

**INVESTIGATION OF AMMONIA AND EQUIPMENT
CONFIGURATIONS FOR SUPERMARKET APPLICATIONS**

by

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ABSTRACT

The need for replacement refrigerants has been of increasing concern with the international agreements which legislated the phasing-out of many of the refrigerants currently being used in the world. The refrigerants that are currently used to provide the cooling for the refrigerated cases in supermarkets are being phased-out and the refrigerants R22 and ammonia (R717) are two of the refrigerants that have been proposed as replacements. The choice of refrigerants for supermarket applications is important because approximately 4% of the national electrical power consumption is by supermarkets. It is important to ensure that the new refrigerants do not have significantly lower system performance. The use of R22 is a concern because it still has a potential for contributing to global warming. Ammonia does not contribute to ozone depletion or global warming but it is toxic at low concentrations. An ammonia system would have to employ a secondary heat transfer fluid to provide the cooling to the refrigerated cases. The work documented in this thesis studies the best design method for ammonia - secondary fluid systems and compares their performance to the performance of R22 systems.

Both R22 and ammonia have high discharge temperatures leaving the compressor and it is necessary to utilize staged compression. Three different methods of staging the

compression were compared for both of the refrigerants: basic staged compression, staged compression and evaporation, and staged compression with a flash tank. The best method for R22 systems is staged compression and evaporation and the best method for ammonia systems is staged compression with flash tank.

Six different secondary fluids were evaluated for use with ammonia in the supermarket system. The fluids are a propylene glycol fluid called Dowfrost, an ethylene glycol fluid called Dowtherm SR-1, a mineral oil called Multitherm 503, a silicon based heat transfer fluid called Syltherm XLT, ethanol, and propane. When the system performance was evaluated for the fluids, the system using propane yielded the highest performance with Dowfrost and Dowtherm SR-1 yielding the second highest performance. However, in a application, such as a supermarket, where the general public could come into contact with the secondary fluid, the toxicity and flammability of the secondary fluids need to be considered. Propane is highly flammable and explosive and Dowtherm SR-1 is toxic, so Dowfrost is the best choice of the six fluids evaluated.

The performance of the R22 system and the ammonia - Dowfrost system was compared. The performance of the R22 system was about 4% higher than the ammonia - Dowfrost system. With further improvements in the two heat exchangers used in the ammonia - Dowfrost system the performance could be even closer.

The overall system performance of the ammonia - secondary fluid refrigeration system is governed by a large set of design parameters. A parametric study was done of the influence that these parameters have on the overall system performance. From this parametric study, design rules for an ammonia - secondary fluid system were developed. When these design rules are utilized, the ammonia - secondary fluid will have the highest performance possible.

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NOMENCLATURE

ROMAN SYMBOLS

Symbol	Definition
caseload	refrigeration load met by refrigerated case
CFC	chlorofluorocarbon
COP	coefficient of performance
C _p	specific heat
Cr	capacitance rate ratio for heat exchangers
C _{sp}	empirical constant for pool boiling
D	diameter
f	friction factor
g	acceleration of gravity
GWP	global warming potential
h	enthalpy
h	heat transfer coefficient

HCFC	non-fully halogenated chlorofluorcarbon
head	pressure head
h_{fg}	heat of fusion
J	joule
K	degrees Kelvin
k	thermal conductivity
kg	kilogram
K_{eq}	equivalent length for minor losses
L	length
\dot{m}	mass flow rate
m	meter
m	clearance ratio
n	polytropic exponent
NTU	number of transfer units
Nu	Nusselt number
ODP	ozone depletion potential
P	pressure
Pa	Pascal
percent	volume percent of glycol
ppm	parts per million
Pr	Prandtl number
Q	heat transfer
rms	root mean square error
rp	pressure ratio

s	seconds
T	temperature
ton	ton of refrigeration
U	overall heat transfer coefficient
UA	loss coefficient
V	Volume
v	specific volume
v	velocity
W	work
W	watts
x	pressure ratio exponent

GREEK SYMBOLS

Symbol	Definition
Δ	difference
ε	effectiveness
ε	pipe roughness
η	efficiency
σ	surface tension
ρ	density

SUBSCRIPTS

Symbol	Definition
air	air properties
ammonia	ammonia properties
comp	compressor
cond	condenser
D	diameter
disp	displacement
evap	evaporator
first stage	first stage of compression
i	inside
in	into compressor
intercool	used for intercooling between compressors
l	length
lm	log mean
o	outside
out	out of compressor
overall	total pressure ratio
pipe	properties of the pipes
ref	refrigerant
sf	secondary fluid
vol	volume
wall	surface between ammonia and secondary fluid in heat exchanger

CHAPTER **ONE**

INTRODUCTION

International agreements have been made that call for the gradual phasing-out of many of the refrigerants currently in use. These include the refrigerants R12 and R502 which are commonly used to provide the cooling for supermarket refrigerated cases. Suitable replacements are needed for these refrigerants and both R22 and ammonia have been proposed. The choice of refrigerant for use in supermarket refrigerated cases is important because supermarkets consume approximately 4% of the national electrical power and approximately 30 to 50% of this electrical power is used to store and display food products. (Progressive Grocer 1985) Thus, it is important to choose a replacement refrigerant that results in the most efficient refrigeration.

1.1 Agreements Concerning Refrigerants

With the growing concern about the environment, international agreements have been struck to eliminate substances that cause both ozone depletion and global warming. This includes 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). As the CFCs and HCFCs migrate to the upper levels of the atmosphere, ultraviolet radiation from the sun decomposes the compounds releasing chlorine. The chlorine, in turn, chain reacts with the ozone reducing the concentration of ozone in the stratosphere. The ozone layer has a role in filtering a portion of the sun's ultraviolet radiation and the increase in ultraviolet radiation at ground level may affect human health, as well as damage crops and aquatic life. There is also concern over the trapping of some of the infrared radiation emitted from the earth's surface by atmospheric gases known as the "greenhouse" effect. Many scientists predict that increases in atmospheric concentrations of these "greenhouse" gases may result in a warming of the atmosphere. The major "greenhouse" gas is carbon dioxide. However, many refrigerants may contribute to the problem by being released into the atmosphere where they absorb infrared radiation or by increasing energy consumption which, in turn, increases the amount of carbon dioxide produced at fossil fuel power plants. (ASHRAE 1992)

The World Meteorological Organization Global Ozone Research and Monitoring Project has quantified the ozone depletion potential (ODP) and global warming potential (GWP) of refrigerants. (Albritten 1989) Table 1.1 lists these potentials where the ODP and GWP of refrigerant R11 is set as unity. The United Nations Environmental Programme (UNEP) established a mechanism for the international agreements concerning the use of CFCs and HCFCs in the form of "protocols". The Montreal Protocol was the

first protocol adopted in 1987. The protocol stepped down future production of CFCs to 50% of 1986 production levels by the end of the century. An amendment to the protocol in 1990 called for the complete phase-out of the production of CFCs by the year 2000 and a declaration of intent was adopted to phase-out the production of HCFCs in the next century, but no specific schedule was adopted. (ASHRAE 1992)

Refrigerant	ODP	GWP
R11	1	1
R12	0.9-1.0	2.8-3.4
R123	0.013-0.022	0.017-0.020
R125	0	0.51-0.65
R22	0.04-0.06	0.32-0.37
R115	0.3-0.5	7.4-7.6
R134a	0	0.24-0.29
R717	0	0

Table 1.1 Environmental effects of various refrigerants (Albritten 1989)

The Montreal Protocol was implemented in the United States by a rule issued by the Environmental Protection Agency (EPA) in 1987. Title VI of the Clean Air Act Amendments of 1990 legislated a production phase-out for CFCs by the end of the century with more decreasing steps than the Montreal Protocol and also included a schedule for the phase-out of HCFC production by the year 2030. Venting of either CFCs or HCFCs was prohibited beginning July 1, 1992. President Bush, in early 1992, requested that CFC producers in the U.S. cut their production in 1992 to 50% of 1986

levels and to cease production, except for small amounts needed for service requirements, by the end of 1995. This request represented an acceleration of the schedules of the Clean Air Act and the Montreal Protocol. (ASHRAE 1992)

1.2 Supermarket Refrigerants

Currently supermarket refrigeration systems utilize either refrigerant R12 or R502. R502 is an azeotropic mixture of 49% (by weight) of R22 and 51% of R115. Table 1.1 shows that R12 has a high ODP and GWP and that R115 has a lower ODP than R12 but a higher GWP. Both R12 and R502 are scheduled to be phased-out in the Montreal Protocol. The proposed replacement for both R12 and R502 in supermarket applications is the refrigerant R22. Table 1.1 shows that R22 has a relatively low ODP and GWP, however, it is a 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 than either R22 or R502. In comparison to both R22 and R502, ammonia has higher cycle efficiencies, higher heat transfer coefficients, and a higher critical temperature. Because ammonia has a pungent odor, it is easy to detect leaks in the system. Water is completely soluble in ammonia and will not freeze like it can if it is mixed with other refrigerants. Water in the system could react with compressor oils, even in an ammonia system, and should still be avoided. Oils are completely insoluble in ammonia, while oils are completely soluble in halocarbon refrigerants. So in ammonia systems, the ammonia will not foam in the crankcase of the reciprocating compressor. Removing oil in the system with ammonia will involve draining the oil off at an inactive point in the system. Table 1.1 shows that ammonia has

an ODP and a GWP of 0. Ammonia has no effect on the ozone layer because it reacts with other chemicals in the air to form benign compounds.

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. Since ammonia behaves differently with oils than other refrigerants, the method of removing oil from the refrigerant in the system is different. This is an additional alteration needed to current refrigeration systems. Ammonia is not compatible with copper and copper bearing alloys so steel and aluminum must be used as the materials in system construction. Also hermetically sealed compressors cannot be used with ammonia because the copper wiring in the motors would be destroyed. 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 compressor or by staging the compression, need to be taken to reduce the temperature. However, the largest drawback to the use of ammonia as a refrigerant is the low concentrations at which it is considered toxic. Stoecker states that the threshold limits of ammonia are 25 parts per million (ppm) in a time weighted average and 35 ppm for short time exposure limit. (Stoecker 1989) These compare with 1000 ppm time weighted average and 1250 ppm short time exposure limit for R22. Table 1.2 shows the results of an informal test of reactions to short-term exposure to ammonia.

Concentration of Ammonia (ppm)	Effects
150 to 200	affected eyes but vision not seriously impaired, breathing not affected
440	affected eyes more quickly but vision not seriously affected
600	eyes streamed within about 30 seconds, still breathable
700	tears to eyes in a few seconds, still breathable
1000	eyes stream instantly and vision impaired but not lost, breathing intolerable to most subjects, skin irritation after several minutes
1500 or greater	instant reaction was to get out of area

Table 1.2 Reactions to short-term exposure to ammonia (Stoecker 1989)

1.3 Supermarket Refrigerated Case Systems

There are many different types of refrigerated cases utilized in a single supermarket. The Hussman Corporation lists three different major uses and temperatures of refrigerated cases. (Hussman 1989) The low temperature (around -29°C (-20°F)) cases are used for storing ice cream. Frozen foods are kept in medium temperature cases that operate around -20°C (-4°F). Meats and dairy products are kept in the high temperature cases that operate around -6°C (25°F). A supermarket will usually have a single mechanical room that will house the refrigeration equipment which provides the refrigerant for all of the refrigerated cases in the supermarket. The typical load for all of the cases at one temperature regime in a single supermarket is around 52.8 kW (15 tons). (EPRI 1992 and Thornton 1991)

A refrigeration system that utilizes R22 as the refrigerant would consist of the condenser and compressor rack in the mechanical room and distribution pipes to the

refrigerated cases which act as the evaporator. The liquid refrigerant leaves the condenser and travels to the refrigerated case where it is expanded to the evaporator pressure. The liquid is then evaporated in the case, cooling the air in the refrigerated case. The vapor travels back to the mechanical room where it is compressed to the condenser pressure in the compressor and then condensed back to liquid form in the condenser. A diagram of the system is shown in Figure 1.1.

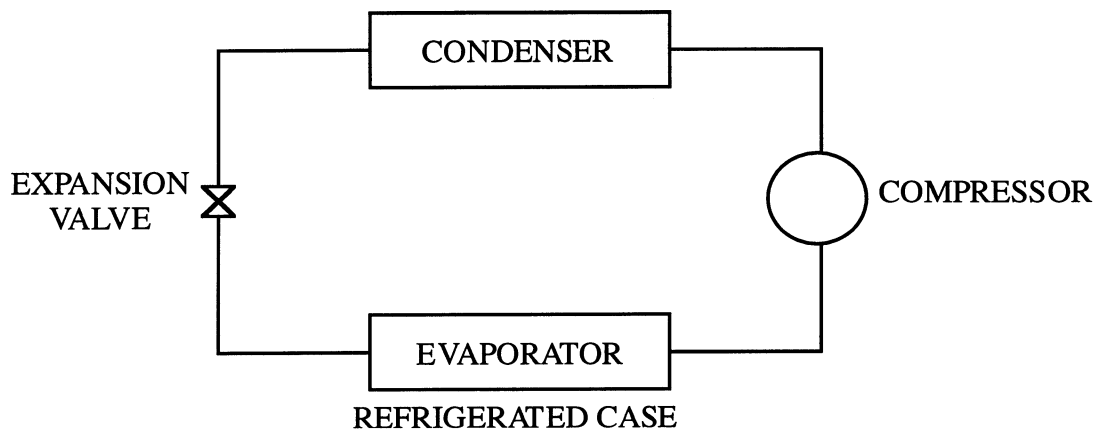


Figure 1.1 Diagram of R22 refrigeration system

Because of the toxicity of ammonia, it would be impossible to circulate ammonia throughout the supermarket. The ammonia would have to be contained in the mechanical room. To provide the cooling to the refrigerated cases, ammonia would be used to cool a secondary heat transfer fluid which would be used to cool the air around the refrigerated case. The liquid ammonia would leave the condenser and be expanded to the evaporator pressure. It would enter a flooded evaporator where the secondary fluid flowing through pipes in the liquid ammonia would evaporate the ammonia. The saturated ammonia vapor would be extracted from the top of the evaporator and compressed to the condenser pressure. The ammonia vapor would then be condensed back to liquid form. The

secondary fluid leaving the evaporator would be pumped to the refrigerated case where a heat exchanger would be used to cool the air with the secondary fluid. The warmed secondary fluid then returns to the evaporator of the ammonia system where it would be cooled. A diagram of the system is shown in Figure 1.2.

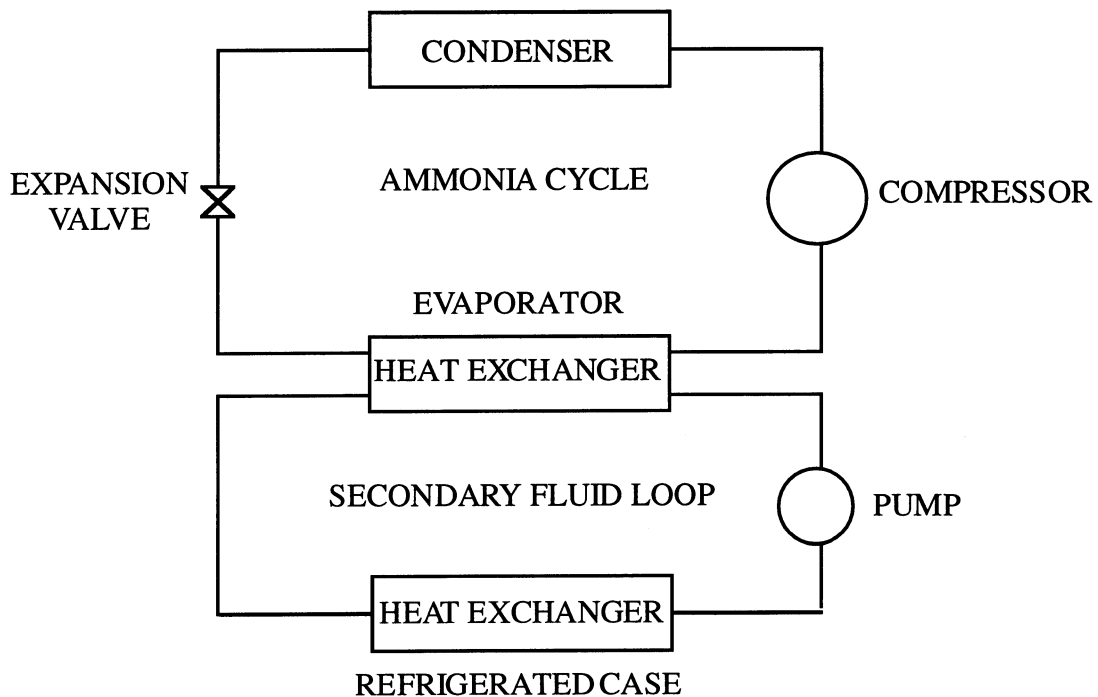


Figure 1.2 Diagram of an ammonia - secondary fluid refrigeration system

The R22 system satisfies the load of each refrigerated case by providing liquid R22 to the individual cases through individual lines. Each refrigerated case is then an individual evaporator for the amount of refrigerant provided to the case. The vapor from all of the refrigerated cases is then combined into a single compressor rack. The ammonia - secondary fluid system will utilize the flooded evaporator as the evaporator for the ammonia system. The ammonia refrigerant always stays together and is not split for each

of the refrigerated cases. The secondary fluid is divided as it is piped to the individual cases and then is recombined before it reenters the flooded heat exchanger.

1.4 Refrigerant Comparison

Because of the new laws which phase-out refrigerants currently used to provide the cooling of the refrigerated cases in supermarkets, it is necessary to select alternatives for use in supermarkets. R22 and ammonia are two refrigerants that are already in use in other applications that could be used in the supermarkets. However, R22 is likely to be phased-out in the future. Therefore, it is important to determine the best method of designing an ammonia - secondary fluid refrigeration system and comparing the performance of such a system with the performance of a system utilizing R22. This study uses computer models to evaluate ammonia - secondary fluid systems and compare the performance to the performance to R22 systems.

**CHAPTER
TWO**

SECONDARY FLUIDS

There are many different fluids available for heat transfer applications. Some work well in applications at high temperatures and some at low temperatures. For supermarket applications it is necessary to find a fluid that balances low temperature heat transfer performance with non-flammability and non-toxicity. In order to determine the best secondary fluid for this application, the thermodynamic and transport properties of six different secondary fluids were compared.

2.1 Secondary Fluid Descriptions

The important properties and information pertaining to toxicity and corrosiveness of the six different secondary fluids compared for use in the supermarket system are discussed in this section.

2.1.1 Propylene Glycol

Many chemical companies produce propylene glycol for use in heat transfer applications. Propylene glycol-water solutions are used in applications where oral toxicity is a concern, such as applications where contact with drinking water is possible or in food processing applications where accidental contact with food or beverage products is possible. These solutions can be inhibited to reduce the corrosiveness of normal propylene glycol. The propylene glycol solution produced by the Dow Chemical Company, called Dowfrost, was the solution considered in this application. (Dow Chemical Company [1]) Dowfrost consists of 95.5 percent propylene glycol with dipotassium phosphate and water, and has an operating temperature range of -46 to 121 °C (-50 to 250 °F). In solutions with water, Dowfrost provides freeze protection to below -51 °C (-60 °F) and prevents pipes from bursting to below -73 °C (-100 °F). The U.S. Department of Agriculture (USDA) lists Dowfrost as chemically acceptable for both defrosting refrigeration coils and for immersion freezing of wrapped meats, poultry and meats, and the FDA recognizes both propylene glycol and dipotassium phosphate as safe food additives under parts 182 and 184 of the Food Additive Regulations. Standard system materials can be used with Dowfrost. Steel, cast iron, copper, brass, bronze, solder, and most plastic piping materials are generally acceptable for use, as well as normal pipe fittings, valves, and pumps. When Dowfrost is mixed with water it is not flammable, as long as the percentage of propylene-glycol remains less than 80%.

Dowfrost contains inhibitors that prevent corrosion by passivating the surface of metals to prevent acids from attacking it and by buffering acids formed from glycol oxidation.

The thermophysical properties of Dowfrost depend on its concentration when mixed with water. The initial concern is that the mixture freezing point must be below the minimum temperature reached by the system. Dow provides data for the freezing point of Dowfrost based on volume percentages. (Dow Chemical Company [1]) The graph of the freezing point of Dowfrost based on the volume percentages is shown in Figure 2.1. For volume percentages of Dowfrost greater than 60%, Dow simply states that the freezing point is lower than $-51\text{ }^{\circ}\text{C}$ ($-60\text{ }^{\circ}\text{F}$).

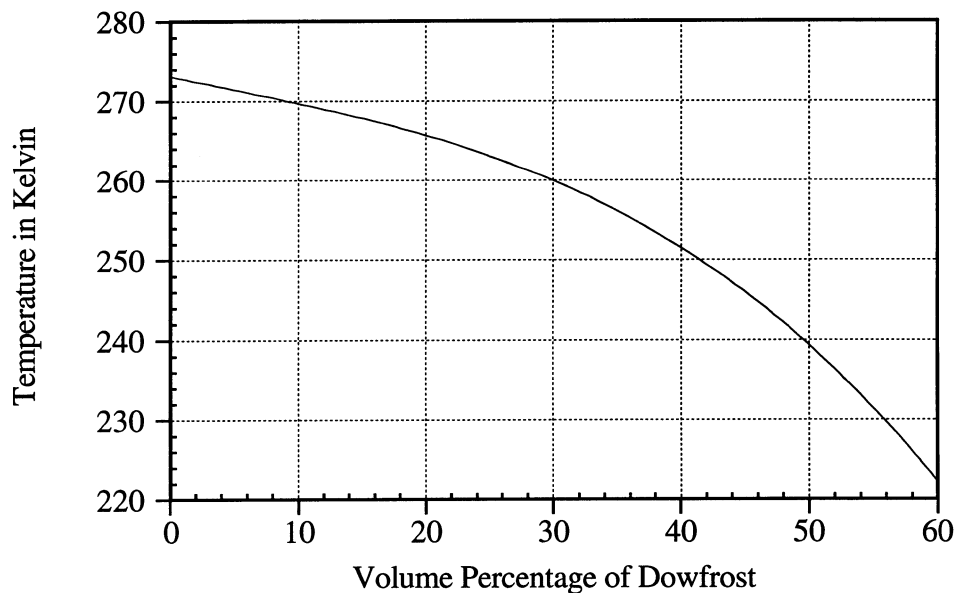


Figure 2.1 The freezing point of Dowfrost based on the volume percentage in water

It is important to ensure that freezing does not take place in the refrigeration system, so it is necessary to select a volume percentage of Dowfrost which allows for a freezing

temperature 10 to 15 °C below the lowest anticipated temperature. For the supermarket application studied here, a volume percentage of 60% is necessary to ensure the proper safety against freezing in the lines.

Other properties also depend on the volume percentage of Dowfrost. These include density, specific heat, thermal conductivity, and viscosity. Each of these properties influences the performance of a refrigeration system. Dow provides data for all of these properties for the different volume percentages of Dowfrost. The graphs of these properties are shown in Figures 2.2 through 2.5.

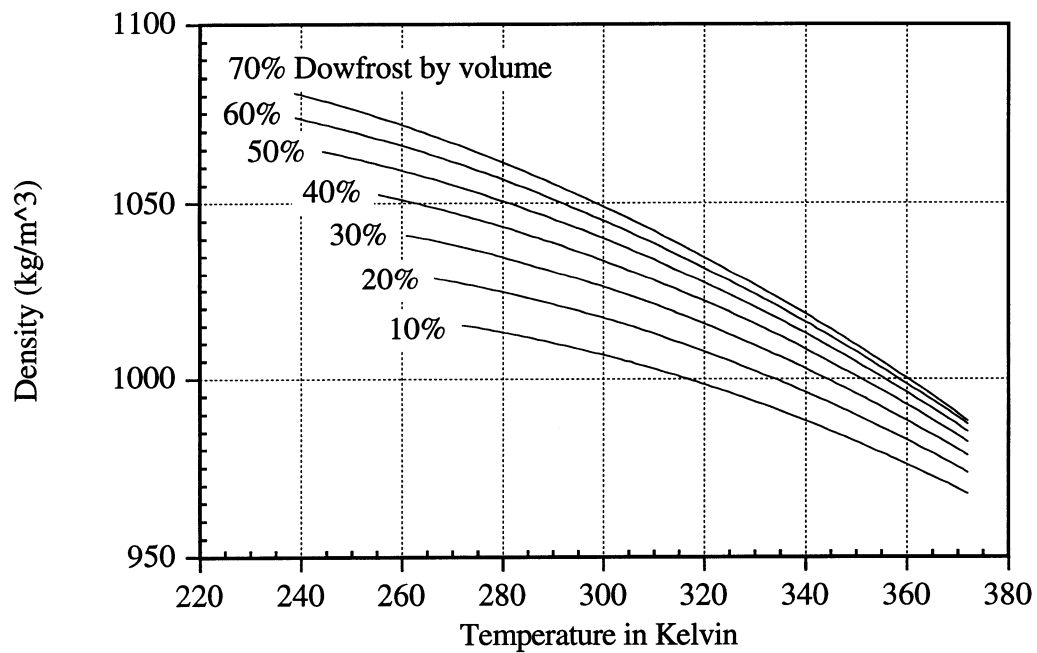


Figure 2.2 Density of Dowfrost

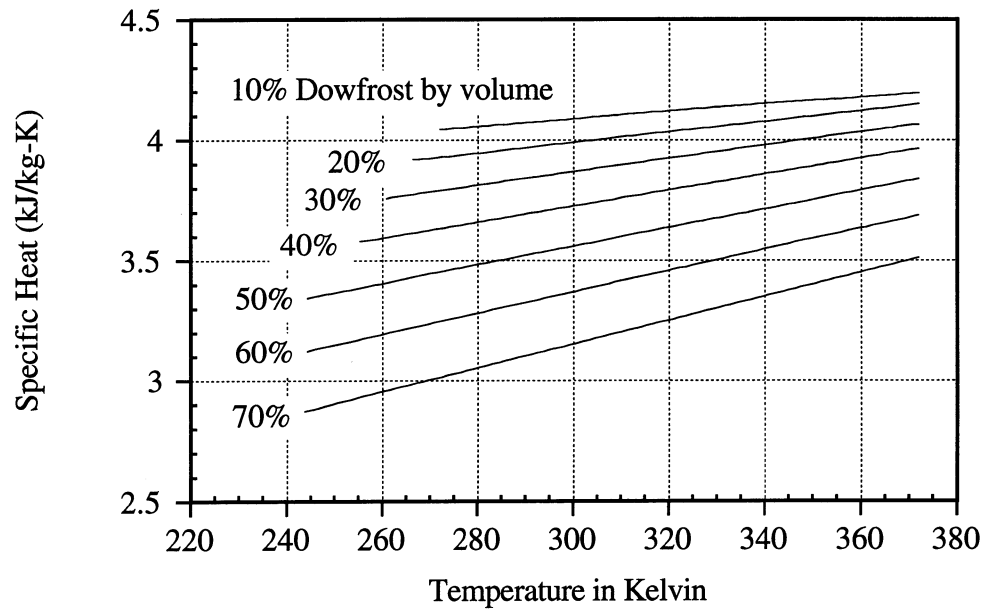


Figure 2.3 Specific heat of Dowfrost

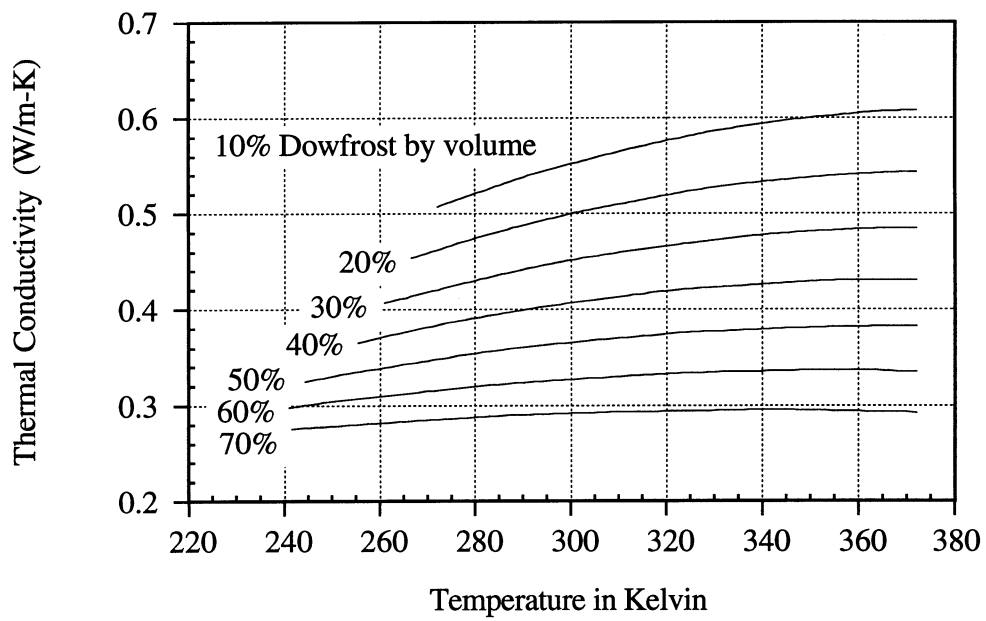


Figure 2.4 Thermal conductivity of Dowfrost

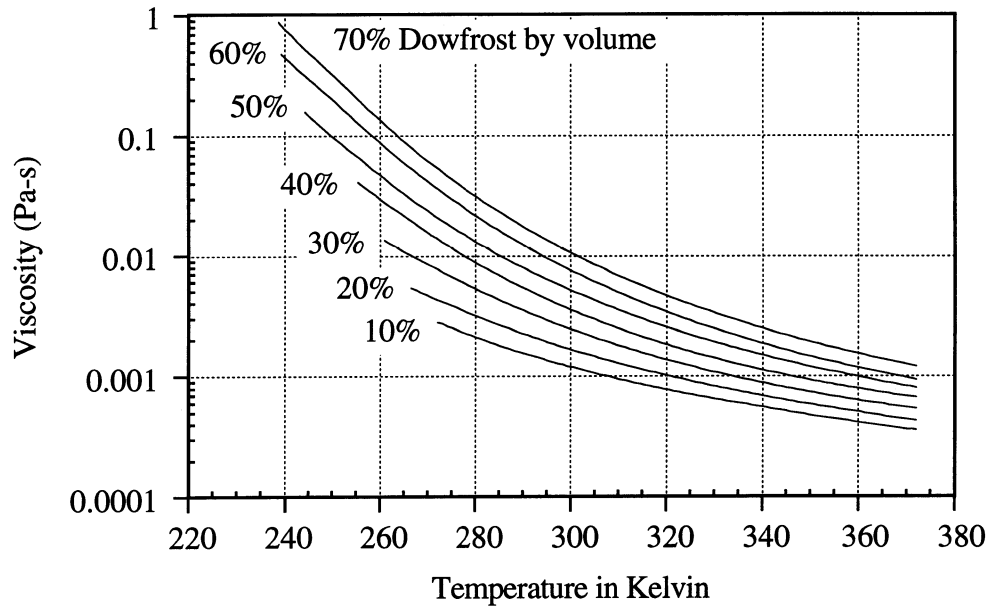


Figure 2.5 Viscosity of Dowfrost

In order to make use of these data in computer simulations, it was necessary to develop equations to calculate the properties based on the volume percentage and the temperature of Dowfrost. The data was fitted not only in terms of volume percentage, but also in terms of the temperature to allow for the properties to be calculated at any temperature for any volume percentage of Dowfrost. The equations that were developed are shown here, where percent stands for the volume percent of Dowfrost and T is the temperature in degrees Kelvin at which the properties are desired. Density is calculated in kg/m^3 , thermal conductivity in W/m-K , specific heat in kJ/kg-K , and viscosity in Pa-s .

$$\begin{aligned} \text{Specific Heat} = & (3.865 - 0.02369\text{percent} - 0.00011278 \text{ percent}^2) \\ & + (0.001024 + 5.6634\text{e-}5\text{percent})T \end{aligned} \quad (2.1)$$

$$\begin{aligned} \text{Density} = & (875.547 + 2.1514\text{percent}) + (1.1191 - 0.00076\text{percent} \\ & - 4.924\text{e-}5\text{percent}^2)T + (-0.002378 - 9.1377\text{e-}6\text{percent} \\ & + 1.0872\text{e-}7\text{percent}^2)T^2 \end{aligned} \quad (2.2)$$

$$\begin{aligned} \text{Thermal Conductivity} = & (-0.78595 + 0.015562\text{percent} - 4.89335\text{e-}5\text{percent}^2) \\ & + (0.0076866 - 0.0001156\text{percent} + 3.66034\text{e-}7\text{percent}^2)T \\ & + (-9.9977\text{e-}6 + 1.4561\text{e-}7\text{percent} - 4.5879\text{e-}10\text{percent}^2)T^2 \end{aligned} \quad (2.3)$$

$$\text{Viscosity} = \exp \left[\begin{aligned} & (71.639 - 0.6698T + 0.001915T^2 - 1.8588\text{e-}6T^3) \\ & + (0.2702 - 0.00123T + 1.5045\text{e-}6T^2)\text{percent} \end{aligned} \right] \quad (2.4)$$

To show how these correlation equations compare with the manufacturer's data, graphs were completed for a temperature range of -34 to 54 °C (-30 to 130 °F) for three different volume percentages of Dowfrost in water. These plots, shown in Figures 2.6-2.9, show that there is a good agreement between the correlation and the data. In order to quantify the quality of the fit between the correlation and the data, a root mean square error was calculated for each. This was done by finding the difference between the correlation and the data at each data point. These differences were then squared and summed. By dividing by the number of data points and taking the square root of the result the root mean square (rms) was found. To normalize the rms, the value was divided by the average of the data. The values of rms for the correlations are indicated on the graphs. The differences should not be great enough to cause major errors in the simulations.

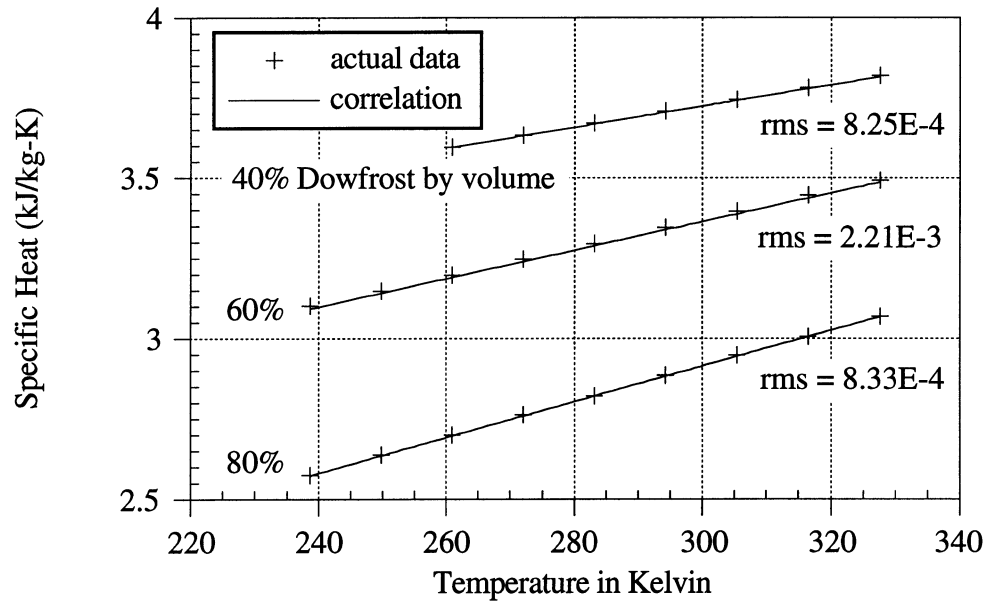


Figure 2.6 Comparison of correlation equation and actual data of the specific heat of a mixture of Dowfrost and water.

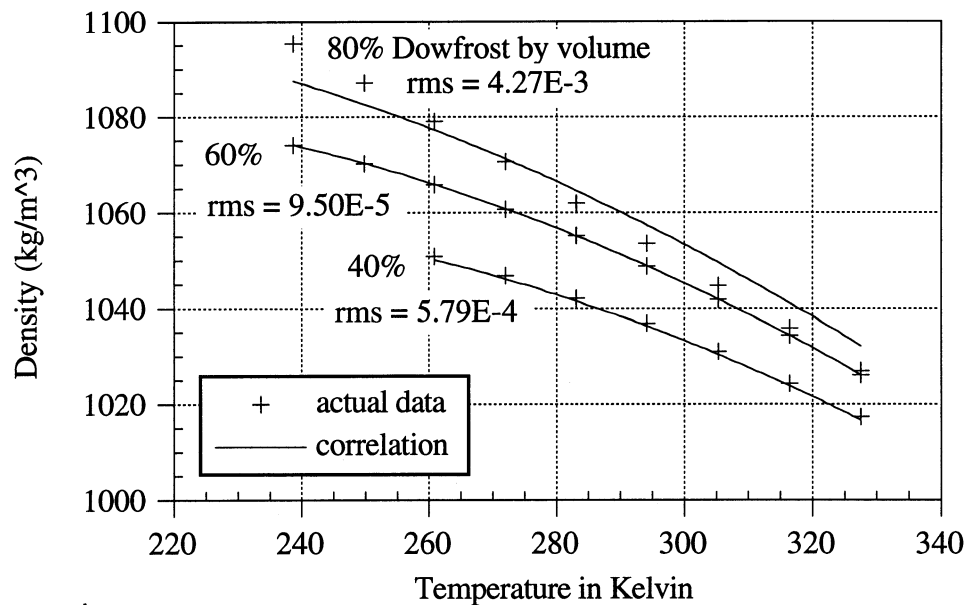


Figure 2.7 Comparison of correlation equation and actual data of the density of a mixture of Dowfrost and water.

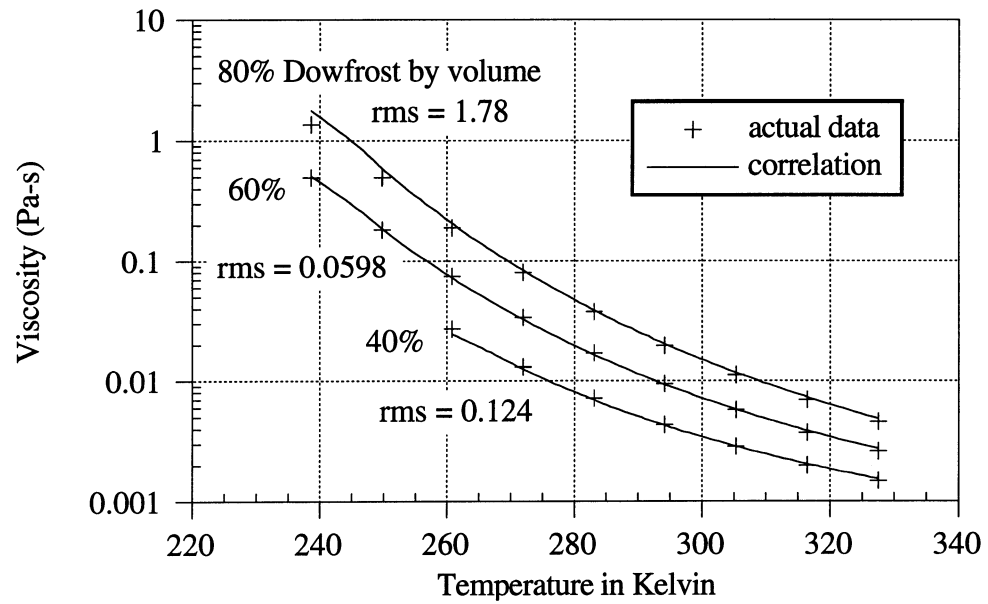


Figure 2.8 Comparison of correlation equation and actual data of the viscosity of a mixture of Dowfrost and water.

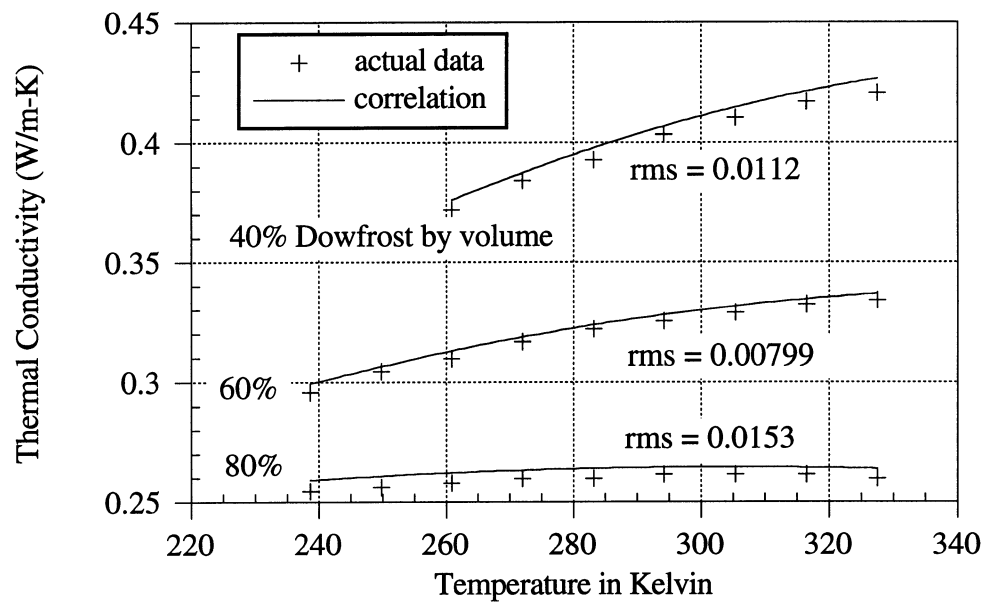


Figure 2.9 Comparison of correlation equation and actual data of the thermal conductivity of a mixture of Dowfrost and water.

2.1.2 Ethylene Glycol

Ethylene glycol is another popular heat transfer fluid that is manufactured by many different chemical companies. Since ethylene glycol is less viscous than propylene glycol, it generally provides greater heat transfer efficiencies and better low temperature performance. It is, however, moderately oral toxic and should be used with caution where accidental contact with food can occur. The Dow Chemical Company manufactures a family of inhibited ethylene glycol under the name of Dowtherm. (Dow Chemical Company [2]) Dowtherm SR-1 consists of 95.4% ethylene glycol along with inhibitors and has an operating range of -51 to 121 °C (-60 to 250 °F). In solutions with water, it provides freeze protection to below -51 °C (-60 °F) and burst protection to below -73 °C (-100 °F). Dowtherm SR-1 is listed as chemically acceptable by the U.S. Department of Agriculture (USDA) for defrosting refrigeration coils in food processing plants assuming that good manufacturing practices are used to prevent direct or indirect contact of the ethylene glycol fluid with edible products. The construction material restrictions for Dowtherm SR-1 are almost exactly the same as for Dowfrost. Steel, cast iron, copper, brass, bronze, solder and most plastics are generally acceptable. Aluminum can be used at temperatures below 66 °C (150 °F) and pipe fittings and valves can be made of ordinary steel or ductile iron. Dowtherm SR-1 is only flammable in volume percentages greater than 80%. The inhibitors against corrosion used in Dowtherm SR-1 work by first passivating the surface of metals to prevent acids from attacking it and second, by buffering any acids formed from glycol oxidation.

The thermophysical properties of Dowtherm SR-1 depend on the concentration of the mixture with water. The freezing point of the mixture is a great concern in the cold temperature applications studied here, and it is necessary to utilize a volume percentage of

Dowtherm SR-1 that will have a low enough freezing point. A graph of the freezing point as a function of the volume percentage of Dowtherm SR-1 is shown in Figure 2.10.

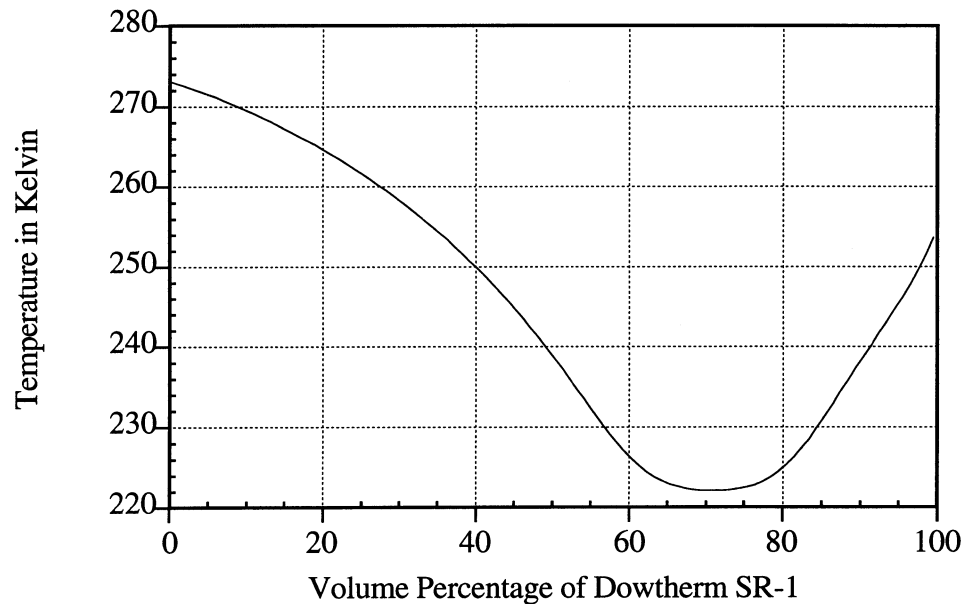


Figure 2.10 Freezing point of Dowtherm SR-1 based on volume percentage in water

Again, to ensure protection from freezing in the lines carrying the ethylene glycol, it is important to pick a volume percentage of Dowtherm SR-1 that allows for a freezing temperature 10 to 15 °C below the lowest anticipated temperature. For this application, volume percentages between 50 and 90 percent give this temperature allowance.

The properties, such as density, thermal conductivity, specific heat and viscosity, also vary with the volume percentage of Dowtherm SR-1 and the temperature of the mixture. Data for these properties were provided by the Dow Chemical Company and graphs of these are shown in Figures 2.11 through 2.14. (Dow Chemical Company [2])

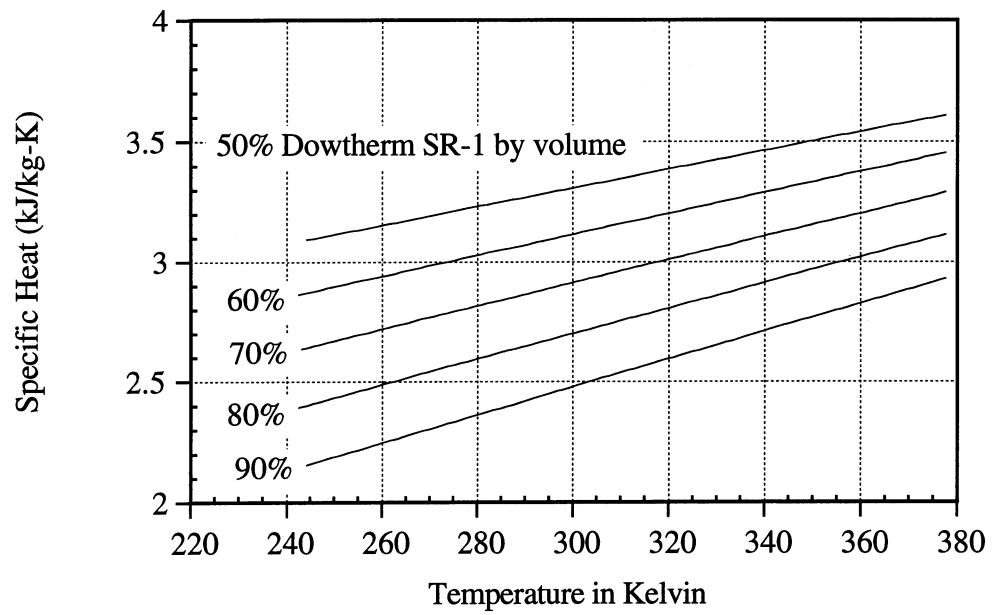


Figure 2.11 Specific heat of Dowtherm SR-1

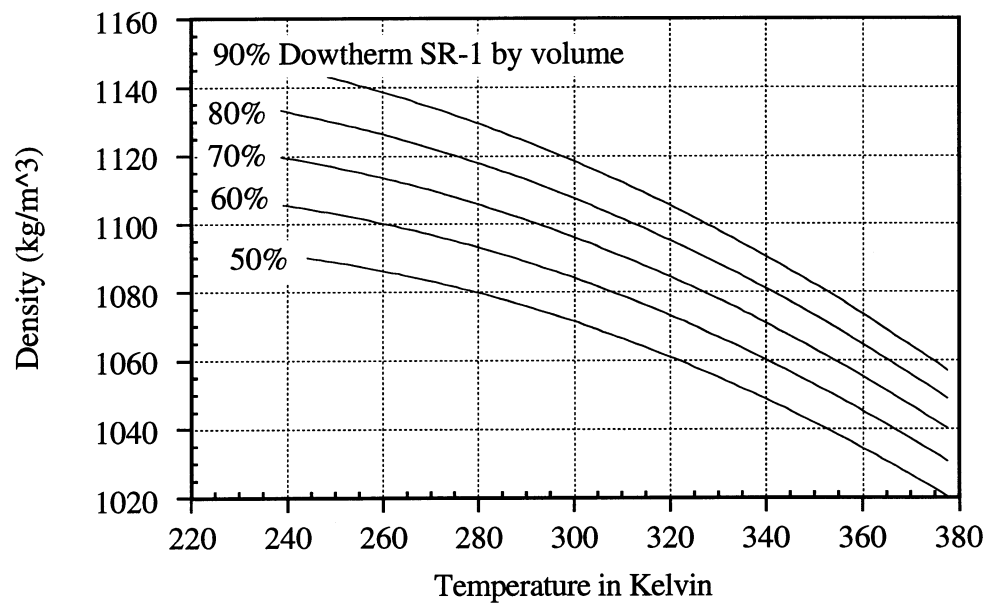


Figure 2.12 Density of Dowtherm SR-1

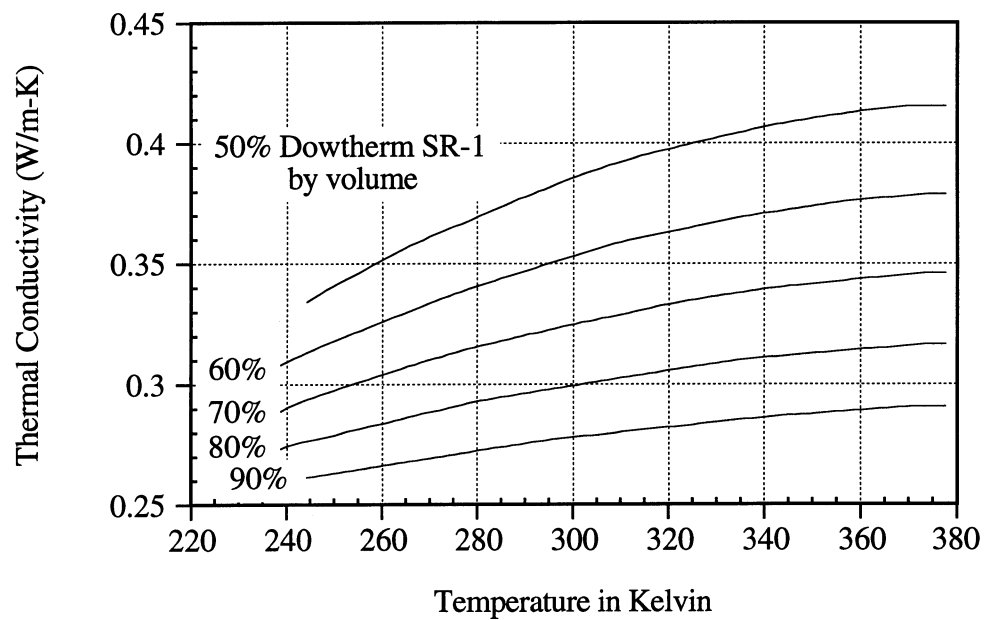


Figure 2.13 Thermal conductivity of Dowtherm SR-1

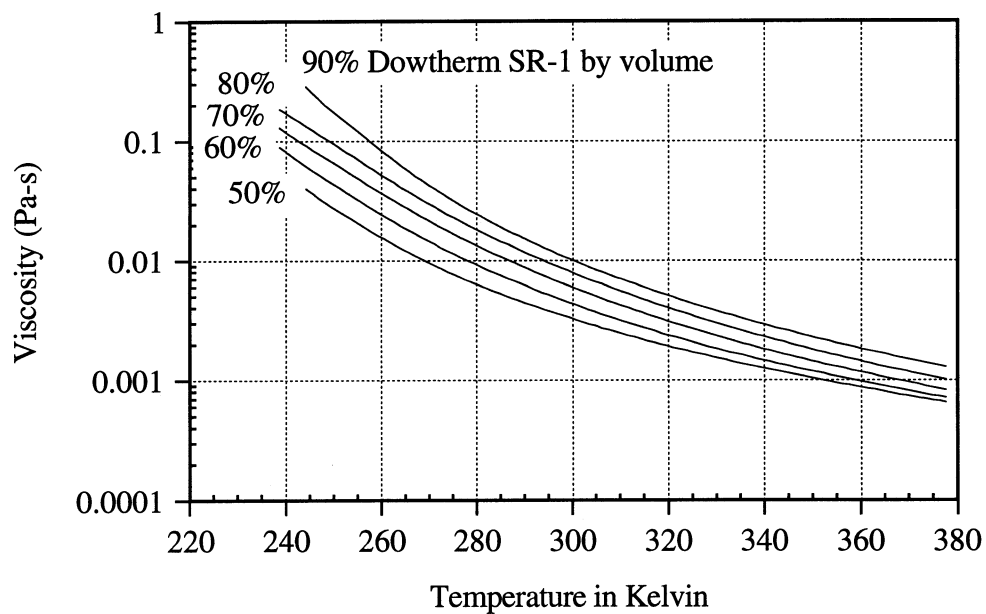


Figure 2.14 Viscosity of Dowtherm SR-1

As with Dowfrost, it was necessary to develop correlation equations for the properties of Dowtherm SR-1 to be used in the computer simulations. The correlation equations are shown here, where percent is the volume percent of Dowtherm SR-1 and T is the temperature in degrees Kelvin. Density is calculated in kg/m³, thermal conductivity in W/m-K, specific heat in kJ/kg-K, and viscosity in Pa-s.

$$\begin{aligned} \text{Density} = & (884.53 + 2.1741\text{percent}) + (1.1613 - 0.0033403\text{percent})T \\ & + (-0.0024393 + 2.994\text{e-}8\text{percent})T^2 \end{aligned} \quad (2.5)$$

$$\text{Specific Heat} = (3.9189 - 0.035267\text{percent}) + (0.0014555 + 4.8423\text{e-}5\text{percent})T \quad (2.6)$$

$$\begin{aligned} \text{Thermal Conductivity} = & (-0.84402 + 0.016948\text{percent} - 6.9969\text{e-}5\text{percent}^2) \\ & + (0.0079877 - 0.00012444\text{percent} + 5.0041\text{e-}7\text{percent}^2)T \\ & + (-1.0647\text{e-}5 + 1.7090\text{e-}7\text{percent} - 7.0658\text{e-}10\text{percent}^2)T^2 \end{aligned} \quad (2.7)$$

$$\text{Viscosity} = \exp \left[\begin{aligned} & (970.432 - 10.0014T + 0.0340567T^2 - 3.86137\text{e-}5T^3) \\ & + (-27.0361 + 0.279956T - 0.000960623T^2 \\ & + 1.09418\text{e-}6T^3)\text{percent} + (0.196245 - 0.00202259T \\ & + 6.92206\text{e-}6T^2 - 7.87103\text{e-}9T^3)\text{percent}^2 \end{aligned} \right] \quad (2.8)$$

Graphs were prepared to show the fit between the correlation equations and the manufacturer's data for these properties. The graphs of the properties for three different volume percentages of Dowtherm SR-1 at a temperature range of -34 to 54 °C (-30 to 130 °F) are shown here in Figures 2.15 to 2.18. The root mean square error was calculated for Dowtherm SR-1 in the same manner as for Dowfrost and the values are shown on the graphs. The graphs display a very close agreement between the manufacturer's data and the correlation equations.

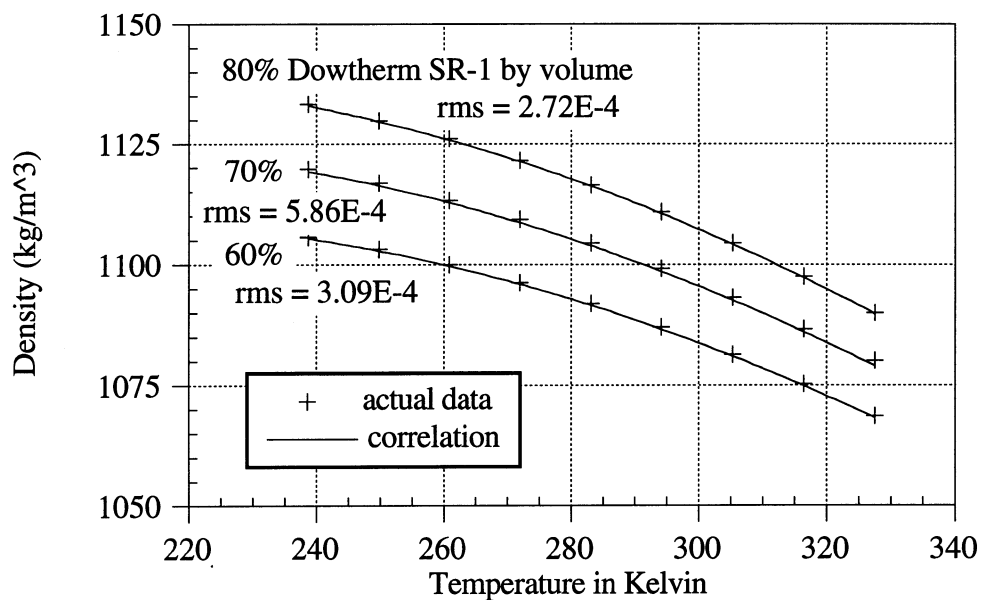


Figure 2.15 Comparison of correlation equation and actual data of the density of mixtures of Dowtherm SR-1 and water.

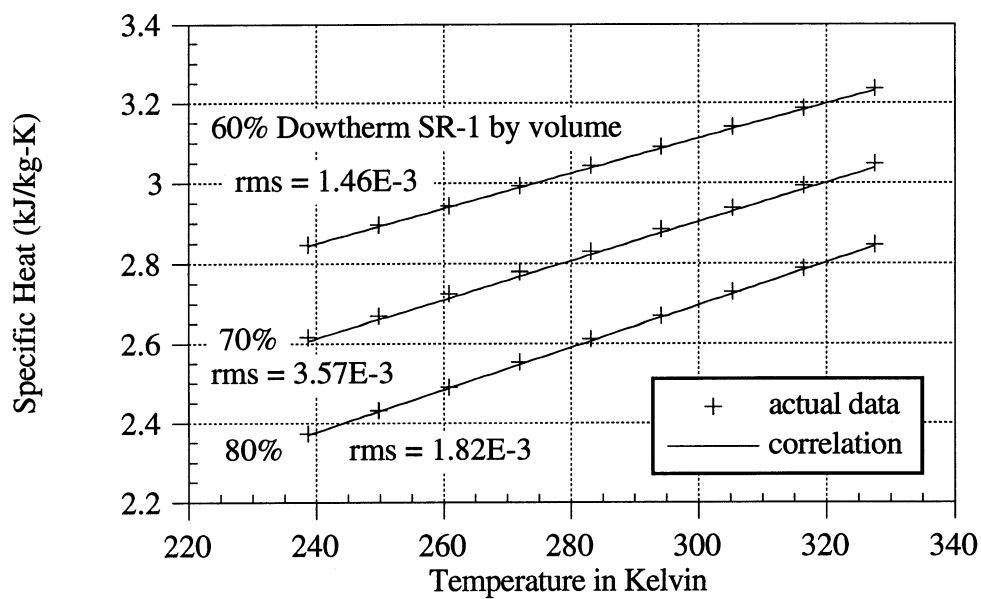


Figure 2.16 Comparison of correlation equation and actual data of the specific heat of mixtures of Dowtherm SR-1 and water.

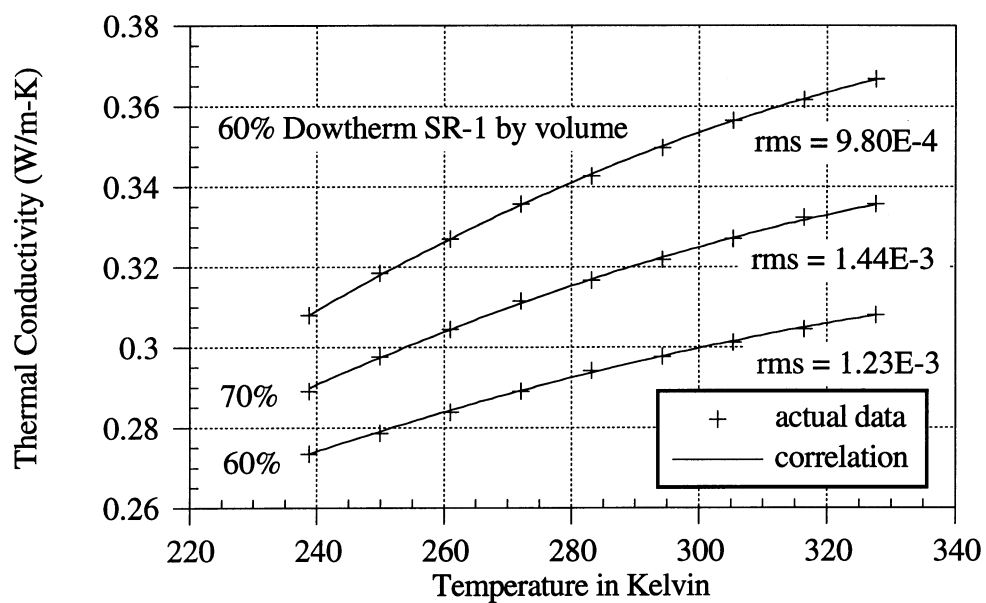


Figure 2.17 Comparison of correlation equation and actual data of the thermal conductivity of mixtures of Dowtherm SR-1 and water.

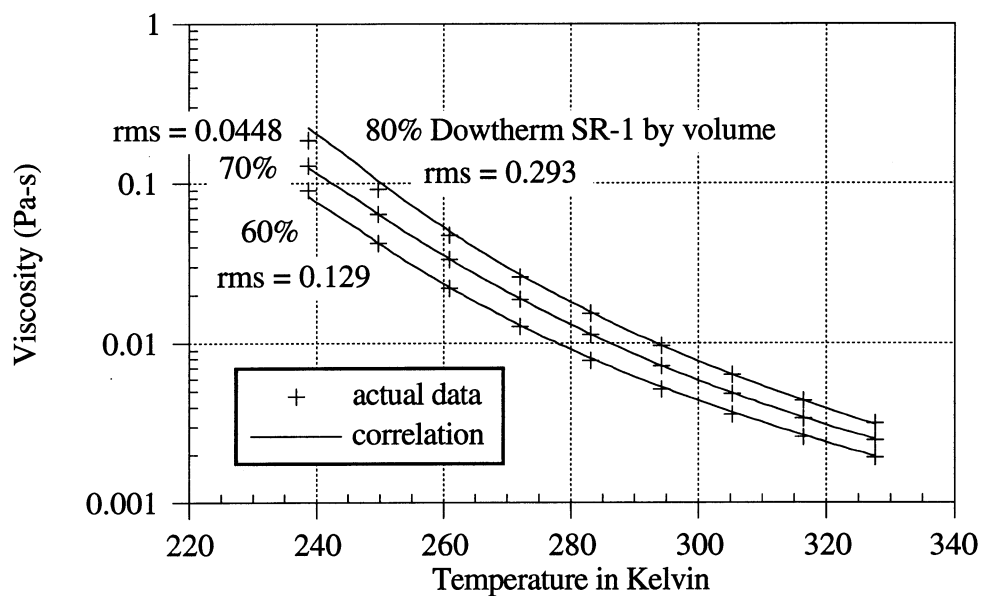


Figure 2.18 Comparison of correlation equation and actual data of the viscosity of mixtures of Dowtherm SR-1 and water.

2.1.3 Mineral Oil

A third common heat transfer fluid is mineral oil. There are many different substances that fall under the name mineral oil and each has different properties. The Multitherm Corporation produces a large family of mineral oil fluids for heat transfer purposes. (Multitherm Corporation) Their low temperature fluid, which is polyalphaolefin, is produced under the name Multitherm 503. It is a non-toxic substance that meets the FDA regulation 21CFR178.3620(b) for use as a synthetic white mineral oil for non-food articles in contact with food. It can be specified for many food, beverage, and pharmaceutical applications. It is also non-corrosive, so it can be used with all construction materials. Multitherm 503 does not irritate eyes or skin, should not present an inhalation hazard at ambient temperatures, and ingestion is non-toxic. However, it does act as a laxative when ingested. The flash point of Multitherm 503 is 160 °C (320 °F) and the autoignition point is 324 °C (615 °F). Fires must be extinguished with a dry chemical or foam rather than water. Combustion of Multitherm 503 will produce carbon monoxide and other asphyxiates. However, the fire hazard rating of Multitherm 503 is slight.

The Multitherm Corporation has provided property data for Multitherm 503. (Multitherm Corporation) Multitherm 503 is considered pumpable down to -58 °C (-72 °F). As with Dowfrost and Dowtherm SR-1, correlation equations were written for the data for use in computer simulations. The properties of Multitherm 503 depend only on the temperature of the fluid and are shown here where T stands for temperature in degrees Kelvin. Density is calculated in kg/m³, thermal conductivity in W/m-K, specific heat in kJ/kg-K, and viscosity in Pa-s.

$$\text{Viscosity} = \exp \left[\begin{array}{l} 94.38577 - 0.934043T + 0.00328677T^2 \\ - 5.18449e-6T^3 + 3.05544e-9T^4 \end{array} \right] \quad (2.9)$$

$$\text{Thermal Conductivity} = 0.17183 - 9.9229e-5T \quad (2.10)$$

$$\text{Density} = 994.31 - 0.68367T \quad (2.11)$$

$$\text{Specific Heat} = 1.1729 + 0.0034758T \quad (2.12)$$

To check the accuracy of these correlation equations, the manufacturer's data for the properties were plotted against the correlation equations. Again to show the accuracy of the correlations, the root mean square error is shown. The graphs are shown in Figures 2.19 through 2.22.

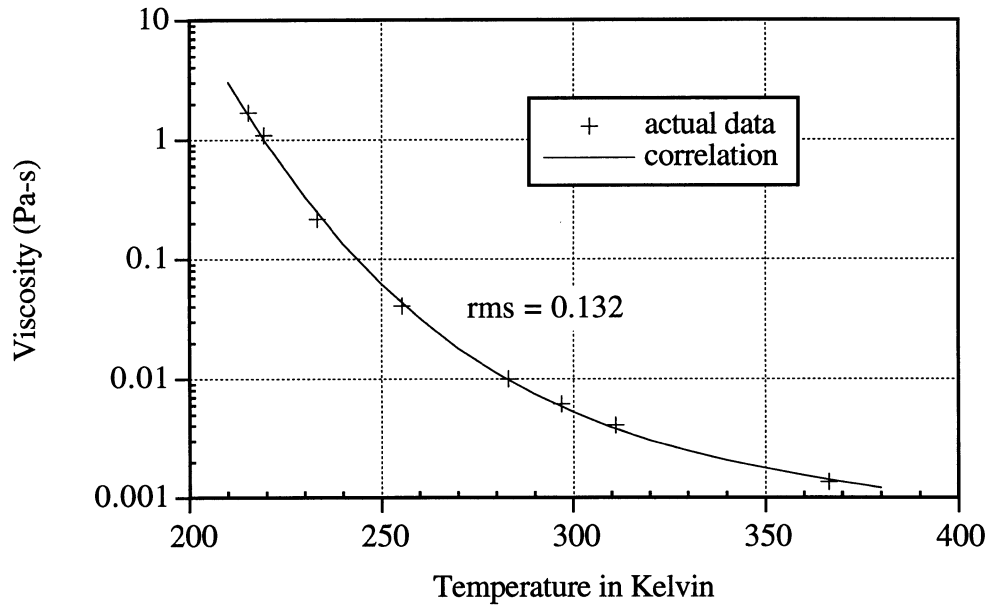


Figure 2.19 Comparison of correlation equation and actual data of the viscosity of Multitherm 503

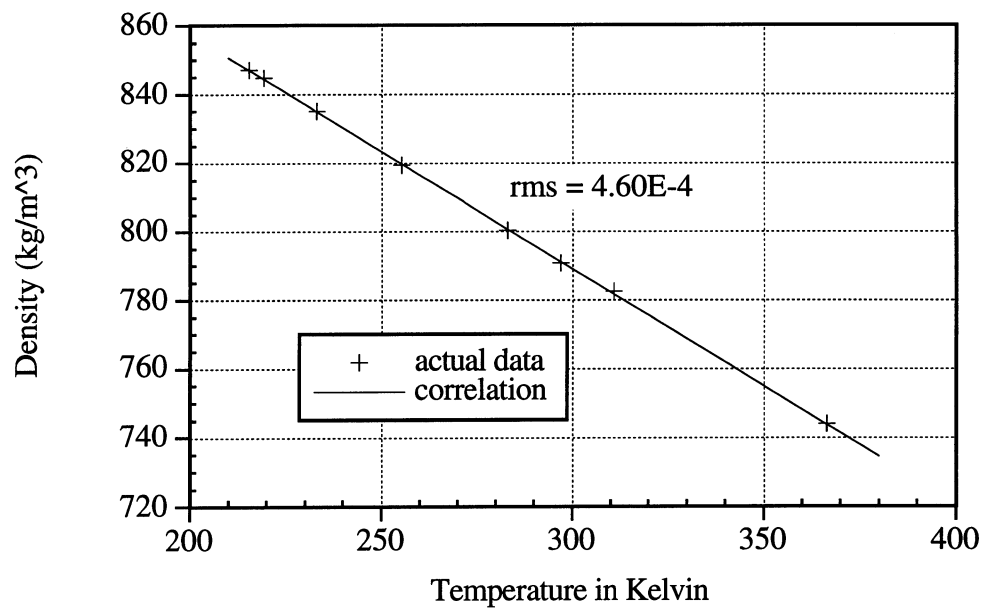


Figure 2.20 Comparison of correlation equation and actual data of the density of Multitherm 503

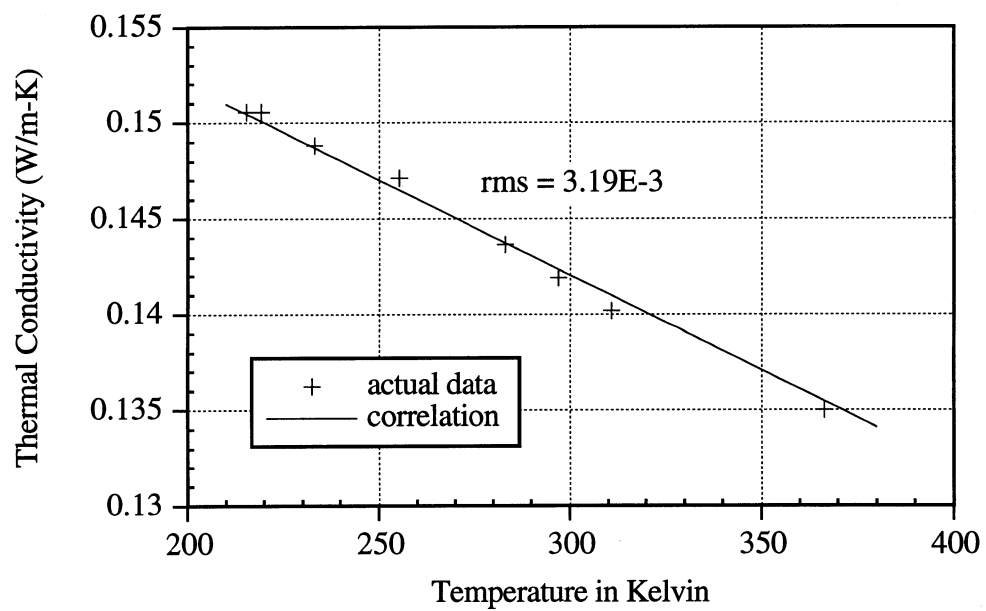


Figure 2.21 Comparison of correlation equation and actual data of the thermal conductivity of Multitherm 503

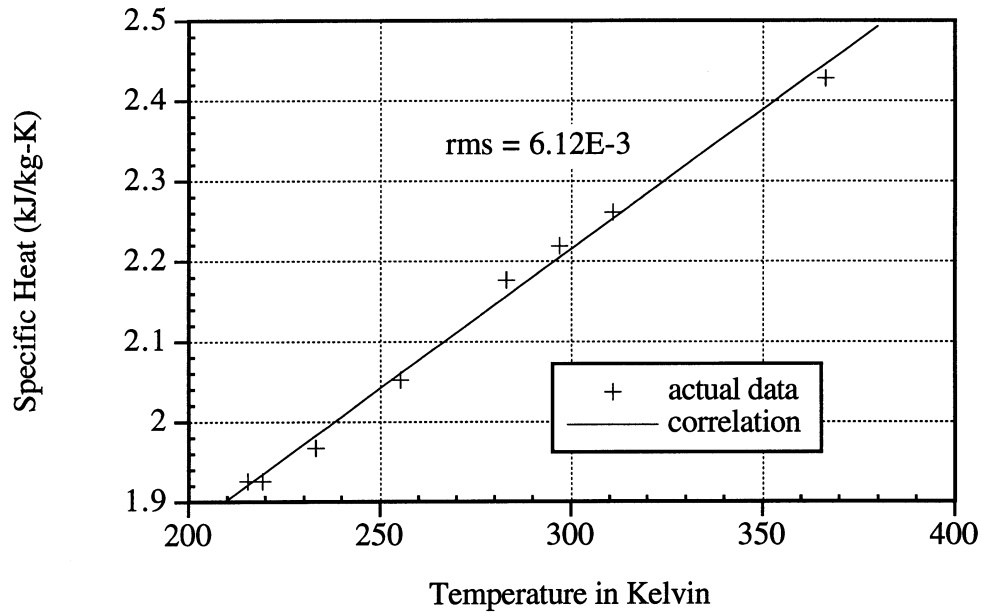


Figure 2.22 Comparison of correlation equation and actual data of the specific heat of Multitherm 503

2.1.4 Syltherm

The Dow Chemical Company produces a silicone based family of heat transfer fluids under the name Syltherm. (Dow Chemical Company [3]) The low temperature, liquid phase heat transfer medium is a specially formulated silicone polymer, Dimethyl Polysiloxane, and is called Syltherm XLT. Syltherm XLT has an operating range of -73 to 260 °C (-100 to 500 °F) and is low in oral toxicity. The freezing point of Syltherm XLT is -111 °C (-168 °F) and is non-corrosive to normal construction materials. It is not sensitive to contamination by common piping contaminants; such as water, rust, mill scale, lubricants and pipe dope, but is somewhat more sensitive to acids and bases at elevated temperatures. Syltherm XLT has a flash point of 47 °C (116 °F), a fire point of 60 °C (140 °F), and an auto-ignition temperature of 350 °C (662 °F). Because Syltherm

XLT is a poor electrical conductor, static charges can build up and discharge static electricity. Therefore, it is necessary to exclude oxygen from the headspace of the expansion tank. Syltherm XLT has been studied for acute toxicology under the Federal Hazardous Substance Act (FSHA) guidelines. It was classified as non-toxic with regard to acute oral ingestion and dermal absorption and having minimal potential for eye or skin irritation. Studies also indicate that repeated, prolonged skin contact should not result in irritation. Syltherm XLT has minimal odor and no airborne exposure limits, however, vapors released into the air at temperatures above 149 °C (300 °F) may cause temporary eye and/or respiratory irritation.

Property data for Syltherm XLT was provided by Dow and correlation equations were developed for use in computer simulations. (Dow Chemical Company [3]) The properties depend on the temperature of the fluid and the resulting equations are shown here where T stands for temperature in degrees Kelvin. Density is calculated in kg/m³, thermal conductivity in W/m-K, specific heat in kJ/kg-K, and viscosity in Pa-s.

$$\text{Thermal Conductivity} = 0.12569 - 6.5011e-5T \quad (2.13)$$

$$\text{Viscosity} = \exp[5.1231 - 0.070469T + 0.00013078T^2 - 8.7607e-8T^3] \quad (2.14)$$

$$\text{Density} = 1001.1 - 0.16603T - 0.0012262T^2 \quad (2.15)$$

$$\text{Specific Heat} = 0.89721 + 0.0025653T \quad (2.16)$$

Again, to check the accuracy of the correlation equations, the manufacturer's data were plotted against the results from the correlations. The graphs, with the root mean square error, are shown in Figures 2.23 through 2.26.

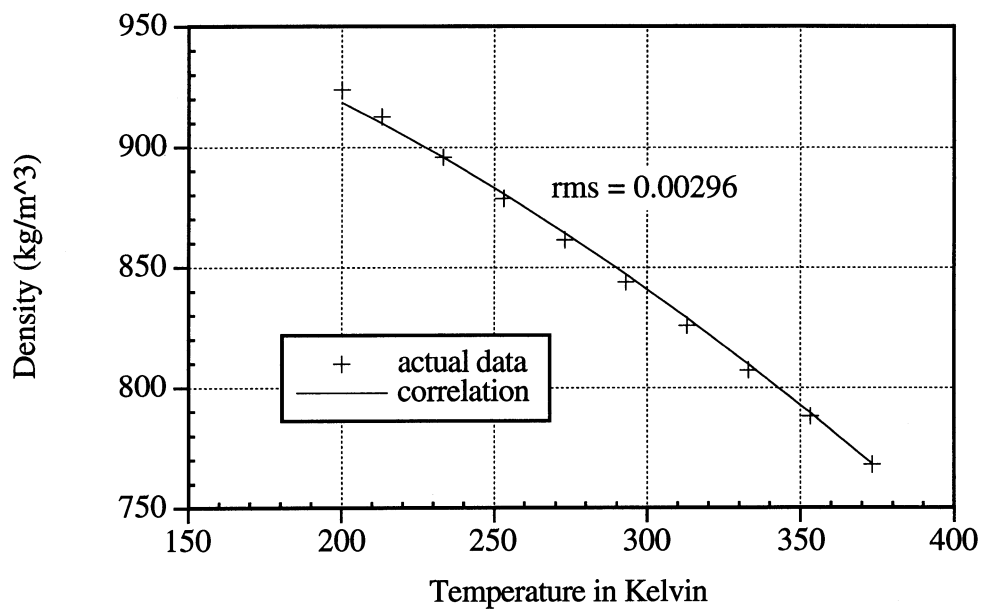


Figure 2.23 Comparison of correlation equation and actual data of the density of Syltherm XLT

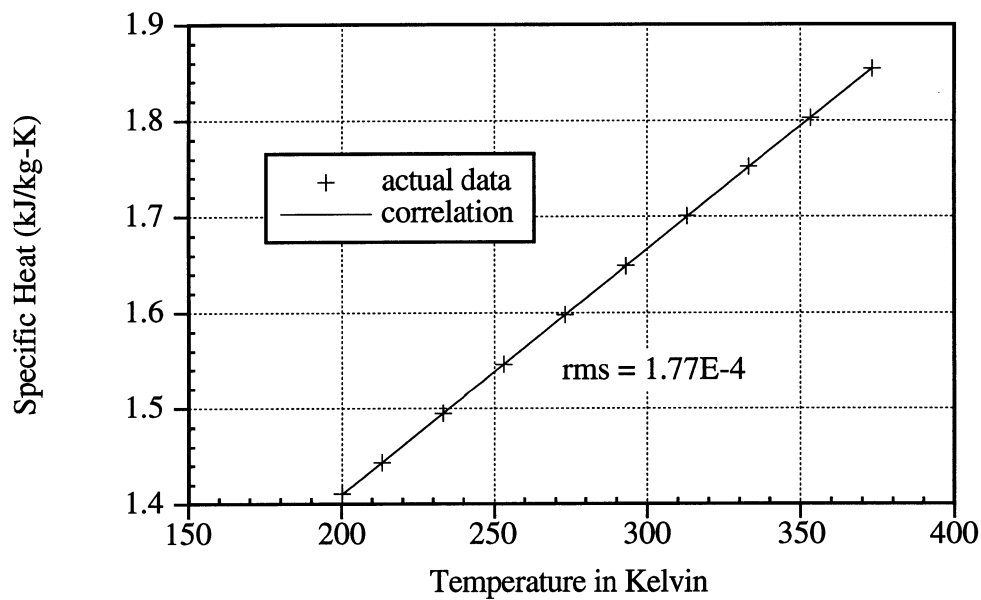


Figure 2.24 Comparison of correlation equation and actual data of the specific heat of Syltherm XLT

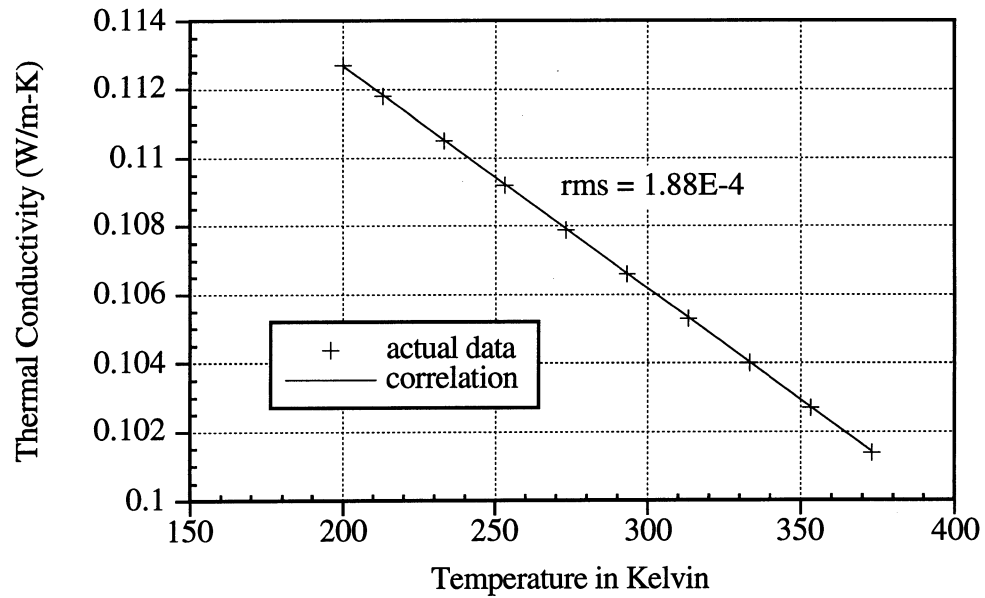


Figure 2.25 Comparison of correlation equation and actual data of the thermal conductivity of Syltherm XLT

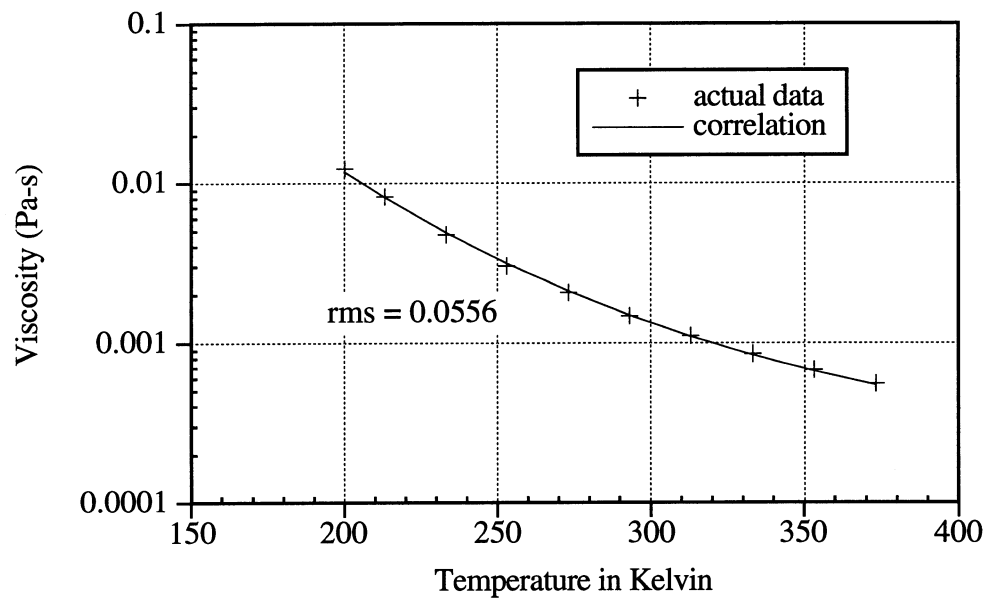


Figure 2.26 Comparison of correlation equation and actual data of the viscosity of Syltherm XLT

2.1.5 Ethanol

Ethanol, or ethyl alcohol, is a common heat transfer fluid. Rather than use the properties of ethanol from a chemical company, the correlations were developed from data taken from Liley, Yaws, and Natarajan. (Liley 1970, Yaws 1992, and Natarajan 1989) It is also important to note that ethanol is both flammable and explosive. When denatured, toxicity with ethanol is not a concern. The correlation equations used for ethanol are shown here, where T stands for temperature in Kelvin.

$$\text{Thermal Conductivity} = 0.25715 - 0.00030645T \quad (2.17)$$

$$\text{Viscosity} = \frac{\exp[11.664554604 - 0.057902271482T + 6.3561331711e-5T^2]}{1000} \quad (2.18)$$

$$\text{Density} = 0.2567e3 * 0.240^{\left[1 - \frac{T}{515.8}\right]^2} \quad (2.19)$$

$$\text{Specific Heat} = \frac{100.92 - 111.839e-3T + 498.54e-6T^2}{46} \quad (2.20)$$

2.1.6 Propane

Propane is a refrigerant that can also be used as a secondary fluid. It is necessary with propane to ensure that the pressure is high enough that the propane remains in liquid form throughout the system. Propane is also a highly flammable and explosive substance and needs to be used with caution. The equation solving program that was used to run the computer simulations, EES, calculates the properties of refrigerants, including propane. (Klein 1993) The properties are calculated using data from the ASHRAE handbook of fundamentals and the Martin-Hou equation of state. (ASHRAE 1989 and Martin 1955) These were used for the properties for propane.

2.2 Comparison of Properties of Secondary Fluids

The first step in determining the best secondary coolant for use in the supermarket system is to compare the properties of the different secondary coolants. Density, thermal conductivity, specific heat and viscosity were calculated for the six fluids for temperatures between 230 and 315 °K.

2.2.1 Density Comparison

The graph showing the comparison of the densities of the fluids is shown in Figure 2.27. The fluids fall into three groups. The two glycol based fluids, Dowfrost and Dowtherm SR-1, are grouped between 1000 and 1100 kg/m³ and are significantly higher than the other fluids. Propane lies between 600 and 500 kg/m³ and is significantly lower than the other fluids. The remaining fluids: ethanol, Multitherm 503 and Syltherm XLT, are grouped in the middle between 750 and 900 kg/m³.

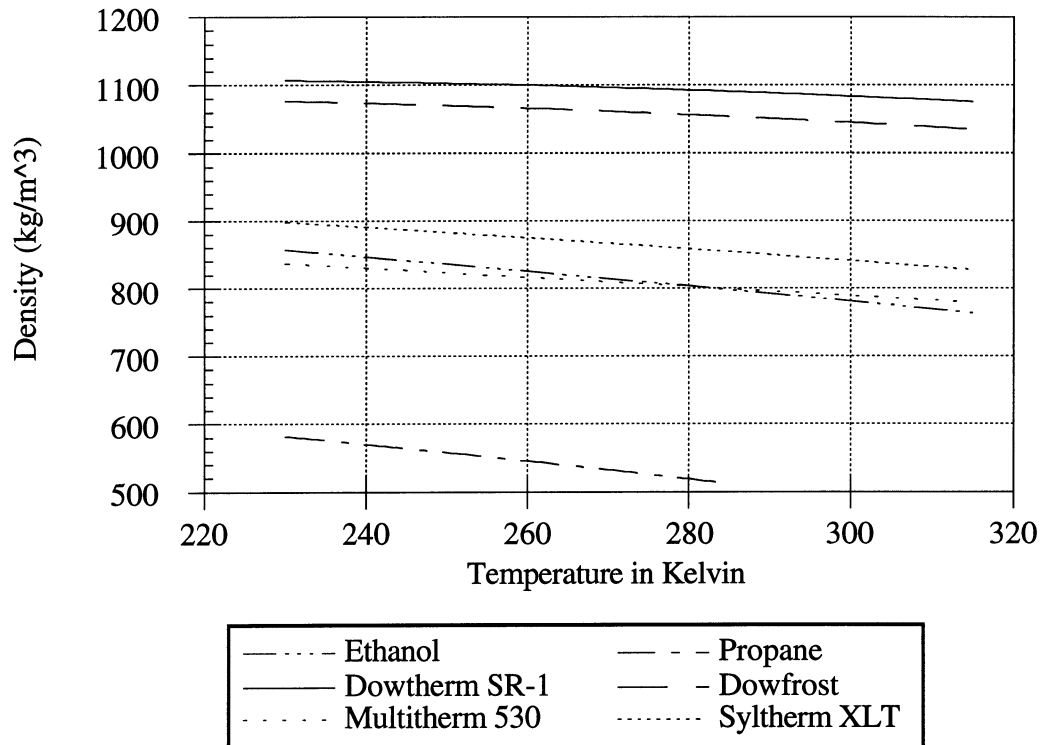


Figure 2.27 Density comparison

2.2.2 Thermal Conductivity Comparison

Figure 2.28 shows the graph of the comparison of the thermal conductivity of the six fluids. The fluids separate into two groups when the thermal conductivities are compared. Again, Dowfrost and Dowtherm SR-1 have thermal conductivities that are significantly higher than the other fluids. They both have thermal conductivities that vary from around 0.3 to 0.35 W/m-K. The other four fluids are spread out between 0.1 and 0.2 W/m-K. Ethanol has the highest thermal conductivity of the four lower fluids, followed by Multitherm 503, propane, and Syltherm XLT, respectively.

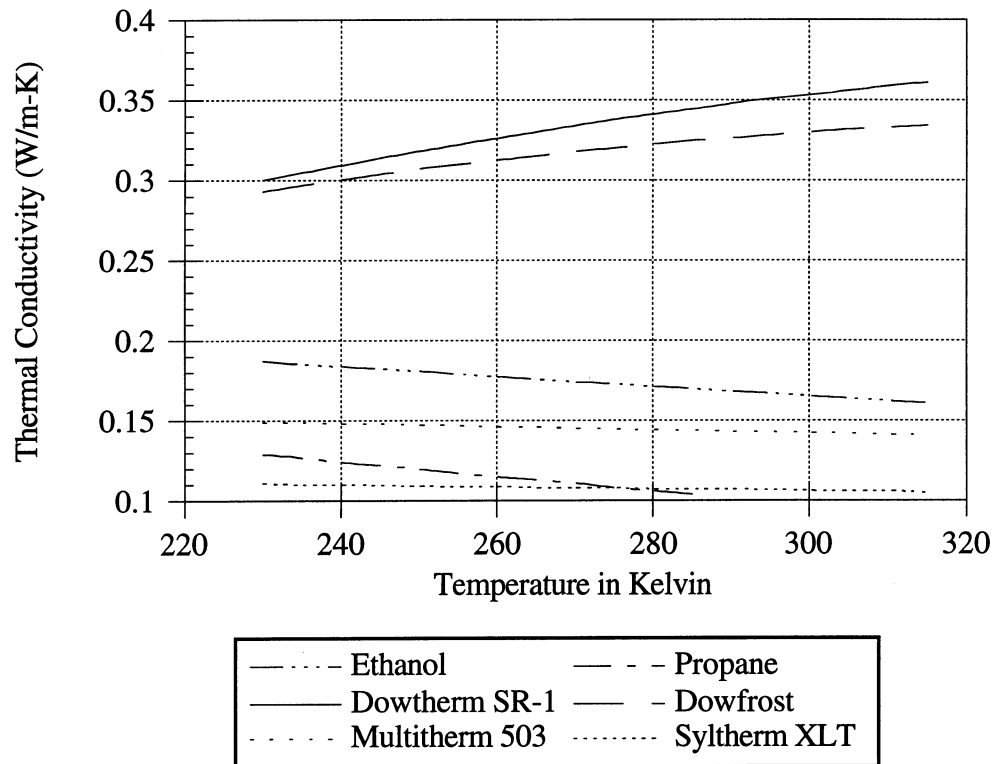


Figure 2.28 Thermal conductivity comparison

2.2.3 Specific Heat Comparison

The graph of the comparison of the specific heats of the secondary fluids is shown in Figure 2.29. As with density, the fluids divide into three distinct groups in this comparison. Dowtherm SR-1 and Dowfrost again constitute the group with the highest specific heats. Dowfrost is slightly higher than Dowtherm SR-1, but both vary between 2.7 and 3.6 kJ/kg-K. The middle group varies between 2.0 and 2.6 kJ/kg-K and consists of propane, ethanol and Multitherm 503, in that order. Syltherm XLT has the lowest specific heats varying between 1.5 and 1.7 kJ/kg-K.

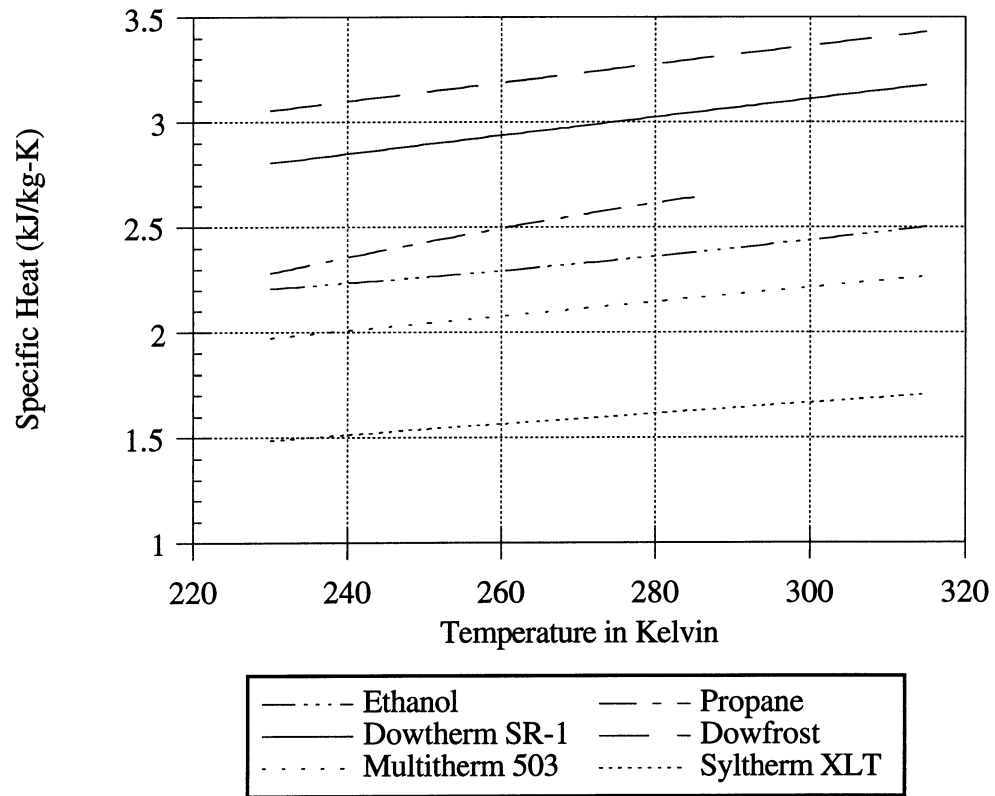


Figure 2.29 Specific heat comparison

2.2.4 Viscosity Comparison

Figure 2.30 shows the graph of the comparison of the viscosities of the six fluids. Again the fluids divide into three groups in this comparison. The group with the highest viscosities is made up of Dowfrost, Multitherm 503 and Dowfrost SR-1 and their viscosities vary from 1 to 0.005 Pa-s. The middle group varies from 0.001 to 0.007 Pa-s and consists of Syltherm XLT and ethanol. Propane has the lowest viscosity lying between 0.0001 and 0.0002 Pa-s.

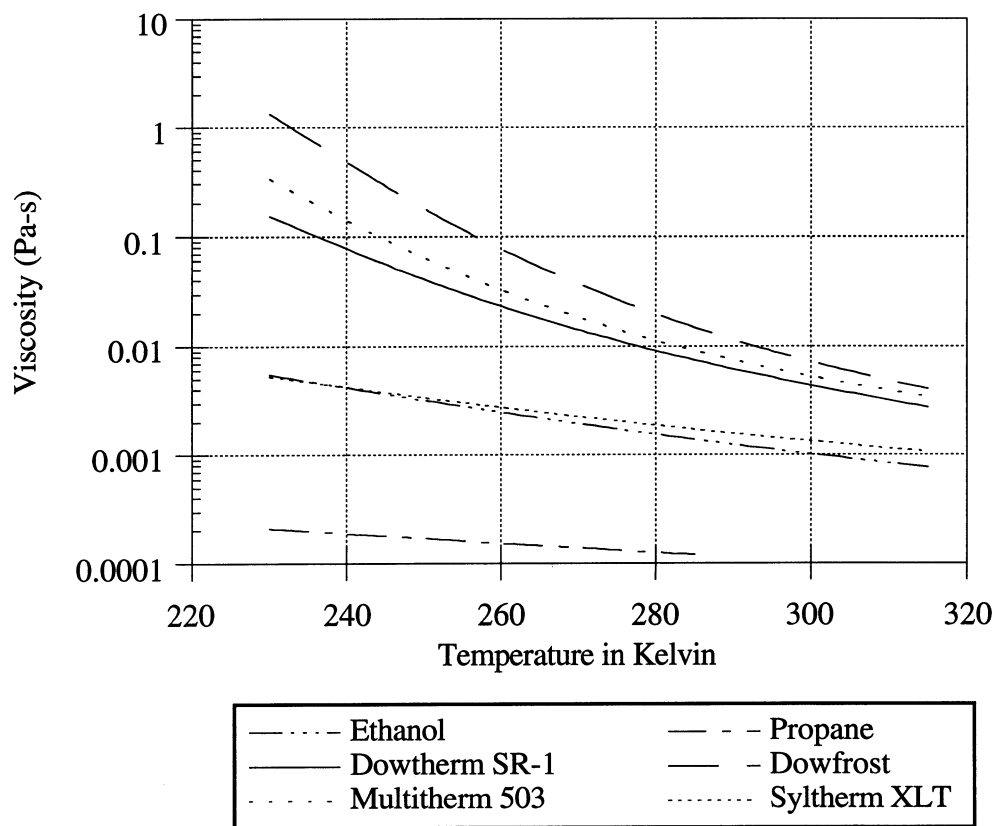


Figure 2.30 Viscosity comparison

2.2.5 Comparison of Hazards

In applications where the general public may come into contact with a secondary fluid, it is necessary to evaluate the hazards of that secondary fluid. Table 2.1 summarizes the toxicity, corrosiveness, and flammability of the six secondary fluids compared here.

Secondary Fluid	Toxic	Corrosive	Flammable
Dowfrost	no	inhibited	no
Dowtherm SR-1	yes	inhibited	no
Multitherm 503	no	no	slightly
Syltherm XLT	no	no	yes
Ethanol	no	no	yes
Propane	no	no	yes

Table 2.1 Hazards of secondary fluids

2.2.6 Property Comparison Conclusions

Different properties influence system performance in different manners. The higher the product of density and specific heat the lower the flow rate needed to meet the load. The lower the viscosity the lower the pumping power needed to circulate the secondary fluid. The higher the thermal conductivity the better the heat transfer. However, the fluids that have favorable properties for heat transfer do not have favorable properties for pump work. The selection of the best secondary fluid will have to be made when a comparison of the pumping work and heat transfer in the supermarket system is made. Comparisons of system performance with the different secondary fluids appear in Chapter 6.

CHAPTER **THREE**

REFRIGERANT CYCLE

The basis of the supermarket refrigeration system is the vapor compression cycle which cools the refrigerated case with a direct expansion coil, or by cooling a secondary fluid which in turn cools the refrigerated case. The first step in analyzing the supermarket system is to determine how to configure the system involving the vapor compression refrigeration cycle without the secondary fluid. This analysis was carried out for both R22 and ammonia. Because both of these refrigerants have high discharge temperatures, it is necessary to use staged compression for both refrigerants to reduce the temperatures. Three different methods of staging the compression were evaluated to determine the best method for each refrigerant.

3.1 Model Description

A computer model was written for the refrigeration cycles using the equation solving program EES. (Klein 1993) Each component of the system (evaporator, condenser, compressor, and expansion valve) was modeled using fundamental equations for each type of apparatus.

3.1.1 Condenser

The condenser model is representative of an air-cooled condenser. The flow rate of air is calculated using 0.14 kg/s of air per kW of cooling. This method of determining the flow rate is from the ASHRAE handbook of fundamentals. (ASHRAE 1989) The UA value for the condenser is a design input and is used to calculate the number of transfer units (NTU) of the condenser.

$$UA = \dot{m}_{\text{air}} C_{p\text{air}} \text{NTU} \quad (3.1)$$

The NTU can then be used to calculate the effectiveness of the condenser. The refrigerant entering the condenser is a superheated vapor. First the condenser brings the superheated refrigerant vapor to a saturated vapor and then condenses the vapor to a saturated fluid. The total heat transfer in the process includes both the desuperheating and the condensing. However, the calculation of heat transfer during desuperheating is complex and a simplifying assumption is that the heat transfer in the desuperheating process is small compared with the heat transfer in the condensing process. With this assumption, the process is isothermal on the refrigerant side of the heat exchanger and the effectiveness-NTU equation for an isothermal phase change can be used.

$$\varepsilon = 1 - \exp(-NTU) \quad (3.2)$$

The heat transferred in the condenser can be calculated from the temperature difference between the ambient air and the refrigerant flowing through the condenser.

$$Q = \varepsilon \dot{m}_{\text{air}} C_{p\text{air}} (T_{\text{ref}} - T_{\text{amb}}) \quad (3.3)$$

The calculated heat transfer can be used to calculate the temperature change in the air through the condenser.

$$Q = \dot{m}_{\text{air}} C_{p\text{air}} \Delta T_{\text{air}} \quad (3.4)$$

Then the heat transfer can also be used with the flow rate of the refrigerant to calculate the enthalpy change of the refrigerant in the condenser.

$$Q = \dot{m}_{\text{ref}} \Delta h_{\text{ref}} \quad (3.5)$$

3.1.2 Evaporator

The desired temperature of the refrigerated case is the temperature of the evaporator. It is assumed that the temperature of the refrigerant is constant across the evaporator and that the refrigerant leaves the evaporator with a specified amount of superheat. The number of degrees of superheat is a parameter of the model. The pressure in the evaporator is the pressure at which the refrigerant would be vapor at the evaporator temperature, and it is assumed that the pressure is constant. Then, knowing the

temperature and the pressure of the refrigerant leaving the evaporator, the properties of the refrigerant are set at this state.

3.1.3 Expansion

The expansion valve for the refrigeration system is assumed to be isenthalpic. This means that the enthalpy of the refrigerant does not change between the evaporator and the condenser.

3.1.4 Compression

The compression method is different depending on the method of staged compression chosen, and will be discussed in Section 3.2. The individual compressor model is the same in each staged compression method. Originally, the compressor was modeled using an isentropic model and an isentropic efficiency. This type of model can give accurate results for a particular compressor, but it does not include the physical dimensions of the compressor itself. Instead, a polytropic model based on the physical dimensions of the compressor was used. The techniques and the equations used in this model are from Threlkeld and Chlumsky. (Threlkeld 1970 and Chlumsky 1965) The refrigerant enters the compressor at a certain pressure and specific volume. The polytropic model uses the polytropic exponent (n) of the refrigerant to determine the relationship between the entering state and exiting state of the refrigerant.

$$P_{in}V_{in}^n = P_{out}V_{out}^n \quad (3.6)$$

The displacement volume of the compressor depends on the bore and stroke of the cylinder and the number of cylinders being operated.

$$V_{\text{disp}} = (\text{number of cylinders}) \left(\pi \left(\frac{\text{bore}}{2} \right)^2 \text{stroke} \right) \quad (3.7)$$

The volumetric efficiency of the compressor is based on the clearance ratio(m) of the compressor, as well as the pressure ratio and the polytropic exponent.

$$\eta_{\text{vol}} = 1 - \left[\frac{m}{1-m} \right] \left[\left(\frac{P_{\text{out}}}{P_{\text{in}}} \right)^{\frac{1}{n}} - 1 \right] \quad (3.8)$$

The flow rate of the refrigerant through the compressor is dependent on the displacement volume, the speed of the compressor, and the volumetric efficiency.

$$\dot{m}_{\text{ref}} = \left[\frac{1}{v_{\text{in}}} \right] V_{\text{disp}} [\text{speed}] \eta_{\text{vol}} \quad (3.9)$$

The power used by the compressor can be calculated from the flow rate of the refrigerant, the enthalpy change in the refrigerant, and the polytropic efficiency of the compressor.

$$W_{\text{comp}} = \frac{\dot{m}_{\text{ref}} \Delta h_{\text{ref}}}{\eta_{\text{polytropic}}} \quad (3.10)$$

The reason that a polytropic efficiency needs to be used is that when the ideal polytropic exponent is used, the result is an isentropic model. An isentropic model will under-predict the amount of compressor work needed because the entropy of the refrigerant in the compressor is not constant. To reduce this under-prediction, it is necessary to either alter the polytropic exponent, or utilize a polytropic efficiency. In this manner an accurate

value of the compressor work is calculated. The outgoing pressure of the refrigerant from the compressor is the same as the condenser pressure.

3.1.5 Other Equations

The refrigerant mass flow rate is dependent on the heat transfer rate of the refrigerated cases (caseload) of the system and the enthalpy change of the refrigerant through the evaporator.

$$\dot{m}_{\text{ref}} = \frac{\text{caseload}}{\Delta h_{\text{ref}}} \quad (3.11)$$

For refrigeration cycles, a measure of the performance of a system is the coefficient of performance (COP), which is defined as the caseload divided by the work done by the compressor.

$$\text{COP} = \frac{\text{caseload}}{W_{\text{comp}}} \quad (3.12)$$

3.2 Compression Methods

Because both ammonia and R22 have high discharge temperatures in refrigeration cycles, it is necessary to evaluate different compression staging methods for each. High discharge temperatures are a problem because of the increased superheating that they cause in the compressors. This increased superheating increases the work done by the compressor. Staging the compression reduces the discharge temperatures by allowing intercooling between the stages of compression. Three different methods of staging were modeled along with the cycle with no staging. These models are discussed here.

3.2.1 No Staging

The basic vapor compression refrigeration cycle uses one stage of compression to bring the refrigerant from the evaporator pressure to the condenser pressure. A diagram of the basic cycle is shown in Figure 3.1.

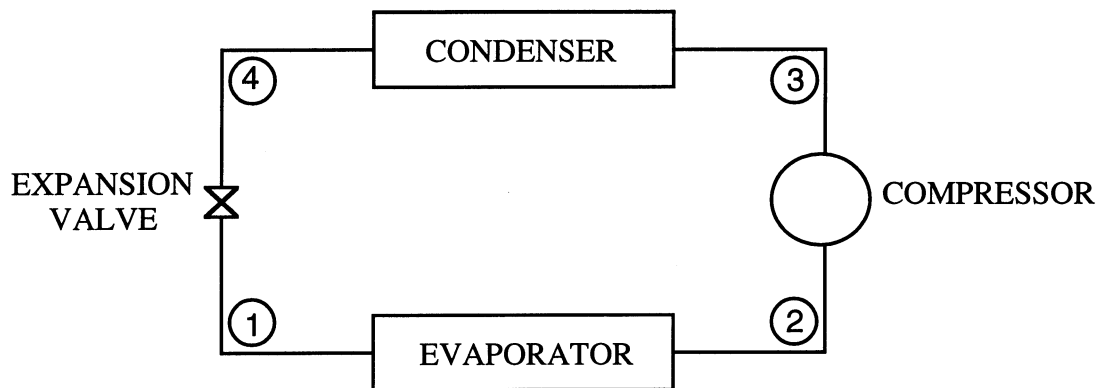


Figure 3.1 Refrigeration cycle with one stage of compression.

In the single cycle, the temperature of the refrigerant at state 1 is the desired refrigerated case temperature. The pressure of state 1 can be calculated from this temperature. State 2 is at the same pressure as state 1 and the temperature is a given number of degrees of superheat higher than the temperature at state 1. State 4 is the liquid leaving the condenser and the pressure at state 4 must be high enough that the exiting state is liquid. The pressure of the gas leaving the compressor at state 3 is the same as the pressure in state 4. The cycle can be viewed on a pressure-enthalpy diagram to show the different states, and a diagram for the cycle with a single stage of compression is shown in Figure 3.2.

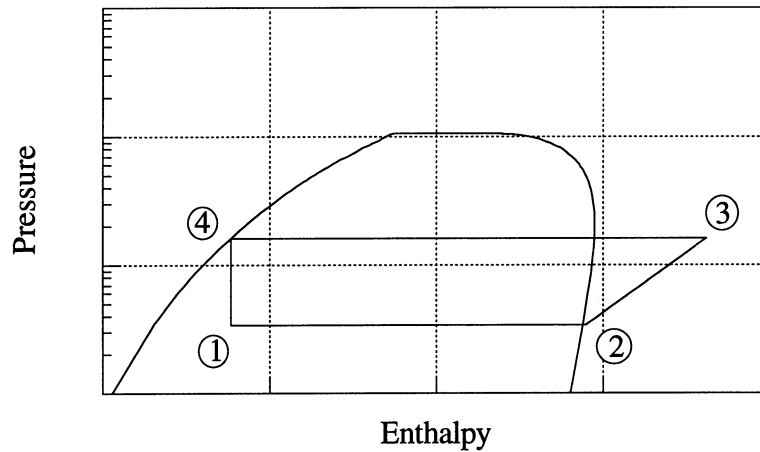


Figure 3.2 Pressure-Enthalpy diagram for single stage of compression

3.2.2 Staged Compression

The first method for staging the compression of the refrigerant 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 schematic of the cycle is shown in Figure 3.3. The advantages of this type of staging is that the refrigerant is desuperheated between the two compressors. This causes the compressed gas to exit the second stage of compression at a lower temperature. The lower temperatures also lead to a higher volumetric efficiency in the second compressor. The lower temperature gas entering the second compressor has a smaller specific volume than the gas would if no desuperheating took place. This smaller volume allows a smaller compressor to be used in the second stage of compression.

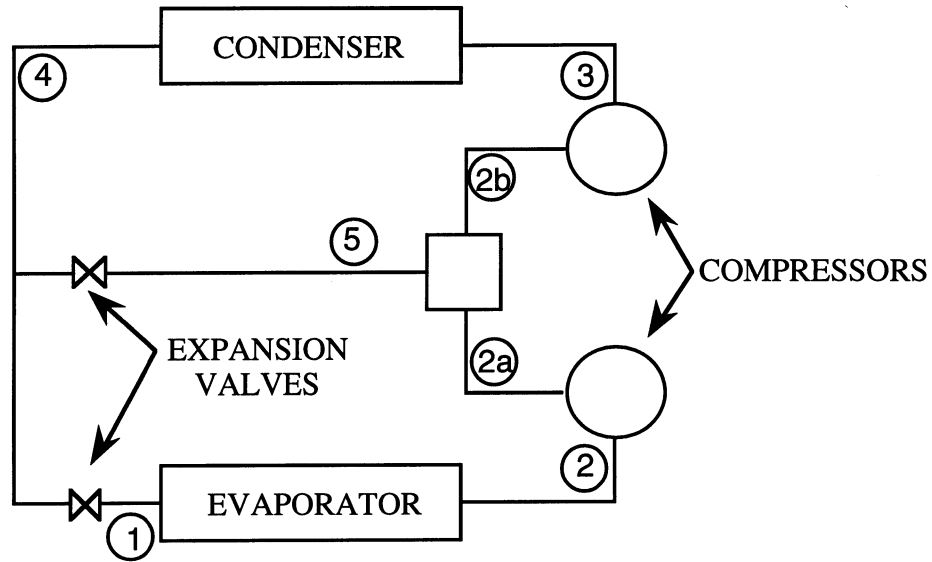


Figure 3.3 Refrigeration cycle with staged compression

The introduction of the staging changes the refrigerant mass flow rates through the components. Along with the flow needed to meet the load in the evaporator, additional flow is needed through the condenser to be extracted for intercooling. The mixing equations for the two refrigerant streams mixing between the two compressors can be used to determine the necessary flow rates. These equations are included here where y is the flow rate of the intercooling refrigerant if there is unit flow rate through the evaporator.

$$y = \frac{h_{2a} - h_{2b}}{h_{2b} - h_4} \quad (3.13)$$

$$1+y = \frac{h_{2a} - h_4}{h_{2b} - h_4} \quad (3.14)$$

$$\dot{m}_{\text{cond}} = (1+y)\dot{m}_{\text{evap}} \quad (3.15)$$

The selection of the intermediate pressure is important. It is necessary to select the intermediate pressure so that the mixture of the refrigerant leaving the compressor and the refrigerant extracted from the condenser is saturated vapor. The states 1 and 2 are the same for the staged compression as in the compression in one stage. The pressure at states 3 and 4 is also the same, however, the temperature at state 3 has been reduced. With staged compression, the refrigerant in state 2 is compressed to the intermediate pressure at state 2a. The refrigerant extracted from the condenser is expanded at the intermediate pressure at state 5. The refrigerant at states 5 and 2a is mixed to produce saturated vapor at the intermediate pressure in state 2b. This vapor is finally compressed from state 2b to state 3. The pressure-enthalpy diagram for this cycle is shown in Figure 3.4.

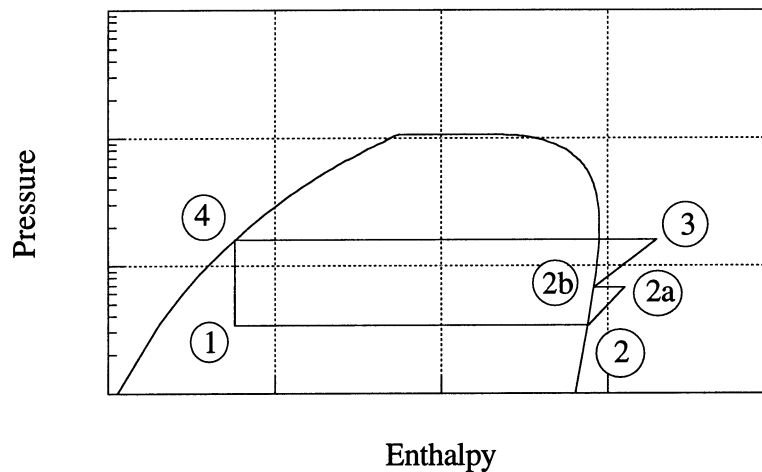


Figure 3.4 Pressure-Enthalpy diagram of staged compression

3.2.3 Staged Compression and Evaporation

The second method of staging the compression, known as staged compression and evaporation, differs from the 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 intercool the vapor leaving the first compressor. The diagram of the staged compression and evaporation is shown in Figure 3.5. This type of staging utilizes the same desuperheating advantages as mentioned in the staged compression section, but to a smaller degree. The amount of desuperheating is less so the advantages are not as significant. There is an additional advantage due to the staging of the expansion. Because the refrigerant is expanded twice the enthalpy difference across the evaporator is greater and less mass flow rate of refrigerant is needed to meet the refrigeration load.

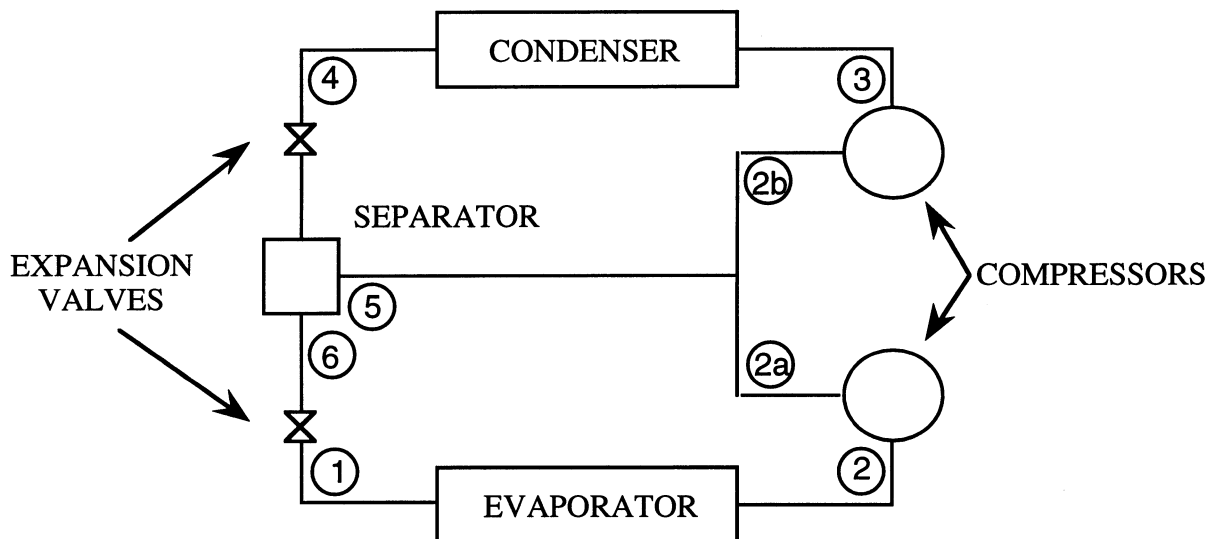


Figure 3.5 Refrigeration cycle with staged compression and evaporation

Like the staged compression method, the flow rates through the components changes with this method. Again the mixing equations are used to determine the flow rates.

$$\dot{m}_{\text{cond}} = \dot{m}_{\text{evap}} + \dot{m}_{\text{intercool}} \quad (3.16)$$

$$\dot{m}_{\text{cond}}h_4 = \dot{m}_{\text{evap}}h_6 + \dot{m}_{\text{intercool}}h_5 \quad (3.17)$$

$$\dot{m}_{\text{cond}}h_{2b} = \dot{m}_{\text{evap}}h_{2a} + \dot{m}_{\text{intercool}}h_5 \quad (3.18)$$

With this type of staging, the intermediate pressure can be any pressure between the pressures at state 2 and at state 3. There is an optimal intermediate pressure that will give the best performance; and this optimal pressure depends on the refrigerant used. The optimal occurs at a pressure ratio which results in the highest volumetric efficiency for each compressor. This may not be the highest volumetric efficiency that either compressor could attain on its own, but it is the combination of volumetric efficiencies for both compressor which is the highest. State 1 with this method is no longer the same as the state 1 in the single stage compression method. Since the evaporation is now staged, the refrigerant entering the evaporator is at the same temperature and pressure as the single stage case, but the quality has changed. State 2 has remained the same, as well as the pressure in states 3 and 4. Once again the temperature in state 3 is reduced from the temperature in the single stage case. The separator in the refrigerant stream leaving the condenser expands the refrigerant at the intermediate pressure and sends the vapor to the compressors and the liquid to the evaporator. The first stage of compression compresses the refrigerant leaving the evaporator to the intermediate pressure where it is mixed with the vapor from the separator. The resulting mixture is then compressed to the

state 3 pressure in the second stage compressor. The pressure-enthalpy diagram for the staged compression and evaporation method is shown in Figure 3.5.

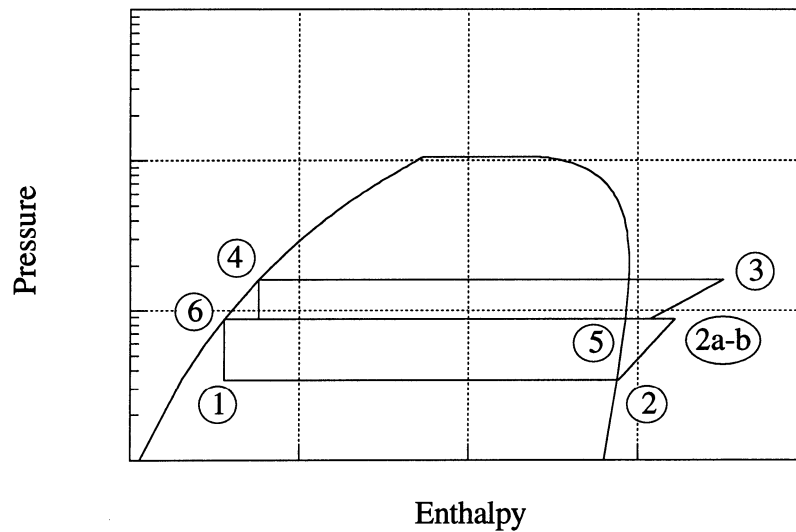


Figure 3.6 Pressure-Enthalpy diagram of staged compression and evaporation.

3.2.4 Staged Compression with Flash Tank

The final method of staged compression utilizes a flash tank between the condenser and the evaporator and between the two stages of compression. (Stoecker 1982) The diagram of the staged compression with flash tank is shown in Figure 3.7.

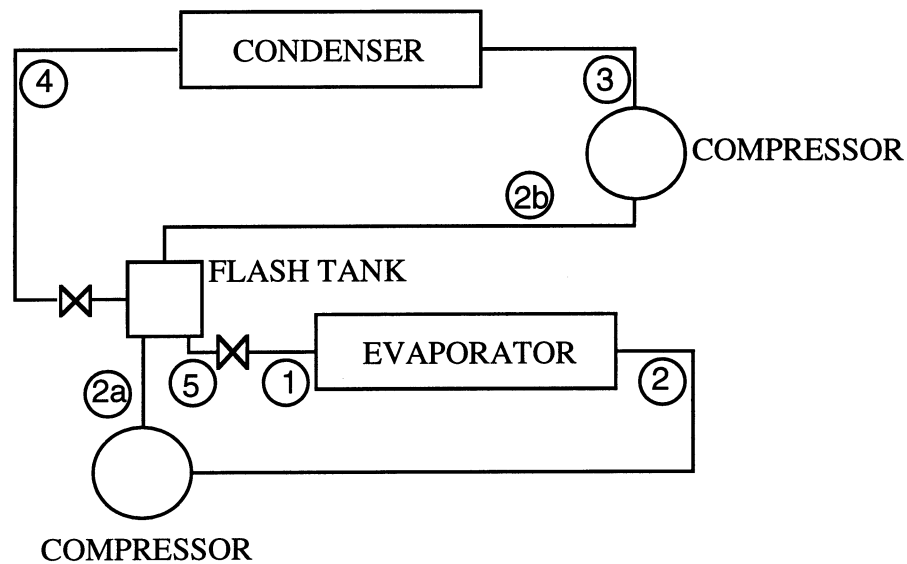


Figure 3.7 Refrigeration cycle with staged compression and flash tank

This method is similar to the staged compression and evaporation because the refrigerant leaving the condenser is expanded at an intermediate pressure and the resulting liquid is used in the evaporator. The vapor resulting from the expansion of the refrigerant leaving the condenser is not used to intercool the compressors directly in this method. Instead, the liquid and vapor from the expansion are held in a flash tank. Some of the liquid is extracted and used in the evaporator and once the refrigerant leaves the evaporator it is compressed back to the intermediate pressure. The refrigerant leaving the compressor is bubbled through the remaining liquid and vapor from the expansion of the refrigerant leaving the condenser. The saturated vapor in the flash tank is then compressed to the condenser pressure. This method of staged compression combines the advantages of both the first two methods. The refrigerant is desuperheated to the saturation point as in the staged compression method. This means that it has all of the same advantages of increased volumetric efficiency and decreased size as the staged compression method.

The expansion is staged like in the staged compression and evaporation method and has the same refrigeration capacity advantage. There is only one mixing equation for this method, because there are only two different flow rates in the system. The mixing involves the flow into and out of the flash tank.

$$\dot{m}_{\text{cond}}h_4 + \dot{m}_{\text{evap}}h_{2a} = \dot{m}_{\text{evap}}h_5 + \dot{m}_{\text{cond}}h_{2b} \quad (3.19)$$

Again, any intermediate pressure can be selected for this process, but there is an optimal pressure that maintains a mixture of liquid and vapor refrigerant in the flash tank and gives the highest performance. The state points of this cycle are similar to the points in the other cycles. State 5 is saturated liquid at the intermediate pressure and state 2b is saturated vapor at the intermediate pressure. The pressure-enthalpy diagram for this method is shown in Figure 3.8.

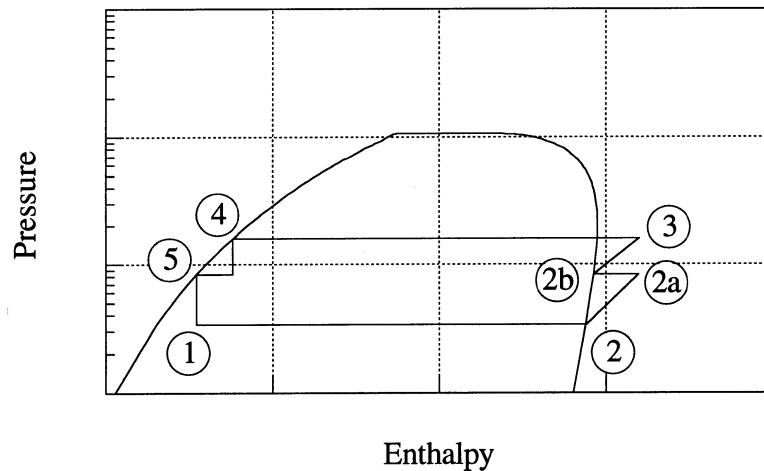


Figure 3.8 Pressure-Enthalpy diagram of staged compression with flash tank

3.3 Performance Comparison

Since the refrigerants have different characteristics, it was necessary to evaluate all of the staging methods with both ammonia and R22 as the refrigerant. Listings of the models used to evaluate the methods are in Appendix A and the results are summarized here.

3.3.1 R22 Comparison

The models for the R22 cycles were run for evaporator temperatures of 267 and 244 °K. The comparison was done based on the COPs for the different cycles. The results are shown in Table 3.1.

Method	COP	
	Temperature	
	244 °K	267 °K
single stage of compression	1.44	2.50
staged compression	1.43	2.49
staged compression and evaporation	1.69	2.76
staged compression with flash tank	1.68	2.74

Table 3.1 R22 staging comparison

From this comparison it can be seen that the best method for staging a R22 refrigeration system is to use staged compression and evaporation.

3.3.2 Ammonia Comparison

As with R22, the models of the different methods with ammonia as the refrigerant were run for evaporator temperatures of 244 and 267 °K. The table of the COP comparison is shown in Table 3.2.

Method	COP	
	Temperature	
	244 °K	267 °K
single stage of compression	1.52	2.90
staged compression	1.61	3.03
staged compression and evaporation	1.69	3.09
staged compression with flash tank	1.84	3.25

Table 3.2 Ammonia staging comparison

Unlike R22, ammonia systems give the highest performance when staged compression with a flash tank is utilized.

3.4 Conclusions

The analysis of the four different compression staging methods for R22 and ammonia systems shows that for R22 systems, the staged compression and evaporation method is best, and for ammonia systems, the staged compression with flash tank method is the best. These conclusions agree with those mentioned by Stoecker and Gosney. (Stoecker 1982 and Gosney 1982) The optimal methods for R22 and ammonia will be used in the analysis of the supermarkets systems in the following chapters.

**CHAPTER
FOUR**

AMMONIA WITH SECONDARY FLUID SYSTEM

The addition of a secondary fluid to the refrigeration system increases the complexity of the model necessary to simulate the performance. The evaporator for the vapor compression cycle is replaced with an heat exchanger and the components needed for the secondary fluid are added. This chapter discusses these components and how they are modeled.

4.1 Evaporator Heat Exchanger Model

The model of the vapor compression part of the ammonia-secondary fluid system is very similar to the models used for the single cycle comparison discussed in Chapter 3. The staged compression with flash tank method was used and the model of the individual compressor was the same. The condenser was also air-cooled and used the same ASHRAE equations as shown in Equation 3.1 to 3.5. (ASHRAE 1989) All expansion valves were modeled as isenthalpic.

The evaporator in the ammonia-secondary fluid system is completely different than the evaporator used in the single cycle system. In the single cycle system, the evaporator for the refrigerant is the refrigerated case, and in the ammonia-secondary fluid system, the evaporator is a heat exchanger that uses the ammonia to cool the secondary fluid. The model of that heat exchanger is discussed here.

The heat exchanger is a flooded, shell and tube heat exchanger with the secondary fluid flowing through the tubes and ammonia in the shell. A diagram of the heat exchanger is shown in Figure 4.1. The secondary fluid enters the tube bundle of the heat exchanger after leaving the refrigerated case. It is cooled by the ammonia and then circulated to the refrigerated case. The ammonia enters as liquid after leaving the condenser and the expansion valve. As the ammonia cools the secondary fluid, it evaporates and the saturated ammonia vapor collects in the top of the heat exchanger. This vapor is removed and compressed.

For the computer model, it is necessary to determine the heat transferred in the evaporation process. The initial idea for the model was to specify the effectiveness of the heat exchanger and use this to calculate the heat transfer. The problem with this type of model is that when the flow rates of the fluids change in an heat exchanger the effectiveness of the heat exchanger changes because of the change in the capacitance rate

of the streams. So using a constant effectiveness model is equivalent to replacing the heat exchanger for each different simulation. To eliminate this problem, a model was used that calculates the effectiveness of the heat exchanger from the calculated UA product for the specific heat exchanger design being considered.

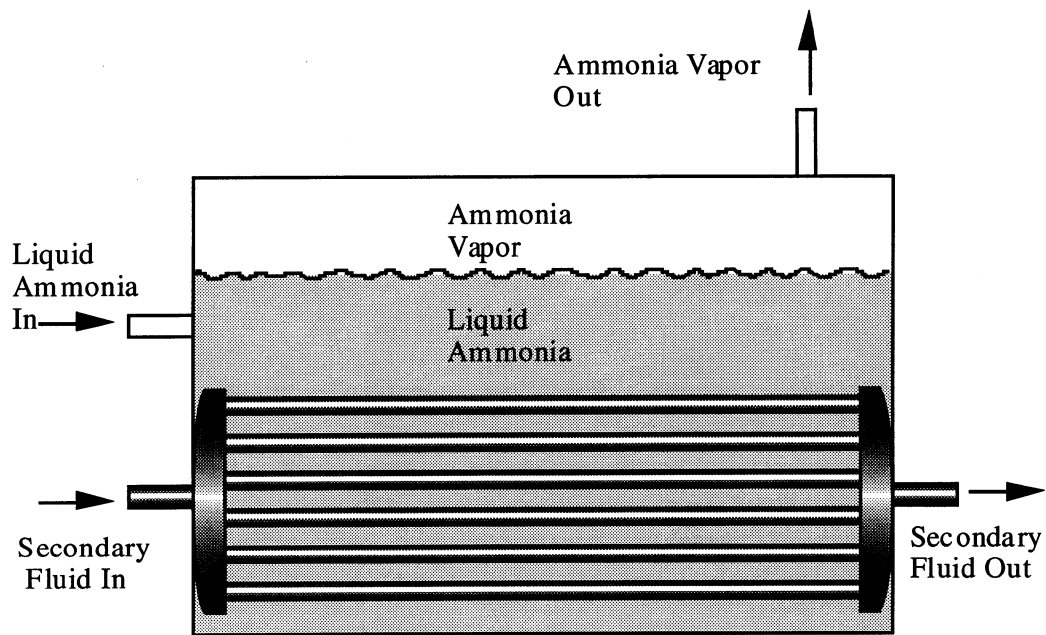


Figure 4.1 Diagram of flooded ammonia-secondary fluid heat exchanger

To calculate the UA product of the heat exchanger it is necessary to calculate the heat transfer coefficients on the inside and outside of the tubes in the tube bundle. The techniques for finding the heat transfer coefficients are described in Chapman and Incropera. (Chapman 1984 and Incropera 1985) On the inside of the pipes, the flow is considered to be developing hydrodynamically and developed thermally. For laminar flow, the Hausen correlation can be used to calculate the Nusselt number. (Chapman 1984)

$$\text{Nu}_D = 3.66 + \frac{0.0668 \left(\frac{D}{L} \right) \text{Re}_D \text{Pr}}{1 + 0.4 \left[\left(\frac{D}{L} \right) \text{Re}_D \text{Pr} \right]^{2/3}} \quad (4.1)$$

For turbulent flow, Nusselt developed an equation that accounts for the entry length effects. (Chapman 1984)

$$\text{Nu}_D = 0.036 \text{Re}_D^{0.8} \text{Pr}^{1/3} \left(\frac{D}{L} \right)^{1/8} \quad (4.2)$$

In both of these equations, Re_D is the Reynolds number of the flow based on the diameter of the pipe and all of the properties are evaluated at the bulk temperature of the fluid. The Reynolds number is used to determine whether the flow is laminar or turbulent, with Reynolds numbers greater than 2300 considered turbulent. The heat transfer coefficient is determined from the calculated Nusselt number.

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

$$h = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{C_{p_l}}{C_{sf} h_{fg} (\text{Pr}_l)^{1.7}} \right]^3 (T_{\text{wall}} - T_{\text{saturated}})^2 \quad (4.3)$$

In Equation 4.3, the subscript l refers to properties of the saturated liquid and the subscript v to properties of the saturated vapor. The latent heat of vaporization is given by h_{fg} , σ is the surface tension of the liquid-vapor interface, and g is the acceleration of gravity. The constant C_{sf} is an empirically determined quantity that depends on the composition and roughness of the heating surface and how the liquid wets the surface.

The value of C_{sf} used for ammonia was 0.01, which was estimated from values in Chapman. (Chapman 1984)

In pool boiling, the temperature difference between the surface of the heat source and the saturation temperature of the liquid is one of the driving forces in the heat transfer. It is necessary to estimate the surface or wall temperature if it is not known. In this simulation the wall temperature is not known, so it was necessary to develop a estimation of the wall temperature. The wall temperature will be somewhere between the inlet and outlet temperature of the secondary fluid. The effectiveness of the heat exchanger was calculated for different values of this temperature. The results are summarized in Table 4.1.

T_{wall} Formula	Effectiveness
$\frac{1}{4}T_{in} + \frac{3}{4}T_{out}$	0.39
$\frac{1}{3}T_{in} + \frac{2}{3}T_{out}$	0.39
$\frac{1}{2}T_{in} + \frac{1}{2}T_{out}$	0.39
$\frac{2}{3}T_{in} + \frac{1}{3}T_{out}$	0.39
$\frac{3}{4}T_{in} + \frac{1}{4}T_{out}$	0.39

Table 4.1 Comparison of wall temperature formulas

As can be seen from Table 4.1, as long as the wall temperature was between the inlet and outlet temperature, the effectiveness was the same. With this conclusion, the formula of the average of the inlet and outlet temperatures was used.

$$T_{\text{wall}} = \frac{1}{2}T_{\text{in}} + \frac{1}{2}T_{\text{out}} \quad (4.4)$$

Once the heat transfer coefficient on the inside of the pipes and on the outside of the pipes is calculated, the overall heat transfer coefficient can be determined.

$$U = \frac{1}{\frac{r_o}{r_i h_i} + \frac{r_o \ln\left(\frac{r_o}{r_i}\right)}{k_{\text{pipe}}} + \frac{1}{h_o}} \quad (4.5)$$

In Equation 4.5, the subscript i refers to the inside of the pipes, the subscript o refers to the outside of the pipes, r stands for the radii, h for the heat transfer coefficients, and k_{pipe} is the thermal conductivity of the pipes. The UA product for the heat exchanger is determined by multiplying the overall heat transfer coefficient by the total area that is utilized for heat transfer.

Once the UA product is determined, the heat transfer and temperature changes are calculated using the effectiveness-NTU method. (Chapman 1984) The NTU can be calculated from the UA product and the minimum capacitance rate of the two streams. Since the ammonia side of the heat exchanger has an infinite specific heat, the minimum capacitance rate is the secondary fluid flow.

$$NTU = \frac{UA}{\dot{m}_{\text{sf}} C_{p\text{sf}}} \quad (4.6)$$

Also, since the ammonia side of the heat exchanger involves a phase change, the effectiveness-NTU equation for one fluid changing phase is used.

$$\varepsilon = 1 - \exp(-NTU) \quad (4.7)$$

It is possible to write the effectiveness of the heat exchanger as a combination of the inlet and outlet temperatures of the two streams. The specific combination depends on which of the streams has the minimum capacitance rate. In this case, the secondary fluid has the minimum capacitance rate and the ammonia is at a constant temperature because the ammonia evaporates and is extracted as a saturated vapor.

$$\varepsilon = \frac{T_{sf,in} - T_{sf,out}}{T_{sf,in} - T_{ammonia}} \quad (4.8)$$

Equation 4.8 determines the temperatures in the heat exchanger and the heat transfer can be calculated from the capacitance rate and temperature change of the secondary fluid stream.

$$Q = \dot{m}_{sf} C_{p,sf} (T_{sf,out} - T_{sf,in}) \quad (4.9)$$

4.2 Secondary Fluid System

The components of the system that are unique to the use of a secondary fluid were also modeled. These components include the heat exchanger in the refrigerated case and the thermal losses and required pump work from the piping system.

4.2.1 Refrigerated Case

In Chapter 3, the refrigerated case was the evaporator of the vapor compression cycle and it was modeled by specifying the required case temperature and the amount of super heating of the refrigerant. With a secondary fluid, the refrigerated case is a heat

exchanger which cools the air that is circulated around the case. In this heat exchanger, the secondary fluid flows through a tube bundle that is oriented cross-flow to the air stream. A diagram of this heat exchanger is shown in Figure 4.2.

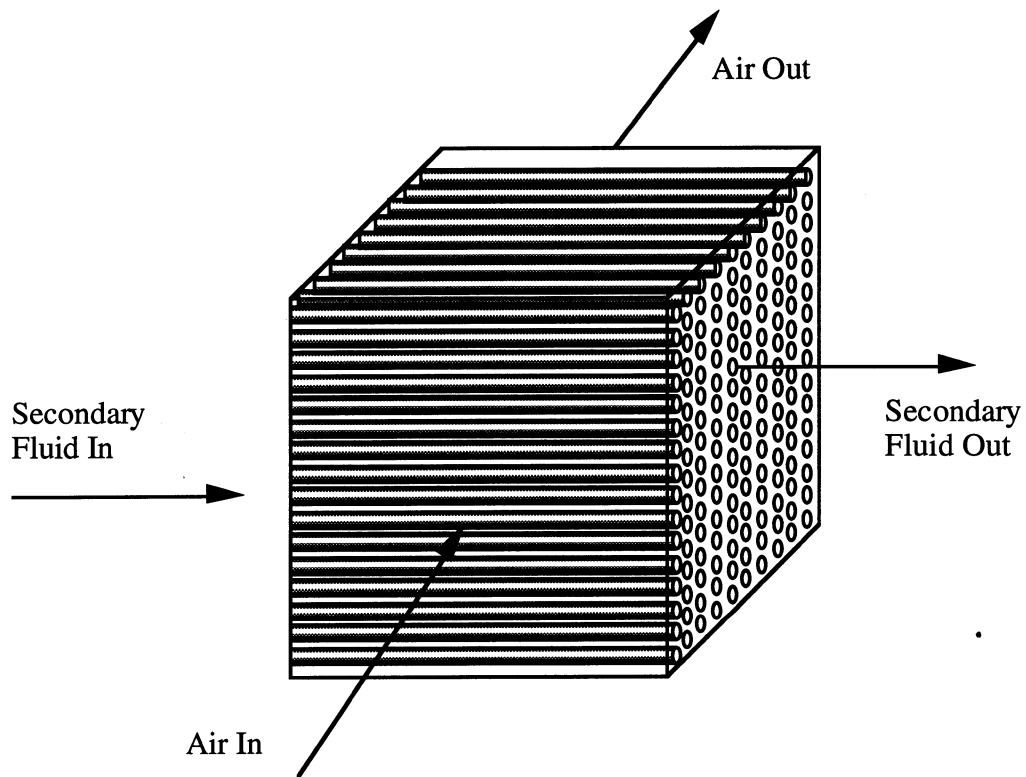


Figure 4.2 Diagram of the refrigerated case heat exchanger

The analysis of the refrigerated case heat exchanger is similar to the analysis of the heat exchanger between ammonia and the secondary fluid. The UA product needs to be determined first before the heat transfer can be calculated. The heat transfer coefficients on the inside and on the outside of the tubes are determined first. The secondary fluid flows through the pipes in the heat exchanger and again the assumption of the flow being developing hydrodynamically and developed thermally is used. This allows the Hausen correlation to be used again for laminar flow. (Chapman 1984)

$$\text{Nu}_D = 3.66 + \frac{0.0668 \left(\frac{D}{L}\right) \text{Re}_D \text{Pr}}{1 + 0.4 \left[\left(\frac{D}{L}\right) \text{Re}_D \text{Pr}\right]^{2/3}} \quad (4.10)$$

With turbulent flow, the Nusselt correlation was used. (Chapman 1984)

$$\text{Nu}_D = 0.036 \text{Re}_D^{0.8} \text{Pr}^{1/3} \left(\frac{D}{L}\right)^{1/18} \quad (4.11)$$

In both of these equations, Re_D is the Reynolds number of the flow based on the diameter of the pipe and all of the properties are evaluated at the average of the inlet and outlet temperature of the fluid. The Reynolds number is used to determine whether the flow is laminar or turbulent, with Reynolds numbers greater than 2300 considered turbulent. From the calculated Nusselt number the heat transfer coefficient is determined. Air is circulated on the outside of the pipes, resulting in heat transfer by forced convection over horizontal tubes. It is assumed that the tubes do not have extended surfaces on the outside. Churchill and Bernstein developed a correlation equation for this type of heat transfer. (Chapman 1984)

$$\text{Nu}_D = 0.3 + \frac{0.62 \text{Re}_D^{1/2} \text{Pr}^{1/3}}{\left[1 + (0.4/\text{Pr})^{2/3}\right]^{1/4}} \left[1 + \left(\frac{\text{Re}_D}{2.82 \times 10^5}\right)^{5/8}\right]^{4/5} \quad (4.12)$$

Re_D is the Reynolds number based on the diameter and all of the properties are evaluated at the mean temperature. The heat transfer coefficients are calculated from the Nusselt numbers and then the overall heat transfer coefficient can be calculated.

$$U = \frac{1}{\frac{r_o}{r_i h_i} + \frac{r_o \ln\left(\frac{r_o}{r_i}\right)}{k_{\text{pipe}}} + \frac{1}{h_o}} \quad (4.13)$$

In this equation the subscript o stands for the outside the pipe and i for the inside. The calculated overall heat transfer coefficient is multiplied by the total heat transfer area to determine the UA product for the heat exchanger.

Once the UA product is calculated, it is necessary to determine which of the two streams has the lower capacitance rate. Since the specific heat of air is significantly lower than the specific heat of any of the secondary fluids, the air capacitance rate is likely to be the minimum capacitance rate. The NTU for the heat exchanger is determined based on this minimum capacitance rate.

$$NTU = \frac{UA}{\dot{m}_{\text{air}} C_{p\text{air}}} \quad (4.14)$$

Since the flows on both sides of the heat exchanger are single phase, the effectiveness-NTU formula for an isothermal phase change, Equation 3.2, is not applicable. Instead the formula for cross-flow heat exchangers with both fluids unmixed is used. (Incropera 1985)

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right)(NTU)^{0.22} \left\{\exp[-C_r(NTU)^{0.78}] - 1\right\}\right] \quad (4.15)$$

The effectiveness is used with the minimum capacitance rate and the temperature difference of the two incoming streams to determine the heat transfer.

$$Q = \epsilon \dot{m}_{\text{air}} C_{p,\text{air}} (T_{\text{air},\text{in}} - T_{\text{sf},\text{in}}) \quad (4.16)$$

This heat transfer can then be used to calculate the temperature change of the air and secondary fluid streams.

$$Q = \dot{m}_{\text{air}} C_{p,\text{air}} \Delta T_{\text{air}} \quad (4.17)$$

$$Q = \dot{m}_{\text{sf}} C_{p,\text{sf}} \Delta T_{\text{sf}} \quad (4.18)$$

The calculated heat transfer in the case heat exchanger is the same as the load met by the refrigerated case. Either the temperature change of the fluid can be given to calculate the load met, or the required load can be given to determine the necessary temperature change.

4.2.2 Thermal Losses from Piping System

The secondary fluid needs to be distributed from the heat exchanger where it is cooled by the ammonia to the refrigerated case where it provides the cooling to the air. The piping system 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 secondary fluid inside the pipes is assumed to be fully developed both hydrodynamically and thermally and to be in turbulent flow. The heat transfer coefficient on the inside of the pipes can be calculated using the Dittus-Boelter correlation. (Incropera 1985)

$$\text{Nu}_i = 0.023 \text{Re}_D^{0.8} \text{Pr}^{0.4} \quad (4.19)$$

All properties are calculated from the average bulk temperature of the secondary fluid. The bulk temperature is determined as the average of the inlet and outlet secondary fluid temperatures. The pipes carrying the secondary fluid are assumed to be exposed to the air of the supermarket. The heat transfer in this case would be forced convection due to the air currents inside of the store. However, this type of heat transfer is dependent on the velocity of the air around the pipes. Since this is not an easily determined value and since it depends on many factors, such as people walking by the pipes, a constant heat transfer coefficient was used for the outside of the pipes. Chapman states that for convection into air, the heat transfer coefficient is often very close to 6 W/m²-C and this value is used in this study. (Chapman 1984) With the inside and outside heat transfer coefficients determined, the overall heat transfer coefficient can be calculated for the pipes. The calculation is done based on the assumption that there is some amount of insulation on the pipes. Conduction through the pipe walls is neglected because the heat transfer resistance through the wall would be significantly lower than the resistance through the insulation.

$$U = \frac{1}{\frac{r_{total}}{r_i h_i} + \frac{r_{total} \ln \left[\frac{r_{total}}{r_i} \right]}{k_{ins}} + \frac{1}{h_o}} \quad (4.20)$$

In Equation 4.20, r_{total} is the diameter of the pipe plus the insulation. Subscript i refers to the inside of the pipe, subscript o refers to the outer surface of the insulation, and k_{ins} is the thermal conductivity of the insulation.

With the overall heat transfer coefficient calculated, the heat transfer between the pipes and the environment can be determined. Since the heat transfer properties are

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.

$$\Delta T_{lm} = \frac{T_{sf,out} - T_{sf,in}}{\ln \left[\frac{T_{air} - T_{sf,in}}{T_{air} - T_{sf,out}} \right]} \quad (4.21)$$

This log mean temperature difference is used with the overall heat transfer coefficient and the total heat transfer area to determine the heat transfer.

$$Q = UA\Delta T_{lm} \quad (4.22)$$

The heat transfer area (A) is the surface area of the insulation exposed to the supermarket air. The energy balance of the secondary fluid stream can be expressed.

$$Q = \dot{m}_{sf} C_{p,sf} \Delta T_{sf} \quad (4.23)$$

With these two different equations, it is possible to determine the thermal losses of the piping system to the environment.

4.2.3 Pump Work

To circulate the secondary fluid through the piping system, it is necessary to overcome the pressure drops from friction in the pipes. 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) The correlation equations of the Moody diagram depend on whether the flow is laminar or turbulent. For laminar flow, the friction factor is inversely proportional to the Reynolds number.

$$f = \frac{64}{Re_D} \quad (4.24)$$

Re_D is the Reynolds number based on the diameter. For turbulent flow, a correlation equation developed by Haaland is utilized. (White 1986)

$$\frac{1}{f^{1/2}} = -1.8 \log \left[\frac{6.9}{Re_D} + \left(\frac{\epsilon/D}{3.7} \right)^{1.11} \right] \quad (4.25)$$

Re_D is the Reynolds number based on diameter, ϵ is the roughness of the pipe, and D is the diameter of the pipe. The frictional head losses can be calculated once the friction factor has been determined.

$$\text{head} = f \left(\frac{L}{D} \right) \left(\frac{v^2}{2g} \right) \quad (4.26)$$

L is the length of the pipe, D is the diameter, v is the velocity of the fluid flow through the pipes, and g is the acceleration due to gravity.

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. The method for determining the minor losses is outlined by White. (White 1986) 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 from all of the bends and valves. This total equivalent loss coefficient is then used to calculate the head losses from the minor losses.

$$\text{head} = K_{eq} \left(\frac{v^2}{2g} \right) \quad (4.27)$$

The total head loss is the sum of the losses due to friction and the minor losses. With the total head losses, the pressure drop through the pipes can be calculated.

$$\Delta P = (\text{head}) \rho g \quad (4.28)$$

Where head is the total head losses, g is the acceleration of gravity, and ρ is the density of the fluid flowing through the pipes.

Once the pressure drop is determined, the pump work needed can be calculated.

$$W_{\text{pump}} = \dot{m}_{\text{sf}} \left(\frac{\Delta P}{\rho} \right) \quad (4.29)$$

The pump work is calculated for pressure drop in the distribution lines, as well as through each component that the secondary fluid flow through. Thus, there is pump work associated with the distribution lines, the heat exchanger in the refrigerated case, and the heat exchanger with ammonia. The pump work is calculated using motor and mechanical efficiencies of one. The addition of pump work into the system model alters the coefficient of performance (COP) equation. The pump work is added to the compressor work to penalize the additional circulation needed.

$$\text{COP} = \frac{\text{caseload}}{W_{\text{compressor}} + W_{\text{pump}}} \quad (4.30)$$

**CHAPTER
FIVE**

PARAMETRIC STUDY

The computer model written to simulate the performance of the supermarket refrigeration system utilizes many design parameters that influence the performance of the system. The highest COP means the refrigeration load is met by the least amount of compressor and pump power. It is necessary to determine the influence that each of these parameters has on the performance of the whole system. In order to evaluate the influence, a parametric study of the system was performed by first determining a base case of all of the parameters and then varying the parameters one at a time to observe how the system performance is affected.

5.1 Parameter List

Each of the components in the system model have parameters that would be specified by the system designer and these parameters are described here. The parameters have been divided according to the component group which they influence.

5.1.1 System Parameters

The first set of parameters affect the system as a whole and not an individual component. The total refrigeration load met by the system is the first parameter. The overall heat transfer coefficient (UA) value of the condenser is a specified variable that primarily depends on the size and design of the condenser. The ambient air temperature is used to determine the performance of the condenser. The performance of the staged compression depends on the pressure ratio of the compressors. The intermediate pressure is usually specified by giving a pressure ratio for the first compressor that is some power of the overall pressure ratio. The equation for determining the pressure ratio (r_p) of the first stage of compression from the overall pressure ratio of the system is shown here, where x is the exponent (between 0.0 and 1.0) that is specified. (Chlumsky 1965)

$$r_{p_{\text{first stage}}} = (r_{p_{\text{overall}}})^x \quad (5.1)$$

The mass flow rate of the secondary fluid is the last system parameter in the model. The flow rate will determine the necessary temperature drops in the heat exchangers and the piping system losses.

5.1.2 Refrigerated Case Parameters

The refrigerated case is modeled as an air to secondary fluid cross-flow heat exchanger. The physical dimensions of the heat exchanger are necessary for the calculations. The temperature of the air leaving the heat exchanger is the desired temperature of the refrigerated case. Values for the output air of refrigerated cases can be obtained from the manufacturers of refrigerated cases. The values used in this study were determined from data from the Hussman Corporation that gave the output air temperatures for the Hussman refrigerated cases using R22 and R502 as the refrigerants. (Hussman 1989) On the air side of the heat exchanger, it is necessary to specify the air velocity and the cross sectional area of the heat exchanger that the air passes through. The air velocity in the cases can also be found in the case manufacturer's data. The heat exchanger cross sectional area depends on the case design. The number, the inside diameter, the thickness, and the length of the pipes used to carry the secondary fluid through the air stream are all specified in the model. Finally, the thermal conductivity of the material used for the pipes and the K_{eq} of the heat exchanger need to be specified. As discussed in Section 4.2.3, the K_{eq} is used to determine the minor losses of the flow through the heat exchanger that are part of the pressure drop calculations.

5.1.3 Ammonia-Secondary Fluid Heat Exchanger Parameters

The parameters for this flooded heat exchanger are similar to the ones for the case heat exchanger discussed above. Again the dimensions of the pipes used need to be specified. These dimensions are the number, length, inside radius, and thickness of the pipes. Also the thermal conductivity of the pipe material and the K_{eq} of the heat exchanger for minor losses need to be specified.

5.1.4 Piping System Parameters

A piping system is used to supply and return the secondary fluid from the ammonia-secondary fluid heat exchanger to the case heat exchanger. For the piping system used in this model, parameters are provided separately for the supply and return piping to allow the system to be modeled with different pipes and insulation for the two different pipe runs. For the pipes themselves, the length and the inside diameter need to be specified. The amount of insulation, as well as the thermal conductivity of the insulation, are parameters. In order to determine the losses to the store environment, the store temperature needs to be provided. The roughness of the pipe material is needed for the pump work calculations and the K_{eq} for each pipe run must again be specified.

5.1.5 Compressor Parameters

Since two stage compression is used in this system model, two sets of compressor parameters are needed. The bore and stroke of the piston in the compressor are parameters, as well as the speed at which the compressor operates and the clearance ratio. The polytropic model used for the compressors requires a polytropic exponent and polytropic efficiency. Finally, the number of cylinders of the compressor can be specified, but does not need to be. In the system parameters discussed in Section 5.1.1, the pressure ratio and secondary fluid flow rate parameters are discussed. If these two parameters are specified, then the number of cylinders being used is determined. If the number of cylinders being used is given, then the pressure ratio and the flow rate are calculated.

5.2 Influence on Performance by Parameters

The parameters discussed in Section 5.1 were varied to determine how specific parameters influence the performance of the system. In order to have a basis of comparison, it was first necessary to develop a base case of the parameters. The calculations to evaluate the influence of the parameters were done using EES. (Klein 1993)

5.2.1 Base Case

The parameters for the base case are listed here by component group. The system parameters are listed in Table 5.1. The secondary fluid flow rate is varied to observe the performance of the system over a range of different flow rates.

Parameter	Base Value
Refrigeration load	52.8 kW (15 tons)
UA value of the condenser	8.0 kW/K
Ambient temperature	300 K
Pressure ratio exponent	0.53
Secondary fluid flow rate	variable

Table 5.1 System parameter list

The parameter list for the refrigerated case heat exchanger is shown in Table 5.2. There are two temperatures listed for the air temperature leaving the case heat exchanger, these refer to the two different temperature regimes that were studied. It is important to note that the refrigeration load being studied represents many individual cases at one

temperature regime. The parameters for the refrigerated case heat exchanger are for a heat exchanger that would provide the cold air for all of the cases, not a single case. A single case would have a smaller load and would require a smaller heat exchanger. However, the approach used here correctly models the effect of the refrigerated cases on the refrigeration system.

Parameter	Base Value
Air temperature leaving the case	267 or 244 K
Number of pipes in heat exchanger	700
Inside radius of pipes	0.01 m
Thickness of pipes	0.0015 m
Length of pipes	2.0 m
Air velocity	5.0 m/s
Air cross sectional area	0.5 m ²
Thermal conductivity of pipes	43 W/m-C

Table 5.2 Refrigerated case parameter list

The parameter list for the heat exchanger between the ammonia and the secondary fluid is shown in Table 5.3. This heat exchanger is assumed to be a single large heat exchanger for cooling all the secondary fluid needed to cool all of the refrigerated cases at a single temperature regime.

Parameter	Base Value
Number of pipes in heat exchanger	300
Inside radius of pipes	0.005 m
Thickness of pipes	0.0015 m
Length of pipes	2.0 m
Thermal conductivity of pipes	43 W/m-C

Table 5.3 Ammonia-secondary fluid heat exchanger parameter list

The piping system parameters are listed in Table 5.4. It is assumed that the same material would be used for both the supply and return pipes so there is only a single pipe roughness parameter.

The parameters for the compressor are not needed in this simulation because the flow rate and the pressure ratio were used as parameters instead. By specifying the pressure ratio and the flow rate, the number of cylinders of the compressor is allowed to float to the needed values. If the parameters describing the dimensions of the compressor are changed, the number of cylinders needed will change so that the flow rate and pressure ratio are at the specified values.

Parameter	Base Value
Length of pipe to case	30 m
Length of pipe from case	30 m
Diameter of pipe to case	0.12 m
Diameter of pipe from case	0.12 m
Thickness of insulation on pipe to case	0.025 m
Thickness of insulation on pipe from case	0.025 m
Thermal conductivity of insulation to case	0.04 W/m-C
Thermal conductivity of insulation from case	0.04 W/m-C
Pipe roughness	0.000046 m
Air temperature in store	300 K

Table 5.4 Piping system parameter list

5.2.2 Base Case Performance

In order to evaluate the influence of the different parameters on the system performance, it was necessary to first look at the performance of the base case discussed in Section 5.2.1. The performance calculations used a 60% by volume propylene glycol solution as the secondary fluid. The COP of the system was determined for a range of mass flow rates of the propylene glycol solution from 0.75 to 15 kg/s. The COP is determined using Equation 4.30. The equation set used for the performance study is contained in Appendix B. In order to normalize the calculated results for different loads, the calculated performance was plotted against the ratio of the capacitance rate of the propylene glycol solution through the refrigerated case heat exchanger to the refrigeration

load met by the case. The graph showing how the COP varies with changing capacitance rate - caseload ratios is shown in Figure 5.1.

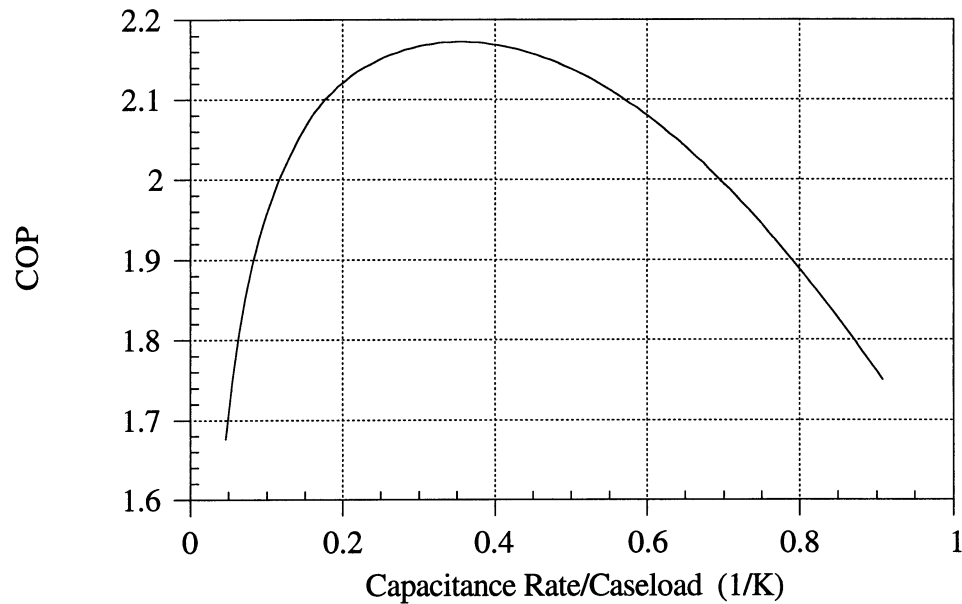


Figure 5.1 Performance of base case

5.2.3 Influence of System Parameters

The system parameters were evaluated first. The calculated performance of the system was plotted at refrigerated case loads of 61.5 kW (17.5 tons), 52.8 kW (15 tons), 44.0 kW (12.5 tons), and 35.2 kW (10 tons) and the resulting graph is shown in Figure 5.2. The graph shows that the lower the load the higher the COP, with all other parameters held constant. The increased load causes a greater temperature difference of the secondary fluid for the same mass flow rate, which in turn lowers the required temperature of the ammonia in the ammonia - secondary fluid heat exchanger. This lower ammonia temperature leads to lower system performance. The graph also indicates that there exists

an optimal capacitance rate - caseload ratio which is a function of the required temperature and mass flow rate of the propylene glycol solution, as well as the refrigeration load of the case. For all of the plotted case loads, the optimal ratio was between 0.3 and 0.5.

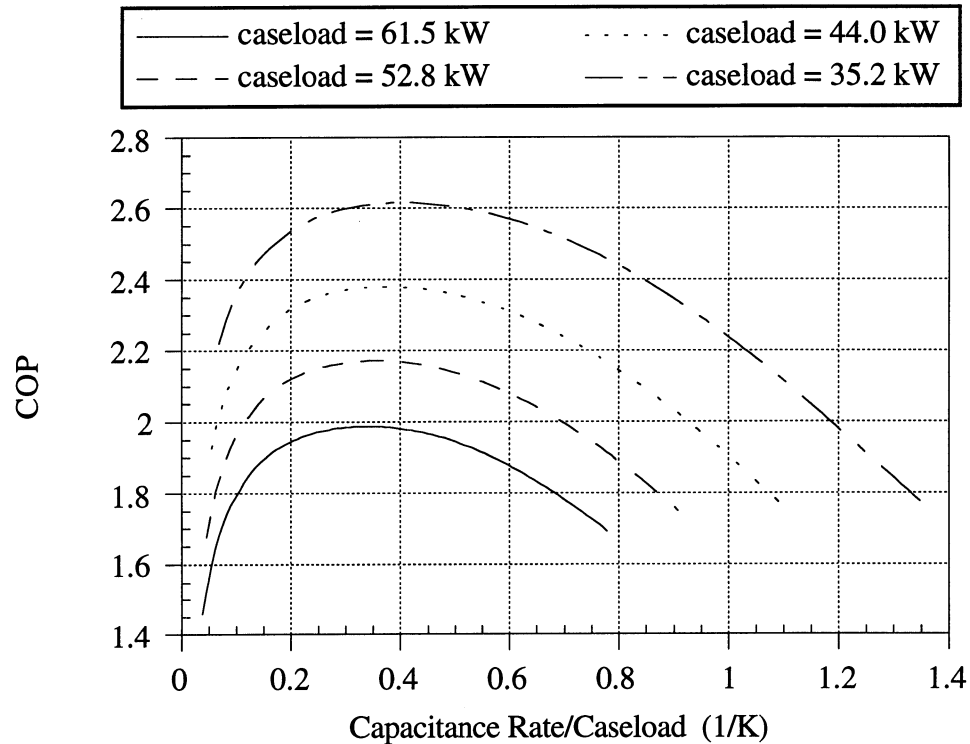


Figure 5.2 Influence of refrigerated case load on system performance

The influence on the system performance of the UA product is shown in Figure 5.3. In this graph, the system COP was evaluated with condenser UA values between 7.0 and 9.0 kW/K. The expected response of higher performance with higher UA values was observed. This means that when designing a system it is important to use the condenser with the highest UA value that is economically feasible. The optimal capacitance rate - caseload ratio for the system performance did not change with the changing UA values of the condenser. The ratio remained between 0.3 and 0.4.

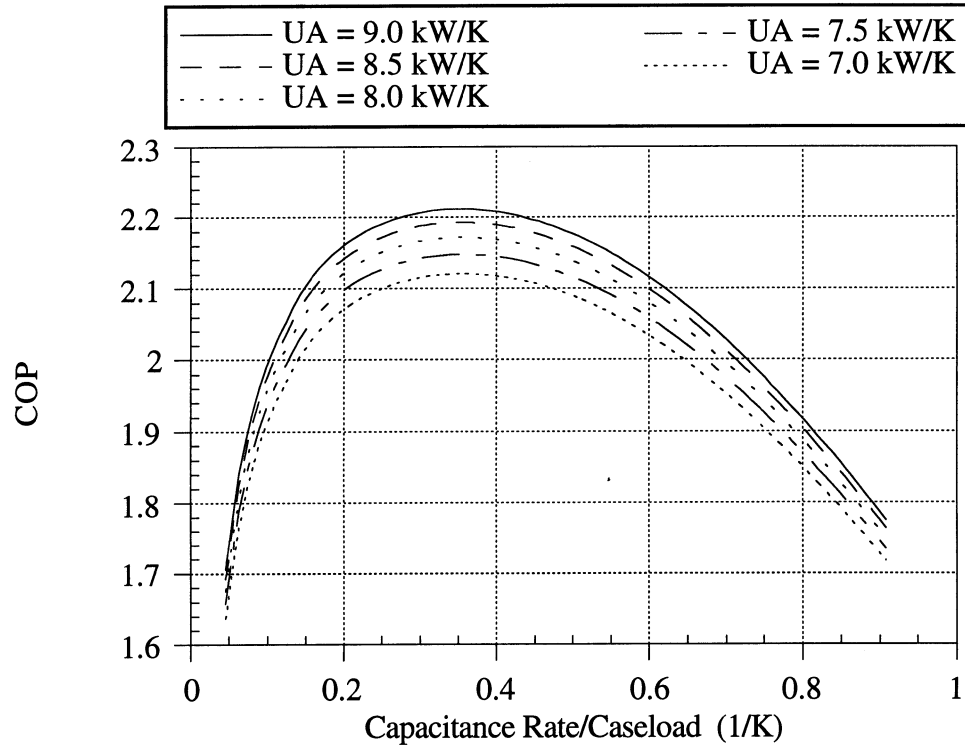


Figure 5.3 Influence of the UA value of the condenser on system performance

The influence of the ambient temperature on the system performance was calculated using temperatures from 270 to 310 °K (26.3 to 98.3 °F) and the results are shown in Figure 5.4. The COP of the system increases as the ambient temperature decreases, as expected. The ambient temperature influences the performance of the condenser so these results suggest that it is possible to increase the performance of the system by using a water-cooled or evaporative condenser rather than the air-cooled condenser assumed in this system analysis. The change in ambient temperature has only a minor influence on the optimal capacitance rate - caseload ratio, as the ratio remains between 0.3 and 0.4 regardless of the ambient temperature.

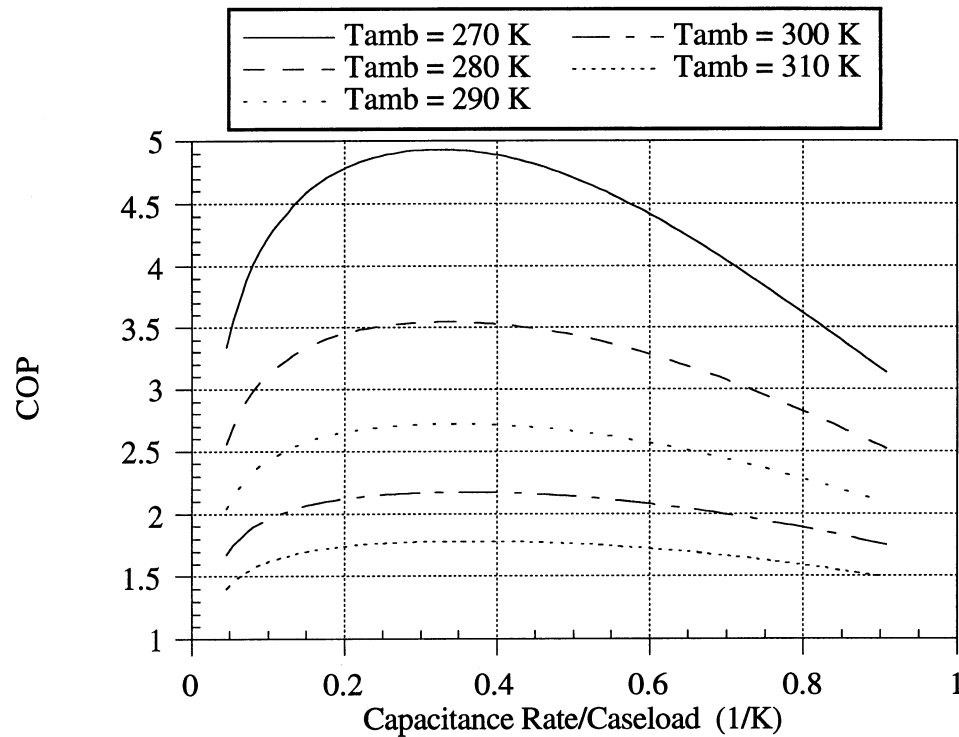


Figure 5.4 Influence of ambient temperature on system performance

The final system parameter that was analyzed for influence on the system performance is the pressure ratio exponent. This is the variable x in Equation 5.1 and discussed in Section 5.1.1. This exponent determines the intermediate pressure of the first stage of compression. The graph of the performance of the system with pressure ratio exponents from 0.2 to 0.8 is shown in Figure 5.5 and shows that the exponents 0.5 and 0.6 have the highest performance. As long as the intermediate pressure is chosen so that the pressure ratio exponent remains between 0.5 and 0.6, the performance will be relatively unaffected. The change in the pressure ratio exponent does very little to influence the optimal capacitance rate - caseload ratio, as it again remains between 0.3 and 0.4 for all of the exponents.

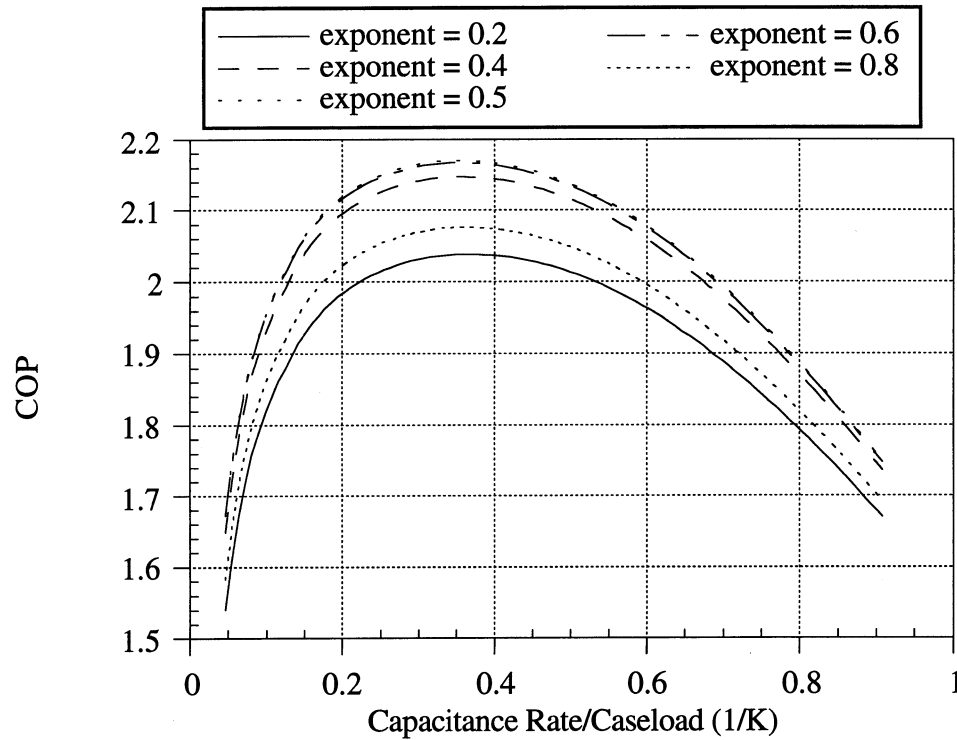


Figure 5.5 Influence of pressure ratio exponent on system performance

5.2.4 Influence of Refrigerated Case Parameters

The main parameter for the refrigerated case is the temperature of the air leaving the heat exchanger which is used to provide the cooling for the case itself. The two temperature regimes used in this study are air temperatures of 267 and 244 °K. Figure 5.6 shows that reducing the air temperature dramatically reduces the system performance. However, the optimum COP at either case temperature occurred at a capacitance rate - caseload ratio between 0.3 and 0.4. The other parameters studied in this chapter were analyzed with the air temperature leaving the case set at 267 °K.

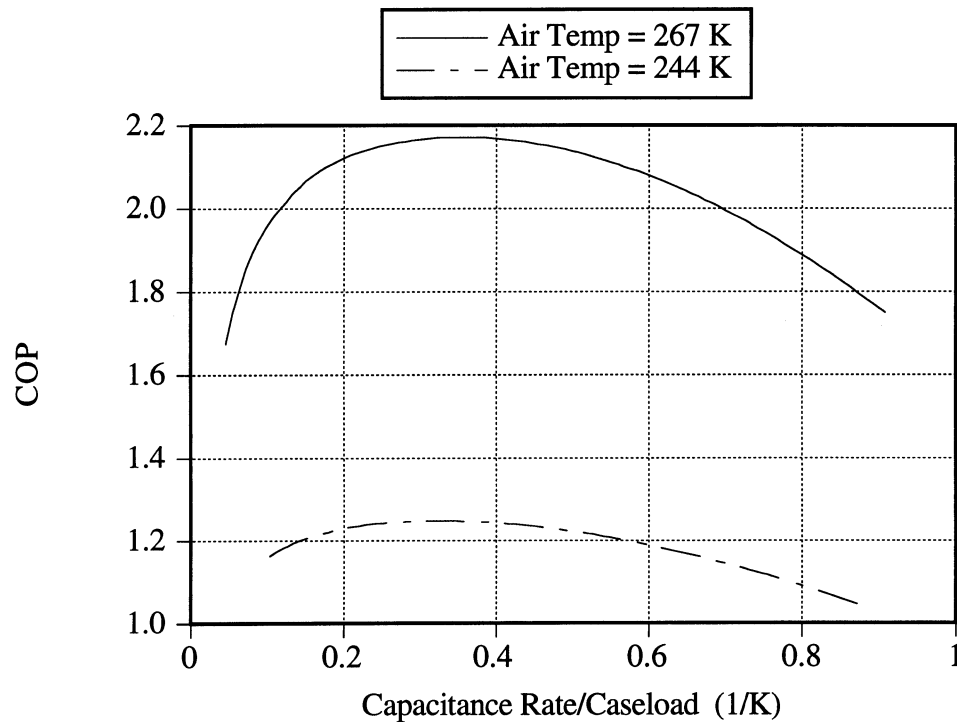


Figure 5.6 Influence of the temperature leaving the refrigerated case on system performance.

The other parameters used in the refrigerated case model influence the performance of the system by changing the UA value of the heat exchanger used to cool the air in the refrigerated case. The first of these parameters is the number of pipes in the refrigerated case heat exchanger. The system performance was plotted for the same capacitance rate - caseload ratios as used in the system parameter evaluations in Section 5.2.3. The system COP was calculated for 650 to 800 tubes, all of the same size, and the graph is shown in Figure 5.7. The graph shows that an increased number of pipes will slightly increase the system COP. The increase occurs because of an increase in the UA value of the heat exchanger. The highest performance is at a capacitance rate - caseload ratio between 0.3 and 0.4 for all of the different number of pipes.

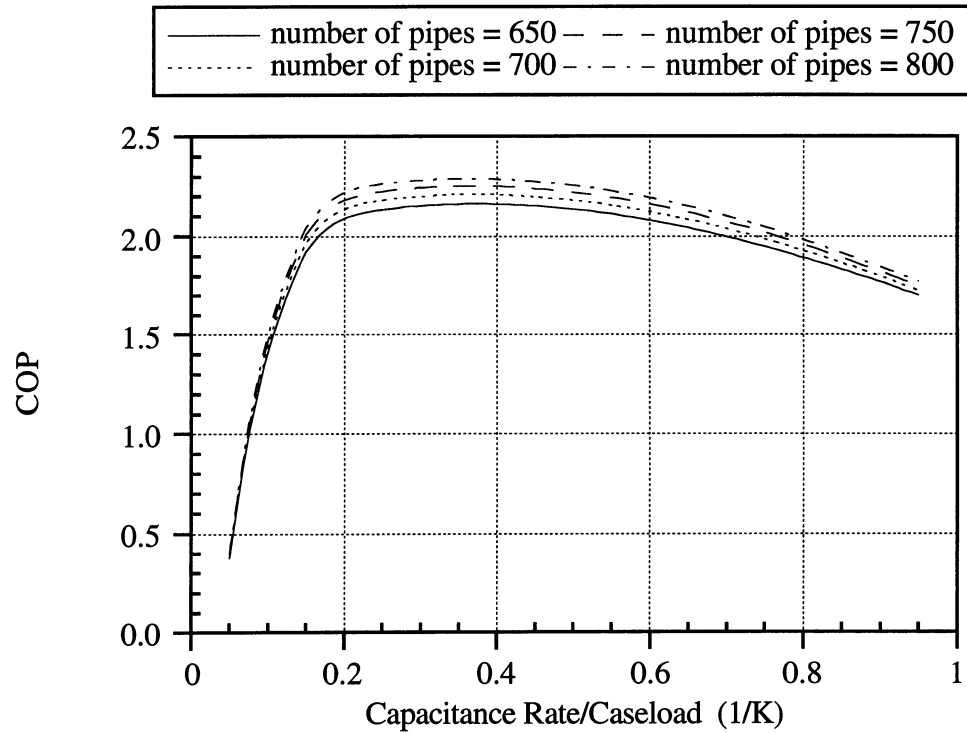


Figure 5.7 Influence of the number of pipes in the case heat exchanger on system performance

The inside radius of the pipes in the refrigerated case heat exchanger was varied from 0.0075 to 0.015 meters. The graph showing the dependence of the system performance on the inside radius of the pipes in the heat exchanger is shown in Figure 5.8. As the radius of the pipes increases, the UA value of the heat exchanger increases which increases the system COP. Figure 5.8 also shows the same optimal capacitance rate - caseload ratio between 0.3 and 0.4.

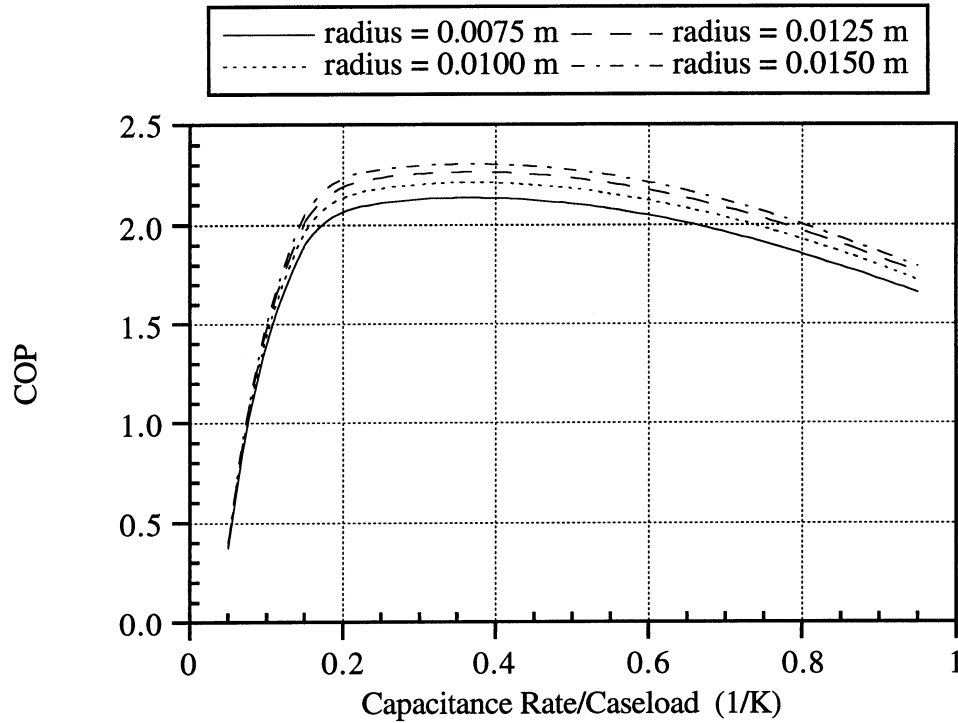


Figure 5.8 Influence of the radius of the pipes in the case heat exchanger on system performance

The analysis of the thickness of the pipes showed that if the pipes are of the same material (steel was used for these calculations) the thickness of the pipes in the refrigerated case heat exchanger has essentially no effect on the system performance. The influence of the length on the system performance is shown in Figure 5.9. The lengths of the pipes in the refrigerated case heat exchanger ranged from 1.75 to 2.75 meters. The system COP increases as the length is increased. The increase occurs because of an increase in UA value without an equivalent increase in required pumping power. As with the other parameters, the optimal capacitance rate - caseload ratio is between 0.3 and 0.4 for the length of the pipes.

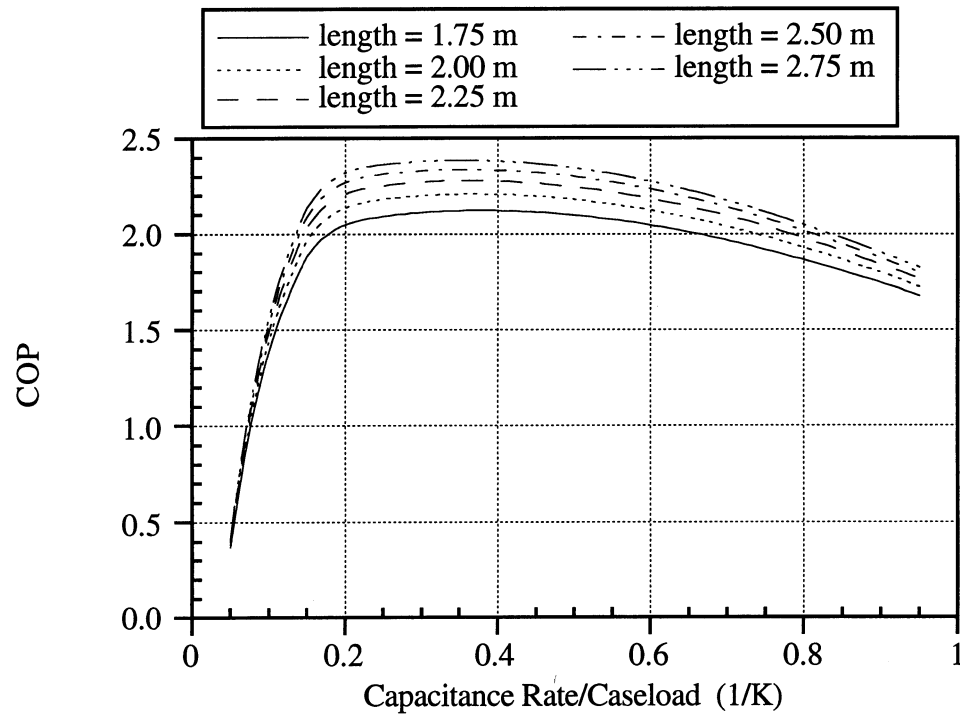


Figure 5.9 Influence of the length of the pipes in the case heat exchanger on system performance

The parameters affecting the air side of the refrigerated case are the air velocity and the cross sectional area of the air passage. The comparison for these parameters is done in the same manner as the other refrigerated case parameters, with the system COP plotted versus the changing capacitance rate - caseload ratios for different values of the parameter. The plot of the influence of the air velocity, varied from 4.0 to 6.0 meters per second, is shown in Figure 5.10. The plot shows that the changing velocity has no effect on the overall system performance. The plot also shows that, as with the pipe parameters, the optimal capacitance rate - caseload ratio is between 0.3 and 0.4.

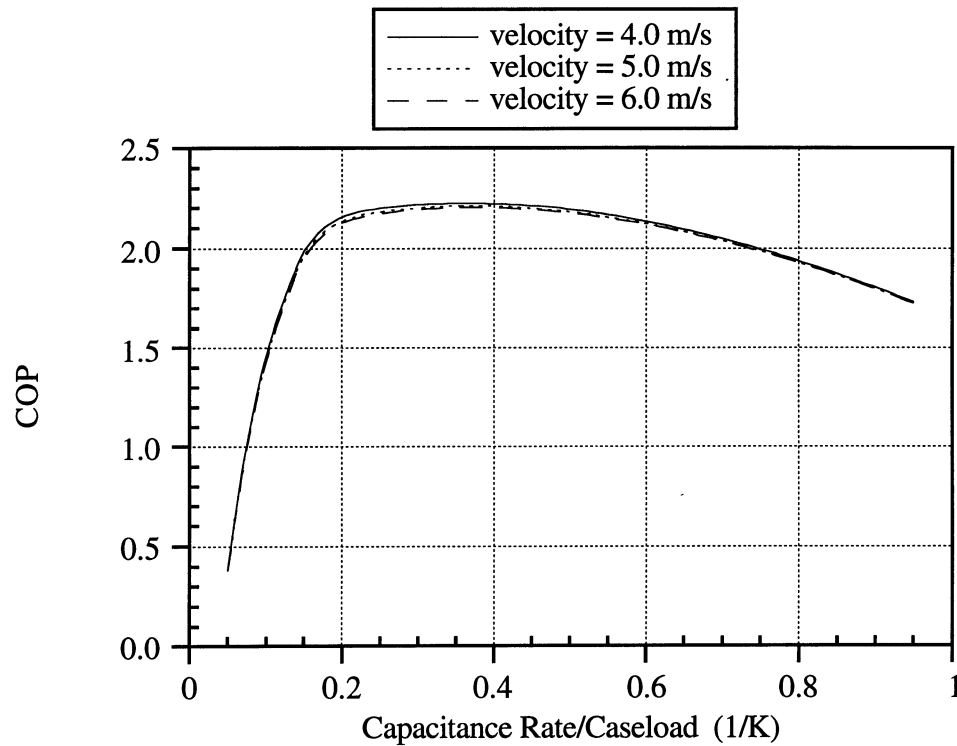


Figure 5.10 Influence of the velocity of the air through the refrigerated case heat exchanger on the system performance

The cross sectional area in the refrigerated case that the air flows through was varied from 0.4 to 0.6 m² and the plot of the influence on the system performances is shown in Figure 5.11. The plot shows that as the area increases the COP decreases. The increase in area increases the capacitance rate of the air through the case which decreases the system COP. The curves have the same general shape of all of the other curves from the refrigerated case parameters. So for the best performance of the case heat exchanger, the capacitance rate - caseload ratio should be between 0.3 and 0.4. For every parameter evaluated in this section, the optimal capacitance rate - caseload ratio was between 0.3 and 0.4.

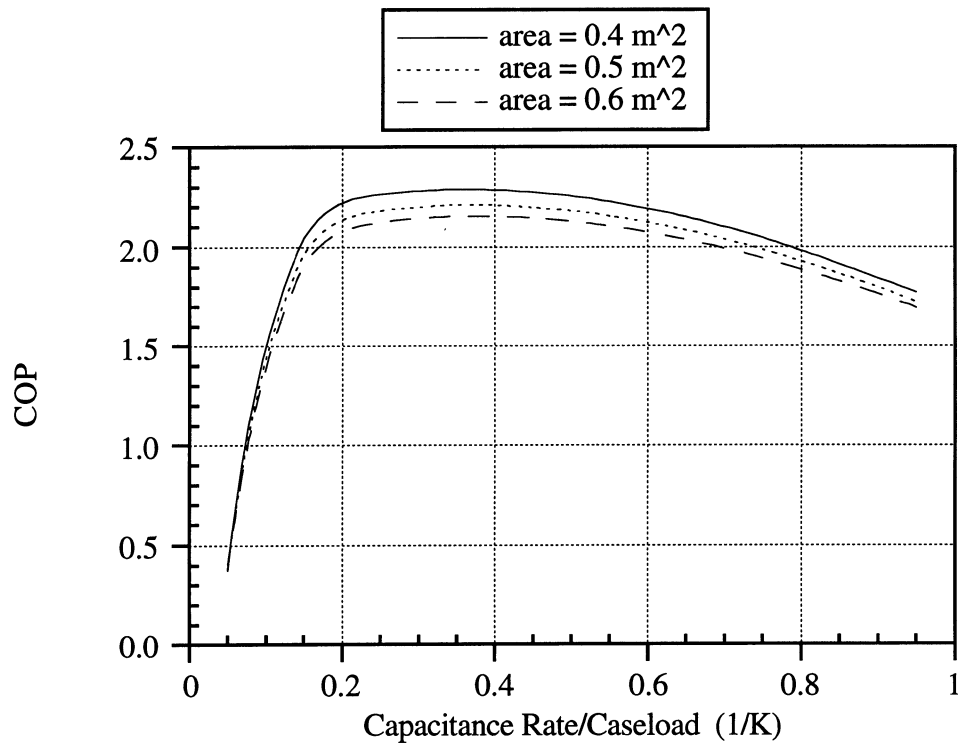


Figure 5.11 Influence of the cross sectional area that the air flows through in the refrigerated case heat exchanger on the system performance

5.2.5 Influence of Ammonia - Secondary Fluid Heat Exchanger Parameters

The parameters of the ammonia - secondary fluid heat exchanger affect the performance of the system through the effectiveness of the heat exchanger. Lower effectiveness values result in reduced evaporator temperature and pressures and thus lower system COP values.

The first parameter is the number of pipes in the heat exchanger. These are the pipes that the propylene glycol flows through to evaporate the ammonia on the outside of the pipes. The number of pipes was varied from 250 to 400. The plot of the system performance as it varies with the capacitance rate - caseload ratio for different numbers of pipes is shown in Figure 5.12. COP increases as the number of pipes increases from the

increased UA value for the heat exchanger. It also shows that the optimal capacitance rate - caseload ratio is between 0.3 and 0.4.

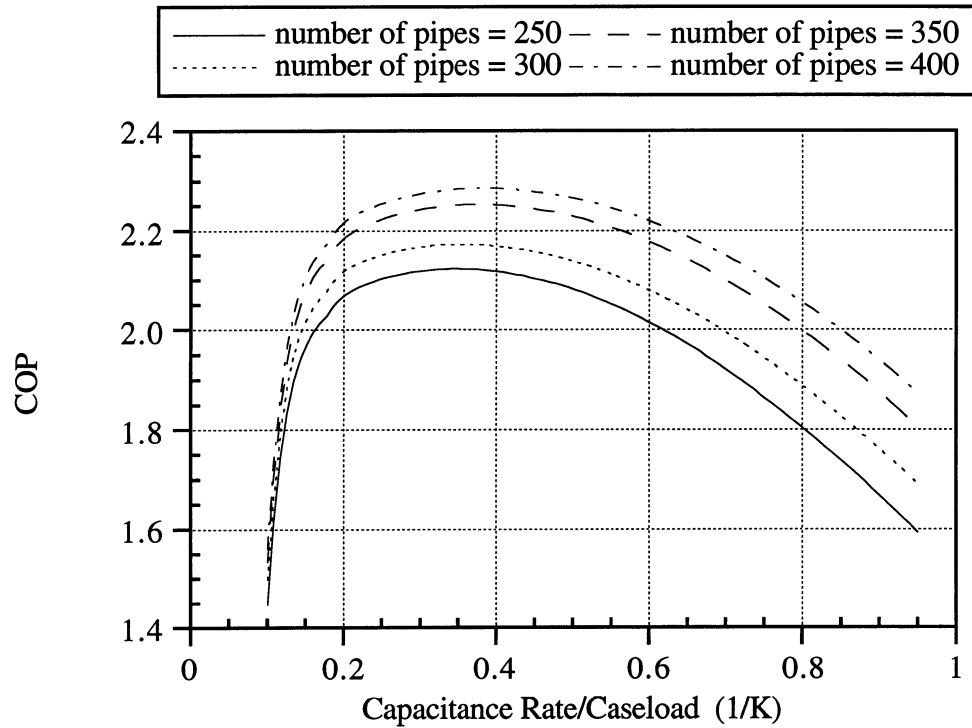


Figure 5.12 Influence of the number of pipes in the ammonia - secondary fluid heat exchanger on system performance

To evaluate the influence of the inside radius of the pipes in the heat exchanger, the radius was varied from 0.0025 to 0.0100 m. The plot showing the change in the system performance with the change in pipe radius is shown in Figure 5.13. The plot indicates that as the inside radius of the pipes increases the system COP increases. The increase in COP is due to the increase in heat exchanger UA value. The plot also indicates that, if a too small radius pipe is chosen for the system, the performance will be inhibited. This is caused by an increase in the friction in the pipes which increases the needed pumping power. The optimal capacitance rate - caseload ratio also changes with the increase in

radius, however, at the radius of 0.0050 m and above, the lines are flat and a wide range of capacitance rate - caseload ratios will give near optimal performance.

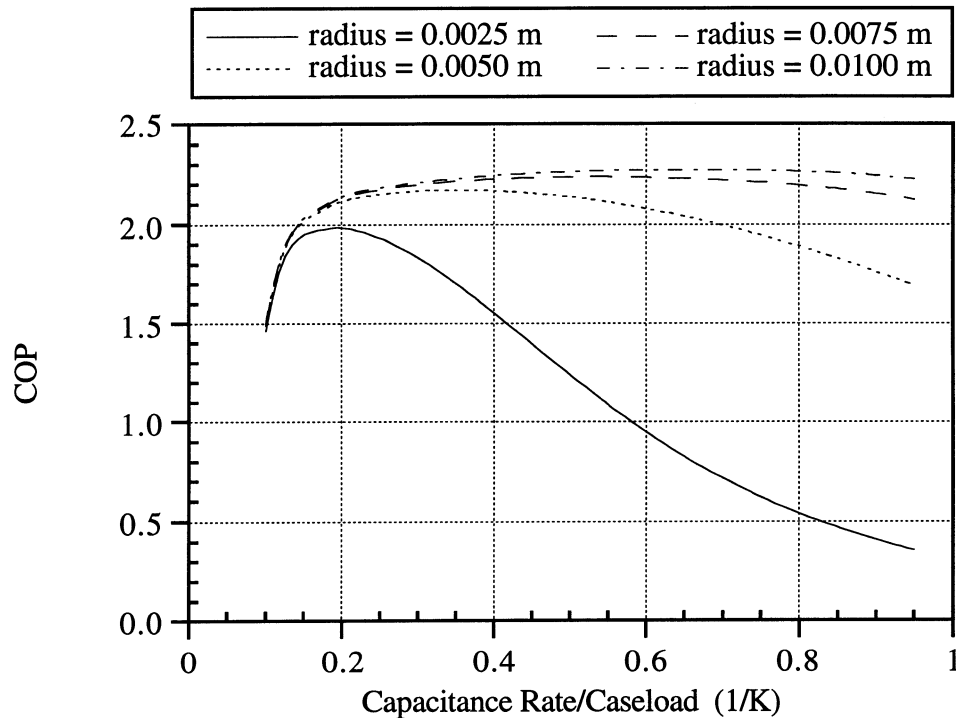


Figure 5.13 Influence of the radius of the pipes in the ammonia - secondary fluid heat exchanger on the system performance

The thickness of the pipe showed very little effect on the performance of the system. For the analysis the pipes were considered to be made of steel and the thermal conductivity did not change. The length of the pipes in the ammonia - secondary fluid heat exchanger was varied from 1.5 to 3.0 meters to determine the influence that the length has on the system performance. The plot of the system COP as it varies with pipe length is shown in Figure 5.14. The plot shows that the longer the pipes the higher the COP. Again the increase in the length of the pipes increases the UA value of the heat

exchanger. The plot shows that the optimal capacitance rate - caseload ratio is between 0.3 and 0.4.

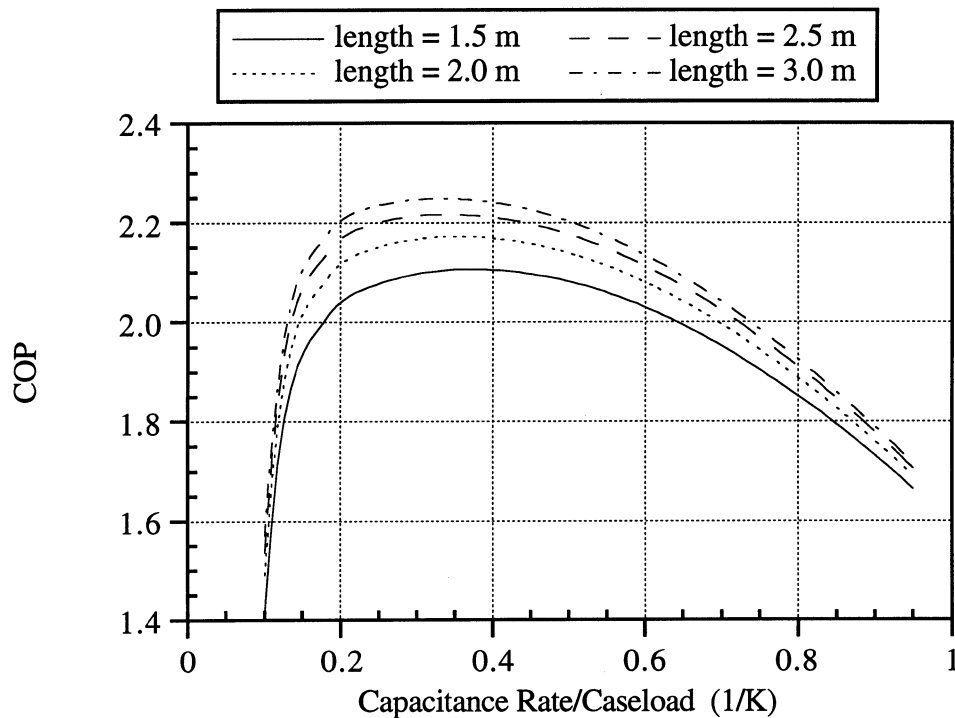


Figure 5.14 Influence of the length of the pipes in the ammonia - secondary fluid heat exchanger on the system performance

5.2.6 Influence of Piping System Parameters

The length of the pipe from the ammonia - secondary fluid heat exchanger to the refrigerated case was varied from 10 to 50 meters and the plot showing the system COP as it varies with length is shown in Figure 5.15. The graph shows that as the pipe length increases the performance decreases. The increased pipe length increases the pressure drop in the pipes and increases the thermal losses into the store environment. The optimal capacitance rate - caseload ratio is between 0.3 and 0.4 for all of the lengths.

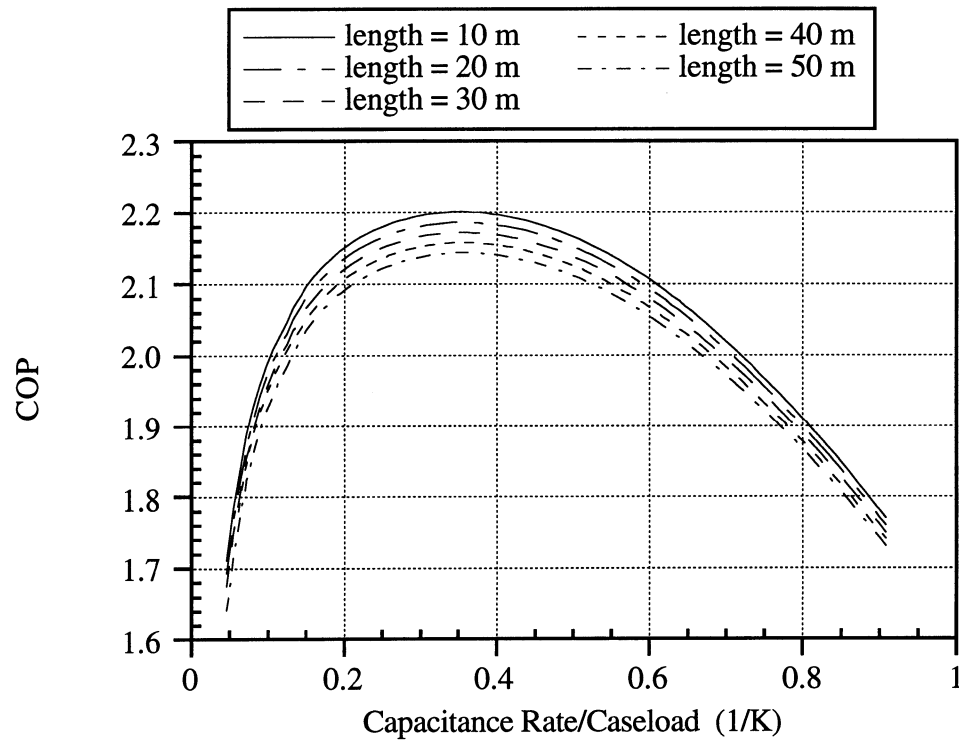


Figure 5.15 Influence of the length of the pipe to the refrigerated case on system performance

The plot of the influence of the length of the pipe from the refrigerated case to the ammonia - propylene glycol heat exchanger being varied from 10 to 50 meters is shown in Figure 5.16. As with the pipe length to the refrigerated case, the performance decreases as the pipe length increases. The optimal capacitance rate - caseload ratio is again between 0.3 and 0.4 for all of the lengths.

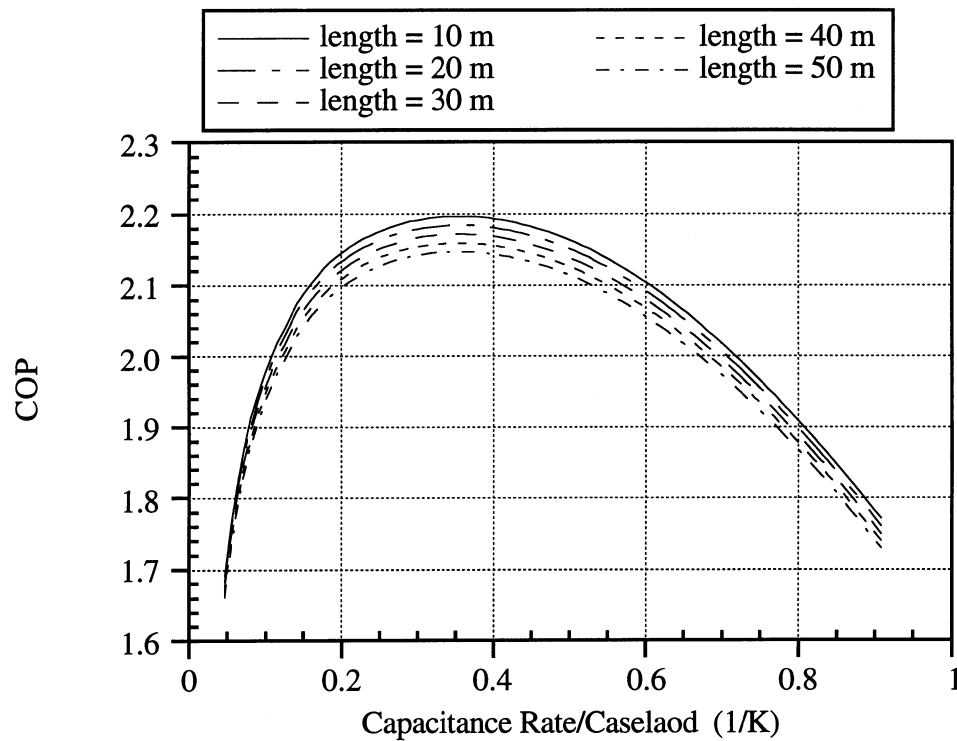


Figure 5.16 Influence of the length of the pipe from the refrigerated case on system performance

It is likely that the length of the pipes to and from the refrigerated case would be the same. The influence of the change in length for both pipes from 10 to 80 meters is shown in Figure 5.17. With both of the pipes being varied, the same decrease in performance as the length increases is seen, but it is a stronger dependence. The optimal capacitance rate - caseload ratio remains between 0.3 and 0.4 even when both lengths are varied.

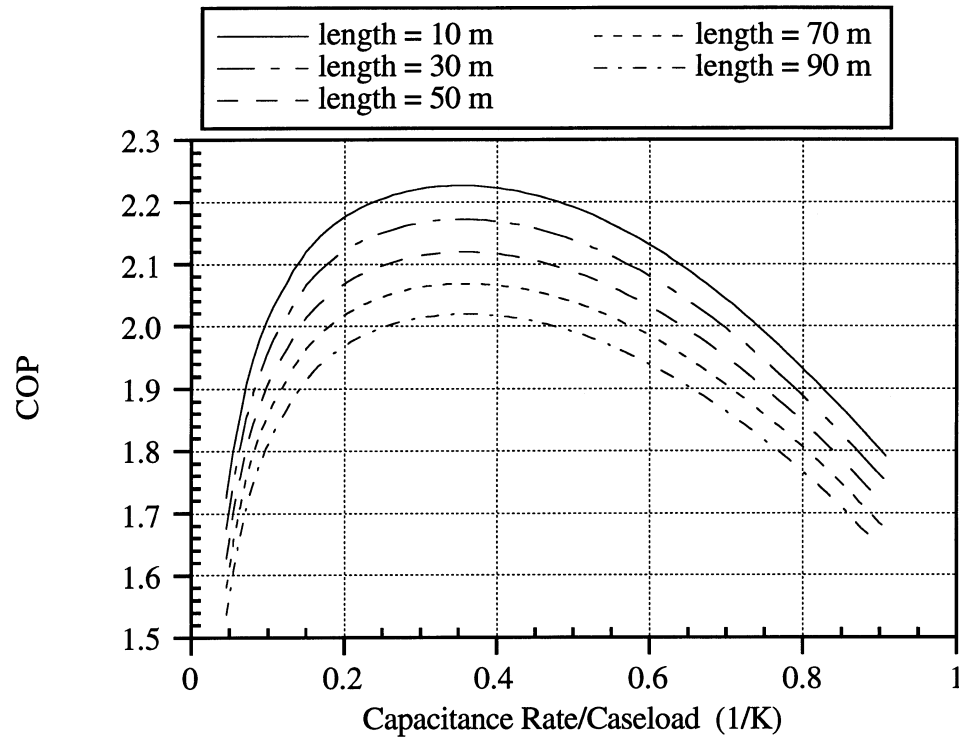


Figure 5.17 Influence of the length of the pipes to and from the refrigerated case on system performance

To evaluate the influence of the diameter of the pipes from the ammonia - secondary fluid heat exchanger to the refrigerated case, the values were varied from 0.04 to 0.20 meters and the system COP was calculated. The graph of the dependence of the system performance on the diameter of the pipes to the refrigerated case is shown in Figure 5.18. The graph shows that all of the pipe diameters give approximately the same performance except for the 0.04 m diameter. It is important to have a large enough pipe diameter, but after a certain size an increase in size will not increase performance. The 0.04 m diameter pipe also has a peak performance at a capacitance rate - caseload ratio of between 0.2 and 0.3, while the other diameters peak performance is between 0.3 and 0.4.

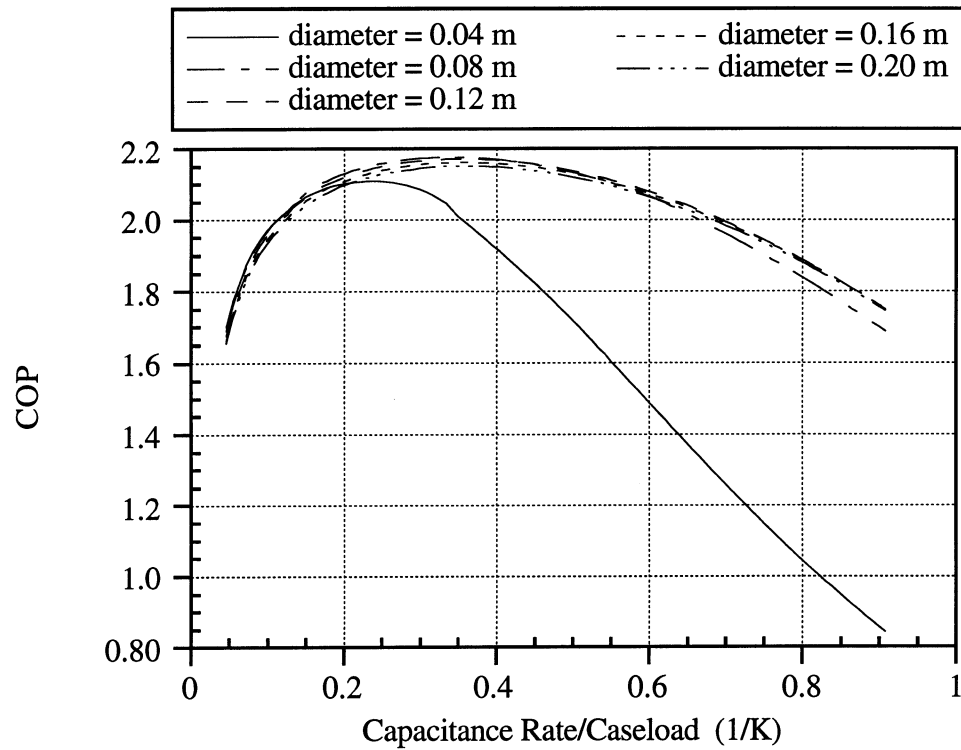


Figure 5.18 Influence of the diameter of the pipe to the refrigerated case on system performance

As for the diameter of the pipe to the refrigerated case, the diameter of the supply pipe was varied from 0.04 to 0.20 meters. The graph of the dependence of the system performance on the pipe diameter is shown in Figure 5.19. The influence of the diameter of the pipe from the refrigerated case is similar to the influence of the pipe to the refrigerated case. Again, all of the diameters have almost the same performance except for the 0.04 m diameter, which has a significantly lower performance. The optimal capacitance rate - caseload ratio is between 0.02 and 0.3 for the 0.04 m diameter and between 0.3 and 0.4 for the other diameters.

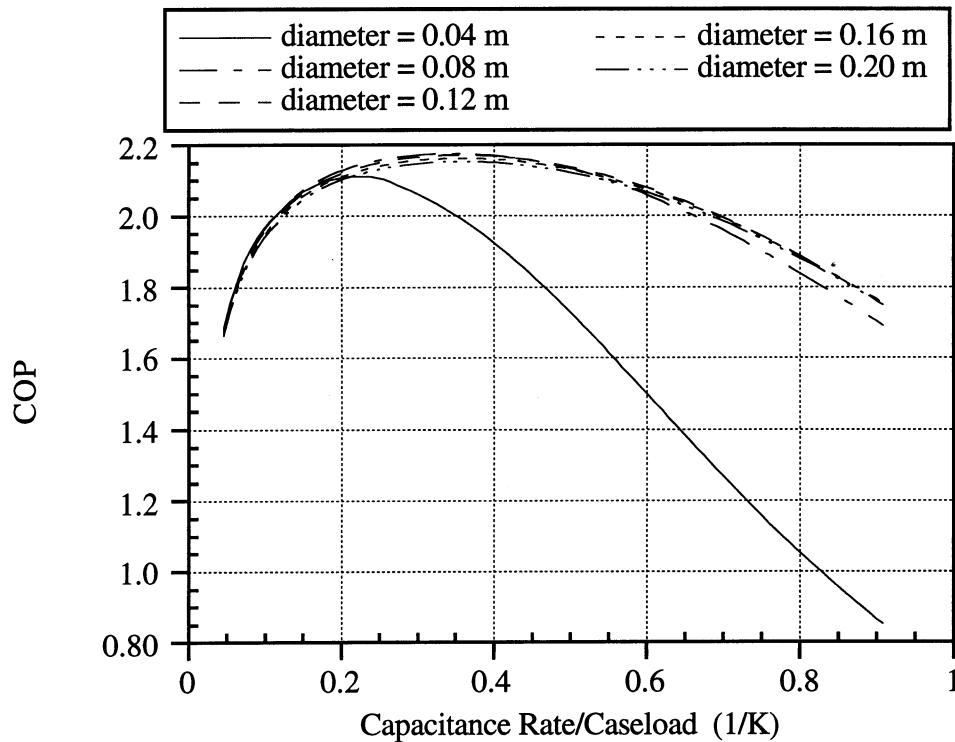


Figure 5.19 Influence of the diameter of the pipe from the refrigerated case on system performance

Because the supply and return pipes will both carry liquid, the two pipes will likely have the same diameter. To test the influence of changing both the pipe diameters, the performance was analyzed for diameters from 0.04 to 0.20 meters. The plot is shown in Figure 5.20. As with the individual pipe diameter plots, the pipe diameter of 0.04 m has a significantly lower performance, while the other diameters are approximately equal. The optimal capacitance rate - caseload ratio is between 0.3 and 0.4 for the diameters, except for the 0.04 m diameter.

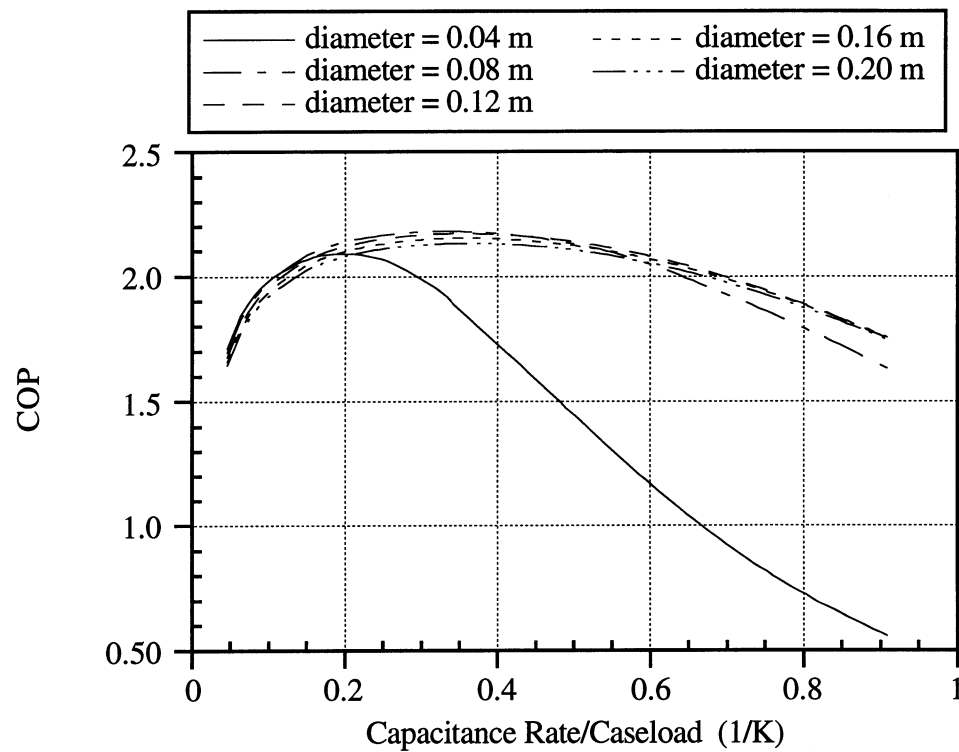


Figure 5.20 Influence of the diameter of the pipes to and from the refrigerated case on system performance

To determine the influence that the amount of insulation on the pipes from the ammonia - secondary fluid heat exchanger to the refrigerated case, the system performance was calculated for insulation thicknesses of 0.005 to 0.035 meters. The graph of the dependence of the system performance on the amount of insulation is shown in Figure 5.21. The graph shows that as the amount of insulation increases the performance increases. As the amount insulation increases from a low amount the performance increases significantly, but as more insulation is added the increases in performance becomes negligible. The change in amount of insulation does not affect the optimal capacitance rate - caseload ratio which remains between 0.3 and 0.4.

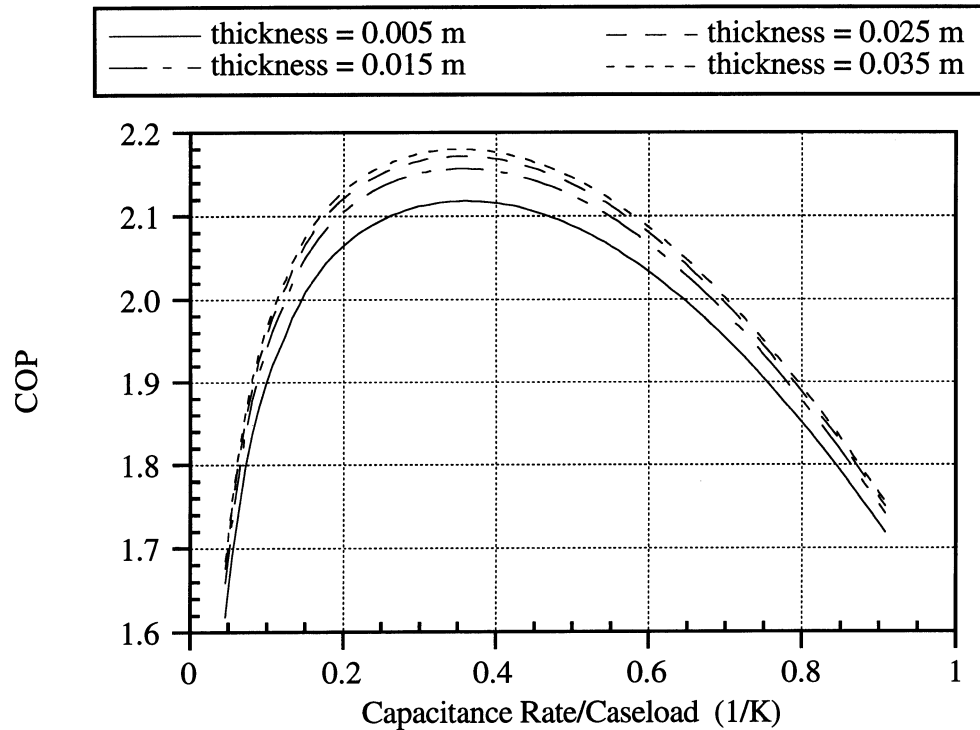


Figure 5.21 Influence of the thickness of the insulation on the pipes to the refrigerated case on system performance

The graph showing the influence of the amount of insulation on the pipes from the refrigerated case to the ammonia - secondary fluid heat exchanger, varied from 0.005 to 0.035 meters, is shown in Figure 5.22. The graph shows that, as with the amount of insulation on the pipes to the refrigerated case, as the insulation thickness increases, the system performance increases. The dependence is strong as insulation is first added and the dependence decreases as the amount of insulation increases. The amount of insulation has no impact on the optimal capacitance rate - caseload ratio which remains between 0.3 and 0.4.

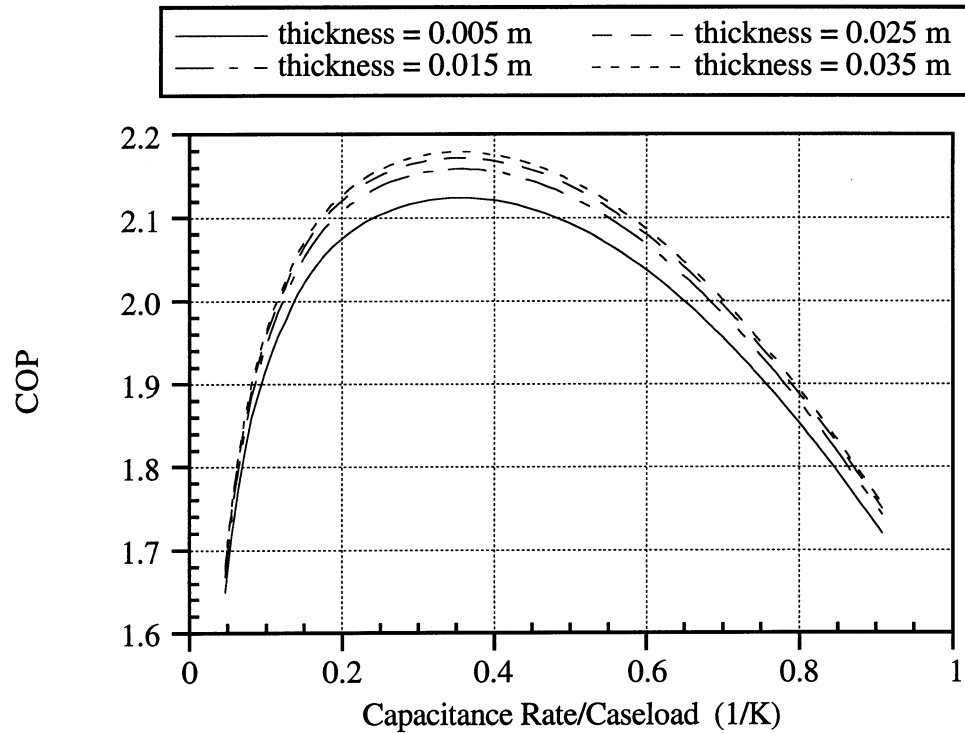


Figure 5.22 Influence of the thickness of the insulation on the pipes from the refrigerated case on system performance

Figure 5.23 shows the influence of the store temperature, varied from 290 to 305 °K, on the system performance. The graph shows that the performance increases slightly as the temperature decreases. The decrease in store temperature decreases the amount of losses from the piping system to the store environment. The change in performance is not great as long as the temperature does not increase dramatically. The store temperature does not affect the optimal capacitance rate - caseload ratio which remains between 0.3 and 0.4 for all of the store temperatures.

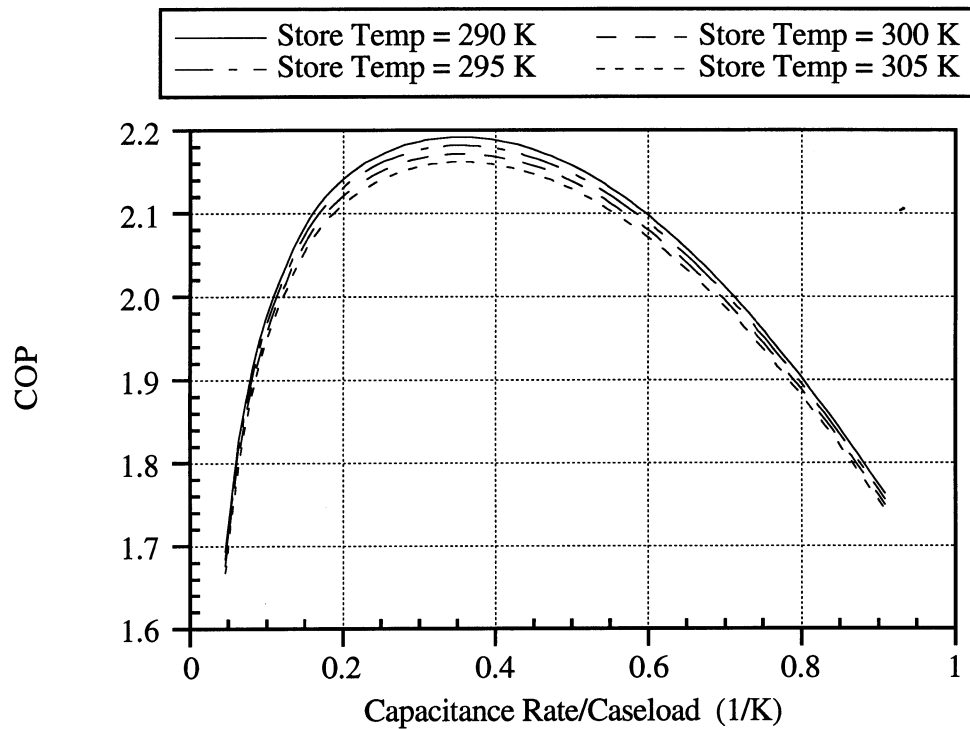


Figure 5.23 Influence of the store temperature on system performance

5.3 R22 System

Ammonia - secondary fluid refrigeration systems will only be practical 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 penalties associated with the heat exchangers and the distribution of the secondary fluid throughout the supermarket. To compare the performance to the performance of a R22 system, it was necessary to write a model of a R22 system that included an heat exchanger in the refrigerated case and the pumping costs of the R22.

5.3.1 R22 Base Case

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 the equations used in the ammonia - secondary fluid system model. These equations are discussed in Sections 4.2.2 and 4.2.3. The refrigerated case is assumed to have the 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 and shown in Equation 4.12. The R22 flow in the pipes is changing phase and the correlation equations become confounded by the forced convection and the phase change. An assumption was made that the inside heat transfer coefficient would be an order of magnitude greater than the heat transfer coefficient of the outside of the tubes due to the phase change of the R22 on the inside of the pipes. The heat transfer coefficients are used to calculate the overall heat transfer coefficient and the UA product for the heat exchanger. The effectiveness of the heat exchanger is determined using the effectiveness - NTU equation for heat exchangers with one stream changing phase, as shown in Equation 4.7.

5.3.2 R22 and Ammonia Performance Comparison

The models of the ammonia - secondary fluid system and the R22 system contain the same penalties for the refrigerated case heat exchangers and the piping of the fluids. The models are included in Appendix C. The performance of the R22 system and the ammonia with Dowfrost system were compared at a refrigerated case temperature of 267 °K. The results of the comparison are shown in Table 8.1. The system COP for the R22 system is around 4% higher than the system COP for the ammonia with Dowfrost system. This is a small difference between the two system and improvements in the heat

exchangers used in the ammonia - secondary fluid system could easily overcome the difference. The similar performance of the two systems implies that the ammonia - secondary fluid system is a viable replacement for the R22 system.

Refrigerant	COP
R22	2.84
Ammonia with Dowfrost	2.72

Table 5.5 Performance comparison

The analysis done in this study indicates that the ammonia - secondary fluid systems are sensitive to design variables. For each individual supermarket, the design will need to be carefully analyzed to establish the best possible combination of parameters. If the design is carried out with a combination of parameters that are not optimal, the system can have a very low performance. However, it also indicates that with the careful selection of these parameters, the ammonia - secondary fluid system can be expected to have performance that rivals the performance of an R22 refrigeration system.

5.4 Conclusions

This Chapter has examined the influence of the different model parameters on the modeled performance of the system. The system designer will not have control over all of these parameters because some will be limited by the availability of equipment from the manufacturers and others, such as pipe lengths, will be determined by constraints in the store layout. This is especially true of the heat exchangers used in the system. The computer model can still be used to evaluate the best system design using the manufacturer's equipment in the model. Design rules for the parameters that the designer

will have the most control over are presented in Chapter 7. With the use of these rules, a carefully designed ammonia - secondary fluid system can have a performance that is comparable with the performance of a R22 system.

**CHAPTER
SIX**

SECONDARY FLUID PERFORMANCE COMPARISON

The six different secondary fluids analyzed in the study were discussed in Chapter 2. The properties and the hazards of each fluid were enumerated. However, the performance of the ammonia - secondary fluid system for each of the different fluids was not discussed. To analyze the overall system performance, the correlation equations for the properties were entered into the base case model and the performance of the ammonia - secondary fluid refrigeration system was calculated for each secondary fluid.

6.1 Secondary Fluid Comparison

The secondary fluid correlation equations from Chapter 2 were entered into the base case computer model for the ammonia - secondary fluid system and the performance of the system was evaluated using EES[1]. The comparison of the performance was done for each fluid in four different manners: ignoring thermal losses and required pump work, with thermal losses and without pump work, with pump work and without thermal losses, and with both pump work and thermal losses. Using the base case parameters as listed in Section 5.2.1, the comparison was done at refrigerated case temperatures of 267 and 244 °K. The model equation sets are included in Appendix D.

6.1.1 Performance Comparison without thermal losses and pump work

This comparison did not include thermal losses of the piping system to the store environment or the pumping power needed to circulate the secondary fluid in order to isolate the effects of the two heat exchangers on the secondary fluid distribution system. The plots of the performance of the different secondary fluids are shown in Figures 6.1 and 6.2. At a refrigerated case temperature of 267 °K, propane yields the highest performance. Ethanol, Dowfrost, and Dowtherm SR-1 have about the same performance and Syltherm XLT and Multitherm 503 have the lowest performance. Since all other factors except for the heat exchangers are neglected here it is necessary to look at the performance of the two heat exchangers to determine the reason that propane has the highest performance. The Reynolds numbers of the secondary fluid flows through the ammonia - secondary fluid heat exchanger are around 100 for Dowfrost, 400 for Dowtherm SR-1, 45000 for propane, and 3000 for ethanol. This means that Dowfrost and Dowtherm SR-1 are laminar flow, ethanol is transitional flow, and propane is turbulent flow. The heat transfer in turbulent flow is much higher than the heat transfer

in laminar flow. This can be seen in the effectiveness of the ammonia - secondary fluid heat exchanger. For Dowfrost the effectiveness is 0.34, for Dowtherm SR-1 it is 0.35, for propane it is 0.68, and for ethanol it is 0.56. The high Reynolds number of propane can be attributed to the fact that propane has lower viscosities than the other secondary fluids. The comparison of the viscosities of the secondary fluids is shown in Figure 2.30. The turbulent flow of propane brings forth the concept that turbulators or treated surfaces on the heat exchanger could be used to create turbulent flow for the other secondary fluids.

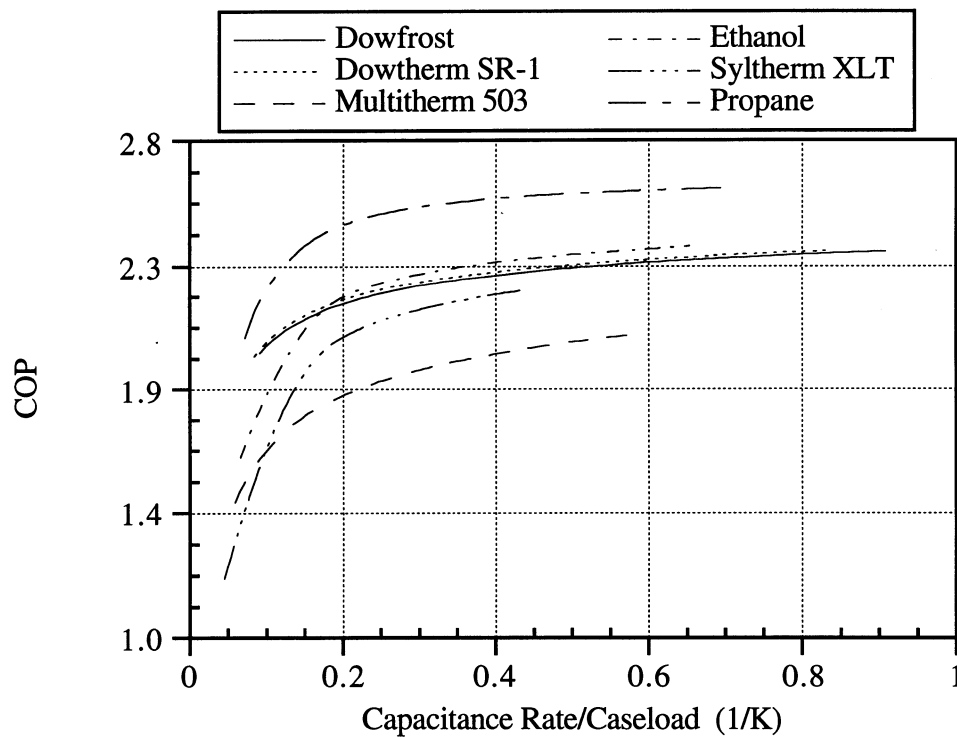


Figure 6.1 Performance comparison of the secondary fluids ignoring thermal losses and pumping power at a refrigerated case temperature of 267 °K

The comparison at a case temperature of 244 °K has performance curves that are very similar to the ones at a case temperature of 267 °K. Again propane has the highest

performance and Multitherm 503 has the lowest. Ethanol, Dowfrost, Dowtherm SR-1, and Syltherm XLT are grouped in between the other two fluids.

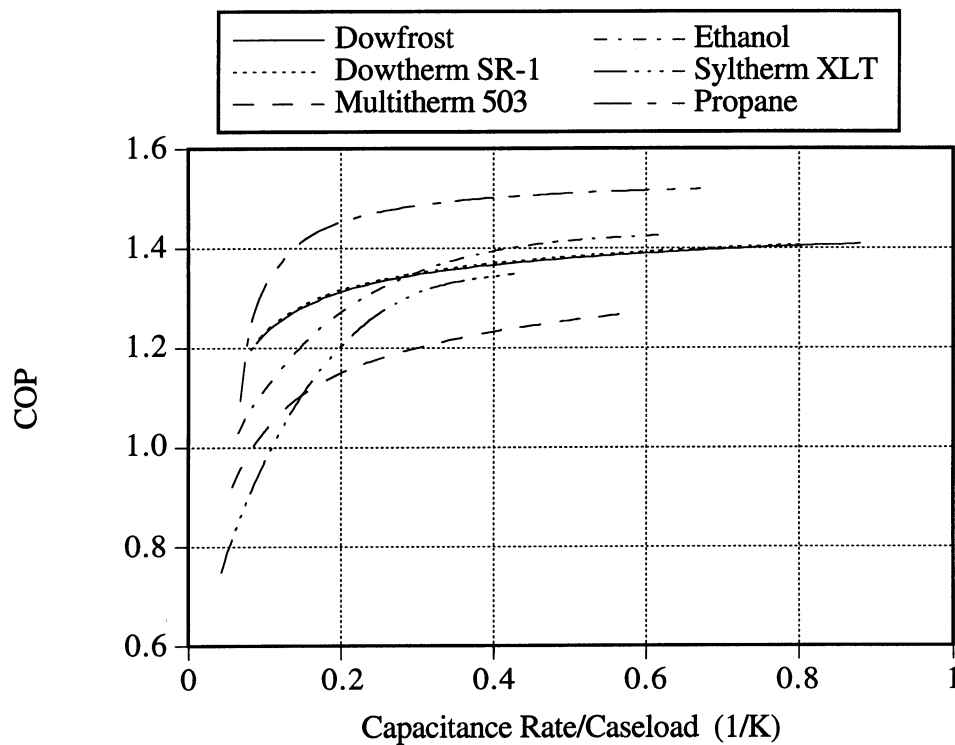


Figure 6.2 Performance comparison of the secondary fluids ignoring thermal losses and pumping power at a refrigerated case temperature of 244 °K

6.1.2 Performance Comparison without pump work

Figures 6.3 and 6.4 show the comparison of the performance of the ammonia with secondary fluid system when thermal losses are included and pumping power is neglected. At a refrigerated case temperature of 267 °K, the performance curves are similar to the curves with thermal losses ignored. Propane has the highest performance with Dowfrost, Dowtherm SR-1, ethanol, and Syltherm XLT grouped in the middle and Multitherm 503 yielding the lowest performance.

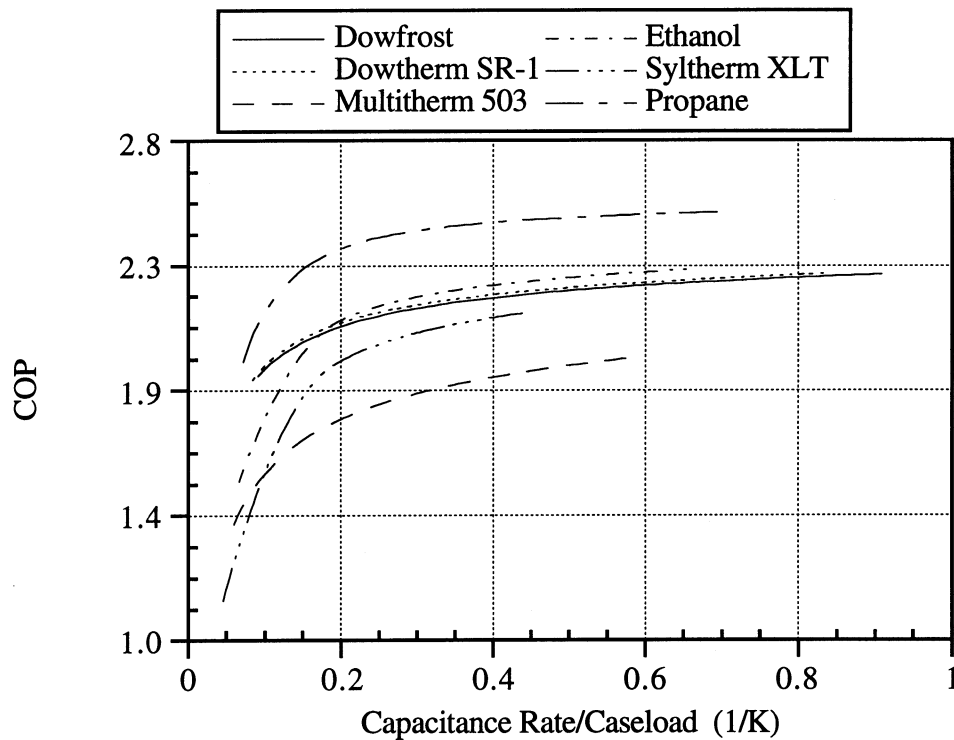


Figure 6.3 Performance comparison of the secondary fluids ignoring pumping power at a refrigerated case temperature of 267 °K

The comparison at a case temperature of 244 °K again displays performance curves similar to those of the higher case temperature. Propane continues to yield the highest performance. Looking at the curves ignoring both thermal losses and pumping power and the curves ignoring just pumping power shows that the system performance decreases slightly with the inclusion of thermal losses for all of the secondary fluids. The reduction in performance is essentially the same for all of the secondary fluids.

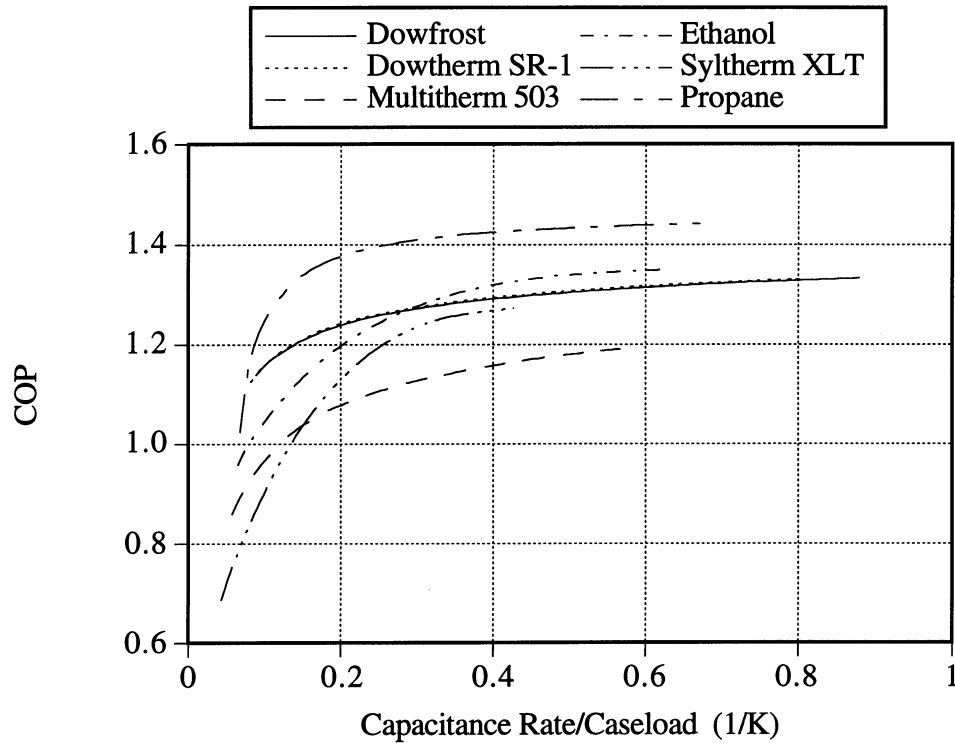


Figure 6.4 Performance comparison of the secondary fluids ignoring pumping power at a refrigerated case temperature of 244 °K

6.1.3 Performance Comparison without thermal losses

The performance curves have a much different shape when the pumping power is added and the thermal losses are neglected. The graphs of the performance are shown in Figures 6.5 and 6.6. The increased pumping power with increased capacitance rate - caseload ratios reduces the COP of the system. With increased capacitance rate - caseload ratios, the temperature difference of the secondary fluid across the refrigerated case heat exchanger needs to be larger than for the lower ratios. With the larger temperature difference, the temperature of the secondary fluid needs to be lower for the same case temperature. The fluids that have a steeper degradation in performance with increased capacitance rate - caseload ratios also have densities that increase rapidly with decreasing

temperatures. The comparison of the density of the secondary fluids is shown in Figure 2.27.

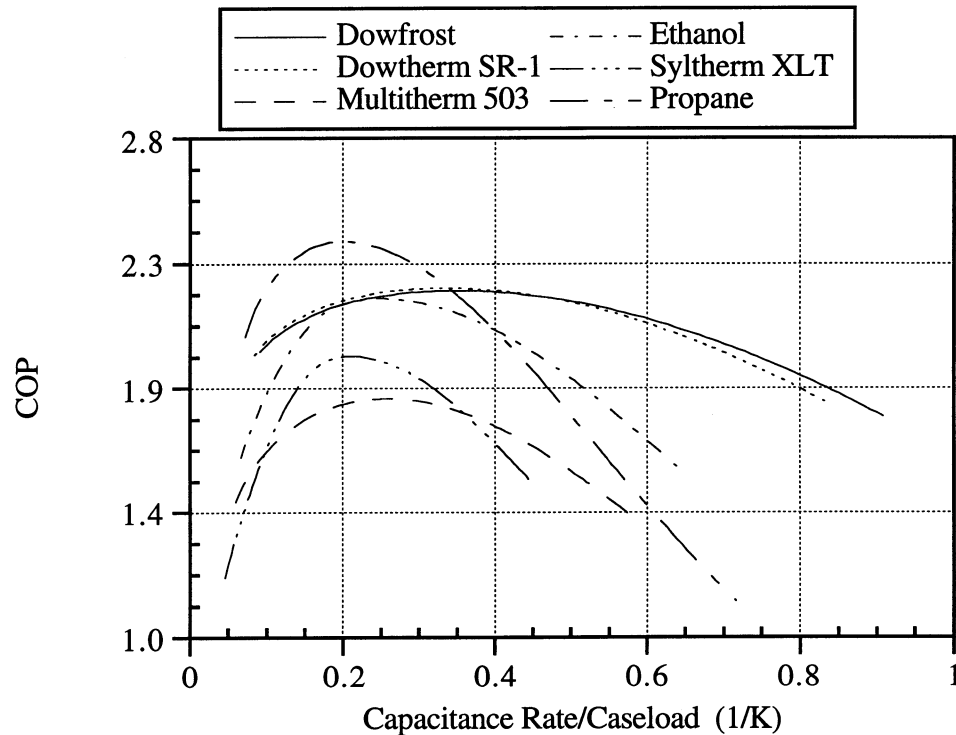


Figure 6.5 Performance comparison of the secondary fluids ignoring thermal losses at a refrigerated case temperature of 267 °K

Propane still has the highest performance, but the performance curve has a sharp peak around a capacitance rate - caseload ratio of 0.2 and the COP declines steeply at higher ratios. Dowfrost and Dowtherm SR-1 are the next highest performing fluids and the curves are much flatter than the other fluids. The flat curves indicate that if the system does not operate at the optimal capacitance rate - caseload ratio the performance will not be affected as strongly for Dowfrost and Dowtherm SR-1 as for the other secondary fluids. Ethanol has a peak performance that is about as high as Dowfrost and Dowtherm

SR-1 but declines much more quickly at higher capacitance rate - caseload ratios. Syltherm XLT and Multitherm 503 have the lowest performance.

The performance curves at case temperatures of 244 °K are similar to the curves at 267 °K. The performance of Dowtherm SR-1 is slightly above the performance of Dowfrost at the colder temperature, but propane still has the highest performance. The curves for both Dowtherm SR-1 and Dowfrost are again flatter than for the other secondary fluids. This behavior can be attributed to the fact that the density of Dowtherm SR-1 and Dowfrost is not as temperature dependent as it is for the other fluids.

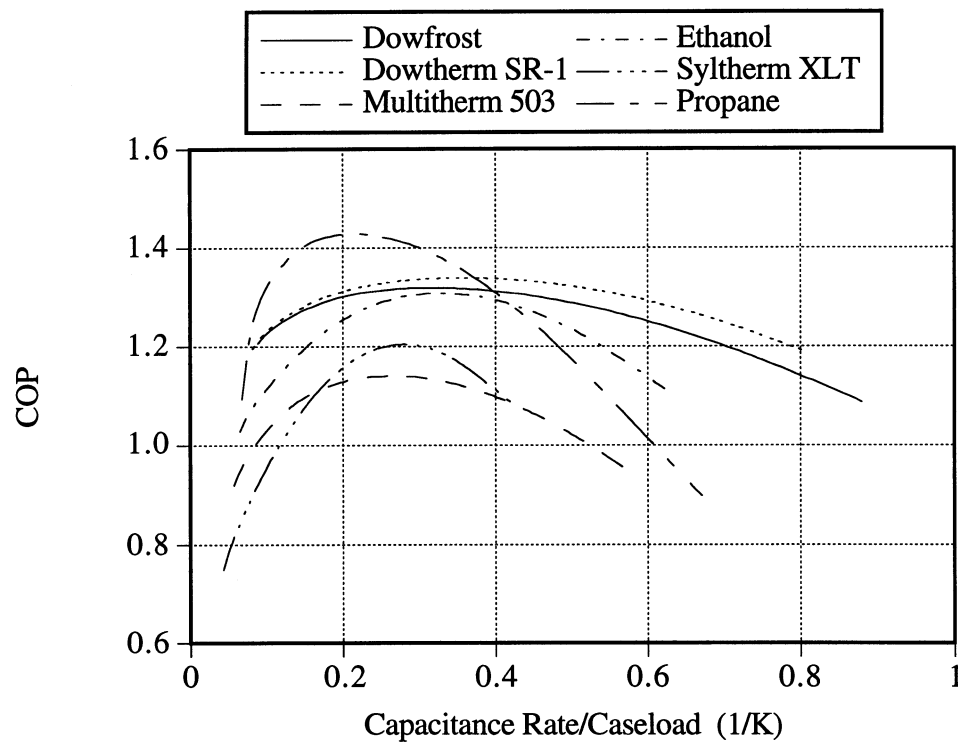


Figure 6.6 Performance comparison of the secondary fluids ignoring thermal losses at a refrigerated case temperature of 244 °K

6.1.4 Performance Comparison with pump work and thermal losses

Figures 6.7 and 6.8 show the performance curves of the secondary fluids with both pumping power and thermal losses included. Again adding the thermal losses onto the performance curves decreases the performance curves of all of the secondary fluids about the same amount. With the refrigerated case temperature of 267 °K, propane has the same high performance with a sharp peak. Dowfrost and Dowtherm SR-1 have the next highest performance and have flat curves that give high performance over a wide range of capacitance rate - caseload ratios. Ethanol has a performance which is almost as high as Dowfrost and Dowtherm SR-1, but the performance drops off at higher capacitance rate - caseload ratios. Syltherm XLT and Multitherm 503 have the lowest performance. The curves at a refrigerated case temperature of 244 °K are similar.

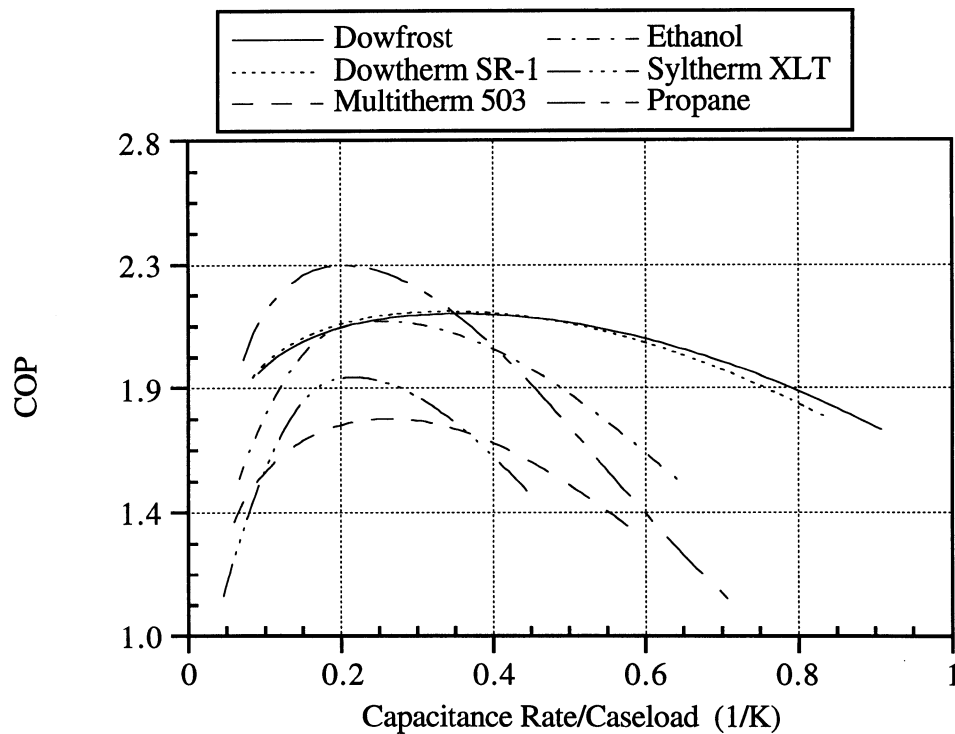


Figure 6.7 Performance comparison of the secondary fluids at a refrigerated case temperature of 267 °K

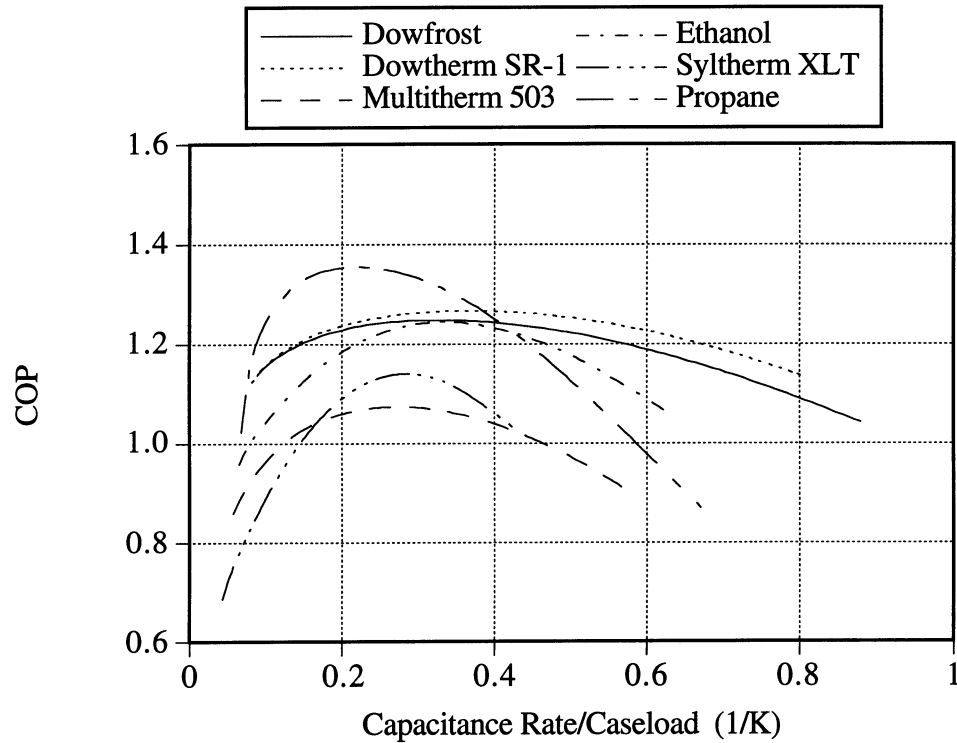


Figure 6.8 Performance comparison of the secondary fluids at a refrigerated case temperature of 244 °K

6.2 Secondary Fluid Selection

The performance curves show that the choice of propane as the secondary fluid will yield the highest performance by about 10%. However, the selection of a secondary fluid involves more than just looking at the peak performance. The performance of propane is high at a narrow band of capacitance rate - caseload ratios. A narrow band of capacitance rate - caseload ratios indicates that the system must be operated at these ratios at all times to get this high performance. The capacitance rate - caseload ratio is regulated by adjusting the flow rate of the secondary fluid to adjust the capacitance rate for changing case loads. This adjustment changes the desired temperature difference in the heat exchanger between the ammonia and the secondary fluid, which in turn changes the

required flow rate of the ammonia. This flow rate is adjusted by increasing or decreasing the number of cylinders of the compressor being used. Fine adjustments in the ammonia flow rate cannot be accomplished in this manner. Thus secondary fluids that have performance that does not change as much with changing capacitance rate - caseload ratios will yield better performance at off-design conditions. Both Dowfrost and Dowtherm SR-1 exhibit this type of behavior. Another concern is the toxicity and flammability of the secondary fluid. As discussed in Chapter 2, propane is both flammable and explosive. Dowfrost and Dowtherm SR-1 are both considered non-flammable. Dowfrost is non-toxic and Dowtherm SR-1 is orally toxic.

When the performance curves and the toxicity and flammability concerns are considered, different choices for the secondary fluid are possible. Propane gives the highest design, or peak, performance, Dowtherm SR-1 and Dowfrost yield the highest performance over a range of capacitance rate - caseload ratios, and Dowfrost yields the highest, safest performance. For an application as public as a supermarket, the safety concern is very high and Dowfrost would likely be the best choice of the six secondary fluids examined here.

CHAPTER **SEVEN**

DESIGN RULES

An accurate prediction of a supermarket ammonia - secondary fluid system can only be accomplished with a complete analysis of the entire system. Some of the choices in the design will be controlled by the availability of materials from the manufacturers. Design rules that involve the design parameters which the system designer will have the most control in selecting can be developed to simplify the choices. By using these design rules, the selection of some of the optimal parameters can be made without modeling the complete system.

7.1 Ammonia Cycle Design

The ammonia vapor compression cycle is the backbone of the ammonia - secondary fluid system. As discussed in Chapter 3, ammonia has a high discharge temperature from the compressor and it is necessary to utilize staged compression to reduce the temperature. Three different staging methods are described in Section 3.2 and a performance comparison is shown in Section 3.3. Table 3.2 presents the results of the performance comparison and shows that, with ammonia as the refrigerant, the staged compression with flash tank method will give the highest performance. A diagram of this system is shown in Figure 3.7.

7.2 Secondary Fluid Selection

Six different secondary fluids were compared in this study. Dowfrost, Dowtherm SR-1, Multitherm 503, Syltherm XLT, ethanol, and propane are described in detail in Chapter 2. The descriptions include a comparison of the properties of the fluids. The property comparison is inconclusive because one fluid does not have the best properties for high heat transfer and low pumping costs. In Chapter 6, a comparison of the performance of the ammonia - secondary fluid system with the six different fluids is documented. Figures 6.7 and 6.8 show the performance comparison at refrigerated case temperatures of 267 and 244 °K, respectively. Both of these graphs indicated that propane has a performance that is about 10% higher than both Dowfrost and Dowtherm SR-1. However, since a supermarket refrigeration system is an application where the general public may come in accidental contact with the secondary fluid, the flammability and toxicity of the fluids is a concern. Table 2.1 indicates that propane is both flammable and explosive and Dowtherm SR-1 is orally toxic. Dowfrost is neither flammable nor toxic and with the safety concerns of the supermarket system would likely be the best choice of

the six secondary fluids examined in this study. Dowfrost is the inhibited propylene glycol heat transfer fluid manufactured by the Dow Chemical Company.

7.3 Operating Parameters

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. The parametric study shown in Chapter 5, included an analysis of the influence of the exponent in the pressure ratio equation (Equation 5.1) on the overall system performance. Figure 5.5 shows the results of the analysis. The highest performance occurs when the exponent is between 0.5 and 0.6.

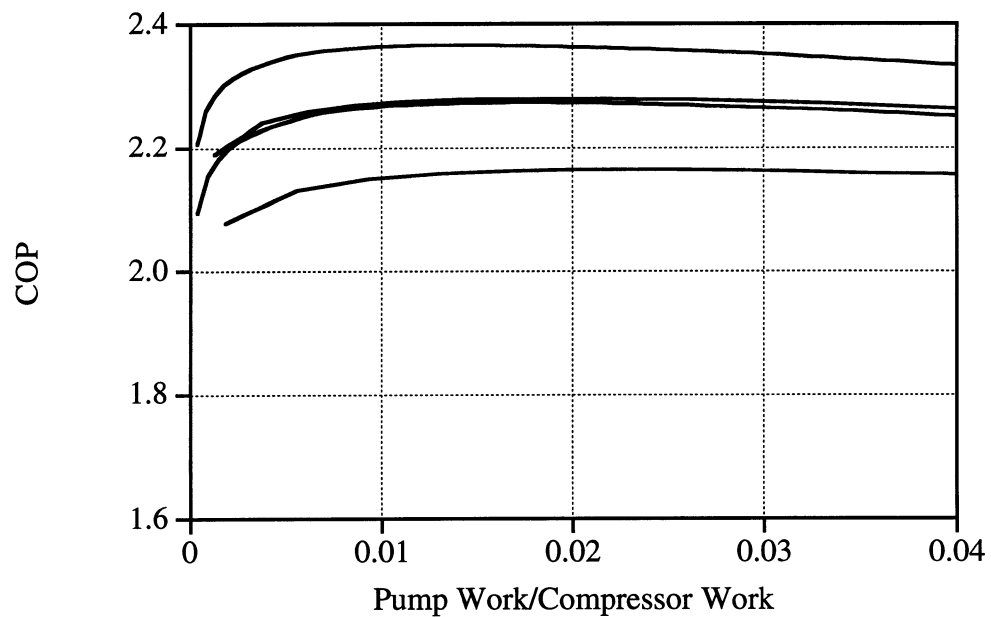


Figure 7.1 System performance as a function of compressor work - pump work ratio

The optimal capacitance rate - caseload ratio provides another design rule. The ratio influences the system performance by changing the ratio between the amount of compressor work and the pump work needed. The influence of the ratio of the compressor work and the pump work was calculated and the graph is shown in Figure 7.1. The graph indicates that the highest overall system performance occurs when the ratio of pump work to compressor work is around 0.02. When selecting the operating capacitance rate - caseload ratio, it is important that the compressor work - pump work ratio is in this neighborhood to ensure that the highest system performance possible is achieved.

7.4 Heat Exchangers

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. In a system with one heat exchanger, the design mass flow rate of a system is selected to be at the point where the highest heat exchanger effectiveness would be achieved. This would be true with a system involving two heat exchangers only if both heat exchangers have the highest effectiveness at the same flow rate.

In the supermarket system, the heat exchangers do not have the same behavior with changing flow rate. For the base case system described in Chapter 5, Figure 7.2 shows the changing effectiveness with changing flow rate. The flow rate is normalized by using the capacitance rate - caseload ratio. Figure 7.2 shows that as the capacitance rate - caseload ratio increases, the effectiveness of the refrigerated case heat exchanger increases and the effectiveness of the ammonia - secondary fluid heat exchanger decreases. The maximum COP for this parameter set occurs at a capacitance rate -

caseload ratio of about 0.35. This maximum does not occur at a point that is discernible from the plots of the effectivenesses. The maximum is not at the maximum effectiveness of either heat exchanger and is also not at the point where the effectivenesses added together are at a maximum.

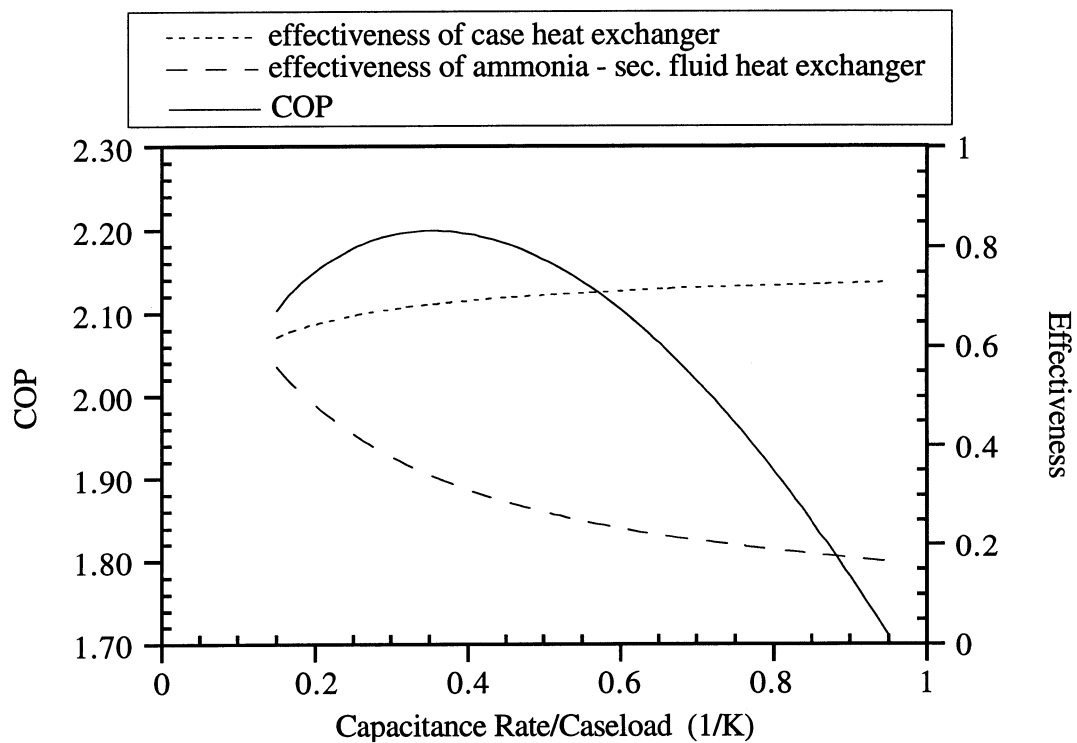


Figure 7.2 Heat exchanger effectivenesses and system COP

Another method of looking at the same heat exchanger sizing problem is to compare the overall loss coefficients (UA) of the two heat exchangers. An analysis comparing the system performance based on the ratio of the UA value of the refrigerated case heat exchanger to the UA value of the ammonia - secondary fluid heat exchanger was done for four different 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 are shown in Figure 7.2. The highest system performance occurs when the UA ratio is around 0.475. The heat exchangers should be sized in such a manner that the UA value for the ammonia - secondary fluid heat exchanger is about 0.45 to 0.5 times the UA value of the refrigerated case heat exchanger.

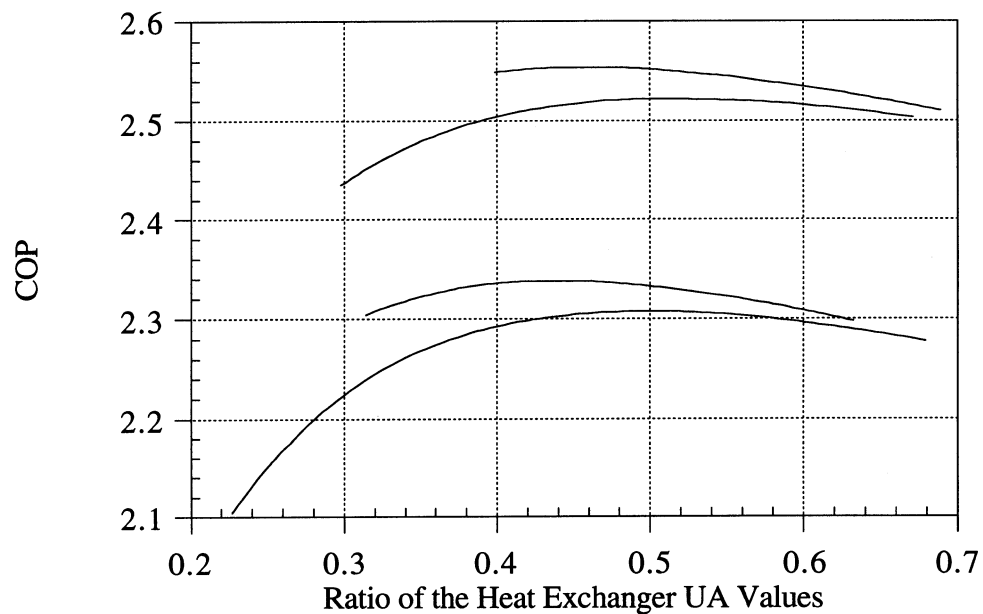


Figure 7.3 System performance as a function of UA value ratio

7.5 Secondary Fluid Piping System

The system designer will have the most control in the design of the secondary fluid supply and return pipes. The length of the pipes, the diameter of the pipes, and the amount of insulation on the pipes will all be selected by the system designer. Since the supply and return pipes will both carry liquid secondary fluid, it is reasonable to assume that the same length, diameter, and amount of insulation will be used for both pipes. The

combination of these three parameters would influence the system more than any one of them separately. A designer would probably know the length of the pipes necessary to reach from the ammonia - secondary fluid heat exchanger to the refrigerated case heat exchanger and would simply want to know which combination of pipe diameter and amount of insulation would yield the highest performance.

Maps of the performance of an ammonia - propylene glycol system at a refrigerated case temperature of 267 °K with changing pipe length and diameter and amount of insulation were developed. The ranges of the parameters were pipe length from 10 to 80 meters, pipe diameter from 0.05 and 0.30 meters, and insulation thickness from 0.01 to 0.03 meters. The other parameters were held constant at their base case values, and the capacitance rate - caseload ratio was set to 0.35, which is the optimal ratio for the base case. The maximum system COP in this range was calculated at a pipe length of 10 m, a pipe diameter of 0.10 m, and 0.03 m of insulation.

To develop the performance maps, the combinations of the pipe length, pipe diameter, and amount of insulation that yielded a system COP within 2.5%, 5.0%, 7.5%, 10%, and 20% of the maximum were determined and plotted. The performance maps are shown here in three parts: Figure 7.4 shows the map of diameters and lengths with 0.01 meters of insulation, Figure 7.5 shows the map with 0.02 meters of insulation, and Figure 7.6 shows the map with 0.03 meters of insulation.

The maps provide an easy way to estimate an optimal design. Assume for example, the designer needs to use 40 meters of pipe between the two heat exchangers. Using 0.01 meters of insulation, the system can attain a COP between 5 and 7.5% of the maximum COP by using pipe diameters between 0.05 and 0.12 meters. With 0.02 meters of insulation, performance between 2.5 and 5% of maximum can be attained with diameters between 0.06 and 0.14 meters and between 5 and 7.5% of maximum with

diameters between 0.05 and 0.06 or 0.14 and 0.21 meters. With 0.03 meters of insulation, performance between 2.5 and 5% of maximum can be attained with diameters between 0.06 and 0.18 meters and performance between 5 and 7.5% with diameters between 0.05 and 0.06 or 0.18 and 0.29 meters.

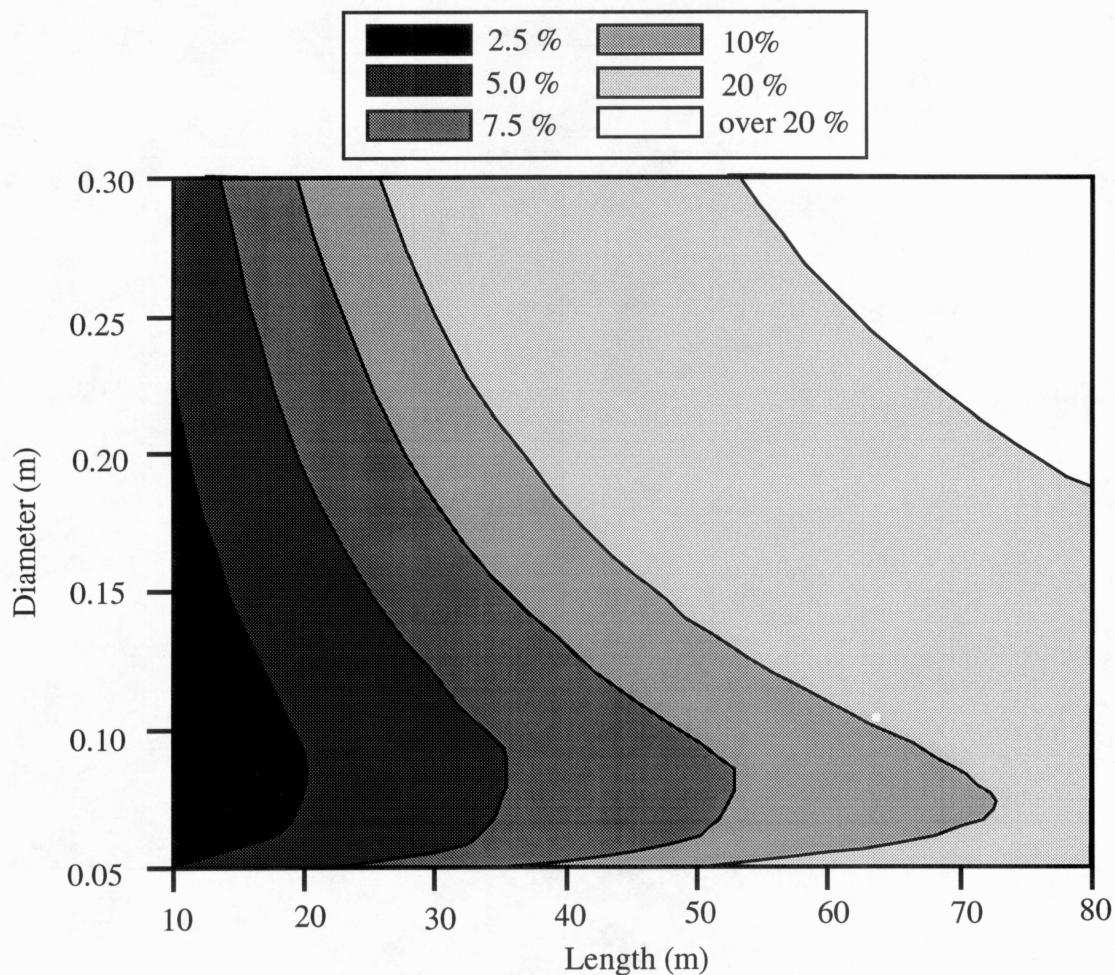


Figure 7.4 Performance map with 0.01 m of insulation

Another use of the maps is to determine the amount of insulation needed to attain a certain performance level with a specific pipe length and diameter. If a pipe length of 25 meters and a diameter of 0.17 meters is to be used, with 0.01 meters of insulation will

give performance between 5 and 7.5% of the maximum, with 0.02 meters of insulation between 2.5 and 5%, and with 0.03 meters of insulation within 2.5% of maximum.

By utilizing these performance maps, the system designer can determine the piping system that will give the highest performance. However, if an accurate determination of the actual performance of the system is desired or if the basic system being utilized is different from the one modeled to produce these performance maps, then the designer should model the entire system to ensure that the behavior is similar to the behavior of the system used here.

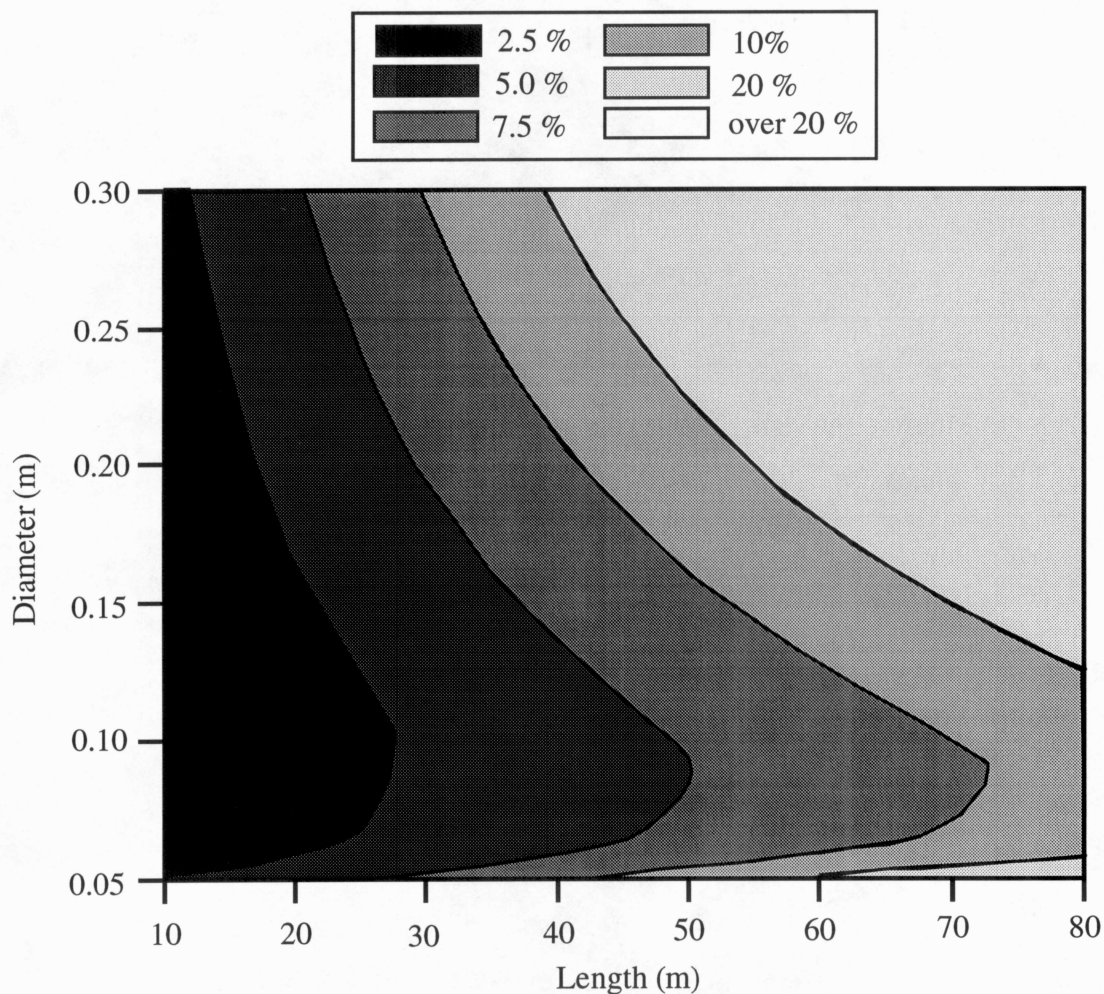


Figure 7.5 Performance map with 0.02 m of insulation

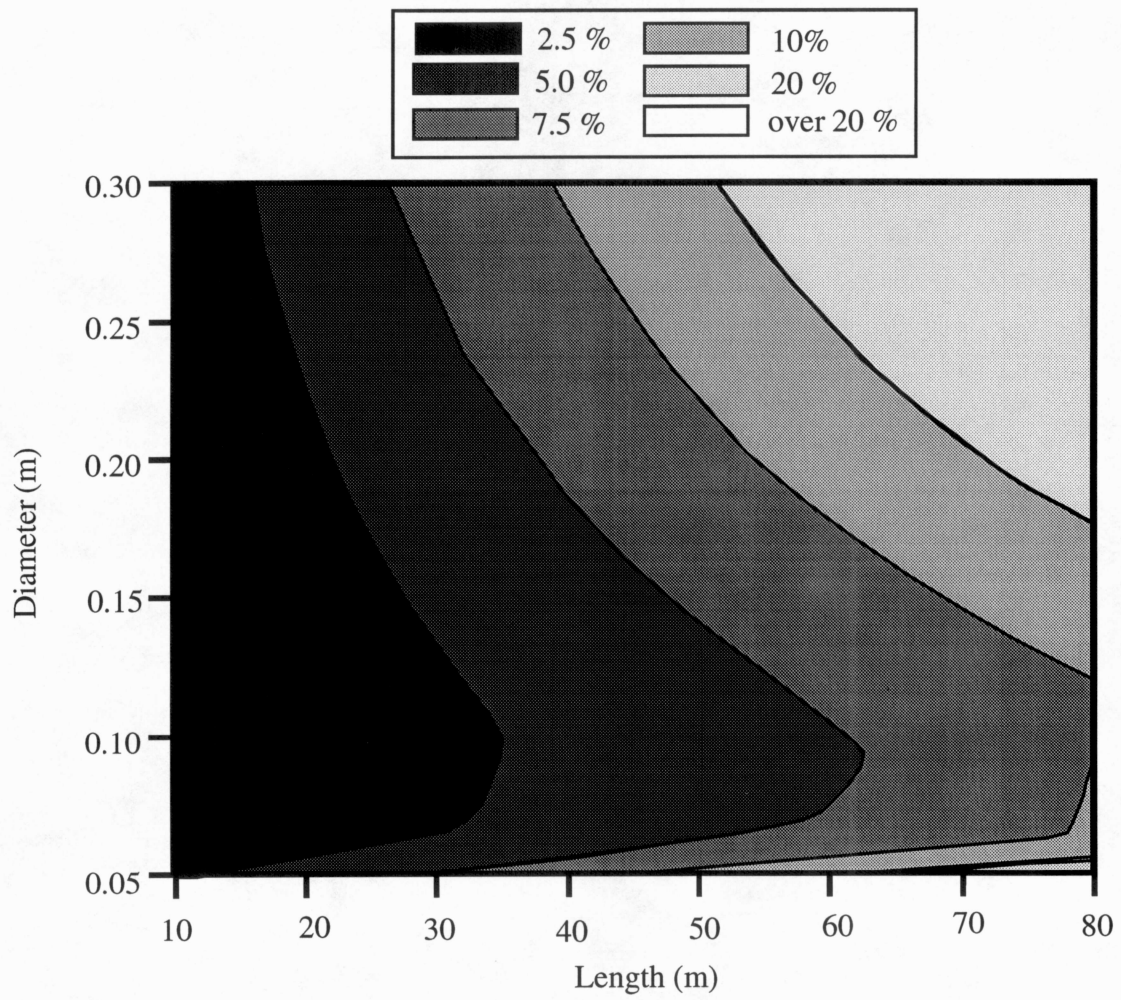


Figure 7.6 Performance map with 0.03 m of insulation

CHAPTER **EIGHT**

CONCLUSIONS AND RECOMMENDATIONS

The implementation of ammonia - secondary fluid supermarket refrigeration system will require careful design considerations. It is important to ensure that the installed system will be designed well enough to provide optimal performance. This study has looked at the basic refrigeration systems. Further system alterations might make the use of ammonia - secondary fluid systems more attractive.

8.1 Design Guidelines

In Chapter 7 the guidelines for the design of an ammonia - secondary fluid supermarket refrigeration system were discussed in depth. It is important to use these guidelines to ensure that the highest performance possible is attained. These guidelines include the ratio of refrigerated case heat exchanger UA value to ammonia - secondary fluid heat exchanger UA value of 0.475, the correct intermediate pressure selection, the use of Dowfrost or a similar propylene glycol solution as the secondary fluid, and a capacitance rate - caseload ratio that yields a pump work to compressor work ratio of around 0.02. It is also important to utilize the performance maps in Chapter 7 to determine the secondary fluid piping system that gives the highest performance.

If the design guidelines outlined in Chapter 7 are utilized and a system is constructed which produces the highest performance possible, the ammonia - secondary fluid system should have a performance which is comparable to the R22 refrigeration systems that would be replaced.

8.2 Future Research

There are some additional modifications that could be made to the ammonia - secondary fluid system that might increase the overall performance of the system. One such modification is to produce two temperatures of secondary fluid with a single ammonia system. This is accomplished by adding a second ammonia - secondary fluid heat exchanger between the flash tank and the second compressor in the ammonia system. The temperature requirements of the second heat exchanger would determine the pressure ratio that could be used in the staged compression. If the pressure ratio that is needed to produce the correct temperature for the additional secondary fluid is different enough from the optimal pressure ratio, the performance of the two secondary fluid

system could be lower than the performance of two individual ammonia - secondary fluid systems.

The addition of mechanical sub-cooling would increase the performance of the system. In mechanical sub-cooling a second vapor compression cycle is added that utilizes a heat exchanger to further cool the refrigerant as it leaves the evaporator. Mechanical sub-cooling has been proven to increase performance and is used in many applications. However, it can be added to a R22 refrigeration system, as well as the ammonia - secondary fluid system. The performance of both system will be higher with the mechanical sub-cooling, however, without quantifying the increase, it is impossible to know if the improvement will be greater for one of the systems.

The replacement of the air-cooled condenser with an evaporative or water-cooled condenser should increase the performance of the ammonia cycle. It may also be beneficial to utilize some of the secondary fluid in a heat exchanger that would condense the ammonia. The reduction in condenser temperature will increase the ammonia system performance, but the increased amount of secondary fluid needed may reduce the performance enough to negate the increase.

Storage of the secondary fluid is another possibility to increase to performance of the ammonia - secondary fluid system. The ammonia cycle would be run at night and all of the secondary fluid stored to provide the needed cooling of the refrigerated cases during the day. The performance of the system will increase slightly because the condenser will be more efficient in the colder temperatures and will decrease with increased storage losses. However, the advantage of storage is that the electrical load is shifted to the night time hours where electrical demand charges are less, lowering the cost of operating the system. The storage could be accomplished by either storing all of the secondary fluid at one temperature to provide the cooling for all of the refrigerated cases or by storing

different temperature secondary fluids for each of the temperature regimes needed in the supermarket.

In a supermarket, the refrigeration system and the air conditioning systems are closely linked. The use of the ammonia - secondary fluid system to provide the air conditioning for the supermarket in addition to providing cooling for the refrigeration cases needs to be analyzed. Changing the store conditions changes the refrigeration loads and the refrigeration loads influence the needed air conditioning for the store. Simulations that link the performance of the two systems could be used to determine if certain changes in either system improve the combined performance of the two systems

APPENDIX A

COMPRESSION METHOD MODELS

This Appendix contains the EES models used to compare the different staged compression methods.

NO STAGING METHOD

R22 MODEL:

{ Model of single stage R22 refrigeration system }

{ parameters }

Tevap = 244 {K} {evaporator temperature}
 degsupheat = 4 {K} {number of degrees of superheat}
 caseload = 15*3.51685 {kW} {refrigeration load met by case}
 UAcond = 8.0 {kW/K} {UA value of the condenser}
 Tamb = 300 {K} {ambient temperature}

{ compressor parameters }

bore = 2.38*2.54e-2 {m}
 stroke = 1.75*2.54e-2 {m}
 speed = 1700/60 {rps}
 clearratio = 0.05 {clearance ratio}
 polyexp = 1.15 {polytropic exponent}
 polyeff = 0.65 {polytropic efficiency}

mref = caseload/(h2-h1) {kg/s} {determines mass flow rate of R22 from refrigeration load and the enthalpy change in evaporator}

{ evaporator }

T1 = Tevap {K} {temperature entering evaporator}
 T2 = T1 + degsupheat {K} {temperature leaving the evaporator}
 P2 = Pressure(R22,T=Tevap,x=1.0) {kPa} {pressure leaving evaporator}
 h2 = Enthalpy(R22,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}
 s2 = Entropy(R22,T=T2,P=P2) {kJ/kg-K} {entropy leaving evaporator}
 P2 = P1 {kPa} {constant pressure across evaporator}

{ compressor }

v2 = Volume(R22,P=P2,h=h2) {m³/kg} {specific volume entering compressor}
 P2*(v2)^{polyexp} = P3*(v3)^{polyexp} {polytropic compression equation}
 h3 = Enthalpy(R22,P=P3,v=v3) {kJ/kg} {enthalpy leaving compressor}
 T3 = Temperature(R22,P=P3,v=v3) {K} {temperature leaving compressor}
 Vdisp = numcyl*(pi*((bore/2)²)*stroke) {m³} {displacement volume of compressor}

```

voleff = 1-(clearratio/(1-clearratio))*((P3/P2)^(1/polyexp)-1)
                                     {volumetric efficiency of compressor}
mref = (1/v2)*Vdisp*speed*voleff {kg/s} {mass flow rate of R22 through compressor}
Wcomp = (mref*(h3-h2))/polyeff {kW} {compressor work}
P3 = P4 {kPa} {constant pressure through condenser}

{condensor}
mair = 15*0.47879 {kg/s} {mass flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates the number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mref*(h3-h4) {calculates enthalpy change of R22 in condenser}
Qcond = mair*Cpair*(Tairout-Tamb) {calculates temperature change of air}

P4 = Pressure(R22,T=T4,x=0.0) {kPa} {pressure leaving condenser}
h4 = Enthalpy(R22,T=T4,x=0.0) {kJ/kg} {enthalpy leaving condenser}

{expansion}
h1 = h4 {kJ/kg} {isenthalpic expansion}

COP = caseload/Wcomp {coefficient of performance of refrigeration system}

```

AMMONIA MODEL:

```

{Model of single stage of compression with ammonia as the refrigerant}

{parameters}
Tevap = 244 {K} {evaporator temperature}
degssupheat = 4 {K} {number of degrees of superheat}
caseload = 15*3.51685 {kilowatts} {refrigeration load met by case}
UAcond = 8.0 {kW/K} {UA value of the condenser}
Tamb = 300 {K} {ambient temperature}

{compressor parameters}
bore = 2.38*2.54e-2 {m}
stroke = 1.75*2.54e-2 {m}
speed = 1700/60 {rpm}
clearratio = 0.05 {clearance ratio of the compressor}
polyexp = 1.3 {polytropic efficiency}
polyeff = 0.65 {polytropic effectiveness}

mref = caseload/(h2-h1) {kg/s} {determines mass flow rate of ammonia from
                                     refrigeration load and the enthalpy change in condenser}

```

```

{evaporator}
T1 = Tevap {K} {temperature entering the evaporator}
T2 = T1 + degsupheat {K} {temperature leaving the evaporator}
P2 = Pressure(AmmoniaTim,T=Tevap,x=1.0) {kPa} {pressure leaving evaporator}
h2 = Enthalpy(AmmoniaTim,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}
s2 = Entropy(AmmoniaTim,T=T2,P=P2) {kJ/kg-K} {entropy leaving evaporator}
P2 = P1 {kPa} {constant pressure across the evaporator}

{compressor}
v2 = Volume(AmmoniaTim,P=P2,h=h2) {m^3/kg}
                                     {specific volume entering compressor}
P2*(v2)^polyexp = P3*(v3)^polyexp {polytropic compression equation}
h3 = Enthalpy(AmmoniaTim,P=P3,v=v3) {kJ/kg} {enthalpy leaving compressor}
T3 = Temperature(AmmoniaTim,P=P3,v=v3) {K} {temperature leaving compressor}
Vdisp = numcyl*(pi*((bore/2)^2)*stroke) {m^3} {displacement volume of compressor}
voleff = 1-(clearratio/(1-clearratio))*((P3/P2)^(1/polyexp)-1)
                                     {volumetric efficiency of compressor}
mref = (1/v2)*Vdisp*speed*voleff {ammonia mass flow rate through compressor}
Wcomp = (mref*(h3-h2))/polyeff {kW} {compressor work}
P3 = P4 {kPa} {constant pressure through evaporator}

{condensor}
mair = 15*0.47879 {kg/s} {mass flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg-K} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates the number of transfer units of the condenser}
eff = 1-exp(-NTU) {effectiveness of the condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mref*(h3-h4) {calculates enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairout-Tamb) {calculates the temperature change of the air}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa} {pressure leaving the condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg} {enthalpy leaving the condenser}

{expansion}
h1 = h4 {kJ/kg} {isenthalpic expansion}

COP = caseload/Wcomp {coefficient of performance of refrigeration system}

```

STAGED COMPRESSION METHOD

R22 MODEL:

{ Model of R22 system with staged compression }

{ parameters }

Tevap = 267 {K} {evaporator temperature}

degsupheat = 4 {K} {number of degrees of superheat}

caseload = 15*3.51685 {kW} {refrigeration load met by case}

UAcond = 8.0 {kW/K} {UA value of condenser}

Tamb = 300 {K} {ambient temperature}

{ compressor parameters }

{ first compressor }

borelow = 2.38*2.54e-2 {m}

strokelow = 1.75*2.54e-2 {m}

speedlow = 1700/60 {rps}

clearratiolow = 0.05 {clearance ratio}

{ second compressor }

borehigh = 2.38*2.54e-2 {m}

strokehigh = 1.75*2.54e-2 {m}

speedhigh = 1700/60 {rps}

clearratiohigh = 0.05 {clearance ratio}

polyexp = 1.15 {polytropic exponent}

polyeff = 0.65 {polytropic efficiency}

mref = caseload/(h2-h1) {kg/s} {determines mass flow rate of R22 from refrigeration load and the enthalpy change in evaporator}

{ evaporator }

T1 = Tevap {K} {temperature entering evaporator}

T2 = T1 + degsupheat {K} {temperature leaving evaporator}

P2 = Pressure(R22,T=Tevap,x=1.0) {kPa} {pressure leaving evaporator}

h2 = Enthalpy(R22,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}

s2 = Entropy(R22,T=T2,P=P2) {kJ/kg-K} {entropy leaving evaporator}

P2 = P1 {kPa} {constant pressure across evaporator}

{ compressor }

rp = P3/P2 {ratio of condenser pressure to evaporator pressure}

rps = P2a/P2 {ratio of intermediate pressure to evaporator pressure}

v2 = Volume(R22,P=P2,h=h2) {m^3/kg} {specific volume entering first compressor}

P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}

h2a = Enthalpy(R22,P=P2a,v=v2a) {kJ/kg} {enthalpy leaving first compressor}

T2a = Temperature(R22,P=P2a,v=v2a) {K} {temperature leaving first compressor}

Vdisplow = numcyl*low*(pi*((borelow/2)^2)*strokelow) {m^3}

{displacement volume of first compressor}

volefflow = 1-(clearratiolow/(1-clearratiolow))*((P2a/P2)^(1/polyexp)-1)

{volumetric efficiency of first compressor}

mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}

{R22 flow rate through first compressor}

Wcomplow = (mref*(h2a-h2))/polyeff {kW} {work of first compressor}

$P2b = P2a$ {kPa} {intermediate pressure}
 $T2b = \text{Temperature}(R22, P=P2b, x=1.0)$ {K} {temperature entering second compressor}
 $h2b = \text{Enthalpy}(R22, P=P2b, x=1.0)$ {kJ/kg} {enthalpy entering second compressor}
 $v2b = \text{Volume}(R22, P=P2b, x=1.0)$ {m³/kg} {specific volume entering second compressor}
 $P2b \cdot (v2b)^{\text{polyexp}} = P3 \cdot (v3)^{\text{polyexp}}$ {polytropic compression equation}
 $h3 = \text{Enthalpy}(R22, P=P3, v=v3)$ {kJ/kg} {enthalpy leaving second compressor}
 $T3 = \text{Temperature}(R22, P=P3, v=v3)$ {K} {temperature leaving second compressor}
 $V_{\text{disphigh}} = \text{numcylhigh} \cdot (\pi \cdot (\text{borehigh}/2)^2 \cdot \text{strokehigh})$ {m³} {displacement volume of second compressor}
 $v_{\text{oleffhigh}} = 1 - (\text{clearratiohigh}/(1 - \text{clearratiohigh})) \cdot ((P3/P2b)^{(1/\text{polyexp})} - 1)$ {volumetric efficiency of second compressor}
 $\dot{m}_{\text{tot}} = (1/v2b) \cdot V_{\text{disphigh}} \cdot \text{speedhigh} \cdot v_{\text{oleffhigh}}$ {kg/s} {R22 flow rate through second compressor}
 $W_{\text{comphigh}} = (\dot{m}_{\text{tot}} \cdot (h3 - h2b)) / \text{polyeff}$ {kW} {work of second compressor}

 $P3 = P4$ {kPa} {constant pressure through condenser}
 $W_{\text{comp}} = W_{\text{complow}} + W_{\text{comphigh}}$ {kW} {total compressor work}

{condensor}
 $\dot{m}_{\text{air}} = 15 \cdot 0.47879$ {kg/s} {mass flow rate of air through condenser}
 $C_{\text{pair}} = \text{SpecHeat}(\text{Air}, T=T_{\text{amb}})$ {kJ/kg} {specific heat of air}
 $U A_{\text{cond}} = \dot{m}_{\text{air}} \cdot C_{\text{pair}} \cdot \text{NTU}$ {calculates the number of transfer units of condenser}
 $\text{eff} = 1 - \exp(-\text{NTU})$ {effectiveness of condenser}

 $Q_{\text{cond}} = \text{eff} \cdot \dot{m}_{\text{air}} \cdot C_{\text{pair}} \cdot (T4 - T_{\text{amb}})$ {kW} {heat transfer in condenser}
 $Q_{\text{cond}} = \dot{m}_{\text{tot}} \cdot (h3 - h4)$ {calculates the enthalpy change of R22 in condenser}
 $Q_{\text{cond}} = \dot{m}_{\text{air}} \cdot C_{\text{pair}} \cdot (T_{\text{airout}} - T_{\text{amb}})$ {calculates temperature change in air}

 $P4 = \text{Pressure}(R22, T=T4, x=0.0)$ {kPa} {pressure leaving condenser}
 $h4 = \text{Enthalpy}(R22, T=T4, x=0.0)$ {kJ/kg} {enthalpy leaving condenser}

{expander}
 $h1 = h4$ {kJ/kg} {isenthalpic expansion}

 $\text{COP} = \text{caseload} / W_{\text{comp}}$ {coefficient of performance of refrigeration system}

{mixing equations}
 $y = (h2a - h2b) / (h2b - h4)$
 $1 + y = (h2a - h4) / (h2b - h4)$
 $\dot{m}_{\text{tot}} = (1 + y) \cdot \dot{m}_{\text{ref}}$

AMMONIA MODEL:

{Model of ammonia system with staged compression}


```

{parameters}
Tevap = 267 {K} {evaporator temperature}
degsupheat = 4 {K} {number of degrees of superheat}
caseload = 15*3.51685 {kW} {refrigeration load met by case}
UAcond = 8.0 {kW/K} {UA value of the condenser}
Tamb = 300 {K} {ambient temperature}

{compressor parameters}
{first stage compressor}
borelow = 2.38*2.54e-2 {m}
strokelow = 1.75*2.54e-2 {m}
speedlow = 1700/60 {rpm}
clearratiolow = 0.05 {clearance ratio}
{second stage compressor}
borehigh = 2.38*2.54e-2 {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rpm}
clearratiohigh = 0.05 {clearance ratio}
polyexp = 1.3 {polytropic exponent}
polyeff = 0.65 {polytropic efficiency}

mref = caseload/(h2-h1) {kg/s} {determines the flow rate of ammonia from
                                refrigeration load and the enthalpy change in condenser}

{evaporator}
T1 = Tevap {K} {temperature entering the evaporator}
T2 = T1 + degsupheat {K} {temperature leaving the evaporator}
P2 = Pressure(AmmoniaTim,T=Tevap,x=1.0) {kPa} {pressure leaving evaporator}
h2 = Enthalpy(AmmoniaTim,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}
s2 = Entropy(AmmoniaTim,T=T2,P=P2) {kJ/kg-K} {entropy leaving evaporator}
P2 = P1 {kPa} {constant pressure across the evaporator}

{compressor}
rp = P3/P2 {ratio of condenser pressure to evaporator pressure}
rps = P2a/P2 {ratio of intermediate pressure to evaporator pressure}

v2 = Volume(AmmoniaTim,P=P2,h=h2) {m^3/kg}
                                {specific volume entering the first compressor}
P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}
h2a = Enthalpy(AmmoniaTim,P=P2a,v=v2a) {kJ/kg}
                                {enthalpy leaving first compressor}
T2a = Temperature(AmmoniaTim,P=P2a,v=v2a) {K}
                                {temperature leaving first compressor}
Vdisplow = numcyllow*(pi*((borelow/2)^2)*strokelow) {m^3}
                                {displacement volume of first compressor}
volefflow = 1-(clearratiolow/(1-clearratiolow))*((P2a/P2)^(1/polyexp)-1)
                                {volumetric efficiency of first compressor}
mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}

```

$W_{comlow} = (m_{ref} \cdot (h_{2a} - h_2)) / \text{polyeff}$ {kW} {work of first compressor}

$P_{2b} = P_{2a}$ {intermediate pressure}

$T_{2b} = \text{Temperature}(\text{AmmoniaTim}, P=P_{2b}, x=1.0)$ {K}

$h_{2b} = \text{Enthalpy}(\text{AmmoniaTim}, P=P_{2b}, x=1.0)$ {kJ/kg}

$v_{2b} = \text{Volume}(\text{AmmoniaTim}, P=P_{2b}, x=1.0)$ {m³/kg}

$P_{2b} \cdot (v_{2b})^{\text{polyexp}} = P_3 \cdot (v_3)^{\text{polyexp}}$ {polytropic compression equation}

$h_3 = \text{Enthalpy}(\text{AmmoniaTim}, P=P_3, v=v_3)$ {kJ/kg}

$T_3 = \text{Temperature}(\text{AmmoniaTim}, P=P_3, v=v_3)$ {K}

$V_{disphigh} = \text{numcylhigh} \cdot (\pi \cdot ((\text{borehigh}/2)^2) \cdot \text{strokehigh})$ {m³}

$v_{oleffhigh} = 1 - (\text{clearratiohigh} / (1 - \text{clearratiohigh})) \cdot ((P_3/P_{2b})^{(1/\text{polyexp})} - 1)$

$mtot = (1/v_{2b}) \cdot V_{disphigh} \cdot \text{speedhigh} \cdot v_{oleffhigh}$ {kg/s}

$W_{comphigh} = (mtot \cdot (h_3 - h_{2b})) / \text{polyeff}$ {kW} {work of second compressor}

$P_3 = P_4$ {kPa} {constant pressure through evaporator}

$W_{comp} = W_{comlow} + W_{comphigh}$ {kW} {total compressor work}

{condensor}

$m_{air} = 15 \cdot 0.47879$ {kg/s} {mass flow rate of air though the condenser}

$C_{pair} = \text{SpecHeat}(\text{Air}, T=T_{amb})$ {kJ/kg-K} {specific heat of air}

$UA_{cond} = m_{air} \cdot C_{pair} \cdot NTU$ {calculates the number of transfer units of condenser}

$eff = 1 - \exp(-NTU)$ {effectiveness of condenser}

$Q_{cond} = eff \cdot m_{air} \cdot C_{pair} \cdot (T_4 - T_{amb})$ {kW} {heat transfer in condenser}

$Q_{cond} = mtot \cdot (h_3 - h_4)$ {calculates enthalpy change of ammonia in condenser}

$Q_{cond} = m_{air} \cdot C_{pair} \cdot (T_{airout} - T_{amb})$ {calculates the temperature change of air}

$P_4 = \text{Pressure}(\text{AmmoniaTim}, T=T_4, x=0.0)$ {kPa} {pressure leaving the condenser}

$h_4 = \text{Enthalpy}(\text{AmmoniaTim}, T=T_4, x=0.0)$ {kJ/kg} {enthalpy leaving the condenser}

{expansion}

$h_1 = h_4$ {kJ/kg} {isenthalpic expansion}

$COP = \text{caseload} / W_{comp}$ {coefficient of performance of refrigeration system}

{mixing equations for intercooling}

$y = (h_{2a} - h_{2b}) / (h_{2b} - h_4)$

$1 + y = (h_{2a} - h_4) / (h_{2b} - h_4)$

$mtot = (1 + y) \cdot m_{ref}$

STAGED COMPRESSION AND EVAPORATION METHOD

R22 MODEL:

{ Model of R22 system with staged compression and evaporation }

{ parameters }

Tevap = 244 {K} {evaporator temperature}

degshupheat = 4 {K} {number of degrees of superheat}

caseload = 15*3.51685 {kW} {refrigeration load met by case}

UAcond = 8.0 {kW/K} {UA value of condenser}

Tamb = 300 {K} {ambient temperature}

{ compressor parameters }

{ first compressor }

borelow = 2.38*2.54e-2 {m}

strokelow = 1.75*2.54e-2 {m}

speedlow = 1700/60 {rps}

clearratio_low = 0.05 {clearance ratio}

{ second compressor }

borehigh = 2.38*2.54e-2 {m}

strokehigh = 1.75*2.54e-2 {m}

speedhigh = 1700/60 {rps}

clearratio_high = 0.05 {clearance ratio}

polyexp = 1.15 {polytropic exponent}

polyeff = 0.65 {polytropic efficiency}

mref = caseload/(h2-h1) {kg/s} {determines mass flow rate of R22 from refrigeration load and the enthalpy change in evaporator}

{ evaporator }

T1 = Tevap {K} {temperature entering evaporator}

T2 = T1 + degshupheat {K} {temperature leaving evaporator}

P2 = Pressure(R22,T=Tevap,x=1.0) {kPa} {pressure leaving evaporator}

h2 = Enthalpy(R22,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}

s2 = Entropy(R22,T=T2,P=P2) {entropy leaving evaporator}

P2 = P1 {kPa} {constant pressure across evaporator}

{ compressor }

rp = P3/P2 {ratio of condenser pressure to evaporator pressure}

rps = rp^x {relationship between pressure ratios}

rps = P2a/P2 {ratio of intermediate pressure to evaporator pressure}

v2 = Volume(R22,P=P2,h=h2) {m^3/kg} {specific volume entering first compressor}

$P2*(v2)^{\text{polyexp}} = P2a*(v2a)^{\text{polyexp}}$ {polytropic compression equation}
 $h2a = \text{Enthalpy}(R22, P=P2a, v=v2a)$ {kJ/kg} {enthalpy leaving first compressor}
 $T2a = \text{Temperature}(R22, P=P2a, v=v2a)$ {K} {temperature leaving first compressor}
 $Vdisplow = \text{numcyllo} * (\pi * ((borelow/2)^2) * \text{strokelow})$ {m³}
 {displacement volume of first compressor}
 $volefflow = 1 - (\text{clearratio}low / (1 - \text{clearratio}low)) * ((P2a/P2)^{(1/\text{polyexp}) - 1})$
 {volumetric efficiency of first compressor}
 $mref = (1/v2) * Vdisplow * \text{speedlow} * volefflow$ {kg/s}
 {R22 flow rate through first compressor}
 $Wcomplow = (mref * (h2a - h2)) / \text{polyeff}$ {kW} {work of first compressor}

$P2b = P2a$ {kPa} {intermediate pressure}
 $T2b = \text{Temperature}(R22, P=P2b, h=h2b)$ {K} {temperature entering second compressor}
 $v2b = \text{Volume}(R22, P=P2b, h=h2b)$ {m³/kg}
 {specific volume entering second compressor}
 $P2b*(v2b)^{\text{polyexp}} = P3*(v3)^{\text{polyexp}}$ {polytropic compression equation}
 $h3 = \text{Enthalpy}(R22, P=P3, v=v3)$ {kJ/kg} {enthalpy leaving second compressor}
 $T3 = \text{Temperature}(R22, P=P3, v=v3)$ {K} {temperature leaving second compressor}
 $Vdisphigh = \text{numcylhigh} * (\pi * ((borehigh/2)^2) * \text{strokehigh})$ {m³}
 {displacement volume of second compressor}
 $voleffhigh = 1 - (\text{clearratio}high / (1 - \text{clearratio}high)) * ((P3/P2b)^{(1/\text{polyexp}) - 1})$
 {volumetric efficiency of second compressor}
 $mtot = (1/v2b) * Vdisphigh * \text{speedhigh} * voleffhigh$ {kg/s}
 {R22 flow rate through second compressor}
 $Wcomphigh = (mtot * (h3 - h2b)) / \text{polyeff}$ {kW} {work of second compressor}

$P3 = P4$ {kPa} {constant pressure through condenser}
 $Wcomp = Wcomplow + Wcomphigh$ {kW} {total compressor work}

{condensor}
 $mair = 15 * 0.47879$ {kg/s} {mass flow rate of air through condenser}
 $Cpair = \text{SpecHeat}(\text{Air}, T=Tamb)$ {kJ/kg} {specific heat of air}
 $UAcond = mair * Cpair * NTU$ {calculates the number of transfer units of the condenser}
 $eff = 1 - \exp(-NTU)$ {effectiveness of condenser}

$Qcond = eff * mair * Cpair * (T4 - Tamb)$ {kW} {heat transfer in condenser}
 $Qcond = mtot * (h3 - h4)$ {calculates the enthalpy change of R22 in condenser}
 $Qcond = mair * Cpair * (Tairout - Tamb)$ {calculates temperature change of air}

$P4 = \text{Pressure}(R22, T=T4, x=0.0)$ {kPa} {pressure leaving condenser}
 $h4 = \text{Enthalpy}(R22, T=T4, x=0.0)$ {kJ/kg} {enthalpy leaving condenser}

{expansion}
 $P5 = P2a$ {kPa} {intermediate pressure}
 $P6 = P2a$ {kPa} {intermediate pressure}
 $h5 = \text{Enthalpy}(R22, P=P5, x=1.0)$ {kJ/kg} {enthalpy of ammonia vapor for intercooling}
 $h6 = \text{Enthalpy}(R22, P=P6, x=0.0)$ {kJ/kg} {enthalpy of liquid ammonia for evaporation}
 $h1 = h6$ {kJ/kg} {isenthalpic expansion}

$COP = \text{caseload}/W_{\text{comp}}$ {coefficient of performance of refrigeration system}

{mixing equations for separator and intercooling}

$m_{\text{tot}} = m_{\text{ref}} + m_{\text{vap}}$

$m_{\text{tot}}*h_4 = m_{\text{ref}}*h_6 + m_{\text{vap}}*h_5$

$m_{\text{tot}}*h_{2b} = m_{\text{ref}}*h_{2a} + m_{\text{vap}}*h_5$

AMMONIA MODEL:

{Model of ammonia system with staged compression and evaporation}

{parameters}

$T_{\text{evap}} = 267$ {K} {evaporator temperature}

$\text{deg_superheat} = 4$ {K} {number of degrees of superheat}

$\text{caseload} = 15*3.51685$ {kW} {refrigeration load met by case}

$UA_{\text{cond}} = 8.0$ {kW/K} {UA value of the condenser}

$T_{\text{amb}} = 300$ {K} {ambient temperature}

{compressor parameters}

{first compressor}

$b_{\text{orelow}} = 2.38*2.54e-2$ {m}

$s_{\text{trokelow}} = 1.75*2.54e-2$ {m}

$s_{\text{peedlow}} = 1700/60$ {rps}

$c_{\text{learratiolow}} = 0.05$ {clearance ratio}

{second compressor}

$b_{\text{orehigh}} = 2.38*2.54e-2$ {m}

$s_{\text{trokehigh}} = 1.75*2.54e-2$ {m}

$s_{\text{peedhigh}} = 1700/60$ {rps}

$c_{\text{learratiohigh}} = 0.05$ {clearance ratio}

$p_{\text{olyexp}} = 1.3$ {polytropic exponent}

$p_{\text{olyeff}} = 0.65$ {polytropic efficiency}

$m_{\text{ref}} = \text{caseload}/(h_2 - h_1)$ {kg/s} {determines mass flow rate of ammonia from refrigeration load and the enthalpy change in evaporator}

{evaporator}

$T_1 = T_{\text{evap}}$ {K} {temperature entering the evaporator}

$T_2 = T_1 + \text{deg_superheat}$ {K} {temperature leaving the evaporator}

$P_2 = \text{Pressure}(\text{AmmoniaTim}, T=T_{\text{evap}}, x=1.0)$ {kPa} {pressure leaving evaporator}

$h_2 = \text{Enthalpy}(\text{AmmoniaTim}, T=T_2, P=P_2)$ {kJ/kg} {enthalpy leaving evaporator}

$s_2 = \text{Entropy}(\text{AmmoniaTim}, T=T_2, P=P_2)$ {kJ/kg-K} {entropy leaving evaporator}

$P_2 = P_1$ {kPa} {constant pressure in evaporator}

{compressor}

$r_p = P_3/P_2$ {ratio of condenser pressure to evaporator pressure}

$rps = rp^x$ {determines relationship between pressure ratios}
 $rps = P2a/P2$ {ratio of intermediate pressure to evaporator pressure}

$v2 = \text{Volume}(\text{AmmoniaTim}, P=P2, h=h2)$ {m³/kg}
 {specific volume entering first compressor}
 $P2*(v2)^{\text{polyexp}} = P2a*(v2a)^{\text{polyexp}}$ {polytropic compression equation}
 $h2a = \text{Enthalpy}(\text{AmmoniaTim}, P=P2a, v=v2a)$ {kJ/kg}
 {enthalpy leaving first compressor}
 $T2a = \text{Temperature}(\text{AmmoniaTim}, P=P2a, v=v2a)$ {K}
 {temperature leaving first compressor}
 $Vdisplow = \text{numcyllo} * (\pi * ((\text{borelow}/2)^2) * \text{strokelow})$ {m³}
 {displacement volume of first compressor}
 $\text{volefflow} = 1 - (\text{clearratio} / (1 - \text{clearratio})) * ((P2a/P2)^{(1/\text{polyexp})} - 1)$
 {volumetric efficiency of first compressor}
 $mref = (1/v2) * Vdisplow * \text{speedlow} * \text{volefflow}$ {kg/s}
 {ammonia mass flow rate through first compressor}
 $Wcomplow = (mref * (h2a - h2)) / \text{polyeff}$ {kW} {work of first compressor}

$P2b = P2a$ {kPa} {intermediate pressure}
 $T2b = \text{Temperature}(\text{AmmoniaTim}, P=P2b, h=h2b)$ {K}
 {temperature entering second compressor}
 $v2b = \text{Volume}(\text{AmmoniaTim}, P=P2b, h=h2b)$ {m³/kg}
 {specific volume entering second compressor}
 $P2b*(v2b)^{\text{polyexp}} = P3*(v3)^{\text{polyexp}}$ {polytropic compression equation}
 $h3 = \text{Enthalpy}(\text{AmmoniaTim}, P=P3, v=v3)$ {kJ/kg}
 {enthalpy leaving second compressor}
 $T3 = \text{Temperature}(\text{AmmoniaTim}, P=P3, v=v3)$ {K}
 {temperature leaving the second compressor}
 $Vdisphigh = \text{numcylhigh} * (\pi * ((\text{borehigh}/2)^2) * \text{strokehigh})$ {m³}
 {displacement volume of second compressor}
 $\text{voleffhigh} = 1 - (\text{clearratiohigh} / (1 - \text{clearratiohigh})) * ((P3/P2b)^{(1/\text{polyexp})} - 1)$
 {volumetric efficiency of second compressor}
 $mtot = (1/v2b) * Vdisphigh * \text{speedhigh} * \text{voleffhigh}$ {kg/s}
 {ammonia mass flow rate through second compressor}
 $Wcomphigh = (mtot * (h3 - h2b)) / \text{polyeff}$ {kW} {work of second compressor}

$P3 = P4$ {kPa} {constant pressure through the condenser}
 $Wcomp = Wcomplow + Wcomphigh$ {kW} {total compressor work}

{condensor}
 $mair = 15 * 0.47879$ {kg/s} {mass flow rate of air through condenser}
 $Cpair = \text{SpecHeat}(\text{Air}, T=Tamb)$ {kJ/kg} {specific heat of air}
 $UAcond = mair * Cpair * NTU$ {calculates the number of transfer units of the condenser}
 $eff = 1 - \exp(-NTU)$ {effectiveness of the condenser}

$Qcond = eff * mair * Cpair * (T4 - Tamb)$ {kW} {heat transfer in condenser}
 $Qcond = mtot * (h3 - h4)$ {calculates the enthalpy change of ammonia in condenser}
 $Qcond = mair * Cpair * (Tairout - Tamb)$ {calculates temperature change of air}

```

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa} {pressure leaving condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg} {enthalpy leaving condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
P6 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=1.0) {kJ/kg}
                                     {enthalpy of ammonia gas for intercooling}
h6 = Enthalpy(AmmoniaTim,P=P6,x=0.0) {kJ/kg}
                                     {enthalpy of ammonia vapor for evaporator}
h1 = h6 {isenthalpic expansion}

COP = caseload/Wcomp {coefficient of performance of refrigeration system}

{mixing equations for separator and intercooling}
mtot = mref + mvap
mtot*h4 = mref*h6 + mvap*h5
mtot*h2b = mref*h2a + mvap*h5

```

STAGED COMPRESSION WITH FLASH TANK METHOD

R22 MODEL:

{ Model of R22 system with staged compression and flash tank }

```

{parameters}
Tevap = 267 {K} {evaporator temperature}
degssupheat = 4 {K} {number of degrees of superheat}
caseload = 15*3.51685 {kW} {refrigeration load met by case}
UAcond = 8.0 {kW/K} {UA value of condenser}
Tamb = 300 {K} {ambient temperature}

{compressor parameters}
{first compressor}
borelow = 2.38*2.54e-2 {m}
strokelow = 1.75*2.54e-2 {m}
speedlow = 1700/60 {rps}
clearratiolow = 0.05 {clearance ratio}
{second compressor}
borehigh = 2.38*2.54e-2 {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rps}
clearratiohigh = 0.05 {clearance ratio}
polyexp = 1.15 {polytropic exponent}

```

polyeff = 0.65 {polytropic efficiency}

mref = caseload/(h2-h1) {kg/s} {determines mass flow rate of R22 from refrigeration load and enthalpy change in evaporator}

{evaporator}

T1 = Tevap {K} {temperature entering evaporator}

T2 = T1 + degsupheat {K} {temperature leaving evaporator}

P2 = Pressure(R22,T=Tevap,x=1.0) {kPa} {pressure leaving evaporator}

h2 = Enthalpy(R22,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}

s2 = Entropy(R22,T=T2,P=P2) {kJ/kg-K} {entropy leaving evaporator}

P2 = P1 {kPa} {constant pressure across evaporator}

{compressor}

rp = P3/P2 {ratio of condenser pressure to evaporator pressure}

rps = rp^x {relationship of pressure ratios}

rps = P2a/P2 {ratio of intermediate pressure to evaporator pressure}

v2 = Volume(R22,P=P2,h=h2) {m^3/kg} {specific volume entering first compressor}

P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}

h2a = Enthalpy(R22,P=P2a,v=v2a) {kJ/kg} {enthalpy leaving first compressor}

T2a = Temperature(R22,P=P2a,v=v2a) {K} {temperature leaving first compressor}

Vdisplow = numcyllo*(pi*((borelow/2)^2)*stroke) {m^3}

{displacement volume of first compressor}

volefflow = 1-(clearratiolow/(1-clearratiolow))*((P2a/P2)^(1/polyexp)-1)

{volumetric efficiency of first compressor}

mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}

{R22 flow rate through first compressor}

Wcomplow = (mref*(h2a-h2))/polyeff {kW} {work of first compressor}

P2b = P2a {kPa} {intermediate pressure}

h2b = Enthalpy(R22,P=P2b,x=1.0) {kJ/kg} {enthalpy entering second compressor}

T2b = Temperature(R22,P=P2b,x=1.0) {K} {temperature entering second

compressor}

v2b = Volume(R22,P=P2b,x=1.0) {m^3/kg}

{specific volume entering second compressor}

P2b*(v2b)^polyexp = P3*(v3)^polyexp {polytropic compression equation}

h3 = Enthalpy(R22,P=P3,v=v3) {kJ/kg} {enthalpy leaving second compressor}

T3 = Temperature(R22,P=P3,v=v3) {K} {temperature leaving second compressor}

Vdisphigh = numcylhigh*(pi*((borehigh/2)^2)*stroke) {m^3}

{displacement volume of second compressor}

voleffhigh = 1-(clearratiohigh/(1-clearratiohigh))*((P3/P2b)^(1/polyexp)-1)

{volumetric efficiency of second compressor}

mcond = (1/v2b)*Vdisphigh*speedhigh*voleffhigh {kg/s}

{R22 flow rate through second compressor}

Wcomphigh = (mcond*(h3-h2b))/polyeff {kW} {work of second compressor}

P3 = P4 {kPa} {constant pressure through condenser}

Wcomp = Wcomplow + Wcomphigh {kW} {total compressor work}


```

{condensor}
mair = 15*0.47879 {kg/s} {mass flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates the number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates the enthalpy change of R22 in condenser}
Qcond = mair*Cpair*(Tairout-Tamb) {calculates the temperature change of air}

P4 = Pressure(R22,T=T4,x=0.0) {kPa} {pressure leaving condenser}
h4 = Enthalpy(R22,T=T4,x=0.0) {kJ/kg} {enthalpy leaving condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(R22,P=P5,x=0.0) {kJ/kg} {enthalpy of liquid R22 for evaporation}
h1 = h5 {kJ/kg} {isenthalpic expansion}

COP = caseload/Wcomp {coefficient of performance of refrigeration system}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

```

AMMONIA MODEL:

{ Model of ammonia system with staged compression and flash tank }

```

{parameters}
Tevap = 267 {K} {evaporator temperature}
degssupheat = 4 {K} {number of degress of superheat}
caseload = 15*3.51685 {kW} {refrigeration load met by case}
UAcond = 8.0 {kW/K} {UA value of condenser}
Tamb = 300 {K} {ambient temperature}

{compressor parameters}
{first compressor}
borelow = 2.38*2.54e-2 {m}
strokelow = 1.75*2.54e-2 {m}
speedlow = 1700/60 {rps}
clearratiolow = 0.05 {clearence ratio}
{second compressor}
borehigh = 2.38*2.54e-2 {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rps}
clearratiohigh = 0.05 {clearence ratio}

```

polyexp = 1.3 {polytropic exponent}
 polyeff = 0.65 {polytropic efficiency}

mref = caseload/(h2-h1) {kg/s} {determines mass flow rate of ammonia from
 refrigeration load and the enthalpy change in evaporator}

{evaporator}

T1 = Tevap {K} {temperature entering evaporator}
 T2 = T1 + degsupheat {K} {temperature leaving evaporator}
 P2 = Pressure(AmmoniaTim,T=Tevap,x=1.0) {kPa} {pressure leaving evaporator}
 h2 = Enthalpy(AmmoniaTim,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}
 s2 = Entropy(AmmoniaTim,T=T2,P=P2) {kJ/kg-K} {entropy leaving evaporator}
 P2 = P1 {kPa} {constant pressure across the evaporator}

{compressor}

rp = P3/P2 {ratio of condenser pressure to evaporator pressure}
 rps = rp^x {determine relationship of pressure ratios}
 rps = P2a/P2 {ratio of intermediate pressure to evaporator pressure}

v2 = Volume(AmmoniaTim,P=P2,h=h2) {m^3/kg}
 {specific volume entering first compressor}
 $P2 \cdot (v2)^{\text{polyexp}} = P2a \cdot (v2a)^{\text{polyexp}}$ {polytropic compression equation}
 h2a = Enthalpy(AmmoniaTim,P=P2a,v=v2a) {kJ/kg}
 {enthalpy leaving first compressor}
 T2a = Temperature(AmmoniaTim,P=P2a,v=v2a) {K}
 {temperature leaving first compressor}
 $V_{\text{displow}} = \text{numcyllo} \cdot (\pi \cdot (\text{borelow}/2)^2 \cdot \text{stroke low})$ {m^3}
 {displacement volume of first compressor}
 $v_{\text{olefflow}} = 1 - (\text{clearratiolow} / (1 - \text{clearratiolow})) \cdot ((P2a/P2)^{(1/\text{polyexp})} - 1)$
 {volumetric efficiency of first compressor}
 $m_{\text{ref}} = (1/v2) \cdot V_{\text{displow}} \cdot \text{speedlow} \cdot v_{\text{olefflow}}$ {kg/s}
 {ammonia mass flow rate through first compressor}
 $W_{\text{complow}} = (m_{\text{ref}} \cdot (h2a - h2)) / \text{polyeff}$ {kW} {work of first compressor}

P2b = P2a {kPa} {intermediate pressure}

h2b = Enthalpy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg}
 {enthalpy entering second compressor}

T2b = Temperature(AmmoniaTim,P=P2b,x=1.0) {K}
 {temperature entering second compressor}

v2b = Volume(AmmoniaTim,P=P2b,x=1.0) {m^3/kg}
 {specific heat entering second compressor}

$P2b \cdot (v2b)^{\text{polyexp}} = P3 \cdot (v3)^{\text{polyexp}}$ {polytropic compression equation}

h3 = Enthalpy(AmmoniaTim,P=P3,v=v3) {kJ/kg}
 {enthalpy leaving second compressor}

T3 = Temperature(AmmoniaTim,P=P3,v=v3) {K}
 {temperature leaving second compressor}

$V_{\text{disphigh}} = \text{numcylhigh} \cdot (\pi \cdot (\text{borehigh}/2)^2 \cdot \text{stroke high})$ {m^3}
 {displacement volume of second compressor}

$v_{\text{oleffhigh}} = 1 - (\text{clearratiohigh} / (1 - \text{clearratiohigh})) \cdot ((P3/P2b)^{(1/\text{polyexp})} - 1)$

```

                                {volumetric efficiency of second compressor}
mcond = (1/v2b)*Vdisphigh*speedhigh*voleffhigh {kg/s}
                                {ammonia mass flow rate through second compressor}
Wcomphigh = (mcond*(h3-h2b))/polyeff {kW} {work of second compressor}

P3 = P4 {kPa} {constant pressure through condenser}
Wcomp = Wcomplow + Wcomphigh {kW} {total compressor work}

{condensor}
mair = 15*0.47879 {kg/s} {mass flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates the number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates the enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairout-Tamb) {calculates temperature change of air}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa} {pressure leaving condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg} {enthalpy leaving condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=0.0) {kJ/kg}
                                {enthalpy of liquid ammonia for evaporator}
h1 = h5 {isenthalpic expansion}

COP = caseload/Wcomp {coefficient of performance of refrigeration system}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

```

APPENDIX B

BASE CASE MODEL

This Appendix contains the EES models used to perform the parametric study.

{ Ammonia with Dowfrost system model used to perform parametric study }

FUNCTION CPPG(percent,T)

{ calculates the specific heat of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = 3.8649883866 - 0.023691954902 * \text{percent} - 0.00011278222908 * \text{percent}^2$

$C = 0.001023655712 + 5.6633876714e-5 * \text{percent}$

$CPPG = B + C * T$ {kJ/kg-K}

END

FUNCTION DENSITYPG(percent,T)

{ calculates the density of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = 875.54696219 + 2.151387542 * \text{percent}$

$C = 1.1191046068 - 0.0007599907262 * \text{percent} - 4.9236799989e-5 * \text{percent}^2$

$D = -0.002377960199 - 9.1377252136e-6 * \text{percent} + 1.0872237562e-7 * \text{percent}^2$

$DENSITYPG = B + C * T + D * T^2$ {kg/m³}

END

FUNCTION THERMCONDPG(percent,T)

{ calculates the thermal conductivity of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = -0.78595253278 + 0.015561899561 * \text{percent} - 4.8933521576e-5 * \text{percent}^2$

$C = 0.0076866167254 - 0.0001155974176 * \text{percent} + 3.6603360830e-7 * \text{percent}^2$

$D = -9.9976810237e-6 + 1.4560615474e-7 * \text{percent} - 4.5879383578e-10 * \text{percent}^2$

$THERMCONDPG = B + C * T + D * T^2$ {W/m-K}

END

FUNCTION VISCOSITYPG(percent,T)

{ calculates the viscosity of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = 71.639163222 - 0.66981698459 * T + 0.0019150513174 * T^2 - 1.8587687783e-6 * T^3$

$C = 0.27019804611 - 0.0012299975866 * T + 1.5045427918e-6 * T^2$

$VISCOSITYPG = \exp(B + C * \text{percent})$ {Pa s}

END

FUNCTION UVAL(T,mpg,diam,ins,kins)

{ calculates U-value of Dowfrost distribution pipes }

$\text{diamtotal} = \text{diam} + 2 * \text{ins}$ {m} {total outside diameter of pipe and insulation}

$\text{Area} = \pi * (\text{diam}/2)^2$ {m²} {total heat transfer area}

$\text{vel} = (\text{mpg}/DENSITYPG(60,T))/\text{Area}$ {m/s} {velocity of Dowfrost in the pipes}

$\text{kvis} = VISCOSITYPG(60,T)/DENSITYPG(60,T)$ {m²/s} {kinematic viscosity}

$\text{Re} = \text{vel} * \text{diam}/\text{kvis}$ {Reynolds number}

$\text{Pr} = 1000 * VISCOSITYPG(60,T) * CPPG(60,T)/THERMCONDPG(60,T)$

```

{ Prandtl number}
Nui = 0.023*(Re**0.8)*(Pr**0.4) {Dittus - Boelter correlation for Nusselt number}
hi = Nui*THERMCONDPG(60,T)/diam {W/m^2-C} {inside heat transfer coefficient}
ho = 6 {W/m^2-C} {outside heat transfer coefficient}
UVAL = (1/1000)/((diamtotal/(diam*hi))+((diamtotal/2)*ln(diamtotal/diam)/kins)+(1/ho))
{ W/m^2-C} {overall heat transfer coefficient}
END

```

```

FUNCTION PRESSDROP(T,mpg,diam,rough,length,Keq,gravity)
{calculates the pressure drop through pipes}
Area = pi*(diam/2)^2 {m^2} {inside area of pipes}
vel = (mpg/DENSITYPG(60,T))/Area {m/s} {velocity of Dowfrost in pipes}
kvis = VISCOSITYPG(60,T)/DENSITYPG(60,T) {m^2/s} {kinematic viscosity}
Re = vel*diam/kvis {Reynolds number}
{calculates the friction factor from Moody diagram using Haarland correlation}
Reynolds numbers less than 2300 considered laminar}
if (Re<2300) then ff = 64/Re else ff = (1/(-1.8*log10((6.9/Re) +
((rough/diam)/3.7)**1.11)))**2
headf = FF*(length/diam)*(vel^2/(2*gravity))+Keq*(vel^2)/(2*gravity) {m}
{head losses}
PRESSDROP = headf*DENSITYPG(60,T)*gravity {Pa} {pressure drop}
END

```

```

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)
{calculates the overall heat transfer coefficient of the ammonia-Dowfrost heat exchanger}
{inside pipes}
Tb = (Tin+Tout)/2 {K} {bulk temperature}
vel = (mdot/DENSITYPG(percent,Tb))/(num*ri**2) {m/s}
{velocity of Dowfrost through pipes}
Red = vel*(2*ri)/(VISCOSITYPG(percent,Tb)/DENSITYPG(percent,Tb))
{Reynolds number}
Pr = 1000*VISCOSITYPG(percent,Tb)*CPPG(percent,Tb)
/THERMCONDPG(percent,Tb) {Prandtl number}
{Nusselt number calculated using Hausen correlation for lamminar flow and Nusselt
correlation for turbulent flow}
if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
Nud = 3.66 + (0.668*(2*ri/length)*Red*Pr)/(1+0.4*((2*ri/length)*Red*Pr)**(2/3))
hi = NUD*THERMCONDPG(percent,Tb)/(2*ri) {W/m^2-C}
{inside heat transfer coefficient}
{outside pipes}
Csf = 0.01 {empirical constant}
{heat of fusion}
hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
{kj/kg}
g = 9.8 {m/s^2} {acceleration of gravity}
sigma = 0.0213 {N/m} {surface tension}
Pr1 = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*
SpecHeat(AmmoniaTim,T=Tsat,P=Psat+1)/
Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) {Prandtl number}

```

```

Area = 2*pi*ro*length*num    {total heat transfer area}
{boiling heat transfer coefficient from Rohsenow correlation}
ho = 1000*Viscosity(AmmoniaTim,T=Tsats,P=Psat+1)*hfg*
((g*(1/Volume(AmmoniaTim,T=Tsats,x=0.0)-
1/Volume(AmmoniaTim,T=Tsats,x=1.0))/sigma)**0.5)*
((SpecHeat(AmmoniaTim,T=Tsats,P=Psat-1)/(Csf*hfg*Prl**1.7))**3)*(Twall-Tsats)**2
{W/m^2-C}

```

```

{Uvalue}
UHX = (1/1000)/((ro/(ri*hi))+ro*ln(ro/ri)/kpipe) + (1/ho)) {W/m^2-C}
{overall heat transfer coefficient}

```

```

END

```

```

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
{calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
{inside pipe}
vel = (mdot/DENSITYPG(percent,Tref))/(num*ri**2) {m/s}
{velocity of Dowfrost through pipes}
Redi = vel*(2*ri)/(VISCOSITYPG(percent,Tref)/DENSITYPG(percent,Tref))
{Reynolds number}
Pri = 1000*VISCOSITYPG(percent,Tref)*CPPG(percent,Tref)/
THERMCONDPG(percent,Tref) {Prandtl number}
{Nusselt number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow}
if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
else Nudi = 3.66 +
(0.668*(2*ri/length)*Redi*Pri)/(1+0.4*((2*ri/length)*Redi*Pri)**(2/3))
hi = NUDI*THERMCONDPG(percent,Tref)/(2*ri) {W/m^2-C}
{inside heat transfer coefficient}

```

```

{outside pipes}
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
{Reynolds number}
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
{Prandtl number}
{Nusselt number using Churchill Bernstein correlation}
Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))*
(1+(Redo/2.82e5)**(5/8))**(4/5)
ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-C}
{outside heat transfer coefficient}
UCASE = (1/((ro/(ri*hi))+ro*ln(ro/ri)/kpipe) + (1/ho)))/1000 {W/m^2-C}
{overall heat transfer coefficient}

```

```

END

```

```

{The equation set begins here}

```

```

{parameters}
Tairout = 267 {K} {temperature of air leaving refrigerated case}
caseload = 15{tons}*3.51685 {kW} {refrigeration load met by the case}
UAcond = 8.00 {kW/K} {UA value of the condenser}
Tamb = 300 {K} {ambient temperature}

```

$T_{store} = 300 \text{ {K}}$ {store temperature}

{Compressor Parameters}
 {dimensions of the first stage compressor}
 $bore_{low} = 2.38 \times 2.54 \times 10^{-2} \text{ {m}}$
 $stroke_{low} = 1.75 \times 2.54 \times 10^{-2} \text{ {m}}$
 $speed_{low} = 1700/60 \text{ {rev/sec}}$
 $clear_{ratio_{low}} = 0.05$
 {dimensions of the second stage compressor}
 $bore_{high} = 2.38 \times 2.54 \times 10^{-2} \text{ {m}}$
 $stroke_{high} = 1.75 \times 2.54 \times 10^{-2} \text{ {m}}$
 $speed_{high} = 1700/60 \text{ {rev/sec}}$
 $clear_{ratio_{high}} = 0.05$
 $poly_{exp} = 1.3$ {polytropic exponent for ammonia}
 $poly_{eff} = 0.65$ {polytropic efficiency}

{Dowfrost Parameters}
 $mpg = 2.305 \text{ {kg/s}}$ {mass flow rate of Dowfrost}
 $x = 0.54$ {pressure ratio exponent}
 $diam_c = 0.12 \text{ {m}}$ {diameter of pipe to refrigerated case}
 $diam_h = 0.12 \text{ {m}}$ {diameter of pipe from refrigerated case}
 $ins_c = 0.025 \text{ {m}}$ {thickness of insulation on pipe to case}
 $ins_h = 0.025 \text{ {m}}$ {thickness of insulation on pipe from case}
 $kin_c = 0.04 \text{ {W/m-C}}$ {thermal conductivity of insulation on pipe to case}
 $kin_h = 0.04 \text{ {W/m-C}}$ {thermal conductivity of insulation on pipe from case}
 $rough = 0.000046 \text{ {m}}$ {roughness of pipe material}
 $length_c = 30 \text{ {m}}$ {length of pipe to case}
 $length_h = 30 \text{ {m}}$ {length of pipe from case}
 $gravity = 9.81 \text{ {m/s}^2}$ {acceleration due to gravity}
 $diam_{total} = diam_c + 2 \times ins_c \text{ {m}}$ {total diameter of pipe and insulation to case}
 $diam_{total} = diam_h + 2 \times ins_h \text{ {m}}$ {total diameter of pipe and insulation from case}
 $K_{pipe_c} = 8.0$ {for minor losses}
 $K_{pipe_h} = 8.0$ {for minor losses}

{Heat Exchanger Parameters}
 $num_{hx} = 300$ {number of pipes in ammonia-Dowfrost heat exchanger}
 $ri_{hx} = 0.005 \text{ {m}}$ {inside radius of pipes in ammonia-Dowfrost heat exchanger}
 $thick_{hx} = 0.0015 \text{ {m}}$ {thickness of pipes in ammonia-Dowfrost heat exchanger}
 $ro_{hx} = ri_{hx} + thick_{hx} \text{ {m}}$ {outside radius of pipes in ammonia-Dowfrost heat exchanger}
 $len_{hx} = 2.0 \text{ {m}}$ {length of pipes in ammonia-Dowfrost heat exchanger}
 $k_{pipe_{hx}} = 43 \text{ {W/m-C}}$ {thermal conductivity of pipes in ammonia-Dowfrost heat exchanger}
 $K_{hx} = 8 \times num_{hx} + 1.5$ {for minor losses}
 $num_{case} = 700$ {number of pipes in refrigerated case heat exchanger}
 $ri_{case} = 0.01 \text{ {m}}$ {inside radius of pipes in refrigerated case heat exchanger}
 $thick_{case} = 0.0015 \text{ {m}}$ {thickness of pipes in refrigerated case heat exchanger}
 $ro_{case} = ri_{case} + thick_{case} \text{ {m}}$

$\{ \text{outside radius of pipes in refrigerated case heat exchanger} \}$
 $\text{lencase} = 2.0 \{ \text{m} \} \{ \text{length of pipes in refrigerated case heat exchanger} \}$
 $\text{kpipecase} = 43 \{ \text{W/m}^2 \}$
 $\{ \text{thermal conductivity of pipes in refrigerated case heat exchanger} \}$
 $\text{airvel} = 5.0 \{ \text{m/s} \} \{ \text{velocity of the air through the refrigerated case heat exchanger} \}$
 $\text{airarea} = 0.5 \{ \text{m}^2 \} \{ \text{cross sectional area for air flow in refrigerated case heat exchanger} \}$
 $\text{Kcase} = 8 * \text{numcase} + 1.5 \{ \text{for minor losses} \}$

{ Secondary Cycle }

{ equations for the refrigerated case heat exchanger }

$\text{Tair} = (\text{Tairin} + \text{Tairout})/2 \{ \text{K} \}$

$\{ \text{bulk temperature of the air through the refrigerated case heat exchanger} \}$

$\text{Tpg} = (\text{T6} + \text{T5})/2 \{ \text{K} \}$

$\{ \text{bulk temperature of Dowfrost through the refrigerated case heat exchanger} \}$

$\text{Cpaircase} = \text{SpecHeat}(\text{Air}, \text{T} = \text{Tairin}) \{ \text{kJ/kg-K} \} \{ \text{specific heat of air} \}$

$\text{mdotair} = (\text{airvel}/\text{Volume}(\text{Air}, \text{T} = \text{Tairin}, \text{P} = 101.325)) * \text{airarea} \{ \text{kg/s} \} \{ \text{mass flow of air} \}$

$\text{Cpcase} = \text{CPPG}(60, \text{T5}) \{ \text{kJ/kg-K} \} \{ \text{specific heat of Dowfrost} \}$

$\text{Cair} = \text{Cpaircase} * \text{mdotair} \{ \text{kW/K} \} \{ \text{capacitance rate of air stream} \}$

$\text{Cpg} = \text{Cpcase} * \text{mpg} \{ \text{kW/K} \} \{ \text{capacitance rate of Dowfrost stream} \}$

$\text{Cmin} = \text{if}(\text{Cair}, \text{Cpg}, \text{Cair}, \text{Cair}, \text{Cpg}) \{ \text{minimum capacitance rate} \}$

$\text{Cmax} = \text{if}(\text{Cair}, \text{Cpg}, \text{Cpg}, \text{Cair}, \text{Cair}) \{ \text{maximum capacitance rate} \}$

$\text{Cr} = \text{Cmin}/\text{Cmax} \{ \text{ratio of the capacitance rates} \}$

$\{ \text{calls function to calculate the overall heat transfer coefficient} \}$

$\text{Ucasehx} =$

$\text{UCASE}(\text{numcase}, \text{ricase}, \text{rocase}, \text{lencase}, \text{mpg}, \text{airvel}, \text{airarea}, \text{Tpg}, 60, \text{Tair}, \text{kpipecase})$

$\{ \text{W/m}^2\text{-K} \}$

$\text{areacase} = (\text{numcase} * 2 * \pi * \text{rocase} * \text{lencase}) \{ \text{m}^2 \} \{ \text{total heat transfer area} \}$

$\text{UAcase} = \text{Ucasehx} * \text{areacase} \{ \text{W/K} \} \{ \text{UA value} \}$

$\text{NTUcase} = \text{UAcase}/\text{Cmin} \{ \text{number of transfer units} \}$

$\{ \text{calculation of the effectiveness using cross-flow correlation with both fluids unmixed} \}$

$\text{effcase} = 1 - \exp((1/\text{Cr}) * (\text{NTUcase} ** 0.22) * (\exp(-\text{Cr} * (\text{NTUcase} ** 0.78)) - 1))$

$\text{caseload} = \text{effcase} * \text{Cmin} * (\text{Tairin} - \text{T5})$

$\{ \text{heat transferred in refrigerated case (refrigeration load met)} \}$

$\text{caseload} = \text{Cair} * (\text{Tairin} - \text{Tairout}) \{ \text{change in temperature of the air stream} \}$

$\text{caseload} = \text{Cpg} * (\text{T6} - \text{T5}) \{ \text{change in temperature of the Dowfrost stream} \}$

$\{ \text{calls function to calculate pressure drop through the heat exchanger pipes} \}$

$\text{deltaPcase} = \text{PRESSDROP}(\text{Tpg}, \text{mpg}/\text{numcase}, 2 * \text{ricase}, \text{rough}, \text{lencase}, \text{Kcase}, \text{gravity})$

$\{ \text{Pa} \}$

$\{ \text{calculates the pump work needed to overcome pressure drop in refrigerated case heat exchanger} \}$

$\text{Wpumppcase} = \text{mpg} * (\text{deltaPcase}/\text{DENSITYPG}(60, \text{Tpg}))/1000 \{ \text{kW} \}$

{ Piping System }

$\text{Tc} = (\text{T8} + \text{T5})/2 \{ \text{K} \} \{ \text{bulk temperature of Dowfrost in pipes to case} \}$

$\text{Th} = (\text{T6} + \text{T7})/2 \{ \text{K} \} \{ \text{bulk temperature of Dowfrost in pipes from case} \}$

$\{ \text{calls function to calculate overall loss coefficient of pipes to refrigerated case} \}$

$\text{Uc} = \text{UVAL}(\text{Tc}, \text{mpg}, \text{diamc}, \text{insc}, \text{kinsc}) \{ \text{W/m}^2\text{-K} \}$

$\{ \text{calls function to calculate overall loss coefficient of pipes from refrigerated case} \}$

```

Uh = UVAL(Th,mpg,diamh,insh,kinsh) {W/m^2-K}
{calls function to determine pressure drop in pipes to refrigerated case}
deltaPc = PRESSDROP(Tc,mpg,diamc,rough,lengthc,Kpipec,gravity) {Pa}
{calls function to determine pressure drop in pipes from refrigerated case}
deltaPh = PRESSDROP(Th,mpg,diamh,rough,lengthh,Kpipeh,gravity) {Pa}

densityc = DENSITYPG(60,Tc) {kg/m^3} {density of Dowfrost in pipes to case}
densityh = DENSITYPG(60,Th) {kg/m^3} {density of Dowfrost in pipes from case}
{Pump work necessary to overcome pressure drop in pipes to and from case}
Wpumppipe = mpg*(deltaPc/densityc + deltaPh/densityh)/1000 {kW}

{calculates thermal losses of piping system}
Ac = pi*diamctotal*lengthc {m^2} {total heat transfer area of pipes to case}
Ah = pi*diamhtotal*lengthh {m^2} {total heat transfer area of pipes from case}
Cpc = CPPG(60,Tc) {kJ/kg-K} {specific heat of Dowfrost in pipes to case}
Cph = CPPG(60,Th) {kJ/kg-K} {specific heat of Dowfrost in pipes from case}
Qc = mpg*Cpc*(T5-T8) {kW} {heat loss of Dowfrost stream in pipes to case}
Qh = mpg*Cph*(T7-T6) {kW} {heat loss of Dowfrost stream in pipes from case}
deltatlmc = (T5-T8)/(ln((Tstore-T8)/(Tstore-T5)))
{log mean temperature difference of pipes to case}
deltatlmh = (T7-T6)/(ln((Tstore-T6)/(Tstore-T7)))
{log mean temperature difference of pipes from case}
Qc = Uc*Ac*deltatlmc {kW} {heat loss from pipes to case}
Qh = Uh*Ah*deltatlmh {kW} {heat loss from pipes from case}
NTUc = Uc*Ac/(mpg*Cpc) {number of transfer units in flow to case}
NTUh = Uh*Ah/(mpg*Cph) {number of transfer units in flow from case}

{ equation for the heat exchanger between Dowfrost and Ammonia}
Twall = T7-(T7-T8)/2 {K} {surface temperature of pipes in the ammonia}
{calls function to calculate the overall heat transfer coefficient of heat exchanger}
U = UHX(numhxp,rihx,rohr,lehx,mpg,T7,T8,Twall,T1,P1,60,kpipehx) {W/m^2-K}
areahx = (2*rohr*pi*lehx*numhxp) {m^2} {total heat transfer area of heat exchanger}
UAhx = U*areahx {W/K} {UA value of the heat exchanger}
NTUhx = UAhx/(mpg*CPPG(60,T7)) {number of transfer units of heat exchanger}
effhx = 1-exp(-NTUhx) {effectiveness of the heat exchanger}
effhx = (T7-T8)/(T7-T1)
{using effectiveness to calculate the temperatures in the heat exchanger}
Qhx = mpg*CPPG(60,T7)*(T7-T8) {kW} {heat transfer in heat exchanger}
Tmid = (T7+T8)/2 {K} {bulk temperature in Dowfrost in heat exchanger}
T1 = T2 {K} {constant ammonia temperature in heat exchanger}
mref = Qhx/(h2-h1) {kg/s}
{calculates necessary ammonia mass flow rate to meet the load}
{calls function to determine pressure drop of flow through the heat exchanger}
deltaPhx = PRESSDROP(Tmid,mpg,numhxp,2*rihx,rough,lehx,Khx,gravity) {Pa}
{calculates pump work needed to overcome pressure drop in heat exchanger}
Wpumphx = mpg*(deltaPhx/DENSITYPG(60,Tmid))/1000 {kW}

P2 = Pressure(AmmoniaTim,T=T1,x=1.0) {kPa}
{pressure of ammonia leaving heat exchanger}

```

```

h2 = Enthalpy(AmmoniaTim,T=T2,x=1.0) {kJ/kg}
                                     {enthalpy of ammonia leaving heat exchanger}
s2 = Entropy(AmmoniaTim,T=T2,x=1.0) {kJ/kg-K}
                                     {entropy of ammonia leaving heat exchanger}
P1 = P2 {kPa} {ammonia pressure constant in heat exchanger}

{ Ammonia Cycle}
{compressor}
rp = P3/P2 {pressure ratio of evaporator to condenser}
rps = rp^x {exponent used to calculate the pressure ratio of first stage of compression}
rps = P2a/P2 {calculates intermediate temperature}

{ Low Pressure Compressor}
v2 = Volume(AmmoniaTim,P=P2,x=1.0) {m^3/kg}
                                     {specific volume of ammonia entering first compressor}
P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}
h2a = Enthalpy(AmmoniaTim,P=P2a,v=v2a) {kJ/kg}
                                     {enthalpy of ammonia leaving first compressor}
T2a = Temperature(AmmoniaTim,P=P2a,v=v2a) {K}
                                     {temperature of ammonia leaving first compressor}
Vdisplow = numcyllo*(pi*((borelow/2)^2)*strokelow) {m^3}
                                     {displacement volume of first compressor}
volefflow = 1-(clearratio/(1-clearratio))*((P2a/P2)^(1/polyexp) - 1)
                                     {volumetric efficiency of first compressor}
mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}
                                     {mass flow rate of ammonia through first compressor}
Wcomplow = (mref*(h2a-h2))/polyeff {kW} {work of first compressor}

{ High Pressure Compressor}
P2b = P2a {kPa} {pressure of ammonia entering second compressor same as pressure
                                     leaving first compressor}
h2b = Enthalpy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg}
                                     {enthalpy of ammonia entering second compressor}
s2b = Entropy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg-K}
                                     {entropy of ammonia entering second compressor}
T2b = Temperature(AmmoniaTim,P=P2b,x=1.0) {K}
                                     {temperature of ammonia entering second compressor}
v2b = Volume(AmmoniaTim,P=P2b,x=1.0) {m^3/kg}
                                     {specific volume of ammonia entering second compressor}
P2b*(v2b)^polyexp = P3*(vol3)^polyexp {polytropic compression equation}
h3 = Enthalpy(AmmoniaTim,P=P3,v=vol3) {kJ/kg}
                                     {enthalpy leaving second compressor}
T3 = Temperature(AmmoniaTim,P=P3,v=vol3) {K}
                                     {temperature leaving second compressor}
Vdisphigh = numcylhigh*(pi*((borehigh/2)^2)*strokehigh) {m^3}
                                     {displacement volume of second compressor}
voleffhigh = 1-(clearratiohigh/(1-clearratiohigh))*((P3/P2b)^(1/polyexp) - 1)
                                     {volumetric efficiency of second compressor}
mcond = (1/v2b)*Vdisphigh*speedhigh*voleffhigh {kg/s}

```

{mass flow rate of ammonia through second compressor}
 $W_{\text{comphigh}} = (m_{\text{cond}} \cdot (h_3 - h_{2b})) / \text{polyeff} \quad \{\text{kW}\} \quad \{\text{work of second compressor}\}$
 $P_3 = P_4 \quad \{\text{kPa}\} \quad \{\text{constant pressure in condenser}\}$
 $W_{\text{comp}} = W_{\text{complow}} + W_{\text{comphigh}} \quad \{\text{kW}\} \quad \{\text{total compressor work}\}$
 {condensor}
 $m_{\text{air}} = 15 \cdot 0.47879 \quad \{\text{kg/s}\} \quad \{\text{mass flow rate of air through the condenser}\}$
 $C_{\text{pair}} = \text{SpecHeat}(\text{Air}, T = T_{\text{amb}}) \quad \{\text{kJ/kg-K}\} \quad \{\text{specific heat of air in condenser}\}$
 $U A_{\text{cond}} = m_{\text{air}} \cdot C_{\text{pair}} \cdot \text{NTU} \quad \{\text{calculates the number of transfer units of condenser}\}$
 $\text{eff} = 1 - \exp(-\text{NTU}) \quad \{\text{effectiveness of condenser}\}$
 $Q_{\text{cond}} = \text{eff} \cdot m_{\text{air}} \cdot C_{\text{pair}} \cdot (T_4 - T_{\text{amb}}) \quad \{\text{kW}\} \quad \{\text{heat transfer in condenser}\}$
 $Q_{\text{cond}} = m_{\text{cond}} \cdot (h_3 - h_4) \quad \{\text{calculates enthalpy change of ammonia in condenser}\}$
 $Q_{\text{cond}} = m_{\text{air}} \cdot C_{\text{pair}} \cdot (T_{\text{airoutcond}} - T_{\text{amb}}) \quad \{\text{calculates temperature change of air in the condenser}\}$
 $P_4 = \text{Pressure}(\text{AmmoniaTim}, T = T_4, x = 0.0) \quad \{\text{kPa}\} \quad \{\text{pressure of ammonia leaving the condenser}\}$
 $h_4 = \text{Enthalpy}(\text{AmmoniaTim}, T = T_4, x = 0.0) \quad \{\text{kJ/kg}\} \quad \{\text{enthalpy of ammonia leaving the condenser}\}$
 {expansion}
 $P_5 = P_{2a} \quad \{\text{kPa}\} \quad \{\text{intermediate pressure}\}$
 $h_5 = \text{Enthalpy}(\text{AmmoniaTim}, P = P_5, x = 0.0) \quad \{\text{kJ/kg}\} \quad \{\text{enthalpy of saturated liquid ammonia leaving flash tank}\}$
 $h_1 = h_5 \quad \{\text{kJ/kg}\} \quad \{\text{isenthalpic expansion}\}$
 $W_{\text{pump}} = W_{\text{pumpcase}} + W_{\text{pumpphx}} + W_{\text{pumppipe}} \quad \{\text{kW}\} \quad \{\text{total pump work done in system}\}$
 $\text{COP} = \text{caseload} / (W_{\text{comp}} + W_{\text{pump}}) \quad \{\text{coefficient of performance of the system}\}$
 $\text{Cap} = \text{mpg} \cdot \text{CPPG}(60, T_7) / \text{caseload} \quad \{1/\text{K}\} \quad \{\text{capacitance rate - caseload ratio}\}$
 {mixing equation for flash tank}
 $m_{\text{cond}} \cdot h_4 + m_{\text{ref}} \cdot h_{2a} = m_{\text{ref}} \cdot h_5 + m_{\text{cond}} \cdot h_{2b}$

APPENDIX C

R22 - AMMONIA COMPARISON MODELS

This Appendix contains the EES models used to perform the performance comparison of R22 and ammonia with Dowfrost.

R22 MODEL

{ Model of R22 system used to compare with Ammonia with Dowfrost system }

```

FUNCTION UVAL(T,P,mref,diam,ins,kins)
{calculates U-value of R22 distribution pipes}
diamtotal = diam + 2*ins {m} {total outside diameter of pipes and insulation}
Area = pi*(diam/2)^2 {m^2} {total heat transfer area}
vel = (mref/(1/Volume(R22,T=T,P=P)))/Area {m/s} {velocity of R22 in pipes}
kvis = Viscosity(R22,T=T,P=P)/(1/Volume(R22,T=T,P=P)) {m^2/s}
                                         {kinematic viscosity}

Re = vel*diam/kvis {Reynolds number}
Pr = 1000*Viscosity(R22,T=T,P=P)*SpecHeat(R22,T=T,P=P)/
Conductivity(R22,T=T,P=P) {Prandtl number}
Nui = 0.023*(Re**0.8)*(Pr**0.4) {Dittus-Boelter correlation for Nusselt number}
hi = Nui*Conductivity(R22,T=T,P=P)/diam {W/m^2-C}
                                         {inside heat transfer coefficient}

ho = 6 {W/m^2-C} {outside heat transfer coefficient}
UVAL = (1/1000)/((diamtotal/(diam*hi))+((diamtotal/2)*ln(diamtotal/diam)/kins)+(1/ho))
                                         {W/m^2-C} {overall heat transfer coefficient}

END

```

```

FUNCTION PRESSDROP(T,P,mref,diam,rough,length,Keq,gravity)
{calculates the pressure drop though pipes}
Area = pi*(diam/2)^2 {m^2} {inside area of pipes}
vel = (mref/(1/Volume(R22,T=T,P=P)))/Area {m/s} {velocity of R22 in pipes}
kvis = Viscosity(R22,T=T,P=P)/(1/Volume(R22,T=T,P=P)) {m^2/s}
                                         {kinematic velocity}

Re = vel*diam/kvis {Reynolds number}
{calculates the friction factor from Moody diagram using Haarland correlation
  Reynolds numbers less than 2300 considered laminar}
if (Re<2300) then ff = 64/Re else ff = (1/(-1.8*log10((6.9/Re) +
((rough/diam)/3.7)**1.11)))*2
headf = FF*(length/diam)*(vel^2/(2*gravity)) + Keq*(vel^2)/(2*gravity) {m}
                                         {head losses}

PRESSDROP = headf*(1/Volume(R22,T=T,P=P))*gravity {Pa} {pressure drop}

END

```

```

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,Pref,Tair,kpipe)

```

```

{calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
{outside pipes}
{Reynolds number}
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
{Prandtl number}
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
{Nusselt number using Churchill-Bernstein correlation}
Nudo =
0.3+(((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))*(1+(Redo/2.82e5)
**((5/8))**(4/5))
ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-C}
{outside heat transfer correlation}

{inside pipe}
hi = 10*ho {W/m^2-C}
{approximation of inside heat transfer correlation with phase change}
UCASE = (1/((ro/(ri*hi)))+(ro*ln(ro/ri)/kpipe) + (1/ho))/1000 {W/m^2-K}
{overall heat transfer coefficient}

END

{The equation set begins here}

{parameters}
Tairout = 267 {K} {temperature of air leaving refrigerated case}
degssupheat = 4 {K} {number of degrees of superheat}
caseload = 1.5*3.51685 {kW} {refrigeration load met by case}
UAcond = 8.0 {kW/K} {UA value of condenser}
Tamb = 300 {K} {ambient temperature}
Tstore = 300 {K} {store temperature}

{Compressor parameters}
{dimensions of the first stage compressor}
borelow = 2.38*2.54e-2 {m}
strokelow = 1.75*2.54e-2 {m}
speedlow = 1700/60 {rev/sec}
clearratiolow = 0.05 {clearance ratio}
{dimensions of the second stage compressor}
borehigh = 2.38*2.54e-2 {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rev/sec}
clearratiohigh = 0.05 {clearance ratio}
polyexp = 1.15 {polytropic exponent}
polyeff = 0.65 {polytropic efficiency}

{Piping System parameters}
diamc = 0.08 {m} {diameter of pipes to refrigerated case}
diamh = 0.08 {m} {diameter of pipes from refrigerated case}
insc = 0.025 {m} {thickness of insulation on pipes to case}
insh = 0.025 {m} {thickness of insulation on pipes from case}
kinsc = 0.04 {W/m-C} {thermal conductivity of insulation on pipes to case}

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kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipes from case}
 rough = 0.000046 {m} {pipe roughness}
 lengthc = 30 {m} {length of pipes to case}
 lengthh = 30 {m} {length of pipes from case}
 gravity = 9.81 {m/s^2} {acceleration due to gravity}
 diamctotal = diamc + 2*insc {m} {total diameter of pipe and insulation to case}
 diamhtotal = diamh + 2*insh {m} {total diameter of pipe and insulation from case}
 Kpipec = 8.0 {for minor losses of pipes to case}
 Kpipeh = 8.0 {for minor losses of pipes from case}

{Heat Exchanger Parameters}

numcase = 90 {number of pipes in refrigerated case heat exchanger}
 ricase = 0.01 {m} {inside radius of pipes in case heat exchanger}
 thickcase = 0.0015 {m} {thickness of pipes in case heat exchanger}
 rocase = ricase + thickcase {m} {outside radius of pipes in case heat exchanger}
 lencase = 2.0 {m} {length of pipes in case heat exchanger}
 airvel = 5.0 {m/s} {velocity of air through case heat exchanger}
 airarea = 0.5 {m^2} {cross sectional area for air flow in case heat exchanger}
 kpipecase = 43 {W/m^2} {thermal conductivity of pipes in case heat exchanger}
 Kcase = 8*numcase + 1.5 {for minor losses}

mref = caseload/(h2-h1) {kg/s} {determines mass flow rate of R22 from refrigeration
 load and enthalpy change in evaporator}

{evaporator (refrigerated case heat exchanger)}

Tair = (Tairin + Tairout)/2 {K} {bulk temperature of air in case heat exchanger}
 Cpaircase = SpecHeat(Air,T=Tairin) {kJ/kg-K} {specific heat of air}
 mdotair = (airvel/Volume(Air,T=Tairin,P=101.325))*airarea {kg/s}
 {mass flow rate of air}

Cair = Cpaircase*mdotair {kW/K} {capacitance rate of air stream}
 {calls function to calculate overall heat transfer coefficient of case heat exchanger}
 {W/m^2-K}

Ucasehx =
 UCASE(numcase,ricase,rocase,lencase,mref,airvel,airarea,T1,P1,Tair,kpipecase)
 UAcase = Ucasehx*(numcase*2*pi*rocase*lencase) {W/K} {UA value}
 NTUcase = UAcase/Cair {number of transfer units}
 effcase = 1 - exp(-NTUcase) {effectiveness of case heat exchanger}
 caseload = effcase*Cair*(Tairin-T1) {kW} {heat transfer in case heat exchanger}
 caseload = Cair*(Tairin-Tairout) {calculates the temperature change of air}

T2 = T1 + degsupheat {K} {temperature of R22 leaving evaporator}
 P2 = Pressure(R22,T=T1,x=1.0) {kPa} {pressure leaving evaporator}
 h2 = Enthalpy(R22,T=T2,P=P2) {kJ/kg} {enthalpy leaving evaporator}
 s2 = Entropy(R22,T=T2,P=P2) {kJ/kg-K} {entropy leaving evaporator}
 P2 = P1 {kPa} {constant pressure across the evaporator}

{Piping System}

Tc = (T6+T6in)/2 {K} {bulk temperature of R22 in pipes to case}
 Th = (T2+T2in)/2 {K} {bulk temperature of R22 in pipes from case}


```

{calls function to calculate overall heat transfer coefficient of pipes to case}
Uc = UVAL(Tc,P6,mref,diamc,insc,kinsc) {W/m^2-K}
{calls function to calculate overall heat transfer coefficient of pipes from case}
Uh = UVAL(Th,P2,mref,diamh,insc,kinsh) {W/m^2-K}
{calls function to calculate pressure drop in pipes to case}
deltaPc = PRESSDROP(Tc,P6,mref,diamc,rough,lengthc,Kpipecc,gravity) {kPa}
{calls function to calculate pressure drop in pipes to case}
deltaPh = PRESSDROP(Th,P2,mref,diamh,rough,lengthh,Kpipehc,gravity) {kPa}
P2in = P2 - deltaPh {kPa} {adds pressure drop to be overcome by compressor}
P3 = P4 + deltaPc {kPa} {adds pressure drop to condenser pressure}

{calculates thermal losses in distribution pipes}
Ac = pi*diamctotal*lengthc {m^2} {total heat transfer area of pipes to case}
Ah = pi*diamhtotal*lengthh {m^2} {total heat transfer area of pipes from case}
Qc = mref*SpecHeat(R22,T=Tc,P=P6)*(T6in-T6) {kW} {heat loss in pipes to case}
Qh = mref*SpecHeat(R22,T=Th,P=P2)*(T2in-T2) {kW} {heat loss in pipes from case}
deltatlmc = (T6in-T6)/(ln((Tstore-T6)/(Tstore-T6in)))
{log mean temperature difference in pipes to case}
deltatlmh = (T2in-T2)/(ln((Tstore-T2)/(Tstore-T2in)))
{log mean temperature difference in pipes from case}
Qc = Uc*Ac*deltatlmc {kW} {heat loss in pipes to case}
Qh = Uh*Ah*deltatlmh {kW} {heat loss in pipes to case}
NTUc = Uc*Ac/(mref*SpecHeat(R22,T=Tc,P=P6))
{number of transfer units in flow to case}
NTUh = Uh*Ah/(mref*SpecHeat(R22,T=Th,P=P2))
{number of transfer units in flow from case}

{compressor}
rp = P3/P2in {ratio of condenser pressure to evaporator pressure with pressure losses}
rps = rp^x {relationship of the pressure ratios}
rps = P2a/P2in
{ratio of intermediate pressure to evaporator pressure with pressure losses}

v2in = Volume(R22,P=P2in,T=T2in) {m^3/kg}
{specific volume entering first compressor}
h2in = Enthalpy(R22,P=P2in,T=T2in) {kJ/kg} {enthalpy entering first compressor}
P2in*(v2in)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}
h2a = Enthalpy(R22,P=P2a,v=v2a) {kJ/kg} {enthalpy leaving first compressor}
T2a = Temperature(R22,P=P2a,v=v2a) {K} {tempertaure leaving first compressor}
Vdisplow = numcyllo*(pi*((borelow/2)^2)*strokelow) {m^3}
{displacement volume of first compressor}
volefflow = 1-(clearratiolow/(1-clearratiolow))*((P2a/P2in)^(1/polyexp)-1)
{volumetric efficiency of first compressor}
mref = (1/v2in)*Vdisplow*speedlow*volefflow {kg/s}
{mass flow rate through first compressor}
Wcomplow = (mref*(h2a-h2in))/polyeff {kW} {work of first compressor}

P2b = P2a {kPa} {intermediate pressure}
T2b = Temperature(R22,P=P2b,h=h2b) {K}

```

$v_{2b} = \text{Volume}(\text{R22}, P=P_{2b}, h=h_{2b})$ {m³/kg} {temperature entering second compressor}
 $P_{2b} \cdot (v_{2b})^{\text{polyexp}} = P_3 \cdot (v_3)^{\text{polyexp}}$ {polytropic compression equation}
 $h_3 = \text{Enthalpy}(\text{R22}, P=P_3, v=v_3)$ {kJ/kg} {specific volume entering second compressor}
 $T_3 = \text{Temperature}(\text{R22}, P=P_3, v=v_3)$ {K} {enthalpy leaving second compressor}
 $V_{\text{disphigh}} = \text{numcylhigh} \cdot (\pi \cdot (\text{borehigh}/2)^2 \cdot \text{strokehigh})$ {m³} {temperature leaving second compressor}
 $v_{\text{oleffhigh}} = 1 - (\text{clearratiohigh}/(1 - \text{clearratiohigh})) \cdot ((P_3/P_{2b})^{(1/\text{polyexp})} - 1)$ {displacement volume of second compressor}
 $m_{\text{tot}} = (1/v_{2b}) \cdot V_{\text{disphigh}} \cdot \text{speedhigh} \cdot v_{\text{oleffhigh}}$ {kg/s} {volumetric efficiency of second compressor}
 $W_{\text{comphigh}} = (m_{\text{tot}} \cdot (h_3 - h_{2b})) / \text{polyeff}$ {kW} {mass flow rate through second compressor}
 $W_{\text{comp}} = W_{\text{complow}} + W_{\text{comphigh}}$ {kW} {work of second compressor}
 {total compressor work}
 {condensor}
 $m_{\text{air}} = 15 \cdot 0.47879$ {kg/s} {mass flow rate of air through condenser}
 $C_{\text{pair}} = \text{SpecHeat}(\text{Air}, T=T_{\text{amb}})$ {kJ/kg-K} {specific heat of air}
 $U A_{\text{cond}} = m_{\text{air}} \cdot C_{\text{pair}} \cdot \text{NTU}$ {calculates the number of transfer units of condenser}
 $\text{eff} = 1 - \exp(-\text{NTU})$ {effectiveness of condenser}
 $Q_{\text{cond}} = \text{eff} \cdot m_{\text{air}} \cdot C_{\text{pair}} \cdot (T_4 - T_{\text{amb}})$ {kW} {heat transfer in condenser}
 $Q_{\text{cond}} = m_{\text{tot}} \cdot (h_3 - h_4)$ {calculates enthalpy change of R22 in condenser}
 $Q_{\text{cond}} = m_{\text{air}} \cdot C_{\text{pair}} \cdot (T_{\text{airoutcond}} - T_{\text{amb}})$ {calculates temperature change of air in condenser}
 $P_4 = \text{Pressure}(\text{R22}, T=T_4, x=0.0)$ {kPa} {pressure leaving condenser}
 $h_4 = \text{Enthalpy}(\text{R22}, T=T_4, x=0.0)$ {kJ/kg} {enthalpy leaving condenser}
 {expansion}
 $P_5 = P_{2a}$ {kPa} {intermediate pressure}
 $P_6 = P_{2a}$ {kPa} {intermediate pressure}
 $h_5 = \text{Enthalpy}(\text{R22}, P=P_5, x=1.0)$ {kJ/kg} {enthalpy of R22 vapor for intercooling}
 $h_6 = \text{Enthalpy}(\text{R22}, P=P_6, x=0.0)$ {kJ/kg} {enthalpy of liquid R22 for evaporator}
 $T_6 = \text{Temperature}(\text{R22}, P=P_6, x=0.0)$ {K} {temperature of liquid R22 in separator}
 $h_1 = h_6$ {kJ/kg} {isenthalpic expansion}
 $\text{COP} = \text{caseload} / W_{\text{comp}}$ {coefficient of performance of the system}
 {mixing equations for separator and intercooling}
 $m_{\text{tot}} = m_{\text{ref}} + m_{\text{vap}}$
 $m_{\text{tot}} \cdot h_4 = m_{\text{ref}} \cdot h_6 + m_{\text{vap}} \cdot h_5$
 $m_{\text{tot}} \cdot h_{2b} = m_{\text{ref}} \cdot h_{2a} + m_{\text{vap}} \cdot h_5$

AMMONIA WITH DOWFROST MODEL

{ Ammonia with Dowfrost system model used to compare with R22 model }

FUNCTION CPPG(percent,T)

{ calculates the specific heat of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = 3.8649883866 - 0.023691954902 * \text{percent} - 0.00011278222908 * \text{percent}^2$

$C = 0.001023655712 + 5.6633876714e-5 * \text{percent}$

$CPPG = B + C * T$ {kJ/kg-K}

END

FUNCTION DENSITYPG(percent,T)

{ calculates the density of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = 875.54696219 + 2.151387542 * \text{percent}$

$C = 1.1191046068 - 0.0007599907262 * \text{percent} - 4.9236799989e-5 * \text{percent}^2$

$D = -0.002377960199 - 9.1377252136e-6 * \text{percent} + 1.0872237562e-7 * \text{percent}^2$

$DENSITYPG = B + C * T + D * T^2$ {kg/m³}

END

FUNCTION THERMCONDPG(percent,T)

{ calculates the thermal conductivity of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = -0.78595253278 + 0.015561899561 * \text{percent} - 4.8933521576e-5 * \text{percent}^2$

$C = 0.0076866167254 - 0.0001155974176 * \text{percent} + 3.6603360830e-7 * \text{percent}^2$

$D = -9.9976810237e-6 + 1.4560615474e-7 * \text{percent} - 4.5879383578e-10 * \text{percent}^2$

$THERMCONDPG = B + C * T + D * T^2$ {W/m-K}

END

FUNCTION VISCOSITYPG(percent,T)

{ calculates the viscosity of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin }

$B = 71.639163222 - 0.66981698459 * T + 0.0019150513174 * T^2 - 1.8587687783e-6 * T^3$

$C = 0.27019804611 - 0.0012299975866 * T + 1.5045427918e-6 * T^2$

$VISCOSITYPG = \exp(B + C * \text{percent})$ {Pa s}

END

FUNCTION UVAL(T,mpg,diam,ins,kins)

{ calculates U-value of Dowfrost distribution pipes }

$\text{diamtotal} = \text{diam} + 2 * \text{ins}$ {m} {total outside diameter of pipe and insulation}

$\text{Area} = \pi * (\text{diam}/2)^2$ {m²} {total heat transfer area}

$\text{vel} = (\text{mpg}/DENSITYPG(60,T))/\text{Area}$ {m/s} {velocity of Dowfrost in the pipes}

$\text{kvis} = VISCOSITYPG(60,T)/DENSITYPG(60,T)$ {m²/s} {kinematic viscosity}

$\text{Re} = \text{vel} * \text{diam}/\text{kvis}$ {Reynolds number}

$\text{Pr} = 1000 * VISCOSITYPG(60,T) * CPPG(60,T)/THERMCONDPG(60,T)$

{Prandtl number}

$\text{Nui} = 0.023 * (\text{Re}^{0.8}) * (\text{Pr}^{0.4})$ {Dittus - Boelter correlation for Nusselt number}

$\text{hi} = \text{Nui} * THERMCONDPG(60,T)/\text{diam}$ {W/m²-C} {inside heat transfer coefficient}

$\text{ho} = 6$ {W/m²-C} {outside heat transfer coefficient}

UVAL = (1/1000)/((diamtotal/(diam*hi))+((diamtotal/2)*ln(diamtotal/diam)/kins)+(1/ho))
 {W/m²-C} {overall heat transfer coefficient}

END

FUNCTION PRESSDROP(T,mpg,diam,rough,length,Keq,gravity)
 {calculates the pressure drop through pipes}
 Area = pi*(diam/2)^2 {m²} {inside area of pipes}
 vel = (mpg/DENSITYPG(60,T))/Area {m/s} {velocity of Dowfrost in pipes}
 kvis = VISCOSITYPG(60,T)/DENSITYPG(60,T) {m²/s} {kinematic viscosity}
 Re = vel*diam/kvis {Reynolds number}
 {calculates the friction factor from Moody diagram using Haaland correlation
 Reynolds numbers less than 2300 considered laminar}
 if (Re<2300) then ff = 64/Re else ff = (1/(-1.8*log10((6.9/Re) +
 ((rough/diam)/3.7)**1.11)))**2
 headf = FF*(length/diam)*(vel^2/(2*gravity))+Keq*(vel^2)/(2*gravity) {m}
 {head losses}
 PRESSDROP = headf*DENSITYPG(60,T)*gravity {Pa} {pressure drop}
 END

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)
 {calculates the overall heat transfer coefficient of the ammonia-Dowfrost heat exchanger}
 {inside pipes}
 Tb = (Tin+Tout)/2 {K} {bulk temperature}
 vel = (mdot/DENSITYPG(percent,Tb))/(num*ri**2) {m/s}
 {velocity of Dowfrost through pipes}
 Red = vel*(2*ri)/(VISCOSITYPG(percent,Tb)/DENSITYPG(percent,Tb))
 {Reynolds number}
 Pr = 1000*VISCOSITYPG(percent,Tb)*CPPG(percent,Tb)
 /THERMCONDPG(percent,Tb) {Prandtl number}
 {Nusselt number calculated using Hausen correlation for lamminar flow and Nusselt
 correlation for turbulent flow}
 if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
 Nud = 3.66 + (0.668*(2*ri/length)*Red*Pr)/(1+0.4*((2*ri/length)*Red*Pr)**(2/3))
 hi = NUD*THERMCONDPG(percent,Tb)/(2*ri) {W/m²-C}
 {inside heat transfer coefficient}
 {outside pipes}
 Csf = 0.01 {empirical constant}
 {heat of fusion}
 hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
 {kJ/kg}
 g = 9.8 {m/s²} {acceleration of gravity}
 sigma = 0.0213 {N/m} {surface tension}
 Prl = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*
 SpecHeat(AmmoniaTim,T=Tsat,P=Psat+1)/
 Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) {Prandtl number}
 Area = 2*pi*ro*length*num {total heat transfer area}
 {boiling heat transfer coefficient from Rohsenow correlation}
 ho = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*hfg*
 ((g*(1/Volume(AmmoniaTim,T=Tsat,x=0.0)-

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1/Volume(AmmoniaTim,T=Tsats,x=1.0))/sigma)**0.5)*
((SpecHeat(AmmoniaTim,T=Tsats,P=Psat-1)/(Csf*hf*Prl**1.7))**3)*(Twall-Tsats)**2
{W/m^2-C}
{Uvalue}
UHX = (1/1000)/((ro/(ri*hi))+ro*ln(ro/ri)/kpipe) + (1/ho)) {W/m^2-C}
{overall heat transfer coefficient}
END

```

```

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
{calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
{inside pipe}
vel = (mdot/DENSITYPG(percent,Tref))/(num*ri**2) {m/s}
{velocity of Dowfrost through pipes}
Redi = vel*(2*ri)/(VISCOSITYPG(percent,Tref)/DENSITYPG(percent,Tref))
{Reynolds number}
Pri = 1000*VISCOSITYPG(percent,Tref)*CPPG(percent,Tref)/
THERMCONDPG(percent,Tref) {Prandtl number}
{Nusselt number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow}
if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
else Nudi = 3.66 +
(0.668*(2*ri/length)*Redi*Pri)/(1+0.4*((2*ri/length)*Redi*Pri)**(2/3))
hi = NUDI*THERMCONDPG(percent,Tref)/(2*ri) {W/m^2-C}
{inside heat transfer coefficient}
{outside pipes}
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
{Reynolds number}
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
{Prandtl number}
{Nusselt number using Churchill Bernstein correlation}
Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))*
(1+(Redo/2.82e5)**(5/8))**(4/5)
ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-C}
{outside heat transfer coefficient}
UCASE = (1/((ro/(ri*hi))+ro*ln(ro/ri)/kpipe) + (1/ho))/1000 {W/m^2-C}
{overall heat transfer coefficient}
END

```

{ The equation set begins here }

```

{parameters}
Tairout = 267 {K} {temperature of air leaving refrigerated case}
caseload = 15{tons}*3.51685 {kW} {refrigeration load met by the case}
UAcond = 8.00 {kW/K} {UA value of the condenser}
Tamb = 300 {K} {ambient temperature}
Tstore = 300 {K} {store temperature}

```

```

{ Compressor Parameters}
{dimensions of the first stage compressor}

```

```

borelow = 2.38*2.54e-2    {m}
strokelow = 1.75*2.54e-2  {m}
speedlow = 1700/60 {rev/sec}
clearratiolow = 0.05
{dimensions of the second stage compressor}
borehigh = 2.38*2.54e-2   {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rev/sec}
clearratiohigh = 0.05
polyexp = 1.3    {polytropic exponent for ammonia}
polyeff = 0.65   {polytropic efficiency}

{Dowfrost Parameters}
mpg = 2.305 {kg/s}    {mass flow rate of Dowfrost}
x = 0.54      {pressure ratio exponent}
diamc = 0.12 {m}      {diameter of pipe to refrigerated case}
diamh = 0.12 {m}      {diameter of pipe from refrigerated case}
insec = 0.025 {m}      {thickness of insulation on pipe to case}
insh = 0.025 {m}      {thickness of insulation on pipe from case}
kinsec = 0.04 {W/m-C} {thermal conductivity of insulation on pipe to case}
kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipe from case}
rough = 0.000046 {m}  {roughness of pipe material}
lengthc = 30 {m}      {length of pipe to case}
lengthh = 30 {m}      {length of pipe from case}
gravity = 9.81 {m/s^2} {acceleration due to gravity}
diamctotal = diamc + 2*insec {m} {total diameter of pipe and insulation to case}
diamhtotal = diamh + 2*insh {m} {total diameter of pipe and insulation from case}
Kpipec = 8.0 {for minor losses}
Kpipeh = 8.0 {for minor losses}

{Heat Exchanger Parameters}
numhxp = 300    {number of pipes in ammonia-Dowfrost heat exchanger}
rihx = 0.005 {m} {inside radius of pipes in ammonia-Dowfrost heat exchanger}
thickhx = 0.0015 {m} {thickness of pipes in ammonia-Dowfrost heat exchanger}
rohx = rihx + thickhx {m}
                                {outside radius of pipes in ammonia-Dowfrost heat exchanger}
lenhx = 2.0 {m} {length of pipes in ammonia-Dowfrost heat exchanger}
kpipehx = 43 {W/m-C}
                                {thermal conductivity of pipes in ammonia-Dowfrost heat exchanger}
Khx = 8*numhxp + 1.5 {for minor losses}
numcase = 700    {number of pipes in refrigerated case heat exchanger}
ricase = 0.01 {m} {inside radius of pipes in refrigerated case heat exchanger}
thickcase = 0.0015 {m} {thickness of pipes in refrigerated case heat exchanger}
rocase = ricase + thickcase {m}
                                {outside radius of pipes in refrigerated case heat exchanger}
lencase = 2.0 {m} {length of pipes in refrigerated case heat exchanger}
kpipecase = 43 {W/m^2}
                                {thermal conductivity of pipes in refrigerated case heat exchanger}

```

airvel = 5.0 {m/s} {velocity of the air through the refrigerated case heat exchanger}
 airarea = 0.5 {m^2} {cross sectional area for air flow in refrigerated case heat exchanger}
 Kcase = 8*numcase + 1.5 {for minor losses}

{Secondary Cycle}

{equations for the refrigerated case heat exchanger}

Tair = (Tairin + Tairout)/2 {K}

{bulk temperature of the air through the refrigerated case heat exchanger}

Tpg = (T6+T5)/2 {K}

{bulk temperature of Dowfrost through the refrigerated case heat exchanger}

Cpaircase = SpecHeat(Air,T=Tairin) {kJ/kg-K} {specific heat of air}

mdotair = (airvel/Volume(Air,T=Tairin,P=101.325))*airarea {kg/s} {mass flow of air}

Cpcase = CPPG(60,T5) {kJ/kg-K} {specific heat of Dowfrost}

Cair = Cpaircase*mdotair {kW/K} {capacitance rate of air stream}

Cpg = Cpcase*mpg {kW/K} {capacitance rate of Dowfrost stream}

Cmin = if(Cair,Cpg,Cair,Cair,Cpg) {minimum capacitance rate}

Cmax = if(Cair,Cpg,Cpg,Cair,Cair) {maximum capacitance rate}

Cr = Cmin/Cmax {ratio of the capacitance rates}

{calls function to calculate the overall heat transfer coefficient }

Ucasehx =

UCASE(numcase,ricase,rocase,lencase,mpg,airvel,airarea,Tpg,60,Tair,kpipecase)
 {W/m^2-K}

areacase = (numcase*2*pi*rocase*lencase) {m^2} {total heat transfer area}

UAcase = Ucasehx*areacase {W/K} {UA value}

NTUcase = UAcase/Cmin {number of transfer units}

{calculation of the effectiveness using cross-flow correlation with both fluids unmixed}

effcase = 1-exp((1/Cr)*(NTUcase**0.22)*(exp(-Cr*(NTUcase**0.78))-1))

caseload = effcase*Cmin*(Tairin-T5)

{heat transferred in refrigerated case (refrigeration load met)}

caseload = Cair*(Tairin-Tairout) {change in temperature of the air stream}

caseload = Cpg*(T6-T5) {change in temperature of the Dowfrost stream}

{calls function to calculate pressure drop through the heat exchanger pipes}

deltaPcase = PRESSDROP(Tpg,mpg/numcase,2*ricase,rough,lencase,Kcase,gravity)
 {Pa}

{calculates the pump work needed to overcome pressure drop in refrigerated case heat exchanger}

Wpumpcase = mpg*(deltaPcase/DENSITYPG(60,Tpg))/1000 {kW}

{Piping System}

Tc = (T8+T5)/2 {K} {bulk temperature of Dowfrost in pipes to case}

Th = (T6+T7)/2 {K} {bulk temperature of Dowfrost in pipes from case}

{calls function to calculate overall loss coefficient of pipes to refrigerated case}

Uc = UVAL(Tc,mpg,diamc,insc,kinsc) {W/m^2-K}

{calls function to calculate overall loss coefficient of pipes from refrigerated case}

Uh = UVAL(Th,mpg,diamh,insh,kinsh) {W/m^2-K}

{calls function to determine pressure drop in pipes to refrigerated case}

deltaPc = PRESSDROP(Tc,mpg,diamc,rough,lengthc,Kpipec,gravity) {Pa}

{calls function to determine pressure drop in pipes from refrigerated case}

```

deltaPh = PRESSDROP(Th,mpg,diamh,rough,lengthh,Kpipeh,gravity) {Pa}

densityc = DENSITYPG(60,Tc) {kg/m^3} {density of Dowfrost in pipes to case}
densityh = DENSITYPG(60,Th) {kg/m^3} {density of Dowfrost in pipes from case}
{ Pump work necessary to overcome pressure drop in pipes to and from case}
Wpumppipe = mpg*(deltaPc/densityc + deltaPh/densityh)/1000 {kW}

{ calculates thermal losses of piping system}
Ac = pi*diamctotal*lengthc {m^2} {total heat transfer area of pipes to case}
Ah = pi*diamhtotal*lengthh {m^2} {total heat transfer area of pipes from case}
Cpc = CPPG(60,Tc) {kJ/kg-K} {specific heat of Dowfrost in pipes to case}
Cph = CPPG(60,Th) {kJ/kg-K} {specific heat of Dowfrost in pipes from case}
Qc = mpg*Cpc*(T5-T8) {kW} {heat loss of Dowfrost stream in pipes to case}
Qh = mpg*Cph*(T7-T6) {kW} {heat loss of Dowfrost stream in pipes from case}
deltatlmc = (T5-T8)/(ln((Tstore-T8)/(Tstore-T5)))
{log mean temperature difference of pipes to case}
deltatlmh = (T7-T6)/(ln((Tstore-T6)/(Tstore-T7)))
{log mean temperature difference of pipes from case}
Qc = Uc*Ac*deltatlmc {kW} {heat loss from pipes to case}
Qh = Uh*Ah*deltatlmh {kW} {heat loss from pipes from case}
NTUc = Uc*Ac/(mpg*Cpc) {number of transfer units in flow to case}
NTUh = Uh*Ah/(mpg*Cph) {number of transfer units in flow from case}

{ equation for the heat exchanger between Dowfrost and Ammonia}
Twall = T7-(T7-T8)/2 {K} {surface temperature of pipes in the ammonia}
{calls function to calculate the overall heat transfer coefficient of heat exchanger}
U = UHX(numhxp,rihx,rohr,lenhx,mpg,T7,T8,Twall,T1,P1,60,kpipehx) {W/m^2-K}
areahx = (2*rohr*pi*lenhx*numhxp) {m^2} {total heat transfer area of heat exchanger}
UAhx = U*areahx {W/K} {UA value of the heat exchanger}
NTUhx = UAhx/(mpg*CPPG(60,T7)) {number of transfer units of heat exchanger}
effhx = 1-exp(-NTUhx) {effectiveness of the heat exchanger}
effhx = (T7-T8)/(T7-T1)
{using effectiveness to calculate the temperatures in the heat exchanger}
Qhx = mpg*CPPG(60,T7)*(T7-T8) {kW} {heat transfer in heat exchanger}
Tmid = (T7+T8)/2 {K} {bulk temperature in Dowfrost in heat exchanger}
T1 = T2 {K} {constant ammonia temperature in heat exchanger}
mref = Qhx/(h2-h1) {kg/s}
{calculates necessary ammonia mass flow rate to meet the load}
{calls function to determine pressure drop of flow through the heat exchanger}
deltaPhx = PRESSDROP(Tmid,mpg,numhxp,2*rihx,rough,lenhx,Khx,gravity) {Pa}
{calculates pump work needed to overcome pressure drop in heat exchanger}
Wpumphx = mpg*(deltaPhx/DENSITYPG(60,Tmid))/1000 {kW}

P2 = Pressure(AmmoniaTim,T=T1,x=1.0) {kPa}
{pressure of ammonia leaving heat exchanger}
h2 = Enthalpy(AmmoniaTim,T=T2,x=1.0) {kJ/kg}
{enthalpy of ammonia leaving heat exchanger}
s2 = Entropy(AmmoniaTim,T=T2,x=1.0) {kJ/kg-K}
{entropy of ammonia leaving heat exchanger}

```


$P1 = P2$ {kPa} {ammonia pressure constant in heat exchanger}

{ Ammonia Cycle}

{compressor}

$rp = P3/P2$ {pressure ratio of evaporator to condenser}

$rps = rp^x$ {exponent used to calculate the pressure ratio of first stage of compression}

$rps = P2a/P2$ {calculates intermediate temperature}

{ Low Pressure Compressor}

$v2 = \text{Volume}(\text{AmmoniaTim}, P=P2, x=1.0)$ {m³/kg}

{specific volume of ammonia entering first compressor}

$P2*(v2)^{\text{polyexp}} = P2a*(v2a)^{\text{polyexp}}$ {polytropic compression equation}

$h2a = \text{Enthalpy}(\text{AmmoniaTim}, P=P2a, v=v2a)$ {kJ/kg}

{enthalpy of ammonia leaving first compressor}

$T2a = \text{Temperature}(\text{AmmoniaTim}, P=P2a, v=v2a)$ {K}

{temperature of ammonia leaving first compressor}

$V\text{displow} = \text{numcyllo}*(\pi*((\text{borelow}/2)^2)*\text{strokelow})$ {m³}

{displacement volume of first compressor}

$\text{volefflow} = 1 - (\text{clearratio} / (1 - \text{clearratio})) * ((P2a/P2)^{(1/\text{polyexp})} - 1)$

{volumetric efficiency of first compressor}

$m\text{ref} = (1/v2)*V\text{displow}*\text{speedlow}*\text{volefflow}$ {kg/s}

{mass flow rate of ammonia through first compressor}

$W\text{complow} = (m\text{ref}*(h2a-h2))/\text{polyeff}$ {kW} {work of first compressor}

{ High Pressure Compressor}

$P2b = P2a$ {kPa} {pressure of ammonia entering second compressor same as pressure leaving first compressor}

$h2b = \text{Enthalpy}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {kJ/kg}

{enthalpy of ammonia entering second compressor}

$s2b = \text{Entropy}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {kJ/kg-K}

{entropy of ammonia entering second compressor}

$T2b = \text{Temperature}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {K}

{temperature of ammonia entering second compressor}

$v2b = \text{Volume}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {m³/kg}

{specific volume of ammonia entering second compressor}

$P2b*(v2b)^{\text{polyexp}} = P3*(v3)^{\text{polyexp}}$ {polytropic compression equation}

$h3 = \text{Enthalpy}(\text{AmmoniaTim}, P=P3, v=v3)$ {kJ/kg}

{enthalpy leaving second compressor}

$T3 = \text{Temperature}(\text{AmmoniaTim}, P=P3, v=v3)$ {K}

{temperature leaving second compressor}

$V\text{disphigh} = \text{numcylhigh}*(\pi*((\text{borehigh}/2)^2)*\text{strokehigh})$ {m³}

{displacement volume of second compressor}

$\text{voleffhigh} = 1 - (\text{clearratiohigh} / (1 - \text{clearratiohigh})) * ((P3/P2b)^{(1/\text{polyexp})} - 1)$

{volumetric efficiency of second compressor}

$m\text{cond} = (1/v2b)*V\text{disphigh}*\text{speedhigh}*\text{voleffhigh}$ {kg/s}

{mass flow rate of ammonia through second compressor}

$W\text{comphigh} = (m\text{cond}*(h3-h2b))/\text{polyeff}$ {kW} {work of second compressor}

$P3 = P4$ {kPa} {constant pressure in condenser}

$W\text{comp} = W\text{complow} + W\text{comphigh}$ {kW} {total compressor work}

```

{condensor}
mair = 15*0.47879 {kg/s} {mass flow rate of air through the condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg-K} {specific heat of air in condenser}
UAcond = mair*Cpair*NTU {calculates the number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kw} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairoutcond-Tamb)
{calculates temperature change of air in the condenser}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa}
{pressure of ammonia leaving the condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg}
{enthalpy of ammonia leaving the condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=0.0) {kJ/kg}
{enthalpy of saturated liquid ammonia leaving flash tank}
h1 = h5 {kJ/kg} {isenthalpic expansion}

Wpump = Wpumpcase+Wpumpha+Wpumppipe {kW}
{total pump work done in system}
COP = caseload/(Wcomp+Wpump) {coefficient of performance of the system}
Cap = mpg*CPPG(60,T7)/caseload {1/K} {capacitance rate - caseload ratio}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

```

APPENDIX D

SECONDARY FLUID COMPARISON MODELS

This Appendix contains the EES models used to perform the comparison of the different secondary fluids.

DOWTHERM SR-1 MODEL

{ Ammonia with Dowtherm SR-1 system model }

FUNCTION DENSITYEG(percent,T)

{ calculates the density of Dowtherm, a Ethylene-Glycol solution,
given temperatures in Kelvin and percent (between 55 and 85) of
Ethylene-glycol by volume }

$$B = 884.53 + 2.1741 * \text{percent}$$

$$C = 1.1613 - 0.0033403 * \text{percent}$$

$$D = -0.0024393 + 2.994e-8 * \text{percent}$$

$$\text{DENSITYEG} = B + C * T + D * T^2 \quad \{ \text{kg/m}^3 \}$$

END

FUNCTION CPEG(percent,T)

{ calculates the Specific Heat of Dowtherm, a Ethylene-Glycol solution,
given temperatures in Kelvin and percent (between 55 and 85) of
Ethylene-glycol by volume }

$$B = 3.9189 - 0.035267 * \text{percent}$$

$$C = 0.0014555 + 4.8423e-5 * \text{percent}$$

$$\text{CPEG} = B + C * T \quad \{ \text{kJ/kg-K} \}$$

END

FUNCTION THERMCONDEG(percent,T)

{ calculates the Thermal Conductivity of Dowtherm, a Ethylene-Glycol solution,
given temperatures in Kelvin and percent (between 55 and 85) of
Ethylene-glycol by volume }

$$B = -0.84402 + 0.016948 * \text{percent} - 6.99691e-5 * \text{percent}^2$$

$$C = 0.0079877 - 0.00012444 * \text{percent} + 5.00412e-7 * \text{percent}^2$$

$$D = -1.06474e-5 + 1.708955e-7 * \text{percent} - 7.065844e-10 * \text{percent}^2$$

$$\text{THERMCONDEG} = B + C * T + D * T^2 \quad \{ \text{W/m-K} \}$$

END

FUNCTION VISCOSITYEG(percent,T)

{ calculates the Viscosity of Dowtherm, a Ethylene-Glycol solution,
given temperatures in Kelvin and percent (between 55 and 85) of
Ethylene-glycol by volume }

$$B = 970.43146598 - 10.001392253 * T + 0.034056662648 * T^2 - 3.8613683343e-5 * T^3$$

```

C = -27.036068044 + 0.27995557712*T - 0.00096062280174*T^2 + 1.0941819338e-
6*T^3
D = 0.19624504556 - 0.0020225892738*T + 6.9220560583e-6*T^2 - 7.8710335530e-
9*T^3
VISCOSITYEG = exp(B + C*percent + D*percent^2) {Pa s}
END

```

```

FUNCTION UVAL(T,meg,diam,ins,kins)
{calculates the U-value of the Dowtherm pipe}
diamtotal = diam + 2*ins {m} {total outside diameter of pipe and insulation}
Area = pi*(diam/2)^2 {m^2} {total heat transfer area}
vel = (meg/DENSITYEG(60,T))/Area {m/s} {velocity of Dowtherm in pipes}
kvis = VISCOSITYEG(60,T)/DENSITYEG(60,T) {m^2/s} {kinematic viscosity}
Re = vel*diam/kvis {Reynolds number}
Pr = 1000*VISCOSITYEG(60,T)*CPEG(60,T)/THERMCONDEG(60,T)
{Prandtl number}
Nui = 0.023*(Re**0.8)*(Pr**0.4) {Dittus-Boelter correlation for Nusselt number}
hi = Nui*THERMCONDEG(60,T)/diam {W/m^2-K} {inside heat transfer coefficient}
ho = 6 {W/m^2-K} {outside heat transfer coefficient}
UVAL = (1/1000)/((diamtotal/(diam*hi))+((diamtotal/2)*ln(diamtotal/diam)/kins)+(1/ho))
{W/m^2-K} {overall heat transfer coefficient}
END

```

```

FUNCTION PRESSDROP(T,meg,diam,rough,length,Keq,gravity)
{calculates the pressure drop through pipes}
Area = pi*(diam/2)^2 {m^2} {inside area of pipes}
vel = (meg/DENSITYEG(60,T))/Area {m/s} {velocity of Dowtherm in pipes}
kvis = VISCOSITYEG(60,T)/DENSITYEG(60,T) {m^2/s} {kinematic viscosity}
Re = vel*diam/kvis {Reynolds number}
{calculates the friction factor from Moody diagram using Haarland correlation}
Reynolds numbers less than 2300 considered laminar}
if (Re<2300) then ff = 64/Re else ff = (1/(-1.8*log10((6.9/Re) +
((rough/diam)/3.7)**1.11)))**2
headf = FF*(length/diam)*(vel^2/(2*gravity)) + Keq*(vel^2)/(2*gravity) {m}
{head losses}
PRESSDROP = headf*DENSITYEG(60,T)*gravity {Pa} {pressure drop}
END

```

```

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)
{calculates the overall heat transfer coefficient of the ammonia-Dowtherm heat
exchanger}
{inside pipes}
Tb = (Tin+Tout)/2 {K} {bulk temperature}
vel = (mdot/DENSITYEG(percent,Tb))/(num*ri**2) {m/s} {velocity of Dowtherm}
Red = vel*(2*ri)/(VISCOSITYEG(percent,Tb)/DENSITYEG(percent,Tb))
{Reynolds number}
Pr = 1000*VISCOSITYEG(percent,Tb)*CPEG(percent,Tb)/
THERMCONDEG(percent,Tb) {Prandtl number}

```

```

{Nusslet number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow}
if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
Nud = 3.66 + (0.668*(2*ri/length)*Red*Pr)/(1+0.4*((2*ri/length)*Red*Pr)**(2/3))
hi = NUD*THERMCONDEG(percent,Tb)/(2*ri) {W/m^2-K}
                                     {inside heat transfer coefficient}

{outside pipes}
Csf = 0.01 {empirical constant}
hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
                                     {kJ/kg} {heat of fusion}

g = 9.8 {m/s^2} {acceleration due to gravity}
sigma = 0.0213 {N/m} {surface tension}
Pr1 = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*
SpecHeat(AmmoniaTim,T=Tsat,P=Psat+1)/
Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) {Prandtl number}
Area = 2*pi*ro*length*num {m^2} {total heat transfer area}
{boiling heat transfer correlation using Rohsenow correlation}
ho = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*hfg*
((g*(1/Volume(AmmoniaTim,T=Tsat,x=0.0)-
1/Volume(AmmoniaTim,T=Tsat,x=1.0))/sigma)**0.5)*
((SpecHeat(AmmoniaTim,T=Tsat,P=Psat-1)/(Csf*hfg*Pr1**1.7))**3)*(Twall-Tsat)**2
                                     {W/m^2-K}

{Uvalue}
UHX = (1/1000)/((ro/(ri*hi))+ (ro*ln(ro/ri)/kpipe) + (1/ho)) {W/m^2-K}
                                     {overall heat transfer coefficient}

END

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
{calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
{inside pipe}
vel = (mdot/DENSITYEG(percent,Tref))/(num*ri**2) {m/s} {velocity of Dowtherm}
Redi = vel*(2*ri)/(VISCOSITYEG(percent,Tref)/DENSITYEG(percent,Tref))
                                     {Reynolds number}

Pri = 1000*VISCOSITYEG(percent,Tref)*CPEG(percent,Tref)/
THERMCONDEG(percent,Tref) {Prandtl number}
{Nusselt number calculated using Nausen correlation fpr laminar flow and Nusselt
correlation for turbulent flow}
if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
else Nudi = 3.66 +
(0.668*(2*ri/length)*Redi*Pri)/(1+0.4*((2*ri/length)*Redi*Pri)**(2/3))
hi = NUDI*THERMCONDEG(percent,Tref)/(2*ri) {W/m^2-K}
                                     {inside heat transfer coefficient}

{outside pipes}
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
                                     {Reynolds number}
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
                                     {Prandtl number}
{Nusselt number calculated using Churchill-Bernstein correlation}

```

```

Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))*
(1+(Redo/2.82e5)**(5/8))**(4/5)
ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-K}
                                         {outside heat transfer coefficient}
UCASE = (1/((ro/(ri*hi)))+(ro*ln(ro/ri)/kpipe) + (1/ho))/1000 {W/m^2-K}
                                         {overall heat transfer coefficient}
END

```

```

{The equation set begins here}

```

```

{parameters}
Tairout = 267 {K} {temperature of the air leaving the refrigerated case}
caseload = 15{tons}*3.51685 {kW} {refrigeration load met by the case}
UAcond = 8.00 {kW/K} {UA value of the condenser}
Tamb = 300 {K} {ambient temperature}
Tstore = 300 {K} {store temperature}

```

```

{Compressor Parameters}
{dimensions of the first stage compressor}
borelow = 2.38*2.54e-2 {m}
strokelow = 1.75*2.54e-2 {m}
speedlow = 1700/60 {rev/sec}
clearratiolow = 0.05 {clearance ratio}
{dimensions of the second stage compressor}
borehigh = 2.38*2.54e-2 {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rev/sec}
clearratiohigh = 0.05 {clearance ratio}
polyexp = 1.3 {polytropic exponent}
polyeff = 0.65 {polytropic efficiency}

```

```

{Dowtherm Parameters}
meg = 2.305 {kg/s} {mass flow rate of Dowtherm}
x = 0.54 {pressure ratio exponent}
diamc = 0.12 {m} {diameter of pipe to case}
diamh = 0.12 {m} {diameter of pipe from case}
insc = 0.025 {m} {thickness of insulation on pipe to case}
insh = 0.025 {m} {thickness of insulation on pipe from case}
kinsc = 0.04 {W/m-C} {thermal conductivity of insulation on pipe to case}
kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipe from case}
rough = 0.000046 {m} {pipe roughness}
lengthc = 30 {m} {length of pipe to case}
lengthh = 30 {m} {length of pipe from case}
gravity = 9.81 {m/s^2} {acceleration due to gravity}
diamctotal = diamc + 2*insc {m} {total diameter of pipe and insulation to case}
diamhtotal = diamh + 2*insh {m} {total diameter of pipe and insulation from case}
Kpipec = 8.0 {for minor losses}
Kpipeh = 8.0 {for minor losses}

```

```

{Heat Exchanger Parameters}
numhxp = 300 {number of pipes in ammonia-Dowtherm heat exchanger}
rihxp = 0.005 {m} {inside radius of pipes in ammonia-Dowtherm heat exchanger}
thickhxp = 0.0015 {m} {thickness of pipes in ammonia-Dowtherm heat exchanger}
rohx = rihxp + thickhxp {m} {outside radius of pipes in ammonia-Dowtherm heat exchanger}
lenhxp = 2.0 {m} {length of pipes in ammonia-Dowtherm heat exchanger}
kpipehxp = 43 {W/m-C}
      {thermal conductivity of pipes in ammonia-Dowtherm heat exchanger}
Khp = 8*numhxp + 1.5 {for minor losses}
numcase = 700 {number of pipes in refrigerated case heat exchanger}
ricase = 0.01 {m} {inside radius of pipes in refrigerated case heat exchanger}
thickcase = 0.0015 {m} {thickness of pipes in refrigerated case heat exchanger}
rocase = ricase + thickcase {m}
      {outside radius of pipes in refrigerated case heat exchanger}
lencase = 2.0 {m} {length of pipes in refrigerated case heat exchanger}
airvel = 5.0 {m/s} {air velocity in refrigerated case heat exchanger}
airarea = 0.5 {m^2}
      {cross sectional area for air flow in refrigerated case heat exchanger}
kpipecase = 43 {W/m^2}
      {thermal conductivity of pipes in refrigerated case heat exchanger}
Kcase = 8*numcase + 1.5 {for minor losses}

{Secondary Cycle}
{equations for the refrigerated case heat exchanger}
Tair = (Tairin + Tairout)/2 {K} {bulk temperature of air in case heat exchanger}
TEG = (T6+T5)/2 {K} {bulk temperature of Dowtherm in case heat exchanger}
Cpaircase = SpecHeat(Air,T=Tairin) {kJ/kg-K} {specific heat of air}
mdotair = (airvel/Volume(Air,T=Tairin,P=101.325))*airarea {kg/s} {mass flow of air}
Cpcase = CPEG(60,T5) {kJ/kg-K} {specific heat of Dowtherm}
Cair = Cpaircase*mdotair {kW/K} {capacitance rate of air}
CEG = Cpcase*meg {kW/K} {capacitance rate of Dowtherm}
Cmin = if(Cair,CEG,Cair,Cair,CEG) {minimum capacitance rate}
Cmax = if(Cair,CEG,CEG,Cair,Cair) {maximum capacitance rate}
Cr = Cmin/Cmax {capacitance rate ratio}
{calls function to calculate the overall heat transfer coefficient} {W/m^2-K}
Ucasehx =
UCASE(numcase,ricase,rocase,lencase,meg,airvel,airarea,TEG,60,Tair,kpipecase)
UAcase = Ucasehx*(numcase*2*pi*rocase*lencase) {W/K} {UA value of case}
NTUcase = UAcase/Cmin {number of transfer units of case heat exchanger}
{calculates the effectiveness using cross-flow correlation with both streams unmixed}
effcase = 1-exp((1/Cr)*(NTUcase**0.22)*(exp(-Cr*(NTUcase**0.78))-1))
caseload = effcase*Cmin*(Tairin-T5) {kW}
      {heat transfer in case (refrigeration load met)}
caseload = Cair*(Tairin-Tairout) {change in temperature of the air stream}
caseload = CEG*(T6-T5) {change in temperature of the Dowtherm}
{calls function to calculate pressure drop through refrigerated case} {Pa}
deltaPcase = PRESSDROP(TEG,meg/numcase,2*ricase,rough,lencase,Kcase,gravity)
{calculates the pump work needed to overcome pressure drop in case}

```


$$W_{pumpcase} = meg * (\Delta P_{case} / \text{DENSITYEG}(60, TEG)) / 1000 \quad \{kW\}$$

{Piping System}

$$T_c = (T_8 + T_5) / 2 \quad \{K\} \quad \{\text{bulk temperature of Dowtherm in pipes to case}\}$$

$$T_h = (T_6 + T_7) / 2 \quad \{K\} \quad \{\text{bulk temperature of Dowtherm in pipes from case}\}$$

{calls function to calculate overall heat transfer coefficient of pipes to case}

$$U_c = \text{UVAL}(T_c, meg, diam_c, in_s, k_{in_s}) \quad \{W/m^2-K\}$$

{calls function to calculate overall heat transfer coefficient of pipes from case}

$$U_h = \text{UVAL}(T_h, meg, diam_h, in_s, k_{in_s}) \quad \{W/m^2-K\}$$

{calls function to calculate pressure drop of pipes to case}

$$\Delta P_c = \text{PRESSDROP}(T_c, meg, diam_c, rough, length_c, K_{pipe_c}, gravity) \quad \{Pa\}$$

{calls function to calculate pressure drop of pipes to case}

$$\Delta P_h = \text{PRESSDROP}(T_h, meg, diam_h, rough, length_h, K_{pipe_h}, gravity) \quad \{Pa\}$$

$$\text{density}_c = \text{DENSITYEG}(60, T_c) \quad \{kg/m^3\} \quad \{\text{density of Dowtherm in pipes to case}\}$$

$$\text{density}_h = \text{DENSITYEG}(60, T_h) \quad \{kg/m^3\} \quad \{\text{density of Dowtherm in pipes from case}\}$$

{calculates pump necessary to overcome pressure drop in piping system}

$$W_{pumppipe} = meg * (\Delta P_c / \text{density}_c + \Delta P_h / \text{density}_h) / 1000 \quad \{kW\}$$

{calculates thermal losses of piping system}

$$A_c = \pi * diam_{total} * length_c \quad \{m^2\} \quad \{\text{total heat transfer area of pipes to case}\}$$

$$A_h = \pi * diam_{total} * length_h \quad \{m^2\} \quad \{\text{total heat transfer area of pipes from case}\}$$

$$C_{pc} = \text{CPEG}(60, T_c) \quad \{kJ/kg-K\} \quad \{\text{specific heat of Dowtherm in pipes to case}\}$$

$$C_{ph} = \text{CPEG}(60, T_h) \quad \{kJ/kg-K\} \quad \{\text{specific heat of Dowtherm in pipes from case}\}$$

$$Q_c = meg * C_{pc} * (T_5 - T_8) \quad \{kW\} \quad \{\text{heat loss of Dowtherm in pipes to case}\}$$

$$Q_h = meg * C_{ph} * (T_7 - T_6) \quad \{kW\} \quad \{\text{heat loss of Dowtherm in pipes from case}\}$$

$$\Delta T_{lm_c} = (T_5 - T_8) / (\ln((T_{store} - T_8) / (T_{store} - T_5)))$$

{log mean temperature difference of pipes to case}

$$\Delta T_{lm_h} = (T_7 - T_6) / (\ln((T_{store} - T_6) / (T_{store} - T_7)))$$

{log mean temperature difference of pipes from case}

$$Q_c = U_c * A_c * \Delta T_{lm_c} \quad \{kW\} \quad \{\text{heat loss of pipes to case}\}$$

$$Q_h = U_h * A_h * \Delta T_{lm_h} \quad \{kW\} \quad \{\text{heat loss of pipes from case}\}$$

$$NTU_c = U_c * A_c / (meg * C_{pc}) \quad \{\text{number of transfer units of pipes to case}\}$$

$$NTU_h = U_h * A_h / (meg * C_{ph}) \quad \{\text{number of transfer units of pipes from case}\}$$

{equations for heat exchanger between Dowtherm and Ammonia}

$$T_{wall} = T_7 - (T_7 - T_8) / 2 \quad \{K\} \quad \{\text{surface temperature of pipes in ammonia}\}$$

{calls function to calculate overall heat transfer coefficient} $\{W/m^2-K\}$

$$U = \text{UHX}(\text{numhxp}, r_{ihx}, r_{ohx}, len_{hx}, meg, T_7, T_8, T_{wall}, T_1, P_1, 60, k_{pipe_{hx}})$$

$$UA_{hx} = U * (2 * r_{ohx} * \pi * len_{hx} * \text{numhxp}) \quad \{W/K\} \quad \{UA \text{ value of heat exchanger}\}$$

$$NTU_{hx} = UA_{hx} / (meg * \text{CPEG}(60, T_7)) \quad \{\text{number of transfer units of heat exchanger}\}$$

$$\text{eff}_{hx} = 1 - \exp(-NTU_{hx}) \quad \{\text{effectiveness of heat exchanger}\}$$

$$\text{eff}_{hx} = (T_7 - T_8) / (T_7 - T_1)$$

{using effectiveness to calculate temperatures in heat exchanger}

$$Q_{hx} = meg * \text{CPEG}(60, T_7) * (T_7 - T_8) \quad \{kW\} \quad \{\text{heat transfer in heat exchanger}\}$$

$$T_{mid} = (T_7 + T_8) / 2 \quad \{K\} \quad \{\text{bulk temperature of Dowtherm in heat exchanger}\}$$

$$T_1 = T_2 \quad \{K\} \quad \{\text{constant ammonia temperature in heat exchanger}\}$$

$$m_{ref} = Q_{hx} / (h_2 - h_1) \quad \{kg/s\} \quad \{\text{calculates necessary ammonia flow rate to meet load}\}$$

{calls function to calculate the pressure drop in heat exchanger}

$\Delta P_{hx} = \text{PRESSDROP}(T_{mid}, \text{meg}/\text{numhxp}, 2 \cdot r_{ihx}, \text{rough}, \text{lenhx}, K_{hx}, \text{gravity})$ {Pa}
 {calculates pump work necessary to overcome pressure drop in heat exchanger}

$W_{pumphx} = \text{meg} \cdot (\Delta P_{hx} / \text{DENSITYEG}(60, T_{mid})) / 1000$ {kW}

$P_2 = \text{Pressure}(\text{AmmoniaTim}, T=T_1, x=1.0)$ {kPa}
 {pressure of ammonia leaving heat exchanger}

$h_2 = \text{Enthalpy}(\text{AmmoniaTim}, T=T_2, x=1.0)$ {kJ/kg}
 {enthalpy of ammonia leaving heat exchanger}

$s_2 = \text{Entropy}(\text{AmmoniaTim}, T=T_2, x=1.0)$ {kJ/kg-K}
 {entropy of ammonia leaving heat exchanger}

$P_1 = P_2$ {kPa} {ammonia pressure constant in heat exchanger}

{Ammonia Cycle}

{compressor}

$r_p = P_3/P_2$ {pressure ratio of evaporator to condenser}

$r_{ps} = r_p^x$ {pressure ratio relationship}

$r_{ps} = P_{2a}/P_2$ {calculates intermediate pressure}

{Low Pressure Compressor}

$v_2 = \text{Volume}(\text{AmmoniaTim}, P=P_2, x=1.0)$ {m³/kg}
 {specific volume of ammonia entering first compressor}

$P_2 \cdot (v_2)^{\text{polyexp}} = P_{2a} \cdot (v_{2a})^{\text{polyexp}}$ {polytropic compression equation}

$h_{2a} = \text{Enthalpy}(\text{AmmoniaTim}, P=P_{2a}, v=v_{2a})$ {kJ/kg}
 {enthalpy of ammonia leaving first compressor}

$T_{2a} = \text{Temperature}(\text{AmmoniaTim}, P=P_{2a}, v=v_{2a})$ {K}
 {temperature of ammonia leaving first compressor}

$V_{\text{displow}} = \text{numcyllo} \cdot (\pi \cdot (\text{borelow}/2)^2 \cdot \text{strokelow})$ {m³}
 {displacement volume of first compressor}

$v_{\text{olefflow}} = 1 - (\text{clearratio}_{\text{low}} / (1 - \text{clearratio}_{\text{low}})) \cdot ((P_{2a}/P_2)^{(1/\text{polyexp})} - 1)$
 {volumetric efficiency of first compressor}

$\dot{m}_{\text{ref}} = (1/v_2) \cdot V_{\text{displow}} \cdot \text{speedlow} \cdot v_{\text{olefflow}}$ {kg/s}
 {mass flow of ammonia through first compressor}

$W_{\text{complow}} = (\dot{m}_{\text{ref}} \cdot (h_{2a} - h_2)) / \text{polyeff}$ {kW} {work of first compressor}

{High Pressure Compressor}

$P_{2b} = P_{2a}$ {kPa} {intermediate pressure}

$h_{2b} = \text{Enthalpy}(\text{AmmoniaTim}, P=P_{2b}, x=1.0)$ {kJ/kg}
 {enthalpy of ammonia entering second compressor}

$s_{2b} = \text{Entropy}(\text{AmmoniaTim}, P=P_{2b}, x=1.0)$ {kJ/kg-K}
 {entropy of ammonia entering second compressor}

$T_{2b} = \text{Temperature}(\text{AmmoniaTim}, P=P_{2b}, x=1.0)$ {K}
 {temperature of ammonia entering second compressor}

$v_{2b} = \text{Volume}(\text{AmmoniaTim}, P=P_{2b}, x=1.0)$ {m³/kg}
 {specific heat of ammonia entering second compressor}

$P_{2b} \cdot (v_{2b})^{\text{polyexp}} = P_3 \cdot (v_{3b})^{\text{polyexp}}$ {polytropic compression equation}

$h_3 = \text{Enthalpy}(\text{AmmoniaTim}, P=P_3, v=v_{3b})$ {kJ/kg}
 {enthalpy of ammonia leaving second compressor}

$T_3 = \text{Temperature}(\text{AmmoniaTim}, P=P_3, v=v_{3b})$ {K}
 {temperature of ammonia leaving second compressor}

```

Vdisphigh = numcylhigh*(pi*((borehigh/2)^2)*strokehigh) {m^3}
                                     {displacement volume of second compressor}
voleffhigh = 1-(clearratiohigh/(1-clearratiohigh))*((P3/P2b)^(1/polyexp) - 1)
                                     {volumetric efficiency of second compressor}
mcond = (1/v2b)*Vdisphigh*speedhigh*voleffhigh {kg/s}
                                     {mass flow of ammonia through second compressor}
Wcomphigh = (mcond*(h3-h2b))/polyeff {kW} {work of second compressor}
P3 = P4 {kPa} {constant pressure across condenser}
Wcomp = Wcompow + Wcomphigh {kW} {total compression work}

{condensor}
mair = 15*0.47879 {kg/s} {flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg-K} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairoutcond-Tamb) {calculates temperature change of air}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa}
                                     {pressure of ammonia leaving condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg}
                                     {enthalpy of ammonia leaving condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=0.0) {kJ/kg}
                                     {enthalpy of saturated liquid ammonia leaving the flash tank}
h1 = h5 {kJ/kg} {isenthalpic expansion}

Wpump = Wpumpcase+Wpumphx+Wpumppipe {kW} {total pump work done by
system}
COP = caseload/(Wcomp+Wpump) {coefficient of performance of refrigeration system}
Cap = meg*CPEG(60,T7)/caseload {1/K} {capacitance rate - caseload ratio}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

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DOWFROST MODEL

{ Ammonia with Dowfrost system model }

FUNCTION CPPG(percent,T)

{calculates the specific heat of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin}

$B = 3.8649883866 - 0.023691954902 * \text{percent} - 0.00011278222908 * \text{percent}^2$

$C = 0.001023655712 + 5.6633876714e-5 * \text{percent}$

$\text{CPPG} = B + C * T \quad \{\text{kJ/kg-K}\}$

END

FUNCTION DENSITYPG(percent,T)

{calculates the density of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin}

$B = 875.54696219 + 2.151387542 * \text{percent}$

$C = 1.1191046068 - 0.0007599907262 * \text{percent} - 4.9236799989e-5 * \text{percent}^2$

$D = -0.002377960199 - 9.1377252136e-6 * \text{percent} + 1.0872237562e-7 * \text{percent}^2$

$\text{DENSITYPG} = B + C * T + D * T^2 \quad \{\text{kg/m}^3\}$

END

FUNCTION THERMCONDPG(percent,T)

{calculates the thermal conductivity of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin}

$B = -0.78595253278 + 0.015561899561 * \text{percent} - 4.8933521576e-5 * \text{percent}^2$

$C = 0.0076866167254 - 0.0001155974176 * \text{percent} + 3.6603360830e-7 * \text{percent}^2$

$D = -9.9976810237e-6 + 1.4560615474e-7 * \text{percent} - 4.5879383578e-10 * \text{percent}^2$

$\text{THERMCONDPG} = B + C * T + D * T^2 \quad \{\text{W/m-K}\}$

END

FUNCTION VISCOSITYPG(percent,T)

{calculates the viscosity of Dowfrost given percent composition of Dowfrost and the temperature in degrees Kelvin}

$B = 71.639163222 - 0.66981698459 * T + 0.0019150513174 * T^2 - 1.8587687783e-6 * T^3$

$C = 0.27019804611 - 0.0012299975866 * T + 1.5045427918e-6 * T^2$

$\text{VISCOSITYPG} = \exp(B + C * \text{percent}) \quad \{\text{Pa s}\}$

END

FUNCTION UVAL(T,mpg,diam,ins,kins)

{calculates U-value of Dowfrost distribution pipes}

$\text{diamtotal} = \text{diam} + 2 * \text{ins} \quad \{\text{m}\} \quad \{\text{total outside diameter of pipe and insulation}\}$

$\text{Area} = \pi * (\text{diam}/2)^2 \quad \{\text{m}^2\} \quad \{\text{total heat transfer area}\}$

$\text{vel} = (\text{mpg}/\text{DENSITYPG}(60,T))/\text{Area} \quad \{\text{m/s}\} \quad \{\text{velocity of Dowfrost in the pipes}\}$

$\text{kvis} = \text{VISCOSITYPG}(60,T)/\text{DENSITYPG}(60,T) \quad \{\text{m}^2/\text{s}\} \quad \{\text{kinematic viscosity}\}$

$\text{Re} = \text{vel} * \text{diam}/\text{kvis} \quad \{\text{Reynolds number}\}$

$\text{Pr} = 1000 * \text{VISCOSITYPG}(60,T) * \text{CPPG}(60,T)/\text{THERMCONDPG}(60,T)$

{Prandtl number}

$\text{Nui} = 0.023 * (\text{Re}^{0.8}) * (\text{Pr}^{0.4}) \quad \{\text{Dittus - Boelter correlation for Nusselt number}\}$

$\text{hi} = \text{Nui} * \text{THERMCONDPG}(60,T)/\text{diam} \quad \{\text{W/m}^2\text{-C}\} \quad \{\text{inside heat transfer coefficient}\}$

$\text{ho} = 6 \quad \{\text{W/m}^2\text{-C}\} \quad \{\text{outside heat transfer coefficient}\}$

$\text{UVAL} = (1/1000)/((\text{diamtotal}/(\text{diam} * \text{hi})) + ((\text{diamtotal}/2) * \ln(\text{diamtotal}/\text{diam})/\text{kins}) + (1/\text{ho}))$
 $\quad \quad \quad \{\text{W/m}^2\text{-C}\} \quad \{\text{overall heat transfer coefficient}\}$

END

```

FUNCTION PRESSDROP(T,mpg,diam,rough,length,Keq,gravity)
{calculates the pressure drop through pipes}
Area = pi*(diam/2)^2 {m^2} {inside area of pipes}
vel = (mpg/DENSITYPG(60,T))/Area {m/s} {velocity of Dowfrost in pipes}
kvis = VISCOSITYPG(60,T)/DENSITYPG(60,T) {m^2/s} {kinematic viscosity}
Re = vel*diam/kvis {Reynolds number}
{calculates the friction factor from Moody diagram using Haarlend correlation}
Reynolds numbers less than 2300 considered laminar}
if (Re<2300) then ff = 64/Re else ff = (1/(-1.8*log10((6.9/Re) +
((rough/diam)/3.7)**1.11)))**2
headf = FF*(length/diam)*(vel^2/(2*gravity))+Keq*(vel^2)/(2*gravity) {m}
{head losses}
PRESSDROP = headf*DENSITYPG(60,T)*gravity {Pa} {pressure drop}
END

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)
{calculates the overall heat transfer coefficient of the ammonia-Dowfrost heat exchanger}
{inside pipes}
Tb = (Tin+Tout)/2 {K} {bulk temperature}
vel = (mdot/DENSITYPG(percent,Tb))/(num*ri**2) {m/s}
{velocity of Dowfrost through pipes}
Red = vel*(2*ri)/(VISCOSITYPG(percent,Tb)/DENSITYPG(percent,Tb))
{Reynolds number}
Pr = 1000*VISCOSITYPG(percent,Tb)*CPPG(percent,Tb)
/THERMCONDPG(percent,Tb) {Prandtl number}
{Nusselt number calculated using Hausen correlation for lamminar flow and Nusselt
correlation for turbulent flow}
if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
Nud = 3.66 + (0.668*(2*ri/length)*Red*Pr)/(1+0.4*((2*ri/length)*Red*Pr)**(2/3))
hi = NUD*THERMCONDPG(percent,Tb)/(2*ri) {W/m^2-C}
{inside heat transfer coefficient}

{outside pipes}
Csf = 0.01 {empirical constant}
{heat of fusion}
hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
{kJ/kg}

g = 9.8 {m/s^2} {acceleration of gravity}
sigma = 0.0213 {N/m} {surface tension}
Prl = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*
SpecHeat(AmmoniaTim,T=Tsat,P=Psat+1)/
Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) {Prandtl number}
Area = 2*pi*ro*length*num {total heat transfer area}
{boiling heat transfer coefficient from Rohsenow correlation}
ho = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*hfg*
((g*(1/Volume(AmmoniaTim,T=Tsat,x=0.0)-
1/Volume(AmmoniaTim,T=Tsat,x=1.0))/sigma)**0.5)*
((SpecHeat(AmmoniaTim,T=Tsat,P=Psat-1)/(Csf*hfg*Prl**1.7))**3)*(Twall-Tsat)**2
{W/m^2-C}

{Uvalue}

```

UHX = (1/1000)/((ro/(ri*hi))+ (ro*ln(ro/ri)/kpipe) + (1/ho)) {W/m^2-C}
 {overall heat transfer coefficient}

END

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
 {calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
 {inside pipe}

vel = (mdot/DENSITYPG(percent,Tref))/(num*ri**2) {m/s}
 {velocity of Dowfrost through pipes}

Redi = vel*(2*ri)/(VISCOSITYPG(percent,Tref)/DENSITYPG(percent,Tref))
 {Reynolds number}

Pri = 1000*VISCOSITYPG(percent,Tref)*CPPG(percent,Tref)/
 THERMCONDPG(percent,Tref) {Prandtl number}
 {Nusselt number calculated using Hausen correlation for laminar flow and Nusselt
 correlation for turbulent flow}

if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
 else Nudi = 3.66 +

(0.668*(2*ri)/length)*Redi*Pri)/(1+0.4*((2*ri)/length)*Redi*Pri)**(2/3))

hi = NUDI*THERMCONDPG(percent,Tref)/(2*ri) {W/m^2-C}
 {inside heat transfer coefficient}

{outside pipes}

Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
 {Reynolds number}

Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
 {Prandtl number}

{Nusselt number using Churchill Bernstein correlation}

Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))*
 (1+(Redo/2.82e5)**(5/8))**(4/5)

ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-C}
 {outside heat transfer coefficient}

UCASE = (1/((ro/(ri*hi))+ (ro*ln(ro/ri)/kpipe) + (1/ho)))/1000 {W/m^2-C}
 {overall heat transfer coefficient}

END

{The equation set begins here}

{parameters}

Tairout = 267 {K} {temperature of air leaving refrigerated case}

caseload = 15{tons}*3.51685 {kW} {refrigeration load met by the case}

UAcond = 8.00 {kW/K} {UA value of the condenser}

Tamb = 300 {K} {ambient temperature}

Tstore = 300 {K} {store temperature}

{Compressor Parameters}

{dimensions of the first stage compressor}

borelow = 2.38*2.54e-2 {m}

strokelow = 1.75*2.54e-2 {m}

speedlow = 1700/60 {rev/sec}

clearratiolow = 0.05

{dimensions of the second stage compressor}

borehigh = $2.38 \times 2.54 \times 10^{-2}$ {m}

strokehigh = $1.75 \times 2.54 \times 10^{-2}$ {m}

speedhigh = 1700/60 {rev/sec}

clearratiohigh = 0.05

polyexp = 1.3 {polytropic exponent for ammonia}

polyeff = 0.65 {polytropic efficiency}

{Dowfrost Parameters}

mpg = 2.305 {kg/s} {mass flow rate of Dowfrost}

x = 0.54 {pressure ratio exponent}

diamc = 0.12 {m} {diameter of pipe to refrigerated case}

diamh = 0.12 {m} {diameter of pipe from refrigerated case}

insec = 0.025 {m} {thickness of insulation on pipe to case}

insh = 0.025 {m} {thickness of insulation on pipe from case}

kinsec = 0.04 {W/m-C} {thermal conductivity of insulation on pipe to case}

kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipe from case}

rough = 0.000046 {m} {roughness of pipe material}

lengthc = 30 {m} {length of pipe to case}

lengthh = 30 {m} {length of pipe from case}

gravity = 9.81 {m/s²} {acceleration due to gravity}

diamctotal = diamc + 2*insec {m} {total diameter of pipe and insulation to case}

diamhtotal = diamh + 2*insh {m} {total diameter of pipe and insulation from case}

Kpipec = 8.0 {for minor losses}

Kpipeh = 8.0 {for minor losses}

{Heat Exchanger Parameters}

numhxp = 300 {number of pipes in ammonia-Dowfrost heat exchanger}

rihx = 0.005 {m} {inside radius of pipes in ammonia-Dowfrost heat exchanger}

thickhx = 0.0015 {m} {thickness of pipes in ammonia-Dowfrost heat exchanger}

rohx = rihx + thickhx {m}

{outside radius of pipes in ammonia-Dowfrost heat exchanger}

lenhx = 2.0 {m} {length of pipes in ammonia-Dowfrost heat exchanger}

kpipehx = 43 {W/m-C}

{thermal conductivity of pipes in ammonia-Dowfrost heat exchanger}

Khx = 8*numhxp + 1.5 {for minor losses}

numcase = 700 {number of pipes in refrigerated case heat exchanger}

ricase = 0.01 {m} {inside radius of pipes in refrigerated case heat exchanger}

thickcase = 0.0015 {m} {thickness of pipes in refrigerated case heat exchanger}

rocase = ricase + thickcase {m}

{outside radius of pipes in refrigerated case heat exchanger}

lencase = 2.0 {m} {length of pipes in refrigerated case heat exchanger}

kpipecase = 43 {W/m²}

{thermal conductivity of pipes in refrigerated case heat exchanger}

airvel = 5.0 {m/s} {velocity of the air through the refrigerated case heat exchanger}

airarea = 0.5 {m²} {cross sectional area for air flow in refrigerated case heat exchanger}

Kcase = 8*numcase + 1.5 {for minor losses}

{Secondary Cycle}

{equations for the refrigerated case heat exchanger}

Tair = (Tairin + Tairout)/2 {K}

{bulk temperature of the air through the refrigerated case heat exchanger}

Tpg = (T6+T5)/2 {K}

{bulk temperature of Dowfrost through the refrigerated case heat exchanger}

Cpaircase = SpecHeat(Air,T=Tairin) {kJ/kg-K} {specific heat of air}

mdotair = (airvel/Volume(Air,T=Tairin,P=101.325))*airarea {kg/s} {mass flow of air}

Cpcase = CPPG(60,T5) {kJ/kg-K} {specific heat of Dowfrost}

Cair = Cpaircase*mdotair {kW/K} {capacitance rate of air stream}

Cpg = Cpcase*mpg {kW/K} {capacitance rate of Dowfrost stream}

Cmin = if(Cair,Cpg,Cair,Cair,Cpg) {minimum capacitance rate}

Cmax = if(Cair,Cpg,Cpg,Cair,Cair) {maximum capacitance rate}

Cr = Cmin/Cmax {ratio of the capacitance rates}

{calls function to calculate the overall heat transfer coefficient }

Ucasehx =

UCASE(numcase,ricase,rocase,lencase,mpg,airvel,airarea,Tpg,60,Tair,kpipecase)

{W/m²-K}

areacase = (numcase*2*pi*rocase*lencase) {m²} {total heat transfer area}

UAcase = Ucasehx*areacase {W/K} {UA value}

NTUcase = UAcase/Cmin {number of transfer units}

{calculation of the effectiveness using cross-flow correlation with both fluids unmixed}

effcase = 1-exp((1/Cr)*(NTUcase**0.22)*(exp(-Cr*(NTUcase**0.78))-1))

caseload = effcase*Cmin*(Tairin-T5)

{heat transferred in refrigerated case (refrigeration load met)}

caseload = Cair*(Tairin-Tairout) {change in temperature of the air stream}

caseload = Cpg*(T6-T5) {change in temperature of the Dowfrost stream}

{calls function to calculate pressure drop through the heat exchanger pipes}

deltaPcase = PRESSDROP(Tpg,mpg/numcase,2*ricase,rough,lencase,Kcase,gravity)

{Pa}

{calculates the pump work needed to overcome pressure drop in refrigerated case heat exchanger}

Wpumpcase = mpg*(deltaPcase/DENSITYPG(60,Tpg))/1000 {kW}

{Piping System}

Tc = (T8+T5)/2 {K} {bulk temperature of Dowfrost in pipes to case}

Th = (T6+T7)/2 {K} {bulk temperature of Dowfrost in pipes from case}

{calls function to calculate overall loss coefficient of pipes to refrigerated case}

Uc = UVAL(Tc,mpg,diamc,insc,kinsc) {W/m²-K}

{calls function to calculate overall loss coefficient of pipes from refrigerated case}

Uh = UVAL(Th,mpg,diamh,insh,kinsh) {W/m²-K}

{calls function to determine pressure drop in pipes to refrigerated case}

deltaPc = PRESSDROP(Tc,mpg,diamc,rough,lengthc,Kpipec,gravity) {Pa}

{calls function to determine pressure drop in pipes from refrigerated case}

deltaPh = PRESSDROP(Th,mpg,diamh,rough,lengthh,Kpipeh,gravity) {Pa}

densityc = DENSITYPG(60,Tc) {kg/m³} {density of Dowfrost in pipes to case}

densityh = DENSITYPG(60,Th) {kg/m³} {density of Dowfrost in pipes from case}


```

{ Pump work necessary to overcome pressure drop in pipes to and from case }
Wpumppipe = mpg*(deltaPc/densityc + deltaPh/densityh)/1000 {kW}

{ calculates thermal losses of piping system }
Ac = pi*diamctotal*lengthc {m^2} {total heat transfer area of pipes to case}
Ah = pi*diamhtotal*lengthh {m^2} {total heat transfer are of pipes from case}
Cpc = CPPG(60,Tc) {kJ/kg-K} {specific heat of Dowfrost in pipes to case}
Cph = CPPG(60,Th) {kJ/kg-K} {specific heat of Dowfrost in pipes from case}
Qc = mpg*Cpc*(T5-T8) {kW} {heat loss of Dowfrost stream in pipes to case}
Qh = mpg*Cph*(T7-T6) {kW} {heat loss of Dowfrost stream in pipes from case}
deltatlmc = (T5-T8)/(ln((Tstore-T8)/(Tstore-T5)))
{log mean temperature difference of pipes to case}
deltatlmh = (T7-T6)/(ln((Tstore-T6)/(Tstore-T7)))
{log mean temperature difference of pipes to case}
Qc = Uc*Ac*deltatlmc {kW} {heat loss from pipes to case}
Qh = Uh*Ah*deltatlmh {kW} {heat loss from pipes from case}
NTUc = Uc*Ac/(mpg*Cpc) {number of transfer units in flow to case}
NTUh = Uh*Ah/(mpg*Cph) {number of transfer units in flow from case}

{ equation for the heat exchanger between Dowfrost and Ammonia }
Twall = T7-(T7-T8)/2 {K} {surface temperature of pipes in the ammonia}
{calls function to calculate the overall heat transfer coefficient of heat exchanger}
U = UHX(numhxp,rihx,rohr,lenhx,mpg,T7,T8,Twall,T1,P1,60,kpipehx) {W/m^2-K}
areahx = (2*rohr*pi*lenhx*numhxp) {m^2} {total heat transfer area of heat exchanger}
UAhx = U*areahx {W/K} {UA value of the heat exchanger}
NTUhx = UAhx/(mpg*CPPG(60,T7)) {number of transfer units of heat exchanger}
effhx = 1-exp(-NTUhx) {effectiveness of the heat exchanger}
effhx = (T7-T8)/(T7-T1)
{using effectiveness to calculate the temperatures in the heat exchanger}
Qhx = mpg*CPPG(60,T7)*(T7-T8) {kW} {heat transfer in heat exchanger}
Tmid = (T7+T8)/2 {K} {bulk temperature in Dowfrost in heat exchanger}
T1 = T2 {K} {constant ammonia temperature in heat exchanger}
mref = Qhx/(h2-h1) {kg/s}
{calculates necessary ammonia mass flow rate to meet the load}
{calls function to determine pressure drop of flow through the heat exchanger}
deltaPhx = PRESSDROP(Tmid,mpg/numhxp,2*rihx,rough,lenhx,Khx,gravity) {Pa}
{calculates pump work needed to overcome pressure drop in heat exchanger}
Wpumphx = mpg*(deltaPhx/DENSITYPG(60,Tmid))/1000 {kW}

P2 = Pressure(AmmoniaTim,T=T1,x=1.0) {kPa}
{pressure of ammonia leaving heat exchanger}
h2 = Enthalpy(AmmoniaTim,T=T2,x=1.0) {kJ/kg}
{enthalpy of ammonia leaving heat exchanger}
s2 = Entropy(AmmoniaTim,T=T2,x=1.0) {kJ/kg-K}
{entropy of ammonia leaving heat exchanger}
P1 = P2 {kPa} {ammonia pressure constant in heat exchanger}

{ Ammonia Cycle }
{ compressor }

```

$rp = P3/P2$ {pressure ratio of evaporator to condenser}
 $rps = rp^x$ {exponent used to calculate the pressure ratio of first stage of compression}
 $rps = P2a/P2$ {calculates intermediate temperature}

{Low Pressure Compressor}

$v2 = \text{Volume}(\text{AmmoniaTim}, P=P2, x=1.0)$ {m³/kg}
 {specific volume of ammonia entering first compressor}
 $P2*(v2)^{\text{polyexp}} = P2a*(v2a)^{\text{polyexp}}$ {polytropic compression equation}
 $h2a = \text{Enthalpy}(\text{AmmoniaTim}, P=P2a, v=v2a)$ {kJ/kg}
 {enthalpy of ammonia leaving first compressor}
 $T2a = \text{Temperature}(\text{AmmoniaTim}, P=P2a, v=v2a)$ {K}
 {temperature of ammonia leaving first compressor}
 $V\text{displow} = \text{numcyllo}*(\pi*((\text{borelow}/2)^2)*\text{strokelow})$ {m³}
 {displacement volume of first compressor}
 $\text{volefflow} = 1 - (\text{clearratio} / (1 - \text{clearratio})) * ((P2a/P2)^{(1/\text{polyexp})} - 1)$
 {volumetric efficiency of first compressor}
 $m\text{ref} = (1/v2)*V\text{displow}*\text{speedlow}*\text{volefflow}$ {kg/s}
 {mass flow rate of ammonia through first compressor}
 $W\text{complow} = (m\text{ref}*(h2a-h2))/\text{polyeff}$ {kW} {work of first compressor}

{High Pressure Compressor}

$P2b = P2a$ {kPa} {pressure of ammonia entering second compressor same as pressure leaving first compressor}
 $h2b = \text{Enthalpy}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {kJ/kg}
 {enthalpy of ammonia entering second compressor}
 $s2b = \text{Entropy}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {kJ/kg-K}
 {entropy of ammonia entering second compressor}
 $T2b = \text{Temperature}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {K}
 {temperature of ammonia entering second compressor}
 $v2b = \text{Volume}(\text{AmmoniaTim}, P=P2b, x=1.0)$ {m³/kg}
 {specific volume of ammonia entering second compressor}
 $P2b*(v2b)^{\text{polyexp}} = P3*(v3)^{\text{polyexp}}$ {polytropic compression equation}
 $h3 = \text{Enthalpy}(\text{AmmoniaTim}, P=P3, v=v3)$ {kJ/kg}
 {enthalpy leaving second compressor}
 $T3 = \text{Temperature}(\text{AmmoniaTim}, P=P3, v=v3)$ {K}
 {temperature leaving second compressor}
 $V\text{disphigh} = \text{numcylhigh}*(\pi*((\text{borehigh}/2)^2)*\text{strokehigh})$ {m³}
 {displacement volume of second compressor}
 $\text{voleffhigh} = 1 - (\text{clearratiohigh} / (1 - \text{clearratiohigh})) * ((P3/P2b)^{(1/\text{polyexp})} - 1)$
 {volumetric efficiency of second compressor}
 $m\text{cond} = (1/v2b)*V\text{disphigh}*\text{speedhigh}*\text{voleffhigh}$ {kg/s}
 {mass flow rate of ammonia through second compressor}
 $W\text{comphigh} = (m\text{cond}*(h3-h2b))/\text{polyeff}$ {kW} {work of second compressor}
 $P3 = P4$ {kPa} {constant pressure in condenser}
 $W\text{comp} = W\text{complow} + W\text{comphigh}$ {kW} {total compressor work}

{condensor}

$m\text{air} = 15*0.47879$ {kg/s} {mass flow rate of air through the condenser}
 $C\text{pair} = \text{SpecHeat}(\text{Air}, T=T\text{amb})$ {kJ/kg-K} {specific heat of air in condenser}

```

UAcond = mair*Cpair*NTU {calculates the number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kw} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairoutcond-Tamb)
{calculates temperature change of air in the condenser}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa}
{pressure of ammonia leaving the condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg}
{enthalpy of ammonia leaving the condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=0.0) {kJ/kg}
{enthalpy of saturated liquid ammonia leaving flash tank}
h1 = h5 {kJ/kg} {isenthalpic expansion}

Wpump = Wpumpcase+Wpumphx+Wpumppipe {kW}
{total pump work done in system}
COP = caseload/(Wcomp+Wpump) {coefficient of performance of the system}
Cap = mpg*CPPG(60,T7)/caseload {1/K} {capacitance rate - caseload ratio}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

```

MULTITHERM 503 MODEL

{ Ammonia with Multitherm 503 system model }

FUNCTION VISCOSITYMO(T)

{calculates the viscosity of Multitherm503, a mineral oil solution, given temperatures in Kelvin}

$B = 94.385773521 - 0.93404292344*T + 0.0032867668094*T^2 - 5.1844790662e-6*T^3 + 3.055440215e-9*T^4$

VISCOSITYMO = exp(B) {Pa s}

END

FUNCTION THERMCONDMO(T)

{calculates the thermal conductivity of Multitherm503, a mineral oil solution, given temperatures in Kelvin}

THERMCONDMO = $0.17183 - 9.9229e-5*T$ {W/m-K}

END

FUNCTION DENSITYMO(T)

{calculates the density of Multitherm503, a mineral oil solution, given temperatures in Kelvin}

DENSITYMO = $994.31 - 0.68367 \cdot T$ {kg/m³}

END

FUNCTION CPMO(T)

{calculates the specific heat of Multitherm503, a mineral oil solution, given temperatures in Kelvin}

CPMO = $1.1729 + 0.0034758 \cdot T$ {kJ/kg-K}

END

FUNCTION UVAL(T,mmo,diam,ins,kins)

{calculates the U-value of the Multitherm pipe}

diamtotal = diam + 2*ins {m} {total outside diameter of pipe and insulation}

Area = $\pi \cdot (\text{diam}/2)^2$ {m²} {total heat transfer area}

vel = (mmo/DENSITYMO(T))/Area {m/s} {velocity of Multitherm in pipes}

kvis = VISCOSITYMO(T)/DENSITYMO(T) {m²/s} {kinematic viscosity}

Re = vel*diam/kvis {Reynolds number}

Pr = $1000 \cdot \text{VISCOSITYMO}(T) \cdot \text{CPMO}(T) / \text{THERMCONDMO}(T)$ {Prandtl number}

Nui = $0.023 \cdot (\text{Re}^{0.8}) \cdot (\text{Pr}^{0.4})$ {Dittus-Boelter correlation for Nusselt number}

hi = Nui*THERMCONDMO(T)/diam {W/m²-K} {inside heat transfer coefficient}

ho = 6 {W/m²-K} {outside heat transfer coefficient}

UVAL = $(1/1000) / ((\text{diamtotal}/(\text{diam} \cdot \text{hi})) + ((\text{diamtotal}/2) \cdot \ln(\text{diamtotal}/\text{diam})/\text{kins}) + (1/\text{ho}))$
{W/m²-K} {overall heat transfer coefficient}

END

FUNCTION PRESSDROP(T,mmo,diam,rough,length,Keq,gravity)

{calculates the pressure drop through pipes}

Area = $\pi \cdot (\text{diam}/2)^2$ {m²} {inside area of pipes}

vel = (mmo/DENSITYMO(T))/Area {m/s} {velocity of Multitherm in pipes}

kvis = VISCOSITYMO(T)/DENSITYMO(T) {m²/s} {kinematic viscosity}

Re = vel*diam/kvis {Reynolds number}

{calculates the friction factor from Moody diagram using Haarland correlation}

Reynolds numbers less than 2300 considered laminar}

if (Re<2300) then ff = 64/Re else ff = $(1/(-1.8 \cdot \log_{10}((6.9/\text{Re}) + ((\text{rough}/\text{diam})/3.7)^{1.1})))^{**2}$

headf = $\text{FF} \cdot (\text{length}/\text{diam}) \cdot (\text{vel}^2/(2 \cdot \text{gravity})) + \text{Keq} \cdot (\text{vel}^2)/(2 \cdot \text{gravity})$ {m}
{head losses}

PRESSDROP = headf*DENSITYMO(T)*gravity {Pa} {pressure drop}

END

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)

{calculates the overall heat transfer coefficient of the ammonia-Multitherm heat exchanger}

{inside pipes}

Tb = (Tin+Tout)/2 {K} {bulk temperature}

vel = (mdot/DENSITYMO(Tb))/(num*ri**2) {m/s} {velocity of Multitherm}

Red = $\text{vel} \cdot (2 \cdot \text{ri}) / (\text{VISCOSITYMO}(\text{Tb}) / \text{DENSITYMO}(\text{Tb}))$ {Reynolds number}

Pr = $1000 \cdot \text{VISCOSITYMO}(\text{Tb}) \cdot \text{CPMO}(\text{Tb}) / \text{THERMCONDMO}(\text{Tb})$

```

{ Prandtl number }
{ Nusslet number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow }
if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
Nud = 3.66 + (0.668*(2*ri/length)*Red*Pr)/(1+0.4*((2*ri/length)*Red*Pr)**(2/3))
hi = NUD*THERMCONDMO(Tb)/(2*ri) { W/m^2-K } { inside heat transfer coefficient }
{ outside pipes }
Csf = 0.01 { empirical constant }
hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
{ kJ/kg } { heat of fusion }

g = 9.8 { m/s^2 } { acceleration due to gravity }
sigma = 0.0213 { N/m } { surface tension }
Prl = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*
SpecHeat(AmmoniaTim,T=Tsat,P=Psat+1)/
Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) { Prandtl number }
Area = 2*pi*ro*length*num { m^2 } { total heat transfer area }
{ boiling heat transfer correlation using Rohsenow correlation }
ho = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*hfg*
((g*(1/Volume(AmmoniaTim,T=Tsat,x=0.0)-1/Volume(AmmoniaTim,T=Tsat,x=1.0))
/sigma)**0.5)*((SpecHeat(AmmoniaTim,T=Tsat,P=Psat-1)/(Csf*hfg*Prl**1.7))**3)
*(Twall-Tsat)**2 { W/m^2-K }
{ Uvalue }
UHX = (1/1000)/((ro/(ri*hi))+(ro*ln(ro/ri)/kpipe) + (1/ho)) { W/m^2-K }
{ overall heat transfer coefficient }

END

```

```

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
{ calculates the overall heat transfer coefficient of the refrigerated case heat exchanger }
{ inside pipe }
vel = (mdot/DENSITYMO(Tref))/(num*ri**2) { m/s } { velocity of Multitherm }
Redi = vel*(2*ri)/(VISCOSITYMO(Tref)/DENSITYMO(Tref)) { Reynolds number }
Pri = 1000*VISCOSITYMO(Tref)*CPMO(Tref)/THERMCONDMO(Tref)
{ Prandtl number }
{ Nusselt number calculated using Nausen correlation fpr laminar flow and Nusselt
correlation for turbulent flow }
if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
else Nudi = 3.66 +
(0.668*(2*ri/length)*Redi*Pri)/(1+0.4*((2*ri/length)*Redi*Pri)**(2/3))
hi = NUDI*THERMCONDMO(Tref)/(2*ri) { W/m^2-K }
{ inside heat transfer coefficient }
{ outside pipes }
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
{ Reynolds number }
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
{ Prandtl number }
{ Nusselt number calculated using Churchill-Bernstein correlation }
Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))
*(1+(Redo/2.82e5)**(5/8))**(4/5)

```

$ho = Nudo * Conductivity(Air, T=Tair) / (2 * ro)$ {W/m²-K}
 {outside heat transfer coefficient}
 $UCASE = (1 / ((ro / (ri * hi)) + (ro * \ln(ro / ri) / kpipe) + (1 / ho))) / 1000$ {W/m²-K}
 {overall heat transfer coefficient}
 END

{ The equation set begins here }

{ parameters }

Tairout = 267 {K} { temperature of the air leaving the refrigerated case }
 caseload = 15 {tons} * 3.51685 {kW} {refrigeration load met by the case}
 UAcond = 8.00 {kW/K} {UA value of the condenser}
 Tamb = 300 {K} {ambient temperature}
 Tstore = 300 {K} {store temperature}

{ Compressor Parameters }

{ dimensions of the first stage compressor }
 borelow = 2.38 * 2.54e-2 {m}
 strokelow = 1.75 * 2.54e-2 {m}
 speedlow = 1700 / 60 {rev/sec}
 clearratio_low = 0.05 {clearance ratio}
 { dimensions of the second stage compressor }
 borehigh = 2.38 * 2.54e-2 {m}
 strokehigh = 1.75 * 2.54e-2 {m}
 speedhigh = 1700 / 60 {rev/sec}
 clearratio_high = 0.05 {clearance ratio}
 polyexp = 1.3 {polytropic exponent}
 polyeff = 0.65 {polytropic efficiency}

{ Multitherm Parameters }

mmo = 2.305 {kg/s} {mass flow rate of Multitherm}
 x = 0.54 {pressure ratio exponent}
 diamc = 0.12 {m} {diameter of pipe to case}
 diamh = 0.12 {m} {diameter of pipe from case}
 insc = 0.025 {m} {thickness of insulation on pipe to case}
 insh = 0.025 {m} {thickness of insulation on pipe from case}
 kinsc = 0.04 {W/m-C} {thermal conductivity of insulation on pipe to case}
 kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipe from case}
 rough = 0.000046 {m} {pipe roughness}
 lengthc = 30 {m} {length of pipe to case}
 lengthh = 30 {m} {length of pipe from case}
 gravity = 9.81 {m/s²} {acceleration due to gravity}
 diamctotal = diamc + 2 * insc {m} {total diameter of pipe and insulation to case}
 diamhtotal = diamh + 2 * insh {m} {total diameter of pipe and insulation from case}
 Kpipec = 8.0 {for minor losses}
 Kpipeh = 8.0 {for minor losses}

{ Heat Exchanger Parameters }

numhxp = 300 {number of pipes in ammonia-Multitherm heat exchanger}

```

rihx = 0.005 {m} {inside radius of pipes in ammonia-Multitherm heat exchanger}
thickhx = 0.0015 {m} {thickness of pipes in ammonia-Multitherm heat exchanger}
rohx = rihx + thickhx {m}
      {outside radius of pipes in ammonia-Multitherm heat exchanger}
lenhx = 2.0 {m} {length of pipes in ammonia-Multitherm heat exchanger}
kpipehx = 43 {W/m-C}
      {thermal conductivity of pipes in ammonia-Multitherm heat exchanger}
Khx = 8*numhxp + 1.5 {for minor losses}
numcase = 700 {number of pipes in refrigerated case heat exchanger}
ricase = 0.01 {m} {inside radius of pipes in refrigerated case heat exchanger}
thickcase = 0.0015 {m} {thickness of pipes in refrigerated case heat exchanger}
rocase = ricase + thickcase {m}
      {outside radius of pipes in refrigerated case heat exchanger}
lencase = 2.0 {m} {length of pipes in refrigerated case heat exchanger}
airvel = 5.0 {m/s} {air velocity in refrigerated case heat exchanger}
airarea = 0.5 {m^2}
      {cross sectional area for air flow in refrigerated case heat exchanger}
kpipecase = 43 {W/m^2}
      {thermal conductivity of pipes in refrigerated case heat exchanger}
Kcase = 8*numcase + 1.5 {for minor losses}

{Secondary Cycle}
{equations for the refrigerated case heat exchanger}
Tair = (Tairin + Tairout)/2 {K} {bulk temperature of air in case heat exchanger}
TMO = (T6+T5)/2 {K} {bulk temperature of Multitherm in case heat exchanger}
Cpaircase = SpecHeat(Air,T=Tairin) {kJ/kg-K} {specific heat of air}
mdotair = (airvel/Volume(Air,T=Tairin,P=101.325))*airarea {kg/s} {mass flow of air}
Cpcase = CPMO(T5) {kJ/kg-K} {specific heat of Multitherm}
Cair = Cpaircase*mdotair {kW/K} {capacitance rate of air}
CMO = Cpcase*mmo {kW/K} {capacitance rate of Multitherm}
Cmin = if(Cair,CMO,Cair,Cair,CMO) {minimum capacitance rate}
Cmax = if(Cair,CMO,CMO,Cair,Cair) {maximum capacitance rate}
Cr = Cmin/Cmax {capacitance rate ratio}
{calls function to calculate the overall heat transfer coefficient} {W/m^2-K}
Ucasehx =
UCASE(numcase,ricase,rocase,lencase,mmo,airvel,airarea,TMO,60,Tair,kpipecase)
UAcase = Ucasehx*(numcase*2*pi*rocase*lencase) {W/K} {UA value of case}
NTUcase = UAcase/Cmin {number of transfer units of case heat exchanger}
{calculates the effectiveness using cross-flow correlation with both streams unmixed}
effcase = 1-exp((1/Cr)*(NTUcase**0.22)*(exp(-Cr*(NTUcase**0.78))-1))
caseload = effcase*Cmin*(Tairin-T5) {kW}
      {heat transfer in case (refrigeration load met)}
caseload = Cair*(Tairin-Tairout) {change in temperature of the air stream}
caseload = CMO*(T6-T5) {change in temperature of the Multitherm}
{calls function to calculate pressure drop through refrigerated case} {Pa}
deltaPcase = PRESSDROP(TMO,mmo/numcase,2*ricase,rough,lencase,Kcase,gravity)
{calculates the pump work needed to overcome pressure drop in case}
Wpumppcase = mmo*(deltaPcase/DENSITYMO(TMO))/1000 {kW}

```

{Piping System}

$T_c = (T_8 + T_5)/2$ {K} {bulk temperature of Multitherm in pipes to case}

$T_h = (T_6 + T_7)/2$ {K} {bulk temperature of Multitherm in pipes from case}

{calls function to calculate overall heat transfer coefficient of pipes to case}

$U_c = UVAL(T_c, mmo, diamc, insc, kinsc)$ {W/m²-K}

{calls function to calculate overall heat transfer coefficient of pipes from case}

$U_h = UVAL(T_h, mmo, diamh, insh, kinsh)$ {W/m²-K}

{calls function to calculate pressure drop of pipes to case}

$\Delta P_c = PRESSDROP(T_c, mmo, diamc, rough, lengthc, Kpipe_c, gravity)$ {Pa}

{calls function to calculate pressure drop of pipes to case}

$\Delta P_h = PRESSDROP(T_h, mmo, diamh, rough, lengthh, Kpipe_h, gravity)$ {Pa}

$density_c = DENSITYMO(T_c)$ {kg/m³} {density of Multitherm in pipes to case}

$density_h = DENSITYMO(T_h)$ {kg/m³} {density of Multitherm in pipes from case}

{calculates pump necessary to overcome pressure drop in piping system}

$W_{pump\ pipe} = mmo * (\Delta P_c / density_c + \Delta P_h / density_h) / 1000$ {kW}

{calculates thermal losses of piping system}

$A_c = \pi * diam_{total} * length_c$ {m²} {total heat transfer area of pipes to case}

$A_h = \pi * diam_{total} * length_h$ {m²} {total heat transfer area of pipes from case}

$C_{pc} = CPMO(T_c)$ {kJ/kg-K} {specific heat of Multitherm in pipes to case}

$C_{ph} = CPMO(T_h)$ {kJ/kg-K} {specific heat of Multitherm in pipes from case}

$Q_c = mmo * C_{pc} * (T_5 - T_8)$ {kW} {heat loss of Multitherm in pipes to case}

$Q_h = mmo * C_{ph} * (T_7 - T_6)$ {kW} {heat loss of Multitherm in pipes from case}

$\Delta T_{lmc} = (T_5 - T_8) / (\ln((T_{store} - T_8) / (T_{store} - T_5)))$

{log mean temperature difference of pipes to case}

$\Delta T_{lmh} = (T_7 - T_6) / (\ln((T_{store} - T_6) / (T_{store} - T_7)))$

{log mean temperature difference of pipes from case}

$Q_c = U_c * A_c * \Delta T_{lmc}$ {kW} {heat loss of pipes to case}

$Q_h = U_h * A_h * \Delta T_{lmh}$ {kW} {heat loss of pipes from case}

$NTU_c = U_c * A_c / (mmo * C_{pc})$ {number of transfer units of pipes to case}

$NTU_h = U_h * A_h / (mmo * C_{ph})$ {number of transfer units of pipes from case}

{equations for heat exchanger between Multitherm and Ammonia}

$T_{wall} = T_7 - (T_7 - T_8) / 2$ {K} {surface temperature of pipes in ammonia}

{calls function to calculate overall heat transfer coefficient} {W/m²-K}

$U = UHX(numhxp, rihx, rohx, lenhx, mmo, T_7, T_8, T_{wall}, T_1, P_1, 60, kpipe_hx)$

$UA_{hx} = U * (2 * rohx * \pi * lenhx * numhxp)$ {W/K} {UA value of heat exchanger}

$NTU_{hx} = UA_{hx} / (mmo * CPMO(T_7))$ {number of transfer units of heat exchanger}

$eff_{hx} = 1 - \exp(-NTU_{hx})$ {effectiveness of heat exchanger}

$eff_{hx} = (T_7 - T_8) / (T_7 - T_1)$

{using effectiveness to calculate temperatures in heat exchanger}

$Q_{hx} = mmo * CPMO(T_7) * (T_7 - T_8)$ {kW} {heat transfer in heat exchanger}

$T_{mid} = (T_7 + T_8) / 2$ {K} {bulk temperature of Multitherm in heat exchanger}

$T_1 = T_2$ {K} {constant ammonia temperature in heat exchanger}

$m_{ref} = Q_{hx} / (h_2 - h_1)$ {kg/s} {calculates necessary ammonia flow rate to meet load}

{calls function to calculate the pressure drop in heat exchanger}

$\Delta P_{hx} = PRESSDROP(T_{mid}, mmo / numhxp, 2 * rihx, rough, lenhx, Khx, gravity)$ {Pa}

{calculates pump work necessary to overcome pressure drop in heat exchanger}


```

Wpumphx = mmo*(deltaPhx/DENSITYMO(Tmid))/1000 {kW}

P2 = Pressure(AmmoniaTim,T=T1,x=1.0) {kPa}
                                     {pressure of ammonia leaving heat exchanger}
h2 = Enthalpy(AmmoniaTim,T=T2,x=1.0) {kJ/kg}
                                     {enthalpy of ammonia leaving heat exchanger}
s2 = Entropy(AmmoniaTim,T=T2,x=1.0) {kJ/kg-K}
                                     {entropy of ammonia leaving heat exchanger}
P1 = P2 {kPa} {ammonia pressure constant in heat exchanger}

{ Ammonia Cycle}
{compressor}
rp = P3/P2 {pressure ratio of evaporator to condenser}
rps = rp^x {pressure ratio relationship}
rps = P2a/P2 {calculates intermediate pressure}

{Low Pressure Compressor}
v2 = Volume(AmmoniaTim,P=P2,x=1.0) {m^3/kg}
                                     {specific volume of ammonia entering first compressor}
P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}
h2a = Enthalpy(AmmoniaTim,P=P2a,v=v2a) {kJ/kg}
                                     {enthalpy of ammonia leaving first compressor}
T2a = Temperature(AmmoniaTim,P=P2a,v=v2a) {K}
                                     {temperature of ammonia leaving first compressor}
Vdisplow = numcyllo*(pi*((borelow/2)^2)*strokelow) {m^3}
                                     {displacement volume of first compressor}
volefflow = 1-(clearratiolow/(1-clearratiolow))*((P2a/P2)^(1/polyexp) - 1)
                                     {volumetric efficiency of first compressor}
mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}
                                     {mass flow of ammonia through first compressor}
Wcomplow = (mref*(h2a-h2))/polyeff {kW} {work of first compressor}

{High Pressure Compressor}
P2b = P2a {kPa} {intermediate pressure}
h2b = Enthalpy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg}
                                     {enthalpy of ammonia entering second compressor}
s2b = Entropy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg-K}
                                     {entropy of ammonia entering second compressor}
T2b = Temperature(AmmoniaTim,P=P2b,x=1.0) {K}
                                     {temperature of ammonia entering second compressor}
v2b = Volume(AmmoniaTim,P=P2b,x=1.0) {m^3/kg}
                                     {specific heat of ammonia entering second compressor}
P2b*(v2b)^polyexp = P3*(vol3)^polyexp {polytropic compression equation}
h3 = Enthalpy(AmmoniaTim,P=P3,v=vol3) {kJ/kg}
                                     {enthalpy of ammonia leaving second compressor}
T3 = Temperature(AmmoniaTim,P=P3,v=vol3) {K}
                                     {temperature of ammonia leaving second compressor}
Vdisphigh = numcylhigh*(pi*((borehigh/2)^2)*strokehigh) {m^3}
                                     {displacement volume of second compressor}

```

```

voleffhigh = 1-(clearratiohigh/(1-clearratiohigh))*((P3/P2b)^(1/polyexp) - 1)
                                     {volumetric efficiency of second compressor}
mcond = (1/v2b)*Vdisphigh*speedhigh*voleffhigh {kg/s}
                                     {mass flow of ammonia through second compressor}
Wcomphigh = (mcond*(h3-h2b))/polyeff {kW} {work of second compressor}
P3 = P4 {kPa} {constant pressure across condenser}
Wcomp = Wcomplow + Wcomphigh {kW} {total compression work}

{condensor}
mair = 15*0.47879 {kg/s} {flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg-K} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairoutcond-Tamb) {calculates temperature change of air}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa}
                                     {pressure of ammonia leaving condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg}
                                     {enthalpy of ammonia leaving condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=0.0) {kJ/kg}
                                     {enthalpy of saturated liquid ammonia leaving the flash tank}
h1 = h5 {kJ/kg} {isenthalpic expansion}

Wpump = Wpumpcase+Wpumphpx+Wpumppipe {kW}
                                     {total pump work done by system}
COP = caseload/(Wcomp+Wpump) {coefficient of performance of refrigeration system}
Cap = mmo*CPMO(T7)/caseload {1/K} {capacitance rate - caseload ratio}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

```

SYLTHERM XLT MODEL

{ Ammonia with Syltherm XLT system model}

FUNCTION THERMCONDST(T)

{calculates the thermal conductivity of Syltherm given temperatures in Kelvin}

THERMCONDST = 0.12569 - 6.5011e-5*T {W/m-K}

END

FUNCTION VISCOSITYST(T)

{calculates the viscosity of Syltherm given temperatures in Kelvin}

$B = 5.123113 - 0.07046865 \cdot T + 0.0001307815 \cdot T^2 - 8.7607645e-8 \cdot T^3$

VISCOSITYST = $\exp(B)$ {Pa s}

END

FUNCTION DENSITYST(T)

{calculates the density of Syltherm given temperatures in Kelvin}

DENSITYST = $1001.1504 - 0.1660318 \cdot T - 0.00122623 \cdot T^2$ {kg/m³}

END

FUNCTION CPST(T)

{calculates the specific heat of Syltherm given temperatures in Kelvin}

CPST = $0.89721 + 0.0025653 \cdot T$ {kJ/kg-K}

END

FUNCTION UVAL(T,mst,diam,ins,kins)

{calculates the U-value of the Syltherm pipe}

diamtotal = diam + 2*ins {m} {total outside diameter of pipe and insulation}

Area = $\pi \cdot (\text{diam}/2)^2$ {m²} {total heat transfer area}

vel = (mst/DENSITYST(T))/Area {m/s} {velocity of Syltherm in pipes}

kvis = VISCOSITYST(T)/DENSITYST(T) {m²/s} {kinematic viscosity}

Re = vel*diam/kvis {Reynolds number}

Pr = $1000 \cdot \text{VISCOSITYST}(T) \cdot \text{CPST}(T) / \text{THERMCONDST}(T)$ {Prandtl number}

Nui = $0.023 \cdot (\text{Re}^{0.8}) \cdot (\text{Pr}^{0.4})$ {Dittus-Boelter correlation for Nusselt number}

hi = Nui*THERMCONDST(T)/diam {W/m²-K} {inside heat transfer coefficient}

ho = 6 {W/m²-K} {outside heat transfer coefficient}

UVAL = $(1/1000) / ((\text{diamtotal}/(\text{diam} \cdot \text{hi})) + ((\text{diamtotal}/2) \cdot \ln(\text{diamtotal}/\text{diam})/\text{kins}) + (1/\text{ho}))$
{W/m²-K} {overall heat transfer coefficient}

END

FUNCTION PRESSDROP(T,mst,diam,rough,length,Keq,gravity)

{calculates the pressure drop through pipes}

Area = $\pi \cdot (\text{diam}/2)^2$ {m²} {inside area of pipes}

vel = (mst/DENSITYST(T))/Area {m/s} {velocity of Syltherm in pipes}

kvis = VISCOSITYST(T)/DENSITYST(T) {m²/s} {kinematic viscosity}

Re = vel*diam/kvis {Reynolds number}

{calculates the friction factor from Moody diagram using Haaland correlation}

Reynolds numbers less than 2300 considered laminar}

if (Re<2300) then ff = 64/Re else ff = $(1/(-1.8 \cdot \log_{10}((6.9/\text{Re}) + ((\text{rough}/\text{diam})/3.7)^{1.11})))^{0.25}$

headf = $\text{FF} \cdot (\text{length}/\text{diam}) \cdot (\text{vel}^2/(2 \cdot \text{gravity})) + \text{Keq} \cdot (\text{vel}^2)/(2 \cdot \text{gravity})$ {m}
{head losses}

PRESSDROP = headf*DENSITYST(T)*gravity {Pa} {pressure drop}

END

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)

{calculates the overall heat transfer coefficient of the ammonia-Syltherm heat exchanger}
{inside pipes}

```

Tb = (Tin+Tout)/2 {K} {bulk temperature}
vel = (mdot/DENSITYST(Tb))/(num*ri**2) {m/s} {velocity of Syltherm}
Red = vel*(2*ri)/(VISCOSITYST(Tb)/DENSITYST(Tb)) {Reynolds number}
Pr = 1000*VISCOSITYST(Tb)*CPST(Tb)/THERMCONDST(Tb) {Prandtl number}
{Nusselt number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow}
if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
Nud = 3.66 + (0.668*(2*ri/length)*Red*Pr)/(1+0.4*((2*ri/length)*Red*Pr)**(2/3))
hi = NUD*THERMCONDST(Tb)/(2*ri) {W/m^2-K} {inside heat transfer coefficient}
{outside pipes}
Csf = 0.01 {empirical constant}
hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
{kJ/kg} {heat of fusion}

g = 9.8 {m/s^2} {acceleration due to gravity}
sigma = 0.0213 {N/m} {surface tension}
Pr1 = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*
SpecHeat(AmmoniaTim,T=Tsat,P=Psat+1)/
Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) {Prandtl number}
Area = 2*pi*ro*length*num {m^2} {total heat transfer area}
{boiling heat transfer correlation using Rohsenow correlation}
ho = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*hfg*((g*
(1/Volume(AmmoniaTim,T=Tsat,x=0.0)-1/Volume(AmmoniaTim,T=Tsat,x=1.0))/
sigma)**0.5)*((SpecHeat(AmmoniaTim,T=Tsat,P=Psat-1)/(Csf*hfg*Pr1**1.7))**3)
*(Twall-Tsat)**2 {W/m^2-K}
{Uvalue}
UHX = (1/1000)/((ro/(ri*hi))+(ro*ln(ro/ri)/kpipe) + (1/ho)) {W/m^2-K}
{overall heat transfer coefficient}

END

```

```

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
{calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
{inside pipe}
vel = (mdot/DENSITYST(Tref))/(num*ri**2) {m/s} {velocity of Syltherm}
Redi = vel*(2*ri)/(VISCOSITYST(Tref)/DENSITYST(Tref)) {Reynolds number}
Pri = 1000*VISCOSITYST(Tref)*CPST(Tref)/THERMCONDST(Tref)
{Prandtl number}
{Nusselt number calculated using Nausen correlation fpr laminar flow and Nusselt
correlation for turbulent flow}
if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
else Nudi = 3.66 +
(0.668*(2*ri/length)*Redi*Pri)/(1+0.4*((2*ri/length)*Redi*Pri)**(2/3))
hi = NUDI*THERMCONDST(Tref)/(2*ri) {W/m^2-K}
{inside heat transfer coefficient}
{outside pipes}
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
{Reynolds number}
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
{Prandtl number}
{Nusselt number calculated using Churchill-Bernstein correlation}

```

```

Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))
*(1+(Redo/2.82e5)**(5/8))**(4/5)
ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-K}
                                         {outside heat transfer coefficient}
UCASE = (1/((ro/(ri*hi)))+(ro*ln(ro/ri)/kpipe) + (1/ho))/1000 {W/m^2-K}
                                         {overall heat transfer coefficient}
END

```

{The equation set begins here}

{parameters}

```

Tairout = 267 {K} {temperature of the air leaving the refrigerated case}
caseload = 15{tons}*3.51685 {kW} {refrigeration load met by the case}
UAcond = 8.00 {kW/K} {UA value of the condenser}
Tamb = 300 {K} {ambient temperature}
Tstore = 300 {K} {store temperature}

```

{Compressor Parameters}

{dimensions of the first stage compressor}

```

borelow = 2.38*2.54e-2 {m}
strokelow = 1.75*2.54e-2 {m}
speedlow = 1700/60 {rev/sec}
clearratiolow = 0.05 {clearance ratio}
{dimensions of the second stage compressor}
borehigh = 2.38*2.54e-2 {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rev/sec}
clearratiohigh = 0.05 {clearance ratio}
polyexp = 1.3 {polytropic exponent}
polyeff = 0.65 {polytropic efficiency}

```

{Syltherm Parameters}

```

mst = 2.305 {kg/s} {mass flow rate of Syltherm}
x = 0.54 {pressure ratio exponent}
diamc = 0.12 {m} {diameter of pipe to case}
diamh = 0.12 {m} {diameter of pipe from case}
insh = 0.025 {m} {thickness of insulation on pipe to case}
insh = 0.025 {m} {thickness of insulation on pipe from case}
kinsc = 0.04 {W/m-C} {thermal conductivity of insulation on pipe to case}
kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipe from case}
rough = 0.000046 {m} {pipe roughness}
lengthc = 30 {m} {length of pipe to case}
lengthh = 30 {m} {length of pipe from case}
gravity = 9.81 {m/s^2} {acceleration due to gravity}
diamctotal = diamc + 2*insh {m} {total diameter of pipe and insulation to case}
diamhtotal = diamh + 2*insh {m} {total diameter of pipe and insulation from case}
Kpipec = 8.0 {for minor losses}
Kpipeh = 8.0 {for minor losses}

```

```

{Heat Exchanger Parameters}
numhxp = 300 {number of pipes in ammonia-Syltherm heat exchanger}
rihx = 0.005 {m} {inside radius of pipes in ammonia-Syltherm heat exchanger}
thickhx = 0.0015 {m} {thickness of pipes in ammonia-Syltherm heat exchanger}
rohx = rihx + thickhx {m} {outside radius of pipes in ammonia-Syltherm heat exchanger}
lenhx = 2.0 {m} {length of pipes in ammonia-Syltherm heat exchanger}
kpipehx = 43 {W/m-C}
           {thermal conductivity of pipes in ammonia-Syltherm heat exchanger}
Khx = 8*numhxp + 1.5 {for minor losses}
numcase = 700 {number of pipes in refrigerated case heat exchanger}
ricase = 0.01 {m} {inside radius of pipes in refrigerated case heat exchanger}
thickcase = 0.0015 {m} {thickness of pipes in refrigerated case heat exchanger}
rocase = ricase + thickcase {m}
           {outside radius of pipes in refrigerated case heat exchanger}
lencase = 2.0 {m} {length of pipes in refrigerated case heat exchanger}
airvel = 5.0 {m/s} {air velocity in refrigerated case heat exchanger}
airarea = 0.5 {m^2}
           {cross sectional area for air flow in refrigerated case heat exchanger}
kpipecase = 43 {W/m^2}
           {thermal conductivity of pipes in refrigerated case heat exchanger}
Kcase = 8*numcase + 1.5 {for minor losses}

{Secondary Cycle}
{equations for the refrigerated case heat exchanger}
Tair = (Tairin + Tairout)/2 {K} {bulk temperature of air in case heat exchanger}
TST = (T6+T5)/2 {K} {bulk temperature of Syltherm in case heat exchanger}
Cpaircase = SpecHeat(Air,T=Tairin) {kJ/kg-K} {specific heat of air}
mdotair = (airvel/Volume(Air,T=Tairin,P=101.325))*airarea {kg/s} {mass flow of air}
Cpcase = CPST(T5) {kJ/kg-K} {specific heat of Syltherm}
Cair = Cpaircase*mdotair {kW/K} {capacitance rate of air}
CST = Cpcase*mst {kW/K} {capacitance rate of Syltherm}
Cmin = if(Cair,CST,Cair,Cair,CST) {minimum capacitance rate}
Cmax = if(Cair,CST,CST,Cair,Cair) {maximum capacitance rate}
Cr = Cmin/Cmax {capacitance rate ratio}
{calls function to calculate the overall heat transfer coefficient} {W/m^2-K}
Ucasehx =
UCASE(numcase,ricase,rocase,lencase,mst,airvel,airarea,TST,60,Tair,kpipecase)
UAcase = Ucasehx*(numcase*2*pi*rocase*lencase) {W/K} {UA value of case}
NTUcase = UAcase/Cmin {number of transfer units of case heat exchanger}
{calculates the effectiveness using cross-flow correlation with both streams unmixed}
effcase = 1-exp((1/Cr)*(NTUcase**0.22)*(exp(-Cr*(NTUcase**0.78))-1))
caseload = effcase*Cmin*(Tairin-T5) {kW}
           {heat transfer in case (refrigeration load met)}
caseload = Cair*(Tairin-Tairout) {change in temperature of the air stream}
caseload = CST*(T6-T5) {change in temperature of the Syltherm}
{calls function to calculate pressure drop through refrigerated case} {Pa}
deltaPcase = PRESSDROP(TST,mst/numcase,2*ricase,rough,lencase,Kcase,gravity)
{calculates the pump work needed to overcome pressure drop in case}

```

$$W_{pumpcase} = mst * (\Delta P_{case} / \text{DENSITYST}(TST)) / 1000 \quad \{kW\}$$

{Piping System}

$$T_c = (T_8 + T_5) / 2 \quad \{K\} \quad \{\text{bulk temperature of Syltherm in pipes to case}\}$$

$$T_h = (T_6 + T_7) / 2 \quad \{K\} \quad \{\text{bulk temperature of Syltherm in pipes from case}\}$$

{calls function to calculate overall heat transfer coefficient of pipes to case}

$$U_c = \text{UVAL}(T_c, mst, diam_c, insc, kinsc) \quad \{W/m^2-K\}$$

{calls function to calculate overall heat transfer coefficient of pipes from case}

$$U_h = \text{UVAL}(T_h, mst, diam_h, insh, kinsh) \quad \{W/m^2-K\}$$

{calls function to calculate pressure drop of pipes to case}

$$\Delta P_c = \text{PRESSDROP}(T_c, mst, diam_c, rough, length_c, K_{pipe_c}, gravity) \quad \{Pa\}$$

{calls function to calculate pressure drop of pipes to case}

$$\Delta P_h = \text{PRESSDROP}(T_h, mst, diam_h, rough, length_h, K_{pipe_h}, gravity) \quad \{Pa\}$$

$$\text{density}_c = \text{DENSITYST}(T_c) \quad \{kg/m^3\} \quad \{\text{density of Syltherm in pipes to case}\}$$

$$\text{density}_h = \text{DENSITYST}(T_h) \quad \{kg/m^3\} \quad \{\text{density of Syltherm in pipes from case}\}$$

{calculates pump necessary to overcome pressure drop in piping system}

$$W_{pumppipe} = mst * (\Delta P_c / \text{density}_c + \Delta P_h / \text{density}_h) / 1000 \quad \{kW\}$$

{calculates thermal losses of piping system}

$$A_c = \pi * diam_{ctotal} * length_c \quad \{m^2\} \quad \{\text{total heat transfer area of pipes to case}\}$$

$$A_h = \pi * diam_{htotal} * length_h \quad \{m^2\} \quad \{\text{total heat transfer area of pipes from case}\}$$

$$C_{pc} = \text{CPST}(T_c) \quad \{kJ/kg-K\} \quad \{\text{specific heat of Syltherm in pipes to case}\}$$

$$C_{ph} = \text{CPST}(T_h) \quad \{kJ/kg-K\} \quad \{\text{specific heat of Syltherm in pipes from case}\}$$

$$Q_c = mst * C_{pc} * (T_5 - T_8) \quad \{kW\} \quad \{\text{heat loss of Syltherm in pipes to case}\}$$

$$Q_h = mst * C_{ph} * (T_7 - T_6) \quad \{kW\} \quad \{\text{heat loss of Syltherm in pipes from case}\}$$

$$\Delta T_{lm_c} = (T_5 - T_8) / (\ln((T_{store} - T_8) / (T_{store} - T_5)))$$

{log mean temperature difference of pipes to case}

$$\Delta T_{lm_h} = (T_7 - T_6) / (\ln((T_{store} - T_6) / (T_{store} - T_7)))$$

{log mean temperature difference of pipes from case}

$$Q_c = U_c * A_c * \Delta T_{lm_c} \quad \{kW\} \quad \{\text{heat loss of pipes to case}\}$$

$$Q_h = U_h * A_h * \Delta T_{lm_h} \quad \{kW\} \quad \{\text{heat loss of pipes from case}\}$$

$$NTU_c = U_c * A_c / (mst * C_{pc}) \quad \{\text{number of transfer units of pipes to case}\}$$

$$NTU_h = U_h * A_h / (mst * C_{ph}) \quad \{\text{number of transfer units of pipes from case}\}$$

{equations for heat exchanger between Syltherm and Ammonia}

$$T_{wall} = T_7 - (T_7 - T_8) / 2 \quad \{K\} \quad \{\text{surface temperature of pipes in ammonia}\}$$

{calls function to calculate overall heat transfer coefficient} {W/m^2-K}

$$U = \text{UHX}(\text{numhxp}, \text{rihx}, \text{rohx}, \text{lenhx}, mst, T_7, T_8, T_{wall}, T_1, P_1, 60, k_{pipehx})$$

$$UA_{hx} = U * (2 * rohx * \pi * lenhx * \text{numhxp}) \quad \{W/K\} \quad \{UA \text{ value of heat exchanger}\}$$

$$NTU_{hx} = UA_{hx} / (mst * \text{CPST}(T_7)) \quad \{\text{number of transfer units of heat exchanger}\}$$

$$\text{effhx} = 1 - \exp(-NTU_{hx}) \quad \{\text{effectiveness of heat exchanger}\}$$

$$\text{effhx} = (T_7 - T_8) / (T_7 - T_1)$$

{using effectiveness to calculate temperatures in heat exchanger}

$$Q_{hx} = mst * \text{CPST}(T_7) * (T_7 - T_8) \quad \{kW\} \quad \{\text{heat transfer in heat exchanger}\}$$

$$T_{mid} = (T_7 + T_8) / 2 \quad \{K\} \quad \{\text{bulk temperature of Syltherm in heat exchanger}\}$$

$$T_1 = T_2 \quad \{K\} \quad \{\text{constant ammonia temperature in heat exchanger}\}$$

$$m_{ref} = Q_{hx} / (h_2 - h_1) \quad \{kg/s\} \quad \{\text{calculates necessary ammonia flow rate to meet load}\}$$

{calls function to calculate the pressure drop in heat exchanger}

deltaPhx = PRESSDROP(Tmid,mst/numhxp,2*rihx,rough,lenhx,Khx,gravity) {Pa}
 {calculates pump work necessary to overcome pressure drop in heat exchanger}
 Wpumphx = mst*(deltaPhx/DENSITYST(Tmid))/1000 {kW}

P2 = Pressure(AmmoniaTim,T=T1,x=1.0) {kPa}
 {pressure of ammonia leaving heat exchanger}
 h2 = Enthalpy(AmmoniaTim,T=T2,x=1.0) {kJ/kg}
 {enthalpy of ammonia leaving heat exchanger}
 s2 = Entropy(AmmoniaTim,T=T2,x=1.0) {kJ/kg-K}
 {entropy of ammonia leaving heat exchanger}
 P1 = P2 {kPa} {ammonia pressure constant in heat exchanger}

{ Ammonia Cycle}
 {compressor}
 rp = P3/P2 {pressure ratio of evaporator to condenser}
 rps = rp^x {pressure ratio relationship}
 rps = P2a/P2 {calculates intermediate pressure}

{ Low Pressure Compressor}
 v2 = Volume(AmmoniaTim,P=P2,x=1.0) {m^3/kg}
 {specific volume of ammonia entering first compressor}
 P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}
 h2a = Enthalpy(AmmoniaTim,P=P2a,v=v2a) {kJ/kg}
 {enthalpy of ammonia leaving first compressor}
 T2a = Temperature(AmmoniaTim,P=P2a,v=v2a) {K}
 {temperature of ammonia leaving first compressor}
 Vdisplow = numcyllo*(pi*((borelow/2)^2)*strokelow) {m^3}
 {displacement volume of first compressor}
 volefflow = 1-(clearratilow/(1-clearratilow))*((P2a/P2)^(1/polyexp) - 1)
 {volumetric efficiency of first compressor}
 mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}
 {mass flow of ammonia through first compressor}
 Wcomplow = (mref*(h2a-h2))/polyeff {kW} {work of first compressor}

{ High Pressure Compressor}
 P2b = P2a {kPa} {intermediate pressure}
 h2b = Enthalpy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg}
 {enthalpy of ammonia entering second compressor}
 s2b = Entropy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg-K}
 {entropy of ammonia entering second compressor}
 T2b = Temperature(AmmoniaTim,P=P2b,x=1.0) {K}
 {temperature of ammonia entering second compressor}
 v2b = Volume(AmmoniaTim,P=P2b,x=1.0) {m^3/kg}
 {specific heat of ammonia entering second compressor}
 P2b*(v2b)^polyexp = P3*(vol3)^polyexp {polytropic compression equation}
 h3 = Enthalpy(AmmoniaTim,P=P3,v=vol3) {kJ/kg}
 {enthalpy of ammonia leaving second compressor}
 T3 = Temperature(AmmoniaTim,P=P3,v=vol3) {K}
 {temperature of ammonia leaving second compressor}


```

Vdisphigh = numcylhigh*(pi*((borehigh/2)^2)*strokehigh) {m^3}
                                     {displacement volume of second compressor}
voleffhigh = 1-(clearratiohigh/(1-clearratiohigh))*((P3/P2b)^(1/polyexp) - 1)
                                     {volumetric efficiency of second compressor}
mcond = (1/v2b)*Vdisphigh*speedhigh*voleffhigh {kg/s}
                                     {mass flow of ammonia through second compressor}
Wcomphigh = (mcond*(h3-h2b))/polyeff {kW} {work of second compressor}
P3 = P4 {kPa} {constant pressure across condenser}
Wcomp = Wcompow + Wcomphigh {kW} {total compression work}

{condensor}
mair = 15*0.47879 {kg/s} {flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg-K} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairoutcond-Tamb) {calculates temperature change of air}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa}
                                     {pressure of ammonia leaving condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg}
                                     {enthalpy of ammonia leaving condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=0.0) {kJ/kg}
                                     {enthalpy of saturated liquid ammonia leaving the flash tank}
h1 = h5 {kJ/kg} {isenthalpic expansion}

Wpump = Wpumpcase+Wpumphx+Wpumppipe {kW}
                                     {total pump work done by system}
COP = caseload/(Wcomp+Wpump) {coefficient of performance of refrigeration system}
Cap = mst*CPST(T7)/caseload {1/K} {capacitance rate - caseload ratio}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

```

ETHANOL MODEL

{ Ammonia with ethanol system model }

FUNCTION THERMCONDEA(T)

{calculates the thermal conductivity of Ethanol given temperatures in Kelvin}

THERMCONDEA = 0.25715 - 0.00030645*T {W/m-K}

END

FUNCTION VISCOSITYEA(T)

{calculates the viscosity of Ethanol given temperatures in Kelvin}

$B = 11.664554604 - 0.057902271482 \cdot T + 6.3561331711e-5 \cdot T^2$

VISCOSITYEA = $\exp(B)/1e3$ {Pa s}

END

FUNCTION DENSITYEA(T)

{calculates the density of Ethanol given temperatures in Kelvin}

DENSITYEA = $0.2567e3 \cdot (0.240^{-(1-T/515.8)^{(2/7)}})$ {kg/m³}

END

FUNCTION CPEA(T)

{calculates the specific heat of Ethanol given temperatures in Kelvin}

CPEA = $(100.92 - 111.839e-3 \cdot T + 498.54e-6 \cdot T^2)/46$ {kJ/kg-K}

END

FUNCTION UVAL(T,mea,diam,ins,kins)

{calculates the U-value of the Ethanol pipe}

diamtotal = diam + 2*ins {m} {total outside diameter of pipe and insulation}

Area = $\pi \cdot (\text{diam}/2)^2$ {m²} {total heat transfer area}

vel = (mea/DENSITYEA(T))/Area {m/s} {velocity of Ethanol in pipes}

kvis = VISCOSITYEA(T)/DENSITYEA(T) {m²/s} {kinematic viscosity}

Re = vel*diam/kvis {Reynolds number}

Pr = $1000 \cdot \text{VISCOSITYEA}(T) \cdot \text{CPEA}(T) / \text{THERMCONDEA}(T)$ {Prandtl number}

Nui = $0.023 \cdot (\text{Re}^{**0.8}) \cdot (\text{Pr}^{**0.4})$ {Dittus-Boelter correlation for Nusselt number}

hi = Nui*THERMCONDEA(T)/diam {W/m²-K} {inside heat transfer coefficient}

ho = 6 {W/m²-K} {outside heat transfer coefficient}

UVAL = $(1/1000) / ((\text{diamtotal}/(\text{diam} \cdot \text{hi})) + ((\text{diamtotal}/2) \cdot \ln(\text{diamtotal}/\text{diam})/\text{kins}) + (1/\text{ho}))$
{W/m²-K} {overall heat transfer coefficient}

END

FUNCTION PRESSDROP(T,mea,diam,rough,length,Keq,gravity)

{calculates the pressure drop through pipes}

Area = $\pi \cdot (\text{diam}/2)^2$ {m²} {inside area of pipes}

vel = (mea/DENSITYEA(T))/Area {m/s} {velocity of Ethanol in pipes}

kvis = VISCOSITYEA(T)/DENSITYEA(T) {m²/s} {kinematic viscosity}

Re = vel*diam/kvis {Reynolds number}

{calculates the friction factor from Moody diagram using Haarland correlation}

Reynolds numbers less than 2300 considered laminar}

if (Re<2300) then ff = $64/\text{Re}$ else ff = $(1/(-1.8 \cdot \log_{10}((6.9/\text{Re}) +$

$((\text{rough}/\text{diam})/3.7)^{**1.11}))^{**2}$

headf = $\text{FF} \cdot (\text{length}/\text{diam}) \cdot (\text{vel}^2/(2 \cdot \text{gravity})) + \text{Keq} \cdot (\text{vel}^2)/(2 \cdot \text{gravity})$ {m}
{head losses}

PRESSDROP = headf*DENSITYEA(T)*gravity {Pa} {pressure drop}

END

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)

```

{calculates the overall heat transfer coefficient of the ammonia-Ethanol heat exchanger}
{inside pipes}
Tb = (Tin+Tout)/2 {K} {bulk temperature}
vel = (mdot/DENSITYEA(Tb))/(num*ri**2) {m/s} {velocity of Ethanol}
Red = vel*(2*ri)/(VISCOSITYEA(Tb)/DENSITYEA(Tb)) {Reynolds number}
Pr = 1000*VISCOSITYEA(Tb)*CPEA(Tb)/THERMCONDEA(Tb) {Prandtl number}
{Nusslet number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow}
if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
Nud = 3.66 + (0.668*(2*ri)/length)*Red*Pr/(1+0.4*((2*ri)/length)*Red*Pr)**(2/3))
hi = NUD*THERMCONDEA(Tb)/(2*ri) {W/m^2-K} {inside heat transfer coefficient}
{outside pipes}
Csf = 0.01 {empirical constant}
hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
                                         {kJ/kg} {heat of fusion}

g = 9.8 {m/s^2} {acceleration due to gravity}
sigma = 0.0213 {N/m} {surface tension}
Pr1 = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*
SpecHeat(AmmoniaTim,T=Tsat,P=Psat+1)/
Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) {Prandtl number}
Area = 2*pi*ro*length*num {m^2} {total heat transfer area}
{boiling heat transfer correlation using Rohsenow correlation}
ho = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*hfg*((g*
(1/Volume(AmmoniaTim,T=Tsat,x=0.0)-1/Volume(AmmoniaTim,T=Tsat,x=1.0))/
sigma)**0.5)*((SpecHeat(AmmoniaTim,T=Tsat,P=Psat-1)/(Csf*hfg*Pr1**1.7))**3)*
(Twall-Tsat)**2 {W/m^2-K}
{Uvalue}
UHX = (1/1000)/((ro/(ri*hi))+(ro*ln(ro/ri)/kpipe) + (1/ho)) {W/m^2-K}
                                         {overall heat transfer coefficient}

END

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
{calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
{inside pipe}
vel = (mdot/DENSITYEA(Tref))/(num*ri**2) {m/s} {velocity of Ethanol}
Redi = vel*(2*ri)/(VISCOSITYEA(Tref)/DENSITYEA(Tref)) {Reynolds number}
Pri = 1000*VISCOSITYEA(Tref)*CPEA(Tref)/THERMCONDEA(Tref)
                                         {Prandtl number}
{Nusselt number calculated using Nausen correlation fpr laminar flow and Nusselt
correlation for turbulent flow}
if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
else Nudi = 3.66 +
(0.668*(2*ri)/length)*Redi*Pri/(1+0.4*((2*ri)/length)*Redi*Pri)**(2/3))
hi = NUDI*THERMCONDEA(Tref)/(2*ri) {W/m^2-K}
                                         {inside heat transfer coefficient}
{outside pipes}
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
                                         {Reynolds number}
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)

```

```

{ Prandtl number}
{ Nusselt number calculated using Churchill-Bernstein correlation}
Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))*
(1+(Redo/2.82e5)**(5/8))**(4/5)
ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-K}
UCASE = (1/((ro/(ri*hi)))+(ro*ln(ro/ri)/kpipe) + (1/ho))/1000 {W/m^2-K}
{outside heat transfer coefficient}
{overall heat transfer coefficient}
END

```

```

{The equation set begins here}

```

```

{parameters}
Tairout = 267 {K} {temperature of the air leaving the refrigerated case}
caseload = 15{tons}*3.51685 {kW} {refrigeration load met by the case}
UAcond = 8.00 {kW/K} {UA value of the condenser}
Tamb = 300 {K} {ambient temperature}
Tstore = 300 {K} {store temperature}

```

```

{Compressor Parameters}
{dimensions of the first stage compressor}
borelow = 2.38*2.54e-2 {m}
strokelow = 1.75*2.54e-2 {m}
speedlow = 1700/60 {rev/sec}
clearratiolow = 0.05 {clearance ratio}
{dimensions of the second stage compressor}
borehigh = 2.38*2.54e-2 {m}
strokehigh = 1.75*2.54e-2 {m}
speedhigh = 1700/60 {rev/sec}
clearratiohigh = 0.05 {clearance ratio}
polyexp = 1.3 {polytropic exponent}
polyeff = 0.65 {polytropic efficiency}

```

```

{Ethanol Parameters}
mea = 2.305 {kg/s} {mass flow rate of Ethanol}
x = 0.54 {pressure ratio exponent}
diamc = 0.12 {m} {diameter of pipe to case}
diamh = 0.12 {m} {diameter of pipe from case}
insc = 0.025 {m} {thickness of insulation on pipe to case}
insh = 0.025 {m} {thickness of insulation on pipe from case}
kinsc = 0.04 {W/m-C} {thermal conductivity of insulation on pipe to case}
kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipe from case}
rough = 0.000046 {m} {pipe roughness}
lengthc = 30 {m} {length of pipe to case}
lengthh = 30 {m} {length of pipe from case}
gravity = 9.81 {m/s^2} {acceleration due to gravity}
diamctotal = diamc + 2*insc {m} {total diameter of pipe and insulation to case}
diamhtotal = diamh + 2*insh {m} {total diameter of pipe and insulation from case}
Kpipec = 8.0 {for minor losses}

```

$K_{pipeh} = 8.0$ {for minor losses}

{Heat Exchanger Parameters}

$numhxp = 300$ {number of pipes in ammonia-Ethanol heat exchanger}

$rihx = 0.005$ {m} {inside radius of pipes in ammonia-Ethanol heat exchanger}

$thickhx = 0.0015$ {m} {thickness of pipes in ammonia-Ethanol heat exchanger}

$rohx = rihx + thickhx$ {m}

{outside radius of pipes in ammonia-Ethanol heat exchanger}

$lenhx = 2.0$ {m} {length of pipes in ammonia-Ethanol heat exchanger}

$k_{pipehx} = 43$ {W/m-C}

{thermal conductivity of pipes in ammonia-Ethanol heat exchanger}

$K_{hx} = 8 * numhxp + 1.5$ {for minor losses}

$numcase = 700$ {number of pipes in refrigerated case heat exchanger}

$ricase = 0.01$ {m} {inside radius of pipes in refrigerated case heat exchanger}

$thickcase = 0.0015$ {m} {thickness of pipes in refrigerated case heat exchanger}

$rocase = ricase + thickcase$ {m}

{outside radius of pipes in refrigerated case heat exchanger}

$lencase = 2.0$ {m} {length of pipes in refrigerated case heat exchanger}

$airvel = 5.0$ {m/s} {air velocity in refrigerated case heat exchanger}

$airarea = 0.5$ {m²}

{cross sectional area for air flow in refrigerated case heat exchanger}

$k_{pipecase} = 43$ {W/m²-K}

{thermal conductivity of pipes in refrigerated case heat exchanger}

$K_{case} = 8 * numcase + 1.5$ {for minor losses}

{Secondary Cycle}

{equations for the refrigerated case heat exchanger}

$T_{air} = (T_{airin} + T_{airout})/2$ {K} {bulk temperature of air in case heat exchanger}

$TEA = (T_6 + T_5)/2$ {K} {bulk temperature of Ethanol in case heat exchanger}

$C_{paircase} = \text{SpecHeat}(\text{Air}, T=T_{airin})$ {kJ/kg-K} {specific heat of air}

$\dot{m}_{dotair} = (\text{airvel}/\text{Volume}(\text{Air}, T=T_{airin}, P=101.325)) * \text{airarea}$ {kg/s} {mass flow of air}

$C_{pcase} = \text{CPEA}(T_5)$ {kJ/kg-K} {specific heat of Ethanol}

$C_{air} = C_{paircase} * \dot{m}_{dotair}$ {kW/K} {capacitance rate of air}

$C_{EA} = C_{pcase} * \dot{m}_{dotair}$ {kW/K} {capacitance rate of Ethanol}

$C_{min} = \text{if}(C_{air}, C_{EA}, C_{air}, C_{air}, C_{EA})$ {minimum capacitance rate}

$C_{max} = \text{if}(C_{air}, C_{EA}, C_{EA}, C_{air}, C_{air})$ {maximum capacitance rate}

$Cr = C_{min}/C_{max}$ {capacitance rate ratio}

{calls function to calculate the overall heat transfer coefficient} {W/m²-K}

$U_{casehx} =$

$U_{CASE}(\text{numcase}, \text{ricase}, \text{rocase}, \text{lencase}, \text{mea}, \text{airvel}, \text{airarea}, \text{TEA}, 60, T_{air}, k_{pipecase})$

$U_{Acase} = U_{casehx} * (\text{numcase} * 2 * \pi * \text{rocase} * \text{lencase})$ {W/K} {UA value of case}

$NTU_{case} = U_{Acase}/C_{min}$ {number of transfer units of case heat exchanger}

{calculates the effectiveness using cross-flow correlation with both streams unmixed}

$\text{effcase} = 1 - \exp((1/Cr) * (NTU_{case} ** 0.22) * (\exp(-Cr * (NTU_{case} ** 0.78)) - 1))$

$\text{caseload} = \text{effcase} * C_{min} * (T_{airin} - T_5)$ {kW}

{heat transfer in case (refrigeration load met)}

$\text{caseload} = C_{air} * (T_{airin} - T_{airout})$ {change in temperature of the air stream}

$\text{caseload} = C_{EA} * (T_6 - T_5)$ {change in temperature of the Ethanol}

{calls function to calculate pressure drop through refrigerated case} {Pa}

$\Delta P_{case} = \text{PRESSDROP}(\text{TEA}, \text{mea}/\text{numcase}, 2 \cdot \text{ricase}, \text{rough}, \text{lencase}, \text{Kcase}, \text{gravity})$

{calculates the pump work needed to overcome pressure drop in case}

$W_{pumpcase} = \text{mea} \cdot (\Delta P_{case} / \text{DENSITYEA}(\text{TEA})) / 1000 \text{ {kW}}$

{Piping System}

$T_c = (T_8 + T_5) / 2 \text{ {K}}$ {bulk temperature of Ethanol in pipes to case}

$T_h = (T_6 + T_7) / 2 \text{ {K}}$ {bulk temperature of Ethanol in pipes from case}

{calls function to calculate overall heat transfer coefficient of pipes to case}

$U_c = \text{UVAL}(T_c, \text{mea}, \text{diamc}, \text{insc}, \text{kinsc}) \text{ {W/m}^2\text{-K}}$

{calls function to calculate overall heat transfer coefficient of pipes from case}

$U_h = \text{UVAL}(T_h, \text{mea}, \text{diamh}, \text{insh}, \text{kinsh}) \text{ {W/m}^2\text{-K}}$

{calls function to calculate pressure drop of pipes to case}

$\Delta P_c = \text{PRESSDROP}(T_c, \text{mea}, \text{diamc}, \text{rough}, \text{lengthc}, \text{Kpipec}, \text{gravity}) \text{ {Pa}}$

{calls function to calculate pressure drop of pipes to case}

$\Delta P_h = \text{PRESSDROP}(T_h, \text{mea}, \text{diamh}, \text{rough}, \text{lengthh}, \text{Kpipeh}, \text{gravity}) \text{ {Pa}}$

$\text{densityc} = \text{DENSITYEA}(T_c) \text{ {kg/m}^3}$ {density of Ethanol in pipes to case}

$\text{densityh} = \text{DENSITYEA}(T_h) \text{ {kg/m}^3}$ {density of Ethanol in pipes from case}

{calculates pump necessary to overcome pressure drop in piping system}

$W_{pumppipe} = \text{mea} \cdot (\Delta P_c / \text{densityc} + \Delta P_h / \text{densityh}) / 1000 \text{ {kW}}$

{calculates thermal losses of piping system}

$A_c = \pi \cdot \text{diamctotal} \cdot \text{lengthc} \text{ {m}^2}$ {total heat transfer area of pipes to case}

$A_h = \pi \cdot \text{diamhtotal} \cdot \text{lengthh} \text{ {m}^2}$ {total heat transfer area of pipes from case}

$C_{pc} = \text{CPEA}(T_c) \text{ {kJ/kg-K}}$ {specific heat of Ethanol in pipes to case}

$C_{ph} = \text{CPEA}(T_h) \text{ {kJ/kg-K}}$ {specific heat of Ethanol in pipes from case}

$Q_c = \text{mea} \cdot C_{pc} \cdot (T_5 - T_8) \text{ {kW}}$ {heat loss of Ethanol in pipes to case}

$Q_h = \text{mea} \cdot C_{ph} \cdot (T_7 - T_6) \text{ {kW}}$ {heat loss of Ethanol in pipes from case}

$\Delta T_{lmc} = (T_5 - T_8) / (\ln((T_{store} - T_8) / (T_{store} - T_5)))$

{log mean temperature difference of pipes to case}

$\Delta T_{lmh} = (T_7 - T_6) / (\ln((T_{store} - T_6) / (T_{store} - T_7)))$

{log mean temperature difference of pipes from case}

$Q_c = U_c \cdot A_c \cdot \Delta T_{lmc} \text{ {kW}}$ {heat loss of pipes to case}

$Q_h = U_h \cdot A_h \cdot \Delta T_{lmh} \text{ {kW}}$ {heat loss of pipes from case}

$NTU_c = U_c \cdot A_c / (\text{mea} \cdot C_{pc})$ {number of transfer units of pipes to case}

$NTU_h = U_h \cdot A_h / (\text{mea} \cdot C_{ph})$ {number of transfer units of pipes from case}

{equations for heat exchanger between Ethanol and Ammonia}

$T_{wall} = T_7 - (T_7 - T_8) / 2 \text{ {K}}$ {surface temperature of pipes in ammonia}

{calls function to calculate overall heat transfer coefficient} {W/m²-K}

$U = \text{UHX}(\text{numhxp}, \text{rihx}, \text{rohx}, \text{lenhx}, \text{mea}, T_7, T_8, T_{wall}, T_1, P_1, 60, \text{kpipehx})$

$UA_{hx} = U \cdot (2 \cdot \text{rohx} \cdot \pi \cdot \text{lenhx} \cdot \text{numhxp}) \text{ {W/K}}$ {UA value of heat exchanger}

$NTU_{hx} = UA_{hx} / (\text{mea} \cdot \text{CPEA}(T_7))$ {number of transfer units of heat exchanger}

$\text{effhx} = 1 - \exp(-NTU_{hx})$ {effectiveness of heat exchanger}

$\text{effhx} = (T_7 - T_8) / (T_7 - T_1)$

{using effectiveness to calculate temperatures in heat exchanger}

$Q_{hx} = \text{mea} \cdot \text{CPEA}(T_7) \cdot (T_7 - T_8) \text{ {kW}}$ {heat transfer in heat exchanger}

$T_{mid} = (T_7 + T_8) / 2 \text{ {K}}$ {bulk temperature of Ethanol in heat exchanger}

$T_1 = T_2 \text{ {K}}$ {constant ammonia temperature in heat exchanger}

```

mref = Qhx/(h2-h1) {kg/s} {calculates necessary ammonia flow rate to meet load}
{calls function to calculate the pressure drop in heat exchanger}
deltaPhx = PRESSDROP(Tmid,mea/numhxp,2*rihx,rough,lenhx,Khx,gravity) {Pa}
{calculates pump work necessary to overcome pressure drop in heat exchanger}
Wpumphx = mea*(deltaPhx/DENSITYEA(Tmid))/1000 {kW}

P2 = Pressure(AmmoniaTim,T=T1,x=1.0) {kPa}
{pressure of ammonia leaving heat exchanger}
h2 = Enthalpy(AmmoniaTim,T=T2,x=1.0) {kJ/kg}
{enthalpy of ammonia leaving heat exchanger}
s2 = Entropy(AmmoniaTim,T=T2,x=1.0) {kJ/kg-K}
{entropy of ammonia leaving heat exchanger}
P1 = P2 {kPa} {ammonia pressure constant in heat exchanger}

{Ammonia Cycle}
{compressor}
rp = P3/P2 {pressure ratio of evaporator to condenser}
rps = rp^x {pressure ratio relationship}
rps = P2a/P2 {calculates intermediate pressure}

{Low Pressure Compressor}
v2 = Volume(AmmoniaTim,P=P2,x=1.0) {m^3/kg}
{specific volume of ammonia entering first compressor}
P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}
h2a = Enthalpy(AmmoniaTim,P=P2a,v=v2a) {kJ/kg}
{enthalpy of ammonia leaving first compressor}
T2a = Temperature(AmmoniaTim,P=P2a,v=v2a) {K}
{temperature of ammonia leaving first compressor}
Vdisplow = numcyllo*(pi*((borelow/2)^2)*strokelow) {m^3}
{displacement volume of first compressor}
volefflow = 1-(clearratiolow/(1-clearratiolow))*((P2a/P2)^(1/polyexp) - 1)
{volumetric efficiency of first compressor}
mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}
{mass flow of ammonia through first compressor}
Wcomplow = (mref*(h2a-h2))/polyeff {kW} {work of first compressor}

{High Pressure Compressor}
P2b = P2a {kPa} {intermediate pressure}
h2b = Enthalpy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg}
{enthalpy of ammonia entering second compressor}
s2b = Entropy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg-K}
{entropy of ammonia entering second compressor}
T2b = Temperature(AmmoniaTim,P=P2b,x=1.0) {K}
{temperature of ammonia entering second compressor}
v2b = Volume(AmmoniaTim,P=P2b,x=1.0) {m^3/kg}
{specific heat of ammonia entering second compressor}
P2b*(v2b)^polyexp = P3*(vol3)^polyexp {polytropic compression equation}
h3 = Enthalpy(AmmoniaTim,P=P3,v=vol3) {kJ/kg}
{enthalpy of ammonia leaving second compressor}

```

```

T3 = Temperature(AmmoniaTim,P=P3,v=vol3) {K}
                                     {temperature of ammonia leaving second compressor}
Vdisphigh = numcylhigh*(pi*((borehigh/2)^2)*strokehigh) {m^3}
                                     {displacement volume of second compressor}
voleffhigh = 1-(clearratiohigh/(1-clearratiohigh))*((P3/P2b)^(1/polyexp) - 1)
                                     {volumetric efficiency of second compressor}
mcond = (1/v2b)*Vdisphigh*speedhigh*voleffhigh {kg/s}
                                     {mass flow of ammonia through second compressor}
Wcomphigh = (mcond*(h3-h2b))/polyeff {kW} {work of second compressor}
P3 = P4 {kPa} {constant pressure across condenser}
Wcomp = Wcompow + Wcomphigh {kW} {total compression work}

{condensor}
mair = 15*0.47879 {kg/s} {flow rate of air through condenser}
Cpair = SpecHeat(Air,T=Tamb) {kJ/kg-K} {specific heat of air}
UAcond = mair*Cpair*NTU {calculates number of transfer units of condenser}
eff = 1-exp(-NTU) {effectiveness of condenser}

Qcond = eff*mair*Cpair*(T4-Tamb) {kW} {heat transfer in condenser}
Qcond = mcond*(h3-h4) {calculates enthalpy change of ammonia in condenser}
Qcond = mair*Cpair*(Tairoutcond-Tamb) {calculates temperature change of air}

P4 = Pressure(AmmoniaTim,T=T4,x=0.0) {kPa}
                                     {pressure of ammonia leaving condenser}
h4 = Enthalpy(AmmoniaTim,T=T4,x=0.0) {kJ/kg}
                                     {enthalpy of ammonia leaving condenser}

{expansion}
P5 = P2a {kPa} {intermediate pressure}
h5 = Enthalpy(AmmoniaTim,P=P5,x=0.0) {kJ/kg}
                                     {enthalpy of saturated liquid ammonia leaving the flash tank}
h1 = h5 {kJ/kg} {isenthalpic expansion}

Wpump = Wpumpcase+Wpumpphx+Wpumppipe {kW}
                                     {total pump work done by system}
COP = caseload/(Wcomp+Wpump) {coefficient of performance of refrigeration system}
Cap = mea*CPEA(T7)/caseload {1/K} {capacitance rate - caseload ratio}

{mixing equation for flash tank}
mcond*h4 + mref*h2a = mref*h5 + mcond*h2b

```

PROPANE MODEL

{ Ammonia with propane system model }

FUNCTION UVAL(T,mpr,diam,ins,kins)


```

{calculates the U-value of the propane distribution pipes}
density = 1/Volume(Propane,T=T,P=700) {kg/m^3} {density of propane}
diamtotal = diam + 2*ins {m} {total outside diameter of pie and insulation}
Area = pi*(diam/2)^2 {m^2} {total heat transfer area}
vel = (mpr/density)/Area {m/s} {velocity of propane}
kvis = Viscosity(Propane,T=T,P=700)/density {m^2/s} {kinematic viscosity}
Re = vel*diam/kvis {Reynolds number}
Pr = 1000*Viscosity(Propane,T=T,P=700)*SpecHeat(Propane,T=T,P=700)/
Conductivity(Propane,T=T,P=700) {Prandtl number}
Nui = 0.023*(Re**0.8)*(Pr**0.4) {Dittus-Boelter correlation for Nusselt number}
hi = Nui*Conductivity(Propane,T=T,P=700)/diam {W/m^2-K}
{inside heat transfer coefficient}
ho = 6 {W/m^2-K} {outside heat transfer coefficient}
UVAL = (1/1000)/((diamtotal/(diam*hi))+((diamtotal/2)*ln(diamtotal/diam)/kins)+(1/ho))
{W/m^2-K} {overall heat transnfer coefficient}
END

```

```

FUNCTION PRESSDROP(T,mpr,diam,rough,length,Keq,gravity)
{calculate the pressure drop through pipes}
density = 1/Volume(Propane,T=T,P=700) {kg/m^3} {density of propane}
Area = pi*(diam/2)^2 {m^2} {inside area of pipes}
vel = (mpr/density)/Area {m/s} {velocity of propane}
kvis = Viscosity(Propane,T=T,P=700)/density {m^2/s} {kinematic viscosity}
Re = vel*diam/kvis {Reynolds number}
{calculates the friction factor from Moody diagram using Haarland correlation}
Reynolds numbers less than 2300 considered laminar}
if (Re<2300) then ff = 64/Re else ff = (1/(-1.8*log10((6.9/Re) +
((rough/diam)/3.7)**1.11)))**2
headf = FF*(length/diam)*(vel^2/(2*gravity)) + Keq*(vel^2)/(2*gravity) {m}
{head losses}
PRESSDROP = headf*density*gravity {Pa} {pressure drop}
END

```

```

FUNCTION UHX(num,ri,ro,length,mdot,Tin,Tout,Twall,Tsat,Psat,percent,kpipe)
{calculates the overall heat transfer coefficient of the ammonia-propane heat exchanger}
{inside pipes}
Tb = (Tin+Tout)/2 {K} {bulk temperature of propane}
vel = (mdot/(1/Volume(Propane,T=Tb,P=700)))/(num*ri**2) {m/s}
{velocity of propane}
Red = vel*(2*ri)/(Viscosity(Propane,T=Tb,P=700)/(1/Volume(Propane,T=Tb,P=700)))
{Reynolds number}
Pr = 1000*Viscosity(Propane,T=Tb,P=700)*SpecHeat(Propane,T=Tb,P=700)/
Conductivity(Propane,T=Tb,P=700) {Prandtl number}
{Nusselt number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow}
if (Red>2300) then Nud = 0.036*(Red**0.8)*(Pr**(1/3))*((2*ri)/length)**(1/18) else
Nud = 3.66 + (0.668*(2*ri/length)*Red*Pr)/(1+0.4*((2*ri/length)*Red*Pr)**(2/3))
hi = NUD*Conductivity(Propane,T=Tb,P=700)/(2*ri) {W/m^2-K}
{inside heat transfer coefficient}

```

```

{outside pipes}
Csf = 0.01 {empirical constant}
hfg = Enthalpy(AmmoniaTim,T=Tsat,x=1.0)-Enthalpy(AmmoniaTim,T=Tsat,x=0.0)
                                         {kJ/kg}   {heat of fusion}

g = 9.8 {m/s^2} {acceleration due to gravity}
sigma = 0.0213 {N/m} {surface tension}
Pr1 = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*SpecHeat(AmmoniaTim,
T=Tsat,P=Psat+1)/Conductivity(AmmoniaTim,T=Tsat,P=Psat+1) {Prandtl number}
Area = 2*pi*ro*length*num {m^2} {total heat transfer area}
{boiling heat transfer coefficient calculated using Rohsenow correlation}
ho = 1000*Viscosity(AmmoniaTim,T=Tsat,P=Psat+1)*hfg*((g*(1/Volume
(AmmoniaTim,T=Tsat,x=0.0)-1/Volume(AmmoniaTim,T=Tsat,x=1.0))/sigma)**0.5)*
((SpecHeat(AmmoniaTim,T=Tsat,P=Psat-1)/(Csf*hfg*Pr1**1.7))**3)*(Twall-Tsat)**2
                                         {W/m^2-K}

{Uvalue}
UHX = (1/1000)/((ro/(ri*hi))+(ro*ln(ro/ri)/kpipe) + (1/ho)) {W/m^2-K}
                                         {overall heat transfer coefficient}

END

FUNCTION UCASE(num,ri,ro,length,mdot,airvel,airarea,Tref,percent,Tair,kpipe)
{calculates the overall heat transfer coefficient of the refrigerated case heat exchanger}
{inside pipe}
vel = (mdot/(1/Volume(Propane,T=Tref,P=700)))/(num*ri**2) {m/s}
                                         {velocity of propane}

Redi = vel*(2*ri)/(Viscosity(Propane,T=Tref,P=700)/
(1/Volume(Propane,T=Tref,P=700))) {Reynolds number}
Pri = 1000*Viscosity(Propane,T=Tref,P=700)*SpecHeat(Propane,T=Tref,P=700)/
Conductivity(Propane,T=Tref,P=700) {Prandtl number}
{Nusselt number calculated using Hausen correlation for laminar flow and Nusselt
correlation for turbulent flow}
if (Redi>2300) then Nudi = 0.036*(Redi**0.8)*(Pri**(1/3))*((2*ri)/length)**(1/18)
else Nudi = 3.66 +
(0.668*(2*ri/length)*Redi*Pri)/(1+0.4*((2*ri/length)*Redi*Pri)**(2/3))
hi = NUDI*Conductivity(Propane,T=Tref,P=700)/(2*ri) {W/m^2-K}
                                         {inside heat transfer coefficient}

{outside pipes}
Redo = airvel*(2*ro)/(Viscosity(Air,T=Tair)/(1/Volume(Air,T=Tair,P=101.325)))
                                         {Reynolds number}
Pro = 1000*Viscosity(Air,T=Tair)*SpecHeat(Air,T=Tair)/Conductivity(Air,T=Tair)
                                         {Prandtl number}
{Nusselt number calculated using Churchill-Bernstein correlation}
Nudo = 0.3+((0.62*Redo**(1/2)*Pro**(1/3))/((1+(0.4/Pro)**(2/3))**(1/4)))
*(1+(Redo/2.82e5)**(5/8))**(4/5)
ho = Nudo*Conductivity(Air,T=Tair)/(2*ro) {W/m^2-K}
                                         {outside heat transfer coefficient}
UCASE = (1/((ro/(ri*hi))+(ro*ln(ro/ri)/kpipe) + (1/ho)))/1000 {W/m^2-K}
                                         {overall heat transfer coefficient}

END

```

{The equation set begins here}

{parameters}

Tairout = 267 {K} {temperature of the air leaving the refrigerated case}
 caseload = 15{tons}*3.51685 {kW} {refrigeration load met by the case}
 UAcond = 8.00 {kW/K} {UA value of the condenser}
 Tamb = 300 {K} {ambient temperature}
 Tstore = 300 {K} {store temperature}

{Compressor Parameters}

{dimensions of the first stage compressor}

borelow = 2.38*2.54e-2 {m}

stroke low = 1.75*2.54e-2 {m}

speedlow = 1700/60 {rev/sec}

clearratio low = 0.05 {clearance ratio}

{dimensions of the second stage compressor}

borehigh = 2.38*2.54e-2 {m}

stroke high = 1.75*2.54e-2 {m}

speedhigh = 1700/60 {rev/sec}

clearratio high = 0.05 {clearance ratio}

polyexp = 1.3 {polytropic exponent}

polyeff = 0.65 {polytropic efficiency}

{Propane Parameters}

mpr = 2.305 {kg/s} {mass flow rate of propane}

x = 0.54 {pressure ratio exponent}

diamc = 0.12 {m} {diameter of pipe to case}

diamh = 0.12 {m} {diameter of pipe from case}

insec = 0.025 {m} {thickness of insulation on pipe to case}

insh = 0.025 {m} {thickness of insulation on pipe from case}

kinsec = 0.04 {W/m-C} {thermal conductivity of insulation on pipe to case}

kinsh = 0.04 {W/m-C} {thermal conductivity of insulation on pipe from case}

rough = 0.000046 {m} {pipe roughness}

lengthc = 30 {m} {length of pipe to case}

lengthh = 30 {m} {length of pipe from case}

gravity = 9.81 {m/s^2} {acceleration due to gravity}

diamctotal = diamc + 2*insec {m} {total diameter of pipe and insulation to case}

diamhtotal = diamh + 2*insh {m} {total diameter of pipe and insulation from case}

Kpipec = 8.0 {for minor losses}

Kpipeh = 8.0 {for minor losses}

{Heat Exchanger Parameters}

numhxp = 300 {number of pipes in ammonia-propane heat exchanger}

rihxp = 0.005 {m} {inside radius of pipes in ammonia-propane heat exchanger}

thickhx = 0.0015 {m} {thickness of pipes in ammonia-propane heat exchanger}

rohxp = rihxp + thickhx {m}

{outside radius of pipes in ammonia-propane heat exchanger}

lenhx = 2.0 {m} {length of pipes in ammonia-propane heat exchanger}

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kpipehx = 43 {W/m-C}
               {thermal conductivity of pipes in ammonia-propane heat exchanger}
Khx = 8*numhxp + 1.5 {for minor losses}
numcase = 700 {number of pipes in refrigerated case heat exchanger}
ricase = 0.01 {m} {inside radius of pipes in refrigerated case heat exchanger}
thickcase = 0.0015 {m} {thickness of pipes in refrigerated case heat exchanger}
rocase = ricase + thickcase {m}
               {outside radius of pipes in refrigerated case heat exchanger}
lencase = 2.0 {m} {length of pipes in refrigerated case heat exchanger}
airvel = 5.0 {m/s} {air velocity in refrigerated case heat exchanger}
airarea = 0.5 {m^2}
               {cross sectional area for air flow in refrigerated case heat exchanger}
kpipecase = 43 {W/m^2}
               {thermal conductivity of pipes in refrigerated case heat exchanger}
Kcase = 8*numcase + 1.5 {for minor losses}

{Secondary Cycle}
{equations for the refrigerated case heat exchanger}
Tair = (Tairin + Tairout)/2 {K} {bulk temperature of air}
TPR = (T6+T5)/2 {K} {bulk temperature of propane}
Cpaircase = SpecHeat(Air,T=Tairin) {kJ/kg-K} {specific heat of air}
mdotair = (airvel/Volume(Air,T=Tairin,P=101.325))*airarea {kg/s} {mass flow of air}
Cpcase = SpecHeat(Propane,T=T5,P=700) {kJ/kg-K} {specific heat of propane}
Cair = Cpaircase*mdotair {kW/K} {capacitance rate of air}
CPR = Cpcase*mpr {kW/K} {capacitance rate of propane}
Cmin = if(Cair,CPR,Cair,Cair,CPR) {minimum capacitance rate}
Cmax = if(Cair,CPR,CPR,Cair,Cair) {maximum capacitance rate}
Cr = Cmin/Cmax {capacitance rate ratio}
{calls function to calculate the overall heat transfer coefficient} {W/m^2-K}
Ucasehx =
UCASE(numcase,ricase,rocase,lencase,mpr,airvel,airarea,TPR,60,Tair,kpipecase)
UAcase = Ucasehx*(numcase*2*pi*rocase*lencase) {W/K} {UA value of case}
NTUcase = UAcase/Cmin {number of transfer units of case}
{effectiveness of case calculated as cross-flow with both streams unmixed}
effcase = 1-exp((1/Cr)*(NTUcase**0.22)*(exp(-Cr*(NTUcase**0.78))-1))
caseload = effcase*Cmin*(Tairin-T5) {kW}
               {heat transfer in case (refrigeration load met)}
caseload = Cair*(Tairin-Tairout) {calculates change in air temperature}
caseload = CPR*(T6-T5) {calculates change in propane temperature}
{calls function to calculate the pressure drop in case} {Pa}
deltaPcase = PRESSDROP(TPR,mpr/numcase,2*ricase,rough,lencase,Kcase,gravity)
{calculates the pump work needed to overcome pressure drop in case}
Wpumpcase = mpr*(deltaPcase/(1/Volume(Propane,T=TPR,P=700)))/1000 {kW}

{Piping System}
Tc = (T8+T5)/2 {K} {bulk temperature of propane in pipes to case}
Th = (T6+T7)/2 {K} {bulk temperature of propane in pipes from case}
{calls function to calculate the overall heat transfer coefficient of pipes to case}
Uc = UVAL(Tc,mpr,diamc,insc,kinsc) {W/m^2-K}

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{calls function to calculate the overall heat transfer coefficient of pipes from case}
Uh = UVAL(Th,mpr,diamh,insh,kinsh) {W/m^2-K}
{calls function to calculate the pressure drop of pipes to case}
deltaPc = PRESSDROP(Tc,mpr,diamc,rough,lengthc,Kpipec,gravity) {Pa}
{calls function to calculate the pressure drop of pipes from case}
deltaPh = PRESSDROP(Th,mpr,diamh,rough,lengthh,Kpipeh,gravity) {Pa}

densityc = 1/Volume(Propane,T=Tc,P=700) {kg/m^3}
{density of propane in pipes to case}
densityh = 1/Volume(Propane,T=Th,P=700) {kg/m^3}
{density of propane in pipes from case}
{calculates the pump work needed to overcome pressure drop in piping system}
Wpumppipe = mpr*(deltaPc/densityc + deltaPh/densityh)/1000 {kW}

{calculates the thermal losses of the piping system}
Ac = pi*diamctotal*lengthc {m^2} {total heat transfer area of pipes to case}
Ah = pi*diamhtotal*lengthh {m^2} {total heat transfer area of pipes from case}
Cpc = SpecHeat(Propane,T=Tc,P=700) {kJ/kg-K}
{specific heat of propane in pipes to case}
Cph = SpecHeat(Propane,T=Th,P=700) {kJ/kg-K}
{specific heat of propane in pipes from case}
Qc = mpr*Cpc*(T5-T8) {kW} {heat loss from pipes to case}
Qh = mpr*Cph*(T7-T6) {kW} {heat loss from pipes from case}
deltatlmc = (T5-T8)/(ln((Tstore-T8)/(Tstore-T5)))
{log mean temperature difference of pipes to case}
deltatlmh = (T7-T6)/(ln((Tstore-T6)/(Tstore-T7)))
{log mean temperature difference of pipes from case}
Qc = Uc*Ac*deltatlmc {kW} {heat loss from pipes to case}
Qh = Uh*Ah*deltatlmh {kW} {heat loss from pipes from case}
NTUc = Uc*Ac/(mpr*Cpc) {number of transfer units of pipes to case}
NTUh = Uh*Ah/(mpr*Cph) {number of transfer units of pipes from case}

{heat exchanger between Propane and Ammonia}
Twall = T7-(T7-T8)/2 {K} {surface temperature of pipes in ammonia}
{calls function to calculate the overall heat transfer coefficient} {W/m^2-K}
U = UHX(numhxp,rihx,rohr,lehx,mpr,T7,T8,Twall,T1,P1,60,kpipehx)
UAhx = U*(2*rohr*pi*lehx*numhxp) {W/K} {UA value of heat exchanger}
NTUhx = UAhx/(mpr*SpecHeat(Propane,T=T7,P=700))
{number of transfer units of heat exchanger}
effhx = 1-exp(-NTUhx) {effectiveness of heat exchanger}
effhx = (T7-T8)/(T7-T1) {uses effectiveness to calculate temperatures in heat exchanger}
Qhx = mpr*SpecHeat(Propane,T=T7,P=700)*(T7-T8) {kW}
{heat transfer in heat exchanger}
Tmid = (T7+T8)/2 {K} {bulk temperature of propane}
T1 = T2 {K} {constant ammonia temperature in heat exchanger}
mref = Qhx/(h2-h1) {kg/s}
{calculates the required ammonia mass flow to meet refrigeration load}
{calls function to calculate the pressure drop in case heat exchanger} {Pa}
deltaPhx = PRESSDROP(Tmid,mpr/numhxp,2*rihx,rough,lehx,Khx,gravity)

```

```

{calculates the needed pump work to overcome pressure drop in heat exchanger}
Wpumphx = mpr*(deltaPhx/(1/Volume(Propane,T=Tmid,P=700)))/1000 {kW}

P2 = Pressure(AmmoniaTim,T=T1,x=1.0) {kPa}
                                     {pressure of ammonia leaving heat exchanger}
h2 = Enthalpy(AmmoniaTim,T=T2,x=1.0) {kJ/kg}
                                     {enthalpy of ammonia leaving heat exchanger}
s2 = Entropy(AmmoniaTim,T=T2,x=1.0) {kJ/kg-K}
                                     {entropy of ammonia leaving heat exchanger}
P1 = P2 {kPa} {constant ammonia pressure in heat exchanger}

{Ammonia Cycle}
{compressor}
rp = P3/P2 {pressure ratio of evaporator to condenser}
rps = rp^x {pressure ratio relationship}
rps = P2a/P2 {intermediate pressure}

{Low Pressure Compressor}
v2 = Volume(AmmoniaTim,P=P2,x=1.0) {m^3/kg}
                                     {specific volume of ammonia entering first compressor}
P2*(v2)^polyexp = P2a*(v2a)^polyexp {polytropic compression equation}
h2a = Enthalpy(AmmoniaTim,P=P2a,v=v2a) {kJ/kg}
                                     {enthalpy of ammonia leaving first compressor}
T2a = Temperature(AmmoniaTim,P=P2a,v=v2a) {K}
                                     {tempertaure of ammonia leaving first compressor}
Vdisplow = numcyllo*(pi*((borelow/2)^2)*strokelow) {m^3}
                                     {displacement volume of first compressor}
volefflow = 1-(clearratiolow/(1-clearratiolow))*((P2a/P2)^(1/polyexp) - 1)
                                     {volumetric efficiency of first compressor}
mref = (1/v2)*Vdisplow*speedlow*volefflow {kg/s}
                                     {mass flow rate of ammonia in first compressor}
Wcomplow = (mref*(h2a-h2))/polyeff {kW} {work of first compressor}

{High Pressure Compressor}
P2b = P2a {kPa} {intermediate pressure}
h2b = Enthalpy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg}
                                     {enthalpy of ammonia entering second compressor}
s2b = Entropy(AmmoniaTim,P=P2b,x=1.0) {kJ/kg-K}
                                     {entropy of ammonia entering second compressor}
T2b = Temperature(AmmoniaTim,P=P2b,x=1.0) {K}
                                     {temperature of ammonia entering second compressor}
v2b = Volume(AmmoniaTim,P=P2b,x=1.0) {m^3/kg}
                                     {specific volume of ammonia entering second compressor}
P2b*(v2b)^polyexp = P3*(vol3)^polyexp {polytropic compression equation}
h3 = Enthalpy(AmmoniaTim,P=P3,v=vol3) {kJ/kg-K}
                                     {enthalpy of ammonia leaving second compressor}
T3 = Temperature(AmmoniaTim,P=P3,v=vol3) {K}
                                     {temperature of ammonia leaving second compressor}
Vdisphigh = numcylhigh*(pi*((borehigh/2)^2)*strokehigh) {m^3}

```

$$\text{voleffhigh} = 1 - (\text{clearratiohigh} / (1 - \text{clearratiohigh})) * ((P3/P2b)^{(1/\text{polyexp})} - 1)$$
 {displacement volume of second compressor}
 {volumetric efficiency of second compressor}

$$\text{mcond} = (1/v2b) * V\text{disphigh} * \text{speedhigh} * \text{voleffhigh} \text{ {kg/s}}$$
 {mass flow of ammonia through second compressor}

$$W\text{comphigh} = (\text{mcond} * (h3 - h2b)) / \text{polyeff} \text{ {kW}} \text{ {work of second compressor}}$$

$$P3 = P4 \text{ {kPa}} \text{ {constant pressure in condenser}}$$

$$W\text{comp} = W\text{compow} + W\text{comphigh} \text{ {kW}} \text{ {total compression work}}$$

{condensor}

$$\text{mair} = 15 * 0.47879 \text{ {kg/s}} \text{ {mass flow of air in condenser}}$$

$$C\text{pair} = \text{SpecHeat}(\text{Air}, T = T\text{amb}) \text{ {kJ/kg-K}} \text{ {specific heat of air}}$$

$$U\text{Acond} = \text{mair} * C\text{pair} * \text{NTU} \text{ {calculates the number of transfer units of condenser}}$$

$$\text{eff} = 1 - \exp(-\text{NTU}) \text{ {effectiveness of condenser}}$$

$$Q\text{cond} = \text{eff} * \text{mair} * C\text{pair} * (T4 - T\text{amb}) \text{ {kW}} \text{ {heat transfer in condenser}}$$

$$Q\text{cond} = \text{mcond} * (h3 - h4) \text{ {calculates enthalpy change of ammonia in condenser}}$$

$$Q\text{cond} = \text{mair} * C\text{pair} * (T\text{airoutcond} - T\text{amb}) \text{ {calculates the change in air temperature}}$$

$$P4 = \text{Pressure}(\text{AmmoniaTim}, T = T4, x = 0.0) \text{ {kPa}}$$
 {pressure of ammonia leaving condenser}

$$h4 = \text{Enthalpy}(\text{AmmoniaTim}, T = T4, x = 0.0) \text{ {kJ/kg}}$$
 {enthalpy of ammonia leaving condenser}

{expansion}

$$P5 = P2a \text{ {kPa}} \text{ {intermediate pressure}}$$

$$h5 = \text{Enthalpy}(\text{AmmoniaTim}, P = P5, x = 0.0) \text{ {kJ/kg}}$$
 {enthalpy of saturated liquid ammonia leaving the flash tank}

$$h1 = h5 \text{ {kJ/kg}} \text{ {isenthalpic expansion}}$$

$$W\text{pump} = W\text{pumpcase} + W\text{pumphx} + W\text{pumppipe} \text{ {kW}} \text{ {total pump work in system}}$$

$$\text{COP} = \text{caseload} / (W\text{comp} + W\text{pump}) \text{ {coefficient of performance of refrigeration system}}$$

$$\text{Cap} = \text{mpr} * \text{SpecHeat}(\text{Propane}, T = T7, P = 700) / \text{caseload} \text{ {1/K}}$$
 {capacitance rate - caseload ratio}

{mixing equation for flash tank}

$$\text{mcond} * h4 + \text{mref} * h2a = \text{mref} * h5 + \text{mcond} * h2b$$

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