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Nicholas Hugh
Marquette University

Margaret Mathison
Marquette University, margaret.mathison@marquette.edu

Anthony Bowman
Marquette University, anthony.bowman@marquette.edu

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Nicholas Hugh
Student Member ASHRAE

Margaret Mathison, Ph.D.
Associate Member ASHRAE

Anthony Bowman, Ph.D.
Associate Member ASHRAE

ABSTRACT

Low temperature refrigeration storage equipment in the biotechnology industry typically uses cascade refrigeration to achieve evaporating temperatures of -80 °C (-112 °F) or below. Current systems utilize multiple compressors leading to high energy consumption. Equipment operating costs contribute significantly to the total operating costs of biotechnology companies and therefore motivate the development of more efficient alternatives for low temperature refrigeration. This paper describes a single compressor R-23/R-134a mixed refrigerant cycle that has been designed to extract a load of 0.256 kW (873.5 Btu/hr) from a conditioned space at -80 °C (-112 °F). The designed system compresses a mixture of the gaseous refrigerants to a high pressure and then condenses the R-134a in a water-cooled separator while the R-23 remains in vapor phase. The stream of liquid R-134a is expanded to the suction pressure and is used to condense the R-23 that remains in vapor phase, operating much like an inter-stage heat exchanger in a cascade cycle. The condensed stream of R-23 then expands to the suction pressure and enters a low-temperature evaporator, where it absorbs energy from the load.

A model of the cycle is developed based upon first and second law principles of thermodynamics and used to refine the design of a mixed refrigerant test apparatus. Theoretical analysis of the prototype system predicts that it will reach an evaporating temperature of -78.6 °C (-109.5 °F) when it operates with a mixture of 33.4% R-23 and 66.6% R-134a by mass. In experiments conducted using the same condensing temperature and mixture composition the mixed refrigerant apparatus reached an evaporating temperature of -75.0 °C (-103 °F), corresponding closely to the predicted temperature of -78.6 °C (-109.5 °F). To reach the desired evaporating temperature of -80 °C (-112 °F) the refrigerant mixture must be altered to increase the amount of R-23.

BACKGROUND

Ultra low temperature refrigeration (ULT) is defined by the American Society of Heating, Refrigerating and Air Conditioning Engineers as those applications requiring refrigeration temperatures in the range from -50°C (-58 °F) to -100°C (-148 °F) (ASHRAE, 2010). Biotechnology applications often use ULT refrigeration for long term sample storage, and the continuous operation of this equipment at such low temperatures consumes significant amounts of energy. The current analysis focuses on applications with a refrigerated space temperature of -80°C (-112 °F), which is a common set point for laboratory refrigeration systems used for long term sample storage (Panasonic 2012; Thermo Fisher Scientific

Nicholas Hugh is a graduate student, Margaret Mathison is an assistant professor, and Anthony Bowman is an assistant research professor in the Department of Mechanical Engineering, Marquette University, Milwaukee, WI.

2007). To maintain a sample temperature of -80°C (-112 °F), typical systems supply refrigerant to the low stage evaporator at a maximum temperature of -85°C (-121 °F). This ensures that the refrigerant absorbs sufficient heat to keep a nearly constant temperature of -80°C (-112 °F) inside the conditioned space.

The low temperatures required in these applications are typically reached through the use of cascade refrigeration cycles, as shown in Figure 1. These cycles consist of two or more independent refrigeration cycles connected through an inter-stage heat exchanger that acts as the evaporator for the high stage and the condenser for the low stage (Cengel & Boles 2010). The multiple stages enable the system to reach low evaporation temperatures with greater efficiency than single stage refrigeration cycles. Because the cycles in the cascade system are independent of one another, different refrigerants can be utilized in each cycle to optimize performance. Refrigerants 404A and 508B are commonly paired for cascade refrigeration equipment operating in the -80°C (-112 °F) temperature range. Zeotropic refrigerant blend 404A is used in the high stage cycle and azeotropic refrigerant blend 508B is used in the low stage for its low critical point (DuPont 2005; DuPont 2004). However, operating a cascade system with two compressors continuously consumes a large amount of electricity and contributes significantly to operating costs. While improving compressor efficiency is an area of ongoing research for refrigeration applications (Ribas, F., Deschamps, C., Fagotti, F., Morriesen, A., & Dutra 2008), recent research has also focused on developing novel refrigeration cycles that operate with a mixture of refrigerants to improve cycle efficiency (Gong et al. 2009).

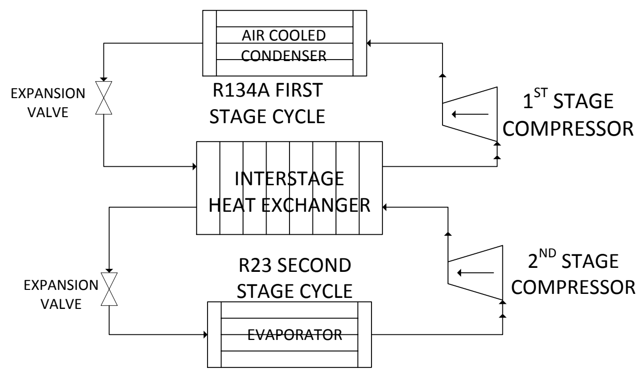


Figure 1: Two-stage cascade refrigeration system diagram.

The commercial refrigerants in use today have been developed to meet the provisions of the Montreal and Kyoto Protocols to reduce the ozone depleting potential and global warming potential of refrigerants (United Nations Environment Program 2000; United Nations 1998). Ongoing research is focused on developing blends of these refrigerants with advantageous thermodynamic properties, such as Dalkilic and Wongwises' 2010 study on the performance of refrigeration systems using alternative refrigerant blends. Kilicarslan and Hosoz (2010) and Missimer (1997) explored mixtures of refrigerants in different ratios to optimize their thermodynamic properties for efficient operation in cascade refrigeration systems. Before the current cascade refrigeration systems using R-404A and R-508B blends were developed to reach evaporator temperatures near -80°C (-112 °F), cascade systems used R-23 and R-134a for the low and high stages, respectively. Refrigerants 23 and 134a are fluorochemicals composed of single constituents trifluoromethane and 1,1,1,2-tetrafluoroethane, respectively. Figure 2 is the pressure versus enthalpy diagram for R-23, which illustrates the thermodynamic properties of the refrigerant at low temperatures and pressures.

Previous studies have also demonstrated the feasibility of using single-stage systems with binary refrigerant mixtures to achieve ULT refrigeration (Podtcherniaev, O., Boiarski, M., and Lunin, A. 2002). The current work extends these concepts to explore a low temperature refrigeration design that operates with a mixture of two simple refrigerants and a single compressor, thus reducing the electricity consumption of the system. It was shown by Chen, J., Hu, P., and Chen, Z. (2008) that the impact of chemical interactions in a binary mixture of two hydrofluorocarbon refrigerants is small compared to the impact of interactions in a binary refrigerant mixture consisting of one hydrofluorocarbon and one hydrofluorocarbon. Therefore, a binary mixture of R-23 and R-134a was chosen for the refrigeration cycle design because each refrigerant is composed of a single chemical and the refrigerants have been effective in previous ULT cascade refrigeration systems. With the use of REFPROP Version 8.0 (NIST 2007), a pressure versus enthalpy diagram has been generated for a binary mixture of 32.2% R-23 and 67.8% R-134a by mass, as shown in Figure 3.

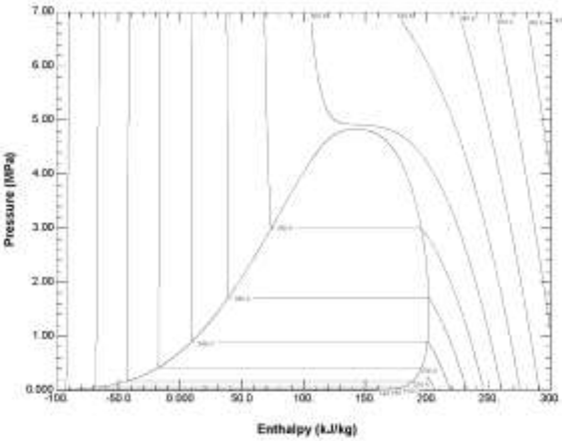


Figure 2: Pressure versus enthalpy diagram for R-23 (NIST 2007).

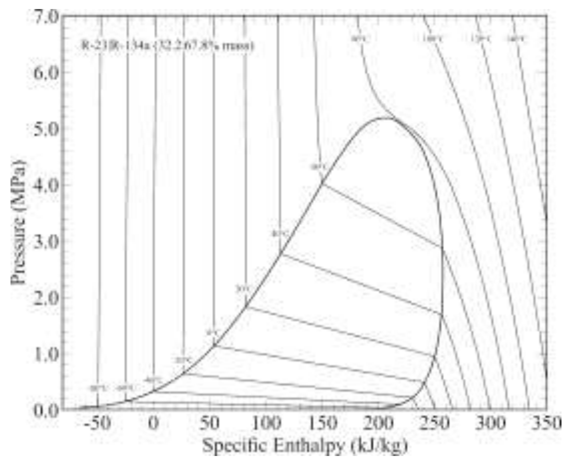


Figure 3: Pressure versus enthalpy diagram for R-23/R-134a refrigerant mixture (NIST 2007).

In this paper, a mixed refrigerant ULT freezer is designed to maintain the cabinet temperature at -80°C (-112 °F) by removing a constant heat load of 0.256 kW, simulating long term sample storage. Using REFPROP, R-23 and R-134a are analyzed at various state points to determine the optimum operating points to reach -80°C (-112 °F) for the ideal mixed refrigerant cycle design. Next, the theoretical refrigeration cycle state points are used to develop a test procedure to determine the evaporating temperature and coefficient of performance of the apparatus. Finally, the theoretical model developed for the mixed refrigeration cycle is compared to the experimental operation of the mixed refrigerant apparatus. The alternative cycle is developed with the goal of achieving a higher coefficient of performance (COP) than the cascade cycle, thus enhancing the future sustainability of ULT refrigeration products.

MIXED REFRIGERANT SYSTEM TEST APPARATUS

While the design of the mixed refrigerant system test apparatus is related to the two-stage cascade refrigeration system shown in Figure 1, it only uses one single-stage compressor. A Copeland CF12 reciprocating compressor was selected for the mixed gas refrigeration cycle and is modeled using empirical performance data at varying operating conditions provided by the manufacturer (Emerson 2010). The refrigerant mixture enters the suction side of the compressor as a superheated vapor, represented as state 1 in the refrigeration schematic in Figure 4.

Following compression, the vapor mixture of R-23 and R-134a enters a water cooled condenser (state 2) and is condensed at constant pressure. The separator is water cooled in order to maximize the heat removal rate per unit area and condense the refrigerant mixture efficiently (Mahindru & Mahendru 2011). It consists of an outer pipe that contains the refrigerant mixture and an inner pipe through which chilled water flows counter to the direction of refrigerant flow. The inlet for the refrigerant vapor mixture is located to leave room for collection of the liquid condensate at the bottom of the separator. Ideally, as the refrigerant mixture flows vertically upward toward the outlet of the condenser the mass fraction of R-23 in the vapor phase increases as the R-134a is condensed. For initial modeling purposes, it is assumed that pure R-134a exits the separator as saturated liquid at state 3, while pure R-23 exits the separator as superheated vapor at state 6.

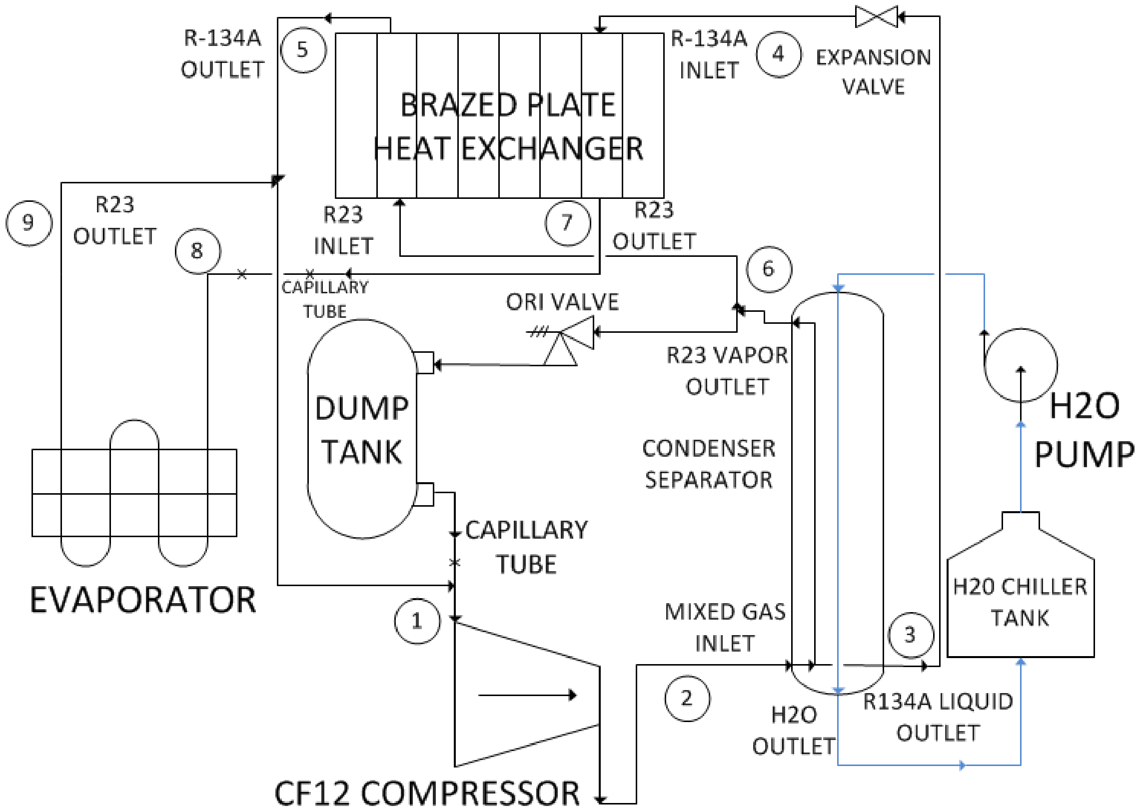


Figure 4: Mixed refrigerant test stand diagram.

The R-134a that leaves the separator as a saturated liquid flows through an automatic expansion valve, which decreases the pressure and temperature of the fluid. Assuming an isenthalpic process, the R-134a will exit the expansion valve as a two-phase liquid-vapor mixture at state 4 in Figure 4. Because the water cooled separator removes only enough energy to condense all of the R-134a to a saturated liquid; the R-23 exits the separator as superheated vapor. Therefore, the two-phase liquid-vapor mixture of R-134a at state 4 will be cool enough to condense the R-23 at state 6. The energy removed from the superheated R-23 as it condenses is absorbed by the R-134a in a brazed plate heat exchanger, causing it to evaporate. Leaving the brazed plate heat exchanger at state 5, the R-134a is assumed to have completely evaporated to a saturated vapor state.

The brazed plate heat exchanger is analogous to the inter-stage heat exchanger in a typical cascade refrigeration system because one fluid evaporates while the second fluid condenses. It is assumed that the R-23 exiting the heat exchanger at state 7 in Figure 4 is completely condensed to a saturated liquid state. The R-23 then flows through a capillary tube to drop the pressure; assuming an isenthalpic process, this also drops the fluid temperature. The resulting R-23 liquid-vapor mixture enters the evaporator (state 8) at a temperature below the controlled cabinet temperature of -80 °C (-112 °F). The evaporator consists of five rectangular plates with a total length of 80 feet of horizontal copper tubing. Silicone heating sheets in between each evaporator plate supply a constant 0.256 kW (873.5 Btu/hr) load representing the energy that must be removed from the storage cabinet. The R-23 evaporates as it flows through the copper tubing of the evaporator and exits the evaporator at state 9, where it is assumed to be a saturated vapor. The R-23 then mixes with the R-134a vapor returning to the suction side of the compressor. The temperature of the R-23 and R-134a vapor mixture will increase as it returns to the compressor due to the heat transfer from the surrounding air, thus ensuring that the vapor is superheated and there is no liquid entering the suction side of the compressor.

MIXED REFRIGERANT SYSTEM MODEL

Achieving complete separation of the mixture of R-134a and R-23 in the separator column is considered ideal because this would ensure that pure fluids enter the inter-stage heat exchanger, just as in a cascade refrigeration cycle. With an ideal separation process and making the assumptions mentioned previously, a first law analysis predicts that a mass flow rate of 5.66 kg/hr (12.5 lbm/hr) of R-23 is required to remove 0.256 kW (873.5 Btu/hr) from the cabinet and maintain evaporating and cabinet temperatures of -87 °C (-124.6 °F) and -80 °C (-112 °F), respectively. The calculated mass flow rate of R-23 can be used to determine the heat load that must be absorbed by the R-134a stream in the brazed plate heat exchanger, which is then used to determine the required R-134a mass flow rate of 11.25 kg/hr (24.8 lbm/hr). Since both refrigerant mass flow rates will flow through the same compressor, the total mass flow rate through the system will be 16.9 kg/hr (37.3 lbm/hr). Therefore, the mass flow rate through the compressor will be a mixture of 33.4 % R-23 and 66.6% R-134a mass.

For the initial modeling of the 33.4% R-23 and 66.6% R-134a refrigerant mixture by mass, it is assumed that the mixture condenses at 21.1 °C (70 °F) in the separator. A binary phase diagram of temperature versus mass fraction of R-134a is generated using REFPROP Version 8.0 to determine how the mixture behaves at constant pressure in the separator (NIST 2007), as shown in Figure 5.

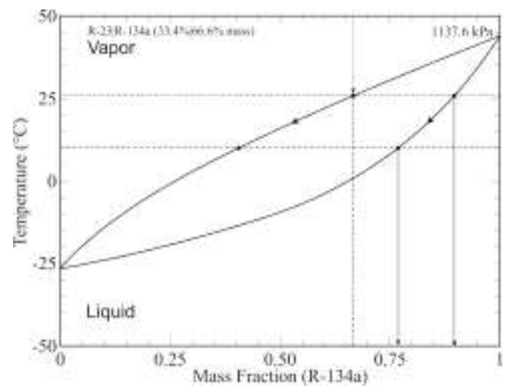


Figure 5: Binary phase diagram for 33.4% R-23 and 66.6% R-134a by mass at 1137.6 kPa (165 psia) (NIST 2007).

The elongated teardrop shown in the binary phase diagram in Figure 5 represents the liquid-vapor region of the mixture at a pressure of 1137.6 kPa (165 psia). The vapor region is situated above the saturated vapor line and the liquid region is below the saturated liquid line. When the mixture with an overall mass fraction of 66.6% R-134a exits the condenser separator at a temperature of 10 °C (50 °F), the phase diagram shows that the condensed liquid will have a mass fraction of 78% R-134a and 22% R-23. This does not match the idealized cycle model in which it was assumed that the condensate would be pure R-134a. Therefore, the phase diagram confirms that assuming the R-23 and R-134a are perfectly separated will only provide an approximation of the actual refrigeration cycle performance.

Empirical data from the Copeland CF12 reciprocating compressor shows that a compression ratio of 15:1 can realistically be achieved at the required mass flow rate (Emerson 2010). Compression ratios (CR) of 15:1 and 12:1 will be considered for this analysis because they will provide data near the working limit of the compressor, well within the operating capabilities. At each compression ratio the heat load will be fixed so that the refrigerant mixture is in a saturated vapor or superheated vapor state leaving the evaporator. With a suction pressure of 75.8 kPa (11 psia), the lowest theoretical temperature this refrigeration cycle could reach is -78.6 °C (-109.5 °F) entering the evaporator.

EXPERIMENTAL OPERATION

The system is analyzed at CR's of 15:1 and 12:1 for suction pressures of 75.8 kPa (11 psia) and 96.5 kPa (14 psia). A variable power supply is used to provide a constant heat load. The heat load varies depending on the suction and discharge pressures being measured. Thermocouples are attached to the inlet and outlet of all five evaporator plates in order to determine the cooling distribution on the evaporator plates. Since the applied load on the evaporator is constant, there is no need for a thermostat to provide feedback to the system from a common set-point. The performance of the refrigeration system will be recorded once the cycle reaches steady state, which is assumed to be achieved when the thermocouples change less than ±1 °C over a 15 minute time span.

A digital power meter is used to measure the power consumed by the compressor at each experimental operating point. The heat load applied to the evaporator plates is calculated from current and voltage measurements, assuming all of the heat provided enters the evaporator plates. Both the refrigeration capacity and the compressor power are needed to determine the overall performance of the mixed refrigerant system. The coefficient of performance (COP) of a refrigeration cycle is defined, in equation 1, as the refrigeration capacity divided by the compressor power:

$$COP = \frac{\dot{Q}_{EVAP}}{\dot{W}_{COMP}} \qquad\qquad COP_{Carnot} = \frac{T_{SOURCE}}{T_{SINK} - T_{SOURCE}} \tag{1), (2}$$

The maximum possible COP for a refrigeration cycle is the Carnot COP, which is defined in terms of the heat source and sink temperatures as shown in equation 2. In this experiment, the cooling water served as the heat sink while electric heaters served as the heat source; therefore, the entering water temperature is used as the sink temperature while the lowest evaporating temperature is used as the source temperature. The second law efficiency of the mixed refrigeration cycle, in equation 3, is found dividing the calculated COP of the system (equation 1) by the reversible, Carnot COP (equation 2):

$$\eta_{II} = \frac{COP}{COP_{Carnot}} \tag{3}$$

EXPERIMENTAL RESULTS

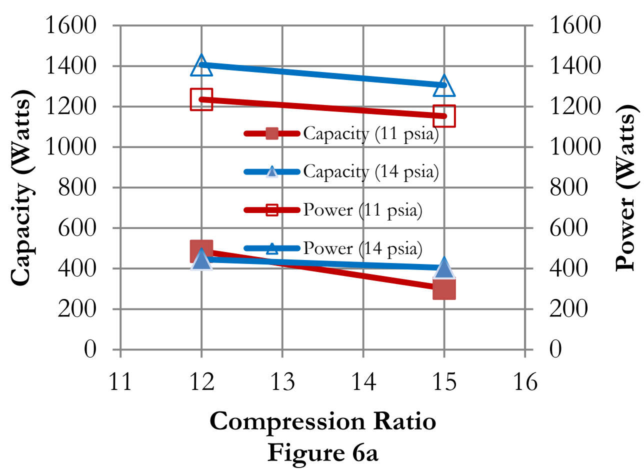
Four experiments were conducted with the refrigerant mixture of 33.4% R-23 and 66.6% R-134a by mass; the suction pressure was set to 75.8 kPa (11 psia) and 96.5 kPa (14 psia), and the compression ratio (CR) was set at 15:1 and 12:1 for each suction pressure. The suction pressure was controlled by changing the expansion valve in the R-134a stream entering the brazed plate heat exchanger. Discharge pressures were set by changing the inlet chilled water temperature and flow rate. Once the temperature of the evaporator plates dropped close to the temperature of the R-23 heavy stream exiting the capillary tube, a heat load was applied to the evaporator plates. The required heat load was estimated from the mixed refrigerant cycle model in order to ensure that the refrigerant mixture leaving the evaporator was heated to at least a saturated vapor state. It is important to minimize the superheat exiting the evaporator in a ULT application in order to maintain a low cabinet temperature, so achieving a saturated vapor state provides the largest refrigeration capacity available at that operating point. Because the refrigerant enters the compressor as a superheated vapor, the measured suction pressure and temperature fix the state and can be used to look up all other thermophysical properties.

Table 1: Performance characteristics at suction pressure and compression ratio.

Suction Pressure (psia|kPa)	Compression Ratio	Evaporator Temperature (° C)	COP	Carnot COP	Second Law Efficiency (η_{II})
11|75.8	12	-73	0.393	2.54	0.155
	15	-75	0.263	2.36	0.111
14|96.5	12	-65	0.317	2.54	0.124
	15	-63	0.310	2.47	0.125

Table 1 displays results for the evaporating temperature, COP, Carnot COP, and second law efficiency achieved at each operating condition. The temperatures of the refrigerant mixture are measured throughout the evaporator and the lowest temperatures recorded are presented in Table 1. At a suction pressure of 75.8 kPa (11 psia) and a discharge pressure of 96.5 kPa (165 psia), the temperature of the refrigerant mixture entering the evaporator reached -75 °C (-103 °F). The model predicted that the evaporator would reach a temperature of -87°C (-124.6 °F) under these conditions, but assumed perfect separation of the R-23 and R-134a streams exiting the separator. To estimate the impact of imperfect separation, the model is modified to assume that the R-23 heavy stream contains 65% R-23 and 35% R-134a, and the R-134a heavy stream contains 22% R-23 and 78% R-134a. With this modification the model predicts an evaporating temperature of -78.6 °C (-109.5 °F), which is very close to the measured temperature of -75°C (-103 °F) considering the thermocouple error of ±1 ° C. The temperature measurements in the evaporator also showed that the mixed refrigerant temperature increases drastically between the inlet and the exit of the evaporator. This phenomenon is supported by the pressure-enthalpy diagram in Figure 3, which shows the temperature glide of the zeotropic mixture in the two-phase region.

The measured refrigeration capacity, power consumption and second law efficiency for the mixed refrigerant system are presented in Figures 6a and 6b as a function of compression ratio and suction pressure. Most of the data follows the expected trend for refrigeration cycles, with the capacity, power and efficiency decreasing as compression ratio increases at a fixed suction pressure. However, for a suction pressure of 14 psia the increase from a compression ratio of 12:1 to 15:1 does not show the expected decrease in capacity. Figure 6b shows that this results in an unexpected increase in second law efficiency as compression ratio increases for a suction pressure of 14 psia.



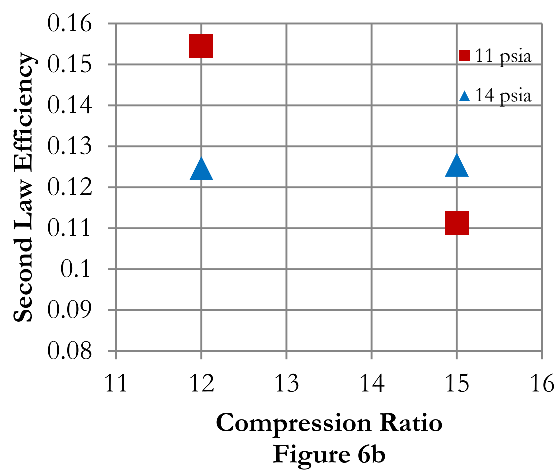


Figure 6: (a) Capacity and power consumption for suction pressure of 11 psia and 14 psia at varying compression ratios and (b) second law efficiency at suction pressures for varying compression ratios.

CONCLUSION

A model of an R-23/R-134a mixed refrigerant system for ULT applications is developed assuming complete separation of the two refrigerants in a water-cooled condensing separator. The analysis predicts that the refrigeration system can reach the target evaporator inlet temperature of -87 °C (-124.6 °F) using a mixture of 33.4% R-23 and 66.6% R-134a by mass. Further analysis of the separation column with the selected mixture of R-23 and R-134a showed that complete separation cannot be achieved realistically within the current operating range. Due to the incomplete separation, the evaporating temperature in the experimental refrigeration system was higher than the prediction from the theoretical analysis, reaching a low temperature of -75.0 °C (-103 °F). In order to reach a cabinet control temperature of -80 °C (-112 °F), the vapor stream leaving the separator must contain a larger portion of R-23 by mass. This could be achieved by improving the separation of the refrigerant mixture in the water cooled separator or by increasing the fraction of R-23 in the total mixture.

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NOMENCLATURE

COMP	=	Compressor
COP	=	Coefficient of Performance
EVAP	=	Evaporator
NIST	=	National Institute of Standards and Technology
ULT	=	Ultra Low Temperature

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