Static and Dynamic Optimization of Radiant Cooling Systems

SinBerBEST Annual Meeting

Singapore

January 9, 2013

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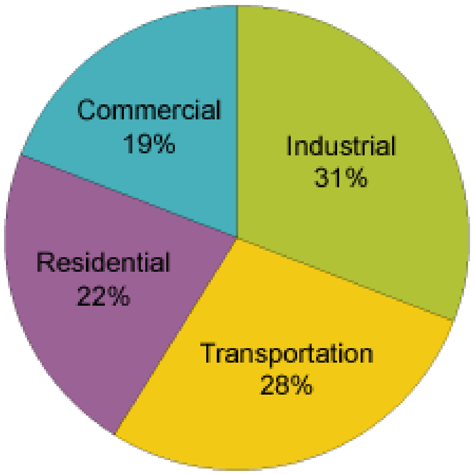
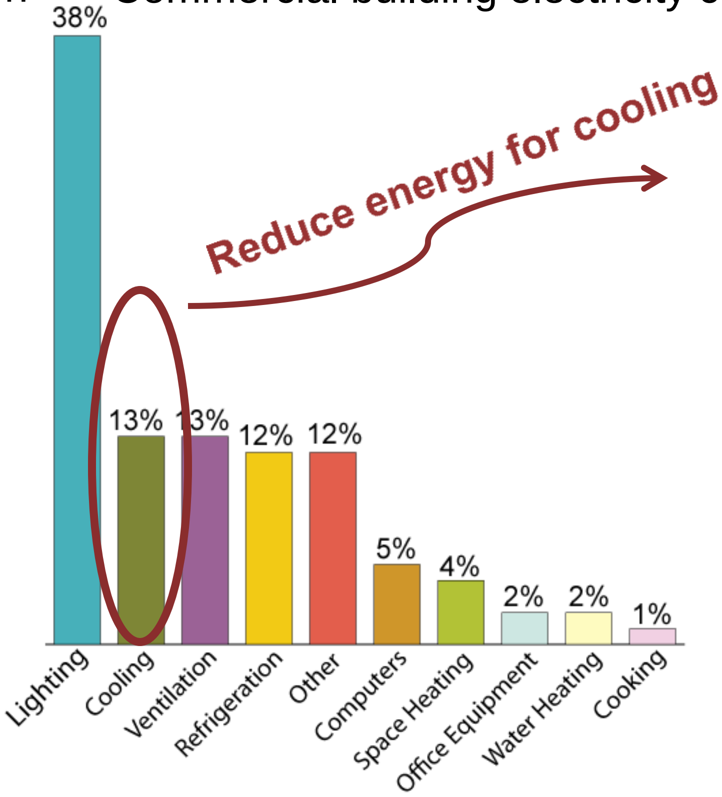
Motivation

The role of cooling in very low energy buildings

Total US energy consumption



Commercial building electricity consumption[1]



Low-lift cooling technology

⇩

30 – 70% savings in energy for cooling[2]

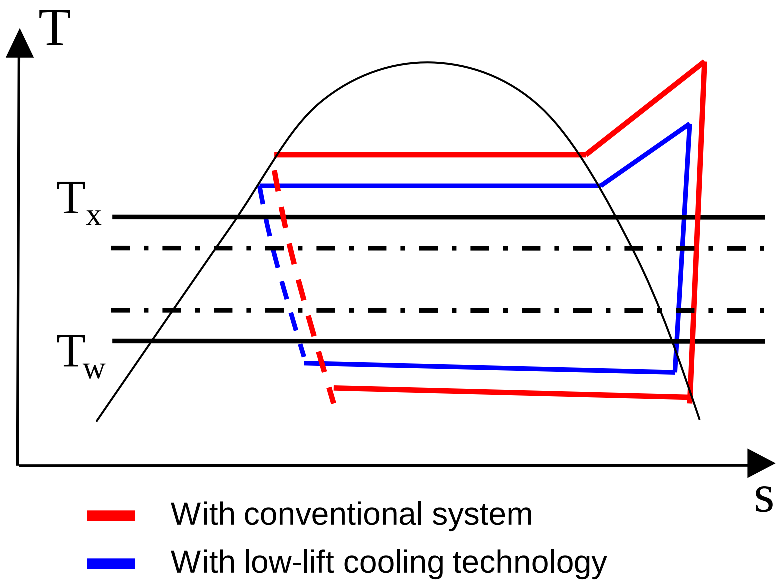
[2]Source: Pacific Northwest National Laboratory analysis

[1]Source: U.S. Energy Information Administration, Annual Energy Review 2008

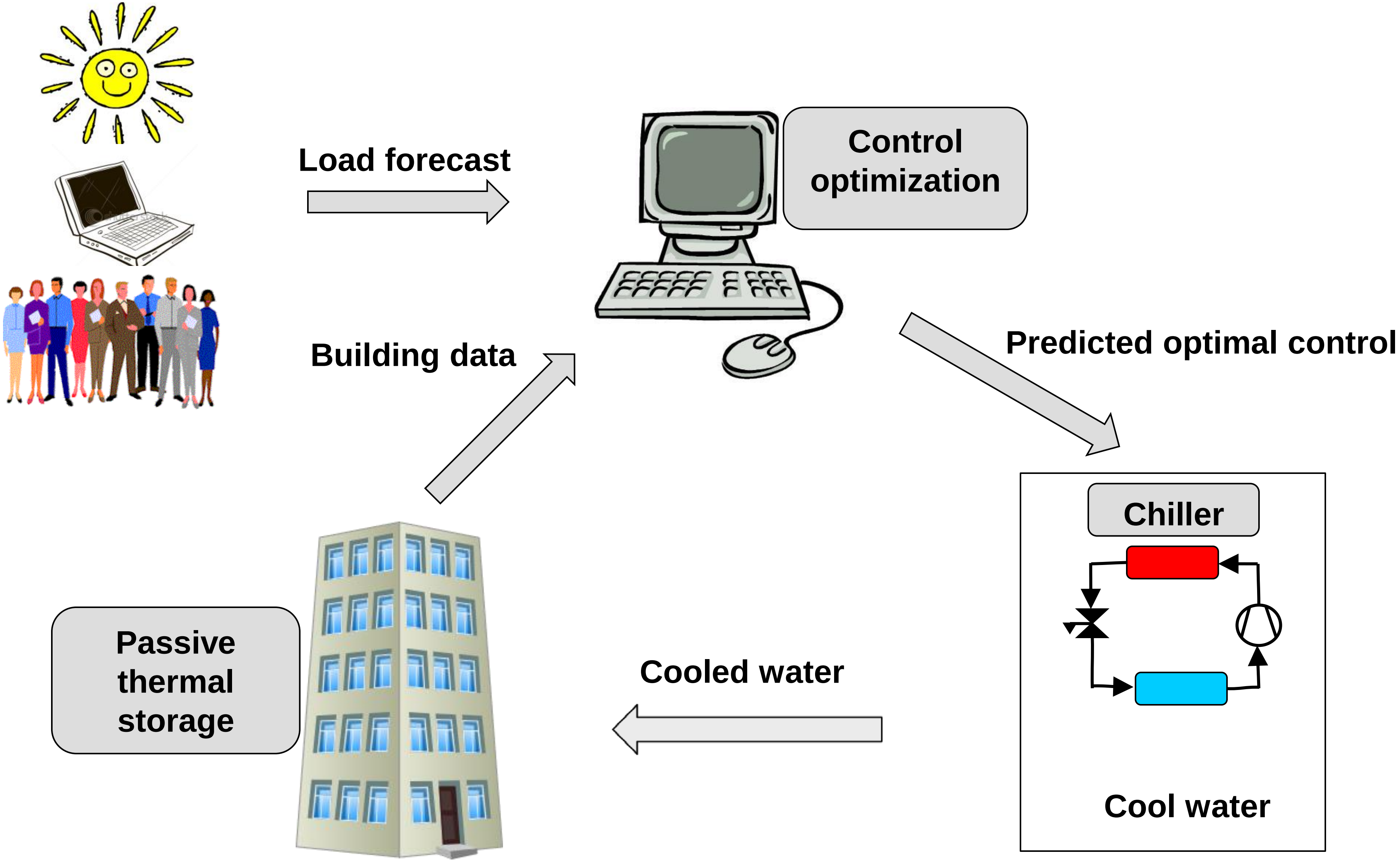
Low-lift cooling technology

- **Radiant hydronic cooling** – reduces transport energy and increases evaporating temperature
- **Thermal storage** – reduces condensing temperature, peak loads and daytime loads
- **Variable speed drive** compressor and fans – reduces flow losses and allows efficient operation at part load
- **Dedicated outdoor air system** – provides ventilation air and dehumidification
- **Building thermal model identification** – allows accurate prediction of cooling loads for pre-cooling control
- **Smart building control** – enables monitoring, system identification and predictive control

Cooling cycle in T-s diagram



Low-lift cooling technology



Load forecast

Control optimization

Building data

Predicted optimal control

Chiller

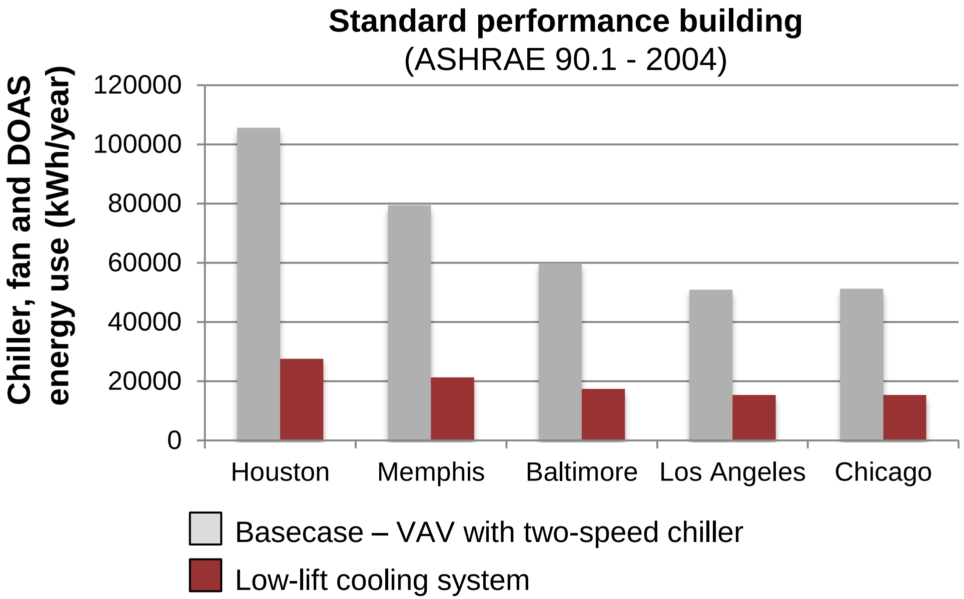
Cool water

Cooled water

Passive thermal storage

Low-lift cooling technology

Pacific Northwest National Laboratory analysis: Office building prototype analysis for five US climates and three envelope performances (standard, mid and high)



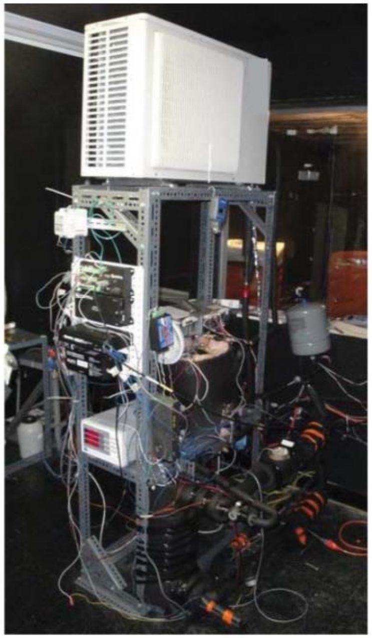
➔ 30 – 70 % savings in annual energy for cooling

Armstrong et al. 2009. Efficient low-lift cooling with radiant distribution, thermal storage and variable-speed chiller controls – Parts I and II.
Katipamula et al. 2010. Cost-effective integration of efficient low-lift baseload cooling equipment.

Experimental work

Nick Gayeski, PhD Thesis, 2010

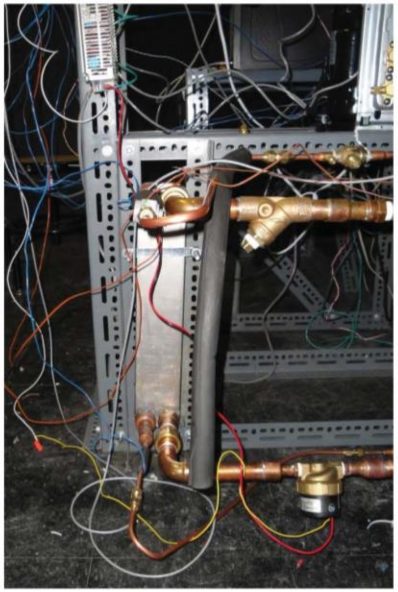
Chiller/heat pump



Radiant concrete floor

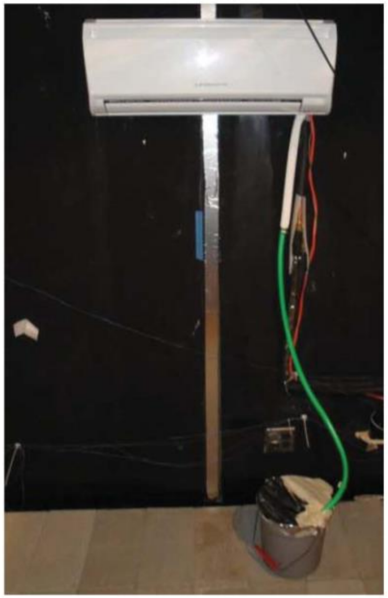
LLCS chiller

Brazed plate heat exchanger



SSAC (SEER~16)

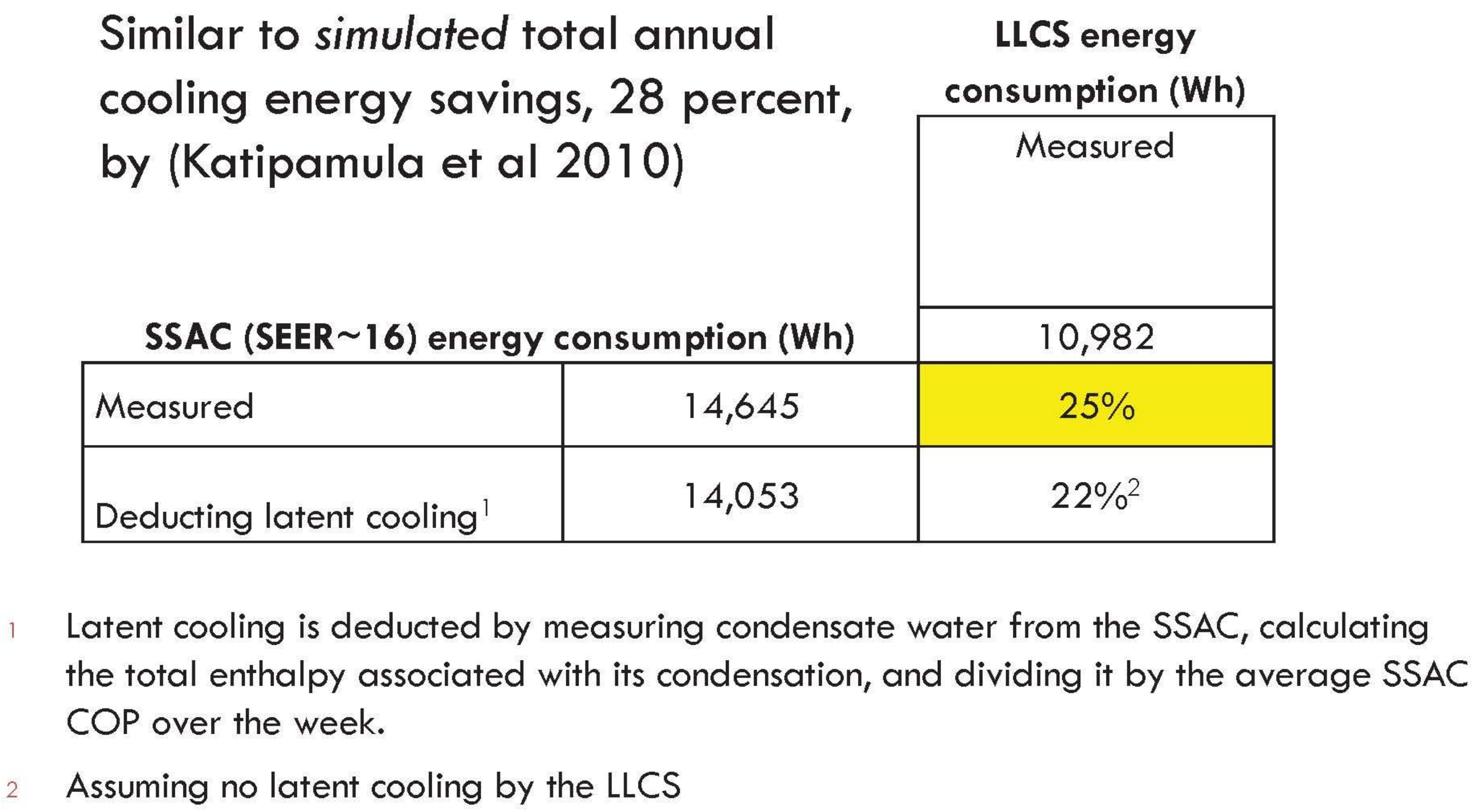
Standard mini-split indoor unit



Experimental work

LLC energy savings relative to split-system

(for Atlanta, subject to standard office loads)



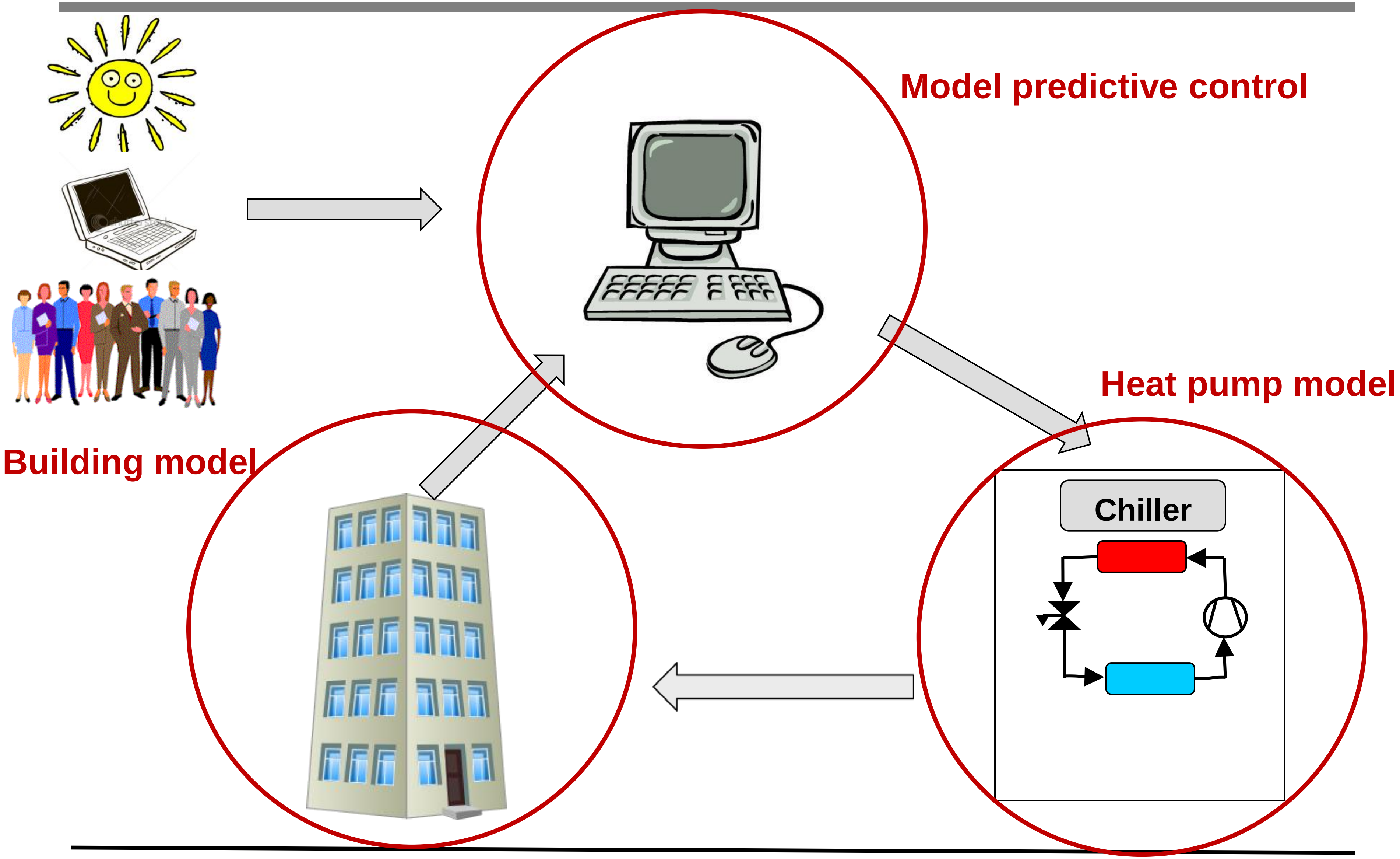
Similar to *simulated* total annual cooling energy savings, 28 percent, by (Katipamula et al 2010)

		LLCS energy consumption (Wh)
		Measured
SSAC (SEER~16) energy consumption (Wh)		10,982
Measured	14,645	25%
Deducting latent cooling[1]	14,053	22%[2]

[1] Latent cooling is deducted by measuring condensate water from the SSAC, calculating the total enthalpy associated with its condensation, and dividing it by the average SSAC COP over the week.

[2] Assuming no latent cooling by the LLCS

Computer simulation



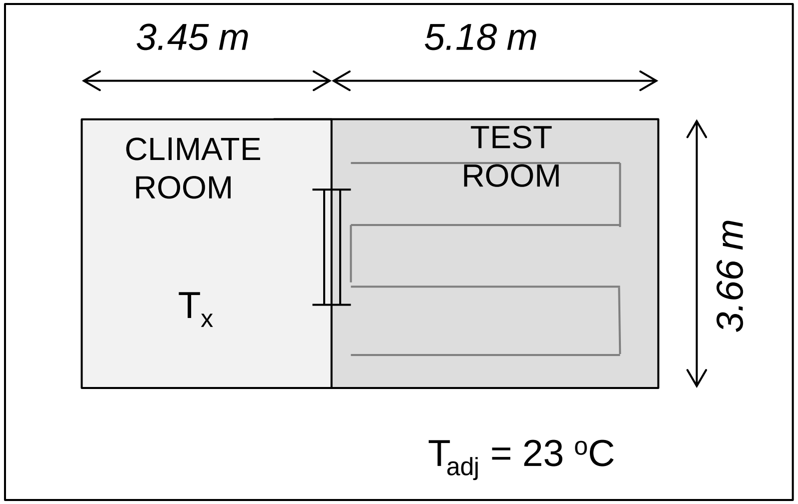
Model predictive control

Heat pump model

Building model

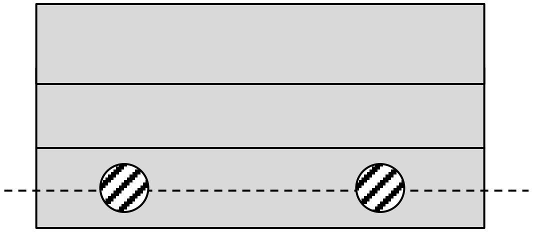
Building model

TRNSYS model of the experimental room



Real TABS construction

Inputs	Outputs
Internal loads	Zone temperature
Water flow rate	Operative temperature
Water supply temperature	Water return temperature
Air flow rate	Floor temperature
Supply air temperature	
Supply air humidity	
Cooling rate	
Heating rate	



TRNSYS TABS construction

Building model

Transfer function model of the experimental room

Model proposed by Armstrong et al. (2009)

For zone, operative and floor temperature:

$$T = \sum_{k=1}^{n} a^k T^k + \sum_{k=0}^{n} b^k T_{adj}^k + \sum_{k=0}^{n} c^k T_x^k + \sum_{k=0}^{n} d^k Q_{load}^k + \sum_{k=0}^{n} e^k Q_c^k$$

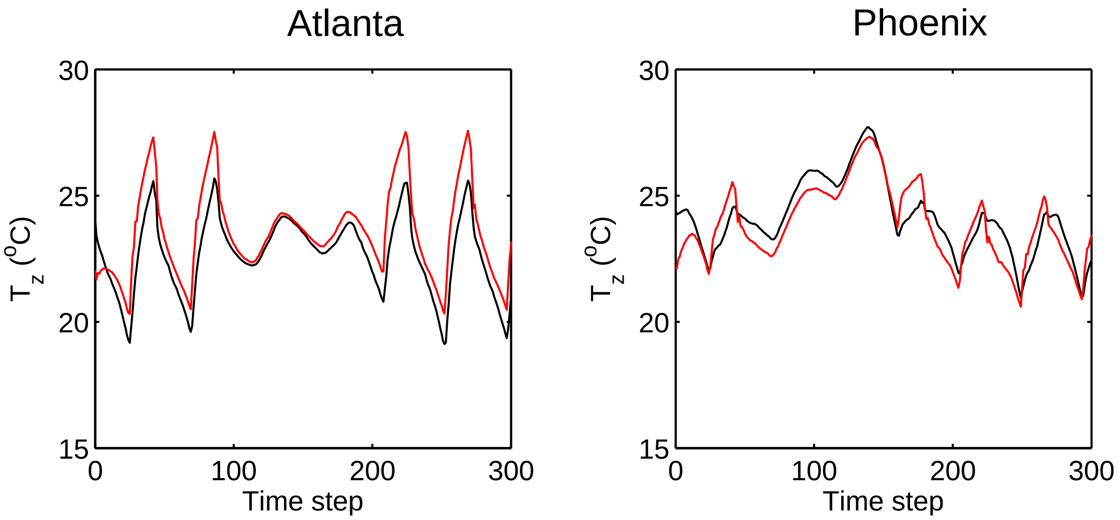
For water return temperature:

$$T_{w,out} = \sum_{k=1}^{n} f^k T_{w,out}^k + \sum_{k=0}^{n} g^k T_{floor}^k + \sum_{k=0}^{n} h^k Q_c^k$$

Coefficients are found by linear regression to TRNSYS data.

Building model

Validation for TRNSYS model



* N. Gayeski

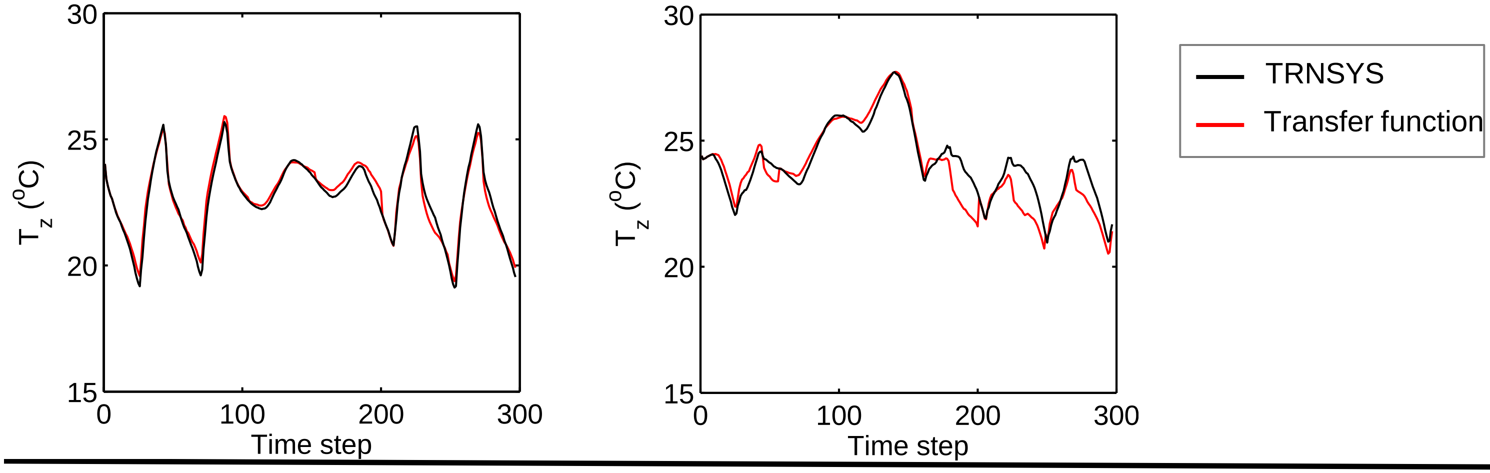
INPUTS

T_x … climate room temperature
$T_{w,in}$ … supply water temperature
m_w … water flow rate
$Q_{internal}$ … internal load

OUTPUTS

T_z … zone temperature
T_f … floor temperature
$T_{w,return}$ … water return temperature

Validation for transfer function model



Heat pump model

Model flowchart

Assume h_{liq_as} and $P_{comp\_out\_as}$

⇩

Call evaporator model (1)

⇩

Call compressor model (2)

⇩

Call condenser model (3)

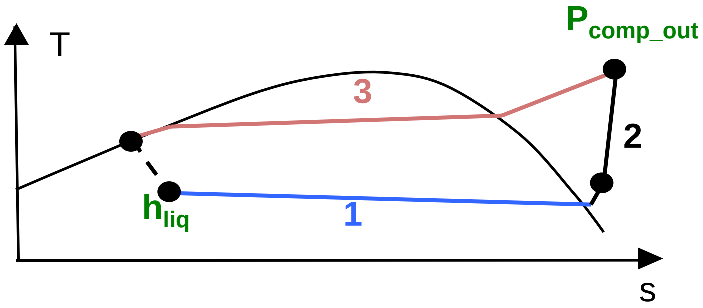
⇩

Is $h_{liq\_as} = h_{liq}$ and $P_{comp\_out\_as} = P_{comp\_out}$?

NO ⇧ (return to: Assume h_{liq_as} and $P_{comp\_out\_as}$)

⇩ YES

Calculate Coefficient of Performance (COP)



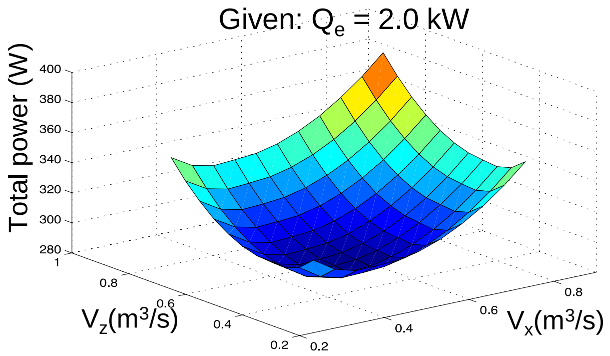
$$COP = \frac{Q_e}{E_{comp} + E_{evap,fan} + E_{cond,fan}}$$

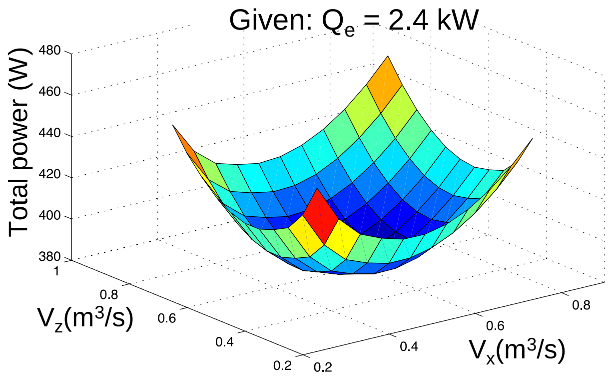
Zakula T., Gayeski N., Armstrong P. and Norford L . 2011. Variable-speed Heat Pump Model for a Wide Range of Cooling Conditions and Loads. *HVAC&R Research* 17(5).

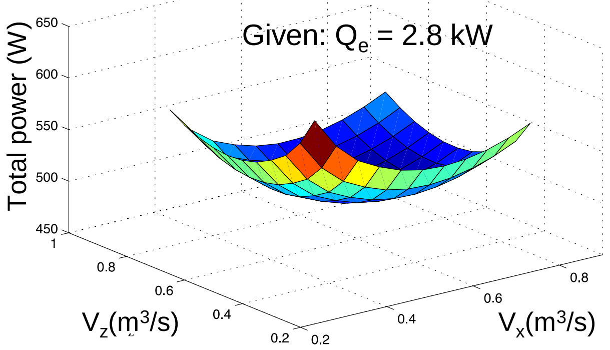
Heat pump model

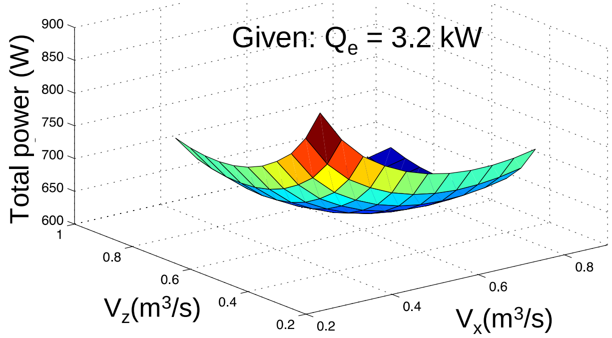
Heat pump static optimization

Finding the optimal evaporator ($V_{z\ opt}$) and condenser ($V_{x\ opt}$) air flows for minimum power consumption if cooling rate, room temperature and outside temperature are given.





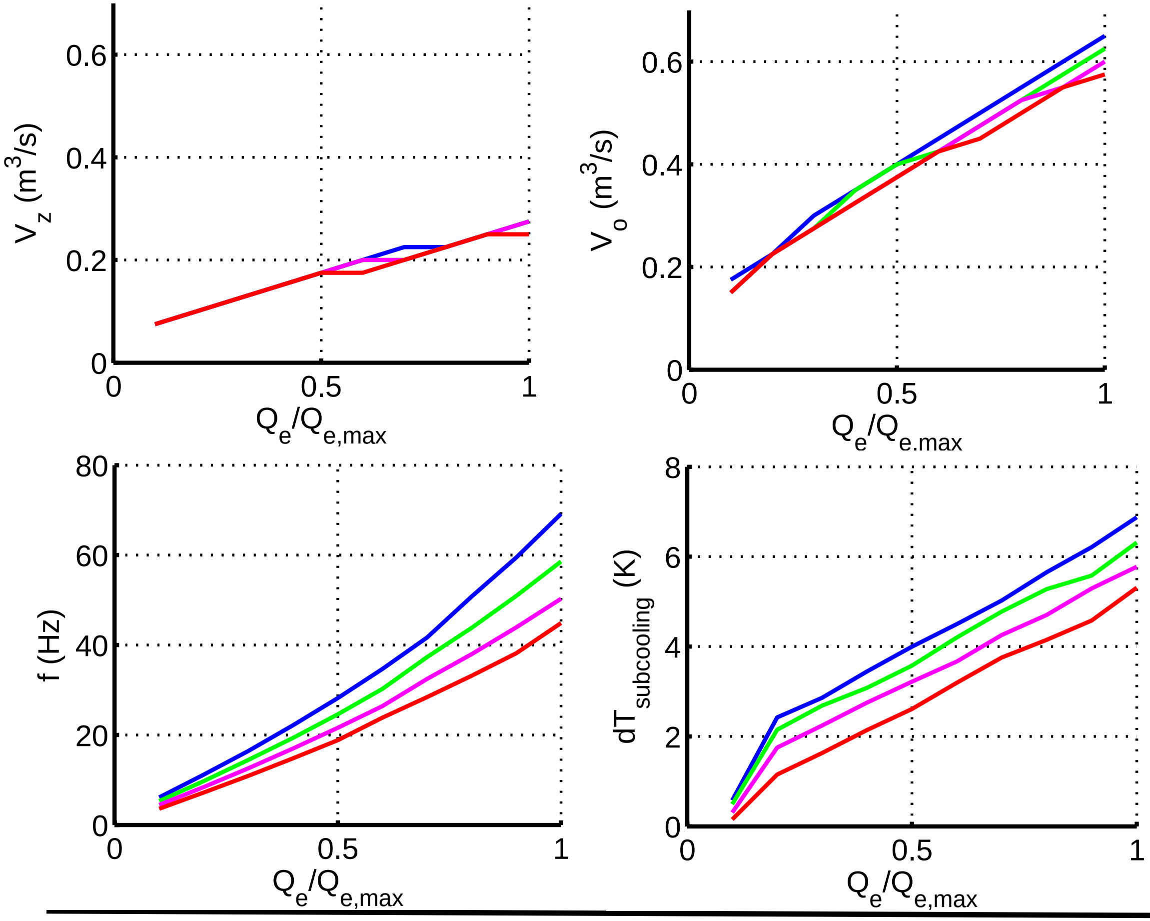




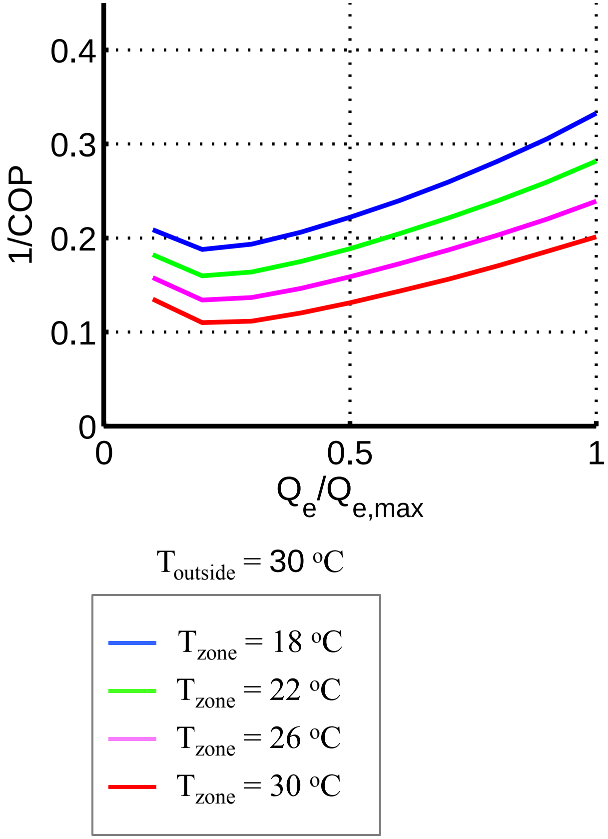
Heat pump

The results of the heat pump optimization for a range of cooling conditions.

Optimal parameters



Power consumption



Heat pump

For non-optimized case:

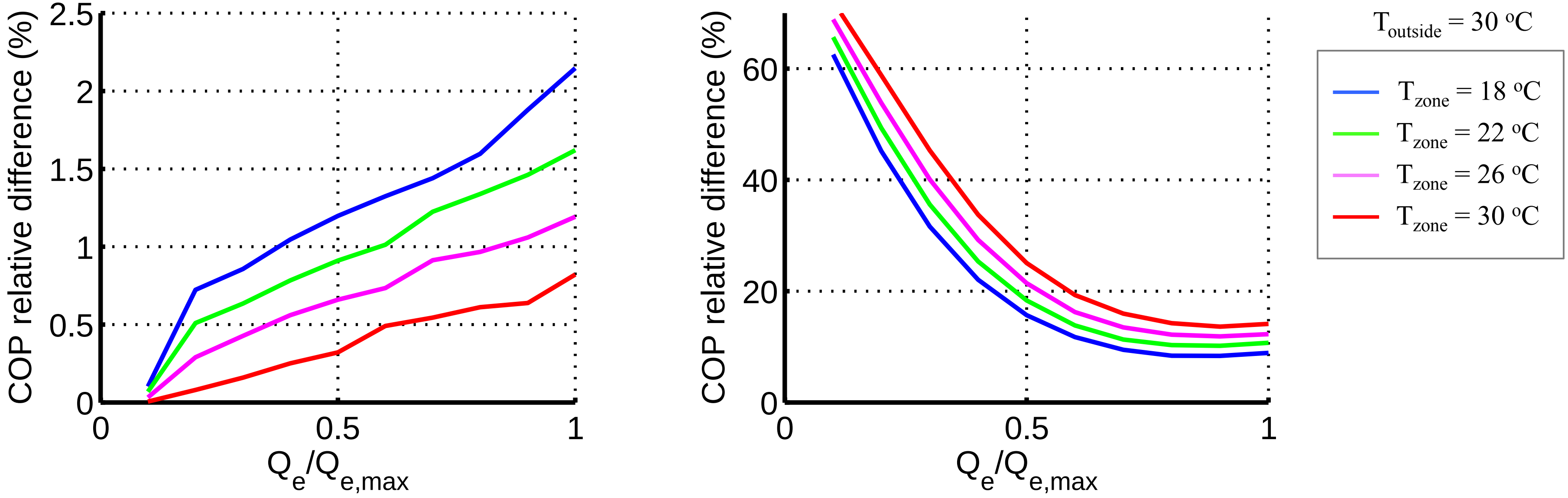
$V_{z\_max} = 0.15\ m^3/s$
$V_{o\_max} = 0.77\ m^3/s$

Maximum airflows for Mr. Slim

Current models have fixed evaporator and variable condenser fan speeds. Note that the evaporator fan speed is in the lower portion of the optimal range because current equipment must remove latent and sensible heat whereas the LLC heat pump removes only sensible heat. In current models, the condenser fan speed is varied and it is important to do so.

Heat pump model

Optimized versus non-optimized heat pump


Difference in COP for optimal versus zero subcooling case

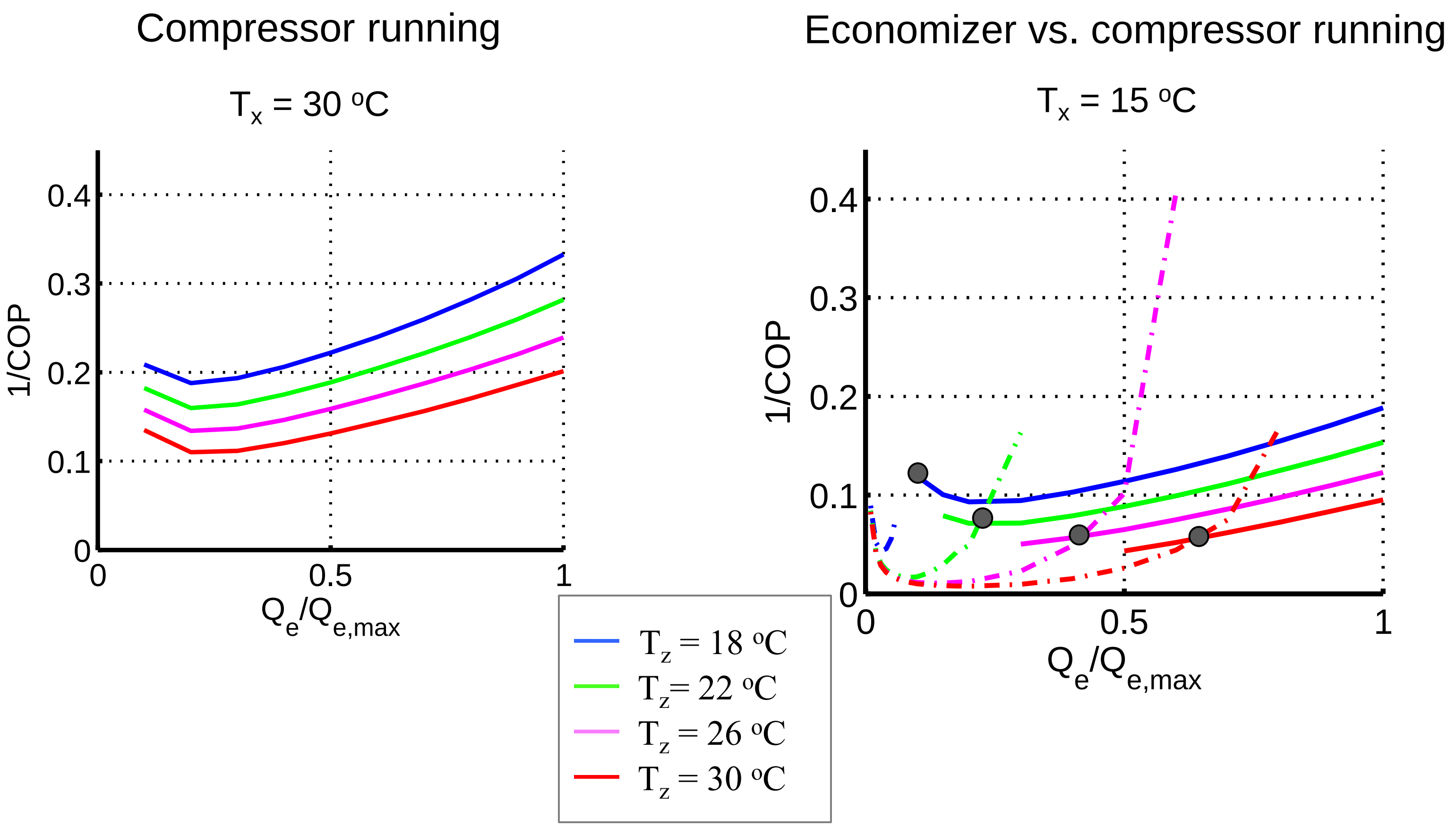
Difference in COP for optimal versus fixed airflow case

$$COP_{relative\_difference} = \frac{COP_{opt\_sub} - COP_{zero\_sub}}{COP_{opt\_sub}} * 100$$

$$COP_{relative\_difference} = \frac{COP_{opt\_airflow} - COP_{fixed\_airflows}}{COP_{opt\_airflow}} * 100$$

Heat pump model

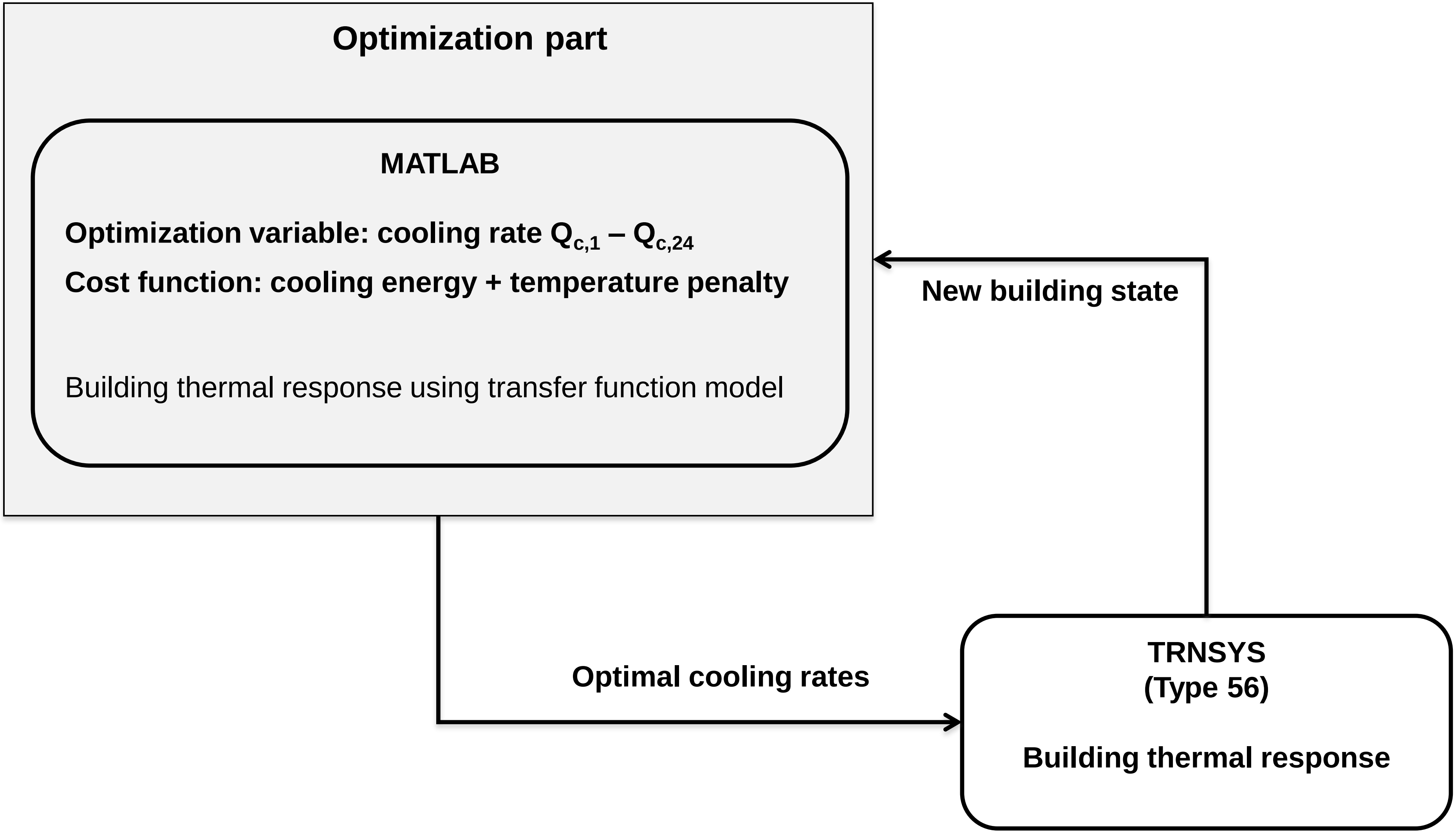
Heat pump performance maps


Compressor running

Economizer vs. compressor running

Zakula T., Armstrong P. and Norford L. 2012. Optimal Coordination of Compressor, Fan and Pump Speeds Over a Wide Range of Loads and Conditions. *HVAC&R Research* 18(06)

Model predictive control



Optimization part

MATLAB

Optimization variable: cooling rate $Q_{c,1} – Q_{c,24}$

Cost function: cooling energy + temperature penalty

Building thermal response using transfer function model

New building state

Optimal cooling rates

TRNSYS
(Type 56)

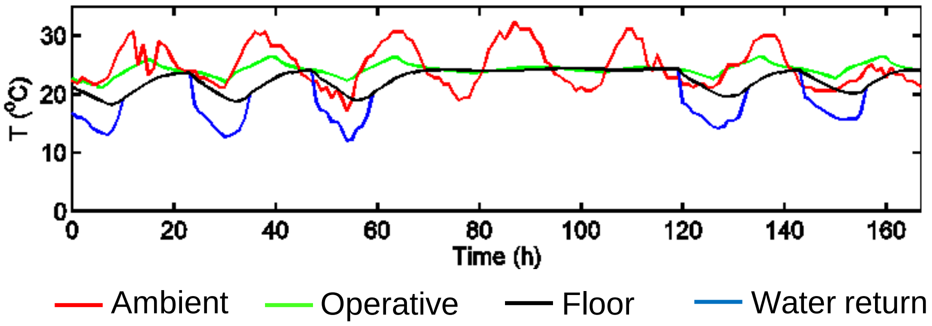
Building thermal response

Current work

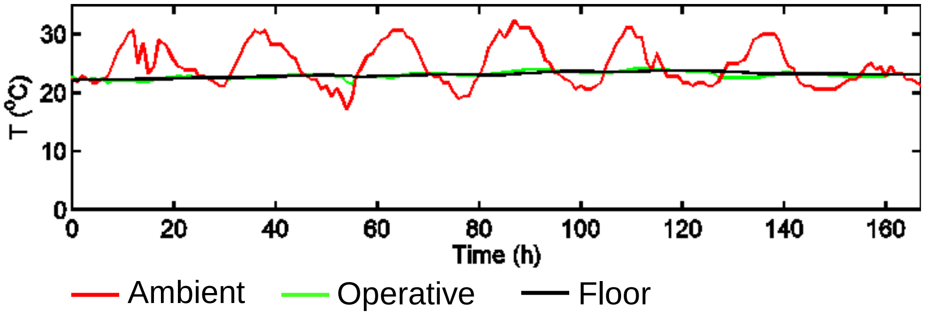
LLC and split-system simulation results

(for one summer week in Atlanta, sensible only)

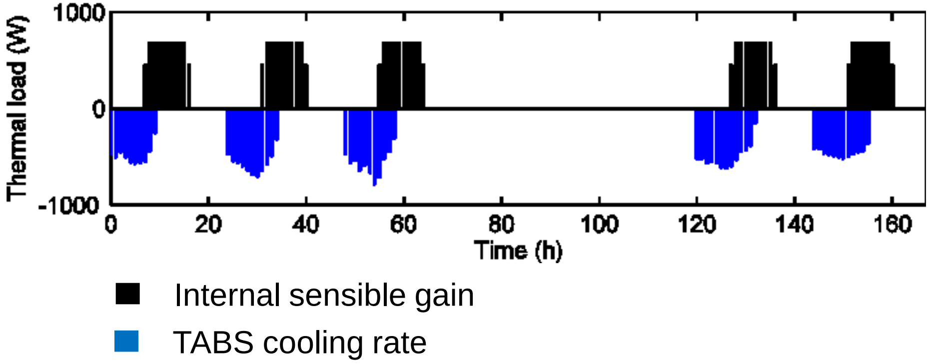
Temperature profiles for LLC



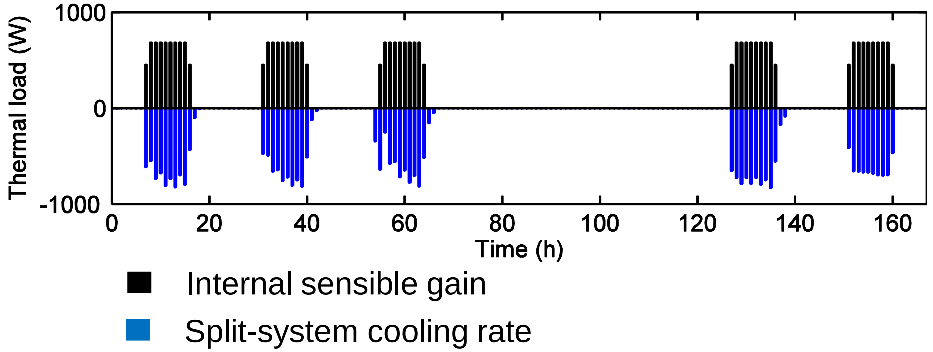
Temperature profiles for split-system



Load profiles for LLC



Load profiles for split-system



Current work

LLC energy savings relative to split-system

(for one summer week, sensible only)

Atlanta

Phoenix

Original Mr. Slim (Q_{max} = 3.0 kW)

Table 1A: Power consumption relative differences

	20/26	20/24
24	-0.9 %	-61.5 %
23	16.8 %	-33.1 %
22	30.2 %	-11.7 %

Table 2P: Power consumption relative differences

	20/26	20/24
24	-11.2 %	-48.6 %
23	1.3 %	-32.0 %
22	13.9 %	-15.2 %

Sized Mr. Slim (Q_{max} = 1.5 kW)

Table 1A: Power consumption relative differences

	20/26	20/24
24	8.8 %	-49.6 %
23	25.6 %	-22.0 %
22	38.3 %	-1.3 %

Table 2P: Power consumption relative differences

	20/26	20/24
24	-4.9 %	-46.8%
23	9.5 %	-26.7 %
22	21.2 %	-10.3 %

Sized Mr. Slim and modified TABS (15 cm pipe spacing)

Table 1A: Power consumption relative differences

	20/26	20/24
24	21.6 %	-25.5 %
23	36.0 %	-2.4 %
22	46.9 %	15.1 %

Table 2P: Power consumption relative differences

	20/26	20/24
24	13.7 %	-21.9 %
23	25.5 %	- 5.2 %
22	35.1 %	8.4 %

Relative difference = (Split – LLC)/Split

And… results for Singapore

LLC energy savings relative to split-system

(for 1 summer week, sensible only)

Singapore

Sized Mr. Slim and modified TABS (15 cm pipe spacing)

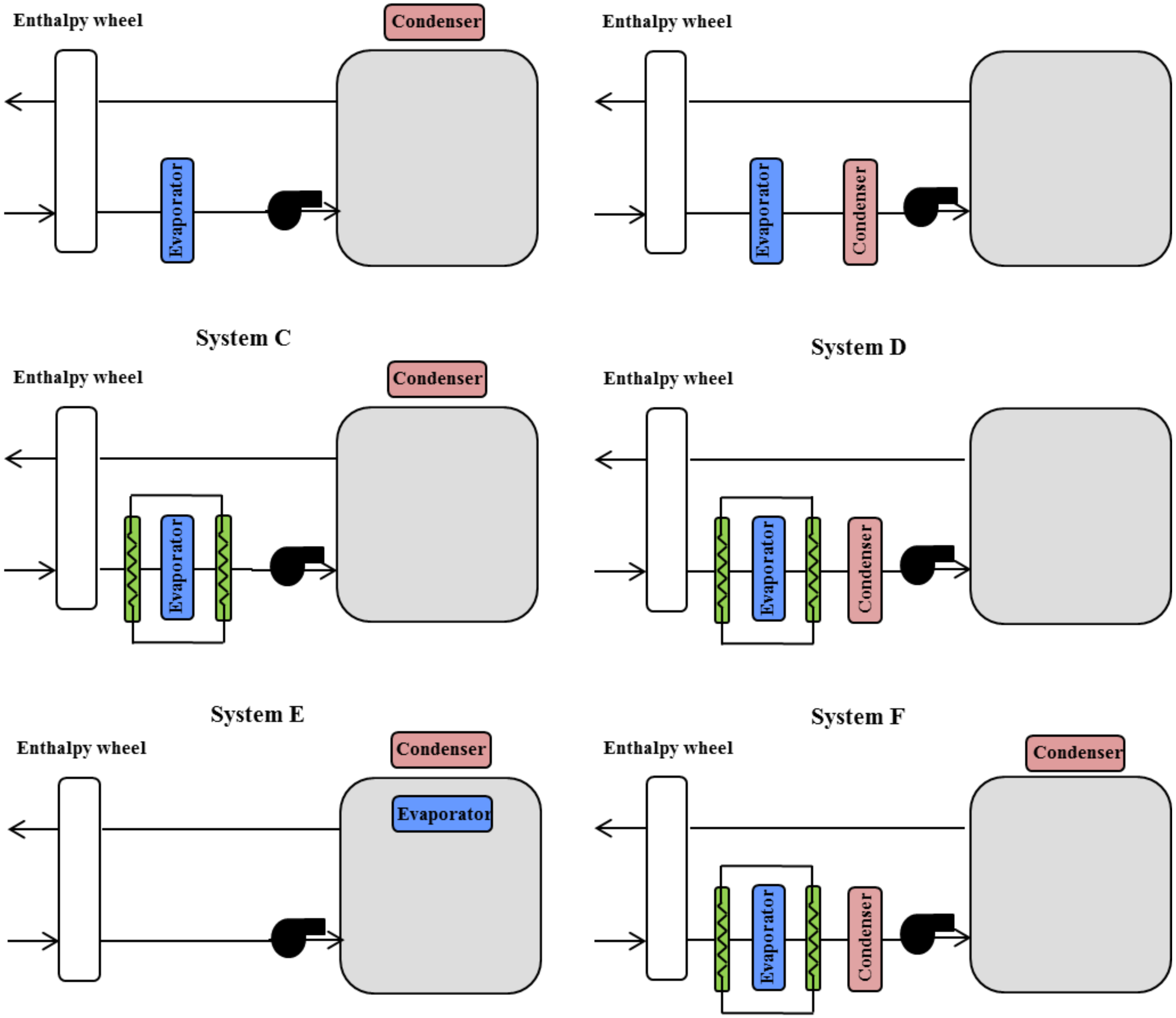
Table 1A: Power consumption relative differences

	20/26	**20/24**
24	14.72	-28.56
23	28.70	-7.49
22	39.58	8.91

Relative difference = (Split – LLC)/Split

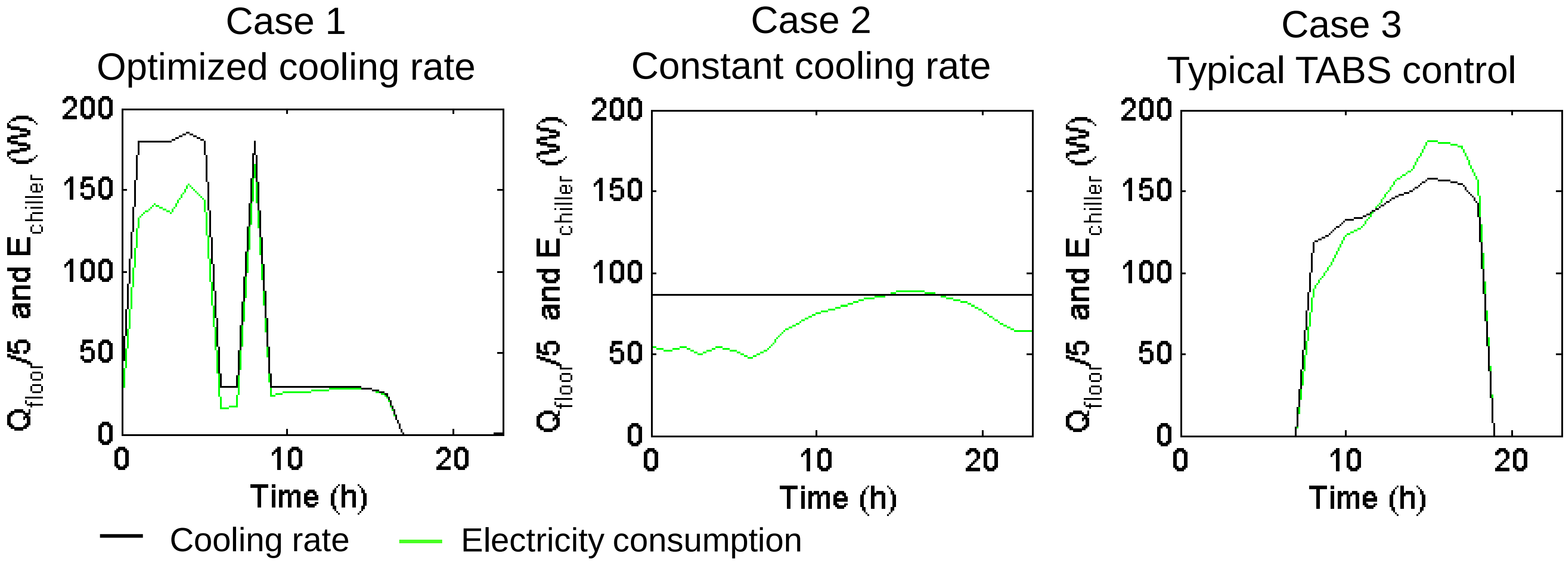
Current work

Proposed dehumidification options



Current work: estimating demand response

Bidding into ancillary services market for frequency support


Case 1
Optimized cooling rate

Case 2
Constant cooling rate

Case 3
Typical TABS control

Weekly energy consumption (kWh)

Bid	No bid	7	8	9	10	11	12	13	14	15	16
Case 1	3.11										
Case 2	4.70	4.75	4.75	4.73	4.72	4.71	4.70	4.69	4.65	4.63	4.61
Case 3	5.92	5.92	5.81	5.76	5.73	5.70	5.75	5.77	5.72	5.73	5.72

Future steps

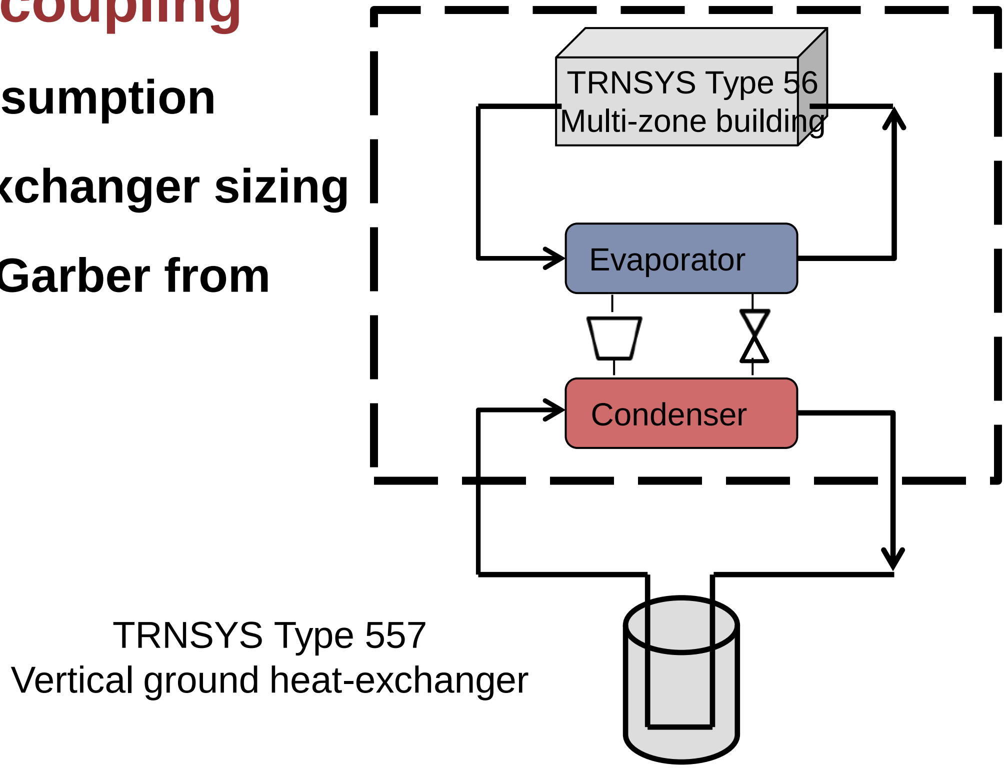
Annual optimization

- VAV versus LLC system
- VAV with precooling versus LLC system

Ground-source heat pump coupling

- Annual cooling energy consumption
- Appropriate ground heat exchanger sizing

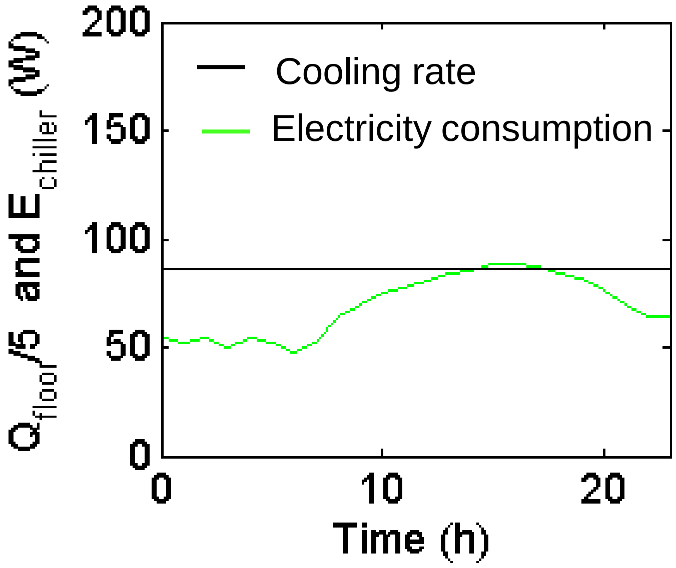
(in collaboration with Dennis Garber from Cambridge University, UK)



Future steps

Ground-source heat pump coupling

Expected optimal cooling control: constant cooling rate



In general:

COP=function($T_{w,\ return}$, T_x, Q_c, $Q_{c.max}$)

For ground coupling and Q_c = constant:

$T_{w,return}$ ≈ constant

$T_x = T_{ground,return}$ ≈ constant

COP ≈ constant

Cooling rates equally spread through day and night

⇩

Good potential to bid into an ancillary service market during peak-hours

Thank you

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