Performance Evaluation of a Flooded Ice Rink Chiller Retrofit from R-22 to R-449A

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ABSTRACT

The selection criteria of working fluids for industrial refrigeration, dictated by both safety and environmental factors, has evolved continuously since the advent of refrigerant technology. With each advancement in working fluid class, considerations are made around the compatibility with regards to retrofitting existing equipment. This is still the case for the transition to HFO blends. The combination of the R-22 phase out with a large majority of well-maintained R-22 ice rink chillers currently in operation creates a need for a retrofit gas that will meet or exceed existing performance criteria while complying with current regulatory requirements. However, it has long been assumed that when dealing with flooded evaporators, such as the case with many ice rink chillers, the predicted decrease in performance of zeotropic retrofit refrigerant blends compared to their incumbents is prohibitive. In the absence of a single component or azeotropic refrigerant retrofit option that matches R-22 energy performance, has an ozone depletion potential of zero, and has a low global warming potential, zeotropic blends must be evaluated.

This paper discusses the retrofit from R-22 to R-449A of an indirect ice rink refrigeration machine, also referred to as an ice plant, with a flooded evaporator. Ice plant power measurements were combined with evaporator heat loads to compare relative system efficiencies. A chiller model was employed to demonstrate estimated annual energy consumption between R-22 and R-449A based on the collected run data. Finally, refrigerant samples one-month and one-year post retrofit were used to visualize the refrigerant blend composition shift throughout the system.

1. INTRODUCTION

The search for a "drop-in" alternative refrigerant to R-22 in systems with flooded evaporators has been a sort of "holy grail" for the industrial refrigeration industry for many years. Now, with the end of R-22 production in the US and Canada, owners and operators of chillers are exploring refrigerant conversion options in existing systems. The only other choices to date have been to continue to operate with R-22 (risking pricing and availability issues), making significant system changes to adopt a refrigerant with properties unlike R-22, or entirely replace the system at significant capital cost.

This paper reports the results of an R-22 to R-449A refrigerant conversion in an ice rink chiller with a flooded (shell and tube) evaporator. R-449A is a non-ozone depleting, low GWP HFO refrigerant blend with moderate temperature glide. Historically, flooded chillers have employed only azeotropic or pure fluids, due to performance and operation concerns over blends with glide. The successful conversion reported here indicates that R-449A can be a viable replacement for R-22 in certain systems with flooded evaporators.

2. ICE PLANT LAYOUT

The facility in this study consists of two sheets of ice served by a common refrigeration system originally designed for R-22. The system (Figure 1) utilizes three open-type reciprocating compressors, a common coalescing oil filter, an evaporative condenser, a shell and tube flooded evaporator, and various ancillary components.

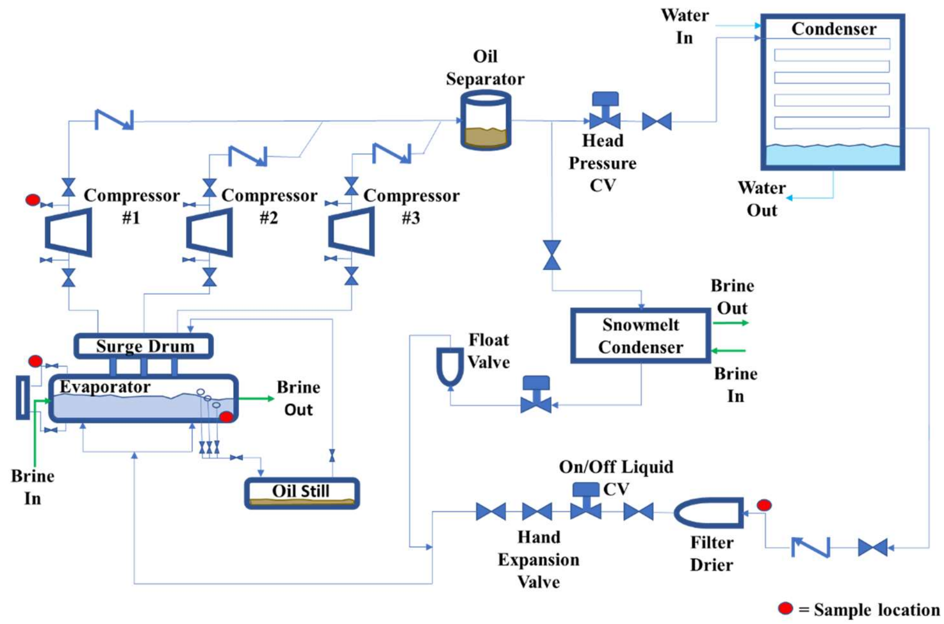


Figure 1: Ice plant layout and sample locations

Calcium chloride brine flows through the evaporator tubes and is circulated by centrifugal pumps to provide cooling to the rink floor. Each sheet of ice has a dedicated brine pump with one standby pump between the two rinks. The brine pumps utilize variable frequency drives which modulate brine flow based on the measured ice temperature. Likewise, the lead compressor utilizes a variable frequency drive to maintain adequate suction pressure and thereby evaporator temperature. The other two compressors come on as the load dictates with capacity control through cylinder loading and unloading.

3. RETROFIT PROCESS

The retrofit process from R-22 to R-449A took place over two days. The first day consisted of an oil change from mineral to POE (polyolester) oil as well as the replacement of the compressor oil filters. A transition away from mineral oil is recommended due to the poor miscibility of mineral oil with HFC/HFO blends. At the same time, easily accessible critical elastomeric seals were replaced. When elastomers come in contact with R-22, the seals swell as the refrigerant is absorbed into the material. When the R-22 is removed from the system, these seals degas causing them to shrink. The deformation of the seals due to expansion and contraction creates a potential future leak point when a new refrigerant is charged into the system. Seal locations in refrigeration systems include Schrader valve pins and caps, liquid level sight glasses, ball valves, solenoid valves, flanges, and shaft seals on open-drive compressors. The term critical is typically applied to seals that cannot be easily isolated and would require a complete removal of the refrigerant charge to fix. Typically, seals on the high-pressure side of the system are considered critical. Specific locations where elastomers were replaced in this study included oil return solenoid valve o-rings and high side control valve o-rings. Internal compressor seals were not replaced and have not exhibited signs of leakage.

Day two of the retrofit began with the recovery of refrigerant. Liquid refrigerant was first fed into a recovery cylinder from the high-pressure liquid line out of the condenser while the system was running. The system compressors were run down to around 0 psig. The ice plant was then deenergized and a recovery machine was deployed to evacuate the remaining vapor.

Once the system was fully evacuated, evaporator liquid sight glass o-rings were replaced as they were also deemed critical. A few system modifications were then made at the request of the owner. Prior to the retrofit, a level indicator and a hand expansion valve in series with an on/off control valve were used to maintain a set temperature and liquid level of refrigerant in the evaporator shell. An electronic expansion valve was installed in parallel with the existing hand expansion and level control valve. The electronic valve was programmed to operate if the evaporator liquid dropped below a prescribed level while the hand expansion valve maintains a minimum load. A new level probe was also installed to replace the existing float. Additional work included the replacement of filter drier cores and the relocation of the brine outlet temperature sensor further away from the evaporator exit. The original sensor, located at the top of the brine outlet pipe, was thought to be misrepresentative of the bulk fluid temperature. Being a single pass exchanger, it is possible for temperature stratification exiting the evaporator where a measurement at the top of the pipe could be warmer than the bottom depending on the evaporator liquid refrigerant level. This measurement is critical in quantifying evaporator load.

4. DATA COLLECTION AND ANALYSIS

Only one sheet of ice was in operation during the data collection period. As a result, only one brine pump and one compressor were enabled. System data was collected every three minutes. R-22 run data was obtained for the month leading up to the retrofit. Data was then collected for a two-week period immediately post-retrofit, and a two-week period approximately one-year post-retrofit.

Due to the lack of instrumentation related to refrigerant pressures, temperatures, and flows, brine side data was used to quantify performance. Efficiency differences pre- and post-retrofit are presented as relative values as there was no brine flowmeter nor was information available on the underground brine piping circuits. In the absence of flow data and the ability to generate a hydraulic model to quantify friction losses, relative mass flows and subsequent evaporator loads were calculated neglecting friction and velocity heads. Experimental values of pump operation and the manufacturers pump curve were used to correlate the measured brine pump motor frequency to a relative brine volumetric flowrate. Corresponding mass flowrates were then calculated assuming a fixed calcium chloride density. The rate of heat transfer, Q, in the evaporator was then found using relative mass flowrate ($\dot{m}$), a fixed brine specific heat (c_p), and brine temperature difference (ΔT) through the evaporator (Lindeburg, 2013).

$$Q = \dot{m} c_p \Delta T \tag{1}$$

A relative chiller efficiency in kW/Ton was calculated for each data set acquired by dividing the total chiller power readings by the calculated evaporator loads. Total chiller power readings taken from the system controller incorporated draw from the compressor motors, condenser water circulation pump, condenser fan motor, brine pumps, snowmelt pump, and ancillary equipment (i.e. crankcase heaters and controls). Relative system efficiencies for pre- and post-retrofit can be seen in Figure 2 below.

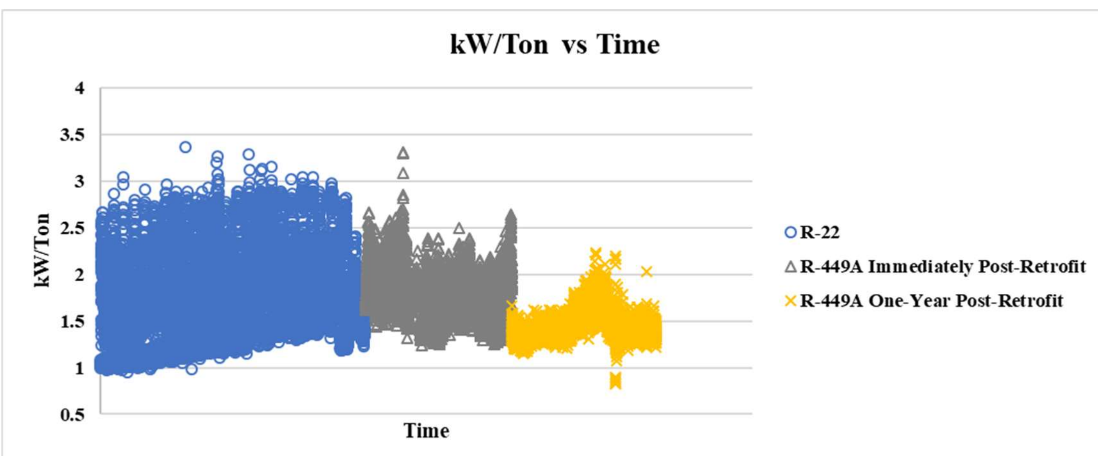


Figure 2: System kW/Ton versus Time

A slight efficiency improvement was noted when comparing the R-449A one-month data to one-year post-retrofit. This is in part due to system optimization by the technician through controls tuning and setpoint adjustments. However, the one-year data also coincided with a period of relatively minimal rink usage and an associated low system load. The key take-away is that relative power requirement per ton of refrigeration did not increase as a result of the retrofit.

When comparing system performance to ambient dry bulb temperature (Figure 3), it was found that efficiencies were consistent both before and after the retrofit over a range of ambient temperatures.

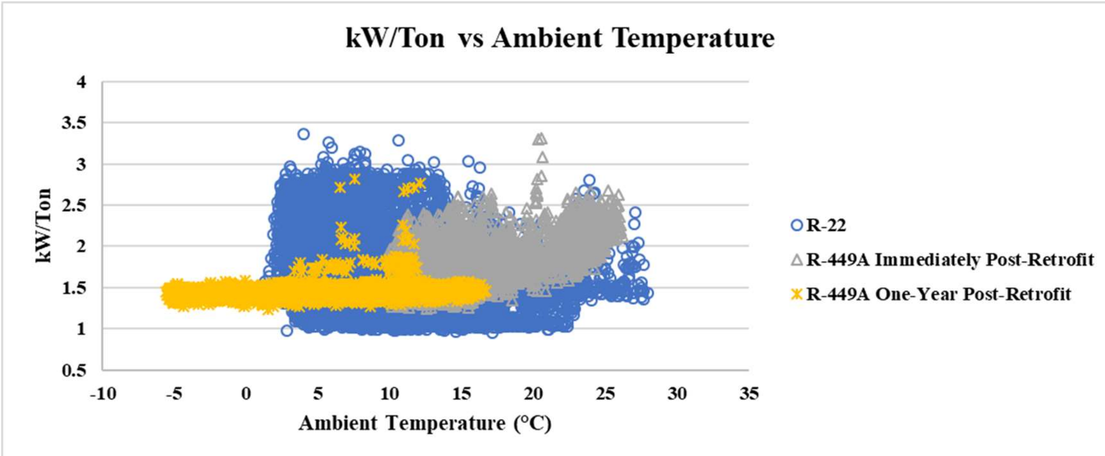


Figure 3: System kW/Ton versus Ambient Temperature

Suction and discharge pressures were slightly increased from R-22 to R-449A as expected. However, R-449A operating pressures were still within the design envelope of the R-22 equipment. Also as expected, discharge temperatures of R-449A were lower than that of R-22. Experimental values can be seen in Table 1.

Table 1: Average Experimental Operating Values

	R-22	*R-449A*
Suction Pressure - Average psig (MPa)	31.14 (0.215)	32.45 (0.224)
Discharge Pressure - Average psig (MPa)	149.85 (1.03)	165.44 (1.14)
Discharge Temperature - Average °F (°C)	151.90 (66.61)	133.75 (56.53)

Ice plant load is heavily dependent upon rink usage, resurfacing, and facility temperature and humidity. Due to the inherent nature of ice rink operation, no two days will have the exact same load profile, thereby making retrofit comparisons difficult. This is only compounded by the fact that ambient temperature ranges present pre-retrofit do not entirely line up with temperature ranges post-retrofit. To address the variability of ice plant load and ambient temperatures when comparing performance before and after, the collected data was used to generate a system model using the methodology proposed by Gordon and Ng (GN) (Jian and Reddy, 2003).

The form of the GN model employed can be seen in Equation 2 below.

$$\left(\frac{1}{COP}+1\right)\frac{T_{CHWout}}{T_{CONDin}}-1=c_1\frac{T_{CHWout}}{Q_{evap}}+c_2\frac{(T_{CONDin}-T_{CHWout})}{T_{CONDin}Q_{evap}}+c_3\frac{\left(\frac{1}{COP}+1\right)Q_{evap}}{T_{CONDin}} \tag{2}$$

Where T_{CHWout} is the evaporator brine outlet temperature, T_{CONDin} is the condenser air inlet temperature, Q_{evap} is the evaporator load, and c_1, c_2, and c_3 are calculated model coefficients.

By defining COP as evaporator load divided by ice plant power consumption and fitting the model to a linear regression, an expression was generated (Equation 3) for the dependent chiller power based on inputs of condenser inlet air temperature (ambient dry bulb temperature), evaporator brine outlet temperature, and evaporator load.

$$P = Q_{evap}\left[\frac{T_{CONDin}\left[1 + c_1\frac{T_{CHWout}}{Q_{evap}} + c_2\frac{(T_{CONDin} - T_{CHWout})}{T_{CONDin}Q_{evap}}\right] + c_3Q_{evap} - T_{CHWout}}{T_{CHWout} - c_3Q_{evap}}\right] \tag{3}$$

After validating the goodness-of-fit for the R-22 and R-449A models, ice plant energy consumption was modeled from binned weather data (ASHRAE, 2017) nearest to the ice rink location using a constant 35 Ton (123.1 kW) load with a 17°F (-8.33°C) brine outlet temperature and a minimum condensing temperature of 50°F (10°C). Figure 4 shows the energy consumption profiles for R-22 and R-449A as well as the yearly totals for both fluids.

	Yearly kWh
R-22	476,914
R-449A	478,181
Difference	-1,267

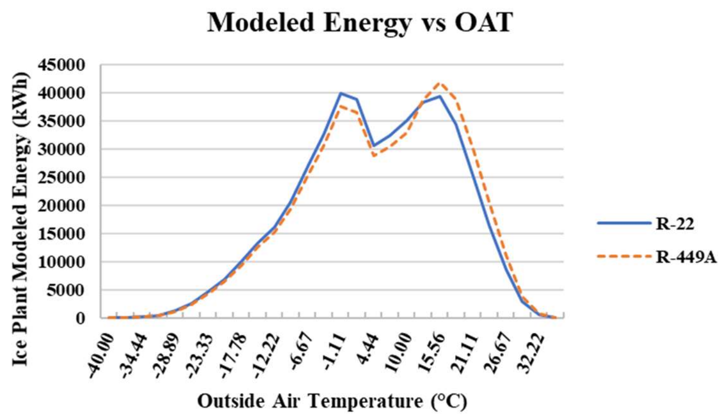


Figure 4: Modeled ice plant energy consumption versus outside air temperature

The R-449A model was developed using the data collected during the two weeks immediately post-retrofit since the one-year data was primarily at low load. The model outputs show negligible differences in energy consumption between the two fluids over a year of operation.

5. REFRIGERANT SAMPLE ANALYSIS

Refrigerant sampling was conducted one-month post-retrofit and one-year post-retrofit. Sample locations included evaporator liquid, evaporator (or suction) vapor, discharge vapor, and condenser liquid outlet. While the concentrations vary slightly between the two sample dates, the general trends were the same and can be visualized using the one-month results in Figure 5 below. The nominal (or initial charge) composition of R-449A is 24.3% R-32, 24.7% R-125, 25.3% R-1234yf, and 25.7% R-134a.

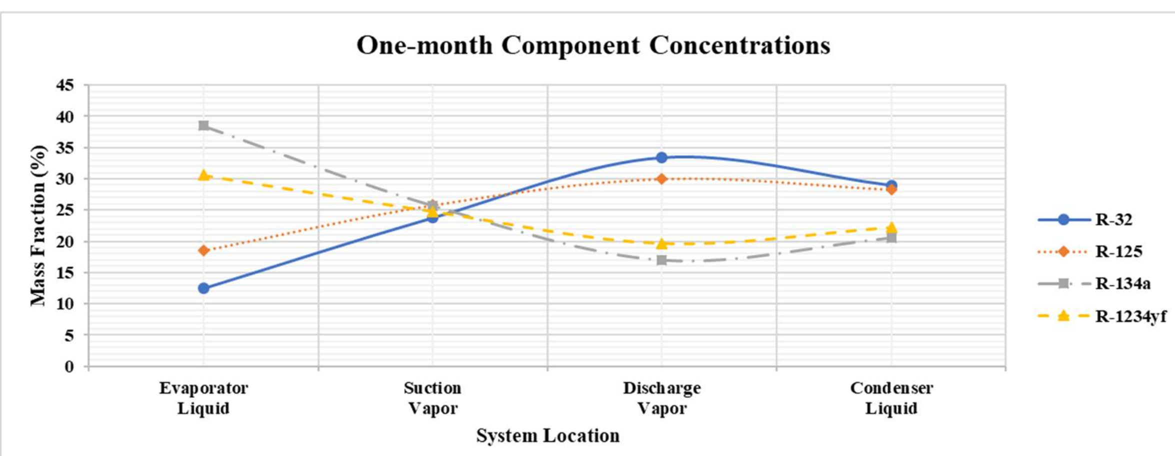


Figure 5: One-month sample results

As expected, the low boilers (i.e. R-32 and R-125) preferentially exit the evaporator and circulate through the system while the high boilers (i.e. R-134a and R-1234yf) tend to preferentially remain in the liquid phase within the evaporator. To take a closer look at the samples, R-32 was used in Figure 6 to represent the low boilers and R-1234yf was used in Figure 7 to represent the high boilers. Nominal component values for R-449A were also plot as a reference.

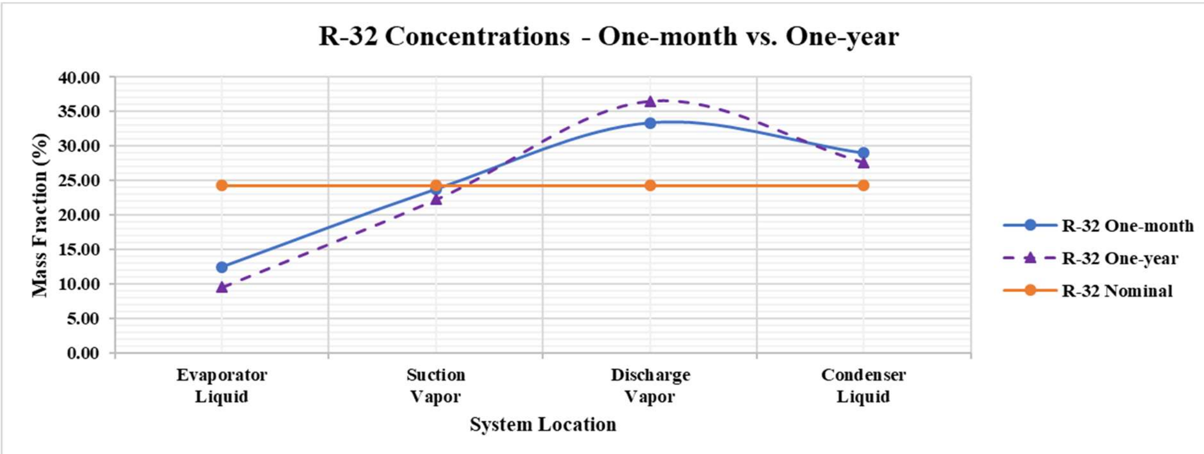


Figure 6: R-32 Concentrations – One-month vs. One-year

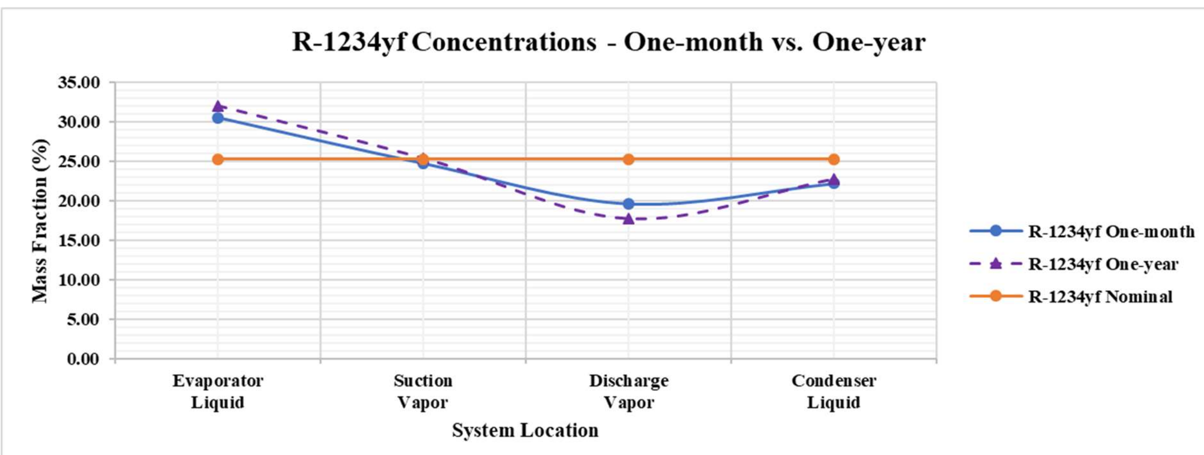
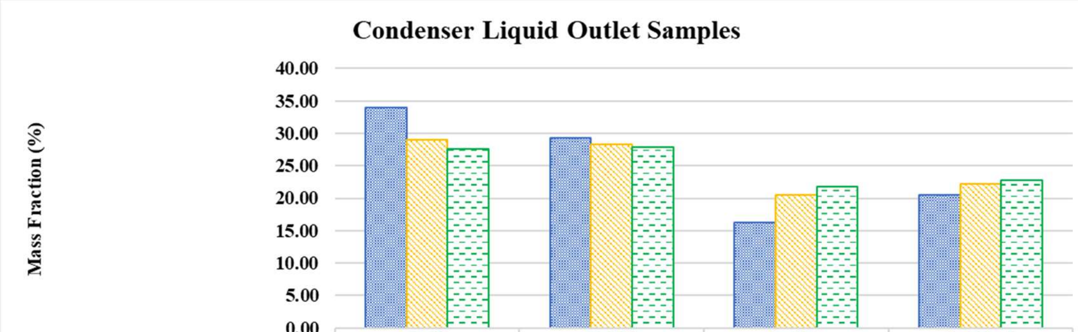


Figure 7: R-1234yf Concentrations – One-month vs. One-year

The small variations found between the one-month and one-year samples were not a surprise due to the dynamic nature of system operation. Note that both refrigerant components reach the nominal concentration of R-449A near the sample location titled "Suction Vapor". The increase in R-32 concentration and associated decrease in R-1234yf concentration from the suction vapor sample to the discharge vapor sample could be explained by the suction sample location within the headspace of the evaporator and the possibility for the presence of liquid refrigerant. It is also important to note that the R-32 concentration decreases while the R-1234yf concentration increases between the compressor discharge and the condenser exit. One explanation for this phenomenon could be due to the lower boiling point substances (i.e. R-32) being trapped within the condenser. A more likely perspective is that if the stream exiting the condenser is not saturated or subcooled, the liquid sampled in the lab could have been the liquid mass fraction of a liquid-vapor mixture.

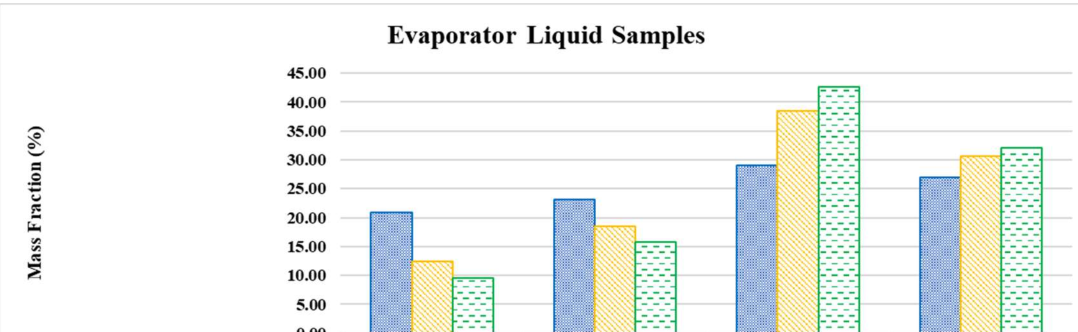
To understand our ability to predict the refrigerant behavior, calculated compositions were generated using the average R-449A field suction and discharge pressure readings. Using the average discharge pressure, evaporator inlet quality was calculated assuming a saturated liquid exiting the condenser and an isenthalpic expansion process. Quality and suction pressure were then used as inputs to NIST REFPROP 10 (Lemmon *et al.*, 2018) to obtain predicted vapor and liquid mass fractions of R-449A. While this calculation could be iterated or further refined using a specific chiller thermodynamic model and known refrigerant inventories, it was found to be a good first approximation.



	R-32	R-125	R-134a	R-1234yf
R-449A Condenser Liquid - Predicted	33.96	29.23	16.28	20.54
Condenser Liquid One-month	28.98	28.25	20.56	22.21
Condenser Liquid One-year	27.59	27.89	21.75	22.76

Figure 8: Condenser liquid outlet samples and predicted compositions

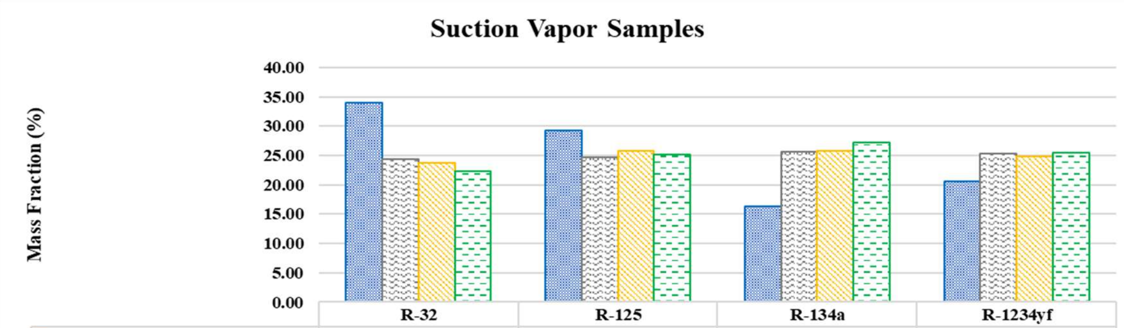
The condenser liquid outlet samples are compared in Figure 8 above to the R-449A predicted composition under the assumptions that the condenser outlet is a saturated liquid and there is no opportunity for the refrigerant components to separate or accumulate.



	R-32	R-125	R-134a	R-1234yf
R-449A Evaporator Liquid - Predicted	20.83	23.07	29.08	27.01
Evaporator Liquid One-month	12.46	18.49	38.47	30.58
Evaporator Liquid One-year	9.58	15.71	42.62	32.09

Figure 9: Evaporator liquid samples and predicted compositions

The difference in composition of the evaporator liquid samples from the predicted values as seen in Figure 9 above is likely an effect of the samples consisting of a liquid-vapor mixture as boiling is continuously occurring in the evaporator.

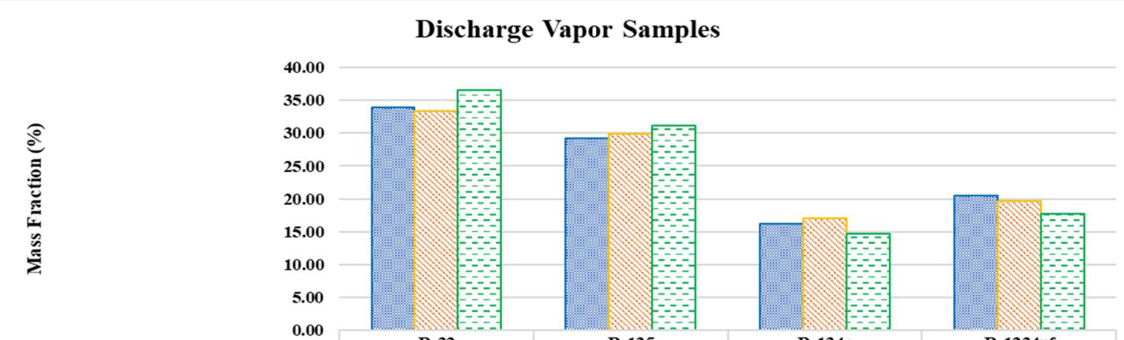


	R-32	R-125	R-134a	R-1234yf
R-449A Suction Vapor - Predicted	33.96	29.23	16.28	20.54
R-449A Nominal Composition	24.3	24.7	25.7	25.3
Suction Vapor One-month	23.76	25.75	25.72	24.77
Suction Vapor One-year	22.25	25.20	27.13	25.41

Figure 10: Suction vapor samples and predicted compositions

As stated earlier, the suction vapor samples were likely liquid-vapor mixtures and not saturated vapor as they were taken from the evaporator vapor disengagement chamber. In fact, the sample results seen in Figure 10 show a closer match to the nominal R-449A composition than the predicted vapor mass fraction.

As the refrigerant continues to absorb heat and travels through the disengagement chamber and into the compressor suction, the vapor composition begins to match the predicted values. This is evident in the discharge vapor sample results in Figure 11.



	R-32	R-125	R-134a	R-1234yf
R-449A Discharge Vapor - Predicted	33.96	29.23	16.28	20.54
Discharge Vapor One-month	33.40	29.96	17.00	19.64
Discharge Vapor One-year	36.46	31.11	14.72	17.71

Figure 11: Discharge vapor samples and predicted compositions

Refrigerant compositions of a zeotropic blend in a chiller employing a flooded evaporator are dictated by the location within the system, the relative refrigerant inventories throughout the system, and operating conditions such as evaporator temperature and associated expansion valve loading. Ambient temperature profiles, system load, and the use of ancillary equipment such as snowmelt condensers and subfloor heating all affect the dynamics of refrigerant inventories and operating conditions in an ice rink chiller. With these complexities, variations in refrigerant composition from field samples over time would be expected. Additional modeling efforts coupled with measurements related to heat transfer and pressure drop would aid in refrigerant composition and overall system performance predictions. The takeaway messages from these samples should be that the nominal R-449A concentration is achieved somewhere within the evaporator, the discharge vapor composition can be closely approximated using field data, and the composition profile has not changed over time.

6. CONCLUSIONS

The results shown in this study identify R-449A as a viable retrofit solution for an indirect flooded R-22 chiller. Ice rink chillers lend themselves well to this type of retrofit mainly due to their simplistic design. Additionally, the specific load profiles of these systems are relatively steady and temperature tolerances are less stringent than some precision cooling applications.

In the absence of improved refrigerant instrumentation and a means to replicate load profiles pre- and post-retrofit, the following general conclusions can be made:

- R-449A was shown to exhibit equal or improved relative system efficiencies (kW/Ton) compared to R-22.
- Suction and discharge pressures and temperatures of R-449A were shown to be within the limits of existing system components.
- Employing a chiller performance model based on first principles, total system energy usage was demonstrated to be nearly equal as compared to R-22 for a given system load.
- R-449A sample analysis shows consistent composition profiles over a year of operation.
- R-449A is a viable replacement for R-22 in ice rink refrigeration systems with flooded evaporators.

NOMENCLATURE

Azeotropic Refrigerant	Blend of two or more refrigerants whose equilibrium vapor-phase and liquid-phase compositions are the same at a given pressure.
Chiller	Refers to the entire refrigeration system in this study, not just the evaporator. Synonymous with industry terminology such as "Ice Plant."
Flooded Evaporator	For the purposes of this study, this is a type of shell and tube exchanger where refrigerant occupies the shell side and process fluid occupies the tubes.
Glide	For the purposes of this study, glide refers to the difference between refrigerant dew and bubble point temperatures at a given pressure.
GWP	Global warming potential
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
Indirect Chiller	Refrigerant is used to cool a secondary fluid which in-turn circulates to cool the process load (or the ice rink floor in the application of a rink chiller.)
Zeotropic Refrigerant	Blend of two or more refrigerants whose equilibrium vapor-phase and liquid-phase compositions are different at a given temperature.

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