Heating, ventilation, and air conditioning

Purdue ME 597, Distributed Energy Resources

Kevin J. Kircher

Outline

Heating with electricity

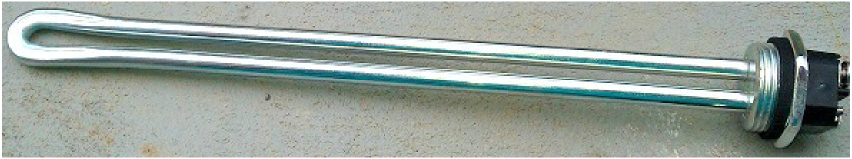
Air conditioning

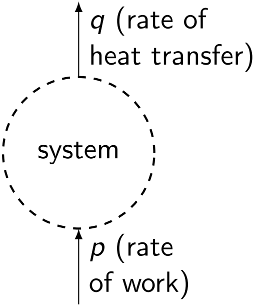
Sizing heating and cooling equipment

Common HVAC system configurations

Thermal distribution models

Electric resistance heating





q (rate of heat transfer)

system

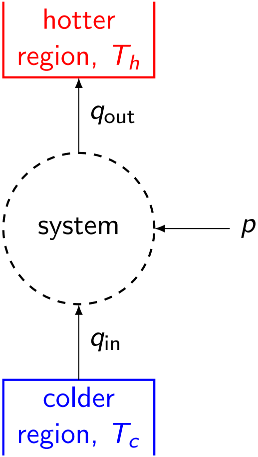
p (rate of work)

- (steady-state) 1st law: $p = q$
- coefficient of performance:

$$\eta = \frac{\text{heat transfer output}}{\text{net work input}} = \frac{q}{p} = 1$$

- Joule heating: $p = I^2 R$
- dirt-cheap to install, lasts ~forever
- but inefficient/expensive to run

Heat pump thermodynamic cycles



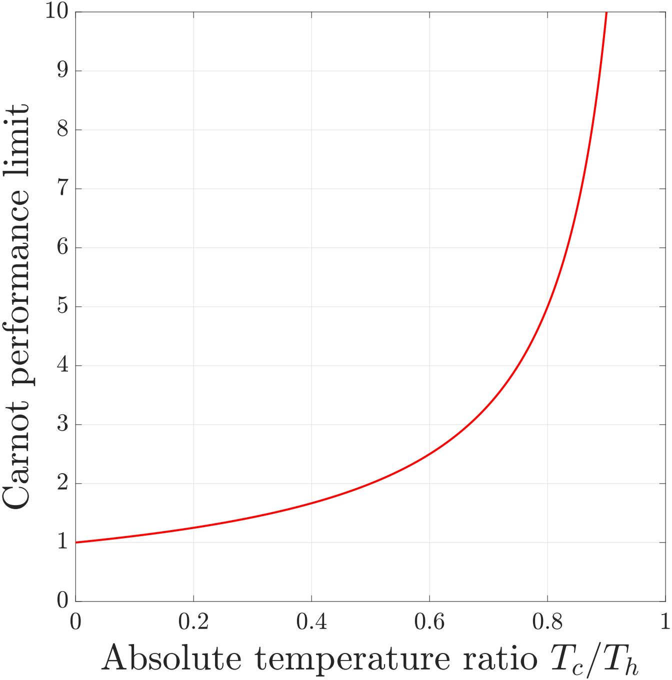
- 1st law: $q_{\text{in}} + p = q_{\text{out}}$
- coefficient of performance:

$$\begin{aligned}\eta &= \frac{\text{heat transfer output}}{\text{work input}} \\ &= \frac{q_{\text{out}}}{p} = \frac{q_{\text{out}}}{q_{\text{out}} - q_{\text{in}}} \\ &= \frac{1}{1 - q_{\text{in}}/q_{\text{out}}}\end{aligned}$$

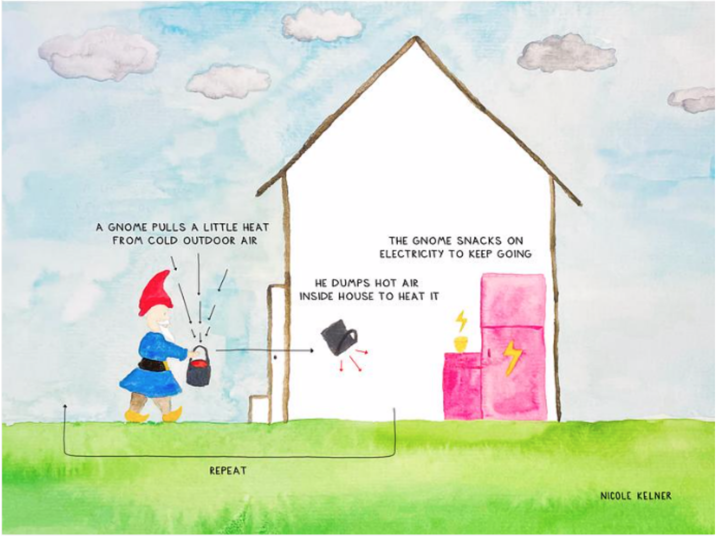
- Carnot performance limit:

$$\eta \leq \frac{1}{1 - T_c/T_h}$$

with T_c, T_h in Kelvin

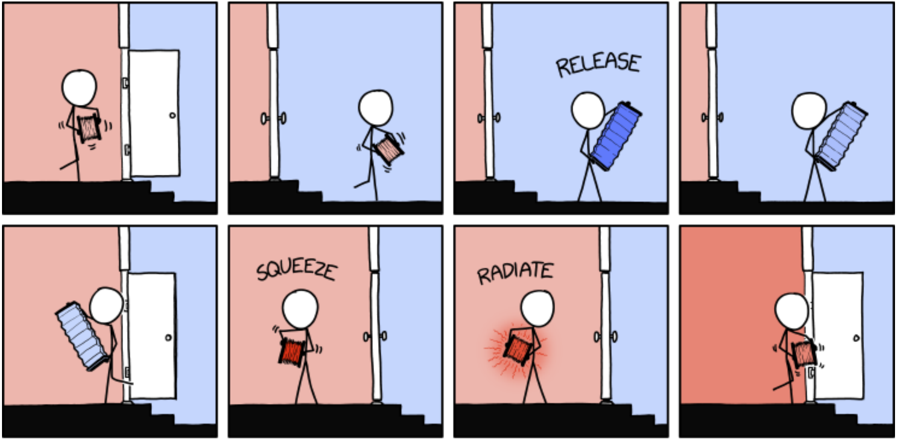

Carnot performance limit
0
1
2
3
4
5
6
7
8
9
10
0
0.2
0.4
0.6
0.8
1
Absolute temperature ratio T_c/T_h

How to pump heat



Nicole Kelner

How to pump heat



MANUAL HEAT PUMPS ARE SUCH A PAIN.

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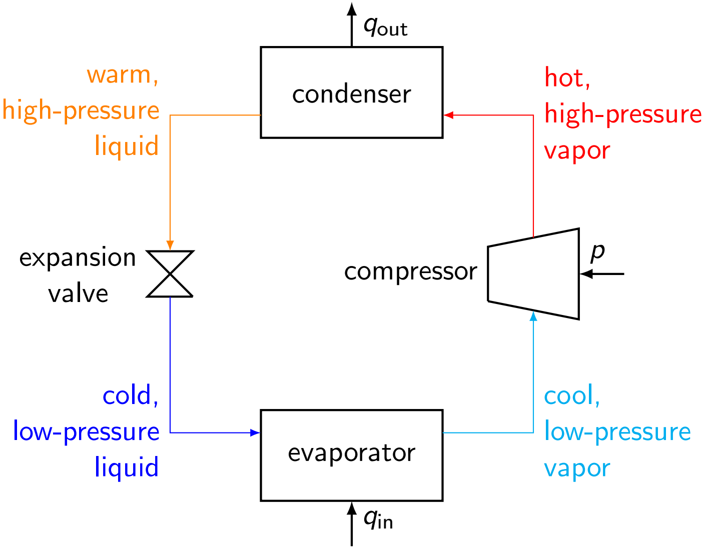
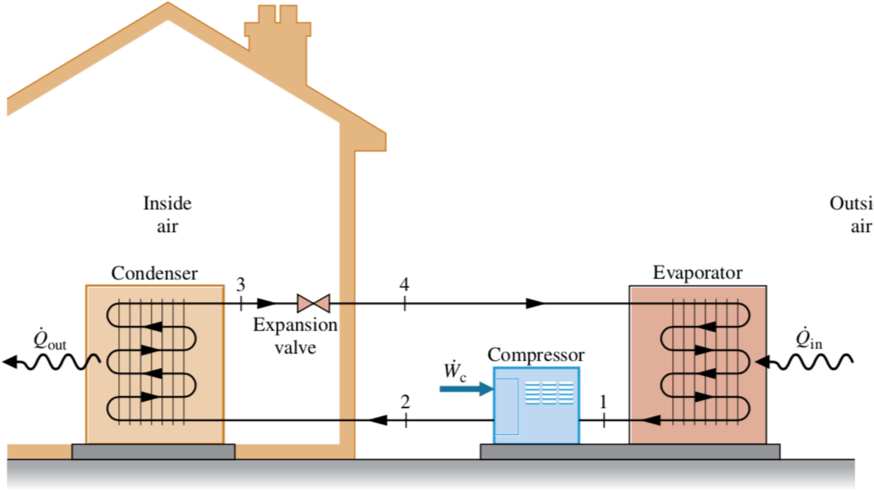
xkcd: Heat Pump

How to pump heat (vapor compression cycle)

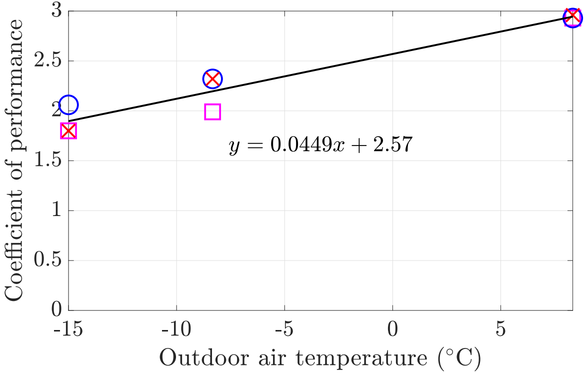




Moran (2018): *Fundamentals of Engineering Thermodynamics*

Fitting real heat pump COPs to manufacturer data

- real heat pumps COPs don't come close to Carnot limit
- NEEP collects manufacturer-reported steady-state COP data



central ducted units from 3 manufacturers, 21 °C indoor air

Northeast Energy Efficiency Partnerships: Cold-climate air-source heat pumps

Simple heat pump simulation

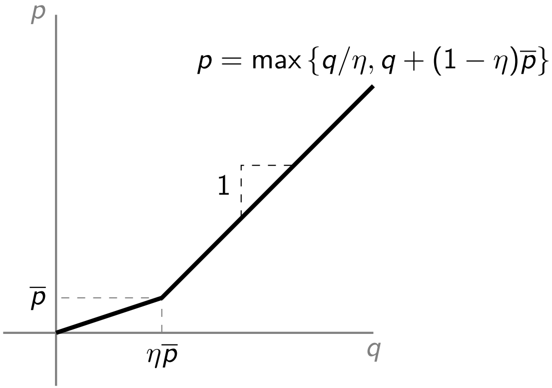
- assume ~constant indoor temperature
- model COP as a function ~only of outdoor temperature θ
- fit COP curve $\eta : \mathbf{R} \to \mathbf{R}$ to manufacturer data, such as

$$\eta(\theta) = \max\{1, 0.0449\theta + 2.57\}$$

  (most heat pumps switch to resistance [COP 1] at low θ)
- read off cold-weather compressor power limit $\overline{p}$ from data
- simulate building with constraint $q(t) \in [0, \eta(\theta(t))\overline{p}]$
- set input electric power to $p(t) = q(t)/\eta(\theta(t))$

Heat pumps with resistance backup

- heat pumps are expensive to install but cheap to run
- resistance is cheap to install but expensive to run
- hybrid systems pair heat pumps with resistance backup



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Common HVAC system configurations

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Air conditioners are just one-way heat pumps

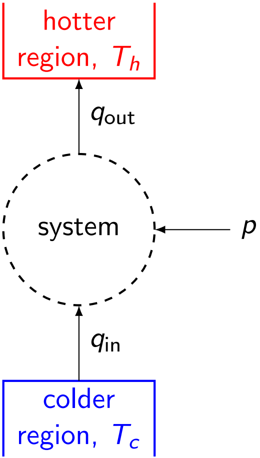
- most heat pumps can run in reverse to cool and dehumidify

$\Longrightarrow$ lower up-front cost than heater + (one-way) air conditioner

Heat pump vocabulary

heat source	heat sink	device name
refrigerator air	kitchen air	refrigerator
freezer air	kitchen air	freezer
outdoor air	indoor air	air-source heat pump (ASHP) (or air-to-air heat pump)
indoor air	outdoor air	air conditioner or ASHP
outdoor ground	indoor air	ground-source heat pump (or geothermal heat pump)
outdoor air	indoor water	heat-pump water heater
indoor air	indoor water	heat-pump water heater
indoor water	outdoor air	chiller
outdoor water	indoor air	water-source heat pump

Refrigeration thermodynamic cycles



hotter region, T_h

q_{out}

system

p

q_{in}

colder region, T_c

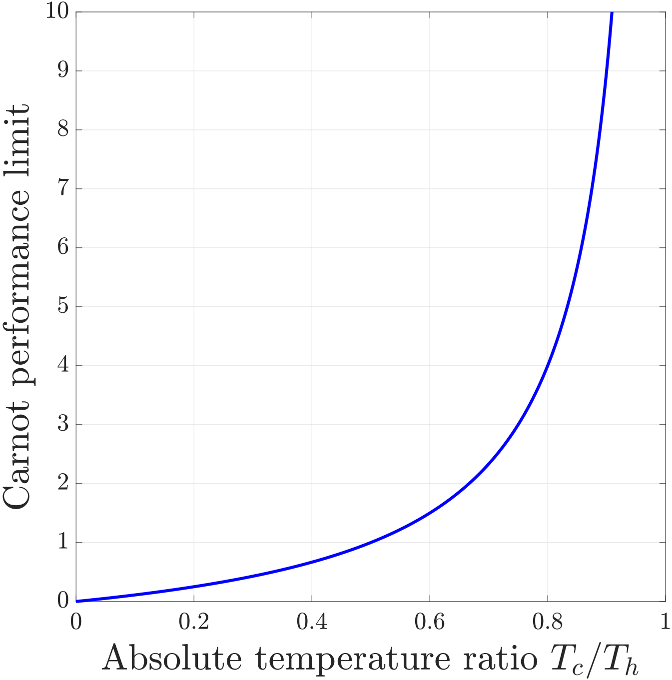
- 1st law: $q_{\text{in}} + p = q_{\text{out}}$
- coefficient of performance:

$$\eta = \frac{\text{heat transfer } \textbf{input}}{\text{work input}} = \frac{q_{\text{in}}}{p} = \frac{q_{\text{in}}}{q_{\text{out}} - q_{\text{in}}} = \frac{q_{\text{in}}/q_{\text{out}}}{1 - q_{\text{in}}/q_{\text{out}}}$$

- Carnot performance limit:

$$\eta \leq \frac{T_c/T_h}{1 - T_c/T_h}$$

with T_c, T_h in Kelvin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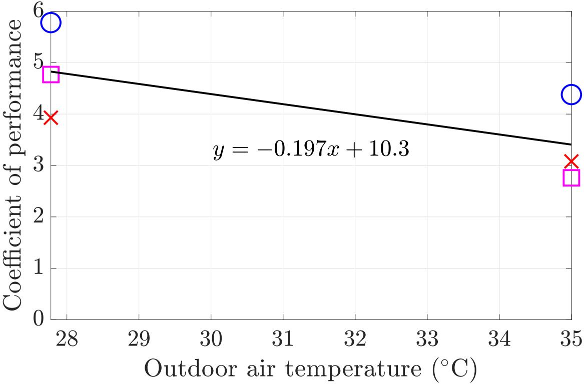
1

Real air conditioner COPs

NEEP database also has cooling COP data



central ducted units from 3 manufacturers, 21 °C indoor air

Northeast Energy Efficiency Partnerships: Cold-climate air-source heat pumps

Dehumidification

- air conditioners
  - ◇ reduce indoor air temperature (**sensible** load)
  - ◇ condense water out of indoor air (**latent** load)
- total load = sensible load + latent load
- **sensible heat ratio** s is ratio of sensible load to total load
- building simulations may produce sensible load $q(t)$ only
- to account for dehumidification, estimate s and set

$$p(t) = \frac{q_{\text{tot}}(t)}{\eta(\theta(t))} = \frac{q(t)}{s\eta(\theta(t))}$$

- in reality, s depends on weather, building, occupant behavior
- first cut: set $s \approx 70$ to 95% for humid to dry climates

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Sizing for heating

- estimate overall indoor-outdoor thermal resistance R
- get design outdoor temperature θ^{des}
- set design indoor temperature T^{des} to occupant preference
- pick plausible q_e^{des} for $\sim$4 AM
- size to steady-state heat load in design conditions:

$$\overline{p}_h = \frac{r}{\eta(\theta^{\text{des}})} \left(\frac{T^{\text{des}} - \theta^{\text{des}}}{R} - q_e^{\text{des}} \right)$$

- oversize ratio $r \approx 1.2$ to 1.5, typically

Sizing for cooling

- like heating, but

$$\overline{p}_c = \frac{r}{s\eta(\theta^{\mathrm{des}})} \left(\frac{\theta^{\mathrm{des}} - T^{\mathrm{des}}}{R} + q_e^{\mathrm{des}} \right)$$

- q_e^{des} should be plausible for sunny afternoon

Sizing two-way heat pumps

- calculate $\overline{p}_h$ and $\overline{p}_c$ for heating and cooling design conditions
- if $\overline{p}_h \leq \overline{p}_c$, set $\overline{p} = \overline{p}_c$ (size for cooling)
- if $\overline{p}_h > \overline{p}_c$, options:
  1. set $\overline{p} = \overline{p}_h$ (size for heating)
  2. set $\overline{p} = \overline{p}_c$ and add backup $\geq \eta(\theta^{\mathrm{des}})(\overline{p}_h - \overline{p})$
  3. get biggest available unit and add backup $\geq \eta(\theta^{\mathrm{des}})(\overline{p}_h - \overline{p})$
- backup heat could be
  - ◇ another heat pump
  - ◇ resistance
  - ◇ heat storage
  - ◇ wood
  - ◇ propane
  - ◇ heating oil
  - ◇ natural gas

Sizing example for a house in Lafayette

	θ^{des} (°C)	T^{des} (°C)	q_e^{des} (kW)	$\eta(\theta^{\text{des}})$
heating	-16	21	1	1.8
cooling	32	24	4	4

- input parameters: $R = 3\ ^\circ\text{C}/\text{kW}$, $r = 1.3$, $s = 0.8$
- sizing results: $\overline{p}_h = 8.2$ kW, $\overline{p}_c = 2.7$ kW
- biggest available residential heat pumps have $\overline{p} \approx 7.5$ kW

$\Longrightarrow$ need some form of backup heat

ASHRAE: Design conditions for Lafayette, IN, USA

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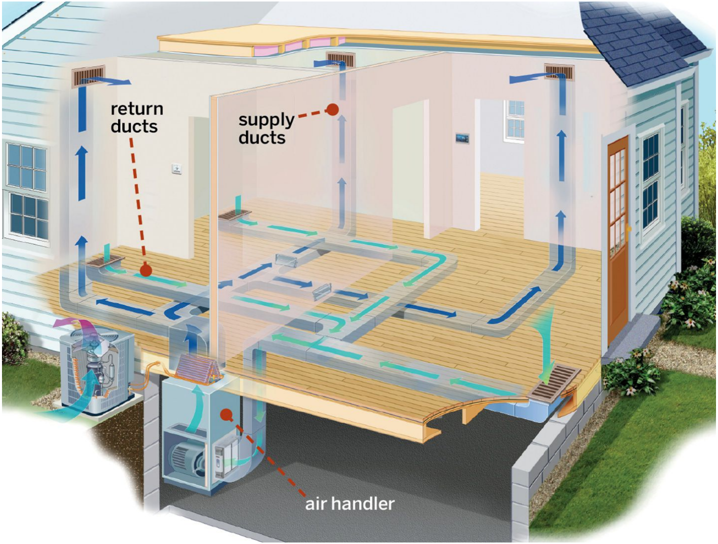
Central ducted residential systems

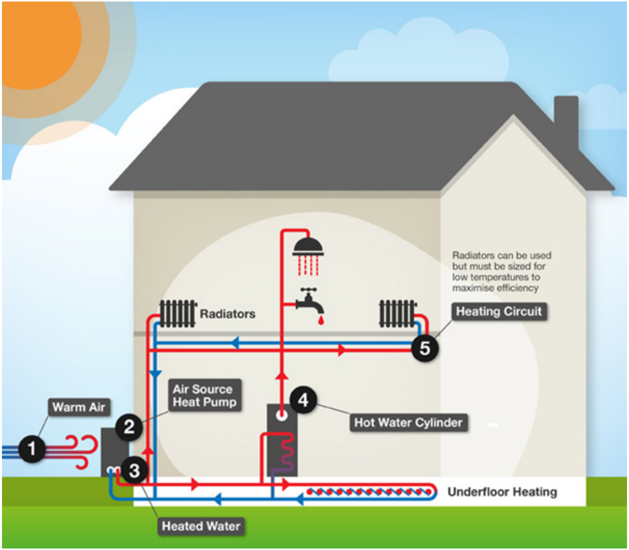


This Old House: Central air conditioning

Ductless mini-split residential systems

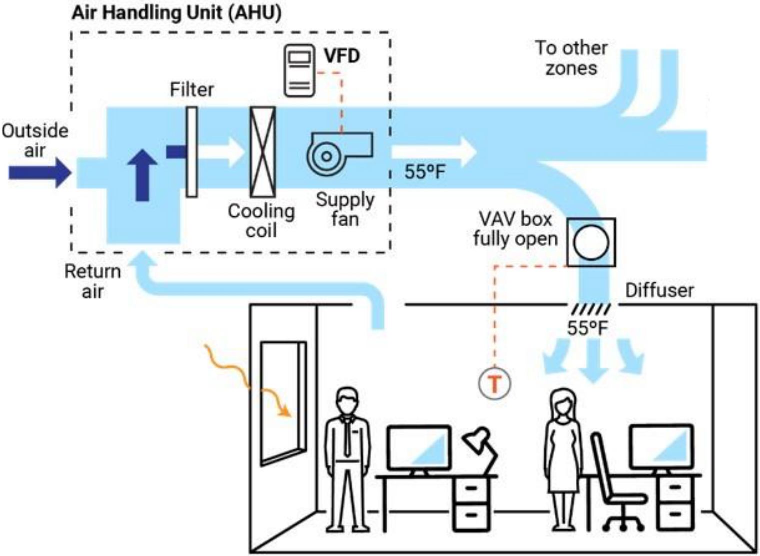
New Hampshire Electric Co-Op: Ductless mini-split heat pumps

Hydronic residential systems



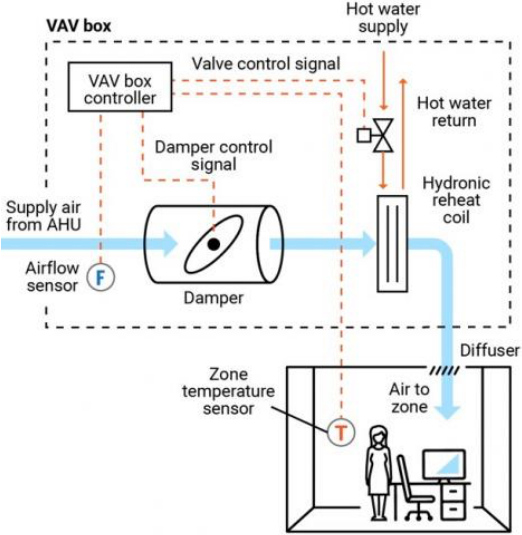
Energy.nl: Heat pump – Air to water

Variable air volume commercial systems



Pacific Northwest National Laboratory: Variable air volume systems

Variable air volume boxes



Pacific Northwest National Laboratory: Variable air volume systems

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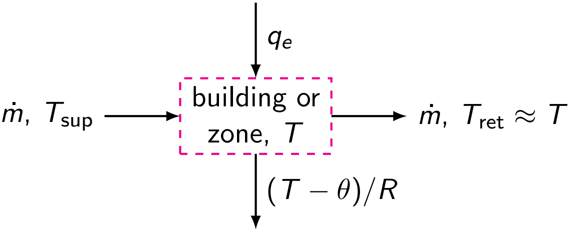
Forced-air heat transfer

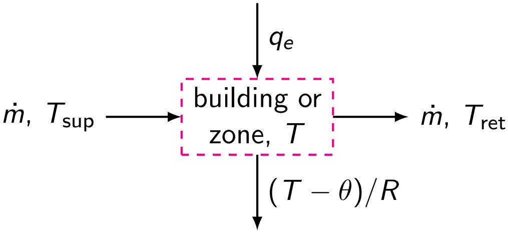


- $\dot{m}(t)$ (kg/s) is mass flow rate of supply air
- power balance:

$$C\frac{\mathrm{d}T(t)}{\mathrm{d}t} = \frac{\theta(t) - T(t)}{R} + \underbrace{\dot{m}(t)c_p(T_{\text{sup}}(t) - T(t))}_{q_c(t)} + q_e(t)$$

- $c_p = 1$ kJ/(kg$^\circ$C) is specific heat of air at constant pressure

Hydronic heat transfer

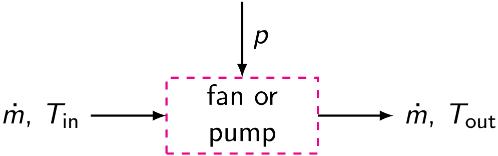


- $\dot{m}(t)$ (kg/s) is mass flow rate of supply water
- power balance:

$$C\frac{\mathrm{d}T(t)}{\mathrm{d}t} = \frac{\theta(t)-T(t)}{R} + \underbrace{\dot{m}(t)c(T_{\text{sup}}(t)-T_{\text{ret}}(t))}_{q_c(t)} + q_e(t)$$

- $c = 4.2$ kJ/(kg°C) is specific heat of water

Fans and pumps



p

$\dot{m},\ T_{\text{in}}$

fan or pump

$\dot{m},\ T_{\text{out}}$

- (rough) fan power balance:

$$p(t) \approx \dot{m}(t) c_p (T_{\text{out}}(t) - T_{\text{in}}(t))$$

- pump:

$$p(t) \approx \dot{m}(t) \left[c(T_{\text{out}}(t) - T_{\text{in}}(t)) + \frac{P_{\text{out}}(t) - P_{\text{in}}(t)}{\rho} \right]$$

- $P_{\text{in}}(t)$, $P_{\text{out}}(t)$ (kPa) are inlet, outlet pressures
- $\rho = 1000$ kg/m^3 is density of water

Pump and fan affinity laws

- in theory, pumps and fans follow the affinity law

$$p(t) = \alpha \dot{m}(t)^3$$

  where $\alpha = p_{\text{rated}} / \dot{m}_{\text{rated}}^3$

- in practice, usually fit a model to $(\dot{m}, p)$ data, such as

$$p(t) = \beta_0 + \beta_1 \dot{m}(t) + \beta_2 \dot{m}(t)^2 + \beta_3 \dot{m}(t)^3$$