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THERMODYNAMIC PERFORMANCE LIMIT AND EVAPORATOR DESIGN CONSIDERATIONS FOR NARM-BASED DOMESTIC REFRIGERATOR-FREEZER SYSTEMS

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ABSTRACT

Non-azeotropic refrigerant mixtures (NARMs) are investigated for a two-temperature level heat exchange process found in a domestic refrigerator-freezer. Ideal (constant air temperature) heat exchange processes are assumed. The results allow the effects of intercooling between the evaporator refrigerant stream and the condenser outlet stream to be examined in a systematic manner. For the conditions studied, an idealized NARM system has a limiting coefficient of performance (COP) that is less than that of the best performing pure refrigerant component. However, for non-ideal heat exchange processes (gliding air temperature), the NARM-based system has a higher limiting COP than a system running on either pure NARM component. Intercooling significantly affects the COP of NARM-based systems; however, depending on the location of "pinch points" in the heat exchangers, only one intercooling heat exchanger may be needed to obtain a NARM's maximum refrigerator COP. Three pairs, R22/R142b, R22/R123 and R32/R142b, were studied, but only the results for R22/R123 will be presented because of its unique temperature glide curvature.

Practical implementation of a Lorenz cycle constrains evaporator design. An evaporator module design is presented which meets the NARM system constraints. Preliminary NARM evaporator data are presented and compared to standard, pure refrigerant evaporators. The air side efficiency criterion indicates the feasibility of the NARM design, and the direction of further research is suggested.

INTRODUCTION

The U.S. domestic refrigerator industry is currently interested in methods that improve refrigerator performance in order to meet recently announced U.S. government energy standards [1]. Simultaneously, the refrigerator industry must choose satisfactory replacements for R12 (thermodynamic cycle fluid) and R11 (blowing agent for urethane foam). One of the options proposed for improving thermodynamic cycle efficiency with non-ozone depleting compounds is the use of Non-Azeotropic Refrigerant Mixtures (NARMs). Figure 1 is a schematic of a refrigerator utilizing a NARM based on the Lorenz cycle [2]. The main features in the system are two evaporators and two refrigerant-to-refrigerant heat exchangers (intercoolers). The heat exchange processes will be described in detail in a later section.

An early study in this area by Lorenz and Meutzner [2] indicated significant performance gains in an experiment that utilized a NARM in a modified domestic refrigerator. Subsequent work by Stoecker and co-workers [3,4,5,6,7,8] examined NARMs through numerical simulation and experiment. While simulation results show improvements to cycle efficiency, the experiments were unable to quantitatively prove significant performance gains. Actual experiments are difficult because the compressor and other components have to be shifted to non-optimal conditions as mixture compositions are changed.

The emphasis of this work is an idealized examination of NARMs in a refrigerator heat exchange configuration and development of a preliminary design for a practical NARM evaporator. The heat transfer investigation is ideal in two ways. First, the air side of the evaporators is assumed to be at a constant temperature level. This is equivalent to assuming that the air flow rate through the evaporator is large (in terms of its energy capacitance) compared to the overall evaporator heat transfer coefficient. Second, it is assumed that all heat exchange processes continue to the limit allowed by the second law of thermodynamics (infinite heat exchange area). That is, heat exchange in a particular section can occur until a 'pinch point' of zero temperature difference is obtained between two fluid streams exchanging thermal energy.

The purpose of the idealized study is to find an upper limit for NARM refrigerator performance and to compare this performance to that of a pure refrigerant. The results can then be used to examine the limiting factors of 'real' systems in which changing air temperature, non-ideal heat exchange, pressure drops, and other effects detract from system performance. The previous studies confined the examinations to finite heat exchange processes. The question remains, will a NARM system perform as well as a pure refrigerant if the heat transfer is not restricted to non-ideal performance? Traditionally, idealizing the heat transfer processes has been used only to evaluate the thermodynamic merit of a system.

Figure 2 illustrates the main differences between a pure refrigerant and a NARM in a two-temperature evaporator process. No refrigerant-to-refrigerant heat exchange is shown in the NARM system for simplicity. When the air-side heat exchange process is such that no significant drop in air temperature occurs across the evaporator (the graphs on the left), the pure refrigerant is able to move through the low temperature evaporator with a very small temperature difference between it and the air. A significant temperature difference is realized between the pure refrigerant and the air in the high temperature evaporator section. A NARM system will tend to have a pinch point in its heat exchange with the air at either the exit of the low or high temperature evaporators. The pinch point depends on the slope of the NARM's temperature glide, the enthalpy change and the fraction of energy exchange allocated to the low and high temperature evaporators.

Non-ideal air heat transfer processes will tend to tilt the air temperature line as energy is removed (the graphs on the right). A pure refrigerant will experience a drop in performance because the change in air temperature forces a pinch point to occur at the air-side exit (evaporator inlet) of the low temperature evaporator. A NARM-based system, however, does not necessarily experience a drop in performance as long as the tilt in the air side temperature (in either high or low temperature evaporators) does not exceed that of the NARM. As will be discussed, the studied NARMs are poorer than pure refrigerants in terms of thermodynamic performance (ideal heat exchange); however, as the heat exchange process becomes less ideal, a pure refrigerant's performance can be reduced to a level lower than that of the NARM. The improvement of air-side heat exchanger performance (to compensate for lower, non-ideal air flow rates) is achieved at the expense of higher air flow rates (greater internal fan heat loading) and larger heat exchanger areas.

Simulation of Refrigerator Performance

A numerical simulation model has been developed which utilizes thermodynamic property information from an equation-of-state subroutine library developed by the National Institute of Science and Technology [9]. In Figure 1, the section within the dotted line is modeled. The simulation program successively adds tube lengths to each section of the heat exchangers until both the desired amount of heat transfer is obtained in each evaporator and the system performance can no longer be increased due to thermodynamic (second law) limitations.

The components shown in Figure 1 represent the heat exchangers that are generally considered necessary for optimal operation of a NARM system. The two refrigerant intercoolers that exchange energy between the condenser outlet refrigerant and the refrigerant in the evaporator sections help to match the refrigerant temperature with the air temperature in the low and high side evaporators. Allocation of optimal heat exchange in the intercoolers is dependent on the NARM mixture and the operating conditions.

Three refrigerant pairs, R22/R142b, R32/R142b and R22/R123, were chosen based on screening runs that indicated favorable performance assuming non-ideal heat exchange [10]. The results for R22/R123 will be the only ones presented. The temperature glide curvature of R22/R123 was such that significant performance gains could be realized by the implementation of intercooling; whereas, the R22/R142b and R32/R142b pairs temperature glide did not have much curvature and intercooling performance gains were not significant. For all of the simulation runs, it was assumed that the low air-side temperature was -15 C (freezer) and the high air-side temperature was 5 C (fresh food). All cases were assumed to exit the condenser as a saturated liquid at 30 C. The evaporation pressure was maintained at a constant level with the exit condition of the refrigerant assumed to be a saturated vapor. The COP was calculated based on the energy exchange in the evaporators and the isentropic work required to compress the refrigerant to the condensing pressure. Although the refrigerant was assumed to exit as a saturated vapor without any superheating, the addition of a superheat region energy exchange to the overall energy exchange process are not expected to affect the trends observed in this study. Superheat studies will be investigated later.

An important factor in the thermodynamic performance of a NARM is the distribution of energy exchange in the low and high temperature evaporators. Three evaporator loadings were assumed in which the freezer requires 25, 50, and 75 percent of the total refrigerator energy load. It can be observed from Figure 2 that a pure refrigerant's COP will not change as the evaporator loading is varied between the freezer and the freshfood sections.

The simulation results presented in the following section examine the NARM system COPs with and without refrigerant intercoolers. The successive addition of refrigerant intercoolers to the process allows one to examine the significance of the intercoolers and to assess the amount of energy transfer in the intercooler relative to the desired energy transfer in the evaporators.

Simulation of Results

A variety of simulation results have been produced based on desired evaporator loadings and intercooler combinations. Four intercooler combinations have been simulated: no intercoolers (intercooler configuration designation "CASE 00"), an intercooler between the low and high temperature evaporators only ("CASE 10"), an intercooler at the end of the high temperature evaporator only ("CASE 01") and two intercoolers with placement as shown in Figure 1 ("CASE 11"). After discussing the effects of the various intercooler combinations on system performance, a set of temperature/enthalpy diagrams will be used to describe the details of the energy exchange processes.

Figure 3 illustrates the change of COP for R22/R123 simulated over a range of mixture compositions for all four cases mentioned above. The three curves on each graph represent freezer load fractions of 25, 50, and 75 percent. For CASE 00 (no intercoolers), the combination is very sensitive to composition and evaporator loading because it has a relatively steep temperature glide that can pinch the heat exchange process in either the high or low temperature evaporator section. The COP at intermediate compositions is lower than the COP of either pure refrigerant. For CASE 10 (intermediate intercooler only) significant performance improvements are realized because the intercooling lowers the temperature of the refrigerant from the condenser outlet and allows the evaporator pressure to be increased. The intercooling is significant enough to raise the COP of the 25 percent freezer load above that of the pure refrigerant with the lower COP (R22). The 50 and 75 percent freezer loadings are lower than both pure refrigerant COPs for all mixture combinations. CASE 01 (end intercooler only) shows some interesting effects caused by a switch in the heat transfer pinch points in the process. Freezer loadings of 25 and 50 percent have higher COPs with an intercooler at the end of the high temperature evaporator than the COPs found for an intercooler placed between the two evaporators. CASE 11 (two intercoolers) showed a higher COP for cases in which heat exchanger pinch points occurred at the end of both evaporators.

The occurrence of a pinch point at the exit of both evaporators is a unique condition and further discussion will describe the factors governing the performance variations of a NARM system. A detailed investigation revealed that at a 65 percent R22 composition, the pinch point shifts from the high temperature evaporator to the low temperature evaporator as the freezer load changes from 25 to 75 percent. Figure 4 shows the shifting of the pinch point for the case in which no intercoolers are present. At a freezer loading of 50 percent, the exits of both high and low evaporators are pinched.

Freezer loadings less than 50 percent (with no intercoolers) have the same COP because the refrigerant's temperature variation with energy exchange remains the same. Freezer loadings greater than 50 percent reduce the COP because the pinch point at the exit of the low temperature evaporator causes the entering temperature of the refrigerant to be reduced (resulting in lower evaporator pressures).

Figure 5 illustrates the effect of adding intercoolers to the system with a mixture composition of 65 percent and a freezer loading of 50 percent. Adding an intercooler between the low and high temperature evaporators at this condition does not cause any improvement to the system's COP. Comparing this case to the case with no intercoolers in Figure 4, the end of the high temperature evaporator remains as the pinch point of the system. Therefore, the effect of the intermediate intercooler for this case is that it recycles some of the energy from the condenser to the intermediate saturation region. The energy exchange of the low temperature evaporator occurs in a somewhat lower quality region. On the practical side, the intermediate heat exchange causes a higher temperature difference to occur between the low temperature air and the refrigerant which would result in a smaller heat exchanger.

Alternatively, adding an intercooler to the exit of the high temperature evaporator causes a significant increase in COP. Although the intercooler at the end of the high temperature evaporator (see Figure 5) can only bring the high pressure liquid down to 5 C, the heat transfer process with the low pressure refrigerant increases its exit temperature from 5 C to 10 C. Finally, adding both intercoolers allows an additional increase in COP because the intermediate intercooler can slide both evaporator pinch points towards the liquid region. This situation results from the curvature of the NARM temperature glide, the temperature levels of the evaporators, and the load split of the freezer and the fresh food sections.

Additional simulation and testing will be performed to further investigate the practical limits to NARM system performance. At the present time, a facility is being constructed which will test NARM Lorenz cycle system performance with two evaporators. Computer control of the system components will allow quick characterization of overall system performance.

Practical Implementation of a NARM System

Practical implementation of a NARM-based domestic refrigerator system requires several design considerations which are not encountered with existing pure refrigerant systems. First, the evaporator temperature glide must be preserved to achieve the potential thermodynamic benefits that a NARM system offers. Practically, this means that each serpentine tube in the evaporator must be thermally isolated from the others. Second, separate freezer and fresh food counterflow evaporators will be required to match air and refrigerant temperature glides in each compartment. Finally, two refrigerant intercoolers may be needed to assist in matching the performance-enhancing temperature glides. Refrigerator manufacturers will be reluctant to use a NARM evaporator that is more complex (and thus more costly and less reliable) than current evaporators, so a single module NARM evaporator design with few refrigerant tube connections is imperative. None of the above considerations are encountered when designing evaporators for pure refrigerant systems. The current technology provides evaporators which exhibit continuous plate fins with "interference fit" tube/fin contact, a single evaporator for both compartments, and refrigerant tubing without intercoolers.

Current research efforts are to analyze the performance of a NARM evaporator design which will meet the preceding constraints. As shown in Figure 6, corrugated fins are epoxied to refrigerant-carrying tubes in a manner that prevents thermal communication between adjacent tube passes, thus providing a cross-counter flow geometry and preserving the refrigerant temperature glide. A freezer evaporator, a fresh food evaporator and two refrigerant intercoolers are provided within the single module. One potential benefit of this design is improved fin/tube contact over the "interference fit" contact found in single refrigerant system evaporators. Another benefit is that staggering adjacent fin rows may increase convective heat transfer by disrupting the boundary layers which form along the fins. Finally, the NARM evaporator design may allow the use of thinner fin material because the rigidity of the module will not be dependent upon fin stiffness.

Before evaluating the NARM evaporator design, a performance criterion must be found that will compare simulation results with previous research on standard evaporators. The criterion chosen must account for contact resistance between the fins and the refrigerant tube which can contribute as much as 22% of the overall resistance to heat transfer in a standard domestic refrigerator evaporator [11].

Manzoor et. al. have examined the effect of gaps between the fins and a base surface in terms of a heat transfer augmentation factor [12]. Other studies have examined the effect of differential fin/tube thermal expansion rates on heat exchanger performance [13]. Recently, studies on the effects of mechanically expanded refrigerant tubing on collared fin heat exchanger performance have been conducted by Nho and Yovanovich [14]. Since NARM evaporator designs must meet different constraints than standard evaporators, no research is available in which an attempt was made to quantify the performance of an evaporator similar to the design proposed in this paper. Thus a design criterion which would be applicable to the NARM evaporator design was developed.

A parameter often used to quantify fin performance is fin efficiency, η_{fin}, which is defined as $\eta_{fin} = \frac{q}{q_{maxfb}}$ where q is the actual fin heat transfer, and q_{maxfb} is the fin heat transfer that would occur if the entire fin were at the fin's base temperature [15]. The fin/tube contact resistance in standard refrigerator evaporators indicates that a significant temperature difference may exist between the base of the fin and the refrigerant tube. With this in mind, the fin efficiency, η_{fin}, is an inadequate method of measuring evaporator air side performance. A fin could be 100% efficient yet be in poor thermal communication with the refrigerant due to the high fin/tube contact resistance, resulting in low evaporator heat transfer and poor evaporator performance. Because it neglects the effect of fin/tube contact resistance, fin efficiency is a poor measure of evaporator performance.

A more appropriate criterion for evaluating the performance of an evaporator may be an overall air side efficiency. The air side efficiency, η_{air}, is defined as $\eta_{air} = \frac{q}{q_{maxt}}$ where q is the actual heat transfer from the fin, and q_{maxt} is the fin heat transfer that would occur if the entire fin were at the refrigerant tube's external temperature. For the thermal circuit shown in Figure 6, η_{air} is given by

$$\eta_{air} = \frac{1 - \phi}{1 + \frac{R_{cond}}{R_{conv}}} \text{ ; where } \phi = \frac{R_{glue}}{R_{total}} .$$

The ratio $\frac{R_{cond}}{R_{conv}}$ is proportional to a fin Biot number, and ϕ gives an indication of the magnitude of the glue resistance to heat transfer relative to the total resistance to heat transfer from the fin. The air side efficiency, η_{air}, is plotted as a function of ϕ and $\frac{R_{cond}}{R_{conv}}$ in Figure 7.

The curve corresponding to $\phi = 0$ in Figure 7, is the upper limit to air side efficiency which could be achieved if perfect thermal contact existed between the fin and the refrigerant tube ($R_{glue} \rightarrow 0$). The introduction of fin/tube resistance increases ϕ and decreases η_{air}. If the fin/tube contact resistance is increased to the point where it dominates all other resistances ($R_{glue} \rightarrow \infty$), $\phi \rightarrow 1$, and $\eta_{air} \rightarrow 0$. The entire fin is at ambient temperature because it is perfectly insulated from the refrigerant tube; thus, no heat transfer exists between the fin and the environment, and the air side efficiency goes to zero. Air side efficiency is a more appropriate measure of evaporator performance than fin efficiency because η_{air} accounts for both the fin to air convective resistance and the fin to tube conductive resistance. For this reason, air side efficiency was used as the performance criterion when evaluating the proposed NARM evaporator design.

O'Neill [11] studied the performance of standard refrigerator evaporators and obtained estimates of the magnitude of each resistance shown in the resistance network of Figure 6. Note that O'Neill assumed the conductive resistance through the fin was zero because the fins were isothermal to within 0.1°C. The operating point for O'Neill's standard evaporator is plotted on Figure 7. Also, if the fin/tube contact resistance was eliminated in a standard evaporator, the total resistance of the fin to heat transfer would be reduced by 33%.

Tests have been conducted on a single pass of a NARM evaporator. Heat generated inside a 0.9525 cm (3/8 in.) aluminum refrigerant tube by an electric heater was dissipated by corrugated aluminum fins epoxied to the refrigerant tube. After the test section was placed in a wind tunnel, fin temperature measurements and an overall heat balance on the system were used to estimate the resistances on a per contact point basis for the NARM evaporator prototype. The results are plotted in Figure 7 for comparison to the standard evaporator. If the glue resistance in the NARM evaporator were eliminated, the total resistance to fin heat transfer would be reduced by approximately 30%. Based on the air side efficiency criterion, the NARM evaporator appears to be feasible as it performs about the same as a standard evaporator. However, the NARM fins have a higher $\frac{R_{cond}}{R_{conv}}$ than standard fins because thinner fin material was used. Since the NARM evaporator data is preliminary, further studies are necessary. The effect of various epoxies and fin geometries on the NARM evaporator module heat transfer and the economics of NARM evaporator production will be investigated.

CONCLUSIONS

The previous discussion indicates that on an ideal thermodynamic basis in which a refrigeration system is operating with two constant temperature levels, NARMs do not offer a performance advantage the better performing pure refrigerant of the pair. NARM performance can vary substantially based on the temperature levels, energy loading, and composition of the NARMs. A careful examination of the pinch points that limit the thermodynamic performance of the system must be studied in order to understand the factors limiting the system's performance. The R22/R123 system is an example of a NARM that can either perform very well or very poorly. Due to the curvature of the temperature glide, the R22/R123 system performance is strongly dependent upon mixture concentration and would have little tolerance for improper charging.

One must consider how real heat transfer processes would affect system COP (as described in previous studies) with finite heat exchange area. Considering the NARM system in Figure 5 with two intercoolers (COP = 4.56), the temperature drop of the air in the freezer is the most restrictive. The temperature of the air stream can drop approximately 4 C across an evaporator at typical air flow rates. For the NARM system shown in Figure 5, the temperature change would not cause a decrease in the system's performance; however, this change would affect the size of the heat exchanger. On the other hand, the COP of a pure R123 system is reduced from 4.92 to 4.38 when the air temperature is reduced from -15 C to -19 C across the evaporator; thus, the NARM system can perform better than a pure refrigerant system for finite air flow rates.

Several constraints control NARM evaporator design, and a design which meets these constraints has been presented. This design has thermally isolated adjacent tube passes, separate freezer and fresh food evaporators and two intercoolers contained in a single module. Preliminary testing has shown that the proposed NARM evaporator will perform as well as standard evaporators based on the air side efficiency criterion for evaluating evaporator performance.

The question of whether NARMs offer a potential means of increasing a refrigerator's performance must be balanced by the practical considerations of heat transfer in the circulating air stream in the cabinet and the configuration/size of the evaporators. Space and cost considerations tend to reduce the size of the evaporator while internal heating of the cabinet by the fan motor adds to the overall load of the system. These practical limitations must be considered before concluding that a NARM system has thermodynamic benefits over a pure refrigerant system.

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Figure 1 Schematic of the components in a NARM Lorenz cycle for a domestic refrigerator.

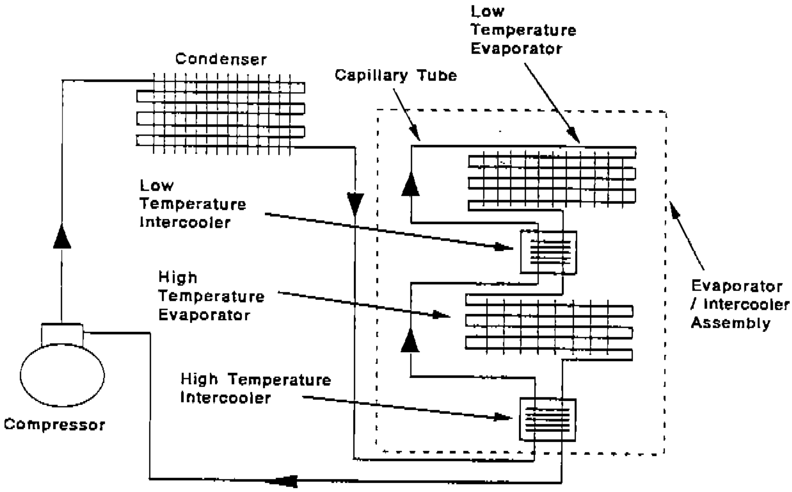

Figure 2 Temperature-enthalpy plot of ideal and non-ideal air-side heat exchange processes for a pure and a NARM refrigerator system.

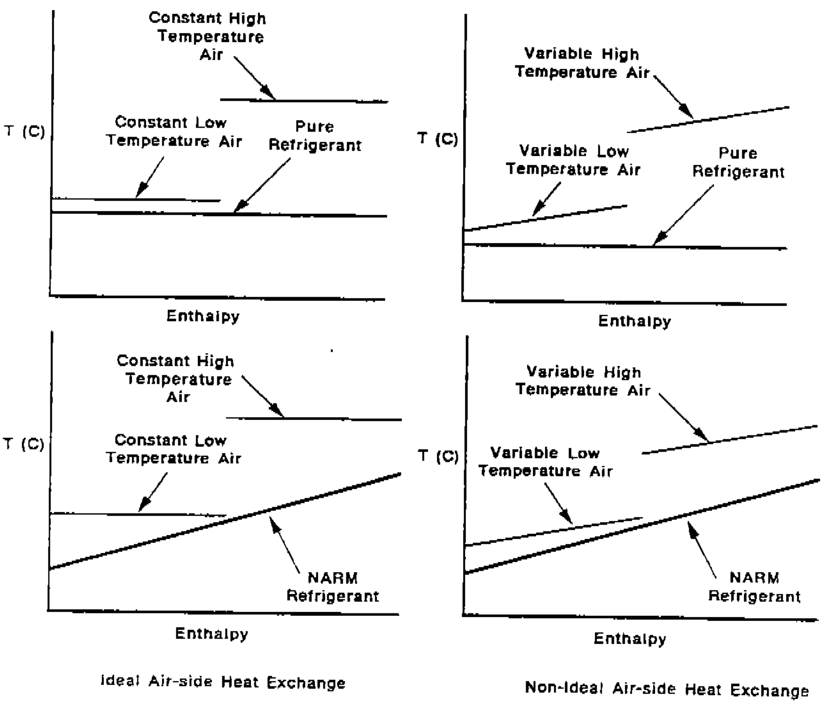

Figure 3 R22/R123: COP vs. Concentration

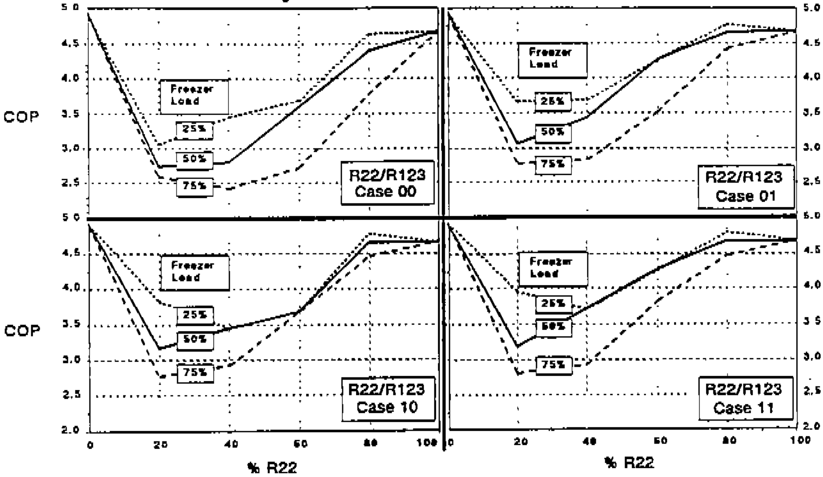

Figure 4 R22/R123, 65% R22 No Intercoolers

Figure 5 R22/R123, 65% R22 Intercooler Combinations

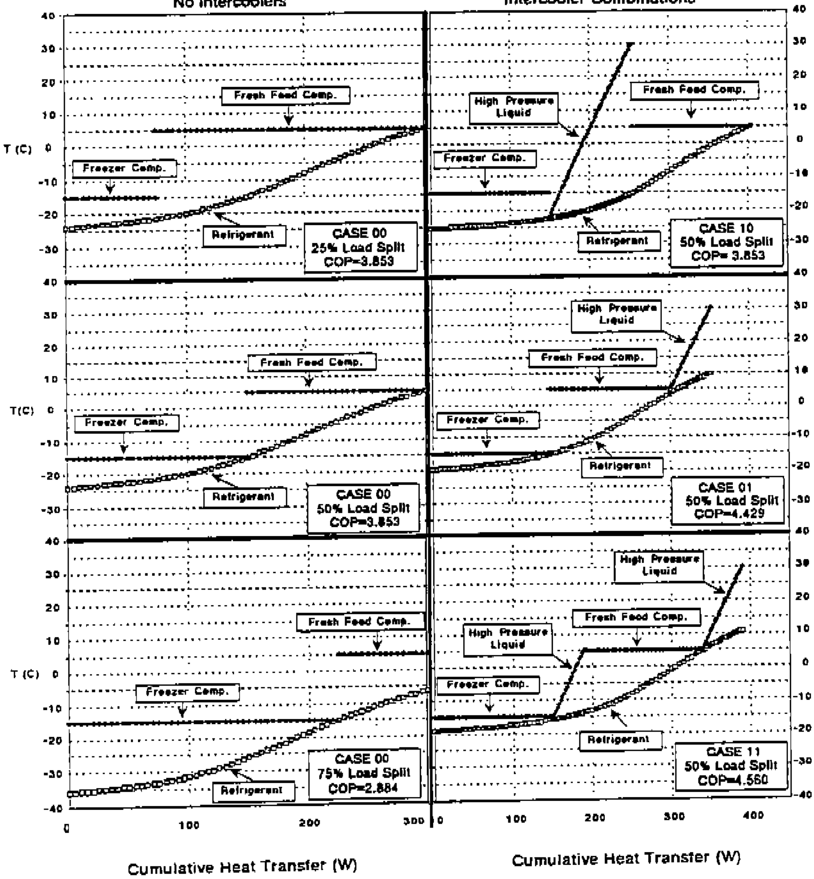

Figure 6 Proposed NARM Evaporator Design and Thermal Circuit

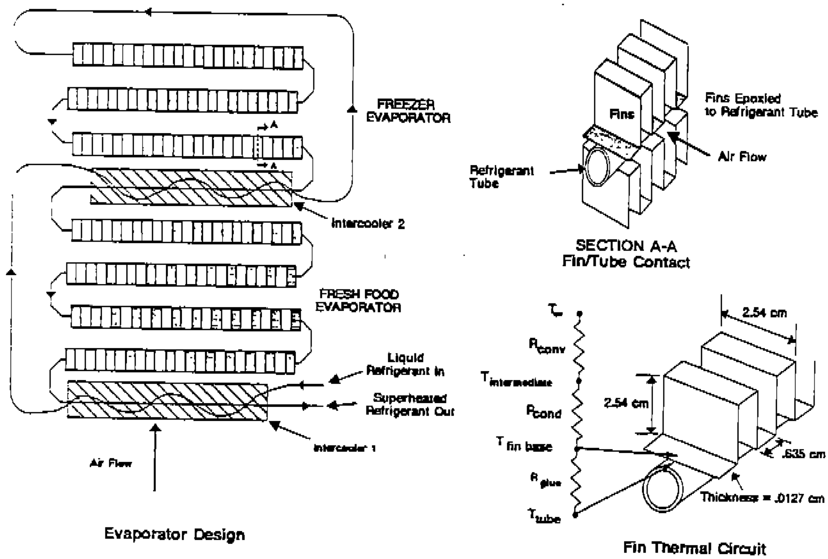

Figure 7 Plot of air side efficiency vs. ϕ and Rcond/Rconv