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LABORATORY EVALUATION OF AN OZONE-SAFE NONAZEOTROPIC REFRIGERANT MIXTURE IN A LORENZ-MEUTZNER REFRIGERATOR FREEZER DESIGN

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ABSTRACT

The Lorenz-Meutzner refrigerator freezer (RF) circuit has been proposed as a design which would operate with nonazeotropic refrigerant mixtures (NARMs) and significantly increase the thermodynamic efficiency of household refrigerators. Several ozone-safe and more environmentally acceptable refrigerants are known which could be blended into a NARM to replace R-12 for this domestic refrigeration application.

Laboratory tests were performed on a Lorenz-Meutzner (L-M) RF using an R-32/R-124 NARM. Comparisons are made between the baseline performance of the refrigerator with R-12 before it was modified to the L-M design and that of the L-M circuit operating with R-12 and the NARM. Circuiting and component changes resulting from initial testing of this unit are described. Computer modeling and compressor calorimeter results for R-12 and the NARM used in the test unit are also presented.

Small performance gains (≈3%) are seen for the NARM over R-12 in the same refrigerator freezer circuit. Modeling results and steady-state data suggest larger improvements (≈15%) are possible. It is felt that the larger improvements predicted from modeling and compressor calorimetry data are not being realized due to poor heat transfer and refrigerant circuiting arrangements.

INTRODUCTION

Global environmental concerns have served to place restrictions on the production and sale of chlorine-containing, fully halogenated compounds (CFCs) (UNEP 1987; NASA 1988). Commercial production of Refrigerant 12 (R-12), which is extensively used in the refrigerating circuit of household refrigerator-freezers (RFs), and Refrigerant 11 (R-11), used as a blowing agent for the insulating foam, will be phased out before 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. It is quite apparent that there are a limited number of pure fluids that can function as substitutes for these restricted compounds (McLinden and Didion 1987). Nonazeotropic refrigerant mixtures (NARMs) offer an approach to developing environmentally acceptable alternatives for pure-compound refrigerants. Using NARMs as refrigeration fluids can also improve the efficiency of vapor-compression refrigeration equipment at the expense of circuit and hardware redesign (Mulroy et al. 1988).

The application of NARMs in domestic refrigerator-freezers has been suggested and experimentally investigated with varied results (Stoecker 1978; ORNL 1984). A RF circuit which incorporates two evaporators and two stages of liquid-line subcooling, Figure 1, was described and tested initially by A. Lorenz and K. Meutzner (Lorenz and Meutzner 1975) in 1975 and by H. Kruse (Kruse et al. 1989) in 1989. This circuit has partially verified the improved efficiency and CFC alternative potential promised by NARMs. Results and design modifications arising from extensive testing of a L-M RF design with pure and NARM refrigerants are presented here.

BACKGROUND

Conventional, single-evaporator RFs maintain internal air temperatures of approximately 5° F in the freezer and 38° F in the fresh food section by cooling all of the air to freezer conditions and allowing some controlled mixing between compartments. This design is simple and reliable, but it has some thermodynamic and operational drawbacks. From 40 to 60% of the total cooling load of a domestic RF is due to the fresh food compartment, and this cooling is affected, in the

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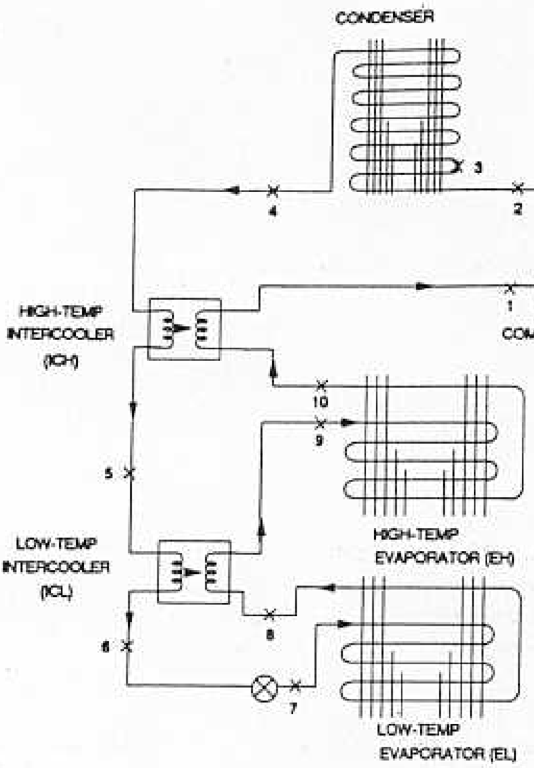


Figure 1 *Schematic diagram of Lorenz-Meutzner refrigerator-freezer circuit.*

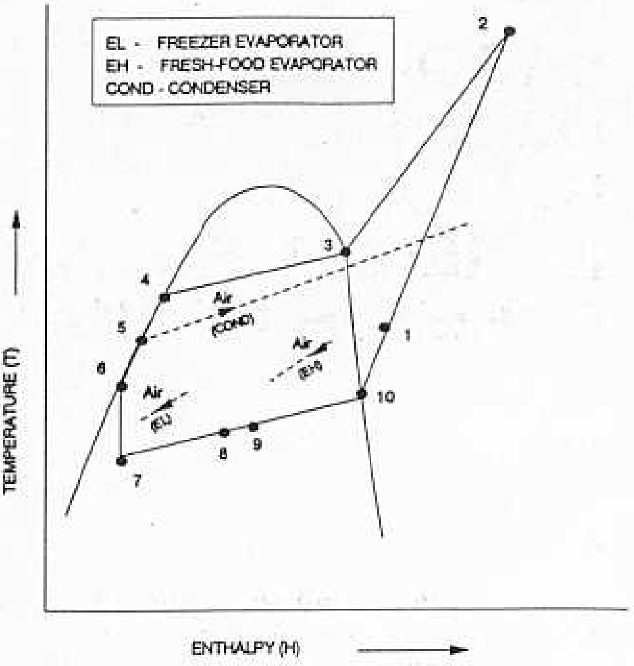


Figure 2 *Refrigerant circuit state-points for the Lorenz-Meutzner, refrigerator-freezer cycle corresponding to the CYCLE-Z model. Superimposed condenser, freezer, and fresh-food compartmentair temperatures.*

conventional design, by rejecting heat over the larger 5° F to ambient temperature lift of the freezer rather than the 38° F to ambient temperature lift of the fresh food compartment.

Another consequence of allowing freezer and fresh food compartment air to mix in a conventional RF is that the relatively humid air from the fresh food section comes into contact with the cold (-10° to -15° F) freezer evaporator. Water vapor condenses out and freezes on this evaporator in the form of frost. Frost build-up interferes with efficient heat transfer and must be removed by periodic defrost cycles which contribute both as an additional parasitic electrical load and as a heat removal load for the RF.

Using NARM refrigerants in the RF circuit as proposed by Lorenz and Meutzner (1975) can improve the cycle efficiency of RFs. The main refrigerant circuit modifications for a L-M RF are a separate, high temperature evaporator (EH) for the fresh food compartment which follows in series after the freezer evaporator (EL) and an additional stage of liquid refrigerant subcooling (ICL) which cools the high pressure liquid refrigerant before the freezer evaporator with low pressure, two phase refrigerant flowing from the freezer to the fresh food evaporator. Since NARMs evaporate over a temperature range due to changing liquid and vapor phase concentrations, the fundamental idea in applying them to a RF is to match the refrigerant evaporation temperature to the required air temperatures in both compartments.

This concept can be further illustrated by referring to the temperature-enthalpy diagram shown in Figure 2. The state points in Figure 2 correspond with those in Figure 1. The temperature increase in the NARM refrigerant from the point where it enters the freezer evaporator (7) to where it leaves the fresh food evaporator (10) can be matched to the air temperatures in the fresh food and freezer compartments. In the L-M RF design, the temperature change of the refrigerant through both evaporators (refrigerant glide) can be adapted to the air temperature requirements of both compartments through selection of the component refrigerants in the NARM and their concentrations (Rice and Sand 1990). Small (1 to 3 psi) refrigerant-side pressure drops in the evaporator will decrease this temperature glide slightly.

Lorenz and Meutzner used an R-22/R-11 NARM in an experimentally modified RF to obtain a 20% improvement in steady-state refrigeration performance over R-12 for their original work in 1975. Dr. Kruse (1989) worked with an R-22/R-142b NARM in a two-evaporator, L-M breadboard system to obtain COP improvements of 10% and a 1.5% improvement in "pull down" efficiency. In later work with an R-22/R-123 mixture, researchers at the University of Hannover were unable to obtain any energy use reduction under the AHAM/DOE 90°F closed-door test conditions over R-12 in a RF designed to United States standards (Tiedman et al. 1991). Cycling losses and different methods

Table 1. Compressor calorimeter test data* for 670 Btu/h compressor
R-12

	Evaporator Temperature		0°F	Evaporator Temperature		-10°F	Evaporator Temperature		-20°F
Condenser temperature (°F)	130	120	110	130	120	110	130	120	110
Capacity (Btu/h)	960.0	1014.6	1017.4	718.3	764.7	805.7	494.4	555.8	591.3
Watts (W)	199.6	198.8	195.2	166.4	167.2	168.4	134.0	139.2	141.6
EER (Btu/Wh)	4.81	5.10	5.37	4.32	4.57	4.78	3.69	3.99	4.18
Speed (rpm)	—	—	—	—	—	—	—	—	—
Refrigerator flow rate/(lb/h)	15.44	16.28	17.52	11.28	12.00	12.76	7.60	8.48	9.24
Oil flow rate (cc/h)	—	—	—	—	—	—	—	—	—
Low-side pressure (psig)	9.2	9.2	9.2	4.5	4.5	4.5	0.6	0.6	0.6
High-side pressure (psig)	181.0	157.7	136.5	181.0	157.7	136.5	181.0	157.7	136.5
Suction tube temperature (4 in. from shell)	89.0	88.6	89.0	88.4	88.5	88.8	88.4	88.2	88.3
Suction gas temperature (intake muff) (°F)	—	—	—	—	—	—	—	—	—
Discharge gas temperature (cylinder heat) (°F)	—	—	—	—	—	—	—	—	—
Discharge tube temperature (4 in. from shell) (°F)	196.0	193.3	190.0	178.8	178.4	179.8	138.4	155.8	164.5
Run winding Temperature (°F)	—	—	—	—	—	—	—	—	—
Shell top temperature (°F)	136.8	136.0	134.2	136.0	134.8	134.3	132.4	132.8	132.6
Shell mid temperature (°F)	148.0	147.7	146.4	144.2	143.7	143.6	138.9	139.6	140.2
Shell bottom temperature (°F)	139.9	138.9	137.8	137.4	135.7	135.4	133.9	132.7	133.2

*Test Conditions:

Air flow rate (ft/s):	3.5
Ambient temperature (°F)	90.0
Volts (V):	115.0
Outlet temperature (subcooled) (°F):	90.0

of computing COPs were cited as reasons for differences from previous results.

A steady-state computer model simulating a two-evaporator, two intercooler RF operating with NARMs was developed to analytically assess the effects of system design and operating parameters on the cycle performance and efficiency of a RF designed around the L-M circuit. This computer model was used to rank the cycle performance of ozone-safe binary refrigerant combinations which could be considered for use in a L-M application (Rice and Sand 1990). Effects of the distribution of heat exchanger area, extent and distribution of intercooler subcooling/superheat, refrigerant mixture composition, and relative refrigerator/freezer loading were also investigated with the model.

A 15 mass% R-32/85 mass% R-124 refrigerant mixture was chosen as the NARM to be used for the experimental L-M RF work because the model predicted a 13-16% improvement in COP for the NARM over R-12 with approximately the same volumetric capacity and operating pressure ratio. Other important considerations were that both refrigerants were available in our laboratory, and using higher concentrations of R-124 simplified compressor lubricant selection. Modeling results with the R-32/R-124 NARM used in the laboratory prototype L-M RF were used to size the fresh food evaporator and check against experimentally measured temperatures and pressures obtained from the test unit.

COMPRESSOR TESTING

Prior to system testing, the performance of RF reciprocating compressors was evaluated with R-12 and the R-32/R-124 NARM on a compressor calorimeter. McLinden and Radermacher have shown that comparisons between NARM and pure refrigerant performance are totally dependent on the temperatures (which correspond directly to pressures set on a compressor calorimeter) chosen to equate NARM and pure refrigerant saturated evaporator and condenser operating temperatures (McLinden and Radermacher 1987). Using an average or midpoint of the NARM glide through the respective heat exchanger appears to be the fairest and most widely accepted way of making this comparison (Boot 1990). Depending on the temperature glide of the NARM pair and the temperature change of secondary fluid through the heat exchanger, this method of simulating equivalent saturated heat exchanger temperatures could create a pinch point in the actual refrigeration application (Smith, et al. 1990).

Equilibrium vapor pressures corresponding to the mean two-phase temperature of the NARM as it evaporates or condenses were calculated to establish the suction and discharge pressure set points for the calorimeter tests. This mean temperature is an average based on ten equal increments of enthalpy change over the two-phase region. Using the mean two-phase temperature rather than the midpoint

Table 2. Calorimeter test data[a] for 670 Btu/h compressor
15% by mass R-32/85 mass% R-124

	Evaporator Temperature 0°F			Evaporator Temperature −10°F			Evaporator Temperature −20°F		
Condenser temperature (°F)	130	120	110[b]	130	120	110[b]	130	120[b]	110[b]
Capacity (Btu/h)	965.2	1026.6	1070.3	679.8	752.2	804.1	465.5	506.5	570.6
Watts (W)	188.0	188.0	186.4	150.4	156.4	157.6	118.0	124.8	130.0
EER (Btu/Wh)	5.13	5.46	5.74	4.52	4.81	5.10	3.95	4.06	4.39
Speed (rpm)	—	—	—	—	—	—	—	—	—
Refrigerant flow rate/(lb/h)	13.04	13.88	14.56	9.28	10.04	10.80	5.68	6.64	7.48
Oil flow rate (cc/h)	—	—	—	—	—	—	—	—	—
Low-side pressure (psig)	8.0	8.0	8.0	3.5	3.5	3.5	-0.3	-0.3	-0.3
High-side pressure (psig)	187.0	161.4	139.0	186.4	161.4	139.0	186.2	161.4	139.0
Suction tube temperature (4 in. from shell)	88.8	88.9	88.6	88.8	88.5	88.2	88.1	88.4	88.6
Suction gas temperature (intake muff) (°F)	—	—	—	—	—	—	—	—	—
Discharge gas temperature (cylinder heat) (°F)	—	—	—	—	—	—	—	—	—
Discharge tube temperature (4 in. from shell) (°F)	190.7	190.1	186.9	172.7	174.0	175.5	146.5	151.6	155.8
Run winding Temperature (°F)	—	—	—	—	—	—	—	—	—
Shell top temperature (°F)	127.5	127.6	125.7	123.9	123.1	122.4	118.6	120.4	120.4
Shell mid temperature (°F)	142.4	141.7	139.0	137.8	137.2	136.6	131.7	134.0	133.9
Shell bottom temperature (°F)	133.2	132.4	129.5	129.8	128.4	127.9	125.9	127.2	126.8

[a]Test Conditions:

Air flow rate (ft/s):	3.5
Ambient temperature (°F)	90.0
Volts (V):	115.0
Outlet temperature (subcooled) (°F):	90.0

[b]Results in this column were obtained with liquid subcooled below 90°F at the calorimeter to prevent two-phase flow at this condenser pressure.

temperature should compensate for any non-linearity in the NARM temperature/enthalpy profile (glide).

Compressor calorimeter results for R-12 refrigerant and the 15 mass% R-32/R-124 NARM refrigerant in the 670 Btu/h compressor used by the baseline RF are summarized in Tables 1 and 2. The composition of the NARM was chosen to approximately match the volumetric capacity (Btu/ft^3, [kJ/m^3]) of R-12. Interestingly, the compressor operates more efficiently with the NARM than it does with the R-12 at every test condition. Figure 3 is a plot of the capacity and EER data from these tables at the 120°F condensing temperature which graphically illustrates this observation.

These data indicate that the R-32/R-124 NARM in this compressor should be able to cool as large a RF load as R-12. Additionally, the higher compressor EER values suggest that it should handle this load more efficiently with the NARM than it did with R-12.

Similar calorimetry results for R-12 and the R-32/R-124 NARM in a larger, 1060 Btu/h, compressor are presented in Tables 3 and 4. The same trends seen in previous data from the smaller compressor are seen in these results. Similar refrigeration capacity is obtained for R-12 and the NARM at each rating condition, but lower compressor wattages are needed with the NARM resulting in higher EERs, Figure 4. As before, the indication is that comparable refrigeration loads would be more efficiently handled with the NARM refrigerant in this compressor than with R-12 refrigerant.

Later refinements to the CSD equation of state coefficients for R-32 and R-124 and better estimates for the binary interaction coefficients necessitated correction of the EER and capacity results presented in Table 2 and Figure 3. These corrected results and a table of condensing and evaporating temperatures for the R-32/R-124 NARM which correspond better with saturated R-12 heat exchanger conditions are given in a paper by Rice and Sand (1993).

REFRIGERATOR-FREEZER TESTING

Power consumption, compressor run time, and compartment operating temperature data from the 90°F, closed door testing performed on the original unmodified RFs, the initial design of the L-M RF, and the final version of the L-M RF are presented in Tables 5-8 and 10-12. The "mid/mid" and "warm/warm" control settings listed in these tables refer to the freezer thermostat and fresh food controller indicators which were adjusted to provide runs which bracket 5°F in the freezer and 38°F in the fresh food compartments. "On" and "off" settings indicate whether the RF mullion heaters were on or off during the power consumption test.

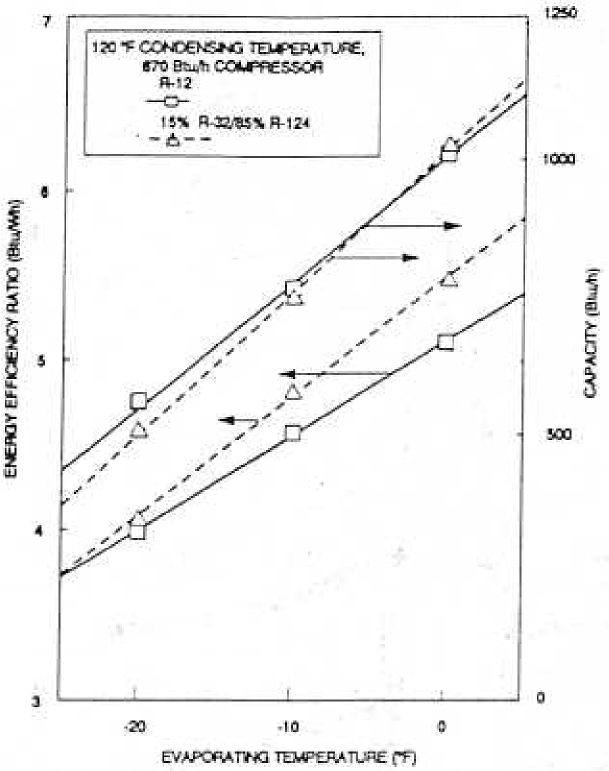


Figure 3 *Compressor capacities and energy efficiency ratios for R-12 and a 15 mass %, R-32/85 mass % R-124 NARM in a 670 Btu/h compressor at 120°F condensing condition.*

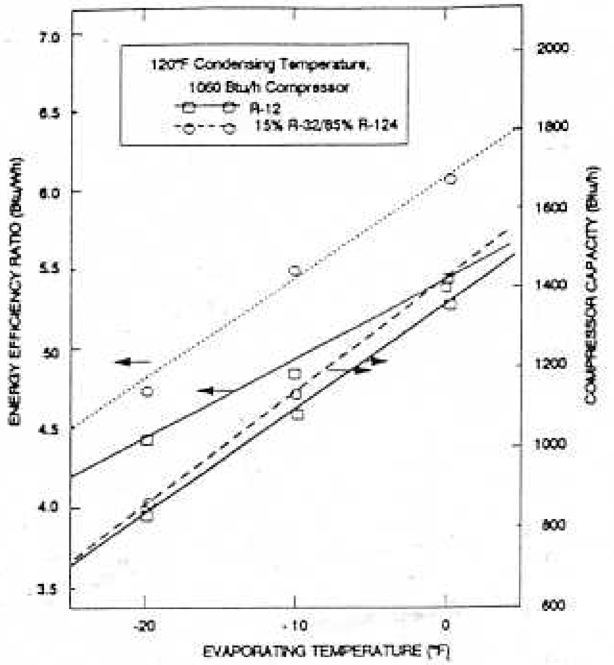


Figure 4 *Compressor capacities and energy efficiency ratios for R-12 and a 15 mass % R-32/R-124 NARM in a 1060 Btu/h compressor at the 120°F condensing condition.*

Baseline Testing

Commercially produced 18 ft^3, top-mount RFs that incorporated separate freezer and fresh food compartments were chosen as the baseline units for this study. This RF was the outcome of a development program conducted in the late 70s-early 80s (Topping 1982). Compared to typical RFs of that time, this product consumed about 40% less energy.

Three unmodified units were baseline tested with R-12 under the AHAM/DOE 90°F closed-door test conditions, Tables 5-7. Run times and compartment temperatures are rather typical of most RFs. The long freezer defrost cycle times reflect one of the energy saving features built into these two-evaporator units. The average of the overall power consumptions measured for all three RFs was 3.00 kilowatt hours per day (kWh/d). The RF which gave the 2.83 kWh/d performance rating was selected for modification to the L-M design.

The first design used for construction of a L-M RF was patterned after a prototype unit tested with an R-22/R-142b NARM at the University of Hannover (Kruse et al. 1989). The more pertinent features of this design were:

- It was built around the production units previously described.
- A large, static evaporator sized on the basis of modeling calculations was used for the fresh food compartment.
- Small, commercial, 200 - 1200 Btu/h, suction-to-liquid-line heat exchangers were used for the liquid line subcoolers (ICH and ICL in Figure 1) on the L-M circuit.
- A fine metering valve was used as a throttling device after the last liquid-line subcooler instead of a capillary tube.
- Compartment temperature control was obtained by using a thermostat to start and stop the compressor based on freezer temperatures and adjusting the refrigerant charge size to obtain typical fresh food temperatures.

Problems with Initial L-M Design

Shakedown tests with R-12 in this initial L-M design revealed several operational problems. Refrigerant pressure drops through the fresh food evaporator at operating conditions were measured at 7 to 9 psi rather than 1 to 3 psi. This excessive pressure drop penalizes the cycle by forcing larger pressure differences across the compressor for any desired evaporator inlet pressure. The static evaporator was replaced with a smaller, fan-forced evaporator to minimize the pressure drop while maintaining compartment cooling capacity with minimized refrigerant charge.

The tests with R-12 also showed that the fine metering valve used for a throttling device in this initial circuit did not work reliably. The micrometer valve could be adjusted to give the desired suction pressure while the compressor was

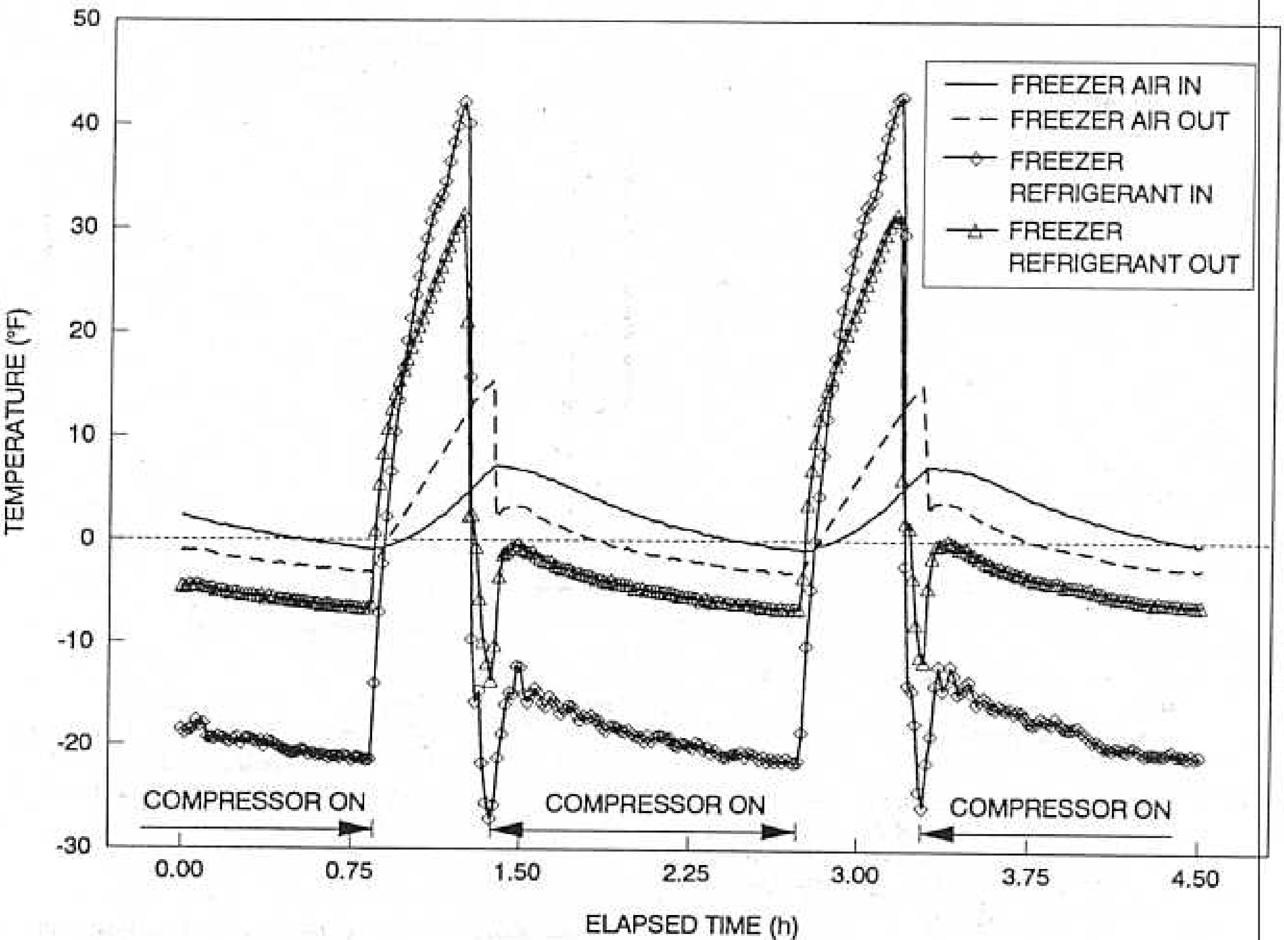


Figure 5 *Lorenz-Meutzner freezer evaporator temperatures. Refrigerant and air temperatures through the freezer (cross-flow evaporator) 15% R-32/R-124 NARM.*

running. However, when the compressor cycled on and off in response to the thermostat, a larger pressure drop was eventually established over the valve at this same setting, thus restricting refrigerant flow to the point where there was insufficient cooling capacity to satisfy the load (Sand et al. 1991).

A manifold that permitted selection of discrete lengths of capillary tube was substituted for the fine metering valves. It was then possible to perform a 90°F closed door power consumption test with the R-32/R-124 NARM in the initial L-M design. The results from that test are briefly summarized in Table 8. The overall power consumption rating of 3.41 kWh/d obtained in this test was much poorer than that of the baseline units. Unfortunately, a full four point closed door test was not performed with R-12 in this initial L-M design.

Steady state efficiencies were contrasted by looking at measured wattages for RF operation at similar compartment air temperatures and refrigerant entering and leaving temperatures near the end of a compressor on cycle with R-12 in the unmodified RF and the R-32/R-124 NARM in the L-M RF. This comparison is predicated on the assumption that R-12 and the 15 mass% R-32/85 mass% R-124 NARM have very similar refrigeration capacities at identical RF rating conditions as indicated by compressor test results described earlier. Table 9 summarizes this comparison.

These steady-state operating data support the compressor calorimeter EER data described earlier and modeled L-M results for this NARM (Rice and Sand 1990). The poor performance of the initial L-M system design with the mixed refrigerant, Table 8, is felt to be attributable to cycling and heat exchanger inefficiencies.

When the compressor shut off in the initial L-M design, relatively warm refrigerant on the high pressure side of the low-temperature intercooler heated up the gaseous refrigerant on the low pressure side of the heat exchanger. This problem with the L-M design was observed by others (Tiedman et al. 1991). The +30° to +40°F refrigerant produced by this process was pushed into the freezer and the fresh food evaporators causing the compartment to warm up and the thermostat to call for cooling. Short cycling of the compressor and increased cabinet heat loads were the net result. Figure 5, plotting freezer air and refrigerant tempera-

Table 3. Calorimeter test data* for 1060 Btu/h compressor

R-12

	Evaporator Temperature 0°F			Evaporator Temperature -10°F			Evaporator Temperature -20°F		
Condenser temperature (°F)	130	120	110	130	120	110	130	120	110
Capacity (Btu/h)	1362.4	1385.6	1430.7	1064.8	1093.5	1129.0	828.6	846.4	883.3
Watts (W)	260.8	254.4	236.0	233.2	224.8	208.4	181.6	189.2	193.2
EER (Btu/Wh)	5.22	5.45	6.06	4.57	4.86	5.42	4.32	4.47	4.57
Speed (rpm)	—	—	—	—	—	—	—	—	—
Refrigerator flow rate/(lb/h)	22.08	22.48	23.20	16.80	17.64	18.16	12.88	13.44	14.08
Oil flow rate (cc/h)	—	—	—	—	—	—	—	—	—
Low-side pressure (psig)	9.2	9.2	9.2	4.5	4.5	4.5	0.6	0.6	0.6
High-side pressure (psig)	181.0	157.5	136.5	181.0	157.7	136.5	181.2	157.8	136.5
Suction tube temperature (4 in. from shell)	88.5	88.7	88.5	88.2	89.2	88.8	88.6	88.8	88.2
Suction gas temperature (intake muff) (°F)	—	—	—	—	—	—	—	—	—
Discharge gas temperature (cylinder heat) (°F)	—	—	—	—	—	—	—	—	—
Discharge tube temperature (4 in. from shell) (°F)	202.9	198.9	190.6	199.7	194.4	186.4	184.9	182.9	177.8
Run winding Temperature (°F)	—	—	—	—	—	—	—	—	—
Shell top temperature (°F)	131.8	129.8	126.0	130.1	128.0	124.4	125.1	123.1	120.3
Shell mid temperature (°F)	125.4	112.7	111.6	113.0	113.4	111.2	120.6	113.7	111.7
Shell bottom temperature (°F)	157.2	152.8	146.7	154.5	151.6	145.7	150.8	147.0	143.3

*Test Conditions:

Air flow rate (ft/s):	3.5
Ambient temperature (°F)	90.0
Volts (V):	115.0
Outlet temperature (subcooled) (°F):	90.0

tures entering and leaving the evaporator, clearly illustrates this phenomena.

This problem was addressed by eliminating the commercial liquid-to-suction-line heat exchangers. Capillary tubes were soldered to the compressor suction line and to the freezer evaporator to fresh food evaporator cross-over line to accomplish the required intracycle heat exchange. Using this conventional approach to subcooling/superheating decreased the mass of hot refrigerant in contact with the low pressure side of the circuit and reduced refrigerant charge size from 20 oz. to approximately 12 oz. The longest capillary tube was equipped with threaded fittings to facilitate changing lengths without disrupting soldered connections.

Compressor short cycling was also addressed by replacing the RF thermostat with relay cards controlled by the data logger. For the freezer, a compressor-on limit of 5°F with a 6°F dead band was used for the mid/mid settings, and a 8°F limit with a 6°F dead band for the warm/warm RF settings.

Another design problem was that the original cross-flow freezer evaporator proved to be very ineffective at using low refrigerant temperatures produced with the NARM to pull down freezer air temperature. The cross-flow design, Figure 6a, is inefficient at transferring energy from an air stream with progressively decreasing temperatures to a refrigerant stream with progressively increasing temperatures.

Figure 5 illustrates the inefficiency associated with the cross-flow heat exchanger. Near the end of the compressor on cycle shown in this figure, refrigerant entered the evaporator at roughly -20°F and left at about -5°F. At the same time, the air entering and leaving this heat exchanger was only showing a 2 to 3 F° temperature change and about a 2 or 3 F° approach to the leaving refrigerant temperature.

Another indication of poor heat transfer shown by the data in Figure 5 is the compressor run time needed to satisfy the RF thermostat. Using this data, the complete compressor cycle took approximately 1.9 hours of which the compressor ran for about 1.45 hours for a 78% run time. The system was running at a mid/mid test condition. A comparison of freezer "pull down" and "warm up" data for this RF operating with R-12 in the unmodified configurations and R-32/R-124 in the L-M configuration, Figure 7, also illustrates the effects of poor heat exchanger performance. The low refrigerant temperatures produced with the NARM were very ineffective in pulling down freezer air temperatures.

To rectify this problem, a freezer heat exchanger with a longer, thinner profile which more closely approximated counter-flow design, Figure 6b, was substituted for the cross-flow evaporator used in the L-M unit.

Performance Testing of Improved L-M Design

After modification of both compartment heat exchangers, the liquid-line intercoolers, and the system thermostat as described previously, a closed door test was conducted with R-12. Results from this test are given in Table 10. The 3.25 kWh/d overall power consumption measured for this configuration with R-12 is about 15% poorer than that given in Table 7 for R-12 in the unmodified RF. It is obvious from this

Table 4. Calorimeter test data* for 1060 Btu/h compressor
15% by mass R-32/85% by mass R-124

	Evaporator Temperature 0°F			Evaporator Temperature −10°F			Evaporator Temperature −20°F		
Condenser temperature (°F)	130	120	110	130	120	110	130	120	110
Capacity (Btu/h)	1396.6	1430.7	1479.8	1103.0	1135.8	1167.2	825.9	857.3	884.6
Watts (W)	254.0	236.8	228.8	215.2	206.0	206.4	186.4	180.4	174.0
EER (Btu/Wh)	5.50	6.04	6.47	5.13	5.51	5.66	4.43	4.75	5.08
Speed (rpm)	—	—	—	—	—	—	—	—	—
Refrigerator flow rate/(lb/h)	1936	1996	20.64	15.00	15.80	16.20	11.32	11.76	12.12
Oil flow rate (cc/h)	—	—	—	—	—	—	—	—	—
Low-side pressure (psig)	8.0	8.0	8.0	3.5	3.5	3.5	-0.3	-0.3	-0.3
High-side pressure (psig)	186.5	161.5	138.8	186.2	161.2	139.0	186.5	161.5	139.0
Suction tube temperature (4 in. from shell)	89.0	88.8	88.6	88.6	88.5	89.2	88.8	88.5	88.5
Suction gas temperature (intake muff) (°F)	—	—	—	—	—	—	—	—	—
Discharge gas temperature (cylinder heat) (°F)	—	—	—	—	—	—	—	—	—
Discharge tube temperature (4 in. from shell) (°F)	198.8	190.9	185.5	191.4	186.0	183.5	186.8	183.3	178.1
Run winding temperature (°F)	—	—	—	—	—	—	—	—	—
Shell top temperature (°F)	132.2	128.5	126.2	128.6	126.2	125.0	125.8	124.2	122.1
Shell mid temperature (°F)	110.1	108.0	107.7	109.0	108.0	107.0	107.8	107.0	106.0
Shell bottom temperature (°F)	151.1	145.7	142.3	147.9	144.2	142.2	146.9	144.4	140.9

*Test Conditions:

Air flow rate (ft/s):	3.5
Ambient temperature (°F)	90.0
Volts (V):	115.0
Outlet temperature (subcooled) (°F):	90.0

Table 5. Unit 1—baseline R-12 testing
AHAM four point, 90°F, closed door test procedure

Temperature control setting	Power consumption (kWh/d)	Compressor run time (%)	Average freezer temperature (°F)	Relative humidity (%)	Average fresh food temperature (°F)	Freezer defrost cycle time (h)
"Mid/Mid-Off"	2.857	52.3	2.77	—	37.44	88.975
"Mid/Mid-On"	3.606	60.2	2.12	—	38.11	78.683
"Warm/Warm-Off"	2.433	43.4	12.74	20.6	43.43	109.163
"Warm/Warm-On"	2.998	47.7	12.78	18.0	44.09	99.558

Overall power consumption rating: 3.10 kWh/d.

data that the compressor run times were about 50% longer than was expected based on earlier test data from this cabinet.

The 670 Btu/h compressor experienced difficulty in achieving suction pressures required for proper chamber temperatures while maintaining adequate refrigerant circulation rates needed to pull down compartment temperatures with reasonable run times. This indicated that its capacity was too small for the L-M configuration. Table 11 shows the results when the R-12 closed door test was repeated after the original compressor was replaced with a 1060 Btu/h compressor. Averaged run times from these data are very comparable to those tabulated in Table 7. The 3.03 kWh/d power consumption rating obtained in Table 11 is essentially equivalent to the averaged power consumption obtained from R-12 baseline testing of the three RFs at the start of this program (Tables 5-7).

The 15 mass% R-32/R-124 NARM showed a 6-7% reduction in energy use over R-12 in the L-M configuration (2.83 kWh/d for the NARM vs. 3.03 kWh/d for R-12). Table

Table 6. Unit 2*—baseline R-12 testing
AHAM four point, 90°F, closed door test procedure

Temperature control setting	Power consumption (kWh/d)	Compressor run time (%)	Average freezer temperature (°F)	Relative humidity (%)	Average fresh food temperature (°F)	Freezer defrost cycle time (h)
"Mid/Mid-Off"	2.895	55.2	1.87	23.0	37.89	85.671
"Mid/Mid-On"	4.018	70.2	−1.90	—	36.79	67.409
"Warm/Warm-Off"	2.340	41.2	10.93	—	47.74	110.430
"Warm/Warm-On"	2.954	48.7	10.55	24.0	44.28	96.870

Overall power consumption rating: 3.07 kWh/d.

* This R/F had a new food defrost kit installed before tests were initiated, and it drew 30–31 watts of power during fresh food defrosts. Units 1 and 3 did not draw any additional power during fresh food defrost cycles.

Table 7. Unit 3—baseline R-12 testing
AHAM four point, 90°F, closed door test procedure

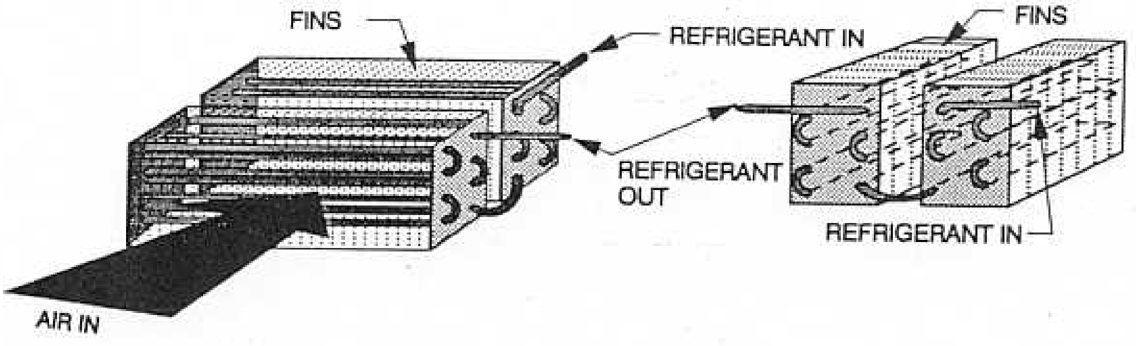
Temperature control setting	Power consumption (kWh/d)	Compressor run time (%)	Average freezer temperature (°F)	Relative humidity (%)	Average fresh food temperature (°F)	Freezer defrost cycle time (h)
"Mid/Mid-Off"	2.882	55.6	−0.81	—	35.80	85.130
"Mid/Mid-On"	3.715	63.9	−2.06	—	36.09	73.837
"Warm/Warm-Off"	2.413	43.7	7.28	31.8	41.45	108.008
"Warm/Warm-On"	2.992	48.5	6.48	28.9	42.09	97.454

Overall power consumption rating: 2.83 kWh/d.

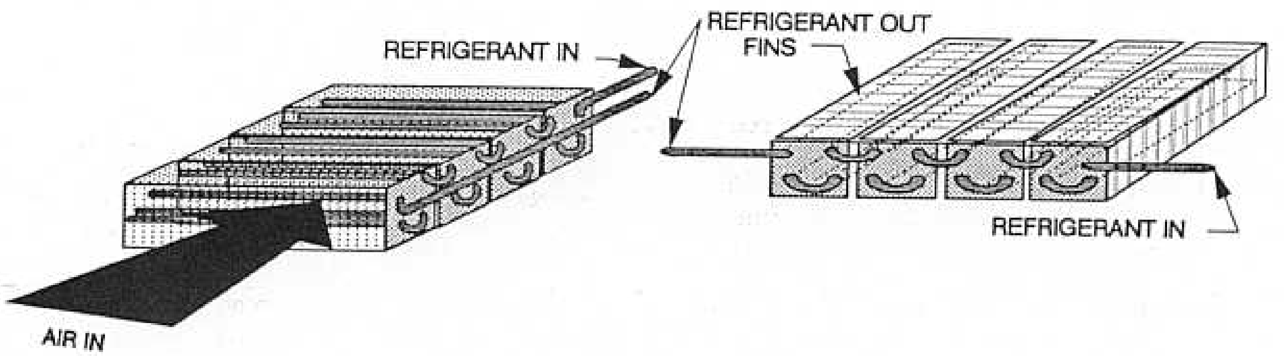
Table 8. Lorenz-Meutzner refrigerator-freezer testing with 15% R-32/R-124
AHAM four-point, 90°F, closed-door test procedure
(original Lorenz-Meutzner design)

Temperature control setting	Power consumption (kWh/d)	Compressor run time (%)	Average freezer temperature (°F)	Average fresh food temperature (°F)	Freezer defrost cycle time (h)
"Mid/Mid-Off"	3.038	73.86	4.89	37.87	62.57
"Mid/Mid-On"	3.797	83.07	4.94	38.20	56.91
"Warm/Warm-Off"	2.618	57.14	10.95	42.19	82.30
"Warm/Warm-On"	3.107	71.29	8.93	42.84	67.54

Overall power consumption rating: 3.41 kWh/d.



(a.) Original cross-flow evaporator and refrigerant circuiting used in initial Lorenz-Meutzner design.



(b.) Longer, thinner counter-flow freezer evaporator and refrigerant circuiting used in final Lorenz-Meutzner design.

Figure 6 *Freezer evaporator heat exchanger used for the Lorenz-Meutzner refrigerator-freezer showing refrigerant circuiting and air flow.*

12. Virtually all of this improvement is due to better mid/mid performance with the NARM, which is surprising in light of the compressor data presented earlier which favored the NARM more at higher evaporator temperatures. Performance with the NARM in the final L-M configuration was about 6% better than the average of the three baseline units and equal to the best one (Tables 5-7).

Very little change in power consumption is seen in going from the mid/mid to the warm/warm conditions with the NARM. In contrast, with the R-12 substantial changes in power consumption and compressor run times are evident between runs at the mid/mid and warm/warm conditions.

Figure 8 shows heat exchanger data for the Lorenz-Meutzner RF operating with the counter-flow freezer evaporator configuration shown in Figure 6b. These data show about the same refrigerant entering and leaving temperatures as Figure 5, but now the temperature change of the air stream is about 5 to 6 F° and the air stream achieves a 1 F° approach to the leaving refrigerant temperature. As a consequence, the complete compressor cycle shown by these data was about 0.84 hours with a compressor on time of 0.45 hours or a 54% compressor run time. Again, these data were taken from a mid/mid RF condition.

Several other important changes to the RF circuit must be taken into consideration when comparing Figures 5 and 8. Most notably, the larger displacement compressor was being used at the time the data for Figure 8 were taken. Using the compressor calorimeter data and the operation pressures of the L-M RF at the time data for Figures 5 and 8 were being recorded, estimated refrigerant flow rates were 5.7 lb/h for Figure 5 and 11.1 lb/h in Figure 8. The increased flow rate resulting from the larger compressor would also significantly improve the refrigerant-side heat transfer performance of the heat exchanger.

Figure 8 also illustrates how redesigning the liquid line intercoolers and decreasing the quantity of warm refrigerant on the high pressure side of these heat exchangers limited the warm refrigerant back-up problems discussed earlier. With

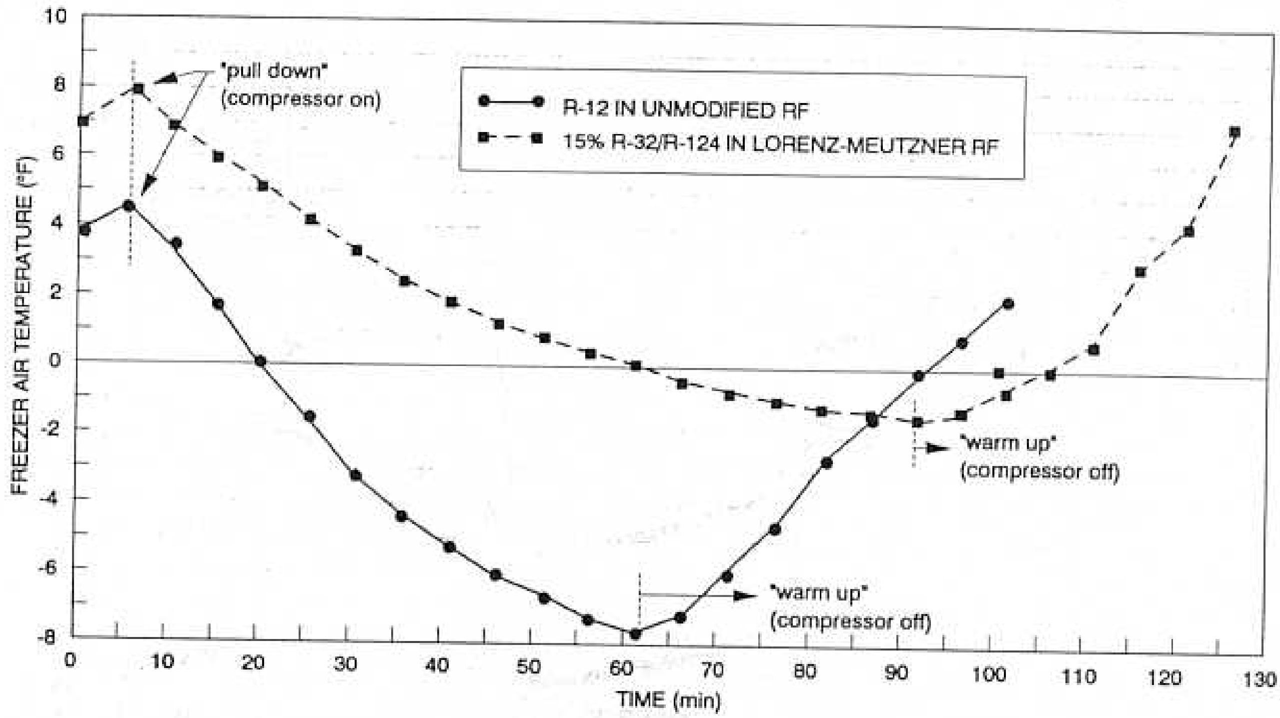


Figure 7 *Comparison of freezer "pull down" and "warm up" of test RF at 90°F. Before and after modification to the Lorenz-Meutzner design.*

Table 9. Refrigerator-freezer steady-state energy use (670 Btu/h compressor)

Temperature Control Setting	R-12 (watts)	R-32 NARM (watts)
"Mid/Mid/On"	235	187
"Mid/Mid/Off"	207	163

this new arrangement, refrigerant temperatures in the +15° to +30°F range are seen during the compressor off cycle. As a result, much less compartment warming occurs when the compressor and evaporator fan come on again.

While improvements over baseline R-12 performance were modest, it should be noted that a fan was added in the fresh food compartment that added to the parasitic power consumption and compartment heat loads, and that more thermal mass in the form of additional heat exchangers and interconnecting tubing were part of the final L-M design. It is felt that significant improvements can still be made in the design and function of the counter-flow heat exchangers and liquid-line subcoolers for this L-M concept.

Rating RF performance based on a single test point can be misleading. For instance, consider the mid/mid/on data from Tables 5, 11, and 12. While the freezer temperatures in Table 5 are slightly lower than those in Tables 11 and 12 which would account for additional energy use, the mid/mid/on power consumption of the L-M RF with the R-32/R-124 NARM in Table 12 (2.956 kWh/d) is 16.6% less than the 3.545 kWh/d value for R-12 in the L-M design given in Table 11 and 18.0% less the 3.606 kWh/d power consumption for R-12 in the unmodified RF, Table 5. Other observations in the laboratory testing of the L-M design indicated that it was quite sensitive to changing operation conditions. The L-M RF tested in our laboratory had a fixed charge, a static throttling device, and two evaporators connected in a

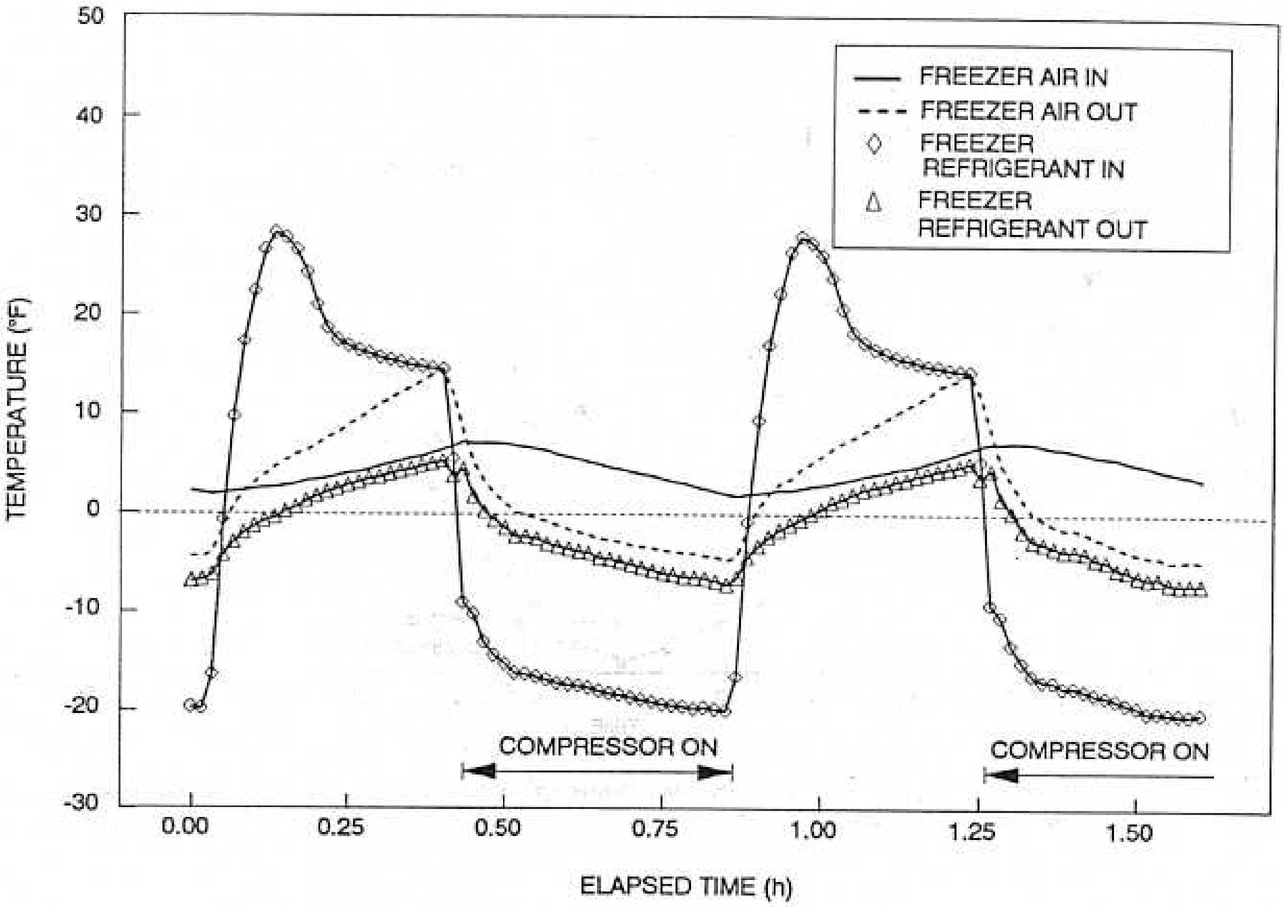


Figure 8 *Lorenz-Meutzner freezer evaporator temperatures. Refrigerant and air temperatures through the freezer (counter-flow evaporator) 15% R-32/R-124 NARM.*

series. Its single-point performance was compromised to obtain balanced temperatures in the fresh food and freezer compartments for the 90°F closed-door test conditions. To be seriously considered as an alternative for the conventional, single-evaporator design, the L-M RF has to show acceptable performance over a wide range of ambient and internal operating conditions. Incorporation of a modulating expansion device and thermostatically controlled evaporator fans into the L-M circuit would help this design meet changes in the fresh food to freezer loadings resulting from changing ambient temperature and off-design operating conditions.

CONCLUSIONS

The energy saving potential of NARM refrigerants in an L-M cycle is supported by steady-state operation results and compressor calorimeter data. Steady-state energy use based on compressor power consumption during on cycles was about 20% less than similar data for the same compressor operating with R-12 at similar compartment and refrigerant temperatures.

A poor correlation was seen between NARM compressor calorimeter, steady state, and modeled performance results and integrated L-M RF power consumption tests. System performance with the NARM is dependent on the efficiencies and design of other system components. Specifically, effective counterflow heat exchangers, an understanding of unique cycling losses, and an awareness of the effects of changing ambient and operating temperatures are needed to realize the efficiency potential of NARM refrigerants in the L-M system.

Counter-flow, refrigerant-to-air heat exchangers are important for the Lorenz-Meutzner design because they are the only way to achieve coil leaving-air temperatures intermediate between the entering and leaving refrigerant temperatures. The absence of an effective counterflow freezer evaporator showed up in our test results as an inability to rapidly "pull down" compartment temperatures despite favorable compressor and steady-state test results.

To function with efficiency comparable to that of the baseline RF, the L-M RF circuit needed a compressor which had about 50% more capacity than the baseline unit's compressor.

Table 10
Lorenz-Meutzner Refrigerator-Freezer testing with R-12
AHAM four point, 90°F, closed door test procedure
(revised L-M design - 670 Btu/h compressor)

Temperature control setting	Power consumption (kWh/d)	Compressor run time (%)	Average freezer temperature (°F)	Average fresh food temperature (°F)	Relative humidity (%)	Freezer defrost cycle time (h)
"Mid-Mid-Off"	3.299	72.40	2.42	33.99	30.70	63.54
"Mid-Mid-On"	4.067	84.53	1.86	32.74	30.08	56.01
"Warm-Warm-Off"	2.895	65.15	5.62	41.08	19.09	71.83
"Warm-Warm-On"	3.436	71.02	5.51	44.30	18.85	66.48

Overall Power Consumption Rating: 3.25 kWh/d

Table 11
Lorenz-Meutzner refrigerator-freezer testing with R-12
AHAM four point, 90°F, closed door test procedure
(revised L-M design - 1060 Btu/h compressor)

Temperature control setting	Power consumption (kWh/d)	Compressor run time (%)	Average freezer temperature (°F)	Average fresh food temperature (°F)	Relative humidity (%)	Freezer defrost cycle time (h)
"Mid-mid-off"	3.148	59.03	4.06	35.32	35.94	80.09
"Mid-mid-on"	3.545	59.98	4.06	36.70	33.08	78.82
"Warm-warm-off"	2.650	46.68	5.81	35.58	25.42	101.27
"Warm-warm-on"	2.947	47.66	5.60	41.35	24.22	99.19

Overall power consumption rating: 3.03 kWh/d

Table 12
Lorenz-Meutzner refrigerator-freezer testing with 15 mass% R-32/R-124 NARM
AHAM four point, 90°F, closed door test procedure
(revised L-M design - 1060 Btu/h compressor)

Temperature control setting	Power consumption (kWh/d)	Compressor run time (%)	Average freezer temperature (°F)	Average fresh food temperature (°F)	Relative humidity (%)	Freezer defrost cycle time (h)
"Mid-mid-off"	2.793	52.29	3.85	36.48	32.03	90.41
"Mid-mid-on"	2.956	56.64	3.94	38.36	33.08	82.61
"Warm-warm-off"	2.470	45.10	9.22	40.68	35.01	103.83
"Warm-warm-on"	2.888	48.42	9.27	42.90	31.94	96.83

Overall power consumption rating: 2.83

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As laboratory technicians for the project, Charles Harden and Randy Linkous set up and performed most of the laboratory tests and helped reduce and organize raw research data.

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DISCUSSION

Chuan Weng, Project Engineer, Revco Scientific, Asheville, NC: What is the comparison between head pressure and discharge temperature vs. a conventional oil system?

James R. Sand: A comparison between the head pressures and discharge temperatures for the conventional and test systems can be obtained by comparing the compressor calorimeter data at similar operating temperatures. Generally, the nonazeotropic refrigerant mixture operated at higher head pressures but lower discharge temperatures.

Mike Kempiak, Senior Project Engineer, Admiral Co., Galesburg, IL: What charge sizes did you use for the three refrigerator-freezer tests?

Sand: The unmodified 18 ft^3 refrigerator-freezers chosen as starting points for this work used about 10 ounces of R-12 for a normal charge size. The initial Lorenz-Meutzner (L-M) refrigerator design with an oversized, static, fresh food evaporator and large liquid-line subcoolers operated with about 20 ounces of refrigerant charge. The final, improved L-M design used approximately 12 ounces of refrigerant for an optimal charge.