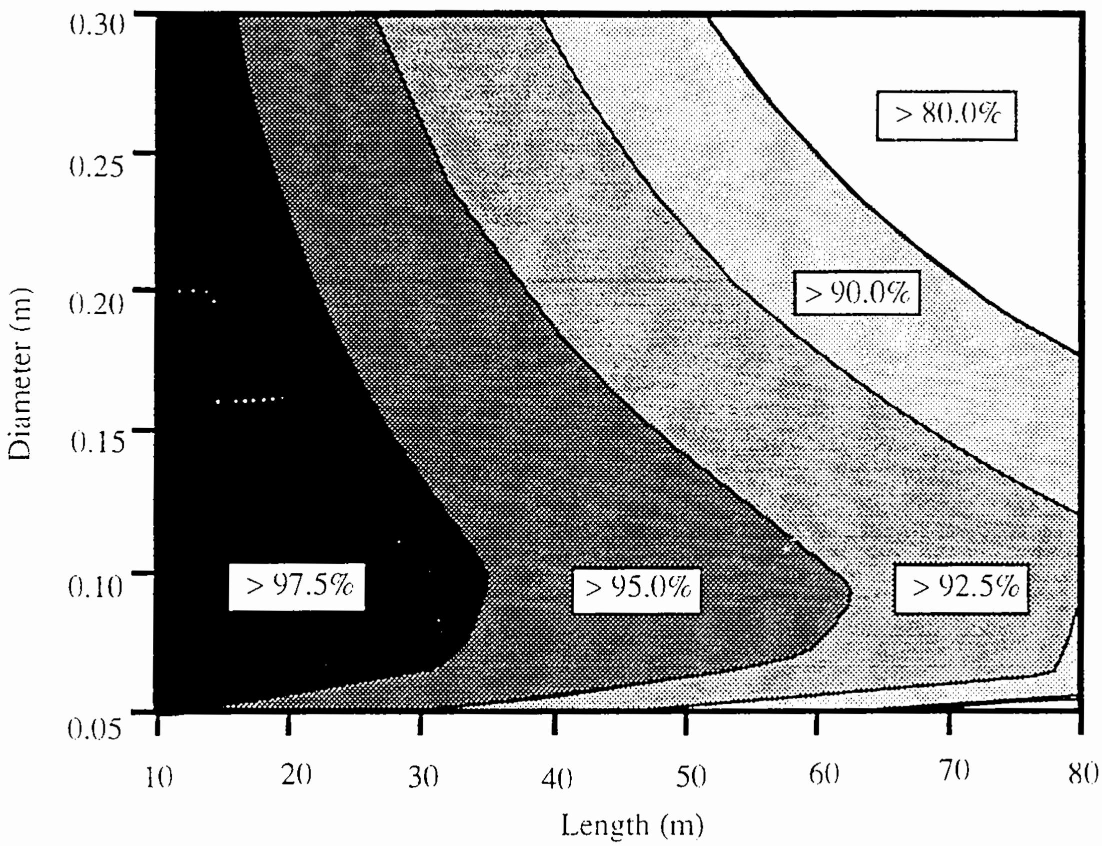

Figure 9: Performance map with 0.03 m (0.0984 ft) of insulation

CONCLUSIONS

The search for replacements for chlorinated refrigerants has focused on finding new refrigerants and mixtures. Ammonia is a proven refrigerant that has been in use for many years. When coupled with a secondary heat transfer fluid, ammonia can be used in applications where its toxicity would be a concern if it is used directly. This study shows that a well designed supermarket system that uses ammonia with propylene glycol can have a performance that is within 4 to 10% of the performance of the R22 systems currently being utilized. 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 (Eq. 4) between 0.5 and

0.6. The pump work is around 0.02 times the compressor work, and the heat exchangers are sized so that the ratio of the UA values is around 0.5. The secondary fluid piping system is selected using the performance maps (Figures 7-9) to achieve the highest performance possible. The calculations indicate that the ammonia with secondary fluid system is a possible replacement for the R22 system, and further research into improvements for the system and heat exchangers could improve the performance of the ammonia with secondary fluid system.

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NOMENCLATURE

Symbol	Definition
COP	coefficient of performance
Cp	specific heat
Cr	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	enthalpy; heat transfer coefficient
h_{fg}	heat of fusion of refrigerant
k	thermal conductivity
K_{eq}	equivalent length for minor losses
L	length
$\dot{m}$	mass flow rate
n	polytropic exponent
NTU	number of transfer units
Nu	Nusselt number
P	pressure
Pr	Prandtl number
Q	heat transfer
Re	Reynolds number
T	temperature
UA	loss coefficient
v	specific volume; velocity
W	work

X	pressure ratio exponent
Δ	difference
ε	heat exchanger effectiveness; pipe roughness
$\eta_{polytropic}$	polytropic efficiency of compressor
σ	surface tension
ρ	density

Subscripts

Symbol	Definition
air	air properties
amb	ambient conditions
comp	compressor
cond	condenser
D	diameter
evap	evaporator
in	into compressor
inter	intermediate pressure
l	liquid state
out	out of compressor
ref	refrigerant
saturated	saturated conditions
v	vapor state
wall	surface between ammonia and secondary fluid in heat exchanger

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16. ABSTRACT The paper discusses equipment configurations for the use of ammonia in supermarket refrigeration applications. International agreements have legislated the phaseout of many refrigerants currently being used in the world, including R502 and R12 which are commonly used to provide the cooling for refrigerated cases in supermarkets. R22 and ammonia (R717) are two of the refrigerants that have been proposed as replacements. This paper identifies the optimal design for ammonia-secondary fluid systems and compares their performance to that of R22 systems. Both R22 and ammonia have high discharge temperatures leaving the compressor, necessitating staged compression. Three methods of staging the compression were compared for both refrigerants. Six secondary fluids were evaluated for use with ammonia in a supermarket system. Overall system performance of the ammonia-secondary fluid refrigeration system is governed by a large set of design parameters The influence of these parameters on overall system performance was studied systematically. From this parametric study, design rules were developed, leading to optimum ammonia-secondary fluid systems. The performance of a well-designed ammonia-secondary fluid system was found to be only about 4% lower than that of an R22 system.

17. KEY WORDS AND DOCUMENT ANALYSIS

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Ammonia	Stationary Sources	07B
Refrigerators	Supermarkets	13A
Refrigerants		
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