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ABSTRACT

Dedicated mechanical subcooling cycles utilize a small mechanical vapor-compression cycle, coupled to the main cycle at the exit of the condenser, to provide subcooling to the main refrigeration cycle. The amount of subcooling, the thermal lift of the subcooling cycle, and consequently the performance of the overall cycle can be related directly to the temperature of the subcooling cycle evaporator. In this paper, the optimum value of the subcooling evaporator temperature is predicted using an ideal dedicated subcooling cycle. These results are then compared to those generated from a property-dependent model. The consideration of this optimum subcooling evaporator temperature leads to a design rule for the optimum distribution of heat exchange area for the dedicated subcooling cycle.

NOMENCLATURE

COP	coefficient of performance
COP_{Carnot}	ideal coefficient of performance
COP_{main}	coefficient of performance of the main refrigeration cycle
COP_{sub}	coefficient of performance of the subcooling refrigeration cycle
COP_{total}	coefficient of performance of the overall dedicated subcooling cycle
$\Delta h_{2\text{-}3}$	main cycle refrigerant enthalpy difference between exit of condenser and exit of subcooler
$\Delta h_{4\text{-}4'}$	enthalpy difference between subcooled and non-subcooled main cycle evaporator inlets
LMTD	Log Mean Temperature Difference
$\dot{m}_{ratio}$	refrigerant flow rate ratio
$\dot{m}_{ref,main}$	refrigerant flow rate for the main refrigeration cycle
$\dot{m}_{ref,sub}$	refrigerant flow rate for the subcooling refrigeration cycle
Q	heat transfer to or from the refrigeration cycle
Q_{evap}	heat transfer to the main cycle evaporator
$Q_{evap,main,nosub}$	heat transfer to the main cycle evaporator if there is no subcooling being performed
$Q_{evap,sub}$	heat transfer to the subcooling cycle evaporator
Q_{sub}	heat transfer across the subcooler
T_L	temperature of the refrigerated space
T_H	refrigeration cycle sink temperature
T_M	intermediate temperature for the ideal subcooling cycle
x	a measure of performance of the subcooler heat exchanger
UA	overall heat transfer coefficient
$W_{comp,main}$	work required to operate main refrigeration cycle
$W_{comp,sub}$	work required to operate subcooling refrigeration cycle

INTRODUCTION

The coefficient of performance (COP) of low-temperature refrigeration cycles can be increased beyond that which is possible through standard vapor-compression cycles by utilizing dedicated mechanical subcooling. Dedicated mechanical subcooling cycles employ a second vapor-compression cycle solely for the purpose of providing subcooling to the main refrigeration cycle. The subcooling cycle is coupled to the main cycle by the use of a subcooler located at the exit of the main cycle condenser (refer to Figure 1). For supermarket applications, the subcooler provides about 70°F of subcooling at design conditions and acts as the evaporator for the subcooling cycle. Because the second cycle provides a lower temperature sink for the subcooler heat transfer than the ambient, the mechanical subcooling cycle is especially effective at high ambient and low evaporator temperatures. In practice, the components of the subcooling cycle are a fraction of the size of the main cycle components and perform through much smaller temperature extremes. For this reason, the COP of the *subcooling* cycle is appreciably higher than that of the main refrigeration cycle. This high subcooling cycle COP can result in an increase in the overall cycle COP.

Considering Figure 2, a pressure-enthalpy diagram for a dedicated mechanical subcooling cycle, subcooling allows the refrigerant to enter the main cycle evaporator with a lower quality (where point 4' represents a typical vapor compression cycle and point 4 represents the dedicated subcooling cycle). The lower quality at the evaporator inlet corresponds to an increase in the refrigeration capacity per unit mass of refrigerant circulated. However, the increase in refrigeration capacity is not without cost. Neglecting losses to the environment, an energy balance on the subcooler reveals that the amount of subcooling provided to the main cycle must equal the heat addition to the subcooling cycle evaporator. The heat addition to the subcooling cycle evaporator must be rejected in the subcooling cycle condenser at the cost of the work of the subcooling cycle compressor. Therefore, there is a trade-off between the amount of subcooling provided to the main cycle and the amount of work performed by the subcooling cycle compressor. This paper investigates this trade-off and explores the concept of the "optimum" temperature for the subcooling cycle evaporator. This "optimum" temperature of the subcooling cycle evaporator is the temperature at which the COP of the overall cycle is maximized. The "optimum" temperature is derived for a thermodynamically ideal mechanical subcooling refrigeration cycle. The results are then compared with those from a more detailed property-dependent system model. Finally, design guidelines for the optimum distribution of heat exchange area for the dedicated mechanical subcooling cycle are developed.

OPTIMUM TEMPERATURE FOR A THERMODYNAMICALLY IDEAL CYCLE

The thermodynamically ideal mechanical subcooling cycle was developed using the theory of Carnot and classic heat exchanger theory. Carnot developed a theoretical upper limit on the performance of a refrigeration cycle that is often called the Carnot COP. The Carnot COP assumes an internally reversible cycle and can be modeled as:

$$COP_{Carnot} = \frac{Capacity}{Work} = \frac{Q_{evap}}{W} = \frac{T_L}{(T_H - T_L)} \qquad (1)$$

where
- Q_{evap} is the heat transfer from the refrigerated space
- W is the work supplied to the refrigeration cycle by the compressor
- T_L is the temperature of the refrigerated space
- T_H is the sink temperature

The following assumptions were made in the development of the ideal mechanical subcooling model:

- Both the main cycle and subcooling cycle condensers reject heat at the sink temperature (T_H)
- The main cycle heat addition occurs at T_L, the refrigerated space temperature
- The subcooling cycle heat addition occurs at T_M, an intermediate temperature ($T_L \leq T_M \leq T_H$)
- The COP of the main cycle and subcooling cycle are assumed to be the Carnot COP if no subcooling is provided
- There is no thermal energy loss to the environment in the subcooler
- The only irreversibility is due to the subcooler heat transfer
- The main cycle compressor work is not influenced by the amount of subcooling provided to the main cycle.
- The exit states of the main cycle condenser and evaporator are unaffected by the amount of subcooling performed
- Isentropic expansion and compression are assumed for both the main and subcooling cycles

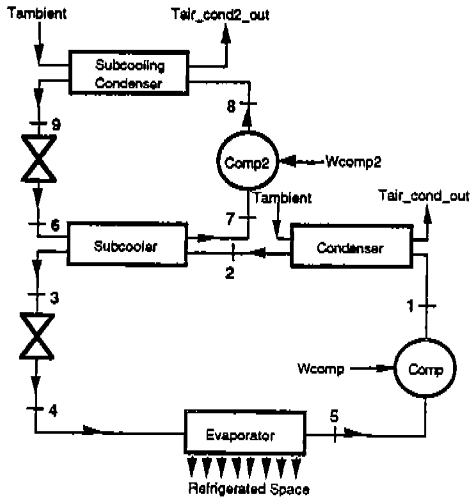

Figure 1: Component diagram for dedicated mechanical subcooling cycle.

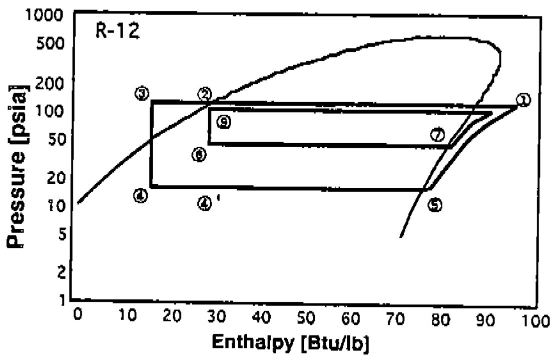

Figure 2: Pressure-enthalpy diagram for a dedicated mechanical subcooling cycle.

If no subcooling is provided to the main cycle, the COP can be written as:

$$COP_{main} = \frac{Q_{evap,main,nosub}}{W_{comp,main}} = \frac{T_L}{(T_H - T_L)} \quad (2)$$

When subcooling is added to the main cycle, the refrigeration capacity will increase due to the reduced quality of the refrigerant entering the main cycle evaporator. However, the main cycle compressor will still provide the same amount of work. Therefore, the main cycle COP increases with additional subcooling.

The subcooling cycle operates between the sink temperature (T_H) and the subcooling cycle evaporator temperature (T_M). Therefore, the COP of the subcooling cycle may now be expressed as:

$$COP_{sub} = \frac{Q_{evap,sub}}{W_{comp,sub}} = \frac{T_M}{(T_H - T_M)} \quad (3)$$

Neglecting losses to the environment, an energy balance on the subcooler shows that the subcooling cycle evaporator heat transfer ($Q_{evap,sub}$) is equal to the amount of subcooling provided to the main cycle (Q_{sub}).

The overall cycle COP may be expressed as the total refrigeration capacity divided by the total work. The capacity of the overall cycle is simply the capacity of the main cycle without subcooling, plus some increment in capacity of the main cycle due to the subcooling performed. With the assumptions given earlier, an energy balance on the main cycle reveals that the amount of subcooling performed is equal to the increment in capacity to the main cycle.

Referring to Figure 2, the energy balance may be seen as $\Delta h_{2-3} = \Delta h_{4-4}$. The total work performed on the cycle is simply the sum of the compressor work for both the subcooling and main cycles. With these definitions, the COP of the overall cycle may be expressed as:

$$COP_{total} = \frac{(Q_{evap,main,nosub} + Q_{sub})}{(W_{comp,main} + W_{comp,sub})} \qquad (4)$$

Before this expression may be further manipulated, an assumption is made to model the heat transfer in the subcooler (the only source of irreversibility in the ideal model). The assumption is that the heat transfer in the subcooler is proportional to the temperature difference between the working fluids in the main and subcooling cycles. For the ideal dedicated mechanical subcooling cycle, the maximum temperature difference in the subcooler is between the sink temperature (T_H) and the subcooling evaporator temperature (T_M). The expression for the subcooler heat transfer becomes:

$$Q_{sub} = Q_{evap,sub} = x * (T_H - T_M) \qquad (5)$$

where x is the effectiveness-C_{min} product as described by the NTU heat exchanger performance calculation method.

The goal of the ideal model is to develop an expression for the overall cycle COP as a function of the subcooling evaporator temperature (T_M) and system parameters. Since T_M is a measure of the amount of subcooling provided <u>and</u> the subcooling cycle thermal lift, there exists a thermodynamic compromise between the two competing effects. The desired expression is obtained by solving equation 2 for the main cycle compressor work (which is independent of the amount of subcooling as described earlier), equation 3 for the subcooling cycle compressor work, and incorporating the subcooler heat transfer (equation 5) into equation 4.

$$COP_{total} = \frac{Q_{evap,main,nosub} + x * (T_H - T_M)}{Q_{evap,main,nosub} * \frac{(T_H - T_L)}{T_L} + x * \frac{(T_H - T_M)^2}{T_M}} \qquad (6)$$

When ($T_M = T_H$), equation 6 reduces to:

$$COP_{total} = \frac{T_L}{(T_H - T_L)} \qquad (7)$$

which is the Carnot COP of a cycle operating between T_H and T_L, as expected. If the subcooling evaporator temperature is the sink temperature, there is no temperature difference between the flow streams in the subcooler. Therefore, there will be no subcooling provided to the main cycle and consequently no work performed by the subcooling cycle compressor. The overall cycle will then act like one cycle operating between T_H and T_L at the Carnot COP.

At the lower extreme ($T_M = T_L$), equation 6 again reduces to equation 7. With the subcooling temperature at the refrigerated space temperature, the maximum amount of subcooling is being performed. However, both cycles are now operating over the same thermal lift and the advantage of using dedicated mechanical subcooling is destroyed.

If there is a subcooling evaporator temperature that maximizes the overall COP, it must lie between the two temperature extremes T_H and T_L. Considering Figures 3 and 4, there exists an optimum temperature of the subcooling evaporator that maximizes the COP of the ideal cycle. Figures 3 and 4 also show that the optimum temperature of the subcooling evaporator is not strongly affected by either x; a measure of the subcooling heat exchanger performance, or Q_{evap}; the desired refrigeration capacity for the selected values of T_H and T_L that are representative of supermarket applications. The only factors influencing the choice of the optimum temperature for the ideal cycle are the sink temperature (T_H) and the refrigerated space temperature (T_L).

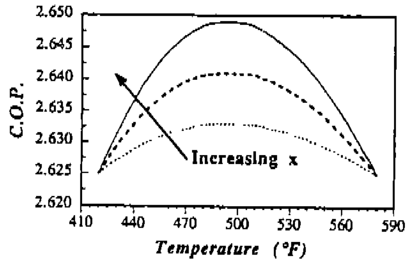

Figure 3: COP as a function of T_M and x for the ideal dedicated subcooling cycle.

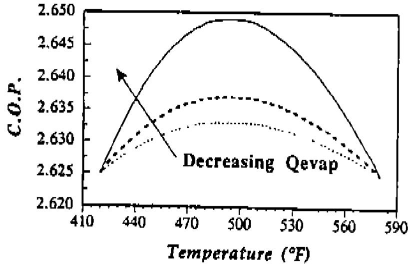

Figure 4: COP as a function of T_M and Qevap for the ideal dedicated subcooling cycle.

OPTIMUM TEMPERATURE FOR A PROPERTY-DEPENDENT CYCLE

Although the results for the ideal cycle suggest that an optimum subcooling cycle evaporator temperature exists and that this optimum temperature is dependent only on the sink and refrigerated space temperatures, there are many irreversibilities that could affect the choice of, or even the existence of, the optimum temperature. To evaluate whether the trends developed in the ideal analysis hold for the non-ideal case, a property-dependent computer model of a dedicated subcooling cycle was developed. The computer simulation was modeled after a supermarket application designed to provide 15 tons of low-temperature refrigeration. The property-dependent model takes into account the irreversibilities due to compression, expansion, and heat exchange. The model was developed using EES; an engineering equation solver that includes built-in thermophysical properties, optimization algorithms and parametric studies. The refrigeration system computer model was created by the integration of the steady-state component models discussed below. The property-dependent model differs from the ideal model in that the refrigeration capacity for the property-dependent model is assumed constant (15 tons).

COMPRESSORS: It was assumed for the simulation that the compressors were reciprocating compressors with negligible heat transfer to the surroundings. Because the isentropic efficiency is relatively independent of reciprocating compressor size for a given refrigerant, the steady-state compressors were modeled using the concept of isentropic efficiency. In this way, the influence of relative compressor size on the simulation results was eliminated.

EVAPORATORS: In most supermarket applications, the refrigerated display cases act as the evaporators for the refrigeration system. Therefore, the refrigerated space temperature dictates the evaporation temperature. For this simulation, the evaporator temperature was set at values of -20°F, 0°F, and 20°F. The refrigerant exiting the evaporator was assumed to leave with seven degrees of superheat.

CONDENSERS: The condensers were assumed to be air-cooled cross-flow heat exchangers with cooling air flow rates of 3800 pounds of air per hour per ton of refrigeration. This value corresponds to approximately 900 CFM per ton of refrigeration, and is representative of current practice. The condensers were modeled using the Log Mean Temperature Difference (LMTD) approach.

EXPANSION VALVES: A typical vapor compression refrigeration cycle contains one expansion device. For this study, it was assumed that the expansion device was a thermostatic expansion valve with negligible heat transfer to the surroundings. Thermostatic expansion valves control the refrigerant flow rate in response to the degrees of superheat exiting the evaporator in order to avoid unevaporated refrigerant being passed to the compressor.

SUBCOOLER: The subcooler, or subcooling heat exchanger, was assumed to be a concentric-tube, counter flow heat exchanger. The subcooling heat exchanger acts as the evaporator for the subcooling cycle and the subcooler for the main cycle and was modeled using the LMTD approach.

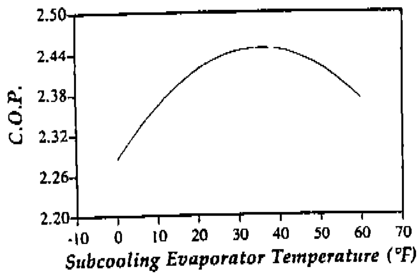

Figure 5: COP as a function of the subcooling evaporator temperature for the property-dependent dedicated subcooling cycle.

The effect of the subcooling evaporator temperature on overall COP was explored using the property-dependent computer model. Referring to Figure 5, there is a noticeable maximum point in the COP versus subcooling evaporator temperature plot as predicted by the ideal model. However, nine variables significantly affected the performance of the dedicated subcooling cycle and the choice of the optimal subcooling evaporator temperature.

- Refrigeration load
- Ambient Temperature
- Degrees of subcooling at exit of evaporators
- Main cycle evaporator temperature
- Compressor isentropic efficiency
- Main cycle condenser size (UA)
- Condenser cooling air flow rates
- Subcooler size (UA)
- Subcooling cycle condenser size (UA)

Of these nine variables, four are constrained by the supermarket application and refrigeration equipment; the refrigeration load, the degrees of subcooling at evaporator exit, the compressor isentropic efficiency, and the condenser cooling air flow rates. Five variables remained that affected the choice of the optimal subcooling evaporator temperature. These variables fell into two groups; heat exchanger size considerations, and refrigeration cycle temperature considerations.

Heat Exchanger Size Considerations

The sensitivity of the optimal subcooling evaporator temperature to the heat exchanger sizes (main cycle condenser, subcooling cycle condenser, subcooler) was explored with the property-dependent computer model. The evaporator UA size was not considered because it is constrained by the choice of refrigerated case. Changing the UA size of a heat exchanger not only affects the heat transfer in that component, but ultimately affects the performance of the entire system.

The consequence of changing the sizes of all three heat exchangers by the same amount was investigated. In this case, the UA product of all three heat exchangers was multiplied by a constant. By multiplying by a constant, the ratio of each heat exchanger to the total remained constant; regardless of the multiplier. In this way, the effect of the total heat exchanger size on the maximum COP point was studied. For this study, the heat exchanger UA products were increased and decreased by 33%. Referring to Figure 6, the COP curves are seen to increase with increasing heat exchanger UA as expected. However, the sizes of the heat exchangers do not affect the maximum COP point if the ratio of the heat exchanger sizes to the total remains constant.

The next problem that required investigation was whether the relative sizes of the heat exchangers affect the optimal subcooling evaporator temperature. In the standard model, the main cycle condenser size (UA) was 300% greater than the subcooling cycle condenser size (UA). Figure 7 was generated by decreasing the main cycle condenser UA by 300% and increasing the subcooling cycle UA by 300% . The inverted UA labeled on the graph represents the switch from the standard condenser UA's. With the main cycle condenser now being only one-third the size of the subcooling cycle condenser, the COP curves were shifted down by approximately 20%. However, the subcooling evaporator temperature at the maximum COP point was left virtually unchanged. This implies that the maximum COP point is not a strong function of the relative sizes of the condensers.

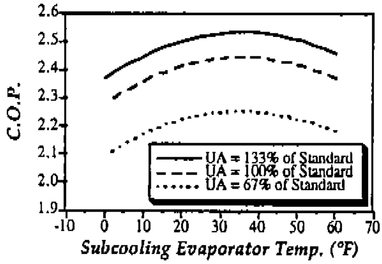

Figure 6: COP as a function of the subcooling evaporator temperature and the total UA product for the property-dependent dedicated subcooling cycle.

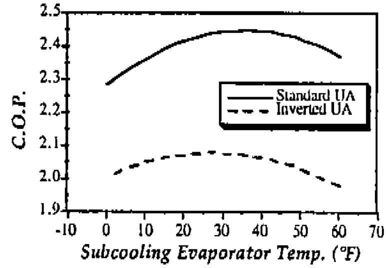

Figure 7: COP as a function of the subcooling evaporator temperature and the condenser UA's for the property-dependent dedicated subcooling cycle.

The ideal model, shows that the subcooler heat exchanger effectiveness (represented by x) had no effect on the optimal choice of the subcooler evaporator temperature. In the property-dependent case, the UA of the subcooler (which is an indirect measure of the heat exchanger effectiveness) is seen to have a slight effect on the choice of the optimal temperature. Referring to Figure 8, the maximum COP point is seen to increase with increasing subcooler UA. However the drift is minimal over a wide range of subcooler thermal sizes.

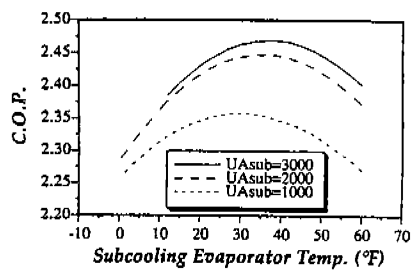

Figure 8: COP as a function of the subcooling evaporator temperature and the subcooler UA product for the property-dependent dedicated subcooling cycle.

Refrigeration Cycle Temperature Extreme Considerations

As predicted by the ideal model, the ambient and main cycle evaporator temperatures affect the choice of the optimal subcooling evaporator temperature. The optimum subcooling evaporator temperature was seen to increase with increasing main cycle evaporator and ambient temperatures (Figures 9 and 10). The result is that the optimum point fluctuates near the middle of the cycle extremes. However, the drift is slight over the normal range of operating temperatures.

Model Conclusions

The ideal dedicated subcooling model (equation 6) exhibits the same tendencies as a property-dependent computer model. The ideal cycle predicts the existence of an optimum subcooling temperature, the importance of the cycle extremes, and the relative unimportance of the heat exchanger's thermal performance on the optimum subcooling evaporator temperature. The ideal model in fact compares well with the property-dependent computer model; regardless of the cycle temperature extremes. Even at the upper extreme of main cycle evaporator temperature, the difference between ideal and property-dependent estimates was approximately four degrees. This four degree difference corresponds to a change in COP of less than 0.1%.

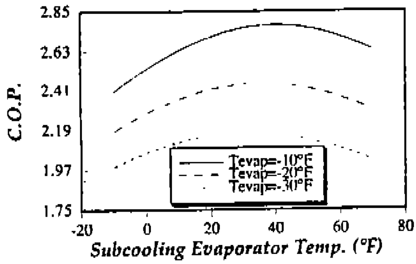

Figure 9: COP as a function of the subcooling and main cycle evaporator temperatures for the property-dependent subcooling cycle.

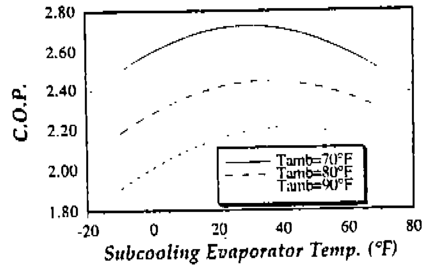

Figure 10: COP as a function of the subcooling evaporator temperature and the ambient temperature for the property-dependent dedicated subcooling cycle.

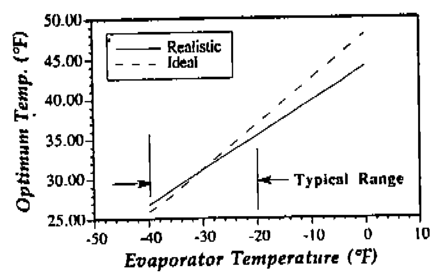

Figure 11: Optimum temperature comparisons between the ideal and property-dependent subcooling cycles as a function of the main cycle evaporator temperature.

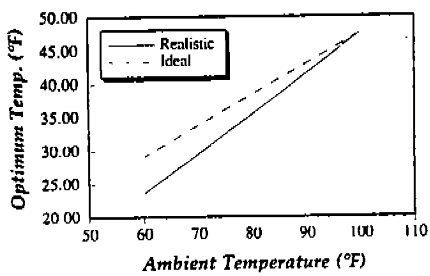

Figure 12: Optimum temperature comparisons between the ideal and property-dependent subcooling cycles as a function of the ambient temperature.

DESIGN STRATEGIES FOR DEDICATED SUBCOOLING SYSTEMS

Design considerations for the dedicated subcooling cycle were derived using the property dependent model. The design considerations were based on an ambient temperature of 80°F, a main cycle evaporator temperature of -20°F, and a subcooling evaporator temperature of 30°F. A subcooling evaporator temperature of 30°F represents a near optimal choice for all ranges of ambient and evaporator temperature (refer to Figures 5 through 10), due to the relative flatness of the COP curves as a function of the subcooling evaporator temperature near the optimal point.

For the earlier sections, the UA products of the three heat exchangers (main cycle condenser, subcooling cycle condenser, and subcooler) were set to values typical of standard practice; a small subcooler, and a subcooling cycle condenser that is a fraction of the size of the main cycle condenser. However, the question arises as to whether this is the optimal distribution of heat exchange area. This section investigates the optimum UA distribution, develops design guidelines, and evaluates these design guidelines over the range of operating conditions.

Since an increase in the total allocated UA product will lead to an increase in the overall cycle COP, the total UA product was constrained to allow the relative effects of heat exchanger distribution to be seen. (Note: this paper makes no attempt to determine the optimum total UA, the total allocated UA product should be determined by application and economics.) With three unknowns (the three heat exchanger thermal sizes) and one constraint (the total allocated UA product), the problem is reduced to a two-variable optimization.

At the optimum heat exchanger distribution, the subcooler represented approximately 10% of the total allocated heat exchange area; which corresponded to a subcooler effectiveness near 1.0. Also at the optimum distribution, the main cycle condenser was 3.3 times as large as the subcooling cycle condenser with effectiveness values of 0.665 and 0.717 respectively. Therefore, the subcooler is seen to be the critical heat exchanger in the dedicated subcooling cycle.

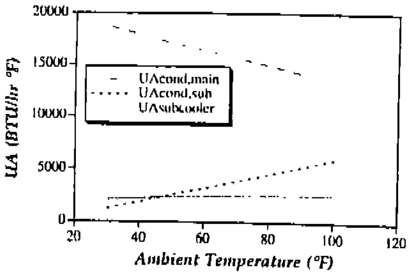

Figure 13 Optimum heat exchanger distribution as a function of the ambient temperature.

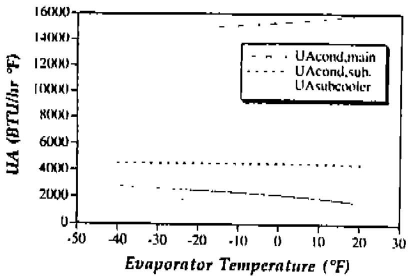

Figure 14 Optimum heat exchanger distribution as a function of the main cycle evaporator temperature.

The results , however, are based on standard conditions of 80°F ambient temperature and -20°F evaporator temperature. Therefore before any conclusions can be drawn about the optimum heat exchange distribution, the results have to be investigated with the changing cycle temperature extremes. As mentioned earlier, the ambient and evaporator temperatures were the only major factors influencing the choice of the optimum subcooling evaporator temperature. These temperatures are also the only major factors influencing the optimum heat exchange distribution, as seen in Figures 13 and 14. The heat exchanger distribution results can be summarized as:

- The optimum subcooler UA is unaffected by the choice of ambient temperature and only slightly affected by the choice of evaporator temperature.
- The ratio of main cycle condenser size to subcooling cycle condenser size decreases as the ambient temperature increases.
- The ratio of main cycle condenser size to subcooling cycle condenser size decreases as the evaporator temperature decreases.

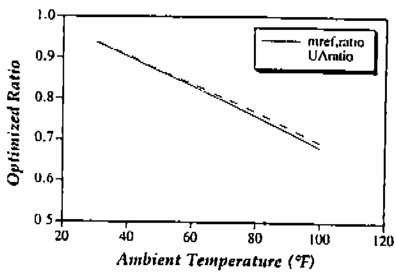

Figure 15 Optimized ratios as a function of the ambient temperature.

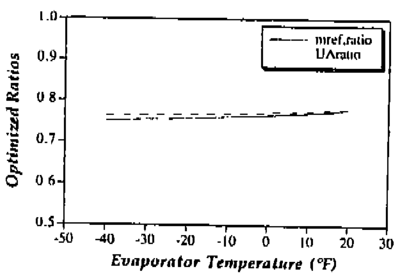

Figure 16 Optimized ratios as a function of the main cycle evaporator temperature.

Before continuing, the system ratios for the dedicated subcooling cycle must be defined. The ratio is defined as the amount of any variable in the main cycle divided by the sum of this variable in the main and subcooling cycles.

For example, the refrigerant flow rate ratio is defined as:

$$\dot{m}_{ratio} = \frac{\dot{m}_{ref,main}}{\dot{m}_{ref,main} + \dot{m}_{ref,sub}}$$

An important trend is revealed when the refrigerant flow rate ratio and the optimized UA ratio are plotted as functions of the ambient and evaporator temperatures. Figures 15 and 16 show that the optimum UA ratio (which is a direct measure of the optimum heat exchange distribution) closely matches the refrigerant flow rate ratio regardless of ambient or evaporator temperature.

Design guidelines for a dedicated mechanical subcooling system can be established using trends developed from the property-dependent computer model.

1) Select the total UA product available for the dedicated subcooling system based on economics.
2) Apportion approximately 10% of the allocated heat exchange area to the subcooler - corresponding to an effectiveness near 1.0.
3) Decide at which ambient temperature the system will most often run (the "design" temp.).
4) Distribute the remaining UA product according to the expected refrigerant flow rates at this "design" temperature.

CONCLUSIONS

An ideal mechanical subcooling cycle was developed from Carnot theory and heat transfer relations. This ideal cycle predicted the existence and location of the "optimum" subcooling temperature for the dedicated subcooling cycle. The ideal cycle also predicted that the "optimum" temperature was strongly dependent on the sink and refrigerated space temperatures, and weakly dependent on the subcooler heat exchanger parameters. A model of a property-dependent dedicated subcooling cycle was created. The property-dependent model took into account the irreversibilities due to compression, expansion, and heat exchange. The property-dependent model proved the trends predicted by the ideal model; the existence and location of an "optimum" subcooling evaporator temperature, the strong dependence on cycle temperature extremes, the weak dependence on subcooler heat exchanger parameters, and the relative unimportance of the condenser thermal sizes.

The property-dependent model allowed the optimal heat exchanger distribution to be developed and design guidelines to be established. The optimum heat exchanger distribution and design guideline for a dedicated subcooling cycle can be summarized as follows:

• Since the subcooler thermal size is relatively independent of ambient and main cycle evaporator temperatures, apportion 10% of the allocated UA product to the subcooler.
• Although the ratio of main cycle condenser thermal size to subcooling cycle thermal size <u>decreases</u> as the cycle temperature extremes increase, the ratio of the condenser UA's mirrors the ratio of refrigerant flow rates. Therefore, the optimal distribution of condenser thermal sizes will be in the same ratio as the design refrigerant flow rates.

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