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# INITIAL PARAMETRIC RESULTS USING CYCLEZ—AN LMTD-SPECIFIED, LORENZ-MEUTZNER CYCLE REFRIGERATOR-FREEZER MODEL

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## ABSTRACT

A computer model representing a two-evaporator, two-intercooler refrigerator-freezer operating at steady-state with nonazeotropic refrigerant mixtures (CYCLEZ) has been developed at Oak Ridge National Laboratory (ORNL). This model is being used to assess the effects of system design and operating parameters on the cycle performance of a refrigerator-freezer designed around the Lorenz-Meutzner (L-M) circuit. Separate evaporators for the freezer and fresh-food compartments are modeled, as well as two intercoolers that subcool liquid refrigerant from the condenser by heat transfer with low-pressure refrigerant. The CYCLEZ refrigerator/freezer model is derived from the CYCLE7 heat-pump model developed originally by the National Institute of Standards and Technology (NIST). CYCLEZ currently uses the Carnahan-Starling-DeSantis (CSD) equation-of-state to compute refrigerant thermodynamic properties, so that new refrigerants can easily be added.

Condenser and evaporator heat-exchanger performance are defined by user-specified *overall* LMTDs which allow equivalent heat-exchanger sizing *per unit refrigeration load* to be maintained for different refrigerant mixtures. A more consistent formulation of *overall* heat-exchanger LMTD is applied across the condenser superheated and two-phase regions as well as over the two evaporators. Source and sink conditions are specified in terms of inlet and outlet temperatures of the external fluid streams. Intercooler high-side (subcooling) $\Delta$Ts and relative fresh-food-to-freezer load ratio are also user-specified. These features make this model well suited for evaluating the optimal thermodynamic cycle requirements of the five heat exchangers used in the L-M refrigerator/freezer circuit.

Parametric investigations involved nineteen ozone-safe refrigerant combinations which are under consideration for this application. Effects of the distribution of heat-exchanger area, extent and distribution of intercooler subcooling/superheat, refrigerant mixture composition, and relative refrigerator/freezer loading were investigated. The model indicates that improvements of 10 to 20% in COP (coefficient of performance) are possible using mixed refrigerants in this cycle configuration compared with the standard refrigerator/freezer circuit using R12. Most of this improvement is due to closer matching between the air and refrigerant temperature profiles across the evaporators. However, intercooler subcooling also results in a decreased pressure ratio across the compressor.

## INTRODUCTION

Global environmental concerns have served to place restrictions on the production and sale of chlorine-containing, fully halogenated compounds (CFCs) [1,2]. Commercial production of refrigerant 12 (R12), which is extensively used in the refrigerating circuit of household refrigerator-freezers (RFs), and refrigerant 11 (R11), used as a blowing agent for the insulating foam, is likely to be phased out by the year 2000.

Replacements for the refrigerants and blowing agents used in this application must be found, and energy efficiency has to be a primary consideration. Using nonazeotropic refrigerant mixtures (NARMs) as refrigeration fluids can improve the efficiency of vapor-compression refrigeration equipment at the expense of circuit and hardware redesign [3].

The application of NARMs in domestic refrigerator-freezers has been suggested and experimentally investigated with varied results [4,5]. An RF circuit described and tested initially by A. Lorenz and K. Meutzner [6] in 1975 and by H. Kruse [7] in 1981 has verified the promise shown by NARM refrigerants. The CYCLEZ RF program was developed at ORNL for modeling the Lorenz-Meutzner circuit with newer, environmentally acceptable NARM refrigerants.

### Background

A fundamental paper on methods for the analytical comparison of pure and mixed refrigerants in vapor-compression cycles was published in 1987 by McLinden and Radermacher [8]. They concluded that meaningful comparisons between pure and mixed refrigerants should include the application

temperatures and required temperature changes of the external heat-transfer streams. In addition, a constant total heat-exchanger surface area *per unit of output capacity* for pure and mixed refrigerants was recommended for the rigorous evaluation of comparative cycle performance.

McLinden and Radermacher further observed that modeling heat exchangers with equal LMTDs gave nearly equivalent results (for a simple heat-pump cycle with equal glides) compared to the more rigorous requirements they set forth [8]. This was the basis of the CYCLE7 heat pump model, which has been widely used for screening candidate refrigerant mixtures for different applications [9,10]. Recently, however, McLinden has noted some of the limitations of the CYCLE7 model when used for refrigerant performance screening in refrigerator cycles [11].

A number of models specifically intended for the L-M cycle are presently under development at the U. of Hannover, the U. of Maryland, the U. of Illinois at Urbana-Champaign, Purdue, NIST, and Arthur D. Little, Inc. Based on the information available to the authors, a mix of heat-exchanger approaches and levels of complexity are being used—ranging from assumptions of infinite heat-exchanger area to UA-specified-LMTD-calculated and heat-exchanger effectiveness/NTU-based methods.

The approach taken in the present work was to adapt the CYCLE7 model to the L-M refrigerator/freezer application while preserving and enhancing the unique advantages of the original LMTD-based formulation. Consistent definitions for *overall* evaporator and condenser LMTDs were developed and a method was devised to maintain constant total heat-exchanger area *per unit of refrigeration capacity.* The sequential solution technique of successive substitution used in CYCLE7 was extended to incorporate the two high-side-to-low-side intercoolers and the split evaporator.

The program input requirements, *overall* LMTD definitions, solution methodology, and program output are described, as well as a summary of the perceived advantages of the resultant model. This is followed by a review of initial parametric investigations using CYCLEZ; effects of the distribution of total heat-exchanger area, relative RF loading, refrigerant composition, and extent and distribution of intercooler subcooling/superheat are considered.

## MODEL DESCRIPTION

### Model Input Requirements

The user-supplied input to the CYCLEZ program was kept similar in form to that of CYCLE7 [9]. Both programs require the user to select two refrigerants (which can be the same), a mixture interaction parameter, a mass fraction concentration, inlet and exit temperatures of the external heat-transfer streams, condenser and evaporator LMTDs and total pressure drops, and a compressor isentropic efficiency. The CYCLEZ program, however, provides for user specification of separate evaporators for the freezer (EL) and fresh-food (EH) compartments and low- and high-temperature intercoolers (ICL and ICH) as shown schematically in Figure 1. (All heat exchangers are presently assumed to be pure counterflow.)

For the two evaporators, the individual source-stream inlet and exit temperatures are now required as well as the fresh-food-to-freezer load ratio. As in CYCLE7, a single evaporator LMTD is still specified; however, in CYCLEZ this now refers to an *overall or combined* LMTD of the two evaporators. Specifying the *overall* evaporator LMTD rather than LMTDs for the individual heat exchangers allows the degree of freedom needed for each evaporator LMTD to change with different refrigerant glides while the combined value is held constant. The inverse of this combined value is equal to the overall evaporator area *per unit of total refrigeration capacity* (based on the combined refrigeration done in both evaporators).

The high- and low-temperature intercoolers are specified simply by providing the desired, individual high-side subcooling $\Delta$Ts. An ambient heat-exchange option is also included to allow the user to specify, in place of a subcooling $\Delta T$ for the high-temperature intercooler, an ambient temperature which both fluid streams of the high-temperature intercooler shall approach. This option represents a passive heat-exchange assumption (with no high-side-to-low-side heat transfer), while the specified subcooling choice is an active heat-interchanger approach. The low-temperature intercooler is defined the same way in either case.

For the condenser, the specified LMTD differs from that in CYCLE7 in two ways. First, the equation used in CYCLE7 to determine an *overall* condenser LMTD across superheated and two-phase refrigerant regions has been modified. Second, the *overall* LMTD is redefined relative to the *total refrigeration capacity* in place of the condenser heat load. This allows the user to establish a more generalized heat-exchanger loading, as will be further discussed.

The CYCLEZ program also allows the user to specify compressor-shell heat loss as a fraction of the computed compressor input power. This results in more reasonable estimates of compressor discharge temperature.

Overall LMTD Definitions

Condenser. The equation used to determine the *overall* LMTD across the superheated and two-phase refrigerant regions of a condenser is developed as follows. From the basic LMTD relationship of Q = UA*LMTD and the refrigerant-side energy balance of

$$Q_{cond} = Q_{tp} + Q_{sup}\,, \tag{1}$$

an equation for the overall condenser LMTD can be written as

$$LMTD_{cond} = \frac{(UA)_{tp} \cdot LMTD_{tp}}{(UA)_{cond}} + \frac{(UA)_{sup} \cdot LMTD_{sup}}{(UA)_{cond}}\,, \tag{2}$$

where the subscripts "cond", "tp", and "sup" refer to overall condenser, two-phase, and superheated refrigerant regions, respectively. With the assumption that the overall conductance U across either region of the heat exchanger is the same, and because $A_{cond} = A_{tp} + A_{sup}$, then $(UA)_{cond}$ in Eq. 2 can be replaced by $(UA)_{tp} + (UA)_{sup}$. Replacing the UAs for each region in Eq. 2 with individual Q/LMTD equivalents, and then rearranging gives the equation used in CYCLEZ, where

$$LMTD_{cond} = \left[\frac{1-f_{sup}}{LMTD_{tp}} + \frac{f_{sup}}{LMTD_{sup}}\right]^{-1} \tag{3}$$

and the superheat heat-transfer fraction $f_{sup} = Q_{sup} / Q_{cond}$. It should be noted that Eq. 3 differs from that used in CYCLE7, where $LMTD_{cond}$ is defined as

$$LMTD_{cond} = (1 - f_{sup})*LMTD_{tp} + f_{sup}*LMTD_{sup}\,. \tag{4}$$

Equation 2 can also be rewritten as

$$LMTD_{cond} = (1 - A_{f,sup})*LMTD_{tp} + A_{f,sup}*LMTD_{sup}\,, \tag{5}$$

where $A_{f,sup} = A_{sup}/ A_{cond}$. A comparison of Eqs. 4 and 5 shows that the CYCLE7 equation (Eq. 4) effectively weights the individual LMTDs by the *relative heat transfer* in each region, while in CYCLEZ (Eq. 5) the required *relative heat-exchanger area* is used as the weighting factor to more closely approximate the UA ratios of Eq. 2 . The net effect of this difference is that the much larger LMTD across the superheat region has less weight in CYCLEZ in determining $LMTD_{cond}$, because the required superheat-area fraction $A_{f,sup}$ is much smaller than the relative heat-transfer fraction $f_{sup}$ .

Equation 3 for the overall LMTD of the condenser represents the *inverse* of the overall condenser sizing given by $UA_{cond}/Q_{cond}$. To give the user control of the condenser size per unit *refrigeration output* rather than *condenser output*, the user-specified LMTD value for the condenser was redefined as the normalized quantity given by

$$LMTD_{cond,norm} = LMTD_{cond} * Q_{evap}/Q_{cond}\,. \tag{6}$$

The superheat-area fraction $A_{f,sup}$ is given by

$$A_{f,sup} = \frac{f_{sup}}{LMTD_{sup}}\left[\frac{f_{sup}}{LMTD_{sup}}+\frac{1-f_{sup}}{LMTD_{tp}}\right]^{-1} \quad . \tag{7}$$

The condenser superheat-area fraction from Eq. 7 is used in CYCLEZ to apportion the specified high-side pressure drop between the condenser two-phase and superheated refrigerant regions. (In CYCLE7, this apportioning was done with relative heat-transfer fractions.) The contributions of the low- and high-temperature intercoolers to the total high-side pressure drop are neglected.

Split-Evaporator. Similar equations for the *overall* evaporator LMTD were derived by replacing the superheated and two-phase condenser regions in Eqs. 1–5 , respectively, with the low- and high-temperature evaporators, substituting $Q_{evap} = Q_{el} + Q_{eh}$ for $Q_{cond}$ , and replacing $f_{sup}$ with $f_{el} = Q_{el}/Q_{evap}$. The user-specified heat-load ratio of the fresh-food–to–freezer compartments, $R_{h-l}$ , is related to $f_{el}$ by $f_{el} = 1/(1+R_{h-l})$ .

The area ratio between the low- and high-temperature evaporators is also calculated by an equivalent form of Eq. 7. The apportioning of the specified low-side pressure drop is slightly different from that of the condenser because the relative area of the low-temperature intercooler is also included. The contribution of the high-temperature intercooler to the low-side pressure drop is neglected.

Relation to Constant Total Hx Area. By fixing the values for *overall* evaporator and condenser LMTDs as defined here, the total primary heat-exchange surface *per unit refrigeration capacity* is held constant. This can been seen from the following equation, where $UA_{tot}$ is given by $UA_{evap} + UA_{cond}$ , and

$$UA_{tot}/Q_{evap} = 1/LMTD_{evap} + 1/LMTD_{cond,norm} \, . \tag{8}$$

Solution Methodology

Cycle Assumptions. As in the case of the CYCLE7 model, assumptions are made of saturated liquid at the condenser exit and saturated vapor at the high-temperature evaporator exit. These assumptions are consistent with the evaluation of optimal performance configurations for the L-M cycle, because the most beneficial levels of high-side subcooling and compressor-inlet superheat can be achieved with the two intercoolers. Using the condenser for subcooling and the high-temperature evaporator for superheating serves only to unnecessarily raise the condenser pressure or lower the evaporator pressure, respectively, for a given LMTD. While predicting these off-design situations is necessary for a *simulation* model capable of representing the full range of operating conditions, the present model is better suited as an initial *design* model to define the hardware performance requirements of an optimal configuration.

Sequential Solution. Cycle specification in terms of the heat-exchanger exit-state assumptions and the user input is sufficient to allow the sequential solution technique of successive substitution that was used for CYCLE7 to also work well for CYCLEZ. One key to the success of the sequential solution as applied to the L-M cycle is that the degree of subcooling done in the high- and low-temperature intercoolers is specified. This approach allows the enthalpy of the refrigerant mixture entering the evaporator to be uniquely determined for each condenser saturation-temperature guess, which, to a great degree, uncouples the high- and low-side iteration loops. The energy balance requirements across the two intercoolers are accommodated on the low side by the flexibility of the split evaporators in taking up the remaining enthalpy change required to achieve saturated vapor at the exit of the high-temperature evaporator. This is possible because of the generalized specification of the load ratio between the two evaporators.

Solution Procedure. The solution technique used to determine the cycle operating pressures and temperatures is next described with reference to state points as shown in Fig. 2. The calculations begin with guesses for the condenser exit saturation temperature T(4) and evaporator inlet temperature T(7). With the guessed value for T(4) and specified values of intercooler subcooling, the enthalpy at the evaporator inlet h(7) is determined. This determines refrigerant state point 7 and, in turn, state 10 by applying the specified total evaporator pressure drop and the saturated vapor requirement.

From the known energy transfer across the low- and high-temperature intercoolers and the relative heat load ratio (enthalpy ratio) between the two evaporators, state points 8, 9, and 1 are next determined. At appropriate stages along this process, the low-side refrigerant temperatures are compared to the adjacent specified sink temperatures and current high-side refrigerant temperatures to ensure proper direction for heat flow. If inconsistent temperatures are found, T(7) is decremented and the low-side calculation sequence is repeated.

Once a plausible value for T(7) is found, values are computed for the individual low-side heat-exchanger LMTDs and for the *overall* evaporator LMTD. Relative areas required for each heat exchanger are calculated and are used to apportion the low-side pressure drop between the two evaporators and the low-temperature intercooler, for use in the next iteration on T(7). The calculated *overall* LMTD values are compared to the user-specified value, and the iteration is repeated until the two agree or until a minimum LMTD limit is reached. (Minimum possible *overall* evaporator LMTDs result from pinch points which occur when too much subcooling is specified across either the low- or high-temperature intercooler.) In either case, the low-side iteration is exited and the compressor work, shell heat loss, and exit state are determined before moving to the condenser iteration loop.

With the condenser inlet-state at point 2 known, the LMTDs for the superheated and two-phase refrigerant regions and for the overall condenser can be calculated. The calculated *overall* condenser LMTD from Eq. 3 is then combined with the $Q_{evap}/Q_{cond}$ ratio for the current estimates of T(4) and T(7) to obtain the *normalized overall* condenser LMTD from Eq. 6. The value of $LMTD_{cond,norm}$ is then tested for convergence with the user-specified value. A new guess for T(7) is made and the entire process is repeated until convergence in both condenser and evaporator loops is achieved.

Ambient Heat Exchange Option. For the option of ambient heat exchange in lieu of the high-temperature intercooler, the basic solution logic of the program remains unchanged. In this case, points 1 and 5 are set at the user-specified ambient temperature (which must be less than or equal to the condenser-inlet sink temperature), and the required enthalpy changes to the respective saturation points are computed. With this option, T(6) is fixed by the user-specified subcooling in the low-temperature intercooler and the evaporator inlet enthalpy is constant for all values of T(4). The evaporator pressure is thus totally independent of the condenser conditions. As such, the program runs more quickly than in the general case with two true intercoolers.

Program Output

Once the program has achieved convergence on *overall* evaporator and condenser LMTDs, a cycle summary is printed. The CYCLEZ output follows the basic format of the CYCLE7 program with a number of additions, and includes the following:

- individual and overall heat-exchanger LMTDs;
- individual and overall heat-exchanger sizings (both per unit heat-exchanger load and per unit total refrigeration load);
- relative heat-exchanger area and UA ratios;
- thermodynamic state conditions at the ten cycle points;
- refrigerant glides;
- estimated maximum plausible low and high-temperature subcooling;
- compressor pressure ratio, work, shell heat loss and exit superheat; and
- refrigeration COP and enthalpic and volumetric capacities.

Advantages of CYCLEZ Model

Temperature-Based. The CYCLEZ approach gives an entirely temperature-based cycle specification by using overall LMTDs and intercooler subcooling $\Delta$Ts. Specifying the cycle configuration in this manner is more intuitive than using UA- or effectiveness/NTU-based approaches in that it is easier to relate the basic task of refrigerant-glide matching to the application at hand. Also, with a temperature-based specification, the program input can be more straightforwardly set up to compare analytical and experimental results.

Constant Heat-Exchanger Loading. The overall heat-exchanger sizing of the major heat-transfer surfaces is specified in a general fashion on the basis of *per unit refrigeration capacity*. Such a specification is fluid and hardware independent. No UAs or external/internal fluid flow rates need to be defined (but can be calculated as output if desired). The heat exchangers are essentially designated on a performance basis (by specifying heat-exchanger loading) and in a manner which allows the total available heat-transfer area of the condenser and the two evaporators to be held constant for a required unit capacity.

Fast Program Execution. The temperature-based approach as applied to the L-M cycle intercoolers also results in relatively quick cycle convergence (about 10–15 sec on a 386-based IBM-compatible).

This is because the intercooler high-to-low-side heat transfers are uncoupled from the low-side iterations, again by use of performance rather than hardware specifications. In the optional case of ambient heat exchange in place of an active high-side intercooler, the solution speed is even faster.

Design Rather Than Simulation Model. CYCLEZ has advantages in evaluating optimal thermal cycle configurations for two reasons. First, the assumptions built into the solution logic maintain optimum cycle conditions at the condenser and fresh-food evaporator outlets (zero subcooling and superheat, respectively). It is more difficult to constrain simulation models with fixed hardware to follow a particular optimal thermodynamic design path. Secondly, use of thermal performance specifications as opposed to hardware specifications allows more of the general cycle parameters to be controlled directly (e.g., intercooler subcooling and total heat-exchanger loading), while the program automatically adjusts the hardware requirements internally as needed with different fluids.

Because of these characteristics, CYCLEZ is well-suited for comparative screening of refrigerant mixtures for application to the L-M cycle. The capability to maintain constant heat-exchanger sizing per unit refrigeration capacity and desired condenser and evaporator exit conditions simplifies the determination of optimum cycle requirements for each refrigerant mixture. These requirements include the distribution of heat-exchanger area, relative and total intercooler subcooling, and most advantageous mixture compositions. Once optimal configurations have been determined for all candidate pairs in a consistent manner, fair performance rankings can be made and optimum hardware requirements can be defined.

## INITIAL PARAMETRIC INVESTIGATIONS AND RESULTS

The implied reason for developing a computer model is to determine the effects of parameter variations analytically rather than experimentally. The L-M circuit is sufficiently different from the conventional, pure-refrigerant RF configuration so that many new system variables must be investigated.

### NARM Advantages

Before outlining results from the parametric investigation of the L-M cycle, it is important to specify how NARMs provide an advantage over pure refrigerants for the domestic RF application. With mixed refrigerants, some of the heat exchange irreversibility can be decreased by matching the evaporating and condensing temperatures of the refrigerant with the temperature-glide requirements of the secondary fluid (RF air temperatures). Also, liquid-line subcooling of a mixed refrigerant serves to produce a given evaporator temperature at a higher suction pressure, thereby decreasing the pressure ratio across the compressor [12]. Several chemical compounds with less severe ozone-depletion potentials (ODPs) and greenhouse-warming potentials (GWPs) than the currently-used CFC refrigerants have been identified as potential NARM components [10].

### Relative Heat-Exchanger Area Distribution

The relative distribution of heat-exchanger area between evaporating and condensing functions was varied by changing the *overall* evaporator and condenser LMTD values according to Eq. 8 while maintaining a constant $UA_{tot}/Q_{evap}$ value of 0.2 /°C. This value was chosen to be small enough so as not to unduly penalize NARMs with high-glides. In Fig. 3, the system COP is shown for NARM pairs with small, medium, and large temperature glides (relative to a total evaporator glide of 23 °C) and with nominal values of intercooler subcooling (20°C each). These results indicate that a $UA_{evap}/UA_{tot}$ ratio of approximately 0.5 gave optimal performance for the range of glides considered, so this relative area distribution was used for all subsequent calculations.

A fresh-food–to–freezer loading ratio of 1/1 is assumed for the parametric investigations based on recommendations in the ASHRAE equipment handbook [13]. Analytical runs with different relative fresh-food/freezer loadings showed that larger relative fresh-food loadings gave better system performance, primarily due to the smaller temperature lift required for these conditions.

CYCLEZ calculates the required distribution of evaporator area between the freezer and fresh-food compartments. This is a strong function of the fresh-food/freezer load distribution and a lesser function of the refrigerant constituents and composition of the NARM pair. NARMs with larger temperature glides (larger boiling-point differences) and those which benefit most from larger amounts of subcooling require larger fresh-food compartment evaporators.

Other conditions which were held constant for all of the CYCLEZ runs are summarized below.

- low-temp evaporator air-inlet and -outlet temperatures . -15 °C and -20 °C

- high-temp evaporator air-inlet and -outlet temperatures .... 3 °C and -2°C
- condenser air-inlet and -outlet temperatures ......... 32 °C and 40 °C

- overall evaporator LMTD .............................. 10 °C
- overall condenser LMTD (normalized from Eq. 6) ............ 10 °C

- evaporator and condenser pressure drop (kPa) ............. 0.0
- compressor isentropic efficiency ........................ 0.55
- fraction of input work lost through compressor can .......... 0.10

Where experimentally-measured interaction coefficients were not available for the CSD refrigerant-property routines, a conservative value of 0.01 was used. Variations of system COP, refrigerant volumetric capacities, evaporator and condenser glides, and compressor pressure ratios were determined for 19 refrigerant mixtures at concentrations ranging from 0 to 1.0 mass fraction of the lower-boiling-temperature component. Optimal results from these calculations are presented in Table 1. In this table, comparisons are normalized to the performance of R12 in a conventional, single-evaporator refrigerator/freezer.

<u>NARM Concentration Variation</u>
The concentration of components in a NARM controls the resulting refrigerant temperature "glide" as the NARM evaporates or condenses, and matching the specified total air-side glide with the refrigerant is an important aspect of NARM efficiency in this application. Hence, the composition of the refrigerant mixture has a significant effect on system performance. Figures 4(a) and 4(b) are typical results for the variation in predicted RF performance and refrigerant-side glides for a NARM with moderate glide. Also plotted on Fig. 4(a) are the volumetric capacity of the refrigerant mixture, and the "baseline" performance and volumetric capacity of R12 in a single-evaporator RF. A similar analysis for the other NARM combinations were used to select the optimal results listed in Table 1.

The corresponding Fig. 4(b) shows the evaporator and condenser refrigerant-side glides in relation to the air-side glides in these two heat exchangers as a function of concentration. The model is predicting maximum system performance at a NARM concentration where the best correspondence occurs between refrigerant and air-side glides, especially on the evaporator side with the larger air-side glide. Heat-exchanger pressure drops and other deviations that result from actual running conditions will modify these results to some degree.

<u>Subcooling/Superheating Variation</u>
Setting suction-line superheat and first stage of liquid subcooling to approach ambient temperature (a condition sought in most conventional single-evaporator designs) puts a rather severe constraint on the extent of high-temperature intercooling in this system. Model runs using this option did not produce system COPs as high as those in which both high- and low-temperature subcooling/superheating were individually specified and controlled. Clearly, the extent and division of the subcooling and superheating accomplished with the intercooler stages on the L-M design are important parameters for controlling system performance. An automated series of model runs for each of the NARMs in Table 1 provided information on how COP changes as a result of the subcooling and subcooling/superheating obtained in these intercoolers.

Figures 5(a), 5(b), and 5(c) are topographical plots of COP versus intercooler subcooling for high-, low-, and moderate-glide NARMs, respectively, in an L-M refrigerator-freezer. From the slopes of the contour lines for constant COP, it is apparent that high-temperature subcooling/superheating is more effective than low-temperature subcooling, and that subcooling accomplished in the lower-temperature intercooler is much more effective at improving system efficiency for NARM pairs with a higher-temperature glide. This is consistent with the previously stated advantages of NARMs over pure refrigerants. This low-temperature subcooling effectively decreases the compressor pressure ratio at the same evaporator temperature and causes the two-phase refrigerant temperature to better match the corresponding evaporator air temperatures. Figure 5(d) is included to show that the low-temperature intercooler does nothing to improve the efficiency of a pure refrigerant in this circuit [12].

Figure 6 further illustrates that subcooling with the low-temperature intercooler facilitated temperature-glide matching in the RF evaporators. The air temperature changes across the two evaporators remain constant, while increased subcooling by the low-temperature intercooler serves to decrease the LMTD of the high-temperature (fresh-food) evaporator at the expense of a slightly larger LMTD for the low-temperature (freezer) evaporator.

## SUMMARY AND CONCLUSIONS

CYCLEZ extends the advantages of the CYCLE7 model to the L-M refrigerator/freezer cycle. It has the added capability of maintaining a constant, total heat-exchanger area *per unit of refrigeration capacity* over the primary heat exchangers (the condenser and the two evaporators) for different refrigerants and refrigerant mixtures. As such, CYCLEZ is well suited for ranking the performance of different fluid combinations under optimum cycle conditions without the need for and possible limitations of specific hardware characteristics.

NARMs offer performance advantages over pure refrigerants in a household refrigerator application through better temperature glide matching and advantageous use of high-side subcooling. COP gains of 10 to 20% over R12 in a conventional RF are predicted for NARMs operating in a modified RF with separate fresh-food and freezer evaporators and two stages of refrigerant intercooling.

An optimum distribution of heat-exchanger area between the two evaporators and the condenser for the Lorenz-Meutzner RF cycle was predicted using the CYCLEZ model. This ratio worked well for NARM pairs with low, moderate, and large temperature glides.

With NARMs, obtaining additional subcooling with suction gas and two-phase refrigerant between the low- and high-temperature evaporators is a very effective method of improving refrigeration performance. The total amount of subcooling and the split between intercoolers for best performance depend on the NARM constituents and composition. Less than optimal performance was observed when the suction and liquid lines were allowed to come to ambient temperature in lieu of the first stage of subcooling/superheating. Such an assumption appears to limit the performance potential of the L-M cycle.

Determining the best NARM combinations, the optimum cycle conditions, and the necessary hardware configurations in the five-heat-exchanger L-M refrigeration cycle is a challenging task. The CYCLEZ model is capable of providing control of the basic design parameters with a simple yet generalized approach. The initial investigations conducted using CYCLEZ show progress toward determining the general heat-exchanger requirements of an optimal L-M system.

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13. *ASHRAE Handbook, 1989 Equipment*, Amer. Soc. Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1989.

Figure 1. Schematic diagram of the Lorenz-Meutzner refrigerator-freezer circuit

Figure 2. Temperature-enthalpy state point representation of Lorenz-Meutzner refrigerator-freezer cycle and adjacent air-side temperatures

Figure 3. Variations in Lorenz-Meutzner refrigerator-freezer COP as a result of heat-exchanger area distribution

(a) Lorenz-Meutzner refrigerator-freezer COP and refrigerant volumetric capacity as a function of NARM concentration;

(b) NARM temperature glides in the evaporator and condenser of a Lorenz-Meutzner refrigerator-freezer as a function of composition

Figure 4. NARM concentration effects

Figure 5. Topographical plots of system COP of a Lorenz-Meutzner refrigerator-freezer circuit as a function of subcooling obtained from the high temperature (H.T.) and low temperature (L.T.) intercoolers

Figure 6. Effects of subcooling in low-temperature intercooler (ICL) on the temperature glide matching in the evaporator of a Lorenz-Meutzner refrigerator-freezer

## TABLE 1

## COP AND CAPACITY DATA FROM CYCLEZ MODEL

### MODELING CALCULATIONS FOR THE A. LORENZ/K. MEUTZNER REFRIGERATOR/FREEZER CIRCUIT

| | Refrigerant | COP | COP[1] Increase Over R-12 (%) | Relative Volumetric Capacity (R-12 = 1.0) | Relative Suction Pressure (R-12 = 1.0) | Compressor Pressure Ratio |
|---|---|---|---|---|---|---|
| | R12 (Single Evaporator) | 1.353 | — | 1.0 | 1.0 | 9.2 |
| | R12 (Dual Evaporator) | 1.379 | 1.9 | 1.0 | 1.0 | 8.9 |
| LOW GLIDE | MIXTURE 1 | 1.473 | 8.9 | 1.1 | 0.9 | 10.2 |
| LOW GLIDE | MIXTURE 2 | 1.485 | 9.8 | 1.2 | 1.1 | 9.5 |
| LOW GLIDE | MIXTURE 3 | 1.481 | 9.5 | 1.2 | 1.1 | 9.6 |
| LOW GLIDE | MIXTURE 4 | 1.496 | 10.6 | 1.1 | 0.9 | 9.0 |
| LOW GLIDE | MIXTURE 5 | 1.525 | 12.7 | 1.6 | 1.4 | 8.8 |
| LOW GLIDE | MIXTURE 6 | 1.547 | 14.3 | 1.0 | 0.8 | 9.6 |
| LOW GLIDE | MIXTURE 7 | 1.468 | 8.5 | 1.1 | 1.0 | 8.6 |
| MODERATE GLIDE | MIXTURE 8 | 1.469 | 8.6 | 1.2 | 1.1 | 9.2 |
| MODERATE GLIDE | MIXTURE 9 | 1.546 | 14.3 | 1.1 | 1.0 | 8.8 |
| MODERATE GLIDE | MIXTURE 10 | 1.548 | 14.4 | 1.0 | 0.9 | 9.4 |
| MODERATE GLIDE | MIXTURE 11 | 1.575 | 16.4 | 1.7 | 1.6 | 8.0 |
| MODERATE GLIDE | MIXTURE 12 | 1.600 | 18.3 | 1.5 | 1.3 | 8.7 |
| MODERATE GLIDE | MIXTURE 13 | 1.712 | 26.5 | 0.7 | 0.5 | 10.4 |
| MODERATE GLIDE | MIXTURE 14 | 1.529 | 13.0 | 1.4 | 1.2 | 9.7 |
| MODERATE GLIDE | MIXTURE 15 | 1.671 | 23.5 | 0.7 | 0.5 | 11.1 |
| HIGH GLIDE | MIXTURE 16 | 1.430 | 5.7 | 1.0 | 0.9 | 11.5 |
| HIGH GLIDE | MIXTURE 17 | 1.651 | 22.0 | 1.3 | 1.1 | 8.5 |
| HIGH GLIDE | MIXTURE 18 | 1.604 | 18.6 | 1.6 | 1.6 | 7.4 |
| HIGH GLIDE | MIXTURE 19 | 1.516 | 12.0 | 2.5 | 2.3 | 7.5 |

[1] Improvement over R-12 in a single-evaporator configuration.

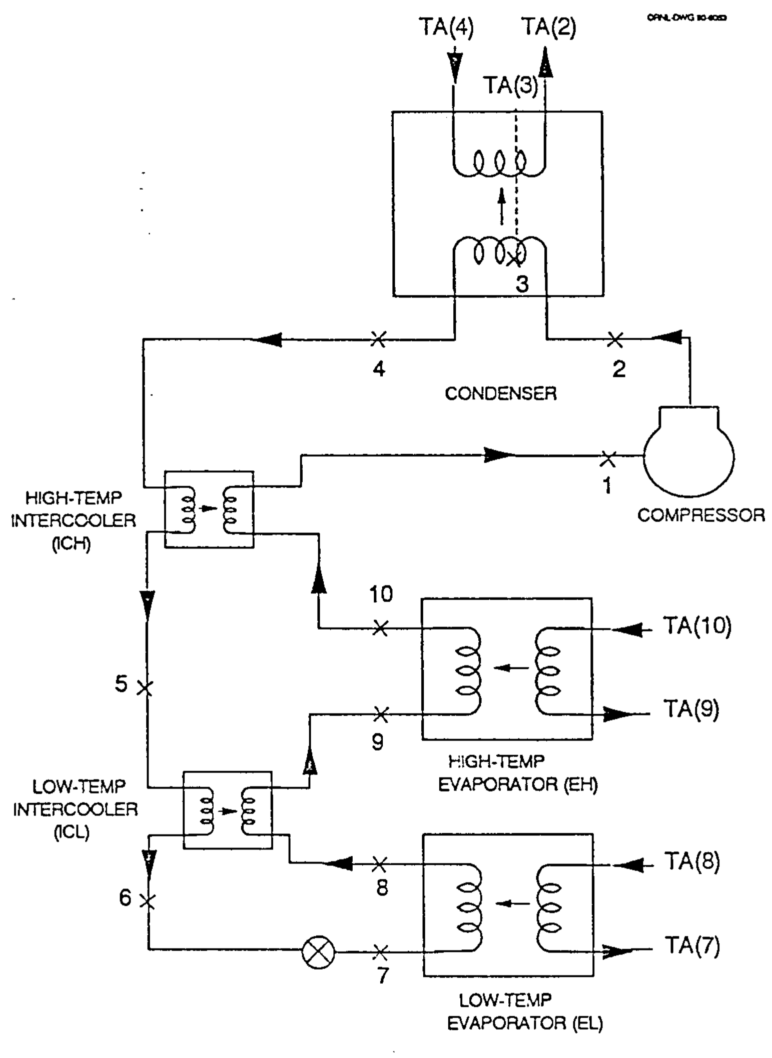

ORNL-DWG 90-6050
TA(4)
TA(2)
TA(3)
3
4
2
CONDENSER
1
COMPRESSOR
HIGH-TEMP
INTERCOOLER
(ICH)
10
TA(10)
5
TA(9)
9
HIGH-TEMP
EVAPORATOR (EH)
LOW-TEMP
INTERCOOLER
(ICL)
8
TA(8)
6
TA(7)
7
LOW-TEMP
EVAPORATOR (EL)

Figure 1

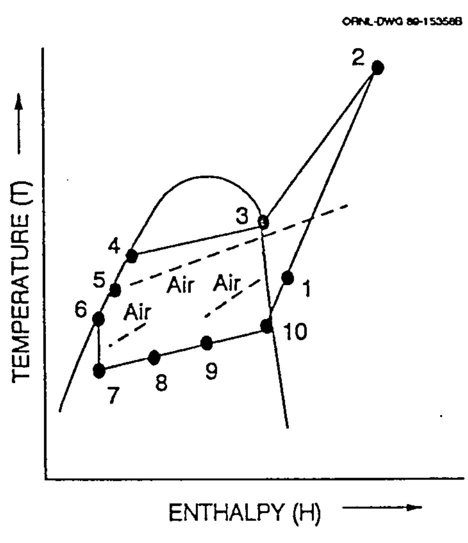
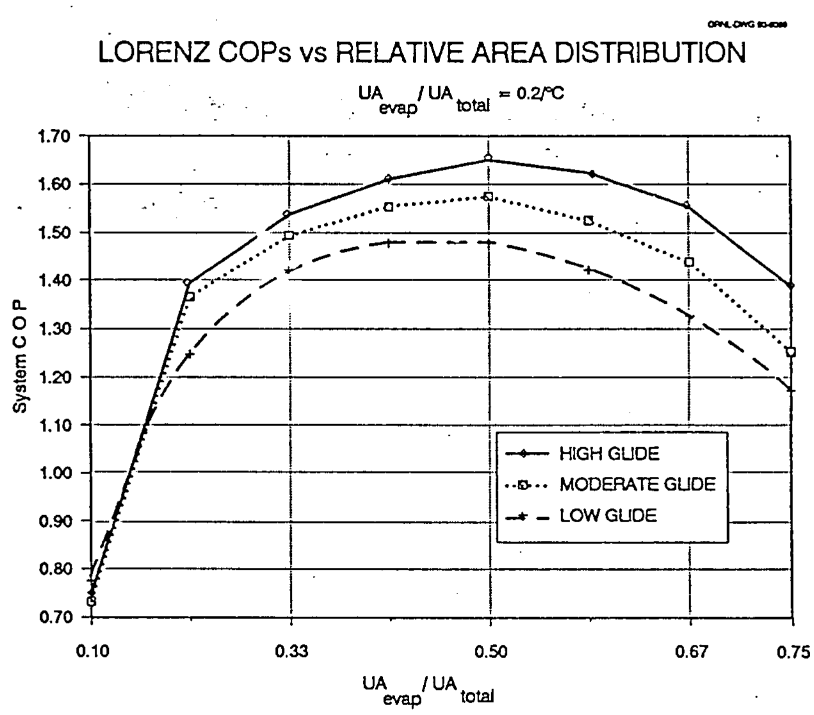

ORNL-DWG 89-15356B
2
3
4
5
6
Air
Air
Air
1
10
7
8
9
TEMPERATURE (T)
ENTHALPY (H)

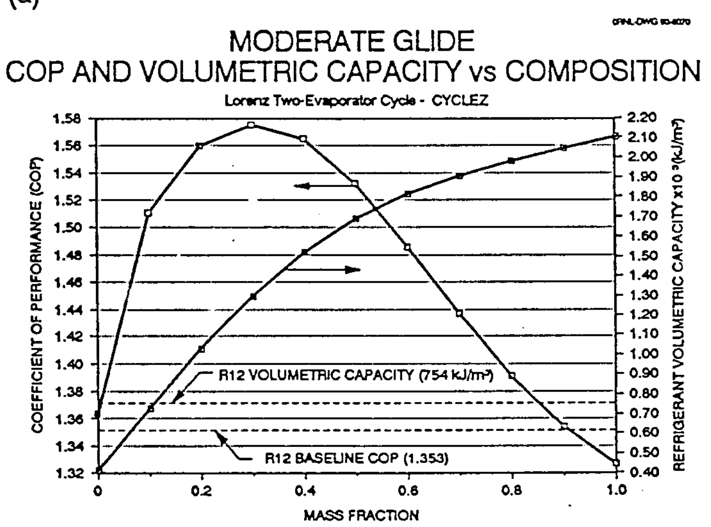
Figure 2



LORENZ COPs vs RELATIVE AREA DISTRIBUTION
UA evap / UA total = 0.2/°C
System C O P
1.70
1.60
1.50
1.40
1.30
1.20
1.10
1.00
0.90
0.80
0.70
0.10
0.33
0.50
0.67
0.75
UA evap / UA total
HIGH GLIDE
MODERATE GLIDE
LOW GLIDE


Figure 3

(a)


MODERATE GLIDE
COP AND VOLUMETRIC CAPACITY vs COMPOSITION
Lorenz Two-Evaporator Cycle - CYCLEZ
COEFFICIENT OF PERFORMANCE (COP)
REFRIGERANT VOLUMETRIC CAPACITY x10 ³(kJ/m³)
R12 VOLUMETRIC CAPACITY (754 kJ/m³)
R12 BASELINE COP (1.353)
MASS FRACTION
1.58
1.56
1.54
1.52
1.50
1.48
1.46
1.44
1.42
1.40
1.38
1.36
1.34
1.32
2.20
2.10
2.00
1.90
1.80
1.70
1.60
1.50
1.40
1.30
1.20
1.10
1.00
0.90
0.80
0.70
0.60
0.50
0.40
0
0.2
0.4
0.6
0.8
1.0

(b)

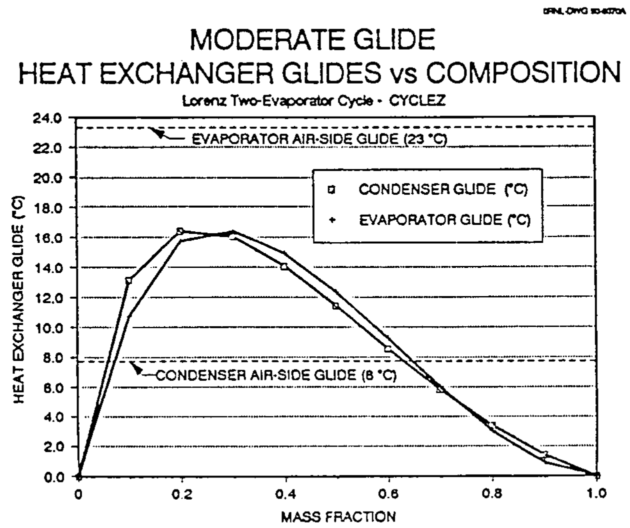

MODERATE GLIDE
HEAT EXCHANGER GLIDES vs COMPOSITION
Lorenz Two-Evaporator Cycle - CYCLEZ
EVAPORATOR AIR-SIDE GLIDE (23 °C)
CONDENSER GLIDE (°C)
EVAPORATOR GLIDE (°C)
CONDENSER AIR-SIDE GLIDE (8 °C)
HEAT EXCHANGER GLIDE (°C)
MASS FRACTION
24.0
22.0
20.0
18.0
16.0
14.0
12.0
10.0
8.0
6.0
4.0
2.0
0.0
0
0.2
0.4
0.6
0.8
1.0

Figure 4

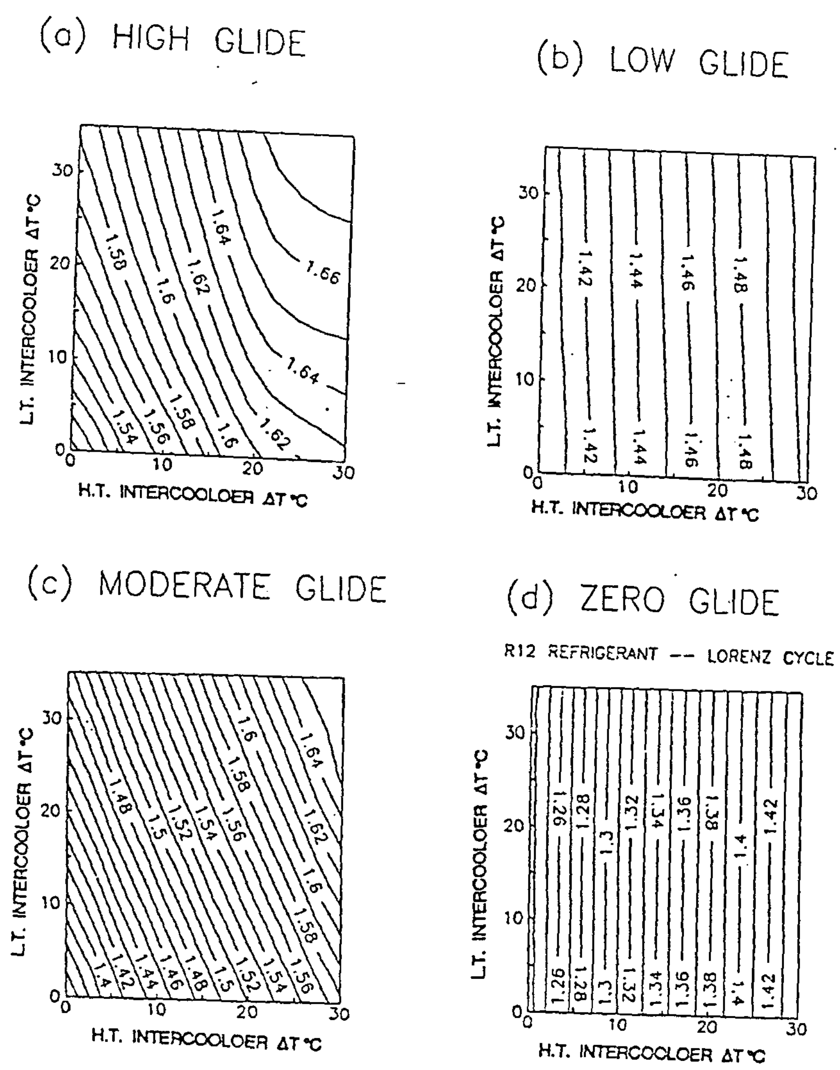
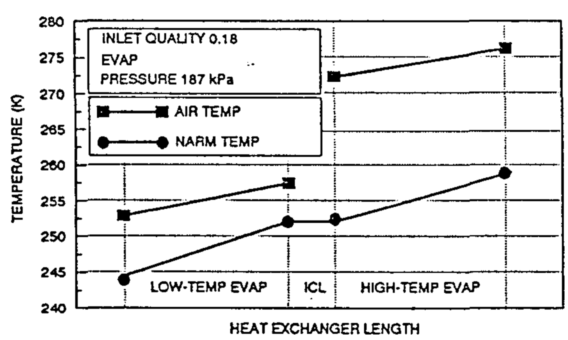

(a) HIGH GLIDE
L.T. INTERCOOLOER ΔT °C
H.T. INTERCOOLOER ΔT °C
1.50
1.52
1.54
1.56
1.58
1.6
1.62
1.64
1.66
(b) LOW GLIDE
L.T. INTERCOOLOER ΔT °C
H.T. INTERCOOLOER ΔT °C
1.42
1.44
1.46
1.48
(c) MODERATE GLIDE
L.T. INTERCOOLOER ΔT °C
H.T. INTERCOOLOER ΔT °C
1.4
1.42
1.44
1.46
1.48
1.5
1.52
1.54
1.56
1.58
1.6
1.62
1.64
(d) ZERO GLIDE
R12 REFRIGERANT -- LORENZ CYCLE
L.T. INTERCOOLOER ΔT °C
H.T. INTERCOOLOER ΔT °C
1.26
1.28
1.3
1.32
1.34
1.36
1.38
1.4
1.42

Figure 5

## 0°C SUBCOOLING LOW-TEMP INTERCOOLER (ICL)

MODERATE GLIDE NARM




## 30°C SUBCOOLING IN LOW-TEMP INTERCOOLER (ICL)

MODERATE GLIDE NARM

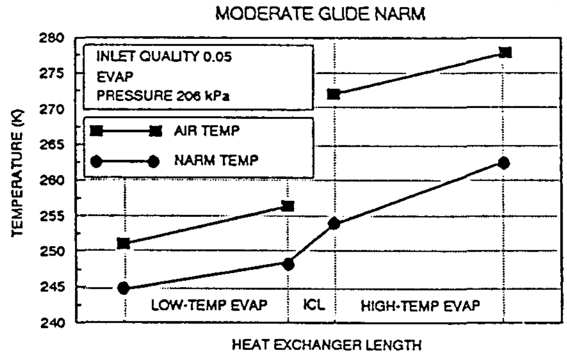



Figure 6