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## A Reduced GWP Replacement for HFC-134a in Centrifugal Chillers: XP10 Measured Performance and Projected Climate Impact

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### ABSTRACT

The performance of XP10, a new refrigerant with a reduced GWP, in a centrifugal chiller designed for HFC-134a with a cooling capacity of 1969.44 kW (560 tons) was measured under full and part load conditions and compared to performance with HFC-134a. Measured chiller energy efficiencies with XP10 were comparable to those with HFC-134a. They resulted in 0.6% higher energy consumption for XP10 when integrated over a representative profile of partial loads as described by AHRI Standard 550/590. Based on the measured chiller performance, XP10 could be considered a near drop-in replacement for HFC-134a in centrifugal chillers. It could replace HFC-134a in existing chillers or enable optimized new chiller designs without extensive equipment and no flammability code modifications. The Total Equivalent Warming Impact (TEWI) of a chiller with XP10 was estimated under representative scenarios and compared to TEWI with HFC-134a. Use of XP10 could significantly reduce chiller global warming impact relative to HFC-134a, when chiller refrigerant losses are unavoidably high or when electricity is generated with low GHG emissions. XP10 has the potential to be a more environmentally sustainable future option for medium pressure centrifugal chillers.

### 1. INTRODUCTION

Air conditioning of large commercial and institutional buildings is commonly provided through medium pressure centrifugal chillers using HFC-134a as the working fluid. The main components of a typical centrifugal chiller are an evaporator, a condenser, an expansion valve, a centrifugal compressor, a compressor drive and a controller. The condenser typically includes a section to subcool the liquid. The evaporator and condenser typically use tubes with enhanced heat transfer surfaces. Further details of a centrifugal chiller can be found in the ASHRAE Handbook (2008).

Chillers consume a substantial fraction of the electrical energy globally and contribute significantly to global greenhouse gas (GHG) emissions. Increasing awareness of the risks to the earth's climate posed by anthropogenic GHG emissions and emerging climate protection regulations are motivating a search for new refrigerants that would reduce the global warming impact from chillers. Hydro-Fluoro-Olefins (HFOs) have been identified as a new class of compounds that could enable the formulation of refrigerants with Global Warming Potentials (GWPs) substantially lower than those of incumbent refrigerants. A new refrigerant, XP10, was recently proposed as a potential reduced-GWP replacement for HFC-134a in centrifugal chillers by Kontomaris et al. (2010) and Kontomaris (2011). (XP10 is a slightly reformulated version of developmental refrigerant DR-11 described by Kontomaris et al. (2010).)

Table 1 compares key properties of XP10 to HFC-134a. XP10 is an azeotropic blend containing HFO-1234yf (CF<sub>3</sub>CF=CH<sub>2</sub>). XP10 is non-flammable at 60 °C according to ASTM 681-01. It has no ozone depletion potential (ODP) and a 100 year horizon GWP of about 600, i.e. about 58% lower than HFC-134a. The boiling temperature,

$T_b$ , of XP10 under atmospheric pressure is about 3 °C lower than that of HFC-134a. The critical temperature,  $T_{cr}$ , of XP10 is also slightly lower than that of HFC-134a but it remains comfortably higher than typical chiller working temperatures.

**Table 1:** Basic properties of XP10 compared to HFC-134a

Property	HFC-134a	XP10
Chemical Formula	CH <sub>2</sub> F-CF <sub>3</sub>	Azeotrope
ODP	none	none
GWP (100 yr horizon)	1,430	about 600
$T_b$ [°C]	-26.1	-29.2
$T_{cr}$ [°C]	101.1	97.5
$P_{cr}$ [MPa]	4.06	3.82

HFC-134a is commonly used with polyol ester (POE) type lubricating oils. The miscibility of XP10 with commercially available chiller POE lubricants was tested over a wide range of concentrations and temperatures that covers the operating ranges typically encountered in centrifugal chillers; it was found to be comparable to that of HFC-134a. The stability of XP10 in the presence of materials that it would likely encounter in practical use was scrutinized according to the sealed tube testing methodology of ANSI/ASHRAE Standard 97-2007. At test conditions, XP10 and XP10/POE blends in the presence of steel, copper and aluminum showed thermal stability comparable to that of HFC-134a. The chemical compatibility of the components of XP10 with a wide range of plastics and elastomers has been thoroughly tested and found to be comparable to HFC-134a.

Table 2 compares predicted cycle performance of XP10 to HFC-134a for typical centrifugal chiller conditions. The coefficient of performance, COP, with XP10 is predicted to be about 2.3% lower than HFC-134a. The volumetric cooling capacity, VCC, with XP10 is predicted to be 1.8% higher than HFC-134a. The components of XP10 form nearly azeotropic mixtures over a wide range of compositions and temperatures. The XP10 composition is azeotropic at typical chiller evaporator and condenser temperatures with predicted temperature glides lower than 0.02 °C.

**Table 2:** Predicted thermodynamic cycle performance of XP10 relative to HFC-134a at representative chiller conditions:  $T_{evap} = 4.44$  °C (40 °F),  $T_{cond} = 37.78$  °C (100 °F),  $\Delta T_{subc} = 0$  °C (0 °F),  $\Delta T_{suph} = 0$  °C (0 °F),  $\eta_{is} = 0.70$ , negligible pressure drops

Property	HFC-134a	XP10	XP10 vs HFC-134a [%]
$COP_{theo}$	4.849	4.738	-2.29
$VCC_{theo}$ [kJ/m <sup>3</sup> ]	2,482.78	2,528.51	1.84
Evaporator & Condenser Temp Glide [°C]	N/A	<0.02	

The global warming impact of a cooling application is one of the criteria used to select among competing refrigerant choices. It depends on several factors including the GWP of the selected refrigerant, the refrigerant emission rate, the energy efficiency with the selected refrigerant, ambient conditions and the primary energy mix used to generate the electricity consumed. It is often quantified in terms of the Life Cycle Climate Performance (LCCP) defined as the total amount of CO<sub>2</sub>, in kg, that would produce a global warming impact equivalent to that of all GHGs emitted in the realization of an application over its lifetime (“cradle to grave”). The GHG emissions associated with the operating energy consumed,  $EM_{NRG}$ , and the refrigerant emissions during,  $EM_{RFG}$ , and at the end of equipment life,  $EM_{RFG-EOL}$ , are usually the dominant contributions to global warming resulting from chiller air conditioning applications. These contributions are added to estimate the Total Equivalent Warming Impact (TEWI) of an application, an easier to estimate and almost as informative a metric as LCCP:

$$TEWI = EM_{NRG} + EM_{RFG} + EM_{RFG-EOL} \quad (1)$$

The TEWI of chillers operating with XP10 relative to HFC-134a under possible scenarios was evaluated by Kontomaris (2011). In the absence of measurements, energy consumption for XP10 chillers was estimated by adjusting HFC-134a energy consumption data according to the theoretical XP10 and HFC-134a COP values. XP10 could enable significant chiller warming impact reductions relative to HFC-134a, except in the case of

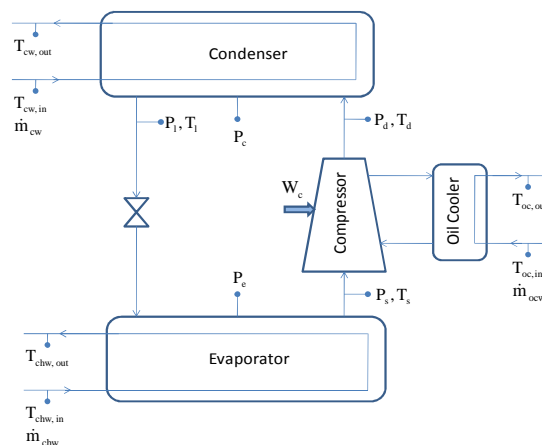
simultaneously high electricity carbon intensity and minimal refrigerant emissions. The primary objective of the work reported in this paper was to measure the performance of XP10 in a centrifugal chiller and to assess the potential of XP10 to replace HFC-134a in new or existing centrifugal chillers. A second objective of this paper was to evaluate the reductions in chiller global warming impact that could be enabled by replacing HFC-134a with XP10 under realistic scenarios. Only water-cooled chillers driven by grid electricity were considered.

## 2. METHODS

The performance of a test chiller with XP10 was measured and compared to HFC-134a. The energy consumption and fluid charge data were used in a TEWI analysis to estimate the global warming impact of XP10 relative to HFC-134a.

### 2.1 Chiller Performance Testing

A centrifugal chiller in an AHRI approved chiller testing facility was used to test the performance of XP10 and HFC-134a. The test chiller main components, configuration and selected measurement locations are depicted in Figure 1. The chiller compressor was equipped with a variable speed drive (VSD) and pre-rotation (or inlet guide) vanes (PRVs). The evaporator and condenser were horizontal shell-and-tube type with refrigerant flooded on the shell side. They had heat transfer tubes with surfaces enhanced on both the refrigerant and the water side. The condenser tube surfaces on the refrigerant side had notched fin enhancements. The refrigerant side surface enhancements were specifically optimized for HFC-134a.



**Figure 1:** Test chiller configuration and key measurement locations

Test chiller flows, temperatures, pressures and power consumption were measured with instruments meeting AHRI (2003) requirements calibrated in place against NIST standards. The evaporator was instrumented with two resistance temperature detectors (RTDs) to measure the incoming water temperature, two RTDs to measure the exiting water temperature and three pressure transducers on its shell to determine saturation pressure. The condenser was instrumented with two RTDs to measure the incoming water temperature, one RTD to measure the exiting water temperature, two pressure transducers to measure the saturation pressure and a pressure transducer and one RTD to determine the state of the refrigerant at the outlet of the sub-cooling section. The evaporator and condenser water flows were measured using turbine flow meters. The compressor power consumption was measured with a power analyzer designed for use with VSDs. Redundant measurements were averaged before use in data reduction and analysis. The instrument measurement uncertainties are listed in Table 3. Both HFC-134a and XP10 were tested without any instrument adjustments.

**Table 3:** Measurement Uncertainties

Parameter	Uncertainty
Temperature	$\pm 0.06^\circ\text{C}$
Pressure	$\pm 1.7 \text{ kPa}$
Volume Flow Rate	$\pm 0.15\%$
Electrical Power	$\pm 0.15\%$

The chiller was charged with the respective refrigerants so as to fill the condenser up to a specified level and the flooded evaporator up to the top of the heat transfer tube bundle as confirmed by direct observation through a sight glass. The required XP10 charge mass was **97.4%** of the required HFC-134a charge mass. The steady-state chiller electrical power draw and the condenser and evaporator overall heat transfer coefficients were measured with both HFC-134a and XP10 at eleven cooling load levels including the levels prescribed by AHRI Standard 550/590 (2003) for the calculation of the Integrated Part-Load Value (IPLV) for chillers with water-cooled condensers (100%, 75%, 50% and 25% of full load). The evaporator and condenser water flow rates were fixed for all the tests. The temperatures of the chilled water and condenser water were varied in accordance with AHRI Standard 550/590 (2003) for part load conditions. At full load conditions, the compressor impeller speed with each fluid was set so as to maximize compressor efficiency with PRVs fully open. The compressor speed was **4.8%** lower with XP10 than with HFC-134a due to differences in fluid properties. The test chiller cooling capacity at full load conditions and peak compressor efficiency was 1969.44 kW (**560 ton**). At test loads higher than 30% of full load, the impeller speed for each fluid was kept at the respective full-load value and the position of the PRVs was adjusted to vary the chiller capacity. At test loads at or below 30% of full load, the position of the PRVs was held fixed and the impeller speed was reduced to reduce cooling capacity while ensuring stable operation with both fluids. The compressor power consumption at all load levels was measured at the output of the VSD to eliminate the effect of varying VSD energy losses at different impeller speeds. Measurements taken after steady operation was established at each load level were checked for consistency before calculating state and performance variables. Mass and energy balances were used to verify the stability of the test conditions and data consistency. The rates of energy input to and output from the chiller were compared as described in [Table 4](#). The refrigerant mass flow rate calculated using evaporator data (heat transfer rate and calculated refrigerant enthalpies at inlet and outlet) was compared to the flow rate calculated using condenser data as shown in [Table 4](#).

**Table 4:** Energy and mass balance calculations

Energy Balance	Mass Balance
$Q_e = \dot{m}_{chw} \cdot c_{pchw} \cdot (T_{chw,in} - T_{chw,out})$	$h_{r,c,in} = h(P_d, T_d); h_{r,c,out} = h(P_l, T_l)$
$Q_c = \dot{m}_{cw} \cdot c_{pcw} \cdot (T_{cw,out} - T_{cw,in})$	$\dot{m}_{r,c} = \frac{Q_c}{(h_{r,c,in} - h_{r,c,out})}$
$Q_{oc} = \dot{m}_{ocw} \cdot c_{poc} \cdot (T_{oc,out} - T_{oc,in})$	$h_{r,e,out} = h(P_s, T_s)$
$\dot{Q}_{in} = Q_e + W_c$	$\dot{m}_{r,e} = \frac{Q_e}{(h_{r,e,out} - h_{r,c,out})}$
$\dot{Q}_{out} = Q_c + Q_{oc}$	
$Q_{av} = 0.5 \cdot (Q_{in} + Q_{out})$	$\dot{m}_{r,av} = 0.5 \cdot (\dot{m}_{r,e} + \dot{m}_{r,c})$
$EB = \frac{(Q_{out} - \dot{Q}_{in})}{Q_{av}}$	$MB = \frac{(\dot{m}_{r,c} - \dot{m}_{r,e})}{(\dot{m}_{r,av})}$

**Table 5:** Calculation of overall heat transfer coefficients

Evaporator	Condenser
$LMTD_e = \frac{(T_{chw,in} - T_{sat,e}) - (T_{chw,out} - T_{sat,e})}{\ln((T_{chw,in} - T_{sat,e}) / (T_{chw,out} - T_{sat,e}))}$	$LMTD_c = \frac{(T_{sat,c} - T_{cw,in}) - (T_{sat,c} - T_{cw,out})}{\ln((T_{sat,c} - T_{cw,in}) / (T_{sat,c} - T_{cw,out}))}$
$U_{o,e} = \frac{Q_e}{A_e \cdot LMTD_e}$	$U_{o,c} = \frac{Q_c}{A_c \cdot LMTD_c}$

Evaporator and condenser performance with each refrigerant was quantified in terms of overall heat transfer coefficients calculated as shown in [Table 5](#). The temperatures of water streams entering and leaving the evaporator

and the condenser were directly measured. The saturation temperatures were calculated from the shell saturation pressures averaged over measurements with multiple transducers. HFC-134a thermophysical properties were evaluated using NIST Refprop 8.0; XP10 thermophysical properties were evaluated using in-house data.

## 2.2 TEWI Evaluation

**2.2.1 Energy-Related Emissions,  $EM_{NRG}$ :** The equivalent CO<sub>2</sub> emissions,  $EM_{NRG}$ , from electricity consumption to operate the chiller over its lifetime, were estimated as shown in [Table 6](#); they depend on the chiller energy efficiency and the primary energy sources used. Chiller energy efficiency varies with daily and seasonally varying weather conditions. Power consumption for HFC-134a and XP10 was based on the measured Integrated Part Load Values (IPLV). The mix of primary energy sources used to generate the electricity supplied to a chiller connected to a regional grid determines the amount of equivalent CO<sub>2</sub> emitted per unit of electricity consumed, referred to as [Carbon Intensity \(CI\)](#). CI varies regionally, seasonally and daily; average CI values in Switzerland (**0.0150** kgCO<sub>2</sub>-eq/kwh) and China (**0.8445** kgCO<sub>2</sub>-eq/kwh), reported by the World Resource Inst. (2006), were used as representative low and high CI levels, respectively.

**Table 6:** Calculation of equivalent emissions from electricity consumption

$COP_{actual}$	IPLV Actual Coefficient of Performance (measured)
$Q_{evap}$ [kw]	Cooling rate: <b>1,142.27</b> kw (324.8 ton <sup>(1)</sup> )
$W_{ch}$ [kw] = $Q_{evap}$ [kw] / $COP_{actual}$	Power drawn by chiller
$W_{cd}$ [kw] = $0.02206 \cdot Q_{evap}$ [kw] $(1 + 1/COP_{actual})$	Power to condenser water pumps & cooling tower fans <sup>(2)</sup>
$W = W_{ch} + W_{cd}$ [kw]	Total electric power consumed by chiller operation
HRS [hr/yr]	Number of hours of chiller operation per year: <b>2,125</b> <sup>(3)</sup>
N [yrs]	Chiller life: <b>30</b> yrs <sup>(3)</sup>
$E$ [kwh] = $W \times HRS \times N$	Electricity consumed to operate the chiller over its lifetime
CI [kgCO <sub>2</sub> -eq/kwh]	Electricity Carbon Intensity: Low: <b>0.0150</b> <sup>(4)</sup> ; High <b>0.8445</b> <sup>(4)</sup>
$EM_{NRG}$ [kgCO <sub>2</sub> -eq] = $CI \times E$	Equivalent emissions from electricity use over chiller lifetime

<sup>(1)</sup>Test chiller capacity averaged over IPLV part-load levels with IPLV weighting factors; it does not affect relative TEWI values; <sup>(2)</sup>Sand F., Baxter V. D. (1997) Appendix F; <sup>(3)</sup>A.D. Little (2002); <sup>(4)</sup>World Resource Inst. (2006)

**Table 7:** Calculation of equivalent emissions from refrigerant leakage

$M_r$ [kg]	Refrigerant charge: $M_{r\_HFC-134a} = 787.44$ [kg]; $M_{r\_XP10} = 766.96$ [kg]
$S_{ANN}$ [%/yr]	Average Percentage of charge emitted annually: Low= <b>1%</b> ; High= <b>5%</b>
$EM_{RFG} = M_r \cdot (S_{ANN}/100) \cdot N \cdot GWP$	CO <sub>2</sub> emissions equivalent to refrigerant escaping during chiller oper. life
$S_{EOL}$ [%]	Percentage of charge emitted at end of chiller life: Low= <b>5%</b> ; High= <b>25%</b>
$EM_{RFG-EOL} = M_r \cdot (S_{EOL}/100) \cdot GWP$	CO <sub>2</sub> emiss. equival. to refrigerant escaping upon chiller decommissioning

**2.2.2 Refrigerant Emissions over Chiller Operating Life,  $EM_{RFG}$ , and End of Life,  $EM_{RFG-EOL}$ :** After equipment installation, refrigerant can escape into the atmosphere as a result of continuous slow leakage or permeation, equipment defects, equipment servicing, unexpected and occasionally catastrophic events and accidental or deliberate venting. The amount of CO<sub>2</sub> emissions equivalent to the aggregate amount of refrigerant emitted due to the above causes over the equipment operating life,  $EM_{RFG}$ , was calculated as summarized in [Table 7](#). The HFC-134a charge was prescribed a value of **1.404 Kg** (3.1 lb) per ton (applicable for any heat exchanger design according to LEED EAC4); the XP10 charge was specified as **97.4%** of the HFC-134a charge in accordance to actual observation with the test chiller. The average annual refrigerant emission rate as a percentage of charge,  $S_{ANN}$  [%/yr], was assigned representative values (A.D. Little (2002), Calm (2002) and Calm (2006)). The impact of refrigerant loss at the end of equipment life,  $EM_{RFG-EOL}$ , was accounted, as shown in [Table 7](#). The fraction of refrigerant charge emitted,  $S_{EOL}$ , depends on local practices, regulations and technician training; it was assigned representative values. When the candidate refrigerant has a lower GWP but leads to higher energy consumption than the incumbent refrigerant, it is informative to calculate the minimum rate of annual refrigerant emissions above which the candidate refrigerant would reduce TEWI relative to the incumbent refrigerant:

$$S_{ANN\_min} = 100 \cdot \frac{(EM_{NRG\_cand} - EM_{NRG\_incumb})}{(N + 5) \cdot (M_{r\_incumb} \cdot GWP_{incumb} - M_{r\_cand} \cdot GWP_{cand})} \quad (2)$$

It was assumed for convenience in deriving eq. (2) that refrigerant losses at the end of chiller life are equal to annual losses over five years (i.e.  $S_{EOL} \sim 5 S_{ANN}$ ).

### 3. RESULTS

Figure 2 shows the eleven operating points (i.e. load levels) with each fluid at which chiller performance was measured on the performance map of the test chiller compressor. It also shows contours of compressor efficiency levels,  $\eta_i$ . Compressor performance was quantified in terms of a dimensionless head and a dimensionless refrigerant volumetric flow rate at the compressor inlet delivered by the compressor at a given rotational speed and PRV angle (ASHRAE Handbook (2008)). Figure 2 is depicted in coordinates of dimensionless head and dimensionless flow rate relative to their values at the operating point with peak compressor efficiency. Use of dimensionless parameters allows the depiction of operating points with different fluids on the same compressor performance map. Figure 2 shows that the chiller tests were conducted at operating conditions under which the compressor operated close to its peak efficiency. It also shows that XP10 enables chiller operating conditions largely similar to HFC-134a over a wide range of capacities.

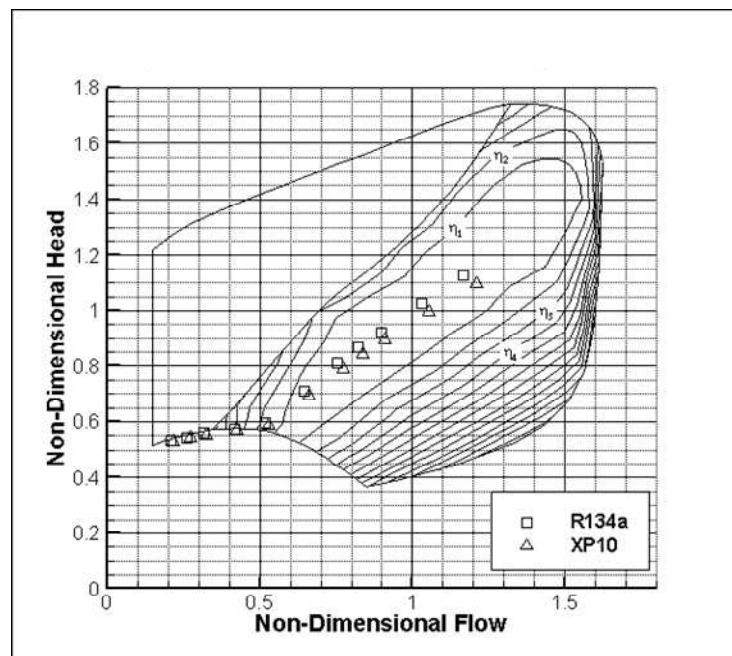


Figure 2: Test compressor performance map and operating points

#### 3.1 Chiller Performance Measurements

The energy and mass balance discrepancies, shown in Figure 3, were well within the tolerances specified in AHRI Standard 550/590 (2003). The largest discrepancies (up to ca. 4% to 5%) were observed at the lower cooling loads. The accuracy of the mass balances suggests that XP10 enthalpy values were of accuracy comparable to HFC-134a.

Figure 4 shows the ratio of chiller COP with XP10 to that with HFC-134a at the same capacity. Chiller COP with XP10 at design chiller capacity is 2.4% lower than HFC-134a, in good agreement with the theoretical prediction. COP with XP10 increases relative to HFC-134a at lower capacities. At capacities 30% of full load or lower, COP with XP10 was higher than HFC-134a. At the lowest capacity of 20% of full load, the COP of XP10 was 7.6% higher than HFC-134a. Average chiller energy consumption, in kw/ton, is often estimated as an Integrated Part Load Value (IPLV) over a representative profile of conditions defined in AHRI Standard 550/590 (2003). The test chiller IPLV energy consumption in kw/ton, calculated using the measured COP data in Figure 4, was 0.6% higher with XP10 than with HFC-134a (with an uncertainty of  $\pm 0.8\%$ ).

The evaporator and condenser overall heat transfer coefficients,  $U_{o,e}$ , and  $U_{o,c}$ , with XP10 relative to HFC-134a are shown in Figure 5. The evaporator overall heat transfer coefficients with XP10 were 5-10% higher than with HFC-

134a at the higher capacities tested and comparable with HFC-134a at the lower capacities tested. The condenser overall heat transfer coefficients with XP10 were 10-20% lower than with HFC-134a at the capacity levels tested.

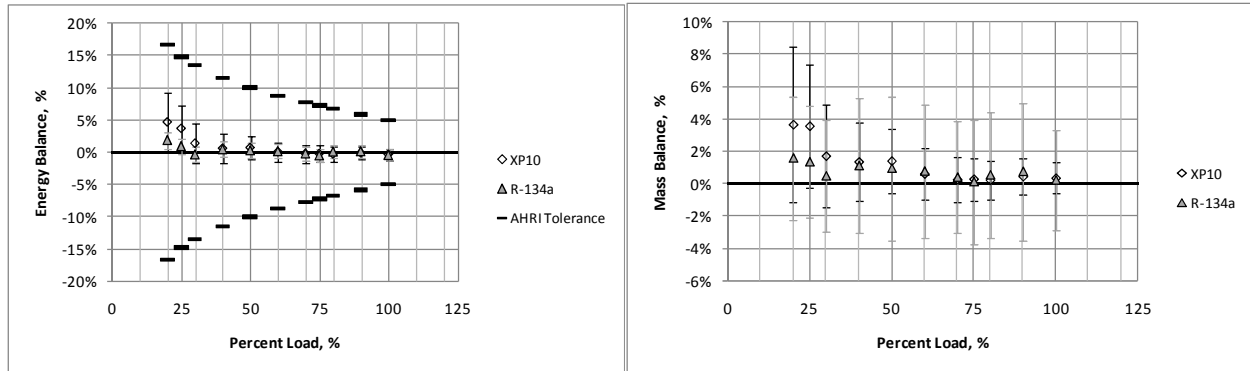


Figure 3: Magnitude of discrepancies in test energy and mass balances

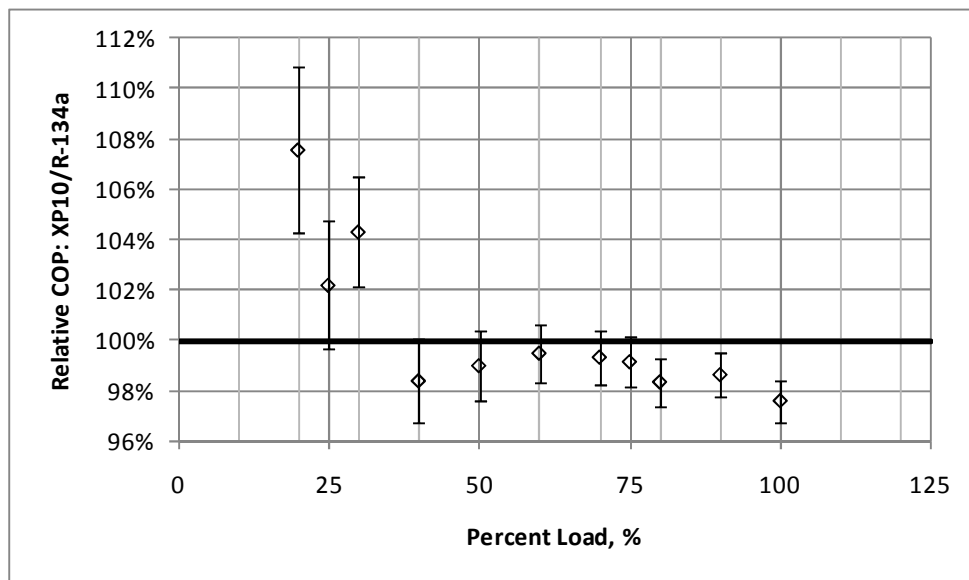


Figure 4: Chiller COP with XP10 relative to HFC-134a

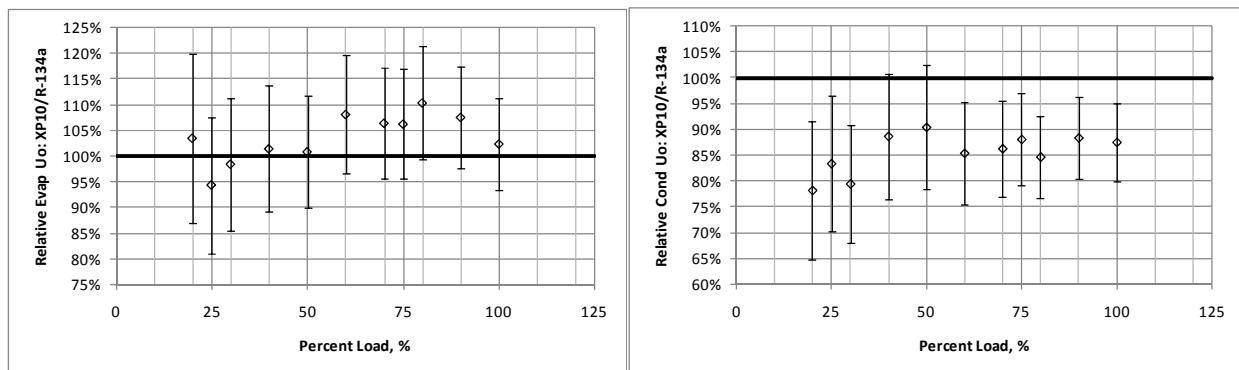


Figure 5: Evaporator and condenser overall heat transfer coefficient with XP10 relative to HFC-134a

### 3.2 TEWI Evaluations

TEWIs of chillers operated with XP10 or HFC-134a were calculated under four scenarios specified according to the levels of electricity carbon intensity ( $CI=0.0150$  or  $0.8445 \text{ kgCO}_2\text{-eq/kwh}$ ) and of refrigerant emission rates ( $S_{ANN}=1.0\%/yr$  and  $S_{EOL}=5.0\%$  or  $S_{ANN}=5\%/yr$  and  $S_{EOL}=25\%$ ). The experimentally determined IPLVs were used to



estimate energy consumption. The results, normalized with the highest TEWI value (for high CI, high refrigerant emission rates with HFC-134a), are shown in Figure 6 (notice different scales for high and low CI cases). In regions where the electricity carbon intensity is high, XP10 would enable reductions in chiller TEWI relative to HFC-134a when the refrigerant emission rates exceed the following values:  $S_{ANN\_min} = 0.402$  %/yr and  $S_{EOL\_min} = 2.010$ %. XP10 could enable a **5.3%** TEWI reduction versus HFC-134a under the scenario of high refrigerant emissions or a modest **0.8%** TEWI reduction under the scenario of low refrigerant emissions. In regions where the electricity carbon intensity is low, XP10 would enable reductions in chiller TEWI relative to HFC-134a when the refrigerant emission rates exceed the following values:  $S_{ANN\_min} = 0.007$  %/yr and  $S_{EOL\_min} = 0.036$ %. Under the high and low chiller refrigerant emission scenarios, XP10 could enable drastic TEWI reductions versus HFC-134a: **49%** and more than **31%**, respectively.

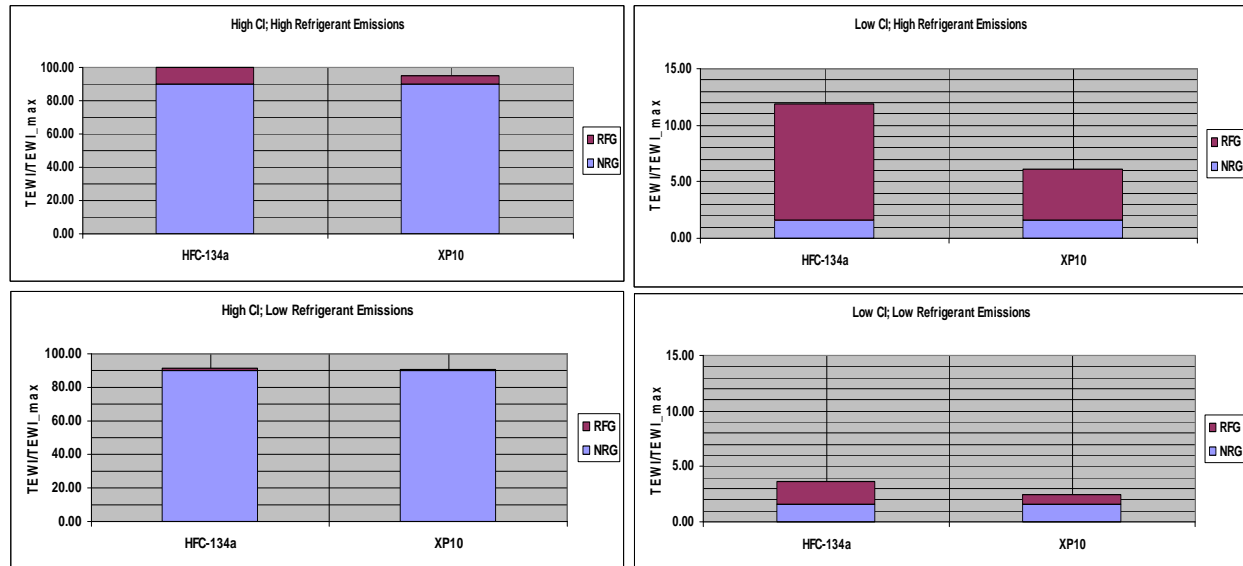


Figure 6: XP10 and HFC-134a Chiller TEWIs under four scenarios

#### 4. DISCUSSION-CONCLUSION

A key result of this work was that the measured energy efficiencies over a range of cooling loads with XP10 in a chiller designed for HFC-134a were comparable to those with HFC-134a. The required XP10 charge was 2.6% lower than the HFC-134a charge. Based on the measured chiller performance, XP10 could be considered a near drop-in replacement for HFC-134a in centrifugal chillers. Given its non-flammability and performance proximity to HFC-134a, XP10 could replace HFC-134a in existing chillers or enable optimized new chiller designs without extensive equipment modifications. Surprisingly, despite the use of refrigerant side enhanced heat transfer surfaces optimized for HFC-134a, the evaporator overall heat transfer coefficients at the higher capacity levels were higher with XP10. The condenser overall heat transfer coefficients with XP10 were lower than with HFC-134a.

Reducing the electricity carbon intensity remains the most effective means for reducing chiller warming impacts even when low GWP refrigerants are considered. Reducing the carbon intensity from levels representative of China (0.8445 kgCO<sub>2</sub>-eq/kwh) to those of Switzerland (0.0150 kgCO<sub>2</sub>-eq/kwh) would reduce HFC-134a chiller warming impact by about 88-96% over the range of refrigerant emission rates considered in this paper. In regions with high electricity carbon intensity, XP10 could effect chiller TEWI reductions in the range of 0.8-5.3% depending on refrigerant emission rates. In regions with low electricity carbon intensity, now and more prevalently in the future, XP10 would enable chiller TEWI reductions in the range of 31-48% depending on refrigerant emission rates.

#### NOMENCLATURE

$A_c$ : Heat transfer area in the condenser (m<sup>2</sup>)  
 $A_e$ : Heat transfer area in the evaporator (m<sup>2</sup>)

$COP_{theo}$ :	Coefficient of Performance (ratio of the rate of heat withdrawal at the evaporator (i.e. useful cooling delivered) and the power consumed by the compressor)
$c_{p,chw}$ :	Specific heat capacity of chilled water through the evaporator (kJ/kg)
$c_{p,cw}$ :	Specific heat capacity of chilled water through the condenser (kJ/kg)
EB:	Imbalance between measured energy transferred into and out of the chiller
$EM_{NRG}$ :	CO <sub>2</sub> and other GHG emissions from the use of energy to operate the chiller (e.g. compressors, condenser water pumps, cooling tower fans, etc.) throughout its useful life
$EM_{RFG}$ :	Refrigerant continuous, regular or intermittent emissions throughout the chiller operating life from installation completion to just before chiller retirement
$EM_{RFG-EOL}$ :	Refrigerant emissions at the end of chiller life
$h(P,T)$	refrigerant specific enthalpy as a function of pressure and temperature
$h_{r,c,in}$ :	Specific enthalpy of refrigerant entering the condenser (kJ/kg.K)
$h_{r,c,out}$ :	Specific enthalpy of refrigerant exiting the condenser (kJ/kg.K)
$h_{r,e,out}$ :	Specific enthalpy of refrigerant exiting the evaporator (kJ/kg.K)
$LMTD_c$ :	Log mean temperature difference in the condenser (K)
$LMTD_e$ :	Log mean temperature difference in the evaporator (K)
MB:	Imbalance between measured refrigerant mass flow rates in the evaporator and condenser
$\dot{m}_{chw}$ :	Mass flow rate of chilled water through the evaporator (kg/s)
$\dot{m}_{cw}$ :	Mass flow rate of cooling water through the condenser (kg/s)
$\dot{m}_{ocw}$ :	Mass flow rate of cooling water through the oil cooler (kg/s)
$\dot{m}_{r,av}$ :	Average of the refrigerant mass flow rates measured in the evaporator and condenser
$\dot{m}_{r,c}$ :	Refrigerant mass flow rate through the condenser (kg/s)
$\dot{m}_{r,e}$ :	Refrigerant mass flow rate through the evaporator (kg/s)
$P_c$ :	Pressure of refrigerant in the condenser (kPa)
$P_d$ :	Pressure of refrigerant at compressor discharge (kPa)
$P_e$ :	Pressure of refrigerant in the evaporator (kPa)
$P_i$ :	Pressure of refrigerant at compressor outlet (kPa)
$P_s$ :	Pressure of refrigerant at compressor suction (kPa)
$Q_c$ :	Condenser heat transfer rate (kW)
$Q_e$ :	Evaporator heat transfer rate (kW)
$Q_{in}$ :	Total energy transfer rate into the chiller (kW)
$Q_{oc}$ :	Compressor oil cooler heat transfer rate (kW)
$Q_{out}$ :	Total energy transfer rate out of the chiller (kW)
$T_{chw,in}$ :	Chiller water temperature at evaporator inlet (K)
$T_{chw,out}$ :	Chiller water temperature at evaporator outlet (K)
$T_{cw,in}$ :	Cooling water temperature at condenser inlet (K)
$T_{cw,out}$ :	Cooling water temperature at condenser outlet (K)
$T_d$ :	Temperature of refrigerant at compressor discharge (K)
$T_i$ :	Temperature of refrigerant at condenser outlet (K)
$T_{oc,in}$ :	Cooling water temperature at oil cooler inlet (K)
$T_{oc,out}$ :	Cooling water temperature at oil cooler outlet (K)
$T_s$ :	Temperature of refrigerant at compressor suction (K)
$T_{sat,c}$ :	Saturation temperature of refrigerant in the condenser (K)

$T_{\text{sat,e}}$ :	Saturation temperature of refrigerant in the evaporator (K)
$U_{\text{o,c}}$ :	Overall heat transfer coefficient in the condenser (kW/m <sup>2</sup> K)
$U_{\text{o,e}}$ :	Overall heat transfer coefficient in the evaporator (kW/m <sup>2</sup> K)
$VCC_{\text{theo}}$ :	Volumetric Cooling Capacity (kJ/m <sup>3</sup> )
$W_c$ :	Electrical power input to compressor motor (kW)
$\eta_{1,2,3,4...}$ :	Compressor efficiency levels

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