A Free-Open-Source Modelica Simulation Package for Air Conditioning and Refrigeration Cycle (SPARC)

Chenjiyu Liang, Donghun Kim, Jiacheng Ma

Lawrence Berkeley National Laboratory | Carrier Corporation

chenjiyuliang@lbl.gov

Presentation Outline

01 **Motivation & Research Gap**
Need for free open-source vapor-compression cycle simulation tools

02 **Package Structure**
Modular component models

03 **System Implementation**
Connecting models to complete vapor-compression cycles

04 **Model Validation**
SPARC predictions against experimental data

05 **Conclusions & Future Work**
Overall contributions and development roadmap

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Motivation & Research Gap

01 / Challenge

Growing Need for Dynamic Simulation

- Advanced control & FDD require high-fidelity dynamic models
- Dynamic models support system design and optimization

02 / Current Gap

Lack of Open Vapor-compression Cycle Modelica Libraries

- Commercial tools are costly and less flexible
- Existing open-source libraries have limited refrigerant and HX support
- Custom connectors reduce compatibility with MSL-based models

03 / Solution

The SPARC Modelica Library

- Open-source Modelica package
- Supports diverse refrigerants and HX configurations
- Uses standard connectors compatible with MSL

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01 **Motivation & Research Gap**
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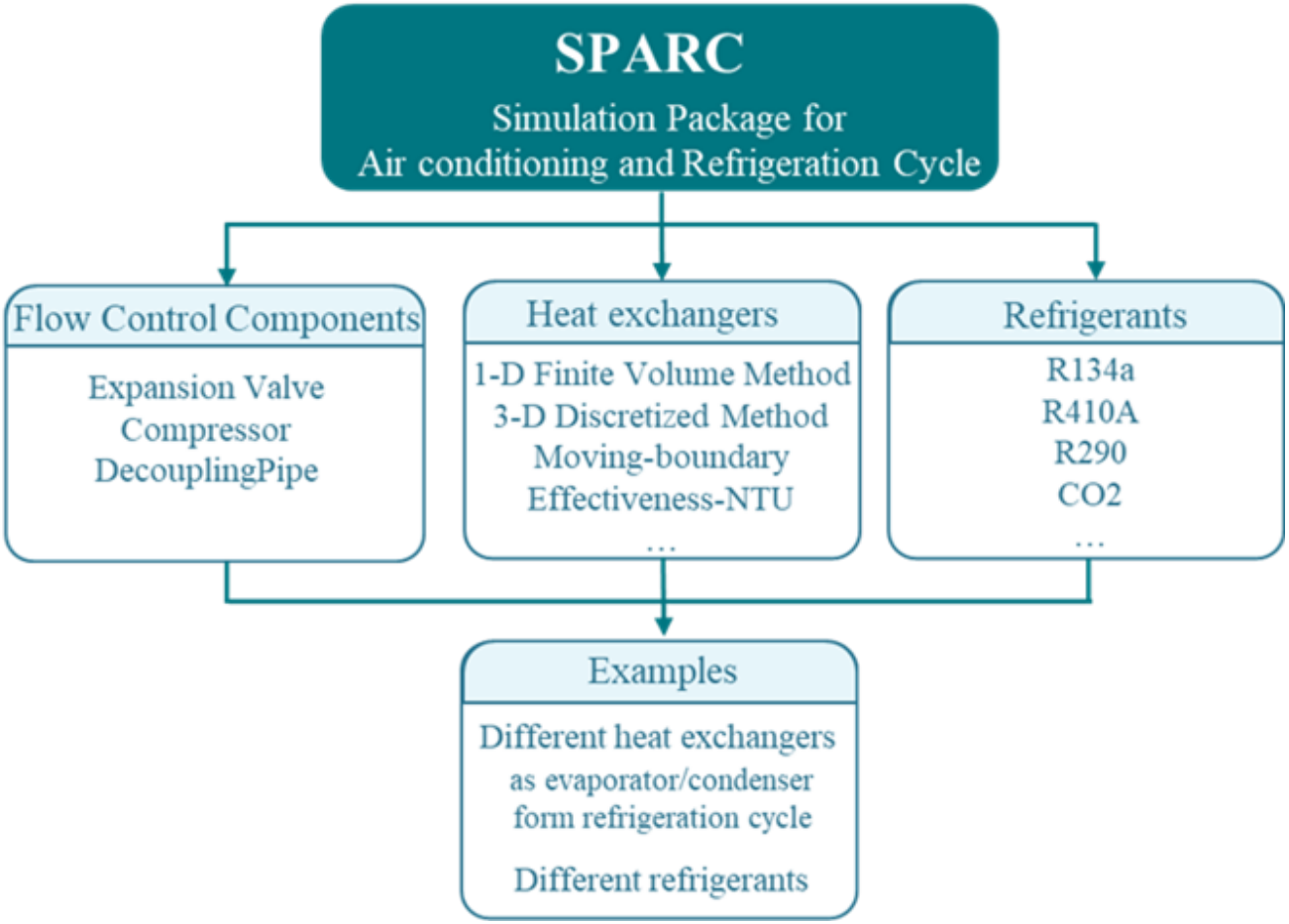
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Package Structure



SPARC is organized into modular sub-packages

Compressor Models

01 / FIXED-SPEED

Ten-Coefficient Model

- Uses evaporating temperature (T_e) and condensing temperature (T_c) as inputs
- Mathematical formulation:
$$Y = a_1 + a_2 T_e + a_3 T_c + a_4 T_e^2 + a_5 T_e T_c + a_6 T_c^2 + a_7 T_e^3 + a_8 T_e^2 T_c + a_9 T_c^2 T_e + a_{10} T_c^3$$
- Predicts mass flow rate, power, and capacity

02 / VARIABLE-SPEED

Efficiency-Based Model

- Refrigerant mass flow rate relation:
$$\dot{m}_{ref} = \eta_v \rho_{suc} V_{dis} \mathrm{N}$$
- Compressor power consumption:
$$W_{comp} = \frac{\dot{m}_{ref}(h_{dis,s} - h_{suc})}{\eta_{is}}$$
- Volumetric and isentropic efficiencies (η_v and η_{is}) calculated from pressure ratio correlations

ANSI/AHRI. (2015). Standard for Performance Rating of Positive Displacement Refrigerant Compressors and Compressor Units. Standard 540. Arlington.
Liang, C., Li, X., Shi, W., & Wang, B. (2021). A direct expansion air handling unit assisted by liquid desiccant for different sensible and latent heat ratios. Energy and Buildings, 238, 110672.

Expansion Valve Models

Orifice-Based Flow

Flow relation driven by pressure difference.

$$\dot{m}_{ref} = C_{eff}\sqrt{\rho_{in}(p_{in} - p_{out})}$$

Effective Flow Coeff.

Dynamically controlled by the superheat signal.

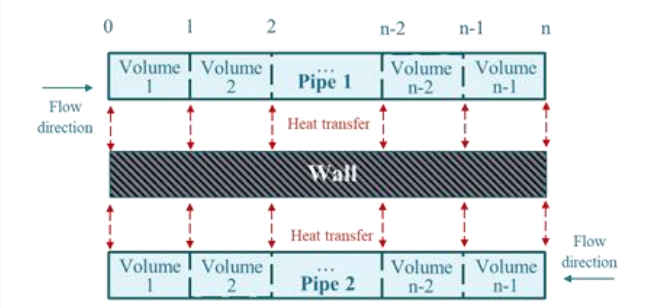
$$C_{eff} = C \cdot Adj \cdot f(SH)$$

Isenthalpic Throttling

Modeling refrigerant phase transitions.

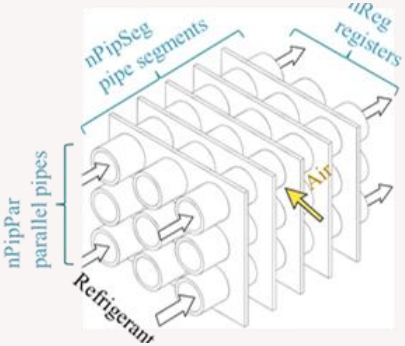
$$h_{in} = h_{out}$$

Heat Exchanger Model Portfolio



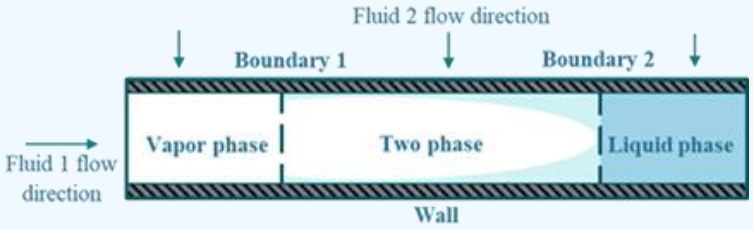
Finite Volume Method

One-dimensional spatial thermodynamic state



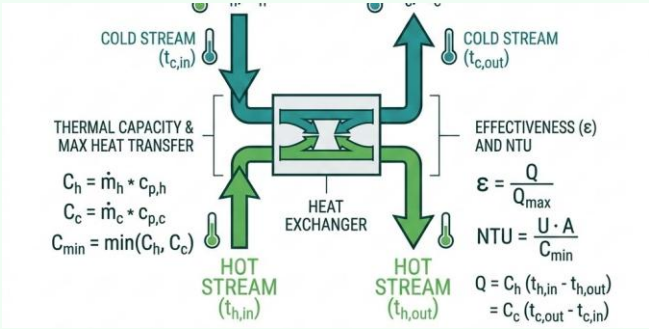
3-D Fin-Tube Discretized

Utilizing highly localized geometric characteristics



Moving Boundary Method

Divides into time-varying based on phases



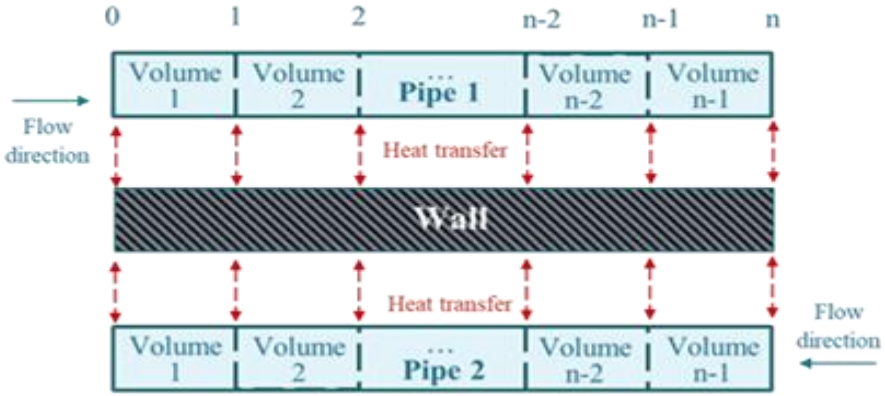
Effectiveness-NTU Method

Lumped performance calculations

Finite Volume Method (FVM) Principles

Model Description & Discretization

- The model is developed based on the Modelica.Fluid.BasicHX library.
- Discretizes pipe into $n-1$ control volumes along the flow path.
- Counter-flow or parallel-flow configurations to capture precise spatial distributions.



Model structure of the FVM heat exchanger model

Conservation Equations

Mass Conservation

$$\frac{d}{dt}(\rho_i V_i) = \dot{m}_{b,i}$$

Energy Conservation

$$\frac{d}{dt}(\rho_i V_i u_i) = \dot{H}_{b,i} + \dot{Q}_{b,i} + W_{b,i}$$

Species Conservation

$$\frac{d}{dt}(\rho_i V_i X_{i,k}) = \dot{m}_{X,b,i,k}$$

Local Heat Transfer

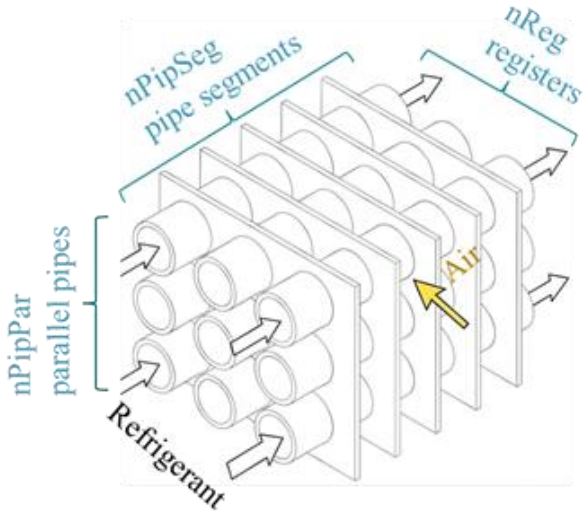
$$\dot{Q}_{b,i} = \alpha_i A_i (T_{w,i} - T_{f,i})$$

- V_i , ρ_i , u_i , $X_{i,k}$ represent volume, density, internal energy, and species mass fraction of the *i*-th cell.
- α_i, A_i, $T_{w,i}$, $T_{f,i}$ represent the heat transfer coefficient, area, wall temp, and fluid temp.

3-D Fin-Tube Discretized HX Principles

Model Description & Discretization

- Developed based on the Modelica Buildings library (DryCoil/WetCoil discretized models).
- Consists of n_{Reg} registers perpendicular to air flow, each with n_{PipPar} parallel pipes.
- Each pipe is divided into n_{PipSeg} segments modeled as a control volume.



Conservation Equations

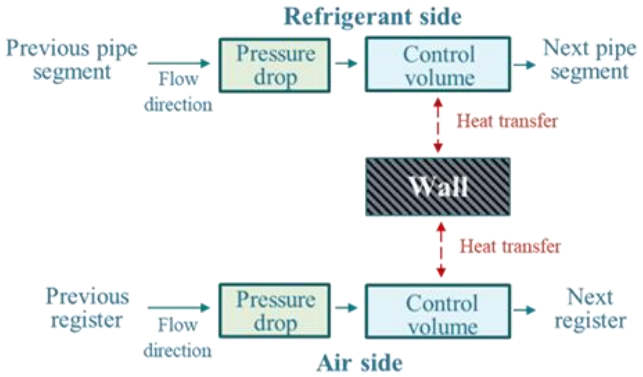
Convective Heat Transfer

$$Q_1 = G_{c,1}(T_w - T_1), \quad Q_2 = G_{c,2}(T_w - T_2)$$

Wall Thermal Dynamics

$$C\frac{dT_w}{dt} = Q_1 + Q_2$$

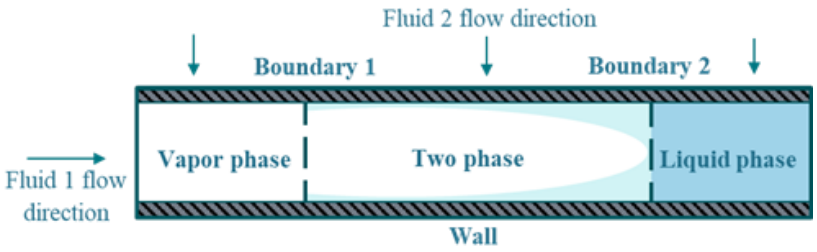
- $G_{c,1}$,$G_{c,2}$ represent convective thermal conductance for the two fluids.
- T_w ,T_1 ,T_2 represent wall temperature and fluid temperatures respectively.
- C is the wall thermal capacitance.



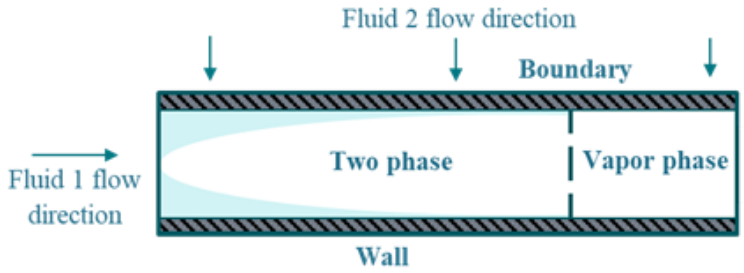
Model structure of the 3-D discretized heat exchanger model

Moving Boundary Method HX Principles

Model Description & Zoning



Condenser model



Evaporator model

Conservation Equations

Overall Mass Balance

$$\frac{d}{dt}(m_{tp} + m_v) = \dot{m}_{in} - \dot{m}_{out}$$

Overall Energy Balance

$$\frac{d}{dt}(U_{tp} + U_v) = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + Q$$

Heat Transfer Formulation

$$Q = \alpha_{tp}A_{tp}(T_w - T_{tp}) + \alpha_v A_v(T_w - T_v)$$

- m_{tp} and m_v are the mass of the two-phase and vapor regions.
- U_{tp} and U_v are the internal energy of the two-phase and vapor regions;Q is the heat transfer between the wall and the refrigerant.
- α_{tp} and α_v, A_{tp} and A_v are heat transfer coefficients and areas in the two-phase and vapor regions; T_w, T_{tp}, and T_v are the temperature of the wall, two-phase and vapor regions.

Song et al. (2024). Power-hardware-in-the-loop validation of air-source heat pump for fast frequency response applications. Electric Power Systems Research, 235, 110754.
Rasmussen & Alleyne (2006). Dynamic modeling and advanced control of air conditioning and refrigeration systems. Air Conditioning and Refrigeration Center TR-244.

Effectiveness-NTU Method HX Principles

Model Description & Relations

- **Lumped Parameter Approach:**

Describes performance based on overall heat transfer characteristics without resolving spatial distributions.

- **Effectiveness:**

Actual heat transfer to the thermodynamic maximum potential.

Effectiveness (Constant Temperature Phase Change)

$$\varepsilon = 1 - \exp(-NTU)$$

Number of Transfer Units (NTU)

$$NTU = \frac{KF}{C_{min}}$$

- K and F represent the heat transfer coefficient and area of the heat exchanger.
- $\mathrm{C_{min}}$ and $\mathrm{C_{max}}$ represent the smaller and larger heat capacities of fluids within the heat exchanger.

Dehumidification Models for Heat Exchangers

- **Two Wet-Coil Model Implementations:** 1-D FVM HX and 3-D discretized HX
- **Condensation:** Water vapor condenses on the coil surface when the surface temperature falls below the air dew point.
- **Mass transfer:** Calculated using the heat and mass transfer analogy based on the Lewis number.
- **Latent heat:** Condensation heat is incorporated into the heat exchanger energy balance.

Heat and mass transfer analogy

$$\frac{h}{h_m} = \rho c_p Le^{1-n}$$

Water vapor condensation rate

$$\dot{m}_{cond} = \frac{G_c}{c_p Le^{1-n}} (X_\infty - X_s)$$

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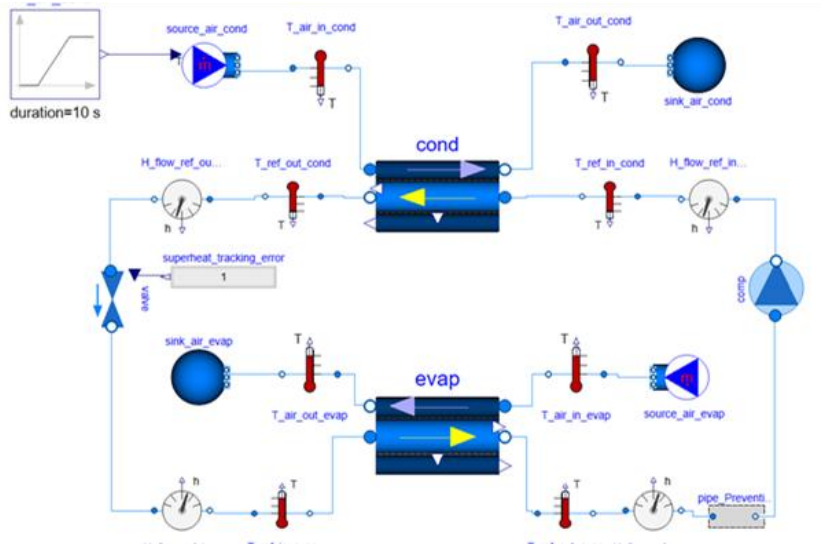
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Modular component models in the library

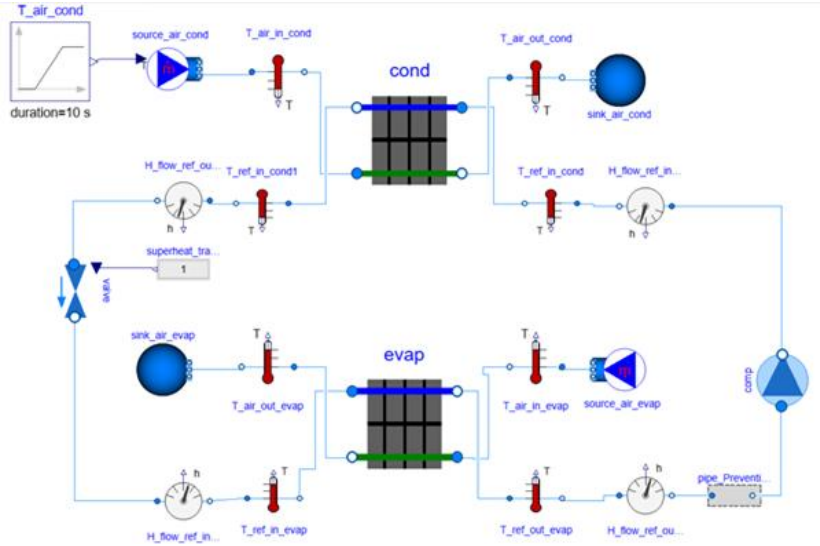
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Overall contributions and development roadmap

System-level Implementation





Vapor-Compression Cycles

Built by interconnecting component models (compressor, evaporator, condenser, expansion valve)

Charge Tracking

Calculated by summing local refrigerant mass using localized volume and densities

Medium properties

Refrigerant and air/water properties calculated using Modelica Standard Library (MSL) and ExternalMedia library

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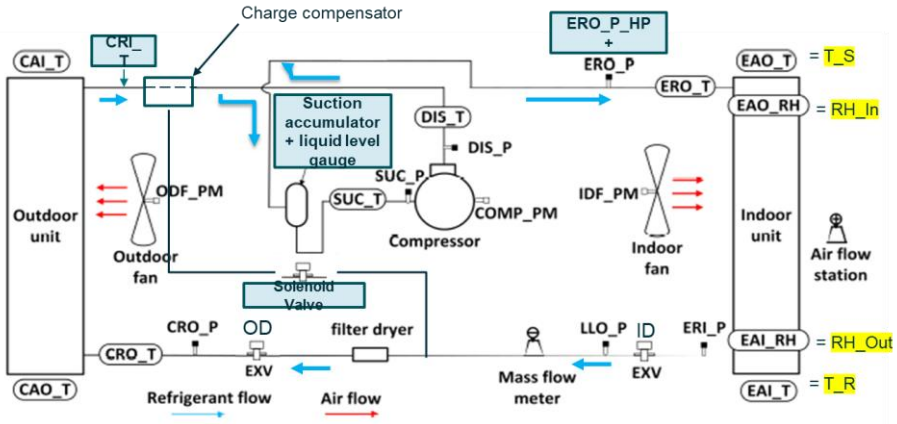
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Validation Case: Residential R410A Heat Pump

Experiment Validation

Steady-state performance comparison against the experiment results.

- Simulation models are exported as FMUs and executed in Python.



Studied residential heat pump:

- Nominal heating capacity: ~10 kW (34,121 Btu/hr).
- Nominal refrigerant charge: 6.5 kg (14.3 lb).

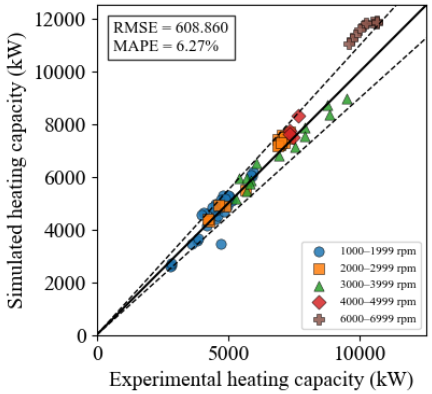
Tested Boundary Conditions

The comparison covers a comprehensive range of critical system operating parameters:

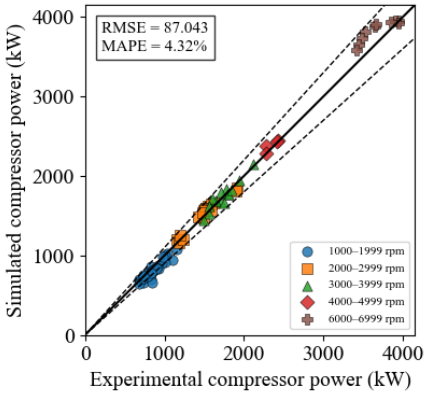
Parameter	Range
Evaporator inlet air temp	-8.3 to 11.1 °C
Indoor fan volume	380 to 1300 CFM
Compressor speed	1500 to 6000 rpm
Charge level	4.1 to 6.85 kg

Validation Results Across Operating Conditions

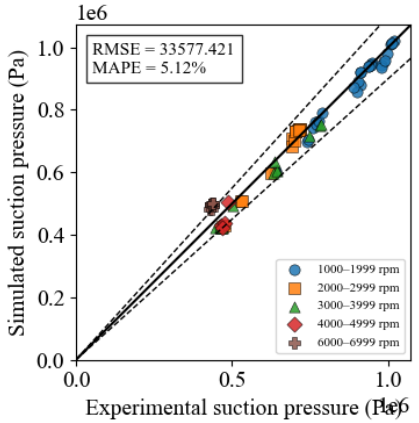
Validation of FVM-HX based system model



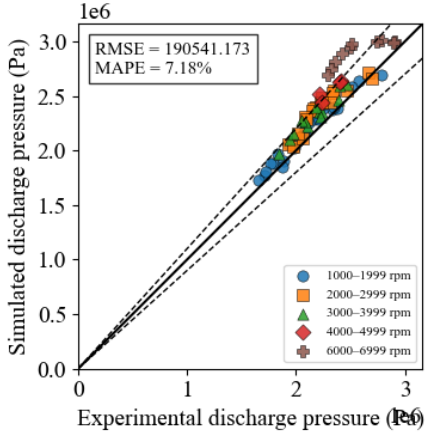
Heating capacity



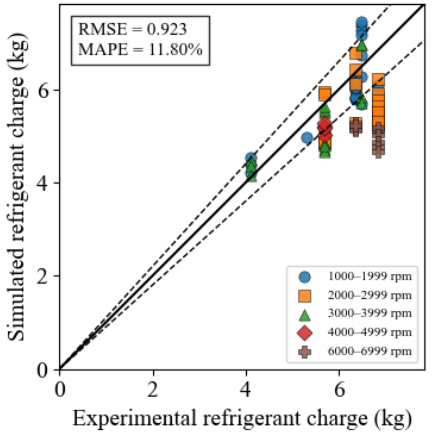
Compressor power



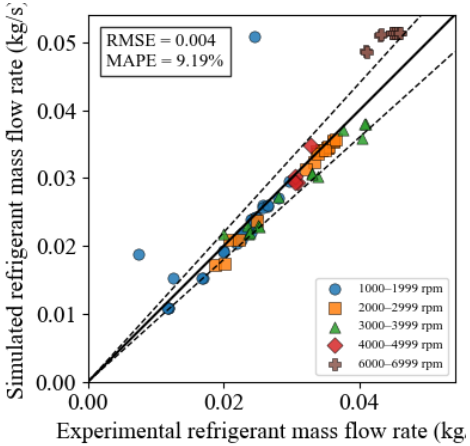
Suction pressure



Discharge pressure



Charge level



Refrigerant mass flow

Validation Results: Overall Agreement

Overall, good agreement is observed between SPARC predictions and experimental results

The model is highly accurate and suitable for further cycle analysis.

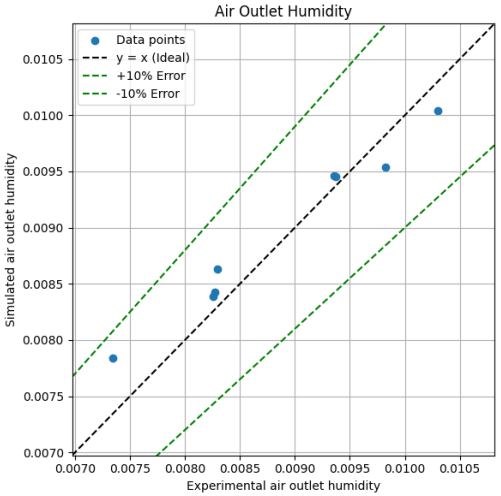
Deviation Matrix (MAPE)

Variable	1-D FVM	3-D Fin-Tube	Moving boundary	Effectiveness-NTU
Heating capacity	6.19%	8.06%	9.78%	14.74%
Compressor power	3.74%	5.48%	7.30%	10.18%
Suction pressure	5.16%	11.74%	5.48%	9.52%
Discharge pressure	7.35%	17.94%	10.12%	10.20%
Refrigerant charge	11.8%	12.26%	9.06%	/
Refrigerant mass flow rate	9.06%	12.20%	6.46%	/

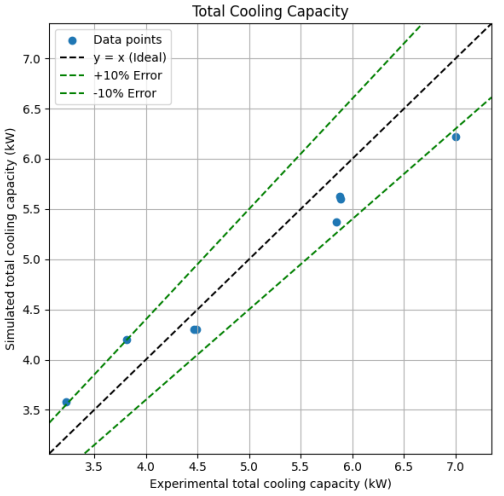
Validation Results: Dehumidification of 1-D FVM HX

Experimental Validation Datasets

1. Refrigerant-to-Air Evaporator
Source: NISTIR 7225
Coil 9, R410A refrigerant
Varying air and refrigerant operating conditions

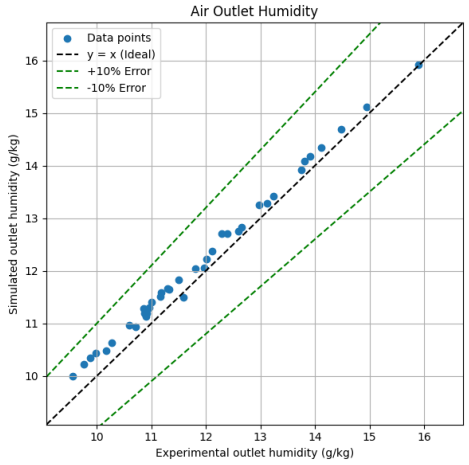


Air outlet humidity

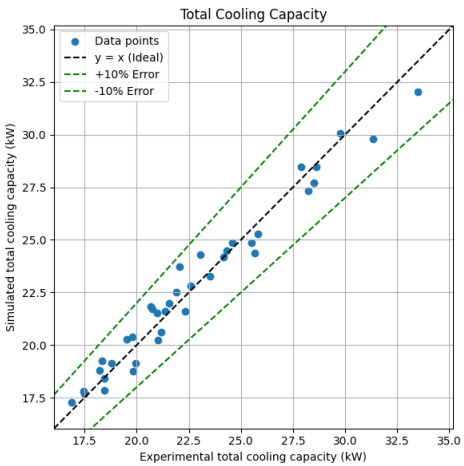


Total cooling capacity

2. Water-to-Air Cooling Coil
Source: NRCC Report 16509
4-row chilled-water coil
Varying air and chilled-water operating conditions



Air outlet humidity

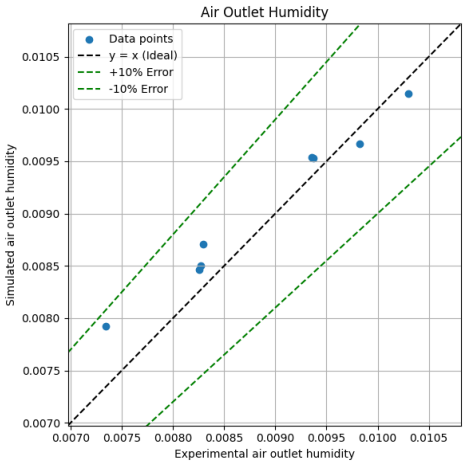


Total cooling capacity

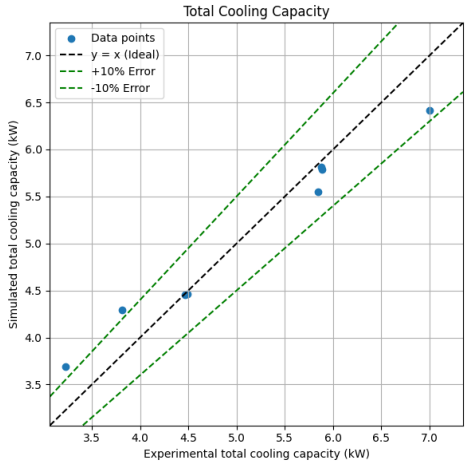
Validation Results: Dehumidification of 3-D fin-tube discretized HX

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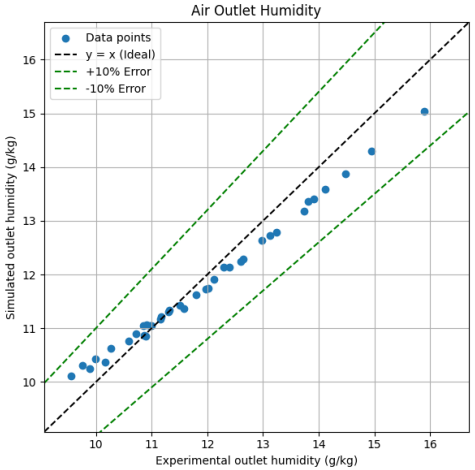


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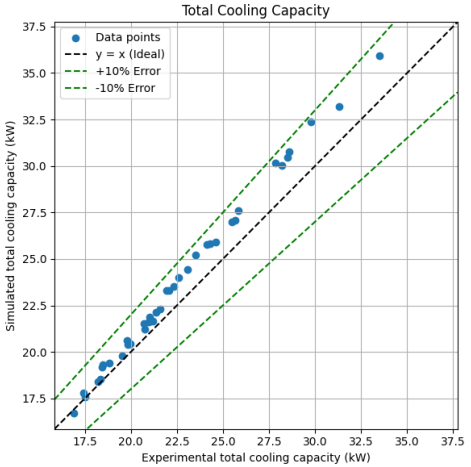


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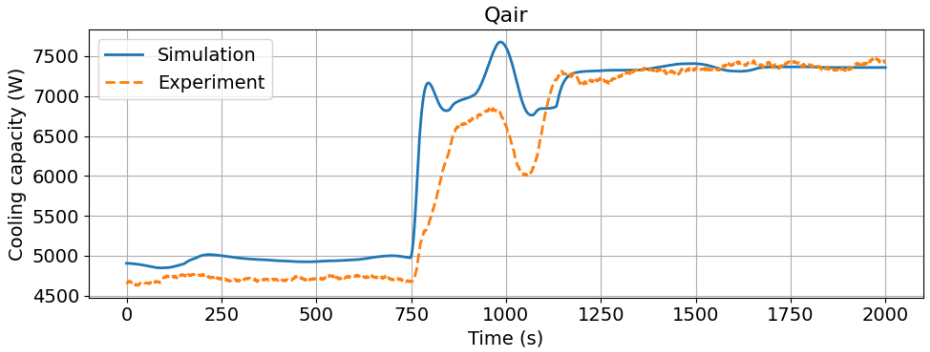


Total cooling capacity

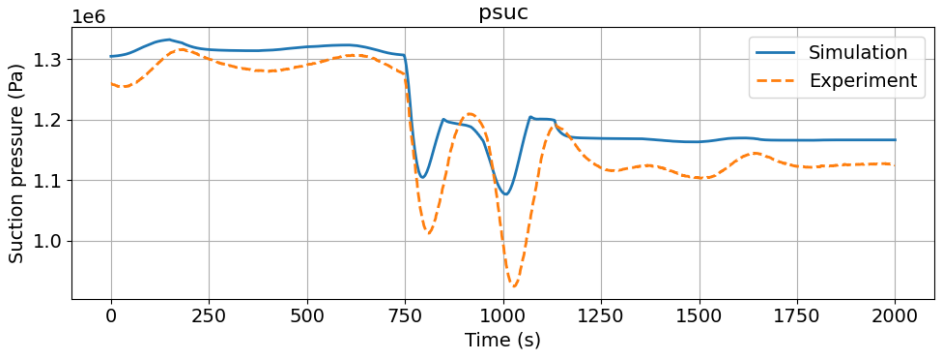
Transient validation when compressor speed changes

Under a cooling condition (Condition A): 80.0 °F (26.7 °C) indoor and 95.0 °F (35 °C) outdoor

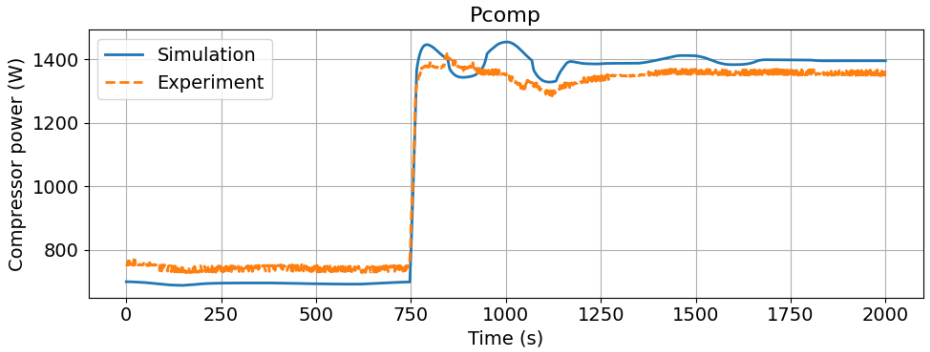
- Compressor speed changes from 1300 to 2300 rpm, and EXV changes accordingly



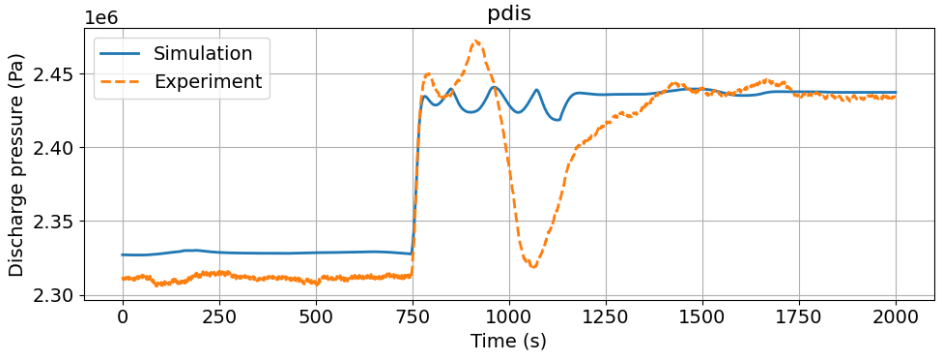
Cooling capacity



Suction pressure



Compressor power

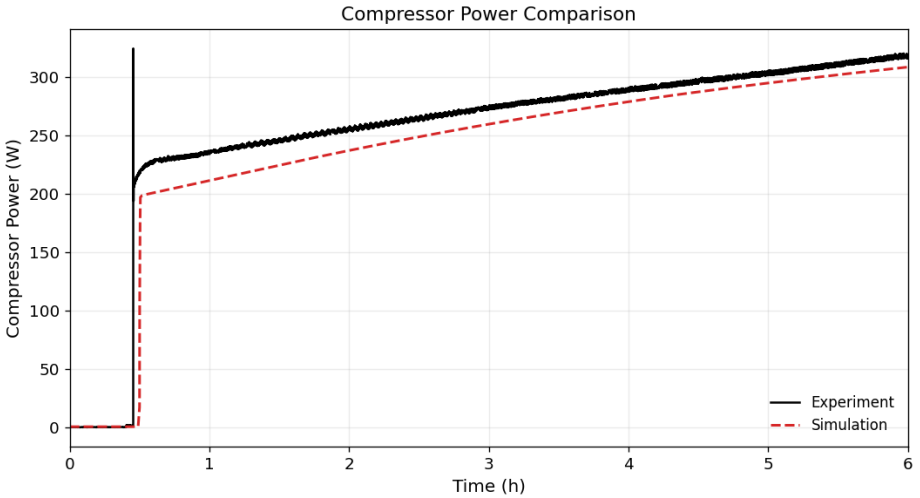


Discharge pressure

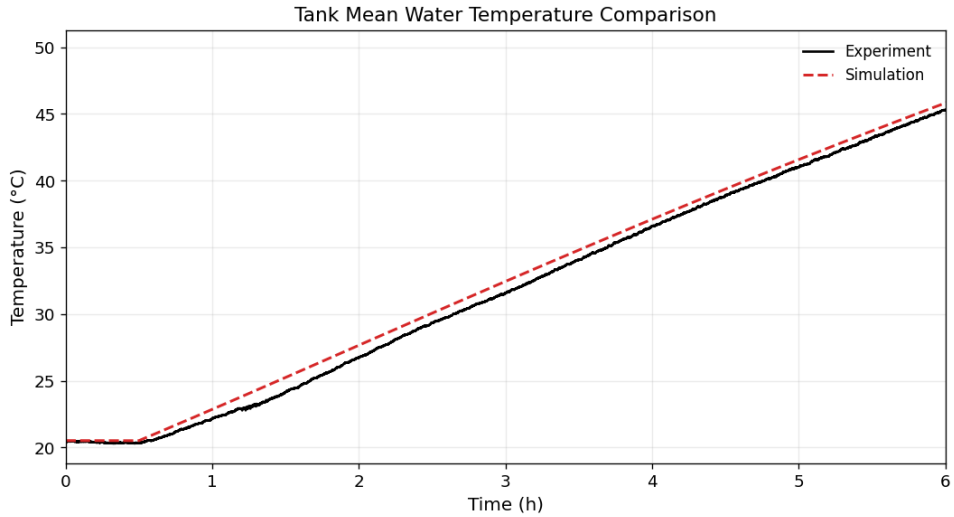
Transient validation using an R1234yf HPWH

Ambient conditions: 67.5 °F and 50% RH

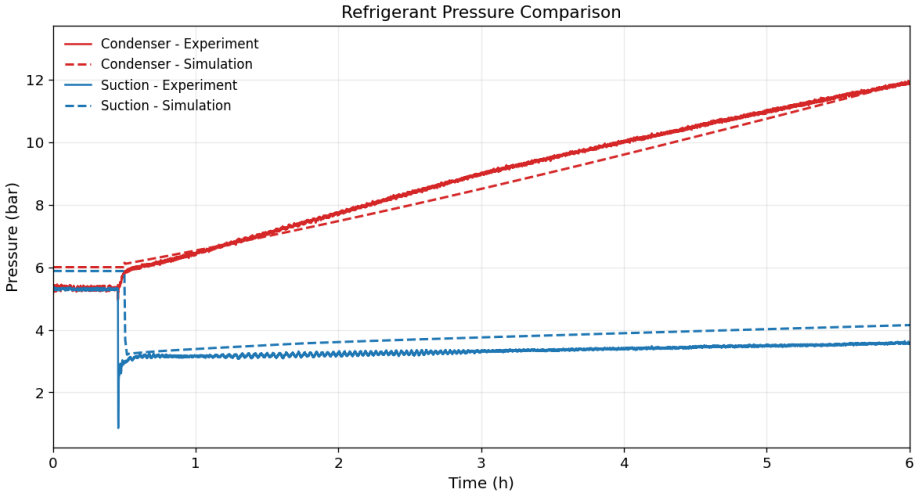
- The compressor starts at 30 min to heat the water tank



Compressor power consumption



Tank water temperature



Refrigerant pressure

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Conclusions and Future Work

SPARC Library Development

- Developed a **free open-source** Modelica package for **dynamic** vapor compression cycle simulation.
- Integrates multiple types of heat exchanger models and refrigerants.

Model Validation

- Validated against experimental data under steady-state and transient operating conditions.
- Demonstrated reasonable agreement across multiple heat exchanger modeling approaches.

Future Research Directions

01 **Frosting & Defrosting Processes**
Incorporate transient frosting/defrosting models on heat exchanger surfaces to expand application scope.

02 **Transient Validation**
Validate components and systems under dynamic/transient conditions using experimental dataset.

03 **Advanced Control & FDD**
Apply the model-based framework for control optimization and fault detection algorithm development.

Thank you

Questions?

ChenjiyuLiang@lbl.gov