#506: Experimental Investigation Into The Effect Of Charge Optimization With Different Heat Exchanger Configuration On The Seasonal Performance In An R410A Heat Pump System

Sugun Tej INAMPUDI[1], Stefan ELBEL[1,2,3]

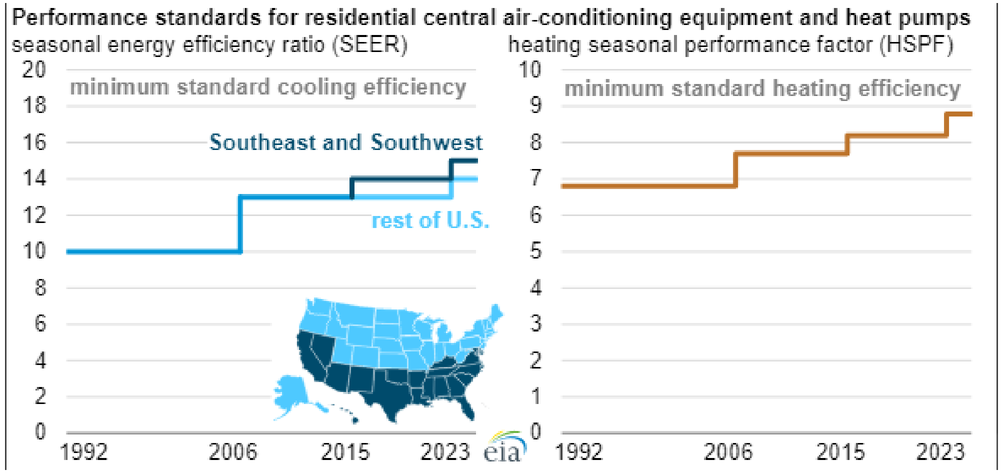
[1]University of Illinois at Urbana-Champaign, MechSE, Air Conditioning and Refrigeration Center

[2]Creative Thermal Solutions, Inc., Urbana, Illinois, USA

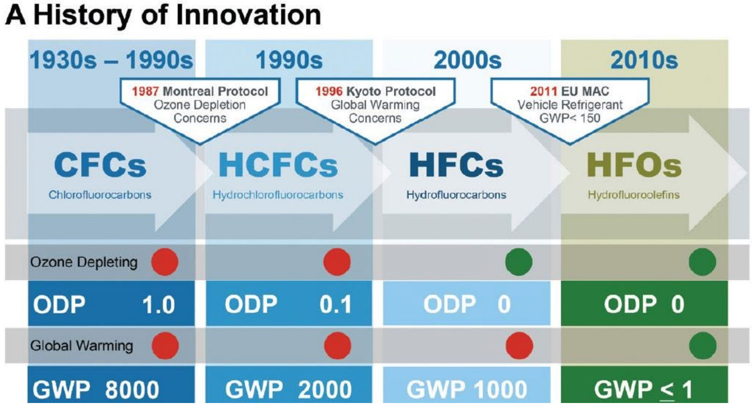
[3]Technische Universität Berlin , Institut für Energietechnik, FG Wärmeübertragung und Wandlung, Germany

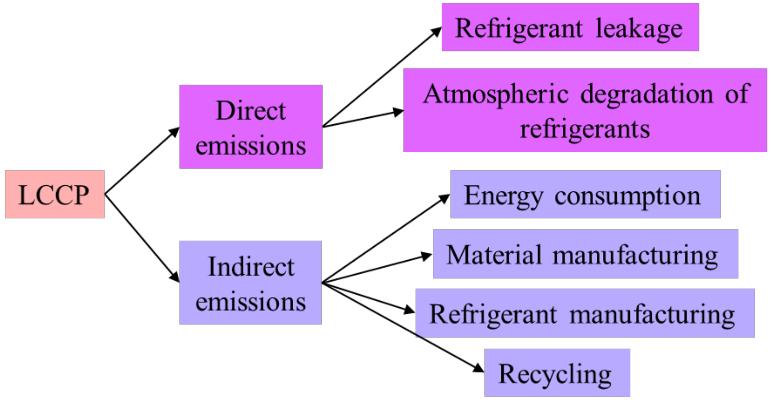
Background

- Global energy efficiency standards are becoming more stringent
- Beginning in 2023, all new residential HVAC systems sold in US will be required to meet new energy efficiency standard
- More focus on seasonal performance and part load conditions
- Increased focus on using low GWP refrigerants and improving system efficiency to reduce LCCP (Life Cycle Climate Performance)
  - Direct and indirect emissions



Source: U.S. Energy Information Administration

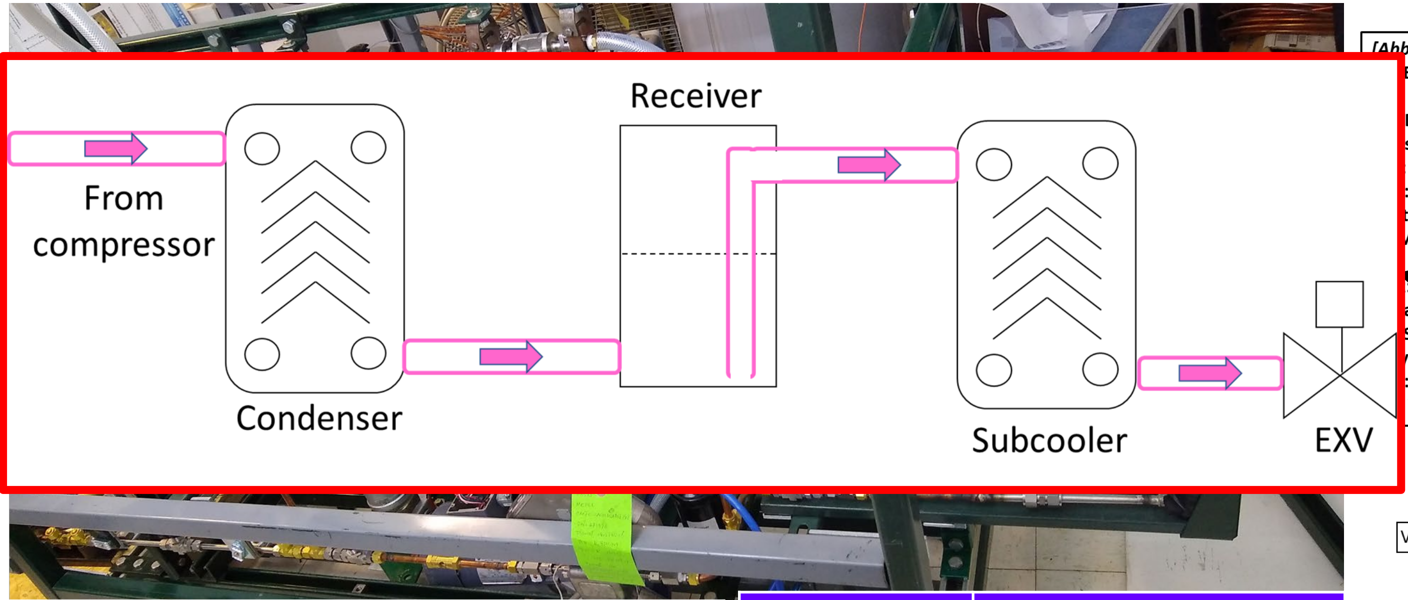


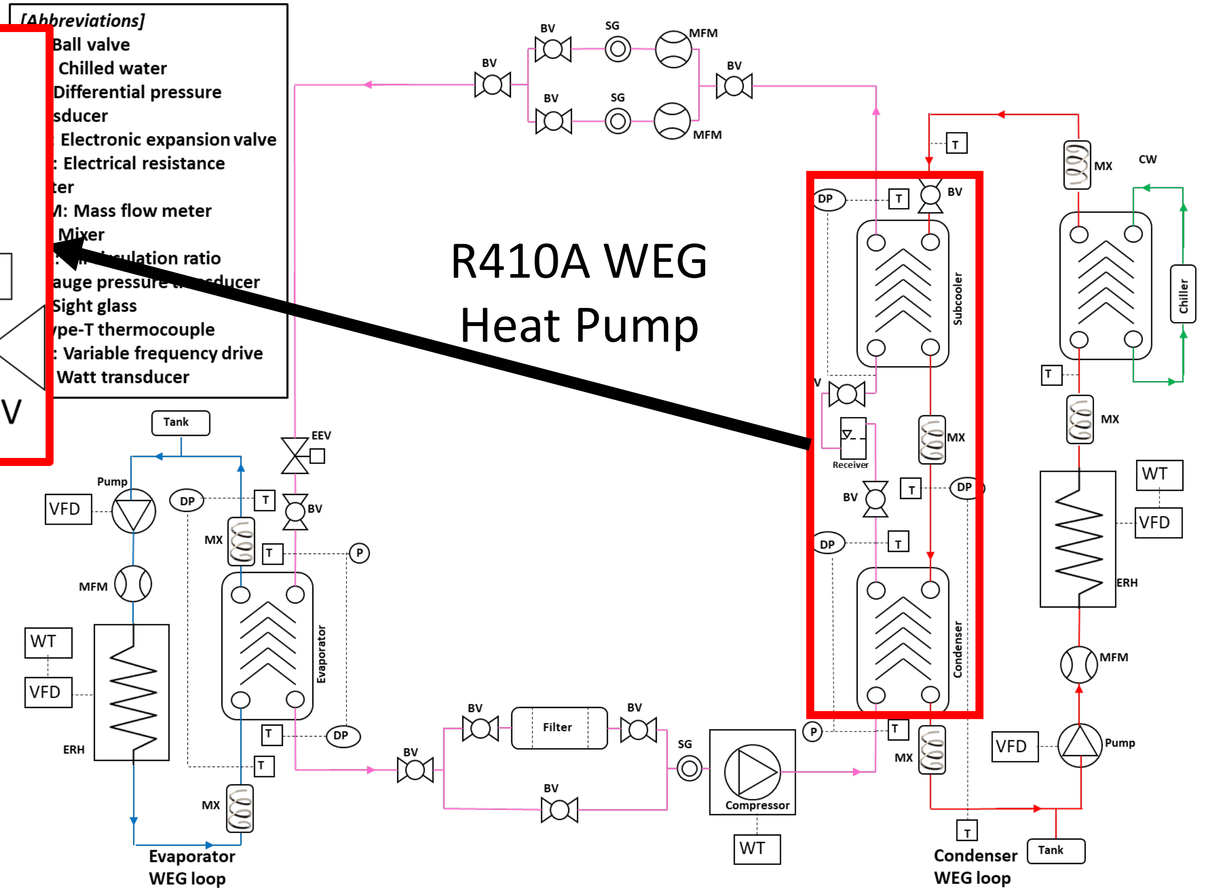


Motivation and objectives

- Seasonal performance of variable speed compressor under heating operation
  - Effect of different test conditions in the standard
  - Limitation due to compressor operating envelope
- Significance of the charge optimization in a dedicated subcooler system
  - A peak in COP was observed when receiver was empty
  - Why does this peak occur?
- Effect of charge optimization on the seasonal performance
  - Variable speed compressor operated at two different charges

Experimental facility





Parameter	Uncertainty
Temperature (°C)	±0.2
Pressure (kPa)	±0.2%
Mass flow rate (g/s)	±0.2%
Power (kW)	±0.5%
Capacity (kW)	±3.0%
COP (-)	±3.1%

Component	Description
Compressor	Scroll compressor
Refrigerant	R410A
Heat exchanger	Brazed plate heat exchanger
Secondary fluid	20% Water Ethylene Glycol (WEG)
Expansion valve	Electronic expansion valve

EU 14825 – Standard to be used

- Used for determination of the seasonal performance of a water heat pump
- USA standard AHRI 551/591 available for chiller application, but not for water heat pump
- Defines the seasonal performance using SCOP, seasonal coefficient of performance

Note

- 5°C is used instead of 3°C in the outdoor BPHX
  - Experimental facility limitations
- All test points: Outdoor HX inlet temperature remains constant
- E&F→A →B→C→D: Part load ratio drops, but
  - Indoor outlet temperature remains constant OR
  - Indoor outlet temperature ↓

Test point	Part load ratio (%)	Outdoor (Evaporator) BPHX temperature [°C]	Indoor (Condenser) fixed outlet temperature [°C]	Indoor (Condenser) variable outlet temperature [°C]
E&F	100	10/7	30/35	30/35
A	88	10/*	*/35	*/34
B	54	10/*	*/35	*/30
C	35	10/*	*/35	*/27
D	15	10/*	*/35	*/24

* The water flow rate must be held the same as A condition

EU 14825 – Cycling losses

- If the provided capacity is greater than the required load, the compressor must cycle to match the load
- Cycling losses are estimated using capacity ratio (CR) and degradation factor (C_d)
- The system is operated at steady state, the measured COP_{dec} is degraded to COP_{PL}

$$CR = \frac{Required\ heating\ load}{Provided\ heating\ capacity}$$

$$C_d = 0.9$$

Experimental result

$$COP_{dec} = \frac{\dot{Q}_{co+sc,avg}}{\dot{W}_{cp}}$$

→

Calculated result

$$COP_{PL} = COP_{dec}\frac{CR}{C_d \cdot CR + (1 - C_d)}$$

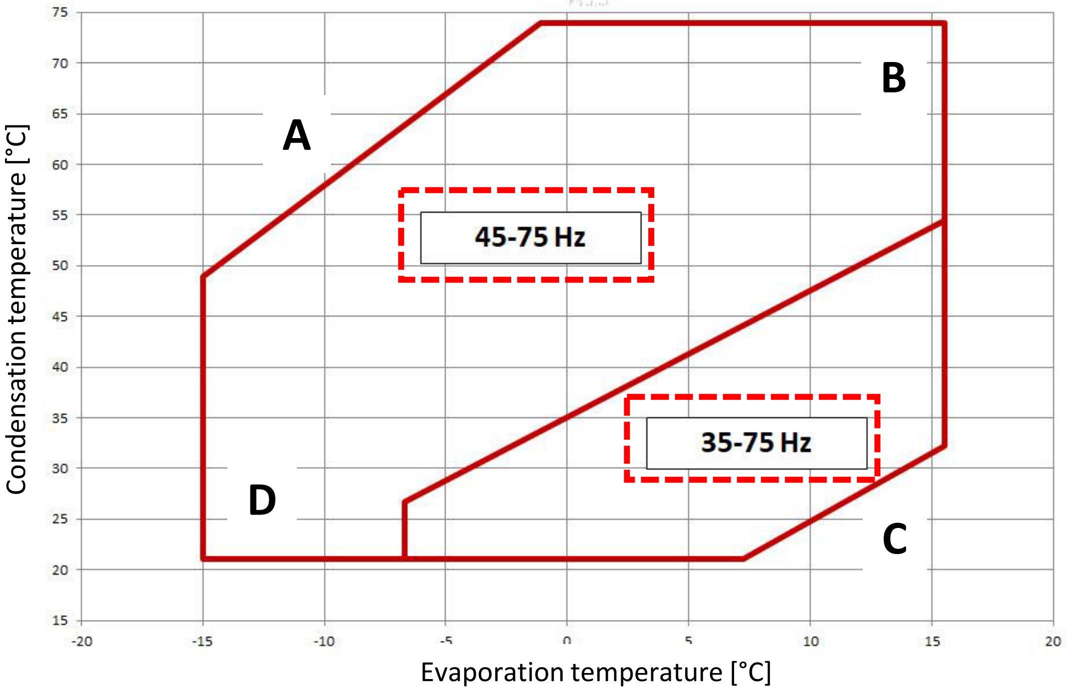
$$\dot{Q}_{co+sc} = \dot{Q}_{condenser} + \dot{Q}_{subcooler}$$

$$\dot{Q}_{co+sc,weg} = \dot{m}_{weg} c_p \Delta T$$

$$\dot{Q}_{co+sc,ref} = \dot{m}_{ref} \Delta h$$

$$\dot{Q}_{co+sc,avg} = \frac{\dot{Q}_{co+sc,weg} + \dot{Q}_{co+sc,ref}}{2}$$

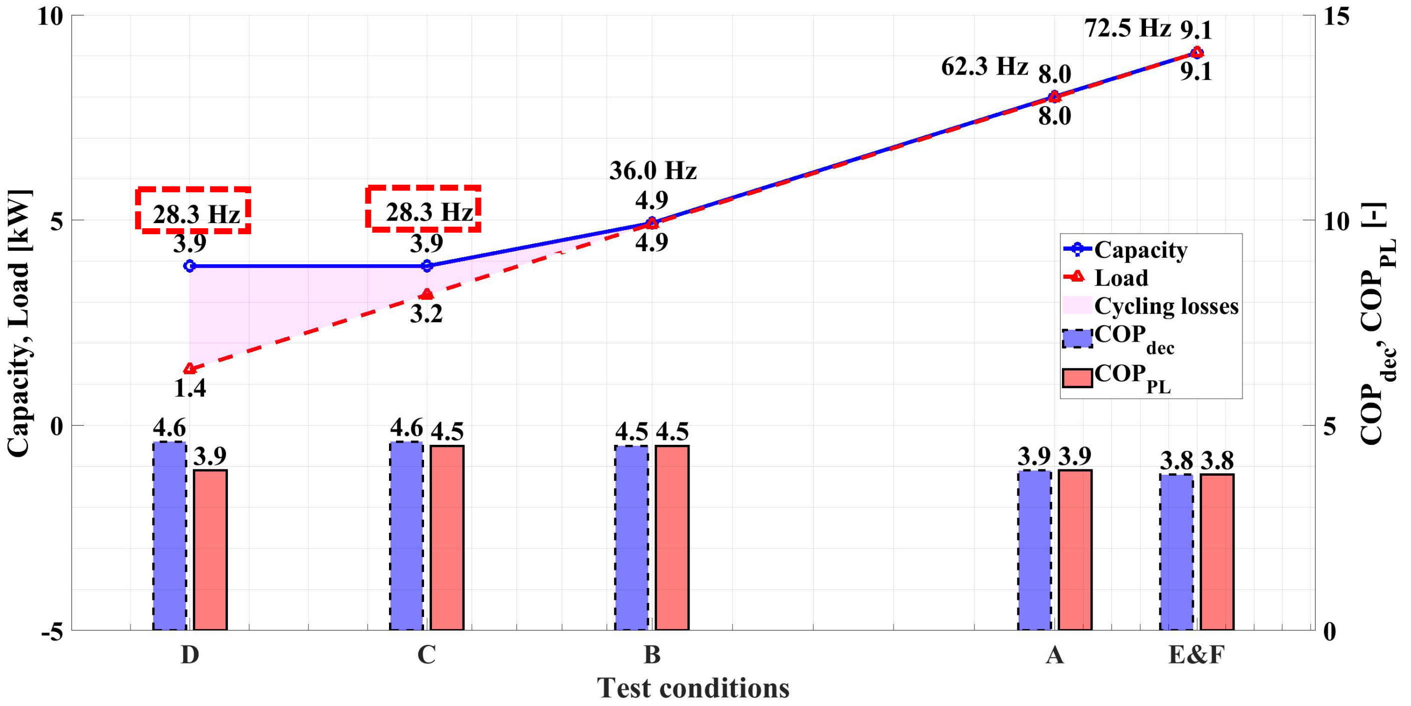
Typical compressor operating envelope



- The compressor envelope is more restrictive at lower speeds to prevent overloading and overheating of the compressor motor
  - Reduced motor cooling due to lower refrigerant flow rate
  - High superheat at compressor suction can cause high discharge temp.

- Zone A: High $P_{cond.}$ and low $P_{evap.}$
  - Cause: Extreme ambient conditions
  - Risk: High compressor discharge temperature and potential compressor burn out
- Zone B: High $P_{cond.}$ and high $P_{evap.}$
  - Cause: High cooling demand during pull-down operation
  - Risk: High current consumption and increased load on compressor motor
- Zone C: Low $P_{cond.}$ and high $P_{evap.}$
  - Cause: High evaporating temperature leads to low pressure ratio
  - Risk: Min. pressure ratio required to keep the scrolls loaded. Noisy operation
- Zone D: Low $P_{cond.}$ and low $P_{evap.}$
  - Reduction in compressor efficiency
  - Low pressure difference causes EXV to malfunction

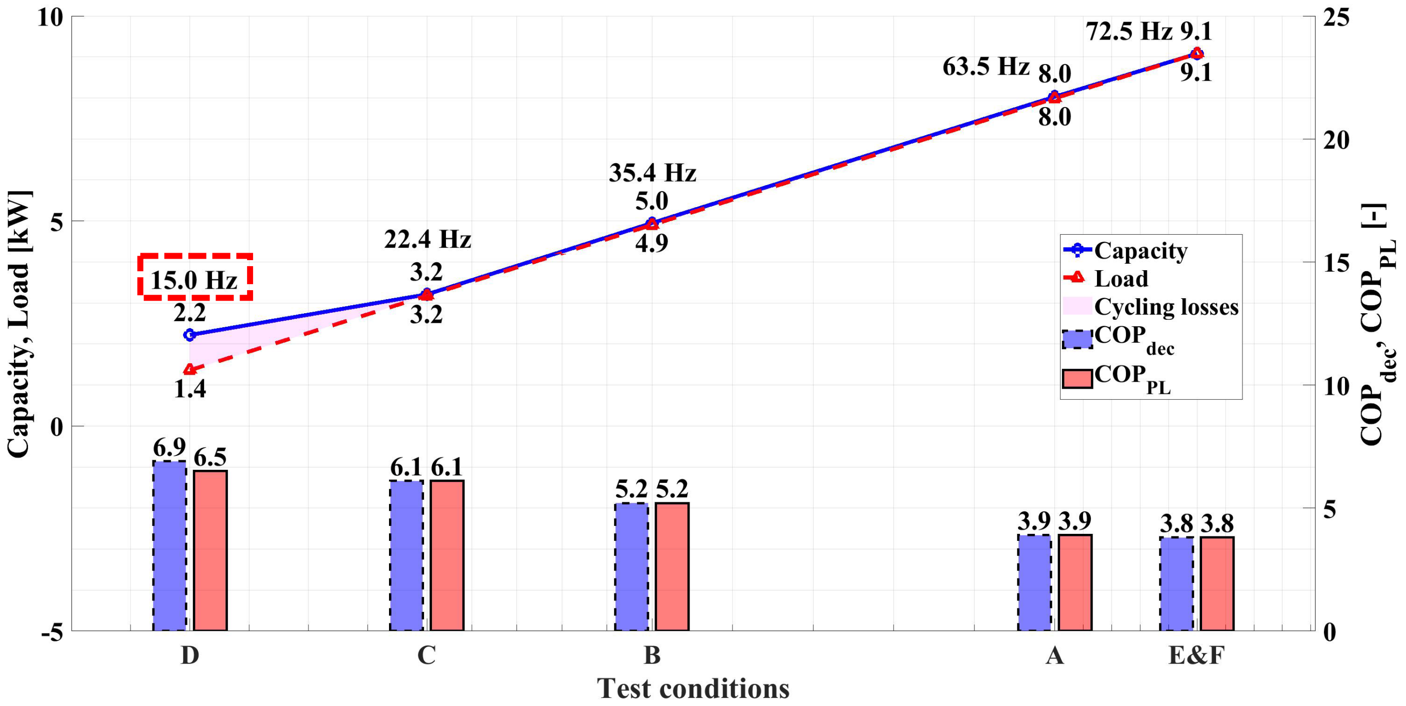
Variable speed compressor – **Fixed condenser outlet** condition



$$SCOP_{avg} = 4.33$$

- The system is charged with enough refrigerant so that receiver is half full
- The compressor can operate between 15 and 100 Hz
- Condenser WEG inlet temperature remains constant at 35°C
- Compressor matches the load at E, A, and B condition by reducing the speed
- But it undergoes cycling losses at C and D conditions
  - If the compressor operates at a frequency below 28.3 Hz, the required pressure ratio falls outside the reduced compressor operating envelope
  - Operation is limited to 28.3 Hz at low loads

Variable speed compressor – **Variable condenser outlet** condition



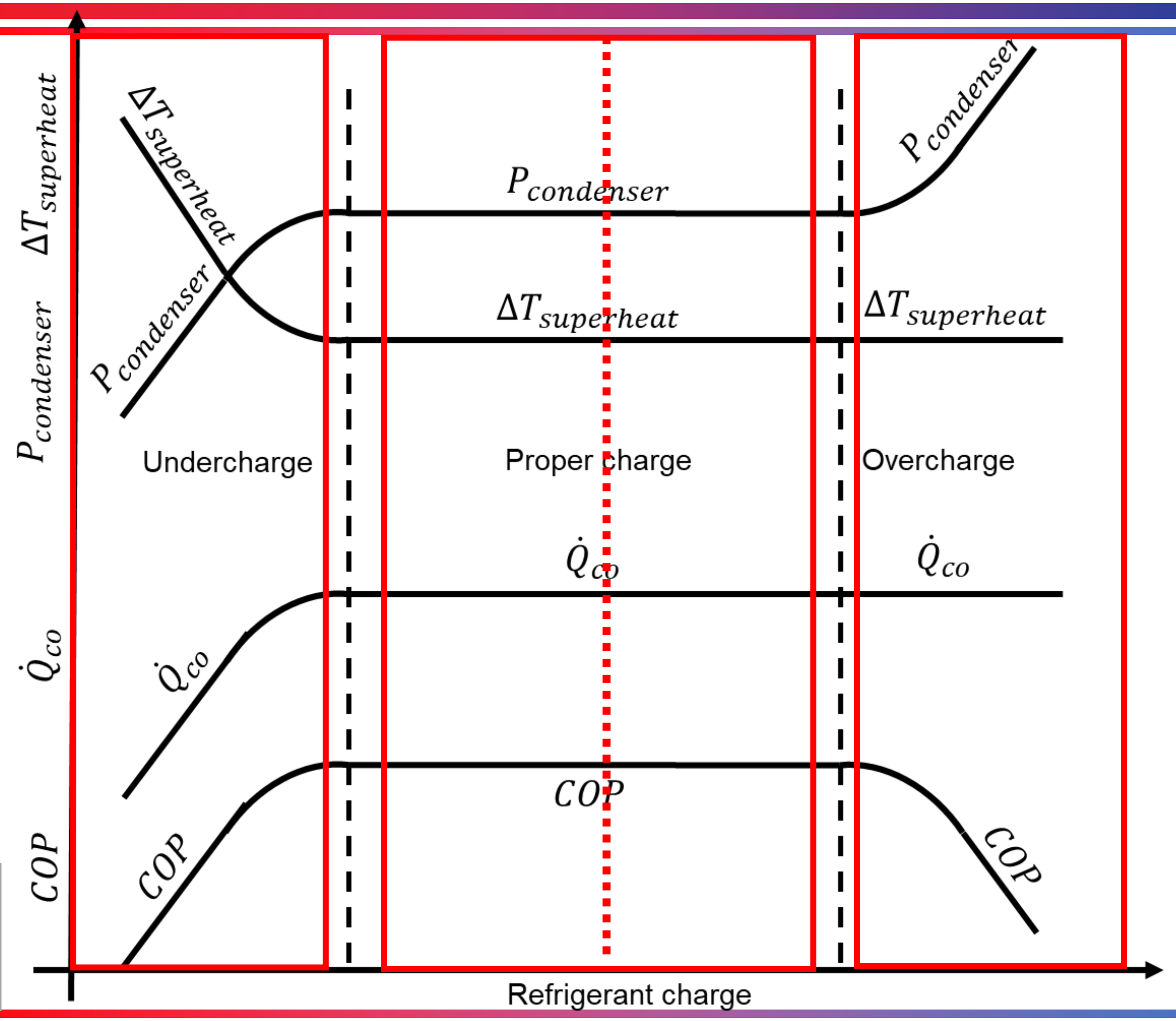
$SCOP_{avg} = 5.26$

- Same refrigerant charge as previous case
- Condenser WEG inlet temperature drops from 35°C at E&F condition to 24°C at D condition
- This drop in the condenser temperature causes the pressure ratio to be within the compressor operating envelope
  - Thus, compressor matches the load at all the conditions except D cond.
- The compressor can operate between 15 and 100 Hz
  - Hence it undergoes cycling losses at D condition
- The higher $SCOP_{avg}$ is due to reduced cycling losses and lower condenser WEG temp. leads to an operation at lower pressure ratios

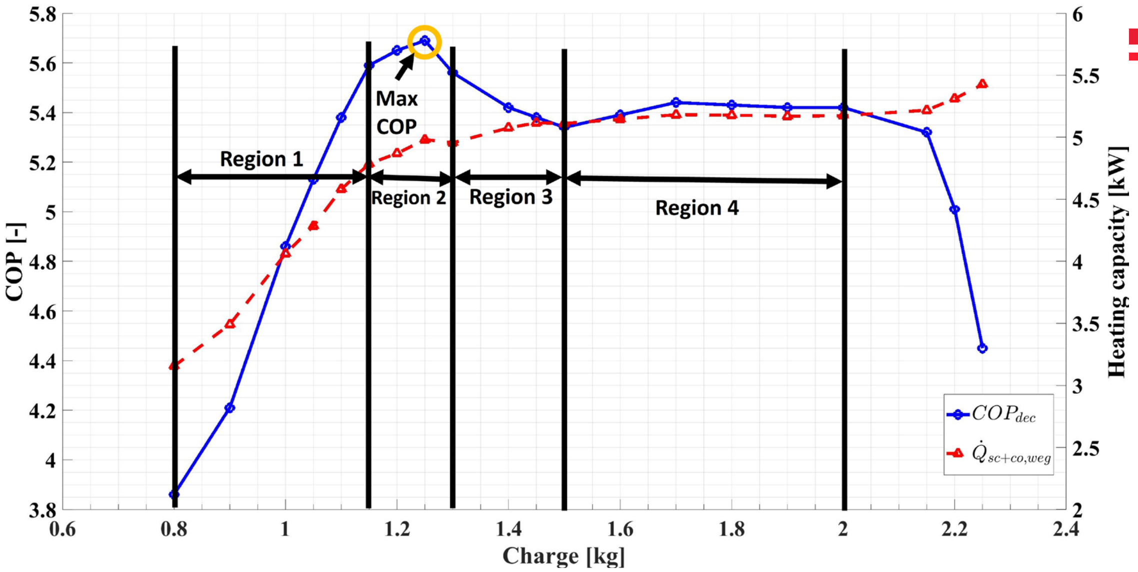
Typical charge test – TXV/EXV system

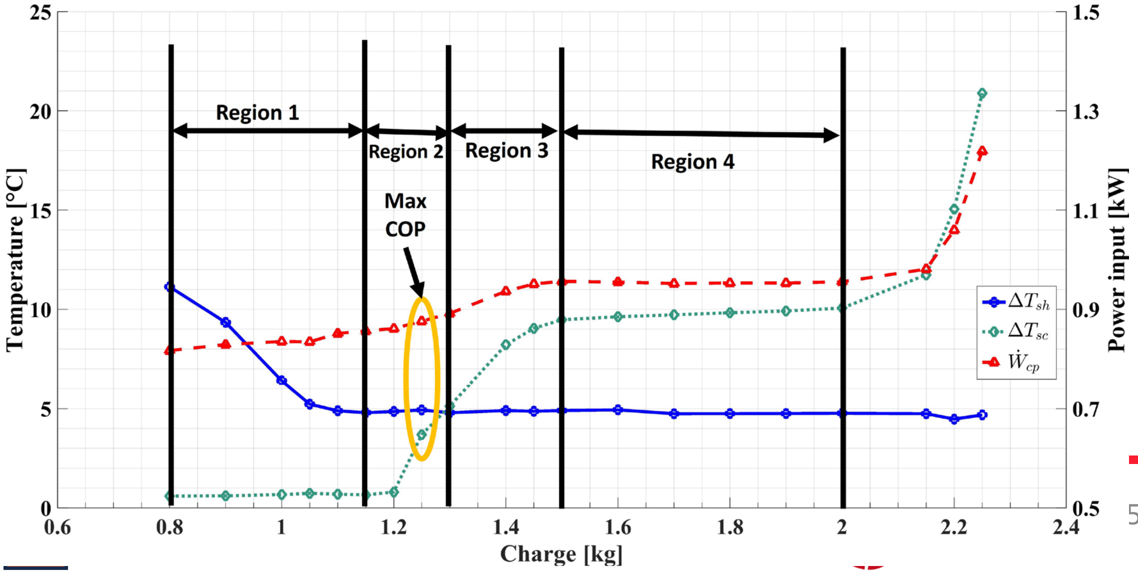
- Refrigerant is added to the system
- Secondary fluid conditions are not changed
- At steady state, $\dot{Q}_{co}$, W_{cp}, P and T at different points are measured
- Undercharge: Low mass, High SPHT, Low P_{high}
- Proper charge: Medium mass. The mass just increases in the receiver, does not do anything to the system performance
- Overcharge: High mass. Charge increases in the condenser

System is usually operated in properly charged region with the receiver half full



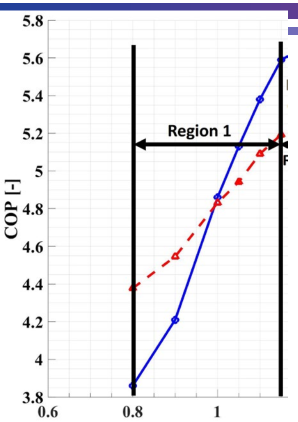
Variable speed compressor – Charge opt. B condition

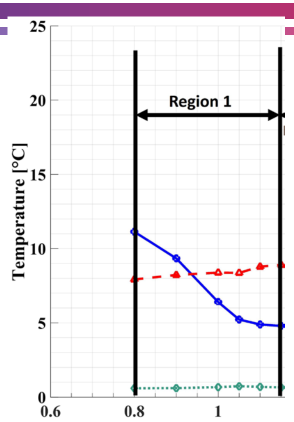




- B condition (54%) with variable condenser outlet
  - Compressor operates at 35.4 Hz
  - Condenser outlet temperature is 30°C
  - Looks like a typical charge opt. curve
  - But COP peak occurs when receiver is empty
- In the coming slides
  - Why does this peak in COP occur?
  - How does the SCOP change if the system is operated at the peak COP charge?
- To explain why this peak occurs, divide into 4 regions
  - Region 1
  - Max COP occurs in Region 2
  - Region 3
  - Region 4

Variable speed charge opt. B condition – **Region 1**





$h_{3'} > h_3$

↓

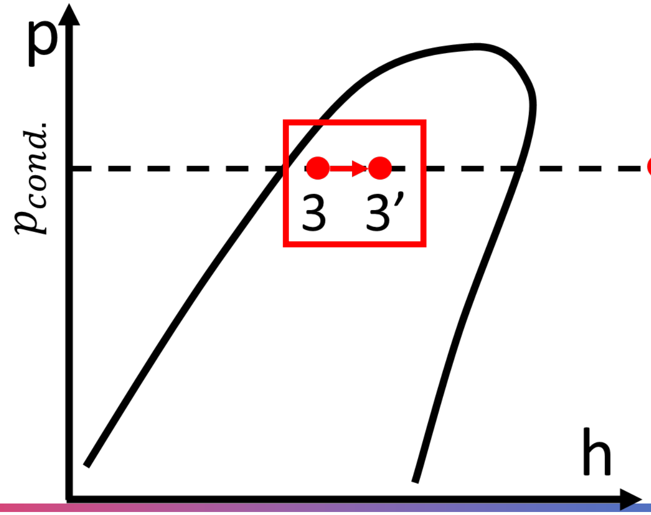
$\dot{q}_{co+sc} = (h_2 - h_{3'})$ ↓

- COP ↑ $\dot{Q}_{co+sc} = \dot{m}_{ref}(h_1 - h_{4'})$
- Added charge → Evaporator
  - Superheat↓ → UA_{ev}↑
  - Heating capacity↑ W_{cp}↑
  - $h_{sc,ro}$↑ = Specific heating capacity↓
  - $\dot{m}_{ref}$↑ = Heating capacity↑

Region 1

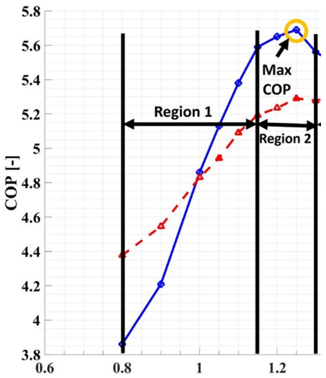
Charge [kg]	$COP_{dec,weg}$ [-]	$\frac{\Delta \dot{Q}_{co}}{\dot{Q}_{co}}$ [%]	$\frac{\Delta W_{cp}}{W_{cp}}$ [%]	$\Delta \dot{Q}_{co}/\dot{Q}_{co} - \Delta W_{cp}/W_{cp}$ [%]	ΔT_{sh} [°C]	UA_{ev} [kW/°C]	h_{scro} [kJ/kg]
0.80	3.86	-	-	-	11.1	0.34	251
0.90	4.21	+10.65	+1.49	+9.16	9.3	0.44	252
1.00	4.86	+16.35	+0.75	+15.59	6.4	0.72	253
1.05	5.13	+5.48	-0.09	+5.57	5.2	0.83	254
1.10	5.38	+6.94	+2.02	+4.92	4.9	0.72	255
1.15	5.59	+4.48	+0.60	+3.88	4.8	0.70	255

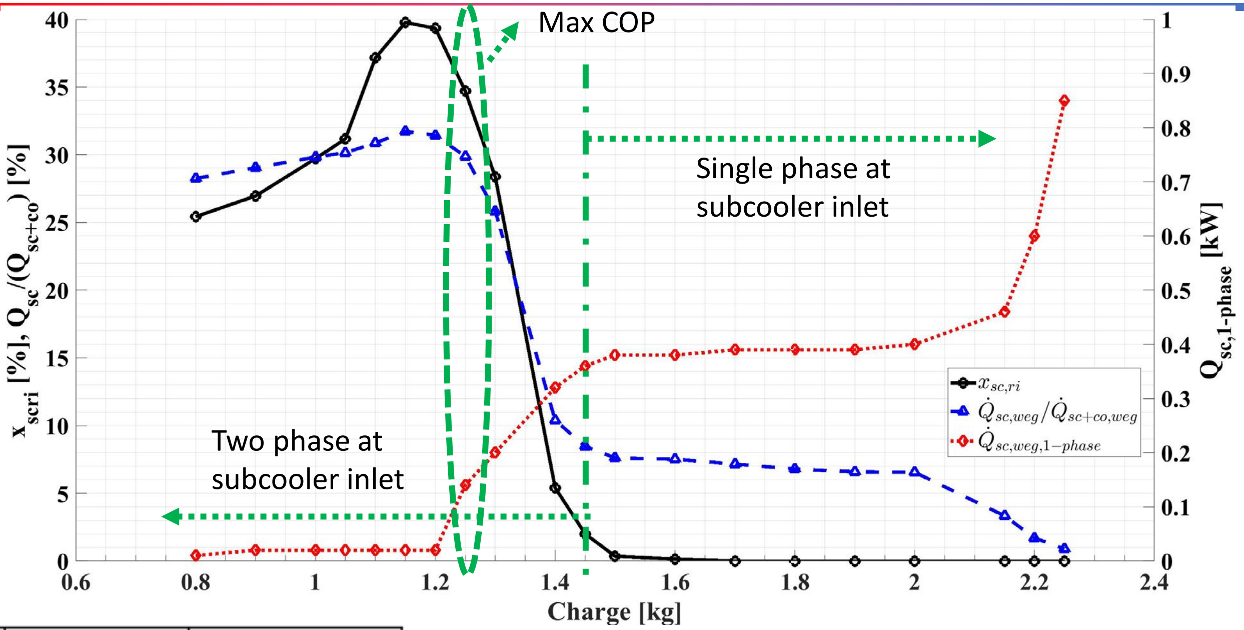
Increases | +ve increase | +ve increase | +ve increase | Reduces | Increases | +ve increase



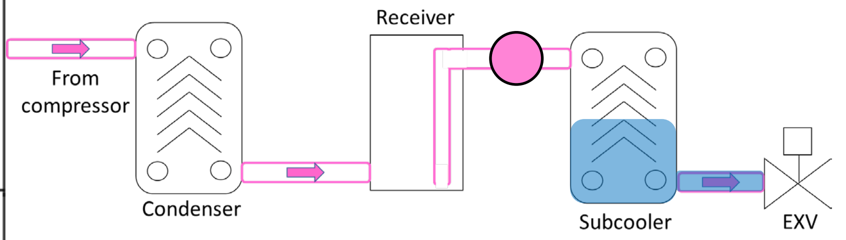
Variable speed charge opt. B condition – Max COP in Region 2

- Superheat, UA_{ev} constant
- $h_{sc,ro}$ ↓
  - Heating capacity↑
- W_{cp}↑ but $\left|\frac{\Delta Q_{co}}{Q_{co}}\right| > \left|\frac{\Delta W_{cp}}{W_{cp}}\right|$
- Quality at subcooler inlet, Q_{sc}
  - Subcooler inlet = Two phase; Part of two−phase heat transfer happens in the subcooler
  - $Q_{sc,1-phase}$, $Q_{sc,two-phase}$, $\frac{Q_{sc}}{Q_{sc}+Q_{co}}$



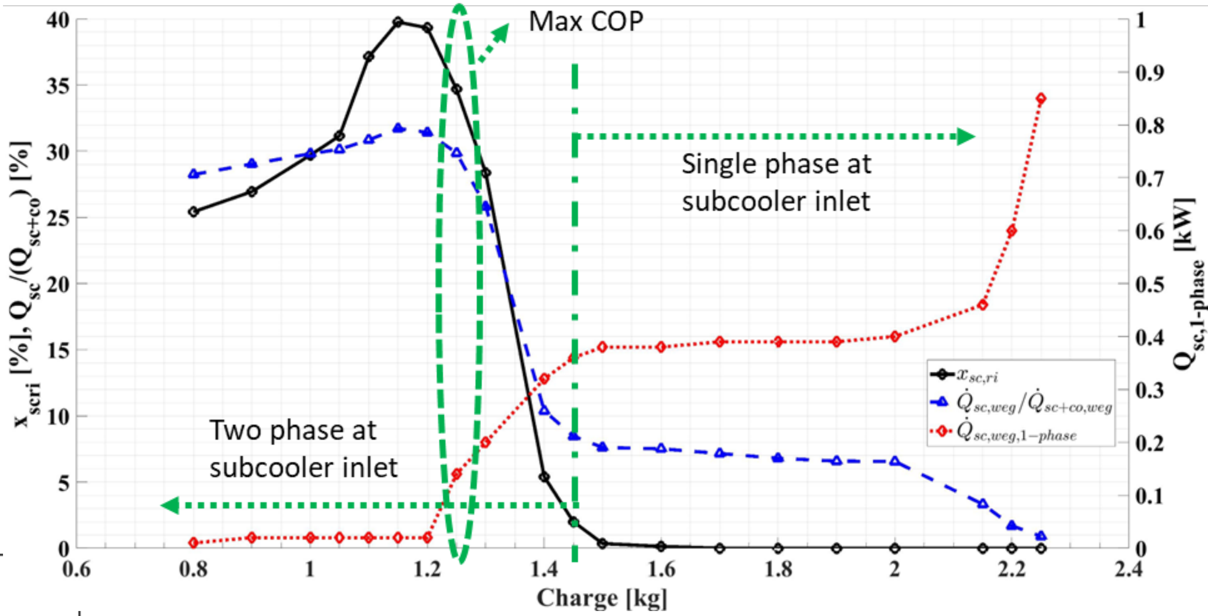


	Charge [kg]	$COP_{dec,weg}$ [-]	$\frac{\Delta \dot{Q}_{co}}{\dot{Q}_{co}}$ [%]	$\frac{\Delta W_{cp}}{W_{cp}}$ [%]	$\Delta \dot{Q}_{co}/\dot{Q}_{co} - \Delta W_{cp}/W_{cp}$ [%]	ΔT_{sh} [°C]	UA_{ev} [kW/°C]	h_{scro} [kJ/kg]
Max COP	1.20	5.65	+1.73	+0.57	+1.17	4.9	0.70	255
	1.25	5.69	+2.29	+1.71	+0.58	4.9	0.71	251



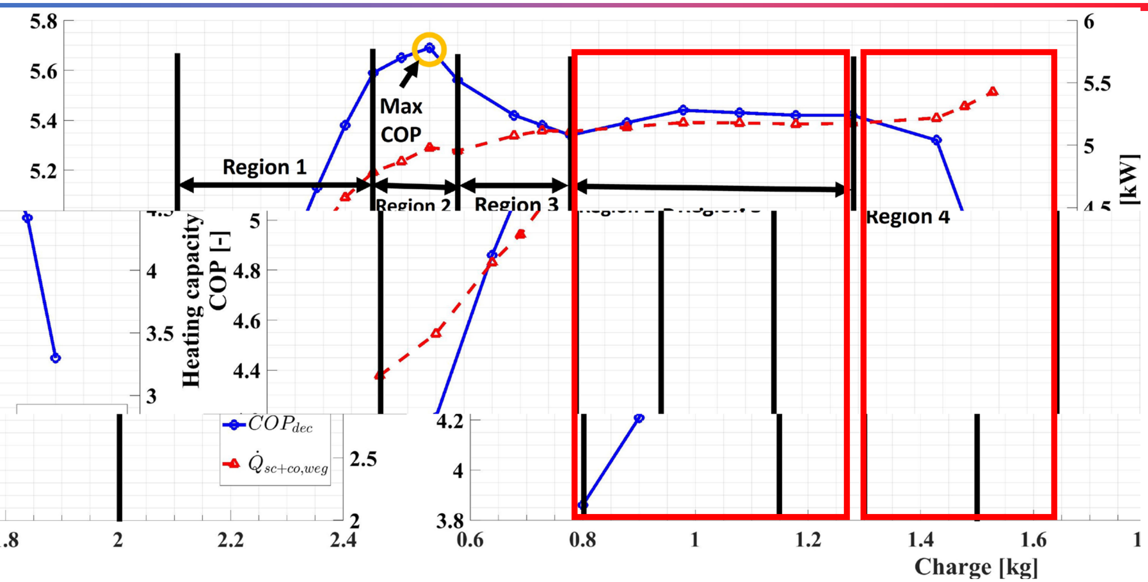
Variable speed charge opt. B condition – Region 3 (Beyond Max COP)

- $h_{sc,ro} \downarrow$
  - But $W_{cp} \uparrow$ and $\left|\frac{\Delta Q_{co}}{Q_{co}}\right| < \left|\frac{\Delta W_{cp}}{W_{cp}}\right|$
- Quality at Subcooler inlet $\downarrow$
  - $Q_{sc,two-phase} \downarrow, Q_{sc,1-phase} \uparrow \rightarrow \frac{Q_{sc}}{Q_{sc}+Q_{co}} \downarrow$
  - $Q_{cond.,two-phase} \uparrow$
  - Same $A_{cond.} \rightarrow LMTD_{co}$ & $P_{co} \uparrow \rightarrow$ COP$\downarrow$

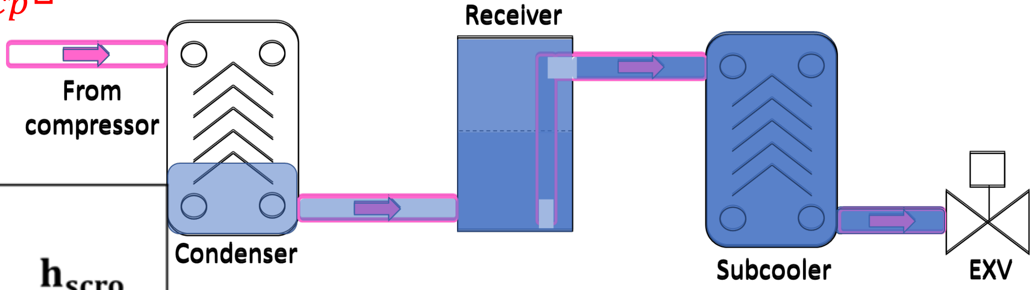


Region	Charge [kg]	$COP_{dec,weg}$ [-]	$\frac{\Delta\dot{Q}_{co}}{\dot{Q}_{co}}$ [%]	$\frac{\Delta W_{cp}}{W_{cp}}$ [%]	$\Delta\dot{Q}_{co}/\dot{Q}_{co} - \Delta W_{cp}/W_{cp}$ [%]	ΔT_{sh} [°C]	UA_{ev} [kW/°C]	h_{scro} [kJ/kg]
Region 3	1.30	5.56	-0.52	+1.79	-2.31	4.8	0.71	249
	1.40	5.42	+2.43	+5.01	-2.58	4.9	0.72	247
	1.45	5.38	+0.79	+1.58	-0.79	4.9	0.73	246
	1.50	5.34	-0.10	+0.54	-0.64	4.9	0.72	246

Variable speed charge opt. B condition – **Region 4 and beyond**



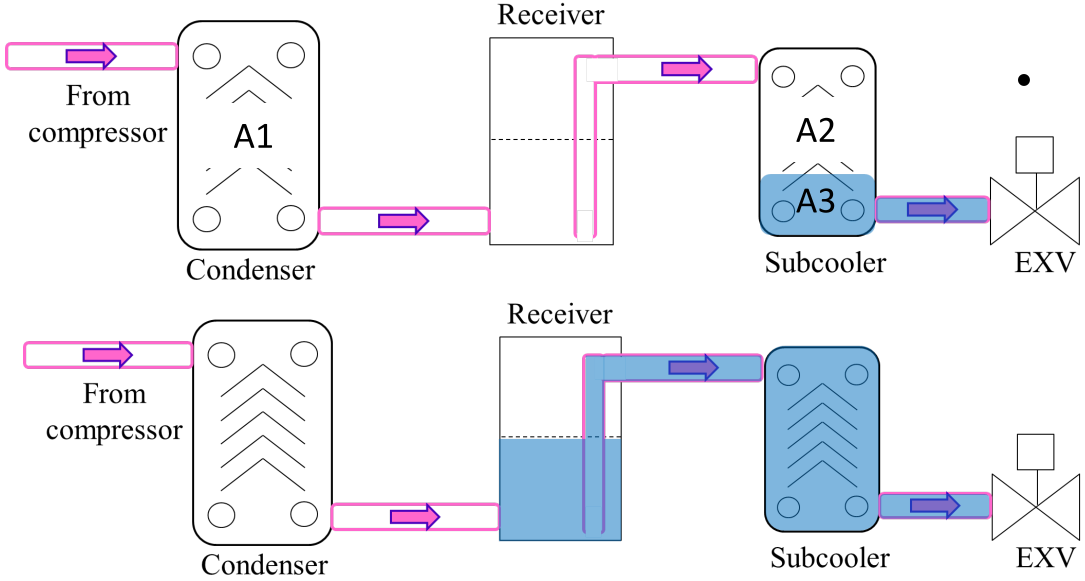
- Added charge → Receiver
  - No significant change in system performance
- Beyond Region 4
  - Receiver is completely full
  - Added charge → Condenser
  - P_{co} ↑ = W_{cp}
  - COP ↓



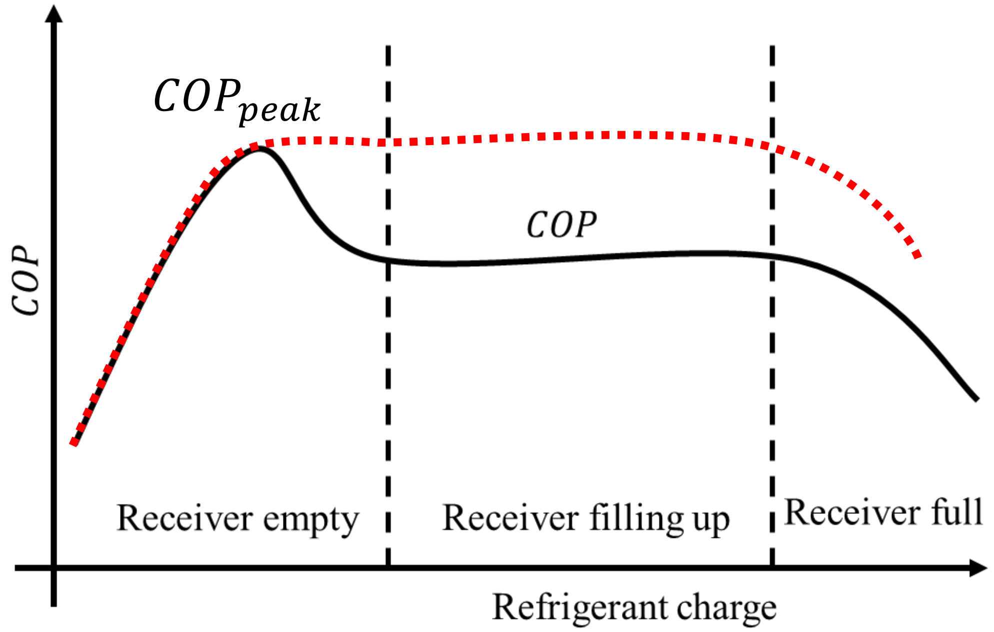
Region 4

Charge [kg]	$COP_{dec,weg}$ [-]	$\frac{\Delta\dot{Q}_{co}}{\dot{Q}_{co}}$ [%]	$\frac{\Delta W_{cp}}{W_{cp}}$ [%]	$\Delta\dot{Q}_{co}/\dot{Q}_{co} - \Delta W_{cp}/W_{cp}$ [%]	ΔT_{sh} [°C]	UA_{ev} [kW/°C]	h_{scro} [kJ/kg]
1.60	5.39	+0.66	-0.15	+0.81	4.9	0.72	246
1.70	5.44	+0.72	-0.30	+1.02	4.7	0.74	246
1.80	5.43	-0.07	0.12	-0.19	4.7	0.74	245
1.90	5.42	-0.17	-0.02	-0.15	4.8	0.74	245
2.00	5.42	+0.09	0.25	-0.16	4.8	0.74	245

Significance of the peak and effect of BPHX size



- Smaller subcooler
  - This will ensure the receiver will start filling up at lower charge
  - But reduced heat transfer area will decrease the COP



- Peak occurs when some part of two-phase heat transfer happens in the subcooler
- How can we achieve the ideal charge opt. curve?
  - Oversized condenser vs smaller subcooler

- Oversized condenser
  - Increases two-phase heat transfer area
  - But COP will ONLY plateau when subcooler is completely filled with subcooled liquid
  - The subcooling obtained when the subcooler is filled might not be the opt. subcooling or cause peak COP
  - Thus, the peak COP will occur before subcooler is filled i.e., empty receiver

- Oversized Cond. + Smaller Subcooler
  - Remove two-phase A2 area from Subcooler and add it to Cond. area (A1)
  - Only optimum subcooling (A3) happens in the dedicated subcooler

Seasonal performance for different cases

Case #	Charge [kg]	Fixed/ Variable Condenser WEG outlet	$SCOP_{avg}$ [-]	Difference in $SCOP_{avg}$ [%]	$\dot{Q}_{co+sc,avg}$ at full load condition [kW]	Difference in full load $\dot{Q}_{co+sc,avg}$ [%]
1	1.50	Fixed	4.33	-	9.08	-
2	1.25	Fixed	4.56	+5.3%	8.85	-2.5%
3	1.50	Variable	5.26	+21.5%	9.08	-
4	1.25	Variable	5.47	+26.3%	8.85	-2.5%

- Compare Case #1 and #2
  - Operation at opt. charge increases the $SCOP_{avg}$ by 5.3%. But the $\dot{Q}_{co+sc}$ drops by 2.5%
  - If both these cases are operated to match the $\dot{Q}_{co+sc}$, the 5.3% would drop
  - Similar conclusions can be seen comparing cases #3 and #4
- Compare Case #1 and #3
  - The variable condenser WEG outlet temperature causes #3 to operate at lower pressure ratio
  - #1 undergoes higher cycling loses because of reduced compressor operating envelope
  - Similar comparison can be done for cases #2 and #4

- The variable speed compressor is operated at two different refrigerant charges
  - 1.50 kg = Receiver is half full
  - 1.25 kg = Charge corresponding to peak COP (Receiver is empty)
- At both these charges, the variable speed compressor is tested with variable/fixed condenser inlet conditions

Conclusions

- Experimentally studied the effect of charge and the fixed/variable outlet condition on the seasonal performance of an R410A water heat pump with a variable speed compressor
- The $SCOP_{avg}$ for the variable speed compressor with variable outlet condition was 21.5% higher than the fixed outlet condition
  - Lower cycling losses and reduced pressure ratio
- Peak in COP = Before receiver starts filling up
  - When the variable speed compressor was operated at this optimum charge, the $SCOP_{avg}$ improved by 5.3% for the fixed outlet condition and 4.8% for the variable outlet condition.
- The charge optimization curve divided into different regions to provide a possible explanation for this peak in COP
  - Increased power consumption due to reduced two phase heat transfer in the subcooler vs. increased cooling capacity due to higher subcooling in the subcooler
- Limitations of this charge opt.
  - Heating capacity is reduced when operating at this peak COP charge
  - Any refrigerant leak can drop the COP significantly
  - TXV/EXV might have problems with lower charge due to reduced subcooling

Acknowledgements

- Thanks to ACRC member companies for their support

- Thanks to
  - Parker Hannifin for the following components – EEV, filters, receiver, solenoid valves
  - Emerson / Copeland for compressor samples
  - Danfoss for brazed plate heat exchangers
  - Creative Thermal Solutions for the technical support

Thank you

Any Questions?

sti2@illinois.edu